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CURRICULUM: MATHEMATICAL ENGINEERING AND SIMULATION

**A cycloidal propeller simulation model and its application to
low and high speed ship manoeuvres**

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1 INTRODUCTION

Until the past century, the only way that naval architects had to predict the behaviour of the system (intended as the ship or a part of it, such as the propulsion plant, or the auxiliary systems) they were working on, was to build a prototype and test it. They had to afford the whole project hoping that it would meet the initial request. Nowadays, thanks to the development and the progress of computer science, we are able to shape not prototypes but simulators, based on mathematical laws, that can predict the behaviour not only of a singular system, but of the whole ship as an ensemble of each subsystem it is made of.

This allow to reduce the cost of the preliminary design since we don't need to physically construct anything but the final product, to train the crew and to make improvements to the project in a short time.

Making use of mathematical equations we can simulate how the hull will perform under certain weather conditions, if the machinery we want to install on board can guarantee the power we need and we can also predict the magnitude and the direction of the propellers thrust.

We can model different kinds of propellers: the traditional skew propellers, waterjets, azimuth thruster or cycloidal propellers.

Marine cycloidal propellers can represent a good alternative to traditional propellers since they can generate almost the same thrust in all directions. They were invented more than 80 years ago and they are mainly used on water tractors and ferries. Depending on their eccentricity value e , namely the ratio between the distance of the steering centre from the propeller axis and the radius of the circular orbit described by the blade axes (the rotor radius), they can be classified into true cycloidal ($e = 1$), epicycloidal ($e < 1$, e.g. Voith Schneider Propeller) and trochoidal propellers ($e > 1$). Cycloidal propellers are made of a set of vertical blade (generally from 3 up to 6 blades) protruding from the hull and performing two main rotations: one around the rotor axis and an oscillatory one around the axis passing through the blade pivoting centre. This kind of propellers are very suitable for Dynamic Positioning (DP) operations since they can generate almost the same thrust in every direction.

The aim of this work is to study the application of a cycloidal propulsion system to a hull whose behaviour, when equipped with a bow thruster and traditional propellers and rudders at the stern, is known.

Starting from the kinematical model for each single blade, a simulator of an epicycloidal propeller has been developed on a Matlab/Simulink platform.

This kind of simulation involves a reliable representation of the cycloidal propellers, the manufacturers of which unfortunately do not publicly share their performance maps for confidential reasons. Therefore, simplified simulation approaches, as possible for traditional propellers or waterjets (Altosole et al. 2012a), are quite difficult to be developed. The present numerical modelling is based on a mixture of theoretical and empirical considerations: in particular, the propeller thrust and torque evaluation is based on the kinematics of the blades, taking into account suitable correction factors in order to properly consider “dissipative” phenomena (such as interference between blades, the shielding induced by the half of the rotor which receives the incoming flow, and the slight reduction of back thrust). The calibration of the

so obtained simulator has been carried out by comparing simulation outputs with real data found in open source. The result is a simulation approach able to represent the performance prediction of an epicycloidal propeller, avoiding demanding computations (e.g. CFD methods) that would not allow an effective simulation of the whole DP system.

As a first application of the implemented simulation platform, different thrust allocation logics of a DP system for the surface vessel equipped with a bow thruster and two cycloidal propellers at stern are presented. The examined ship for this work is the same unit with conventional propulsion system, for which a dynamic positioning system has already been developed and installed on board. A comparison between the performances of the two distinct propulsion configurations has been then possible and is here proposed.

The first results of this static analysis (simply obtained by balancing forces and moments generated by environmental disturbances) have been very optimistic and they have been confirmed by the dynamic analysis.

A dynamic model for the simulation of the manoeuvrability, both at low and high speed, of the unit equipped with cycloidal propellers is presented together with some results.

1.1 *Cycloidal Propellers*

Cycloidal propellers consist of a set of vertical blades that rotate both around their own vertical axes and around the vertical axis of the main rotor. They represent a very good alternative to traditional propellers, since they can generate almost the same thrust in all directions, combining both propulsion and steering in a single unit.

When the blades rotate at uniform angular velocity around the centre of the rotor, the centre of the rotor itself advances at uniform rectilinear translational velocity through the water, so the centre of each blade shaft follows a cycloidal path.

On the basis of the pitch ratio value

$$p = e\pi \quad (1.1)$$

being e the eccentricity of the propeller, defined as

$$e = \frac{a}{R} \quad (1.2)$$

where R is the radius of blade orbit diameter and a is the distance between the steering centre and blade axis, different types of cycloidal propellers can be distinguished as shown in Figure 1.1 (Bose 2008):

- epicycloidal: $p < \pi$ and $e < 1$
- true cycloidal: $p = \pi$ and $e = 1$
- trochoidal propellers: $p > \pi$ and $e > 1$

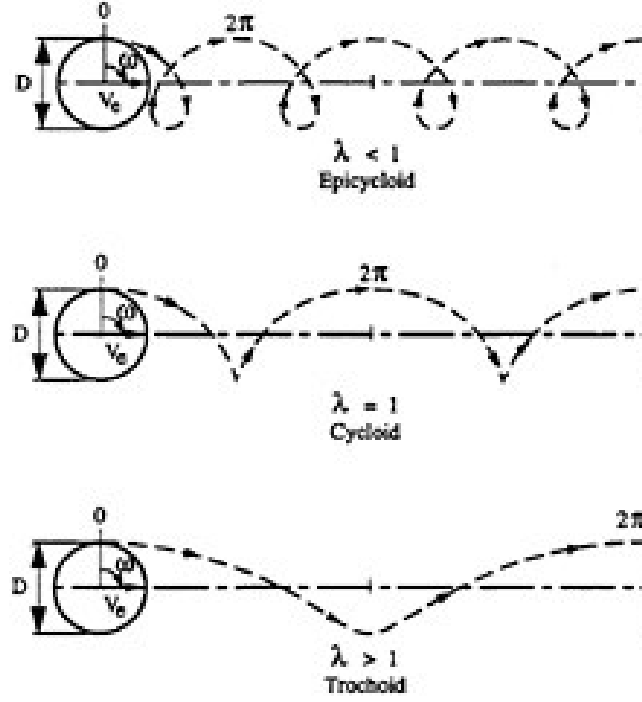


Figure 1.1: Types of cycloidal propellers (Bose 2008)

Taking into account the working point of the propeller, characterized by the velocity ratio

$$\lambda = \frac{V_0}{\pi n D} \quad (1.3)$$

where V_0 is the propeller speed of advance, n is the propeller rotation rate and D is the propeller diameter, three distinct types of blade trajectories can be singled out: for values of $\lambda < 1$, we have an epicycloidal blade path trajectory, for values of $\lambda = 1$ we have a cycloidal blade path trajectory and for values of $\lambda > 1$ we have a trochoidal blade path trajectory. The blade trajectory of a conventional propeller is necessarily epicycloidal, in order to generate a positive thrust. The same happens for cycloidal propellers with eccentricity $e < 1$, while for pitch ratios $e > 1$, both types of blade trajectories can be used in order to yield a positive thrust.

The present work focus on epicycloidal. The most famous company that has been designing this type of propellers for over 85 years is the Voith Turbo Schneider Propulsion GmbH and for this reason epicycloidal propellers are usually known as VSP (Voith Schneider Propeller). VSPs were developed by the Austrian engineer Ernst Schneider. Initially, they were installed on ferries and tugboats but, especially in recent years, VSPs have spread all over the world and today they are also installed on larger units, such as minesweepers and offshore units.

Despite this, about VSP very little material is available in literature and there are very few mathematical models suited for their numerical simulation.

1.2 Main features of epicyclicoidal propellers

The cycloidal propeller is a valid alternative to the most common propeller-rudder system as it merges both the propulsion and the government in a single unit and it allows very wide manoeuvrability margins since it is able to change the direction of the thrust in a very short time.



Figure 1.2: Cycloidal propeller (Intelligent Propulsion System for Safe Shipping, Voith Turbo Schneider Propulsion, brochure).

The cycloidal propeller (Figure 1.2) can generate thrust in all directions by means of vertical blades mounted in a rotor casing which is flush with the bottom of the vessel. While the blades rotate together around the vertical axis of the rotor (revolution motion), each single blade undergoes a rotation motion around its own vertical axis. During the revolution motion, the perpendiculars to the chords of each blade meet at a point which is the centre of thrust (Figure 1.3).

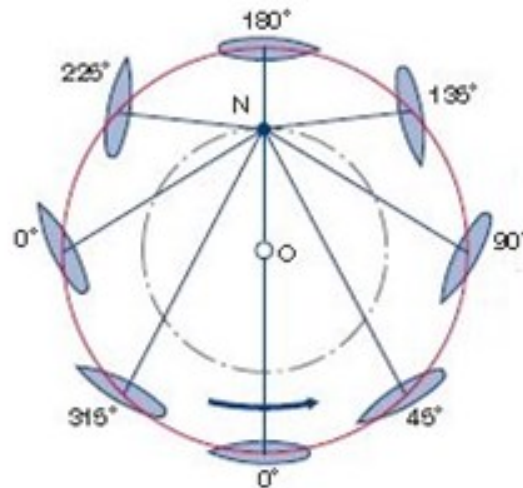


Figure 1.3: Blade positions during the revolution motion (Intelligent Propulsion System for Safe Shipping, Voith Turbo Schneider Propulsion, brochure).

In this way, an almost identical thrust can be generated in any desired direction. Both thrust magnitude and thrust direction are controlled by the hydraulically activated kinematics (gear and control rod) of the propeller (Figure 1.4), with a minimum of power consumption: two servomotors per propeller enable steering to X/Y coordinates (corresponding to the axes of the ship) through the variation of the driving pitch (that controls the magnitude of the thrust) and the steering pitch (related to the direction of the thrust).

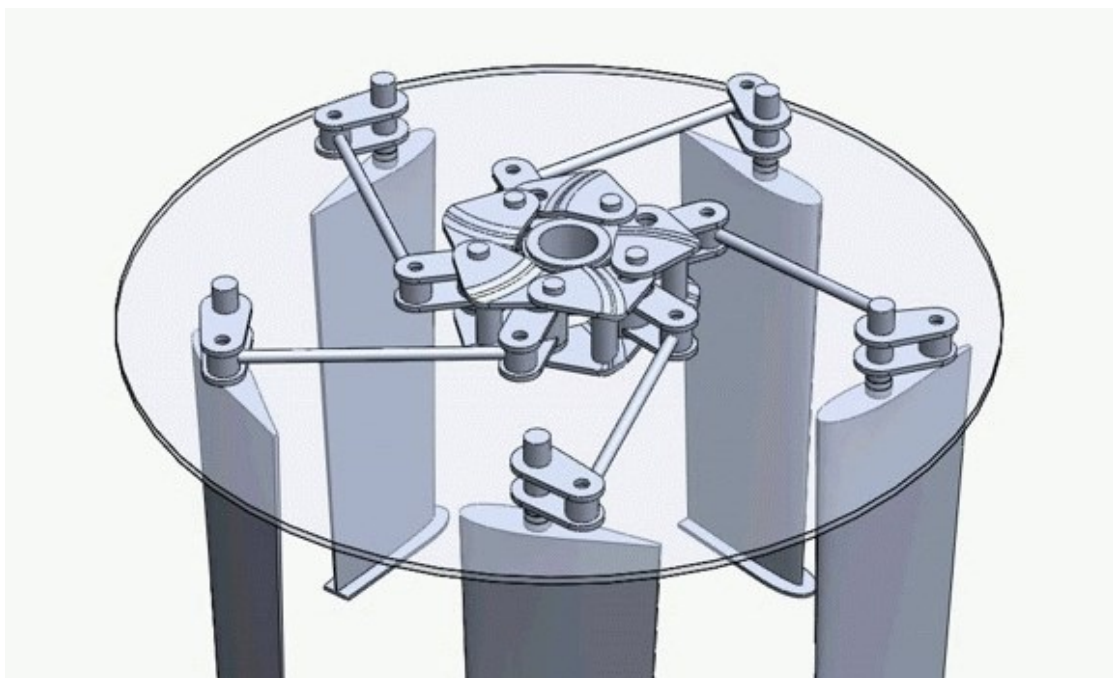


Figure 1.4: Epicycloidal propeller kinematics (<https://grabcad.com>)

Since the VSP simultaneously generates propulsion and steering forces, there is no need for additional appendages such as propeller brackets, rudders, shafts, etc.

A significant difference between the Voith Schneider Propeller and a screw propeller consists in the direction of the axis of rotation with respect to the direction of thrust: for screw propellers, the axis of rotation and the direction of thrust are identical, while for VSP they are perpendicular to each other (Figure 1.5). Thus VSPs have no preferential direction of thrust and allow stepless variations of thrust magnitude and direction.

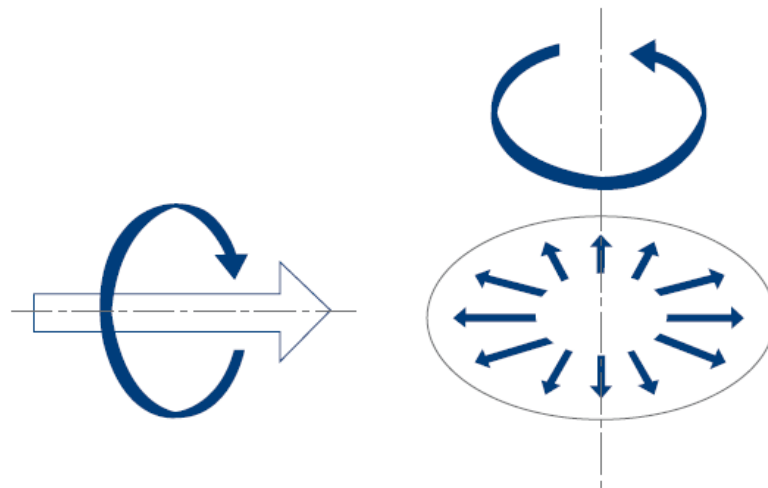


Figure 1.5: Axis rotation and direction of thrust (Precise and safe manoeuvring, Voith Turbo Schneider Propulsion, brochure)

Another important peculiarity of epicycloidal propellers is their DP capability. Thanks to the short response time they can easily help to react immediately to environmental disturbances. If the ship starts moving away from the desired position, an immediate adjustment of the pitches allows to balance wind, waves or current actions. Since VSP works almost at constant engine speed these manoeuvres can be done with low mechanical impact on the thrusters and low fuel consumption. The Voith Schneider Propeller receives mechanical power from a Diesel engine or an electric engine. Figure 1.6 shows the mechanical components of a cycloidal propeller. A flanged-on reduction gear (7), a gear box used to reduce the rpm of the main axis, and a bevel gear (8), a second smaller gear box inside the propeller with a fixed ratio of 76/17, provide the power required for thrust generation to the rotor casing (1). The blade shafts are supported by gland bearings or special roller bearings, while the rotor casing is axially supported by a thrust plate (4) and centred radially by a roller bearing. Due to kinematics system (3), the blades (2) perform an oscillating motion. Amplitude and phase of the blade motion are determined by the position of the steering centre. Thrust magnitude and direction can therefore be varied via control rod (10) that is actuated by two orthogonally arranged servomotors (11). There are two servomotors: the propulsion servomotor is used to adjust the pitch for longitudinal thrust (forward and reverse motion of the ship), the rudder servomotor is used for transverse thrust (motion to port and starboard). The two servomotors allow to change the direction of thrust from full ahead to full astern at a constant

speed of rotation without creating disturbing transverse forces. An oil pump (12) is flanged to the input gear.

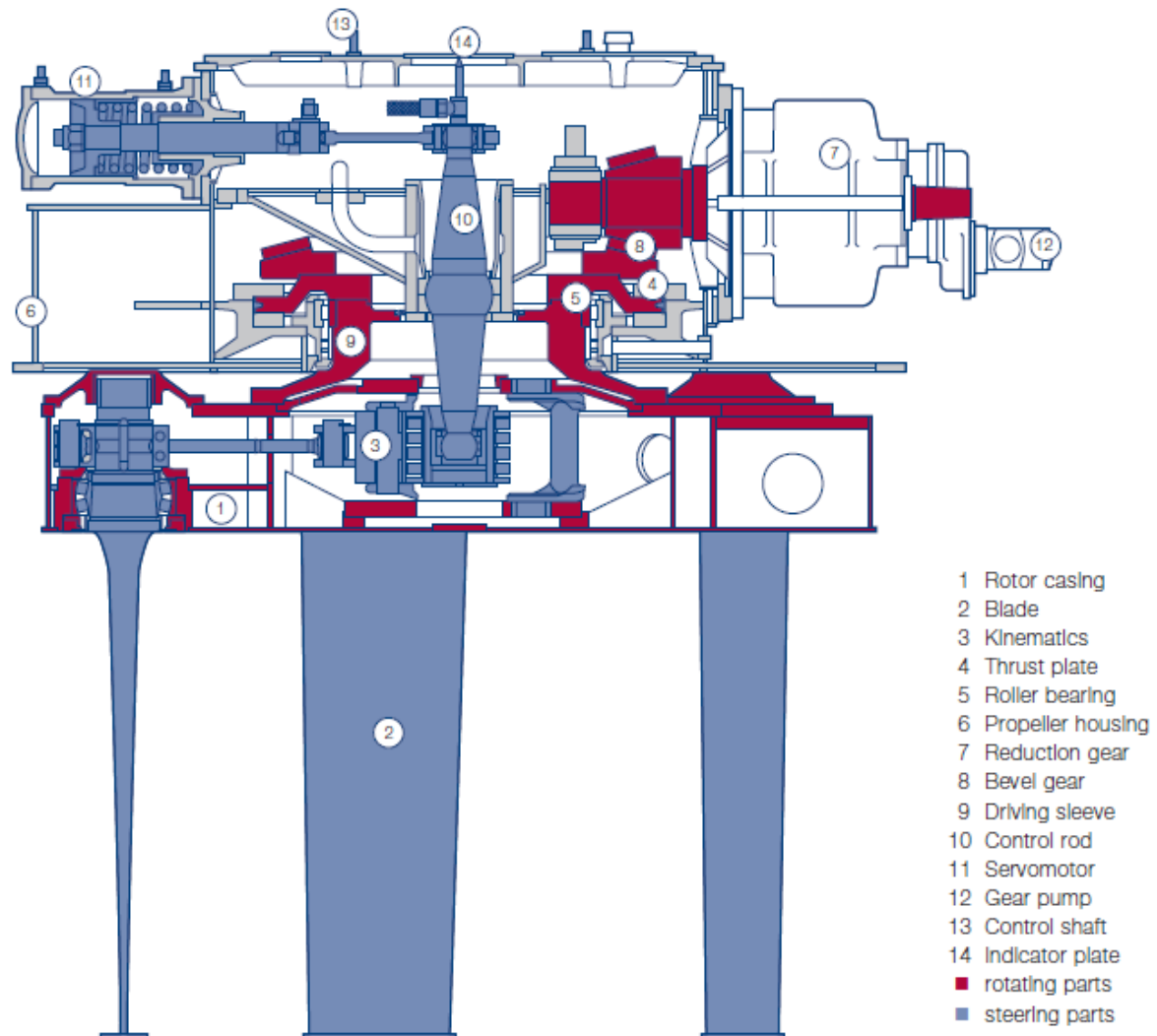


Figure 1.6: Mechanical components (Precise and safe manoeuvring, Voith Turbo Schneider Propulsion, brochure)

1.3 Current applications

The conventional arrangement consists of two VSPs aft. This solution is typical for vessels up to 100 m length and speed range around 16 kn. On larger ships three VSPs aft have been installed. All three propellers contribute to roll stabilization and thus generate a safe and stable working environment on board. The largest ships ever built with VSP propulsion are 160 m long, as the offshore construction vessel (OCV) North Sea Giant. Such ships have even five VSPs with a total installed power up to 19 MW. Three VSPs are installed aft and two in a recess in the bow. These kinds of ships have an excellent DP capability. In some cases, like Voith Water Tractor, VSPs are installed forward in the bow.

1.3.1 Voith Water Tractor (VWT)

VWT (Figure 1.7) are ship-handling tugs propelled by VSPs. Today there are more than 870 VWT in more than 145 port. The reason is that water tractors propelled by VSP are regarded as reliable, speed precise and efficient in towing and escorting service, salvaging, firefighting, oil pollution control and offshore supply.

The VWT is driven and controlled by epicycloidal propellers which ensure propulsion, steering and, in some cases, active roll damping. Since the thrust variation is independent of the ship's speed, VWT ensure maximum safety for the tow, the tug, the crew and the port facilities.

They are characterized by a bow steering (propellers are located in the bow section) that enable the interactive hydrodynamic forces between the tow and the tractor to be counteracted, even at high speed, and allow a stable equilibrium between the two forces acting on the tractor, propellers thrust and towing force.



Figure 1.7: Water tractor equipped with epicycloidal propellers (<http://www.voith.com>)

1.3.2 Double-ended ferries

Ferries priorities are passengers and cargo safety and short manoeuvring time. The propulsion and steering systems of a modern ferry must cope with currents, crosswinds and tidal changes in various water depths.

Double-ended ferries do not have to turn in port, cars can leave the deck in the same direction as they drove on and, if propelled by VSPs, the docking and undocking times is reduced thanks to their precise steerability and a high degree of safety for passengers and cargo is provided, even in heavy shipping traffic.

Voith Schneider Propellers are ideally suited for double-ended ferries (Figure 1.8) as they can be arranged to suit the specific vessel requirements.

Double-ended ferries with screw propellers have to compensate transverse forces due to wind and currents during crossing by means of a drift angle, which leads to higher drag and consequent requirements for additional power. VSP is able to generate sufficient transverse thrust components to compensate environmental forces without the need for the ferry to adopt a drift angle. In 1937, the first double-ended ferry equipped with VSPs was put into service. The "Lymington" successfully operated in Great Britain for 60 years.



Figure 1.8: Double ended ferry equipped with epicycloidal propellers (Tailor-made propulsion solutions for double-ended ferries, Voith Turbo Schneider Propulsion, brochure).

1.3.3 Mine Counter Measure Vessel (MCMV)

MCMVs equipped with VSPs (Figure 1.9) have a single propulsion system for mine hunting operations, for clearing sea mines and for travelling in open waters to the operation area. Extreme manoeuvrability, direction stability and good DP ability are required to MCMVs.

During mine hunting operations, signals that can be detected from the sensors of the mines are not allowed. The aim is to minimize the underwater noise caused by the propeller that can be radiated into the water by the ship structure. Since the sound pressure of VSP is proportional to the product $n^3 D^4$ (where n and D are the propeller revolutions and rotor diameter respectively), its acoustical behaviour is favourable because of the low operating speed of epicycloidal propellers. Moreover, VSPs installed on MCMVs are made up to 90% with non-magnetic steels and the few magnetic components are degaussed or magnetically neutralized.

Nowadays, there are more than 70 MCMVs fitted with VSPs sailing worldwide.



Figure 1.9: MCMV equipped with epicycloidal propellers (The Voith Schneider Propeller Current Applications and New Developments, Voith Turbo Schneider Propulsion, brochure).

1.4 *State of art*

As it has been mentioned above, available data about tests, simulations and mathematical models of epicycloidal propellers in literature are very few.

General descriptions and working principles of cycloidal propellers have been given by Harvald in 1983, van Manen and van Oossanen in 1988, Bartels and Jürgens (in 2000 they presented performance data for some VSP designs; most of the experiments were open water tests, but some works have been done in cavitation tunnels) and by Jürgens and Moltrecht in 2002.

In 1991 Brockett made a review of the hydrodynamic analysis of cycloidal propellers. Before him, Dobay and Dickerson (1975) and Nakonechny in 1974 described systematic experimental investigations of cycloidal propulsion, while Von Manen (in Netherlands in 1966, and in the United States, at the David Taylor Model Basin (DTMB) in 1969) made a review of the hydrodynamic analysis of cycloidal propellers as well.

All the significant works rely on the Taniguchi's method (1962), which gives reliable results, but only over a limited range of the advance ratio and for propellers with low pitch ratios (less than about 0.6π). Some other authors have tried to approach different solutions, making a review of the performance prediction methods for cycloidal propellers, concluding that Taniguchi's method was the best one.

In 1981, Zhu extended Taniguchi's method in order to improve the predictions and extend them over a wide range of advance coefficients for propellers with higher pitch ratios. Both of the methods made use of empirical correction factors derived from tests on cycloidal propellers in order to obtain the influence of the propeller blades on the incoming flow.

In 1973, Mendenhall and Spangler developed a time-domain numerical approach for 2D cycloidal propellers based on discrete vortex representation of the propeller blades and their wakes (Li 1991). This method accounts for the strength and position of the bound and wake vortices of all blades at each time step, and approximates the dissipation of the strength of the wake vortices due to viscosity. Also, stall and blade friction are modeled approximately. It was found out that predictions for epicycloidal propellers were in better agreement with experimental data than those for trochoidal propellers.

In 1987, Bose developed a simple method to calculate the performance of trochoidal propellers over the full range of advance ratios adapting a multiple streamtube theory (this theory was originally used for studying vertical axis wind turbines by Sharpe in 1984). The method gave underestimate values of thrust and torque coefficients at low advance ratios.

Brockett (1991) developed an approximate 3D-analysis technique for cycloidal propellers making use of elements from a lifting-line model and an actuator disk model. It almost is a quasi-steady analysis of the 2D flow at each blade section with addition for unsteady pitch variation in the incoming flow due to the blade motion, for axial-induced velocity and for added mass of the blade section. Comparing his data with experimental ones, Brockett concluded that this method is suitable for preliminary design since it is more accurate in predicting performance of high than low pitch ratio propellers. Moreover, a fully 3D potential flow model, which accounts for all singularities in the flow, should better predict loads and minimum pressure points.

In 1994, Riijarvi showed that the main limitation on accuracy of prediction method is due to the difficulty to define accurately the behavior of the flow near the blades and the stall in unsteady flows. Comparing experimental results of the performance of a trochoidal propeller with those obtained by theoretical predictions methods like vortex theory (Mendenhall and Spangler 1973 and its modification with Li 1991) and multiple streamtube momentum theory (Bose 1987), he obtained that for high pitch ratio values, the theoretical prediction methods gave good results in the approximation of induced velocities. Great influence is associated with the right choice of section data, especially the values of lift and drag coefficient at stall. The influence of stall behavior is higher for low pitch ratios.

In order to improve and correct the model as what concerns unsteady and turbulent flows through the propeller and over the blades at high angles of attack, the Voith Company realized a commercial RANS equation (RANSE) code called COMET, that has been used to model the behaviour of their epicycloidal propellers.

Since it is regarded as the most affordable, Taniguchi's method, as well as some attempts to improve it, will be here outlined.

1.4.1 *Taniguchi's method*

Taniguchi's method numerically evaluates the performance characteristics of vertical axis propellers and it is based on the assumption of a quasi-steady state motion. The total thrust and torque of the propeller are obtained by integrating the exerted lift and drag forces on each blade section.

The main hypotheses of this method are: only the longitudinal velocities induced by the trailing vortex system, i.e. those in the direction of propeller advance, contribute to the thrust and torque of the propeller and the magnitude of induced velocities are constant over the length of the blade. Each propeller rotates with a constant angular velocity around the vertical axis through the rotor centre O which advances at constant speed u_o . Depending on the incoming flow, each blade generates a hydrodynamic force that can be decomposed into two components: the lift $\underline{l}$, perpendicular to the direction of the incoming flow, and the drag $\underline{d}$, parallel to the incoming flow. Their intensities are respectively expressed as:

$$\underline{l} = c_l \frac{1}{2} \rho v^2 c \quad \underline{d} = c_d \frac{1}{2} \rho v^2 c \quad (1.4)$$

where v is the speed of the incoming flow, c is the chord of the blade, ρ the sea water density and c_l and c_d are lift and drag coefficients defined by Taniguchi (by wind tunnel tests on airfoil sections and experiments conducted on a 6-blade cycloidal propeller) as:

$$c_l = 5,34 \alpha \quad c_d = 0,019 + 2,24 \alpha^2 \quad (1.5)$$

where α is the attack angle of the blade section. Denoting θ the blade orbit angle, φ the blade angle, the thrust force generated by each blade section (Figure 1.10) is calculated as:

$$\underline{t} = \underline{l} \cos(\theta - \varphi + \alpha) + \underline{d} \sin(\theta - \varphi + \alpha) \quad (1.6)$$

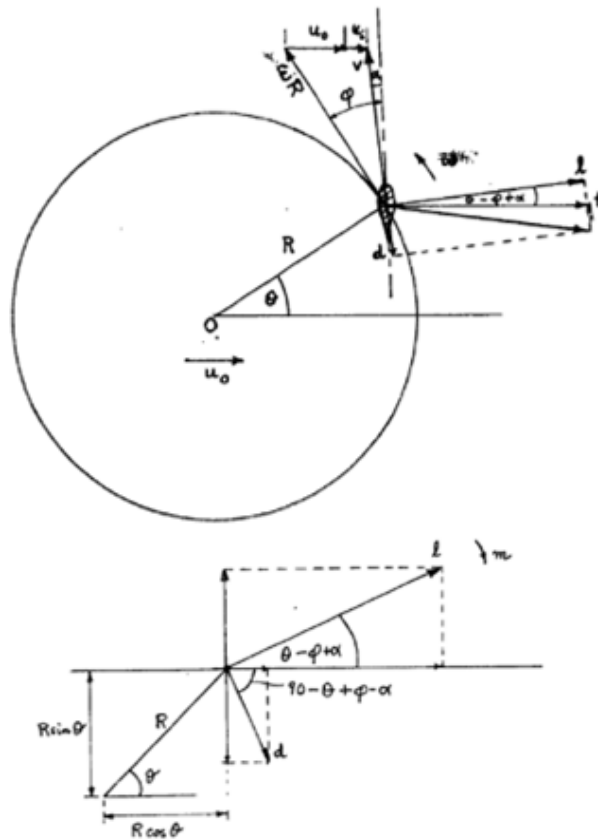


Figure 1.10: Taniguchi's Method - Forces generated in a blade section (Haberman et al. 1961).

Taniguchi decided to neglect the drag term since its contribution to the thrust is small in comparison to the lift contribution. Inserting the first of Eqs. (1.4) into Eq. (1.6), the expression of the thrust becomes:

$$\underline{t} = c_L \frac{1}{2} \rho v^2 c \cos(\theta - \varphi + \alpha) \quad (1.7)$$

After some trigonometric simplifications, the moment (with respect to the point O) acting on each blade section can be expressed as, as:

$$\underline{m} = \underline{l} R \sin(\theta - \alpha) + \underline{d} R \cos(\theta - \alpha) \quad (1.8)$$

The variation of the speed v at each blade section with blade orbit angle θ is calculated considering the sum of tangential speed ($\pi n D$) and the velocity given by the sum of advance velocity u_0 and induced velocity u_i (introduced by Taniguchi to take into account the fact that the trajectory of the flow is modified by the profile of the blades and assumed to be constant over the entire blade and independent of blade orbit position):

$$v^2 = (\pi n D)^2 + (u_0 + u_i)^2 - 2(u_0 + u_i)\pi n D \sin \theta \quad (1.9)$$

Knowing v and φ at every blade orbit position, it is possible to obtain the following relationship:

$$\tan \varphi = \frac{\eta \cos \theta}{1 - \eta \sin \theta} \quad (1.10)$$

Where η is the eccentricity of the cycloidal propeller, related to the maximum blade angle as indicated in Figure 1.11.

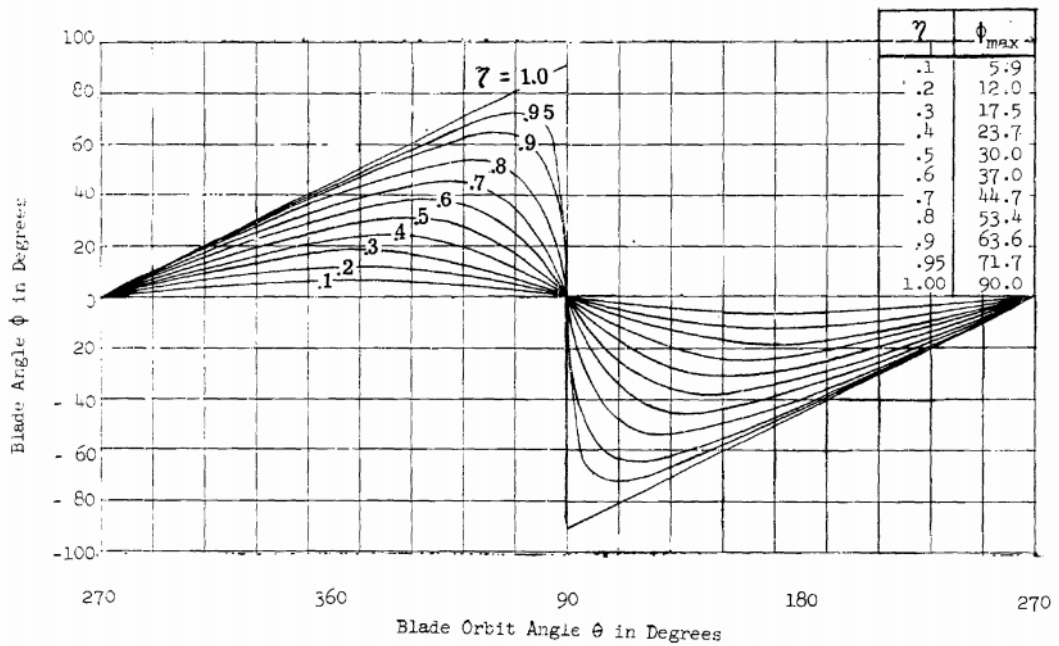


Figure 1.11: Variation of blade orbital angle with orbital blade position for various eccentricities (Haberman et al. 1961).

The thrust of each blade at a given orbital position is obtained by integrating t over the blade length:

$$T(\theta) = \int_0^b t \, db \quad (1.11)$$

For blades with semi-elliptic outline, the thrust of each blade is:

$$T(\theta) = \frac{\pi^2}{8} \rho c_L c n^2 D^2 (\eta - u_0 - u_i) \frac{\sqrt{1 + (\lambda_0 + \lambda_i)^2 - 2(\lambda_0 + \lambda_i) \sin \theta}}{1 + (\lambda_0 + \lambda_i) - (\eta + \lambda_0 + \lambda_i) \sin \theta} \cos \theta^2 \quad (1.12)$$

where λ is the advance coefficient.

The thrust of each blade is symmetric in the forward and aft half of the circle. In order to obtain the average thrust of each blade T_{AV} , $T(\theta)$ is integrated between $\pi/2$ and $-\pi/2$. The total thrust of the propeller is then obtained multiplying T by the number of blades. The total torque of the propeller is then given by:

$$Q = z \frac{1}{\pi} \int_{-\pi/2}^{\pi/2} \int_0^b m \, db \, d\theta \quad (1.13)$$

where z denotes the number of blades. Taniguchi defined thrust and torque coefficients and propeller efficiency as:

$$K_T = \frac{T}{\rho n^2 D^3 b} \quad K_Q = \frac{Q}{\rho n^2 D^4 b} \quad e = \frac{K_T \lambda}{K_Q 2} \quad (1.14)$$

and introduced the solidity of the blade with the term $\sigma = zc/\pi D$. Taniguchi made use of a correction factor K (estimated after experimental tests as 1,321) in order to take into account non-uniformity of induced velocity over the blade length and redefined thrust coefficient as:

$$K_T = 2\pi^2 \frac{\lambda_0 + \lambda_i}{K} \lambda \quad (1.15)$$

To obtain performance characteristics of the cycloidal propeller, the two expressions (1.14) and (1.15) for K_T must be solved simultaneously. The intersection of the curves at λ constant and σ constant gives the value of the desired thrust coefficient (Figure 1.12).

