# Design of a close power loop test bench for contra-rotating propellers

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**Abstract.** The aim of the research is to develop an azimuthing contra-rotating propeller for commercial applications with a power of 2000 kW. The thruster system is designed especially to be installed on high speed crafts (HSCs) for passenger transport with a cruising speed of about 35-40 knots. The topic is very useful because the azimuth thruster solutions currently do not find commercial applications in naval units for passenger transport. The latter are heavy, not very efficient from a hydrodynamic point of view and suitable for maximum cruising speed of about 18-20 knots. The study is interesting because among the advantages that these solutions provide are the possibility of transmitting very high torques and to guarantee a much longer life cycle. In more detail, the propulsion is realized by using a C-drive configuration, with a first mechanical transmission realized by using bevel gears mounted in a frame inside the hull, and a second transmission realized by bevel gears housed in a profiled hull at the lower end of a support structure. In the profiled hull will be installed the shafts of the propellers, in a contra-rotating configuration. In order to optimize the system before its industrial use, a close power loop test bench has been studied and designed to test high power transmissions. The test configuration allows to implement a back-to-back connection between two identical azimuthing contra-rotating propellers. Moreover, the particular test bench allows to size the electric motor simply based on the dissipated power by the kinematic mechanisms. Since the efficiency of these systems are very high, it is not necessary to use large electric motors, thus managing to contain the operating costs of the testing phase. The most significant disadvantage is the need to have two identical transmissions with consequent increase in installation costs. Through the back-to-back test bench it was possible to study the increase in efficiency compared to traditional systems.

**Keywords.** Thruster, POD, azimuthing contra-rotating propeller, experimental test, cad modeling, finite element analysis.

## 1. Introduction

In the last ten years, the *High-Speed Crafts* (HSCs, in particular hydrofoils) have once again attracted the interest of shipping companies, especially for short-haul passenger transport. The renewed interest in this type of naval unit is mainly due to the lower operating costs [1].

After several decades in which the hydrofoils have not shown any particular evolution and they have been set aside by the market, important technological

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innovations have allowed the HSC to become competitive again. This is also demonstrated by important results obtained by several experimental tests, in towing tank and at sea, performed by Ruggiero et al. [2] on a new class of hydrofoils designed and built by Liberty Lines SpA shipping company. In addition, the use of light but strong materials such as fiber metal laminates or composite sandwich materials allow to reduce the weight [3-6]. In [7], Giallanza et al. show the study results that have allowed to optimize the hull geometry and the wing system of a hydrofoil capable to carry 250 passengers at a speed of 35 knots, operating among Sicilian island.

New technological solutions have also been used for the stabilization system in order to reduce the undesirable effects of roll motion of ships and, consequently, to increase the passenger comfort [8, 9]. In order to optimize the kinematic control of the flaps of modern hydrofoils, an innovative servo-mechanical system was developed in [10] constituted by a slider-crank mechanism that is driven by a double-acting hydraulic cylinder positioned above the waterline. This results in a reduction in the drag force and the optimization of hydrodynamic flows in the area connecting the wings to the strut, as well as a considerable reduction in vessel downtime. Recently, several studies are related on assessment of additive manufacturing potential in the maritime construction sector. It is found that the most interesting opportunities concern the weight reduction and the performance improvement [11, 12].

Now the technological challenge is focused on to the propulsion system thanks to the "green thrust" that requires significant further reductions in pollution levels and consequently an optimization of the system as well as the approach to ship fuel consumption monitoring. [13, 14].

There are currently on the market a number of ship propulsion systems that are based on the steering azimuth thrusters, but none of them are commercially available for light and fast ferries. In fact, the main thrusters on the market are realized for commercial applications on cargo vessel and maximum speeds from 18 to 20 knots. They are heavy, with a very large gearbox and not very efficient from a hydrodynamic point of view. The propeller and consequently the propeller hub are also of considerable size and, therefore, not suitable for fast applications. On the other hand, this type of propulsion can transmit very high torques and can reach a very long service life (greater than 20000 hours) before a total maintenance and repair program.

Volvo Penta's IPS contra-rotating propeller thrusters, or Mercruiser's ZEUS or ZF's POD are very light but, today, they are exclusively used in the yachting sector (up to 24 m hull length), with very high rotation speeds but very low torques. Moreover, the total life limit for such an application is very low: in many cases the total life of the component does not exceed 300 hours per year, for a total of 1500 hours. In the case of military or para-military fast patrol boats, with life limits of 1000/1500 hours per year and for a total of 4000-5000 hours, the use of hydrojets or surface propellers is preferred.

The studied thruster is realized in a C-drive configuration, with engine power close to 2000 kW and rotation speed close to 1000 rpm, with a first reduction obtained by a bevel gear positioned in an aluminum alloy box placed inside the hull, and a second reduction realized by bevel gears housed in a hydrodynamically profiled box at the lower end of the stern strut. In the lower gearbox will be installed the propeller shafts in a contra-rotating configuration.

