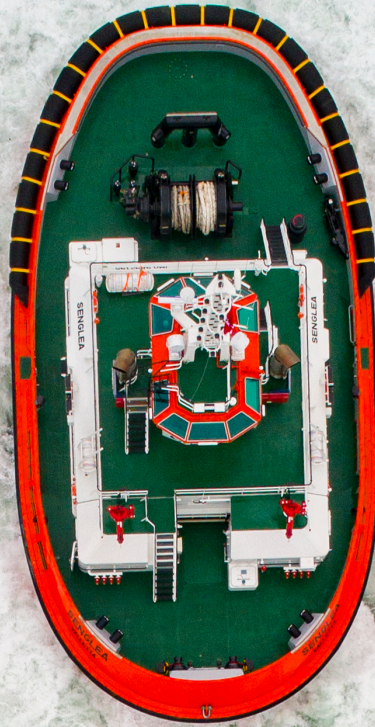


# Assessment of the Dynamic Stability of Tugs in Escort Using CFD

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Thesis for the degree of MSc in Marine Technology in the specialization of Ship  
Hydromechanics

# **Assessment of the Dynamic Stability of Tugs in Escort Using CFD**

By

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Performed at

Damen Shipyards

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# 1 Abstract

**Note on confidentiality:** This is the public version of this thesis for the repository of TU Delft. Important data about the RSD 2513 Damen tug and its performance are left out.

In this thesis, the influence of a combined drift angle, heeling angle and forward speed on the stability of a tug is researched. This is done to gain insight in the stability of tugs during escorting, where they operate at high drift angles, heeling angles and forward speed. The Damen RSD 2513 tug was simulated in steady CFD simulation using the software FINE/Marine. The simulations were performed for 0, 15, 30 and 45 degrees drift angle, 0 to 10 degrees heeling angle for every 2 degrees of heel, and at 8 knots forward speed. Curves of the righting moment and heeling moment were created with the results of these simulations. These curves were analysed to see the effect of the drift angle, heeling angle and forward speed. Furthermore, a comparison was made between the restoring moment of the RSD 2513 calculated by SARC PIAS. The definition of the righting-, heeling- and restoring moment in this thesis is as follows. *stat* refers to a hydrostatic- and *dyn* to a hydrodynamic force or arm.

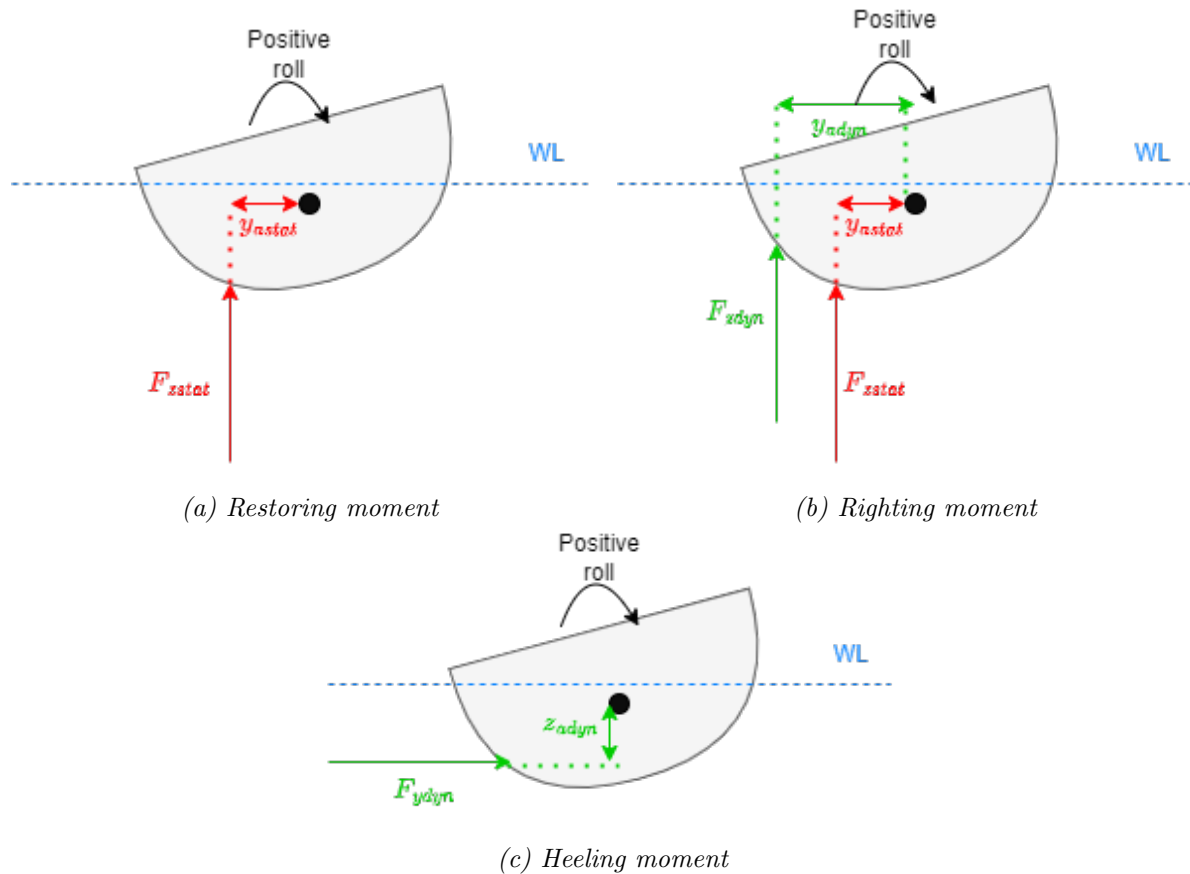


Figure 1: Schematic of the definition of the restoring-, righting- and heeling moment

The following conclusions were drawn from the righting moment:

1. Without a drift angle, forward speed doesn't influence the righting moment significantly. This results a minimal difference between the restoring and the righting moment of the tug.
2. Introducing a drift angle will result in a decrease of the righting moment.
3. Even with a drift angle, increasing the heeling angle will result in an increasing righting moment. In fact, the heeling angle and drift angle can be assumed to have independent

effects on the righting moment.

Furthermore, two conclusions were drawn from the heeling moment:

1. The heeling moment increases when a drift angle is introduced.
2. Increasing the heeling angle will result in a slight increase in the heeling moment.

There are however concerns about the result's numerical uncertainties and convergence. Firstly, the error estimation in the verification exercise is unreliable and therefore has a large safety factor leading to high uncertainties. Secondly, issues with convergence were experienced for higher drift angles. This was caused by large separation areas with high turbulent kinetic energy. Because of these two things, the results have limited reliability and the conclusions should be viewed with scepticism.



## 2 Preface

Firstly, I would like to thank Damen Shipyards for the opportunity to do my thesis at their company and for providing me with all the resources I needed. It was a great experience with many highlights. All the members of the hydromechanics research team have helped me enormously with tackling a topic that I knew so little about. A special thanks goes out to Renske Marijnissen. As my daily supervisor she was always there to patiently answer the millions of questions I had and to give me a little push in the right direction when I needed it. Scott Terry also deserves credit for this. Not only as team leader who gave some extra advice when necessary, but also as stand-in for Renske when she was off for a month. Secondly, I want to thank Tomasz Gornicz from Marin. Whenever there was a CFD problem we didn't know how to fix, a quick call with Tomasz was enough to resolve the issue we had struggled with for days. And finally, I want to thank Tom van Terwisga for all the expertise and advice he brought to the table. Even though we didn't meet often, he made sure I had enough capable supervisors to guide me through this process. At meetings, his insight often showed the way forward, and with his positive mindset he knew how to motivate me till the end.

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# Nomenclature

## Greek symbol

|                    |                                               |
|--------------------|-----------------------------------------------|
| $\beta$            | Drift angle                                   |
| $\Delta$           | Displaced volume                              |
| $\Delta_\phi$      | Data range parameter                          |
| $\epsilon$         | Dissipation of the turbulent kinetic energy   |
| $\epsilon_\varphi$ | Discretization error                          |
| $\gamma$           | Growth ratio                                  |
| $\lambda$          | Scaling factor                                |
| $\mu$              | Dynamic viscosity                             |
| $\nu$              | Kinematic viscosity                           |
| $\omega$           | Specific dissipation                          |
| $\phi$             | Heeling angle                                 |
| $\phi_0$           | Estimated exact solution of the flow quantity |
| $\phi_i$           | Flow quantity of a grid $i$                   |
| $\phi_{Exp}$       | Flow quantity measured in model test          |
| $\phi_{fit}$       | Flow quantity estimated with a regression     |
| $\rho$             | Density                                       |
| $\sigma$           | Standard deviation                            |
| $\tau$             | Trim                                          |
| $\tau_w$           | Shear stress                                  |
| $\varphi$          | Pressure due to body forces                   |

## Roman symbol

|                 |                 |
|-----------------|-----------------|
| $\underline{u}$ | Velocity vector |
| $dyn$           | Dynamic         |
| $stat$          | Static          |



|           |                                           |
|-----------|-------------------------------------------|
| $A_{lat}$ | Lateral area of the ship                  |
| $B$       | Maximum breadth of the ship               |
| $B_{wl}$  | Breadth moulded on water line             |
| $C_D$     | Drag coefficient                          |
| $C_K$     | Roll coefficient                          |
| $C_L$     | Lift coefficient                          |
| $C_X$     | Non-dimensional $X$ force                 |
| $C_Y$     | Non-dimensional $Y$ force                 |
| $d_i$     | Initial cell height in the boundary layer |
| $d_m$     | Mean draft                                |
| $E$       | Comparison error                          |
| $F_i$     | Force in direction $i$                    |
| $F_n$     | Froude number                             |
| $F_S$     | Safety factor                             |
| $GZ$      | Restoring moment arm                      |
| $h_i$     | Typical relative cell size                |
| $k$       | Turbulent kinetic energy                  |
| $L$       | Ship length                               |
| $l$       | Length scale of turbulence                |
| $L_{pp}$  | Ship length between perpendiculars        |
| $L_{wl}$  | Ship length moulded on water line         |
| $M_H$     | Heeling moment                            |
| $M_i$     | Moment around axis $i$                    |
| $M_R$     | Righting moment                           |
| $n_g$     | Number of grids                           |
| $N_i$     | Number of cells in a certain direction    |

|             |                                                                    |
|-------------|--------------------------------------------------------------------|
| $p$         | Static pressure                                                    |
| $Re$        | Reynolds number                                                    |
| $r_i$       | Refinement ratio                                                   |
| $T_A$       | Draught moulded on aft perpendicular                               |
| $T_F$       | Draught moulded on forward perpendicular                           |
| $U$         | Ship speed in x direction                                          |
| $u^+$       | Non-dimensional velocity parallel to the wall                      |
| $U_\phi$    | Numerical uncertainty                                              |
| $U_t$       | Velocity parallel to the wall                                      |
| $u_\tau$    | Shear velocity                                                     |
| $U_{Exp D}$ | Experimental uncertainty of the drag force                         |
| $U_{Exp K}$ | Experimental uncertainty of the roll moment                        |
| $U_{Exp L}$ | Experimental uncertainty of the lift force                         |
| $U_{Exp X}$ | Experimental uncertainty of the $X$ force                          |
| $U_{Exp Y}$ | Experimental uncertainty of the $Y$ force                          |
| $U_{Exp}$   | Model test uncertainty                                             |
| $U_{Val}$   | Validation uncertainty                                             |
| $V$         | Ship speed in y direction                                          |
| $V_S$       | Ship speed                                                         |
| $y^+$       | Wall unit                                                          |
| $y_a$       | Distance in y direction between COG and application point of force |
| $z_a$       | Distance in z direction between COG and application point of force |

### 3 Introduction

**Note on confidentiality:** *This is the public version of this thesis for the repository of TU Delft. Important data about the RSD 2513 Damen tug and its performance are left out.*

In recent years the size of merchant vessels has increased and with that the operational requirements for tugs have become more demanding. The required bollard pull, and therefore installed power, has increased as well [19]. Due to this increase, the risk of self-tripping, where a ship capsizes due to a combination of towline pull and steering forces by the thrusters, has increased significantly. Partly due to this, the number of tug capsizing events is persistently high [20]. It is therefore somewhat surprising that there were no international intact stability criteria for towing and escorting until 2016. These criteria all concern the restoring moment lever, GZ, and the heeling lever. The heeling lever is based on the bollard pull of the tug and the distance between the towing point and the centre line of the propeller. It therefore represents the heeling lever due to the towline forces and propeller forces. GZ is the lever for the restoring moment due to the hydrostatic buoyancy force which is created when a ship is heeled at 0 forward speed. However, there is reason for concern about these criterion since with the hydrostatic restoring moment, they do not consider the hydrodynamic forces acting on the tug. Escort tugs operate in very dynamic situations with a relatively high forward speed, drift angles and heeling angles combined. The effect of these factors on the operational stability of the tug could therefore be significant.

As mentioned, the stability requirements are all concerning the hydrostatic restoring moment that is only dependent on the heeling angle. However, according to Kianejad [22], hydrostatic approximations of the restoring moment overlook the dynamic conditions and the effects of pressure distributions over the hull. De Jong [21] explains that stability for tugboats is connected to speed because of hydrodynamic forces which can be generated when sailing. Furthermore, forward velocity and drift angle will create waves around a hull and the hydrostatic restoring arm GZ is determined assuming undisturbed water. Caraddu [7] shows that waves have an effect on the GZ of a ship. All this is reason to believe that forward velocity influences the stability of a tugboat significantly. It is also evident that tugboats operate under large drift angles during escort. According to Islam [17], forces on the hull in the ship's transverse direction increase with increasing drift angle. Since this force would create a roll moment, it affects the ship's transverse stability. Ommani [12] also states that drift angle and forward speed are of interest in dynamic stability of ships. Especially for escort tugs, that operate under high drift angles, it could therefore be argued that the drift angle should be taken into consideration when assessing the stability of a tug.

These arguments all suggest that assessing the stability of an escort tug solely through the restoring moment and heeling moment created, is not sufficient. Only the hydrostatic forces are taken into account with the restoring moment, and the effect of hydrodynamic forces on the stability of the tug are unknown. These hydrodynamic forces are created due to a combination of forward speed, heeling angle and drift angle. That is why, in this research, the effect of these factors on the stability of escort tugs will be investigated. This will be done by reviewing their effect on the righting- and heeling moment. The research question that will be answered is:

*To what extent are the practical definitions of the righting moment and heeling moment of a tug dependent on forward speed, heeling angle and drift angle?*

#### 3.1 Research methodology

The goal of this research is to investigate the effect of the heeling angle, drift angle and forward speed on the stability of an escort tug. In order to do this, the forces and moments around the tug will need to be known at forward speed, drift angles, and heeling angles. These forces and moments were calculated using the CFD software FINE/Marine. A series of simulations were run

in the software at a speed of 8 knots. The ship simulated was the Damen RSD 2513. These simulations were run at combinations of drift- and heeling angles. The heeling angles would vary between 0- and 15 degrees, and the drift angles between 0- and 45 degrees. From the results of these simulations, the righting moment and heeling moment around the ship were calculated. The righting- and heeling moment were then plotted against the heeling angle and the drift angle. These results were used to draw conclusions on the effect of the forward speed, heeling angle and drift angle on the stability of escort tugs. Furthermore, a grid convergence study was performed to estimate the numerical uncertainty of the simulations. With these uncertainties and model tests for the RSD 2513 at drift, a validation exercise was performed.

### 3.2 State of the art

Important work to consider for this research is the thesis of Vink [36]. Vink researched the validation and verification of an RSD 2513 tug at drift. Furthermore, he compared multiple turbulence models using steady- and unsteady simulations. Due to the similarity between his research and this research, his work is incredibly useful as a reference. However, Vink does not include heeling angles in his simulations and the focus of his work is on verification and validation, whereas this research's main focus is on the effect of different factors on the stability of the tug.

Schot [31] also did similar research. He verified and validated drift simulations for the RSD 2513 as well. He compared different turbulence models, as well as different wall treatment. His work differs from Vink's by the fact that he only uses steady simulations. Again, the major difference between his work and this research is that no heeling angles were simulated in his work and that the focus of his research is towards verification and validation.

### 3.3 Report outline

This report begins with a literature review. In this review the background of some aspects of the CFD simulations will be given to give insight in decisions made in this research. Furthermore, theory about the computation of discretization errors and regression methods will be given. Afterwards, the numerical setup of the simulations will be discussed. In this setup, decision in the meshing process will be discussed as well as simulation settings. Furthermore, the decision between model scale and full scale CFD and the convergence of the results will be reviewed. After reviewing the setup of the simulations, the verification process will be discussed. This chapter will contain an estimation of the error and the numerical uncertainty of the simulations. In the subsequent section, these results will be used to validate the meshes. Furthermore, the model test and its uncertainties will be discussed. Afterwards, the results of the simulations will be presented. A discussion of these results will follow, after which conclusions for this research will be drawn. Finally, a set of recommendations will be made for the classification and design of escort tugs, as well as future research into this subject.



## 4 Literature review

The literature review for this research focuses on providing background information for some important decision made in the numerical setup of this research. Furthermore it strives to give insight in how the numerical uncertainties can be computed. This literature review contains seven parts. The first part gives background to wall treatment and reviews the findings of other research concerning high drift angle CFD simulations. The goal is to have enough information choose the wall treatment applied in the simulations used in the results. In the next section, turbulence models are discussed. The way turbulence is modelled can have a large impact on the outcome of the simulations. It is therefore important to analyse the options to explain why the model used in this research was chosen. Afterwards, adaptive grid refinement (AGR) will be discussed. This section aims to inform about the decision to use AGR and what the benefits are. The next section strives to inform about the data set that will be created with the simulations. It gives multiple options how to fill a simulation matrix and gather the appropriate data points to create the results for this thesis. Next, it is discussed how the numerical uncertainties for the simulations are calculated in the verification study in subsection 6.1. Afterwards, regression methods are discussed to inform about which method was used to generate the righting- and heeling moment curves in subsection 8.1 and why. Lastly, some background in the main difference between steady- and unsteady simulations is explained. This is because in subsection 5.7, a comparison will be made between steady- and unsteady simulations in terms of convergence of the simulations, results and computational cost.

### 4.1 Wall treatment

In the past it has proven difficult to simulate tug boats at high drift angles due to the difficulties with accurately simulating separating flow in the boundary layer. Therefore, two important aspects to model the separation in the boundary layer are discussed in this literature review. The first one is the wall treatment. This answers how the flow in these boundary layers at the ship hull is modelled. The second one is what turbulence model is used and will be reviewed in subsection 4.2

As discussed by Eça [11], there are two main approaches to determine the shear stress at a wall: implementing the wall resolved boundary condition, or using a wall function. The wall resolved condition is physically preferable but may lead to high computational costs and convergence problems. This is because the wall resolved condition will lead to extremely small cell size near the wall which leads to a higher total number of cells. Wall functions on the other hand use semi-empirical equations to describe the velocity profile along the body. These functions may be less accurate, but are less computationally heavy. Eça concludes that wall functions may be useful to model completely turbulent flows around complex geometries. However, he states that it is still only a numerical advantage over the direct wall resolved condition. Finally, wall function boundary conditions should be avoided if friction forces are dominant.

An important parameter when talking about wall treatment is  $y^+$ . It is determined as shown in Equation 1 [11].  $u_\tau$  is the shear velocity,  $\tau_w$  the shear stress and  $\rho$  the density.  $y$  is the absolute distance to the wall,  $\nu$  is the kinematic viscosity. Two different examples of this situation are shown in Figure 2 [24].  $y^+$  is a non-dimensional value often referred to as wall unit. It is used to determine the appropriate cell size near the wall. A low  $y^+$  indicates that the vertical distance of the cells near the wall is small and vice versa. In Figure 2 the  $y^+$  value of the left picture is higher than that of the right picture. As a result of that the vertical distance of the cells centres to the wall is bigger and the cells size is bigger as well. The shear stress is determined as shown in Equation 2 if the near wall cell size is sufficiently small ( $y^+ < 1$ ). This is the case for the direct wall resolved condition where  $y^+ \simeq 1$ . In this equation  $\mu$  is the dynamic viscosity and  $U_t$  is the velocity parallel to the wall. The subscript  $c$  means that it is the cell centre value. For wall functions,  $\tau_w$  is determined differently than shown in Equation 2. This can be done with multiple different

semi-empirical equations. Wall function meshes have a significantly higher  $y^+$  value than 1. The requirement that  $y^+ < 1$  could lead to difficulties with finding iterative convergence which creates the incentive to use a wall function instead and increase the near wall cell size. [11]

$$y^+ = \frac{u_\tau y}{\nu}$$

$$u_\tau = \sqrt{\frac{\tau_w}{\rho}}$$
(1)

$$\tau_w = \mu \frac{\partial U_t}{\partial y} \simeq \mu \frac{(U_t)_c}{y_c}$$
(2)

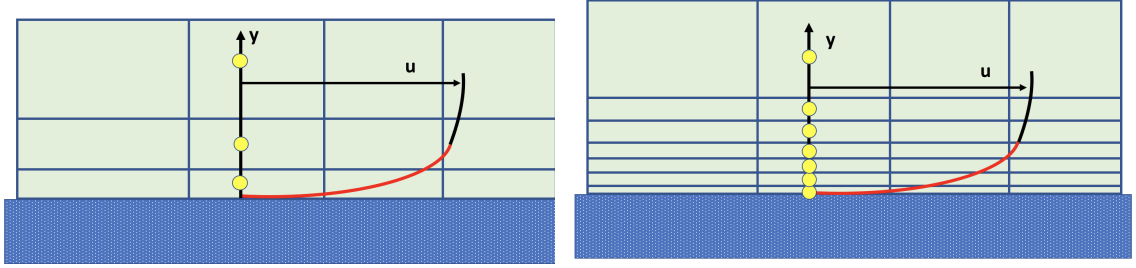


Figure 2: Schematic of the velocity ( $u$ ) profile at a wall for two different  $y^+$  values [24]

The reason why  $y^+ \simeq 1$  for the wall resolved condition is that it should be able to accurately calculate the velocity gradient on the wall. Therefore the mesh should be sufficiently refined there. Wall functions don't require such a small  $y^+$  since they use semi-empirical laws to estimate this gradient. These semi-empirical laws are shown in Figure 3. The figure shows the non-dimensional velocity,  $u^+$ , plotted against the wall unit. In Figure 3,  $u^+$  is the non-dimensional velocity calculated as shown in Equation 3. The red line is the actual velocity profile we try to imitate with the wall function. The blue lines are the wall functions. As is shown in that figure, the wall function changes somewhere in the buffer layer in order to continue to estimate the velocity profile accurately. [24]

$$u^+ = \frac{u}{u_\tau}$$
(3)

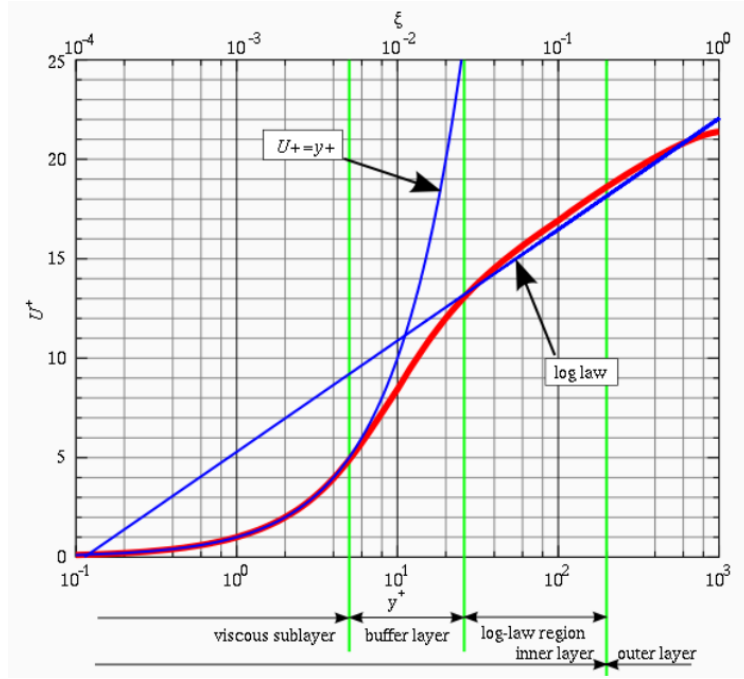


Figure 3: Non-dimensional velocity against  $y^+$  [24]

A drift simulations manual from Damen [30] states that the  $y^+$  value should be set to 1.0 for drift simulations. This means that wall resolved condition is used. It is stated that this approach is preferred because of the turbulence model. In the work of Lungu [23] and Vink [36], the wall resolved treatment is used as well. Lungu did this for large drift angle simulations with a submarine model and Vink for drift simulations with the RSD 2513 tug. In both cases the wall resolved treatment was preferred as they expected it to capture flow separation more accurately. Flow separation occurs heavily on the RSD tug at large drift angles as illustrated in Figure 4. In this figure, the purple parts represent the flow separation.

However, an argument for using wall function was found by Schot [31]. He researched the RSD 2513 tug at full scale at a number of constant drift angles. When he compared the results of wall resolved- and wall function simulations to validation material, he did not find any improvement.

