HYDRODYNAMIC MANOEUVRING ASPECTS OF PLANING CRAFT

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1 ABSTRACT

To get a better insight in hydrodynamic forces and moments acting on a planing hull during a manoeuvre in the horizontal plane oscillation runs have been performed. During these tests the model was fully constrained and forced into a manoeuvring motion (pure sway, pure yaw and yaw with drift). Forces and moments were measured in six degrees of freedom. Draught, trim angle, forward speed and sway- and yaw velocity have been varied systematically. Based on the measured forces and moments a mathematical model has been formulated by performing regression analysis with the varied coefficients as input variables. Subsequently, the mathematical model has been implemented in a simulation program, which has been developed earlier to describe the motional behaviour of a planing hull in six degrees of freedom. A number of simulation runs has been performed to observe the behaviour of a planing hull. Hydrodynamic terms as added mass appear to depend on forward speed.

2 NOMENCLATURE

b_ϕ	Damping term for roll	[-]
b_θ	Damping term for pitch	[-]
c_ϕ	Spring term for roll	[-]
c_θ	Spring term for pitch	[-]
I_{xx}	Moment of inertia in longitudinal direction	[kgm^2]
I_{yy}	Moment of inertia in transverse direction	[kgm^2]
$I_{\psi\psi}$	Moment of inertia in longitudinal direction	[kgm^2]
K	Moment in longitudinal direction, ship-fixed	[Nm]
M	Moment in transverse direction, ship-fixed	[Nm]
M_{yy}	Hydrodynamic mass in transverse direction	[kg]
N	Moment in vertical direction, ship-fixed	[Nm]
p	Roll velocity	[rad/s]
q	Pitch velocity	[rad/s]
r	Yaw velocity	[rad/s]
T	Draught of model at Centre of Reference	[m]
U	Towing speed	[m/s]
u	Forward speed, ship-fixed	[m/s]
v	Sway velocity	[m/s]
w	Heave velocity	[m/s]
X	Force in longitudinal direction, ship-fixed	[N]
Y	Force in transversal direction, ship-fixed	[N]
Z	Force in vertical direction, ship-fixed	[N]
β	Drift angle, around earth-fixed z-axis	[°]
θ	Trim angle, around ship-fixed y axis	[°]
ϕ	Roll angle, around ship-fixed x axis	[°]
κ_ϕ	Roll damping coefficient	[-]
κ_θ	Pitch damping coefficient	[-]
$\dot{v}$	Sway acceleration	[m/s^2]
$\dot{r}$	Yaw acceleration	[rad/s^2]

3 INTRODUCTION

The present paper is the result of a fruitful cooperation between Delft University of Technology and the Maritime Research Institute Netherlands (MARIN). The research aims of both the University and MARIN could be combined leading to a study on the manoeuvring behaviour of planing hulls in six degrees of freedom.

In this 2.5 years project, two series of tests were performed. The first series consisted of static drift tests with two different planing hull forms, during which the forward speed, draught and roll, trim and drift angles were varied. In total, 304 static drift tests have been conducted. Using the results of these tests, combined with hydrodynamic terms obtained from literature, a computer simulation program called VesSim has been developed. This program is capable of simulating the manoeuvring behaviour of a planing ship in six degrees of freedom. The results of this first study were presented earlier by Toxopeus et al [ref. 1].

The second test series consisted of dynamic oscillation tests with pure sway, pure yaw and yaw with drift, for several oscillation frequencies and drift angles. The forces were measured for six degrees of freedom. During the tests the trim angle, draught and forward speed were varied. In this way, coupling, for example between sway and pitch motion, could be determined.
The aim of this paper is to focus mainly on the series of dynamic oscillation tests.

Experimental data was the main objective. However, it turned out to be very helpful to also include video observations and still pictures for a better understanding of the flow about the model. The analytical work was mainly concerned with the regression analysis of the test data to obtain the manoeuvring coefficients.

The analysed data was implemented into the computer simulation program VesSim in order to increase the accuracy of strongly non-linear motions. With VesSim simulations were carried out in which the planing craft was shown to be capable to execute standard manoeuvres as zig-zag tests and turning circles together with coupling terms as roll angle due to turning.

4 COORDINATE SYSTEM

The coordinate systems used in this study are Cartesian coordinate systems. The coordinate system is ship-fixed and right-handed. The x-axis points forward, the y-axis to the starboard and the z-axis downwards. All forces and moments have been measured with respect to an arbitrary centre of reference. The formulations are formed around Newton’s law.

5 MODEL TESTS

5.1 Model particulars

The model used for this study was model 233 of Delft University of Technology. Model 233 was used by Keuning et al [ref. 2] during experiments with models with twisted bottoms as a part of the Delft Systematic Deadrise Series and by Toxopeus [ref. 3] during the static drift experiments. The main particulars and the body plan of the model are presented in Table 1 and Figure 1.

