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Development of a dynamic measuring apparatus for AUV in towing tank experiments

Tesi di Laurea Magistrale in
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Mirco Pozzoli, 966949

PoliMi Advisor:
Prof. Paolo Schito

TU Berlin Advisor:
Dipl.-Ing. Matthias Golz

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Abstract: Underwater unmanned vehicles are nowadays cutting-edge technologies that will find multiple applications in the near future. They permit the exploration of the deepest oceans, widening the possibility of research in the maritime field. This thesis is part of the MUM research project, whose ultimate goal is the development of a new class of modular vehicles. The construction of an extra-large autonomous full-scale submarine is planned for 2025. For the design phase, the hydrodynamic optimization of its shape is a requirement to achieve excellent manoeuvrability. This can be carried out starting with towing tank tests. The innovative design of an automized measuring apparatus for the execution of towing tank experiments with AUV is presented here. It allows the dynamic control of the model in five degrees of freedom while measuring reaction forces and torques. First, the mechanical design is introduced, including the selection of actuators and sensors and their implementation into the system. PLC connections and the first development stages of the control algorithm are then described. The construction phase of the apparatus has started and the creation of a mock-up has proved the functionalities of the system. Initial results are promising and the measuring platform is expected to meet the project requirements, allowing the evaluation of the hydrodynamic parameters.

Key-words:
UUV, XLUUV, MUM, Dynamic testing platform, Towing tank experiments, Hydrodynamic parameters

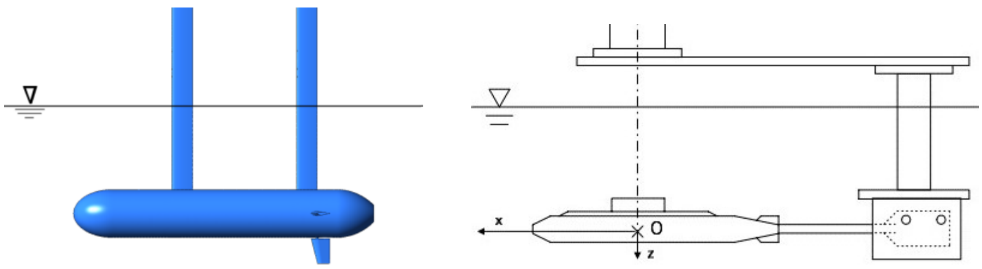
1. Introduction

This paper describes the design phase and the current implementation state of an innovative dynamic testing platform for autonomous underwater vehicles (AUV) in towing tank experiments. An AUV is an unmanned vehicle able to perform multiple tasks without real-time control from a human operator. It can operate in risky environments and in the deepest oceans. Nowadays, the employment of this class of vehicles is arising thanks to the improvements they introduce in collecting superior quality data or performing assignments that were not feasible with old technologies. They can be used in different fields, from military to scientific applications, for example, to perform geophysical data collection, deep water mining, seafloor mapping, ocean exploration, underwater pipes maintenance, offshore constructions management and many other related activities.

With all these possible applications in mind, manoeuvrability is an important aspect to consider in the designing phase. Unlike a surface ship, a submarine can move in all six degrees of freedom, and the required motion stability may differ between vertical and horizontal planes [18]. Uncertainties in the AUV dynamics are typically due to the hydrodynamic effort involved in high-amplitude or high-rate manoeuvres [10]. Therefore, manoeuvrability is one of the major aspects affecting the efficiency of underwater operations [9]. By analyzing the equations of motion, it is possible to recognize the details of the control and the stability of the specific AUV. Reaction torques and forces are expressed, in the equation of motion, as Taylor series as a function of kinematic parameters, also called hydrodynamic coefficients. Their complete understanding enables a better design, better control, and optimal path planning of autonomous underwater vehicles in deeply submerged conditions [17]. Theoretical models, numerical methods and computational fluid dynamics (CFD) simulations are deployed nowadays to evaluate the hydrodynamic performances in many scenarios, commonly considering an infinite domain and with the hypotheses of vehicle operating far from the free surface [12]. The typically encountered errors resulting from numerical simulations are related to inaccuracies in physical models and numerical errors. Experimental tests, with the consideration of the scale effect, are therefore still considered one of the most reliable methods to estimate the hydrodynamic parameters. Thus, allowing the validation and improvement of the mathematical models [10, 14]. Nevertheless, practical methods involve many sources of errors and uncertainties; a measure can be affected by errors in the test setup, scale effects, model fabrication inaccuracies, calibration errors, tank wall effects, and others. Deploying both methods, and comparing the results, seems to be the optimal solution [7].

A towing tank is a water basin, usually a few meters wide and hundreds of metres long, with rails on both lateral sides on which a towing carriage runs. The towing carriage is generally connected to the towed model by a measuring platform, such as the one discussed in this paper. The carriage drives the model in the forward motion, without any propulsion force from the latter during the experiment. The real-time measure of exchanged forces and torques, position, velocity and acceleration, and the post-processing analysis will enable the evaluation of the parameters. After the testing phase, the optimization of the mathematical model is carried out. As a result, it is possible to design an improved control system and a more efficient aerodynamic shape.

In the literature, it is possible to identify two main procedures to secure the AUV to the carriage of the towing tank. One possibility is to use one or two vertical struts to connect the upper surface of the vehicle to the carriage (*Figure 1a*), the resultant reaction forces are measured with a load cell which can be connected between AUV and strut or at the attachment strut-carriage; alternatively a sting mechanism is adopted (*Figure 1b*). In this last case, a long shaft connects the downstream end of the model to a vertical strut placed at the back of the vehicle. By doing so, the connection does not disturb the surrounding flow. In this last configuration, load cells can be placed either inside the model between the model itself and the sting or at the connection strut-carriage. The presence of struts influences the streamlines of the water surrounding the AUV. The sting solution is therefore preferred as it generates a lower disturbance. However, it is mainly suitable for torpedo-shaped vehicles and may result in the inability to investigate the propeller when it is located at the end of the structure [7].



(a) Connection with struts [7]. (b) Connection with sting [16].

Figure 1: Types of AUV-Carriage connection.

Most towing tank experiments, at the state of the art, do not involve a dynamic movement of the vehicle during the test to evaluate its hydrodynamic parameters. A fixed orientation is instead kept in each experiment. Therefore, AUVs are under-actuated and do not achieve the final manoeuvrability of the real-scale model or operate at a significantly lower speed. The only facility published in literature which can be employed to perform dynamic tests with AUV in a towing tank is the Marine Dynamic Test Facility (MDTF) at the National Research Council Canada (*Figure 2*). It consists of a single platform able to control the motion in six degrees of freedom. The device can manoeuvre a vehicle up to 6 m long, which is secured to the apparatus employing the sting configuration. Reaction forces and torques are measured using a six components balance. The control system allows the MDTF to impose all types of motions, including pure or combined manoeuvres, as discussed in by Mackay et al. in [11]. Due to the adopted sting configuration, the MDTF platform is mainly suitable for submarines with length as the principal dimension.

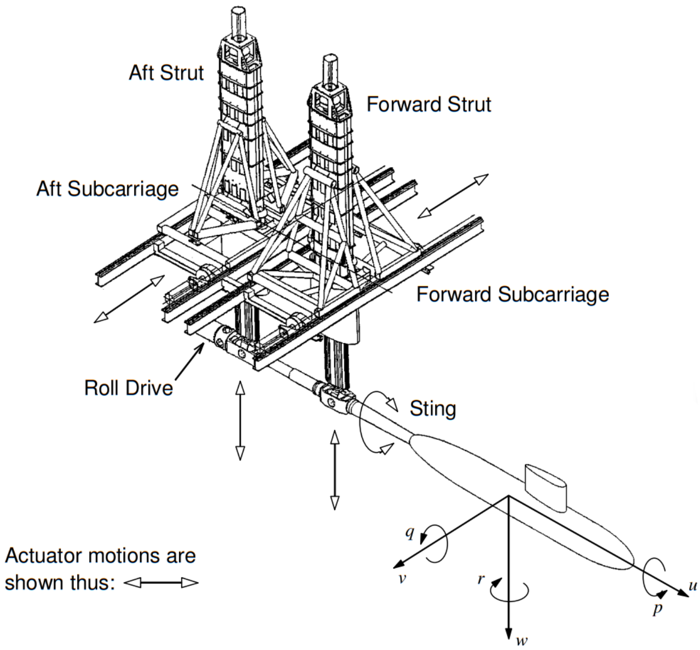


Figure 2: MDTF General arrangement [11].

1.1. Research project and motivations

The presented thesis work is part of the MUM (Modifiable Underwater Mothership) research project funded by the German Federal Ministry for Economic Affairs and Energy. The realization of a new class of modular unmanned underwater vehicles is the aim of the study. These vehicles will operate below the water surface in almost constant environmental conditions, avoiding the uncertainty of the weather in oceans, without onboard operators and therefore using highly automated technology. The innovation comes with a modular design which leads to cost-efficiently customization of the vehicle for every target mission. MUM is a modular extra-large unmanned underwater vehicle (XLUUV) that intends to solve multiple assignments in the underwater field. It consists of basic modules necessary for the proper vehicle operation, and specific task modules that can optionally be placed on board to enable a mission-objective configuration. A few examples of the applications for which MUM can be used are sea mining, offshore energy, payload transport and, in general, maritime science tasks involving exploration missions and stationary jobs in the deep sea.

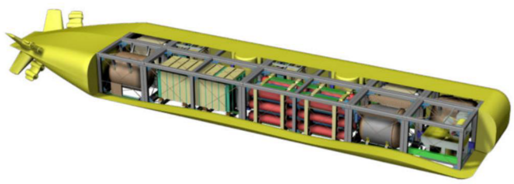


Figure 3: Example of MUM vehicle configuration [2].

The reusability of the modules leads to a reduction in costs and a faster development cycle compared to the conventional vehicle concept, considering that it will be possible to easily integrate new mission modules for future needs. The vehicle will be powered by an air-independent fuel cell propulsion system making it completely emission-free and allowing its use in ecologically protected areas. Continuous operations at depths down to 5000 m for a time range of several weeks are foreseen [3].

The initial research project MUM1, 2017-2020, demonstrated the project feasibility with the realization of a small-scale functional model (1:5), named Monique, now available at the Technische Universität Berlin for the experimental phase. It consists of six modules with overall dimensions L 3.6 m x W 1.2 m, a weight of 900 kg and a maximum operating depth of 500 m. The research project MUM2, 2021-2025, has the purpose to develop and build a full-scale demonstrator with a length of approximately 25 m [15]. The presented thesis is carried out in the framework of the MUM2 research project at the Technische Universität Berlin, Institute for Land and Sea Transport, department of Design and Operation of Maritime Systems. The project network, coordinated by ThyssenKrupp Marine Systems GmbH, also consists of ATLAS Elektronik GmbH, EvoLogics GmbH, and the University of Rostock (Institute for Automation Technology).

The aim of the present work is the design and construction of a dynamic measuring platform for underwater vehicle tests in the towing tank facility of the technical university of Berlin. The main requirement for the apparatus is the capability to drive an AUV model to produce independent or coordinated motions in six degrees of freedom. In particular, it will be deployed to perform experiments with scaled-down models of the MUM2 demonstrator to evaluate its hydrodynamic parameters and achieve the profile optimization of the vehicle. The final shape for the full-scale MUM2 demonstrator will therefore be designed. Evaluation of these parameters will also allow for improved vehicle control and mission management. Considering the large size of the full-scale AUV, and the wide range of manoeuvring operations required for different assignable tasks, a thorough hydrodynamic analysis is necessary. Thus, the work aimed to design a measurement apparatus allowing the evaluation of forces and torques exchanged by the model with the water during towing tank experiments. Significance has been given to obtaining a universally usable device which can also be employed for other tests and research projects. For example, it is planned to deploy the platform for the AUV of the research project CIAM (Comprehensive Integrated and Fully Autonomous Subsea Monitoring) carried out at TU Berlin.

2. Problem statement

The apparatus has to be mounted on the towing tank carriage of the research institute for hydraulic engineering and shipbuilding at TU Berlin (Versuchsanstalt für Wasserbau und Schiffbau - VWS). The large-scale test facility is equipped with a 250 m long, 8.1 m wide and 4.8 m averagely deep towing channel making it Europe's third largest water towing tank [8]. The maximum reachable speed is 12.5 m/s with an acceleration of 1 m/s^2 and a deceleration of 3 m/s^2. The maximum carriage load capacity is instead 2 t in the vertical direction (lift) and 1 t in the horizontal one (drag). For underwater vehicles, the blockage setup of the model must also be analyzed. To avoid the free surface effect the model has to be sufficiently submerged, but at the same time far enough from the bottom and the lateral walls of the basin to avoid external interaction. Several experiments must be performed at different depths and speeds to find the optimal combination that leads to the possibility of neglecting the surface effect on the hydrodynamic coefficients [7]. An AUV model has generally neutral buoyancy and it will be first placed in the water and then mounted to the measurement platform. Consequently, the structure does not have to handle the vehicle with its weight in the air.

