

CFD SIMULATION OF THE INFLUENCE OF DRIFT AND HEELING ON CALM WATER RESISTANCE AND ADDED WAVE RESISTANCE

Cedric Fallet^{1,*}, Sören Ehlers¹, Marco Klein¹,

¹German Aerospace Center (DLR), Institute of Maritime Technologies and Propulsion Systems

ABSTRACT

In order to meet latest and future regulatory requirements with respect to efficiency and emissions, new technologies like wind-assisted propulsion systems (WAPS) will play a vital role for the shipping industry. Although WAPS promise high fuel savings for a broad range of concepts, the application and diffusion of this new technology is still limited. An important reason for that is the high uncertainty with respect to the actual efficiency gains. This uncertainty arises from the complex interactions between WAPS, ship and environmental conditions. These interactions become especially relevant for sail-based technologies which affect the ship's performance by inflicting substantial drift and heeling. The extent of drift and heeling defines the so called "sailing condition".

This work focuses on a structured assessment of the sailing condition on the ship's resistance and other relevant forces. For that, CFD simulations are conducted to determine the influence of drift and heeling on a ship's calm water resistance and the added wave resistance in regular head waves. The study is applying the commercial finite-volume solver STAR-CCM+ to solve the RANS equations to simulate the performance of the established KVLCC2 tanker geometry for multiple varying drift angles as well as heel angles. For the calm water resistance it is shown that the influence of both drift and heeling is small on the overall resistance, while the yaw moment is increasing under drift and heeling conditions. Moreover, the simulations in waves revealed that the added wave resistance decreases with increasing heel and drift angle in long waves. In short waves instead, added wave resistance is increased by heeling as well as by high drift angles.

Keywords: CFD, wave resistance, sailing, heeling, drift

1. INTRODUCTION

The international shipping industry has to overcome major challenges in the future in order to reach legislative goals with

respect to greenhouse gas (GHG) emissions and efficiency standards. To reduce the impact of international shipping as one of the main contributors to global green house emissions, the International Maritime Organization (IMO) intends to implement a regulatory plan to continuously cut GHG emissions [1] with the ultimate goal of zero-emission shipping by 2050. Along indicative check-points for the emission reduction goals and measures as the uptake of zero-emission fuels, an important tool is the Energy Efficiency Design Index (EEDI). It imposes increasingly strict minimum efficiency levels for new ships. These requirements create strong incentives for new technologies which are not only able to reduce emissions like alternative fuels but at the same time increase efficiency through fuel savings. One of the most promising group of technologies are wind assisted propulsion systems (WAPS) that harvest wind energy to reduce the required engine power. The various systems types range from towing kites to Flettner rotors and wing-sails. The application of these technologies to new ships is limited so far, although their potential is estimated to be between a few percent up to 40% under optimal conditions [2]. Newer data from implementations on new ships show that even under real world conditions, fuel savings above 10% can be achieved [3]. Although the latest trends indicate an increasing number of installations, the diffusion of this technology is still inhibited due to several factors [3]. One of the most relevant factors for ship owners and operators is the uncertainty with respect to the actual benefit from installing WAPS onboard of a ship. Since it requires high capital investments and increases operational risks, a reliable and accurate estimation of potential savings is mandatory to assess the profitability of such an economic case. In recent years, the increasing attention for wind-assisted ship propulsion led to growing effort in developing tools to assess the performance of ship equipped with WAPS. Besides high fidelity methods like Computational Fluid Dynamics (CFD), which is able to provide accurate results for a limited set of operational conditions due to its computational costs [4, 5], reduced order methods are a popular approach e.g. for route-based performance assessments [6–8]. While CFD methods allow to

*Corresponding author: cedric.fallet@dlr.de

cover almost all relevant physical effects, especially interactions between the WAPS and the ship, reduced order models (ROMs) usually rely on semi-empirical models for the hydrodynamics and aerodynamics. Besides the limited, intrinsic accuracy of the ROMs, another shortcoming is that interactions between the original ship and the WAPS-equipped ship are not fully covered. Hydrodynamic and aerodynamic forces influence each other in various, complex ways which are often times just neglected in simplified models.

The focus of this work is the “sailing condition” which becomes relevant for systems that can produce significant forces in directions aside from the main propulsion direction [9]. Such systems are, for example but not exclusively, lift-based wing sails whose force is usually not fully aligned with the longitudinal direction of the ship. Thus, leading to a side force that simultaneously exerts a roll moment on the ship. As a result, the force equilibrium implies a heeling of the ship until the hydrostatic forces compensate the exerted heeling moment. Moreover, the side force leads to lateral drift of the ship so that the effective flow direction is changed by a certain drift angle. Both, drift and heel angle, define the sailing condition, whose influence on the hydrodynamic forces will be investigated in this paper. Although the focus of the investigations lies on the effect on the ship’s resistance, side force and yaw moment are also of importance as their compensation, e.g. through rudder deflection, inflicts additional resistance. However, the influence on the rudder and the effect of the varying inflow to the propeller is not part of this investigation. Existing research focuses on the influence of a varying drift angle under calm water conditions [10–12] as a crucial part for the construction of maneuvering models [13]. Substantially less information are present for the interaction of a drifting ship in waves and the impact of heeling under both calm water and wave conditions. This paper addresses this research gap in order to assess the impact of the sailing conditions and thus a relevant effect of WAPS on a ship’s operating conditions.

For this purpose, unsteady RANS simulations are conducted with the help of the commercial finite volume solver *STAR-CCM+* [14]. The simulations consist of calm water simulations and simulations in regular, head waves. In these conditions, drift and heel angle of the established *KVLCC2* test case are varied to study their influence on the ship’s resistance and other relevant force terms. The two-phase flow is resolved with the volume of fluid technique and waves are modelled with Stokes-V theory. A virtual spring system is applied together with external moments to accurately define the drift and heel angle of the ship under wave conditions, so that it is still able to move in all 6 degrees of freedom.

The paper introduces the investigations by first presenting the test case, including geometry description and definitions before explaining the numerical methodology. These explanations provide details on the governing physics, numerical schemes and discretisation procedure of the developed CFD setup. In the following section, the setup is validated with respect to the calm water and added wave resistance under zero drift and heeling conditions. Afterwards, the results of the parameter study are presented, first for the calm water forces, then for the mean wave forces. From that, the main findings are extracted and summa-

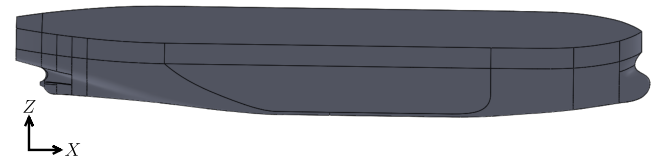


FIGURE 1: KVLCC2 geometry with extruded deck

ized in the conclusion of this paper.

2. TEST CASE

For this investigation, the established *KVLCC2* test case is chosen that represents a generic tanker geometry. The *KVLCC2* has been part of several CFD workshops like the workshop on ship hydrodynamics in Gothenburg [15] or the SIMMAN workshop series for maneuvering [16]. Therefore, a wide range of experimental and simulation results is available e.g. for validation purposes. At the same time, a tanker is a typical candidate for a retro-fit for emission reduction or efficiency increase with the help of WAPS. Hence, it is a suitable and representative test case for the following investigations.