	Symbol	Model 233 (twisted bottom)	
Length between perpendiculars	L_{pp}	1.50	[m]
Max. beam at chine	B_{max}	0.367	[m]
Projected planing area	A_p	0.4589	[m^2]
Centre of planing area forward of ord. 0	C_{AP}	48.8%·LPP	[-]
Length/Beam ratio	L/B	4.09	[-]
Mass model incl. ½ transducer	m	11.5	[kg]
Moment of inertia incl. transducers	$I_{\psi\psi}$	2.8	[kg·m^2]
Longitudinal centre of reference forward of ordinate 0	L_{COR}	0.726	[m]
Vertical centre of reference relative to baseline	V_{COR}	0.080	[m]

Table 1: Main particulars of model 233

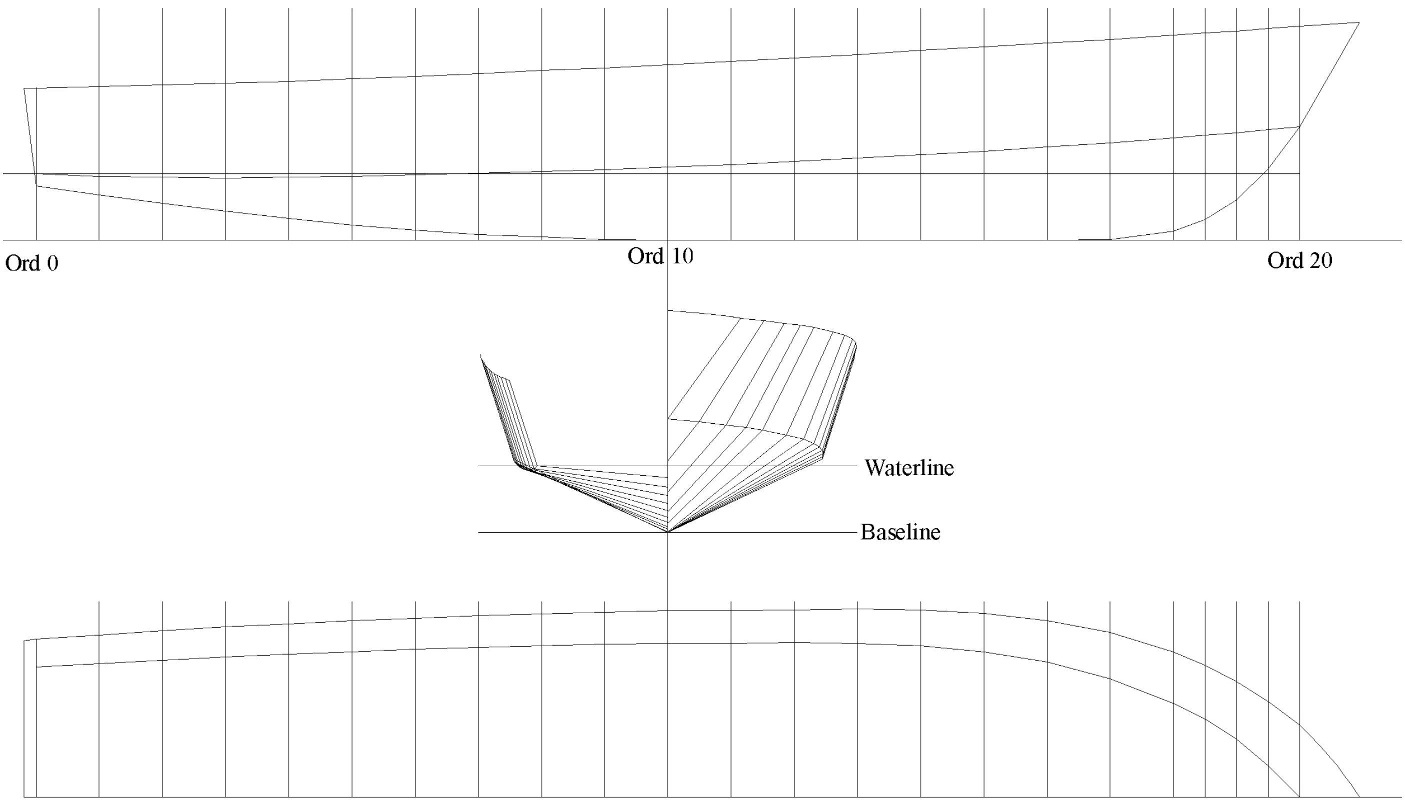

Figure 1: Body plan of model 233

5.2 Experimental set-up

Two six-component transducers were fixed in the model, evenly spaced around the centre of reference. Adding the components of the transducers, three forces and three moments about the centre of reference were derived.

All forces and moments were corrected by the mass properties of the model. These were determined by dry oscillations; the model lifted in air and performing a pure sway and pure yaw motion. The forces and moments corrected for the mass properties resulted in pure hydrodynamic forces and moments.

5.3 Model test program

The static captive model tests and dynamic PMM tests performed were done in the model basin of Delft University of Technology (size model basin 142 m x 4.22 m x 2.50 m). The different oscillatory motions performed to measure the hydrodynamic forces and moments were:

Pure sway
From the pure sway tests the forces and moments related to sway velocities and accelerations are obtained. Added mass, damping and coupling terms can be determined. The sway velocity amplitudes tested were: $v_1 = 0.125$ m/s and $v_2 = 0.250$ m/s.