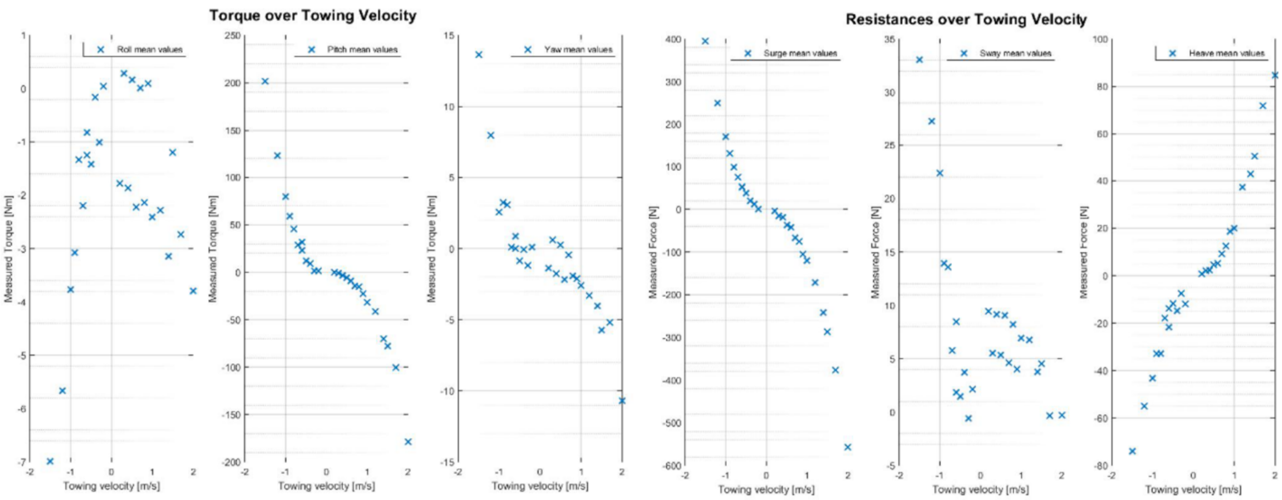


Figure 4: Measured loads acting on Monique in the direction of travel, averaged values [2].

Tests conducted as part of the MUM1 project showed that specific measurement equipment had to be designed. Small forces and large torques were simultaneously occurring during the experiment. By using a single multiaxial load cell, this difference in range made it difficult to observe forces and torques with the same accuracy. Therefore, it was desirable to implement a device able to decouple torques and forces and measure them separately with appropriate instruments that could provide optimal accuracy and precision. The measurement results of the towing tank tests conducted on Monique are depicted in *Figure 4*. The experimental setup involved the functional model mounted to the carriage in a fixed orientation through a 6-axis force and torque sensor able to withstand up to 10 kN and 750 N m (Model K6D110 by ME-Meßsysteme GmbH [19]). Only a minor percentage of the available force measurement range was used. As a result, a poor resolution for the values was obtained. Experiments also pointed out the need for a high-stiffness apparatus, enabling the reduction of the vibrations during the test that may affect the measurement campaign. The stiffness of the support system was also a limiting factor in the maximum submergence distance of the model.

These initial hydrodynamic tests allowed the implementation and validation of the mathematical model for Monique and the preliminary shape design for the MUM2 demonstrator. Starting from the analytical model it was possible, through CFD simulations, to foresee the maximum expected loads in the towing tank experiment to be used as input for the design of the testing platform. In addition to the base case, two orientations of the vehicle have been considered as reference scenarios since have led to the maximum expected load values. The loads resulting from CFD simulations are shown in *Table 1.* These values were obtained by increasing the output numbers from the simulations by a safety factor of 1.5; this is because the comparison between towing tank measures and CFD simulation on Monique for case1 showed an initial underestimation of the loads.

- **Case1:** Pure surge motion, without any roll, pitch or yaw angle. Speed of 2 m/s;
- **Case2:** Forward motion with 20° yaw angle, without any roll and pitch inclination. Speed of 1 m/s;
- **Case3:** Forward motion with 20° pitch angle, without any roll and yaw inclination. Speed of 1 m/s.

	Fx [N]	**Fy [N]**	**Fz [N]**	**Mx [Nm]**	**My [Nm]**	**Mz [Nm]**
Case1	600	35	80	7	200	15
Case2	300	400	120	270	75	360
Case3	450	30	1200	15	900	15

Table 1: Reaction loads of Monique obtained from CFD simulations.

A further aspect taken into account is the presence of the sting or struts that influences the flow surrounding the model, generating a change in the streamlines of the fluid; all the measured parameters will be affected by their presence. By applying the superimposition principle, it is possible to obtain an approximation of the parameter values for the AUV model without the influence of the added objects. They can be evaluated by running a test with struts connected to the model and then subtracting, from the measured values, the ones acquired from another test run with the same dynamic parameters but without AUV. Besides the struts, priority has been given to placing pieces of equipment outside of the water or inside the model. This is to limit the interaction between any additional device and the fluid. The small-scale demonstrator will be an empty body allowing components to be placed inside the shape and reducing flow disturbances. Finally, to achieve the desired flexibility of the testing platform and allow its deployment with different vehicles, an easy interchange of load cells was required. This is to fit the force range to the specific application. Moreover, a quick connection between the testing apparatus and the vehicle was considered.

3. Materials and methods

An initial investigation of the state of the art provided an overview of the types of measurement apparatus used for towing tank tests with small-scale underwater vehicles. The possibilities of adaptation and combination of the published concepts to achieve the goals of the thesis, taking into account the project constraints, has been the starting phase of the work. The ability to adjust the angle of attack, changing, in order of importance, the yaw, pitch and roll angles, was considered the most relevant requirement. Having to work in a wet environment, even if not with salty water, corrosion resistance has been a substantial consideration for the overall design. Solutions with fewer devices added underwater were preferred for two reasons: to limit their influence on the hydrodynamic properties of the model and to avoid the need for the highest IP protection class for the electrical components. An analysis of the differences between using one or two struts to attach the vehicle to the platform has been carried out. The influence on the hydrodynamic properties of one single cylindrical body is significantly lower than the interaction of two bodies with the AUV. This behaviour also occurs when structural elements are embedded in a NACA-shaped profile. However, vibrations can significantly be reduced by employing two vertical struts. This generates better model stability allowing the achievement of higher speeds and accelerations [4]. Listed below are the concepts that emerged from the initial design phase, which were selected for further analysis.

- **Steward platform or its redesign.** This parallel manipulator based on linear actuators can control the movement in six degrees of freedom converting torques into axial forces. The fixed base of the device can be attached upside down to the towing carriage. Load cells can be placed as a link between the actuators and the bottom moving plate. The steward platform operates outside the water and is connected to the model using a single vertical beam. With this device, the fulcrum of movements is placed above the water level and away from the centre of mass of the model. As a result, reaching wide pitch angles could drive the vehicle outside of the water. A redesign of the device has been carried out to overcome this problem. The outcome involves a rotary platform, allowing variation in yaw angles, fixed on one side to the carriage and connected on the opposite side to a torque sensor. One flange is mounted on the top surface of the vehicle, and another flange is screwed into the torque sensor. Four cylindrical struts realize the connection between the two plates. The link between the struts and flanges is realized with ball joints. Axial load cells are placed in the upper part of the structure, between the joints and the metal rods. Three struts are connected to linear actuators, while the fourth is placed in the centre of the flange and acts as a fixed fulcrum of rotation. This solution makes it possible to keep the centre of movement close to the model, avoiding issues while moving the vehicle. However, it has been considered not rigid enough and with output measurements affected by the many parts interacting with the flow.

- **Robotic arm.** This concept involves the deployment of an industrial robotic arm mounted upside down on the carriage with the tip connected to the vehicle employing a six-axis load cell. Loads evaluation can also be carried out from the analysis of the electric data of the motors in the joints. The high cost of a waterproof device and the large size and weight required to move the model led to the discarding of the concept. Moreover, the arm length was not long enough to place the tested model at the target depth to avoid the free surface effect. Delta robots have also been considered but proved not suitable for high loads.
- **Adaptation of a testing platform under construction at the department of Dynamics of Maritime Systems at TU Berlin.** This concept is the one selected for the aim of the thesis, and it will be described in detail in this paper. It is based on a project carried out at the Institute for Land and Sea Transport FG Dynamics of Maritime Systems (DMS) to study the behaviour of a ship model, which is free to move and subjected to waves generated in the towing tank basin.

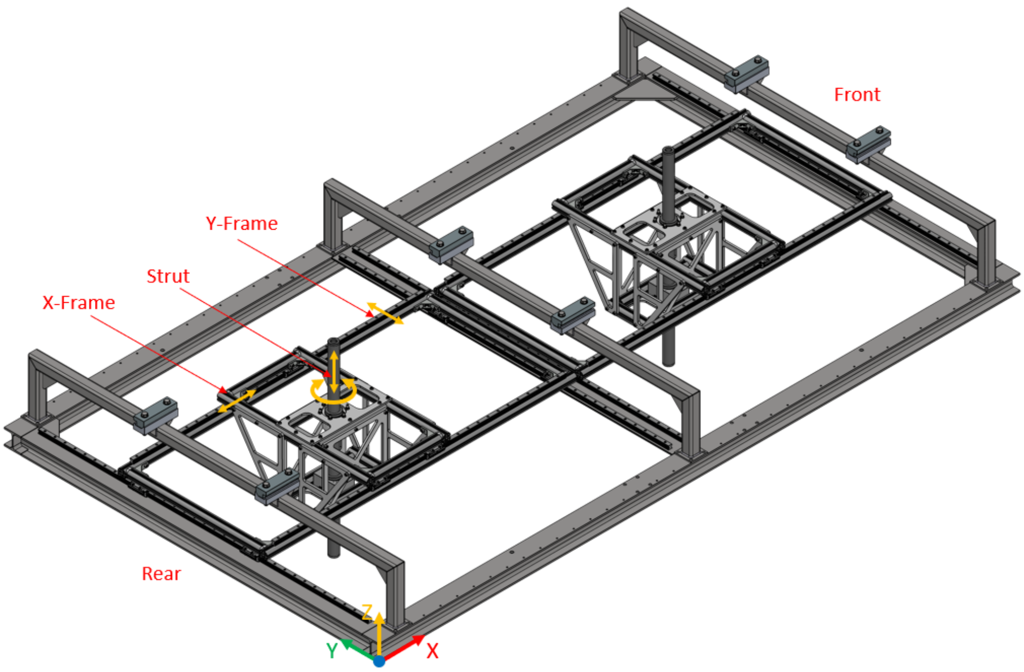


Figure 5: Testing platform designed by DMS department of TU Berlin.

The most promising concept, also considering the ad hoc design already carried out for the specific application in the VWS, has been the adaptation of a testing platform under construction at the DMS department at TU Berlin. The CAD design of the apparatus is depicted in *Figure 5*. The department aims to realize a planar motion mechanism for ship experiments where the model of the boat is free to move in all directions depending on the interaction with water and waves. The sliding frames are kept in the equilibrium position using springs fixed to the non-moving parts of the structure. Displacement sensors are used to evaluate the movement of the components. The design involves a rectangular I-beam frame to be mounted on the carriage, on top of which two identical sliding structures can move. Each chassis is composed of an X-Frame and Y-Frame, allowing two degrees of freedom, plus a central strut which is free to rotate around its axis and to move in the vertical direction. The allowed strokes are 1200 mm for the X-Frame and 1100 mm for the Y-Frame. The structures are realized with carbon fibre profiles and aluminium plates since reduced inertia in the system was a requirement of the investigation. Needle bearings carts run on aluminium rail guides and guarantee optimal smooth running with a reduced friction force. The spar is placed inside two linear ball bearings, with angular error compensation, that allow the strut to move in the vertical direction and rotate. The bearings are fixed to the X-Frame through an aluminium structure.

The apparatus meets the requirements of this thesis since it can be used to move the vehicle, if connected with the struts through a suitable design solution, in five degrees of freedom:

- By moving each X-Frame of the same distance, the surge movement is executed;
- By moving each Y-Frame of the same distance, the sway movement is executed;
- By moving each strut at the same vertical position, the heave movement is executed;
- By moving independently front and rear sliding frames, the yaw movement is executed;
- By moving independently front and rear struts in the vertical direction, the pitch movement is executed;

The roll movement can instead be implemented as a further joint in the vehicle-struts connection solution. Therefore, the goals of the mechanical design were to analyze possible ways of automatizing the apparatus and to identify the optimal technique to measure reaction forces and toques together with the displacement of the moving frames.