2.1. Geometry

The standard *KVLCC2* geometry is modified for the purpose of preventing extensive green water, similar to the approach presented in [17] and in correspondence with the experiments conducted for [18]. For that, the deck is extruded in normal direction resulting in the geometry depicted in Fig. 1. However, the underwater geometry and the main particulars of the *KVLCC2* geometry used in this paper remain unaltered and are summarized in Tab. 1. It should be noted that there are several papers which strictly adhere to the original geometry including the effects of the green water loads. Moreover, some papers utilize or compare to the *KVLCC2-m* geometry which includes critical geometry changes at the stern leading to inconsistencies between results for the test case.

TABLE 1: Main particulars of the KVLCC2

Quantity	Symbol	Unit	Ship	Model
Length at waterline	L_{wl}	[m]	325.5	7.12
Length btw. perp.	L_{pp}	[m]	320	7.0
Breadth	B	[m]	58.0	1.269
Draft	T	[m]	20.8	0.455
Displacement	Δ	[t]	3.126e5	3.272
Inertia radius X	r_{xx}	[m]	0.38B	0.38B
Inertia radius Y / Z	r_{yy}, r_{zz}	[m]	0.25 L_{pp}	0.25 L_{pp}
Scale	λ_s	[-]	1	45.7143

2.2. Coordinate system and angle definitions

The definition of the drift angle and the heel angle follows general conventions. As shown in Fig. 2, the drift angle β corresponds to the angle between the velocity vector U of the ship and

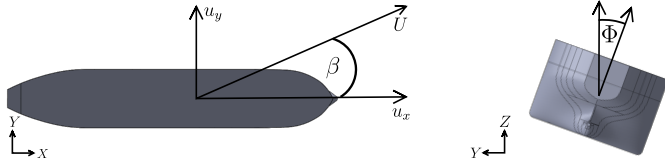


FIGURE 2: Coordinate system and angle definitions

its longitudinal axis from stern to bow. Whereas the heel angle Φ corresponds to the static roll angle around the x-axis of the coordinate system which is located at the center of gravity and follows solely the translations of the body within the calm water plane and the yaw motion. This system is called “horizontal body axis system” in the context of maneuvering and all the forces are evaluated in this coordinate system as it is line with the common experimental setup of resistance tests which measure the resistance force in the longitudinal direction parallel to the calm water surface.

3. NUMERICAL METHODOLOGY

For all simulations, the commercial finite-volume solver *STAR-CCM+* (version 19.06.009-R8) is harnessed to solve the complex two-phase flow around a ship in regular waves. The underlying physical equations are presented together with the most important discretisation approaches and details on the numerical setup.

3.1. Governing equations

The physical flow problem is defined by the Navier-Stokes equations for an incompressible fluid. They include the mass conservation and momentum conservation equation and are given here in their Reynolds-averaged form with the help of the Einstein summation convention. The mass conservation includes the spatial derivative of the fluid velocity u_i

$$\frac{\partial \bar{u}_i}{\partial x_i} = 0 \quad (1)$$

while the conservation of momentum further involves the time derivative, pressure p and the shear stress tensor τ .

$$\frac{\partial \bar{u}_i}{\partial t} + \frac{\partial}{\partial x_j} \left(\bar{u}_i \bar{u}_j + \overline{u'_i u'_j} \right) = -\frac{1}{\rho} \left(\frac{\partial \bar{p}}{\partial x_i} + \frac{\partial \bar{\tau}_{ij}}{\partial x_j} \right). \quad (2)$$

To close this system of equations, a model for the Reynolds stresses $\overline{u'_i u'_j}$ is required, which is usually based on the Boussinesq approximation that relates the turbulent stress tensor to the time-averaged shear rate tensor. This gives at most two free parameters which can be defined by additional conservation equations. In case of the $k - \omega - SST$ model, that is applied for this paper, one conservation equation is solved for the turbulent kinetic energy k and one for the specific turbulent dissipation rate ω . The definition of the equations with all their free parameters exceeds the scope of the paper but can be found in [19]. On top of that, since the flow problem is a two-phase flow of water and air whose solution requires additional techniques, an equation for the

volume fraction of either air or water needs to be solved in order to apply the volume-of-fluid (VoF) approach [20].

$$\frac{\partial \alpha_i}{\partial t} + \frac{\partial (\alpha_i u_j)}{\partial x_j} = 0 \quad (3)$$

Such definition is given by Eq. (3) which describes the conservation of the volume fraction α_i within the flow domain, where the sum of volume fractions of air and water sums up to 1.

3.2. Numerical schemes and methods

Without providing an extensive list of all numerical details, some of the most important information about the numerics are presented in the following. The convective fluxes of the turbulent and main flow variables are based on second order Roe scheme [21] with gradient evaluation utilizing the least-squares approach. Instead, the volume-fraction equation requires a more sophisticated scheme in order to maintain a sharp interface between the two phases. For that, the High Resolution Interface Capturing (HRIC) scheme blends between second order upwind and downwind schemes depending on the orientation of the interface and cell faces [22]. The corresponding angle factor is set to 0.15. The temporal evaluation of the equations is conducted with a second order predictor-corrector approach following the well-known SIMPLE algorithms [23]. Thus, the equations are solved sequentially in an uncoupled manner with an equation for a pressure correction that replaces the mass equation within the iterative solution process.

To account for the ship motions, the overset approach is used which couples the motions of the ship to an additional overset mesh, that moves within a static background mesh. The flow quantities are interpolated at the interface of overset and background mesh so that a continuous flow domain can be represented by the two meshes. Although this approach is highly versatile it still requires care when calibrating the meshes at their interface to achieve accurate interpolations. Another important cornerstone for the simulations is an approach to implement waves in the domain while preventing excessive reflections at the boundaries of the domain. An effective method is the forcing approach which forces the solution towards analytical solutions of a specified wave theory. In *STAR-CCM+*, the forcing approach is implemented as a source term that is added to the momentum equations and is defined by

$$Q_F(\vec{x}, t) = c_F b(x) (\phi(\vec{x}, t) - \phi^*(\vec{x}, t)). \quad (4)$$

Its strength automatically scales with the deviation of the numerical solution ϕ from the analytical one ϕ^* [24] and the distance to the defined boundary through the smoothly declining function $b(x)$. However, it also requires the constant factor c_F which can be determined by the approach given by Perić in [25]. In the conducted simulations, this forcing constant was increased by a factor of three as it proved beneficial for conserving the wave energy. The described approach forces the solution towards the prescribed solution. By applying the forcing only near the boundaries, waves can be imposed through boundary conditions while reflections can be prevented and the flow of interest around the ship is not directly affected. The underlying wave theory is

chosen to be 5th order Stokes theory [26], which covers regular waves in the nonlinear regime. It defines the boundary values as well as the initial condition and analytic solution for the forcing approach.