Pure yaw
From the pure yaw tests the added moment of inertia and damping for yaw as well as coupling terms are determined. Two yaw velocity amplitudes were tested, $r_1 = 0.040$ rad/s and $r_2 = 0.080$ rad/s.

Yaw with drift
From yaw with drift tests the combined yaw-sway forces and moments can be measured. One drift angle was used combined with one yaw velocity amplitude: $\beta_1 = 5°$ and $r_1 = 0.080$ rad/s.

These three PMM motions were tested with the following variables:

Forward speed U
The model was tested at the speeds of: $U_1 = 2.0$ m/s; $U_2 = 3.0$ m/s and $U_4 = 4.0$ m/s. This corresponds to Froude numbers based on displacement between $1.2 \le Fn_{\nabla^{1/3}} \le 2.7$.

Trim angle θ
The model was tested for two different trim angles; $\theta_1 = 3°$ and $\theta_2 = 5°$. These trim angles are with respect to the baseline of the model.

Draught T
For the influence of the draught T two positions were tested, $T_1 = 0.065$ mm and $T_2 = 0.085$ mm. The design draught of the model is assumed to be T = 0.080 mm.

The combination of $T_2 = 0.085$ mm and $U_3 = 4$ m/s has not been tested, because of expected problems with spray. In total the test program consisted of 340 dynamic PMM runs.

6 RESULTS

6.1 Mass matrix

Plante [ref. 4] has described the forces and moments acting on the hull in the centre of reference in detail. The various expressions for the forces and moments as a function of draught, trim angle, forward speed, sway and yaw velocity and sway and yaw acceleration have been determined by regression analysis.

The full hydrodynamic mass matrix would consist of all hydrodynamic influences of the motions on each other. The symbol used for the added mass of a strip at a position x in direction i for an acceleration in direction k is:

$$M_{ik}(x) \text{ with } i,k = 1...6 \text{ or } x, y, z, \phi, \theta, \psi$$

In the description of the properties of the added mass coefficients Papanikolaou [ref. 5] and Newman [ref. 6] assumed that if a body is symmetrical about one or more axes, the cross coupling added mass coefficients can be taken equal. This means that $m_{ik} = m_{ki}$. (Or, more specific: $Y_{\dot{r}} = N_{\dot{v}}$) A number of terms are considered zero, so for the mass matrix of a conventional ship one assumes:

$$M_{ik}(x) = M_{ki}(x)$$
$$M_{ik}(x) = 0 \text{ for } i + k = \text{odd}$$

In the previous study on manoeuvring of planing craft, Toxopeus [ref. 7] used these assumptions and neglected several terms in the mass matrix because of lack of available data in literature. The total mass matrix was presented like:

$$M = \begin{bmatrix} X_{\dot{u}} & 0 & 0 & 0 & 0 & 0 \\ 0 & Y_{\dot{v}} & 0 & Y_{\dot{p}} & 0 & Y_{\dot{r}} \\ 0 & 0 & Z_{\dot{w}} & 0 & Z_{\dot{q}} & 0 \\ 0 & K_{\dot{v}} & 0 & K_{\dot{p}} & 0 & K_{\dot{r}} \\ 0 & 0 & M_{\dot{w}} & 0 & M_{\dot{q}} & 0 \\ 0 & N_{\dot{v}} & 0 & N_{\dot{p}} & 0 & N_{\dot{r}} \end{bmatrix}$$

From the data and observations during the series of dynamic oscillation tests conducted in the course of this study by Plante [ref. 7] it appeared that the mass matrix could not be taken symmetrical anymore. The Force Y derivative due to a yaw acceleration ($M_{26} = Y_{\dot{r}}$) was found not to be equal to the moment N derivative due to a sway acceleration ($M_{62} = N_{\dot{v}}$):

$$Y_{\dot{r}} \neq N_{\dot{v}}$$

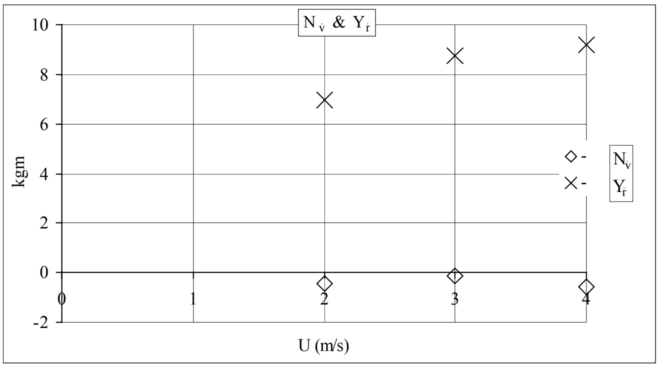

Figure 2: Values of $N_{\dot{v}}$ and $Y_{\dot{r}}$ are not the same

An explanation for this can be sought in the difference in distribution of the acceleration forces and moments on the ship model for pure sway and for pure yaw. For pure sway a pure side accelaration is enforced and the force due to sway acceleration will act around the midship section of the ship model. For pure yaw however, a pure rotational accelaration is enforced and the force will act asymmetrically at the fore and aft sections of the ship model. Therefore, the hydrodynamic coefficients are expected to be different.

