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Proyectos de fin de carrera de **Ingeniería Técnica Naval**

Tug Bollard Pull Test powered with a Voith Schneider Propeller by Computational Fluid Dynamics (CFD)

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Abstract

In the design phase of the dimensioning process of a ship, it is difficult to be precise about the thrust results in the propeller studies. The exact thrust that the propeller will produce and the thrust that will be lost toe to the viscous forces is, nowadays, known only when the vessel is already built and the test of thrust of the boats, called bollard pull test, is performed.

Thanks to the informatics development, the computational fluid dynamics is bringing better results every day. The importance of numerical simulations in various fields of application is by now widely accepted. There are in particular relevant cases where experiments are too costly, too dangerous or even impossible when the process to be simulated is not even observable in experiments. Therefore computational fluid dynamics (CFD) simulations it is an economic and reliable way to determinate the boat capacities before its construction.

This study is focused in simulating the bollard pull test of a concrete tug, check the results and from there analyse the zone requirements from the Classification Societies. These Societies made studies to determinate the requirements of the zone for the bollard pull test performance of boats with regular propellers (screw propeller), but it is not the only kind of existent propellers. In this study many simulations of a tug with a Voith Schneider Propeller will be run and analysed. The effect that makes the characteristics of the zone where the test have to be performed will be studied. Also the requirements of the Classifications Societies will be simulated for this tug with the VSP.

After analysing and comparing all the simulations results, possible requirements for the bollard pull test zone for boats with a VSP will be suggested.

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1. Introducción/Introduction

Introducción

En la actualidad el resultado de la fuerza de empuje de una embarcación se obtiene a través de las pruebas de tracción a punto fijo, ya que en el dimensionamiento del buque no existen aún buenas herramientas y sólo se puede hacer una mera estimación a través de unas formulas expuestas más adelante en este proyecto. Para llevar a cabo esta prueba se requiere que la embarcación ya esté construida, por lo cual al realizarla se puede comprobar si el barco cumple con los requerimientos de empuje que se tenían previamente establecidos. Si la embarcación no los cumple, se requerirá volver a la mesa de diseño para realizar los ajustes pertinentes para solucionar el problema, lo cual causa altos costes adicionales. Por esta razón, es muy prometedora la realización de una metodología que permita comprobar el empuje que tendrá la embarcación durante las etapas de diseño, lo que se consigue realizando una simulación donde se pueda predecir exactamente el comportamiento que se va a obtener durante las pruebas reales de tracción a punto fijo una vez construida la embarcación. Una metodología hace referencia al camino o al conjunto de procedimientos racionales utilizados para alcanzar el objetivo o la gama de objetivos que rige una investigación científica. Es decir, se trata de crear un conjunto de procedimientos que se puedan utilizar para cualquier propulsor conjuntamente con la embarcación correspondiente para obtener la fuerza de empuje exacta en las etapas de diseño. Pudiendo además revertir el proceso y obtener el propulsor más adecuado para el empuje deseado con cualquier embarcación.

El empuje es una fuerza que se produce cuando un sistema expulsa o acelera una determinada cantidad de masa en una dirección, y al realizar esta acción se genera una reacción de igual magnitud pero en sentido contrario tal como lo indica la tercera ley de Newton. Una embarcación se desplaza sobre el agua gracias a un dispositivo de propulsión, el cual normalmente es una hélice, aunque en esta parte del proyecto global, la atención ha sido puesta en el propulsor Voith Schneider. El estudio de propulsores se ha llevado a cabo desde hace años empleando distintos métodos, como el que se propone más adelante en esta parte del proyecto, donde en la mayoría de los casos es muy difícil realizar un análisis preciso. Gracias a los avances informáticos se está incrementado el uso de herramientas computacionales en distintos campos de estudios debido a la versatilidad que ofrecen, ya que es posible realizar modelos de estudio sin la necesidad de construirlo físicamente, lo que representa grandes ahorros económicos.

La condición de tracción a punto fijo en embarcaciones se utiliza como base para determinar la capacidad máxima de una embarcación para producir fuerza de empuje. Actualmente existe poca evidencia de trabajos dedicados a estudiar esta condición en propulsores mediante simulaciones computacionales, así pues se decidió estudiar esta condición. Los métodos analíticos para el estudio del comportamiento hidrodinámico son muy complicados para hacer cálculos en forma manual. Es por este motivo que en los últimos años las grandes empresas de diseño y centros de investigación han implementado la utilización de programas especializados en dinámica de fluidos computacional para el estudio de diversos fenómenos en distintas áreas de la ingeniería. La dinámica de fluidos computacional (CFD por sus siglas en inglés) se está empleando en distintas investigaciones para el análisis de fluidos, obteniendo buenos resultados muy cercanos a los reales. Se está empleando en diversos estudios del área naval, donde se han introducido métodos de solución en CFD tales como Reynolds Averaged Navier-Stokes (conocidos como RANS), que también se aplican en el método de desarrollo de este proyecto, los cuales permiten la solución de las complejas ecuaciones de Navier-Stokes, donde en la mayoría de las ocasiones no es posible resolverlas de forma manual. También el Lepum (Laboratorio de investigación especializado en procesos de unión y manufactura de la Universidad de Panamá, donde se estudia la simulación de procesos ingenieriles: estructural, fluidos, trasferencia de calor, mecánica de fallas, entre otros) decidieron analizar el problema que representa no poder conocer exactamente en la fase de diseño la capacidad de una embarcación, y a partir del problema se está llevando a cabo este proyecto de estudio de propulsores en condición de tracción a punto fijo con el objetivo de obtener la fuerza de empuje exacta de cualquier embarcación antes de construirla mediante una metodología a partir de la dinámica de fluidos computacional.

Como ya he mencionado anteriormente, este proyecto de fin de grado se enmarca dentro de un trabajo de investigación más complejo, es la cuarta parte de un estudio que lleva dos años haciéndose en el LEPUM de la Universidad Tecnológica de Panamá. Actualmente en el LEPUM están trabajando en cinco proyectos de investigación con la SENACYT (Secretaría Nacional de Ciencia, Tecnología e Innovación de la República de Panamá) y muchos otros proyectos de asesoría a empresas privadas. Así pues, este es una parte de uno de los proyecto del Laboratorio LEPUM y está supervisado por el Dr. Adán Vega.

En un primer lugar, Josué René Hernández hizo para su tesis la metodología para la simulación de un remolcador con una hélice para obtener de manera precisa el empuje que tendría una embarcación en la fase de diseño, y así poder predecir y determinar la configuración adecuada de hélice-tobera que se debería establecer para lograr las características deseadas. En la validación de los resultados con los datos teóricos obtuvo un error por debajo de 0,6%. Además también estudió las distancias establecidas por las Sociedades de Clasificación consiguiendo resultados que determinan que esas distancias son las ideales para hacer la prueba de tiro fijo para propulsores con hélices regulares. A continuación, David López Martínez, utilizó la misma metodología para obtener el empuje para un remolcador con dos hélices. Más adelante, Joan Pau Anguera, empezó la metodología para un remolcador con un propulsor Voith Schneider. Él diseño los planos del nuevo propulsor y lo complemento con el remolcador utilizado en los estudios anteriores en el programa CAD Inventor y luego lo importó al programa Ansys Fluent. Una vez en el programa Ansys Fluent, hubo problemas y no pudo acabar. Desde allí empecé yo, buscando el problema que había paralizado el proyecto, solucionándolo y volviendo a programar desde el principio las consideraciones debidas para la prueba de tiro a punto fijo en el programa Ansys Fluent de CFD. Una vez conseguido que las ecuaciones del Ansys convergieran por debajo de 10^-3, que se consideran buenos resultados por convenio, se procede con la validación del modelo con resultados de empuje de buques reales. No es posible calcular los resultados teóricos como lo hizo Josué, porque ciertos gráficos de coeficientes necesarios de este proulsor no están disponibles.

Como ya se ha mencionado anteriormente, las Sociedades de Clasificación tienen unos requerimientos para las pruebas de tracción a punto fijo, pero estos fueron determinados a partir de estudios hechos para embarcaciones con un propulsor convencional. Una vez validado el modelo de estudio, pasé a estudiar el efecto de las características de la zona sobre los resultados de la prueba de tracción a punto fijo del remolcador con el VSP para tener el conocimiento de qué parámetros influencian y en qué medida en estos. Seguidamente se estudian simulaciones hechas con los requerimientos de zona que describen las Sociedades de Clasificación y se comparan con los resultados de los estudios anteriores. Finalmente se sugieren unas nuevas características para la zona de la prueba cuando se trate de embarcaciones con propulsión del tipo Voith Schneider y se hacen simulaciones con las características sugeridas para comprobar la efectividad del estudio de la zona.

Como objetivos específicos de esta tesis se pueden mencionar:

1. Estudiar la teoría para tener una comprensión clara del fenómeno que se va a estudiar
2. A partir del diseño de los componentes de la embarcación a través de programas especializados CAD, competentes con el programa CFD a utilizar (Ansys Fluent), formular un modelo que permita evaluar el empuje de embarcaciones en condición de tracción a punto fijo.
3. Simular el procedimiento de tracción a punto fijo utilizando dinámica de fluidos computacional y obtener resultados.

4. Validar los resultados de la prueba realizada mediante CFD con resultados de reales.
5. Estudiar como influyen las características de la zona de la prueba en la estimación del empuje de la embarcación.
6. Estudiar los resultados de simulaciones que siguen los requerimientos de zona de las Sociedades de Clasificación.
7. Proponer requerimientos de zona para pruebas con embarcaciones con propulsor Voith Schneider.
8. Simular la prueba con los nuevos requerimientos de zona para comprobarlos y validarlos.

Introduction

Nowadays the test of thrust of the boats is made through the called bollard pull test, because in the dimensioning phase of the boat project it does not exist yet good tools to measure it, so it is only possible to make an estimation through equations, which are also explained in this project. To perform this test it is required that the boat is built. It will be checked if the ship meets the requirements of thrust that were previously established. If the boat does not produce the established thrust, it will require return to the drawing board to solve the problem, which causes high additional costs. It is therefore very promising to create a methodology to check the thrust of the boat during the design stages that predicts the behaviour during the actual bollard pull test. A methodology is the path or set of rational procedures used to achieve the target or range of targets governing scientific research. So, this project it is about creating a set of procedures that can be used for any propeller together with the corresponding vessel to obtain the exact thrust force in the design stages. Can also reverse the process and get the right propeller for the concrete desired thrust.

Thrust is a force that occurs when a system expels or accelerates a certain amount of mass in one direction, and a reaction of equal magnitude but opposite direction is generated as indicated by Newton's third law. A boat moves on the water thanks to a propulsion device, which is normally an helix, though in this part of the global project, the attention it has been placed on the Voith Schneider Propeller. Helices studies have been conducted for years using different methods, like the one proposed later on in this project, which in most cases are almost impossible to perform an accurate analysis. Thanks to the advances in computer technology, the use of computational tools in different fields of study is increasing because of the versatility that they offer, since it is possible to study models without the need to physically build it, which represents huge cost savings.

The condition of bollard pull in vessels is used as the basis for determining the maximum capacity of a vessel to produce thrust force. There is currently little evidence of studies dedicated to this propellers condition by computer simulations, therefore it was decided to study this condition. The analytical methods for studying the hydrodynamic behaviour are very difficult to make calculations manually. Therefore, recently, big design firms and research centres have implemented the use of specialized programs in computational fluid dynamics to study various phenomena in different areas of engineering. Computational fluid dynamics (CFD) is being used in various investigations for analysing fluids with good results, very close to the real. It is being used in several studies of the naval area, where they have introduced methods of solution CFD such as Reynolds Averaged Navier-Stokes (known as RANS), which are also used in the methodology of this project, allow the solution of the complex Navier-Stokes equations, which in most cases cannot be resolved manually. Also in LEPUM (Research laboratory specializing in manufacturing and bonding processes of the Technological University of Panama, where the simulation of engineering processes is studied: structural, fluids, heat transfer, mechanical failure, etc.) decided to examine the problem of not being able to meet exactly in the design phase the thrust of a boat, and from the problem this study is propellers study project in the bollard pull condition is being conducted in order to get the exact force of any vessel before building it using a methodology based on computational fluid dynamics.

As I mentioned earlier, this final degree project is part of a complex research work, is the fourth part of a study that is being done already since two years in the LEPUM of the Technological University of Panama. Currently in LEPUM they are working in five research projects of the SENACYT (National Secretariat for Science, Technology and Innovation of the Republic of Panama) and many other advisory projects of private companies. So this is a part of one of the LEPUM Laboratory ptojects and is supervised by Dr. Adam Vega.

At the beginning of the full project, Joshua Rene Hernandez, made the methodology for the simulation of a tugboat with a helix to specify the thrust that the ship will have, and thus be able to predict and determine the proper propeller-nozzle that must be set to achieve the desired characteristics. In the validation of the results with theoretical data it obtained an error below 0.6%. In addition he also studied the distances of the test zone required by the Classification Societies getting results that determine that these distances are ideal for testing the thrust of regular propellers in the bollard pull test. Then David Lopez Martinez used the same methodology for a tugboat with two propellers and analysed the results. Later, Joan Pau Anguera began a methodology for a tug with Voith Schneider Propeller. He designed the drawings for the new propeller and used the same tug as in the previous studies in the program CAD Inventor and then imported the files to the CFD program Ansys Fluent. Once in Ansys Fluent, there were problems and he could not finish. From there I started searching the problem that paralysed the project, solving it and reprogramming the considerations for the bollard pull test in the Ansys Fluent of CFD. Having achieved that the equations of Ansys converged below 10 ^ -3, by convention the results are considered good, I could proceed with the validation of the model with results from real ships. It is not possible to calculate the theoretical results as Joshua did, because certain needed coefficient graphics are not available for this propeller.

As already said, Classification Societies have few requirements for the bollard pull test, but these were determined from studies done for boats with a conventional propeller. Then, once I validated the model, I went on studying the effect that the characteristics of the area make to the results of bollard pull test of a tug with a VSP. Following, simulations with the area requirements of classification societies are studied and compared with the results of the mentioned previous studies. Finally some new features for the test area in the case of vessels with Voith Schneider propulsion type are suggested and simulations using the suggested features to test area are made to check the effectiveness of the study.

The specific objectives of this part of the project can be mentioned as:

1. Study the theory to have a clear understanding of the studied phenomenon.

2. From the design of the vessel components through specialized CAD programs, competent with the CFD program to be used (Ansys Fluent), develop a model to assess the thrust of boats with a VSP in bollard pull condition.

3. Simulate the bollard pull procedure using computational fluid dynamics and obtain results.

4. Validate the results of the test performed by CFD with actual results.

5. Study how the characteristics of the test area influence in estimating the thrust of the boat.

6. Study the results of simulations that follow the area requirements of the Classification Societies.

7. Propose area requirements for testing ships with a Voith Schneider Propeller.

8. Simulate the test with the new requirements of area to check and validate the suggestions.

2. Fluid Dynamics

Fluid Dynamics is a sub discipline of Fluid Mechanics that deals with fluid flow, the natural science of fluids (liquids and gases) in motion, that is to say, it is the part of physics that studies the motion of fluids and the forces that cause it. The main characteristic of the fluids is their inability to resist shear stresses; as a result they have no definite form. To understand and use well the fundamental equations of fluid you must first define the two types of fluids, since their behaviour is different.

The fluids are classified as Newtonian and non-Newtonian; in the graphic we can see the fluids behaviour. Newtonian fluids have a lineal behaviour.

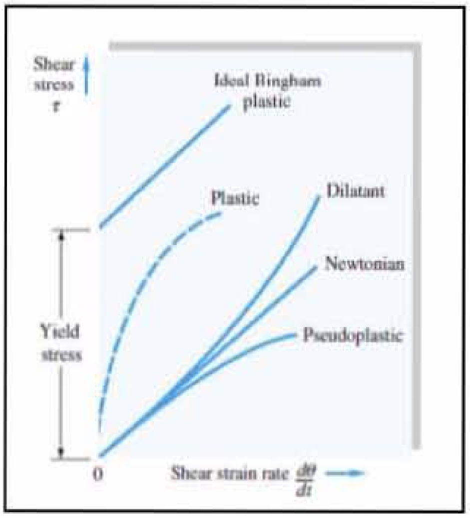


Image 1 - Rheological diagram from wikipedia

· Newtonian fluids are characterized by a direct proportionality between the shear stress and friction velocity gradient. As it is the case of water. The viscosity is constant over time.

· No-Newtonian fluids are characterized by viscosity variation depending on the voltage gradient applied to it. That is, a constant coefficient of viscosity cannot be defined.

For fluids which are sufficiently dense to be a continuum, do not contain ionized species, and have flow velocities small in relation to the speed of light, the momentum equations for Newtonian fluids are the Navier–Stokes equations, later explained in this chapter. Navier-Stroked equations is a non-linear set of differential equations that describes the flow of a fluid whose stress depends linearly on flow velocity gradients and pressure. The not simplified equations do not have a general closed-form solution, so they are primarily of use in Computational Fluid Dynamics. The equations can be simplified in a number of ways.

The solution to a fluid dynamics problem typically involves calculating various properties of the fluid, such as flow velocity, pressure, density, and temperature, as functions of space and time.

2.1 Fluid dynamics equations

The foundational axioms of fluid dynamics are the conservation laws, specifically: conservation of mass, conservation of linear momentum(also known as Newton's Second Law of Motion), and conservation of energy (also known as First Law of Thermodynamics).

In addition to the above, fluids are assumed to obey the *continuum assumption*. Fluids are composed of molecules that collide with one another and solid objects. However, the continuum assumption considers fluids to be continuous, rather than discrete. Consequently, properties such as density, pressure, temperature, and flow velocity are taken to be well defined at infinitesimally small points, and are assumed to vary continuously from one point to another. The fact that the fluid is made up of discrete molecules is ignored.

A Fluid is a substance, which deforms continuously, or flows, when subjected to shearing forces. If a fluid is at rest there are no shearing forces acting. All forces must be perpendicular to the planes, which are acting. When a fluid is in motion shear stresses are developed if the particles of the fluid move relative to one another. When this happens adjacent particles have different velocities. If fluid velocity is the same at every point then there is no shear stress produced: the particles have zero relative velocity.

In addition to the mass, momentum, and energy conservation equations, a thermodynamic equation of state giving the pressure as a function of other thermodynamic variables for the fluid is required to completely specify the problem.

2.2 Conservation laws

Three conservation laws are used to solve fluid dynamics problems. Mathematical formulations of these conservation laws may be interpreted by considering the concept of a *control volume*. A *control volume* is a specified volume in space through which air can flow in and out. Integral formulations of the conservation laws consider the change in mass, momentum, or energy within the control volume.

- ***Conservation of mass***

  The rate of change of fluid mass inside a control volume must be equal to the net rate of fluid flow into the volume. Physically, this statement requires that mass is neither created nor destroyed in the control volume, and can be translated into the integral form of the continuity equation:

$$\frac{\partial}{\partial t}\int\int\int_{v}\rho dV = -\oiint_{s}\rho u \cdot dS$$

Above, ρ is the fluid density, $\vec{V}$ is the flow velocity vector, and t is time. The differential form of the continuity equation is, by the divergence theorem:

$$\frac{D\rho}{Dt} + \nabla \cdot \left(\rho\vec{V}\right) = 0$$

Where,

$$\frac{D\rho}{Dt} = \frac{\partial\rho}{\partial t} + u\frac{\partial\rho}{\partial x} + v\frac{\partial\rho}{\partial y} + w\frac{\partial\rho}{\partial z}$$

$$\nabla \cdot \left(\vec{V}\right) = \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z}$$

- ***Conservation of momentum***

This equation applies Newton's second law of motion to the control volume, requiring that any change in momentum of the air within a control volume be due to the net flow of air into the volume and the action of external forces on the air within the volume. In the integral formulation of this equation, body forces here are represented by f_{body}, the body force per unit mass. Surface forces, such as viscous forces, are represented by F_{surf}, the net force due to stresses on the control volume surface.

$$\frac{\partial}{\partial t}\int\int\int_V \rho u dV = -\oiint_S (\rho u \cdot dS)u - \oiint_S p dS + \iiint_V \rho f_{body} dV + F_{surf}$$

The differential form of the momentum conservation equation is as follows. Here, both surface and body forces are accounted for in one total force, F.

$$\frac{Du}{Dt} = F - \frac{\nabla p}{\rho}$$

The equation above is a vector equation: in a three-dimensional flow, it can be expressed as three scalar equations. The conservation of momentum equations for the compressible, viscous flow case are called the Navier–Stokes equations, which are explained later in this chapter.

- ***Conservation of energy***

Although energy can be converted from one form to another, the total energy in a given closed system remains constant.

$$\rho \frac{Dh}{Dt} = \frac{Dp}{Dt} + \nabla \cdot (k \nabla T) + \Phi$$

Above, *h* is enthalpy, *k* is the thermal conductivity of the fluid, *T* is temperature, and Φ is the viscous dissipation function. The viscous dissipation function governs the rate at which mechanical energy of the flow is converted to heat. The second law of thermodynamics requires that the dissipation term is always positive: viscosity cannot create energy within the control volume. The expression on the left side is a material derivative.

2.3 Navier-Stokes equations

In physics, the Navier-Strokes equations, named after Claude-Louis Navier and Geoier-Strokerge Gabriel Strokes, describe the motion of viscous fluid substances. They represent the base of Fluid Mechanics. These balance equations arise from applying Newton's second law to fluid motion, together with the assumption that the stress in the fluid is the sum of a diffusing viscous term (proportional to the gradient of velocity) and a pressure term—hence describing *viscous flow*. The main difference between them and the simpler Euler equations for inviscid flow is that Navier–Stokes equations also in the Froude limit (no external field) are not conservation equations (but rather a dissipative system) in the sense they can not be put into the almost linear homogeneous form:

$$y_t + A(y)y_x = 0$$

Navier–Stokes equations are useful because they describe the physics of many things of scientific and engineering interest. They may be used to model the weather, ocean currents, water flow in a pipe and airflow around a wing. The Navier–Stokes equations in their full and simplified forms help with the design of aircraft and cars, the study of blood flow, the design of power stations, the analysis of pollution, and many other things. Coupled with Maxwell's equations they can be used to model and study magneto hydrodynamics.

2.4 Characteristics of fluids

- ***Compressible vs. incompressible flow***

All fluids are compressible to some extent, that is, changes in pressure or temperature will result in changes in density. However, in many situations the changes in pressure and temperature are sufficiently small that the changes in density are negligible. In this case the flow can be modelled as an incompressible flow. Otherwise the more general compressible flow equations must be used. Mathematically, incompressibility is expressed by saying that the density ρ of a fluid parcel does not change as it moves in the flow field, i.e.,

$$\frac{D\rho}{Dt} = 0$$

Where *D/Dt* is the substantial derivative, which is the sum of local and convective derivatives. This additional constraint simplifies the governing equations, especially in the case when the fluid has a uniform density.

- ***Inviscid vs. Newtonian and non-Newtonian fluids***

Viscous problems are those in which fluid friction has significant effects on the fluid motion. The Reynolds number, which is a ratio between inertial and viscous forces, can be used to evaluate whether viscous or inviscid equations are appropriate to the problem. Stokes flow is flow at very low Reynolds numbers, *Re*<<1, such that inertial forces can be neglected compared to viscous forces. On the contrary, high Reynolds numbers indicate that the inertial forces are more significant than the viscous (friction) forces. Therefore, we may assume the flow to be an inviscid flow, an approximation in which we neglect viscosity completely, compared to inertial terms. Viscosity often cannot be neglected near solid boundaries because the no-slip condition can generate a thin region of large strain rate (known as Boundary layer) which enhances the effect of even a small amount of viscosity, and thus generating vorticity.

Therefore, to calculate net forces on bodies, such are propellers, we should use viscous flow equations. The standard equations of inviscid flow are the Euler equations. Another often used model, especially in computational fluid dynamics, is to use the Euler equations away from the body and the boundary layer equations, which incorporates viscosity, in a region close to the body.

The Euler equations can be integrated along a streamline to get Bernoulli's equation, both set forth below. When the flow is everywhere irrotational and inviscid, Bernoulli's equation can be used throughout the flow field. Such flows are called potential flows.

Incompressible Euler equation with constant and uniform density (convective form)

$$\rightarrow \quad \frac{\partial u}{\partial t} + u \cdot \nabla u + \nabla w = g$$

$$\nabla \cdot u = 0$$

Where:

- ***u*** -> is the flow velocity vector, with components in a N-dimensional space u_1, u_2 ... u_N,
- $\cdot$ -> Denotes the scalar product,
- ∇ -> is the Nabla operator, here used in the flow velocity divergence (first equation), and in flow velocity and specific pressure gradients (second equation)
- $u \cdot \nabla$ -> is the convective operator,
- *w* -> is the specific (with the sense of *per unit mass*) thermodynamic work, the internal source term.
- g -> represent the body accelerations (per unit mass) acting on the continuum, for example gravity, inertial accelerations, electric field acceleration, and so on.

$$Bernoulli\ ec.\ \rightarrow \frac{V^2\rho}{2} + P + \rho g z = ct$$

Where,

- *v* -> is the fluid flow speed at a point on a streamline,
- g -> is the value of acceleration due to gravity,
- z -> is the elevation of the point above a reference plane, with the positive z-direction pointing upward – so in the direction opposite to the gravitational acceleration,
- P -> is the pressure at the chosen point, and
- ρ -> is the density of the fluid at all points in the fluid.