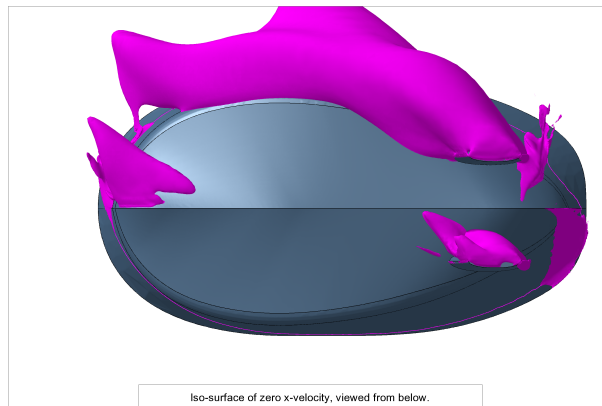


Figure 4: Flow separation on a tug with skegs at 45 degrees drift angle

## 4.2 Turbulence models

As discussed in the previous section, flow separation has made modelling tugs at high drift angles difficult. Therefore, two aspects of modelling the separation in the boundary layer should be

discussed. The first one was wall treatment and was discussed in [subsection 4.1](#), the second one is the turbulence model. Turbulence models can have a major influence on how well turbulence, flow separation and therefore forces on the hull are predicted. This is especially relevant for tug boats during escort as the bluff body and high drift angles are known to create a lot of turbulence and flow separation. There are three ways to solve turbulence in a CFD simulations: Direct Numerical Simulation (DNS), Large Eddy Simulation (LES) and Reynolds Averaged Navier-Stokes (RANS). DNS solves the Navier Stokes equations directly throughout the entire domain. This is very accurate but extremely expensive and computationally heavy. For this reason it is almost never considered in CFD. LES only solves the large scale motions of the turbulent flow directly. LES is less expensive than DNS but still expensive compared to RANS. RANS uses a Reynolds averaged version of Navier-Stokes to solve the turbulent flow in all length scales. This results in a lower computational cost than LES but RANS also tends to be less accurate. The turbulence models that can be used in FINE/Marine are either RANS based, or a combination between LES and RANS. [38](#) [36](#)

Vink [36](#) compared multiple turbulence models for such a case. He used 4 different strategies to solve the turbulence:

- $k-\omega$  SST 2003 model
- Menter's  $k-\sqrt{k}l$  model
- Explicit Algebraic Reynolds Stress Model (EARSM)
- Delayed Detached Eddy Simulations (DDES)

Models such as the  $k-\sqrt{k}l$  model,  $k-\omega$  SST model, and also the EARSM model are RANS based models and try to close the RANS equations by modelling all length scales with RANS. DES (Detached Eddy Simulations) and DDES are a combination between a RANS approach and a LES approach.

$k-\sqrt{k}l$  and  $k-\omega$  are called two-equation models. This is because they use two transport equations to solve two turbulence properties. One of these properties is the turbulent kinetic energy,  $k$ . For the  $k-\sqrt{k}l$  model, the square root of this property is taken and then multiplied by the length scale of the turbulence,  $l$ , to create the property  $\sqrt{k}l$ . For the  $k-\omega$  model  $\omega$  is used, which represents the specific dissipation rate of the turbulence. This can be seen as the turbulence frequency of large eddies. The  $k-\omega$  SST model is different from the  $k-\omega$  model in that it is a blend between the  $k-\omega$  model and the  $k-\epsilon$  model. In the  $k-\epsilon$  model,  $\epsilon$  represents the turbulent dissipation rate of the kinetic energy.  $k-\epsilon$  is used in the free stream flow, and  $k-\omega$  is used near the wall. [38](#) [36](#)

The EARSM model is very similar to the  $k-\omega$  model. The EARSM model is based on the  $k-\omega$  TNT model. This is a version of the  $k-\omega$  model that differs from the  $k-\omega$  SST model by computing the eddy-viscosity slightly different. For the EARSM model, an extra anisotropy tensor is added. This differs from the  $k-\omega$  SST model that has isotropic eddy-viscosity. [35](#)

The DDES model models the near wall region with RANS and the outer- and detached regions with LES. The advantage of using LES is that with LES some turbulence length scales are solved instead of modelled and averaged as is the case with RANS. In theory, this should increase the accuracy. DDES is different from DES because it has an explicit switch between RANS and LES formulations. Because DES doesn't have this, the grid for this model should carefully crafted. [35](#) [36](#)

In his research, Vink finds that the  $k-\sqrt{k}l$  model has the lowest comparison error compared to EFD. However, he concludes that uncertainties are so high that no conclusion could be drawn from the results on which turbulence model had the best performance. Other researchers such as Lungu

[23], Broglia [5] and Guilmineau [14] have also investigated turbulence models for drift simulations. These three all concluded that combined LES and RANS models performed better than RANS based models in terms of accuracy. However, this comes with an increase computational cost. In subsection 5.7 a simulation that uses the  $k-\omega$  SST model and one that uses the DDES model are compared in order to decide what type of model to use.

### 4.3 Free surface adaptive grid refinement

In subsection 5.3, it will be discussed how the grids will be generated and refined. This is important because certain areas in a grid need to be refined more than others to create a good result. These are areas where accurate results are needed and where certain flow phenomena need to be captured accurately. In other words, these are areas where interesting and important flow phenomena occur. One such area is the Kelvin wake of a ship on the free surface. A common way to capture the Kelvin wake is by adding a refinement area behind the ship in the shape of the wake. However, the position and shape of the kelvin wake changes for different drift angles. This is why for drift simulations within Damen, the Kelvin wake refinement is circular as shown in Figure 5 [30].

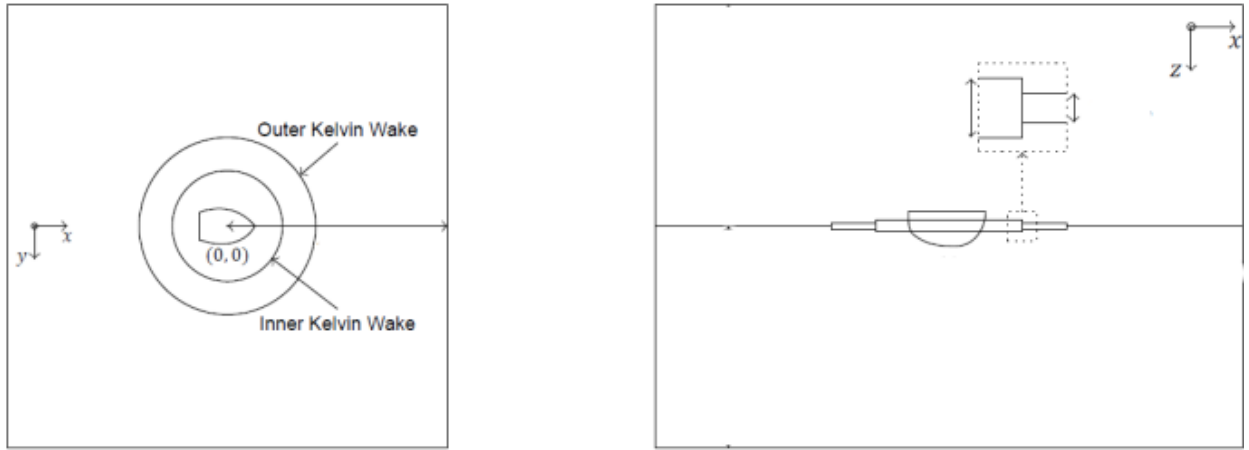


Figure 5: Kelvin wake refinement for drift simulations [30]

A drawback from using a full circle for the refinement is that a lot of this refined area isn't necessary during a simulation. It therefore adds a lot of cells without being useful a lot of the time. For this reason Adaptive Grid Refinement (AGR) is used in this research on the free surface.

AGR is a technique where a mesh is automatically refined during a simulation. For this research, AGR is used around the free surface. There are two reasons why this technique is used in this research: to reduce CPU time and to capture the free surface more accurately. Figure 6 shows the results of using AGR for free surface refinement against using normal free surface refinement when there is a heeling angle. As shown in Figure 6a, when the ship heels the mesh and free surface refinement deform. At one point, the heeling angle becomes too big and the refinement is not overlapping the free surface anymore. With AGR this problem is corrected since the mesh is specifically refined around the free surface during the simulation, as shown in Figure 6b. This will increase the accuracy around the free surface during the simulation.

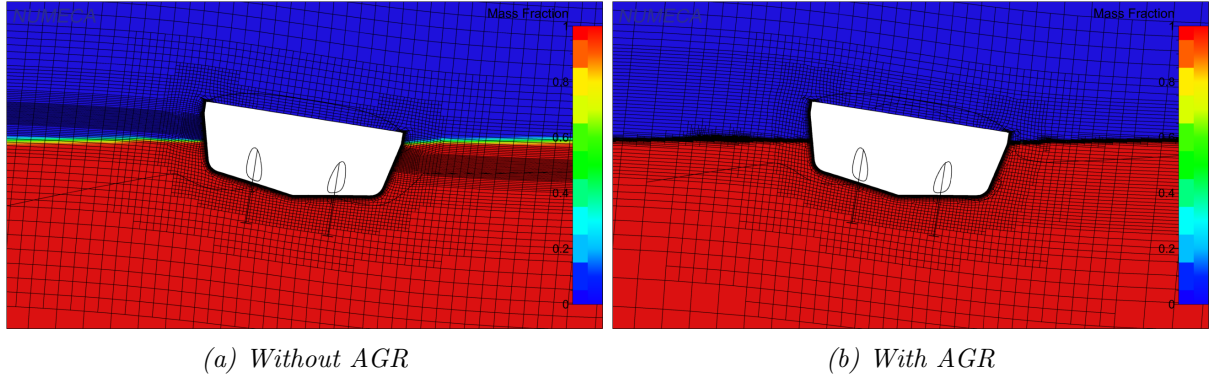


Figure 6: Free surface refinement with and without AGR

#### 4.4 Design of experiment

The objective of this thesis is to create a righting- and heeling moment curve in order to comment on the influence of the drift angle, heeling angle and forward speed on the stability of a tug. To create this curve, simulations will be run at different heeling- and drift angles to create data points for this curve. These data points can be simulated at many different angles and there are different approaches for choosing the location of these point. Therefore, the question in the Design of Experiment (DoE) is: How many data points will be needed and at which angles will they be simulated? A couple of approaches to chose the location of these data points are described in this section. subsection 5.5 provides information on what set of simulations were eventually run to create the results for this research.

To select the heeling- and drift angles for the simulations, there are multiple approaches. The first one is full factorial sampling. Here, the range of each parameter is divided in a number of intervals (uniform or non-uniform) and then all possible combinations are selected as points for the DoE as shown in Figure 7. A drawback of this technique is that the number of sampling points increases exponentially with the number of parameters. A strength of this approach is that it is a very structured approach. This could give insight into what exact combinations of input parameters causes a certain effect in the results of the simulations. For example, the simulations might not converge at a certain combination of a high drift angle and high heeling angle. Having a full factorial sampling space could give insight at which exact combination of heel and drift the simulation wouldn't converge anymore.

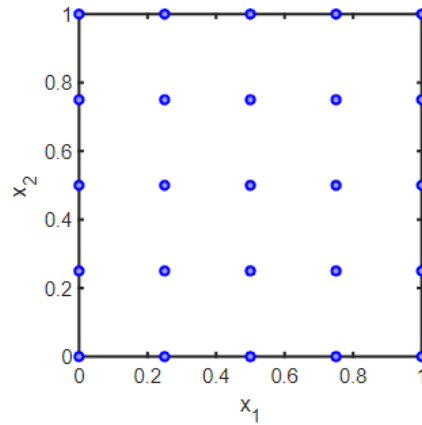


Figure 7: Example of full factorial sampling for two parameters [4]

Another approach is sparse grid sampling. This is a more efficient way of sampling and is therefore



more attractive for cases with a large number of parameters. One interpretation of this method uses a combination of the results of a high resolution full factorial DoE and a low factorial DoE to obtain a DoE where a number of sampling points are omitted in comparison to the high resolution full factorial DoE. This technique is shown in [Figure 8](#)

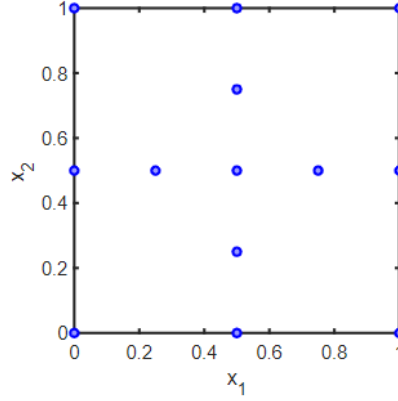


Figure 8: Example of sparse grid sampling for two parameters [\[4\]](#)

Lastly, there is the latin hypercube sampling (LHS) method shown in [Figure 9](#). This method starts with a pseudo-random DoE and then uses an optimising algorithm to maximise the quality of the DoE through the use of the so-called maximin algorithm criterion, which aims to maximise the minimal distance between two samples.

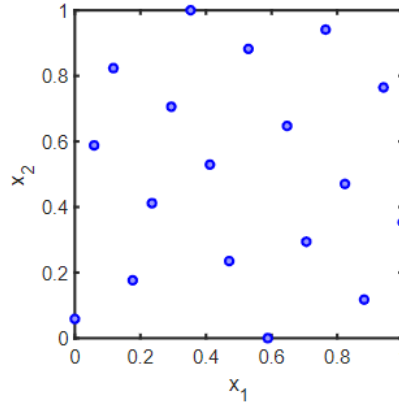


Figure 9: Example of latin hypercube sampling (LHS) for two parameters [\[4\]](#)

It was expected that some simulations at higher heeling angles would either crash or not converge due to the mesh deformation. Using the full factorial sampling would give more insight in when, and due to what parameters, the simulations will crash. Furthermore, there are only two parameters that are being varied: the heeling angle, and the drift angle. Full factorial sampling would therefore not suffer immensely from its biggest drawback. Therefore, that is the preferred approach.

#### 4.5 Theory on numerical uncertainty

In [section 6](#) a grid convergence study will be performed where the numerical uncertainty is calculated. The theory behind this convergence study will be discussed in this section and in [sub-section 5.8](#). This section contains a description of how the error and numerical uncertainty will be estimated.

The verification in this research will focus on solution verification, meaning that the goal is to quantify the numerical uncertainties,  $U_S$ , of the solution. For steady CFD simulations, the uncertainty consists of three parts: the discretization error, the iterative error and the round-off error. The round-off error can be neglected for so called double-precision solvers and is therefore neglected in this research. The iterative uncertainty is also often neglected as the magnitude of this error is often relatively small compared to the discretization error. According to Vink [36], this is correct if the magnitude of the iterative error is two or three orders below the discretization error. Since it is expected that the this error is so much smaller than the discretization error, it is decided to neglect it in this research as well. This means the numerical error will be equal to the discretization error. This error is calculated according to the Eça and Hoekstra method [10].

Equation 4 shows how the discretization error  $\epsilon_\varphi$  is calculated according to the Eça and Hoekstra method.  $\alpha$  and  $p$  will be determined using the least squares method.  $h_i$  is the typical relative cell size. There are four options here to determine  $\epsilon_\varphi$ . It depends on the values of  $p$  which equation will be used. Furthermore,  $\phi_i$  is any flow quantity of a specific grid  $i$ , and  $\phi_0$  is the estimated exact solution of this flow quantity. The flow quantities used in the verification process in section 6 are  $C_D$ ,  $C_L$  and  $C_K$ .

$$\begin{aligned}\epsilon_\varphi &= \delta_{RE} = \phi_i - \phi_0 = \alpha h_i^p \\ \epsilon_\varphi &= \delta_1 = \phi_i - \phi_0 = \alpha h_i \\ \epsilon_\varphi &= \delta_2 = \phi_i - \phi_0 = \alpha h_i^2 \\ \epsilon_\varphi &= \delta_{12} = \phi_i - \phi_0 = \alpha_1 h_i + \alpha_2 h_i^2\end{aligned}\tag{4}$$

Next, the data range parameter,  $\Delta_\phi$ , needs to be determined. A large data range parameter would indicate a large difference in a certain flow quantity between grids, for the number of grids. This can be calculated using Equation 5.  $n_g$  is the number of grids and  $(\phi_i)_{\max}$  and  $(\phi_i)_{\min}$  are a minimum and maximum value of a flow quantity.

$$\Delta_\phi = \frac{(\phi_i)_{\max} - (\phi_i)_{\min}}{n_g - 1}\tag{5}$$

This can then be used to determine the uncertainty. This is done in one of two ways depending on the standard deviation  $\sigma$ .  $\phi_{fit}$  is the value calculated with the least squares method for the specific grid corresponding with  $\phi_i$ .

For  $\sigma < \Delta_\phi$  :

$$U_\phi(\phi_i) = F_S \epsilon_\phi(\phi_i) + \sigma + |\phi_i - \phi_{fit}|\tag{6}$$

For  $\sigma \geq \Delta_\phi$  :

$$U_\phi(\phi_i) = F_S \frac{\sigma}{\Delta_\phi} (\epsilon_\phi(\phi_i) + \sigma + |\phi_i - \phi_{fit}|)\tag{7}$$

## 4.6 Regression methods

In subsection 8.1 the righting moment and heeling moment at different heeling- and drift angles were computed. The result was a scatter of data points. A polynomial regression was applied to create a response surface. Polynomial regression techniques use functions of multiple polynomials up to a certain order. Equation 8 shows the mathematical formulation of this polynomial [9].

$$y = a_0 + a_1x + a_2x^2 + \dots + a_nx^n\tag{8}$$

A polynomial regression is a very intuitive and easy to use method that can be used to describe a wide range of curves [4]. However, it is very sensitive to outliers in the data [29] [13]. Similar drawbacks are associated with radial basis function (RBF) techniques. However, this technique is not as intuitive and simple as polynomial regression. Kriging, on the other hand, is less sensitive to outliers but is a very complex technique. [13]

#### 4.7 RANS vs URANS

In subsection 5.7 an unsteady simulation (URANS) is run to compare the convergence with a steady simulation (RANS). In this literature section, the differences between the two methods will be briefly explained to give insight in why URANS was used as well as RANS.

As the names suggest, the main difference between steady- and unsteady simulations is the unsteady term in the RANS equations. This means that the velocity changes over time in unsteady equations. Equation 9 shows the steady equation and Equation 10 shows the unsteady one [3]. The difference between the two is indeed the derivative of the velocity over time,  $\frac{\partial \bar{u}_l}{\partial t}$ . This part is visualised in Figure 10. Here the velocity magnitude in a point within a turbulent flow is plotted for LES, URANS and RANS. Since URANS and RANS deliver an averaged solution and LES doesn't, URANS and RANS look smoother and less noisy in Figure 10. The reason why with URANS the solution is not a straight line as with RANS, is the unsteady component,  $\frac{\partial \bar{u}_l}{\partial t}$ , which causes the velocity to change over time. Because of this unsteady term, URANS is expected to be able to simulate unsteady flows more accurately than RANS. However, it comes with an increased computational cost as is discussed in subsection 5.7.

$$\bar{u}_j \frac{\partial \bar{u}_l}{\partial x_j} = -\frac{1}{\rho} \frac{\partial \bar{p}}{\partial x_i} + \nu \frac{\partial^2 \bar{u}_l}{\partial x_j^2} - \overline{u_j' \frac{\partial u_l'}{\partial x_j}} \quad (9)$$

$$\frac{\partial \bar{u}_l}{\partial t} + \bar{u}_j \frac{\partial \bar{u}_l}{\partial x_j} = -\frac{1}{\rho} \frac{\partial \bar{p}}{\partial x_i} + \nu \frac{\partial^2 \bar{u}_l}{\partial x_j^2} - \overline{u_j' \frac{\partial u_l'}{\partial x_j}} \quad (10)$$

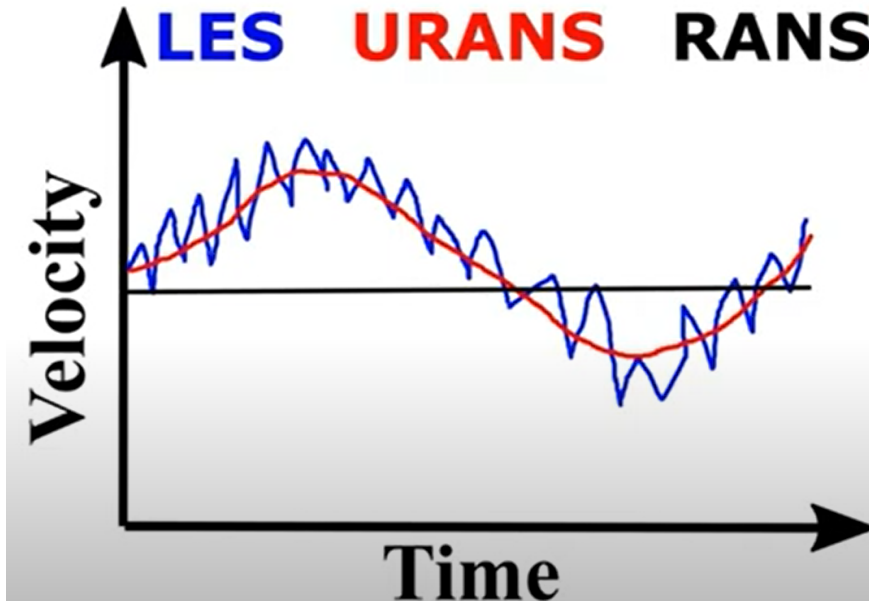


Figure 10: Velocity magnitude in a point within a turbulent flow [8]

## 5 Numerical Setup

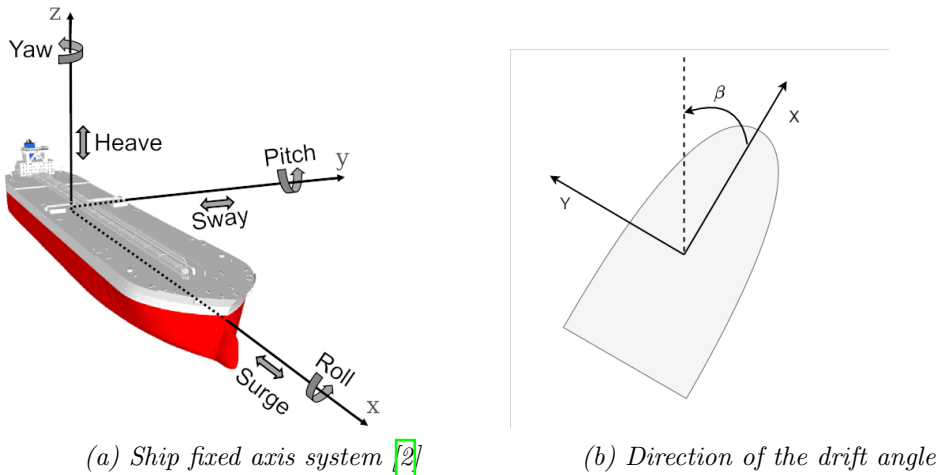
In [section 4](#) background information and theory behind important decision in this research were discussed. With this information choices could be made about the numerical setup of this research. This setup is discussed in this section. [subsection 4.1](#) [subsection 4.2](#) and [subsection 4.3](#) are used in this section to determine the modelling of the boundary layer, the turbulence and the free surface.

Before giving the numerical setup, it should be clear what software will be used in this research. For the creation of the mesh, the meshing software HEXPRESS is used. This software automatically creates unstructured meshes. Unstructured meshes are preferable for this research because it's easier to fit unstructured meshes around a complex structure like the RSD 2513. Furthermore, the software for the CFD simulations is FINE/Marine which uses the ISIS-CFD solver. FINE/Marine is commercial CFD software provided by Cadence.

In this section, firstly, the ship's reference system will be given. Afterwards, the generation of the initial mesh is discussed. Following this, it is explained how this mesh is refined to create the meshes for the verification study. After this, the simulation settings will be discussed. Following this is a chapter on the choice between model scale and full scale simulations. This chapter strives to explain the why full scale simulations were used in this research. Next, a chapter on the convergence of the results is given. In this chapter, a couple of options to improve the convergence are tested and discussed. The conclusions from this chapter are especially important as they play a significant role in the discussion of the results and the recommendations in [section 9](#) and [section 11](#). Afterwards, the trim- and sinkage correction used in the validation will be given. Finally, the method with which the righting- and heeling moment were calculated in [subsection 8.1](#) will be given.

### 5.1 Ship reference frame

The ship's reference frame is shown in [Figure 11a](#). The reference frame originates in the ship's centre of gravity. The figure also shows the names of the translations and rotations around the axis. [Figure 11b](#) shows how a positive drift angle is defined. All experimental and computed quantities are given in this reference frame.



### 5.2 Mesh generation and boundary conditions

An initial grid was created for this study and was refined later on as explained in [subsection 5.3](#). The mesh was generated in the unstructured grid generation software HEXPRESS. The aspects of this grid such as domain size, boundary conditions and refinement boxes are presented in this chapter.

As mentioned, the tug used in this research is the RSD 2513. This is a tugboat with a length of

approximately 25 meters. It normally has two fins at the front and two thrusters in the aft as shown in [Figure 12](#). However, for this research, the vessel was simulated without the thrusters. The model tests are used in the validation exercise are also performed without these thrusters.



Figure 12: Fully appended model of the RSD 2513

In [Figure 13](#) the body is visible in FINE/Marine as well as the boundaries of the domain. The domain needs to be large enough to guarantee that reflection from waves on the boundaries doesn't affect the solution. The domain in this research has the size indicated in [Table 1](#).  $L$  is the ship's length. Furthermore, there is a damping zone at the outflow boundary to ensure waves aren't reflected.

Figure 13: Domain and body in FINE/Marine. *Figure is left out of the public version intentionally due to confidentiality*

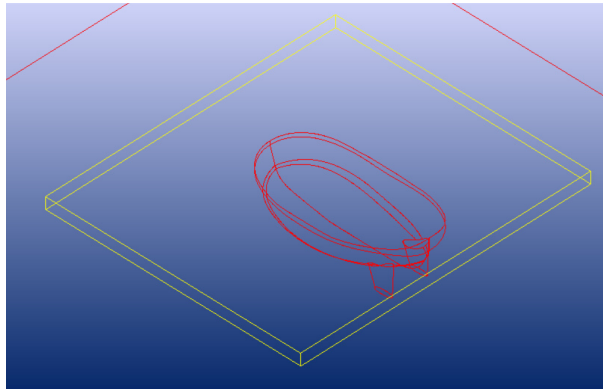
Table 1: Domain size. *Data is left out of the public version intentionally due to confidentiality*

|   | min      | max     |
|---|----------|---------|
| x | $-? * L$ | $? * L$ |
| y | $-? * L$ | $? * L$ |
| z | $-? * L$ | $? * L$ |

The boundaries of the domain all have the far field boundary condition except for the top- and bottom boundary. The top and bottom have a prescribed pressure boundary condition. The prescribed pressure condition imposes a pressure to those boundaries at the initialisation of the simulation [27](#).