3.1. Forces analysis

Starting from a platform already designed and in the implementation phase, the conduction of a stress analysis allowed the evaluation of the maximum loads withstandable from the apparatus, considering the different final applications. The steel frame resulted to tolerate loads well beyond the maximum expected from the test, and the same for the carbon fibre profiles thanks to their performing mechanical properties. The weakest components of the system turned out to be the carts of the aluminium rail guides. The verification procedure of the latter takes into account two parameters: lifespan and safety factors. Considering the particular layout of the sliding guides, it was only possible to perform the verification starting from the applied loads, aiming to calculate the two verification parameters. Therefore, to obtain the maximum loads applicable to the structure, it has been decided to start from the standard load cases reported in *Table 1*, and after having translated the loads to a single strut, gradually increase them. The iteration was stopped when the results of the verification were no longer consistent with the minimum values reported in the product catalogue. Assessing how far the forces can be increased, with a positive verification of the structure, allows determining the minimum scale factor for the testing model to reduce the scale effect when determining the hydrodynamic parameters. The outcome of this analysis is reported in *Table 2*.

Three verification scenarios were identified:

1. Complete verification, both parameters are in the acceptable range of values;
2. Positive verification on the safety factor, meaning the rails can withstand the loads, but negative verification on the lifespan;
3. Negative verification, both parameters are out of the catalogue range.

	-50%			Base load case			+50%			+100%		
	Case1	Case2	Case3	Case1	Case2	Case3	Case1	Case2	Case3	Case1	Case2	Case3
X frame												
Y frame												

Table 2: Outcome of the analysis of the forces: green scenario 1, yellow scenario 2, red scenario 3.

The final purpose of the platform is to carry out tests, the working operations will not be continuous, and the equipment will be used only for a short period of time per year. Therefore, in accordance with the manufacturing company of the rails guides, the second scenario was considered acceptable, even if not ideal. If such loads were ever to be reached, it would only be for short instants of the test time and not as constant stress on the structure. The lifetime can be lower than the recommended one, as long as the load safety factor is greater than the minimum value stated in the catalogue. The maximum loads that can be applied on the bottom end of a single strut, corresponding to loads of the base case transported to a single strut, and then increased by 50%, are reported in *Table 3*. Moreover, they consider the increased length of the spar to place the underwater vehicle at optimal depth, avoiding bottom and surface disturbances, and the suppression of the torques My and Mz generated by the designed connection device that is explained in *Section 4*. Those are the loads considered while carrying out the in-depth mechanical design for the dimensioning procedure and the actuator and sensor selection.

	Fx [N]	Fy [N]	Fz [N]	Mx [Nm]	My [Nm]	Mz [Nm]
Case1	450.0	42.5	455.5	5.5	0	0
Case2	225.0	690.0	187.5	202.5	0	0
Case3	337.5	38.5	2054.25	11.25	0	0

Table 3: Maximum loads allowed at the bottom end of a single strut.

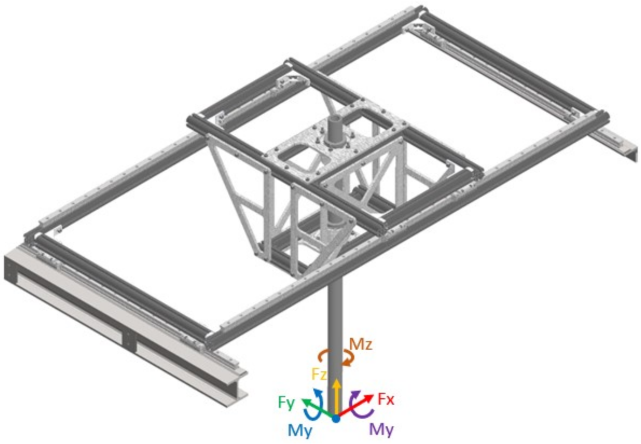


Figure 6: Loads representation on a single strut.

3.2. Actuators selection

One of the challenging aspects of the project has been the selection of the actuators. Intending to automate the movement of the sliding frames, constraints concerning the available space, system loads and needed strokes have been considered. The maximum actuation force was the first parameter used in the selection; in addition to the forces reported in *Table 3*, also weights of the components were considered due to the inertia they generate during the movements. Five actuators were required:

- Z direction: two devices to move the front and rear struts.
- Y direction: two devices to move the front and rear sliding frame.
- X direction: one device to move the front sliding frame; the rear one will move consequently depending on the position of the other frames.

The movement along the basin, in the X direction, is provided by the carriage of the towing tank facility. Therefore, the X actuator is an optional feature which can be mounted, for example, to experience accelerations starting from a constant speed. Eight 5.5 kW DC-motors are equipped to move the carriage train with computer-controlled velocity profiles of arbitrary shape.

Among the considered actuators, the most relevant ones were investigated in detail, contacting manufacturing companies asking for quotes, and configuring them for the specific system in analysis. Several features and functions were considered for each device, and the evaluation matrix reported in *Table 4* was created to assist the selection process. A brief explanation of the analyzed actuators is reported below.

- **Linear actuators.** By converting the rotary motion of the electric motors into linear movements, linear actuators can provide a clean motion with accurate precision and smooth position control. The forces provided were suitable for the applications, and the available strokes match the requirements. Forces measurements can be obtained by mounting a load cell between the piston tip and the sliding frame. The main issue is related to their dimensions. In fact, to achieve 1200 mm stroke, the device has a fixed cylinder of circa 1000 mm, which must be placed outside of the moving area of the frames. In addition, the piston would be overhanging for its entire length. Therefore, the load cell would end up being stressed with bending forces that it cannot withstand. The solution to overcome this issue was to place two additional carts in each moving direction, one on each side rail, connected with a carbon profile perpendicular to the moving direction. The latter is connected on one side to the piston and on the other side to the sliding frame through the load cell. In this way, the forces exchanged by the piston are carried by the added profile, and only the axial force is transmitted to the load cell, enabling its measurement. However, this limits the strokes, especially for the Z direction, where the space occupied by the cylinder was half the permitted travel.

- **Telescopic linear actuators.** To avoid long parts overhanging from the platform and to reduce the occupied space, telescopic linear actuators have been investigated as a possible solution. In particular, with this type of device, it is possible to achieve a stroke of 1200 mm with spatial requirements of only 500 mm. The conversion of the rotatory motion of the motor into the axial translation movement is achieved through two nested precision ball screws. This transmission leads to the highest precision in positioning and avoids repeat inaccuracies. However, such devices were proved too expensive and not corrosion resistant due to the special nitriding steel they are made of. Furthermore, although the occupied space for the Z direction would be smaller, the stroke would have been reduced by one-third.
- **Linear motion system with screw drive.** This linear unit consists of a support frame, guides and a drive. A carriage moves on lateral guides driven by a screw drive, which is connected to a motor at one end and terminated into a bearing on the opposite one. Linear drive systems with screw drive are deployed for applications requiring large forces and high precision. These devices meet the required strokes and provide the needed loads. They are unobtrusive in size, allowing easy mounting in the structure. The presence of a fixed frame on which the carriage is moving permits to obtain only axial forces acting on the load cell and avoiding reduction in the strokes.
- **Linear motion system with toothed belt drive.** This device is similar to the linear motion system with screw drive. The motion transmission is, in this case, achieved with a toothed belt. It is suitable for applications involving low and medium feed forces with low to medium precision. However, the belt cannot guarantee a stable position in case of impulse forces. Moreover, the need for a pulley on both sides increases the occupied space for the same stroke of a screw drive device. Finally, the motor needs to be mounted with the axis perpendicular to the movement direction leading to possible interference with the structure.
- **Scissor jack mechanism.** This type of device allows big strokes with a small occupied space. The mechanism is composed of a screw drive in a horizontal position that lengthens or shortens the horizontal diagonal of a parallelogram consisting of the linkages. Strokes and forces were suitable for the application, but the implementation of this type of actuator in the platform led to issues. First, the weight was too high, secondly, to obtain a purely linear motion, the moving plate has to be constrained in the movement. A possible solution to overcome the issue and to transmit only axial forces through the load cell is to add telescopic linear guides. However, for the lengthy strokes, a long screw driver is needed. The latter may end up impacting the frame of the platform during the movements.
- **Toothed belt/Chain drive system.** This solution involves connecting both ends of a chain, or belt, to the frame, generating a closed loop. By fixing a motor equipped with a pulley or a gear, it is possible to make the platform move. Allowed forces were at the limit of acceptance, but the solution permits easy implementation into the system and large strokes. However, the main issue was the difficulty in precisely measuring the forces, both in traction and compression, having a non-rigid body transmitting the motion.
- **Rack and pinion.** The last investigated solution consists of a motor mounted onto the sliding frame and equipped with a pinion. The latter, engaging a fixed rack placed on the lateral profile, converts the rotary motion into linear motion. This solution is robust, allows large strokes and can withstand big loads. Implementation in the system can be easily achieved, but the direct evaluation of the reaction force between teeth in contact would have been challenging.

In the decision matrix, the most relevant features have been stated and rated on a scale of 1-3, where three is the best. Weights have been assigned to each specification on a scale of 1-5, depending on how the feature is considered valuable for the final application. After calculating the weighted values and having summed up all the scores for each type of actuator, it was possible to identify the most suitable device for the dynamic testing platform object of this thesis. The linear motion system with screw drive has been selected to automize the measurement apparatus. An in-depth design was performed, aiming to incorporate the actuators into the system. Among different manufacturing companies proposing this type of driver solution, the selection was performed considering mechanical characteristics, prices, and delivery time. According to the required strokes, two different sizes of the KK-linear axis from HIWIN have been chosen, the model KK60 for the Z direction, and the model KK100 for the X and Y directions [5]. They are light and compact, with a positioning accuracy of 0.34\0.63 mm, and repeatability of 0.025\0.045 mm.

Servo motors proved to be the optimal solution for the target application. Speeds, accelerations and maximum torques had to meet the requirements of the actuators as well as the flange dimension to be matched to one of the solutions proposed in the actuator catalogue. The apparatus has to keep a fixed position during the test while big loads are transmitted to the actuators. When stopped, the motors cannot provide enough torque to withstand the loads, therefore, a built-in brake was considered a needed feature of the motors. Two models of servomotors, with the respective types of control drivers, have been chosen to match the different technical specifications of the two linear system sizes. In particular, motor SE 80-350-5-55-AK with controller C3-05 for X and Y actuation and motor SE 60-150-3-60-AK with controller C1-05 for the Z direction from item [6]. The selected motors are equipped with robust and precise rotary encoders to ensure reliable positioning of the system and optimal control of the actuator. Finally, for the transmission of motion from the motor to the actuator, metal bellows coupling devices have been chosen thanks to the reduced dimension allowing them to be mounted between different shaft diameters and with only a reduced axial space between the two shafts. The CAD drawing of the complete actuator assembly is depicted in *Figure 7*.

	Maximum forces and torques	Dimensions/Occupied space	Maximum allowed stroke	Ease of implementation	Ease of measuring forces	Position accuracy	Price	Delivery/Realization time	Total Score
Weights	*5*	*3*	*4*	*2*	*5*	*3*	*2*	*3*	
Linear actuators	3	1	2	2	3	3	3	2	19
Telescopic linear actuators	3	3	3	3	3	3	1	1	20
Linear motion system with screw drive	3	3	2	3	3	3	3	3	23
Linear motion systems with toothed belt drive	1	3	2	3	3	2	3	2	19
Lifting jack mechanism	3	1	1	1	2	2	2	1	13
Toothed belt / Chain drive system	2	3	3	2	1	1	3	3	18
Rack and pinion	3	3	3	2	1	1	2	2	17
Weighted scores									
Linear actuators	15	3	8	4	15	9	6	6	66
Telescopic linear actuators	15	9	12	6	15	9	2	3	71
Linear motion system with screw drive	15	9	8	6	15	9	6	9	77
Linear motion system with toothed belt drive	5	9	8	6	15	6	6	6	61
Lifting jack mechanism	15	3	4	2	10	6	4	3	47
Toothed belt / Chain drive system	10	9	12	4	5	3	6	9	58
Rack and pinion	15	9	12	4	5	3	4	6	58

Table 4: Actuators selection matrix.