One can conclude that the assumptions used by Papanikolaou [ref. 5] and Newman [ref. 6] are not valid anymore for planing hulls and therefore:

$$M_{ik}(x) \neq M_{ki}(x)$$
$$M_{ik}(x) \neq 0 \text{ for } i + k = \text{odd}$$

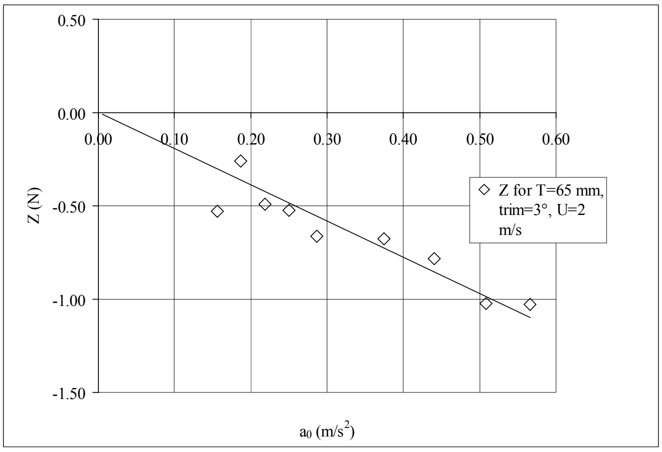

Figure 3: Force Z against sway acceleration

The asymmetry in the mass matrix was also found during the analysis of the test results in for example the vertical force: a force Z due to a sway acceleration (M_{32}) has been measured (see Figure 3). However, the force Y due to the heave acceleration (M_{23}) must be zero if the model is placed with no initial roll angle. This means that the term $M_{23} = Y_{\dot{w}} = 0$ and the term $M_{32} = Z_{\dot{v}} \neq 0$ are not identical anymore. So $M_{23} = Y_{\dot{w}} \neq M_{32} = Z_{\dot{v}}$ and therefore the mass matrix can not be considered symmetrical anymore. The force Z due to sway acceleration can be present due to planing effects.

Because of lack of additional information at this time, the moment K due to an acceleration in direction y and the force Y due to an acceleration in direction ϕ are still assumed symmetrical. This is because an acceleration in sway direction y will cause a moment K and an roll acceleration in direction ϕ will probably cause the same magnitude of force Y. The force and moment distribution on the hull in phase with acceleration will probalbly be the same for both situations and therefore these terms will still be assumed identical.

The mass matrix used for the mathematical model is now:

$$M = \begin{bmatrix} X_{\dot{u}} & X_{\dot{v}} & 0 & 0 & 0 & X_{\dot{r}} \\ 0 & Y_{\dot{v}} & 0 & Y_{\dot{p}} & 0 & Y_{\dot{r}} \\ 0 & Z_{\dot{v}} & Z_{\dot{w}} & 0 & Z_{\dot{q}} & Z_{\dot{r}} \\ 0 & K_{\dot{v}} & 0 & K_{\dot{p}} & 0 & K_{\dot{r}} \\ 0 & M_{\dot{v}} & M_{\dot{w}} & 0 & M_{\dot{q}} & M_{\dot{r}} \\ 0 & N_{\dot{v}} & 0 & N_{\dot{p}} & 0 & N_{\dot{r}} \end{bmatrix}$$

Terms as $Z_{\dot{u}} = M_{31}$ are considered small when compared to other terms and therefore neglected.

During the test runs sway and yaw terms in phase with acceleration for all forces and moments have been measured. This means that the second and sixth columns of the mass matrix are totally determined by regression analysis based on the present study.

6.2 Mathematical model – a qualitative review

For the estimation of the manoeuvrability in still water it can be assumed that all added mass coefficients are frequency independent. Before the model tests started in the basin, it was questioned whether the measured terms would be dependent on the frequencies of the oscillator. This appeared to be not the case for the sway and yaw velocities and accelerations used. A dependency of the added mass terms on the forward speed was found to be present. This dependency was added to the mathematical model and was implemented in the simulation program VesSim.

In the mathematical model, the relation between M_{yy} and forward speed has been taken linear. Since not enough forward speeds have been measured, it is not possible to assume that the hydrodynamic coefficients depend on higher order terms of forward speed U. However, it is visible in Figure 7 that the predictions of added mass using strip theory (Papanikolaou [ref. 5]) fit rather well to the model for zero forward speed when the hydrodynamic term is considered linearly dependent on the forward speed.

Another question risen for this case is whether the dynamic PMM terms in phase with sway velocity were comparable to the static terms of the first series of static drift measurements. If this is the case, this would mean that for small angles of attack (up to a drift angle of about 10°), the static theory will give an accurate representation of the forces and moments. From graphical comparison (see Figure 6) it appeared that the dynamic or quasi-static measurements of Plante [ref. 8] agreed well with the static measurements of Toxopeus [ref. 3]. In this figure, the force Y is divided by the towing speed U and put against the sway velocity v_0.

To give a clear description of the influence on the forces and moments, more forward speeds have to be measured. For the forces and moments measured (except moment K and moment N) a variation appeared due to the variation of the forward speeds. This variation in forces and moments can be a consequence of the varying wave system (due to spray) for different forward speeds. If the wave system differs, the pressure distribution differs and therefore the forces and moments on the model differ. For the motions measured, the assumption of a wave system equal to zero does not exist anymore. The presence of waves was clearly visible during the runs.