Sir Isaac Newton showed how stress and the rate of strain are very close to linearly related for many familiar fluids, such as water and air. A viscosity constant models these Newtonian Fluids, depending only on the specific fluid.

- ***Steady vs. unsteady flow***

Steady-state flow refers to the condition where the fluid properties at a point in the system do not change over time. This roughly means that all statistical properties are constant in time. Otherwise, flow is called unsteady (also called transient). Whether a particular flow is steady or unsteady, can depend on the chosen frame of reference. In a frame of reference that is stationary with respect to a background flow, the flow is unsteady. Turbulent flows are unsteady by definition. A turbulent flow can, however, be statistically stationary.

- ***Laminar vs. turbulent flow***

Turbulence is flow characterized by recirculation, eddies, and apparent randomness. Flow in which turbulence is not exhibited is called laminar. It should be noted, however, that the presence of eddies or recirculation alone does not necessarily indicate turbulent flow—these phenomena may be present in laminar flow as well.

It is believed that turbulent flows can be described well through the use of the Navier–Stokes equations. Direct numerical simulation (DNS), based on the Navier–Stokes equations, makes it possible to simulate turbulent flows at moderate Reynolds numbers. Restrictions depend on the power of the computer used and the efficiency of the solution algorithm.

In order to solve these real-life flow problems, turbulence models will be a necessity for the foreseeable future. Reynolds-averaged Navier-Strokes equations (RANS) combined with turbulence modelling provides a model of the effects of the turbulent flow. Such a modelling mainly provides the additional momentum transfer by the Reynolds stresses, although the turbulence also enhances the heat and mass transfer.

The RANS equations (Reynolds-averaged Navier-Stokes equations) are time-averaged equations of motion for fluid flow, primarily used to describe turbulent flows. The idea behind the equations is Reynolds decomposition, whereby an instantaneous quantity is decomposed into its time-averaged and fluctuating quantities, an idea first proposed by Osborne Reynolds. These equations can be used with approximations based on knowledge of the properties of flow turbulence to give approximate time-averaged solutions to the Navier–Stokes equations. For a stationary, incompressible Newtonian fluid, these equations can be written in Einstein notation as:

$$\rho\bar{u}_j\frac{\partial\bar{u}_i}{\partial x_j}=\rho\bar{f}_i+\frac{\partial}{\partial x_j}\left[-\bar{p}\delta_{ij}+\mu\left(\frac{\partial\bar{u}_i}{\partial x_j}+\frac{\partial\bar{u}_j}{\partial x_i}\right)-\rho\overline{u_i'u_j'}\right]$$

- ***Subsonic vs. transonic, supersonic and hypersonic flows***

While many terrestrial flows (e.g. flow of water through a pipe) occur at low Mach numbers, many flows of practical interest (e.g. in aerodynamics) occur at high fractions of the Mach Number M=1 or in excess of it (supersonic flows). New phenomena occur at these Mach number regimes (e.g. shock waves for supersonic flow, transonic instability in a regime of flows with M nearly equal to 1, non-equilibrium chemical behaviour due to ionization in hypersonic flows) and it is necessary to treat each of these flow regimes separately.

2.5 Static, dynamic and Total Pressure

A pressure can be identified for every point in a body of fluid, regardless of whether the fluid is in motion or not. Pressure can be measured using an aneroid, Bourdon tube, mercury column, or various other methods.

The concepts of *total pressure* and *dynamic pressure* arise from Bernoulli's equation and are significant in the study of all fluid flows. (These two pressures are not pressures in the usual sense - they cannot be measured using an aneroid, Bourdon tube or mercury column.) To avoid potential ambiguity when referring to pressure in fluid dynamics, many authors use the term *static pressure* to distinguish it from *total pressure* and *dynamic pressure*. *Static pressure* is identical to pressure and can be identified for every point in a fluid flow field, it is the pressure at a point on a body moving with the fluid, the actual pressure of the fluid associated with its rate instead of with its motion.

Bernoulli's equation is fundamental to the dynamics of incompressible fluids. In many fluid flow situations of interest, changes in elevation are insignificant and can be ignored. With this simplification, Bernoulli's equation for incompressible flows can be expressed as

$$P + \frac{1}{2}\rho v^2 = P_0$$

Static pressure + dynamic pressure = total pressure

Where,

- P - > is static pressure,
- $\frac{1}{2}\rho v^2$ -> is dynamic pressure,
- ρ -> is the density of the fluid,
- v -> is the flow velocity, and
- P_0 -> is total pressure, also known as the Stagnation Pressure.

Every point in a steadily flowing fluid, regardless of the fluid speed at that point, has its own static pressure, dynamic pressure, and total pressure. Static pressure and dynamic pressure are likely to vary significantly throughout the fluid but total pressure is constant along each streamline. In irrotational flow, total pressure is the same on all streamlines and is therefore constant throughout the flow.

2.6 Hydrodynamic forces

To study this case it is also important to take into account two types of hydrodynamic forces acting on the surfaces; forces due to normal stresses produced by the pressure distribution on the surfaces and the forces produced by viscous stresses and shear dependent viscosity of the fluid. The propeller in rotational movement produces a pushing force while a force is exerted on it. The movement of the fluid exerts forces on the other components of the tug as the hull, the keel stern or bow skate. These surface forces can be calculated by integration:

$$\sum F = \int \bar{\bar{\sigma}}\vec{n}\,dA$$

Where,

- $\bar{\bar{\sigma}}$ is the stress tensor

- $\vec{n}$ is a unit normal vector dA.

- dA is a differential element of area .

Total thrust component is obtained by adding the pressure component and viscous component forces in the forward direction of the boat.

$$F_T = F_P + F_V$$

2.7 Study case

For this study I am going to use seawater in standard conditions, that means at 20°C with 1025Kg/m^3 density and 0,001077 Kg/m dynamic viscosity.

Seawater is a Newtonian fluid, therefore has a linear behaviour. In such fluids the viscous stress is defined by the following equation:

$$\tau = \mu \frac{d\theta}{dt}$$

Where μ is the dynamic viscous and $d\theta/dt$ is the deformation ratio.

We will consider the fluid in the study case isothermal, because the temperature changes in the fluid of the bollard pull test are small. That way is not necessary to use the variable Temperature, which makes the energy conservation law not necessary. Moreover, because the sea water it is a incompressible fluid, the conservation mass equation is reduced to:

$$\nabla \cdot (\vec{V}) = 0$$

The momentum equation is not solvable mathematically yet; this is because it needs to express the stress tensor $\bar{\bar{\sigma}}$ in terms of the primary unknowns, that is, density, pressure and speed.

$$\frac{D(\rho\vec{V})}{Dt} = \nabla \cdot \bar{\bar{\sigma}} + \rho\vec{g}$$

Where

$$\frac{D\vec{V}}{Dt} = \frac{\partial\vec{V}}{\partial t} + u\frac{\partial\vec{V}}{\partial x} + v\frac{\partial\vec{V}}{\partial y} + w\frac{\partial\vec{V}}{\partial z}$$

The constitutive equations allow writing the stress tensor components in terms of the velocity field and pressure field. By introducing these equations can achieve an analytic solution.

Constitutive equation (stress tensor):

$$\sigma_{ij} = -p\delta_{ij} + \tau_{ij}$$

Where,

The tensor of the viscous stresses is:

$$\boldsymbol{\tau_{ij}} = 2\mu D_{ij} - \frac{2}{3}\mu\nabla \cdot (\vec{V})\delta_{ij} = \begin{bmatrix} \boldsymbol{\tau_{xx}} & \boldsymbol{\tau_{yx}} & \boldsymbol{\tau_{zx}} \\ \boldsymbol{\tau_{xy}} & \boldsymbol{\tau_{yy}} & \boldsymbol{\tau_{zy}} \\ \boldsymbol{\tau_{xz}} & \boldsymbol{\tau_{yz}} & \boldsymbol{\tau_{zz}} \end{bmatrix}$$

Strain rate is:

$$D_{ij} = \frac{1}{2}(\frac{\partial v_i}{\partial j} + \frac{\partial v_j}{\partial i})$$

Kronecker delta is:

$$\delta_{ij} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix}$$

Therefore, the constitutive equation for Newtonian fluids is:

$$\sigma_{ij} = -p\delta_{ij} + \mu(\frac{\partial v_i}{\partial j} + \frac{\partial v_j}{\partial i} - \frac{2}{3}\nabla \cdot (\vec{V})\delta_{ij})$$

In this case the fluid is seawater and the Navier-Stokes fluids are incomprehensible

$$\frac{\rho D(u)}{Dt} = -\frac{\partial p}{dx} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right) + \rho g_x$$

$$\frac{\rho D(v)}{Dt} = -\frac{\partial p}{dy} + \mu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2}\right) + \rho g_y$$

$$\frac{\rho D(w)}{Dt} = -\frac{\partial p}{dz} + \mu\left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2}\right) + \rho g_z$$

The Averaged equations of Navier-Strokes method take the variables and divided into two components: the average or middle component, and the fluctuation component:

$$u_i = \bar{u}_i + u'_i$$

Where $\bar{u}\ i$ is the middle component and $u'i$ is the fluctuation component. This model adds to the Navier-Stokes an additional term, the Reynolds tensor, because of this term it is possible to reach a numerical solution by applying a turbulence model.

$$\frac{\rho D(u)}{Dt} = -\frac{\partial p}{dx} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right) + \rho g_x - \rho((\overline{u'u'})_x + (\overline{u'v'})_y + (\overline{u'w'})_z)$$

$$\frac{\rho D(v)}{Dt} = -\frac{\partial p}{dy} + \mu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2}\right) + \rho g_y - \rho((\overline{v'u'})_x + (\overline{v'v'})_y + (\overline{v'w'})_z)$$

$$\frac{\rho D(w)}{Dt} = -\frac{\partial p}{dz} + \mu\left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2}\right) + \rho g_z - \rho((\overline{w'u'})_x + (\overline{w'v'})_y + (\overline{w'w'})_z)$$

3. Computational Fluid Dynamics

Computational fluid dynamics (CFD) is the science that predicts the velocity field of the fluid, heat transfer and mass, chemical reactions and related phenomena through numerical solution of the set of conservation equations. The equations are solved using discretization methods.

3.1 Spatial discretization

In order to calculate the numerical approximation of the convective and diffusive fluxes, it is necessary to apply first a spatial discretization of the domain. Discretization spatial domain can apply the equations that govern the fluid. Since a region of space is discretized creating a computational grid, dividing that way, a region of space in small control volumes. There are three principal methods for the discretization of the problem:

- Finite-differences method discretizes the Navier-Stokes equations in a differential form. It requires a structured mesh of points in which the flow variables are stored.

- Finite-elements method, which for solving the Navier-Stokes equations divides the dominance in 2D triangular elements or 3D tetrahedral elements generating an unstructured mesh.

- Volumes Finite method, where in each generated volume it defines the checkpoints. The control equations are used in the integral way.

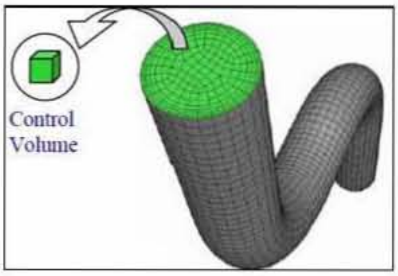


Image 2 - Control Volume

As said before, the computational mesh allows a spatial dominie description in which the numerical simulation will take place.

3.2 Meshing

Meshes can be structured or unstructured. Structured meshes form hexahedral in cells as long as you are working with 3D. And its main advantage lies in the arrangement of the elements, i.e. each grid point is identified by the Cartesian points "i, j, k". But still greater the computational demand than the unstructured meshes demand. Unstructured meshes, both nodes and cells lacking in order, which deprives the identification by their indexes. In this case the cells are tetrahedral.

3.3 Discretization methods

Discretization of the most used equations is through the finite element method (FEM) or finite volume method (FVM).

In the unstructured meshes, both nodes and cells lack of order depriving that way the identification by their indexes. In this case the cells are tetrahedral.

El FVM method creates subdivisions of the domain. Creating thereby a grid composed of a finite number of small control volumes or "cells". The grid defines the limits of the control volumes, while the node is in the centre of the control volume. Once created the control volumes, are resolved in each one of them the discretized conservation equations, so that an algebraic matrix is solved in each cell iteratively until the residual is small enough.

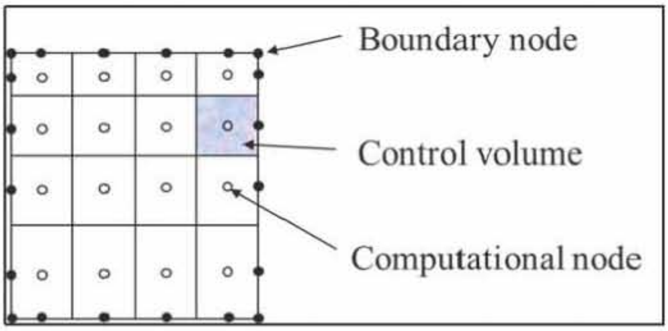


Image 3 - Grid Domain

In the image we can see the grid domain, the control volume in blue and the node inside the grid.

To find the net flow of each control volume a sum of integrals over each of the faces is made. In the case 2D four comprehensive and 3D six comprehensive. The Ansys Fluent program uses the finite volume method.

In our case, for seawater fluid, the Navier-Strokes equations for incompressible fluids are the ones already exposed in chapter 2.7 named "Study case".

For the study of flow generated by the boats and their propellers Navier-Stokes equations are almost impossible to solve. Mathematical models are used to simplify phenomena as viscosity and turbulence. One of the most used models is averaged Navier-Stokes equations (RANS).

3.4 Ansys

Ansys Fluent is the software I am going to use to simulate fluids behaviour in this project. This program uses the *Volumes Finite method* for the spatial discretization.

It contains the broad physical modelling capabilities needed to model flow, turbulence, heat transfer, and reactions for industrial applications. Behind the scenes, fluid dynamics is involved in the design and manufacture of hundreds of products. In any application that involves gas flow, liquid flow or heat transfer, fluid dynamics analysis can help deliver innovation and greater efficiency. In a following chapter I will explain how I used it for my study.

4. Bollard Pull Test

Bollard pull trials are conducted in order to determine the static pull that a tug is able to employ in operating conditions, in order to calculate the tensile force exerted by its thrusts in stationary mode. I.e., it is the thrust developed by the propulsion system of a ship when the speed is zero. To enable fair comparison, the trials are not performed in typical operation conditions (e.g. shallow and confined harbour basin), but in conditions as close to ideal as possible, i.e. in conditions where the environmental and external effects are minimised.
It is a physical test typically performed by fixing a board to a "bollard" on shore with a hawser and measuring the "pull" with a load cell.

Bollard Pull is the thrust developed by the propulsion system of the ship when the speed is zero, because of the connection via cable to a fixed point on earth, which prevents the ship advance.

A measuring device called dynamometer will be installed between the anchor point on earth and the end of the towline. When this is not possible, the dynamometer is placed between the ends of towline and tows the boat. It is the least preferred option of the two and cannot be installed anywhere else.

We distinguish between the following types of Bollard Pull:

Bollard Pull stable or continuous

Known simply as Bollard Pull, is the average of the tension during the test run, usually about 10 minutes. During that time values are taken time intervals, e.g. 30 seconds to the completion of the calculation.

Cash Bollard Pull

Is the shooting that can be achieved in open water. Unable to perform this test are normally considered 78% of Bollard Pull stable.

Maximum static Bollard Pull

The value is the average of the two highest recorded readings recorded during the duration of the test.

Sustained Bollard Pull

Is the average of the tension during the test run, the time between 5 and 10 minutes. During the test measurements at 30-second intervals are performed.

Maximum Static Bollard Pull

It is the highest average intervals of 30 seconds that test.

Maximum Bollard Pull

It is the highest individual value recorded during the test.

To obtain the certificate of "Bollard Pull" they must fulfil minimum standard conditions. Test location, environmental conditions and requirements of the boat. The following guidance notes apply to the bollard pull test of any towing vessel, which the Surveyor is requested to approve or attend.

The safe working load of the test equipment, fittings and any connection points ashore shall be at least 10% in excess of the designed maximum continuous static bollard pull of the vessel.

Bollard pull test certificates issued by Surveyor are acceptable, or by another recognized body provided that acceptable procedures for the tests are produced.

4.1. Location

The water depth at the test location shall be at least 20 meters within a radius of 100 meters of the vessel. If a water depth of 20 meters cannot be obtained at the test location, then a minimum water depth, which is equal to twice the maximum draught of the vessel, may be accepted. The owner of the vessel must be advised that the reduced water depth may adversely affect the test results.

The test location shall be clear of navigational hazards and underwater obstructions within a radius of 300 meters of the vessel.

Whenever possible, the test should be conducted where there is no current. In any case the current should not exceed a 1-knot, i.e. shall be less than 0.5 meters/second from any direction. A current of 1 knot the bow represents a loss of "Bollard Pull" of about 4%. In areas affected by the tide should be given special attention. The tests affected by the tide should start from an hour and a half before high tide, when the current velocity is below 1kN and the water level is slightly below the maximum. They should avoid testing after high tide.

The wind speed shall be less than 5 meters/second from any direction.

The condition of the sea at the test location shall be calm, without swell or waves.

4.2. Vessel

The draught and trim of the vessel shall be as near as possible to the draught and trim under normal operating conditions.

The propellers and fuel used during the tests shall be the same as the propellers and fuel used under normal operating conditions.

All auxiliary equipment such as pumps, generators and other equipment that is driven from the main engine(s) or propeller shaft(s) during normal operation of the vessel shall be connected during the test.

4.3. Test

Avoid, if possible, manoeuvring during testing.

The distance between the stern of the vessel and the shore shall be at least 300 meters. If it is not possible to maintain a distance of 300 meters between the stern of the vessel and the shore, then a minimum distance, which is equal to twice the waterline length of the vessel may be accepted. The owner of the vessel must be advised that the reduced distance between the vessel's stern and the shore may adversely affect the test results.

Adequate communications shall be established between the vessel and instrument recording station.

The Continuous Bollard Pull (CBP) test shall be carried out at the manufacturer's recommended maximum continuous rating of the main engines (100% MCR), for a period of 10 minutes with the vessel on a steady heading.

Whenever possible a maximum (MBP) test shall be carried out at the manufacturer's maximum rating of the main engines (typically 110% MCR), for a period of 5 minutes.

When requested, continuous bollard pull may also be verified at different RPM and/or propeller pitch settings or with fewer propellers or engines in use.

The load cell used for measuring the bollard pull shall have an accuracy of ±2% for the average temperature observed during the test and shall have been calibrated not more than six (6) months prior to the test date. The calibration certificate shall be available.

An autographic recording instrument giving a continuous read-out of the bollard pull shall be connected to the load cell.

If no continuous record of the test is printed, then the bollard pull shall be the mean of consecutive readings recorded at 20-second intervals over the test period.

5. Tugboats

A tugboat (tug) is a boat that manoeuvres vessels by pushing or towing them. Tugs are vessels used for auxiliary use for ships in navigation and manoeuvring or as a tug of floating structures without propulsion. Tugs move vessels that either should not move themselves, such as ships in a crowded harbour or a narrow canal, or those that cannot move by themselves, such as barges, disabled ships, log rafts, or oil platforms. The most common tasks are to assist a ship in the docking and undocking manoeuvres. Help ships keep headed in areas whose nature is difficult to navigate either due to wind, waves, sea traffic, ocean currents or narrow passages such as channels.

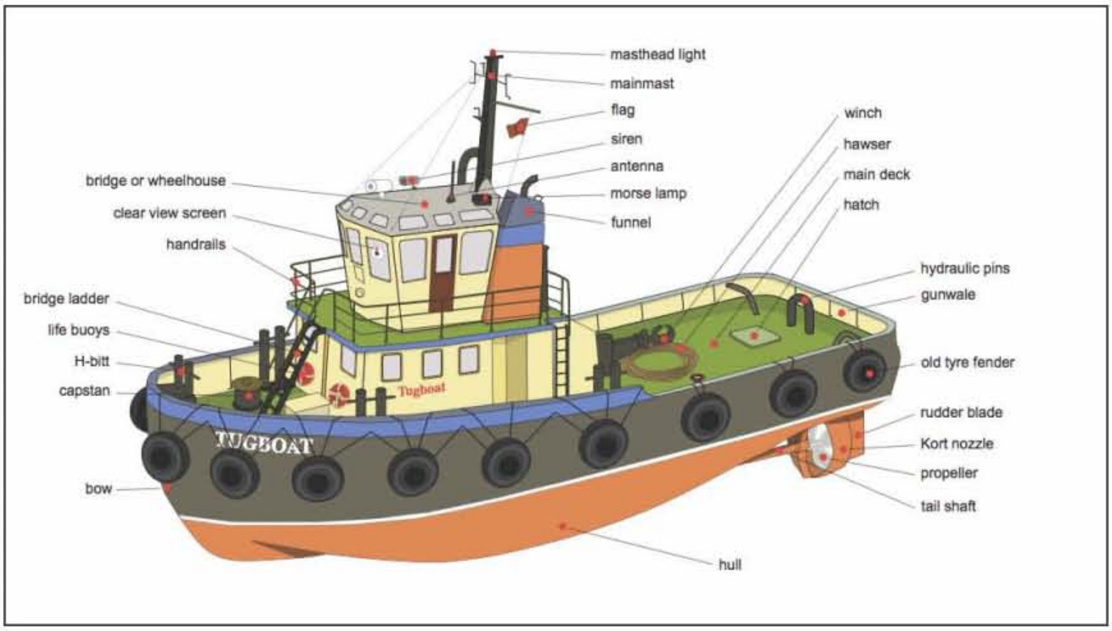


Image 4 - Tugboat parts (wikimedia)

Tugboats are powerful for their size and strongly built, some tugboats serve as icebreakers or salvage boats. Early tugboats had steam engines, but today most have diesel engines. Many tugboats have fire-fighting monitors, allowing them to assist in fire fighting, especially in harbours.

Tugboats are highly stressed given because e of its role and need a quick answer to solve any eventuality. Therefore they must have a favourable stability under all operating conditions, as when the tug pulls produces a torque and power must bear notwithstanding affect stability. For these reason it is really important, that the propulsion equipment must provide an adequate response in the shortest possible time. It is therefore vital to choose a suitable propellant according to their operation. The towing situation creates load all efforts to the hull to be of resist; hence the town is a robust and solid design. Another very important factor is visibility that is essential to give a good response from the wheelhouse; it must be complete fore and aft.

Tug propulsion can be located both fore and aft. It depends on the propulsive system chosen. If it is in the bow you can install an azimuth or Voith Schneider system, while if it is in the stern azimuthal propulsion propeller fixed or variable pitch and usually with nozzle may be installed, as it increases the pressure of the trailer and in some cases adds them to post an azimuthal propeller to increase manoeuvrability.

There are different kinds of Tugboats:

5.1. Seagoing tugboats

Seagoing tugboats (or ocean tugboats) fall into four basic categories:

1) The standard seagoing tugboat with model bow that tows its "payload" on a hawser.

2) The "Notch Tug" which can be secured in a notch at the stern of a specially designed barge, effectively makes the combination of a ship. This configuration is dangerous to use with a barge, which is "in ballast" (no cargo) or in a head or following sea. Therefore, the "notch tugs" are usually built with a towing winch. With this configuration, the barge being pushed might approach the size of a small ship, with interaction of the water flow allowing a higher speed with a minimal increase in power required or fuel consumption.

3) The "integral unit," or "Integrated Tug and Barge" (ITB), comprises specially designed vessels that lock together in such a rigid and strong method as to be certified as such by authorities (classification societies) such as the American Bureau of Shipping, Lloyd's Register of Shipping, Indian Register of Shipping, Det Norske Veritas or several others. These units stay combined under virtually any sea conditions and the "tugs" usually have poor sea-keeping designs for navigation without their "barges" attached.

4) "Articulated Tug and Barge" (ATB) units also utilize mechanical means to connect to their barges. ATBs generally utilize Intercon and Blud worth connecting systems. ATBs are generally staffed as a large tugboat, with between seven to nine crewmembers.

Image 5 - Seagoing Tugboat (www.pinterest.com)

5.2 Harbour tugboats

Image 6 - San Francisco harbour based tractor tug "Delta Deanna" (wikipedia)

Compared to seagoing tugboats, harbour tugboats are generally smaller and the width-to-length ratio is often higher, due to the need for a lower draught. In smaller harbours these are often also termed *lunch bucket boats*, because they are only manned when needed and only at a minimum (captain and deckhand), thus the crew will bring their own lunch with them.

5.3. River tugboats

River tugs are also referred to as towboats or push boats. Their hull designs would make open ocean operation dangerous. River tugs usually do not have any significant hawser or winch. Their hulls feature a flat front or bow to line up with the rectangular stern of the barge, often with large pushing knees.

5.4 Tugboats propulsion

Tugboat engines typically produce 500 to 2,500 kW (~ 680 to 3,400 Hp), but larger boats (used in deep waters) can have power ratings up to 20,000 kW (~ 27,200 Hp) and usually have an extreme power: tonnage-ratio. The engines are often the same as those used in railroad locomotives, but typically drive the propeller mechanically instead of converting the engine output to power electric motors, as is common for diesel-electric locomotives. For safety, tugboat's engines often feature two of each critical part for redundancy.

The Tugboats power is typically stated by its engine's horsepower and its overall bollard pull. The largest commercial harbour tugboats in the 2000s-2010s, used for towing container ships or similar, had around 60-65 tons of bollard pull, which is described as 15 tons above "normal" tugboats.