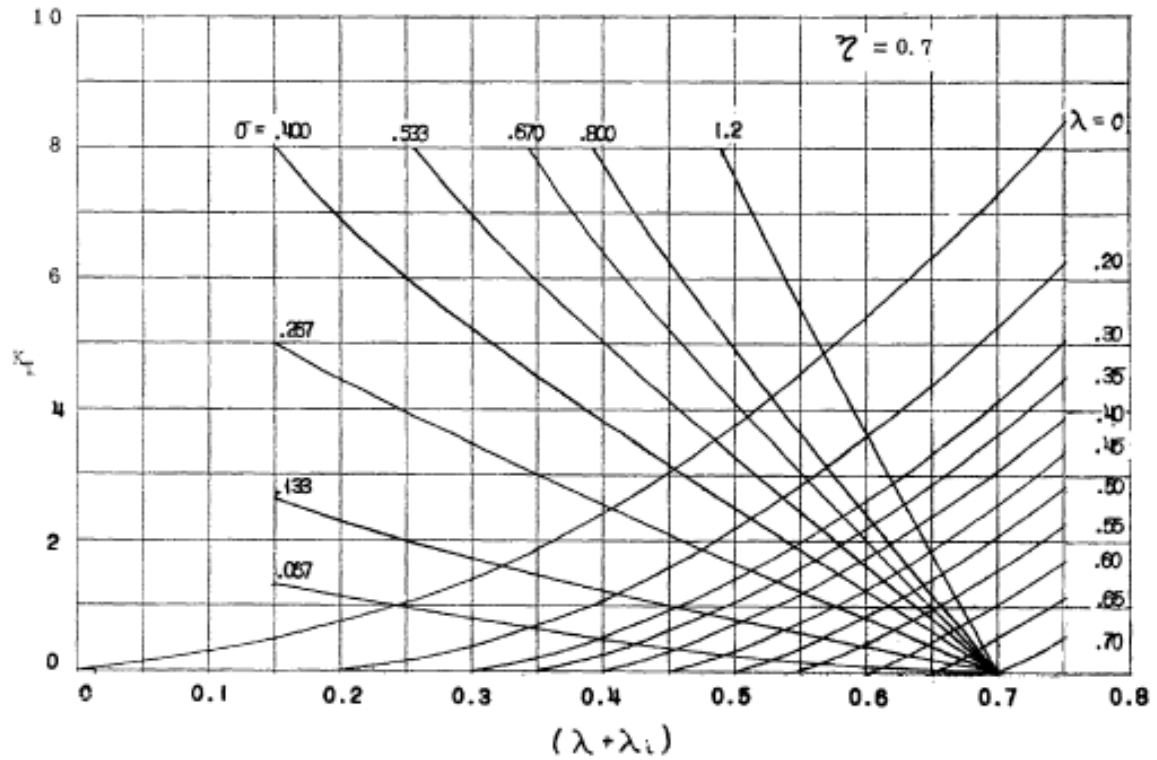


Figure 1.12: K_T as function of advance coefficient and induced velocities factor for various σ (Haberman et al. 1961).

In 1961, Haberman and Harley used Taniguchi's method for a performance analysis: thrust and torque coefficients and efficiency of vertical axis propeller with cycloidal blade motion and semi-elliptic blade outline were evaluated by the David Taylor Model Basin over a certain range of advance coefficients and maximum blade angles. The performance characteristics were evaluated at eccentricities up to 0.95. In some cases, the angle of attack exceeded the angle of stall. For this reason, no reliable predictions of the performance characteristics could be obtained for such cases. The maximum angle of attack occurs at zero advance coefficient. In the following Figure 1.13, the variation of angle of attack of the blade section with orbital position is reported.

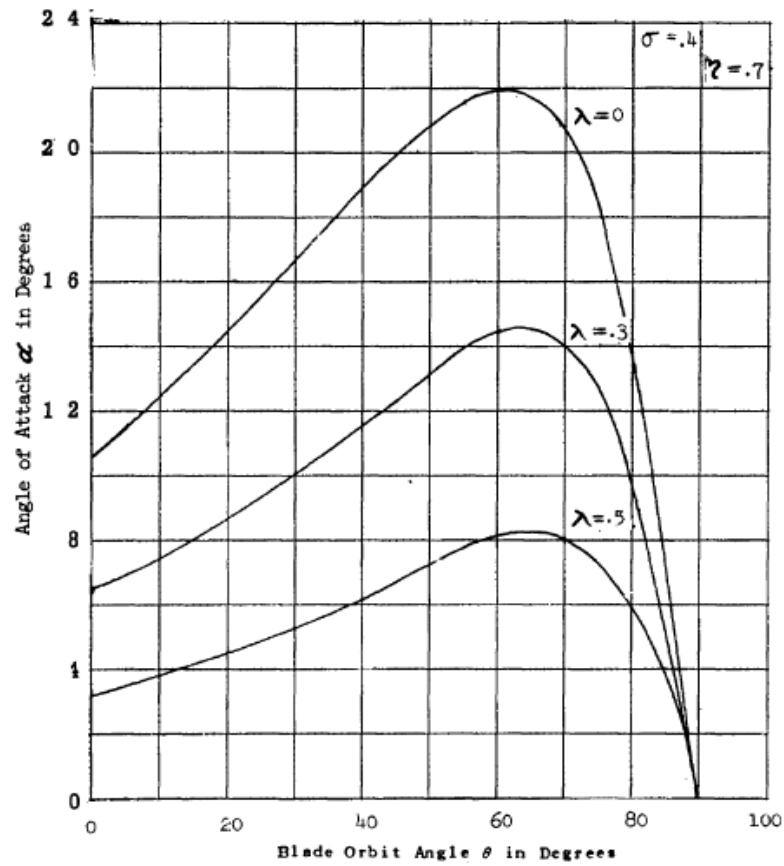


Figure 1.13: Variation of angle of attack of the blade section with orbital position (Haberman et al. 1961).

In their article, they affirm that total thrust, torque and maximum efficiency of cycloidal propellers increase with increase both of maximum blade angle and solidity, but they decrease with increase in number of blades.

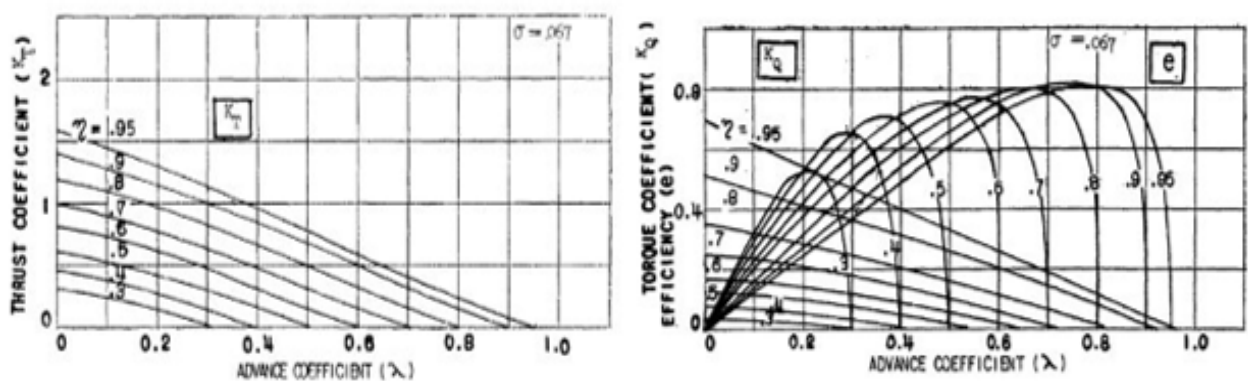


Figure 1.14: Variation of thrust (on the left) and torque coefficients and efficiency (on the right) with advance coefficient (solidity equal to 0,067) (Haberman et al. 1961).

Finally, a comparison was made between the results of the DTMB computations using Taniguchi's method and DTMB experimental measurements of the performance of vertical axis propellers with blades having semi-elliptic outline.

The experiments were conducted with propellers having two, three and six blades respectively. The eccentricity settings were 0,4 and 0,6. The aspect ratio of the blades was the same used by Taniguchi.

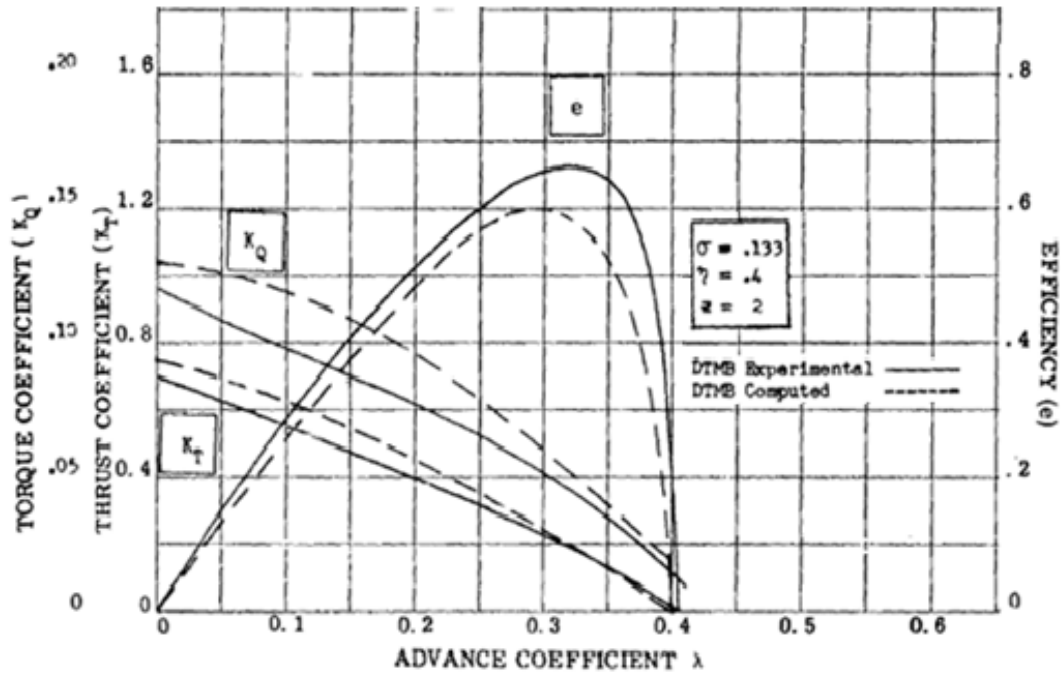


Figure 1.15: Comparison of computed and experimental performance characteristics of a vertical axis propeller with semi-elliptic blades (eccentricity 0,4 – solidity 0,133 – number of blades 2) (Haberman et al. 1961).

Haberman and Harley concluded that Taniguchi's method gave good results for cycloidal propellers with semi-elliptic blades. The main limitation of the method lies in the estimation of induced velocities and, in particular, in the assumption that only longitudinal components of induced velocities contribute to the propeller performance.

1.4.2 Nakonechny's method

The aim of Nakonechny's analysis was to find the open water diagram of a 6-blade Voith Schneider propeller. The results of his experimental investigations, conducted at different pitch values using modified cycloidal and sinusoidal blade motion, are presented in report 1446 of the David Taylor Model Basin. The results are presented in form of performance characteristic curves using new non-dimensional K and λ coefficients, adapted to vertical axis propellers. Nakonechny used Isay's method because in the literature there are few data which allow the design of vertical axis propellers making use of the standard design methods for screw propellers.

Nakonechny defined the non-dimensional coefficients according to:

Advance coefficient	$\lambda = \frac{V_a}{\pi n D}$
Thrust coefficient	$K_T = \frac{T}{\rho L n^2 D^3}$
Torque coefficient	$K_Q = \frac{Q}{\rho L n^2 D^4}$
Efficiency	$e = \frac{T V_a}{2\pi n Q} = \frac{K_T \lambda}{K_Q 2}$

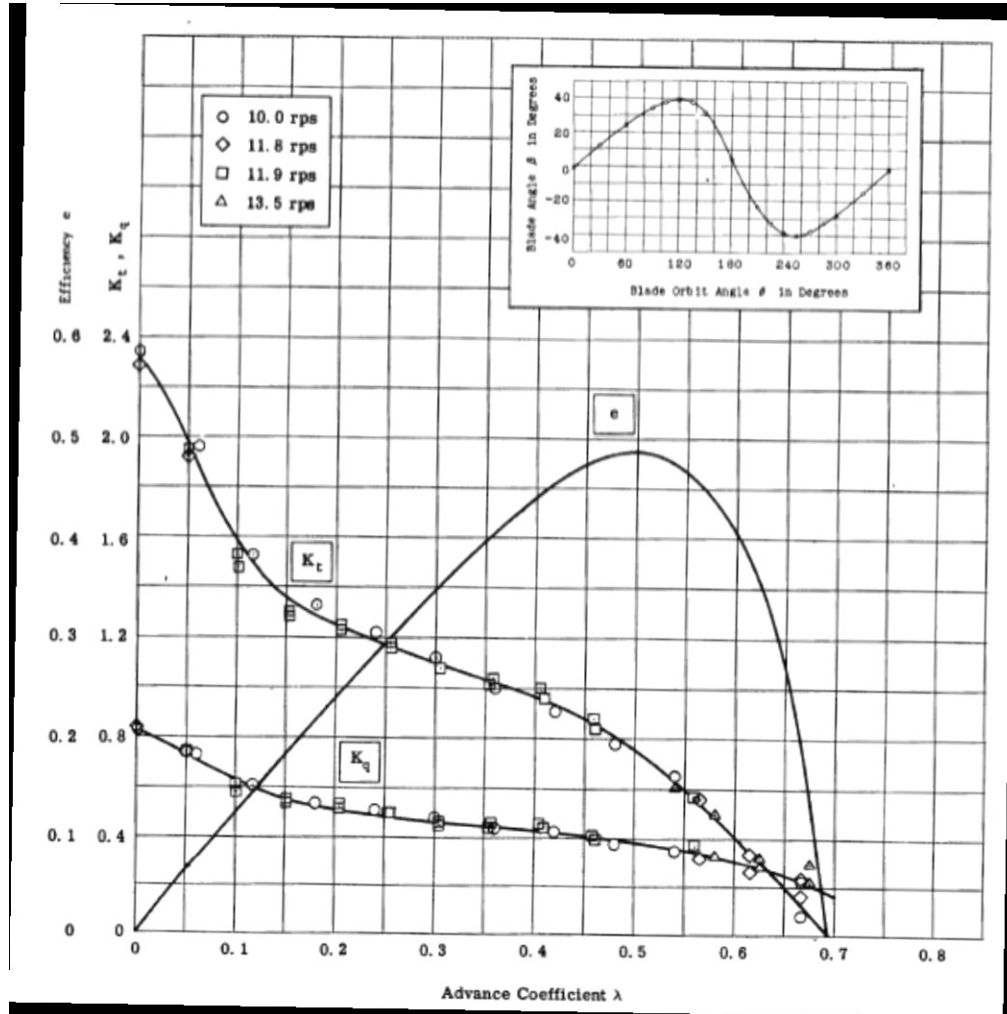


Figure 1.16: K_T , K_Q and e curves for a vertical axis propeller with cycloidal blade motion (Nakonechny 1976).

As a result, he found out that the curves of thrust and torque coefficients (Figure 1.16) are more linear for lower angles of attack and that lift and drag increase with the increases of the pitch, i.e. with the increase of the angle of attack of the blade.

1.4.3 Jürgens and Palm

Jürgens and Palm used numerical simulation and Computational Fluid Dynamics to optimize the efficiency of the VSP using a suitable parameter, called Blade Steering Curve (BSC).

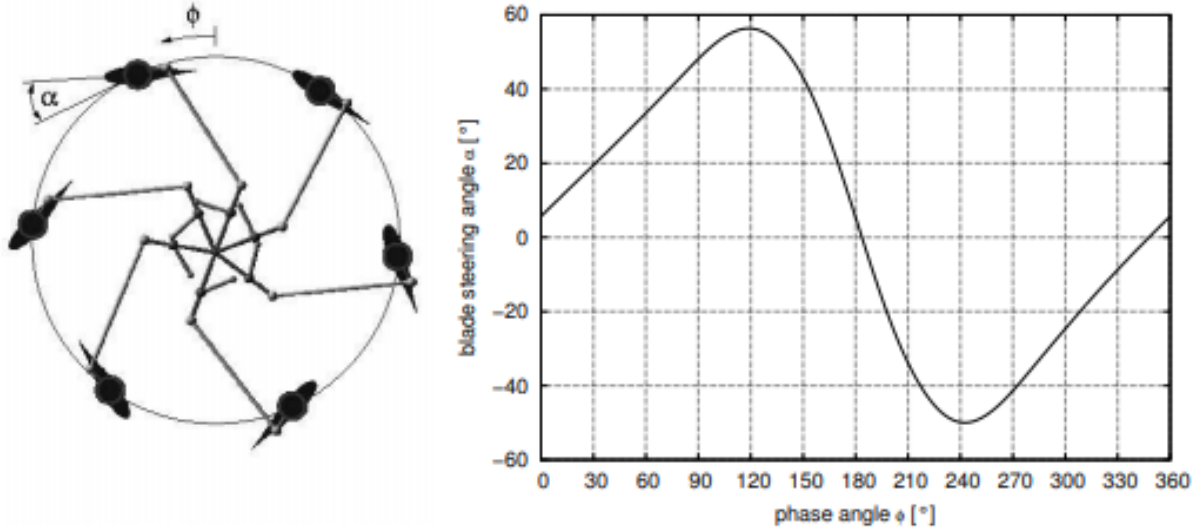


Figure 1.17: Blade steering curve (BSC) (Jürgens et al. 2007).

The BSC could also be investigated by physical experiments, but in order to determine the significant parameters Jürgens and Palm decided to use numerical simulation. In fact, they would have needed a huge amount of experiments for a good optimization and, furthermore, there were scaling problems when comparing model results to real data (for example, the effect of the Reynolds number did not scale appropriately because the experiments were carried out in accordance to Froude's similitude).

They defined a mathematical model for the flow induced by the VSP and a mathematical formulation of the corresponding optimization problem. They described in mathematical terms an appropriate BSC as the periodic function:

$$s: [0, 2\pi] \rightarrow [-\pi, \pi], \Phi \rightarrow s(\Phi) = \alpha \quad (1.16)$$

where Φ and α are related as shown in Figure 1.17.

Technical reasons do not allow rapid changes of the orientation of the blades and this corresponds to smoothness of BSC. For this reason, they introduced the set of valid BSC according to:

$$C_{2\pi}^2: \{ s : [0, 2\pi] \rightarrow [-\pi, \pi] : s \in C^2[0, 2\pi], s(0) = s(2\pi) \} \quad (1.17)$$

To optimize the efficiency of the VSP as a function of the BSC, the objective function was defined as:

$$E: C_{2\pi}^2 \rightarrow \mathbb{R}, \quad E: s \mapsto E(s) := \eta_o(s) \quad (1.18)$$

where η_o is the efficiency. In order to define $E(s)$, the flow induced by the rotation of the cycloidal propeller needed to be described.

The Navier Stokes equations used for studying the flow in the medium of water were simplified under the assumption of incompressible fluid:

$$\frac{\partial}{\partial t} \bar{u} + \bar{u} \nabla \bar{u} - \nu \Delta \bar{u} + \frac{1}{\rho} \nabla p = \bar{f} \quad \text{div } \bar{u} = \nabla \bar{u} = 0 \quad (1.19)$$

where $\bar{u}$ represents the unknown velocity field, p the unknown pressure, $\bar{f}$ exterior forces, ρ is the (constant) density and ν is the kinematic viscosity. All these functions depend on time $t \in [0, T]$ (T corresponds to one full rotation of the propeller) and space $x \in \Omega \subset \mathbb{R}^3$ (Ω is the domain of the flow). In addition, appropriate initial and boundary conditions had to be imposed.

The pressure force is given by the integration of the pressure along the surface of the blades A :

$$\bar{F}_p(t) = \int p(t, \bar{x}) \bar{n}(t, \bar{x}) dA \quad (1.20)$$

$\bar{n}(t, \bar{x})$ is the outward normal vector on the blade surface (Figure 1.18).

The force generated by the shear stress $\bar{\tau}$ is given by :

$$\bar{F}_r(t) = \int \bar{\tau}(t, \bar{x}) dA \quad (1.21)$$

The force in the driving direction $\bar{d}$ of the ship is the thrust force T :

$$T(t) = (\bar{F}_p(t) + \bar{F}_r(t)) \cdot \bar{d} \quad (1.22)$$

The torque M consists of two parts: M_G is the torque related to the global propeller rotation and M_B is the torque necessary to steer the blades accordingly to the BSC s.

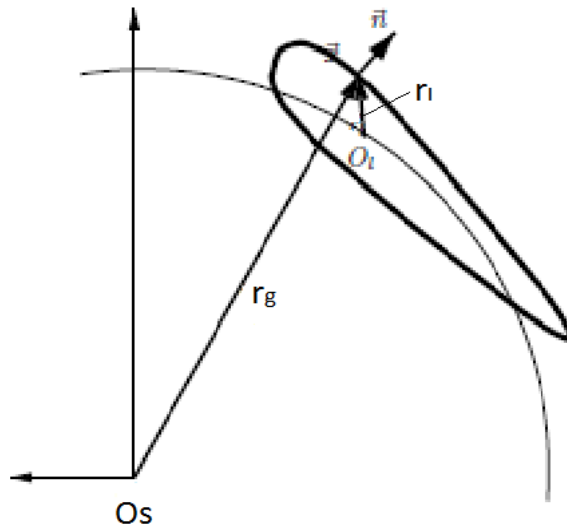


Figure 1.18: System of references of the model (Jürgens et al. 2007).

$$M_G(t) = \left(\int [p(t, \bar{x}) \bar{n}(t, \bar{x}) x \bar{r}_g + \bar{F}_r(t) x \bar{r}_g] dA \right) \bar{z}_D \quad (1.23.a)$$

$$M_B(t) = \left(\int [p(t, \bar{x}) \bar{n}(t, \bar{x}) x \bar{r}_l + \bar{F}_r(t) x \bar{r}_l] dA \right) \bar{z}_D \quad (1.23.b)$$

$$M(t) = M_G(t) + M_B(t) \frac{1}{2\pi n} \frac{\partial s(t)}{\partial t} \quad (1.23.c)$$

where $\bar{r}_g$ is the lever of the global rotation and $\bar{r}_l$ is the lever of the blade rotation.

By integrating with respect to time t over one complete propeller turn (C), total lift and total torque of the propeller are obtained:

$$T = \int_0^{2\pi} T(t) dc \quad (1.24.a)$$

$$M = \int_0^{2\pi} M(t) dc \quad (1.24.b)$$

The propulsion efficiency of the free running propeller is defined as the ratio of the propulsion power Tv_a and the delivered power $M2\pi n$:

$$\eta_0 = \frac{Tv_a}{M2\pi n} \quad (1.25)$$

Lift coefficient, torque coefficient and efficiency are then defined in the following way:

$$k_s = \frac{T}{\frac{1}{2}\rho D L (D\pi n)^2} \quad (1.26.a)$$

$$k_d = \frac{2M}{\frac{1}{2}\rho D^2 L (D\pi n)^2} \quad (1.26.b)$$

$$\eta_0 = \frac{k_s}{k_d} \lambda \quad (1.26.c)$$

The efficiency optimization corresponds to an unconstrained optimization problem with a nonlinear objective function. A common way of solving these equations are Finite Element or Finite Volume methods. For the VSP, Voith has adapted a commercial FV code (Comet) that is used for research and development. However, Jürgens and Palm decided not to use a FV method for Navier Stokes equations. By introducing some simplifications (as shown in Figure 1.19), the flow can be studied by the Euler equation or, for potential flow, by the Laplace equation. For their analysis Jürgens and Palm used a fast Vortex Lattice method.

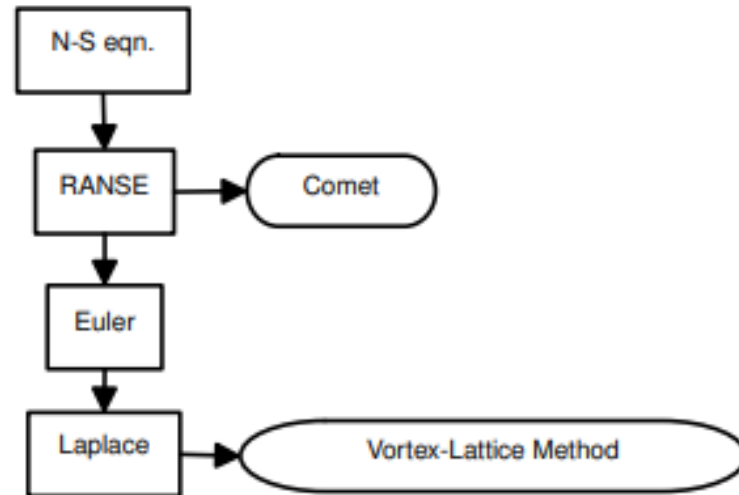


Figure 1.19: Methods for studying flows (Jürgens et al. 2007).

The Vortex Lattice Method considered a potential flow, i.e. an incompressible, inviscid and irrotational fluid. The potential flow is described by the Laplace equation which is linear, so it can be solved using the principle of superposition. The potential problem is studied using vortices on the blade surface (bound vortices) and on the wake (free vortices). Boundary conditions must be imposed, like the Kutta condition which states that the flow is parallel to the trailing edge of the blade.

The problem is discretized for numerical calculation. Vortices are placed on a lattice on the blade surface and wake of the blade.

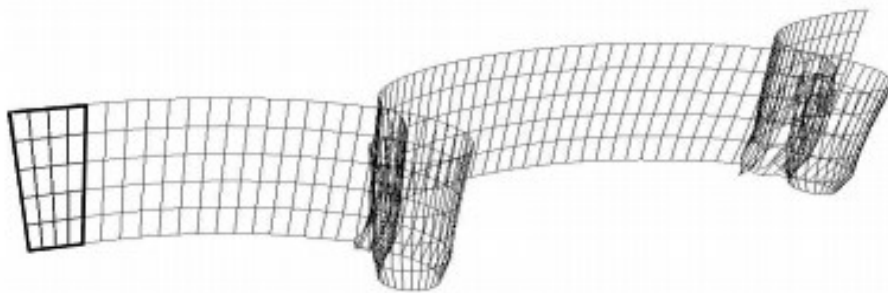


Figure 1.20: Representation of finite elements associated to vortices in two turn of the system (Jürgens et al. 2007).

The equations have to be satisfied on a so called collocation point in each element of the fixed vortex lattice. This linear equation must be solved:

$$A\Gamma = b \quad (1.27)$$

where A denotes a matrix according to the geometry of the blade, Γ is the strength of the bound vortices that must be found and b represents contributions due to the influence of free vortices, the inflow velocity and the boundary conditions.

As a result, a comparison between original and optimized BSC is shown in Figure 1.21:

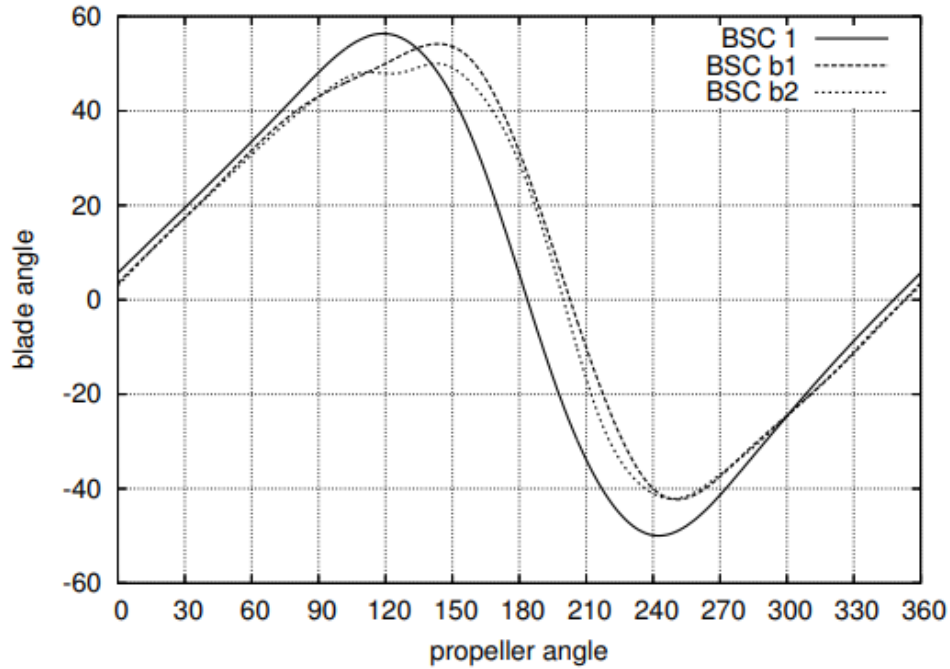


Figure 1.21: Original and optimized BSC (Jürgens et al. 2007).

1.4.4 West Bengal study

Some authors from the Ocean Engineering & Naval Architecture university of West Bengal in India, presented (Nandy S. et al. 2019) in 2019 a simulation study of a PID controller for marine cycloidal propellers.

A sketch of coordinate system and different angles of blades is shown in Figure 1.22.

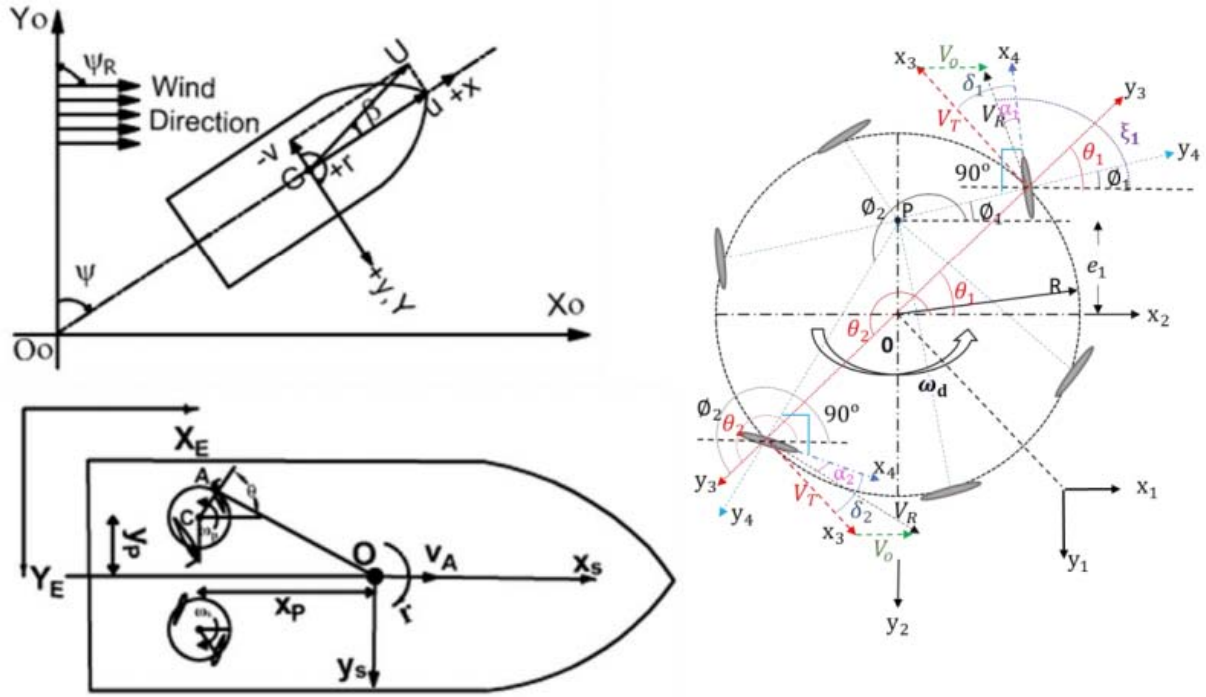


Figure 1.22: Coordinate system and Different angles of blades (Nandy et al. 2019)

Making use of a detailed mathematical model for the kinematics of the problem, lift and drag forces are calculated by means of NACA 0024 data. After this, the generated thrust, torque and transverse force on each blade are evaluated as:

$$T_{i\{S\}} = L_i \sin \xi_{i\{S\}} - F_{Di} \cos \xi_{i\{S\}} \quad (1.28.a)$$

$$Q_{D\{S\}i} = \left[L_i \cos \left(\xi_{i\{S\}} - \theta_{i\{S\}} \right) - F_{Di} \sin \left(\xi_{i\{S\}} - \theta_{i\{S\}} \right) \times R \right] \quad (1.28.b)$$

$$F_{i\{S\}} = L_i \cos \xi_{i\{S\}} - F_{Di} \sin \xi_{i\{S\}} \quad (1.28.c)$$

The moment of hydrodynamic forces with respect to origin of the ship coordinate system is then given by:

$$M_{Oi\{S\}} = \begin{bmatrix} X_{Bi\{S\}} & Y_{Bi\{S\}} & Z_{Bi\{S\}} \end{bmatrix} \times \begin{bmatrix} T_{i\{S\}} & F_{i\{S\}} & 0 \end{bmatrix} \quad (1.29)$$

$\{S\}$ indicates starboard or port propeller, $T_{i\{S\}}$ is the thrust acting on the blade stock due to hydrodynamic action along the i -axis of rotor co-ordinate system, ξ is the angle between resultant flow and thrust direction, L is the lift force on the blade, F_D the drag, θ is the blade orbit angle and R is the radius of propeller disc.

The centrifugal force influence on the total blade thrust is considered insignificant because centrifugal force of opposite blades on the same disc neutralize each other in one turn. Additionally, the torque developed on the propeller blade due to centrifugal force can be minimized by placing the blade stock at the centre of gravity of the blade.

Evidences of the thruster model are not given since the main part of the article is the second one dealing with the PID controller used to govern internal mechanism of the propeller.

1.4.5 *Remarks on the state of art*

The studies presented in the previous paragraphs describe different approaches to deal with the mathematical modelling of cycloid thrusters in order to obtain the open water diagram of the propeller, without showing any application to hull models and without studying the behaviour of ships equipped with cycloidal propellers.

The main purpose of this thesis is first of all to provide a mathematical model (Chapters 2 and 3) based on physical equations which, although based on some empirical hypotheses, provide data that cannot be found in the literature (as they are known only from the manufacturer and not disclosed) and which are necessary to describe the behaviour of the cycloidal propeller. The mathematical model will be used to develop a simulator that, with the exception of some correction coefficients, can be used for different kinds of cycloidal propellers by varying only the construction parameters (rotor diameter, blade length and profile). Once this first objective will be achieved, the matching of the cycloidal propeller model with hull models, already perfected and calibrated, will be studied in order to evaluate both the performance during dynamic positioning manoeuvres (Chapter 4) and the manoeuvring modes during cruising speed and sea trials (Chapter 5), trying to reproduce them in order to better understand how to drive with this type of propellers.