The optimization of the studied thruster is directly related to the evaluation of important parameters such as efficiency, size and cost. Furthermore, it is important to add to these parameters assessments related to the strength and life of primary power transmission components such as, for example, gear wheels and bearings. The high loads could lead to premature damage of the transmission elements due to surface contact fatigue (resulting in increased noise and development of overheating) up to ultimate failure of the gear teeth. It is therefore essential to use a test bench in order to test the transmission components with the same loads that replicate the real operating conditions. The simplest idea is to directly connect the thruster to a test brake but, for high power, it is expensive and it determines excessive energy consumption; also, the space requirements and the installation costs are significant. In fact, when the rated power of the drive train is high, the test with power dissipation becomes complex and, often, it is impossible to realize. A high power motor and a high power braking unit must be used. Not only does the test equipment become expensive, but a great deal of power has to be dissipated in a hydraulic brake during the tests [15]. On the other hand, if a high power alternator is used, high power must be fed into the grid.

Among the various possibilities available on the market, close power loop test benchs are those that allow you to test transmissions with high rated power while containing operating costs. This kind of mechanical test bench requires the implementation of a back to back connection between two identical gearing (main and secondary, see Fig. 1).

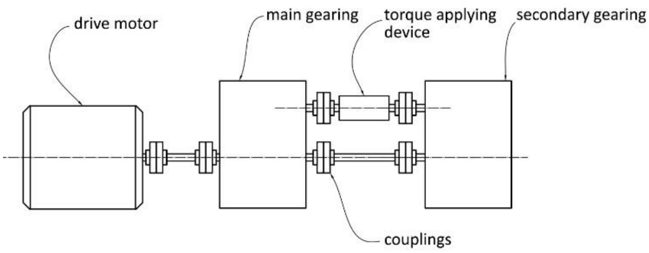


**Figure 1.** General layout of the mechanical close power loop test bench.

This configuration allows the motor to be sized on the power dissipated by the mechanisms. Since the efficiency of these systems is very high, there is no need for big motors and the operating costs can be kept down. The most significant disadvantage is the need to have two identical transmissions with a consequent increase in installation costs. One of the main needs in the testing phase is the introduction of a torque within the system that allows to simulate test conditions as similar as possible to those can be found over the service life of the gearing components. It is therefore necessary to use a mechanical device (torque applying device) that must be specially designed for the studied system.

## 2. State of the art of torque applying devices and close power loop test benches

The existing torque applying devices are very different from each other and mainly differ in totally mechanical and hydraulic systems. Purely mechanical systems (test bench with epicyclic gear train, Harald/Yano bench and Bader bench) are simple from a kinematic point of view but cause high efficiency losses or require complex auxiliary kinematics that cannot be implemented for high power systems and high rotation speeds. Consequently, the torque applying device in the studied test system must necessarily be hydraulic. Among these, the Collins device [16] uses a double-acting piston and a helical spline connection with opposite angles of inclination. The main problem of this configuration is attributable to the friction present on the sides of the splines which, being proportional to the induced torque, does not allow to predict the value of the moment applied by a simple pressure reading. This leads to an increase in efficiency losses and to an increase in energy consumption.

A variant of Collins' patented method is the Hennings test bench (Fig. 2) which, also in this case, uses a helical spline connection on the interconnecting shafts. A hydraulic piston moves the shaft on which one of the splines is fitted, generating an angular offset between the shafts where the test and slave wheels are mounted.

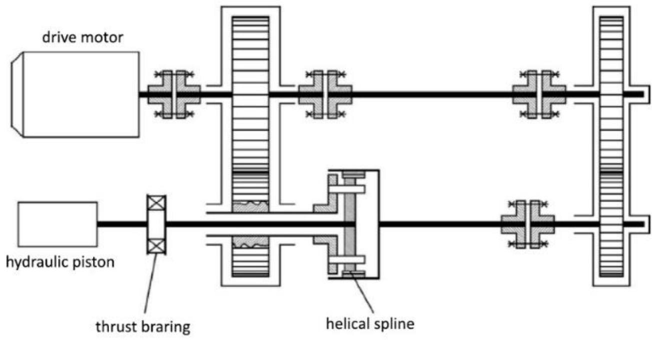


**Figure 2.** General layout of the Hennings test bench.

In order to carry out the experimental tests by means of a power recirculation system, Rodriquez Marine System has patented a device that is able to introduce a torque into a close power loop test bench. This system consists of two flanges that are angularly offset by four hydraulic actuators, while they simultaneously rotate with the input shafts. The cylinders are fed with oil under pressure through a rotating distributor equipped with rings rotating in sealing contact. The system therefore requires four actuators that rotate at a certain distance from the rotation axis. The actuators are connected to the two flanges by a system of hinges attached to the ends of the piston rods and cylinders (Fig. 3).

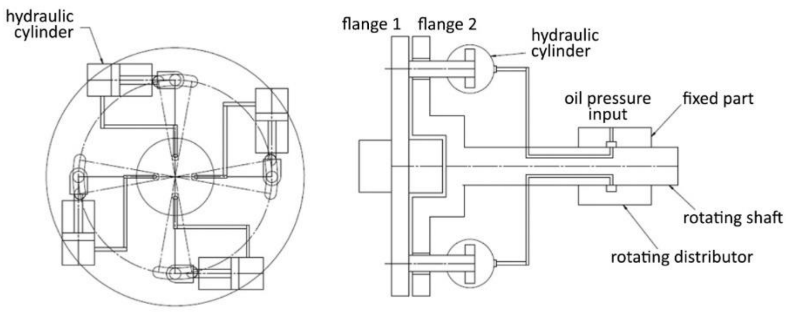


**Figure 3.** General layout of the torque applying device patented by Rodriquez Marine System.

The Rodriquez patent is complex from a kinematic point of view. This is essentially due to the use of four hydraulic cylinders that rotate at a certain distance from the rotation axis. The limit of this technique is the occurrence of high centrifugal forces on the hydraulic elements and high stresses on the connecting hinges.