Tugboats are highly manoeuvrable, and various propulsion systems have been developed to increase manoeuvrability and increase safety. The earliest tugs were fitted with paddle wheels, but these were soon replaced by propeller-driven tugs. Kort nozzles have been added to increase thrust per kW/hp. This was followed by the nozzle-rudder, which omitted the need for a conventional rudder. The cycloid propeller was developed prior to World War II and was occasionally used in tugs because of its manoeuvrability. After World War II it was also linked to safety due to the development of the Voith Water Tractor, a tugboat configuration that could not be pulled over by its tow.

In the late 1950s, the Z-drive or (azimuth thruster) was developed. Although sometimes referred to as the Schottel system, many brands exist: Ulstein, Wärtsilä, Berg Propulsion, etc. The azimuthal propulsion consists of a propeller that can rotate 360 ° thanks to its vertical axis and a nozzle to add more thrust and protects the propeller. This system allows great manoeuvrability and suppression helm as the propeller rotates itself acts helm. These propulsion systems are used on tugboats designed for tasks such as ship docking and marine construction. Conventional propeller/rudder configurations are more efficient for port-to-port towing.

The Kort nozzle is a sturdy cylindrical structure around a special propeller having minimum clearance between the propeller blades and the inner wall of the Kort nozzle. The thrust-to-power ratio is enhanced because the water approaches the propeller in a linear configuration and exits the nozzle the same way. The Kort nozzle is named after its inventor, but many brands exist.

A recent Dutch innovation is the Carousel Tug. The Carousel Tug adds a pair of interlocking rings to the body of the tug, the inner ring attached to the boat, with the outer ring attached to the towed ship by winch or towing hook. Since the towing point rotates freely, the tug is very difficult to capsize. The cycloid propulsion or Voith-Schneider will be explained in more detail in the next chapter. It is a propulsion system consisting of a vertical blades whose pitch and thrust is regulated in the 360th allowing an immediate response. These tugs usually have symmetrical shapes about the amidships section in order to be able to operate both ahead and astern effectively. The Voith Water Tractor, for example, is ideally suited for a wide variety of tasks in everyday port operation. Engineered to perform like a worker bee on water, the complete vessel is developed, designed and built by Voith.

5.5. Project used tugboat design

In this project the design of an already made tug obtained from the final degree work of David Lopez Martinez “Recreation of the bollard pull test of a tug with double propeller using computational fluid dynamics. June 2014" is used. The design of the tug was made by two CAD programs the "Rhinoceros" and "Inventor 3D CAD." The Rhinoceros software was used to import the lines plan of the tugboat in "raster" or "bitmap". "3D CAD Inventor" was used for modelling the hull and appendages. It was conducted in three parts: the hull with the keel, skate forward and the propeller.

In the following image we have the hull of this final work project:

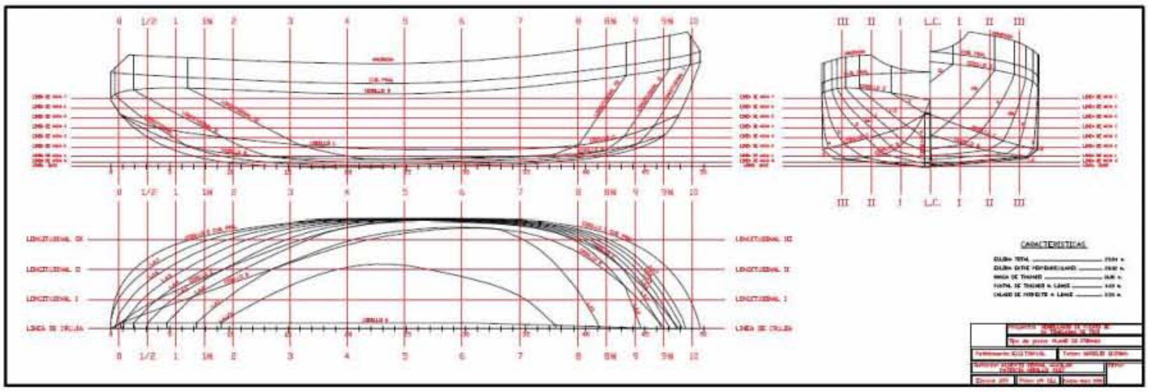


Image 7 - Plane Hull Shapes from the "Bollard pull test recreation of a double propeller tug by CFD" project of David López Martínez

And in the next image, we can see the form plane with the keel and the skate. This skate is not the same one as I am going to use, because it cannot protect the Voith Schneider Propeller toe to its size. For that reason, in the project before that one, another skate was designed to house the VSP.

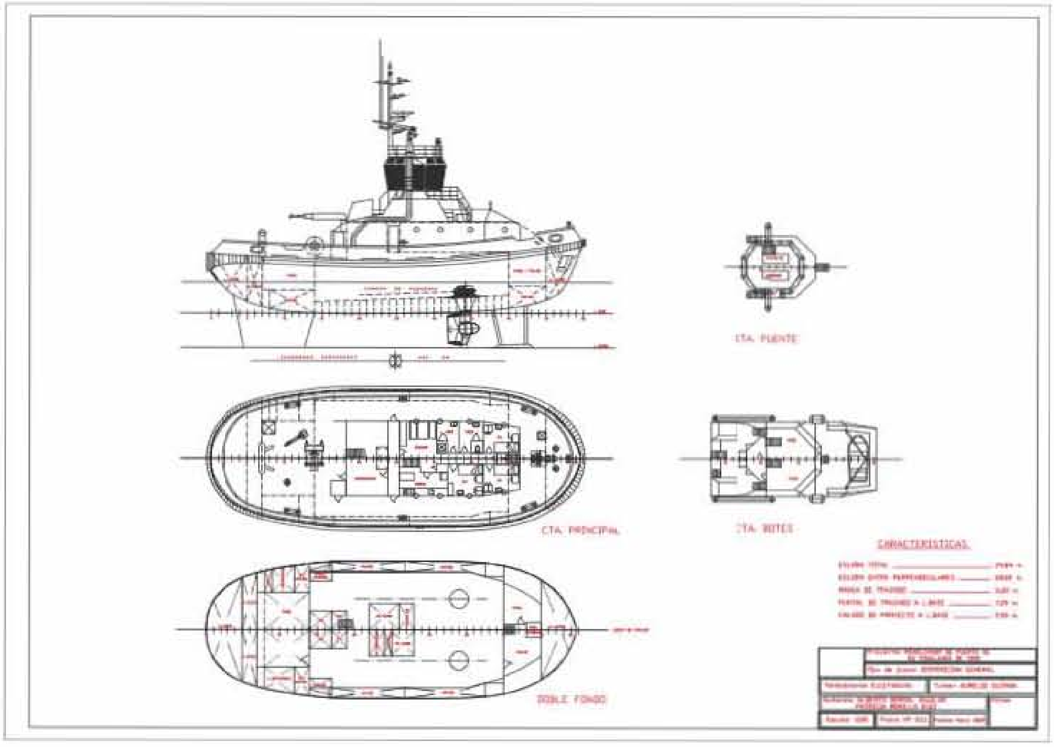


Image 8 - General Arrangement Drawing from the "Bollard pull test recreation of a double propeller tug by CFD" project of David López Martínez

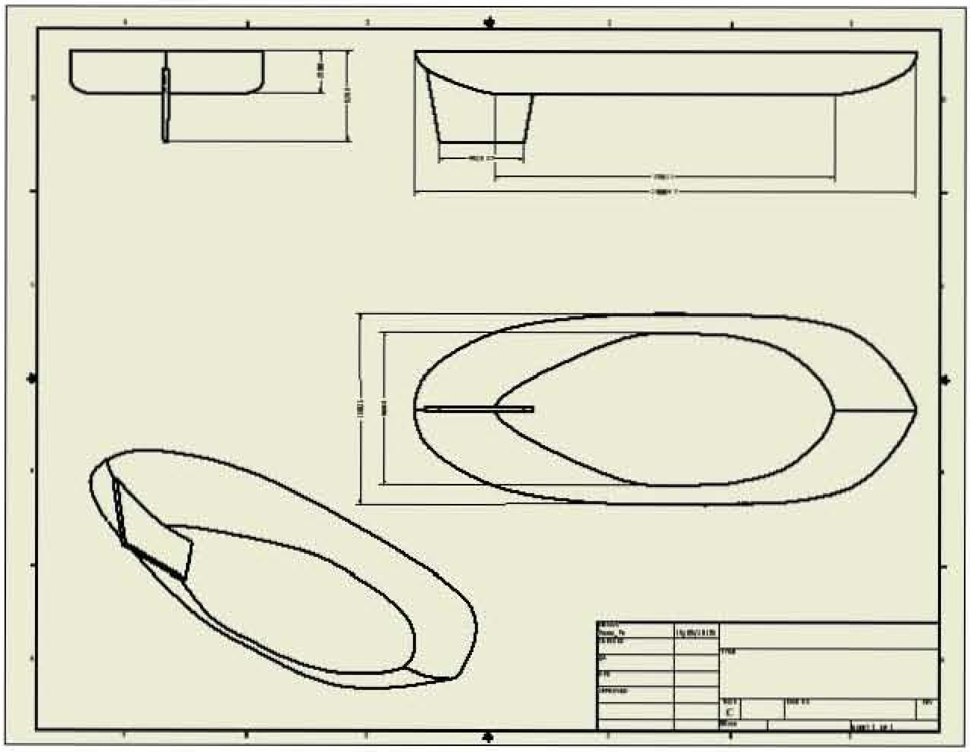


Image 9 - Dimensions of the hull with the new keel from "Recreation of the bollard pull test of a tug with double propeller using computational fluid dynamics" project of Joan Pau Anguera with Autodesk Inventor 2015.

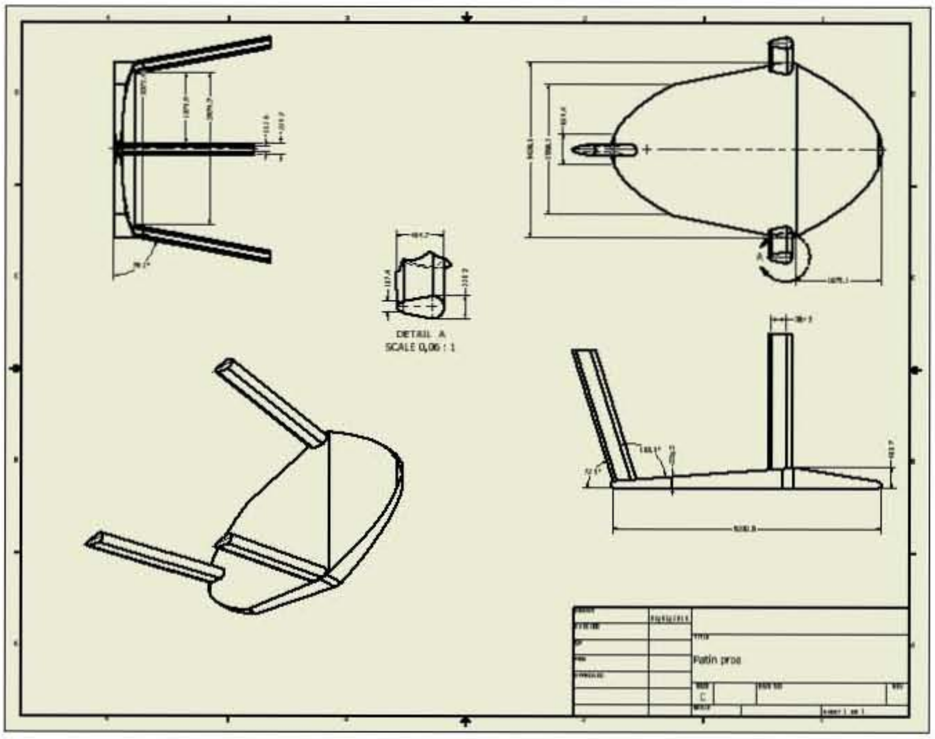


Image 10 - Dimensions of the aft skate with Autodesk Inventor 2015 "Recreation of the bollard pull test of a tug with double propeller using computational fluid dynamics" project of Joan Pau Anguera.

Propulsion

This tugboat have a WÄRTSILÄ 9L20 engine since the power delivered to the brake fits the required one by the tug to be able to provide the project specification power, besides being a lightweight candidate.

It has two engines of 2200 BHP, those proportionate 52 Tones, which is an appropriate value for berthing and towing operations of large container ships and freighters.

6. Voith Schneider Propeller

The Voith Schneider propeller (VSP), also known as a cycloid drive, is a specialized marine propulsion system (MPS). The VSP is a unique propulsion system, offering precise manoeuvrability, because it is a controllable pitch propeller with no preferred pitch direction, being able to change the direction of its thrust almost instantaneously. It combines propulsion and steering in a single unit. It is widely used on tugs and ferries, where extreme manoeuvrability and those improvements of VSP are needed; i.e. precise manoeuvring, high escorting forces, good sea keeping behaviour, no thrust losses in waves, low roll and pitch motion and flexible force generation in all directions. Furthermore from the point of view of maintenance has a low cost and labour because it operates with a low speed regime. Besides providing long lasting life.

Image 11 - VSP (www.ship-technology.com)

The Voith Schneider propeller was originally a design for a hydro-electric turbine. Its Austrian inventor, Ernst Schneider, had a chance meeting with an employee of Voith's. This led to the turbine being investigated by Voith's engineers, who discovered that although it was no more efficient than other water turbines, Schneider's design worked well as a pump by reversing the flow through the device. By changing the orientation of the vertical blades, it could be made to function as a combined propeller and rudder, which provides a drive that can be directed in any direction putting away the need for a rudder. The magnitude of thrust is determined by the rotational speed of the disk and the blade angle determines the direction of thrust.

Its maintenance is minimal, since being variable pitch allowing the main engine to keep the revolutions regime, done to decrease the risk of mechanical failure. Besides the propeller rotates at a low rpm, fact that brings to few losses and prevents cavitation problems.

Regarding the structural point of view the Voith Schneider is more sensitive than the azimuth system but to protect propeller blades and in, itself has a robust defence structure in the bow, which also takes effect nozzle and one keel in case a beached tugboat it is produced. Adding this heavy propeller with protection and keel increases the displacement, but thanks to this draft it gets great stability, because it decrease the centre of gravity. Stability is also improved due to propeller blades requiring a high orbit.

Another important advantage is the hydraulic coupling. Normally one "DTL turbo coupling", wearing a protection system in case the propeller crashes due to a network or any other object. The protection system is to decouple the motor of the axes line emptying the hydraulic oil.

Lastly is a reliable, since the control of the thrusts is mechanical. This, in case a drop of the power plant, allows the tug can to continue operating.

By contrast the main problems of this system compared to a conventional propeller or nozzle are lower efficiency and greater depth.

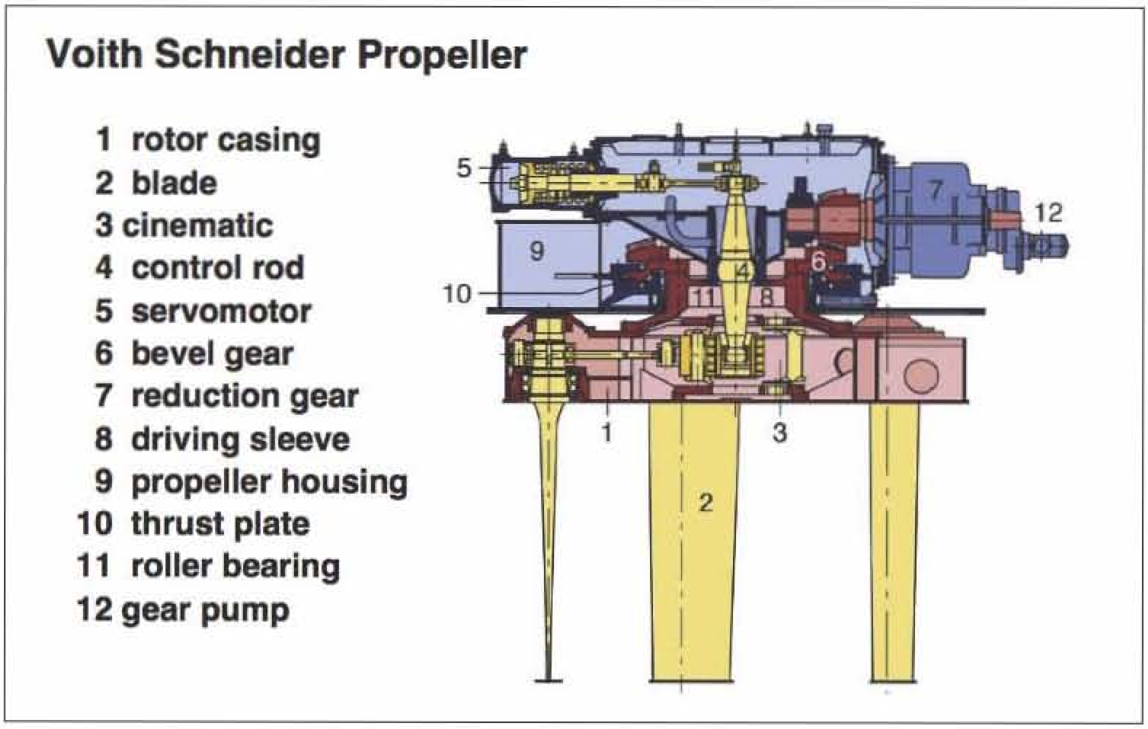


Image 12 - VSP parts (www.fireengines.net)

In the previous sketch we can see the parts of the propeller.

The thrust is created by vertically mounted blades (2)(in the shape of hydrofoils) in a rotor casing (1). While the rotor casing is rotating rotates about a vertical axis, the blades are oscillating, because also each blade can rotate itself around a vertical axis, performing an oscillating motion about its own axis (similar to the motion of the tail of a fish). It is clear to realize that there are two centres of rotation, the centre of the radial bearings and the rotor centre. The blades oscillation is steered by the law of intersecting normals. The internal gear changes the angle of attack of the blades in sync with the rotation of the plate, so that each blade can provide thrust in any direction, very similar to the collective and cyclic of helicopter flight controls. Blade excursion determines the amount of thrust, while the phase angle of between 0 and 360 degrees determines its direction. Unlike the azimuth thruster where a conventional propeller is rotated about the vertical axis to direct its thrust, allowing a vessel to steer without the use of a rudder), the Voith-Schneider drive merely requires changing the pattern of orientation of the vertical blades. Both variables the magnitude and the direction of thrust are controlled by a control rod (4), which is pulled by two servomotors (5) and a mechanical kinematic transmission (3). Servomotors function is to modify the angle of attack by varying eccentricity, i.e., changing the distance between the geometric centre of the propeller and its rotational centre. If the servomotors are not exerting force on the control rod, the blades rotate concentric without a tangent angle relative to the circle that the profile describes. On the other hand, if the servomotors exert force on the radial bearings, it produces a variation of the centre, which differs from the centre of the propeller. Ultimately the maximum angle of attack increases with eccentricity.

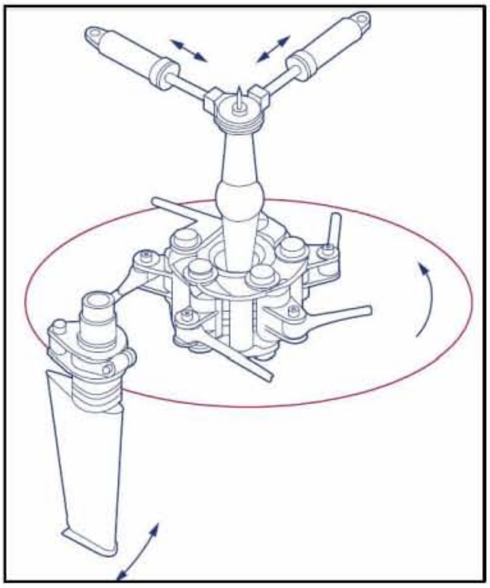


Image 13 - Cinematic principle Voith Schneider Propeller (laoet-63.blogspot.com.es)

The figure on the left shows the movement of the propeller with just a single blade. The red circle is the path describing the blade when the rotor rotates; in this case, the rotation direction is counter clockwise. The arrow front shovel indicates the attacking angle variation on its own axis. At the top end you can see the two thrust servomotors, whose function is to modify the angle of attack by varying eccentricity, i.e., changing the distance between the geometric centres of the propeller and turning centre.

In the figure below on the left it is shown how the direction centre and propeller centre are aligned, situation of zero push. In the circle of the right, the direction centre has changed due to the servomotors action, producing with that a variation between both angles.

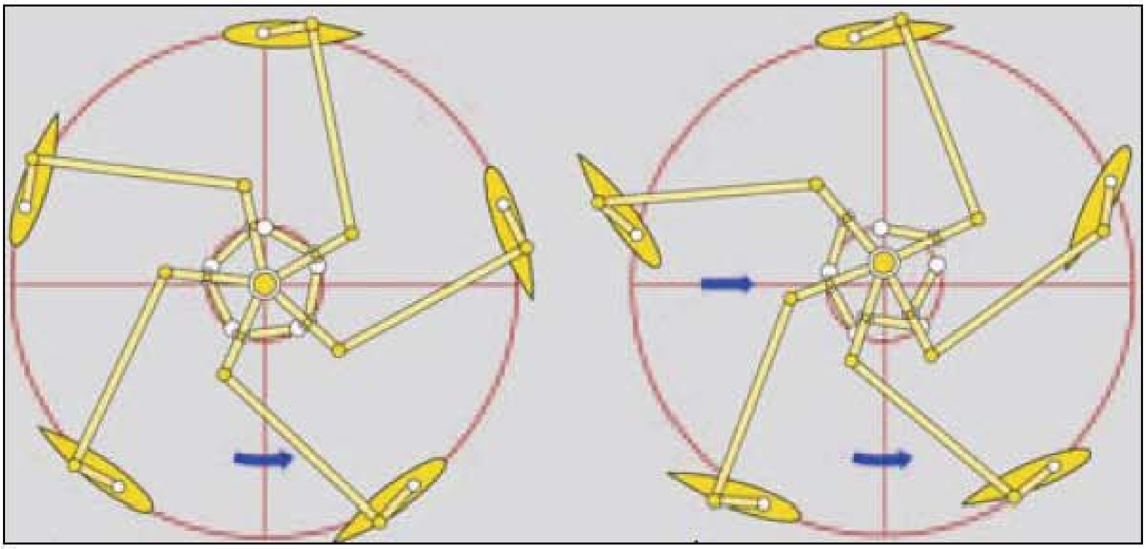


Image 14 - Kinematic transmission system (users.skynet.be)

And in the following figure we can observe the speeds triangle of each profile. The position, where its thrust is zero.

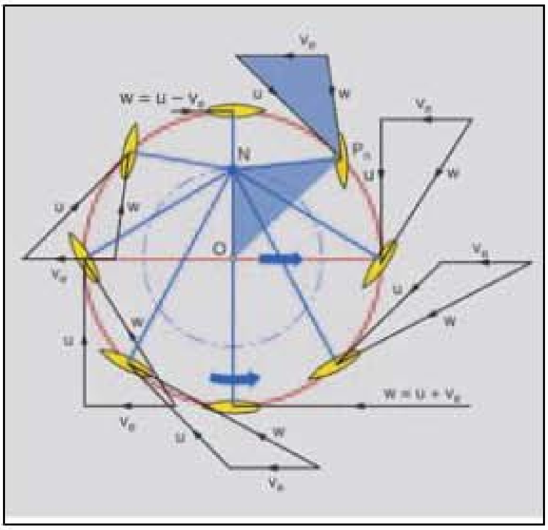


Image 15 - Speed of the blade in no thrust condition

Other important considerations when studying the VSP factor is the hydrodynamic properties. The main parameters influencing its hydrodynamic properties are:

- The angle of attack of the blade during the revolution.

- The blade profile.

- The thickness of the blade.

- The position of the shaft relative to the blade

- The output design of the blade edge.

- The relationship between the length of the blade (L) and the diameter of the orbit (D); (L / D).

- The ratio of chord length (c) and orbit diameter (D); (c / D).

The following figure describes one of the parameters that determinate the behaviour of the engine. Specifically, the curved blade trace direction during the rotational movement or "Blade Steering Curve" (BSC). This figure describes the angle of the propeller blade at every moment. The direction of the blade describes an oscillating motion that can be described as the steering angle α of the blade relative to the rotor position φ. Therefore, this curve defines not only the magnitude and direction of thrust but also has a great influence on the flow around the propeller, or put another way, in their efficiency.