3.3. Flow domain and its discretisation

The extent of the flow domain should cover all relevant flow phenomena but for efficiency reasons it should be as small as possible. Therefore, the flow domain in all simulations is adapted to the ship length and varying wave length λ_i . A crucial criterion is the buffer length which is the distance between the ship and the start of the forcing zones. For side and inlet boundaries, a minimum distance of $\max(0.5L_{pp}, 0.5\lambda_i)$ is chosen, while towards the outlet it is set to $\max(0.5L_{pp}, \lambda_i)$. Moreover, a minimum length of the forcing zones is defined, being at least $\max(L_{pp}, \lambda_i)$. The forcing zones are defined at all boundaries that are perpendicular to the calm water surface. The type of the boundary conditions is depending on the inflow conditions. For the cases, with oblique inflow ($\beta \neq 0$), at all lateral boundaries a velocity-based inflow condition is imposed, while for the other cases, the outlet boundary defines a pressure-based outflow condition. At the bottom boundary a wall boundary is implemented and the top boundary defines either a pressure-based outflow or an velocity-based inflow condition. The discretisation of the domain in form of the mesh incorporates best practices provided in the *STAR-CCM+* user guide [24] and is built up on a previously validated setup [27] for seakeeping problems. An important difference is the spring system that is adapted to produce an additional restoring force / moment for the sway and yaw motion. It is described in more detail in Sec. 4. The mesh is an unstructured, hexahedron-based, trimmed mesh which includes refinements at specific locations like the free surface, the wake, bow and stern of the ship. The base cell size for the background mesh refers to four cells per ship length and for the overset mesh to 65 cells per ship length. The overset mesh is further refined by a factor of two in the water and by a factor of four at bow and transom of the ship. As the resolution at the free surface is crucial to prevent excessive wave energy loss, the mesh contains 20 cells per wave amplitude and 100 cells per wave length. Especially for short waves, this leads to rapidly increasing mesh sizes which is only partially compensated by the scaling of the domain. However, a proper resolution is particularly important for short waves. Furthermore, a proper resolution of the boundary layer at the ship is necessary to accurately predict the shear stress and by that the friction related portion of the resistance. As wall functions can be applied in *STAR-CCM+*, the y^+ value is not limited to be $y^+ < 1$ and is therefore chosen to be on average around 80. An exemplary representation of a mesh visualizing several relevant refinements is given in Fig. 3 and Fig. 4 together with the mesh sizes for all simulated waves in Tab. 2.

3.4. Temporal discretisation

All the simulation are using a second order temporal discretisation scheme (cf. 3.2) with 750 time steps per encounter frequency. This time steps size corresponds to choices from other papers and proved to be suitable also in the context of these investigations. The simulation duration depends on the wave period

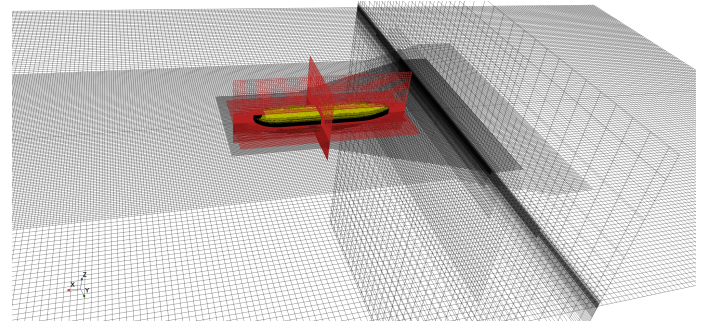


FIGURE 3: Background and overset mesh for case: $\frac{\lambda}{L_{pp}} = 1.4$, $\Phi = \beta = 0^\circ$

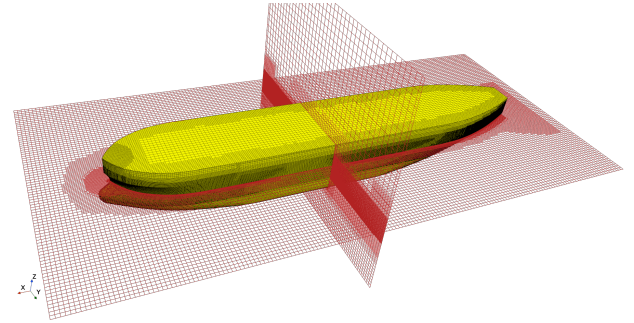


FIGURE 4: Overset mesh for case: $\frac{\lambda}{L_{pp}} = 1.4$, $\Phi = \beta = 0^\circ$

as at least 15 wave encounters are simulated. To increase the efficiency of the calm water simulations, the equilibrium solver of *STAR-CCM+* is utilized with a ten times increased time step size and first order temporal discretisation scheme.

4. SIMULATION AND ANALYSIS PROCEDURE

In the following, several relevant details of the simulation setup and physical parameters defining the test case are described together with the analysis procedure to obtain the added wave resistance.

4.1. Setup details

The investigations are conducted for a Froude number of 0.142 and a Reynolds number of around $9.26e6$. Four different regular head waves are simulated for three different drift and heel angles each. The waves are chosen in such a way that they cover characteristics of the added wave resistance coefficient, like its peak and nearly constant regime in short waves. They are weakly nonlinear waves in deep water conditions that can be well approximated with Stokes-V wave theory. An overview of the wave parameters together with all simulated conditions is given in Tab. 3.

Concerning the drift and heel angles, it has to be noted that the listed target angles are varying over time with small offsets from the defined values. This is because all six degrees of freedom are active during the seakeeping simulations. Although the incoming waves are long-crested head waves, the drift or heeling introduce asymmetry to the flow that leads to natural roll and yaw motions. However, to still approximately reach the defined drift and heel angles, the average moments required

TABLE 2: Mesh sizes

Case No.	λ/L_{pp}	Number of cells		
		Background	Overset	Total (avg. active)
M_1	0.5	44.20e6	14.37e6	52.30e6
M_2	1.1	8.77e6	2.99e6	10.39e6
M_3	1.4	7.24e6	3.55e6	9.46e6
M_4	2.0	10.40e6	4.29e6	12.73e6

TABLE 3: Wave parameters, drift and heel angles applied in simulations

λ/L_{pp} [-]	H/λ [-]	T [s]	β [°]	Φ [°]
0.5	0.01667	1.495	0, 5, 10	0, 5, 10
1.1	0.01667	2.218	0, 5, 10	0, 5, 10
1.4	0.01667	2.502	0, 5, 10	0, 5, 10
2.0	0.01667	2.990	0, 5, 10	0, 5, 10

to hold the ship in the defined position are imposed as artificial moments. These moments are obtained from the prior calm water simulation. Besides that, four virtual springs are connected to the ship in order to fix its average position and orientation. Two springs are fixed 2 m in front of the center of gravity while the other two are fixed 2 m behind the center of gravity. The total assembly is visualized in Fig. 5. The other end point of the springs is set multiple ship lengths away to reduce the influence of the spring forces on the roll, pitch and heave motions. All the springs are relatively stiff with spring constants of 800 N/m to reduce the deviations from the target drift or heel angle through roll or yaw motions.