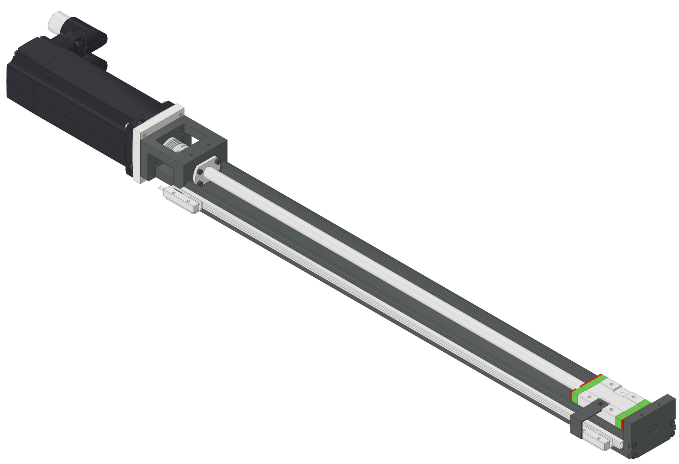
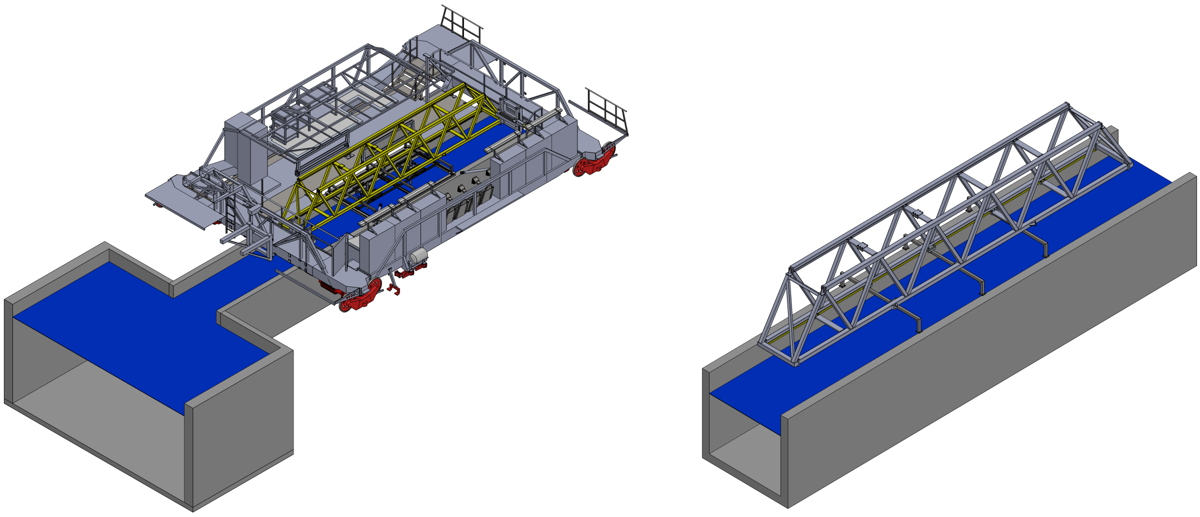


Figure 7: Actuator: screw drive, servomotor, adapter flange, bellow shaft coupling, inductive sensors.

4. Testing platform design

The first step to adapting the DMS platform to the goal of the project has been to increase the distance in the Z direction between the frame of the carriage and the platform. This modification was necessary to extend the vertical stroke for the Z direction and thus to widen the angle range for the pitch movement. In *Figure 8* CAD drawings of the towing tank carriage of the VWS facility at TU Berlin are represented. The frame of the carriage, equipped with rails for the quick mounting of the testing platforms, is shown in *Figure 8b*.



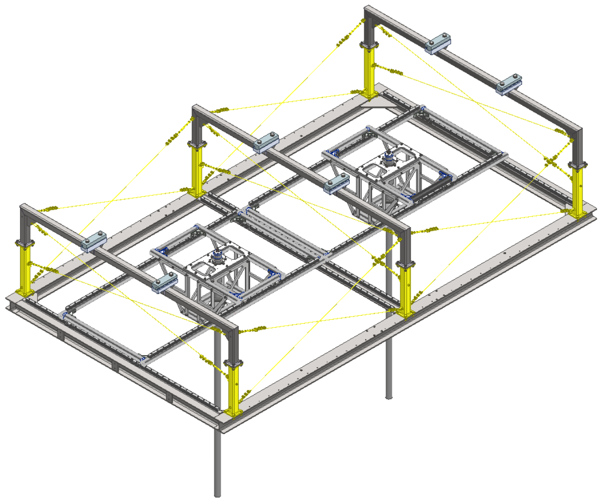
(a) In yellow the frame for testing platform attachment. (b) Attachment frame.

Figure 8: VWS towing tank facility at TU Berlin.

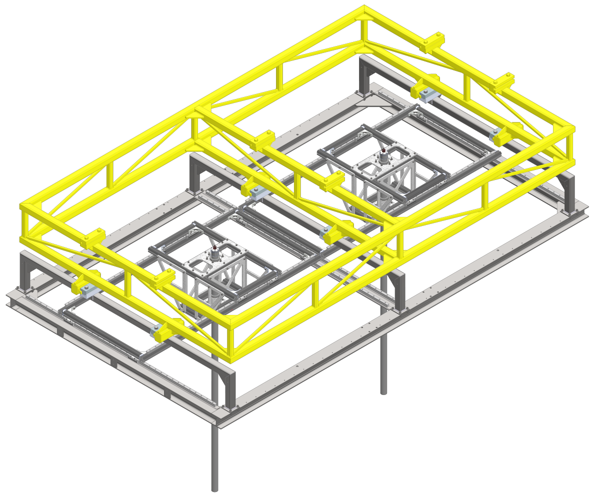
Initially, the maximum upper extension of the strut, which allows the complete movement of the sliding frames, was limited to 150 mm. Overcoming the latter would have led to interference with the upper fixed structure. By increasing this distance, without interposing any obstacle in the movement areas, it has been possible to increase the vertical stroke guaranteeing at the same time complete movements for the frames without any collision with the fixed part of the platform. Two diverse solutions have been presented (*Figure 9*), differing in ease of mounting, weight, stiffness and ease of realisation. The height of the carriage can be adjusted; the maximum distance between the frame itself and the testing platform is limited by the harbour section of the towing tank. The latter corresponds to a portion of the basin where the width of the water channel is narrow, to allow workers to operate on the carriage (i.e. for the mounting operations for testing platforms). Therefore, by setting the frame to the maximum height and considering a safety distance between the bottom of the structure and the concrete side walls, it has been measured a maximum extension distance of 550 mm.

- The solution represented in *Figure 9a* involves six extension flanged beams to be mounted in between the already designed U-shape attachment frame and the rectangular I-beam structure. Seven crossed-wire tensioning systems ensure the stiffness of the construction. The platform, in this case, would have been mounted using the clamping system already installed on the U-shape frame leading to a light, easy-to-mount, cheap and easy-to-build solution.
- The solution represented in *Figure 9b* involves a customized welded frame to be placed as an extension of the upper carriage chassis. Six clamps are welded on the top transversal beam of the structure, allowing it to be attached to the carriage rails. Short sections of rails are welded on the lower part of the frame allowing a quick mounting of the testing devices. The frame has bulky dimensions and weight, but it guarantees a high level of stiffness in the overall structure.

The concept in *Figure 9b* was chosen. It has been considered the optimal solution mainly due to the increased stiffens provided to the structure and the allowed quick mounting procedure. It is possible to fasten the testing apparatus without disassembling the U-shape attachment frame from the I-beam structure. Instead, this modification would have been necessary to fix the extension flange beams whenever there was a change in the testing platform. This operation could affect the measurement procedure by introducing errors due to slight mounting misalignment of the assembly.



(a) Tensioning system solution.

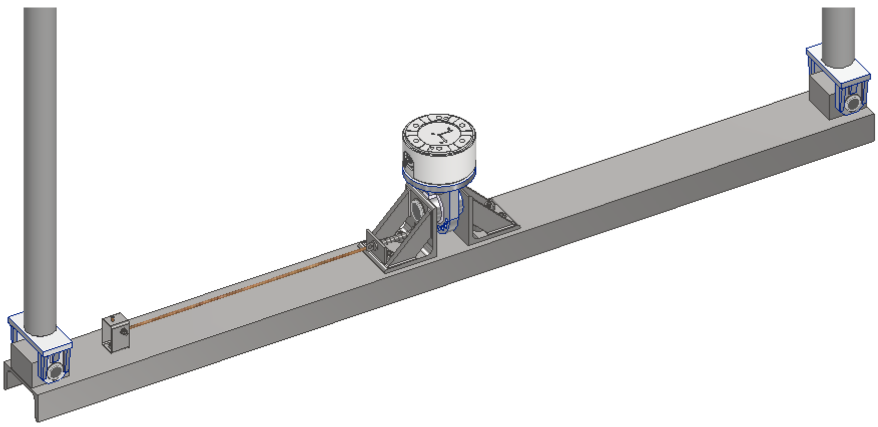


(b) Welded frame solution.

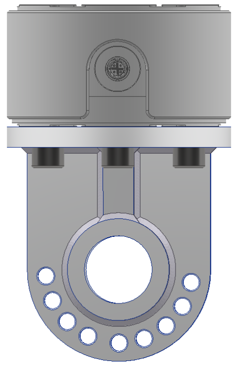
Figure 9: Proposed solution to increase carriage-platform distance.

The second step has been the design of the attachment between the vehicle and struts. The solution required joints to permit movements in all six degrees of freedom, and a concept for the specific application had to be designed. A unique connection point with the model was preferred allowing reaction forces and torque to be measured with the six-axis load cell. The CAD drawing of the final design is shown in *Figure 10a*. A U-shape beam is mounted between the struts and, if possible, is placed inside the testing model. An adaptor flange, represented in *Figure 10b*, has been designed to connect the measuring instrument to the beam through a joint allowing the roll movement. In the initial implementation stage, the dynamic actuation of the roll movement was not included. Instead, it is possible to change the inclination of the vehicle by fixing the flange using one of the realized holes. Starting from the reference hole, which leads to a null roll angle, holes at 20° as angle offset are present on the left side; instead, on the right side, the first hole is at an angular distance of 25° and the following of 20°. This solution gives the possibility to select multiple roll orientations (20°, 25°, 40°, 45°, 60°, 65°, 80°, 85°).

The flange is connected to the beam through a central shaft generating the axis of rotation which withstand the load. The small pin located at the bottom part allows the regulation of the model inclination by being inserted into one of the holes of the flange. Since the rod, and all the attached components, will be placed inside the model, an option was designed to manually move the pin without disassembling the tested object. By vertically pulling a rope, which is then deviated by a pulley, the pin is extracted from the hole allowing the roll angle selection. Once reached the desired orientation, the rope can be released, and a spring, appositely placed, will push the pin inside the new hole preventing at the same time the pin from falling out.



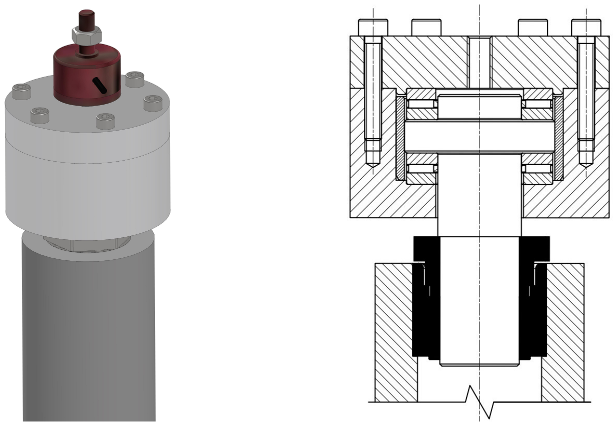
(a) Beam assembly drawing.



(b) Load cell and flange.

Figure 10: Model-Struts connection beam.

The shaft and pin connect the model to the beam using two angle profiles, reinforced with two ribs each. The profiles are attached to the beam with a bolted connection. The attachment with the struts is realized using two fork joints, enabling the rod to be tilted to generate the pitch movement of the model. Specifically, two bushing supports, each equipped with two flanged bronze plain bearings, are attached to the beam on both sides. A shaft links the fork connecting it to the bushing support, which guarantees its rotation while reducing friction and, at the same time, providing rigidity to the connection in a small space. The fork is realized with welded aluminium plates and a threaded shaft on the upper part. The latter permits screwing the fork into the strut. Each strut is free to rotate around its own axis to guarantee the yaw movement of the model and is also free to move vertically.



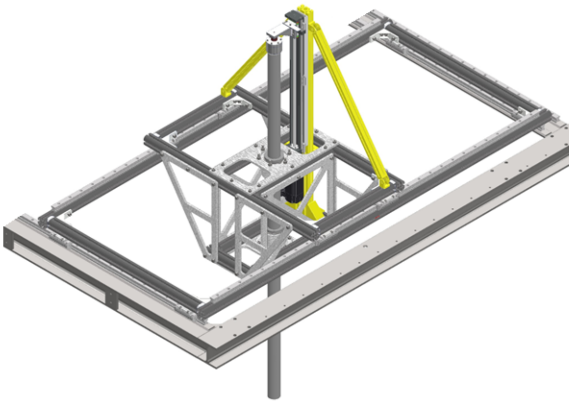
(a) Assembly with load cell. (b) Section view (In black Trantorque device).

Figure 11: Strut upper connection box.