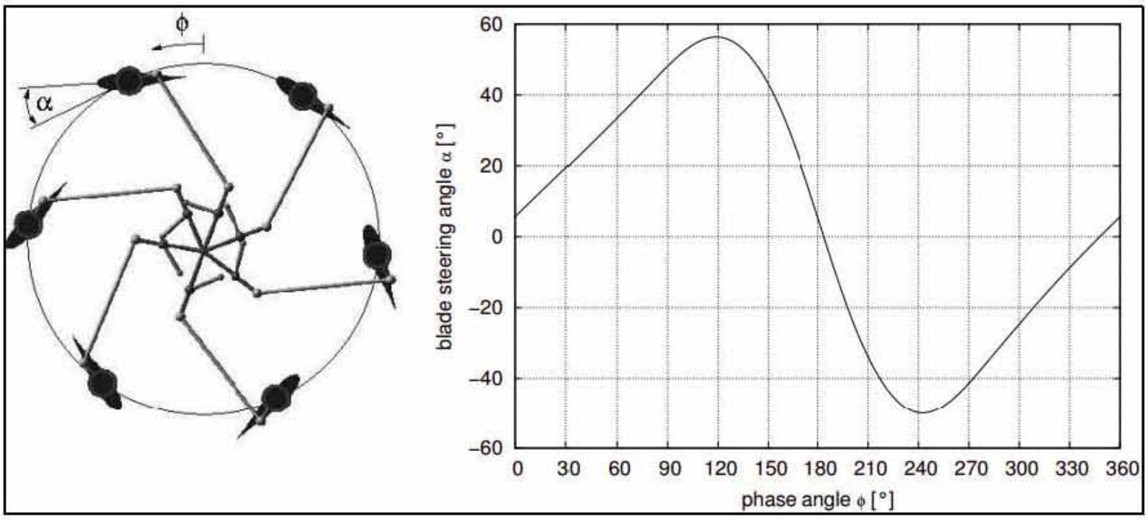


Image 16 - Curvilinear direction of the blade

The towing equipment and the propellers are arranged in a way such that a stable equilibrium is guaranteed, the attack point of the towline force being located directly above the fin, just above deck height. It has good sea keeping behaviour with low roll and pitch motion. The reason for this is the damping effect created by the following:

1) The VSP itself with its vertical axis has a strong stabilising effect. VSP blades have a high lever relative to the rolling centre of the tug and therefore they are efficient even without active anti-roll steering.

2) The large fin and the large guard plate with the struts acts like stabilising fins. The forces that they create are lift forces and forces due to the added mass in water, which give an important contribution; 3 Bilge keels (if installed).

Because of the high lever of the VSP relative to the roll centre of the tug, the VSP can be used efficiently as an active roll stabilising device, as showed in the following image. The roll centre is normally slightly beneath the centre of gravity.

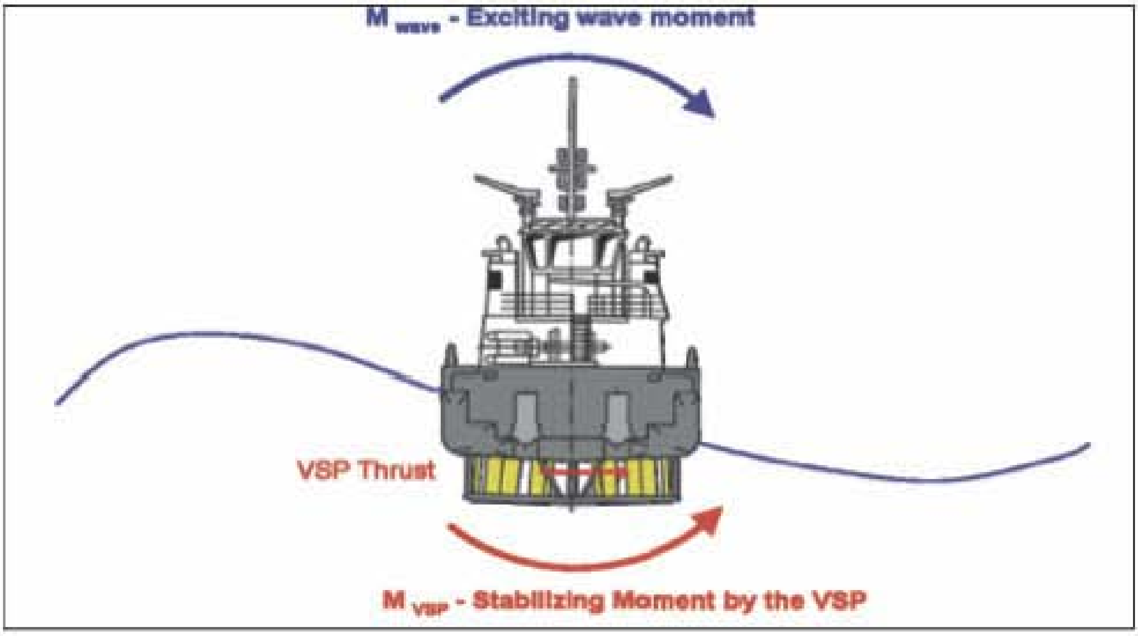


Image 17 - Mechanism of Roll stabilization for the VWT.

Moreover, the low acoustic favours the device's use in minesweepers by minimising cavitation (usually produced at the tips of axial propellers) as the rotor does not need to rotate as fast for a given thrust.

VSPs are offered with an input power range of 160 kW to 3900 kW

There are different propeller types that differ in size, number of blades, shape of blades, blade steering and many other construction details for different boat purposes. Choosing the correct Voith Schneider Propeller is not only a question of hydrodynamic criteria. In each individual case, the mechanical loads on major systems components, such as blades and gear units, as well as kinematics due to the required propeller thrust and the torque to be transmitted, have to be determined and checked.

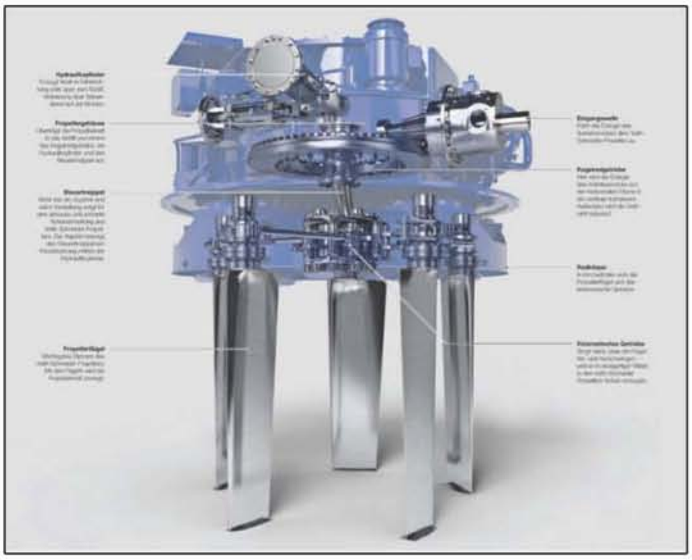


Image 18 - VSP (www.voith.com)

Lift forces imparted to the VSP from the water body:

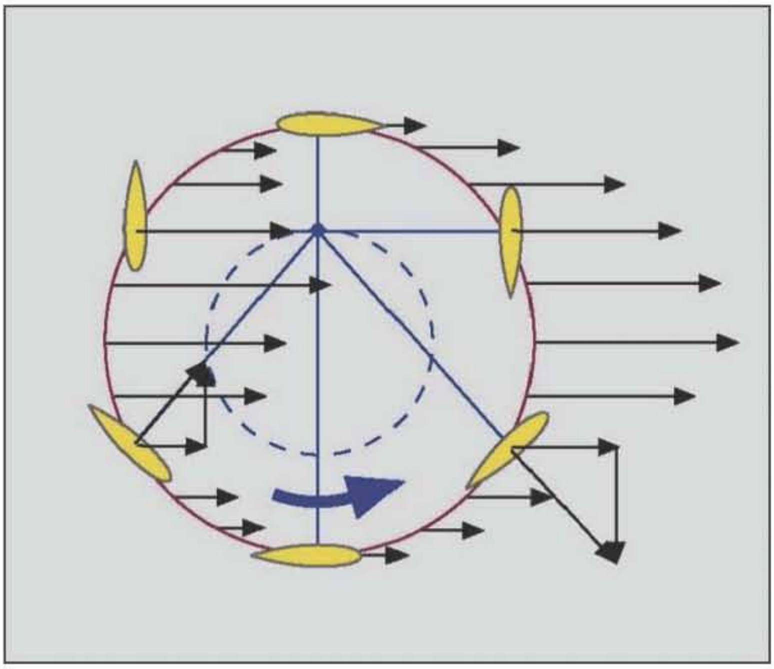


Image 19 - VSP Forces (www.quora.com)

Path of a blade in the water:

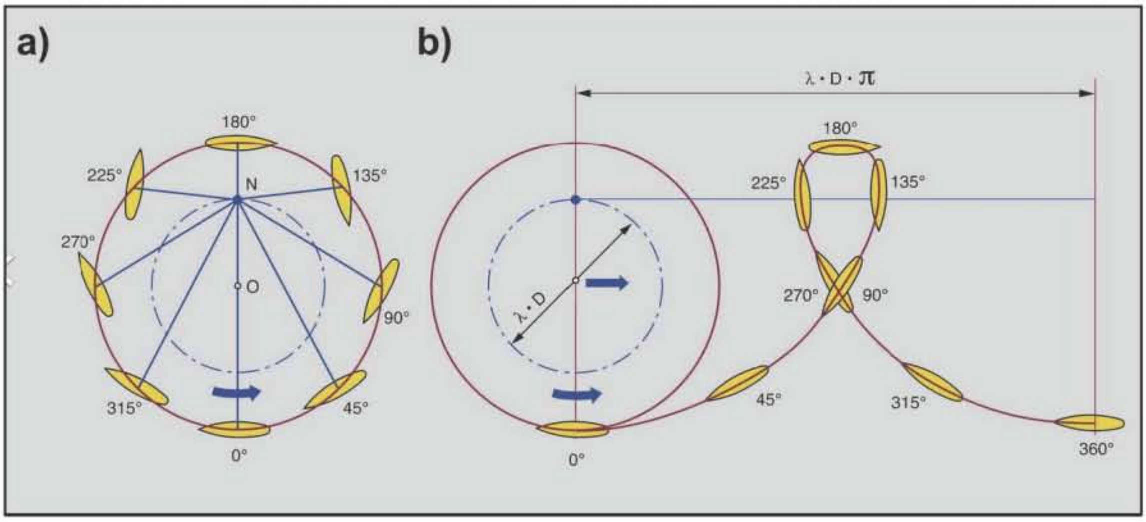


Image 20 - Path of a blade VSP (users.skynet.be)

The red circle is the path described by the blade. In this case, the rotation is counter clockwise. Likewise, you may also notice the angle variation profile attack on its axis indicated by the arrow.

6.1 Generated thrust

For the thrust generated by the propeller the curves of characteristic coefficients are used. The characteristic coefficients relate propellants attack angle, torque, thrust and efficiency. For voiht Schneider propeller characteristic equations coefficients are slightly different from regular propellers, due to the drive movement. However there is a conversion formula for the equivalent in either regular helix or VSP. The main advantage is that they are dimensionless coefficients and aims to estimate the propellant performance:

Advance coefficient (λ)

The advance coefficient relates the angle of attack of the propeller. The equation is defined by the forward speed or wake speed (which is the forward speed of the propeller). In the denominator variables are the diameter (D) and the number of revolutions (N).

$$\lambda = \frac{V_A}{\pi N D}$$

If it is wanted to get the equivalent of a regular propeller the following conversion should be applied.

$$J = \pi\lambda$$

For bollard pull test case, the tug is pulling with fixed shot, i.e. without offset accordingly forward speed (it is zero, and therefore also J).

Thrust coefficient K_S

The thrust coefficient shows the thrust (T) generated by the propellant being the maximum thrust when λ = 0 and no thrust when the helix efficiency is zero. In this paper, the object of study is the value of the thrust generated by the Voiht Schneider. This value is in the graph vertically intersecting with the line = 0. His equation is given by the following formula:

$$K_S = \frac{T}{0{,}5\rho D L u^2}$$

As can be observed K_S is directly proportional to the thrust and inversely proportional to the product by the density ρ, blade length L, diameter of the orbit of the blade D and the circumferential velocity of the blade u.
The circumferential or angular velocity of the blade is described as $u = \pi N D$.
Also the equivalent homologue of a helix is obtained from the following formula:

$$K_T = 0{,}5\pi^2 K_S$$

Torque Coefficient K_D

The torque coefficient defines the pair (Q) delivered by the engine. It can be seen as the torque M will be increased to the fourth power depending on the diameter orbiting the blade and secondly the revolutions number N, which is raised to the second power.

To find their counterpart in the helix the following formula is used:

$$K_D = \frac{M}{\rho N^2 D^4 L}$$

To find its homolog for helix there is the next equation:

$$K_Q = \frac{\pi^2}{4} K_D$$

It is been tried to contact with the Voith Company to obtain the thrust and torque coefficient graphics to be able to also have theoretical calculated thrust and torque results. That way, it would also be possible to validate the model with those results too in chapter 8. Unfortunately it seems that this graphics are confidential data from the company, so it will be no possible to validate the model that way.

6.2 Blades Profile

The Voith Schneider propeller blades are based on the series of profiles of the German companies HSVA and DVL. The most important parameters in the profile are:

- Longitude profile or "Profile length"
- Longitude rope or "Chord length"
- Thickness relative or "Relative thickness"

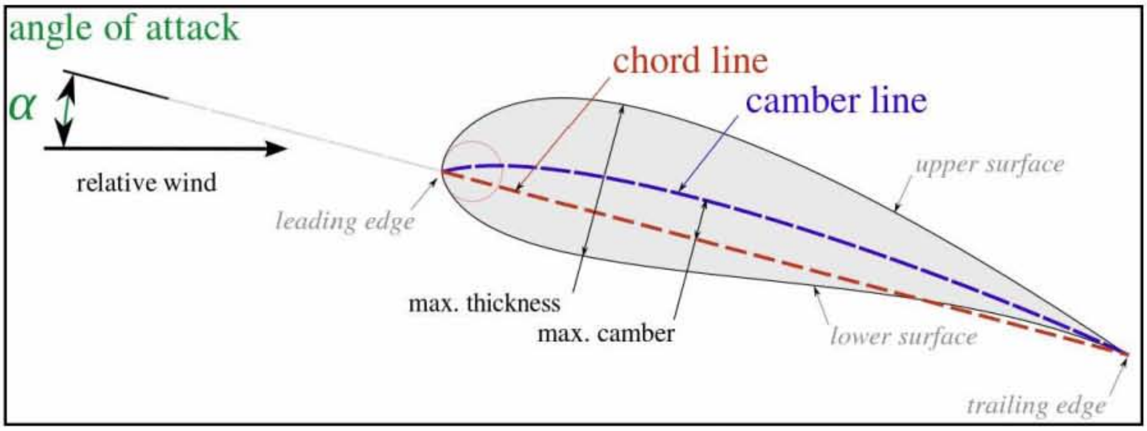


Image 21 - Parts of a profile (www.esru.strath.ac.uk)

This picture shows the main features of a profile. The angle of attack is that one that forms the rope line or "chord line" profile with the direction of incident flow. Its significance lies in that it is a parameter influencing the propeller thrust generation. That is, by increasing the angle of attack increases the pressure till some extent in which it decreases sharply.

The profiles are divided according to their characteristics:

- Four-digit series
- Five-digit series
- Modifications od the series “Five-digit series” y ”Four-digit series “
- 1-series or 16 series
- 6-series
- 7-series
- 8-series

6.3 Project used Voith Schneider Propeller

The propellant chosen is a model created by the Voith Schneider, this is model 28GII / 180. The blades of this propeller have a NACA 0016 shape created by the National Advisory Committee for Aeronautics (NACA).

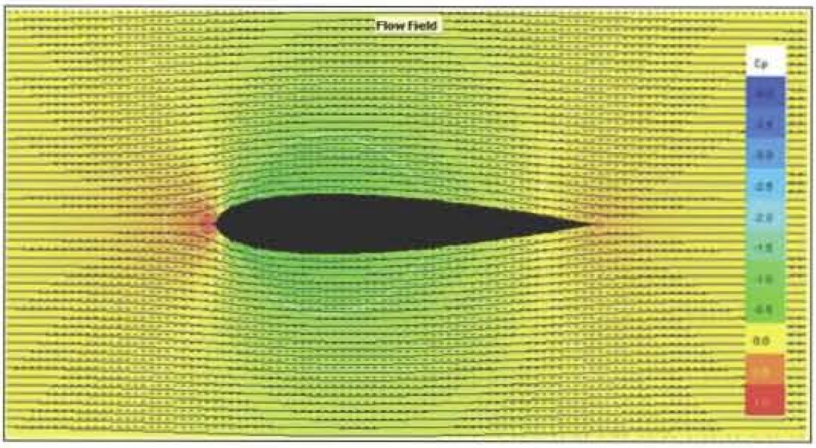


Image 22 - Profile NACA 0016 (www.hantoninnovation.com)

The shape of the NACA airfoils is described using a series of digits following the word "NACA". The parameters in the numerical code can be entered into equations to precisely generate the cross-section of the airfoil and calculate its properties. The NACA 0016 is a symmetrical profile "Four-digit series". To design the profile through inventor can be done in different ways. One is matter the points in an Excel spread sheet, another way is through filed a "bitmap" and another way is creating the equation that defines the profile. In this case the profile equation have been chosen.

After entering the variables and set their value by "sketch" is created and the "Equation curve" tool in which a window where the equation x (t) is written, and (t) will open be used and values t_{min} and t_{max}.

Where,

x(t) = t^2

y(t)=thick*chord*t*5*(0,2969*sqrt(25,40mm/chord)-(0,1260*t*(25,40mm))/chord-(0,3516*t^3*(25,40mm)^2)/(chord)^2+(0,2843*t^5*(25,40mm)^3)/(chord)^3-(0,1015*t^7*(25,40mm)^4)/(chord)^4)

t_{min} = 0 ul

t_{max}. = sqrt(chord / 1 in)

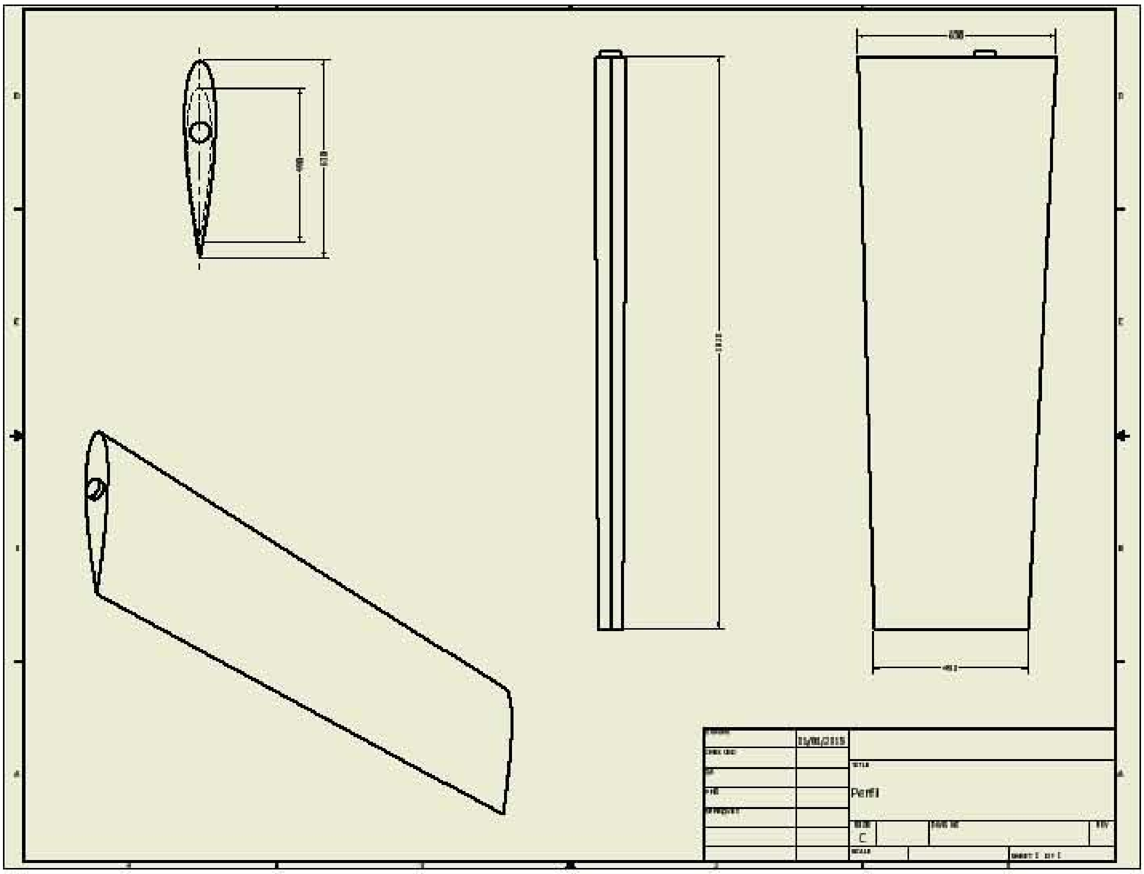


Image 23 - Dimensions of the profile NACA 0016 "Recreation of the bollard pull test of a tug with double propeller using computational fluid dynamics" project of Joan Pau Anguera with Autodesk Inventor 2015.

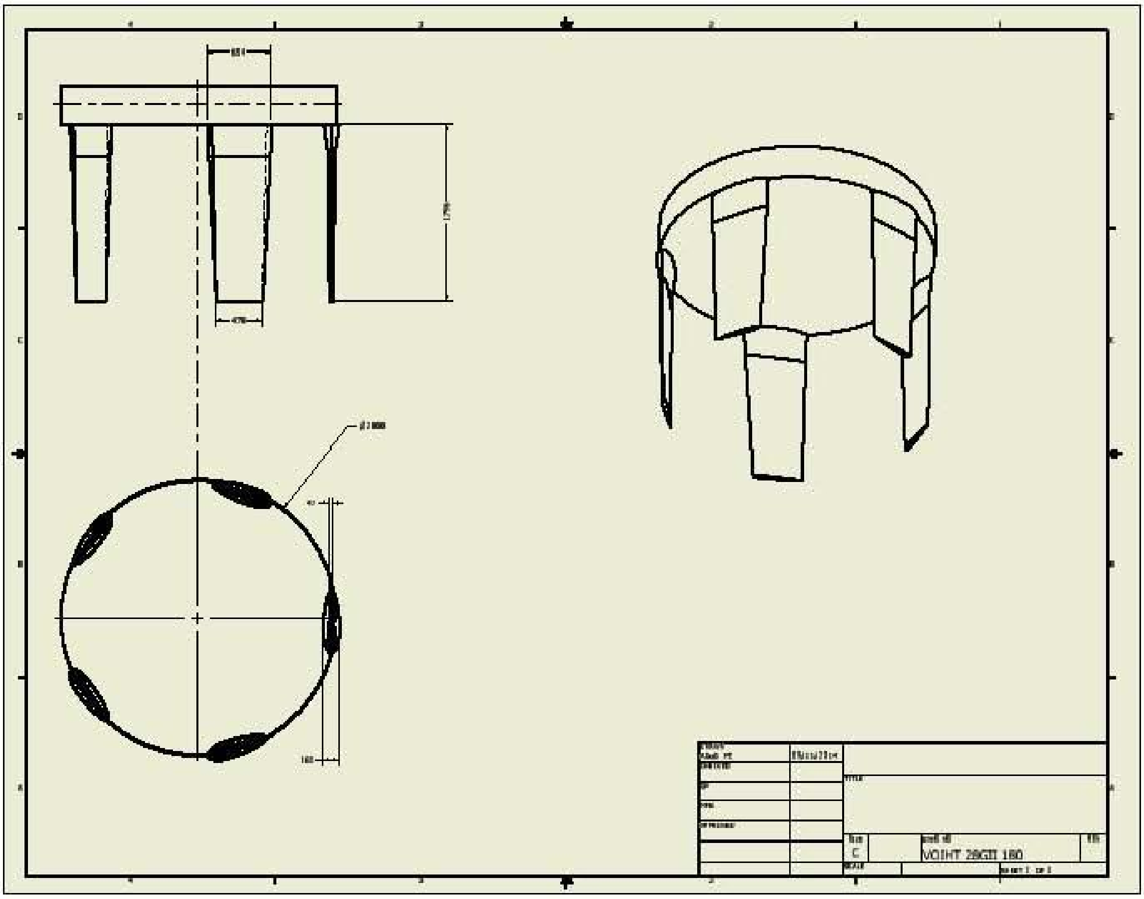


Image 24 - VSP 28GII 180 from the "Recreation of the bollard pull test of a tug with double propeller using computational fluid dynamics" project of Joan Pau Anguera with Autodesk Inventor 2015.

7. Ansys Fluent Program

In this project, for the simulation of the bollard pull test, is going to be used the Ansys Fluent CFD software, as already said. This program consists of a series of modules with a specific function and must be run in order, as showed in the image on the left. Upon completion of the operations in each module, if correct, it will pass to the next module till get the results.

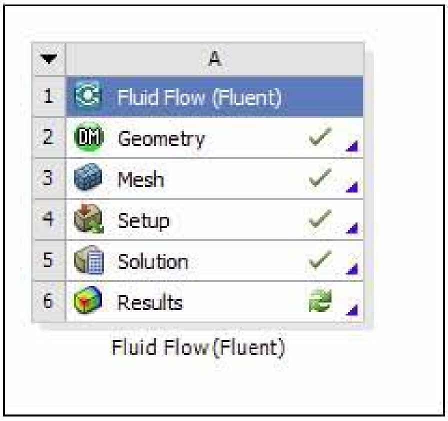


Image 25 - Principal Menu of Ansys Fluent

Comming up next a brief description of the modules and how I developed the study throw them:

7.1 Geometry

Run the program Ansys Design Modeler and allows to import the CAD files which have been created in Inventor and to perform the operations necessary for the study. First of all, all the figures obtained in CAD Inventor were imported to Ansys Design modeller to define the static and rotational domains. The static domain comprises the tug representing the sea area where he conducted the test and which subtracts the areas occupied by the hull, the keel, skate forward, the anchoring elements of the propellers and nozzles.