4.2. Resistance assessment

The total resistance of a ship in waves is the mean of the time varying force in longitudinal direction acting on the ship. To extract the mean value from the periodical force signal, a

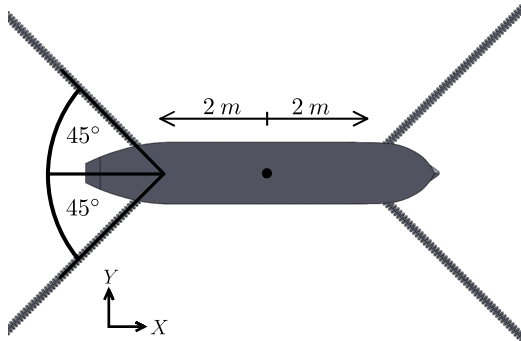


FIGURE 5: Virtual spring system applied in simulations

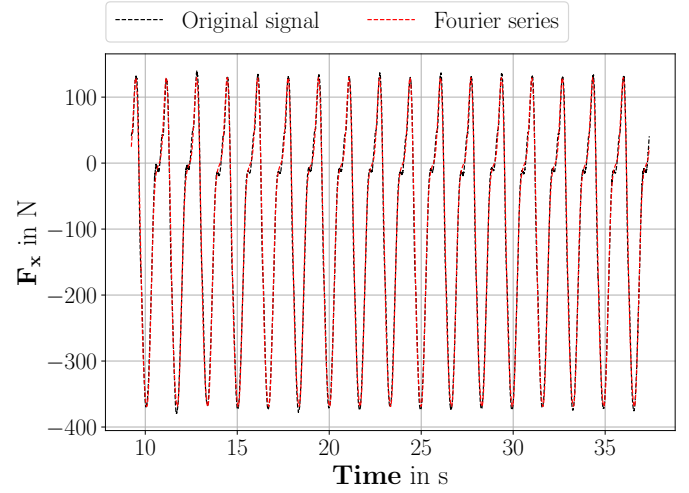


FIGURE 6: Exemplary force signal analysis based on fourier series for the wave with $\frac{\lambda}{L_{pp}} = 1.1$

frequently applied approach is to fit a fourier series like

$$F(t) = a_0 + \sum_{i=1}^n a_i \sin(2\pi\omega t) + b_i \cos(2\pi\omega t) \quad (5)$$

to the signal [28]. Herein, the coefficients a_i and b_i correspond to the amplitude of the i th harmonic. The i th amplitude is defined by $A_i = \sqrt{a_i^2 + b_i^2}$, whereas the amplitude of the 0th harmonic is equal to a_0 which is the sought mean that represents the total ship resistance R_T . For result analysis, a 4th order fourier series is fitted so that the mean squared error between the actual force $R(t)$ and the fitted fourier series $F(t)$ is minimized. To illustrate the procedure and the outcome, Fig. 6 contains the original force history of a simulation and the corresponding result from the fitting of the fourier series.

Besides accurate results for the total resistance from the sea-keeping simulations, a consistent way to determine the calm water resistance R_{CW} is required to obtain the pure added wave resistance R_W according to

$$R_W = R_T - R_{CW}. \quad (6)$$

A common way to reduce the impact of discretisation errors on the wave resistance is to compute the calm water resistance with the same mesh and setup. Thereby, any insufficiently resolved viscous effects do not deteriorate the accuracy of the wave resistance. This approach is applied in this paper together with an additional simulation predicting solely the wave propagation to obtain the wave amplitude at the location of the ship including the inevitable energy loss. These wave simulations are conducted on the same background mesh as the seakeeping simulation and showed overall a small loss in wave amplitude ranging from 1.6% for the longest wave to 3.3% for the shortest wave. The so measured wave amplitude A_w can then be used to compute the wave resistance coefficient C_W :

$$C_W = \frac{R_W}{\rho g A_w^2 B^2 / L_{pp}} \quad (7)$$

where ρ represents the water density and $g = 9.81 \frac{m}{s^2}$ the gravitational acceleration.

4.3. Validation

Before investigating the influence of drift and heeling on the ship's resistance, the setup is validated with respect to calm water and added wave resistance. The importance of high accuracy results is less pronounced for parameter studies as long as the setups are consistent. Nevertheless, the setup is still validated based on an assessment of the uncertainty introduced by the finite spatial discretisation with the help of a mesh convergence study and comparisons to experimental results. This allows to assess the absolute accuracy and quality of the setup and the presented results.

4.3.1. Spatial discretisation error and uncertainty The established method to evaluate the spatial discretisation error and uncertainty is based on the Richardson extrapolation [29] and the Grid Convergence Index (GCI) [30]. The Richardson extrapolation allows to estimate the asymptotic solution on an infinite fine resolved mesh to approximate the discretisation error. In correspondence, the GCI gives a quantification of the uncertainty for a certain solution. The whole analysis procedure is based on solutions from three successively refined meshes. For the calculation of the extrapolated, asymptotic solution and the GCI, the process described in [31] is followed. Its results are presented in Tab. 4 for the calm water resistance and the added wave resistance for the wave with $\lambda = 1.1L_{pp}$. The results include the mesh refinement factor r , the respective solution ϕ , the relative error e_a (to the solution on the fine mesh), the extrapolated relative error e_{ext} and the GCI. It can be seen that the relative errors as well as the GCI are very small for the calm water resistance simulations with values below 1% even for the coarsest mesh. The same meshes are used for the computation of the added wave resistance. For the added wave resistance, e_{ext} on the coarse mesh is rather low with a value of 1.44% which is further reduced to 0.32% on the finest mesh, while the GCI ranges from 4.61% on the coarse to 0.40% on the fine mesh. For both simulation types, oscillatory convergence is observed with decreasing cell sizes. All further computations are conducted on meshes that refer to the refinement level of the medium mesh as the shown results indicate a sufficiently small error on this level, for both calm water resistance and added wave resistance.

4.3.2. Calm water resistance The *KVLCC2* test case has been part of several hydrodynamics CFD workshops like the Gothenburg workshop [15] which provided experimental reference data, published also in [32]. Although the experimental results correspond to a different Reynolds number, the *ITTC* guidelines on resistance model tests can be used to scale the results to the correct Reynolds number [33]. This scaled resistance coefficient corresponds to the experimental calm water resistance shown in Fig. 7 that is compared to the resistance values obtained from the calm water simulations required to compute the added wave resistance. As the the calm water resistance is computed for every wave (cf. Tab. 3) on the mesh for the corresponding seakeeping simulation, the results are shown for four different meshes resulting from the four different simulated waves. It can

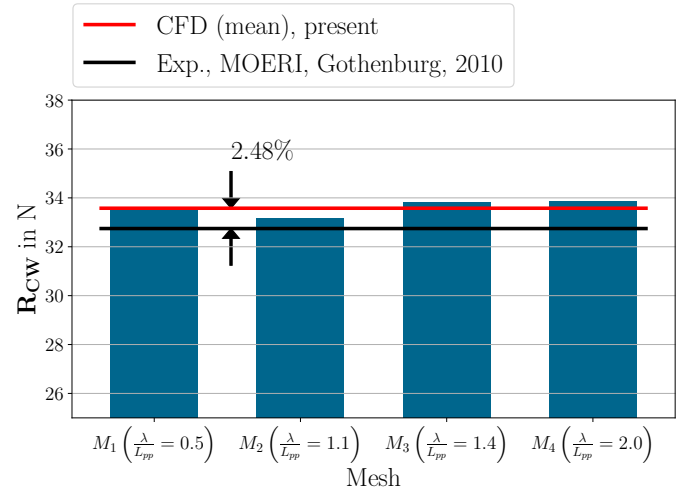


FIGURE 7: Validation results for calm water resistance

be seen that the deviations between the meshes are rather small. Their mean value exhibits a difference of around 2.5% compared to the experiments which is considered to be sufficiently small, since it is within the common level of uncertainty of such experiments.