2 KINEMATICS

The epicycloidal propeller allows precise and uninterrupted thrust generation since propulsion and steering forces can be varied simultaneously. As a result of the rotation around its vertical axis, the same amount of thrust can be provided almost over 360° by blades with hydrodynamically shaped profiles that assure a high degree of efficiency.

2.1 *Blade motion*

The rapid and precise thrust variation of Voith Schneider Propellers is based on the kinematics of the blades (usually from 4 to 6 and equally spaced from each other) that move along a circular path, centred in the rotor centre, and at the same time perform a superimposed pivoting motion around their vertical axis passing through the centre of mass.

When the steering centre overlaps the centre of the rotor casing, the blades are not angled with respect to the tangent to the blade circular trajectory (Figure 2.1b) and no thrust is generated in this circumstance.

If the steering centre is moved away from the centre of the rotor casing, the blades are set at a variable angle with respect to the tangent of their circular path (Figure 2.1a). During the revolution motion, the maximum angle reached by the blades increases with the eccentricity. The eccentricity, that is the geometric pitch, can be varied in a very quick way and with low power and it is defined as:

$$e = \frac{ON}{D/2} \quad (2.1)$$

where ON is the distance between the centre of the rotor and the steering centre and $D/2$ is the radius of the rotor. The motion of the pivot point (assumed as the centre of mass) of the blade, relative to a stationary observer, results from the superimposition of the rotational movement of the rotor casing along a straight line representing the forward motion of the vessel. The pivot point follows a curve of a cycloid. The rolling radius of the cycloid is $D/2$, where λ is the advance coefficient defined as:

$$\lambda = \frac{V_A}{\pi n D} \quad (2.2)$$

During one revolution, the propeller travels a distance $\lambda D \pi$ in the direction of motion of the vessel.

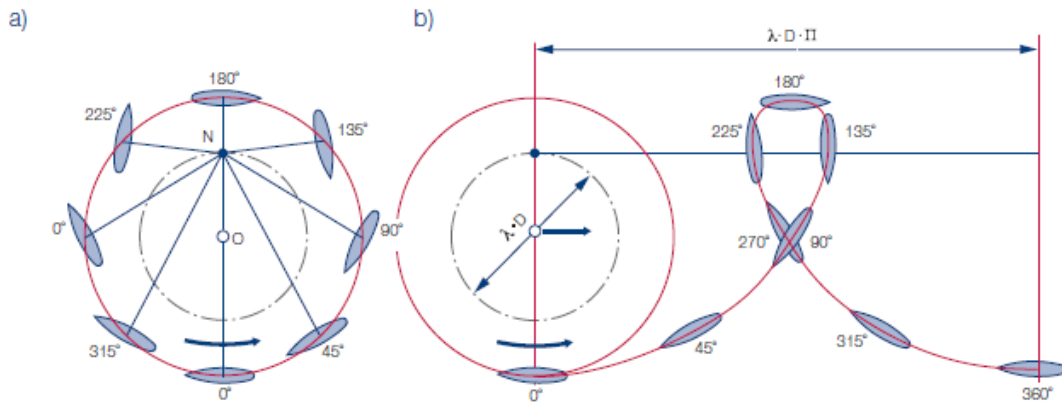


Figure 2.1: Cycloidal path of a VSP blade (Precise and safe maneuvering, Voith Turbo Schneider Propulsion, brochure)

To generate thrust, the blades are set at an angle relative to their path. To achieve this, the steering centre is moved from O to N. The resulting angle of attack leads to the generation of hydrodynamic lift and drag forces on each blade.

The thrust of the propeller is always perpendicular to the line ON. By shifting the steering centre N, it is possible to produce thrust in any direction (Figure 2.2). So, epicycloidal propellers allow to increase and decrease the thrust and change its direction through two parameters (plane polar coordinates):

- the geometric or driving pitch (between 0 and 0.8R for constructive limits): that is the distance (expressed as a percentage of the rotor radius R) between the steering centre N and the centre of the rotor O;
- the steering pitch (between 0° and 360°): the angle between a fixed axis (with respect to the hull) and the line ON.

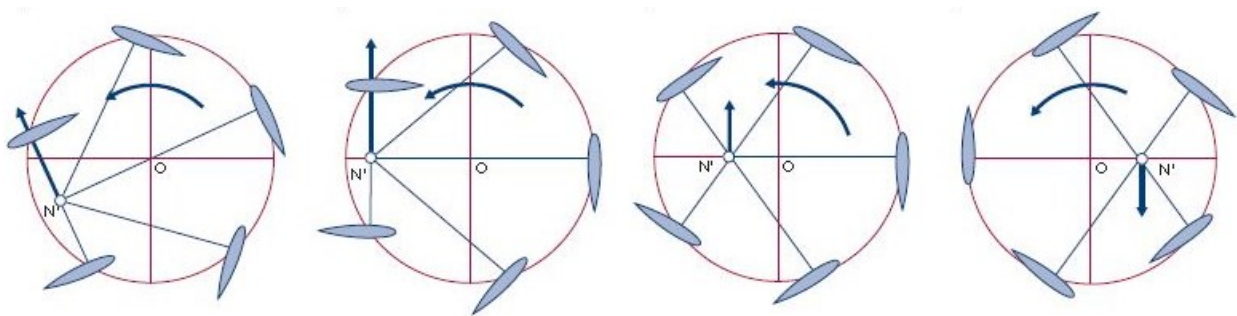


Figure 2.2: Blades position and thrust direction (Precise and safe maneuvering, Voith Turbo Schneider Propulsion, brochure)

Consideration of the processes on each blade position during one revolution provides the simplest explanation of the blades velocities and the resultant hydrodynamic forces.

2.2 Thrust generation

Changing the steering centre (from N to N'), the blades acquire a certain attack angle, so generating corresponding lift and drag forces which give rise to the desired thrust. The hydrodynamic forces components acting transversely to the desired thrust direction cancel each other out (Figure 2.3). It is possible to produce thrust in any direction putting the steering centre in the right position. The zero-thrust condition can be selected at any time, making the ship very safe to handle.

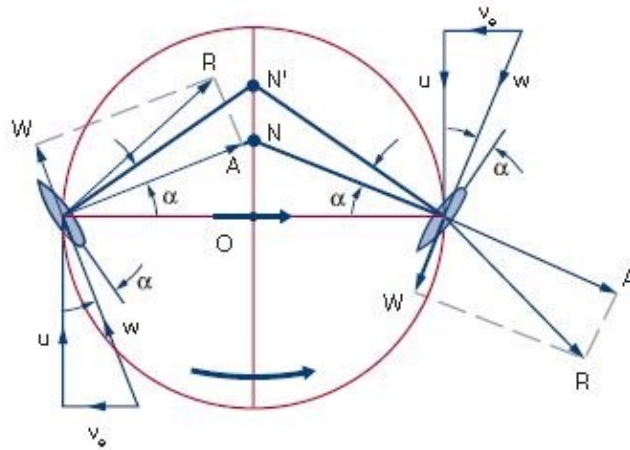


Figure 2.3: Forces generated by the blade for two angular positions (Precise and safe maneuvering, Voith Turbo Schneider Propulsion, brochure)

Hydrodynamic lift is the physical principle involved in the generation of thrust of cycloidal propellers. Its distribution over the blade is shown in Figure 2.4.

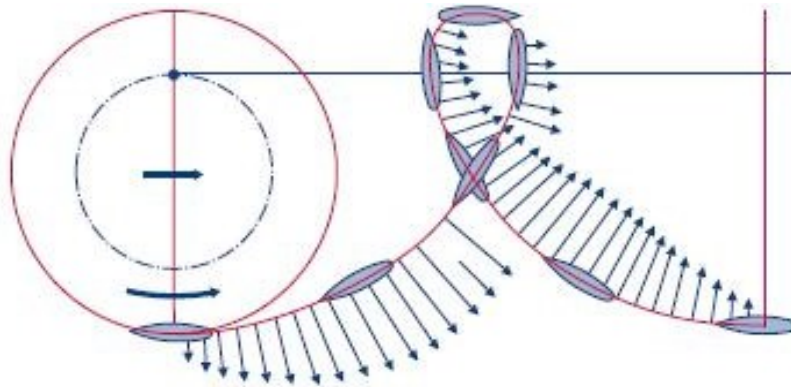


Figure 2.4: Lift distribution over the blade path curve (Precise and safe maneuvering, Voith Turbo Schneider Propulsion, brochure)

Each blade generates instantly a hydrodynamic force which results to be the sum of the lift (component of the hydrodynamic force, perpendicular to the incoming flow) and the drag (parallel

to the incoming flow). The sum of all the hydrodynamic forces generated by all the blades gives rise to the corresponding total thrust.

Also, each blade generates a corresponding torque which contributes to the total torque $\underline{M}$ of the propeller.

For each percentage of pitch, there are corresponding curves of K_S and K_D as functions of the advance coefficient λ . K_S is the thrust coefficient while K_D is the torque coefficient and they are defined in a similar way as for screw propeller, by the following formulae:

	CYCLOIDAL PROPELLER	SCREW PROPELLER
Advance coefficient	$\lambda = \frac{V_A}{\pi n D}$	$J = \frac{V_A}{\pi n D}$
Thrust coefficient	$K_S = \frac{T}{\frac{1}{2} \rho D L u^2}$	$K_T = \frac{T}{\rho n^2 D^4}$
Torque coefficient	$K_D = \frac{4M}{\rho D^2 L u^2}$	$K_Q = \frac{M}{\rho n^2 D^5}$
Efficiency	$\eta_0 = \frac{K_S}{K_D} \lambda$	$\eta_0 = \frac{K_T}{K_Q} \frac{J}{2\pi}$

Table 2.1: Cycloidal and screw propeller coefficient

Where:

- V_A advance velocity
- n rpm
- D rotor diameter
- ρ salted water density
- L blade length
- u tangential speed ($u = \pi n D$)

2.3 Kinematics of the blade

The kinematical model adopted to describe the motion of each blade of a given epicycloidal propeller is sketched. For simplicity, a 2-dimensional plane model is adopted, where two distinguished reference frames are introduced: the first one $(O, \underline{b}_1, \underline{b}_2, \underline{b}_3)$ is fixed to an

hypothetical hull and it has its origin O at the centre of the rotor, the unit vector $\underline{b}_1$ points towards the bow, the unit vector $\underline{b}_2$ points towards starboard and the unit vector $\underline{b}_3 = \underline{b}_1 \wedge \underline{b}_2$ points downwards; the second one $(O, \underline{e}_1, \underline{e}_2, \underline{e}_3)$ rotates clockwise about the vertical axis passing through O and parallel to $\underline{b}_3 = \underline{e}_3$ by an angle $\beta \in [0, 2\pi]$ which determines (the perpendicular of) the steering force direction. The angle β is related to the rudder pitch of the epicycloidal propeller. The steering centre C lies on the straight line passing through O and parallel to $\underline{e}_2$. The linear transformation between the bases $\{\underline{b}_i\}$ and $\{\underline{e}_i\}$ is expressed as

$$\begin{cases} \underline{e}_1 = \cos\beta \underline{b}_1 + \sin\beta \underline{b}_2 \\ \underline{e}_2 = -\sin\beta \underline{b}_1 + \cos\beta \underline{b}_2 \\ \underline{e}_3 = \underline{b}_3 \end{cases} \quad (2.3)$$

During the revolution motion, the projection P of the blade shaft on the plane $\langle O, \underline{b}_1, \underline{b}_2 \rangle$ describes a circumference having centre O e radius R coinciding with the rotor radius. In Cartesian coordinates associated with the frame $(O, \underline{b}_1, \underline{b}_2, \underline{b}_3)$, such a circumference is parameterized by

$$P(\theta) : \begin{cases} x = R\cos\theta \\ y = R\sin\theta \\ z = 0 \end{cases} \quad (2.4)$$

where θ denotes the angle (function of time) describing the revolution motion of the blade. The unit vector $\underline{t}$ tangent to the circular path of P has components in the vector basis $\{\underline{b}_i\}$ of the form

$$\underline{t}(\theta) = \begin{cases} t_1 = -\sin\theta \\ t_2 = +\cos\theta \\ t_3 = 0 \end{cases} \quad (2.5)$$

Introducing the vector

$$(C - O) = s\underline{e}_2 = -s \sin\beta \underline{b}_1 + s \cos\beta \underline{b}_2, \quad s \in [0, 0.8R] \quad (2.6)$$

the vector joining the steering centre C with the point P can be expressed as

$$(P - C) = (R \cos\theta + s \sin\beta)\underline{b}_1 + (R \sin\theta - s \cos\beta)\underline{b}_2 \quad (2.7)$$

The variable s is usually called driving pitch and controls the magnitude of the thrust. The unit vector orthogonal to $(P - C)$ and belonging to the plane $\langle O, \underline{b}_1, \underline{b}_2 \rangle$ identifies with the unit vector of the blade chord and it is given by

$$\frac{(P-C)^\perp}{|(P-C)^\perp|} = \frac{(-R\sin\theta + s\cos\beta)\underline{b}_1 + (R\cos\theta + s\sin\beta)\underline{b}_2}{((-R\sin\theta + s\cos\beta)^2 + (R\cos\theta + s\sin\beta)^2)^{\frac{1}{2}}} \quad (2.8)$$

The pivoting motion of the blade around its own vertical axis can be described by the angle α (function of time) between the unit vectors $\underline{t}$ and $\frac{(P-C)^\perp}{|(P-C)^\perp|}$. Due to the relation

$$\cos\alpha = \frac{(P-C)^\perp}{|(P-C)^\perp|} \cdot \underline{\hat{t}} = \frac{R+s\sin(\beta-\theta)}{((-R\sin\theta+s\cos\beta)^2+(R\cos\theta+s\sin\beta)^2)^{\frac{1}{2}}} \quad (2.9)$$

where the dot denotes the usual scalar product between vectors, choosing anticlockwise the positive direction of rotation around the blade shaft, the pivoting angle α can be defined as

$$\alpha = \pm \arccos\left(\frac{(P-C)^\perp}{|(P-C)^\perp|} \cdot \underline{\hat{t}}\right) \quad \text{where} \quad \begin{array}{l} + \text{ if } \cos(\theta - \beta) \geq 0 \\ - \text{ if } \cos(\theta - \beta) < 0 \end{array} \quad (2.10)$$

The above outlined kinematical model can be summarized by the following figure.

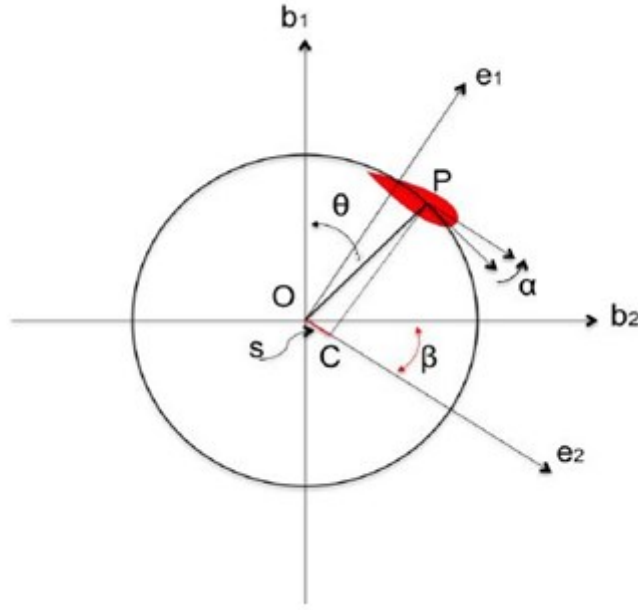


Figure 2.5: Kinematics of the blade

Supposing now that the vessel is moving, let $\underline{v}_O = \hat{u}\underline{b}_1 + \hat{v}\underline{b}_2$ be the velocity of O (with respect to an Earth-fixed frame) expressed in the hull-fixed basis. Denoting by

$$\underline{v}'_P = -R\dot{\theta}\sin\theta\underline{b}_1 + R\dot{\theta}\cos\theta\underline{b}_2 \quad (2.11)$$

the velocity of the point P with respect to the body-fixed frame, the velocity of P with respect to the Earth-fixed frame is given by

$$\underline{v}_P = \underline{v}'_P + \underline{v}_O + \underline{\omega} \wedge (P - O) = [\hat{u} - R(\dot{\theta} + r)\sin\theta]\underline{b}_1 + [\hat{v} + R(\dot{\theta} + r)\cos\theta]\underline{b}_2 \quad (2.12)$$

where $\underline{\omega} = r\hat{b}_3$ is the angular velocity of the vessel. The velocity of the incoming flow experienced at P by a blade-fixed observer is then $-\underline{v}_P$; its unit vector $\underline{\hat{t}}$ is expressed as

$$\underline{\hat{t}} = -\frac{\underline{v}_P}{|\underline{v}_P|} = -\frac{[\hat{u} - R(\dot{\theta} + r)\sin\theta]\underline{b}_1 + [\hat{v} + R(\dot{\theta} + r)\cos\theta]\underline{b}_2}{([\hat{u} - R(\dot{\theta} + r)\sin\theta]^2 + [\hat{v} + R(\dot{\theta} + r)\cos\theta]^2)^{\frac{1}{2}}} \quad (2.13)$$

Making use of the unit vector $\hat{\underline{t}}$ it is possible to characterize the attack angle of the incident flow as

$$\hat{\alpha} = \pi - \arccos \left[\frac{(P-C)^\perp}{|(P-C)^\perp|} \cdot \hat{\underline{t}} \right] \quad (2.14)$$

according to the Figure 2.6:

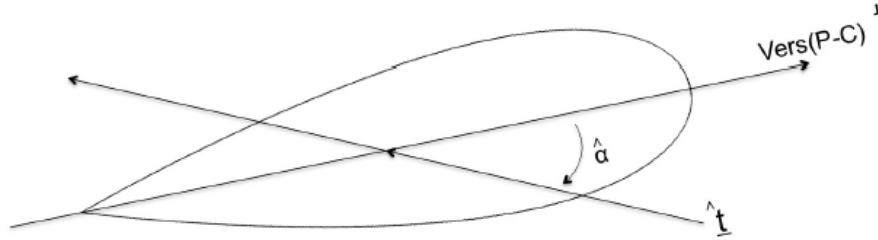


Figure 2.6: Attack angle of the incident flow.

2.3.1 Hydrodynamic forces

Making use of some simplifying assumptions, a suitable model for evaluating the hydrodynamic forces generated by each blade is presented. It is supposed that the velocity of the incident flow is the same on the entire surface of the blade and coincides with $-\underline{v}_P$. Under such a condition, the lift and drag produced by each blade can be expressed as

$$\underline{L} = c_L \frac{1}{2} \rho_w A |\underline{v}_P|^2 \hat{\underline{n}} \quad (2.15)$$

$$\underline{D} = c_D \frac{1}{2} \rho_w A |\underline{v}_P|^2 \hat{\underline{t}} \quad (2.16)$$

where:

c_L = lift coefficient;

c_D = drag coefficient;

ρ_w = sea water density;

A = blade lateral area;

$|\underline{v}_P|$ = incoming flow speed;

$\hat{\underline{t}}$ = unit vector of the lift force (unit vector of the incoming flow at P);

$\hat{\underline{n}}$ = unit vector of the drag force (perpendicular to $\hat{\underline{t}}$).

The unit vector $\hat{\underline{n}}$ can be determined by the following procedure, in which two main scenarios are distinguished:

- the attack angle $\hat{\alpha}$ belongs to the interval $]0, \frac{\pi}{2}[$, namely the incoming flow hits the blade from the front. In such a circumstance, the unit vector $\hat{\underline{n}}$ is determined according to the requirements:

$$\underline{\hat{n}} = \begin{cases} \underline{b}_3 \wedge \underline{\hat{t}} & \text{when } \underline{\hat{t}} \wedge \frac{(P-C)^\perp}{|(P-C)^\perp|} \cdot \underline{b}_3 > 0 \\ -\underline{b}_3 \wedge \underline{\hat{t}} & \text{when } \underline{\hat{t}} \wedge \frac{(P-C)^\perp}{|(P-C)^\perp|} \cdot \underline{b}_3 < 0 \end{cases} \quad (2.17)$$

- $\hat{\alpha} \in \left] \frac{\pi}{2}, \pi \right[$, the incoming flow hits the blade from the back. In this case, $\underline{\hat{n}}$ is singled out by the requests:

$$\underline{\hat{n}} = \begin{cases} -\underline{b}_3 \wedge \underline{\hat{t}} & \text{when } \underline{\hat{t}} \wedge \frac{(P-C)^\perp}{|(P-C)^\perp|} \cdot \underline{b}_3 > 0 \\ \underline{b}_3 \wedge \underline{\hat{t}} & \text{when } \underline{\hat{t}} \wedge \frac{(P-C)^\perp}{|(P-C)^\perp|} \cdot \underline{b}_3 < 0 \end{cases} \quad (2.18)$$

As remaining particular cases, if $\hat{\alpha} = 0$ or $\hat{\alpha} = \pi$ there is no lift while if $\hat{\alpha} = \frac{\pi}{2}$ then $\underline{\hat{n}} = \underline{\hat{t}}$. The above described procedure allows to determine the lift and drag provided by each single blade. The resultant hydrodynamic force generated by the epicycloidal propeller can be computed as the sum of all contributions given by each blade.

2.3.2 Torque acting on the rotor

In order to calculate the torque acting on the rotor, the Newton-Euler moments equations for each single blade and for the rotor are taken into account separately. Developed in the hull-fixed reference frame and with respect to the point O (centre of the rotor), the Newton-Euler moments equation for each blade can be expressed as

$$\underline{M}_O^H + \underline{M}_O^G + \underline{M}_O^R + \underline{M}_O^I = I_G(\underline{\dot{\omega}}) + \underline{\omega} \wedge I_G(\underline{\omega}) + m(G_B - O) \wedge \underline{a}_G \quad (2.19)$$

where $\underline{M}_O^H$, $\underline{M}_O^G$, $\underline{M}_O^R$, and $\underline{M}_O^I$ are the moments acting on the blade, respectively due to hydrodynamic, weight, reactive and inertial forces; I_G is the inertia tensor w.r.t. the gravity centre G_B of the blade (which is assumed to coincide with the pivot point P); $\underline{\omega} = (\dot{\theta} - \dot{\alpha})\underline{b}_3$ is the blade angular velocity w.r.t. the hull-fixed frame; $\underline{a}_G$ is the acceleration of G_B w.r.t. the hull-fixed frame; and m is the blade mass.

The moment of the hydrodynamic force $\underline{M}_O^H$ is given by:

$$\underline{M}_O^H = (P - O) \wedge (\underline{L} + \underline{D}) \quad (2.20)$$

where the hydrodynamic force is described in terms of lift and drag. Expressing all vectors in the basis $\{\underline{b}_i\}$ as $(\underline{L} + \underline{D}) = f_1\underline{b}_1 + f_2\underline{b}_2$ and $(P - O) = R\underline{b}_1 \cos\theta + R\underline{b}_2 \sin\theta$, one has:

$$\underline{M}_O^H = (Rf_2 \cos\theta - Rf_1 \sin\theta) \underline{b}_3 \quad (2.21)$$

The weight force moment is given by:

$$\underline{M}_O^G = (P - O) \wedge m\underline{g} = mg(R \sin\theta \underline{b}_1 - R \cos\theta \underline{b}_2) \quad (2.22)$$

where $\underline{g} = g\underline{b}_3$ is the gravity acceleration.

In order to evaluate the inertial force moment $\underline{M}_O^I$, it is necessary to assess the dragging and Coriolis forces acting on the blade and their associated moments $\underline{M}_O^S$ and $\underline{M}_O^C$. After this, the total inertial forces moment can be expressed as the sum:

$$\underline{M}_O^I = \underline{M}_O^C + \underline{M}_O^S \quad (2.23)$$

The moments $\underline{M}_O^S$ and $\underline{M}_O^C$ are calculated by means of the well-known formulae:

$$\underline{M}_O^C = \underline{M}_P^C + \underline{F}^C \wedge (O - P) \quad (2.24)$$

$$\underline{M}_O^S = \underline{M}_P^S + \underline{F}^S \wedge (O - P) \quad (2.25)$$

The Coriolis force acting on the single blade is:

$$\underline{F}^C = -2 \int_B k \underline{\omega}_\psi \wedge \underline{v}_Q d\tau \quad (2.26)$$

where $\underline{\omega}_\psi = \dot{\psi} \underline{n}_3 = r \underline{b}_3$ is the angular velocity of the vessel, $\underline{v}_Q = \underline{v}_P + \underline{\omega} \wedge (Q - P)$ (with $\underline{\omega} = (\dot{\theta} - \dot{\alpha}) \underline{b}_3$) is the velocity of a generic point Q of the blade w.r.t. the hull-fixed frame and k is the mass density of the blade. After implementing calculations, one gets the expression:

$$\begin{aligned} \underline{F}^C &= -2m \underline{\omega}_\psi \wedge \underline{v}_P - 2m \underline{\omega}_\psi \wedge [\underline{\omega} \wedge (G_B - P)] = -2m \underline{\omega}_\psi \wedge \underline{v}_{G_B} = \\ &= -2m (\dot{\psi} \underline{v}_{G_B1} \underline{b}_2 - \dot{\psi} \underline{v}_{G_B2} \underline{b}_1) \end{aligned} \quad (2.27)$$

$\underline{v}_{G_B} = \underline{v}_{G_B1} \underline{b}_1 + \underline{v}_{G_B2} \underline{b}_2$ denotes the velocity of the gravity centre of the blade w.r.t. the hull-fixed frame.

By definition, the moment $\underline{M}_P^C$ with respect to the pivot point P is given by:

$$\begin{aligned} \underline{M}_P^C &= -2 \int_B k (Q - P) \wedge [\underline{\omega}_\psi \wedge [\underline{v}_P + \underline{\omega} \wedge (Q - P)]] d\tau = \\ &= -2 \int_B k (Q - P) \wedge [\underline{\omega}_\psi \wedge \underline{v}_P] d\tau - 2 \int_B k (Q - P) \wedge [\underline{\omega}_\psi \wedge [\underline{\omega} \wedge (Q - P)]] d\tau \end{aligned} \quad (2.28)$$

Making use of the results shown in (Martelli et al. 2014), the moment $\underline{M}_P^C$ can be rewritten in the form:

$$\underline{M}_P^C = -2m (G_B - P) \wedge [\underline{\omega}_\psi \wedge \underline{v}_P] - \underline{\omega} \wedge I_B(\underline{\omega}_\psi) - \underline{\omega}_\psi \wedge I_B(\underline{\omega}) + I_B(\underline{\omega} \wedge \underline{\omega}_\psi) = 0 \quad (2.29)$$

Expression (2.29) holds in general. In our case, in view of the fact that the considered mathematical model is two-dimensional as well as of the assumption $G_B \equiv P$, it is easily seen that all terms appearing in Eq. (2.29) vanish, so having $\underline{M}_P^C = 0$

For what concerns the drag force, by definition it reads as:

$$\underline{F}^S = - \int_B s [\underline{a}_O + \underline{\omega}_\psi \wedge [\underline{\omega}_\psi \wedge (Q - O)] + \dot{\underline{\omega}}_\psi \wedge (Q - O)] d\tau = \quad (2.30)$$

$$\begin{aligned}
&= -m\underline{a}_O - m\underline{\omega}_\psi \wedge [\underline{\omega}_\psi \wedge (G_B - O)] - m\dot{\underline{\omega}}_\psi \wedge (G_B - O) = \\
&= -m(a_{Ox}\underline{b}_1 + a_{Oy}\underline{b}_2) + m\underline{\omega}_\psi \wedge (R\dot{\psi}\sin\theta\underline{b}_1 - R\dot{\psi}\cos\theta\underline{b}_2) + m(R\ddot{\psi}\sin\theta\underline{b}_1 - \\
&R\ddot{\psi}\cos\theta\underline{b}_2) = \\
&= -m(a_{Ox}\underline{b}_1 + a_{Oy}\underline{b}_2) + m\dot{\psi}^2(R\cos\theta\underline{b}_1 + R\sin\theta\underline{b}_2) + mR\ddot{\psi}(\sin\theta\underline{b}_1 - \cos\theta\underline{b}_2)
\end{aligned}$$

The moment $\underline{M}_P^S$ is given by:

$$\begin{aligned}
\underline{M}_P^S &= -\int_B k(Q - P) \wedge [\underline{a}_O + \underline{\omega}_\psi \wedge [\underline{\omega}_\psi \wedge (Q - O)] + \dot{\underline{\omega}}_\psi \wedge (Q - O)] d\tau = \quad (2.31) \\
&= -m(G_B - P) \wedge \underline{a}_O - \int_B k(Q - P) \wedge \left[\underline{\omega}_\psi \wedge [\underline{\omega}_\psi \wedge [(Q - P) + (P - O)]] \right] d\tau - \\
&\int_B k(Q - P) \wedge [\dot{\underline{\omega}}_\psi \wedge [(Q - P) + (P - O)]] d\tau = \\
&= -m(G_B - P) \wedge \underline{a}_O - \underline{\omega}_\psi \wedge I_B(\underline{\omega}_\psi) - m(G_B - P) \wedge \underline{\omega}_\psi \wedge [\underline{\omega}_\psi \wedge (P - O)] - \\
&I_B(\dot{\underline{\omega}}_\psi) - m(G_B - P) \wedge [\dot{\underline{\omega}}_\psi \wedge (P - O)] = \\
&= -m(G_B - P) \wedge [\underline{a}_O + \underline{\omega}_\psi \wedge [\underline{\omega}_\psi \wedge (P - O)] + \dot{\underline{\omega}}_\psi \wedge (P - O)] - \underline{\omega}_\psi \wedge \\
&I_B(\underline{\omega}_\psi) - I_B(\dot{\underline{\omega}}_\psi)
\end{aligned}$$

Again, since the model is two-dimensional and $G_B \equiv P$, only the last term does not vanish in general, so yielding.

$$\underline{M}_P^S = -I_B(\dot{\underline{\omega}}_\psi) = -I_{33}\dot{\psi}\underline{b}_3 \quad (2.32)$$

In order to implement Eqs. (2.24) and (2.25), we need to calculate the terms $\underline{F}^S \wedge (O - P)$ and $\underline{F}^C \wedge (O - P)$:

$$\underline{F}^S \wedge (O - P) = mR(a_{O1}\sin\theta - a_{O2}\cos\theta)\underline{b}_3 - mR^2\ddot{\psi}\underline{b}_3 \quad (2.33)$$

$$\underline{F}^C \wedge (O - P) = -2mR(\dot{\psi}v_{GB2}\sin\theta + \dot{\psi}v_{GB1}\cos\theta)\underline{b}_3 \quad (2.34)$$

Now, inserting all the obtained results into Eq. (2.23), we end up with the final expression of the moment w.r.t. O of the inertial forces

$$\begin{aligned}
\underline{M}_O^I &= -2mR(\dot{\psi}v_{GB2}\sin\theta + \dot{\psi}v_{GB1}\cos\theta)\underline{b}_3 - I_{33}\dot{\psi}\underline{b}_3 + mR(a_{Ox}\sin\theta - a_{Oy}\cos\theta)\underline{b}_3 \\
&- mR^2\ddot{\psi}\underline{b}_3
\end{aligned} \quad (2.35)$$

For our two-dimensional model, the terms on the right side of Eq. (2.19) are given by:

$$I_G(\underline{\dot{\omega}}) = I_{33}(\ddot{\theta} - \ddot{\alpha})\underline{b}_3 \quad (2.36)$$

$$\underline{\omega} \wedge I_G(\underline{\omega}) = 0 \quad (2.37)$$

where I_G is the inertia moment of the blade with respect to the vertical axis passing through G, and

$$m(G_B - O) \wedge \underline{a}_G = m(R\underline{a}_{G2}\cos\theta - R\underline{a}_{G1}\sin\theta)\underline{b}_3 \quad (2.38)$$

Inserting all the above calculated contributions into Eq. (2.19), we obtain the explicit expression of the reactive moment:

$$\underline{M}_O^R = -\underline{M}_O^H - \underline{M}_O^G - \underline{M}_O^I + I_G(\dot{\underline{\omega}}) + \underline{\omega} \wedge I_G(\underline{\omega}) + m(G_B - O) \wedge \underline{a}_G \quad (2.39)$$

Inserting the reactive moments (2.39) acting on each single blade into the moments equation for the rotor, the engine torque can be calculated as:

$$M_O^E = \sum_{i=1}^n (\underline{M}_O^R)_i \cdot \underline{b}_3 - \underline{M}_O^I \cdot \underline{b}_3 + I_O(\dot{\underline{\omega}}_r) \cdot \underline{b}_3 \quad (2.40)$$

where $\underline{M}_O^I$ is the inertial forces torque acting on the rotor, I_O is the inertia tensor of the rotor and $\underline{\omega}_r = \dot{\theta}\underline{b}_3$ is the angular velocity of the rotor, n is the number of blades.

$\underline{M}_O^I$ can be calculated as already made for the blade, namely as the sum $\underline{M}_O^I = \underline{M}_O^C + \underline{M}_O^S$. In this case, there are more simplifications since the point O is fixed to the rotor itself.

$$\underline{M}_O^C = \underline{M}_G^C + \underline{F}_{rot}^C \wedge (O - G) \quad (2.41)$$

$$\underline{M}_O^S = \underline{M}_G^S + \underline{F}_{rot}^S \wedge (O - G) \quad (2.42)$$

The Coriolis force is:

$$\underline{F}_{rot}^C = -2 \int_B s \underline{\omega}_\psi \wedge \underline{v}_Q d\tau \quad (2.43)$$

where $\underline{v}_Q = \underline{\omega}_r \wedge (Q - O)$ (with $\underline{\omega}_r = \dot{\theta}\underline{b}_3$) is the velocity of a generic point Q of the rotor w.r.t. the hull-fixed frame. After implementing calculations, we get:

$$\underline{F}_{rot}^C = -2 \int_B s \underline{\omega}_\psi \wedge [\underline{\omega}_r \wedge (Q - O)] d\tau = -2 \underline{\omega}_\psi \wedge [\underline{\omega}_r \wedge (G - O)] = -2M \underline{\omega}_\psi \wedge \underline{v}_G \quad (2.44)$$

where G is the gravity centre of the rotor.

The drag force is defined as:

$$\underline{F}^S = -M \underline{a}_O - M \underline{\omega}_\psi \wedge (G - O) - M \dot{\underline{\omega}}_\psi \wedge (G - O) \quad (2.45)$$

Since $O \equiv G$, $\underline{F}_{rot}^C = 0$ and $\underline{F}^S = -M \underline{a}_O$.