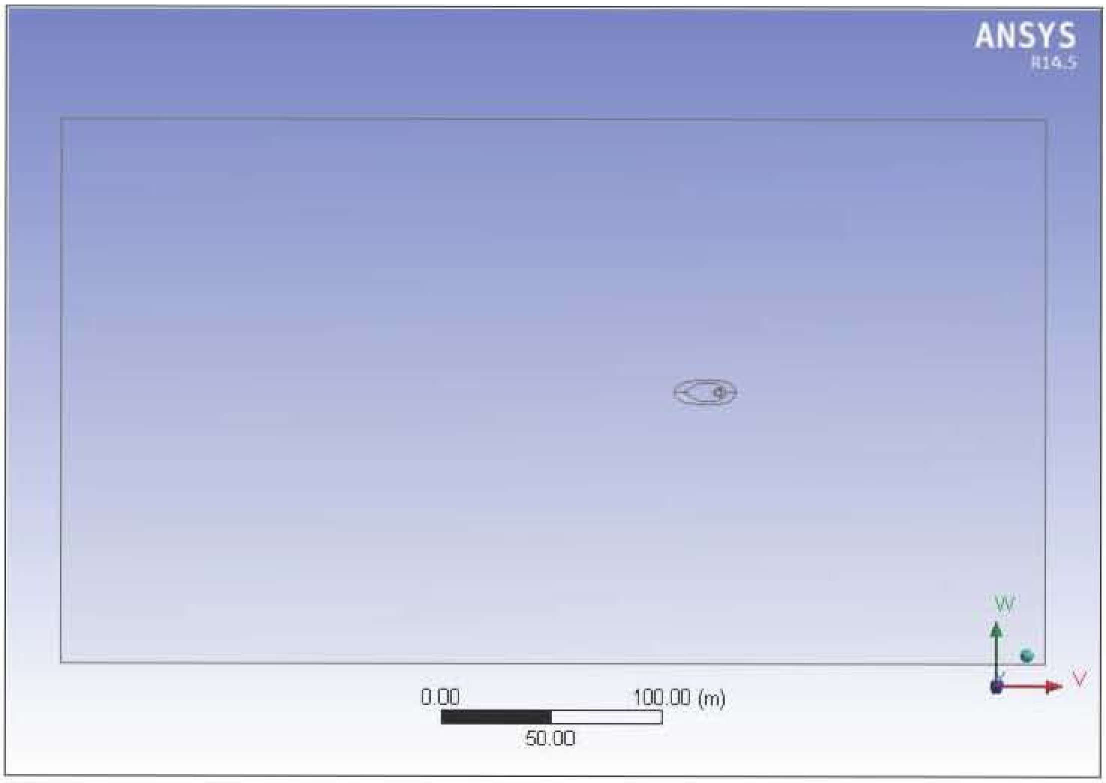


Image 26 - model of the boat and "sea" in the x-y plane from Geometry in Ansys Fluent

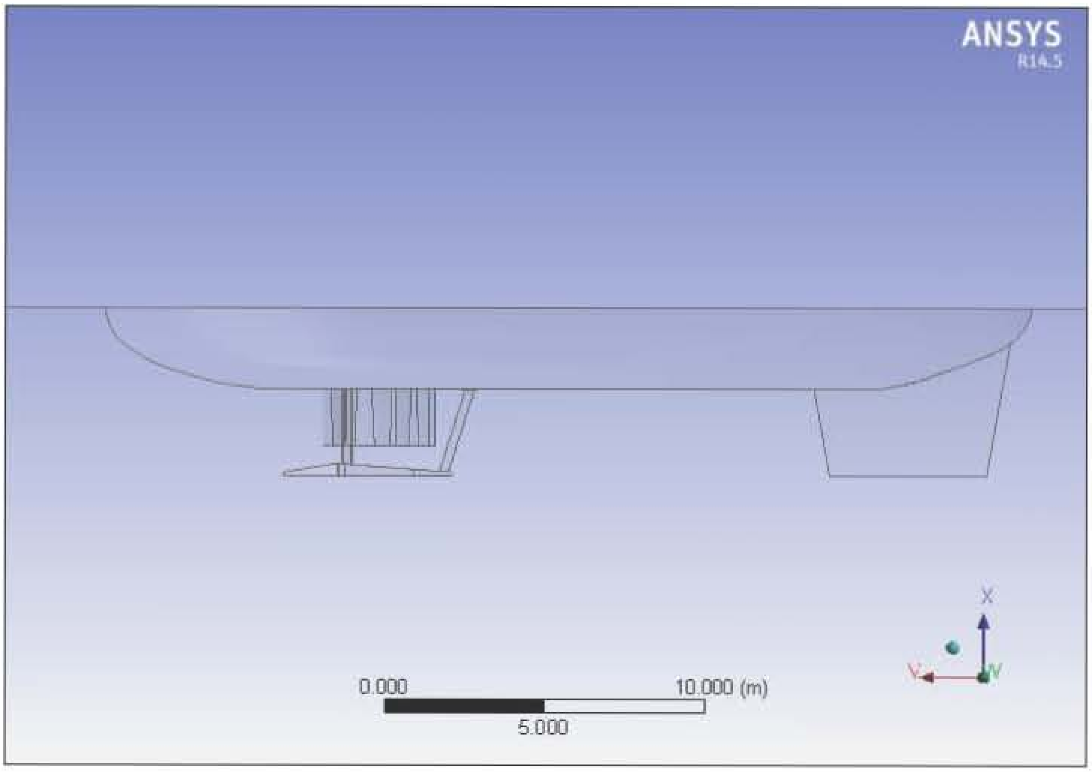


Image 27 - closer view of the tug in Geometry in Ansys Fluent

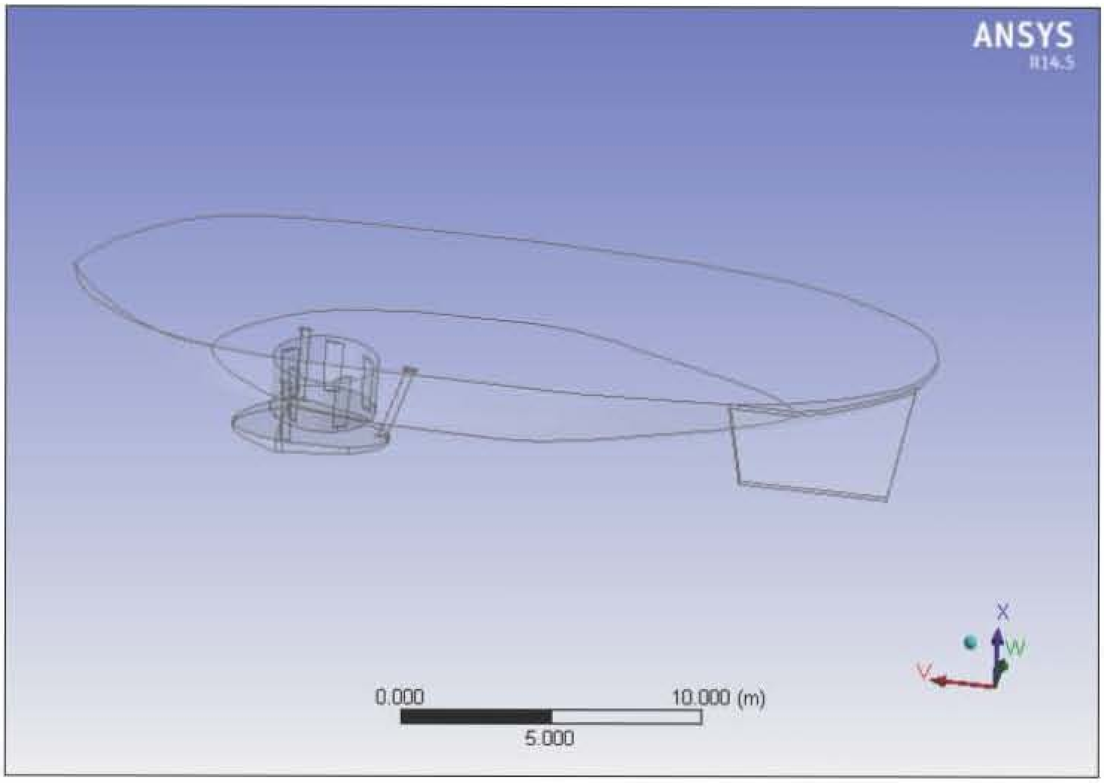


Image 28 - Better view for the VSP and tug in Geometry in Ansys Fluent

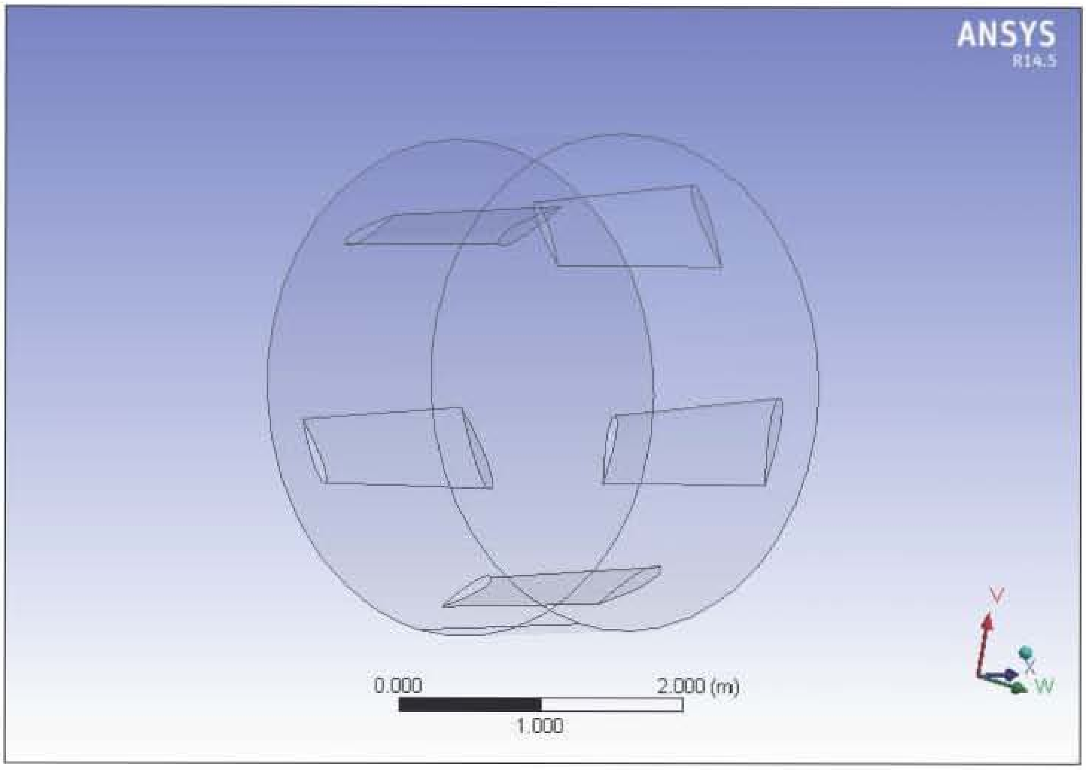


Image 29 - Rotational part of the VSP in Geometry in Ansys

7.2 Mesh

Run the Ansys Meshing, where the mesh corresponding to the geometry obtained in the previous section will be created. The quality of the meshing is critical in order to obtain accurate results. So that, once geometry operations are completed, the next step is to proceed generating the mesh of domains with Ansys Meshing.

Ansys Fluent has several methods to mesh. They can be divided into two groups by using hexahedral or tetrahedral. The hexahedral form a structured mesh while the tetrahedra form an unstructured mesh. It also has many tools to adjust the maximum size of the mesh in certain areas, thus obtaining a higher mesh quality.General mesh properties are shown in the next table:

Table 1 - General mesh properties

Features	
Algorithm	Patch Conforming Method
Method	Tetrahedrons
Use Advanced Size Function	Proximity and Curvature
Relevance Center	Fine
Smoothing	High
Transition	Slow
Min Size	0,001 m
Max Face Size	5 m
Max Size	5 m
Pinch Tolerance	0,007
Defeaturing Tolerance	0,005
Nodes num.	2122808
Element number	10602067
Averaged Asimetry	0,2313
Maximum Asimetry	0,88
Minimum Asimetry	0,000023

In the next picture the difference in mesh size in different areas seen, looking darker areas with the smallest mesh.

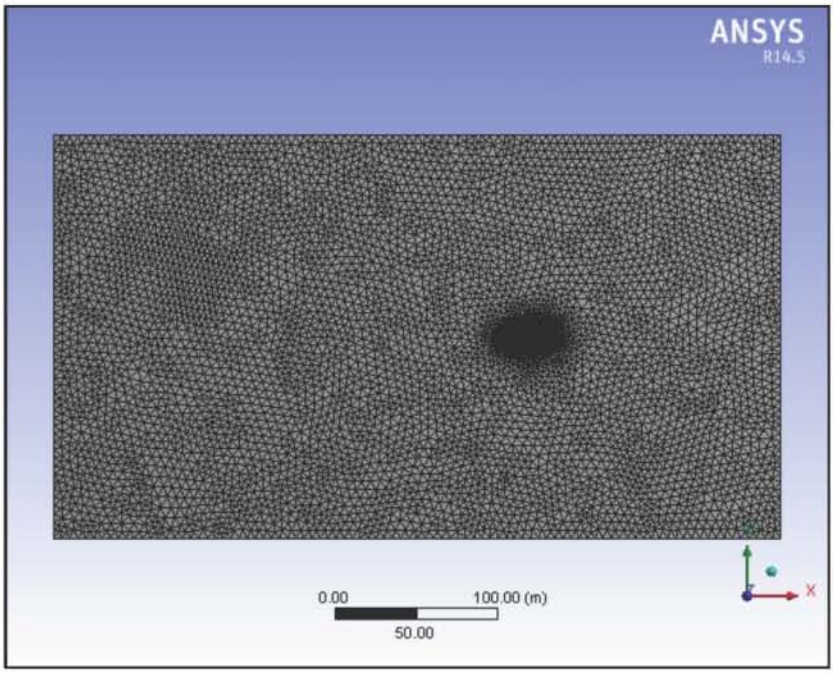


Image 30 - Difference of volume controls size in Mesh in Ansys Fluent

Table 2 - Sizes of the mesh in different parts of the domain

Zones	Face Sizing
Hull	0,1
Skate	0,1
Keel	0,1
Voith schneider	0,02
Interfaces Rotational	0,02
Interfaces Estationary	0,02

The purpose of using the tools of meshing is to adapt the mesh to the problem we want to solve more accurately. The mesh presents more problems where more complex geometry, ie, on the contours of solid objects and regions of interest. In these cases a higher resolution is employed. In this study case is concentrated for example at the rotative domain and skate.

Inadequate mesh can bring to errors in either not converge of the solution equations or not giving correct results. One way to analyze the quality of the mesh is from a visual point of view and observing the asymmetry. To help lower the initial asymmetry of 0.97% I used the following tools and settings from "mesh":

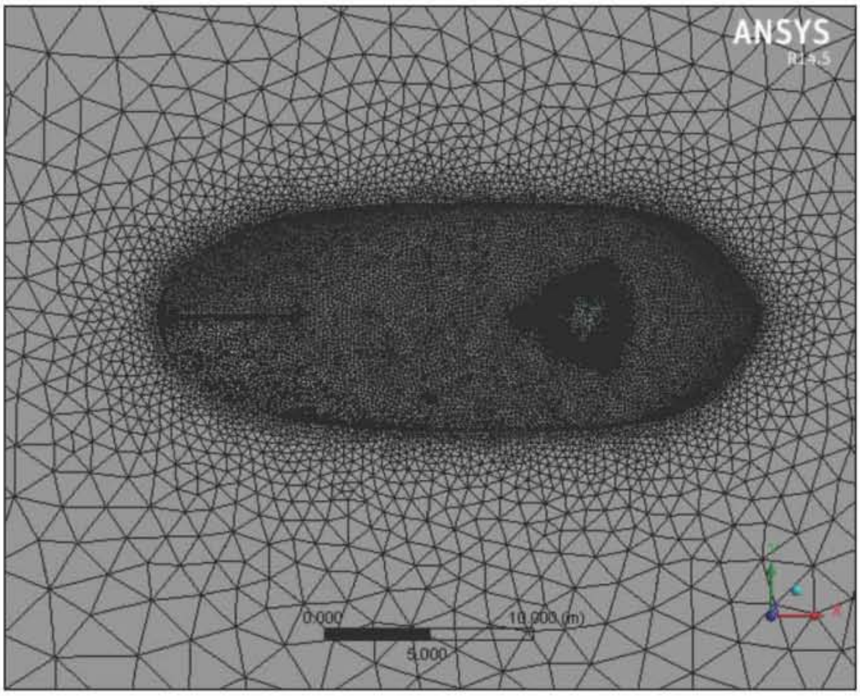


Image 31 - Closer view of hull and VSP mesh

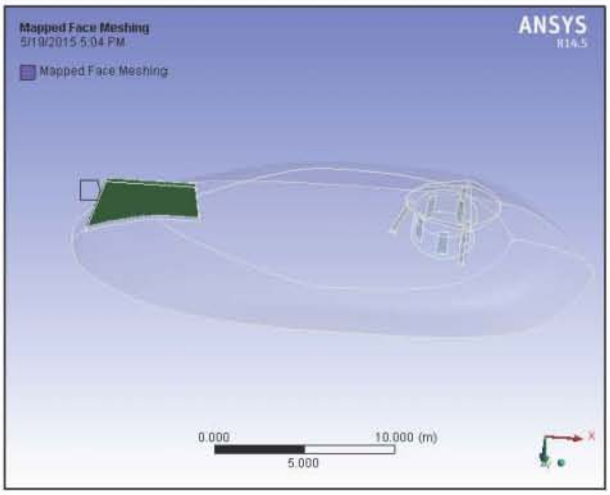


Image 32 - Mapped Face Meshing Ansys Fluent

- Mapped Face Meshing: This tool alproduces an ordered structure on a surface mesh. It is a very useful tool when it comes to reducing the asymmetry and adapt the mesh geometry. This tool has been used in the 5 faces of the keel.

- Patch Independent: This method uses a top-down approach (creates volume mesh and extracts surface mesh from boundaries). The patch-independent method uses the geometry only to associate the boundary faces of the mesh to the regions of interest thereby ignoring gaps, overlaps and other issues that present other meshing tools with countless problems. I apply it to the rotary domain

- Face sizing: Faces requiring greater precision to be sensitive to the validation of the results should have a small element size. With the help of the "Face Sizing" tool, maximum mesh size of the most critical areas are defined. In the case of these simulation, the importance of proof is on the hull and the four interfaces arround the blades, since there is a turbulent regime. Face sizing it is been applied to those and the skid is brought to a value of 0.02m.

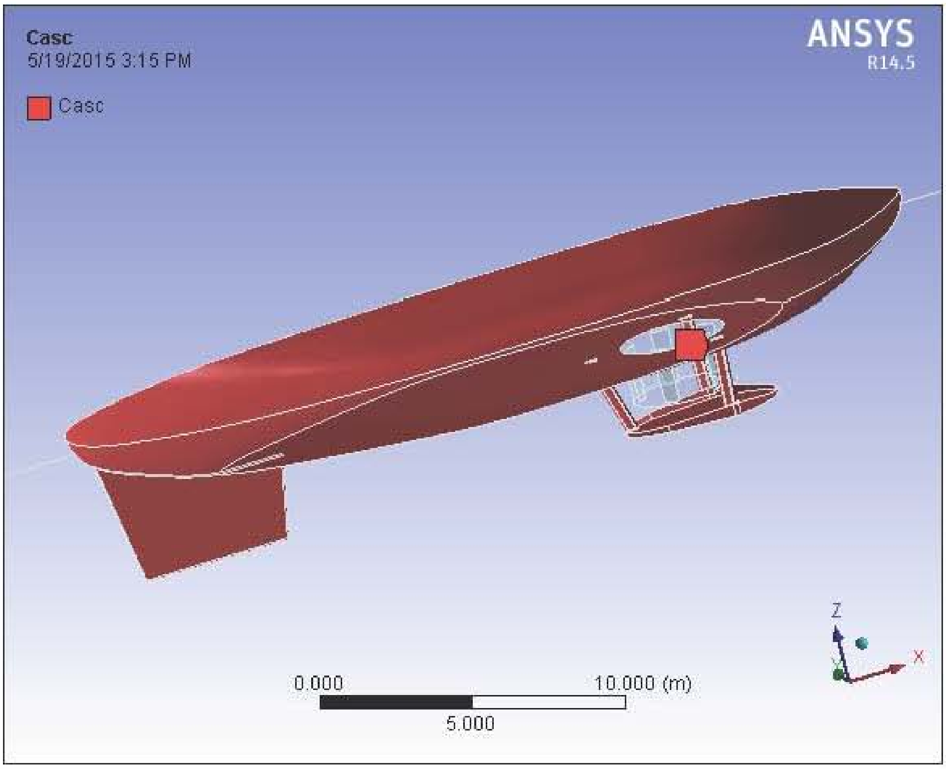


Image 33 - Stationary domain

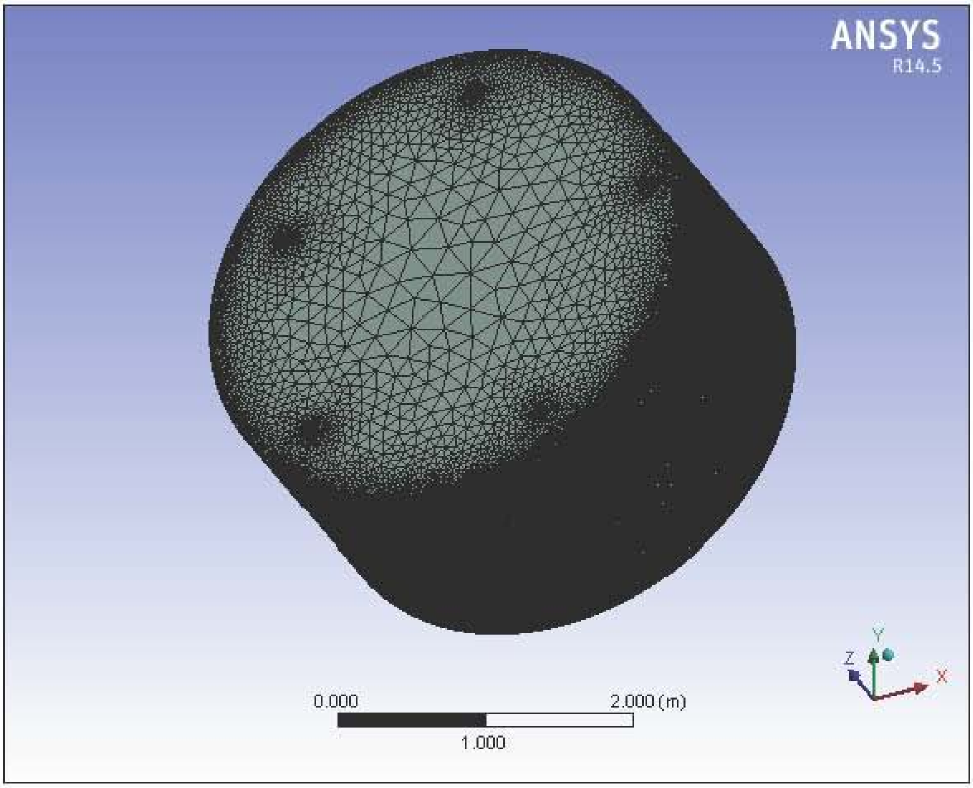


Image 34 - Rotary domain

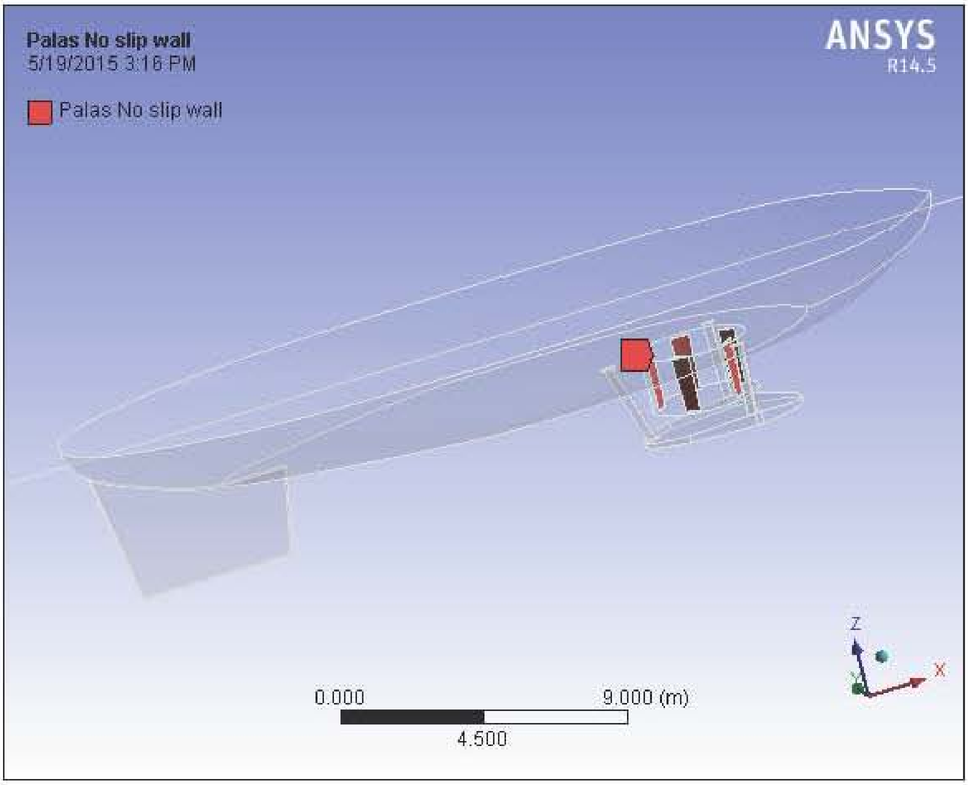


Image 35 - Blades inside the rotary domain

In Ansys Meshing, it is also necessary to name certain areas in order to establish the boundary conditions necessary to perform the simulation settings in the next step *Set up*. For example, the interfaces, the rotary and stationary domain.