4.3.3. Added wave resistance Compared to the calm water resistance, results for the added wave resistance, with both experimental and numerical methods, can not be determined with the same fidelity. This is due to the varying orders of magnitude of the first and second order wave forces. The added wave resistance represented by the second order wave force is usually considerably smaller than the dominant first order wave forces, making the accurate determination of the former more difficult. Therefore, even experimental results from different institutions can exhibit differences easily up to 10% and beyond. However, there are some papers like [18, 34] which systematically assess and quantify the uncertainty in their results. Such information allow for a more differentiated comparison to simulation results. To validate the suitability of the CFD setup to accurately simulate the wave forces acting on the *KVLCC2*, the four reference waves at $\lambda/L_{pp} = 0.5, 1.1, 1.4, 2.0$ are simulated in regular head waves conditions without drift or heeling and compared against experimental results from [18, 35, 36]. The CFD results are given in Fig. 8 as red triangles. The error bars correspond to the uncertainties determined in [18] and a confidence interval with a confidence level of 2σ . It can be seen that the CFD results are in very good agreement with the experimental results overall as well as with the confidence intervals given from [18]. For the waves at $\lambda/L_{pp} = 0.5, 1.1, 2.0$ the results are within the range of uncertainty of the experiments. Solely for the wave at $\lambda/L_{pp} = 1.4$ the calculated resistance values deviates more pronounced from the reference results. Nevertheless, from the author's point of view, it is still in an acceptable range, especially when considering the deviations across the shown experimental results. Moreover, it has to be noted that the wave steepness is not equal across the results, although it can have significant influence on the added wave resistance as it was shown by Seo et al. [37]. Hence,

TABLE 4: Results of mesh convergence studies for calm water resistance and added wave resistance for the wave with $\lambda = 1.1L_{pp}$

Mesh	No. cells	r	R_{CW}				R_W			
			ϕ	e_a	e_{ext}	GCI	ϕ	e_a	e_{ext}	GCI
coarse	2.671e6	—	33.226	0.37%	0.67%	0.38%	16.412	1.76%	1.44%	4.61%
medium	5.854e6	1.30	33.164	0.56%	0.86%	0.14%	16.837	0.79%	1.11%	1.46%
fine	13.523e6	1.32	33.350	—	0.30%	0.38%	16.706	—	0.32%	0.40%

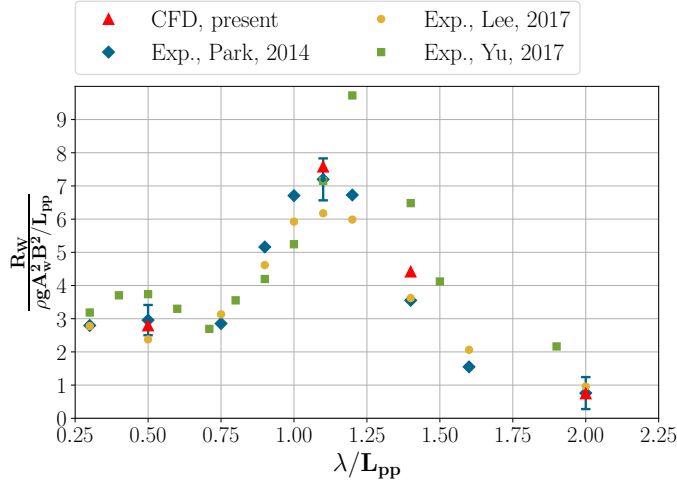


FIGURE 8: Validation results for added wave resistance

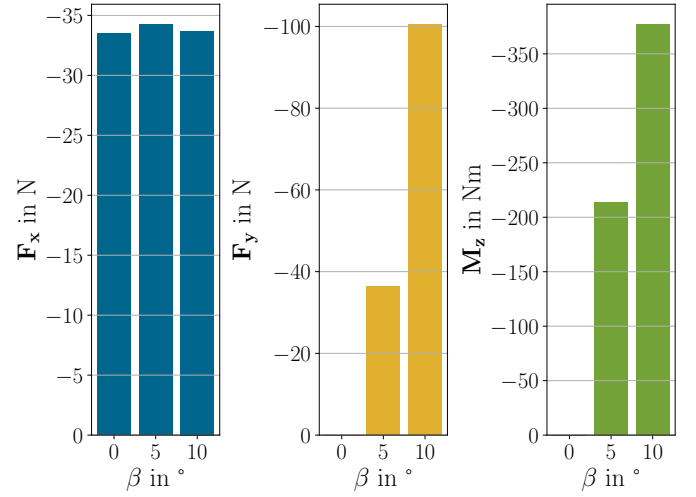


FIGURE 9: Drift study results from calm water simulations

the proposed CFD setup is considered to be validated based on the shown results for the added wave as well as the calm water resistance.

5. RESULTS

Although the ship resistance is as such solely the force in longitudinal direction of the ship, it can not be examined without considering the implications of the side force and the yaw moment. The side force needs to compensate the force produced by the WAPS, while the yaw moment requires active compensation during operation to ensure that the ship stays on its course. For a ship like the *KVLCC2*, this is done through adjustments of the rudder to counter yaw moment and portions of the side force. However, this leads to further resistance caused by the flow around the deflected rudder, which means that a yaw moment would increase the total resistance of the ship. A direct quantification of this mutual influence is not straightforward, which is why the analysis is conducted on a qualitative level. In the following, the result analysis is divided into two parts. The first part covers the calm water results, first for a variation of the drift and then for a varying heeling. In the second part, the influence of drift and heeling on the added wave resistance is presented with the same structure.

5.1. Calm Water simulations

The following results are based on calm water simulations with grid M_1 (cf. Tab. 2), which is chosen since it is the finest

mesh of all and should thus yield the smallest discretisation errors.

5.1.1. Impact of drift on ship forces First, for a variation of the drift angle, the results in Fig. 9 show rather small changes for the resistance. The resistance force is slightly increasing with increasing drift angle but the relative change by 1-2% is very small. This observation is consistent with existing experimental and numerical results [11, 12]. The detailed flow analysis e.g. in [10] proposes that the complexity of vortex structures and the extent of flow separation at the rear grows with increasing drift angles. However, it seems to be compensated by the shift of the stagnation point and resulting pressure distribution, which leads to minor effects on the overall force, although the local flow is significantly influenced. For the side force, a rapid increase with increasing drift angle is observed. It seems to grow faster than a linear relation between side force and drift angle. In contrast, the trend of the yaw moment is almost a linear one, with the yaw moment almost doubling from $\beta = 5^\circ$ to $\beta = 10^\circ$. Both trends are in line with results from known drift tests [13]. From that, it can be concluded that the *KVLCC2* experiences at most a slight increase in the resistance force within realistic drift conditions. However, during operation, the corresponding yaw moment would require counter measures like a rudder deflection that would increase overall resistance of the ship.