By definition, the moment $\underline{M}_G^C$ with respect to the gravity centre G of the rotor is given by:

$$\underline{M}_G^C = -\underline{\omega}_r \wedge I_O^{rot}(\underline{\omega}_\psi) - \underline{\omega}_\psi \wedge I_O^{rot}(\underline{\omega}_r) + I_O^{rot}(\underline{\omega}_r \wedge \underline{\omega}_\psi) \quad (2.46)$$

with I_O^{rot} the inertia tensor of the rotor w.r.t. its centre O (which is assumed to coincide with the gravity centre G); Under the hypothesis of two-dimensional mathematical model, $\underline{M}_G^C = 0$. The moment $\underline{M}_G^S$ is given by:

$$\underline{M}_G^S = -M(G - O) \wedge \underline{a}_O - \underline{\omega}_\psi \wedge I_O^{rot}(\underline{\omega}_\psi) - I_O^{rot}(\underline{\dot{\omega}}_\psi) \quad (2.47)$$

Again, since the model is two-dimensional and $G \equiv O$, only the last term does not vanish in general, so yielding.

$$\underline{M}_G^S = -I_O^{rot}(\underline{\dot{\omega}}_\psi) \quad (2.48)$$

Inserting now all the obtained results into Eq. (2.40), the final result is:

$$M_O^E = \sum_{i=1}^n (\underline{M}_O^R)_i \cdot \underline{b}_3 + I_O(\ddot{\psi} \underline{b}_3) \cdot \underline{b}_3 + I_O(\ddot{\theta} \underline{b}_3) \cdot \underline{b}_3 \quad (2.49)$$

3 NUMERICAL MODELLING AND VALIDATION

The kinematical model illustrated in chapter 2 has been used to develop a Matlab-Simulink simulator for cycloidal propellers. In this section, the main features and the validation of such simulator are presented.

3.1 *Simplifying assumptions*

In order to simplify the simulation platform, some hypotheses have been assumed:

- the propeller is considered in free-running conditions (no hull interference), as in open water, invested from an incoming flow;
- the problem is supposed to be stationary;
- the model is 2D;
- the contributions of each single blade, in terms of generated forces and moments, are separately calculated and then summed. The interference among the blades is taken into account by means of suitable correction factors.

3.2 *Input data*

The Simulink model needs input data, given from Matlab file. These data are:

- the geometry of the propeller (length, chord and orbit diameter of the blade – see Table 1);
- the sea water characteristics (viscosity and density);
- the lift C_L and drag C_D coefficients of the blade (obtained by a previous CFD calculation) (Canaccini et al. 2013);
- the rotor speed and maximum pitch available;
- the steering pitch angle (0° in forward direction, 180° in astern condition) and the driving pitch (expressed as a percentage of the radius of the orbit of the blades).

In the present case study, we have:

N. of blades	5
Diameter (m)	3.2
Blade length (m)	2.65
Blade Chord (m)	0.7744
Max tip thickness	0.242

Table 3.1: Propeller geometric parameters.

3.3 *Simulation*

The simulator allows to generate the curves of the coefficients K_S and K_D , respectively defined by

$$\text{Thrust Coefficient} \quad K_S = \frac{T}{\frac{1}{2}\rho_w D L u^2} \quad (3.1.a)$$

$$\text{Torque Coefficient} \quad K_D = \frac{4M}{\rho_w D^2 L u^2} \quad (3.2.b)$$

and depending on the advance coefficient

$$\lambda = \frac{V_A}{\pi n D} \quad (3.2.c)$$

where T is the total thrust, M is the total torque acting on the rotor, L is the blade height, and $u = \pi n D$ is the revolution speed of the blades.

The whole model consists of a set of identical subsystems, each of them representing the behaviour of a single blade (sketch represented in Figure 3.1). Making use of Eqs. (2.15), (2.16) and (2.49), the components of total thrust and torque are calculated in the basis $\{b_i\}$.

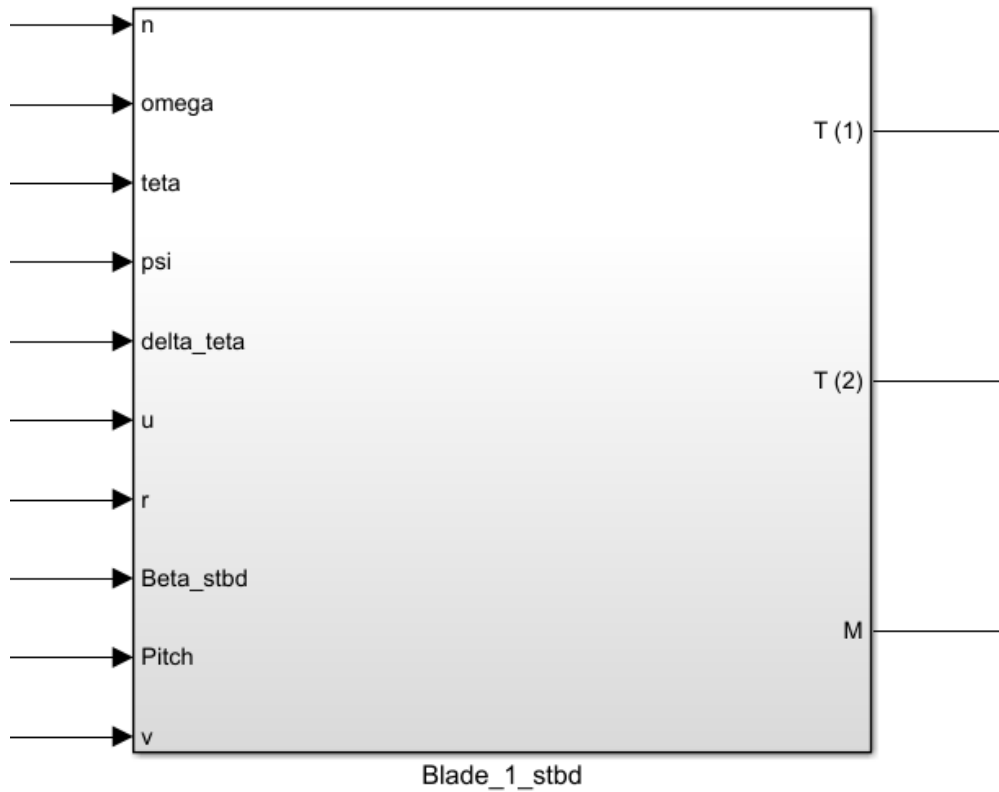


Figure 3.1: Blade subsystem sketch

According to Eqs. (3.1.a) and Eqs. (3.1.b), it is then possible to evaluate the coefficients K_S and K_D .

The simulation of coefficient K_S and K_D has been performed in the pitch range from 40% to 80%, with steps of 10%. In particular, Figure 3.2 shows the comparison between literature data (pertinent to a real existing cycloidal propeller, characterized by the same geometry of the simulated one) and the coefficients K_S and K_D , obtained by simulation without the intervention of any correction factor, for a pitch of 80%.

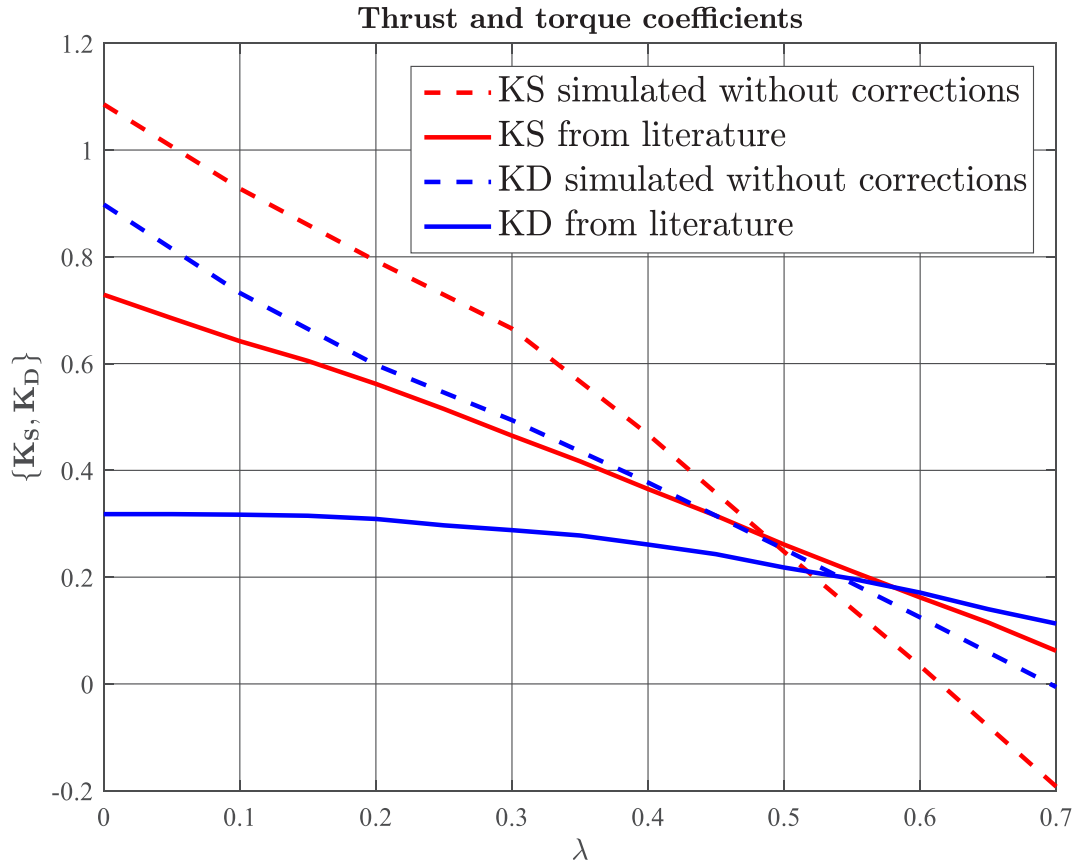


Figure 3.2: Thrust and torque coefficients of the cycloidal thruster

3.4 *Model validation*

The differences between real and simulated data shown in Figure 3.2 are supposed to be due to the stated simplifying assumptions about the interactions among the blades. In order to minimize these differences, two correction factors (properly described in paragraphs 3.4.1 and 3.4.2) have been introduced, taking two main phenomena into account: the shielding of the blades that are in the half circumference not directly exposed to the incoming flow and the interference of a blade with the other. The calibration of such correction factors has been carried out by comparing simulation

results with the performance data of the existing real propeller. The comparison is shown in Figure 3.3.

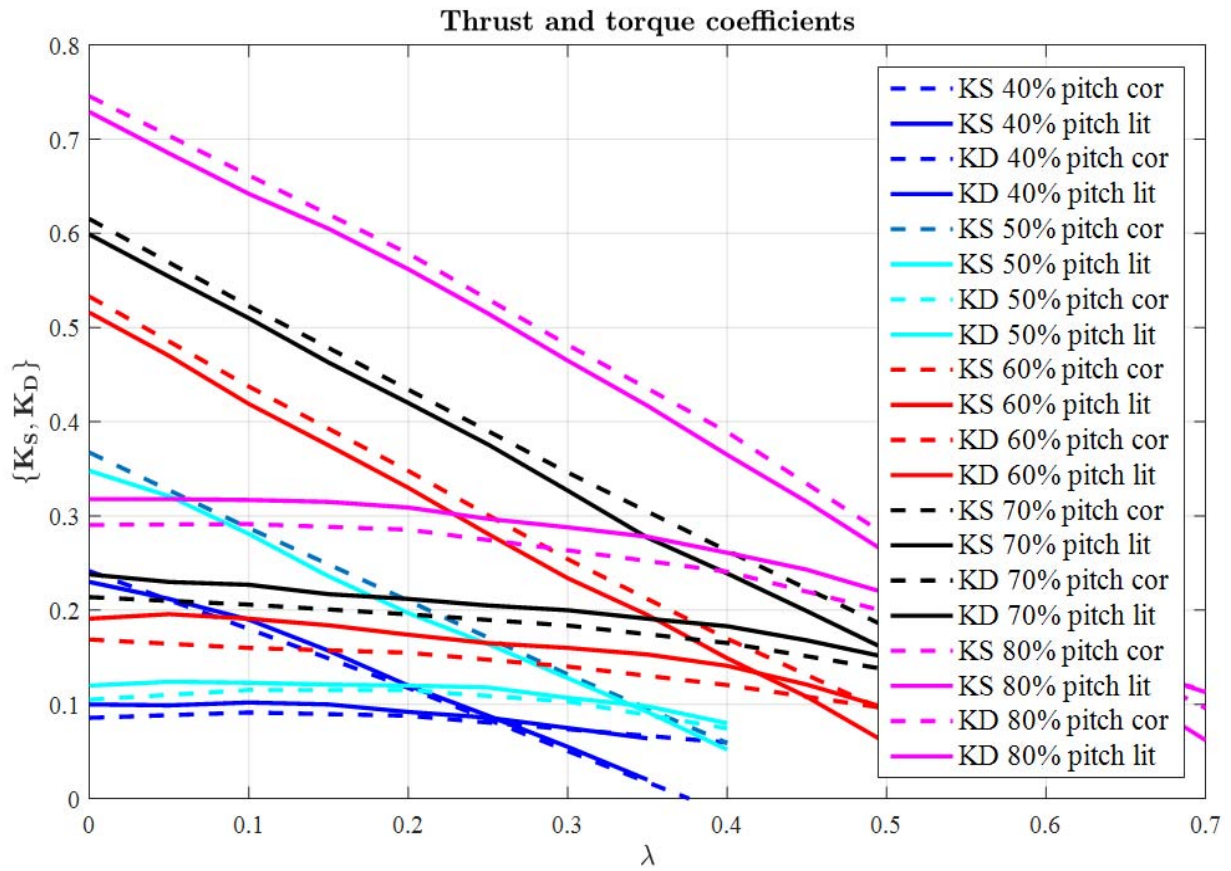


Figure 3.3: Thrust and torque coefficients of the cycloidal thruster.

Once the curves of K_S and K_D have been obtained, the total thrust for advance coefficient values of 0 and 0.4 and different thrust directions has been analysed. Figures 3.4 and 3.5 show the obtained results: the orange line represents simulated results with interference and shielding correction, while the blue one shows the performance data of the existing propeller (brochure of Voith Turbo Schneider Propulsion “The Voith Schneider Propeller Current Applications and New Developments”).

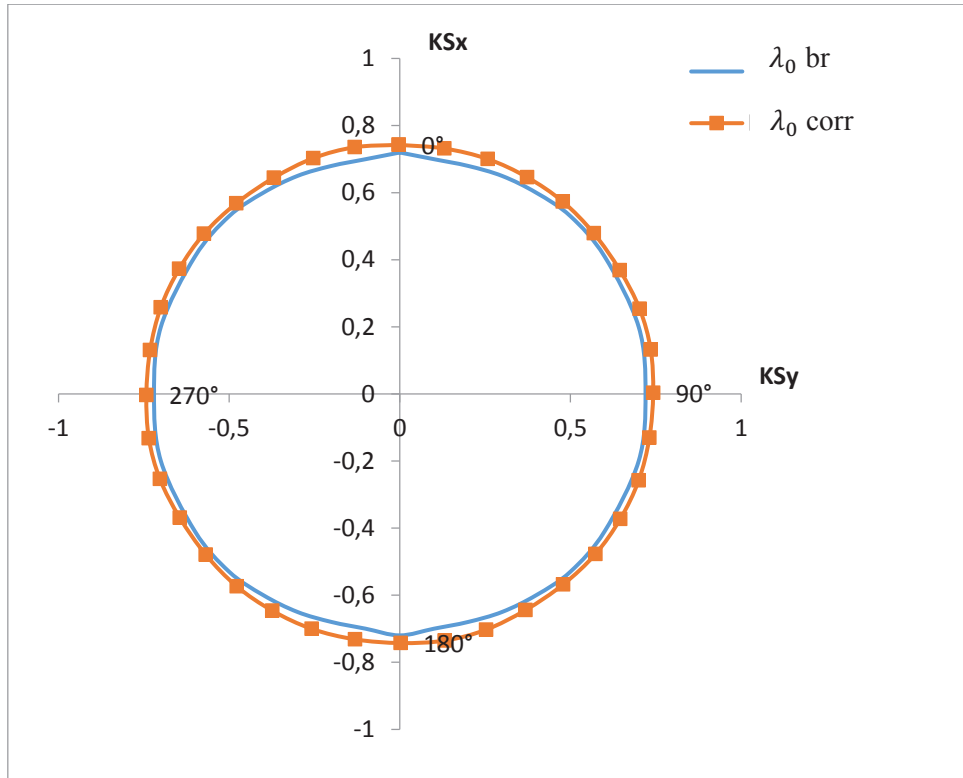


Figure 3.4: Longitudinal and transversal components of K_S for $\lambda = 0$.

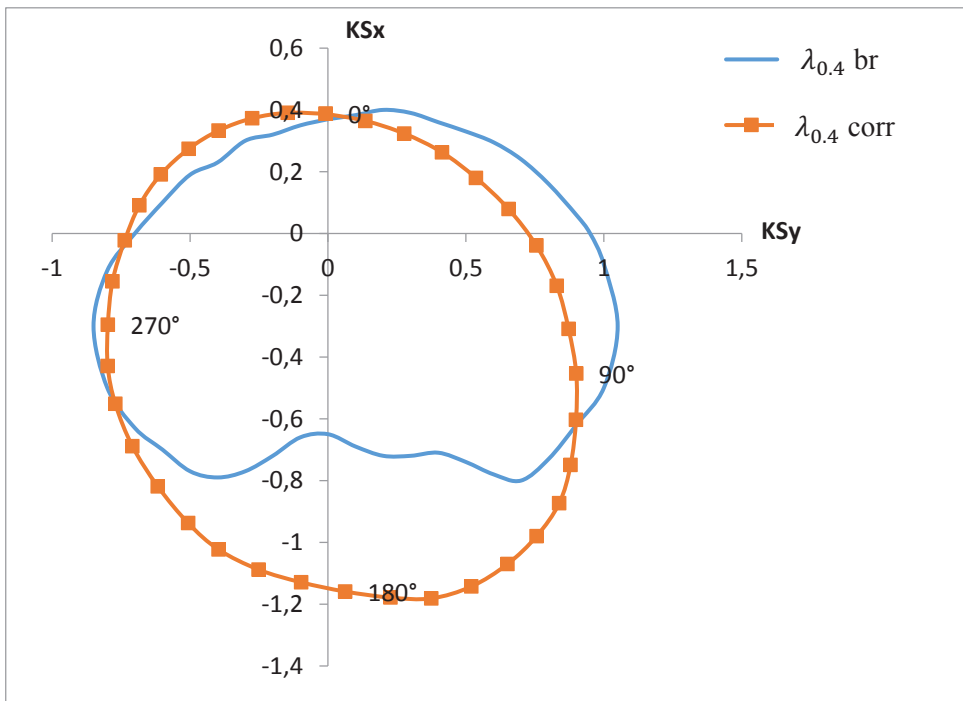


Figure 3.5: Longitudinal and transversal components of K_S for $\lambda = 0.4$.

The numerical modelling does account for a slight reduction of thrust magnitude for the reverse thrust when the advance coefficient λ is more than 0. In order to consider this further phenomenon, another correction factor has been introduced, according to the real propeller performance. The obtained results are illustrated in Figure 3.6.

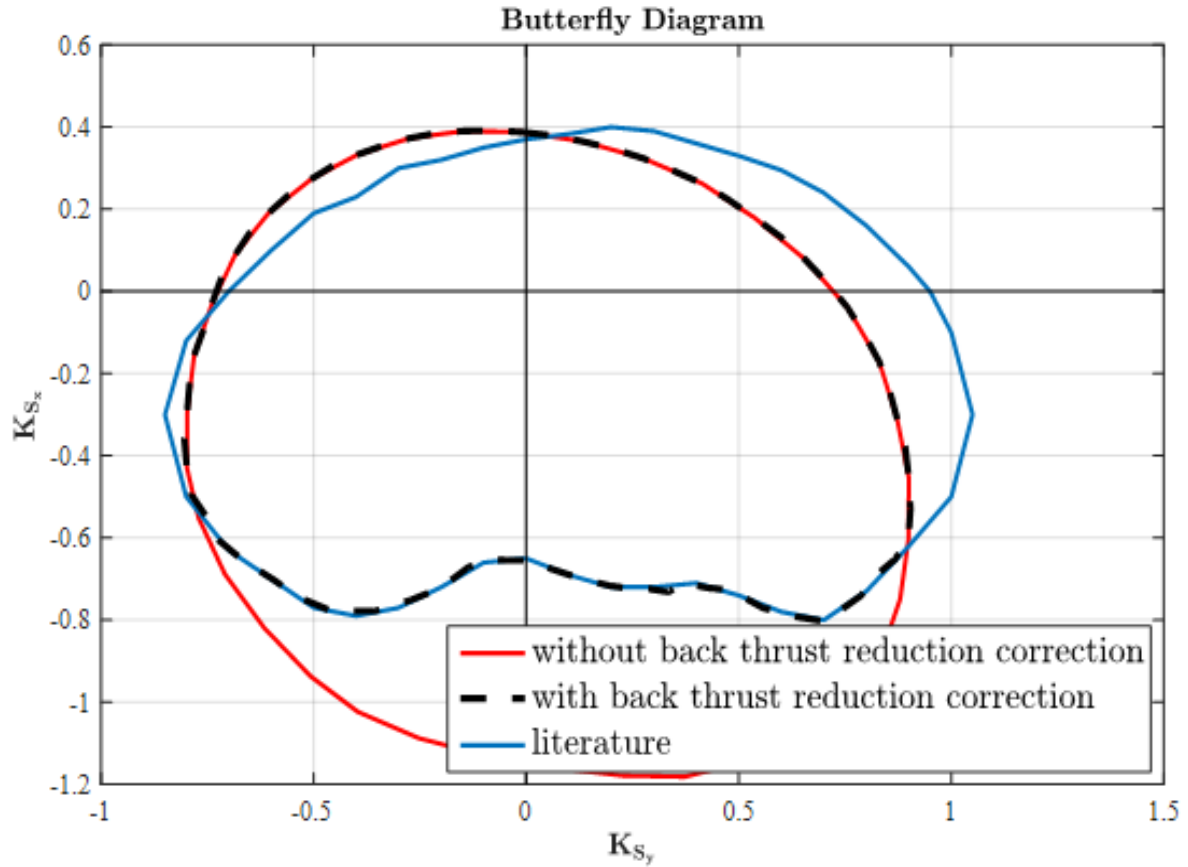


Figure 3.6: Components of thrust coefficients of the cycloidal thruster depending on the direction of the thrust.

The correction factors, empirically estimated for the given propeller, are represented in terms of percentage reductions. In the case of cycloidal propellers with a different number of blades, the same correction values can be considered for an overall estimation of the propeller performance, achieving good results; obviously, for a more reliable simulation, the correction factors should be calibrated again for each specific propeller.

The three considered correction factors are summarized below.

3.4.1 *Shielding correction*

Shielding correction refers to the shielding of the blades that are in the half circumference not directly exposed to the incoming water flow: in the model, this correction factor, depending on the driving pitch values, reduces the velocity of the incoming flow.

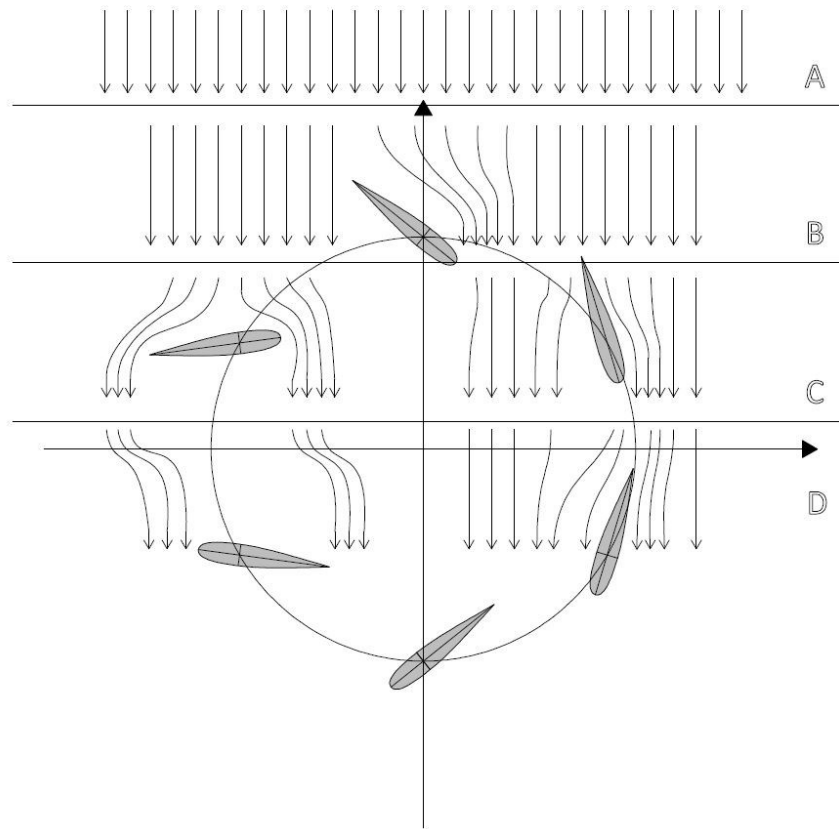


Figure 3.7: Shielding phenomena (Battistoni 2014)

Figure 3.7 gives a qualitative idea how the flow is deviated by the blades.

The matrix of corrective coefficients has been introduced to reduce the magnitude of velocity of the incoming water flow in the half part (between 90° and 270°) of the rotor not directly invested by the flow. The corrective factors depend only on the pitch and not on the advance coefficient λ . Table 2 shows the shielding correction coefficients corresponding to different pitches.

Pitch [%]	SHIELDING CORRECTION COEFFICIENTS
20	0.6
30	0.57
40	0.54
50	0.51
60	0.39
70	0.39
80	0.40
90	0.40
100	0.38

Table 3.2: Shielding correction coefficients

3.4.2 *Interference correction*

Each blade influences the flow of the adjacent blade modifying the angle of attack of the incident flow itself.

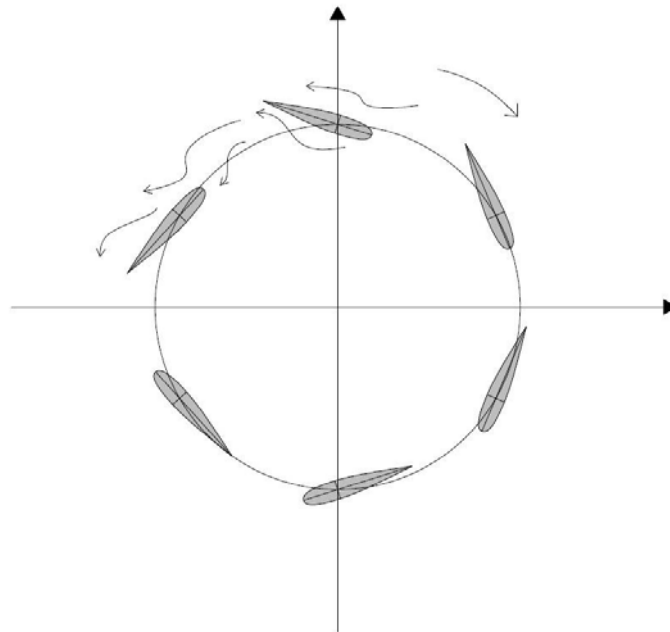


Figure 3.8: Interference phenomena (Battistoni 2014).

The interference among the blades is modelled by reducing the attack angle of the incoming flow; this correction depends on λ and pitch values, according to the following Table 3.

Pitch [%]	ADVANCE COEFFICIENT λ					
	<i>0</i>	<i>0.2</i>	<i>0.4</i>	<i>0.6</i>	<i>0.8</i>	<i>1</i>
<i>20</i>	0.4	0.4	0.4	0.4	0.4	0.4
<i>30</i>	0.31	0.33	0.35	0.35	0.33	0.31
<i>40</i>	0.31	0.33	0.35	0.35	0.33	0.31
<i>50</i>	0.35	0.39	0.4	0.4	0.39	0.35
<i>60</i>	0.37	0.39	0.47	0.47	0.39	0.37
<i>70</i>	0.38	0.44	0.47	0.47	0.44	0.38
<i>80</i>	0.3	0.45	0.52	0.52	0.45	0.3
<i>90</i>	0.78	0.83	0.85	0.85	0.83	0.78
<i>100</i>	0.75	0.84	0.88	0.88	0.84	0.75

Table 3.3: Interference correction coefficients

3.4.3 Reverse thrust correction

Reverse thrust correction represents the reduction of the reverse thrust.

The coefficient has been introduced to reduce magnitude of thrust when the propeller runs backward. The corrective factor depends only on the pitch angle when its value is between 90° and 270° .

Pitch Angle [°]	REVERSE THRUST CORRECTION COEFFICIENTS
<i>0</i>	1
<i>30</i>	1
<i>60</i>	1
<i>90</i>	1
<i>120</i>	0.86
<i>150</i>	0.59
<i>180</i>	0.53
<i>210</i>	0.62
<i>240</i>	0.94
<i>270</i>	1
<i>300</i>	1
<i>330</i>	1
<i>360</i>	1

Table 3.4: Back thrust correction coefficients

4 APPLICATIONS TO DYNAMIC POSITIONING

In order to automatically maintain position and heading of a vessel, DP systems employ in general azimuthal thrusters or rudders and propellers combined with bow and stern thrusters (Donnarumma et al. 2018, Donnarumma et al. 2015), sometimes waterjets (Altosole et al. 2012a). Marine cycloidal propellers, usually driven by diesel-electric propulsion to better handle the large changes in power demand (typical during DP operations), can represent a good alternative to traditional propellers since, as well as the azimuthal propellers, they can generate almost the same thrust in all directions. In the present study, the performance of an epicycloidal propeller is modelled within a DP propulsion simulator, already developed (Alessandri et al. 2014) for a surface vessel equipped with two conventional twin-screw propellers and a bow thruster. This kind of configuration is not very suitable for station-keeping and DP applications, nevertheless, a conventional propulsion configuration could be requested for specific operations characterized by limited DP capabilities. For instance, the mentioned simulator was developed for a patrol vessel designed with a twin propeller-rudder configuration and a single bow thruster, which were requested to provide a certain dynamic positioning performance with moderate weather conditions. The main purpose of the DP simulation model was to validate the Force and Thrust Allocation Logic (FAL, TAL), specifically designed for such propulsion configuration (Donnarumma et al. 2015).

In this work, the same vessel, but supposed equipped with a single bow thruster and two epicycloidal propellers at stern, is simulated to analyse the main differences in the control system behaviour during DP operations. The different location of the two types of propeller relative to the hull is illustrated in Figure 4.1. The epicycloidal propellers are installed more externally with respect to skew propellers for constructive reasons recommended by the manufacturers.

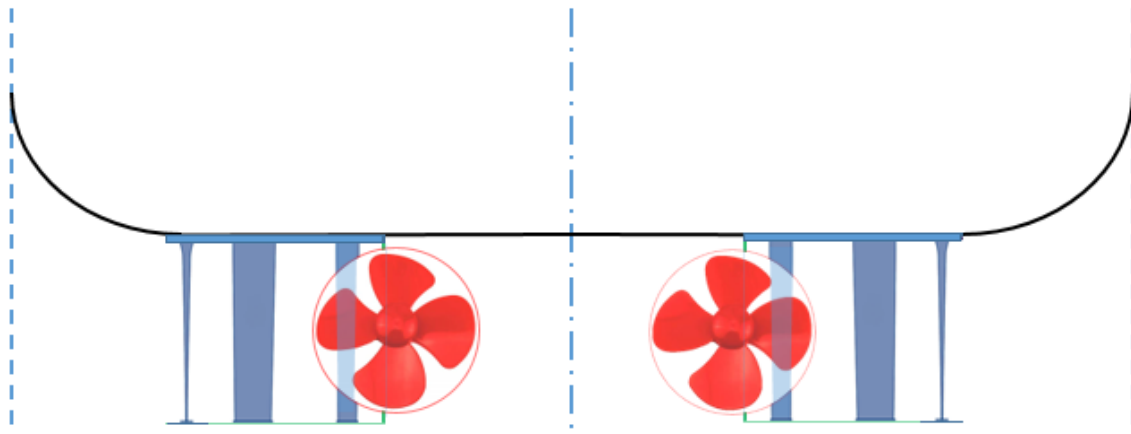


Figure 4.1: propellers transversal position.

The first step in the assessment of the performance of a given DP system is the evaluation of the static capabilities of the vessel. In order to accomplish this task, DP polar plots are a useful tool. In the following, the DP capability polar plots for the vessel considered in the case study are

presented, taking two different allocation logics into account. Finally, a comparison with the DP capability of the real ship (equipped with a standard propulsion configuration) is carried out.

4.1 *Environmental disturbances*

The analysis is performed in static conditions and involves the balance of the forces and moments generated by environmental disturbances. Such disturbances are described as the sum of forces and moments due to wind, current and waves respectively. Forces and moments are expressed by means of the well-known resistance form, depending on non-dimensional coefficients $C_X(\gamma)$, $C_Y(\gamma)$, and $C_N(\gamma)$, related respectively to the longitudinal force, the lateral force and the moment. γ denotes the relative angle between the disturbance incoming direction and the vessel heading. In order to consider the occurring worst condition, all environmental disturbances are supposed to be aligned in the same incoming direction.

The current and the wind speeds are assumed constant, as well as the and wave drift forces that are modelled as proportional to the square of the significant height H_s . Collecting all the (b-basis) components of the force and moment in a unique 3-dimensional array τ , we have:

$$\tau_E = \tau_{current}(\gamma_r) + \tau_{waves}(\gamma_r, H_s) + \tau_{wind}(\gamma_r, V_G) \quad (4.1)$$

$$\tau_{current} = \frac{1}{2} \rho v_c^2 \begin{bmatrix} A_f C_{X_c}(\gamma_{rc}) \\ A_l C_{Y_c}(\gamma_{rc}) \\ A_l L C_{N_c}(\gamma_{rc}) \end{bmatrix} \quad (4.2.a)$$

$$\tau_{waves} = H^2 \begin{bmatrix} C_{X_s}(\gamma_{rs}) \\ C_{Y_s}(\gamma_{rs}) \\ C_{N_s}(\gamma_{rs}) \end{bmatrix} \quad (4.2.b)$$

$$\tau_{wind} = (v_w^2) \begin{bmatrix} C_{X_w}(\gamma_{rw}) \\ C_{Y_w}(\gamma_{rw}) \\ C_{N_w}(\gamma_{rw}) \end{bmatrix} \quad (4.2.c)$$

where

v_c, v_w current and wind speed

A_f e A_l submerged frontal area and submerged longitudinal area

L ship length

$\gamma_{rc}, \gamma_{rs}, \gamma_{rw}$ relative angle between the disturbance incoming direction and the vessel heading

C_X, C_Y, C_N coefficients, related respectively to the longitudinal force, the lateral force and the moment

H_s significant height

4.2 Thrust allocation logic

In order to assess the DP capability of the propulsion configuration with cycloidal propellers, two allocation logics have been developed and compared with the one implemented on the real vessel, equipped with two traditional propellers, two rudders and a bow thruster. Details of the original allocation can be found in (Alessandri et al. 2014).