The moment K due to sway velocity and acceleration and yaw velocity and acceleration gives for all motions steady results. In pure sway, the moments were rather large, when compared to pure yaw. This can be explained regarding the waves induced by the sway and yaw motions as the reason for the existence of the moment K. The (non-linear) wave development was larger for pure sway than for pure yaw, so the moment K due to the sway motion measured is larger than the moment K due to the yaw motion measured.

Force Z and moment M due to sway and yaw velocity and acceleration are small, but present. How the terms due to sway and yaw velocity originate is made visible in Figure 8. Because of the total wave system consisting of waves due to forward motion and waves due to sideway motion the forces in the z-plane arise. These forces due to sway and yaw velocity are reaction forces of the wave system described. The force X is less affected by the interference of the two wave systems. Force X due to yaw velocity and yaw acceleration is almost equal to zero.

7 SIMULATION PROGRAM

The new mathematical model for the forces and moments derived from the dynamic PMM model tests is implemented in the computer program VesSim. The assumptions and values for the forces and moments have been tested in a number of test calculations. The test calculations can be divided in five different types:

1. Sensitivity analysis.
2. Change in initial position or velocity to determine the ability of the modelled ship to return to the equilibrium position.
3. Change in hydrodynamic coefficients, used for stability criteria.
4. Change in manoeuvring mode, to determine the manoeuvrability of the planing model and the behaviour of the model during these motions.
5. Correlation to full-scale data, to compare the results with full-scale tests done with the planing vessel "Voyager".

Ad 1. Sensitivity analysis

The program seems to be stable for the different input changes, i.e. a small change in input results in a small change in output.

Ad 2. Change in initial position or velocity

For the changes in initial position or velocity, the model has to return to its initial equilibrium state.

It appeared that model 233 has become straight-line instable after the adaptation of the mathematical model. After a slight disturbance in the initial sway or yaw velocity the model will keep a certain rate of turn and will not go back to her initial position.

The straight-line instability might be the result of the form of the hull and it is possible that this effect is quite different if the added mass and damping of the appendages such as rudders and propellers are taken into account. Now only the lift and drag of the appendages are calculated by VesSim, but not the damping and added mass of the appendages. However, there is no full-scale information available at this moment and therefore, it can not be concluded whether these results comply with reality.

For the other changes in initial position, the model returned to its initial equilibrium state.

Ad 3. Change in hydrodynamic coefficients

During both the test series, the roll and pitch damping has not been studied. However, the roll and pitch damping can have a strong influence on the manoeuvrability and controllability of a planing ship. The influence of the damping factors κ_ϕ and κ_θ was determined in the third set of calculations. These damping coefficients are defined as:

$$\text{Roll damping factor: } \kappa_\phi = \frac{b_\phi}{2\sqrt{(I_{xx} + M_{\phi\phi}) \cdot c_\phi}}$$

$$\text{Pitch damping factor: } \kappa_\theta = \frac{b_\theta}{2\sqrt{(I_{yy} + M_{\theta\theta}) \cdot c_\theta}}$$

The damping moments are defined as:

$$\text{Roll damping moment: } M_{damp} = K_p \cdot p = -b_\phi \cdot p$$

$$\text{Pitch damping moment: } M_{damp} = M_q \cdot q = -b_\theta \cdot q$$

In the present study, the damping factors were varied in the simulation program in order to ascertain the influence of the roll and pitch damping coefficients. From experimental observations of free running ships sailing a straight course at high speed at an initial non-zero roll or pitch angle, it was found that the decay of the roll would occur during a limited number of oscillations until a stable situation was reached.

The value of the roll and pitch damping coefficients should therefore be chosen such that also during a simulation the number of oscillations is found to be small (approximately one or two). The values of κ_ϕ and κ_θ have been adapted in such a way that this assumption is satisfied. This resulted in a roll damping value $\kappa_\phi = 0.40$ and a the pitch damping $\kappa_\theta = 0.60$. The limits of damping factors can be determined more accurately in future studies.

Ad 4 Change in manoeuvring mode

The diameters of the turning circles [ref. 9] are large when compared to values of Toxopeus [ref. 7]. This can be a result of the smaller values for N_{ur}. Toxopeus used values of N_{ur} as proposed by Hooft [ref. 1] for conventional ships and these appear not to be applicable to planing ships. The values found by Plante [ref. 8] were found to be about one third of the values used in previous assumptions.

Results of the spiral test simulation conducted using the new VesSim are presented in Figure 4.