As discussed in [subsection 4.3](#), the mesh has free surface AGR. For AGR to work, some refinement on the free surface is required [27](#). Firstly, the free surface is refined with grid size **CONFIDENTIAL**.  $L$  is the length of the ship. Furthermore, a refinement box is added with the same cell size in  $z$  direction close to the ship. The goal of this box is to make sure the free surface is captured

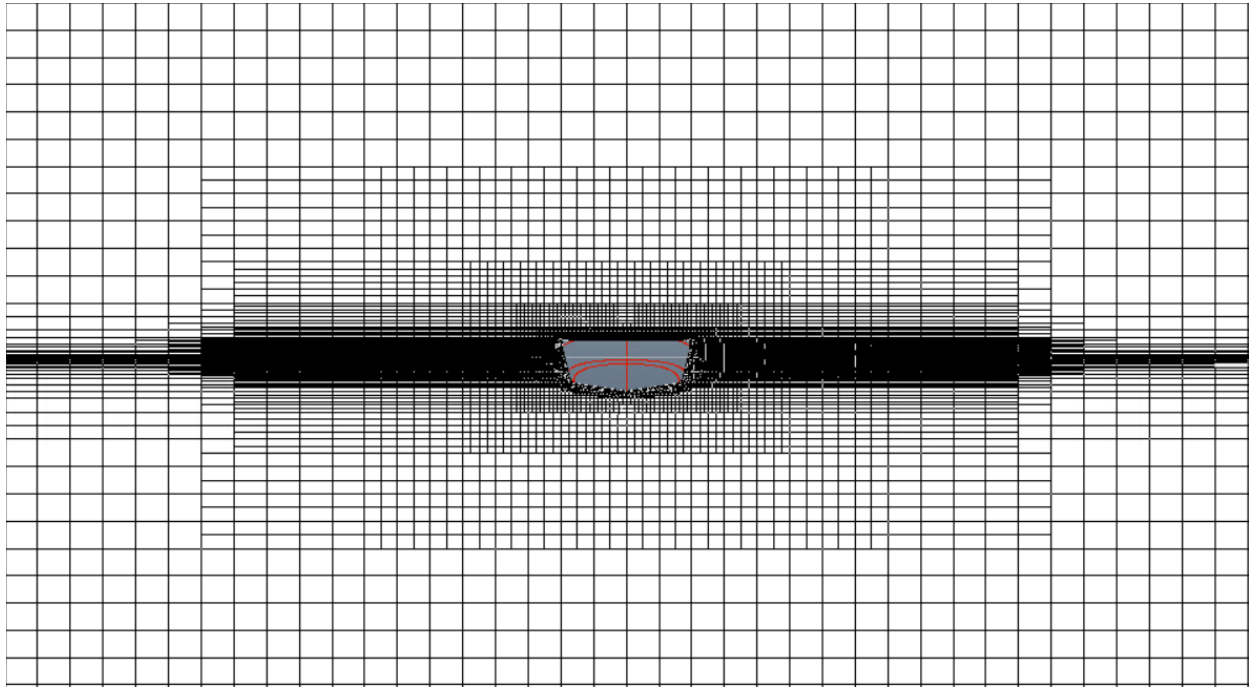
within this box, even if it is elevated due to waves. The dimensions of this box will be **CONFIDENTIAL** in front of the ship, **CONFIDENTIAL** behind it, and **CONFIDENTIAL** to either side in Y-direction from the centreline. The refinement box is shown in a screenshot from HEXPRESS in [Figure 14](#). The guidelines on which this box is base, are given in the FINE/Marine user guide. Further settings for the AGR are discussed in [subsection 5.4](#).



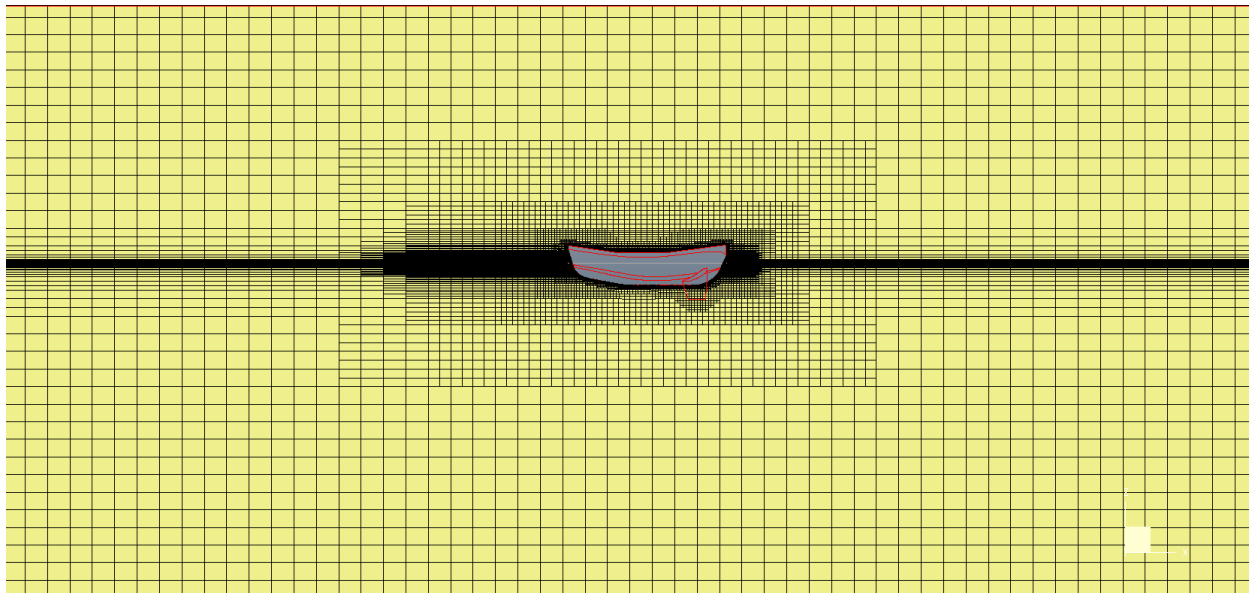
*Figure 14: Refinement box in HEXPRESS*

With the body and the domain in place, an initial mesh is created. To sketch an idea of what this looks like, [Figure 15](#) shows the mesh at cut planes along the length and breadth of the ship, as well as a close-up of the boundary layer. Furthermore, [Figure 16](#) shows the mesh in 3D.

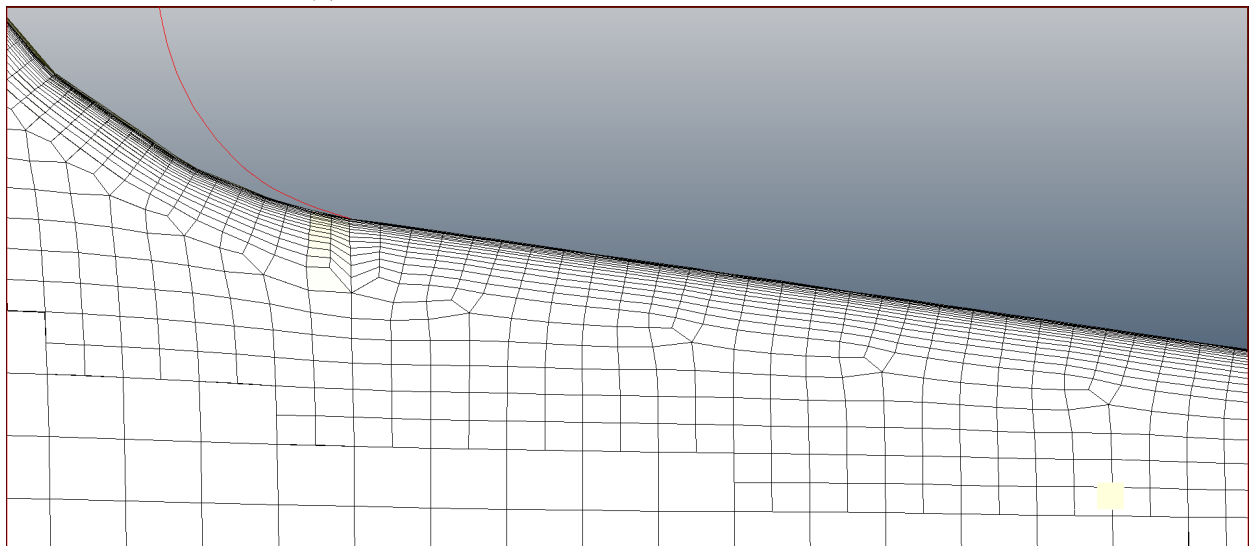




*(a) Cut plane of the front view in the middle of the ship*



*(b) Cut plane of the side view in the middle of the ship*



*(c) Close-up of the boundary layer*

*Figure 15: Global view of the mesh and close-up of the boundary layer*

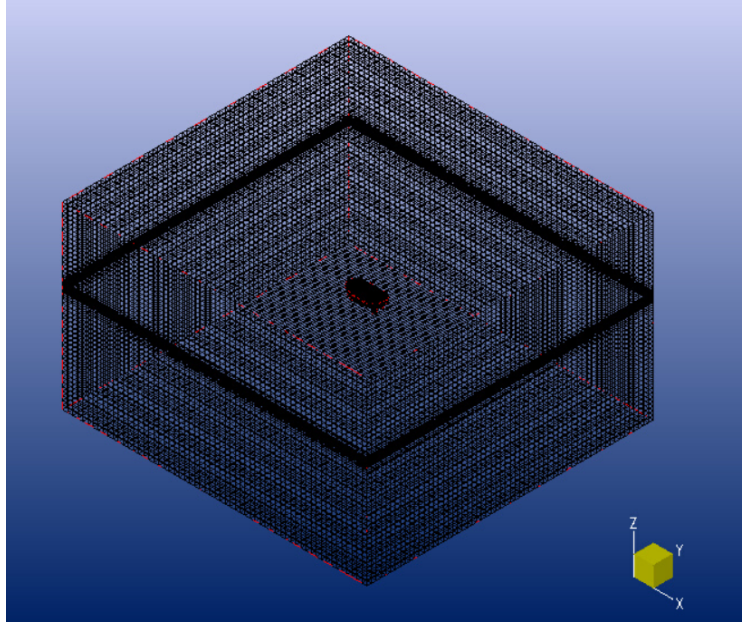


Figure 16: 3D view of the mesh

### 5.3 Grid refinement

To verify the grid with a grid convergence study in subsection 6.1, a series of refined grids needs to be created. These grids are refined from the mesh that was generated in subsection 5.2. This mesh is called mesh 1 from now on. It is important that these grids are geometrically similar to each other. Otherwise scatter is created in the results of the verification study. Geometrically similar grids are, however, difficult to create for unstructured grids. There are methods to keep the grids as geometrically similar as possible. For the initial cell count in all directions, Equation 11 was used [36]. Here  $r_i$  is the refinement ratio and  $N$  is the number of cells in a direction.  $N_1$  is the number of cells in a certain direction for the coarsest grid, mesh 1.

$$N_i = N_1 r_i \quad (11)$$

However, this equation doesn't hold up in the viscous layer. The initial cell size in the boundary layer,  $d_i$ , is determined using equation Equation 12. Where  $\gamma$  is the growth ratio which is calculated with Equation 13. [36]

$$d_i = d_1 \left( \sum_{j=0}^{i-1} \gamma_i^j \right)^{-1} = d_1 \frac{1 - \gamma_i}{1 - (\gamma_i)^i} \quad (12)$$

$$\gamma_i = d_1^{\frac{1}{r_i}} \quad (13)$$

Using these equations, the global grid parameters for the grid refinement are calculated. These are shown in Table 2. With these parameters, four new grids were generated which were used in the verification and validation exercises in this research.

Table 2: Global domain parameters for the 5 grids. *Table is left out of the public version intentionally due to confidentiality*

| Mesh            | 1 | 2 | 3 | 4 | 5 |
|-----------------|---|---|---|---|---|
| $N_x$           |   |   |   |   |   |
| $N_y$           |   |   |   |   |   |
| $N_z$           |   |   |   |   |   |
| Diffusion       |   |   |   |   |   |
| $y^+$           |   |   |   |   |   |
| $d_i$           |   |   |   |   |   |
| Number of cells |   |   |   |   |   |

#### 5.4 Simulation settings

The simulations used in [section 6](#), [subsection 7.3](#) and [subsection 8.1](#) were all run using the same simulation settings. These settings determine, among other things, how the ship's speed, drift angle and heeling angle are determined, what kind of wall treatment is used, what kind of turbulence model is used, and what target cell size is used for AGR. In this chapter, these settings are discussed. The theory behind the decisions can be found in [subsection 4.1](#), [subsection 4.2](#) and [subsection 4.3](#).

The drift angle was not set by rotating the ship, but by changing the direction of the inflow velocity. The ship had a velocity imposed in the surge and sway direction. The vector sum of the velocities in surge and sway added up to the ship's speed. This is visualised in [Figure 17](#). Here,  $V_S$  is the ship speed,  $V$  the speed in sway direction and  $U$  the speed in surge direction. The ship speed for all simulations is set at 8 knots.

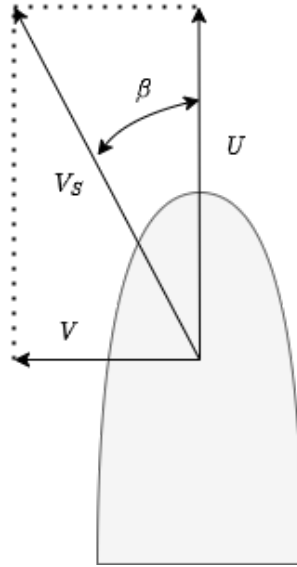


Figure 17: Imposed drift angle in simulation settings

For heeling angles, a motion is also imposed in the simulation settings for certain simulations with smaller angle changes. However, for most simulations the angle is introduced by rotating the ship and remeshing. The roll motion is then fixed to keep the ship locked in that position. Similarly, the yaw motion is fixed. Trim and sinkage, however, are solved. As discussed in [subsection 5.9](#) a trim- and sinkage correction is applied for validating the simulations since in the model test trim and sinkage are fixed.

The boundary layer is solved by imposing a wall function. As discussed in [subsection 4.1](#) in theory,

wall resolved would be more likely to generate accurate results, as turbulence plays a big part in this research. However, it is also mentioned how Schot [31] found little difference between wall resolved and wall function simulation results in his research where he simulated the RSD 2513 for a range of drift angles. Figure 18 shows a comparison between CFD calculations with a wall function and wall resolved to model the boundary layer. The difference that can be observed between the two CFD lines is minimal compared to the difference with the model test. This confirms the conclusion by Schot. Since wall function simulations are less computationally heavy than wall resolved simulations, a wall function was used for in this research.

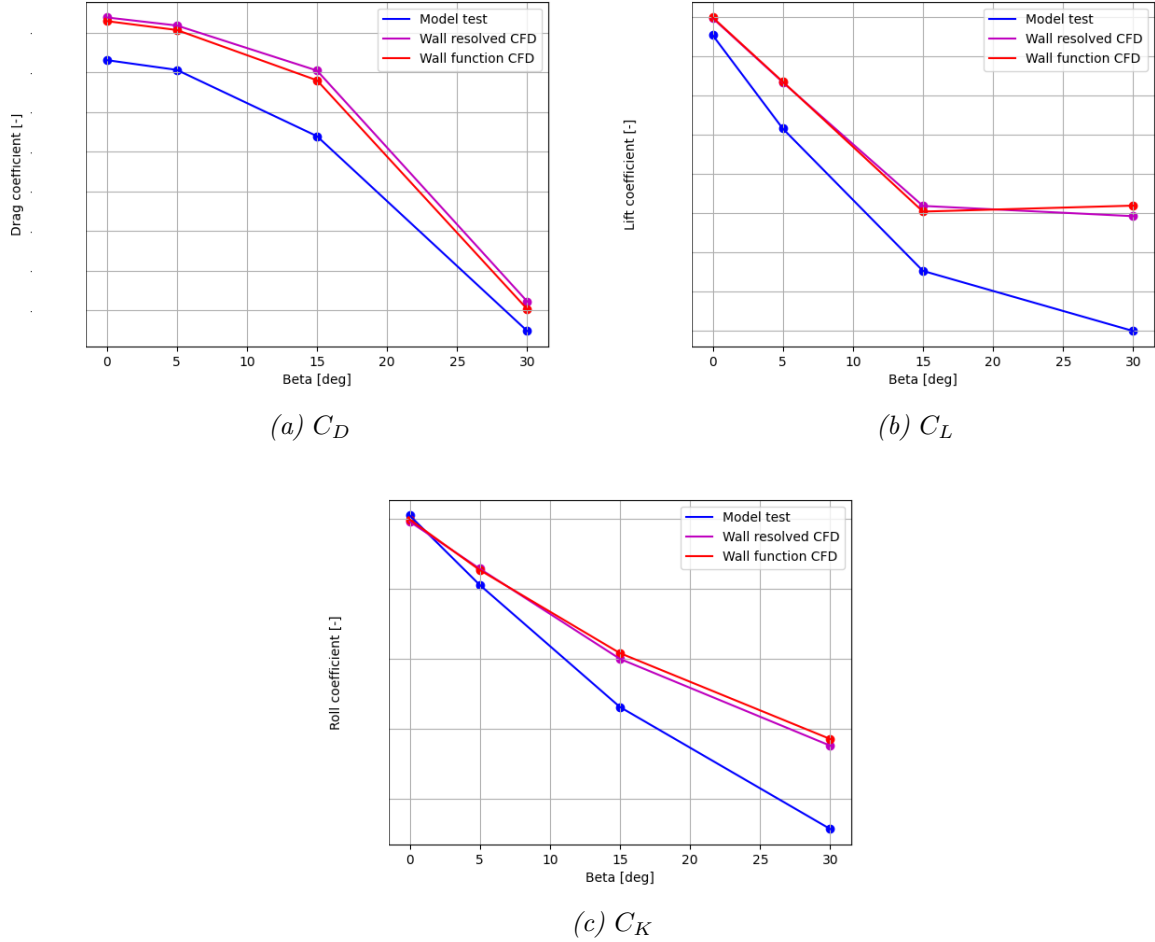


Figure 18: Drag coefficient, lift coefficient and roll coefficient comparison for full scale CFD with wall function and wall resolved, and the model tests results. Simulated at 0 degrees heel, 8 knots forward speed and 0- to 30 degrees drift

As discussed in subsection 4.2 the choice of turbulence model can strongly affect how accurately turbulence is captured and can significantly influence the results of the simulations. The turbulence model for the simulations is set to the  $k - \omega$  SST model. In subsection 5.7 it is explained in detail why this model is chosen as a comparison is made between this model and the DDES model.

subsection 4.3 discusses the theory behind AGR and in subsection 5.2 the refinement box required for AGR is discussed. In the simulation settings, the target cell size for the AGR is set to **CONFIDENTIAL**. This AGR parameter is based on the best practice from the FINE/Marine user guide [27]. Initial test simulations were run to investigate the effect on the solutions by refining the target cell size. However, the amount of cells generated by AGR, their location, and the forces and moment computed by the solver did not change significantly. Therefore, the setting mentioned

above was used in the simulations.

## 5.5 Design of experiment

As discussed in subsection 4.4 the simulation matrix was designed with a full factorial approach. In this chapter it will be discussed which data points were simulated to fill the simulation matrix used in subsection 8.1. It was decided to cover drift angles between 0- and 45 degrees for every 15 degrees, as this represent a normal operating range for an escort tug during indirect towing. For the same reason, the simulations are run at 8 knots. Furthermore, a range of heeling angles was simulated between 0- and 10 degrees heel. At 10 degrees heel, the ship started to get water on deck when combined with a drift angle. for this reason it was decided to simulate up to that angle. The simulations were done for every 2 degrees heel, as this resulted in good resolution for the curves. The simulation matrix is visualised in Figure 19.

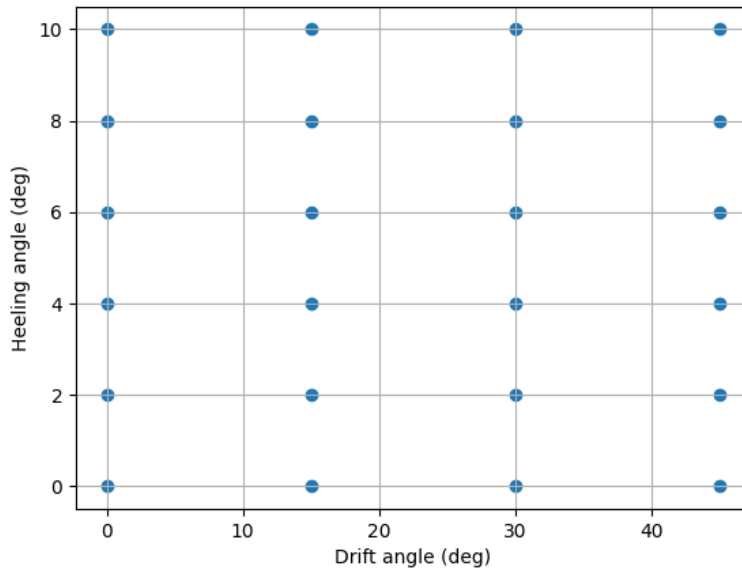


Figure 19: Simulation matrix

## 5.6 Model scale vs full scale

Full scale validation material for CFD in the maritime industry is often not available. This is largely due to costs, but other factors like controlling the parameters also play a role. This is why the validation material used in subsection 7.3 is model scale. Since the CFD results will be applied to full scale ships, it is mostly preferred to run full scale simulations. However, due to the lack of validation material at full scale, it can be difficult to validate these models. Full scale CFD is often validated with model tests since it is the only validation material available. In those cases, it is important to be aware of scale effects and other discrepancies between full scale and model scale.

For this reason, a comparison is made between full scale CFD, model scale CFD and model tests. This is done with mesh 1, a relatively coarse mesh with approximately **CONFIDENTIAL** million cells. The goal is to see if the trend of the full scale- and model scale CFD follow the expected trend from the model tests. The results for  $C_L$ ,  $C_D$  and  $C_K$  are shown in Figure 20. The simulations are performed at 0 degrees heel, and drift angle varying between 0 and 45 degrees. The model scale and model test results are scaled to full scale using  $\lambda^3 \cdot \frac{\rho_{sea}}{\rho_{basin}}$  for the forces, and  $\lambda^4 \cdot \frac{\rho_{sea}}{\rho_{basin}}$  for the moment.  $\lambda$  is 12.189.

The reason why lift- and drag force are reviewed and not surge- and sway force is because the

goal is to review the transverse stability of tugs. This means that the roll moment, and the forces contributing to that moment should be investigated. The surge force, however, cannot contribute to the roll moment. It would therefore be less useful to investigate the surge- and sway force. Another reason for looking at lift- and drag force is that the subject of this thesis are escort tugs. Lift and drag are an important measure for escort tugs since they translate to the steering- and breaking forces a tug can exert on the ship it is escorting.