The first allocation logic is based on a constrained minimum problem. The idea is to minimize a cost function of seven variables

$$\underline{x} = [T_{pt}, T_{sb}, T_{bow}, X_{pt}, Y_{pt}, X_{sb}, Y_{sb}] \in \mathbb{R}^7 \quad (4.3)$$

subject to some suitable constraints. In particular, defining by T_{pt} and T_{sb} respectively the portside and starboard cycloidal propeller thrusts, T_{bow} the thrust of the bow thruster, (X_{pt}, Y_{pt}) and (X_{sb}, Y_{sb}) the components of the portside and starboard thrust forces, the constrained minimum problem is formulated as

$$\min_{\underline{x}} f(\underline{x}) \quad \text{with} \quad h_i(\underline{x}) = 0 \quad \text{and} \quad g_j(\underline{x}) > 0 \quad (4.4)$$

where

$$f(\underline{x}) = \left(\frac{T_{pt}}{T_{max}^{tot}}\right)^2 + \left(\frac{T_{sb}}{T_{max}^{tot}}\right)^2 + \left(\frac{T_{bow}}{T_{max}^{tot}}\right)^2 \quad (4.5)$$

is the cost function to be minimized, and

$$\begin{cases} h_1(\underline{x}) = X_{env} - X_{pt} - X_{sb} \\ h_2(\underline{x}) = Y_{env} - Y_{pt} - Y_{sb} - T_{bow} \\ h_3(\underline{x}) = N_{env} - x_{bow} T_{bow} - x_{pt} Y_{pt} + \\ \quad + y_{pt} X_{pt} - x_{sb} Y_{sb} + y_{sb} X_{sb} \\ h_4(\underline{x}) = T_{pt}^2 - X_{pt}^2 - Y_{pt}^2 \\ h_5(\underline{x}) = T_{sb}^2 - X_{sb}^2 - Y_{sb}^2 \end{cases} \quad (4.6)$$

are the constraints to be satisfied. In Eq. (4.6) $\{X_{env}, Y_{env}, N_{env}\}$ contains the components of the force and the moment due to environmental disturbances, (x_{pt}, y_{pt}) and (x_{sb}, y_{sb}) are the coordinates of the propellers thrust centres and x_{bow} is the longitudinal coordinate of the bow thruster. Moreover, we have

$$g_1(\underline{x}) = T_{pt} \quad , \quad g_2(\underline{x}) = T_{sb} \quad (4.7)$$

Eq. (4.4) and (4.5) require that the sum of the squared desired thrusts is minimum. Eq. (4.6) details the constraints: the first three ones represent the static equilibrium between the environmental disturbances and the delivered forces and moment; $h_4(\underline{x})$ and $h_5(\underline{x})$ correlate the portside and starboard thrust force with their longitudinal and lateral corresponding force components. Finally, Eqs. (4.4) and (4.7) ensure that the modulus of the two aft thrusts is positive.

The second allocation algorithm is based on the idea that one cycloidal propeller is designed to compensate the environmental resultant force, while the other one together with the bow thruster, is devoted to compensate the resultant moment. This allocation configuration is supposed to be more stable when environmental disturbances are relatively small and constant in time. In this case, the force and moment balance is uniquely determined in an algebraic way, whenever the thruster devoted to the force compensation is chosen. The choice of such a thruster relies on the requirement that the moment generated by the thruster itself has opposite sign with respect to the disturbances one. More explicitly, we have

$$\begin{aligned} T_i &= \sqrt{X_{env}^2 + Y_{env}^2} \quad , \quad \delta_i = \tan^{-1} \frac{Y_{env}}{X_{env}} \\ T_{bow} &= -T_j \\ T_j &= \frac{x_i T_i \sin \delta_i - y_i T_i \cos \delta_i + N_{env}}{x_j - x_{bow}} \end{aligned} \quad (4.8)$$

where $i = pt$ and $j = sb$ when $|N_{pt} - N_{env}| > |N_{sb} - N_{env}|$ and $i = sb$ and $j = pt$ otherwise; N_{pt} is the moment generated by the propeller if portside is the one compensating the force and viceversa for N_{sb} .

The criterion adopted to choose the thrust devoted to compensate the disturbances force is based on computing, for both the possible choices, the moment generated by the thruster and adding it to the environmental one. Such resulting moments are compared and the thruster generating the minimum moment is the one chosen for the environmental force compensation.

4.3 *Static Analysis - DPCP*

The station keeping capability of a dynamically positioned vessel is often illustrated by means of polar plots, which present the steady-state performance of the vessel under certain environmental conditions. A capability plot shows the maximum weather conditions in which the vessel is able to maintain its position and heading, obeying upon a certain thrust allocation logic. DP capability plots are drawn assuming that all the environmental disturbances come from the same direction. In this case study, the vessel is supposed to operate in Mediterranean Sea with a significant wave height H_s of 2.5 m and a constant current speed v_C of 1kn. Instead, the wind speed keeps increasing until the vessel is able to sustain the resulting total load, namely until the resultant environmental forces and moments are balanced by the maximum available thrust. Assuming that the aligned environmental disturbances rotate around the vessel (anticlockwise starting from zero which corresponds to stern), in Figure 4.2 the intersection of the curve with the radius of the circumference indicates the maximum wind speed at which the vessel can maintain its position and heading. For sake of reliability, a saturation was added when the maximum wind speed was higher than 60kn. Indeed, higher wind conditions involve sea disturbances that cannot be dealt with in a static analysis.

As it can be seen in Figure 4.2, maximum values of wind speed are reached for disturbances coming from bow directions, while we have lower values for beam and stern directions. That is due to the larger areas exposed to the environmental loads. Regarding the differences due to the two distinct propulsion configurations, the numerical results show that the conventional propulsion

layout is able to ensure only a limited DP capability (the yellow line), while great improvements could come from the adoption of cycloidal propellers as propulsion system. Concerning the different allocation logics adopted for the cycloidal propellers, it is evident that the optimization procedure of the thrust (the red line) gives the best results compared to the second allocation algorithm described by Eqs. 4.8 (the blue line), though this is mainly true for bow and quartering sea states, while for stern sea states the two allocation logics seem to give closer results.

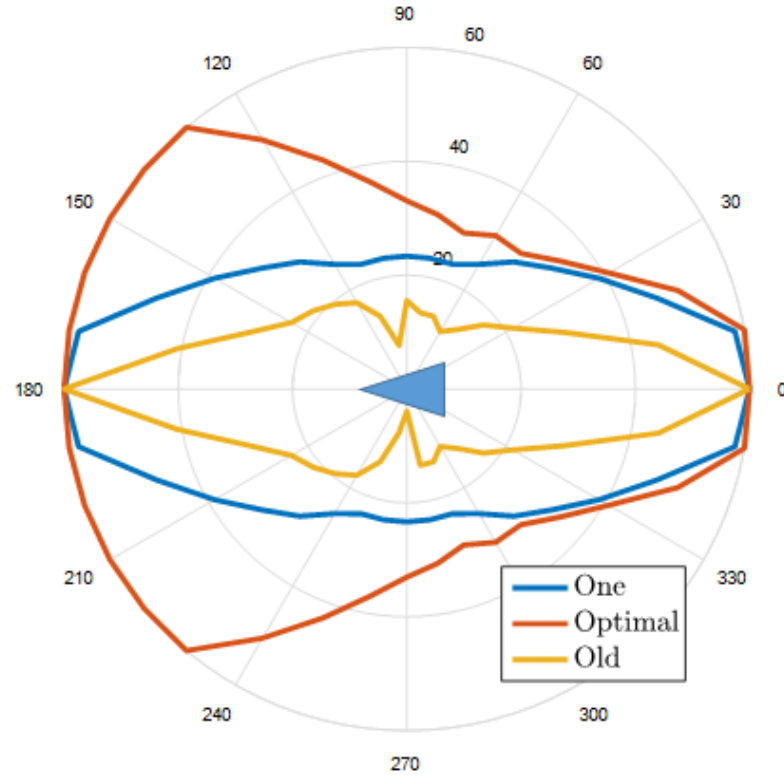


Figure 4.2: DP capability polar plots for different thrust allocation logics.

4.4 *Dynamic analysis*

A dynamic analysis of a propulsion system allows to predict the behaviour of the propulsion plant and the ship dynamics, reducing time and costs of the preliminary design phase and sea trial tests. The proposed model has been designed in order to assess both the regulation strategies and the station keeping capability of the vessel with more realistic results than those obtained by a purely static analysis. The model involves the dynamics of the main engines and their governors, as well as the mathematical description adopted for wind, wave, current forces, and for the controller. The behaviour during DP manoeuvres of the vessel supposed to be equipped with two cycloidal propellers and a bow thruster, is then analysed and compared with the results obtained with the traditional propulsion configuration.

4.4.1 DP Simulation Model

Figure 4.3 provides a sketch of the devised DP-logic circuit, specific for the target ship. The system simulates, through the equations of motion, the presence of a Positioning Reference System (PRS), composed by a DGPS and a Fibre Optic Gyro (FOG), which picks up the instantaneous position, heading and velocity of the vessel. Such measurements are compared with the corresponding desired quantities in order to compute the position and velocity errors.

The circuit extrapolates the low-frequency (LF) components of the errors and send them to the regulator, where the required forces and moments are firstly evaluated and subsequently allocated to the actuators. Environment action and delivered forces are then used within the equations of motion to obtain the new position and velocity.

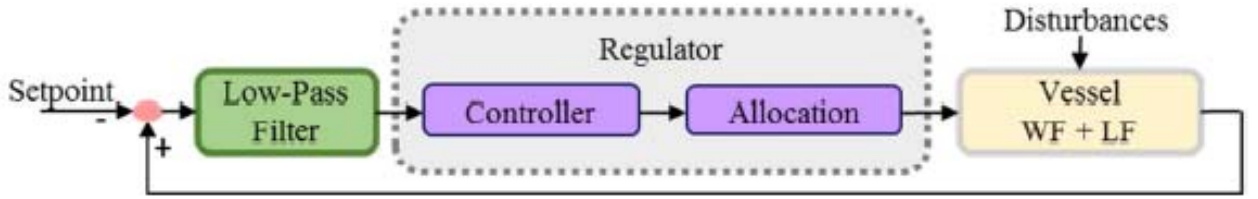


Figure 4.3: DP simulation model (Donnarumma et al. 2015).

The mathematical models adopted for ship motions and controller are briefly illustrated below.

4.4.2 Ship motions

Setting $\eta := [x, y, \psi]^T \in \mathbb{R}^3$, the array of the position (longitudinal and lateral position, and heading) of the vessel w.r.t. the Earth-fixed frame, and $v := [u, v, r]^T \in \mathbb{R}^3$, the array of the components of velocity (linear and angular) expressed in the body-fixed basis, the 3-DOF ship kinematics is described by the relations:

$$\dot{\eta} = R(\psi)v, \quad R(\psi) = \begin{pmatrix} \cos \psi & -\sin \psi & 0 \\ \sin \psi & \cos \psi & 0 \\ 0 & 0 & 1 \end{pmatrix} \quad (4.9)$$

The ship motion equations are given by:

$$M\dot{v} + C(v)v + D_0v + D(v)v = \tau_D + \tau_E \quad (4.10)$$

where the array $\tau := [X, Y, N]^T \in \mathbb{R}^3$ represents the components of the resultant force and moment (τ_D for delivered forces and moments, and τ_E for environmental ones), expressed in the body-fixed basis while M, C, D and D_0 are mass-inertia and added mass, Coriolis and linear and nonlinear damping matrices, respectively defined as follow:

$$M = \begin{bmatrix} m - X_{\dot{u}} & 0 & 0 \\ 0 & m - Y_{\dot{v}} & mx_G \\ 0 & mx_G & I_Z - N_{\dot{r}} \end{bmatrix} \quad (4.11.a)$$

$$C = \begin{bmatrix} 0 & 0 & -m(x_G r + u) \\ 0 & 0 & mu \\ m(x_G r + u) & -mu & 0 \end{bmatrix} \quad (4.11.b)$$

$$D_0 = \begin{bmatrix} 0 & 0 & 0 \\ 0 & Y_v & Y_r \\ 0 & N_v & N_r \end{bmatrix} \quad (4.11.c)$$

$$D(v) = \begin{bmatrix} R(u) & X_{vv}v & X_{vr}v + X_{rr}r \\ 0 & Y_{v|v}|v| & Y_{r|v}|v| + Y_{r|r}|r| + Y_{vrr}vr \\ 0 & N_{v|v}|v| + N_{r|v}|v| & N_{r|r}|r| + N_{vrr}vr \end{bmatrix} \quad (4.11.d)$$

where $X_{\ddot{u}} < 0$, $Y_{\dot{v}} < 0$ and $N_{\dot{r}} < 0$ are the zero frequency added mass coefficients, $R(u)$ is the drag coefficient, while the other coefficients in the definition of D_0 and $D(v)$ are described in accordance with (Ankudinov, Kaplan and Jacobsen, 1993)

4.4.3 Controller

The controller consists of a PD (proportional and derivative) controller, one for each degree of freedom, a wind forces reconstruction, a sea force estimation and a block for the allocation logic (Figure 4.4). The choice of using one controller for each DOF is motivated by (Sørensen 1996 & Fossen 2002) who demonstrated, comparing it with other more complex approaches, the global stability of the system, despite its simplicity. In conventional PID-controllers there usually is the integral term, here replaced by an estimation of the mean values of the environmental disturbances, based on a moving average of the required forces, calculated every given time interval Δt .

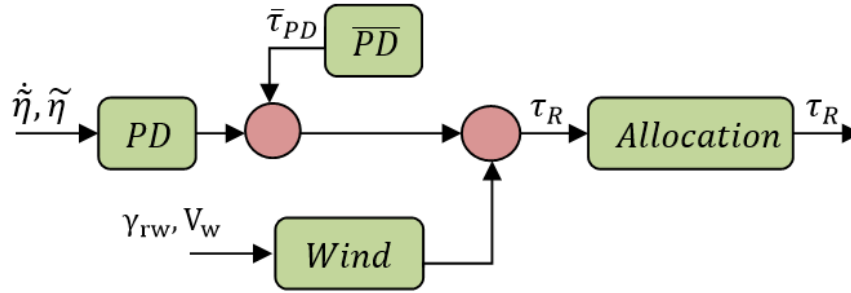


Figure 4.4. Controller Layout.

The controller law is given by:

$$\tau_R = K_P \tilde{\eta} + K_D \dot{\tilde{\eta}} + \bar{\tau}_{PD} - \tau_W \quad (4.12)$$

where the output τ_R represents the required force and moment; in Eq. (4.12), K_P and K_D are constant matrices, $\bar{\tau}_{PD}$ and τ_W are contributions which compensate the environmental disturbances and the quantities $\tilde{\eta} := \eta - \eta_d$ and $\dot{\tilde{\eta}} := \dot{\eta} - \dot{\eta}_d$ are controller input errors, $\eta_d = [x_d, y_d, \psi_d]^T \in$

$\mathbb{R}^3$ and $\dot{\eta}_d = [\dot{x}_d, \dot{y}_d, \dot{\psi}_d]^T \in \mathbb{R}^3$ denoting the desired position and velocity, respectively. Environmental disturbances have been divided into $\bar{\tau}_{PD} - \tau_W$, sea and wind contribution, that simulate the signal of the sensors installed on board.

4.4.4 Ship Simulation Model

Figure 4.5 provides a sketch of the ship model.

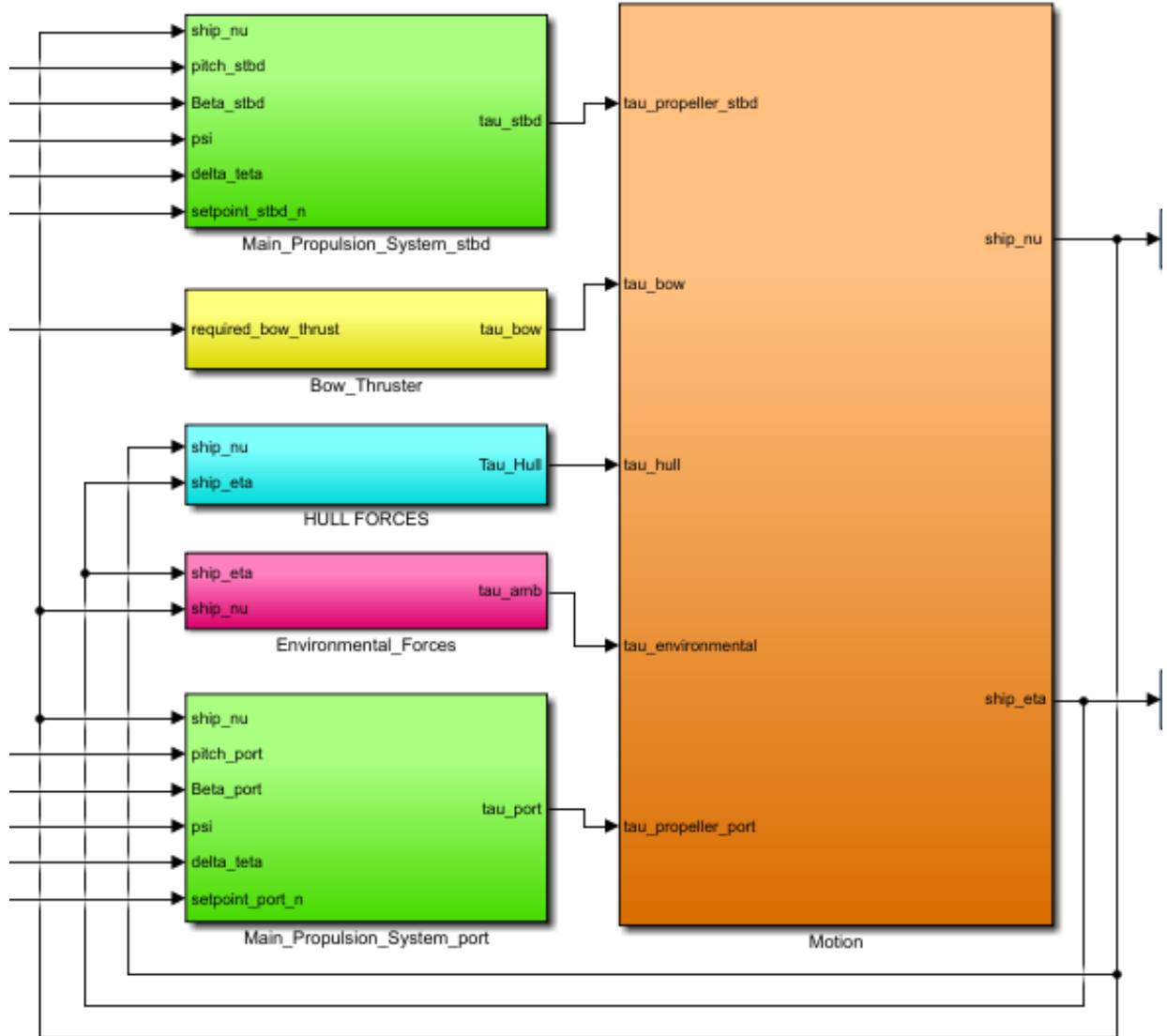


Figure 4.5: Ship model sketch.

The model is the same of that presented in (Alessandri et al. 2014) except for the “main propulsion system” blocks where the propellers are the epicycloidal ones and not the skew propellers. In the model, the motion block receives the delivered thrusts and forces from the actuators and from environmental and hull blocks and computes ship speed and position. From and inside look into the main propulsion system block (Figure 4.6) the main elements of the propulsion plant modelled can be notice: main engine, gearbox and cycloidal propeller.

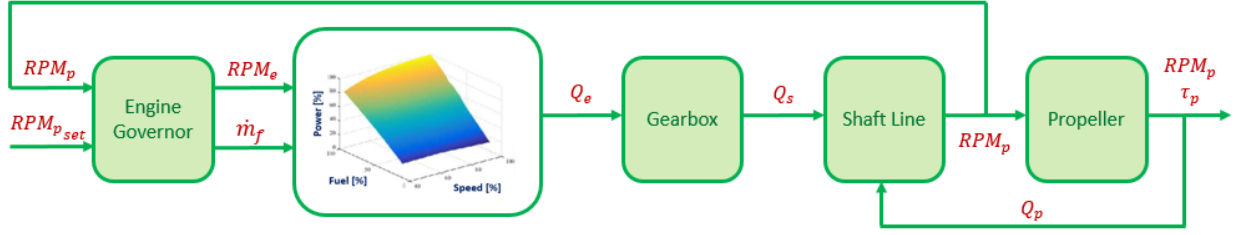


Figure 4.6: Propulsion plant model.

The engine governor, a PI controller, computes the fuel flow requirement $\dot{m}_f$ on the basis of the propeller speed error between RPM_{p_set} (from the combinator law) and RPM_p . The engine is modelled by a response surface whose output is the engine torque Q_e . The last is transformed in shaft torque Q_s through the reduction gear. The output of the line dynamics receives the propeller torque Q_p calculated thanks to Eq. 2.40 and gives RPM_p to the propeller block, where the delivered thrust is computed making use of Eqs. 2.15 and 2.16.

4.5 *Dynamic Positioning Simulation Results*

In this paragraph, some simulation results concerning the vessel equipped with epicycloidal propellers are presented and some of them are compared with those obtained by previous simulations of the same vessel (Alessandri et al. 2014), equipped with a conventional twin screw propulsion system.

4.5.1 *Comparison with traditional propulsion system*

The presented simulation has been carried out assuming the presence of environmental disturbances coming from an angle of 30° with respect to the desired heading, in two distinct cases (Altosole M. et al. 2018): 10 kn and 30 kn. The environmental disturbances have been modelled as detailed in (Donnarumma et al. 2018). In order to consider the worst environmental condition, the disturbances (sea, wind and current) have been considered aligned and coming from the same direction. Sea characteristics have been maintained Mediterranean SS 4 (significant wave height of 1.8 m, wave period of 8.8s) and constant current speed of 1 kn as for the DP capability plots presented in Subsection 4.3, where a static analysis of the ship performance has been presented.

The evaluation of the environmental disturbances mean components $\bar{\tau}_{PD}$ and τ_W requires some minutes of transient that are not relevant for the station keeping performances evaluation. For such a reason, the first few minutes of simulation have been neglected. In Figures 4.7, 4.8 and 4.9 the variations of the ship position and heading are shown in the two different environmental conditions. As we can see, for a wind speed of 10 kn both the propulsion configurations are able to keep the desired position and heading; however, different amplitudes of the oscillations around the desired set-point and then different performances of the two propulsion systems are evident. For a wind speed of 30 kn instead, the conventional propulsion cannot accomplish the desired DP

manoeuvre (see Figure 4.8). The same conclusions are highlighted in the Figures from 4.10 to 4.15, where differences between the two propulsion plants are underlined. The thrusts required to the actuators and thus delivered by the two propulsion plants are very far from each other. Also the required and delivered force and moment are deeply unequal, since the deviation from the desired set-point is very different. These results are reflected in Figures 4.14 and 4.15, where the power required to the engines for the two propulsion configurations is shown.

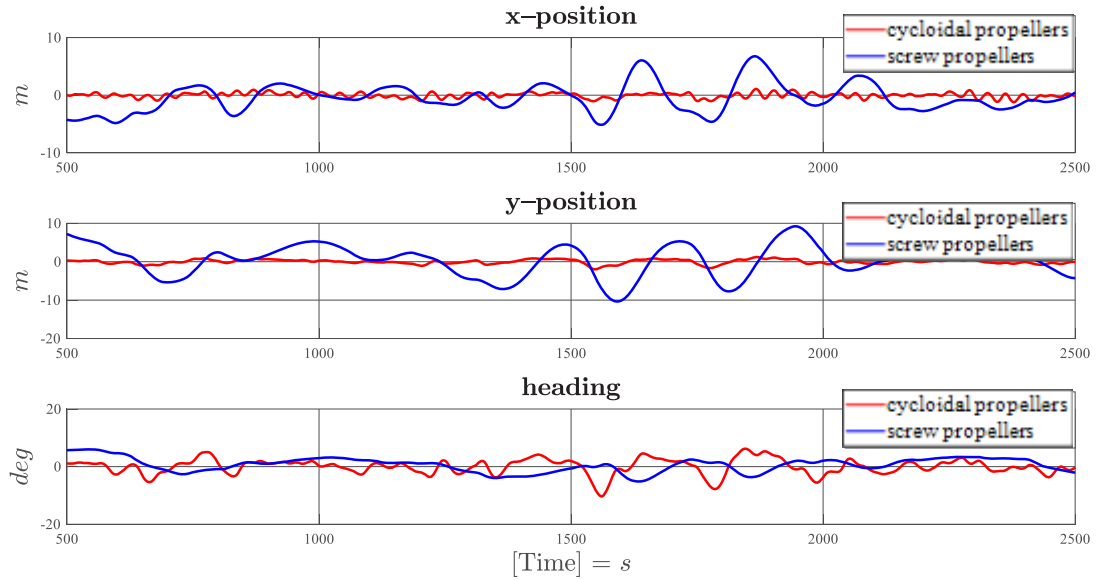


Figure 4.7: Motions time history for wind speed of 10 kn.

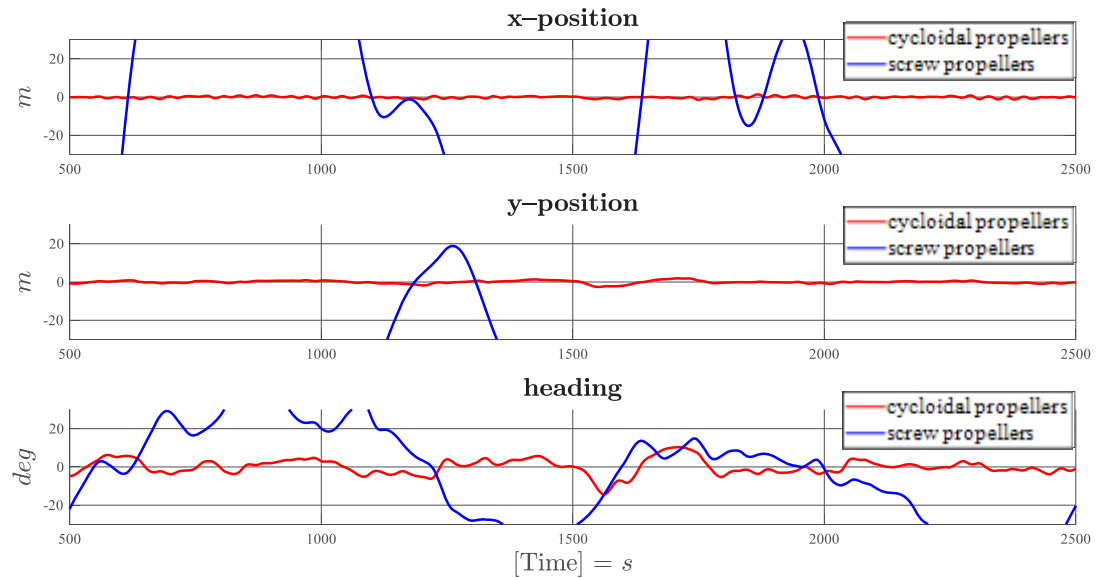


Figure 4.8: Motions time history for wind speed of 30 kn

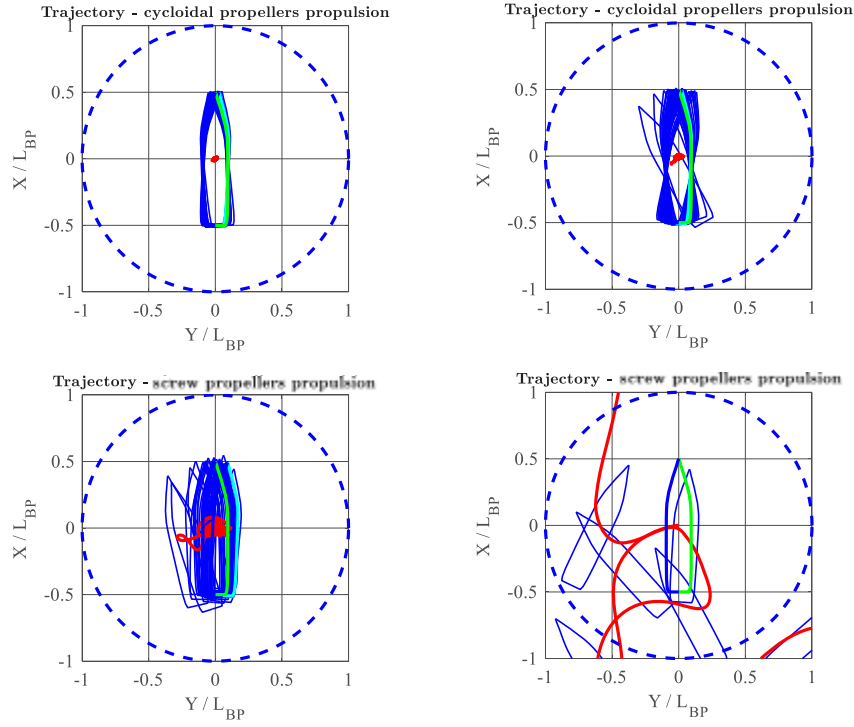


Figure 4.9: Ship position (trajectory of the origin of the body-fixed frame) and orientation variations for wind 10 kn (on the left) and 30 kn (on the right).

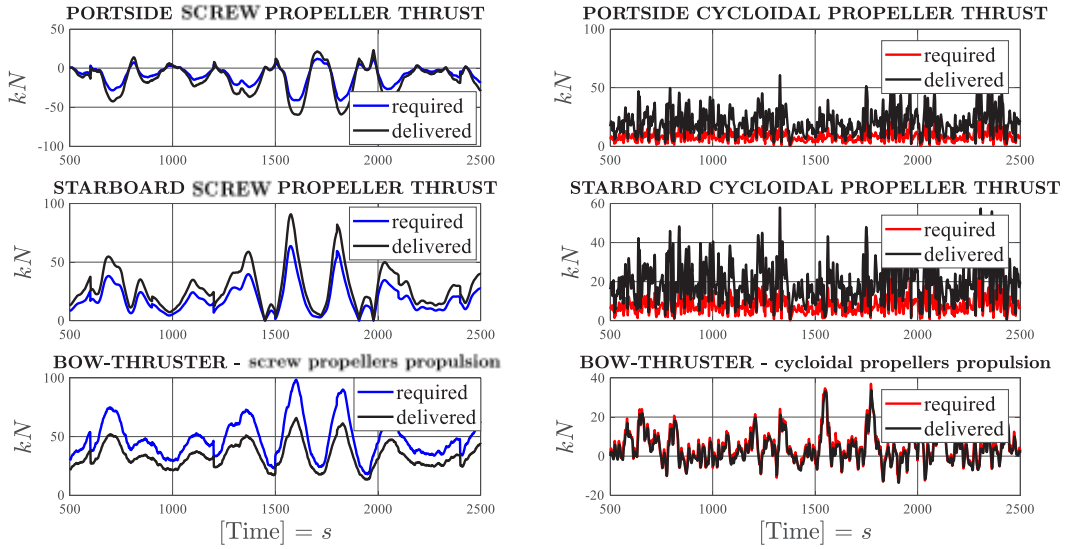


Figure 4.10: Time history of required and delivered thrust for wind speed of 10 kn

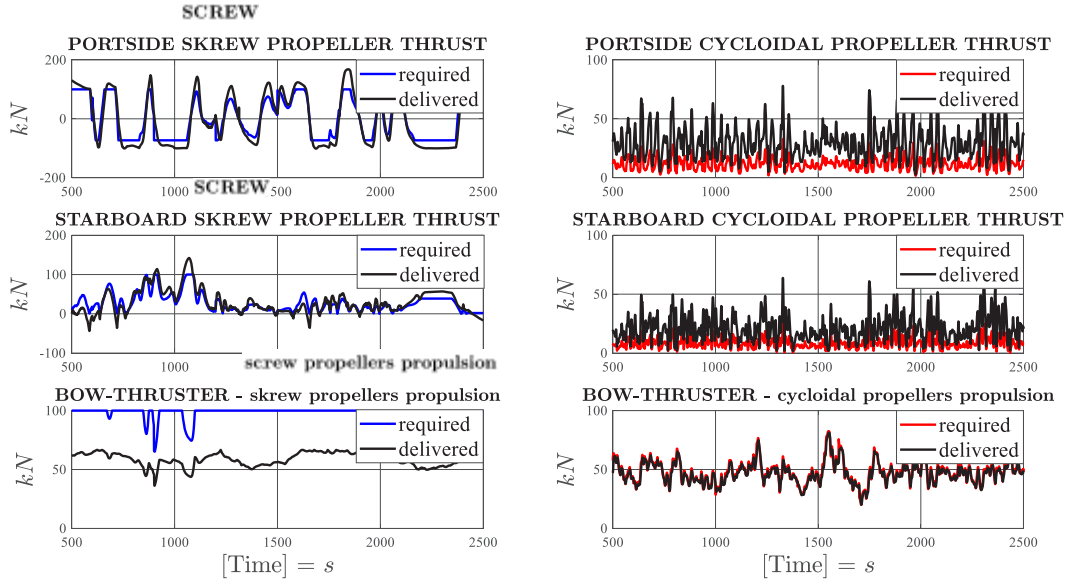


Figure 4.11: Time history of required and delivered thrust for wind speed of 30 kn.

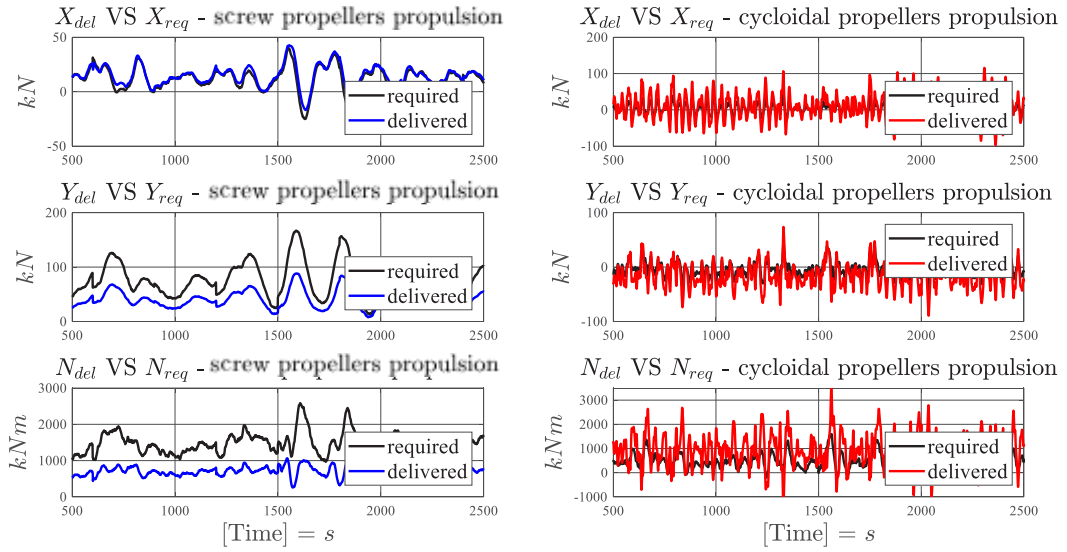


Figure 4.12: Time history of required and delivered forces and moment for wind speed of 10 kn.