## 3. Configuration of the studied thruster system

The thruster system involves two azimuthal PODs characterized by the presence of two contra-rotating propellers (Fig. 4).

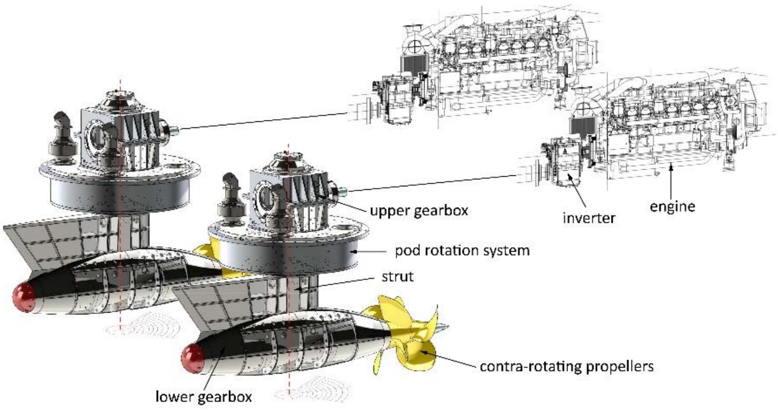


**Figure 4.** Layout of the thruster system.

In detail, the two engines of the HSC in which the PODs will be installed, through drive shafts placed inside the hull, transmit the power to two upper bevel gearboxes. The axes intersect in the gearboxes in an angle of 90° (Fig. 5).

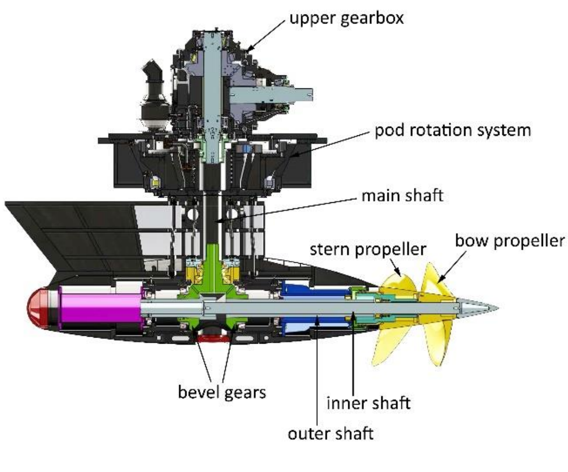


**Figure 5.** Kinematic configuration of the studied POD drive train.

At the output of each gearbox, a hollow shaft transmits motion to the lower part of the thruster passing through the two vertical struts. These struts can rotate around the vertical axis allowing the POD to direct oneself in the desired trajectory making it unnecessary to use the rudders. This system allows, therefore, to vary the angle between the direction of the propeller axis and the longitudinal axis of the HSC. The lower part of the PODs consists of a hydrodynamically profiled structure (lower gearbox in Fig. 4) inside which the mechanisms of the transmission are arranged. In particular, as can be seen in Fig. 5, the motion transmitted by the vertical shaft is transmitted, through bevel gears, to two coaxial drive shafts (one positioned inside the other) which, in turn, transmit the motion to the two contra-rotating propellers. The output speed of the inverter immediately downstream of the engine is equal to 1000 rpm (rotation speed of the engine equal to 2000 rpm with inverter gear ratio $\tau$=1:2). The transmission ratio of the upper bevel gearbox and bevel gears of the lower gearbox is $\tau$=1:1. Consequently, the rotation speed of the contra-rotating propellers is equal to 1000 rpm.

## 4. Engineering of the close power loop test bench for contra-rotating propellers

The studied test bench has been designed to support the components of the kinematic chain and accommodate all the devices and the auxiliary components necessary for the operation of the system. In addition to the thruster components, the test bench also consists of (Fig. 6):

- an electric motor necessary for the movement of the kinematic chain;
- two hydraulic systems, one for each engine, each equipped with screw pumps to ensure gear lubrication;
- an electrical panel equipped with all the instrumentation necessary to control the system's operating parameters;
- brackets that allow the connection of the thruster lower gearbox to the structure.

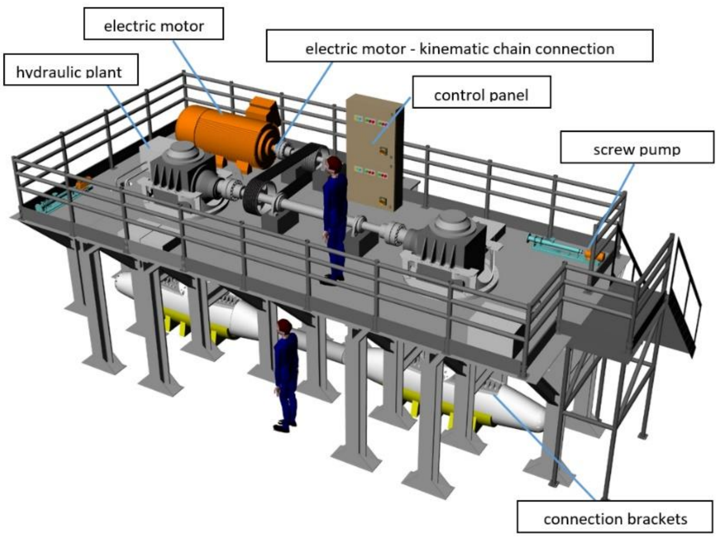


**Figure 6.** Rendering of the close loop test bench.

The structure has been dimensioned not only according to the loads to be supported but also to take into account the maneuvering space needed for the maintenance and the assembly of the mechanical connections. Moreover, the mechanical connections have been dimensioned according to the design speed, in order to guarantee power transmission. The work has been divided into three parts:

- study of the structure;
- study of the connection between electric motor and kinematic chain;
- study of torque applying device.

The fundamental design choices in the final design of the test bench are described below.

### 4.1. *Study of the structure*

The structure, consisting of IPE profiles and S275J0 steel plates welded together, occupies an area of 8397 x 4160 mm$^2$. The total height is equal to 2238 mm and the total weight of about 9500 kg (Fig. 7).

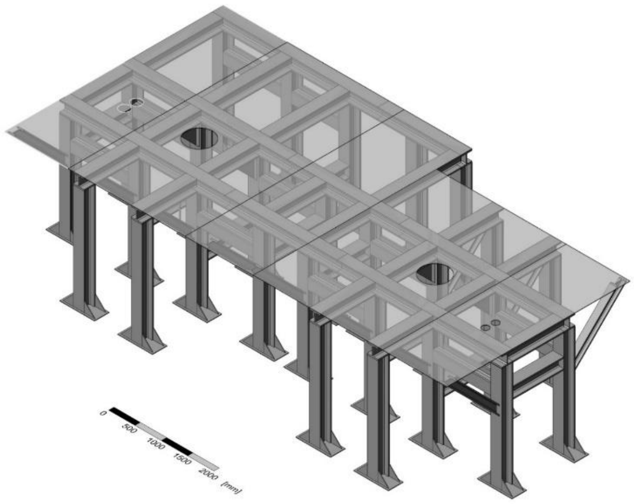


**Figure 7.** Isometric view of the test bench structure.

Width and depth have been selected in order to ensure the optimal distance between the two thruster and to ensure the correct distance between the output shaft of the electric motor and the connecting shaft between the two pods. The height of the structure, on the other hand, has been selected in order to easily insert the torque applying device. The structure has been designed in such a way that it can be divided into three modular blocks (Fig. 8), each of which does not occupy a surface area greater than 4000 X 4500 mm$^2$. In this way, easy transport and subsequent assembly is allowed. The profiles used are IPE 200M and IPE 120M.

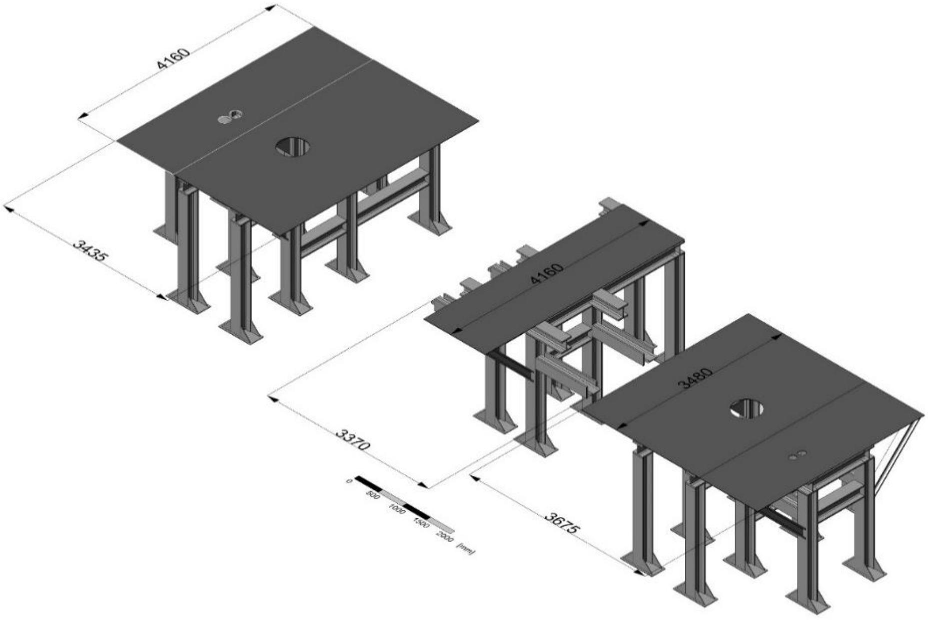


**Figure 8.** Subdivision of the structure into modules.

The structure has been reinforced near the support area of the upper gearboxes in order to support their weight and ensure their fixing. The connection between the lower gearboxes and the structure is realized by means of four brackets (Fig. 9), each of which is welded to the structure and bolted onto the lower gearboxes. The brackets made in such a way as to ensure alignment between the gearboxes and the structure and according to the holes required for gears lubrication.