The lift force, drag force and roll moment are non-dimensionalised by using Equation 14. In Figure 20 the results from the model scale- and full scale CFD are plotted together with the model test results. There is a clear offset in absolute value between the full scale and model test results. However, apart from the 30 degrees drift  $C_L$  value, the trends are in good agreement with the trends of the model tests. The reason why this discrepancy is so large at 30 degrees drift is most likely the high amount of turbulent kinetic energy in the flow around the ship. This is shown and explained in more detail in subsection 5.7. Because the trend of the full scale cfd curves seem to match the model test quite well, there was confidence that with the right grid refinement, full scale simulations could generate results accurate enough to validate with these model tests. For this reason, the simulations in this research were done at full scale.

$$\begin{aligned}
C_X &= \frac{F_X}{\frac{1}{2}\rho A_{lat} V^2} \\
C_Y &= \frac{F_Y}{\frac{1}{2}\rho A_{lat} V^2} \\
C_L &= \frac{F_L}{\frac{1}{2}\rho A_{lat} V^2} = C_Y \cos(\beta) - C_X \sin(\beta) \\
C_D &= \frac{F_D}{\frac{1}{2}\rho A_{lat} V^2} = C_Y \sin(\beta) + C_X \cos(\beta) \\
C_K &= \frac{K}{\frac{1}{2}\rho A_{lat} TV^2}
\end{aligned} \tag{14}$$



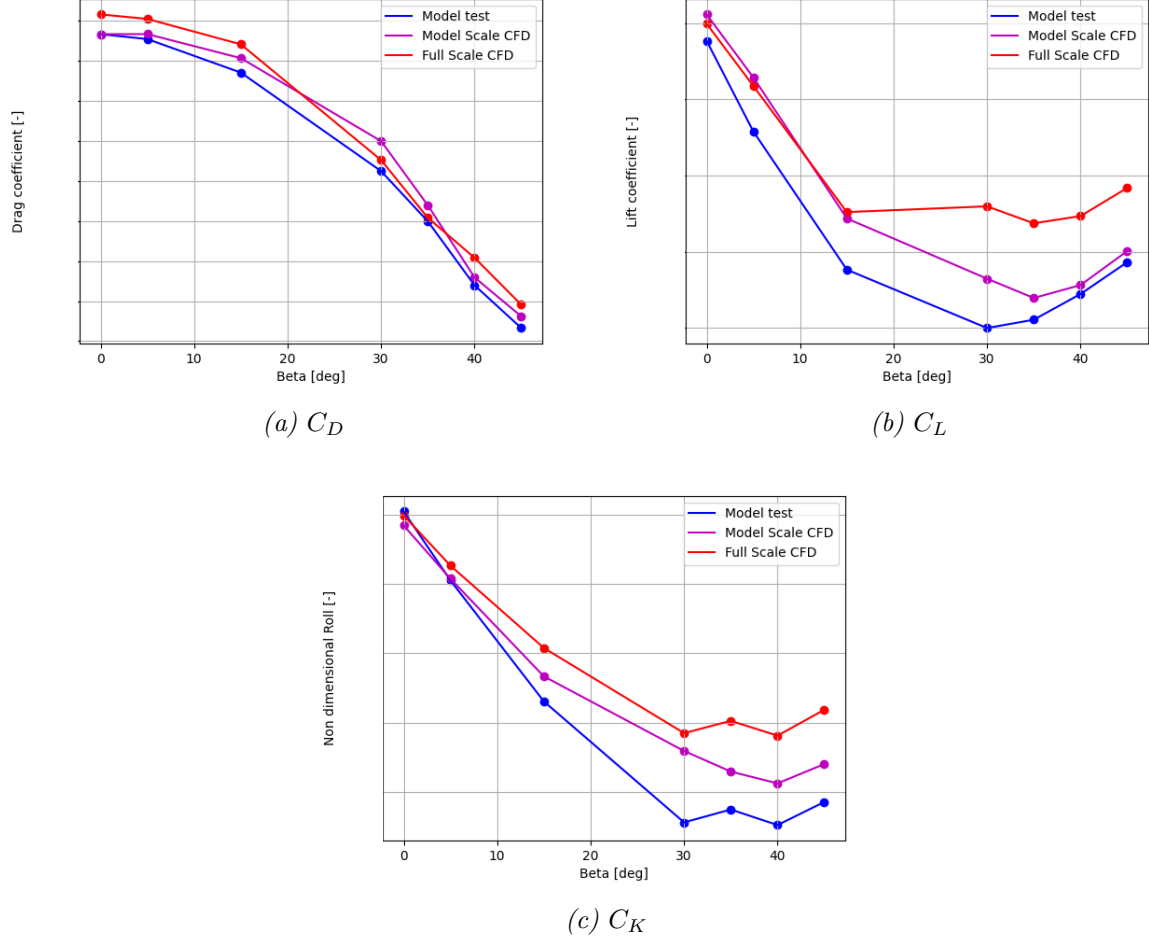
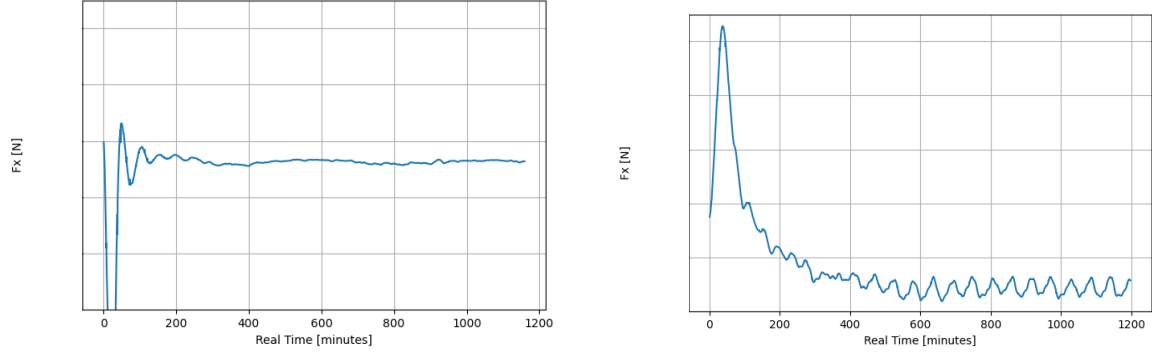


Figure 20: Drag coefficient, lift coefficient and roll coefficient comparison for full scale and model scale CFD and model tests. The simulations are run between 0- and 45 degrees drift, 0 degrees heel and 8 knots forward speed

## 5.7 Convergence

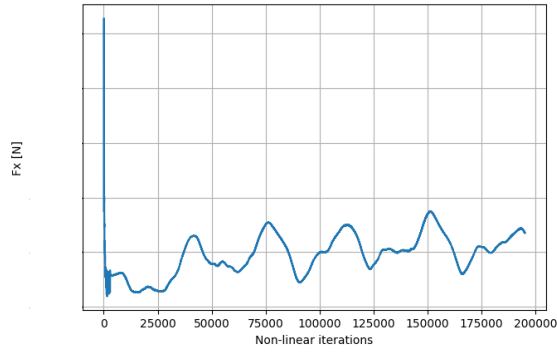
In order to use the results of the simulations in [subsection 8.1](#) the simulations had to converge. If they had not converged the values given by the software would not be reliable, as they could have changed significantly when continuing running the simulations. For lower drift angles, up to 15 degrees, the simulations converged without problems. This is most likely because there is very little turbulence and flow separation. However, for 30 degrees drift and upwards, there were problems with convergence. For some simulations, the solution converged to a point where it was oscillating around a value. The amplitude of this oscillation were around 5% of the result. The residuals were low except for the  $\omega$  residuals which represents the specific dissipation of turbulence. This indicates that it was indeed turbulence that caused these convergence problems. However, it should be noted that even for 0 and 15 degrees drift this residual was high. An example of well converging and less well converging results are given in [Figure 21](#)



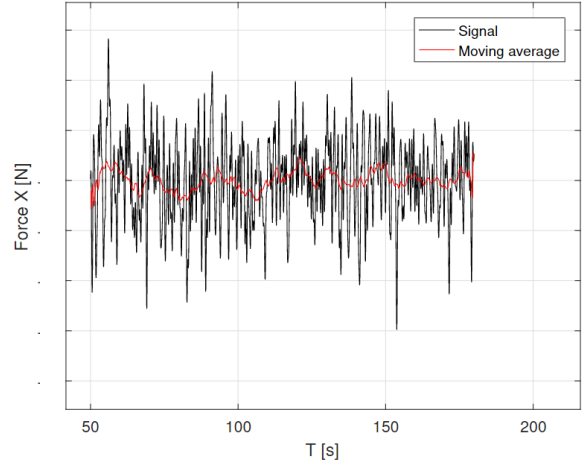
(a) Time trace of  $F_x$  at 15 degrees drift, 0 degrees heel and 8 knots forward speed (b) Time trace of  $F_x$  at 30 degrees drift, 0 degrees heel and 8 knots forward speed

Figure 21: Example of the convergence of simulations at different drift angles

Since it is likely that turbulence caused the convergence problems seen in [Figure 21b](#), it was decided to run an unsteady simulation to compare the results. The result of  $F_x$  for this simulation is shown in [Figure 22a](#). Note that instead of real time, the x axis shows the number of non-linear iterations. It is immediately visible that the result doesn't show ideal convergence and is still rather irregular. It can therefore be concluded that the flow that is being solved is indeed highly unsteady. [Figure 23](#) shows the separation of the unsteady simulation at different times during the simulation, by plotting the area where the x velocity of the flow is 0 or positive. The flow being positive in x direction means that it is going in the same direction as the ship is sailing which indicates turbulence and flow separation. Note that it is only shown at two cut planes and therefore it does not show all the separation. However, it shows that the separation region constantly changes and could therefore be seen as irregular. This is also visualised in [Figure 22b](#) which is a time trace of  $F_x$  as measured in the model tests at 52.5 degrees drift. Even though it is not the same drift angle, it does show how irregular the force signal and the moving average of the model test are. This indicates that the unsteadiness and irregularity of  $F_x$  is not just caused by numerical phenomena but also by physical phenomena. It should be noted that the time step for the unsteady simulation was chosen based on best practice guidelines from the Fine/Marine user guide [\[27\]](#). No temporal refinement study has taken place. It is possible that such a study could improve the results and the convergence from the unsteady simulation. However, since this particular simulation already took over a week to run, it was decided not to continue with unsteady simulations but to run steady simulations instead. It is important to note that there is a difference of about **CONFIDENTIAL** between  $F_x$  with the unsteady simulation and with the steady simulation. This is quite a large difference since it is about 30% of the force from the steady simulation. This should be taken into account in the results and conclusions from these simulations.



(a)  $F_x$  vs non-linear iterations for an unsteady simulation at 30 degrees drift



(b) Time trace of  $F_x$  at 52.5 degrees drift

Figure 22:  $F_x$  for an unsteady simulation at 30 degrees drift and model test at 52.5 degrees drift. Both plots are at 0 degrees heel and 8 knots forward speed

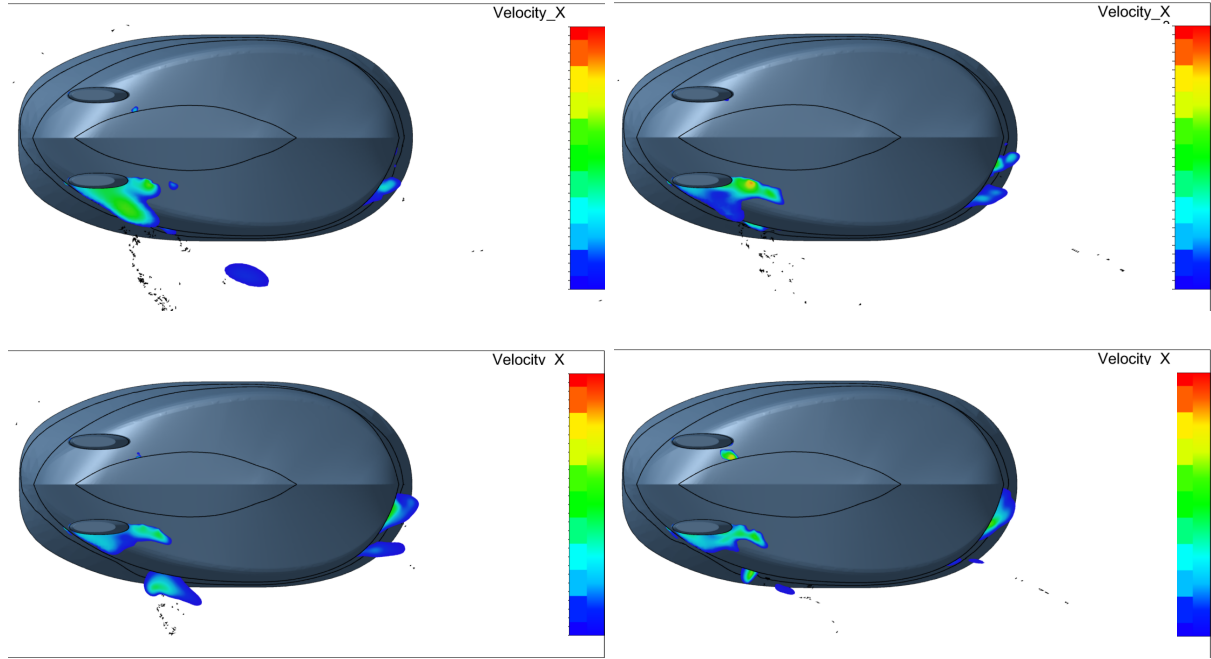


Figure 23: Separation from the unsteady simulation plotted at different times during the simulation on a cut plane. The simulation was run at 30 degrees drift, 0 heel and 8 knots forward speed

In an effort to improve the convergence, the unsteady simulation was rerun with a different turbulence model. A DDES model (see subsection 4.2) was used as it was expected to capture the turbulence and separation on the hull better than the previously used  $k-\omega$  model. The results for  $F_x$  of this simulation are plotted against the non-linear iterations in Figure 24. The results were similar to Figure 22a. The convergence did not improve significantly, even though it was expected that this turbulence model would help improve the convergence. One important remark is that the mesh used for this turbulence model is exactly the same as the mesh from the unsteady simulation with  $k-\omega$  turbulence model and the steady simulation. It is however expected that a slightly different mesh is required to maximise the benefit of using this turbulence model. This is because some of the turbulence length scales will be solved instead of averaged, due to DDES being a

combination of LES and RANS. This is the reason why FINE/Marine recommends basing the cell size on the Taylor turbulence length scale for the DES turbulence model. Using the combination of DDES and unsteady simulations is not a feasible alternative when looking at simulation time. The DDES turbulence model increased the simulation time again. The short amount of time simulated in the figure took well over a week to run.

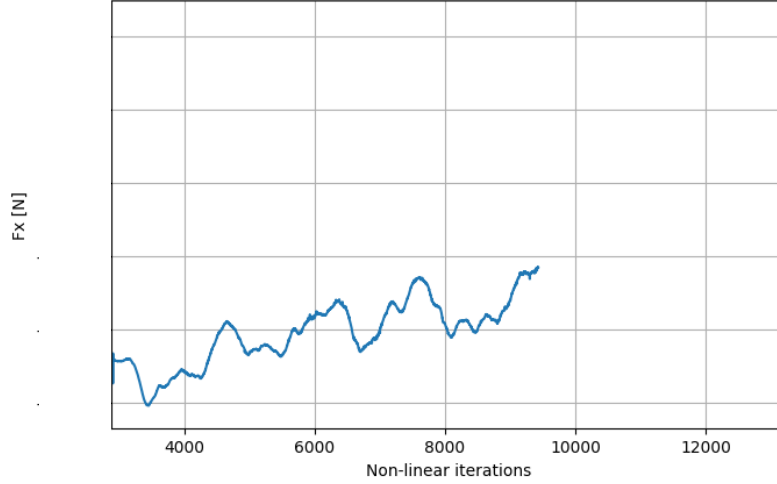


Figure 24:  $F_x$  vs non-linear iteration for an unsteady simulation with DDES at 30 degrees drift

## 5.8 Regression method verification study

subsection 4.5 discusses the theory behind estimating the numerical uncertainty, which is used in the verification study in section 6. The method by Eça en Hoekstra is described and it is mentioned that a least squares method is used to determine  $\phi_0$ ,  $\alpha$  and  $p$ . However, when the least squares method was used, the estimated exact solution  $\phi_0$  was quite far off the results from the meshes  $\phi_i$ . This is visualised in Figure 25a.  $\phi_0$  is the value of  $C_D$  at the point where the relative step size is 0. Even though the results from the CFD have low relative difference to each other, the second order polynomial estimates a  $\phi_0$  that has a large relative difference to the other values. This suggests that the second order polynomial calculated with the least squares method does not result in an accurate estimate of the uncertainty. A possible explanation for why the second order fit does not seem to suit the data is that the grids aren't geometrically similar enough. This is a drawback of using unstructured grids. It seems the grids are not geometrically similar, which leads to scatter in the results. For this reason, it was decided to use a linear polynomial regression instead of the second order least squares fit. The results for  $\phi_0$  seem to be better estimated with this regression technique as shown in Figure 25b.

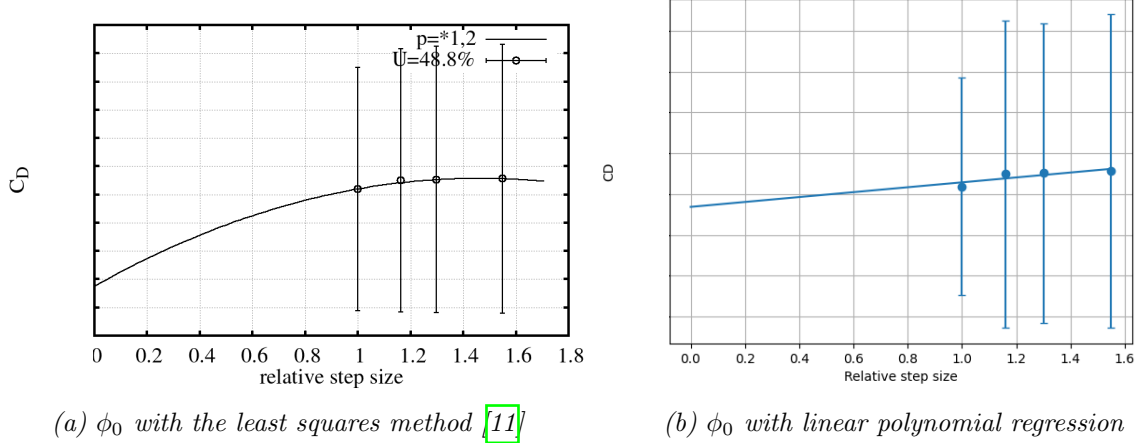


Figure 25:  $\phi_0$  of  $C_D$  calculated with a second order polynomial and a linear regression

Unfortunately, the difference between the value from the CFD,  $\phi_i$ , and the value estimated with the regression,  $\phi_{fit}$ , is larger with the linear polynomial regression than with the second order least squares method. However, the estimate of  $\phi_0$  with the linear regression still results in a lower uncertainty overall.

## 5.9 Trim- and sinkage correction

The model tests used for the validation of the simulations are captive model tests where heave and pitch motions are fixed. However, in the CFD simulations these motions are not fixed. For the validation exercise in subsection 7.3, a trim- and sinkage correction term will be added in order to make it more comparable to the results from the model test. This section describes how this correction is calculated.

Inoue proposed a way to correct the sway force on a vessel for trim [16]. This trim correction was determined based on data for slender ships. Although a tug is far from a slender ship, Damen has used this method in their performance prediction tool for tugs, TugSim. Therefore, this correction will be used to correct the sway force in order to have a better comparison with the model tests. Equation 15 shows the correction as suggested by Inoue.  $\tau$  is the trim, and  $d_m$  is the mean draft of the ship.  $C_{Y\beta}$  is a non-dimensional  $Y$  force which is calculated as shown in Equation 16. The trim of the simulations is around **CONFIDENTIAL** degrees for all the meshes. This is a rather small trim and the influence of the trim correction on the CFD results is therefore around 3%.

$$C_{Y\beta(\tau)} = C_{Y\beta(0)} \left[ 1.0 + \frac{2\tau}{3d_m} \right] \quad (15)$$

$$C_Y = F_y / \left( \frac{1}{2} \rho L d U^2 \right) \quad (16)$$

The sinkage can be corrected using Equation 17.  $A_{lat}$  is the lateral underwater area of the vessel and  $U$  the velocity. This sinkage correction was introduced in Damen and is based on CFD simulations with and without sinkage of an RSD 2513 tug [32]. Only the side force was corrected, since it was concluded from those simulations that the sinkage mainly affects the side force.

$$A_{lat-corrected} = A_{lat} + L * sinkage$$

$$C_Y = \frac{F_y}{0.5 * \rho * U^2 * A_{lat-corrected}} \quad (17)$$

After comparing the results with and without trim- and sinkage correction, it was clear that the trim- and sinkage correction has a big influence on the magnitude of the  $Y$  force. For all meshes at 15 degrees, the difference between the  $Y$  force and the corrected  $Y$  force was 15%. Furthermore, the corrected  $Y$  force was calculated for 0, 5, 15, 30 and 45 degrees drift for mesh 2. This resulted in a difference between the  $Y$  force and the corrected  $Y$  force ranging from 15% to 21%. The influence of the correction on the lift- and drag coefficients can be seen in [Figure 30](#) in the validation chapter. As discussed above, the influence of the trim correction is rather small. The overall effect of the trim- and sinkage correction on the force from CFD is therefore mostly due to the sinkage correction. The overall effect of the trim- and sinkage correction is quite high. Especially since the method behind it was initially based on data from a different ship type. This is important to keep in mind since it significantly affects the  $Y$  force in the validation exercise. However, the correction is not used in the results shown in [section 6](#) and [subsection 8.1](#).

### 5.10 Calculating righting moment from FINE/Marine results

The goal of this thesis is to make an analysis of the dynamic stability of tugs in escort by calculating the righting- and heeling moment of the tug using CFD. In order to achieve this, it is important to be clear on how these moments are calculated. This will be explained in this section. This method will be used to calculate the righting- and heeling moment in [subsection 8.1](#).

When assessing the stability of a ship, the situation is often reviewed statically. When the ship is heeled, a moment is created that makes the ship return to its original position. This moment is called the restoring moment and is created due to additional buoyancy. [Figure 26](#) shows what this situation looks like. The restoring moment is not the same thing as the righting moment, although it is part of the righting moment. The righting moment is the summation of all moments working in the direction opposite to the heeling direction. The restoring moment is always part of it but other moments can contribute as well. [Figure 26](#) doesn't show a force in  $y$  direction because the resultant of the static  $y$  force is zero.

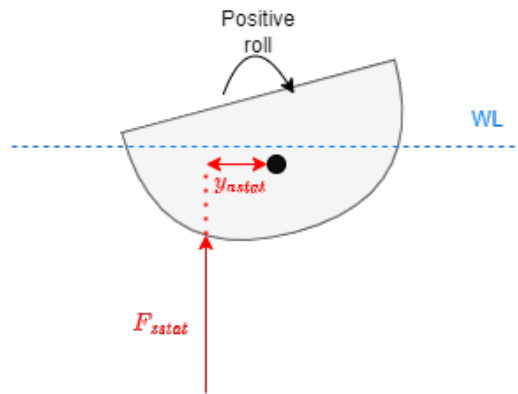


Figure 26: Restoring moment of a ship

The picture above shows the static stability of a ship but in this thesis, the dynamic stability is researched. This situation is shown in [Figure 27](#). The figure is split in two sub-figures to show what is defined as the righting moment and what as the heeling moment. The force in  $z$  direction is now not only consisting of a static part, but also a dynamic part. As discussed above, the  $Y$  force doesn't have static part and is therefore completely dynamic. This picture shows how the

dynamic righting moment is created by the Z force and how the dynamic heeling moment is created by the Y force. This situation is described in [Equation 18](#). The forces are split up into a dynamic- and static part to create more insight into which components contribute to the moments and how much. This creates the following system of equations.  $y_a$  and  $z_a$  are the application points of the forces with respect to the centre of gravity,  $M_R$  is the righting moment and  $M_H$  is the heeling moment. The suffix *stat* indicates a static force or arm, and *dyn* indicates a dynamic force or arm.

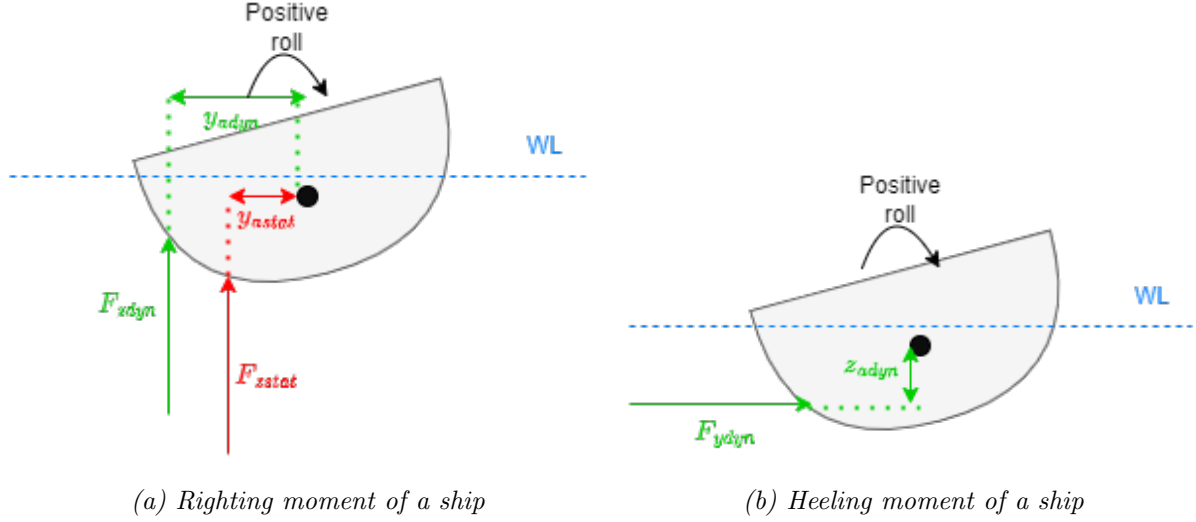


Figure 27: The righting- and heeling moment of a ship

$$\begin{aligned}
 M_R &= F_{zstat} \cdot y_{astat} + F_{zdyn} \cdot y_{adyn} \\
 M_H &= -F_{ydyn} \cdot z_{adyn} \\
 M_x &= M_R + M_H \\
 M_x &= F_{zstat} \cdot y_{astat} + F_{zdyn} \cdot y_{adyn} - F_{ydyn} \cdot z_{adyn}
 \end{aligned} \tag{18}$$

FINE/Marine can output all forces and moments in x-, y- and z direction. Furthermore, it determines the dynamic component of these forces and moments as well. However, the application point of these forces is unknown meaning that you can't directly calculate the contribution of the forces to the roll moment. The dynamic application points can be determined by solving the following system of equations.

$$\begin{aligned}
 M_{xdyn} &= F_{zdyn} \cdot y_{adyn} - F_{ydyn} \cdot z_{adyn} \\
 M_{ydyn} &= F_{xdyn} \cdot z_{adyn} - F_{zdyn} \cdot x_{adyn} \\
 M_{zdyn} &= F_{ydyn} \cdot x_{adyn} - F_{xdyn} \cdot y_{adyn}
 \end{aligned} \tag{19}$$

In order to solve this set of equations, a constraint is needed since there are independent equations. The constraint is shown in equation [Equation 20](#) [\[26\]](#). What this constraint would mean in physical sense, is that a force in x can't create a moment around the x axis. Therefore, a force in x direction has per definition zero arm in x direction and is therefore zero. The same goes for the other directions.

$$0 = F_{xdyn} \cdot x_{adyn} + F_{ydyn} \cdot y_{adyn} + F_{zdyn} \cdot z_{adyn} \tag{20}$$

Having this constraint, [Equation 19](#) can be solved as follows.



$$\begin{aligned}
\text{From } M_{xdyn} \rightarrow z_{adyn} &= \frac{F_{zdyn} * y_{adyn} - M_{xdyn}}{F_{ydy}} \\
\text{From } M_{zdyn} \rightarrow x_{adyn} &= \frac{M_{zdyn} + F_{xdyn} * y_{adyn}}{F_{ydy}}
\end{aligned} \tag{21}$$

Filling in the constraint from [Equation 20](#) gives the following.

$$\begin{aligned}
0 &= \frac{F_{xdyn} * M_{zdyn} + F_{xdyn}^2 * y_{adyn}}{F_{ydy}} + F_{ydy} * y_{adyn} + \frac{F_{zdyn}^2 * y_{adyn} - M_{xdyn} * F_{zdyn}}{F_{ydy}} \\
0 &= F_{xdyn} * M_{zdyn} + F_{xdyn}^2 * y_{adyn} + F_{ydy}^2 * y_{adyn} + F_{zdyn}^2 * y_{adyn} - M_{xdyn} * F_{zdyn} \\
y_{adyn} &= \frac{M_{xdyn} * F_{zdyn} - M_{zdyn} * F_{xdyn}}{F_{xdyn}^2 + F_{ydy}^2 + F_{zdyn}^2}
\end{aligned} \tag{22}$$

Since all the forces and moments in the formula for  $y_{adyn}$  are known,  $y_{adyn}$  is known. With  $y_{adyn}$  known,  $z_{adyn}$  can be calculated from the formula given above.