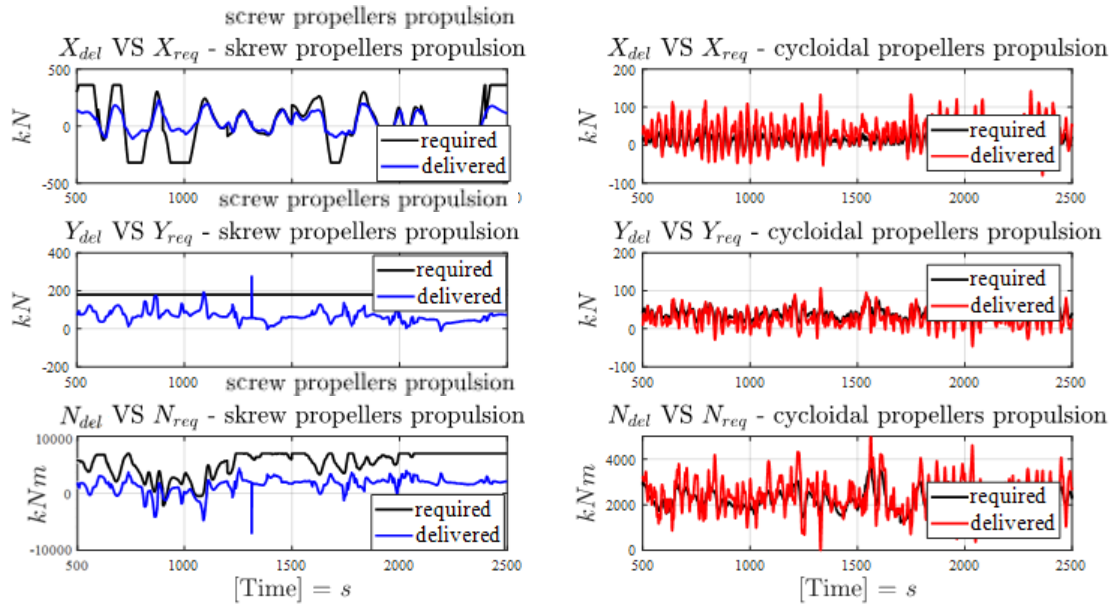


Figure 4.13: Time history of required and delivered forces and moment for wind speed of 30 kn.

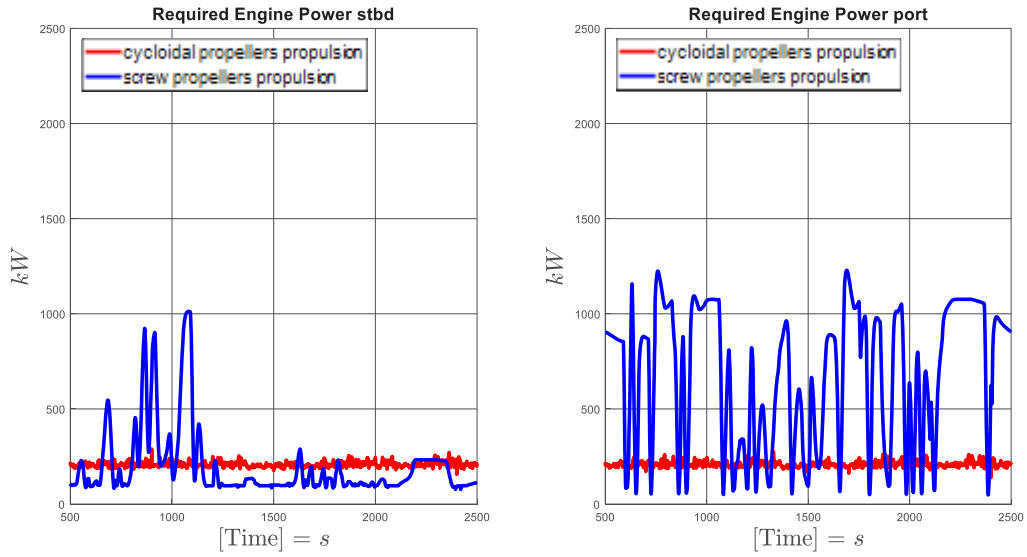


Figure 4.14: Time history of required engine power for wind speed of 10 kn.

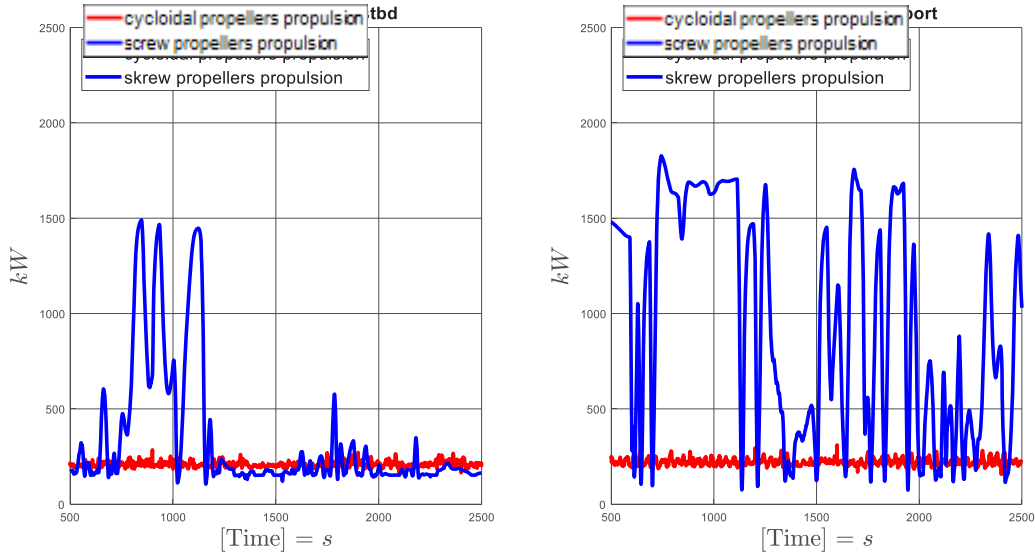


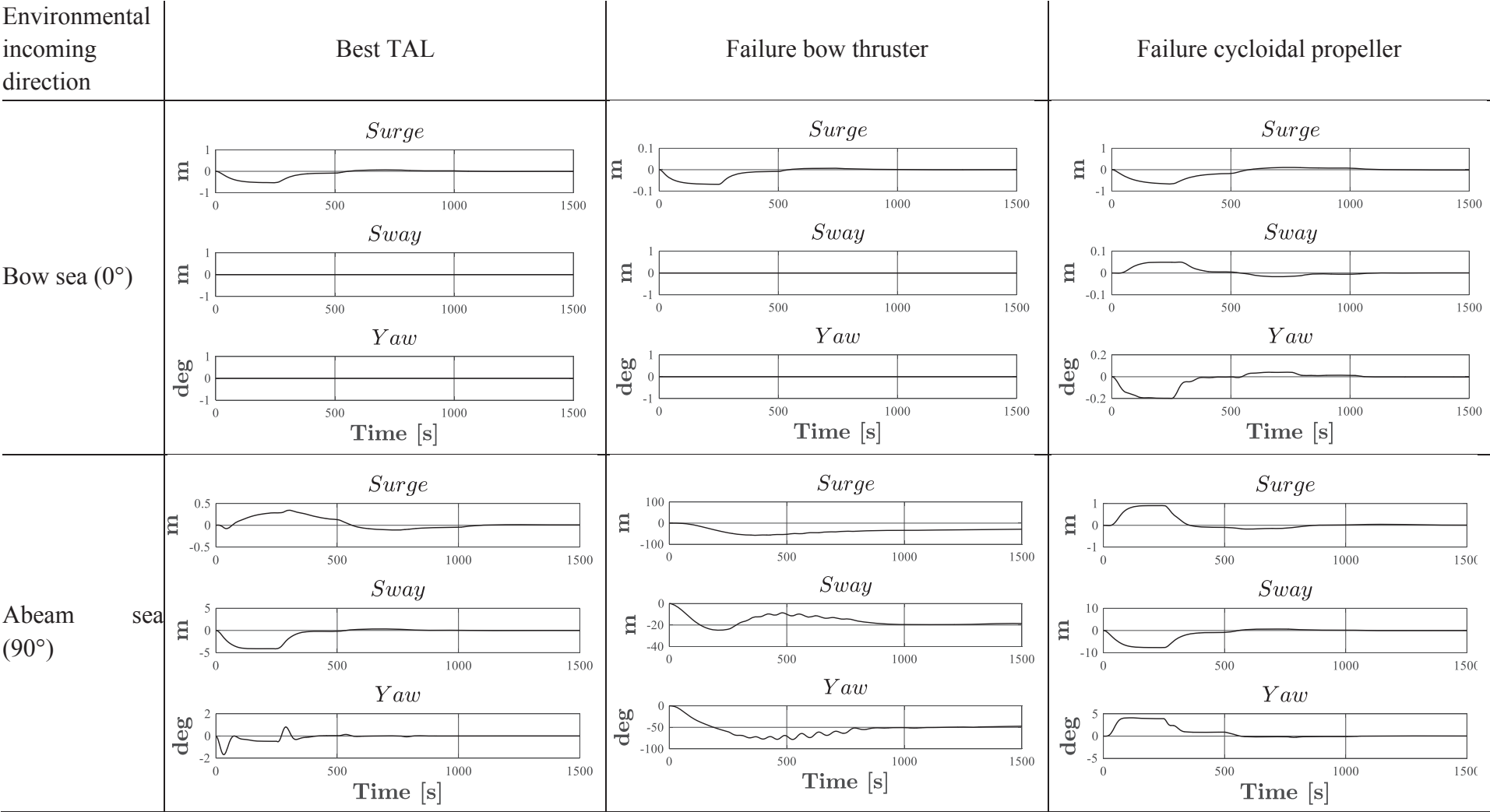
Figure 4.15: Time history of required engine power for wind speed of 30 kn.

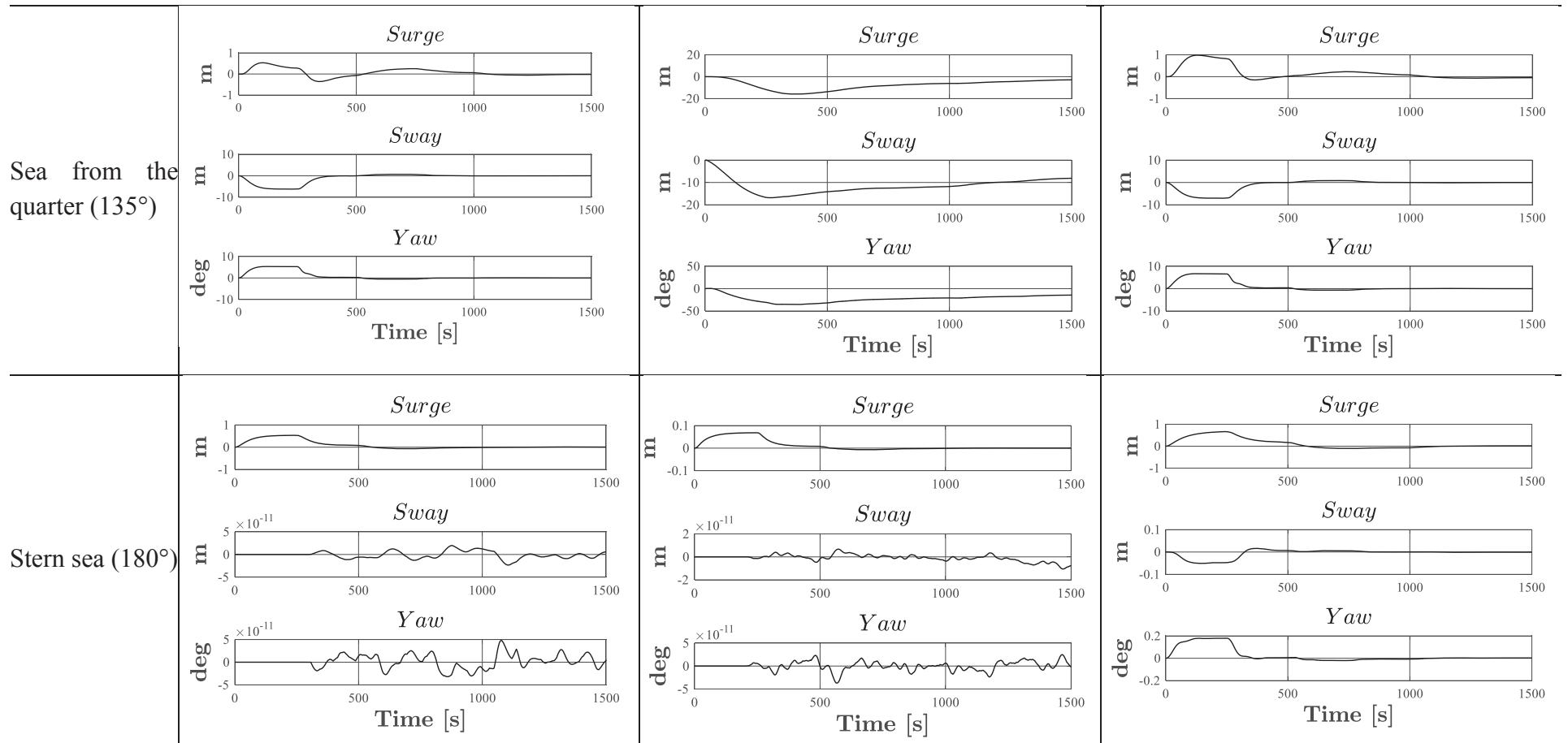
The obtained results confirm the conclusions achieved by the static analysis. As it was expected, cycloidal propellers can be a valid alternative to traditional propellers in case of strong DP requirements. Simulation results also illustrate the effectiveness of the proposed propulsion plant and the corresponding thrust allocation, as well as the reliability of the mathematical and numerical model implemented for cycloidal propellers.

4.5.2 Failure conditions

Various simulations have been carried out considering the optimized thrust allocation logic (best TAL), assuming the failure of the bow thruster and the failure of one of the two cycloidal propellers. As mentioned above, environmental disturbances (SS4, Mediterranean Sea) have been considered aligned and coming from the same direction. The aim was to verify that the unit was able to maintain the position and heading required also in the presence of some failures. The following table, provides results for different environmental disturbances. For sake of conciseness, only the graphs representing the position and heading time history have been reported.

Analysing the results obtained with the optimal allocation logic, we note that the unit is very stable. Even at sea to the cross beam, it reaches the position in a short time and keeps it without oscillations. When the sea come from the stern, the first two cases give good results but the failure of one cycloidal propeller causes a slight initial instability that however restores in a short time. Also for disturbances coming from the bow quad, position and heading errors are small. The case of the sea at the cross is the most difficult condition. In fact, if the bow thruster was not operating, the unit would not be able to guarantee adequate station keeping performances.





4.5.3 Particular DP operations - automatic control mode

In general, the control of low-speed motions belongs to DP system. In order to validate such capabilities for the case study, some particular manoeuvres were simulated such as hold distance, hold area and change position/heading.

For these automatic control modes, the structure of the regulator has been kept the same as described in Paragraph 4.4.3. In particular, the regulator still receives a set of position, heading and speed errors and, with reference to the Figure 4.16, the required forces and moments are then calculated according to the expression:

$$F_{X_0, Y_0, \psi}^R = -K_P^{X, Y, \psi} \cdot E_{X, Y, \psi}^0 - K_D^{X, Y, \psi} \cdot E_{U, V, r}^0 - \left(K_I^{X, Y, \psi} \cdot \int_0^t E_{X, Y, \psi}^0 dt \right) \quad (4.13)$$

where $K_{P,D,I}$ are PID coefficients, $E_{X,Y,\psi}^0$ and $E_{U,V,r}^0$ are respectively the position and speed errors evaluated in the absolute reference system. Consequently, the regulator generates, still in the absolute reference system, a set of required global forces, which, before being passed to the allocation, are translated into the body-fixed system.

With reference to the general conventions shown in the following Figure (4.16), the various control modes listed above will be described, always assuming to start from the hold position condition.

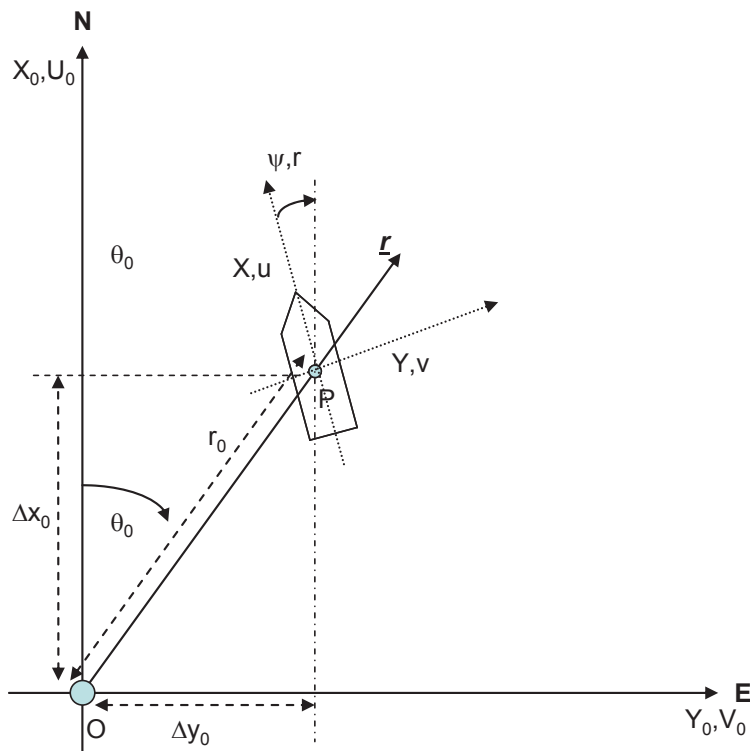


Figure 4.16: Absolute reference system for automatic control mode.

Some simulations made for the different modes will be shown as well for a trial time of 1 hour.

4.5.3.1 Hold Area

This control mode can be defined as an energy-saving DP-mode since it ensures that the ship is stationed within a certain area with the minimum use of thrusters, if there is no need for a tight maintenance of the position.

The area A_0 is defined by a circle of radius R_0 and centre coinciding with the current position of the unit at the beginning of the manoeuvre, as illustrated in the figure below.

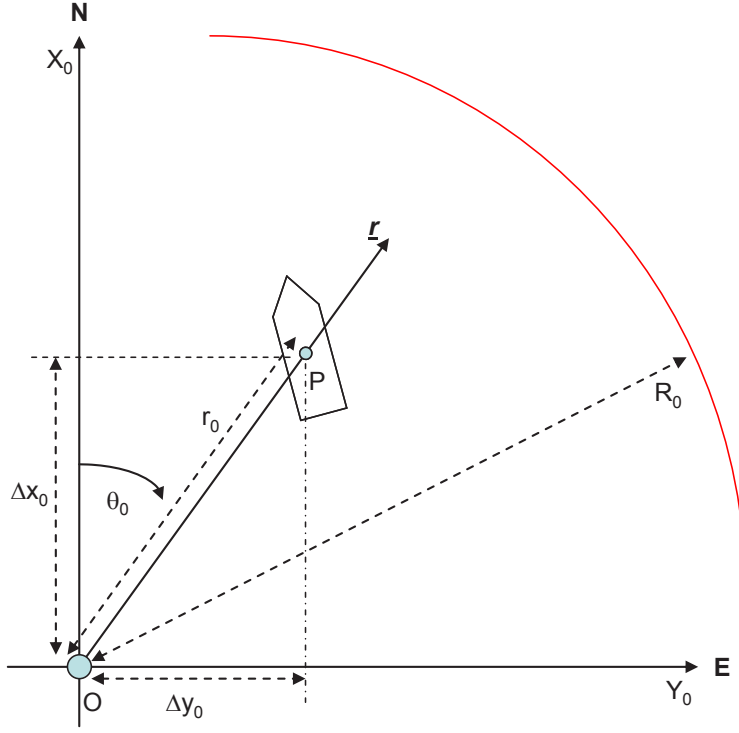


Figure 4.17: Hold area reference system.

This mode can be defined as a “relaxed control” since it does not use thrusters until the distance of the ship from the centre of the reference system is near the limit radius.

The regulator is a PD type (the integral term is omitted) with respect to the radial direction. The radial force $\bar{F}_R$ is calculated as follow:

$$\bar{F}_R = - (K_P \cdot E_r + K_D \cdot E_{U_r}) \cdot \bar{r} \quad (4.14)$$

Where:

$\bar{r} = (\cos\theta_0, \sin\theta_0)$ is the unit vector directed outside the limit area

$E_r = e\left(\frac{r_0}{R_0}\right)$ is the radial error function of dimensionless radius

$E_{U_r} = \bar{U}_0 \cdot \bar{r}$ is the error of the radial speed

Position/speed errors are obtained by:

$$E_X = \cos\theta_0 \cdot e\left(\frac{r_0}{R_0}\right) , \quad E_Y = \sin\theta_0 \cdot e\left(\frac{r_0}{R_0}\right) \quad (4.15)$$

$$E_U = \cos\theta_0 \cdot (U_0 \cdot \cos\theta_0 + V_0 \cdot \sin\theta_0) , \quad E_V = \sin\theta_0 \cdot (U_0 \cdot \cos\theta_0 + V_0 \cdot \sin\theta_0) \quad (4.16)$$

The following figures show the simulation results for the hold area mode with SS2. The heading of the ship has been set to have the opposite direction with respect to the incoming direction of the environmental disturbances using the basic functionality illustrates in Paragraph 4.4.3.

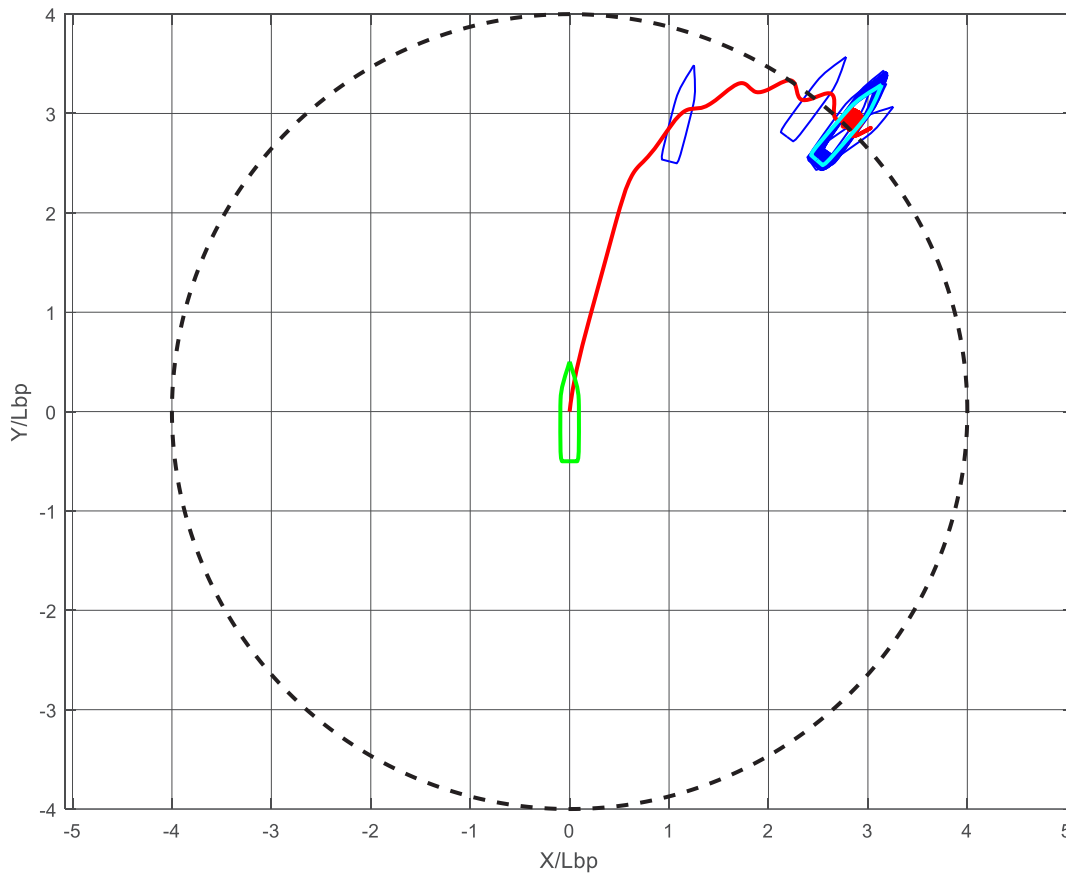


Figure 4.18: Keep Area mode simulation result.

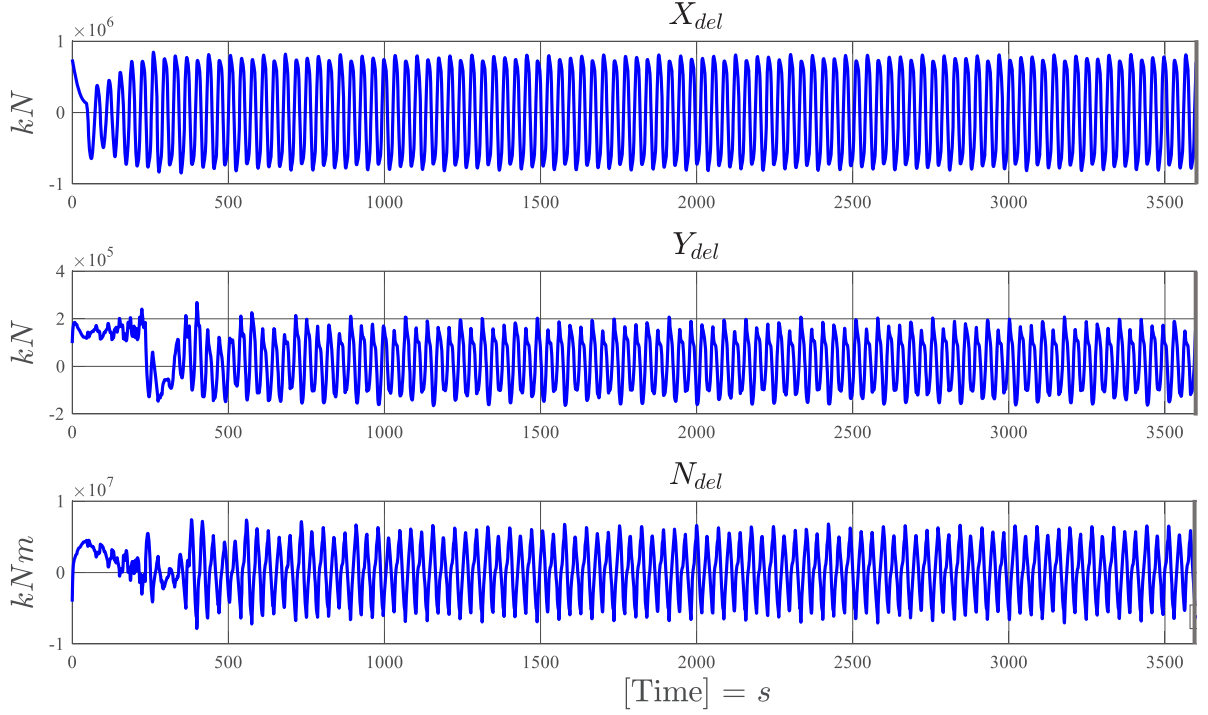


Figure 4.19: Delivered forces and moment time history.

The Figure 4.19 shows how the delivered forces and moment are small (they are not equal to zero since we required the heading to equal environmental direction of 40°) until the vessel reach the boundary line.

4.5.3.2 Hold Distance

This control mode corresponds to the so-called “hovering”, that is to keep constant the distance (R_0^{set}) between the ship and a reference point, without getting too close or moving away from it. In this case, the regulator is assumed to be a PID one, with respect to the radial error. The radial required force $\bar{F}_R$ is defined as:

$$\bar{F}_R = - \left[K_P \cdot E_r + K_D \cdot E_{U_r} + \left(K_I \cdot \int_0^t E_r dt \right) \right] \cdot \bar{r} \quad (4.17)$$

where the terms of the equation are the same of those described in the previous paragraph. Position/speed errors are computed in accordance with:

$$E_X = \cos\theta_0 \cdot (r_0 - R_0^{set}) , \quad E_Y = \sin\theta_0 \cdot e(r_0 - R_0^{set}) \quad (4.18)$$

$$E_U = \cos\theta_0 \cdot (U_0 \cdot \cos\theta_0 + V_0 \cdot \sin\theta_0) , \quad E_V = \sin\theta_0 \cdot (U_0 \cdot \cos\theta_0 + V_0 \cdot \sin\theta_0) \quad (4.19)$$

Since the aim of hold distance mode is only to control the radial deviation from the set point, the heading is left free to move.

In the following figure, the non-dimensional path, obtained from simulation with SS4 and a radius set to 100m, is shown.

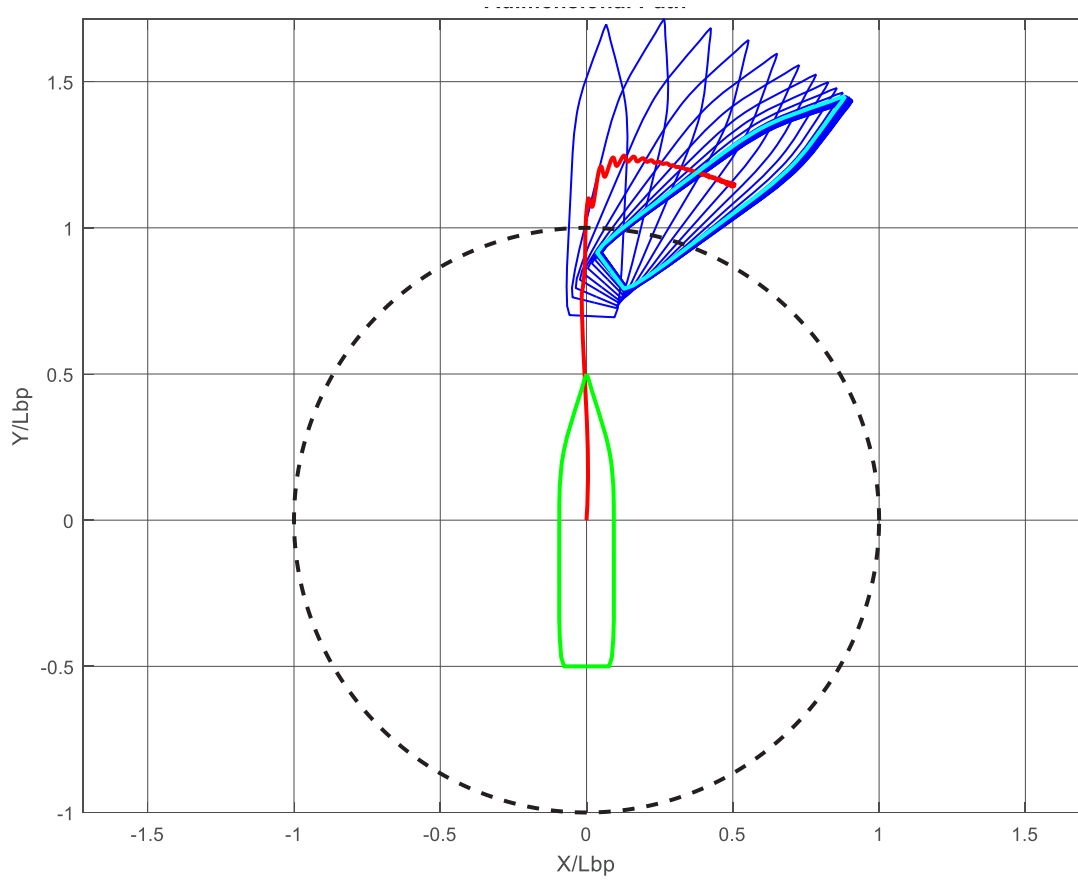


Figure 4.20: Hold Distance mode simulation result.

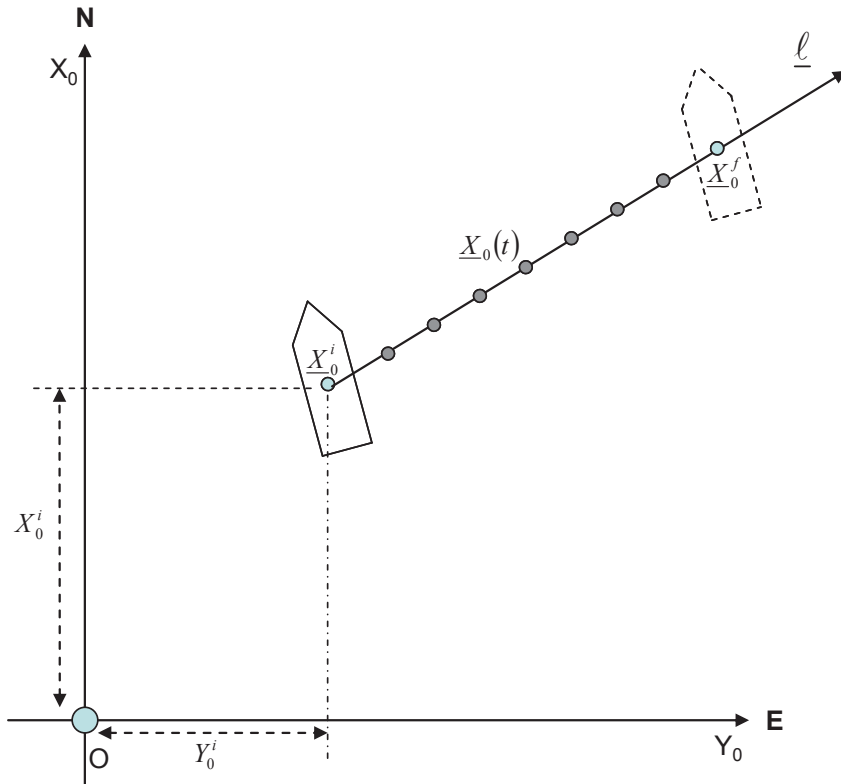
4.5.3.3 Change Position/Heading

Starting from the keep heading/position set-points, the change position/heading mode allows to manually change these set-points.

Even if the function of change position/heading allows to move automatically the ship from one position to another, it does not constitute a mode of auto-pilot/track-keeping. In order to maintain the typical characteristics of DP operations in term of short distances and low speed, the maximum linear deviation from the set point allowed by the system is restricted to half a nautical mile along the two main directions. Furthermore, the speeds that can be reached during these manoeuvres are limited to 1.0 kn for the linear speed and to $0.5^\circ/\text{s}$ for the angular one.

It is assumed that the starting position and heading are stabilized. The operator decides the new position set-point, in terms of the linear deviations ΔX_0 and ΔY_0 from the current reference position, and the new heading set-point, in terms of angular deviation $\Delta\psi_0$ from the current reference heading.

In order to implement the change position/heading function, the hold position/heading is used following a pre-calculated sequence of position/heading set-points.