In the upper part, to connect the spar to the actuator, the connection box represented in *Figure 11* was designed. The device consists of five parts, a connection shaft, a bottom case, an upper cover and two axial needle bearings. The components are kept together using six screws. The connection shaft is attached to the inner diameter of the strut using a Trantorque device. The latter guarantees a zero-backlash frictional connection and can accommodate high-torque, thrust, bending, and radial loads [1]. Being placed inside the strut, this type of coupling does not interfere with its vertical stoke avoiding the reduction of the already limited travel. Furthermore, it allows quick mounting. With the initially considered fastening pin/bolted connection, the available stroke was reduced by circa 150 mm due to the required holes, perpendicular to the strut axis, which can not enter the upper bearing inner diameter. This solution reduced the vertical stroke and the pitch angle range. The two axial needle bearings are placed inside the case to withstand the axial load and guarantee the rotation of the strut. A plain bearing is placed in contact with the large diameter section of the shaft to avoid any offset between the connection shaft and the connection box. This solution made it possible to realize a compact and light design. On the upper cover, a threaded hole allows easy and flexible mounting of the axial load cell. In this way, the sensor is the only element connecting the strut to the actuator, and the only load resulting in the device is an axial force.

Specific supports were designed for actuators to be incorporated into the existing structure without disassembling or modifying the pre-existing configuration. Other requirements such as light weight, ease of assembly, rigidity, and small dimension were considered. Below are described three types of support that have been designed.

- **Support frame for Z direction.** Having already a limited available stroke, here the requirement was to avoid any further reduction of the movement range. Therefore, the solution was to place the motor inside the X sliding structure, permitting having all the available strokes for the movement without interference with the upper carriage frame. The support drawings are depicted in *Figure 12*. Standard aluminium profiles, with dimensional class 6, were used in the design since the centre distance between two nut rows is matching the hole spacing in the actuator frame for fixing purposes. The support frame is attached to the X-sliding structure through six angle connection. Additionally, to increase the stiffness and reduce vibrations, two sloping profiles are connected between the upper part of the actuator and the carbon fibre profile.

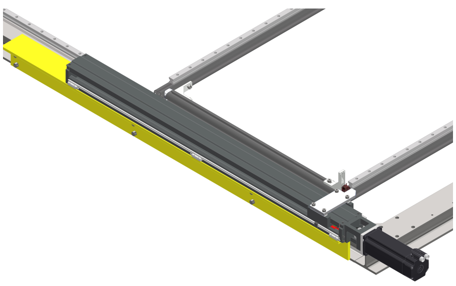


(a) Frame integrated with the sliding structure.

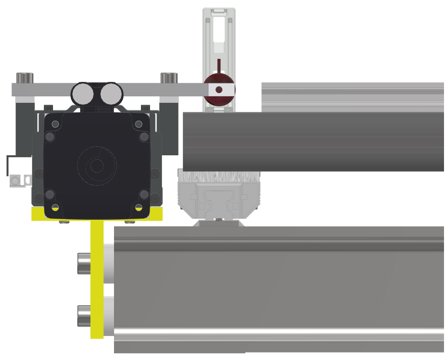
(b) View of the frame only.

Figure 12: Support frame for Z direction.

- **Support frame for Y direction.** Having to be mounted on the I-beam frame, this support structure was not presenting design challenges. Due to the impossibility of placing it in the middle of the Y-sliding frame, it was decided to fix it laterally. The support is realized with a t-shape aluminium profile, with apposite holes designed to match the ones of the actuator frame. The beam is fixed to the frame through a bolted connection using three vertical plates welded on the side of the I-beam structure (They are visible in *Figure 12a*). The actuators for the Y movement have been selected with an aluminium cover to avoid any external piece could enter the screw drive section and generating damage. Spacers are placed between the aluminium profile and the I-beam structure to avoid interference between moving frames and actuators.



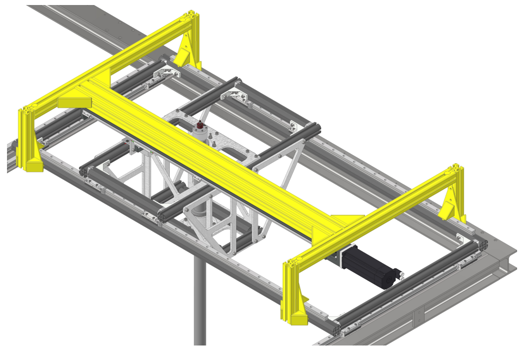
(a) Frame fixed to the I-beam structure.



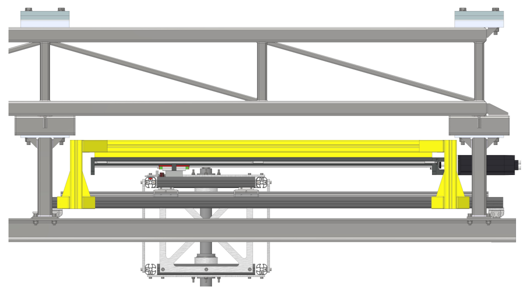
(b) Support frame: lateral view.

Figure 13: Support frame for Y direction.

- **Support frame for X direction.** As previously written, the X actuator is an optional feature for the testing platform. Many constraints are present to place a support frame on the Y sliding structure, mainly due to the few available spaces. The U-shape attachment frames have to be mounted directly on the I-beam construction. The Y sliding structure, with the X rails moving on top of it, is located under those frames. This arrangement leaves only a reduced space in height to mount the support for the X actuator, not even enough to place the actuator itself *Figure 14b*. The only available option was to reduce the stroke and to fix the actuator, through the designed support, to the carbon fibre profile. The chassis is not symmetrical to avoid interference with the vertical strut and the support frame for the Z direction. The structure is composed of standard aluminium profiles with dimensional class 10 since the centre distance between two nut rows is matching the hole spacing in the actuator frame for fixing purposes. It is laterally mounted to the Y-frame using angle connections, and additionally, extra plates are employed to increase the stiffness of the structure *Figure 14a*. However, the reduction of the stroke may not interfere with the test. The connection beam limits the range of the movement in the X direction depending on the dimension of the vehicle. Considering Monique, when the front strut is at half of the X stoke, the rear one is at its maximum X position. Therefore, the entire X run could not be covered in any case.



(a) Frame integrated with the sliding structure.



(b) Lateral view: U-shape frame constraint

Figure 14: Support frame for X direction.

Every component has been verified and validated through FEM analysis using the load cases discussed in *Section 3.1*. The selected material is aluminium EN AW-6060 which is corrosion-resistant and offers good strength values. The described configuration is specific to empty models into which it is possible to place the connection bar. However, it is also possible to position the connection beam above the surface of the vehicle and realize the connection employing only the six-axis load cell. If needed, it is possible to place also the orientable flange for the roll movement. However, increased flow disturbances will be generated by the added devices. Alternatively, one may think of an application in which the connection beam is outside of the water, and a single strut connects it to the underwater model. This flexibility makes it clear that the platform proposed in this paper can be employed for multiple and different applications.

4.1. Instrumentation and data acquisition

It is possible to evaluate hydrodynamic parameters by analysing forces and torques exchanged between the vehicle and the apparatus. Therefore, load cell sensors are essential equipment for the testing platform. As stated in *Section 2*, a decoupling between torques and forces is required to achieve optimal measuring accuracy. The designed apparatus can perform this task by implementing a single-axis loads cell in between each actuator and the respective moving frame. Consequently, in the upper part of the structure, it is possible to measure all the loads exchanged by the underwater model and the struts converted in linear forces. Instead, for the evaluation of the torques, it has been decided to place the 6-axis force and torques cell as the unique attachment point of the functional model with the testing apparatus (*Figure 15a*).

The deployment of this multi-axis sensor allows the evaluation of reaction forces and torques directly on the model. Thus, without any influence of the actuated platform and friction effects of the sliding guides. It is possible to compare underwater measured values with the ones gained on the upper part of the platform. This procedure is expected to bring a better understanding of the influence of struts, and testing platforms in general, on the vehicle behaviour during the experiments. Load cells with a single threaded attachment have been selected to guarantee their easy interchange depending on the expected range of forces and the specific test to be done. In particular, the model KM26z by ME-Meßsysteme GmbH [20] can be easily integrated with the system and is available for nominal force range starting from 20 N up to 5 kN keeping the same thread dimension M6. This feature, in addition to good accuracy and small dimensions, made the model suitable for the application (*Figure 15b*). Specific plates and support structures have been designed to connect the carriage of actuators to the sliding frames by interposing the measuring sensor in between.

Concerning the position of sliding frames and struts, analog draw-wire sensors were identified as the optimal solution. A potentiometer is connected to a wire and a torque spring. Depending on the extension of the cable, a different resistance is imposed on the circuit. When the wire is released, the spring coils it back. By analyzing the electrical output, it is possible to gain information on the real-time extension of the wire. Models WPS-MK30 and WPS-MK46 by Micro-Epsilon Messtechnik GmbH [13] have been selected (*Figure 15c*). They offer a measuring range respectively of 500 mm and 1250 mm with high linearity across the entire measuring range and a compact sensor housing easily embeddable into the system. Two wire sensors are foreseen for each sliding frame guaranteeing the constant presence of a sensor with the cable pulled, avoiding measuring issues due to unstretched wire. Instead, a single sensor has been considered for the strut position measurement due to a lack of space.

(a) K6D110 Six-axis cell [19]. (b) KM26z load cell [20]. (c) Wire sensor [13].

Figure 15: Selected sensors.

Servo motors include capacitive multi-turn absolute encoders allowing a control loop on the angular position of the shaft. Two inductive sensors are positioned along the frame of the actuator. They permit the homing positioning and guarantee safe operation by avoiding movements beyond the limits that could damage the system. The system has two feedback on the position of the frames, one coming from the absolute encoder and one from the wire sensor. The real-time measurement of the struts displacement is performed by the wire sensors being them connected to the centre of the moving frames. Small misalignments between the position of actuators and one of the struts might happen due to the side connection of actuators to the moving frame. This double feedback leads to obtaining high positioning accuracy.

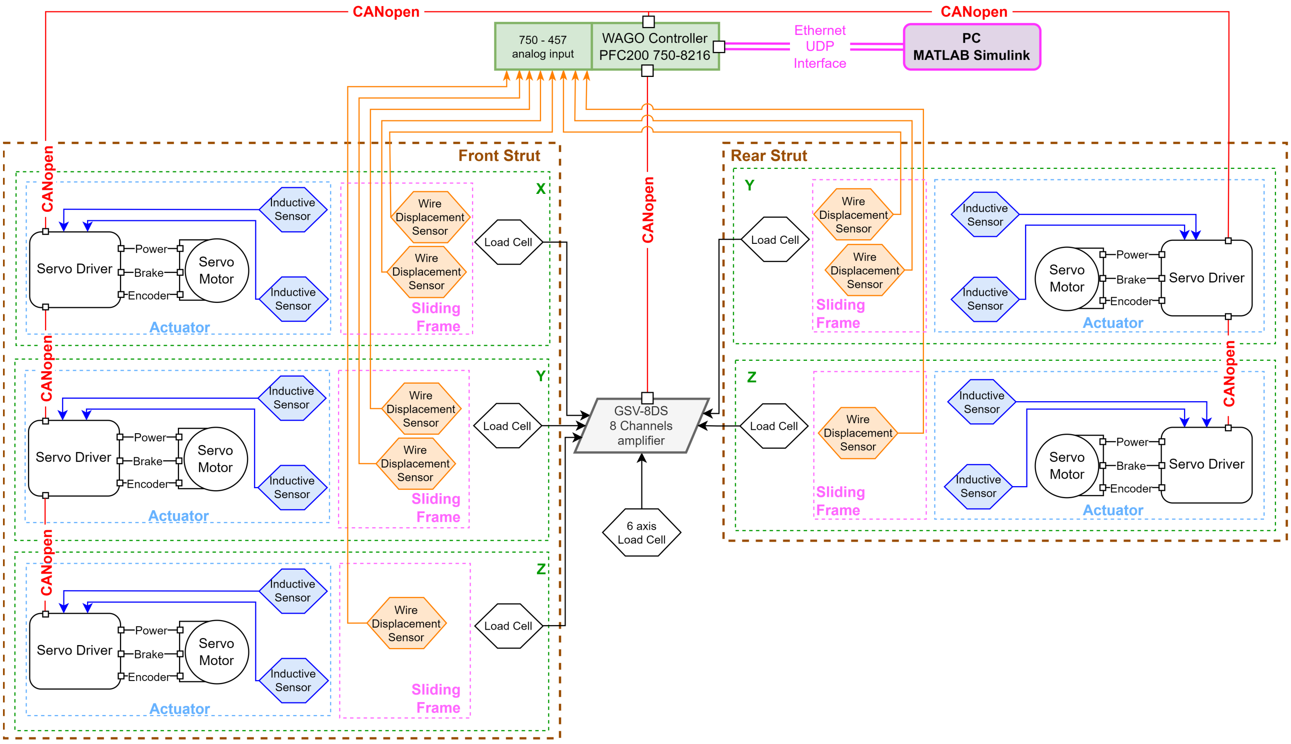


Figure 16: System connection scheme.