Similarly,  $y_{astat}$  can be calculated. The only exception is that there is no static force in y direction. This results in the  $z_{astat}$  as described in [Equation 23](#)

$$y_{a \text{ stat}} = \frac{M_{x \text{ stat}} * F_{z \text{ stat}} - M_{z \text{ stat}} * F_{x \text{ stat}}}{F_{x \text{ stat}}^2 + F_{z \text{ stat}}^2} \tag{23}$$

Now that all moment arms are known, all terms in the roll moment equation are known. This means the heeling and righting moment can be calculated. The results of this can be viewed in [subsection 8.1](#)

## 6 Verification

As mentioned in subsection 4.5, the numerical error in this research only consists of the discretization error as other errors are neglected. In that chapter it is explained how this error is calculated. This is done with a method based on the work of Eça and Hoekstra [10]. However, as explained in subsection 5.8, instead of a second order regression with least squares method, a first order polynomial is used to estimate  $\phi_0$ . With this method, the numerical uncertainty will be calculated for all grids, at 15 degrees drift angle, for the lift coefficient, the drag coefficient and the roll coefficient. The meshes used created for the verification study are discussed in subsection 5.3. Their global parameters are given in Table 2

### 6.1 Verification results

Table 3 shows the relative difference in  $C_D$ ,  $C_L$  and  $C_K$  between the meshes and the finest mesh. It is clear that immediately after the first refinement, the relative difference to the finest mesh becomes a lot smaller. The relative difference for mesh 1 for  $C_D$  and  $C_K$  is especially high. Therefore, the results of mesh 1 are viewed as an outlier compared to the results of the other meshes. Since this could have a significant effect on the uncertainty estimation, it was decided to perform the uncertainty analysis without the results of mesh 1.

Table 3: Relative difference in  $C_D$ ,  $C_L$  and  $C_K$  for the five meshes

|       | Mesh 1/Mesh 5 | Mesh 2/Mesh 5 | Mesh 3/Mesh 5 | Mesh 4/Mesh 5 | Mesh 5/Mesh 5 |
|-------|---------------|---------------|---------------|---------------|---------------|
| $C_D$ | 9.32%         | 4.2%          | 3.79%         | 3.43%         | 0.00%         |
| $C_L$ | 11.8%         | 3.08%         | 2.87%         | 2.17%         | 0.00%         |
| $C_K$ | 5.37%         | -4.55%        | -3.47%        | -4.54%        | 0.00%         |

The results of the uncertainty analysis can be seen in Table 4, Table 5 and Table 6. Furthermore, the curves with the linear polynomial regression are shown in Figure 28. The first thing to notice from the tables is that the uncertainty does decrease with mesh refinement. For all meshes, the biggest decrease is between mesh 4 and mesh 5. The reason for this is that  $\phi_i$  for mesh 5 is closer to  $\phi_0$  for than all other meshes. The difference in uncertainty between the other meshes is clearly a lot less. This is because the difference in  $\phi_i$  is relatively small for those three meshes.

However, overall it can be concluded that the discretization uncertainty is high. All values, except  $C_K$  for mesh 5, have an uncertainty of more than 20%. This is mostly caused by a high safety factor. The high safety factor is due to the standard deviation of the data being higher than the data range parameter. Therefore, the error estimate in this uncertainty analysis should be regarded as unreliable. Using a linear regression instead of the second order least squares regression did result in a more plausible estimate of  $\phi_0$ , which decreased the uncertainty. However, it also resulted in a slight increase in the difference between the regression curve and the data points,  $|\phi_i - \phi_{fit}|$ . Overall, the uncertainties were lower for the linear polynomial regression than with the second order least squares regression. Furthermore, the curves shown in Figure 28 seem to fit the data points better than the second order curves as shown in subsection 5.8

In the end, the conclusion from the verification exercise is that the error estimate in the uncertainty analysis is unreliable and that the numerical uncertainties are very high for  $C_D$ ,  $C_L$  and  $C_K$ . This has an effect on the reliability of the results of this thesis. It could be argued that, because of the high and unreliable uncertainties, the results of the CFD simulations themselves are unreliable. Therefore, the results and conclusions of this thesis should be used carefully and viewed with this criticism in mind. To improve the credibility of the results, the numerical uncertainty and the reliability of the error estimate should be improved.

Table 4: Uncertainty  $C_D$ . *Data is left out of the public version intentionally due to confidentiality*

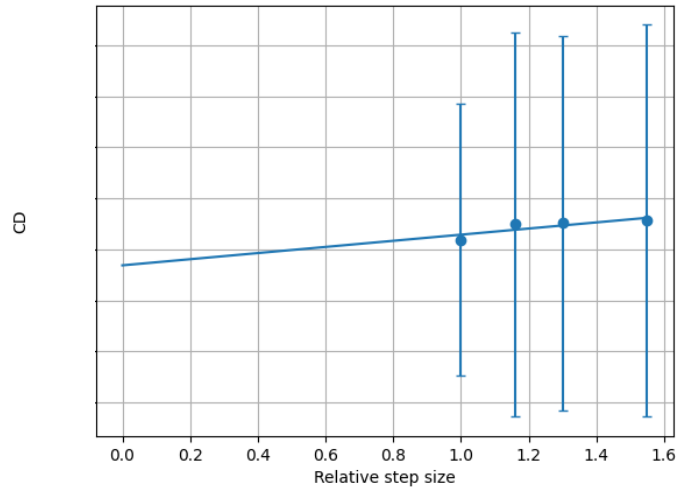
| $C_D$  | $h_i$ | $\phi_0$ | $\phi_i$ | $U_\phi$ |
|--------|-------|----------|----------|----------|
| Mesh 5 | 1.00  |          |          | 30.2%    |
| Mesh 4 | 1.16  |          |          | 44.2%    |
| Mesh 3 | 1.30  |          |          | 43.3%    |
| Mesh 2 | 1.55  |          |          | 45.5%    |

Table 5: Uncertainty  $C_L$ . *Data is left out of the public version intentionally due to confidentiality*

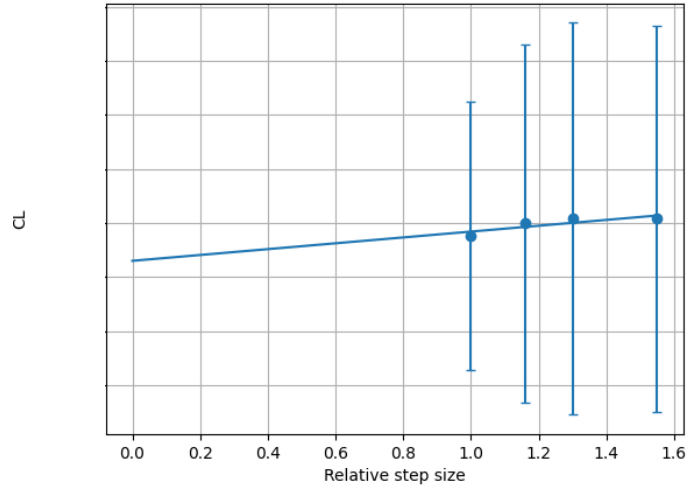
| $C_L$  | $h_i$ | $\phi_0$ | $\phi_i$ | $U_\phi$ |
|--------|-------|----------|----------|----------|
| Mesh 5 | 1.00  |          |          | 22.1%    |
| Mesh 4 | 1.16  |          |          | 30.1%    |
| Mesh 3 | 1.30  |          |          | 33.2%    |
| Mesh 2 | 1.55  |          |          | 32.7%    |

Table 6: Uncertainty  $C_K$ . *Data is left out of the public version intentionally due to confidentiality*

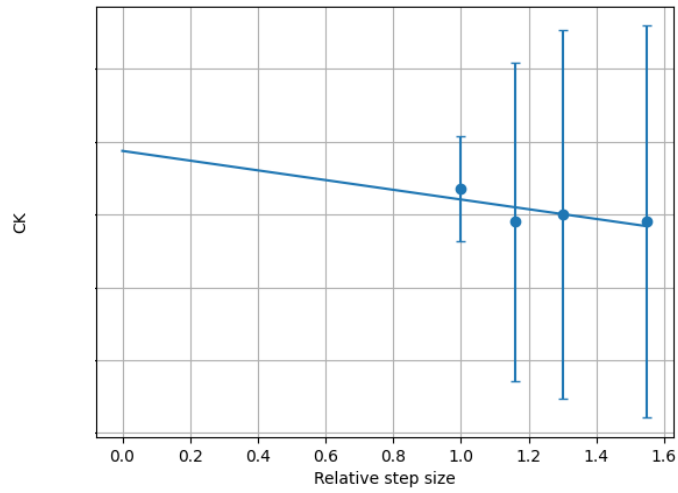
| $C_K$  | $h_i$ | $\phi_0$ | $\phi_i$ | $U_\phi$ |
|--------|-------|----------|----------|----------|
| Mesh 5 | 1.00  |          |          | 7.47%    |
| Mesh 4 | 1.16  |          |          | 21.6%    |
| Mesh 3 | 1.30  |          |          | 25.3%    |
| Mesh 2 | 1.55  |          |          | 26.7%    |



(a) Uncertainty  $C_D$



(b) Uncertainty  $C_L$



(c) Uncertainty  $C_K$

Figure 28: Uncertainties from verification exercise plotted with linear polynomial regression

## 7 Validation

Validation is an exercise where the modelling error is computed in order to determine how accurately the model matches reality. Equation 24 shows how the model will be validated.  $E$  is the comparison error between the experimental results and the CFD.  $\phi$  refers to a flow characteristic. In this case the lift- and drag forces will be used.  $U_\phi$  is the numerical uncertainty. It is explained in section 6 how this uncertainty is calculated.  $U_{Exp}$  is the experimental- or model test uncertainty. The values used for  $U_{Exp}$  were calculated by Vink [36]. In subsection 7.2, it is explained how these model tests were conducted and what the uncertainty is. The input parameter uncertainty is not taken into account in  $U_{Val}$ . This is because, in order to calculate it, unknown quantities in the input settings of a flow problem should be known. Because these quantities are unknown, this term is assumed to be zero.

$$\begin{aligned}
 E &= \frac{\phi_{Exp} - \phi_i}{\phi_{Exp}} \\
 U_{Val} &= \sqrt{U_\phi^2 + U_{Exp}^2} \\
 \text{Validated if : } E &< U_{Val}
 \end{aligned} \tag{24}$$

The meshes will be validated using this approach at 15 degrees drift. Afterwards, one mesh will be simulated at 0, 5, 15, 30, 40 and 45 degrees drift. Using the numerical uncertainties from the 15 degrees drift simulation, and the experimental uncertainties from 7.2, a comparison will be made for these drift angles between the CFD and the model test. This section will give insight in which of the meshes are validated for which parameters. This will help select a mesh to use in the simulations for subsection 8.1. Furthermore, this chapter will give insight in how trustworthy the results from this thesis are based on their uncertainties and difference to the model tests.

### 7.1 Validation case

The model tests that are used as validation material in this research were conducted in the deep water towing tank at MARIN (Maritime Research Institute Netherlands). These were captive model tests that were conducted with a model of the RSD 2513 with a scale ratio  $\lambda$  of 1:12.189. The tests were conducted for a bare hull model, bare hull model with fins and fully appended model. For the validation case the bare hull model with fins was used. Figure 29 shows the fully appended model used in the captive model tests. The main particulars are shown in Table 7

*Table 7: Model test particulars. Table is left out of the public version intentionally due to confidentiality*



Figure 29: Model of the RSD 2513 used in the model test [15]

## 7.2 Model tests uncertainty

The uncertainty in model tests is an accumulation of uncertainties throughout the entire experiment. It is an extensive effort to determine all of the uncertainty terms contributing to the total model test uncertainty. Vink undertook this effort for the same model test as used in this research for validation [36]. To compute the total uncertainty, Vink determined the statistical-, ballasting-, model speed-, 6-component frame-, temperature- and drift angle uncertainty. This was done for surge force, sway force and yawing moment for a range of drift angles between 0 and 180 degrees. Table 8 shows the results from this calculation by Vink. The uncertainties for the surge- and sway force are converted to uncertainties of the drag- and lift force by the method shown in Equation 25

$$\begin{aligned} U_{Exp L} &= \frac{U_{Exp Y} \cdot C_Y \cos(\beta) - U_{Exp X} \cdot C_X \sin(\beta)}{C_L} \\ U_{Exp D} &= \frac{U_{Exp Y} \cdot C_Y \sin(\beta) + U_{Exp X} \cdot C_X \cos(\beta)}{C_D} \end{aligned} \quad (25)$$

It should be noted that Vink calculated these uncertainties for the bare hull model test without fins. However, the simulations in this research are for the RSD tug with fins. On one hand, the fins caused flow separation and could therefore be expected to create additional statistical uncertainty and therefore experimental uncertainty. This is because the flow separation could make the measured force signal more irregular and unsteady. However, it could also be argued that the flow separation would be more well defined and therefore the measured force signal would be less irregular and unsteady. This would mean the statistical uncertainty would be lower for a hull with fins. A similar phenomenon was observed by Vink when comparing higher drift angles uncertainty with lower drift angles. It is therefore deemed more likely that the uncertainty will be lower for the model tests with fins.

The uncertainty of the roll coefficient is equal to the uncertainty of the  $Y$  force. Vink decided to set the uncertainty of a moment equal to the highest uncertainty of the two forces that are causing the moment. The reason he did that is because there are no time traces of the moments and therefore the uncertainty of the mean could not be obtained. For the roll moment, this would mean that the uncertainty is equal the highest uncertainty of either the  $Y$  force, or the  $Z$  force at that drift angle. However, the uncertainty of the  $Z$  force is not known. Therefore, the uncertainty of the  $Y$  force is equal to that of the roll moment. This means that neither the uncertainty of the  $Z$  force,

nor the uncertainty of the application point of the  $Z$ - and  $Y$  forces is taken into account. This is unfortunate, because it would add to the credibility of the experimental uncertainty and therefore the validation exercise, if those were taken into consideration when determining the uncertainty of the roll moment.

Table 8: Experimental uncertainty of  $X$ -,  $Y$ -, lift and drag force for 0 to 45 degrees [36]

| $\beta$     | 0      | 5      | 15    | 30     | 45     |
|-------------|--------|--------|-------|--------|--------|
| $U_{Exp X}$ | 9.69%  | 10.39% | 9.98% | 10.08% | 13.45% |
| $U_{Exp Y}$ | 12.54% | 26.84% | 9.20% | 7.80%  | 7.69%  |
| $U_{Exp D}$ | 9.69%  | 13.29% | 9.38% | 7.85%  | 8.49%  |
| $U_{Exp L}$ | 12.54% | 28.22% | 9.62% | 7.93%  | 11.73% |
| $U_{Exp K}$ | 12.54% | 26.84% | 9.20% | 7.80%  | 7.69%  |

### 7.3 Validation results

The model was validated using the numerical uncertainty from subsection 6.1 and the experimental uncertainty from subsection 7.2. Equation 24 was used to calculate the validation uncertainty and the comparison error between the experiment and the CFD. Table 9 shows the results of these calculations. The last column shows if meshes comply with the validation criterion. From that last column, it can be seen that for mesh 2 and 3 the criterion is met for  $C_D$ ,  $C_L$  and  $C_K$ . For mesh 4 and 5 the criterion is met for  $C_D$  and  $C_L$ , but not for  $C_K$ .

The fact that all meshes comply with the criterion for  $C_D$  is largely due to the high numerical uncertainty  $U_\phi$ . This causes the validation uncertainty to be very high as well. All the while the comparison error for  $C_D$  is not very small, ranging between 13% and 17%. This is enough cause for serious scepticism on whether the meshes are really validated for  $C_D$ .

For  $C_L$ , the numerical uncertainty and therefore the validation uncertainty is still high. However, contrary to  $C_D$ , the comparison error is very low for  $C_L$  at 15 degrees drift. This gives a little more validity to the validation of the meshes for  $C_L$  at this drift angle, since it could be said that it is not solely due to the high uncertainty that the validation criterion is met.

$C_K$  shows the lowest validation uncertainties for all meshes. This is combined with the highest comparison error. Since the validation uncertainty decreases with mesh refinement and the comparison error increases, the validation criterion is not met for mesh 4 and 5. However, the criterion is only just met for mesh 2 and 3, and with high uncertainties and comparison errors. Again, it is doubtful that mesh 2 and 3 should be seen as validated. Especially since the more refined meshes are not validated according to the criterion.



Table 9: Validation exercise for mesh 2 to 5 at 15 degrees drift

| $\beta = 15$ |       | $U_\phi$ | $U_{Exp}$ | $U_{val}$ | E     | $U_{val} > E$ |
|--------------|-------|----------|-----------|-----------|-------|---------------|
| Mesh 2       | $C_D$ | 45.5%    | 9.38%     | 46.5%     | 16.6% | ✓             |
|              | $C_L$ | 32.7%    | 9.62%     | 34.1%     | 2.40% | ✓             |
|              | $C_K$ | 26.7%    | 9.20%     | 28.2%     | 25.0% | ✓             |
| Mesh 3       | $C_D$ | 43.3%    | 9.38%     | 44.3%     | 16.3% | ✓             |
|              | $C_L$ | 33.2%    | 9.62%     | 34.6%     | 2.24% | ✓             |
|              | $C_K$ | 25.3%    | 9.20%     | 26.9%     | 25.7% | ✓             |
| Mesh 4       | $C_D$ | 44.2%    | 9.38%     | 45.2%     | 16.0% | ✓             |
|              | $C_L$ | 30.1%    | 9.62%     | 31.6%     | 1.55% | ✓             |
|              | $C_K$ | 21.6%    | 9.20%     | 23.5%     | 25.0% | ×             |
| Mesh 5       | $C_D$ | 30.2%    | 9.38%     | 31.6%     | 13.3% | ✓             |
|              | $C_L$ | 22.1%    | 9.62%     | 24.1%     | 0.44% | ✓             |
|              | $C_K$ | 7.47%    | 9.20%     | 11.9%     | 28.2% | ×             |

The verification exercise was extended to multiple drift angles for mesh 2. Mesh 2 was chosen because it is one of the meshes that passed all the criterion and, according to [Table 3](#) it has a relatively small comparison error to the more refined meshes. Especially when compared to the numerical uncertainty. Since mesh 2 has less cells than the other meshes, using this mesh for the simulations at multiple drift angles would require the least computational time. Furthermore, since the comparison error between mesh 2 and the other meshes is small compared to the numerical uncertainty, it was not expected to have added value to do this validation exercise for all meshes. [Table 10](#) shows  $C_D$ ,  $C_L$  and  $C_K$  of mesh 2 for multiple drift angles, and the comparison error with the experimental results. The comparison error is calculated for the CFD results with- and without the trim- and sinkage correction from [subsection 5.9](#).  $C_K$  does not have a trim- and sinkage correction, so only one set of comparison errors is given. The highest comparison errors seem to occur at 0 and 5 degrees drift. This is especially true for  $C_L$  and  $C_K$  at 0 degrees where the comparison error is extremely high compared to the other drift angles. Another thing to notice is the low comparison error for  $C_L$  at 15 degrees drift for the CFD with trim- and sinkage correction. Since it is so much lower than the other comparison errors, it gives some more perspective to the earlier statement about [Table 9](#). Here it was noted that the low comparison error for  $C_L$  meant that validation of  $C_L$  could be seen as more acceptable since the comparison error was very low. However, seeing that this is not the case at other drift angles contradicts this statement.

Table 10: Comparison error for  $C_D$ ,  $C_L$  and  $C_K$  for CFD with- and without trim- and sinkage correction, and the model tests. Computed for mesh 2 at a range of drift angles. **Data is left out of the public version intentionally due to confidentiality**

| $\beta$ |               | 0     | 5     | 15    | 30    | 45    |
|---------|---------------|-------|-------|-------|-------|-------|
| $C_D$   | Model test    |       |       |       |       |       |
|         | CFD           |       |       |       |       |       |
|         | E             | 38.5% | 34.4% | 26.7% | 6.6%  | 5.47% |
|         | CFD corrected |       |       |       |       |       |
|         | E corrected   | 38.5% | 32.6% | 16.7% | 6.7%  | 7.92% |
| $C_L$   | Model test    |       |       |       |       |       |
|         | CFD           |       |       |       |       |       |
|         | E             | 96.5% | 27.4% | 15.7% | 43.6% | 29.4% |
|         | CFD corrected |       |       |       |       |       |
|         | E corrected   | 96.0% | 16.4% | 2.47% | 29.9% | 10.9% |
| $C_K$   | Model test    |       |       |       |       |       |
|         | CFD           |       |       |       |       |       |
|         | E             | 90.9% | 41.6% | 25.0% | 29.9% | 28.9% |

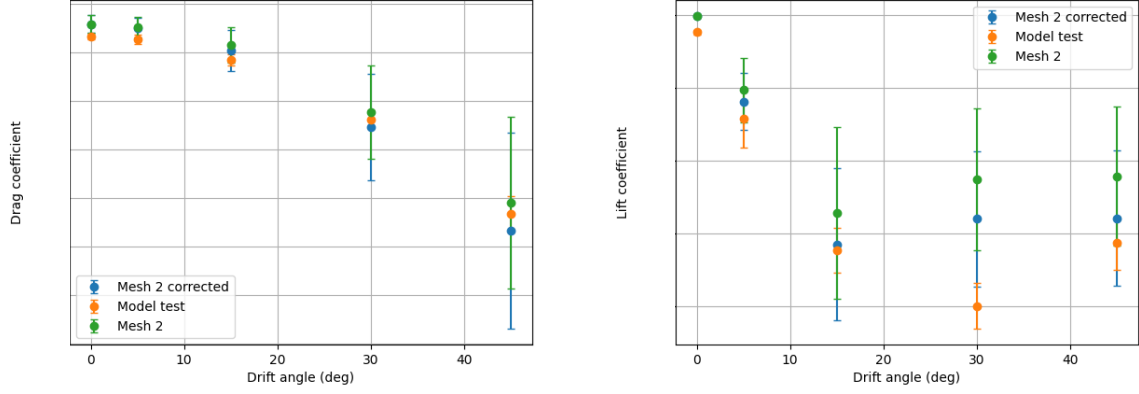
The values from Table 10 are plotted against the model test results in Figure 30a, Figure 30b and Figure 30c. The values come with error bars for the numerical uncertainty (calculated at 15 degrees drift) and the experimental uncertainty from Table 8. There are two plots from CFD results. The green data is the CFD results without the trim- and sinkage correction from subsection 5.9, and the blue data is the CFD results with the trim- and sinkage correction. For  $C_K$ , there is only one plot from the CFD results since the trim- and sinkage correction does not influence  $C_K$ .

Firstly, it should be noted that all values for the CFD with trim- and sinkage correction fall within the error bar of the model, except  $C_L$  at 0 drift and  $C_K$  at 30 and 45 degrees drift. For the CFD without the trim correction, the same is true with the addition of  $C_L$  at 30 degrees drift not falling within the error bars of the model tests. However, as mentioned, most values fall into the error bars because of the numerical uncertainty being really high.

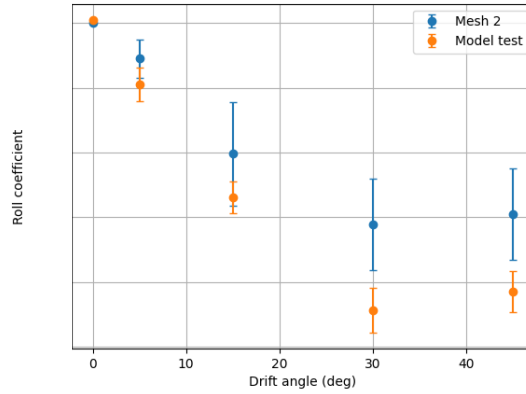
Furthermore, the absolute difference in CFD- and experimental results is quite high at higher drift angles for  $C_K$  and  $C_L$ . This can be explained by the fact that increasing the drift angle increases turbulence and separation in the flow around the ship, which is difficult to simulate accurately. When looking at Figure 30, the CFD calculated  $C_L$  at 30 degrees drift is significantly more off the experimental value than at other drift angles. This could be due to the separation of the flow occurring earlier in the CFD simulations than at the model tests. The separation of the CFD results is plotted in Figure 31. It shows how there is very little separation at 15 degrees drift and lower, compared to 30 and 45 degrees. In Figure 32 the turbulent kinetic energy is plotted at cut planes at several locations on the tug for 15, 30 and 45 degrees drift. The difference in separation and turbulence between 15 degrees drift and 30- and 45 degrees drift is immediately visible. This confirms that the turbulence is most likely the cause of the large discrepancy between the CFD- and experimental results at 30- and 45 degrees drift. From Figure 32 it is visible that at 45 degrees drift, the area of turbulent flow is bigger but the turbulent kinetic energy in that area is lower than at 30 degrees drift. This could be an explanation why the discrepancy between the CFD- and experimental results is higher at 30 degrees drift than at 45 degrees drift. This is because the turbulence is expected to be more irregular and unsteady at the areas with higher turbulent kinetic energy since it is a measure for the intensity of the turbulence [6].

Overall it could be concluded from Figure 30a that the CFD quite accurately compares to the model tests for  $C_D$ . This is both true in terms of the trend of the curve, and the absolute values. For  $C_L$  this is true for most values. However, at 30 degrees drift the trend seems to deviate from the

model tests. Figure 30c shows that for  $C_K$  the trend of the model tests is followed quite accurately. However, at 30 and 45 degrees drift, the absolute difference between the CFD and the model tests seems to be noticeably higher than at lower drift angles. Those are also the drift angles for which the error bars of the CFD don't fall into the error bars of the model tests.



(a) Drag coefficient of mesh 2 with and without trim- and sinkage correction vs. model test results with and sinkage correction vs. model test results with error bars



(c) Roll coefficient of mesh 2 with and without trim- and sinkage vs. model test results with error bars

Figure 30: Comparison mesh 2 with and without trim- and sinkage correction and model test with uncertainties from the second validation exercise

The conclusion of the validation exercises is that the credibility of the results is hindered by the high numerical uncertainty. Though most times the validation criterion is met in Table 9 it is mostly due to the validation uncertainty being so high that the comparison error can be really high as well. From Table 9 and Table 10 it is clear that the comparison error often is quite high. It is likely that validation criterion would not be met if the numerical uncertainty would be lower. The high numerical uncertainty is also shown in Figure 30. The numerical error bars that are plotted here, are so large that they almost always fall within the error bars of the model tests. Because of this, the model should not be considered completely validated. However, it is positive that for the most part the CFD follows the same trend as the model tests.