Typical mistakes

It is normal in the moment to run the simulations that different kind of problems appears. Here below are two of the most common problems that have arisen and how to deal with them:

- "Overlapping geometry in named selections"

  This problem is quite common in interfaces. It can be given either in the "setup" or in the "mesh". It is recalled, that an interface is defined as the interface between two domains and in this case is between the rotating and the stationary domains. What happens is that there is an overlap of surfaces, which causes a conflict between them. When this problem occurs may be due to several problems. The most two common are: the same wall is defined twice. In this case the "selections named" will be again reviewed to ensure that you have not selected a wall more than once. For example in this project that happened between these two interfaces showed in red in the following images; i.e. the interface of the stationary domain on the first picture and the interface of the rotary domain on the second picture.

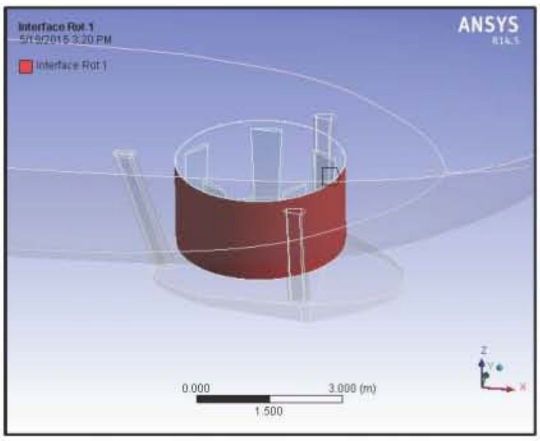


Image 36 - Stationary domain

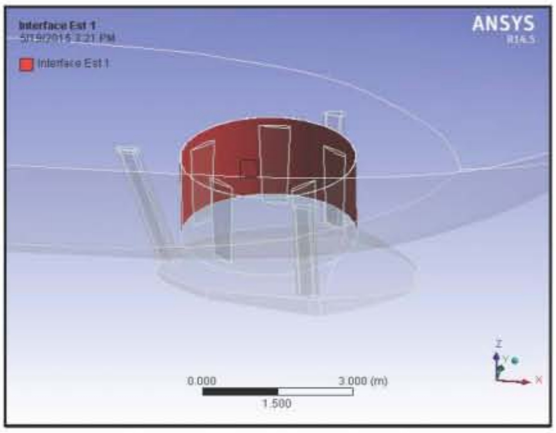


Image 37 - Rotary domain

Or due to the quality of the mesh. If the quality of the mesh is low, for example the "element size" is very large, it can happened that a cell of the stationary domain penetrates in the rotary domain or vice versa. It is accentuated if the interface has right angles. This problem can be corrected by using options in the "sizing". In "relevance centre" it would be better to choose the option "fine" rather than "coarse", choose an "element size" very small and lower the "curvature" angle and check the best meshing option.

7.3 Set up

In these module the type of simulation, boundary conditions and parameters for each simulation conditions will be defined.

Although the Ansys Fluent allows other simulations, this project has been done with the scheme << multiple reference frames (MRF) >>, which is an approximation of steady state in which you can assign different speeds translacional as well as rotational to the different areas of the domain.

A rotation speed is set to the domains of helices (rotational) within a stationary domain. For a frame of reference with constant rotation speed it is possible to make a transformation of the equations of motion of the fluid to a rotating reference frame thus enabling a stationary solution.

In this study has been established rotational speed for the domain of the propeller (rotational domain). One of the main reasons for using a moving frame of reference, is to develop a problem that is unstable in a stationary reference frame. For a frame of reference with constant rotation speed it is possible to make a transformation of the equations of motion of the fluid to a rotating reference frame thus enabling a stationary solution.

A series of names for stationary domain boundaries are established to be able to identify them and see the different boundary conditions established for each one.

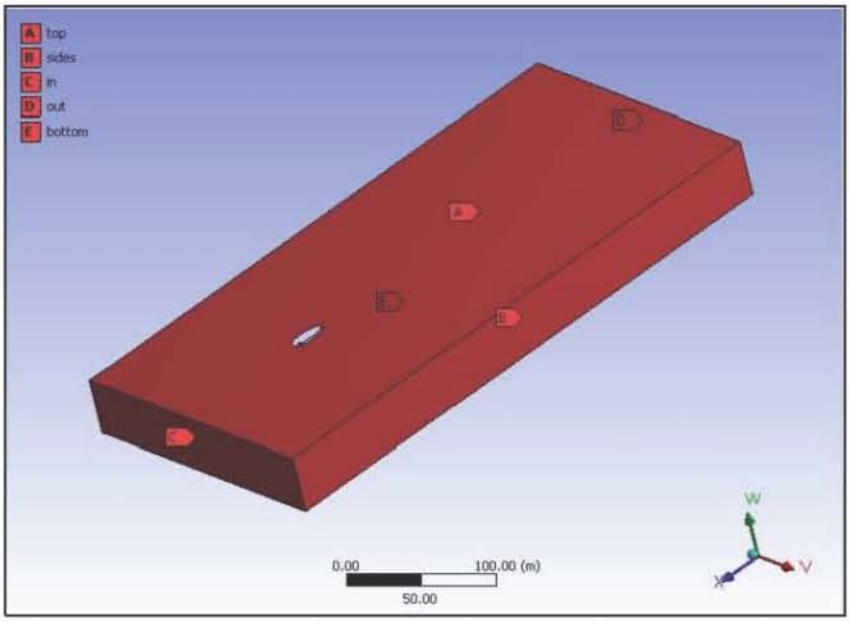


Image 38 – Boundaries

A) Top -> It represents the top domain wall, represents the water surface. It matches the waterline.
B) Sides -> They are the sidewalls of the domain.
C) In -> It is a domain wall located meters forward of tug.

D) Out -> It is a domain wall located aft tug at a distance.

E) Bottom -> Lower domain wall representing the seabed.

Boundary conditions.

Boundary conditions of different types are defined to achieve an appropriate relationship between the fluid and the components of the boat. The domain boundary conditions for the simulations of this project are defined to meet the conditions required from the Classification Societies of the actual bollard pull test.

a) Boundary condition for the surface of the water:

In the water surface does not occur viscous stresses and shear stress is zero. Therefore the boundary condition of *Symmetry type*, which determines that there is no flow through this border, there is no mass transport on this surface and the normal component of velocity is equal to zero, is the perfect condition for it.

b) Boundary condition for the full trailer:

The surfaces of the hull, aft keel, and forward skate thruster elements are of the type wall without sliding (not slip wall). During the test, the tug remains stationary, so that the moving fluid reaches the zero velocity on its surface and has zero speed relative to it, sticking to the surface and generating because of that viscous stresses.

c) Boundary condition for the seabed and boundary domains:

Depending in which are the actual simulation objectives, boundary conditions will change. The sea floor, for example, is an area that is subject to viscous stresses. Although it is an area that can be more or less flat, in reality it is irregular. In the simulation it is approximated to a flat wall, because there is no point in trying to simulate irregularity of a certain seabed. Except in the case of separate studies in which the attention is focused just in the distance of the lateral, bow or stern walls, in which then the condition of the seabed will be *pressure-outlet.* With this option we are simulating that the seabed is really far away an makes no effect to the propeller thrust, turbulences… For the boundary domains free space to bow, stern and sides of the tug or not will be considerate depending if there is the need to simulate a port wall or free sea. Therefore domain boundaries will be in the boundary condition of the type *Pressure Outlet* if it continues being the same fluid without barriers of any kind, or Wall in case we find a wall in a determinate distance.

This two first simulations will prove that the model has been well designed and that the results of the simulation are right and, therefore, we are able to start simulating other cases with this tug and VSP model.

The variable parameters that had been taken for these firsts simulations are the following ones:

- Speed –> To be able to compare my results with results from real bollard pull test I have to take the same velocity that those tugs use. Therefore I run the first simulation at 6,7 rad/s, the second one at 7,6 rad/s and the third one at 8 rad/s. Afterwards other speeds will be simulated for the different studies.

- Sea dimensions -> 450x30x250 m; having 300 meters from aft till the port and 150 meters fore, 125 meter on each side of the boat and 30 meters of depth.

Which boundary condition each -> since in this fist simulation we also want to compare the simulation results with the theoretical results, we want to obtain the best ones so they will be more similar to the theoretical ones and we will obtain less error percentage. In that case we will pretend that there is no walls anywhere that could influence the results of the simulation. In order to obtain this effect all the boundary conditions will be *pressure-outlet*, which acts as if the sea continues beyond that imaginary end. This means that no turbulence due to any modification rebounds against a wall or obstacle.

7.4. Solution

This module allows to analyze the results. Ones all last steps of the program Ansys Fluent are correctly followed you can make the symulation run. In the firsts simulations it is really possible that not all the equations converge due to any little mistake that could have been done in the other parts of the program. After all those little problems are arranged and all equations converge you are able to get the right solutions for what it is been looked for.

The simulations are considered completed with reference to a parameter called "residual". By default, the residual criterion is established at 1x10 -3. Once a value of "waste" is reached below this figure, it is said that the simulation has "converged" and the results are valid.

In the next image we can see the 2000 iterations simulation of the Tug and VSP effect at 8 rad/s represented in a graphic. Each line defines de result of one of the equations and, below, we can see how they already at 500 iterations converge:

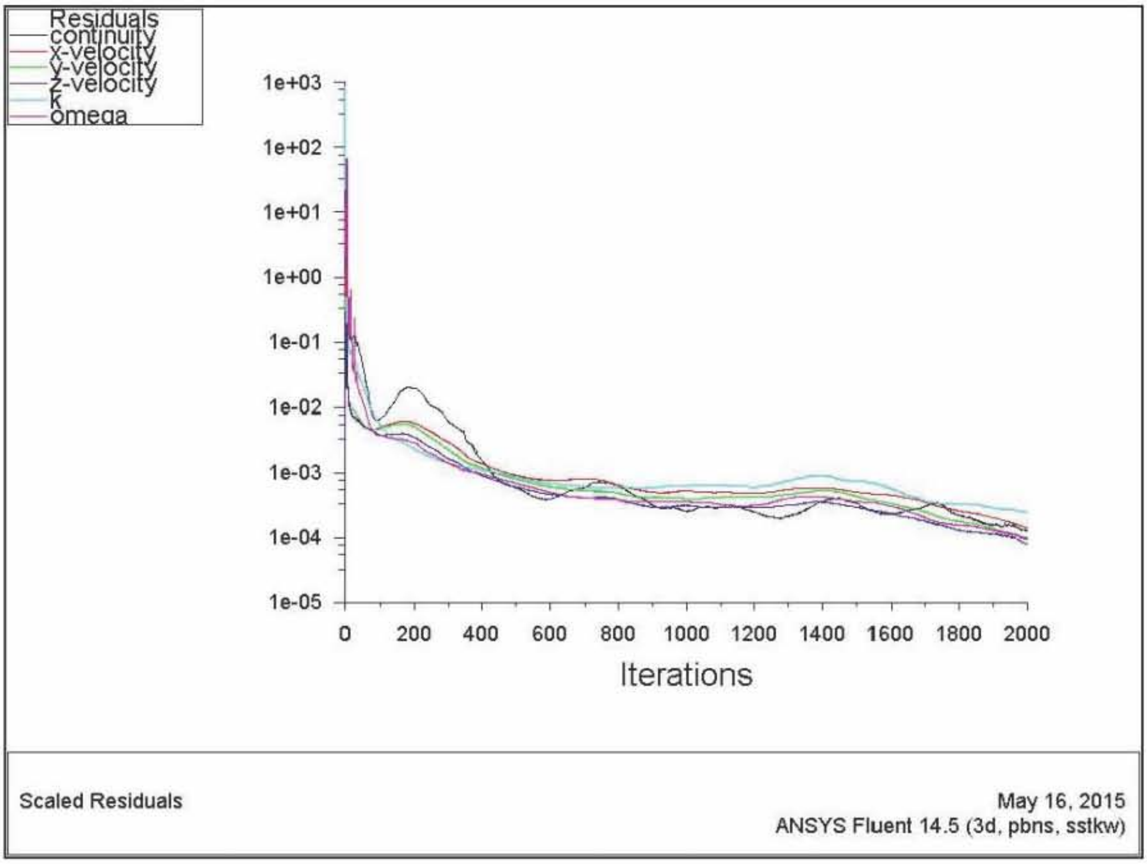


Image 39 - Conversion of the equations in Solutions in Ansys Fluent

Therefore, in following simulations I am just going to run the simulations with 500 iterations, because it is enough to get good results and it consumes between 10 and 15 hours less to finish the iterations. Some converging fails due to the high turbulence are produced, but the residual value approximates the convergence and results are considered but with certain precautions.

There are many results coming out of Ansys Fluent ones the simulation is run, but I am going to focus in the ones that really interest us for this project, and those are:

Static Pressure

Since in the Bollard Pull Test it is supposed to be carried out without current, I would also like to take a look at the effect of the Static Pressure, which, as already said, is the pressure exerted by a fluid that is not moving or flowing.

In the following images we can see how the static pressures are partitioned:

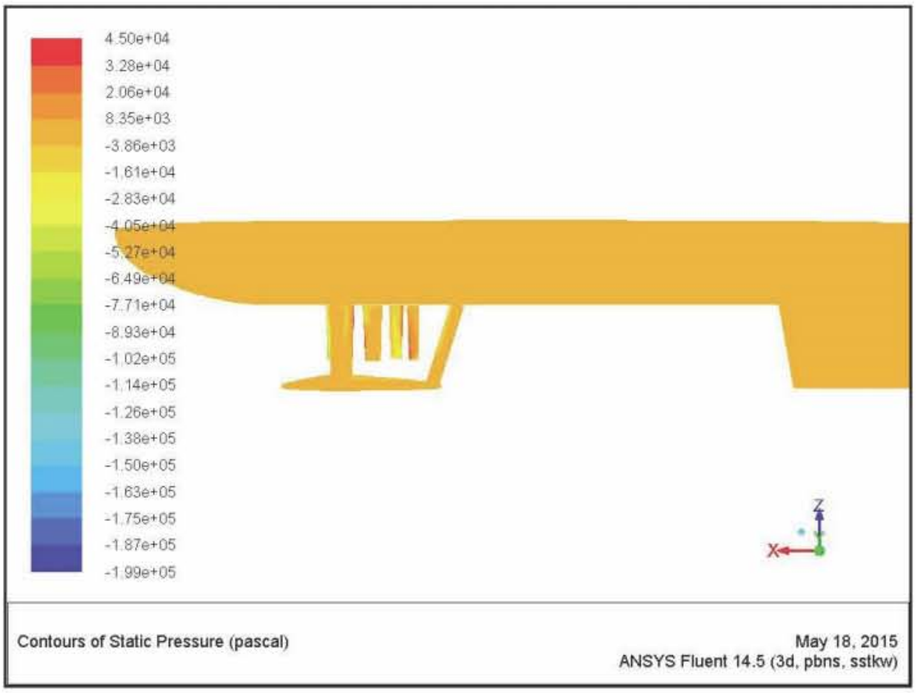


Image 40 - Static Pressure Solutions Ansys Fluent

In the all hull there is no static pressure change, because the fluid have no movement. If we zoom to the rotational domain we see that it is where there is more and less Static pressure (highlighted with yellow and red colours). Some edges, coloured with red, have the maximum static pressure (4,5e+04 pascal) and other blades parts, coloured with yellow, is where there is less static pressure (-2,83e+04 pascal). And, in the next image, we can also appreciate green colour in the inside part of some edges, which represents even less static pressure than the yellow colour.

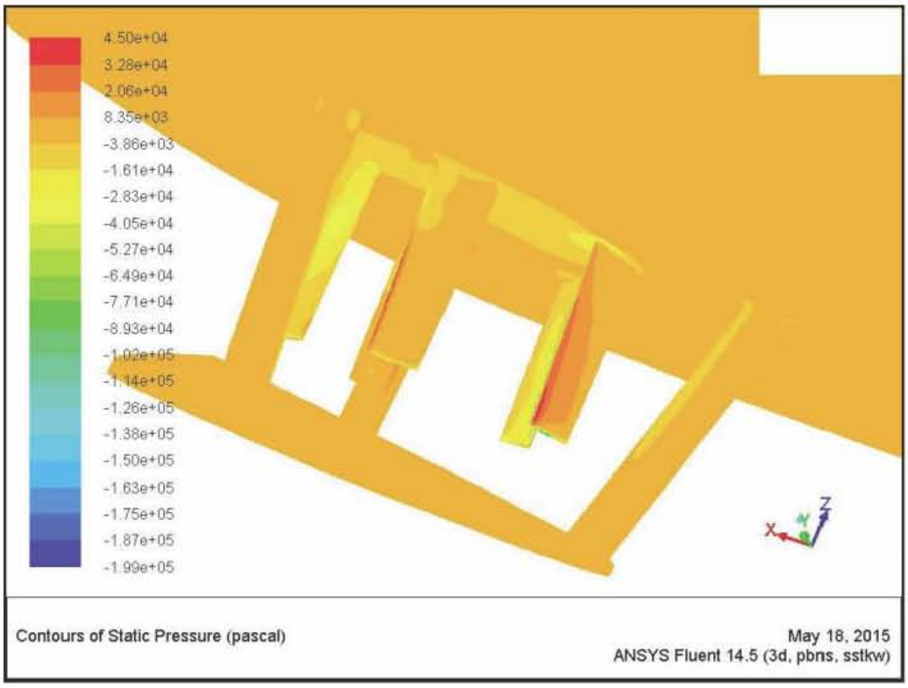


Image 41 - Static Pressure in the Rotary Domain

Following we have an image only of the inside part of the rotational domaine, i.e. the five blades of the VSP and the top part which hold them, where we can appreciate better this chnge of colours:

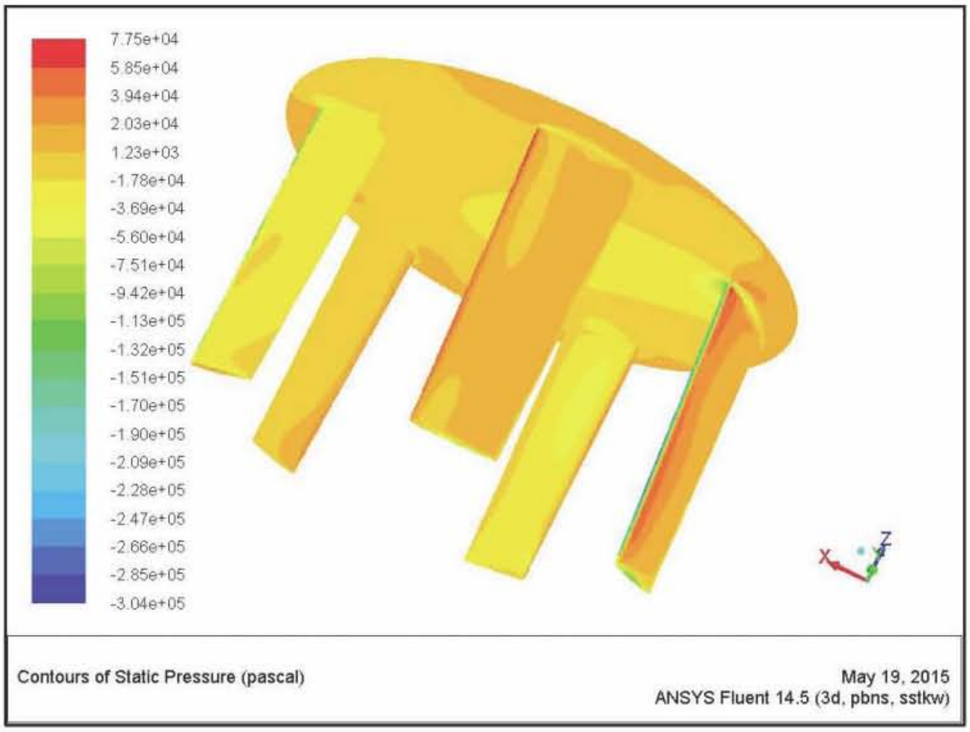


Image 42 - Static Pressure only on the VSP view

Image 43 - Static Pressure only on the VSP view

Total Pressure

The sections forming the blades are hydrodynamic profiles, which generate a difference of speed in the fluid that surrounds them and, therefore, pressures in the active or pressure face and the passive or suction face. However, in the output bode blades in the pressure faces, the highest pressure values that would be represented in red colour are not reached.

The following figure shows the total pressures of the tug. In this case, it is easy to see that there is a pressure change within the hull of the tug, showed with different greens.

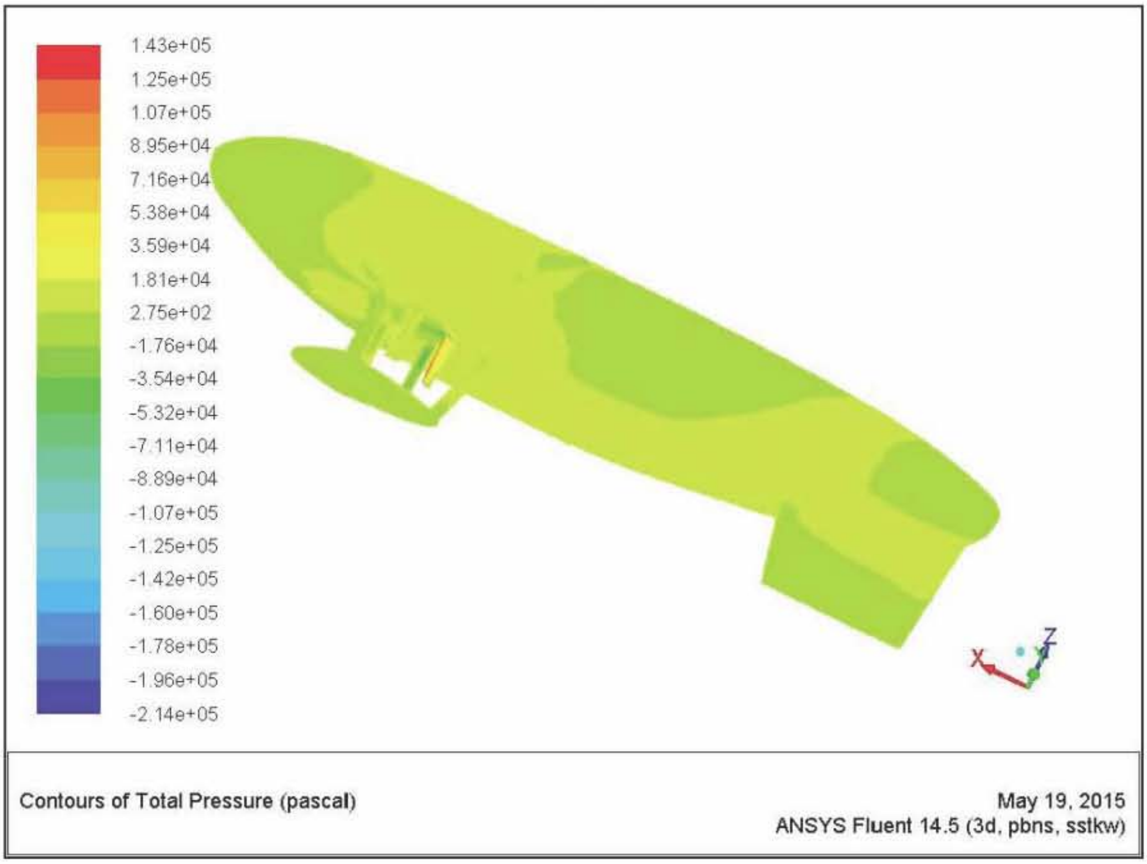


Image 44 - Total Pressure on the hull and propeller

It can be clearly seen in the next image the total pressures partitioned in the propeller blades. Again, as in the static pressure, we find the maximum and minimum pressure values in the edges of the blades (coloured in red and green/blue).