The system connection scheme is shown in *Figure 16*. The selected controller for the platform is a PFC200 750-8216 by WAGO gmbh equipped with a ±10 V DC analog input module (750-457) and end module (750-600) [21]. This compact PLC, among a variety of connection protocols, includes ETHERNET and a CAN\CANopen interfaces that are used for the presented project. The Wago Power supply Pro2 supplies the controller, and all the mounted modules, with 24 V and the Wago 24\10 VDC converter (Module 859-802) powers the wire displacement sensors. The servo drivers are instead supplied with 24 VDC for the control modules and with three-phase power 230-480 VAC and single phase 100-230 VAC respectively for the big and small size motor. Drivers and motors are connected using two wires: a power supply cable for the electric motor and the brake and a data cable for the encoder and the motor temperature sensor. Inductive sensors are directly connected to the drivers using the I/O analog and digital interface.

The signal cables of the wire displacement sensors are plugged into the DC analog input module of the PLC. Instead, load cells are connected to the GSV-8DS 8-channel measuring amplifier, which is powered by a dedicated 12 V power supply. The working principle of the selected load cells is based on strain gauges. Therefore, signal amplification and a post-processing analysis are required. It is essential to remove interference and apply a correction considering the specific analyzed system. The GSV device receives data from the load cells, actuates the post-processing operations and sends the refined data to the controller using the CAN bus. The drivers are also connected to the CAN bus and can communicate with the PLC employing the CANopen protocol. The baud rate value is set to 500 KBaud, allowing a fast exchange of information without overcharging the Fieldbus. Two 120 Ω resistors terminate the CANbus on both sides. Finally, the engineering computer is connected via the ETHERNET interface to the controller using UDP protocol.

4.2. Control system

The fundamental parts of the control software of the platform have been implemented. The PLC is programmed using the automation software e!COCKPIT from Wago [21], based on the open-source programming language CODESYS (Controlled Development System), which is used nowadays in industrial computing since it complies with the industrial standard IEC 61131-3. The control system consists of three components, MATLAB mathematical model, PLC software and motor driver software. The collaboration of these three subsections allows the optimal functioning of the system. The MATLAB model of the platform can communicate in real-time, via a UDP connection, with the PLC through a Simulink interface. By doing so, it is possible to combine the computational power of the software with the robustness and reliability provided by the industrial controller. In *Figure 17*, a simplified representation of the software logic scheme is shown.

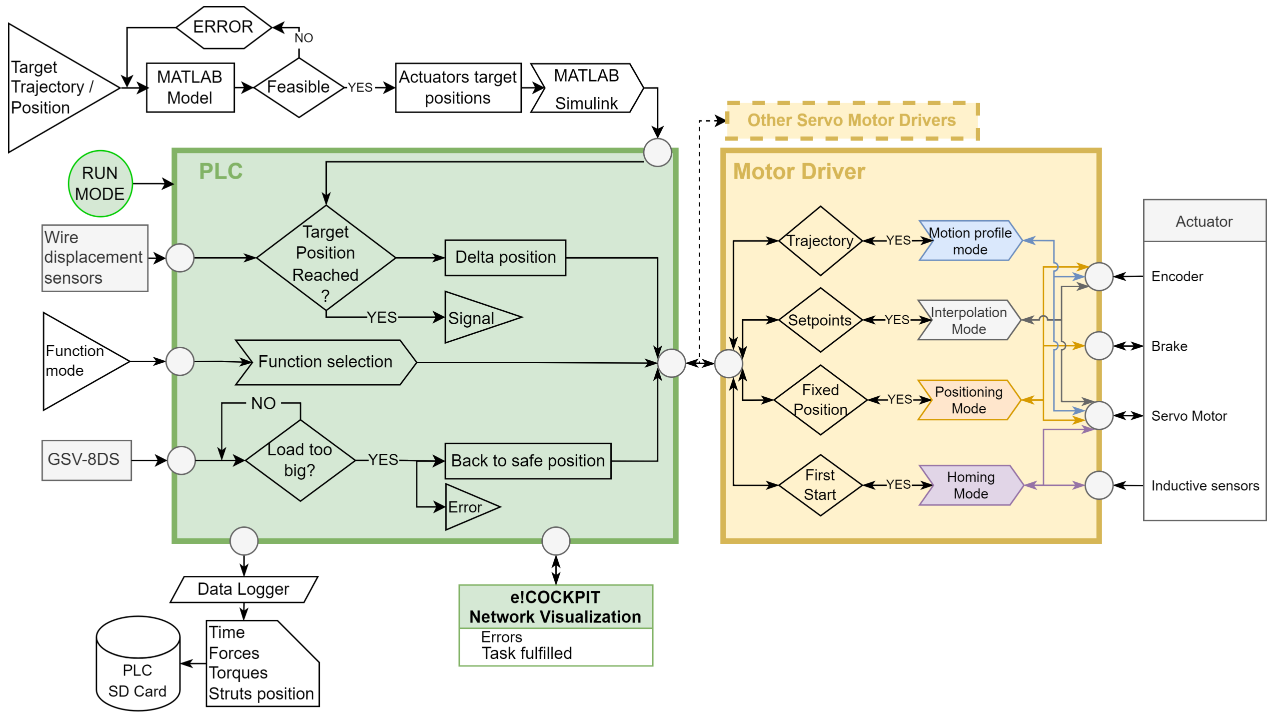


Figure 17: Simplified software logic scheme.

4.2.1 MATLAB model

A mathematical model in MATLAB was created with two main goals, to analyze the kinematic behaviour of the testing platform to find out the maximum attainable angles, and make a tool for the basis of the control algorithm. The model serves as a user interface where the desired target positions and/or trajectories are given as inputs. The model calculates the target position for all the actuators to reproduce the chosen orientation in the towing tank basin. Dimensions of the platform, such as strokes of the actuators, the height of the towing tank carriage, and the basin water level, are written in the model as constant parameters. Furthermore, the sizes of the vehicle and specific dimensions related to the type of model-platform connection are set. Thus, given these dimensions, the mathematical model can graphically reproduce the entire platform.

The model takes as input the desired angles for the vehicle orientation: yaw, pitch and roll. It is first checked whether the required position can be reached, based on the allowed strokes and the mechanical limitations of the frame. The front strut is assumed as the master, while the one on the rear will move accordingly to reach the target angles. If the configuration is not feasible, an error in the command window is generated, along with suggestions on what to change to obtain a feasible orientation. Positive feedback is displayed otherwise. The coordinates of the bottom tip of each strut are provided, together with the target positions of each actuator. In addition, the model provides the minimum distance between the vehicle and the water surface and the distance between AUV and the basin bottom. This information helps to identify possible influences of the free surface effect or any external interaction. If wide angles are givens as input, the vehicle may come out of the water or even impacts the platform structure. Even in these cases, an error to the user is displayed, who can try to enter new parameters following the provided suggestions. A 3D graph is plotted, showing the platform structure with the sliding frame positions, the vehicle orientation, and the water level. An example of this is shown in *Figure 18*, where it is possible to identify the edges that would end up being outside the water during the test. Moreover, three additional graphs are displayed to show, in the three principal planes, the orientation of the connection beam (*Figure 19*).

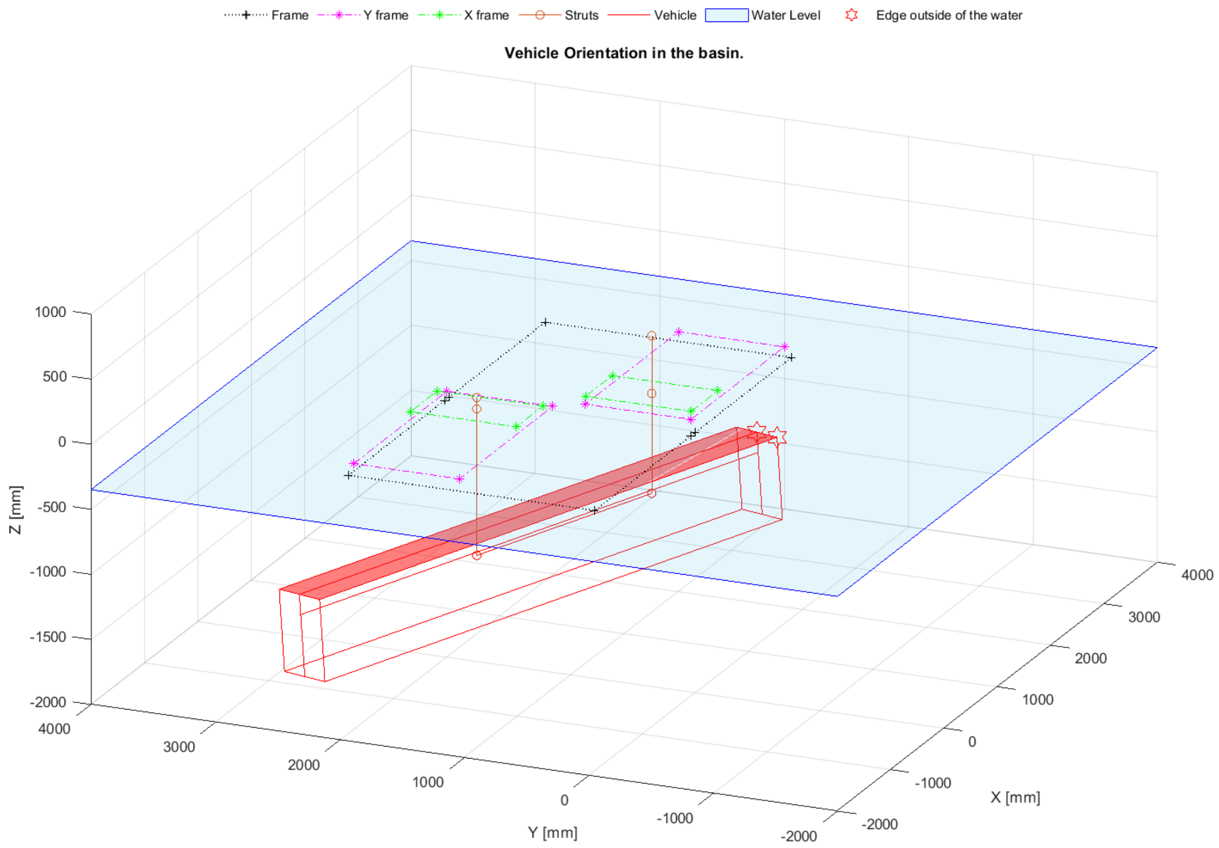
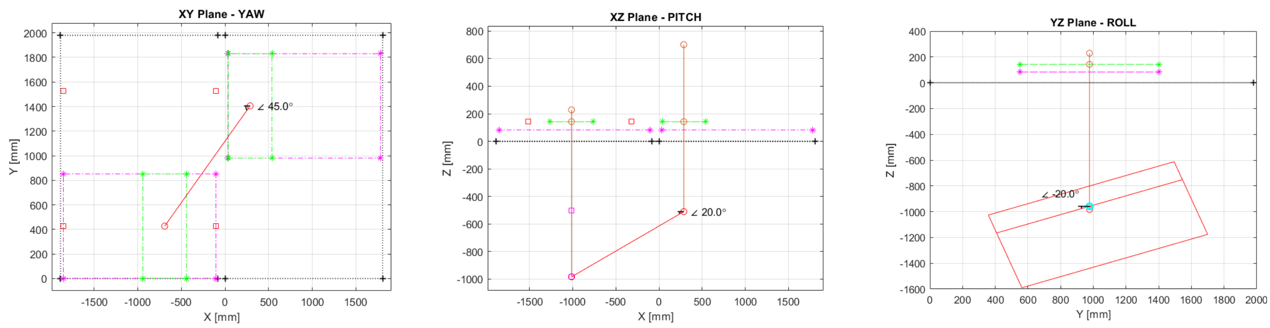


Figure 18: Example of vehicle 3D plot with orientation Yaw: −52°, Pitch: 15°, Roll 15°.



(a) Yaw angle.

(b) Pitch angle.

(c) Roll angle.

Figure 19: MATLAB model, additional plots.

It is also possible to give in input to the model a trajectory. Firstly a discretization in setpoints is performed, and by keeping the connection beam axis tangent to the trail, it is then possible to calculate the orientation angles. Therefore, the evaluation of the feasibility is carried out point by point. The outputs of the model are the actuator target positions, the coordinates of the bottom tip of the struts, and a vector containing the speeds to use while reaching the targets. A MATLAB Simulink interface, using the Simulink real-time library, permits the establishment of a communication MATLAB-PLC. The functions UDPsend \ UDPreceive are used for this purpose. Therefore, after processing the data to make them suitable to be shipped, the values are sent to the PLC.