5.1.2. Impact of heeling on ship forces The results for the heeling study in calm water are presented in Fig. 10. For both forces, resistance and side force, no significant influence of the

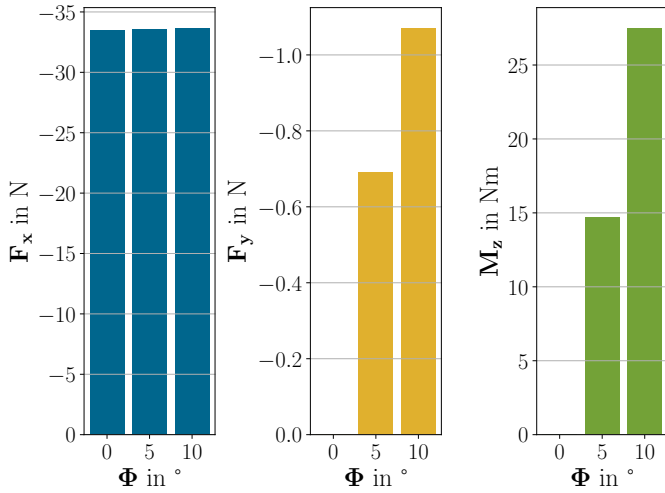


FIGURE 10: Heeling study results from calm water simulations

heel angle is perceived. The resistance is increasing by less than 1%. This is explicable since the main flow conditions do not change while the underwater geometry is influenced very little due to the bulbous form of the hull. The most significant effects are at the bow and the stern where some asymmetries are present in the flow field. Solely for that yaw moment it can be observed, that it linearly increases with increasing heel angle, which is notable as the moment is produced without significant increase in total side force. Thus, the yaw moment might result from small contributions of side force from bow and stern which compensate each other. The implications of the increase in yaw moment on the overall resistance are the same as for the drift influence. However, the yaw moment is around one order of magnitude lower for the *KVLCC2* heeling with $\Phi = 10^\circ$ than drifting with $\beta = 10^\circ$. So, to sum up, the sailing condition has a negligible influence on the resistance force while the yaw moment is mainly increasing due to drift, but heeling would additionally increase it, since in a realistic scenario, a positive drift angle would combine with a negative heel angle.

5.2. Added resistance simulations

It is not possible to specify the exact heeling and drift angles for the ship in waves as all six degrees of freedom are enabled and the spring system only weakly enforces e.g. the yaw / drift angle. Although the average calm water moments are applied as additional moments as well, the influence of the waves can not be determined beforehand. Therefore, the drift and heel angles are oscillating around some mean value which is close to the target values from Tab. 3 but might deviate from it. So, to distinguish between the differing definitions, the index t is used for the target angles while $\bar{\beta}$ and $\bar{\Phi}$ represent the actual time-averaged angles from the simulation. To rate the agreement between target and actual values, Fig. 11 shows the time-averaged values of the heeling and drift angle from the simulations. Every point represents one specific simulation defined by $\frac{\lambda}{L_{pp}}$, β and Φ . Points of the same color belong to one specific wave length but differing target drift and heel angles. Deviations from the

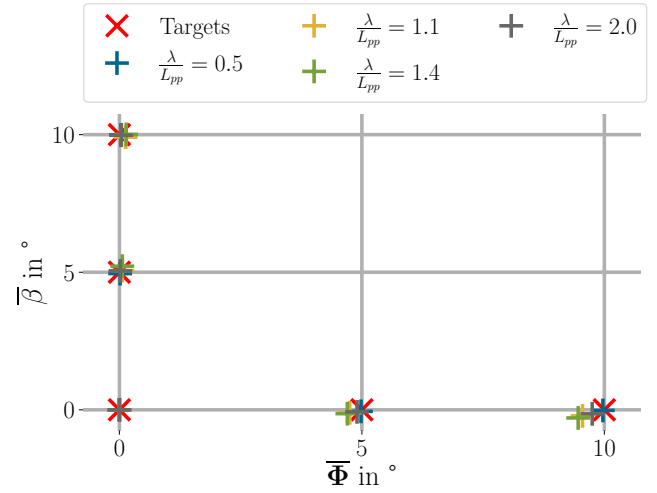


FIGURE 11: Time averaged heeling and drift angles

target angles are encoded as distances to the nearest red cross at $(\beta_{t,i}, \phi_{t,i})$. It can be seen that the deviations are extremely small for the variation of the drift angle. When varying the heel angle, it leads to higher deviations from the target heel angle and rather small deviations from the target drift angle of 0° .

5.2.1. Impact of drift on ship forces When evaluating the added wave resistance for the four different waves and varying the drift angle in the previously described manner, the results depicted in Fig. 12 can be obtained. Figure 12 shows for all four, investigated waves the added wave resistance coefficient C_W with respect to the varying, time-averaged drift angle. It can be seen that the influence of the drift angle on C_W changes with the wave length. So, for the longest wave at $\frac{\lambda}{L_{pp}} = 2.0$, the added wave resistance coefficient is changing only by a small absolute amount while it is significantly decreased for the waves with $\frac{\lambda}{L_{pp}} = 1.4$ and $\frac{\lambda}{L_{pp}} = 1.1$. Exclusively for the shortest wave, the added wave resistance coefficient increases when reaching around 10° . The relative change in C_W is depicted in Fig. 13. For all waves C_W decreases by around 3% to 7% at a drift angle of 5° . Besides for the shortest wave, this trend continues so that a decrease of up to 10% can be found for the two longest waves at a drift angle of 10° . At the same time, the further increase in drift angle leads to a lesser decrease or even increase of C_W for the two shortest waves which indicates a reversing trend with decreasing wave length. So overall, an increased drift angle leads to a reduced added wave resistance, although high drift angles seem to increase the added wave resistance for short waves.