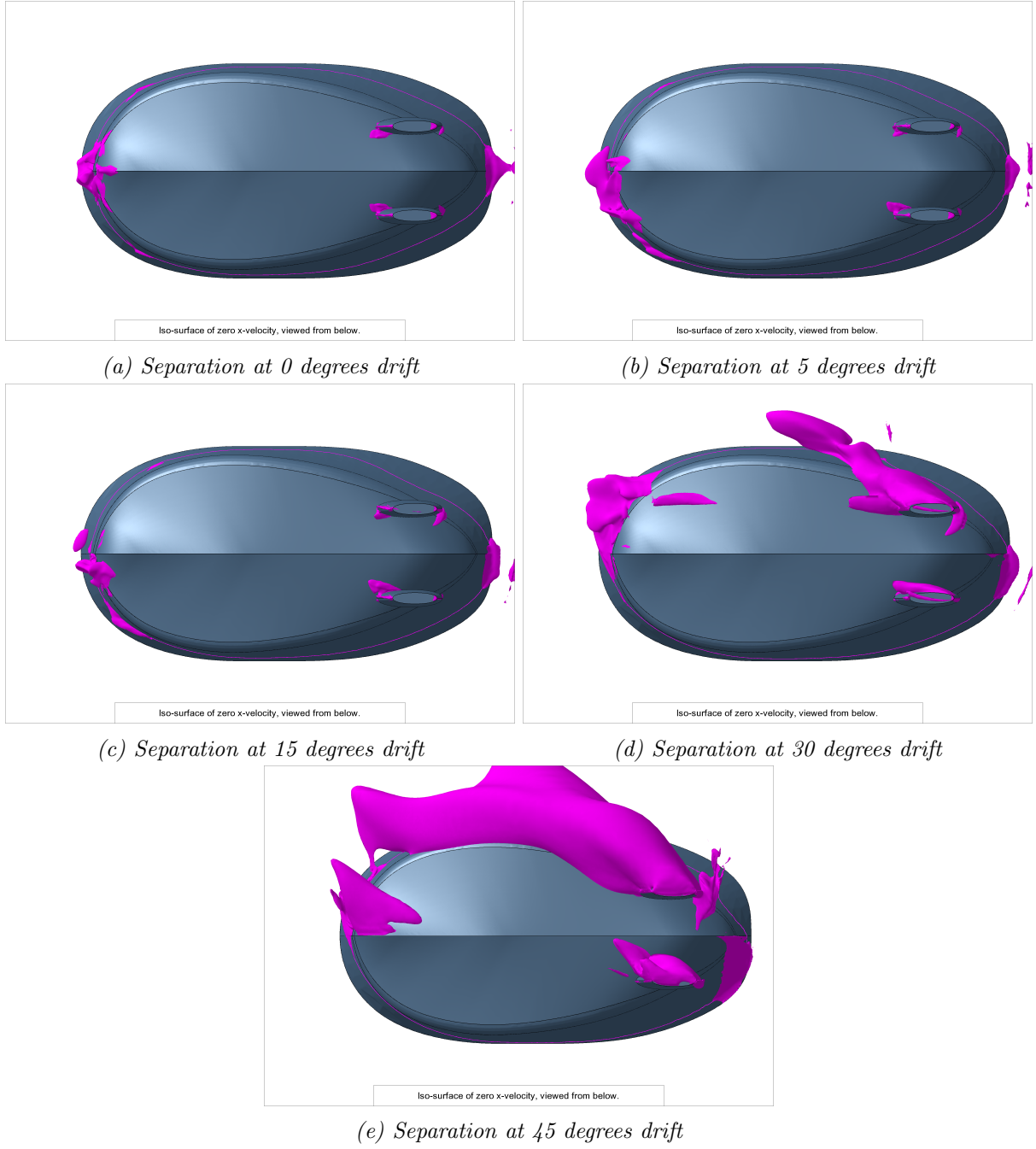
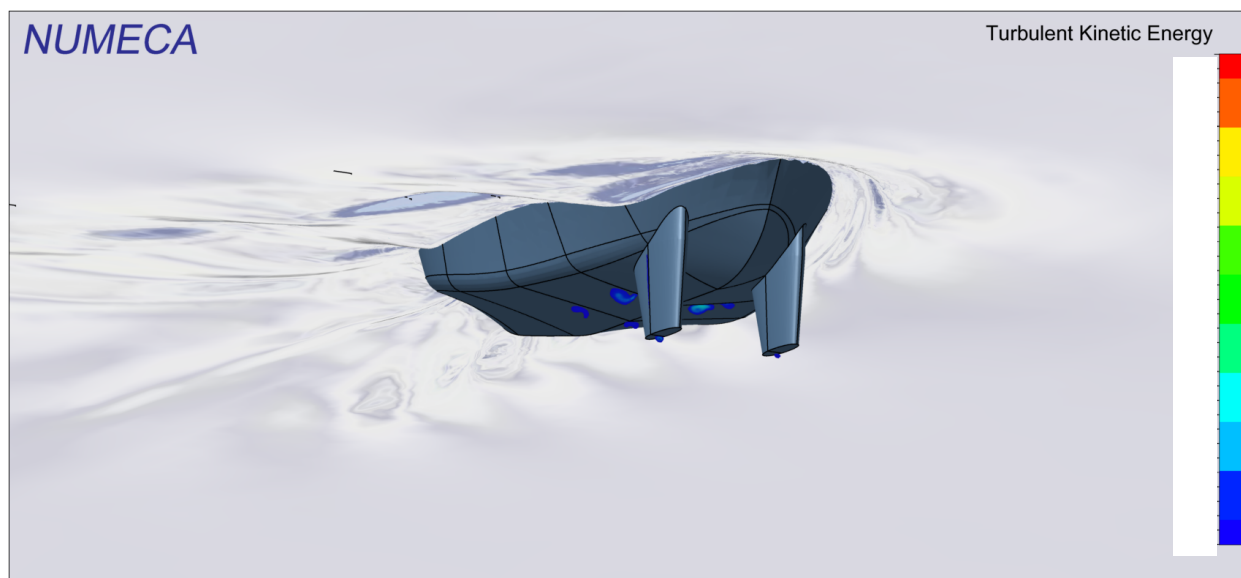
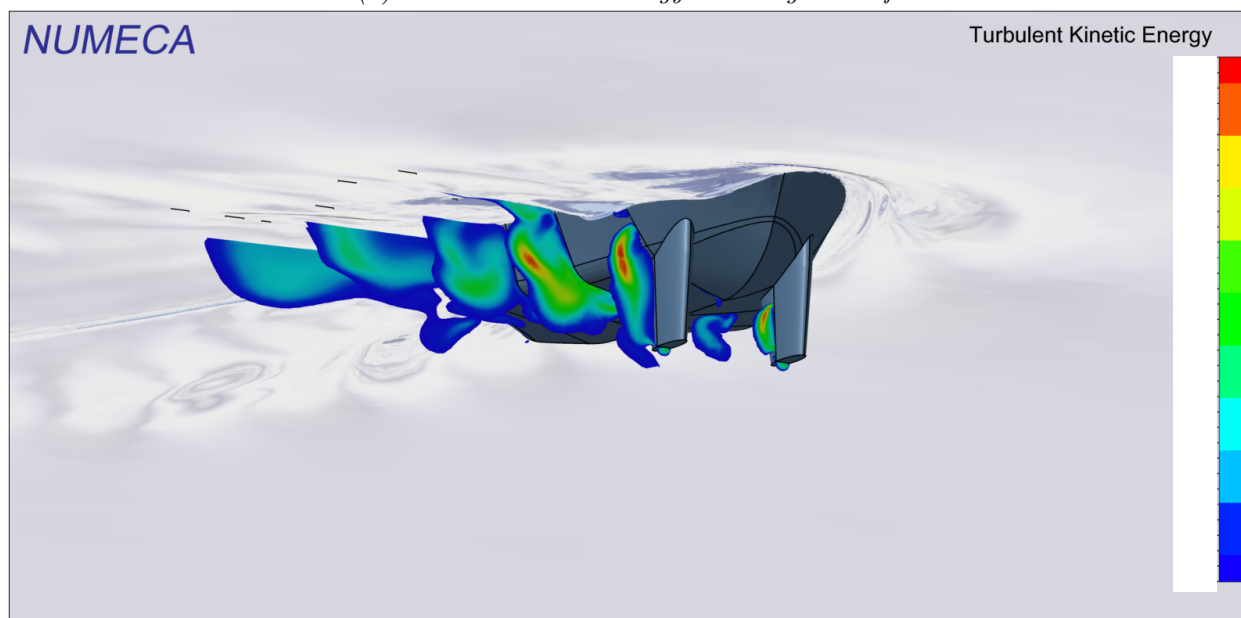


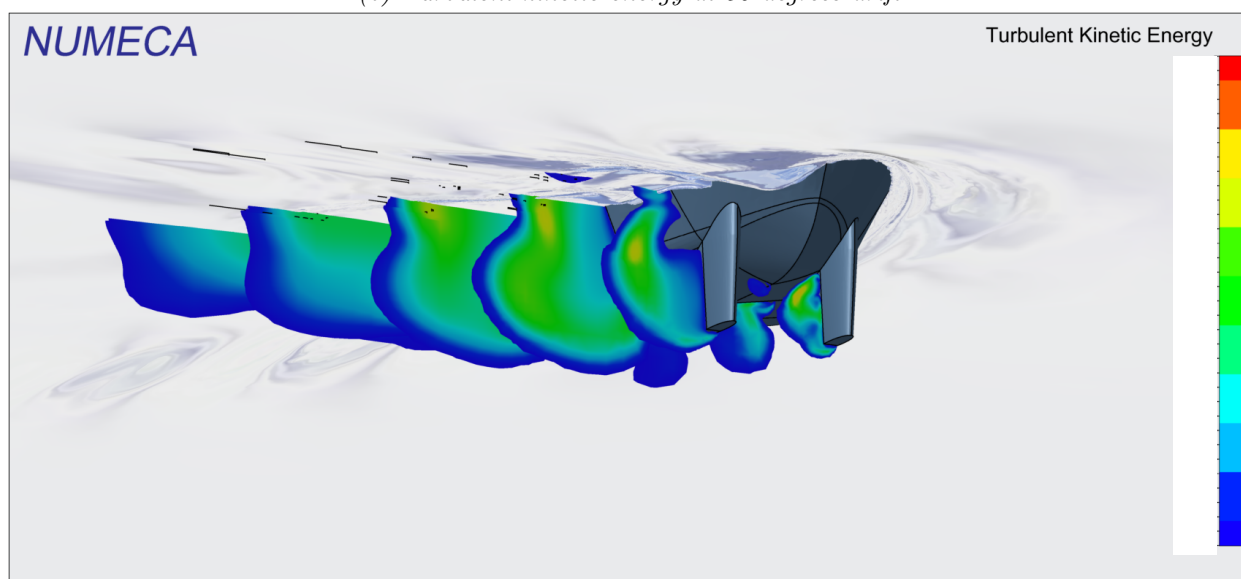
Figure 31: Separation computed using mesh 2 at multiple drift angels



(a) Turbulent kinetic energy at 15 degrees drift



(b) Turbulent kinetic energy at 30 degrees drift



(c) Turbulent kinetic energy at 45 degrees drift

Figure 32: Turbulent kinetic energy at cut planes at different locations on the tug at 0, 30 and 45 degrees drift

## 8 Results

After doing verification and validation exercises, mesh 2 was used to simulate the RSD 2513 at a series of drift- and heeling angles at 8 knots forward speed. The reason why mesh 2 was chosen is that it had a small comparison error to the other meshes, performed relatively well in the verification and validation compared to the other meshes, and has a relatively few amount of cells compared to the other meshes. The simulations were run at 0, 15 , 30 and 45 degrees drift, and at 0, 2, 4, 6, 8 and 10 degrees heel. Important sections to review before reading this chapter are subsection 5.10 and subsection 4.6 which cover the methods used to create the curves. Furthermore, section 6 and subsection 7.3 give insight in the uncertainties and validity of the results used in this chapter. After this chapter, the results will be discussed and conclusions and recommendations will be drawn from them. This will be described in section 9, section 10 and section 11

### 8.1 Results

In this section, a differentiation will be made between three moments. These moments are the righting moment, the heeling moment and the restoring moment. The definition of these moments is as follows.

- **Righting moment:** The righting moment is a moment that is working in positive direction around the x axis. It is therefore working to return the ship to zero heel. It consists of a component created by the static z force,  $F_{zstat}$ , called the static component of the righting moment, and a component created by the dynamic z force,  $F_{zdyn}$ , called the dynamic component of the righting moment. A schematic is given in Figure 46b.
- **Heeling moment:** The Heeling moment is a moment that is working in negative direction around the x axis. It is therefore working to increase the ship's heel. It consists of one component created by the dynamic y force,  $F_{ydyn}$ , called the dynamic component of the heeling moment. A schematic is given in Figure 46c.
- **Restoring moment:** The restoring moment is a moment that is working in positive direction around the x axis. It is therefore working to return the ship to zero heel. It consists of one component created by the static z force,  $F_{zstat}$ . In classic ship stability it is calculated using the GZ curve from SARC PIAS, not CFD. The restoring moment is always calculated for 0 knots forward speed. A schematic is given in Figure 46a.

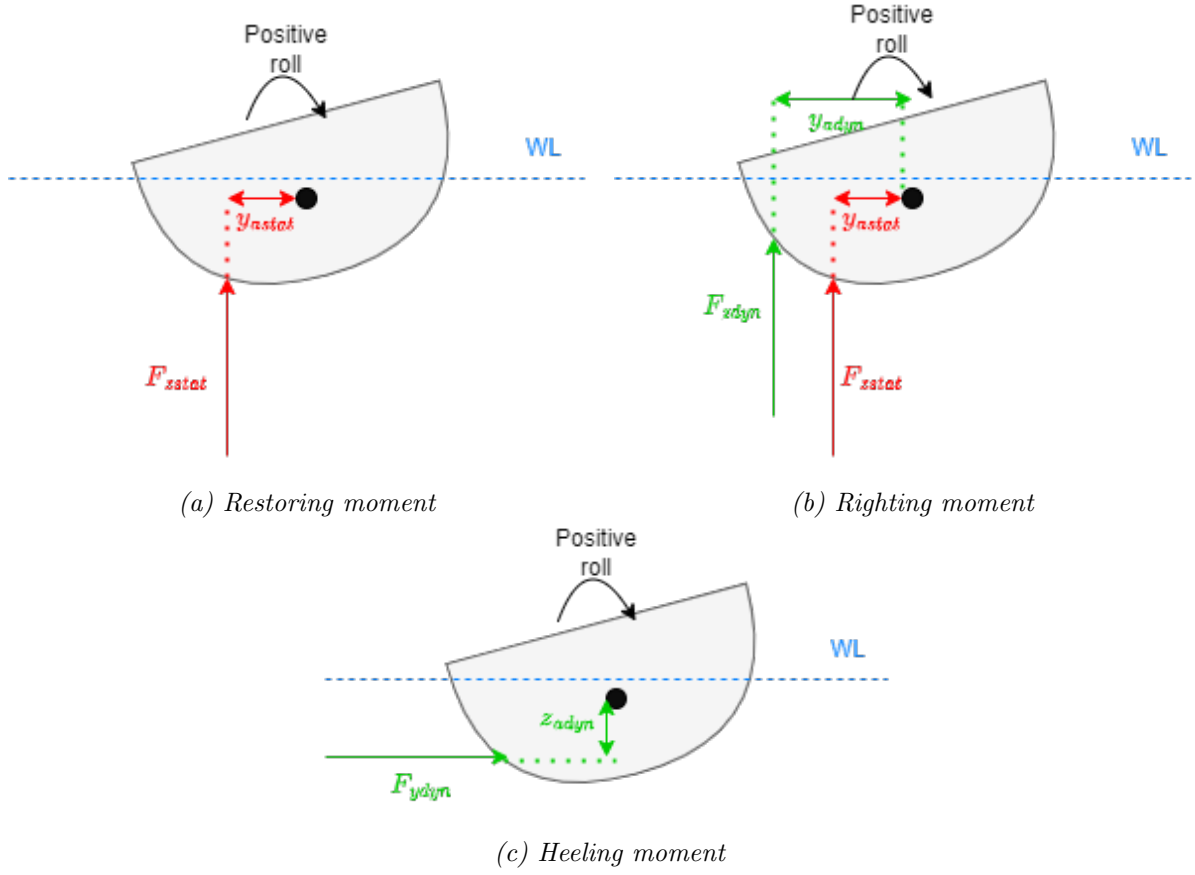
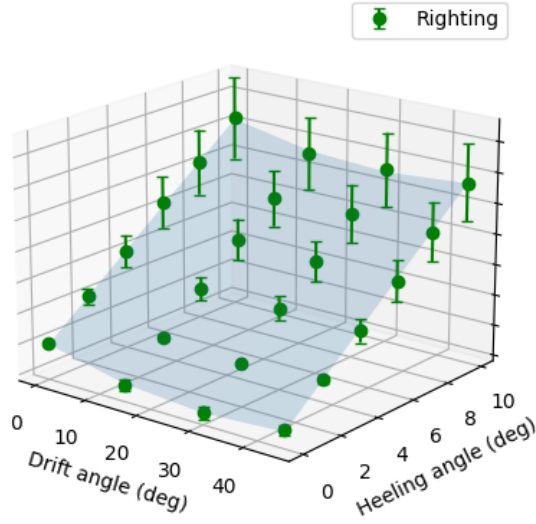


Figure 33: Schematic of the definition of the restoring-, righting- and heeling moment

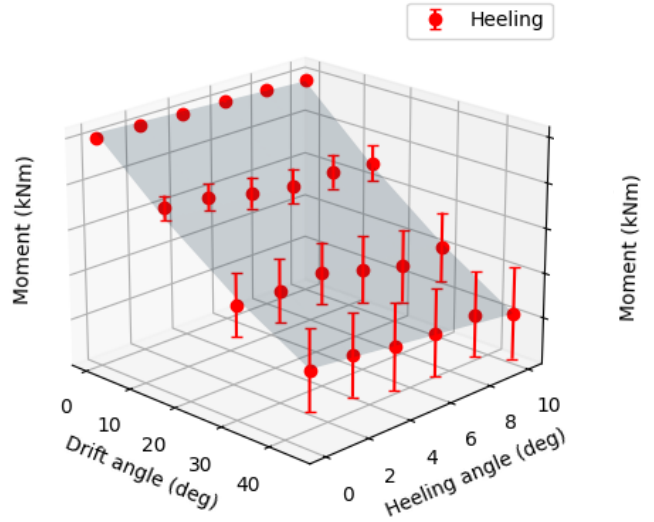
Using the methods described in subsection 5.10 and subsection 4.6, the righting- and heeling moment were calculated. Figure 34c shows the scatter of these curves with a polynomial regression. As mentioned in subsection 4.6 the biggest drawback of the polynomial regression was its sensitivity to outliers. After inspecting the data, no outliers were detected in the righting moment data, and only two relatively moderate outliers in the heeling data, making the polynomial regression a suitable technique. A second order polynomial was used as it was concluded that it fitted the data nicely after visual inspection. To obtain an estimate of the constants,  $a_n$ , in the polynomial regression formula, the least squares method was used [28]. Figure 34a and Figure 34b show only the righting- and heeling moment respectively to give a better view of its slope. In these figures, error bars are added to show the numerical uncertainty of the data points. Note that this is the uncertainty of the roll coefficient,  $C_K$ , at 15 degrees drift, 0 degrees heel and 8 knots forward speed as calculated in subsection 6.1. The uncertainty for every point is therefore the same, namely 26.7%. This is therefore not the exact numerical uncertainty for every drift- and heeling angle. However, it shows how large the uncertainty is, assuming the uncertainty is constant for all drift- and heeling angles. From Figure 34a and Figure 34b it is clear that the absolute uncertainty becomes quite large at the larger righting- and heeling moment values. Within that uncertainty, there is quite a large range where the exact answer could be expected to be. This could possibly change the trend of the righting- and heeling moment curve significantly.

The first thing to note about Figure 34 is that the heeling moment is negative and righting moment is positive for the majority of values. This means that the two moments were working in opposite direction as was expected. Figure 34d shows the righting moment due to the dynamic force in z direction compared to the total righting moment.

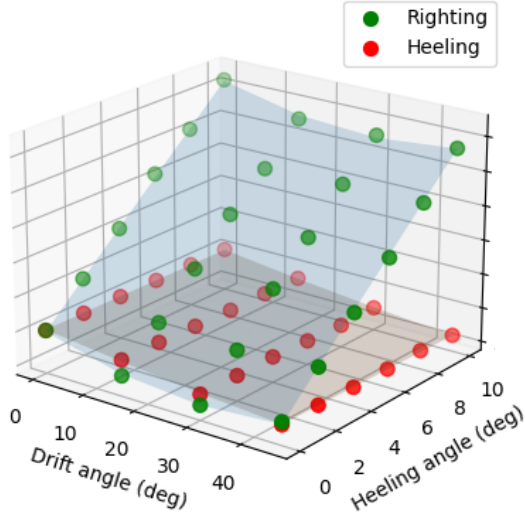




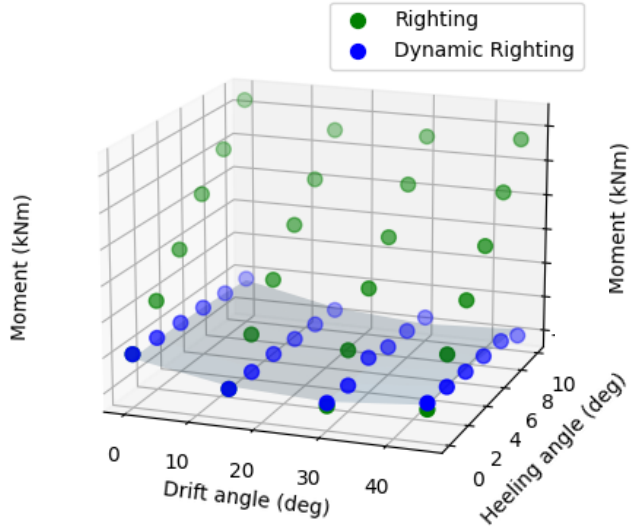
(a) Righting moment



(b) Heeling moment



(c) Righting- and heeling moment



(d) Righting moment due to  $F_{zdyn}$  compared to total righting moment

Figure 34: Heeling- and righting moment between 0 and 45 degrees drift and 0 to 10 degrees heel, at 8 knots forward speed

When looking at the influence of the heeling angle on the righting moment, it is clear that increasing the heeling angle increases the righting moment. This is a familiar phenomenon in ship stability and was expected to happen. A heeling angle shifts the centre of buoyancy to the side of the ship, which leads to a righting moment. This increase in righting moment was due to the static force in  $z$  direction. The dynamic force in  $z$  direction created a negative moment working in the opposite direction. This was due to the force being negative. A plausible explanation for this is that, by increasing the flow's velocity, the pressure it exerts on the hull decreases as can be exemplified by Bernoulli's equation in [Equation 26](#). In this equation,  $\varphi$  represents the pressure due to body forces. An example of a body force can be gravity working on the body. Let's assume, the pressure due to the body forces remain constant. It can then be concluded that, if the velocity  $|u|$  increases, the pressure  $p$  decreases. However, the moment due to the dynamic force in  $z$  direction

was significantly smaller than that of the static force in z direction. This resulted in the overall righting moment being positive.

$$p + \frac{1}{2} \cdot \rho \cdot |\underline{u}|^2 + \varphi = \text{constant along a streamline} \quad (26)$$

The influence of the forward speed could be seen by examining the righting moment at zero degrees drift and 8 knots forward speed, and comparing that to the restoring moment of an RSD 2513. This restoring moment is calculated without CFD but with the GZ curve of the RSD calculated with SARC PIAS. The restoring moment doesn't take dynamic forces into account. Therefore, the difference between the restoring- and righting moment is the forward speed. Figure 35 shows the restoring moment against the righting moment at 0 drift angle and 8 knots forward speed. The data points for the restoring moment are made visible and are named 'restoring moment data' in the legend. The righting moment points fit the restoring moment curve accurately and the difference between the two is minimal. Table 11 shows that the dynamic component of the righting moment increases when the heeling angle is increased. This is mostly due to the fact that the  $y_{dyn}$  increases slightly when the ship is heeled. There is also a very slight and less significant increase in  $F_{zdyn}$ .

The dynamic component of the righting moment at 0 degrees drift is very small compared to the static component of the righting moment for all heeling angles. This is shown in Table 11. The reason for this is partly that  $y_{dyn}$  is smaller than  $y_{astat}$ , but mostly that  $F_{zdyn}$  is smaller than  $F_{zstat}$ . This is visualised in Figure 36 and in Figure 37. In both figures, the hydrostatic- and hydrodynamic pressure in z direction are projected on the hull. In Figure 36 the scale is the same to show that the hydrostatic pressure is significantly larger than the hydrodynamic pressure. In Figure 37 the scales are different so that the symmetry, and therefore the arm for the righting moment, can be reviewed. If the pressure is completely symmetrical on the port side and starboard side of the hull, there is no arm for the righting moment. Although this can't be judged completely accurately by this figure alone, it does show that the hydrodynamic pressure is more symmetrical than the hydrostatic pressure. This visualises why  $y_{dyn}$  is smaller than  $y_{astat}$ .

However, Figure 35 also shows that the uncertainty of the righting moment is quite large. Especially at the larger heeling angles, the righting moment could actually be far off the value of the restoring moment. The observation of the effect of the heeling angle on the righting moment is therefore somewhat weakened due to large numerical uncertainty.