4.2.2 PLC software

PLC programming has been performed with the e!COCKPIT automation software. The connection scheme of the system can be represented in its network window by specifying the communication protocols between each device. The appliances equipped with the CAN bus protocol, are provided by the manufacturing companies with Electronic Data Sheets files (EDS). These electronic documents are used to configure different descriptive and communication data for the hardware. In particular, an EDS file is an ASCII-structured text which includes a detailed definition of the network configurable parameters of the device based on the CANopen standard. The network representation of the testing platform in the e!COCKPIT environment is depicted in *Figure 20*. The computer, with the MATLAB model, is represented by the generic Modbus slave block and is connected to the PLC through UDP protocol via the ETHERNET interface (drawn in grey). The controller, and the equipped modules, are automatically detected and placed in the network view when connected to the computer via ETHERNET. Five motor drivers and the GSV-8DS, uploaded into the software using their EDS file, are connected via CANopen protocol (marked in blue). The configuration of each box was performed, assigning CANopen node ID, IP address and all the required parameters.

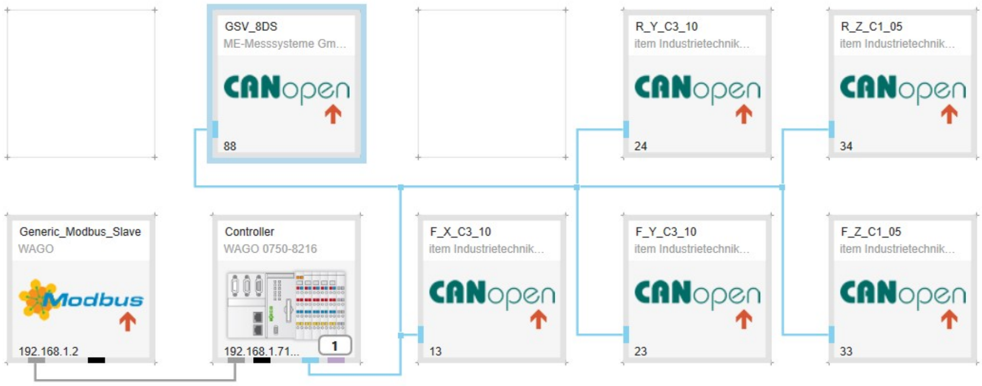


Figure 20: e!COCKPIT network representation of the testing platform.

Through the EDS file, it is possible to map the variables of the appliance with the ones used in the control code. It is also possible to fully communicate with the CAN device and configure it by employing Process Data Objects (PDO) and Service Data Objects (SDO). In particular, a CANopen device employs SDO to read/edit values of another component, while it uses PDO to share real-time operational data across CANopen nodes. Four main scripts, which are described below, have been written using the programming language CODESYS.

- **UDP communication Simulink-PLC.** As described in *Section 4.2.1*, target positions for the actuators, strut coordinates and speeds are sent to the PLC using the UDP protocol. The code allows the exchange of data from and to the computer's IP address. Due to the data type limitation of the UDP send function of MATLAB Simulink, only unsigned integers with 8 bits of information can be sent. Considering three digits after the decimal point, and a maximum value to transfer in the order of thousands, each number consists of seven digits. A Matlab function divides each value into an array, assigning to every position of the vector one of the seven digits making up the values to send. The opposite process is carried out on the PLC side, making it possible to recreate the original numbers. The recomposed values are stored as global variables in the PLC memory, allowing them to be shared among the scripts of the controller. A UDP sending function was also implemented, given the future need of sending information, i.e. error messages, to the MATLAB interface.
- **Data sampling.** All sensors deployed on the platform are connected to the PLC, which constantly reads their output to process the control algorithm. Wire displacement sensors are plugged directly into one of the modules of the controller. Therefore, the PLC receives real-time values without any processing.

The analogue signal generated by the sensor corresponds to the electrical resistance of the potentiometer that constitutes the instrument. A calibration function was employed to correlate the electrical signal to a value representing a distance in millimetres. For both sizes of the sensors, an accuracy of 0.5 mm has been obtained. Load cells are instead connected to the GSV-8DS. The device is programmed to perform signal processing and forward the treated values to the PLC every 30 ms. Measured values are stored as global variables in PLC memory, allowing them to be shared among controller scripts.

- **Data logger.** Permanently storing the measured values, and the data read from the motors is essential to analyze the test results at the end of the experiment. A data logger script has been written to save, every time step set to 30 ms, the parameters of the test. In particular, a comma-separated values file (CSV file) is composed, row by row, including date and hour with milliseconds of the experiment, a row counter, a milliseconds counter, forces and torques from load cells, positions of the sliding frames from the wire sensors, positions and speeds of the actuators. The CSV file is saved regularly on the SD card placed inside the controller. The data logger function already implemented in the CODESYS libraries has 1 s as the minimum sampling time step, which was not meeting the requirements for the calculation of the hydrodynamic parameters. Therefore, a new script employing a buffering logic was adopted. The limitation on the logger sampling time comes from the physical time needed by the program to open the CSV file, add a new row with the values, save the file and close it for each time step. Thus, the solution was to reduce the number of times this operation was carried out. Temporarily storing the data in a buffering array of size 200 and proceeding with the saving operation only when it is filled, permits reducing the computational time. This solution allowed the data logger script to achieve sampling time up to 5 ms, varying, at the same time, the cyclic run time for the PLC tasks.
- **Control algorithm for motors.** Regarding the control of the servomotors, the PLC acts in two steps: first, it transmits the received target positions and speeds to each driver via the CAN bus, where the final control algorithm runs. Secondly, the real-time location of the strut is checked using the output data of the wire displacement sensors and comparing them with the target coordinates received from the MATLAB model. If the values do not match, it is clculated the difference in position, and a new command is sent to the respective driver. At the same time, the maximum loads detected by the load cells are analyzed, and if these numbers reach limit values either for the load cells themselves or for other components of the system (i.e. linear guides or actuators), an error message is generated. Thus, a stop command is sent to the driver. In such a manner, possible damage to the platform can be limited.

When the run command is given to the PLC, it simultaneously processes all these scripts. The application, including the device settings and algorithms, can be downloaded into the controller. Afterwards, it can work autonomously without the support of e!COCKPIT software. However, the connection with the PC via ETHERNET is still needed to allow communication between MATLAB and the PLC.

4.2.3 Driver software

The first step with the servo drivers has been their initial configuration by connecting them to the computer via USB. The software MotionSoft by item [6] allowed the initialization of the programmable servo controllers to adapt their settings for the specific application. By selecting the model of the motor connected to the drivers, the most relevant configuration parameters are uploaded automatically into the control program. Secondly, specifications about the linear unit connected to the motor, including limit sensors, were set. Gear ratio, screw drive pitch, maximum permissible speed and acceleration and travel distance are examples of configuration values to insert into the driver program.

Following the initial configuration phase, function tests were performed, allowing the software to automatically fine-tune the control parameters best suited to the drive system. During this run, the carriage moves from one limit switch to another, measuring the maximum positioning range and registering a safety distance from the limit switch. Lastly, the CAN communication protocol was enabled and configured to match the parameters stored in the PLC network interface. The PLC controls the drivers by sending controlwords and receiving statuswords. With a controlword, it is possible to change the state of the servo controller or directly execute a designated action. Instead, the statusword is the answer the driver sends to the PLC indicating the state of the servo controller or the ongoing operation.

Multiple modes of operation are implemented in the driver control software and can be triggered by sending a controlword. The ones considered for the presented project are listed below.

- **Homing mode.** This operational mode is processed as a first step at the beginning of each run. The reference point for each actuator is the limit sensor placed at the minimum position of the linear drive unit. The user can set the velocity, acceleration, and the kind of homing operation most suitable for the system. The one selected involves a relatively quick motion of the carriage in the negative direction until it reaches the negative limit switch. Afterwards, the drive slowly moves forward to find the exact position of the sensor. The zero position is then correlated to the first zero impulse of the encoder in the positive direction from the limit switch. The absolute encoder in the servomotor allows performing this operation only when the system is booted up. Once the initial position has been recorded, it is possible to drive the servo motor using all the modes of operation set in the driver.

- **Positioning mode.** This operation consists in reaching a fixed target position to be kept for the test execution. Once received the command from the PLC, the driver, using the built-in encoder and the implemented positioning loop, activates the motor to reach the target position. The achieved placement is checked by the PLC employing the wire sensors. If it matches the target, the brake is activated. The driver stops the operations when the target position is achieved. Standard values for speed, acceleration and deceleration are preset in the driver, but specific values for the application can be employed by sending the corresponding controlword via CANopen from the PLC.
- **Interpolation mode.** This operation mode permits the model to follow a trajectory inserted in the MATLAB interface. After the discretization, the PLC receives setpoints of 3D coordinates and forwards them to the corresponding driver. The trajectory generator in the controller takes coordinates and assigned speed as input and, through linear interpolation, can provide a smooth movement. This procedure, performed by all actuators of the testing platform, allows the connected vehicle to recreate the desired trajectory. This mode of operation requires synchronisation telegrams (SNYC) to be shared among drivers via the CAN bus. Every driver sets its clock following the received sync message allowing the achievement of optimal system coordination.
- **Motion profile mode.** In the driver configuration environment, it is possible to create multiple motion profiles that can be saved in the internal memory and subsequently activated by sending a specific controlword from the PLC. Therefore, the idea is to create motion profiles representing, for example, trigonometric functions by assigning amplitude and frequency. Each driver will move the respective axis following the allocated motion profile. The combination of the movements of the frame, and the advancement of the towing tank carriage along the basin, will produce a complex underwater trajectory.

5. Results

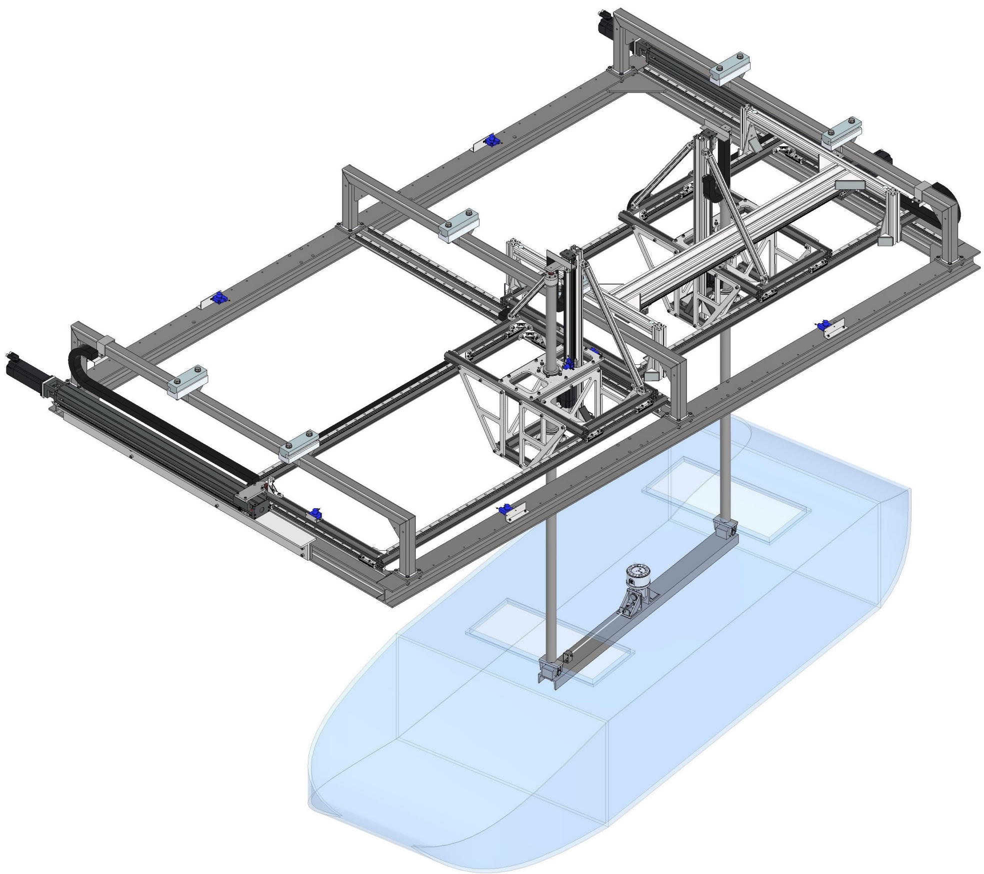


Figure 21: Dynamic test platform for AUV: CAD drawing of the final assembly.

In *Figure 21*, the final assembly CAD drawing of the measuring apparatus, without the upper extension frame, is reported. In the drawing are embedded all the above-described components and devices. In addition, cable carrier chains were placed to support cables during the movements of the frames. They provide smooth functioning without interference and increase the service life of the electrical wires.