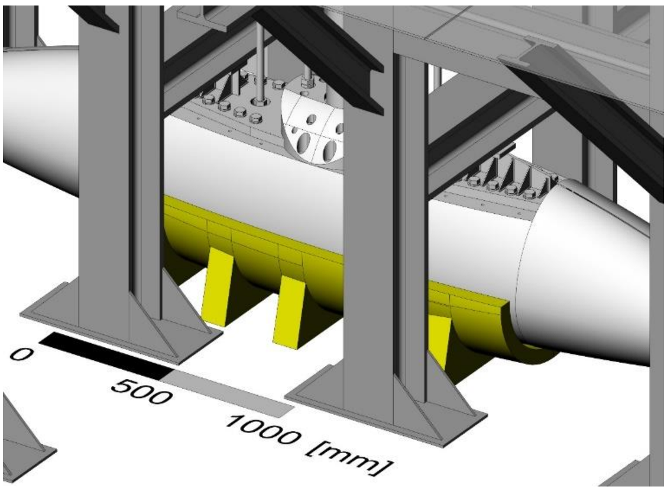


**Figure 9.** Detail of the connection brackets.

The entire structure was verified by means of a finite element analysis conducted in ANSYS Workbench environment [6, 10, 17]. The model was discretized into hexahedral elements (ex-dominant solid elements). Figs 10 and 11 show a detail of the discretization performed on the test bench structure.

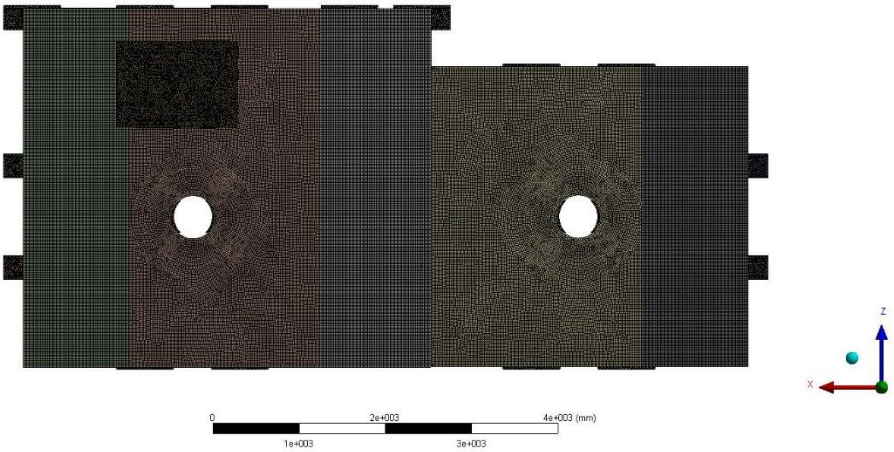


**Figure 10.** Detail of the discretization - top view.

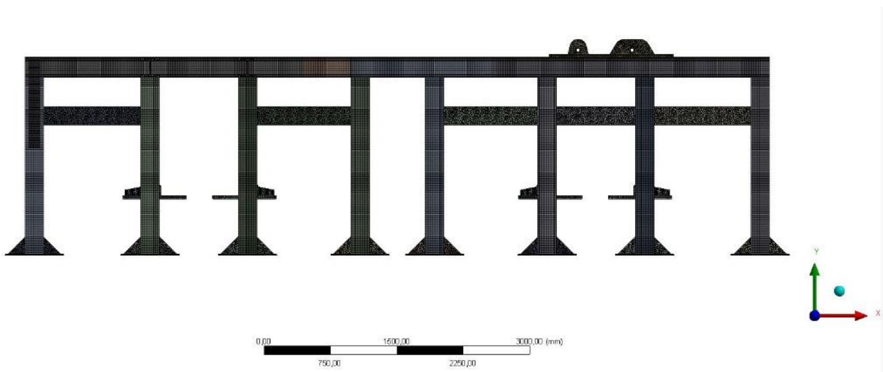

**Figure 11.** Detail of the discretization - front view.

It was chosen to verify the strength of the structure to the weight of the thrusters and of the electric motor, since the other masses are clearly lower than these. In particular, the weight force of a single thruster is equal to 10.5 kN, while that of the electric motor is equal to 25 kN. The weight of the electric motor was subdivided between the supporting surface and the lower surface of the connecting brackets. The weight force of the electric motor, on the other hand, was divided on the four shapes of the motor supports to the structure. *Fixed supports* were used to constrain the entire test bench structure to the base surface (Fig. 12).

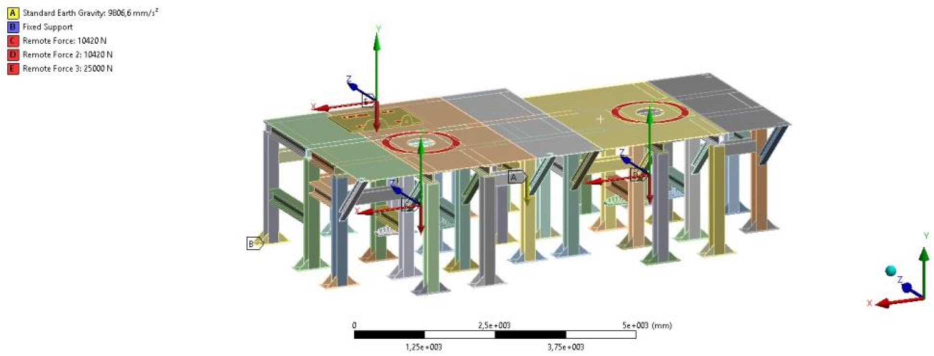


**Figure 12.** Loads and constraints in the numerical model.

Fig. 13 shows the equivalent Von Mises stress distribution. All values are lower than the yield stress value of the material, equal to 275 MPa. As shown in Fig. 13, the most stressed zone occurs in correspondence of the electric motor connection brackets.