*Table 11: Ratio between dynamic component- and static component of the restoring moment at 0 degrees drift and 8 knots forward speed. Data is left out of the public version intentionally due to confidentiality*

| Heeling angle (deg)     | 2    | 4    | 6    | 8    | 10   |
|-------------------------|------|------|------|------|------|
| Dynamic component (kNm) |      |      |      |      |      |
| Static component (kNm)  |      |      |      |      |      |
| ratio (Dynamic/Static)  | 3.4% | 4.3% | 3.9% | 4.0% | 3.7% |

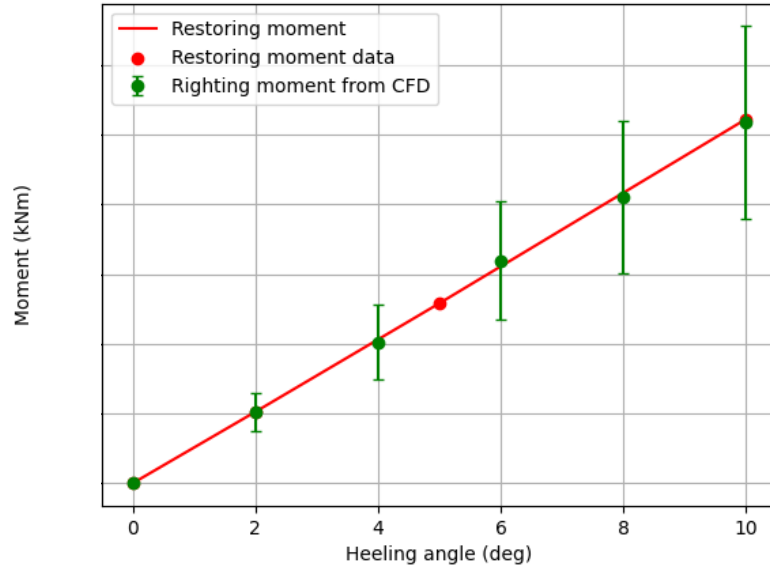


Figure 35: Restoring moment RSD 2513 vs righting moment from CFD at 0 degrees drift and 8 knots forward speed

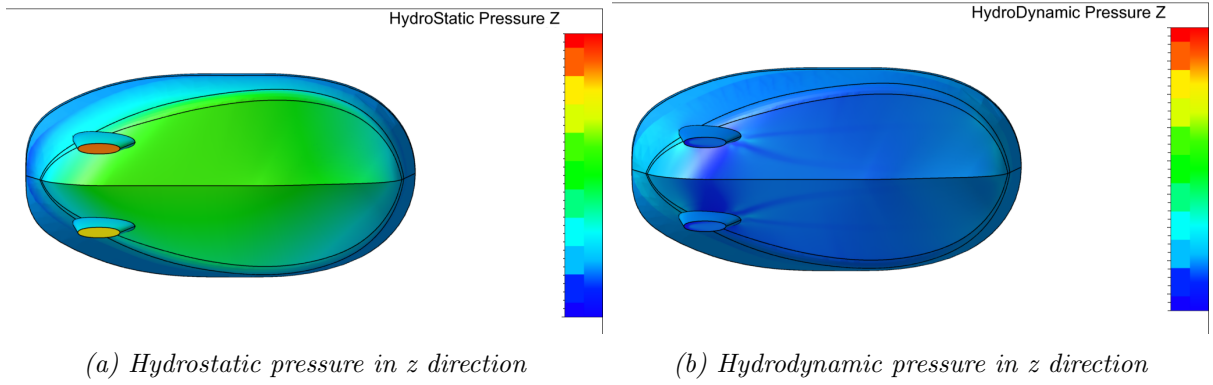


Figure 36: Comparison between the hydrostatic pressure in z direction and the hydrodynamic pressure in z direction at 0 degrees drift, 8 degrees heel and 8 knots forward speed

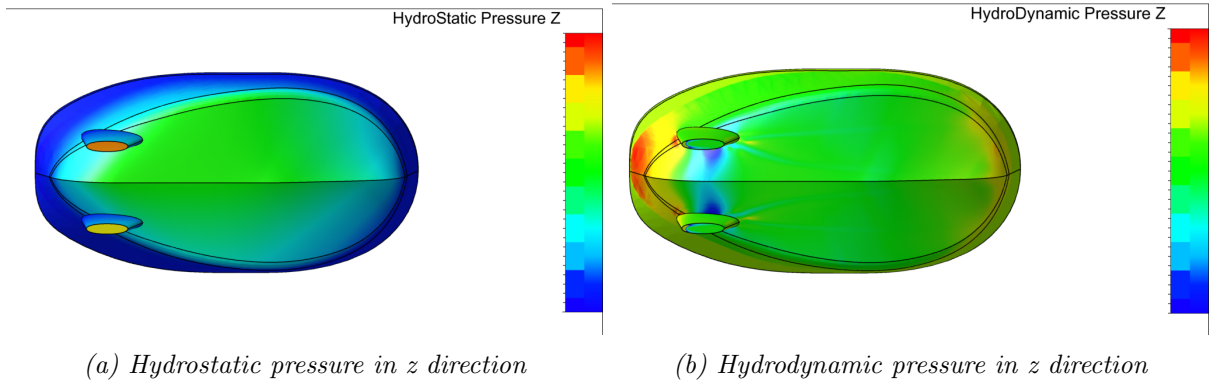


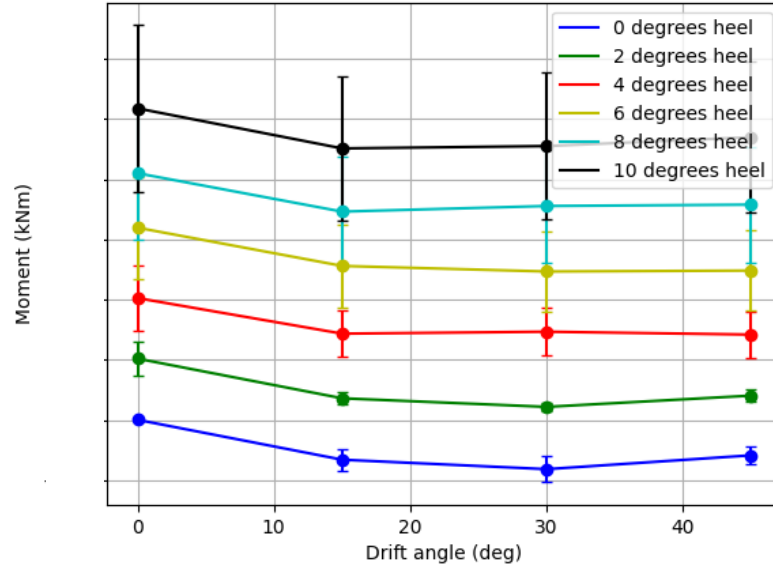
Figure 37: Hydrostatic- and hydrodynamic pressure in z direction scaled so that the symmetry in pressure field on the hull can be compared

Lastly, it should be noticed that the righting moment became smaller when a drift angle was

introduced. This is shown in [Figure 38](#). The initial drop from 0 to 15 degrees drift is due to a sharp increase in the dynamic component of the righting moment. This is illustrated in [Table 12](#). After 15 degrees the dynamic component of the righting moment starts to decrease for most heeling angles as shown in the table. This can also be seen in [Figure 34a](#) by the fact that the curve starts to slightly curve upwards again. A possible explanation for that could be that heavy separation starts to occur at 30 degrees drift. Again, the uncertainties shown in [Figure 38](#) are quite large for the higher heeling angles. This should be kept in mind when drawing conclusions from these results.

*Table 12: Dynamic component of the righting moment at 0, 15 and 30 degrees drift in kNm.  
Data is left out of the public version intentionally due to confidentiality*

| Heeling angle (deg) | 2 | 4 | 6 | 8 | 10 |
|---------------------|---|---|---|---|----|
| 0 degrees drift     |   |   |   |   |    |
| 15 degrees drift    |   |   |   |   |    |
| 30 degrees drift    |   |   |   |   |    |

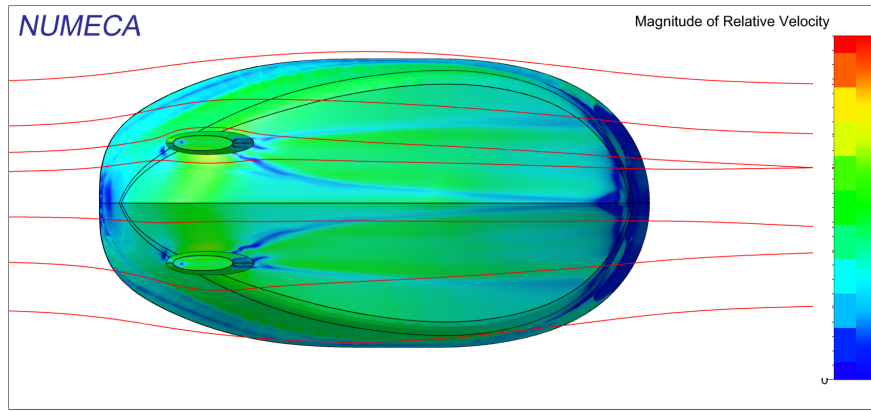


*Figure 38: Change in righting moment due to drift angle for heeling angles between 0 and 10 degrees at 8 knots forward speed*

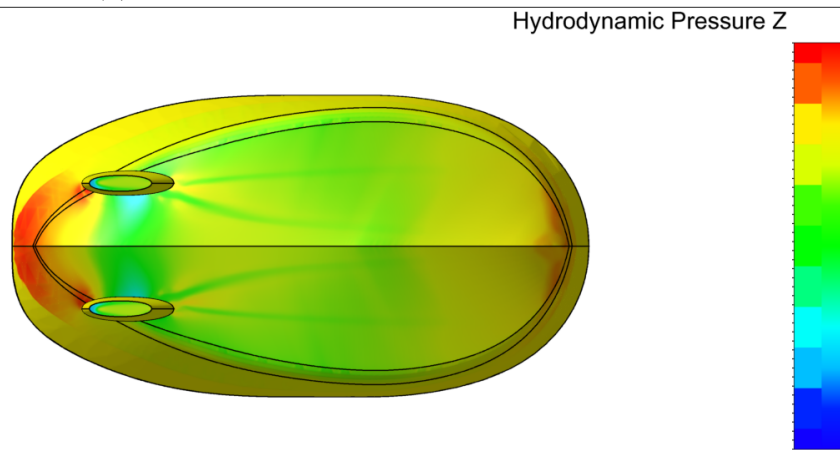
The reason why the dynamic component of the righting moment is negative and increased when a drift angle was introduced, can be seen in [Figure 39](#). In those figures, the magnitude of the velocity was plotted and the corresponding hydrodynamic pressure on the hull. In [Figure 39c](#) and [Figure 39d](#) the ship moves to the top left corner. From [Figure 39c](#) it is clear that at the starboard side on the bottom of the ship, there is a lower flow velocity than at the port side on the bottom. In the picture, the separation is indicated by the low velocity, blue flow on the hull and fins. Furthermore, cross flow is visible by the blue lines coming of the port side fin. [Figure 39a](#) and [Figure 39c](#) show this cross flow by showing how the red streamlines cross over to the other side of the hull at 45 degrees drift. As is shown in the figures, the separation causes areas with a reduced magnitude of relative velocity. These areas of reduced magnitude of relative velocity are mostly present on the starboard side of the ship due to the flow crossing over at 45 degrees. Lower flow velocity increases the pressure and vice versa as can be seen from [Figure 39b](#) and [Figure 39d](#). This causes an asymmetry in the hydrodynamic pressure between starboard and port side at 45

degrees drift. This isn't the case at 0 drift. This asymmetry is further visualised by [Figure 40](#). In this figure, the magnitude of relative velocity is plotted at multiple points in the ship. Only the areas where the flow is slower than 8 knots are shown. From this figure, it is clearly visible that at 45 degrees the flow at the starboard side of the ship is more often slower than 8 knots than at the port side.

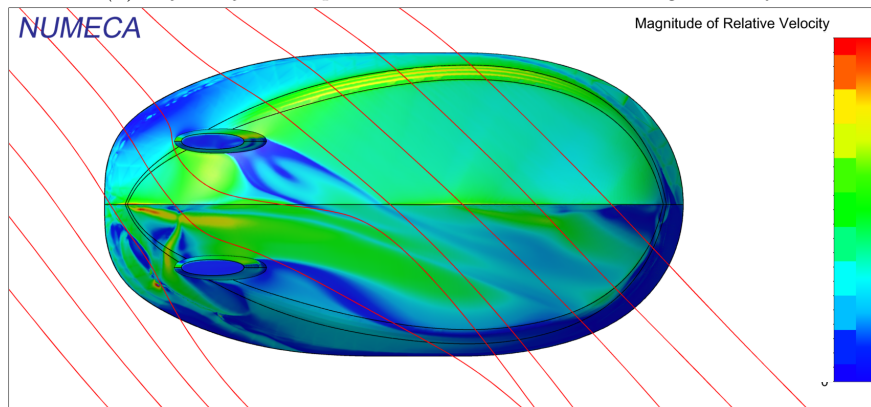
Because of the symmetry of the hydrodynamic pressure at 0 degrees drift, the arm for the dynamic component of the righting moment is very small, causing a small moment. When the flow is asymmetrical at 45 degrees drift, the arm of the moment increases significantly. This means, in the case of [Figure 39c](#) the ship would be experiencing a negative dynamic component of the righting moment.



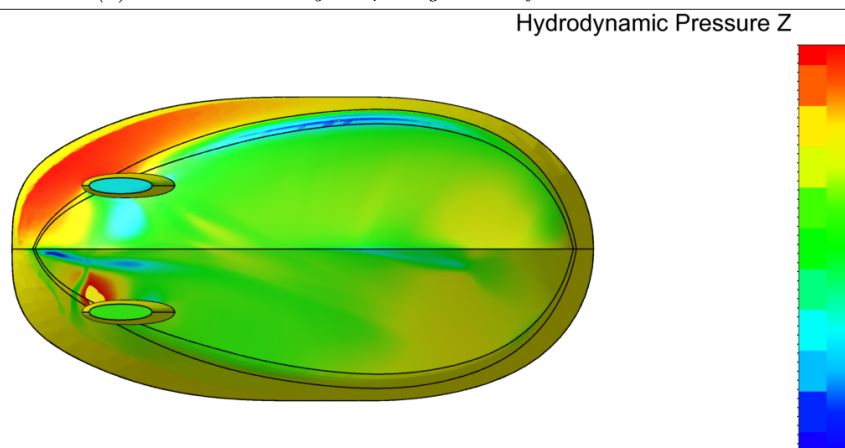
(a) Relative velocity at 0 degrees drift with streamlines



(b) Hydrodynamic pressure in z direction at 0 degrees drift



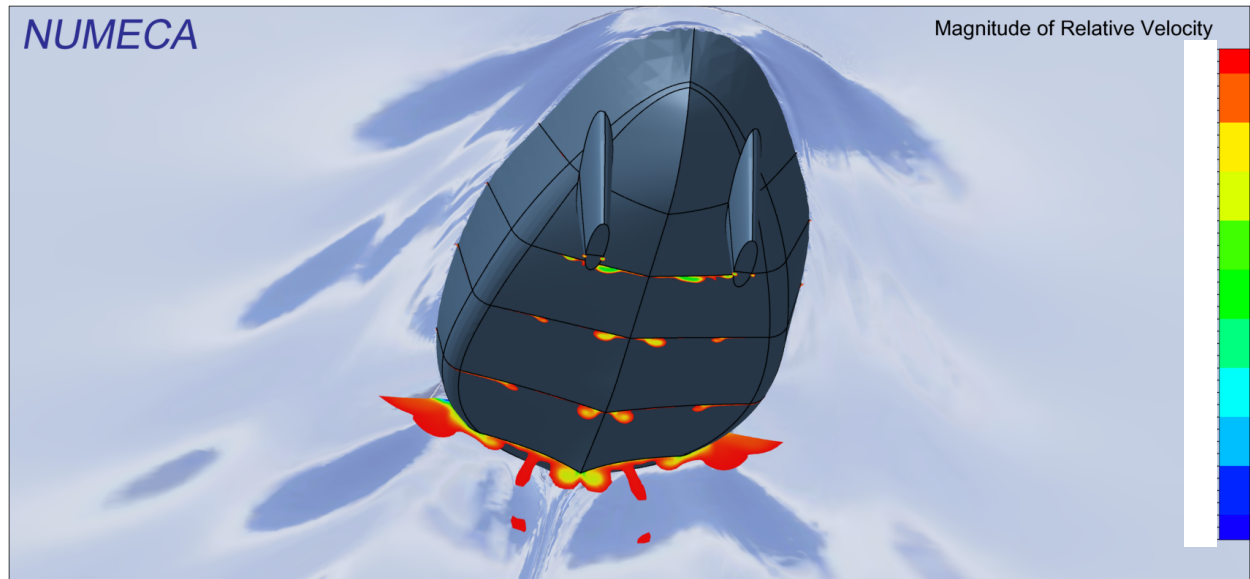
(c) Relative velocity at 45 degrees drift with streamlines



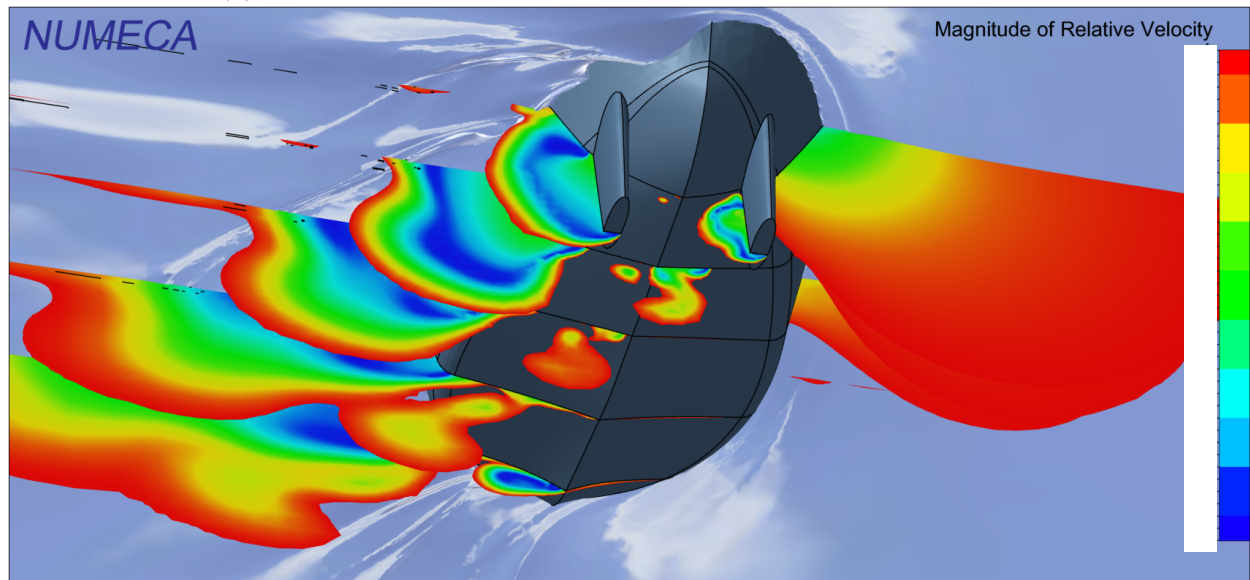
(d) Hydrodynamic pressure in z direction at 45 degrees drift

Figure 39: Magnitude of the relative velocity and hydrodynamic pressure in z direction over the bottom of the ship at 0 and 45 degrees drift, 0 degrees heel and 8 knots forward speed





(a) Relative velocity at 0 degrees drift at several cut planes on the hull



(b) Relative velocity at 45 degrees drift at several cut planes on the hull

Figure 40: Magnitude of the relative velocity at 0 and 45 degrees drift, 0 degrees heel and 8 knots forward speed. The flow is plotted in such a way that only velocities in the flow lower than 8 knots are shown

An interesting finding was that a drift angle does not change the slope of the righting moment created by the heeling significantly. This is shown in [Figure 41](#) where a comparison is made between 0, 15, 30 and 45 degrees drift. The fact that the slope does not change significantly for these angles, indicates that increasing the heeling angle increases the righting moment with approximately the same amount for all drift angles. This leads to the conclusion that it can be assumed that the heeling angle and drift angle have independent effects on the righting moment for the heeling- and drift angles and the forward velocity simulated in this research. As with the other observations in this chapter, this observation should be viewed with caution due to the high uncertainties that are also plotted in [Figure 41](#)



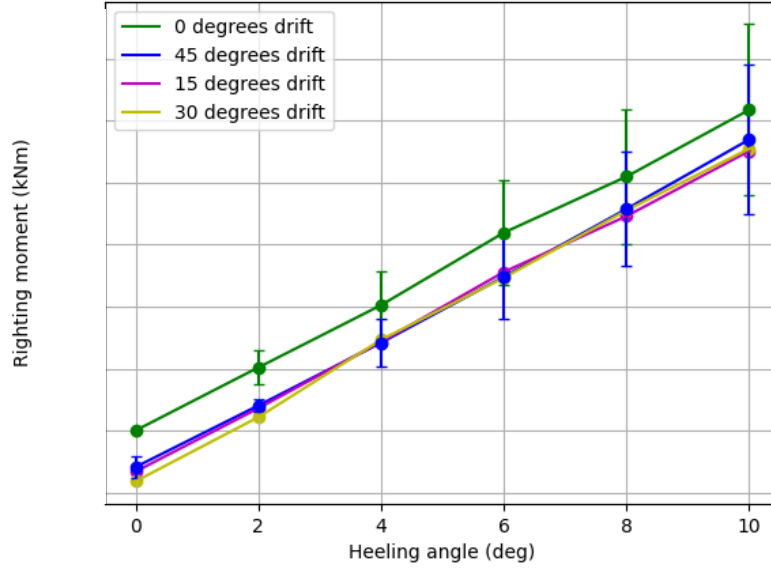


Figure 41: Slope comparison of the righting moment between 0 to 45 degrees drift at 0 degrees heel 8 knots forward speed

Looking at [Figure 34b](#) it is clear that the heeling moment increases with increasing drift angle. A clear explanation for this is the increase of  $F_{y_{dyn}}$  in negative direction. This was due to an increase in hydrodynamic pressure as illustrated in [Figure 42](#) and [Figure 43](#). These figures show the hydrodynamic pressure in positive y direction, meaning that a negative value shows the pressure is working in the negative y direction. The biggest difference between the two is visible in the port side views, [Figure 42a](#) and [Figure 43a](#). A large negative pressure area is visible at 45 degrees. This is the source of the increase of  $F_{y_{dyn}}$  in negative y direction. The negative pressure area is due to hydrodynamic- lift and drag being generated by the shape of the hull. From [Figure 20](#), in which the lift coefficient was plotted, it is expected that the generated lift will be at its maximum at around 30 degrees after which it will slightly decrease. However, the drag continues to decrease between 0 and 45 degrees.

Secondly, there is a slight increase in the heeling moment with an increased heeling angle. This is not due to an increase in the force, but because the arm of the heeling moment becomes slightly larger. This is illustrated in [Table 13](#). It is also plotted in [Figure 44](#), however, the increase is so small and the difference in the picture is not clear.

All the observations made in this chapter, have limited credibility due to high numerical uncertainty. This means it is plausible that the observations made are actually proven wrong when accurate results are obtained with a lower numerical uncertainty. Furthermore, the high numerical uncertainties also mean the validation process that took place in [subsection 7.3](#) is unreliable. Lastly, it should be noted that in [subsection 5.7](#) it was discussed that the results did not always fully converge. This further decreases the reliability of these results. This should be taken into account when reviewing these results and the conclusions.