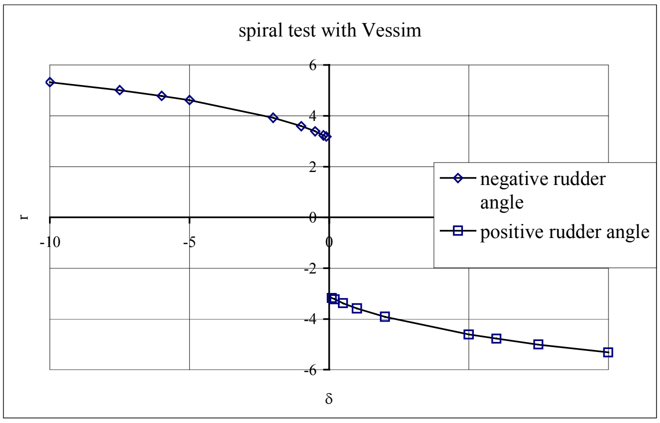

Figure 4: Spiral test

From the spiral test it can be concluded that the model is straight-line instable. The instability loop for rudder angles resulting in rates of turn between 3.5°/sec and –3.5°/sec was not determined.

Ad 5. Correlation to full scale data

To get an impression of the results and the order of magnitude of the accuracy of the simulation results, a comparison has been made with some full-scale measurements. Results of the full-scale tests of the Voyager, a pilot tender with a length between perpendiculars of 15.10 m have been used [refs. 11 and 12].

To be able to compare model 233 with the Voyager, the model had to be scaled up to the correct dimensions. A scale factor of 10 has been taken to compare the two ships. The body plan of the model and the Voyager hull are not exactly similar, the aft body of the Voyager is more twisted and the Voyager has larger spray rails than model 233. The Voyager is also equipped with waterjet propulsion, and model 233 with conventional rudders. Yet for a qualitative comparison the two hulls are similar enough.

In order to compare with the Voyager tests, the turning tests have been simulated with a forward velocity of 20 knots, with a rudder angle of 19°. In Figure 5, results from the simulation calculations are compared to the full-scale results with the Voyager.

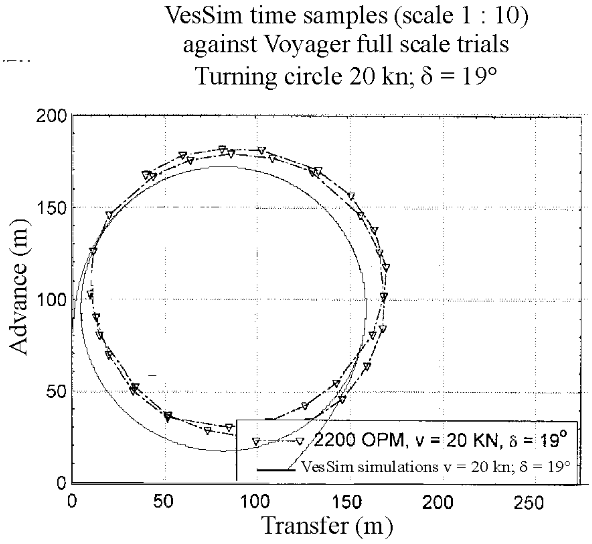

Figure 5: Turning test for model 233, forward speed 20 knots

The continuous line shows the simulation calculations. It seems that these two turning circles are of the same order of magnitude, both a diameter of about 150 m (about $10 \cdot L_{PP}$). If VesSim will be extended in future, it will be possible to develop a powerful tool for the prediction of the manoeuvring behaviour of planing hulls in the design stage.

In future studies, the simulation program VesSim can be validated by doing free sailing model tests with model 233. The diameters of the turning circles and the zig-zag tests can then be measured and compared to the values predicted by the simulation program VesSim. The next step is to do some full-scale experiments on planing vessels and do extensive comparisons between simulation runs and runs in full-scale. Possible scaling problems can be investigated for high forward speeds and the simulation program VesSim can be thoroughly validated.

8 CONCLUSIONS

The main goal of this project was to predict the forces and moments on a planing hull performing manoeuvres. During the investigation, other questions arose about certain aspects of the research. In this chapter, these questions will be formulated and answered as much as possible, together with the conclusions referring to the goal of the investigation. The unanswered problems will form the fundaments of the recommendations.

In model tests, the forces and moments were measured in six degrees of freedom, and a question was whether the forces and moments measured in the other directions (x- and z-direction) would provide any extra information. It appeared that coupling between several terms was clearly visible and the magnitudes of the forces and moments due to sway and yaw velocity and acceleration in for example z-direction were not negligible. The existence of the forces and moments provided new information on the flow profile around a planing hull during a manoeuvre. Forces and moments in x- and z-direction were caused by an asymmetric wetted surface by the sway and yaw velocity.

During the model tests, the question appeared whether the model tests from static drift angle tests and the dynamic PMM tests were interchangeable. Intensive comparison by means of graphs showed that the results from the dynamic model tests correspond very well with the results from the static model tests.

The computer simulation program VesSim was extended with the mathematical model based on the dynamic oscillation tests. To determine whether the change in mathematical model gave a more realistic behaviour of the vessel, a comparison with simulation runs of Toxopeus [ref. 7] was performed. The model appeared to have become straight line instable. Free sailing model tests have to be conducted to ascertain whether the ship model is in fact straight line instable. In case of discrepancies, the model's behaviour in the simulation program can probably be improved by the implementation of added mass and damping terms from rudder and propeller.
When the turning circles for the model simulations were scaled up and compared to full-scale tests with a planing hull the results seemed to be very promising.