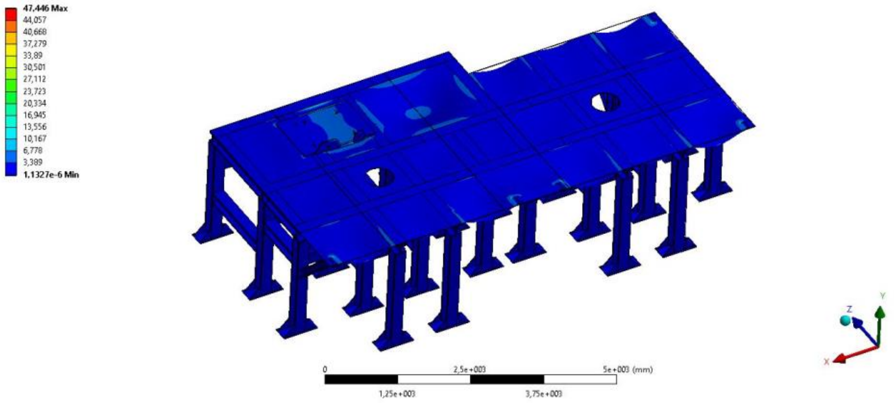


**Figure 13.** Von Mises stresses distribution in the test bench structure.

Fig. 14 shows the total deformation of the structure. The maximum displacement is about 0.65 mm and it is related to the structure's own weight. In fact, this value is located in a part of the structure that is not particularly reinforced causing the bending of the sheet metal. However, since the deformation value is negligible and it is located on an area where heavy loads are not expected to be placed, no further reinforcement of this area is expected.

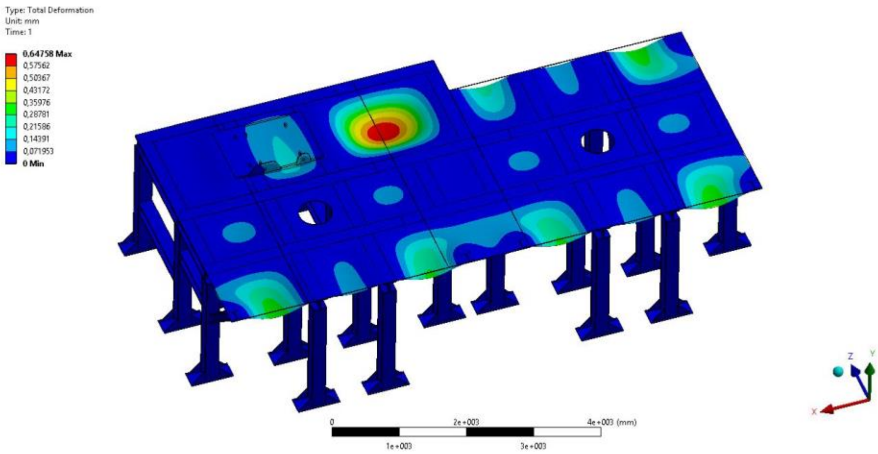


**Figure 14.** Total deformation of the structure.

The static numerical analysis was completed by a modal analysis in order to obtain the resonance frequencies and the associated modal deformations of the system. The study allowed to verify the goodness of the solutions adopted for the design of the test bench structure since all the operating frequencies are lower than the numerical modal frequencies.

*4.2. Study of the connection between electric motor and kinematic chain*

In order to transmit the mechanical power from the electric motor to the connected thrusters, it was decided to use a belt drive. The input power to the system is equal to 330 kW and the output speed is equal to 990 rpm. Fig. 15 shows a detail of the connection between the drive shaft and the test kinematic chain.

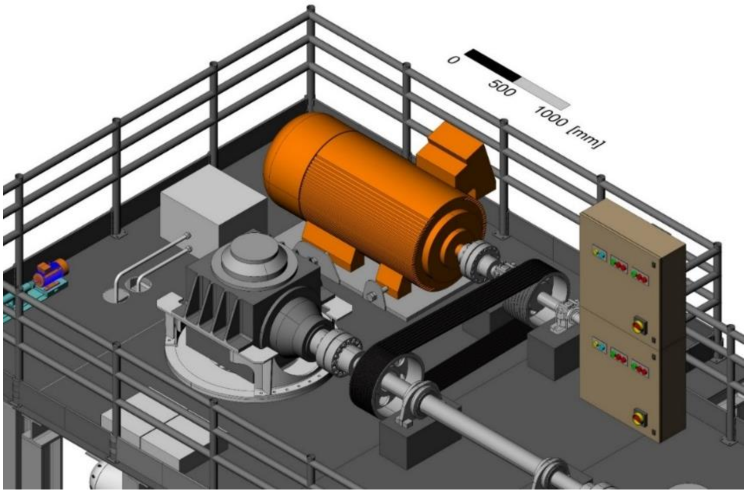


**Figure 15.** Power transmission with V-belts.

The drive pulley has a diameter of 228.6 mm while the driven pulley of 269.4 mm. The distance between the two shafts is equal to 1536 mm (see Figs 16 and 17). The study allowed to determine the optimal number of V-belts (equal to 8) that allow the transmission of power with a safety factor $n_f = 1.5$.