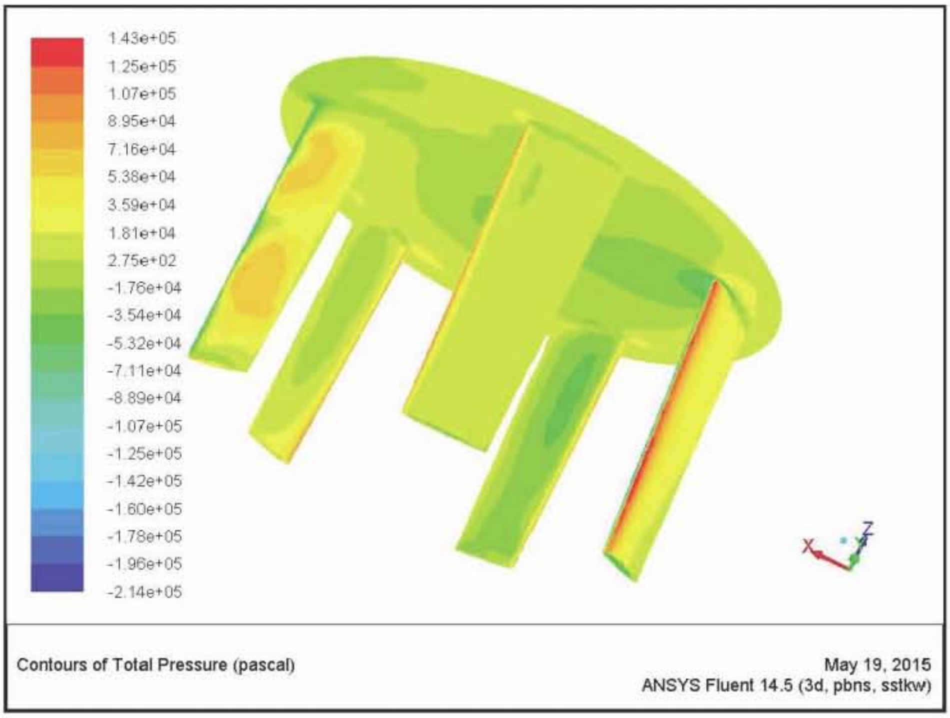


Image 45 - Total Pressure Results only on the VSP

Velocity

The next image shows, the obvious fact, that in the stationary part of the tug the lineal velocity does not change (all coloured in the same dark blue, i.e. 0 rad/s).

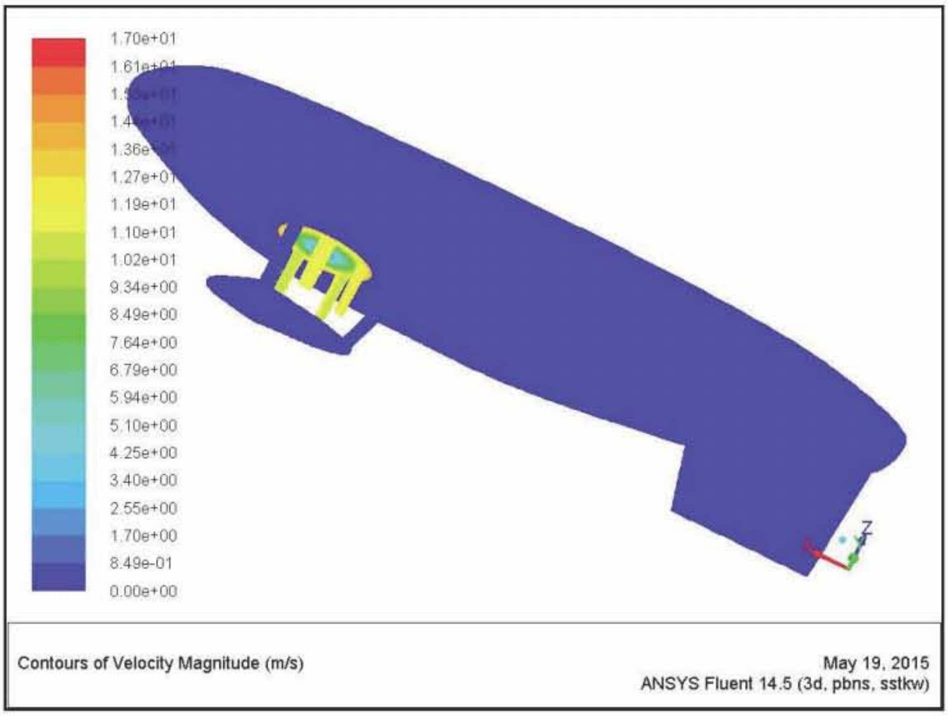


Image 46 - Velocity Magnitude on the hull

In the next picture you can see the distribution of the linear velocity in rad/s in each of the blades and the cover of the propeller. This image corresponds to the linear velocities calculated for an angular velocity of -8 rad/s.

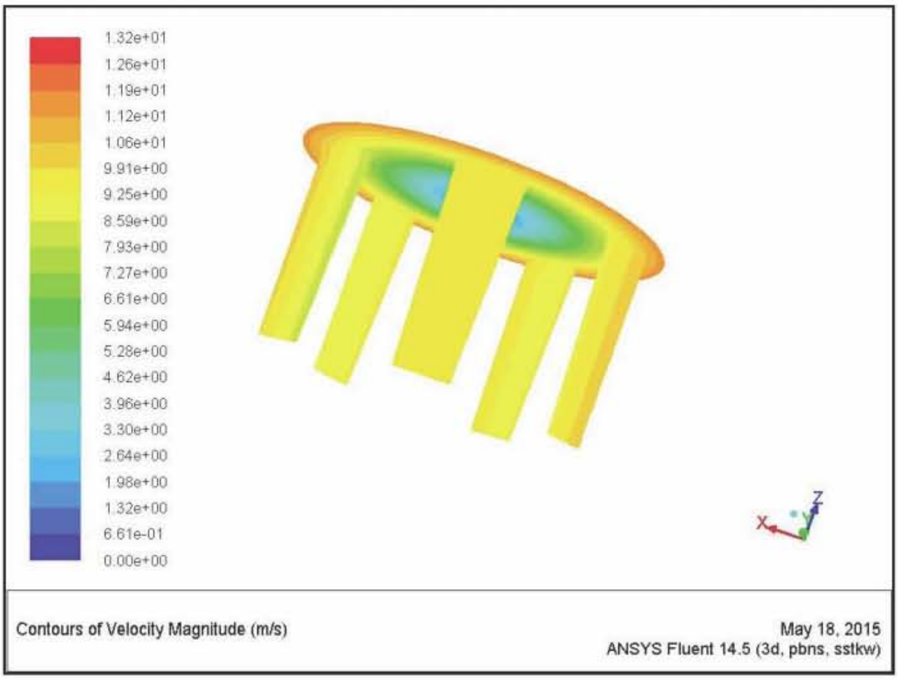


Image 47 - Velocity Magnitude on the Propeller

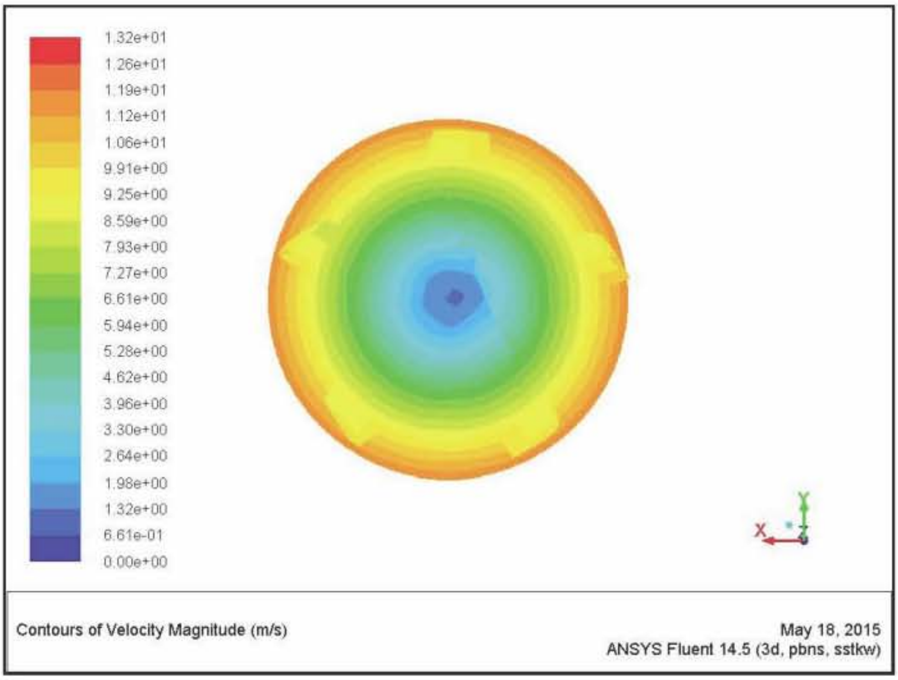


Image 48 - Another view of the Velocity Magnitude on the VSP

Turbulences

Finally, the turbulence produced by the propellers of the vessel will be studied.

As a benchmark the "Turbulent Viscosity Ratio" parameter is used. It is directly proportional to the Reynolds number and is considered high value of about 100 to 1000. The Reynolds number relates the density, viscosity, speed and size of a typical flow in a dimensionless expression. In the wake produced by a ship, the flow is considered turbulent and therefore the Reynolds number is high.

In the following images it is clear to see the turbulences generated by the propeller:

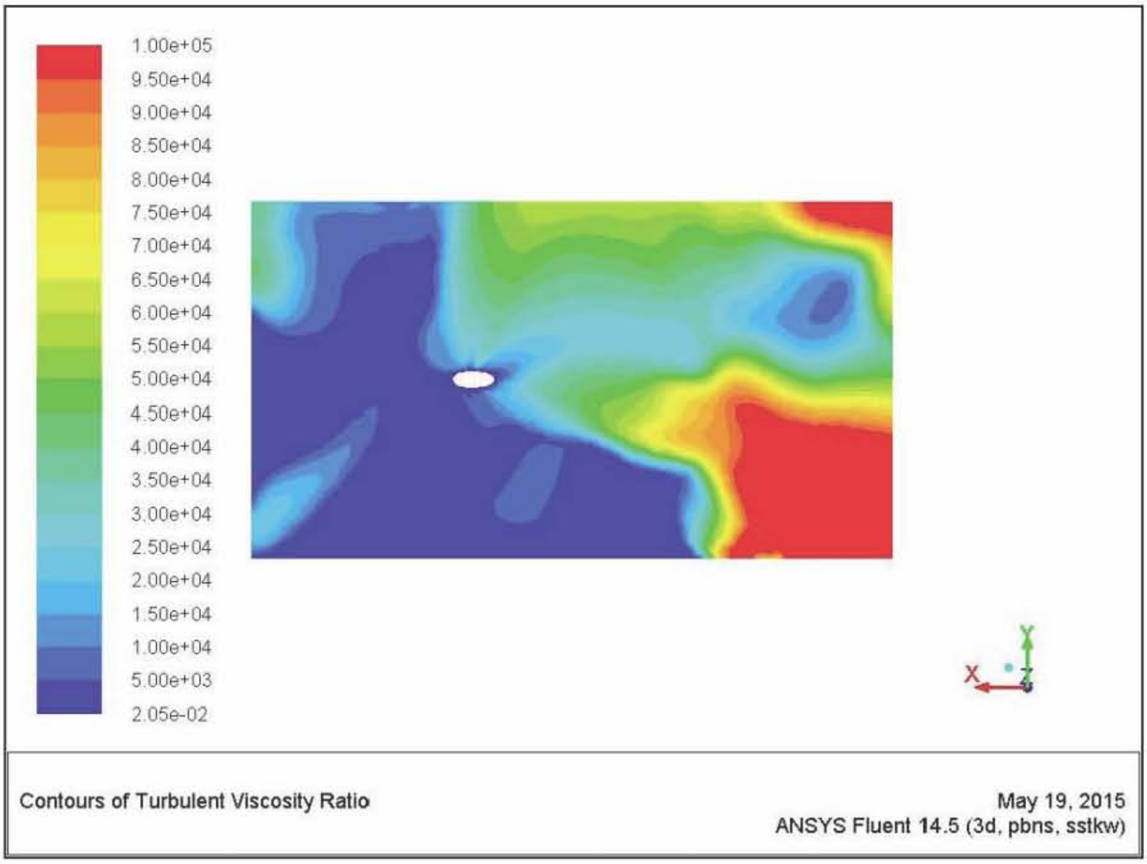


Image 49 - Turbulent Viscosity Ratio on the "sea" produced by the VSP

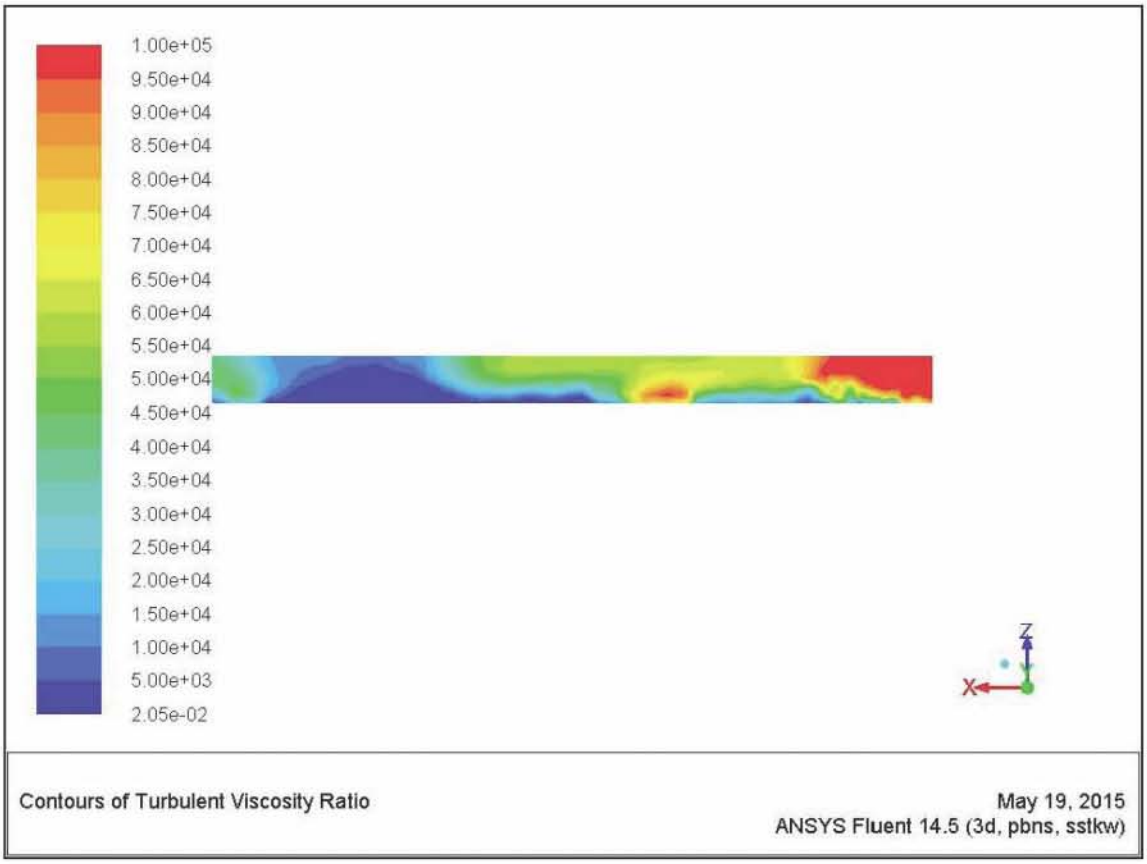


Image 50 - Another view of the Turbulent Viscosity Ratio produced by the VSP

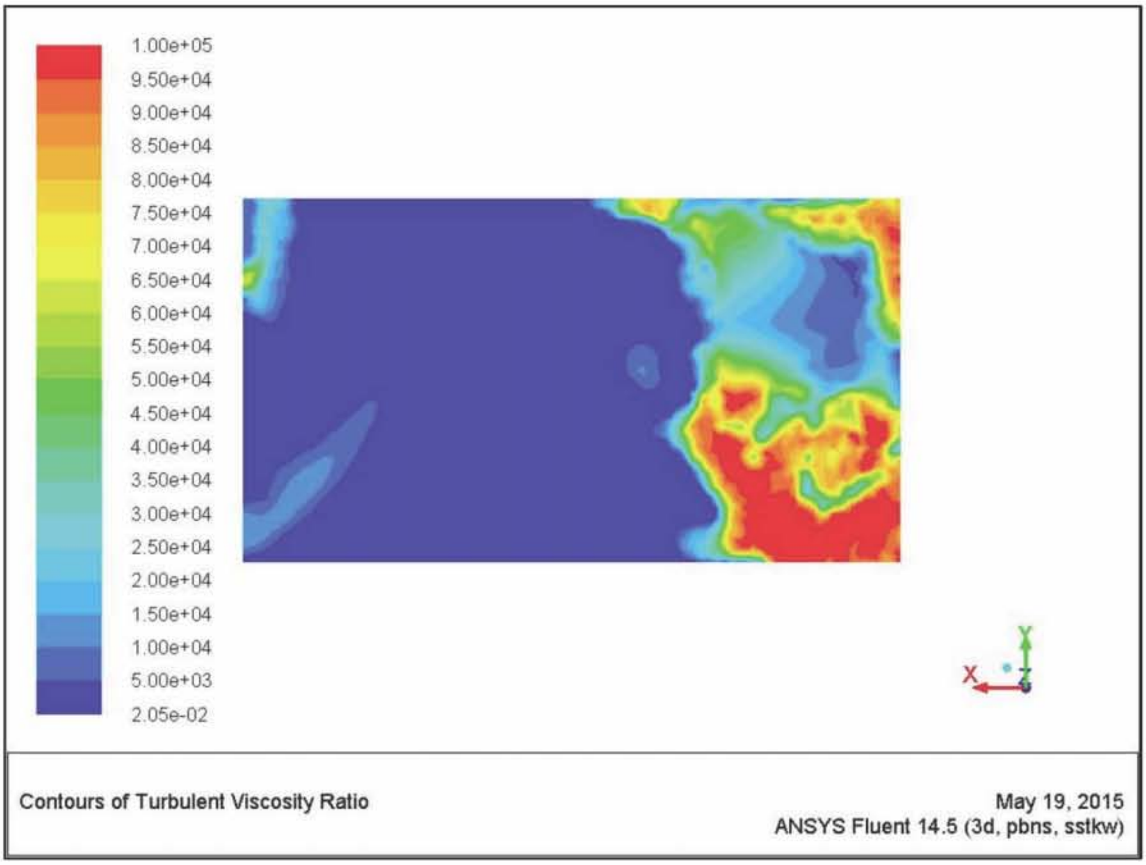


Image 51 - View of the Turbulences at the Sea bottom

Force reports

In the following excel table, copied as a PDF, the forces reports (pressure force, viscose force, total force, normal torque, viscous torque and total torque) of the x, y and z-axis in the hull, blades, the cover of the propeller and the addition of them are to be seen:

Table 3 – Model at -8rad/s Force Report from Ansys Fluent

VSP Velocity Study: -8 rad/s		Forces Report			Pressure Coefficients			
	Component	Pressure Force (N)	Viscose Force (N)	Total Force (N)	Normal Torque	Viscoso Torque	Total Torque	
Hull		-7.941	-1.578	-9518,762	-12964,363	-2576,4732	-15540,836	
Blades		133666,08	-362,539	133303,54	218230,33	-591,901	217638,43	direcc X
Cover Prop		0	-277,816	-277,8158	0	-453,577	-453,577	
Total		125725,41	-2218,445	123506,96	205265,97	-3621,951	201644,02	
	Component	Pressure Force (N)	Viscose Force (N)	Total Force (N)	Normal Torque	Viscoso Torque	Total Torque	
Hull		3.019,1946	187,577	3206,772	4929,297	306,248	5235,545	
Blades		-73910,594	-269,391	-74179,984	-120670,36	-439,822	-121110,18	direcc Y
Cover Prop		0	-181,407	-181,407	0	-296,175	-296,175	
Total		-70891,399	-263,221	-71154,62	-115741,06	-429,748	-116170,81	
	Component	Pressure Force (N)	Viscose Force (N)	Total Force (N)	Normal Torque	Viscoso Torque	Total Torque	
Hull		84.977,867	154,043	85131,911	138739,38	251,5	138990,88	
Blades		5954,691	125,913	6080,604	9721,9444	205,573	9927,517	direcc Z
Cover Prop		-90617,531	0	-90617,531	-147946,99	0	-147946,99	
Total		315,027	279,957	594,984	514,32956	457,073	971,403	

In the next chapter this and the values from other rotational velocities of the propeller simulations will be studied to validate the study model to be able to study the bollard pull test.

8. Validation of the model with real cases

8.1 Check of the rotation direction

Before starting with the validation of the model, a little prove has been done to check if the propeller was properly well design to turn counter clockwise. In the following tables we find the Forces report for 6,7 rad/s turning clockwise and counter clockwise:

Table 4 - Model at 6,7 rad/s Force Report

VSP Velocity Study : 6,7 rad/s		Forces Report			Pressure Coefficients			
	Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	X-axis
Hull		-2,498	176	-2322,223	-4077,9281	286,543	-3791,385	
Blades		-34842,719	221,054	-34621,665	-56886,071	360,904	-56525,167	
Cover Prop		0	29,609	29,609	0	48,341	48,342	
Total		-37340,45	426,17	-36914,279	-60964	695,788	-60268,211	
	Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	Y-axis
Hull		-5.719	-403,647	-6122,4	-9336,744	-659,015	-9995,759	
Blades		-23872,664	54,576	-23818,088	-38975,778	89,104	-38886,674	
Cover Prop		0	-41,071	-41,071	0	-67,054	-67,054	
Total		-29591,419	-390,1412	-29981,561	-48312,522	-636,965	-48949,487	
	Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	Z-axis
Hull		77.659,148	276,051	77935,2	126790,45	450,696	127241,14	
Blades		1292,2615	38,947	1331,209	2109,8147	63,587	2173,402	
Cover Prop		-81438,352	0	-81438,352	-132960,57	0	-132960,57	
Total		-2486,942	314,998	-2171,943	-4060,3129	514,283	-3546,029	

Table 5 - Model at -6,7 rad/s Force Report

VSP Velocity Study: -6,7 rad/s	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	X-axis
Hull	1.016	-947.914	68,138	1658,86	-1547,615	111,246	
Blades	86741,156	-96,87	86644,286	141618,21	-158,156	141460,06	
Cover Prop	0	-27,67	-27,679	0	-45,19048	-45,191	
Total	87757,2	-1072,46	86684,745	143277,08	-1750,961	141526,11	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	Y-axis
Hull	-3.496	69,569	-3426,636	-5708,0899	113,583	-5594,507	
Blades	-48641,001	-153,048	-48794,056	-79413,89	-249,874	-79663,765	
Cover Prop	0	-37,669	-37,669	0	-61,5	-61,5007	
Total	-52137,213	-121,148	-52258,361	-85121,98	-197,793	-85319,77	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	Z-axis
Hull	8.643,442	387,230	9030,679	14111,743	632,222	14743,966	
Blades	2703,105	161,781	2864,886	4413,2322	264,133	4677,365	
Cover Prop	-34910,199	0	-34910,199	-56996,244	0	-56996,244	
Total	-23563,652	549,018	-23014,634	-38471,269	896,356	-37574,913	

As we can see, the pressure force (N) for the x direction (in which is meant to go during a bollard pull test), we obtain a negative and much smaller force result for the clockwise simulation (6,7 rad/s), -37.340,45 N, next to the 87.757,2 N that are obtained from the counter clockwise (-6,7 rad/s) simulation. Therefore, it is clear that the Voith Schneider Propeller for that study project it is made to turn counter clockwise. As logical, turning clockwise the tug would move backwards, which is also really important for some manoeuvres, but in this direction the VSP produce less pressure force.

8.2 First validation

The first validation was carried out with a rotation speed of 6.7 rad/s and is going to be compared with the values obtained from the file "GmbH & Co. KG Gross Harald 13th December 2002":

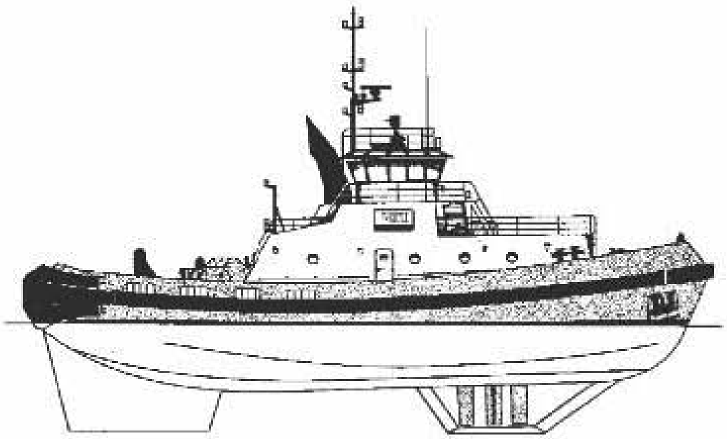
VOITH VOITH-SCHNEIDER-PROPELLER MODELS (950e) Page 4

Similarity of the Model to the Prototype

In our similarity investigation we use our VOITH WATER TRACTOR as an example. The model was built for deployment on the West Coast of the United States.

The principal dimensions are:

Length overall	30.48 m
Length in the waterline	29.26 m
Moulded breadth	10.97 m
Total draught	4.11 m
Draught of hull	2.97 m
Displacement	560 t
Engines	2 * 1100 kW
VSP Type propellers	28GII/185
Rotor speed	64.4 /min
Forward bollard pull	343 kN
Open-water speed	12.7 kn



VOITH WATER TRACTOR with 2 Propellers

Image 52 - Voith Water Tractor

Even this boat has two propellers, we are going to make a first comparison with it because most of the tugboats, which bring this VSP, have two propellers like our one (28GII / 180) and because the size of the it is almost like our one.