It is assumed that for each set point the PD type regulator match the hold position/heading mode one with the following position and speed errors:

$$E_U^0 = U_0^P - U_0^{set} \quad , \quad E_V^0 = V_0^P - V_0^{set} \quad , \quad E_r^0 = r_0^P - r_0^{set} \quad (4.21)$$

	X position	Y position
<i>From 0 to 999s</i>	0	0
<i>From 1000 to 1999</i>	100 m	100 m
<i>From 2000 till the end</i>	50 m	50 m

Table 4.1: Change Position/Heading mode set.

The heading has been set to 45° for the whole simulation. The results are shown below.

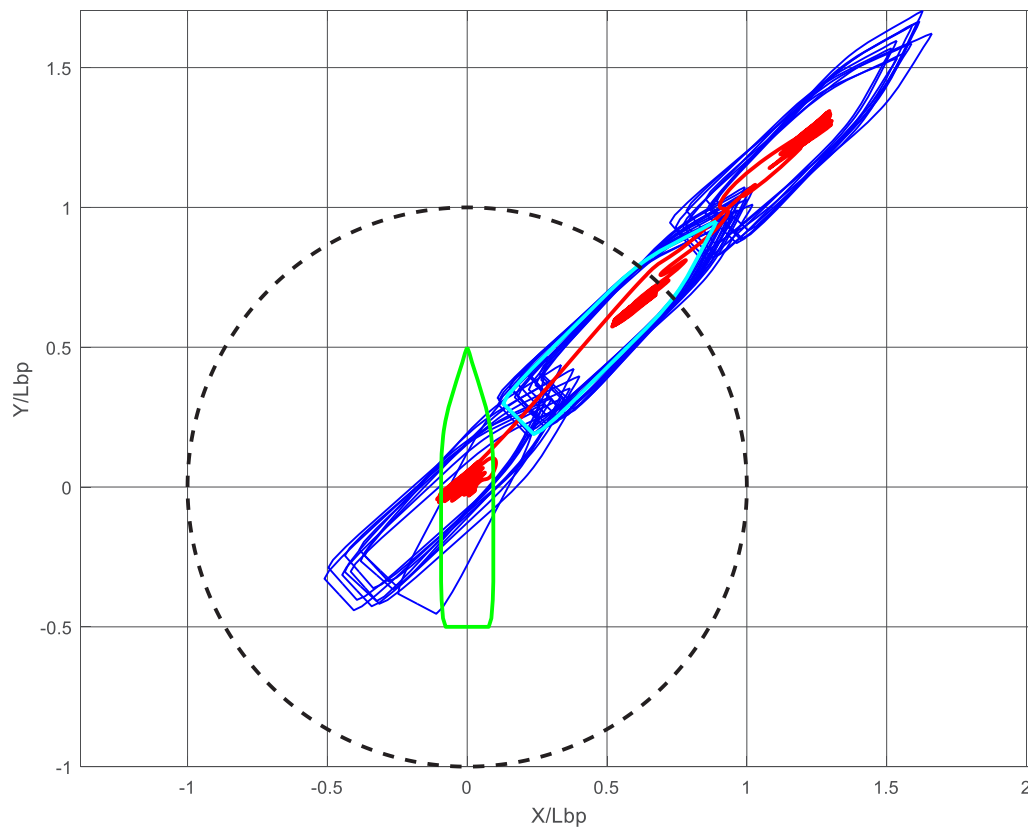


Figure 4.22: Change Position/Heading mode simulation - Non dimensional path.

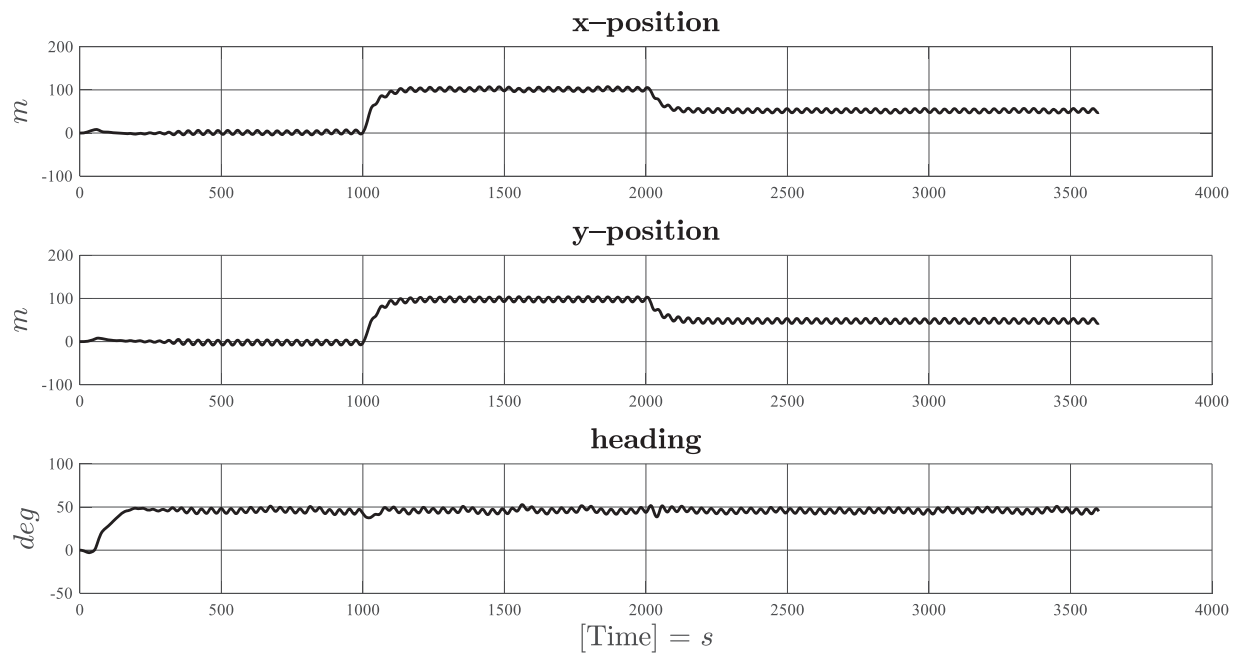


Figure 4.23: Change Position/Heading mode simulation results.

5 MANOEUVRES AT CRUISING SPEED

In order to confirm the goodness of the cycloidal simulation model, some manoeuvres at cruise speed have been performed. In this chapter we will present some results obtained trying to reproduce the real sea trials carried out for a patrol vessel of the Italian Coast Guard, a ship equipped with a conventional twin screw propulsion system for which a DP system has been studied (Alessandri et al. 2014), implemented and installed on board. Using a consolidated ship model, the only uncertainties are related to the epicycloidal propellers and their interaction with the hull.

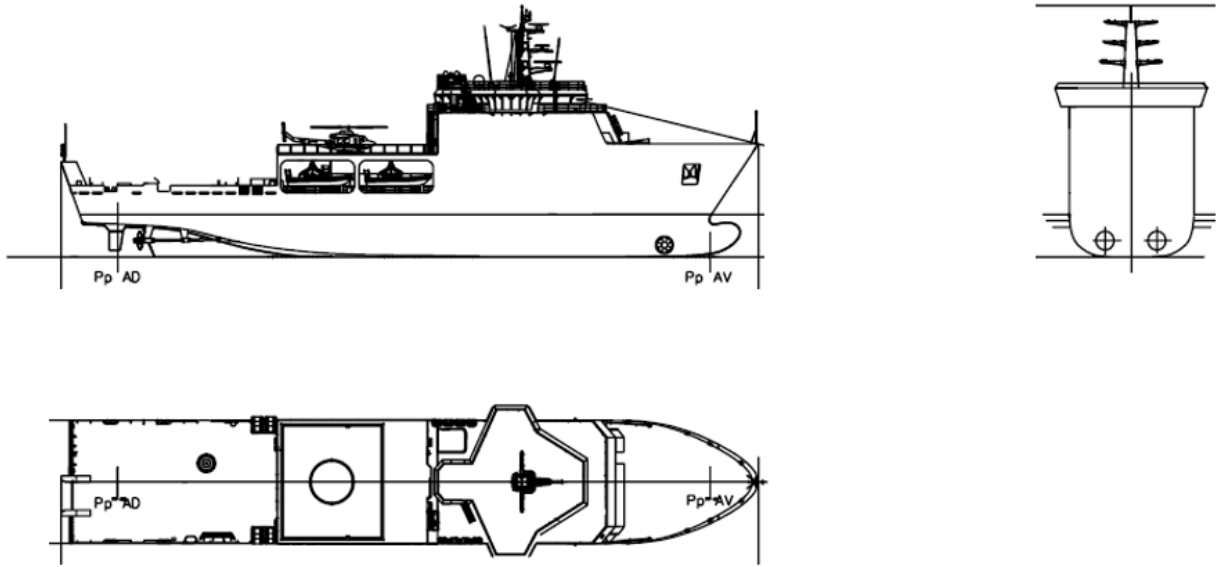


Figure 5.1: Patrol vessel layout.

For confidentiality reasons, the results are given and illustrated in non-dimensional form.

5.1 *Preamble*

First of all, some initial conditions were set:

- no environmental disturbances;
- the cycloidal propeller thrust directions are parallel;
- manoeuvre start time= 200 s (steady speed).

In the following paragraphs a comparison between simulated data and real sea trials results is reported. For the sake of brevity, only the most relevant results are shown. Since the design speed cannot be disclosed for confidentiality reasons, all the results are presented in non-dimensional form, where, in the ratio V_{test}/V_{max} , V_{test} is the speed test and V_{max} is the design speed. The simulated tests are:

- $V_{test}/V_{max} = 0.94$: evolution, pull out, zig zag 20/20, crash stop;
- $V_{test}/V_{max} = 0.72$: evolution, pull out, zig zag 20/20.

5.1.1 Combinator law

As we have pointed out in the description of the epicycloidal propeller model, there are two different pitches (s and β) that control respectively the magnitude and the direction of the thrust, together with the choice of a suitable rotation speed. In this case study, simulations have been performed using three rotation speed set (trying to keep them constant for as many speed choices as possible), so modifying the magnitude of the thrusts only by means of the geometric pitch s . The so-called “combinator law” for cruising speed has to assign set-points at propulsion plant corresponding to the desired speed. Usually, if coupled to electric engines, epicycloidal propellers are driven at constant speed. In case of diesel engines, the rotation speed is keep constant as much as possible. In the presented combinator law, engines rpms were set and then the corresponding pitch was investigated for each one of the three rotation speed steps, in order to obtain the desired speed of the vessel.

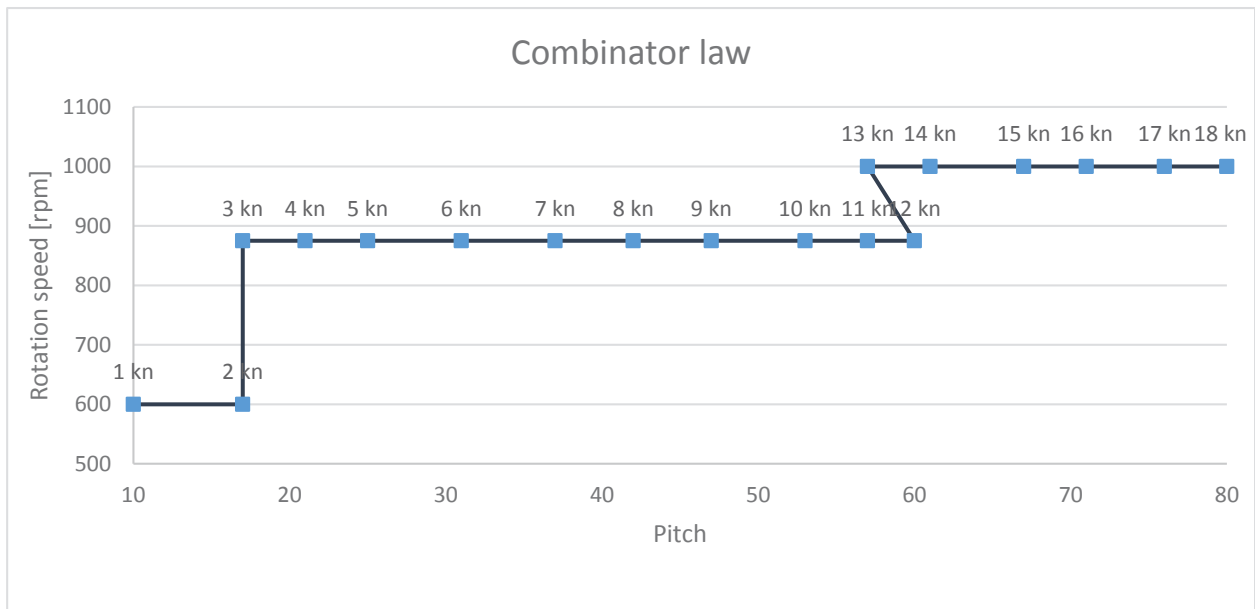


Figure 5.2: Combinator law.

As mentioned before, the pitch β controls the direction of the thrust. In Figure 5.3 the link between β angle and thrust direction is illustrated.

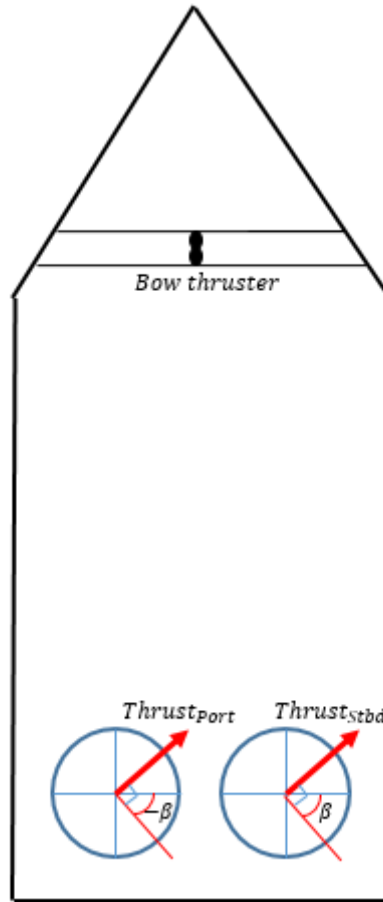


Figure 5.3: β angle and thrust direction.

5.2 Turning circle/Pull out

During this manoeuvre, starting from straight motion at the test speed, the rudder (for traditional propulsion) is turned to 35° or the maximum rudder angle admissible and kept at this angle until the ship has performed a turning circle trajectory for at least 540° . We tried to reproduce the same evolution diameter resulting from real sea trials of the patrol vessel. Since cycloidal propeller thrusts can be directed in any direction, the setting of the direction pitch angle β is then the result of this requirement. The main information obtained from this manoeuvre consist of tactical diameter, maximum advance, transfer at 90° change of heading, and final diameter.

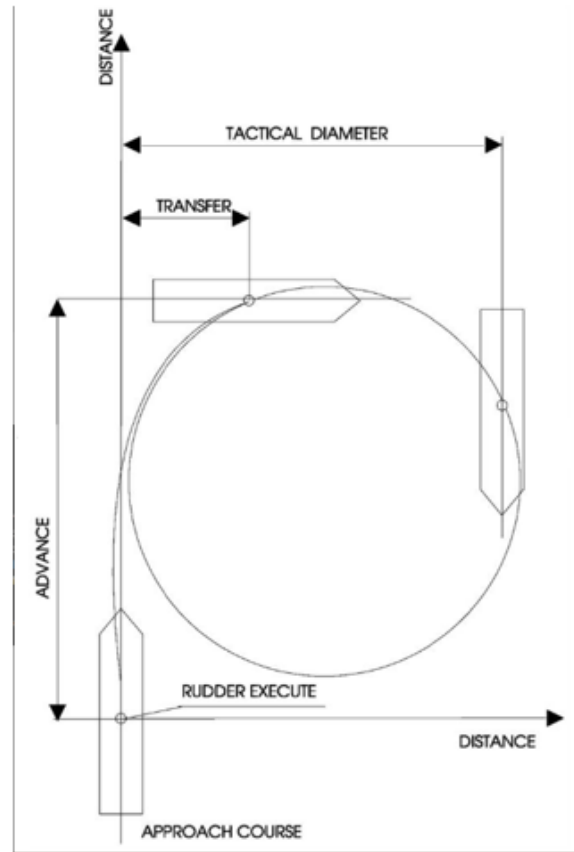


Figure 5.4: Turning circle manoeuvre (www.wartsila.com).

The pull out manoeuvre is performed right after a turning circle manoeuvre. When the heading have completed 720° , the rudder/ β angle is returned to midship/zero.

The following table summarizes the main data obtained by simulation. The comparison with real sea trials data are reported in figures from 5.5 to 5.10.

Speed $V_{\text{test}}/V_{\text{max}}$	β angle stbd [°]	β angle port [°]	Transfer Y midship/ L_{BP}	Advance X midship/ L_{BP}	Tactical Diameter Y midship/ L_{BP}
0,72	-50	50	1,82	1,79	3,15
0,94	-50	50	1,91	-0,06	3,4
	-60	60	1,48	0,14	1,55

Table 5.1: Pull out simulation output.

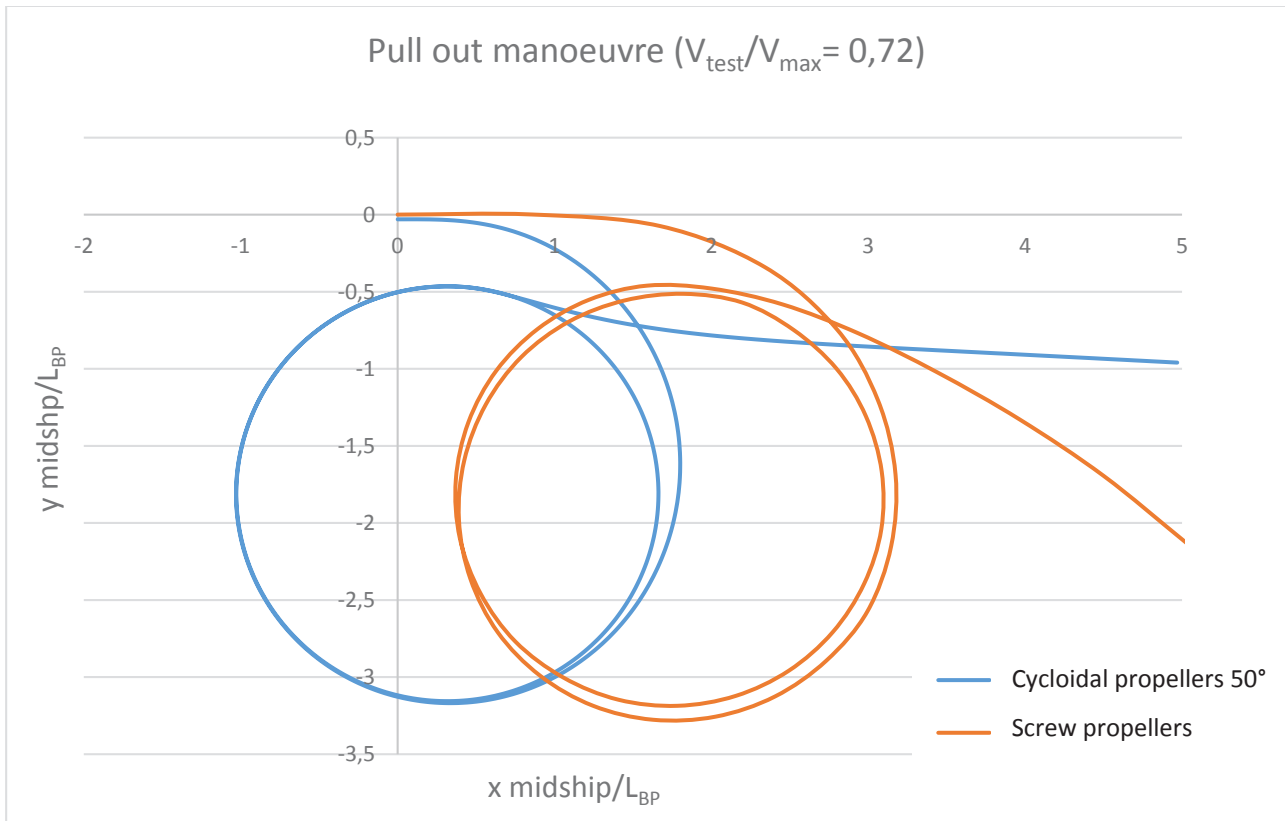


Figure 5.5: Pull out manoeuvre $V_{\text{test}}/V_{\text{max}} = 0,72$ - position of ship centre of gravity in non-dimensional form.

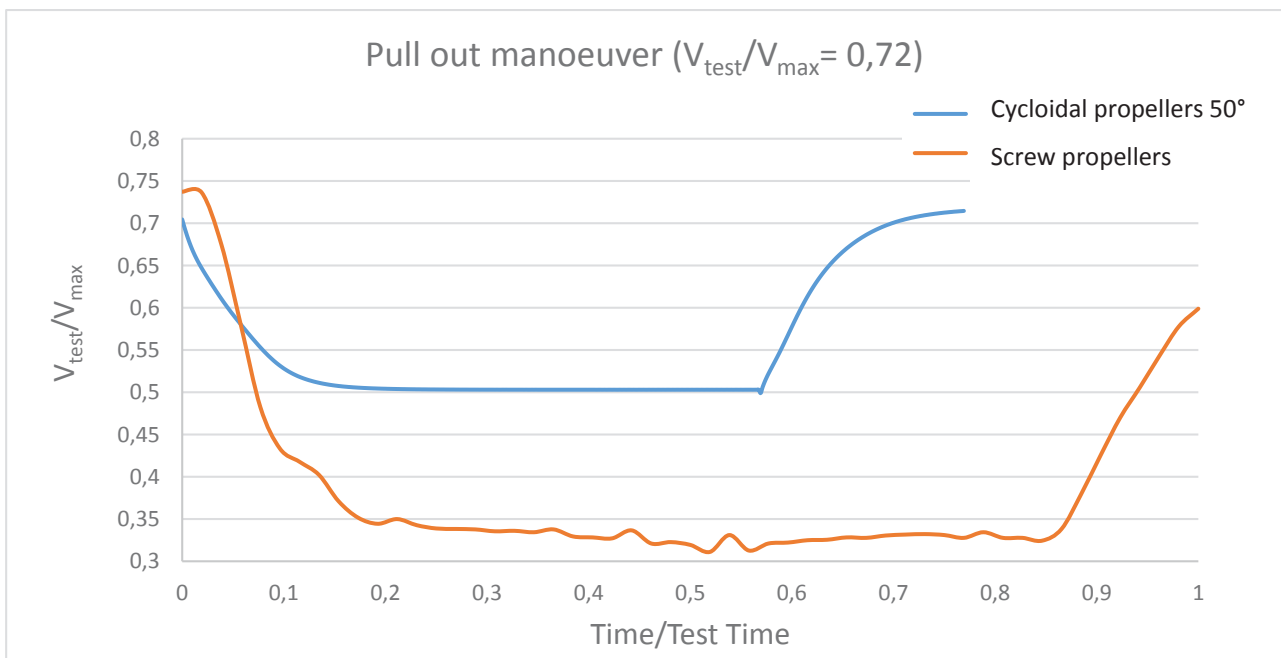


Figure 5.6: Pull out manoeuvre $V_{\text{test}}/V_{\text{max}} = 0,72$ – speed function of time in non-dimensional form.

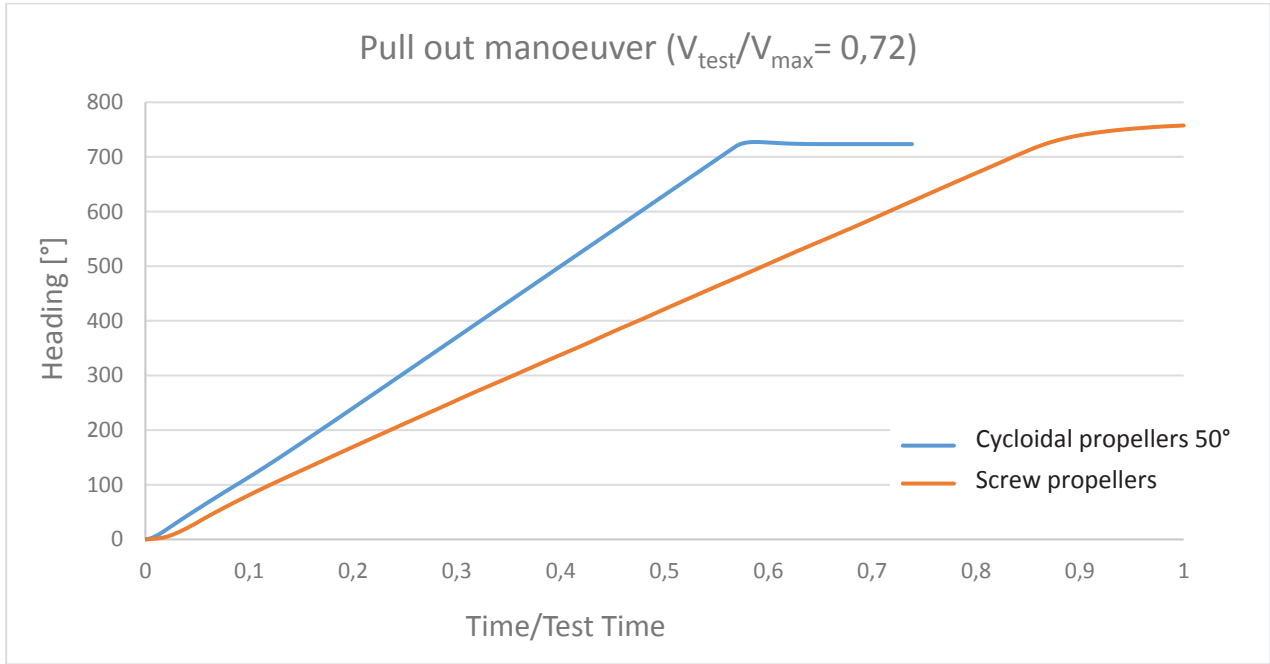


Figure 5.7: Pull out manoeuvre $V_{\text{test}}/V_{\text{max}} = 0,72$ – heading function of time in non-dimensional form.

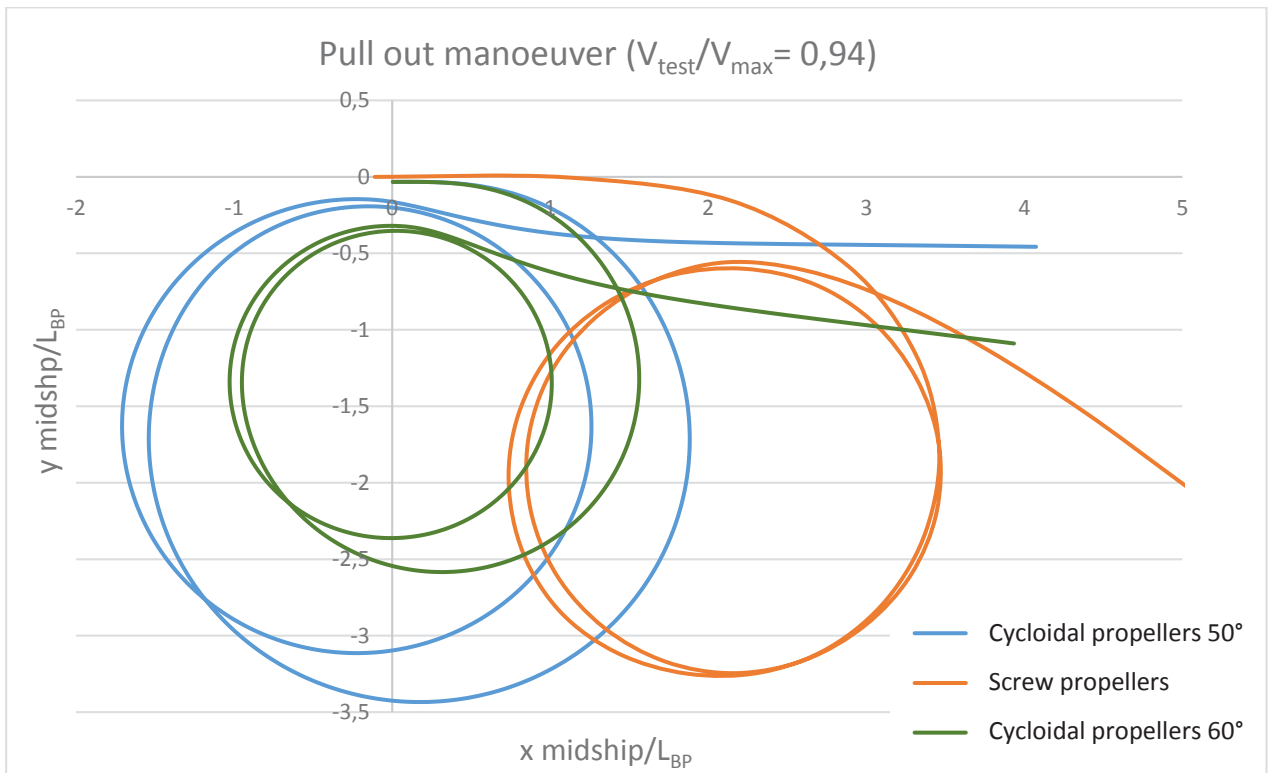


Figure 5.8: Pull out manoeuvre $V_{\text{test}}/V_{\text{max}} = 0,94$ – position of ship centre of gravity in non-dimensional form.

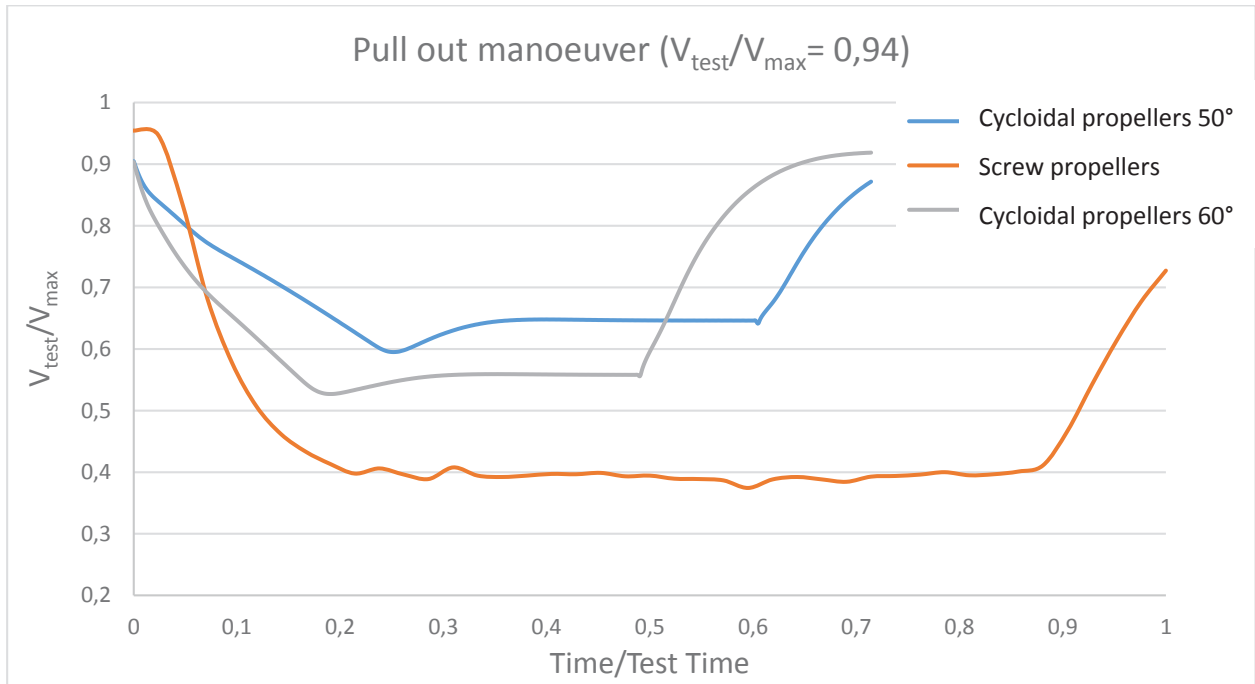


Figure 5.9: Pull out manoeuvre $V_{\text{test}}/V_{\text{max}} = 0,94$ – speed function of time in non-dimensional form.

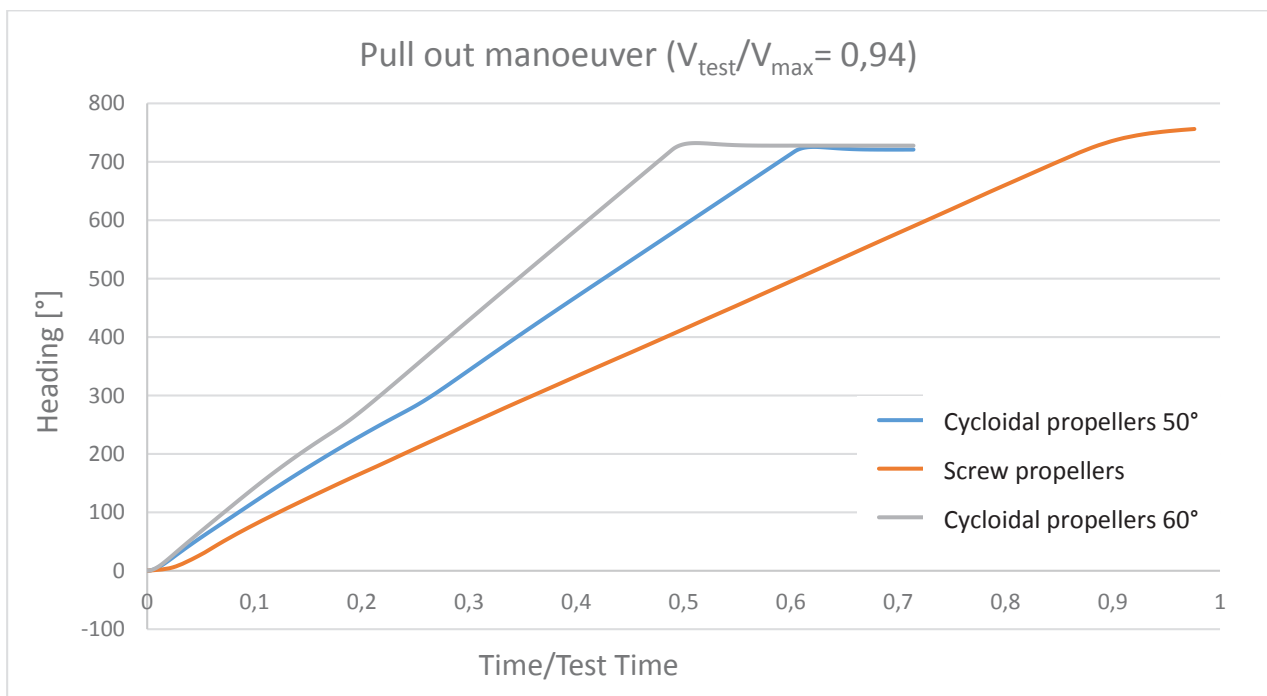


Figure 5.10: Pull out manoeuvre $V_{\text{test}}/V_{\text{max}} = 0,94$ – heading function of time in non-dimensional form.