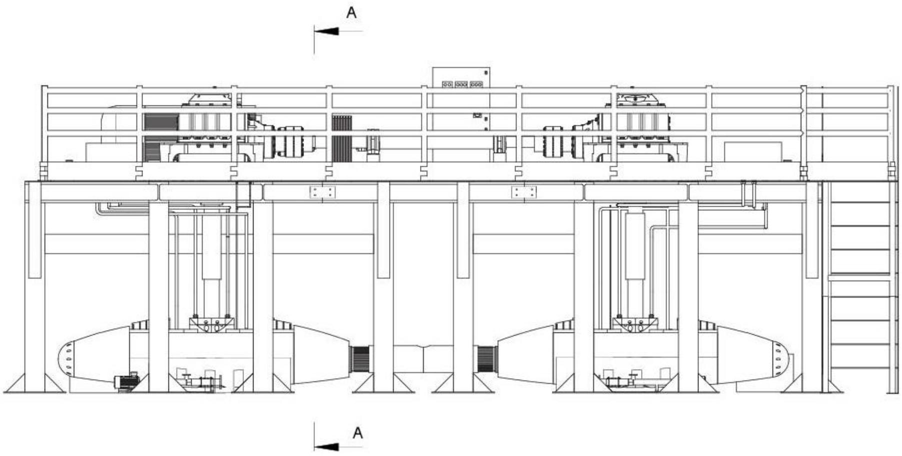


**Figure 16.** Front view of the test bench.

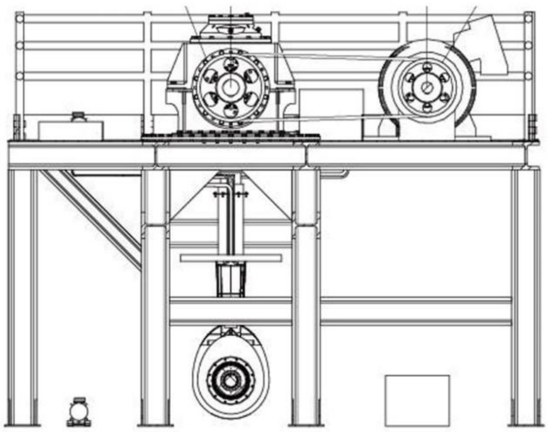

**Figure 17.** Section A-A.

A screw jack system was specially designed to ensure the preload of the belts. In particular, the rotation of two M20 tie rods allows the adjustment of the plate in which the motor is positioned and, consequently, the adjustment of the belts.

*4.3. Study of the torque applying device*

The general layout of the test bench requires that the electric motor transmits the mechanical power to the kinematic chain; at this stage, the power used in the system must only overcome the friction in the bearings and gear wheels.

When the kinematic chain is set in rotation at the design speed the bearings, the gears and all mechanical elements are suitably loaded by means of a torque applying device. An electro-hydraulic system connected to a PC allows the modulation of the loads in order to simulate real operation scenarios at maximum operating power.

The studied torque applying device differs from those previously analyzed in that the relative rotation of the end flanges, which generates the test torque, is achieved by means of an innovative kinematic system (patent application number 102019000011031). Fig. 18 shows a rendering of the device.

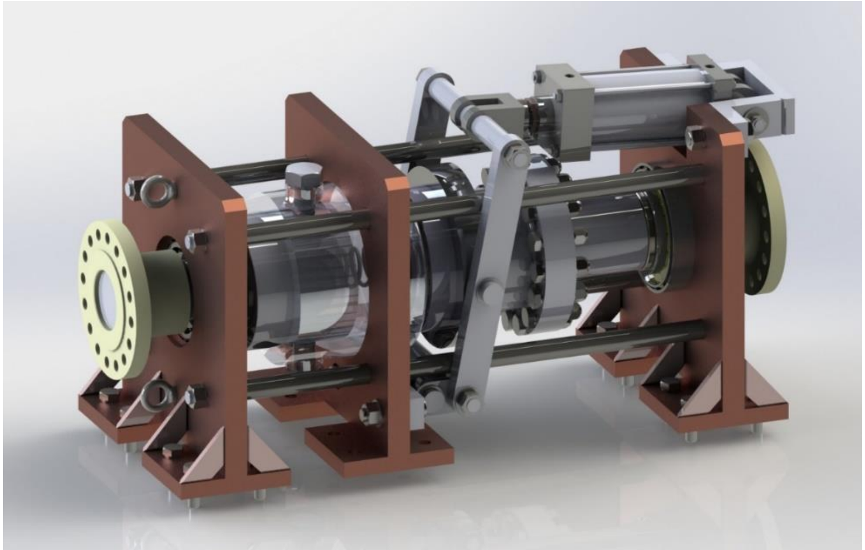

**Figure 18.** Rendering of the torque applying device.

The main advantages in the use of the studied device lie in the fact that all the components, due to the effect of rotation, are not subjected to significant centrifugal forces such as those present in the device patented by Rodriquez Marine System.