The mean wave side force and the mean wave yaw moment are presented in Fig. 14. The side force shows a consistent trend towards an increased magnitude with increasing drift angle. Although the values are rather small compared to the calm water drift force, they would add up, increasing the total side force acting on the drifting *KVLCC2* in waves. For the mean wave yaw moment the trend is not that distinct. It appears that the moment is slightly increasing for smaller drift angles but decreases again with further increasing drift angle. In any case, the change in mean wave yaw moment is small compared to the calm water

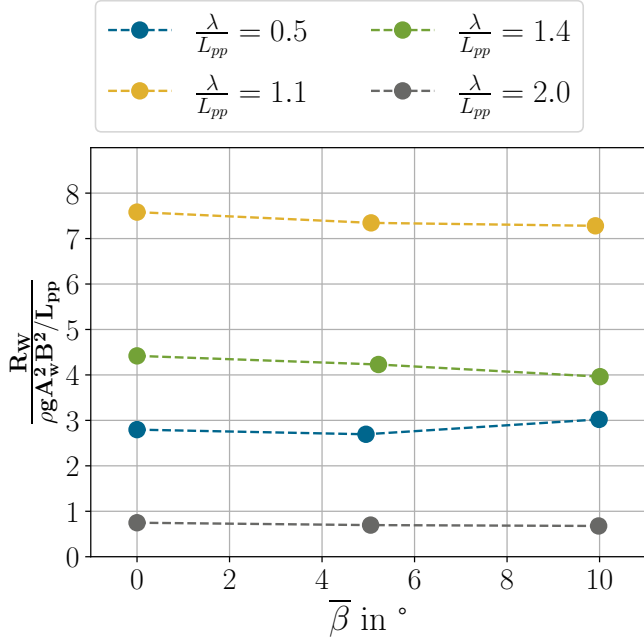


FIGURE 12: Added wave resistance coefficient at different drift angles

drift moment.

5.2.2. Impact of heeling on ship forces The influence of the heeling on the added wave resistance is illustrated in the same way as for the drift study. For all simulated waves and heel angles, the added wave resistance coefficient is shown in Fig. 15, while Fig. 16 depicts the relative change in C_W with increasing heel angle. For the longest wave, the absolute change in the added wave resistance coefficient is almost negligible as even a heeling of 10° does not significantly influence the absolute value. However, the influence of the heel angle is significantly higher for all the other, shorter waves. From Fig. 16 it can be seen that the added wave resistance coefficient decreases by around 2-5% for the two longest waves at $\frac{\lambda}{L_{pp}} = 2.0$ and $\frac{\lambda}{L_{pp}} = 1.4$. For the two shorter waves, instead, C_W increases by 0–2% at a heel angle of 5° and by 6–7% at a heel angle of around 10° . Therefore, we can see differing behaviour of the added wave resistance for the shorter and longer waves. Similar to the effect of drift, heeling leads to a decrease in C_W for the longer, investigated waves and to an increase for the two shorter waves.

With respect to the mean wave side force and yaw moment, Fig. 17 shows the effect of heeling on these. The influence on the side force is extremely small. Nevertheless, there is a clear trend towards an increased side force with increasing heel angle for all waves, except the shortest wave. In contrast to that, the mean wave yaw moment is significantly increased showing an almost linear trend for all investigated waves. The moment resulting from the shortest waves is small for all heel angles while it increases with increasing wave length reaching its maximum for the wave at $\frac{\lambda}{L_{pp}} = 1.4$. With a maximum mean wave yaw moment of over 35N, the effect of the heel angle in waves surpasses that under calm water conditions. However, as the added moment due

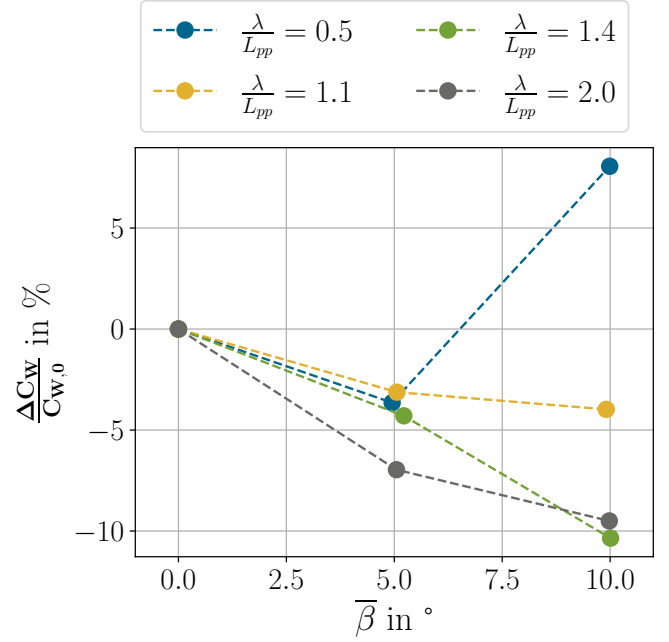


FIGURE 13: Relative change in added wave resistance coefficient for different drift angles

to heeling under calm water conditions (cf. Fig. 10) is acting opposing to the one due to waves, the effect of heeling on the total yaw moment becomes small. So, both, total side force and total yaw moment, are influenced little by heeling.

6. CONCLUSION

In this paper, the influence of drift and heeling was investigated by means of high-fidelity CFD methods to assess the impact of these parameters, which define the sailing conditions, on a ship's resistance. For that, the well-known *KVLCC2* test case was harnessed and simulation results for the calm water resistance and the added wave resistance were validated with existing experimental data, showing convincing agreement between experiments and CFD. Based on the validated setup, calm water simulations were conducted for varying drift and heel angles. The results of these simulation showed that the influence of the drift angle on the resistance is small for the investigated range between 0° and 10° which is in line with existing results. Nevertheless, drift inflicts a substantial side force and yaw moment. In contrast, the side force due to heeling is negligible and the yaw moment resulting from heeling is around one order of magnitude smaller. Moreover, it was shown that the influence on resistance force is small. As the investigated range of drift and heel angles already covers rather extreme operating conditions for a ship with WAPS, it can be concluded that under calm water conditions the main effects would result from the drift of the ship. More precisely, the most relevant effects are an increased side force that helps counteracting the side force from the WAPS and a yaw moment which would require active counter measures like rudder deflection that would increase total resistance. On top of that, a small increase in resistance is to be expected from heeling and drift. The results