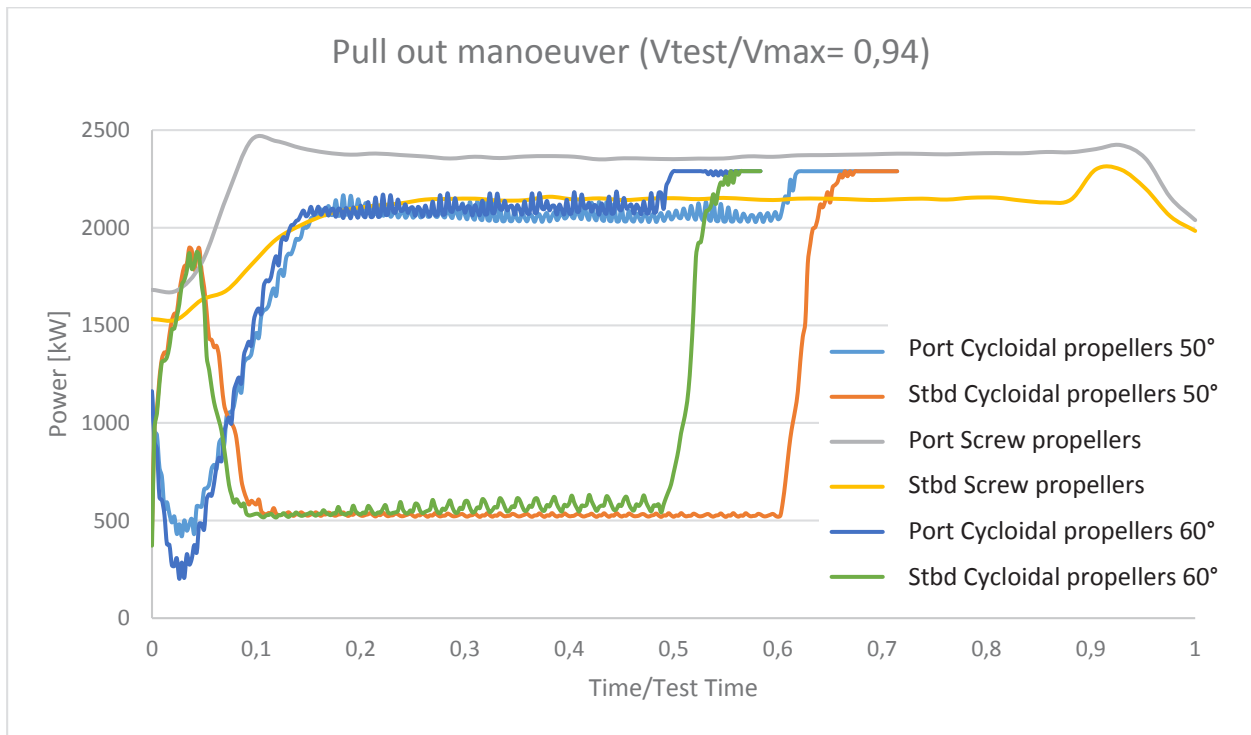


Figure 5.11: Pull out manoeuvre $V_{\text{test}}/V_{\text{max}} = 0,94$ – Engine power function of time in non-dimensional form.

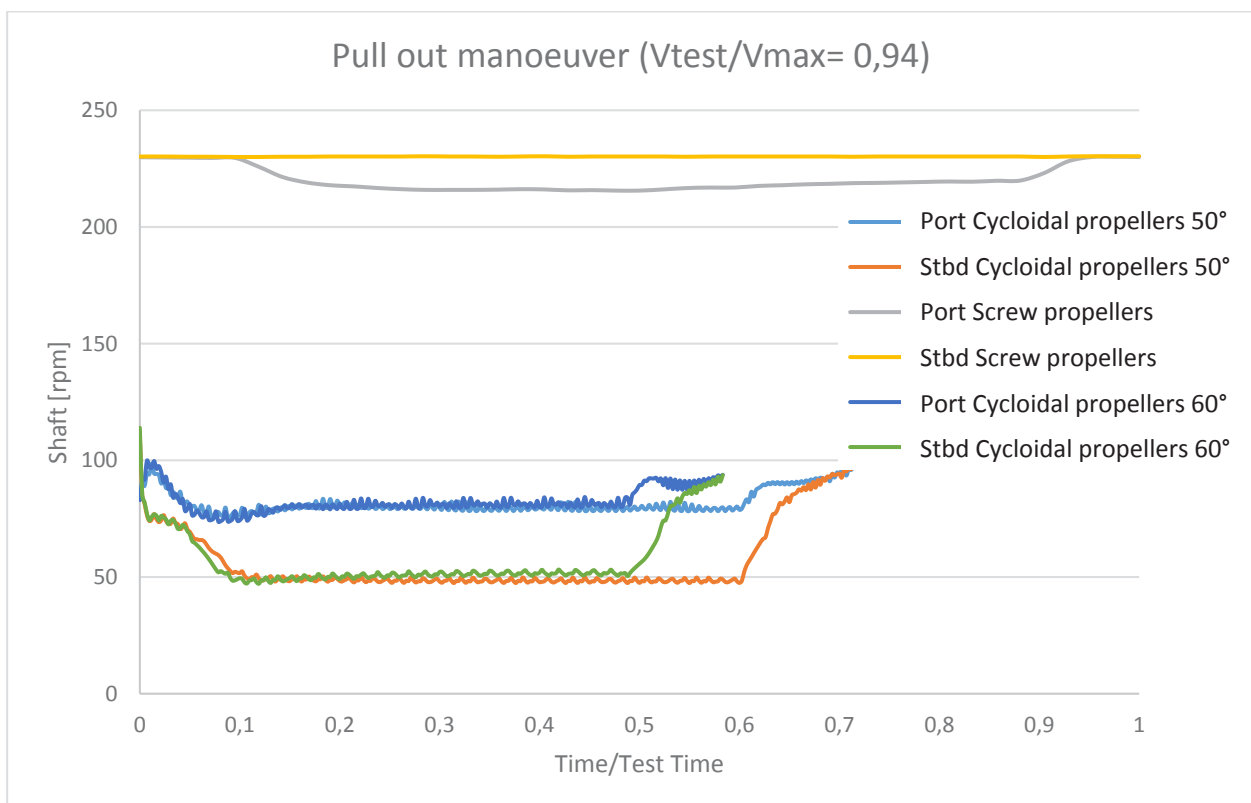


Figure 5.12: Pull out manoeuvre $V_{\text{test}}/V_{\text{max}} = 0,94$ – Shaft speed function of time in non-dimensional form.

The tests performed show how the readiness of the unit propelled by cycloidal propellers is much greater than that obtained with traditional propulsion and that the decrease in speed during these manoeuvres is much lower (as it was largely expected). Moreover, in Figure 5.11 is shown that for all for all propulsion types, the external engines have to work much more than the internal ones. The notable difference in shaft speed (Figure 5.12) is due to the fact that the reduction ratio is different: the revolutions of the epicycloidal propellers are subjected to two reduction stages, one identical to the solution with propellers and an additional one of 3.4 inside the propulsor itself.

5.3 *Zig zag*

Zig-zag manoeuvre, usually 10/10 or 20/20, consists in changing the rudder/ β angle to 20° (in this case) until the heading reach 20° (first execute). When the heading is 20° , the rudder/pitch is reversed to the same rudder/ β angle to the opposite side (second execute). After this command, the ship continues turning in the original direction (overshoot). Once again, when the heading is -20° , the rudder/ β is reversed again. This process continues for five rudder/pitch changes.

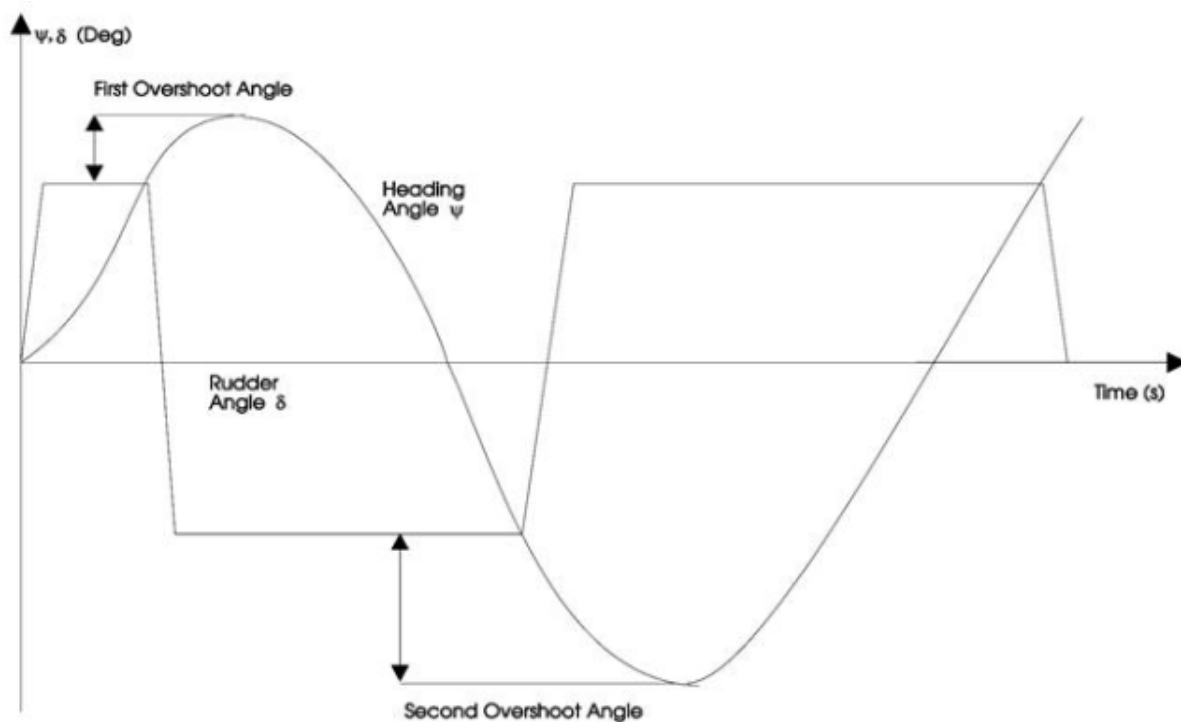


Figure 5.13: Zig-zag manoeuvre (www.wartsila.com).

In the following table, input and output data of the simulation are shown.

		Speed	
		$V_{\text{test}}/V_{\text{max}}$	
		0,72	0,94
Minimum Speed			
$V_{\text{min}}/V_{\text{max}}$		0,65	0,82
First Execute	T_{1start}		
	Time/Test time	0,147	0,105
	T_{1end}		
	Time/Test time	0,424	0,315
	β angle stbd [°]	20	20
	β angle port [°]	-20	-20
	Heading in T_{1end}		
	Time/Test time	-20	-20,1
Second Execute	Overshoot [°]	1,52	1,85
	T_{max heading}		
	Time/Test time	0,441	0,33
	T_{1start}		
	Time/Test time	0,435	0,325
	T_{1end}		
	Time/Test time	0,694	0,505
	β angle stbd [°]	-20	-20
Second Execute	β angle port [°]	20	20
	Heading in T_{1end}		
	Time/Test time	20,4	20,26
	Overshoot [°]	2,14	2,01
	T_{max heading}		
	Time/Test time	0,71	0,53

Table 5.2: Zig zag 20/20 simulation output.

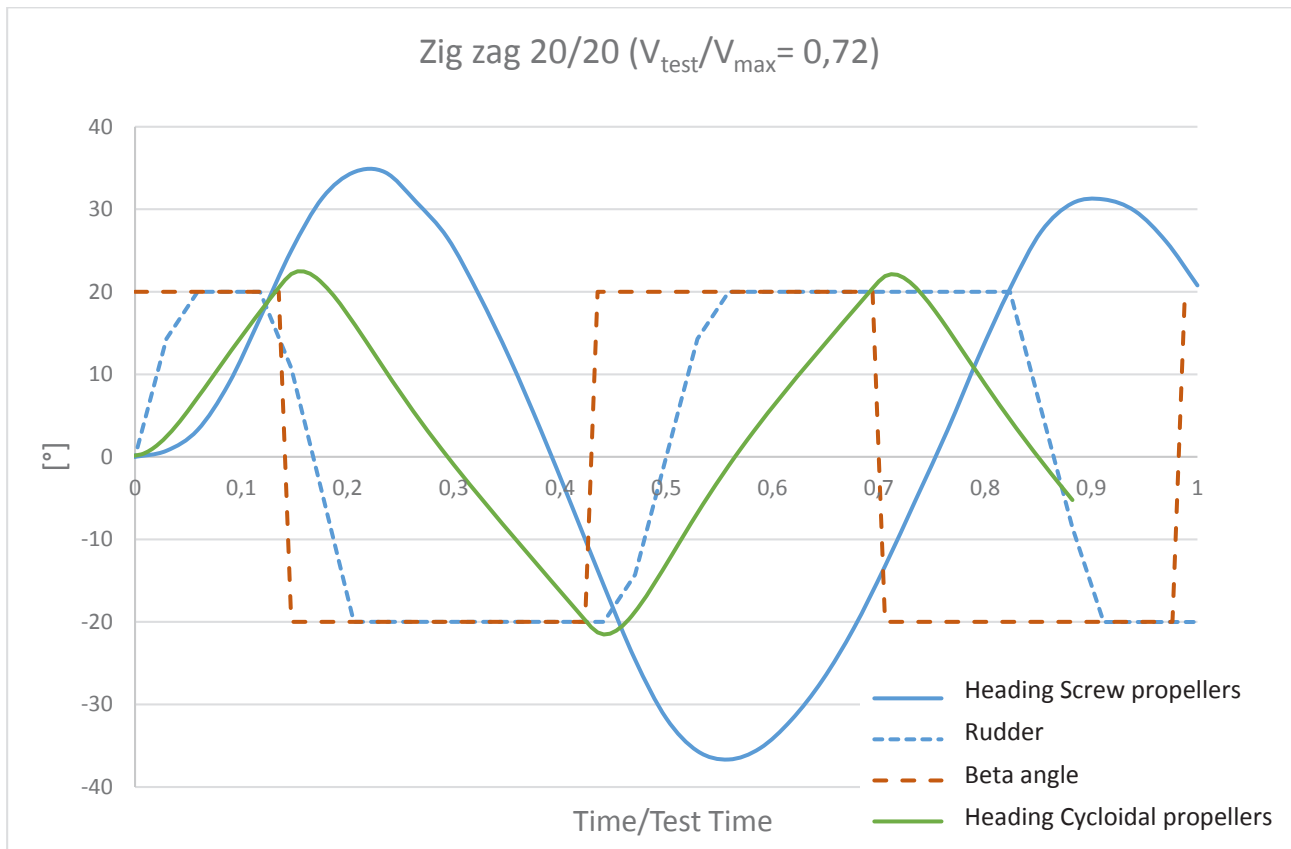


Figure 5.14: Zig zag 20/20 $V_{\text{test}}/V_{\text{max}} = 0,72$ – heading and rudder/Beta function of time in non-dimensional form.

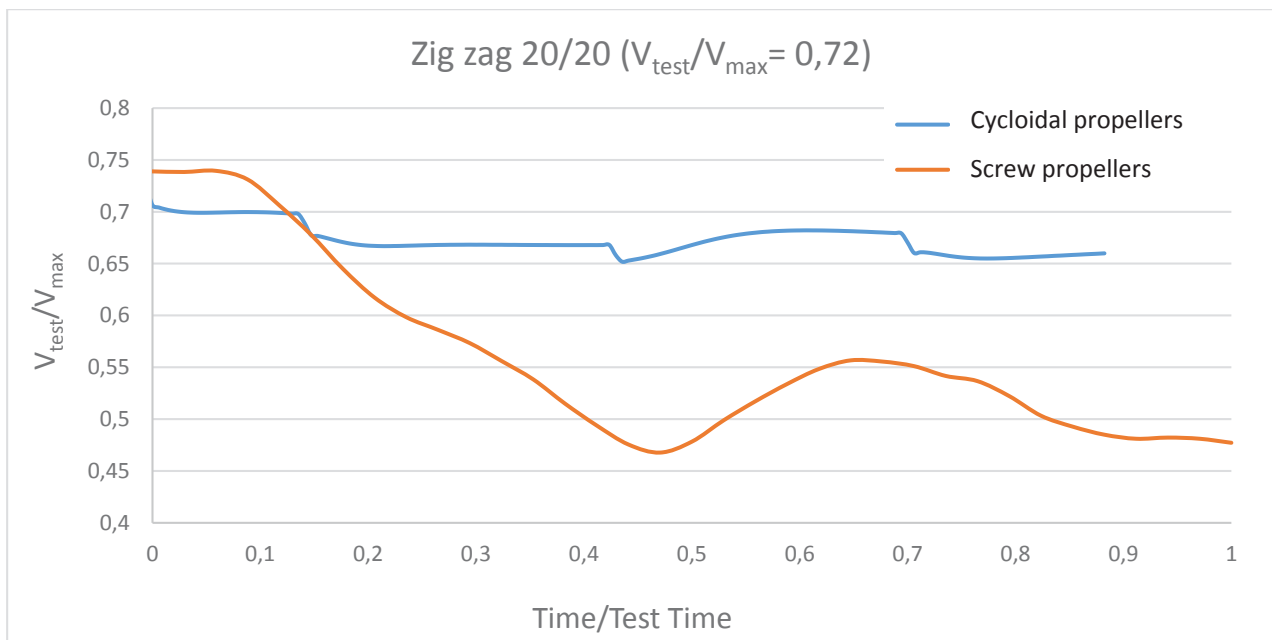


Figure 5.15: Zig zag 20/20 $V_{\text{test}}/V_{\text{max}} = 0,72$ – speed function of time in non-dimensional form.

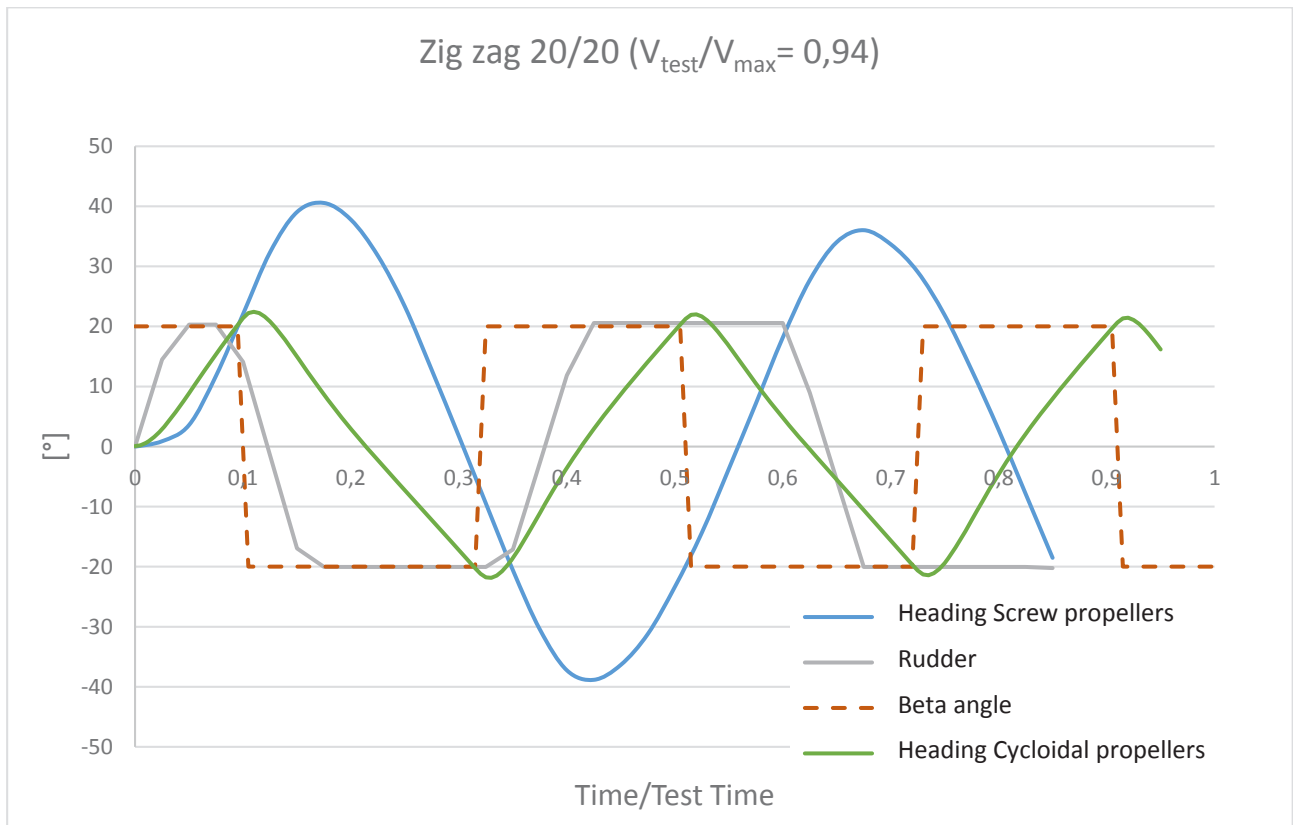


Figure 5.16: Zig zag 20/20 $V_{\text{test}}/V_{\text{max}} = 0,94$ – heading and rudder/Beta function of time in non-dimensional form

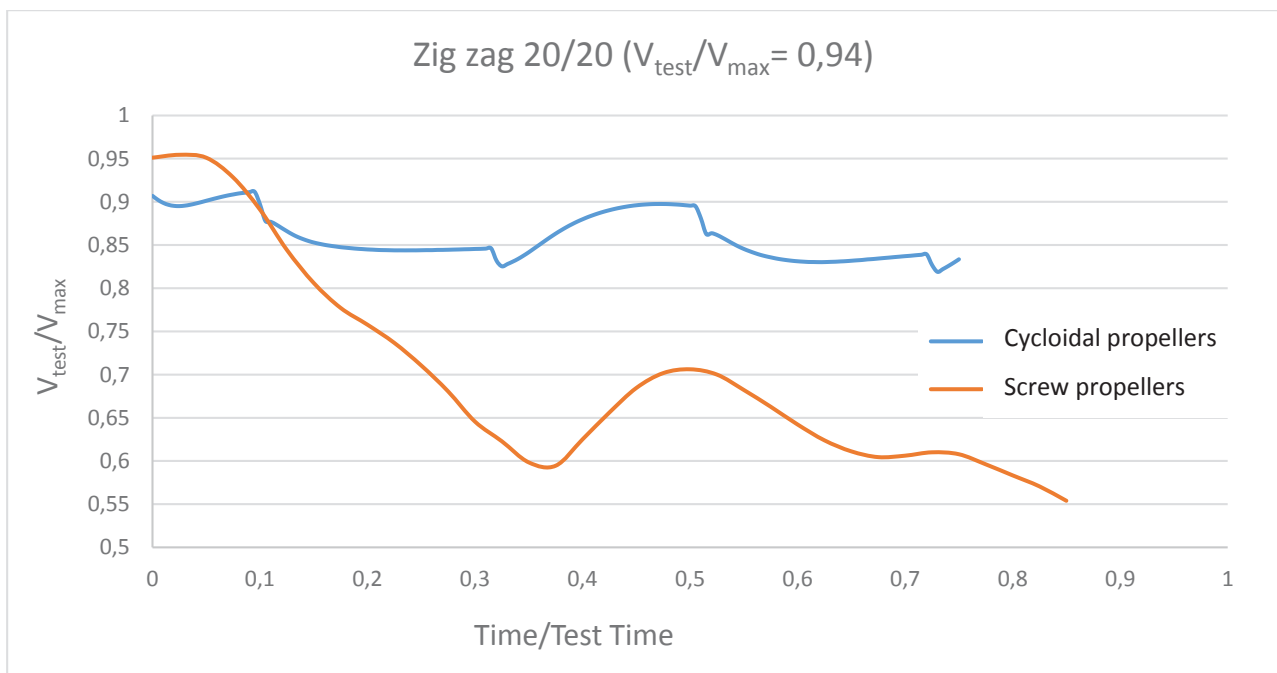


Figure 5.17: Zig zag 20/20 $V_{\text{test}}/V_{\text{max}} = 0,94$ – speed function of time in non-dimensional form.

From the zig-zag tests, there is a further confirmation of the fact that the speed reduction in the case of cycloidal propulsion is negligible. Moreover, it can be noted that the bow does not reach high peaks in the transients of the change thrust direction (overshoot) with a difference with respect to the traditional propulsion of about 10°.

5.4 *Crash stop (Full astern stopping test)*

The aim of a crash stop manoeuvre is to determine the distance necessary for the ship to pass from a given initial speed to zero speed with full astern stopping command.

The following table shows the results obtained with full astern stopping command corresponding to the maximum astern speed reachable (pitch 74% and 750 rpm).

Cycloidal propellers			Screw propellers	
<i>Speed</i> V_{test}/V_{max}	<i>Advance</i> $X_{midship}/L_{BP}$	<i>Time/Test time</i>	<i>Advance</i> $X_{midship}/L_{BP}$	<i>Time/Test time</i>
0,278	1,838	0,392	2,540	0,562
0,333	1,975	0,407	2,913	0,610
0,389	2,050	0,418	3,270	0,654
0,444	2,188	0,428	3,615	0,695
0,500	2,363	0,444	3,949	0,732
0,556	2,513	0,452	4,275	0,768
0,611	2,670	0,459	4,593	0,801
0,667	2,763	0,467	4,903	0,833
0,722	2,888	0,473	5,206	0,864
0,778	3,038	0,479	5,504	0,893
0,833	3,260	0,485	5,796	0,922
0,889	3,375	0,490	6,085	0,949
0,944	3,488	0,492	6,369	0,974
1,000	3,600	0,497	6,648	1,000

Table 5.3: Crash stop simulation output.

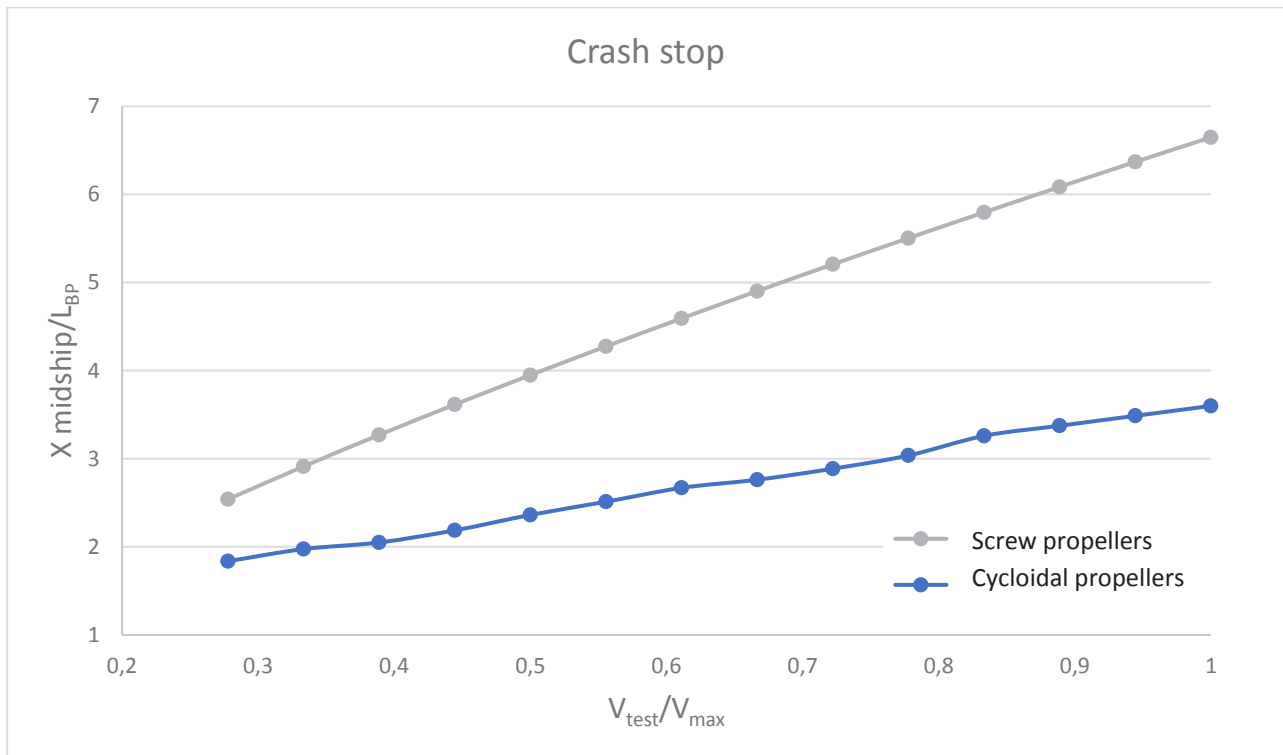


Figure 5.18: Crash stop manoeuvre – advance function of speed in non-dimensional form.

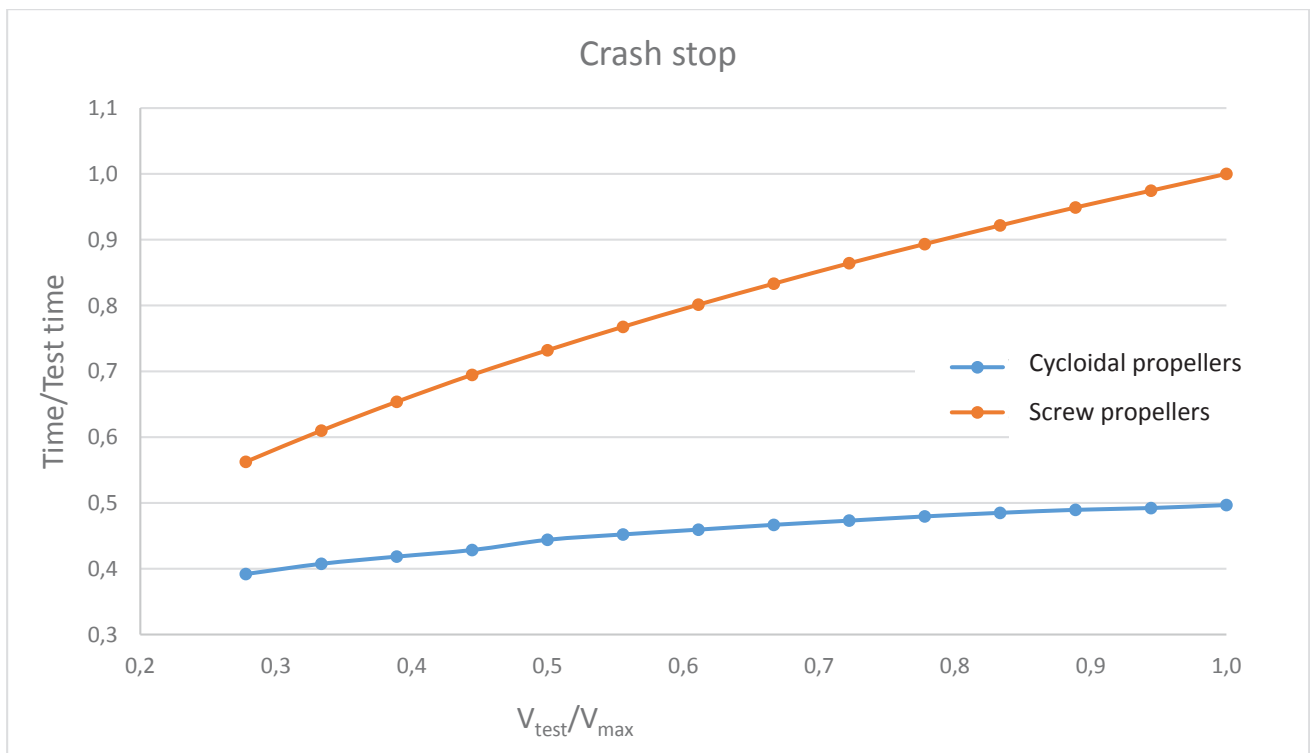


Figure 5.19: Crash stop manoeuvre – stop time function of speed in non-dimensional form.

Since the cycloidal propellers provide almost the same magnitude thrust in any direction, stopping times and distances are significantly different, especially at high speeds.

6 CONCLUSIONS

In this work we have presented a simulation model for both low and high speed manoeuvring of ships equipped with cycloidal propellers. The mathematical and numerical modelling of the free running behaviour of cycloidal propellers, in different operating conditions, is described without using a proper - but also demanding - fluid dynamics computation (CFD method is only used for the evaluation of the lift and drag coefficients of the single blade). The simulator has been calibrated by comparing simulation with experimental data. The core of the developed model is represented by the kinematical description and by the empirical correction factors that can be used for a preliminary performance estimation of several cycloidal propulsion units, characterized by different lengths and number of blades. In this sense, the simulator can be regarded as a parametric one, since we can obtain open water diagrams, not available in open source since the manufacturers do not disseminate this kind of data, for a wide range of cycloidal propellers only changing blade length, rotor diameter and number of blades.

As it has been already mentioned, the main point of strength of the proposed simulation model comes from not having to calculate the propeller fluid dynamics, avoiding demanding computations that would not allow an effective simulation of the whole propulsion plant, but obtaining, however, realistic and reliable results even in terms of real time simulation and evaluation of transient behaviour.

Even if the empirical coefficients used in this work should be validated for the different kinds of existing cycloidal propellers, the output of the present work gives the University of Genoa the availability of a new simulation tool that can be used for a wide range of subsequent applications. The comparison between a traditional propulsion system and the cycloidal one has been carried out, at first instance, in terms of DP capability plots. After that, the obtained results have been validated by means of a further dynamic analysis: the outcome of the simulation shows and confirms the better performance of cycloidal propulsion during dynamic positioning manoeuvres in different operating situations.

The same response has been obtained for manoeuvres at cruise speed: when equipped with cycloidal propellers, the ship gives better results in terms of times and distances needed to carry out the manoeuvre.

Summing up all the obtained results, it can be stated that the developed simulator can reproduce the dynamic behaviour of the ship in a reliable way.

With the appropriate insights concerning the correction coefficients and therefore a dedicated hydrodynamic analysis, the isolated simulator could be used to reproduce the open water behaviour of other kinds of cycloidal propulsors for which the literature and the industry provide a very few data. Right because the lack of data, the implemented simulator could be useful to predict the behaviours of the ship during the design phase in terms of general performances, forces generation, response times and evaluation of fuel/energy consumptions. Moreover, the simulator could be also used to implement a training platform for personnel.

As regards future developments, a first step should include the integration of the hydrodynamic interaction between the hull and the cycloidal propeller and vice versa, as well as the calculation of the hydrodynamic resistance of the cycloidal propellers intended as hull appendages.

Further improvements could include the study of different engine sizes, considering thermodynamic engine models coupled to developed model, in order to better analyse the engine-propulsion dynamics (especially during transient phases).

Finally, the completion of the work done up to now would be obtained through the validation of the simulation model by means of test data of a cycloidal propulsion system installed on board a real ship.

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