Table 13: Arm of the heeling moment at 15 degrees drift and 8 knots forward speed. *Data is left out of the public version intentionally due to confidentiality*

| Heeling angle (deg)                | 0 | 2 | 4 | 6 | 8 | 10 |
|------------------------------------|---|---|---|---|---|----|
| $z_{adyn}$ (m) at 15 degrees drift |   |   |   |   |   |    |

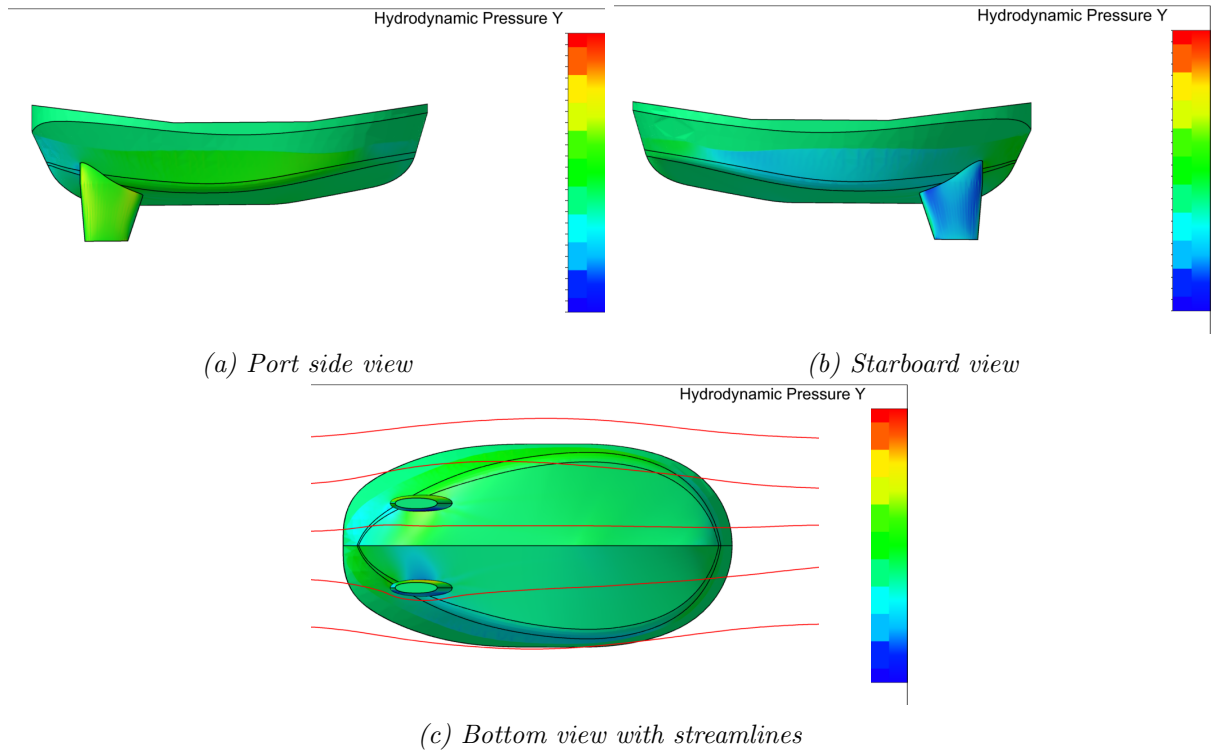


Figure 42: Hydrodynamic pressure in positive  $y$  direction at 0 degrees drift, 0 degrees heel and 8 knots forward speed

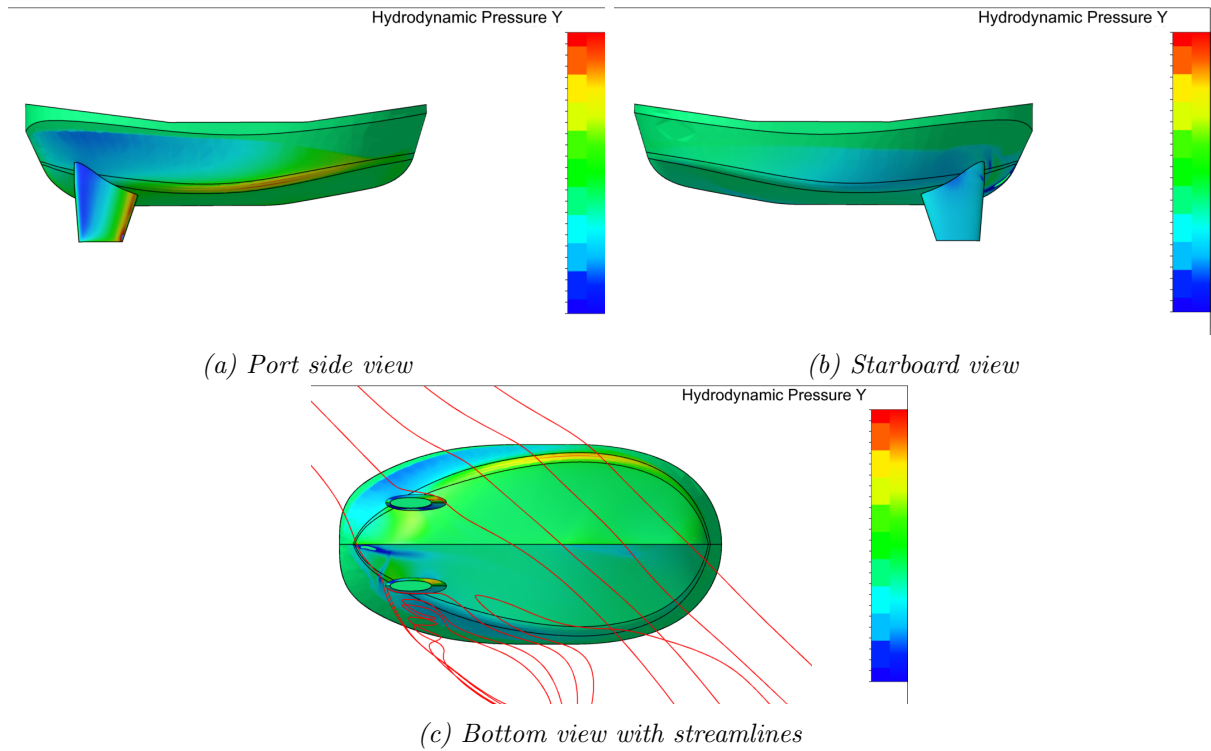
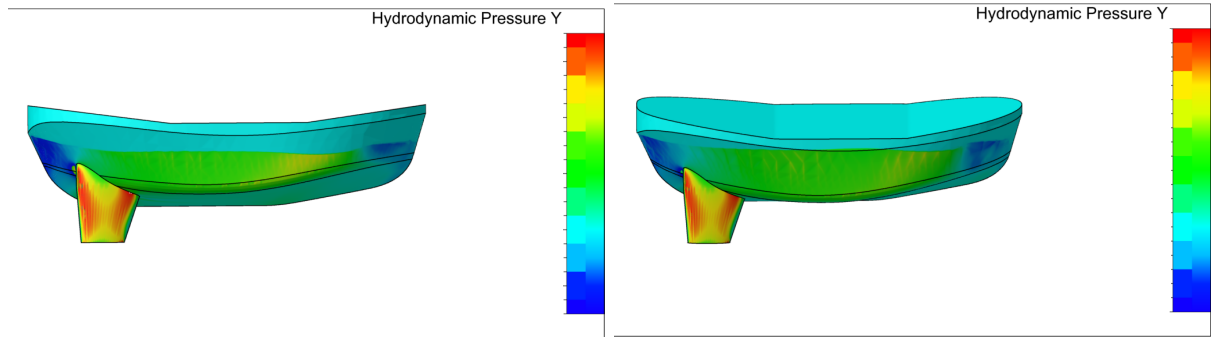


Figure 43: Hydrodynamic pressure in positive  $y$  direction at 45 degrees drift, 0 degrees heel and 8 knots forward speed



(a) Port side view at 0 degrees drift and 0 degrees heel (b) Starboard view at 0 degrees drift and 10 degrees heel

Figure 44: Hydrodynamic pressure in positive  $y$  direction 0 degrees drift, 0- and 10 degrees heel and 8 knots forward speed

## 9 Discussion

In this chapter drawbacks and limitations of the methods used in this research will be discussed. This is done to reflect on the method and put the results discussed in the previous chapter in perspective.

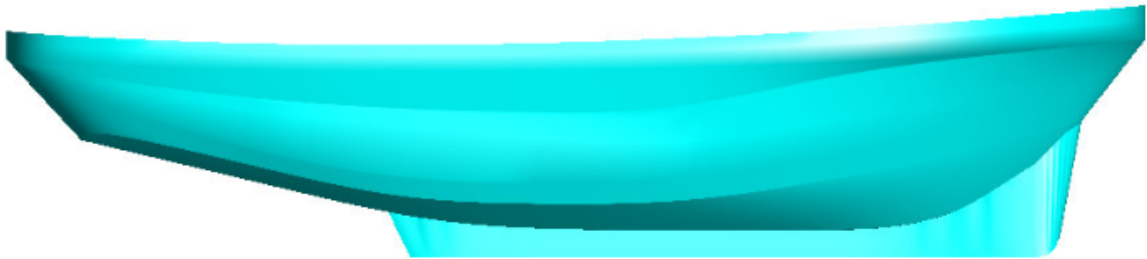
- Convergence proved to be an issue for higher drift angles as discussed in [subsection 5.7](#). This was due to high separation at these angles. Oscillation around a certain value was observed for 30- and 45 degrees drift. This leads to the results being more unreliable than for fully converged solutions. Since separation was the likely cause, an unsteady simulation was run at 30 degrees drift, 0 degrees heel and 8 knot forward speed to compare to the steady simulation. The unsteady simulation did not converge completely but oscillated irregularly around a certain value. This indicated that the flow was indeed unsteady. Better convergence might have been obtained by doing a temporal refinement exercise. However, this was not done since the unsteady simulation took over a week to run. The value computed by the unsteady simulation was quite far off the value computed by the steady simulations. This indicated that the steady simulations might not estimate the forces on the tug very accurately. Another unsteady simulation was run. This time with a DDES turbulence model. The results of this simulation were almost exactly the same as the other unsteady simulation. It was expected that the convergence would be improved by using the DDES turbulence model. This could still be achieved by using different mesh settings as suggested in the FINE/Marine user guide [\[27\]](#). However, both the turbulence model and the unsteady simulations required considerable computational time. Therefore, it was not deemed feasible to use them in this research.
- When the lift- and moment coefficient from mesh 2 were compared to model tests in the validation exercise, the comparison error was significantly larger at 30- and 45 degrees drift than the lower drift angles. This is explained by the amount of separation that starts to occur around 30 degrees drift and the high turbulent kinetic energy. To decrease this comparison error, unsteady simulations and/or a different turbulence model could be considered. However, similar to the first discussion point and to what was observed in [subsection 5.7](#) both could increase the computational hours of the simulations significantly.
- Large numerical uncertainties were computed in the verification exercise for all meshes. Although omitting the coarsest mesh and using a linear polynomial regression to determine  $\phi_0$  resulted in lower uncertainties for the flow quantities, the uncertainties remain high. These high numerical uncertainties also interfere with the reliability of the validation of the meshes. With these uncertainties and the unreliability of the error estimation, the results of this thesis have significantly less credibility.
- The numerical uncertainty that was included in the results of [subsection 8.1](#), is calculated for 15 degrees drift, 0 degrees heel and 8 knots forward speed. However, this uncertainty was used to show the uncertainty on all combinations of drift angles and heeling angles in the results chapter. It would be interesting to calculate this uncertainty for other angles, especially for 30 degrees drift. This is because, the comparison error for  $C_L$  and  $C_K$  is significantly larger at 30 degrees drift than at other drift angles. It would therefore be insightful to know what the uncertainty for those simulations is and how that would affect the results.
- The uncertainties from the model test used for the validation of this research, was calculated by Vink [\[36\]](#). However, these uncertainties were calculated for the bare hull case without fins. How significant the addition of fins is to the uncertainty of the model tests is unknown. However, it could be that it does influence the value of one or more components that make up the total experimental uncertainty. On one hand the uncertainty could be expected to increase as the fins create flow separation. This increased turbulence can interfere with the signal of the measured forces. The signal could be even more unsteady resulting in a

higher statistical uncertainty, which is the uncertainty in the measured signal. This would result in an increased experimental uncertainty. However, on the other hand the uncertainty could decrease due to the flow separation line being stricter and more well defined. This would cause the separation and therefore the force signal to be less irregular, resulting in a lower statistical uncertainty. This was observed for higher drift angles where separation also increases [36]. It is therefore deemed more likely that the uncertainty will be lower for the model tests with fins.

- Furthermore, Vink [36] recommended to improve the uncertainty of the model tests or to obtain better validation material. In combination with more accurate estimates of the numerical error, this could help to improve the validation exercise. This way the accuracy of the CFD compared to real life could be assessed more accurately. However, improving the numerical uncertainties would be the priority. This is because the numerical uncertainties are larger and are therefore the biggest contributor to the validation uncertainty. This would mean improving the numerical uncertainty would have a bigger effect than the experimental uncertainty.
- Furthermore, as mentioned in subsection 5.9 the trim- and sinkage correction used during the validation of this research is based on a data set for large, slender ships. Damen has been using the same correction for their RSD 2513 tug in their performance prediction tool TugSim. However, one could be sceptical about using this approach when it was derived for a different type of vessel. Especially since the effect is quite large, ranging from 15% to 21% depending on the drift angle. It should be noted that this correction is only used during validation, and not for the results from subsection 8.1
- The most extreme combination of heeling- and drift angle used simulated were 10 degrees heel and 45 degrees drift. At this point, water starts to come on deck. For this simulation this did not become problematic but for more extreme simulations where more water will come on deck, this might start giving problems with convergence and the accuracy of the solution. If similar simulations would be used for further research, this should be taken into account.
- Design measures might be taken to lessen the effect of the cross flow which leads to the asymmetry in flow velocity on both sides of the ship. This is the cause of the dynamic component of the righting moment, which decreases the righting moment. A skeg or bilge keels might be installed in an effort to slow the flow at the port side of the ship. By slowing down the flow on the port side of the ship, the hydrodynamic pressure is expected to increase. This would decrease the dynamic component of the righting moment and thereby reduce its negative effect on the ship's righting moment. However, it is not certain that these appendices will have this desired effect. Bilge keels and skegs have other beneficial effects that will affect the performance and stability of the ship. Both have the effect of damping the roll motion of the ship [34] [18]. And skegs can be used to generate extra lift to increase the tug's steering- and breaking force [25]. However, this extra lift force will likely increase the heeling moment of the ship. This should be taken into account when designing the tug.



(a) Bilge keel [37]



(b) ASD tug with skeg [25]

Figure 45: Example of a bilge keel and skeg

## 10 Conclusion

This chapter will discuss the results presented in chapter [subsection 8.1](#) and draw the conclusions of this research. This is done by answering the research question:

*To what extend are the practical definitions of the righting moment and heeling moment of a tug dependent on forward speed, heeling angle and drift angle?*

In order to answer the question above, CFD simulations were conducted to calculate the forces and moments acting on the tug at a range of drift- and heeling angles. These were used to compute the righting- and heeling moment. These moments were defined as follows:

- **Righting moment:** The righting moment is a moment that is working in positive direction around the x axis. It is therefore working to return the ship to zero heel. It consists of a component created by the static z force,  $F_{zstat}$ , called the static component of the righting moment, and a component created by the dynamic z force,  $F_{zdyn}$ , called the dynamic component of the righting moment. A schematic is given in [Figure 46b](#).
- **Heeling moment:** The Heeling moment is a moment that is working in negative direction around the x axis. It is therefore working to increase the ship's heel. It consists of one component created by the dynamic y force,  $F_{ydyn}$ , called the dynamic component of the heeling moment. A schematic is given in [Figure 46c](#).
- **Restoring moment:** The restoring moment is a moment that is working in positive direction around the x axis. It is therefore working to return the ship to zero heel. It consists of one component created by the static z force,  $F_{zstat}$ . In classic ship stability it is calculated using the GZ curve from SARC PIAS, not CFD. The restoring moment is always calculated for 0 knots forward speed. A schematic is given in [Figure 46a](#).



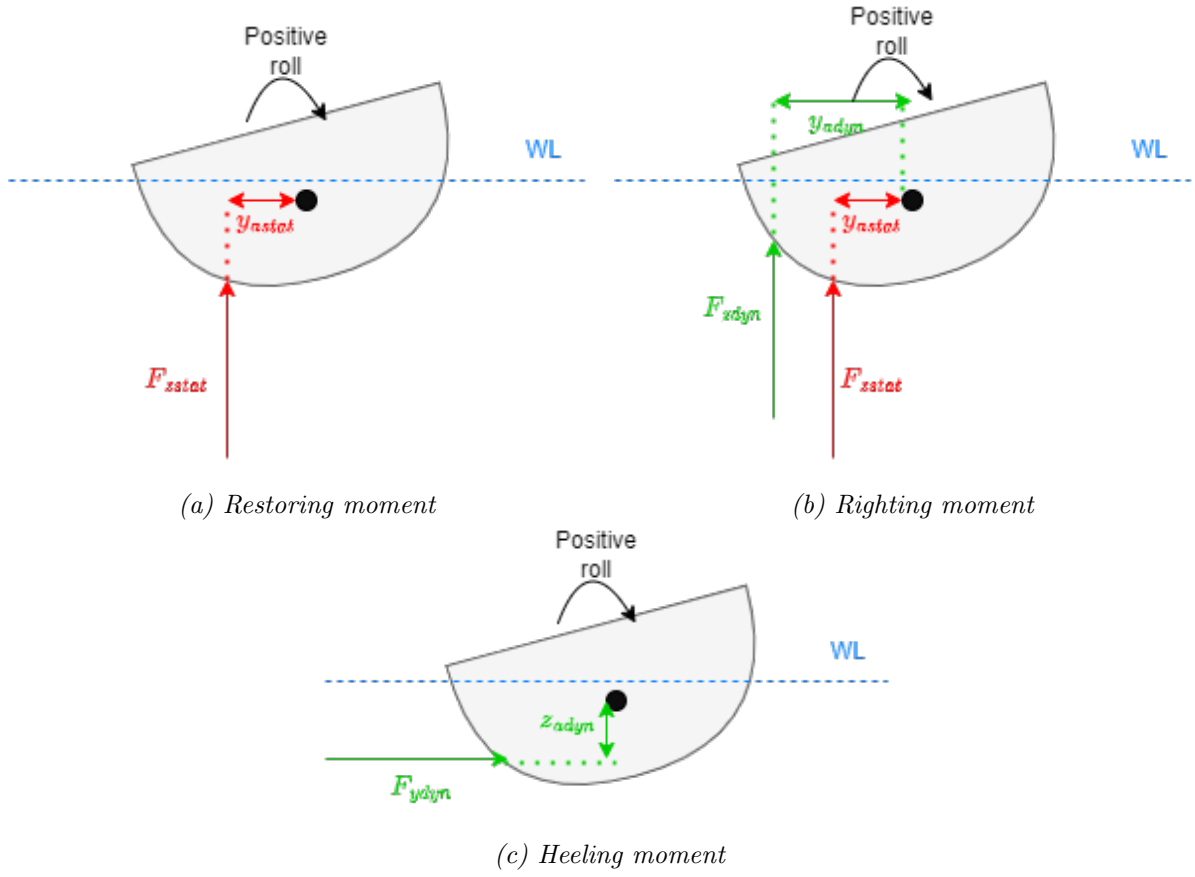


Figure 46: Schematic of the definition of the restoring-, righting- and heeling moment

Curves of these moments were plotted to visualise the results. From these curves a number of conclusions could be drawn. For the righting moment the following conclusions were drawn:

1. Without a drift angle, forward speed doesn't influence the righting moment significantly.
2. Introducing a drift angle will result in a decrease of the righting moment.
3. Even with a drift angle, increasing the heeling angle will result in an increasing righting moment. In fact, between 0 and 10 degrees heel and 0 and 45 degrees drift, the heeling angle and drift angle can be assumed to have independent effects on the righting moment.

The first conclusion could be drawn based on the righting moment at 0 degrees drift, compared to the restoring moment based on the GZ curve of a RSD 2513 tug. This comparison showed that there was no significant difference between these two. This means that, as long as there is no drift angle, the influence of forward speed on the righting moment is negligible. Though a dynamic force in negative z direction was created, it did not result in a significant moment compared to the moment created by the static force in z direction. This was due to the dynamic component of the righting moment being significantly smaller than the static component of the righting moment. This is explained by the fact that  $F_{zdyn}$  is significantly smaller than  $F_{zstat}$ . Furthermore,  $y_{astat}$  is larger than  $y_{adyn}$  at 0 degrees drift. Leading to the static component of the righting moment being so much larger than the dynamic component.

The second conclusion is drawn from inspecting the righting moment curve. From this curve it is clear that the moment drops significantly when a drift angle is introduced. This is because the arm of the dynamic component of the righting moment increases significantly between 0- and 15 degrees, and because the dynamic force increases in negative z direction. This creates a dynamic

component of the righting moment that is negative around the x axis, working in opposite direction to the static component of the righting moment.

The last conclusion from the righting moment curve is that an increasing heeling angle increases the righting moment, even with a drift angle applied. Without a drift angle, this would be a relatively straightforward conclusion as it adheres to classic ship stability. However, the drift angle, combined with forward speed and heeling angle, gave insight in whether this still holds up in the dynamic situation that was being researched. As mentioned the static component of the righting moment increases significantly when a heeling angle is introduced. The dynamic component of the righting moment increases as well but in negative direction. However, the static component of the righting moment is significantly bigger than the dynamic component, leading to an overall increase in righting moment when the heeling angle increases. An interesting finding is that the drift angle didn't significantly affect the rate at which the heeling angle increased the righting moment. Meaning that the slope of the righting moment between 0 and 10 degrees heel, is approximately the same for drift angles between 0 and 45 degrees. This would suggest that the heeling angle and drift angle can be assumed to have independent effects on the righting moment.

However, for all these conclusions the results from the verification and validation study from [subsection 6.1](#) and [subsection 8.1](#) should be considered. The numerical uncertainties are very high and the error estimate is unreliable. Therefore, if the range in which the exact results are expected to be, is very large. Furthermore, these high uncertainties mean that the validation is rather unreliable as well. And lastly, as described in [subsection 5.7](#) the simulations did not always show complete convergence. These things combined mean that the perceived conclusions might be very different if the exact results lie somewhere else in the uncertainty range. These conclusions should therefore be used carefully and with scepticism.

The following was concluded about the heeling moment:

1. The heeling moment increases when a drift angle is introduced.
2. Increasing the heeling angle will result in a slight increase in the heeling moment.

The first conclusion drawn from the heeling moment is explained by the increase in hydrodynamic pressure. This is due to the increased force due to hydrodynamic- lift and drag. This increases the pressure force on the ship acting in the y direction. The second conclusion about the heeling moment isn't due to the increase in hydrodynamic pressure but to a small increase in the arm of the dynamic moment.

Similarly to the conclusions of the righting moment, the conclusions of the heeling moment suffer from high numerical uncertainty and an unreliable error estimate. The range in which the exact results could lie is large. Therefore, the observed trends and conclusions might change drastically when more accurate results are obtained. Again, these conclusion should be viewed sceptically.

To answer the research question, these 5 conclusions can be drawn about the extend to which the righting- and heeling moment are influenced by the forward speed, heeling angle and drift angle. One important thing to note is that increasing the drift angle, increases the heeling moment and decreases the righting moment. Even though the righting moment is still significantly bigger than the heeling moment, this could lead to dangerous situations in escort since the towline forces haven't been taken into account in the heeling moment in this research. In regulations the heeling lever is computed from towline forces only and is expected to decrease with increased heeling angle [1](#). However, as shown in this research, the heeling lever due to hydrodynamic forces is expected to increase slightly with heeling angle. This suggests that it is important to take more than the static GZ curve into account when assessing the stability of escort tugs.

## 11 Recommendations

After discussing the drawbacks and limitations of the methods used in this research and drawing conclusions from the results, recommendations can be made regarding future research, and design and classification of tugboats. Firstly, the recommendations regarding the classification of tugboats will be discussed. Secondly, recommendations for the design of tugboats will be given. And lastly, recommendations for future research will be made.

Classification societies base their requirements for escort tugs on the curve of the heeling arm and the GZ curve. The heeling arm is for most, such as DNV and Bureau Veritas, based on the bollard pull of the tug and the distance between the towing point and centreline propeller. The regulations then state that the ratio between the areas underneath the heeling arm curve and the GZ curve should have a certain minimum value in order for the tug to have sufficient reserve stability [1]. However, the GZ curve is calculated at 0 knots forward speed and 0 degrees drift. In other words, there are no regulations concerning the dynamic forces that a tug encounters during escort. Similarly, the heeling arm is based on the bollard pull of the tug. It does not take into account the additional heeling arm created by dynamic forces. However, in this research it was concluded that, at 8 knots forward speed, the righting moment decreases due to hydrodynamic forces while the heeling moment caused by hydrodynamic forces increases. It is therefore recommended classification societies adapt their regulations concerning escort tugs to incorporate the heeling moment and righting moment created by the hydrodynamic forces. It should be ensured that with the decrease in righting moment and increase in heeling moment due to hydrodynamic forces, the regulations are still sufficient.

Tugboat designers can take some design measures to make the tug safer to use in escort operation. In this thesis, it was concluded that the righting moment decreased when forward speed and a drift angle were introduced. This was caused by the dynamic component of the righting moment. This component became increasingly negative when a drift angle was introduced at forward speed because of asymmetry in the hydrodynamic pressure on both sides of the ship. It was concluded that this was caused by high flow velocity on one side of the ship compared to the other side. In section 9 it was suggested that bilge keels or a skeg could be installed to slow down the flow on one side of the ship hull. This could potentially decrease the dynamic component of the righting moment and thereby reduce its negative effect on the ship's righting moment. Although it's not certain that a bilge keel and skeg could help decrease the dynamic component of the righting moment, it is known that they have the added beneficial effect of damping the roll motion of the ship [34] [18]. Furthermore, skegs are used to generate extra lift to increase the tug's steering- and breaking force [25]. However, this extra lift force will increase the heeling moment of the ship. This should be taken into account when designing the tug.

Since it was concluded that the combination of a drift angle and forward speed increased the heeling moment and decreased the righting moment, it is important that the additional heeling moment that is created by the towline forces on the tug is as small as possible. For this reason, the placement of the towing point is crucial in the design of the ship. This is because this placement determines how far away the application point of the towline force is to the centre of gravity and therefore how large the moment is that it creates. Careful placement of the towline connection point could therefore ensure the ship's safety even when sailing at large drift angles and considerable forward velocity.

The last recommendation on ship design is concerning the ship's hull shape. A designer can influence the stability of a ship greatly with the ratios between the ship's length and the beam of the ship. A low  $L/B$  will mean the beam of the ship is large compared to the length. This will provide the ship with a great restoring moment and will in that sense positively influence the stability of the ship. At the same time, it is known from lift theory that an increase in the  $L/B$  of the ship will result in an increase in the  $C_L$  of the ship [32]. On one hand, this can be beneficial

for a designer, as this will increase the steering force the tug exerts on the ship it is escorting. However, it will also increase the heeling moment created by the hydrodynamic forces. A careful balance is therefore advised.

There are certain recommendations to be made concerning future work and research into this topic. Firstly, the results from the uncertainty analysis in the verification of this research were disappointing. The numerical uncertainties were high and the error estimate was unreliable. This led to an unreliable validation study as well. Therefore, the results of this thesis should be used with caution as they are unreliable due to high uncertainties. It is therefore important that future work will be done to improve the numerical uncertainty and the validity of the results of this thesis. There are certain areas that could be researched to improve the numerical uncertainty. The way the grids are meshed could be reviewed to see whether improvement is possible there. Especially when this is combined with a DDES turbulence model. This model requires some different meshing parameters according to the FINE/Marine user guide. However, in this research this was not implemented. Furthermore, more in depth research can be done into using unsteady simulations instead of steady simulations. A temporal refinement study could be executed in order to verify the time step. These measures might not only improve the numerical uncertainty, but the convergence of the results as well. This will further add to the reliability of the results and conclusions.

Furthermore, the scope of this research is limited to heeling angles between 0- and 10 degrees, and drift angles between 0- and 45 degrees. The reason why it is limited to those angles is because water starts to come on deck at the combination of 10 degrees heel and 45 degrees drift. However, tugs do often experience water on deck during escort operation. This could have a significant effect on the stability of the tug. Therefore, future research should be done into the stability of tugs at higher drift- and heeling angles. Similarly, it is recommended to extend this research to include multiple different velocities. In this research only 8 knots was considered. However, escort tugs operate at different speeds. Therefore, after this research it is not yet clear how the stability will be impacted at different speeds. Furthermore, the effects of a different velocity, drift- and heeling angle on the numerical uncertainty would be interesting to research. In this research the numerical uncertainty is only calculated for 15 degrees drift, 0 degrees heel and 8 knots forward velocity. However, this is only one data point in the simulation matrix. Knowing the uncertainty at other data points as well would be insightful.

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