Moreover, the device does not need a rotating distributor because the hydraulic cylinder is positioned outside the machine, in a fixed position and not subject to any rotation. A leverage allows to generate the necessary thrust to realize the test torque. The system is more compact and less expensive because the hydraulic actuator can be chosen among those available on the market. The device has been engineered, kinematically and structurally verified in all its components.

Fig. 19 shows the installation phase of the two lower gearboxes and of the torque applying device in the test bench frame structure.

**Figure 19.** Installation phase of the two lower gearboxes and of the torque applying device in the test bench frame structure.

Several preliminary tests were conducted on the thrusters by means of the test bench. Figure 20 shows the trend of the power supplied by the electric motor and the PODs rotation speed as a function of the torque applied to the system by the torque applying device.

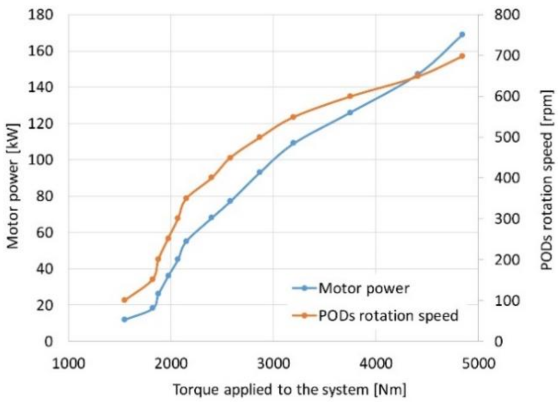


**Figure 20.** Preliminary experimental tests conducted on the test bench.

## 5. Discussions and conclusions

Through the test bench it has been possible to engineer the azimuth thruster, optimizing the design of each single component. In particular, the prototype of the azimuth thruster tested on the test bench was preliminarily manufactured on CNC machine tools, with a consequent high construction cost. For example, the lower gearbox case of the POD, which contains the kinematics of the contra-rotating transmission, was realized from a single block of aluminium alloy. The choice of this solution is due to the fact that in the preliminary design phase it was necessary to verify whether the gearbox structure was able to withstand the high mechanical stress resulting from the loads in operation.

The experimental tests conducted on the test bench have allowed to validate the goodness of the design solutions used for the realization of the azimuth thruster, to refine the production processes as well as to improve the technical and technological solutions to be used for the design of the new azimuth thrusters. For example, the frame of the PODs and the gearboxes cases will be made of cast EN 43300 T6 aluminium in order to reduce costs and speed up the production process. Fig. 21 shows the casting simulation of the lower gearbox.

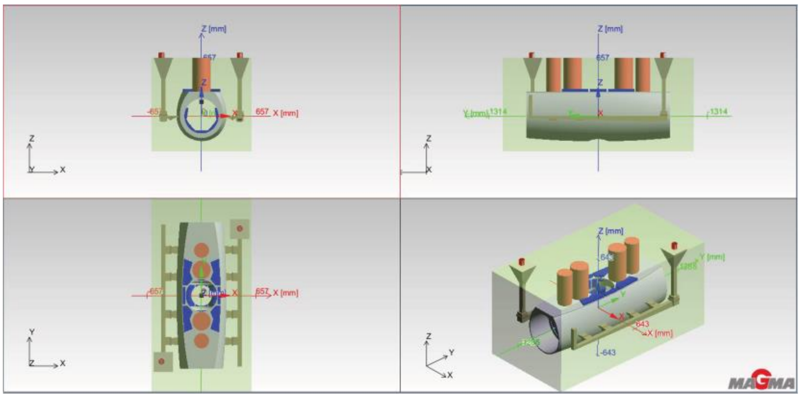


**Figure 21.** EU-STANDARD view of the casting technology

Fig. 22 shows several images of the casting model and the specially realized casting accessories.

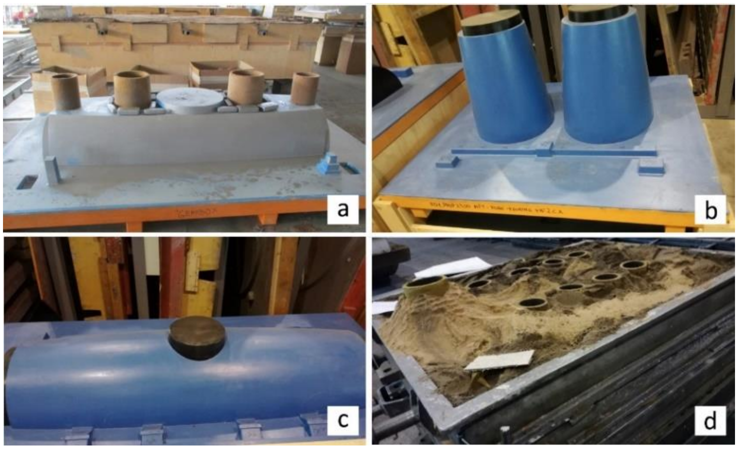


**Figure 22.** (a) Casting pattern - longitudinal view; (b) mould of the casting channels; (c) mould of the pattern; (d) pattern inside the brackets filled with sand.

In conclusion, the test bench has allowed the evaluation and optimization of all the most important parameters that determine goodness of the solutions adopted for the realization of the azimuth thruster. Through the experimental tests carried out on the test bench it was possible to verify the strength of the POD to operating loads, to increase its performance and efficiency as well as to reduce its size and cost.


## Acknowledgment

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