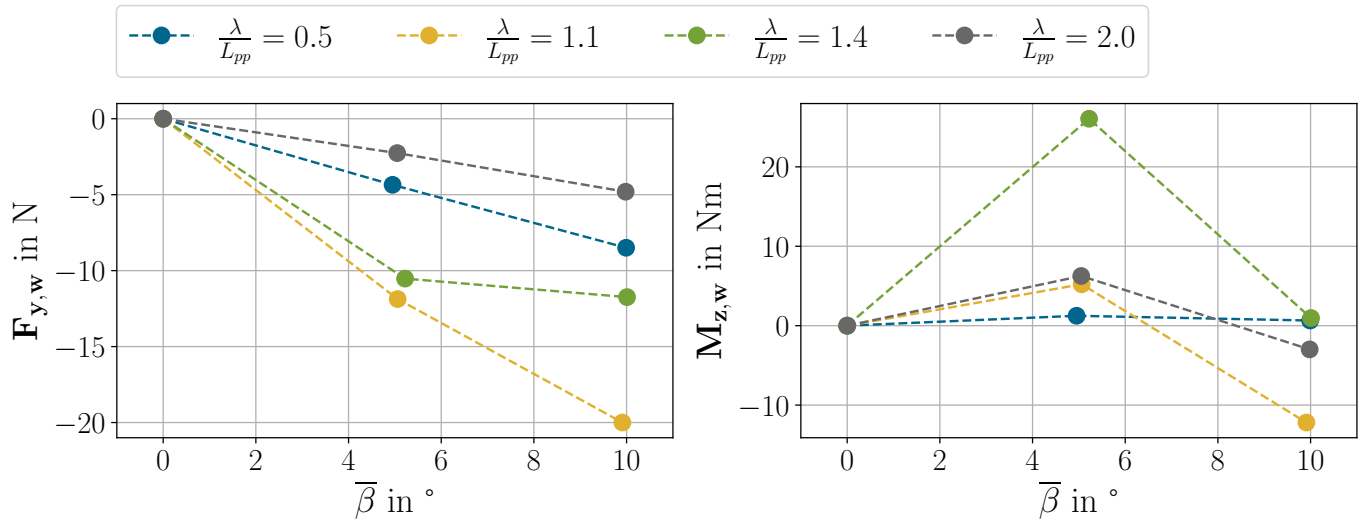


FIGURE 14: Mean wave side force and yaw moment at different drift angles

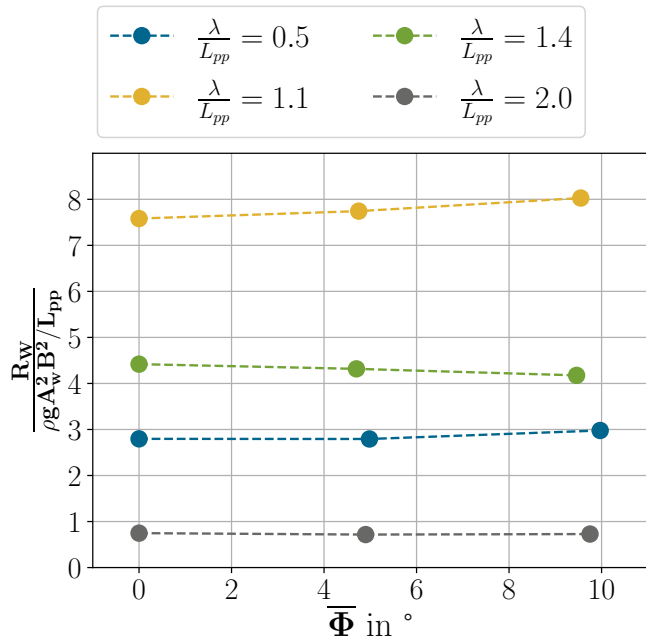


FIGURE 15: Added wave resistance coefficient at different heel angles

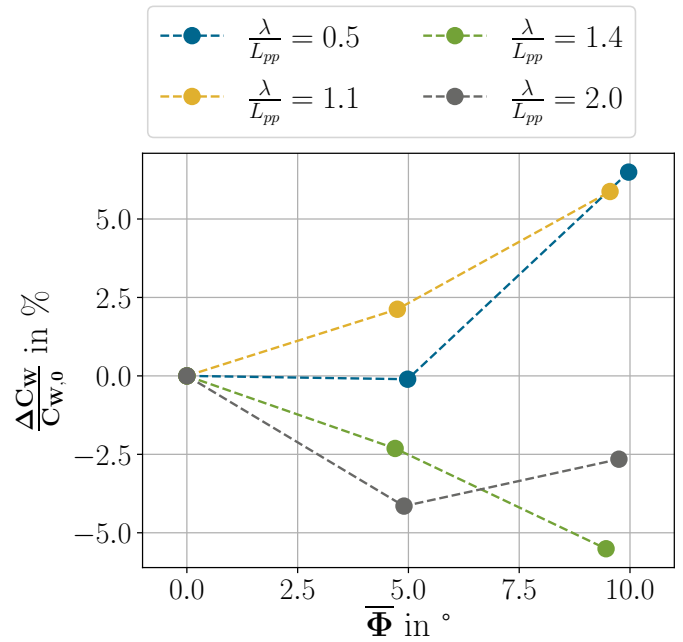


FIGURE 16: Relative change in added wave resistance coefficient for different heel angles

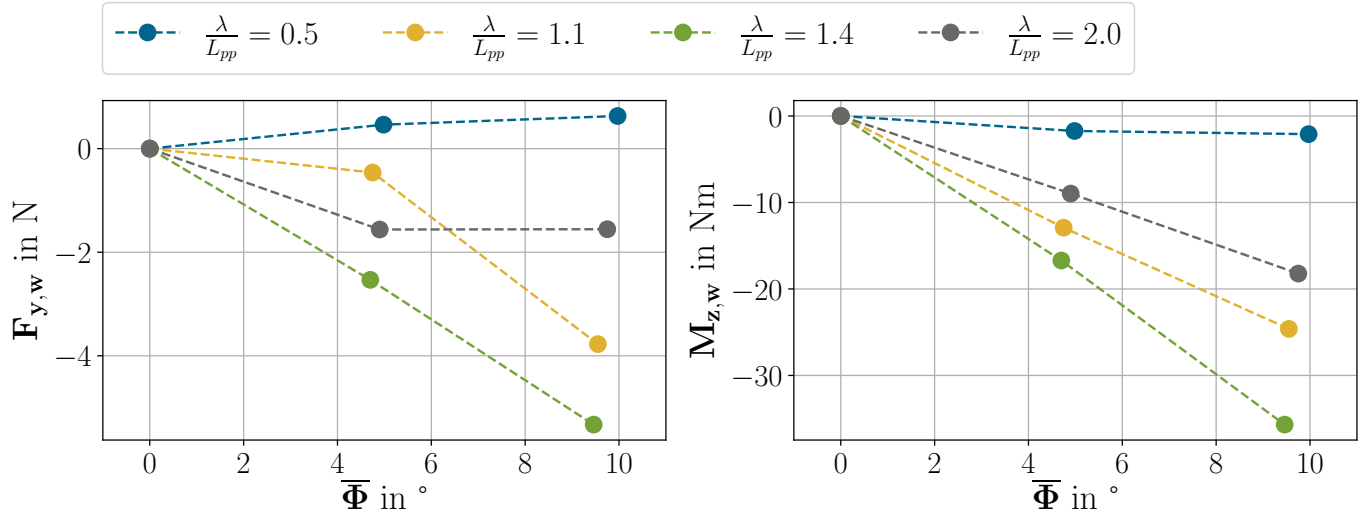


FIGURE 17: Mean wave side force and yaw moment at different heel angles

for the influence of drift and heeling on the mean wave forces, including added wave resistance, depend on the wave parameters. It was shown that the added wave resistance decreases for the longer waves when the ship is drifting or heeling. Whereas the results for the shortest wave show a trend towards increased wave resistance with increasing heel angle and for a high drift angle. The mean wave side force is rather increased by drift than by heeling, thus adding up to an increased total side force. In contrast to that, the mean wave yaw moment due to heeling is counteracting the calm water yaw moment. At the same time, the effect of drift on the mean wave yaw moment is rather small which indicates that the total yaw moment would be influenced mostly by the yaw moment from drift in calm water. Within a realistic scenario for the KVLCC2 with WAPS, sailing with a drift and heel angle would lead to a slightly increased calm water resistance, while the added wave resistance would increase in a sea state where waves dominate that are shorter than the ship. For the KVLCC2 in the North Atlantic, the added wave resistance would be dominated by these short waves which is why further work is required to assess the quantitative influence of the sailing condition in realistic seaway.

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