9 RECOMMENDATIONS

Measurements on the model with rudder (oscillation tests with rudder angle) and propeller can provide additional information on planing hulls. For example, the added mass and damping coefficients of a rudder and a propeller behind a planing hull need to be developed. More information needs to be available for the influence of added mass and damping coefficients of rudder and propeller on the straight-line stability.

Additionally, more information is necessary to simulate the roll and pitch damping more accurately. The damping can be obtained by roll and pitch oscillation tests for various speeds, draughts and frequencies. A full understanding of the behaviour of a planing hull in the manoeuvring mode can only be found if more measurements and observations are conducted for more conditions.

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FIGURES

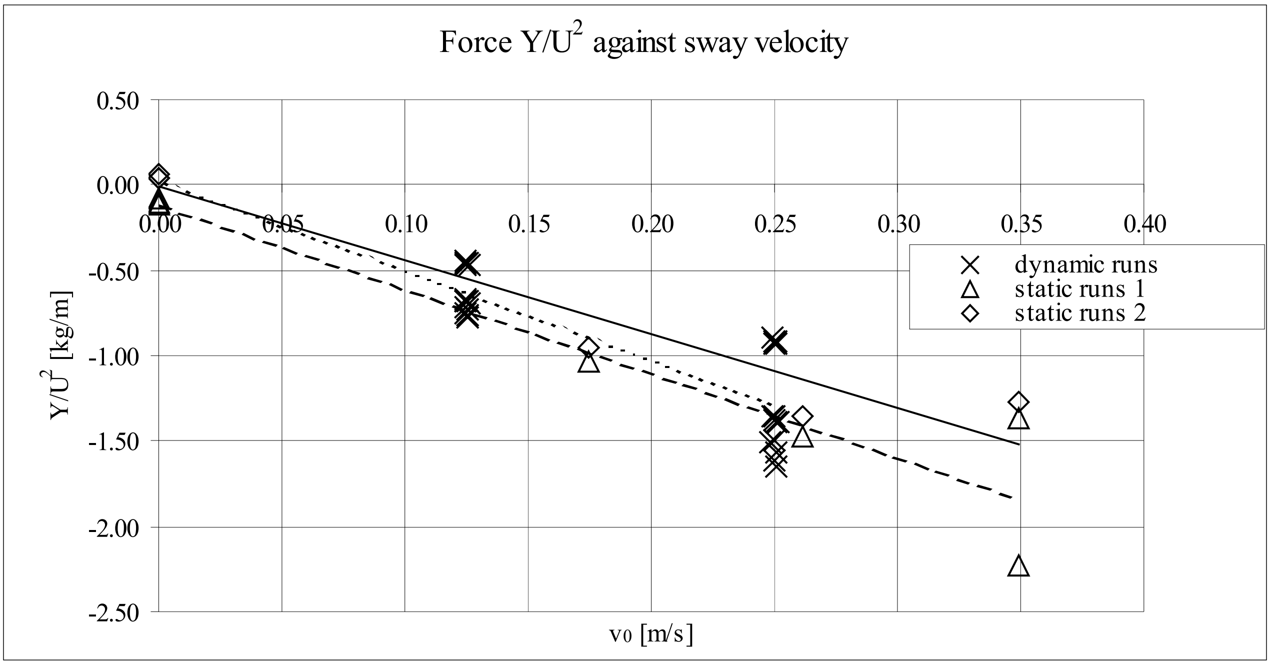

Figure 6 Comparison of static and dynamic test runs

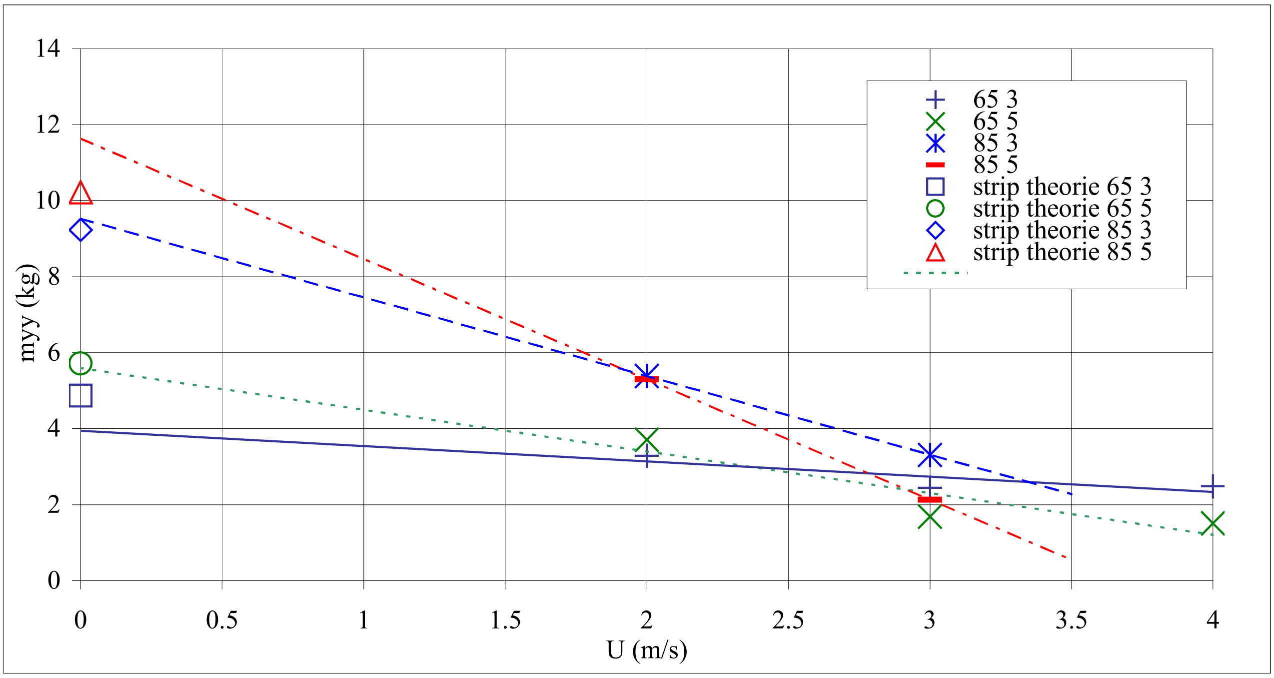

Figure 7 Values of $Y_{\dot{v}}$ against forward speed for draught and trim angle variations and predicted hydrodynamic forces

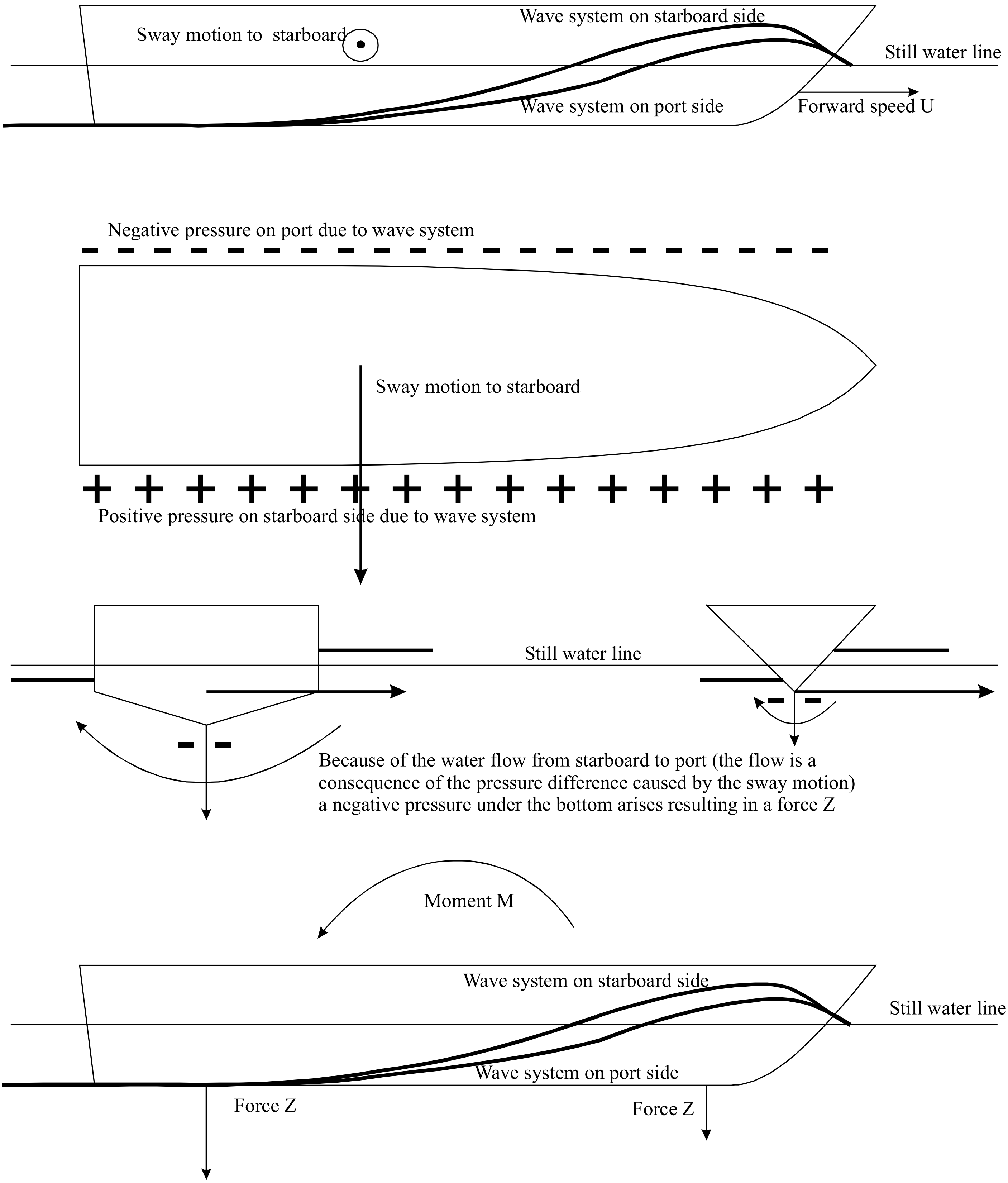

Figure 8 Indication for existence of force Z and moment M