The design has been approved and validated. Thus construction drawings were realized and sent to the workshop, and raw materials, sensors, motors, and all the required components were purchased. In the figure, it is possible to identify wire sensors, coloured in blue, which are fixed to the frames with L-shape aluminium profiles. For the Z axis, an aluminium profile connected to the carriage of the actuator allows the wire terminal and the cable carrier chain to be secured. All the mechanical requirements of the project have been achieved. The platform can be employed to dynamically move the model in five degrees of freedom. The possibility to automize the roll movements is left as an additional feature to implement once the platform will be realized.

The construction of the apparatus could not be completed on time due to long delivery times and issues that arose during the implementation process. The final fabrication is scheduled for 2023 when all the components will be available. However, the design of the DMS department is now ready for the testing phase in the VWS of TU Berlin. In *Figure 21*, the platform is shown during some ship tests carried out at the beginning of November 2022.

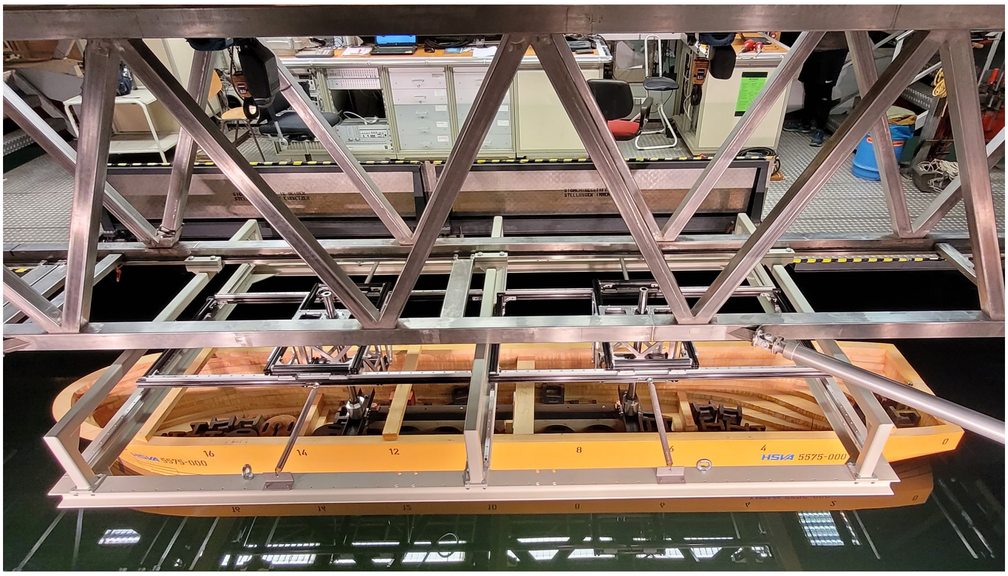
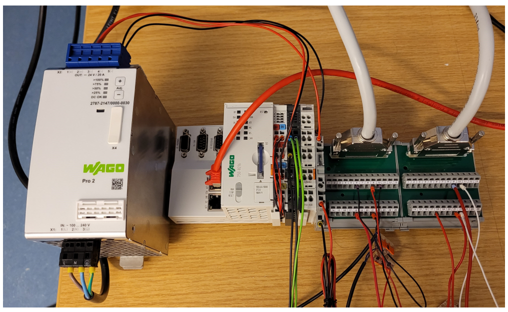
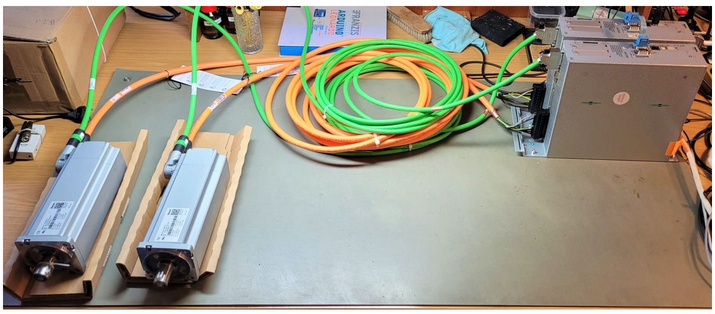


Figure 22: DMS Testing platform in the towing tank of TU Berlin.

A mock-up on a test bench has been implemented with the delivered appliances to prove their performances, and to test the PLC software features. In *Figure 23a* the PLC rail configuration is depicted; starting from the left: 24 V power supply unit, PFC200 750-8216 controller with output interfaces, 24\10 VDC converter and I\O external modules for the motor drivers control. Two servo motors SE80-350-5-55-AK with the respective drivers C3-05 are shown in *Figure 23b*. The orange cable is the one that provides the power supply to the electric motor and the brake, the green one is instead the data cable, and it transmits output values from the encoder and information about the temperature of the device, used for safety reasons.



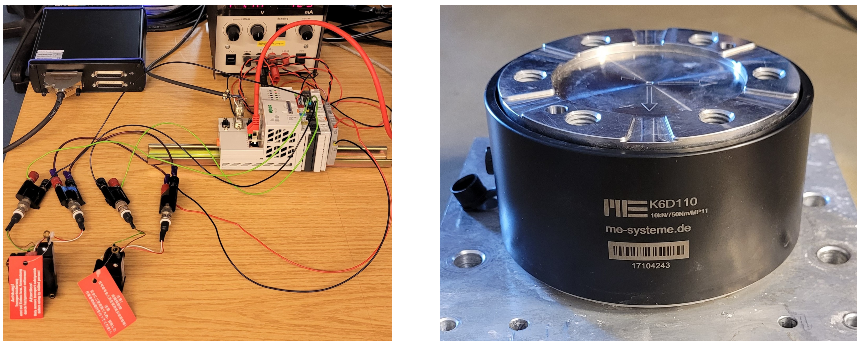
(a) PLC rail setup.



(b) SE80-350-5-55-AK motors with C3-05 drivers.

Figure 23: PLC mock-up configuration and servomotors.

In *Figure 24a* are shown two wire sensors: one WPS-MK30 with a maximum extension of 500 mm and one WPS-MK46 with a maximum extension of 1250 mm. They are connected to the 24\10 VDC converter for power supply and to the ±10 V DC analog input module of the PLC. In the figure, it is also possible to see the GSV-8DS 8-channel measuring amplifier connected to the 6-axis force and torque sensor depicted in *Figure 24b*. Drivers, PLC and GSV8 are connected to the CAN bus, and the 120 Ω terminal resistors have been plugged respectively into the amplifier and to one of the motor drivers. To simulate the inductive limit switches, that will be mounted on the linear drive system, sensors of the same type available in the laboratory were deployed.



(a) Wire sensors and GSV-8DS.

(b) 6-axis force torque sensor.

Figure 24: Mock-up sensors.

The test setup made it possible to practically test some of the software features described in this document. However, it was not possible to test the functionality of all the described modes of operations mainly due to the lack of motors and missing components. The main achievements on the software side are listed below.

- Data sampling and logger. The PLC correctly receives the measurement values from load cells after being processed by the GSV amplifier. The analogue signal received from the wire sensor is treated by the program and converted into distance measure with an accuracy of 0.5 mm. The real-time positions, speeds, and accelerations of the motors received by the PLC from the drivers. The data logger script works as expected and can generate the CSV file and fill it, row by row, with all the mentioned parameters. The date, hour of the test and millisecond counter are also added to the file. The sampling time is set to 30 ms, but it can be changed depending on the need. If this is required, also the setting of the GSV8 amplifier must be adapted so that the load cell outputs are transmitted via CAN bus with the right frequency. The logging will allow an in-depth analysis, following the towing tank experiment, with the ultimate goal of determining the hydrodynamic parameters of the tested model.
- MATLAB model. The analytical model is fully working. It can be deployed to assess whether the input orientation of the vehicle is achievable by the apparatus. The communication between MATLAB Simulink and PLC via UDP protocol has been tested and validated. It permits real-time communication between the two devices, and it is employed to send input data to the control system. The target positions generated for each actuator have been verified and correspond to reality.
- Position control. The positioning mode has been tested and validated. The PLC receives coordinates and sends them to the respective destination motor driver. By deploying the positioning mode, it is possible to reach the target coordinate, and the brake is subsequently enabled. The PLC also considers the measured distance by the wire sensors allowing the reduction of possible small misalignments.
- Homing operation. The homing mode has been fully tested. The zero position can be set, and is stored correctly in the driver, which can therefore work by referring its operations to the absolute position. The optimal configuration of the control parameters in the servo leads to the reduction of the home positioning time by guaranteeing at the same time position accuracy.
- Other modes of operations. The controlling strategies described in *Section 4.2.2* have been implemented in terms of coding and simulated in the programming environment. However, due to the presence of only two motors and without the linear drive systems, the practical validation of their operation was not possible. Basic motion profiles have been generated and saved in the drivers. The motors correctly reproduce the desired behaviour when the motion profile mode is activated. However, without the complete layout of motors and linear motion system with screw drive, it was difficult to estimate and prove the final movement of a hypothetical model attached to the apparatus.

6. Conclusions

A measuring apparatus for towing tank tests with AUV allowing the dynamic control of its orientation in five degrees of freedom, has been fully developed. The platform is designed for the towing tank carriage of the research institute for hydraulic engineering and shipbuilding at TU Berlin (VWS). The in-depth mechanical design resulted in the final concept. The latter was validated using FEM analysis, adopting the loads obtained by increasing the output values of CFD simulations up to the ultimate limit of the aluminium rail guides, the weakest elements of the system. The innovative design allows driving an underwater model in independent or coordinated motions bringing improvement to the traditional testing platform for towing tanks. Thus, it was possible to meet most of the initial requirements. The relization of technical drawings for the workshop, and the purchase of all the components from the respective manufacturing companies, have been carried out after a budget analysis. The kinematic analysis carried out with the analytical MATLAB model proved the ability of the measuring platform to drive the AUV model with the desired orientations. The actuation system, with motors and sensors, was engineered, and a mock-up with the available components was implemented. The functionalities testing of each device allowed optimal parameter configurations to be achieved. The basic parts of the control software have been implemented, and multiple modes of operation are programmed, allowing flexible use of the apparatus.

The final construction of the platform could not be carried out on time by the end of the year due to delivery times and issues which arose during the testing phase. The finalization of the mechanical implementation of the platform is planned for 2023. However, after the initial tests, the testing platform is expected to meet the requirements, both in terms of data collection and manoeuvrability, allowing the hydrodynamic optimization of the shape for the MUM2 demonstrator. Furthermore, it is going to be a new apparatus available at the facility of TU Berlin to be deployed to continue the research in the maritime field.

Only the basic features of the control software have been tested, obtaining promising results and laying the groundwork for future advancements. Improving the performances and testing the whole control algorithm on the finished platform will be the following step of the project. There is room for the deployment of new modes of operation, and one may consider new control strategies by combining MATLAB functions with the easy-to-implement drivers' software. On the mechanical side, once all the functionalities of the presented system are validated, it will be possible to implement the automation of the remaining degree of freedom. Thus, the dynamic control of the roll movement will also be achieved. New sensors can be installed on the tested model for real-time feedback on the actual orientation and motion parameters. As a further development, a mathematical model to identify the optimal vehicle scale, depending on the maximum loads the platform can withstand, is required. This tool will allow obtaining the minimum scale factor that can be considered for the model realization, aiming to reduce the scale effect on the test.

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Abstract in lingua italiana

I veicoli subacquei a guida autonoma rappresentano una tecnologia all'avanguardia che troverà molteplici applicazioni nel futuro prossimo. Essi consentono l'esplorazione degli oceani più profondi, ampliando le possibilità di ricerca in campo marittimo. Questa tesi fa parte del progetto di ricerca MUM, il cui obiettivo finale è lo sviluppo di una nuova classe di veicoli modulari. La costruzione di un sottomarino autonomo extra-large in scala reale è prevista entro il 2025. Il perfezionamento idrodinamico della sua forma è un requisito per la fase di progettazione, con lo scopo di raggiungere la manovrabilità ottimale. Esso può essere perseguito a partire dall'esecuzione di test in vasca navale. In questo documento viene presentato il progetto innovativo di un apparato di misurazione automatizzato per l'esecuzione di esperimenti di traino con veicoli sottomarini. Esso consente il controllo dinamico del modello in cinque gradi di libertà misurando, al contempo, forze e coppie di reazione. In primo luogo, viene introdotta la progettazione meccanica, con un focus sulla selezione e l'implementazione di attuatori e sensori. Vengono poi descritti i collegamenti al PLC e le prime fasi di sviluppo dell'algoritmo di controllo. Il processo di costruzione di questo apparato è stato avviato, e la realizzazione di una configurazione di prova ha permesso di testare le funzionalità del sistema. I risultati iniziali sono promettenti e si prevede che la piattaforma di misura soddisferà i requisiti del progetto, consentendo la valutazione dei parametri idrodinamici.

Parole chiave: UUV, XLUUV, MUM, Piattaforma per test dinamici, Esperimenti in vasca navale, Parametri idrodinamici

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