This tugboat provides a thrust of 343Kn, which divided between two propellers is 171,5 KN each at 64.4 rpm, which is equivalent to 6,7 rad/s. In the following image we find the results of the simulation made with our tug model:

Table 6 - Model at -6,7 rad/s Force Report

VSP Velocity Study: -6,7 rad/s	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	1.016	-947.914	68,138	1658,86	-1547,615	111,246	
Blades	86741,156	-96,87	86644,286	141618,21	-158,156	141460,06	direcc X
Cover Prop	0	-27,67	-27,679	0	-45,19048	-45,191	
Total	87757,2	-1072,46	86684,745	143277,08	-1750,961	141526,11	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-3.496	69,569	-3426,636	-5708,0899	113,583	-5594,507	
Blades	-48641,001	-153,048	-48794,056	-79413,89	-249,874	-79663,765	direcc Y
Cover Prop	0	-37,669	-37,669	0	-61,5	-61,5007	
Total	-52137,213	-121,148	-52258,361	-85121,98	-197,793	-85319,77	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	8.643,442	387,230	9030,679	14111,743	632,222	14743,966	
Blades	2703,105	161,781	2864,886	4413,2322	264,133	4677,365	direcc Z
Cover Prop	-34910,199	0	-34910,199	-56996,244	0	-56996,244	
Total	-23563,652	549,018	-23014,634	-38471,269	896,356	-37574,913	

This first simulation provides forces quite far away from the real Tug thrust. In the simulation I obtained 8,949 Tons (87757,2 N) and in the real Tug bollard pull 17,48 Tons (171420,243 N) for each propeller is obtained. Following the error of the simulation next to the real test is calculated:

$$e_{rel} = \frac{T_s - T_r}{T_r} = \frac{87757{,}2 - 171420{,}243}{171420{,}243} = -0{,}488$$

In that case, the mistake is about the 48,8%, which is really big, but before making any conclusion, I am going to make more comparisons with other tugboats and different speeds.

Looking it in another way, instead of dividing the thrust produced by the two VSP of the tug bollard pull test, another option is to double the rotational speed of our model propeller pretending there is two, and the force report from the program is:

Table 7 - Model at -13,4 rad/s Force Report

VSP Velocity Study: -13,4 rad/s		Forces Report			Pressure Coefficients			
	Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull		-18.162,2380	-3.649,286	-21811,524	-29652,634	-5958,017	-35610,651	
Blades		377368,41	-1133,757	376234,65	616111,68	-1851,033	614260,65	direcc X
Cover Prop		0	-828,026	-828,026	0	-1351,879	-1351,8788	
Total		359206,170	-5611,069	353595,1	586459,05	-9160,929	577298,12	
	Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull		-3.383,8877	-64,042	-3447,9294	-5524,715	-104,558	-5629,273	
Blades		-225292,13	-741,568	-226033,69	-367823,88	-1210,7234	-369034,6	direcc Y
Cover Prop		0	-548,365	-548,366	0	-895,291	-895,291	
Total		-228676,01	-1353,975	-230029,99	-373348,59	-2210,572	-375559,16	
	Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull		207.015,130	379,253	207394,38	337983,88	619,188	338603,07	
Blades		17735,363	424,521	18159,884	28955,695	693,096	29648,791	direcc Z
Cover Prop		-229182,61	0	-229182,61	-374175,69	0	-374175,69	
Total		-4432,121	803,774	-3628,347	-7236,116	1312,284	-5923,832	

$$e_{rel} = \frac{T_s - T_r}{T_r} = \frac{359206{,}17 - 343000}{343000} = 0{,}047$$

And, that way, good values are found with just 4,7% of error.

8.3 Second validation

The other source corresponds to the values of the harbour tugboat Smit Sandom "Bollard Pull Test" (IMO number is 9120152):

CONSTRUCTION/CLASSIFICATION	
Year of construction	1996
Classification	Lloyd's Register 100 A1 Tug, LMC, UMS
IMO number	9120152

FEATURES
Air-conditioned accomodation for 8 persons
Bunker capacity: potable water 30 m³, fuel oil 180 m³

DECK EQUIPMENT	
Towing hook	Mampaey, quick release, 50 t SWL
Aft winch	Hydraulic Double drum - 90 T. brake load Line pull 17 T. @ 11 m/min. on 1st layer

MAIN DATA	
Gross tonnage	398
Length overall	30.54 m
Beam overall	11.52 m
Max. draught	5.31 m
Main engines	2 x Deutz SBV8M628
Total power	3,100 kW
Propulsion	2 x Voith-Schneider
Bollard pull (ahead)	41 t
Speed ahead (max.)	13 kn
Speed ahead (economic)	9 kn

NAVIGATION AND COMMUNICATION EQUIPMENT	
Magnetic compass	Unilux Geomar
Echo sounder	Furuno FCV 5SZL
VHF	Sailor SP3110, R-2100
Radar	Sperry Marine Bridgemaster
GPS	Magnavux MX200

Image 53 - Smit Sandom tugboat information (www.smit.com/uploads/media/smit-sandom.pdf)

This tug boat measures 30 m long and 5.3 m deep with Voith Schneider propulsion 2 x 28 GII / 185, therefore, it is really similar to the study case and allow me to compare the results with the simulations results. Moreover the hull design is also approximate. The value of the "bollard pull" is 41t (402072,651 N) at 73 rpm. Two Voiht Schneider propellers 28 GII / 185 give 41t (402,072.651 N) force to that Tug. That is, each propeller contributes with 20.05t (201036,325 N) versus the 12.39t (121504,393) that the simulation performed that we can see in the next data table:

Table 8 - Model at -7,6 rad/s Force Report

VSP Velocity Study: -7,6 rad/s	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-6.950	-1.293	-8243,233	-11347,231	-2111,108	13458,339	
Blades	128454,393	-368,289	128086,104	202348,55	-601,287	201747,26	direcc X
Cover Prop	0	-150,662	-150,662	0	-245,979	-245,979	
Total	121504,393	-1.812,005	119692,209	191001,319	-2958,374	214959,62	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-931,047	-20,138	-951,185	-1520,077	-32,878	-1552,955	
Blades	-70208,578	-200,608	-70409,186	-114626,25	-327,523	-114953,77	direcc Y
Cover Prop	0	-70,345	-70,345	0	-114,849	-114,849	
Total	-71.139,625	-291,091	-71430,716	-116146,327	-475,25	-116621,574	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	78.284,258	118,023	78402,281	127811,03	192,691	128003,72	
Blades	5257,213	52,681	5309,894	8583,205	86,01	8669,215	direcc Z
Cover Prop	-85911,5	0	-85911,5	-140263,67	0	-140263,67	
Total	-2.370,029	170,704	-2199,325	-3869,435	278,701	-3590,735	

The difference in results aggravated by the fact that this tug has only one propeller blade with a length of 1800 mm in front of Smit Sandom, which length is a little bit more (1850 mm), that fact gives greater pressure on the blades and consequently stronger pressure.

$$e_{rel} = \frac{T_s - T_r}{T_r} = \frac{121504{,}393 - 201036{,}325}{201036{,}325} = -0{,}39561$$

In that case, the error decreases to 39,561%.

8.4 Third validation

In this third validation, we are also making a comparison with a two Voith Shneider 28G Propeller Tugboat, the Smit Waterloo:

CONSTRUCTION/CLASSIFICATION	
Year of construction	1987
Classification	Lloyd's Register of Shipping 100 A1 Tug
IMO number	8610289

FEATURES
Air-conditioned accomodation for 8 persons
Bunker capacity: potable water 15 m³, fuel oil 67 m³

DECK EQUIPMENT	
Towing hook	Britannia, quick release, 40 ton SWL
Aft winch	Hydraulic Split drum - 70 T. brake load Line pull 30 T. @ 6 m/min. on 1st layer

MAIN DATA	
Gross tonnage	298
Length overall	31.10 m
Beam overall	9.30 m
Max. draught	4.50 m
Main engines	2 x Ruston RK270M
Total power	2,532 kW
Propulsion	2 x Voith-Schneider
Bollard pull (ahead)	36 t
Speed ahead (max.)	13 kn
Speed ahead (economic)	9 kn

NAVIGATION AND COMMUNICATION EQUIPMENT	
Magnetic compass	Cooke & Sons 3829
Echo sounder	Elliott E100
VHF	Sailor TR031, R2022
Radar	Koden MDC-410
GPS	Furuno GP31

Image 54 - Smit Waterloo Information (www.smit.com/uploads/media/smit-waterloo.pdf)

As we can see in the tug information, this vessel produces 353,0394 kN (36 Ton) at 13kn, 176,52 KN per propeller, but it does not specify how many rpm. This time I am going to make a comparison with a simulation of our model with a higher rotation speed of the VSP, for example at 8 rad/s:

Table 9 - Model at -8 rad/s Force Report

VSP Velocity Study: -8 rad/s	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscose Force (N)	Total Force (N)	Normal Torque	Viscoso Torque	Total Torque	
Hull	-7.941	-1.578	-9518,762	-12964,363	-2576,4732	-15540,836	
Blades	133666,08	-362,539	133303,54	218230,33	-591,901	217638,43	direcc X
Cover Prop	0	-277,816	-277,8158	0	-453,577	-453,577	
Total	125725,41	-2218,445	123506,96	205265,97	-3621,951	201644,02	
Component	Pressure Force (N)	Viscose Force (N)	Total Force (N)	Normal Torque	Viscoso Torque	Total Torque	
Hull	3.019,1946	187,577	3206,772	4929,297	306,248	5235,545	
Blades	-73910,594	-269,391	-74179,984	-120670,36	-439,822	-121110,18	direcc Y
Cover Prop	0	-181,407	-181,407	0	-296,175	-296,175	
Total	-70891,399	-263,221	-71154,62	-115741,06	-429,748	-116170,81	
Component	Pressure Force (N)	Viscose Force (N)	Total Force (N)	Normal Torque	Viscoso Torque	Total Torque	
Hull	84.977,867	154,043	85131,911	138739,38	251,5	138990,88	
Blades	5954,691	125,913	6080,604	9721,9444	205,573	9927,517	direcc Z
Cover Prop	-90617,531	0	-90617,531	-147946,99	0	-147946,99	
Total	315,027	279,957	594,984	514,32956	457,073	971,403	

As we increase the rotational speed, the thrust produced by the propeller increases, in this case up to 125725,41 N (12,82 Ton).

$$e_{rel} = \frac{T_s - T_r}{T_r} = \frac{125725{,}41 - 176519{,}7}{176519{,}7} = -0{,}288$$

There is an error of 28,8% of the real tug thrust. If we keep increasing the rotational speed of the propeller in the model, for example at 10 rad/s we obtain the following data:

Table 10 - Model at -10 rad/s Force Report

VSP Velocity Study: -10 rad/s	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-11.334	-2.239	-13573,504	-18504,601	-3656,222	-22160,824	
Blades	209483,39	-562,052	208921,34	342013,7	-917,637	341096,06	direcc X
Cover Prop	0	-481,242	-481,242	0	-785,702	-785,702	
Total	198149,32	-3282,731	194866,59	323509,1	-5359,561	318149,54	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	1.620,3481	71,918	1692,266	2645,466	117,417	2762,883	
Blades	-128320,08	-425,842	-128745,92	-209502,17	-695,252	-210197,42	direcc Y
Cover Prop	0	-304,766	-304,766	0	-497,578	-497,578	
Total	-126699,73	-658,69	-127358,42	-206856,7	-1075,413	-207932,11	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	129.965,760	240,358	130206,12	212188,99	392,42	212581,41	
Blades	9597,024	214,992	9812,017	15668,611	351,007	16019,619	direcc Z
Cover Prop	-134632,22	0	-134632,22	-219807,7	0	-219807,7	
Total	4930,564	455,3497	5385,913	8049,899	743,428	8793,328	

With 10 rad/s speed of the propeller the simulation results obtain 198149,32 N (20,206 Ton), higher than the real Smit Waterloo tugboat, which produces 36 Ton with two VSP, i.e. 18 Ton each.

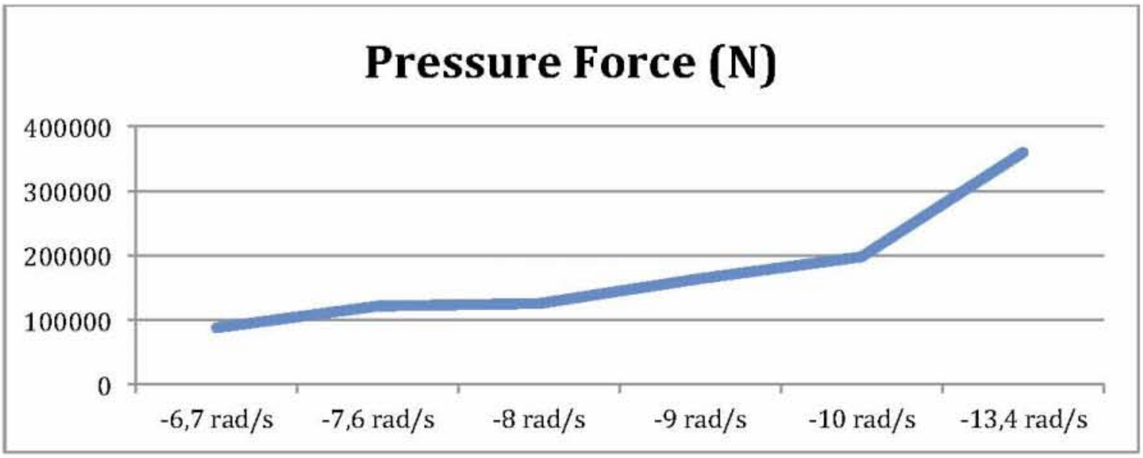
Here we can see that the model is right but it needs a higher rotational speed of the propeller to achieve the same thrust produced in the reality. In the next chapter, there is a study of simulations with different velocities and the operational and maximum power for this model will be chosen.

8.5 Propeller rotational speed study

In this velocity of the VSP study several speeds will be probed. With the graphic made out of the results, will be possible to analyse the thrusts results given for each velocity and to conclude which is the maximum and operational power of this VSP tug model.

To perform this study, the bottom, the laterals, the bow and stern boundary conditions have been established as *pressure outlet* because anything apart from the speed is meant to influence. In the following table there are the results of the simulations made to study the effects of the velocity changes in the X-axis:

Pressure Force



Graphic 1 - Pressure Force (N) at different speeds

As already said, the thrust increases together with the rotational speed of the propeller, as logical. In the reality, the operational speed of that concrete Voith Schneider propeller is between 6,7 rad/s and 7,6 rad/s. But there are so many factors that influence in both the real bollard pull tests and in this model like:

- The 1 VSP model tug is being compared to tugs that bring two of them (because there is no thrust results of 1VSP tug with the same characteristics as the model in internet), that produce already a big error.

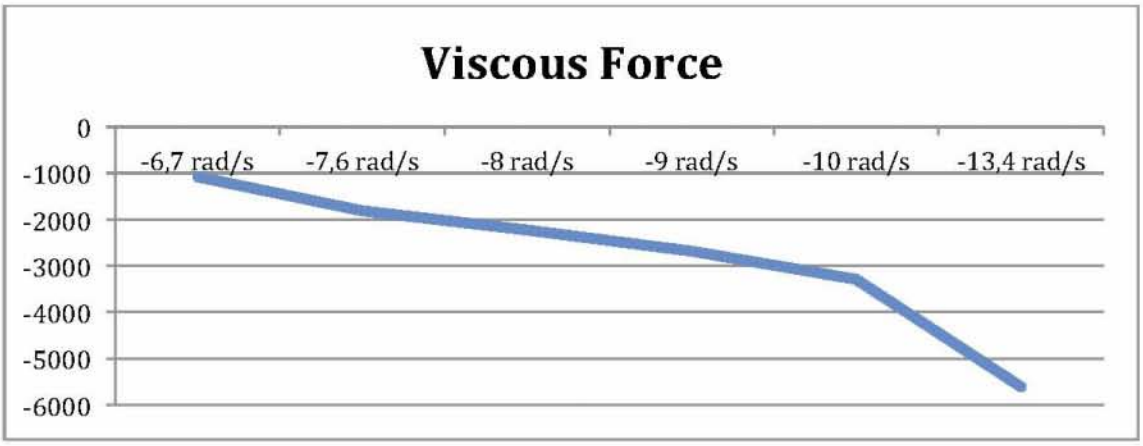
- The factors that influence the test in real sea conditions are different to those ones simulated in the Ansys program. In these simulations there is 0 current, which in the reality is impossible. Moreover, in this study, it has not been taken the wind into account.

- Another fact is that I cannot know, which where the characteristics of the zone where those tugs where tested, i.e. maybe there was not enough depth or maybe there was more than the estimated in the simulations. And like the example of the depth, that later in this project the effect that this produce is studied, there is many more parameters that influence. So, since this is information that just the shipyards have, it is impossible to perfectly recreate the concrete zone of the test in the program in this project to be able to make a more accurate comparison of thrust results.

- Furthermore, I do not have the real results of the tests of those tugs, so the comparison is being done with the only thrust value the shipyard gives at one concrete speed.

So, taking all this error estimations into an account and considering that we want to study the effect of the obstacles in the bollard pull test, not the efficiency of the propeller in this concrete study model, I am going to use the 8rad/s as a operational power because it approaches the real thrust results, and, at the same time, it is not that far away from the 7,6 rad/s (used in some boats as the operational power). And I will use 10 rad/s for the maximum power. To be able to make comparisons with an easier reference from the simulation with 8 rad/s speed and no boundary conditions set as “wall”, we will refer us to it with the name “model simulation”.

Viscous force:

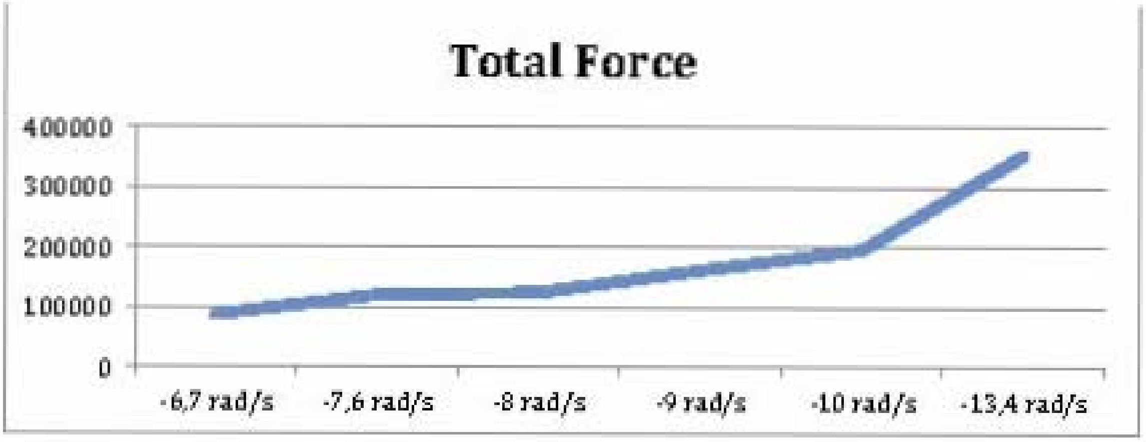


Graphic 2 - Viscouse Forse at different speeds

As we can observe in the results tables, the viscous force also increases with the velocity but with negative sign.

For this reason, the Total Force will be smaller than the Pressure force. Anyway, as we can see in the following image, the graphic would follow the same form as the Pressure Force graphic.

Pressure Force



Graphic 3 - Total Force at different speeds

Torque

The program Ansys, when it gives the forces report, also gives information about the torque, which is following explained and commented within this speeds study.

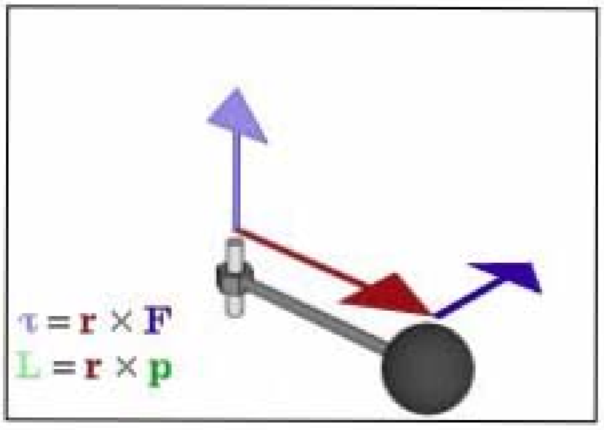


Image 55 - Torque representation (wikipedia)

Torque, moment, or moment of force, is the tendency of a force to rotate an object about an axis or pivot. Just as a force is a push or a pull, a torque can be thought of as a twist to an object. Mathematically, torque is defined as the cross product of the lever-arm distance vector and the force vector, which tends to produce rotation.

The magnitude of torque depends on three quantities: the force applied, the length of the *lever arm* connecting the axis to the point of force application, and the angle between the force vector and the lever arm. In symbols:

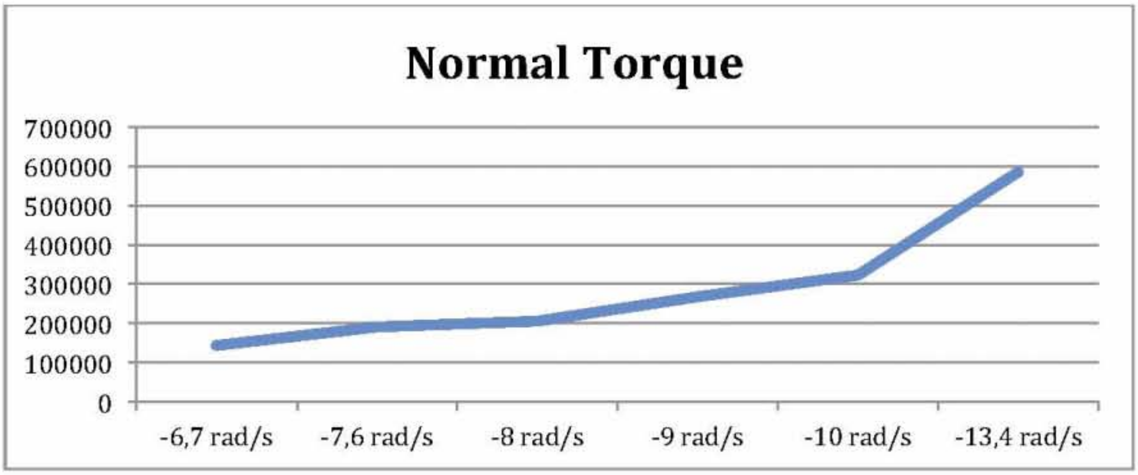
$$\tau = r \times F$$

$$\mathrm{T} = \|r\| \|F\| \cdot sin\theta$$

Where

- τ Is the torque vector and T is the magnitude of the torque
- r is the displacement vector (a vector from the point from which torque is measured (typically the axis of rotation) to the point where force is applied)
- F is the force vector
- $\times$ Denotes the cross product
- θ is the angle between the force vector and the lever arm vector.

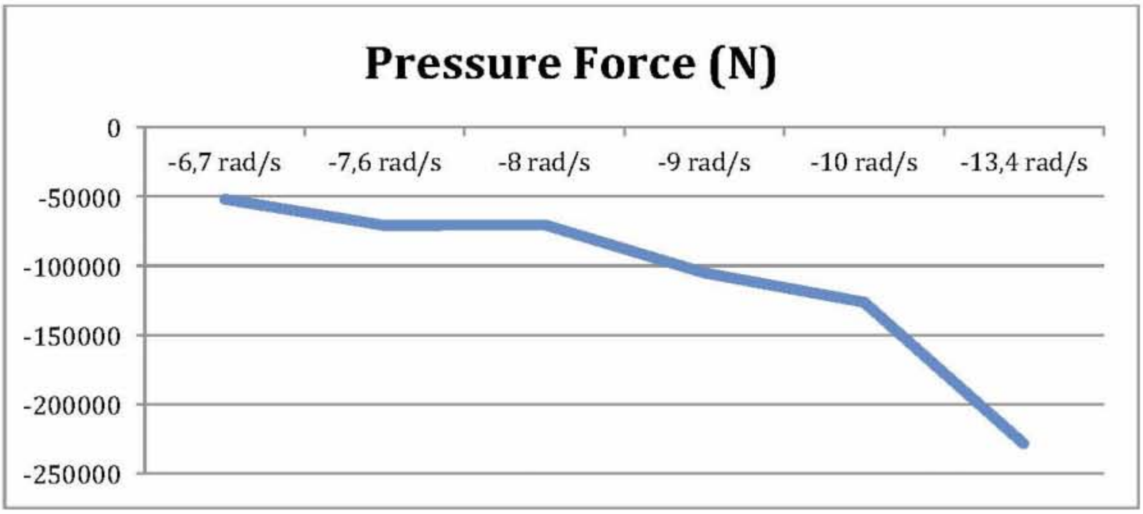
The length of the lever arm is particularly important; choosing this length appropriately lies behind the operation of levers, pulleys, gears, and most other simple machines involving a mechanical advantage. In this study case the lever arm is the distance between the centre of the control rod and the part that attaches each blade. The SI unit for torque is the newton metre (N·m). In the following graphic we find the results of the velocities study simulations normal torque. As in the Pressure Force case, these values will be a little less in the moment to calculate the total torque because the viscous torque is, in the x-axis, negative.



Graphic 4 - Normal Torque at different speeds

8.6 Y-axis and Z-axis

In the graphic the negative pressure force produced by the VSP in the Y-axis is showed:



Graphic 5 - Pressure Force in the Y-axis

That could make us think, that this would produce an infinite turn of the boat, but it actually protects the balance of the boat as it is following explained.

Balance effect

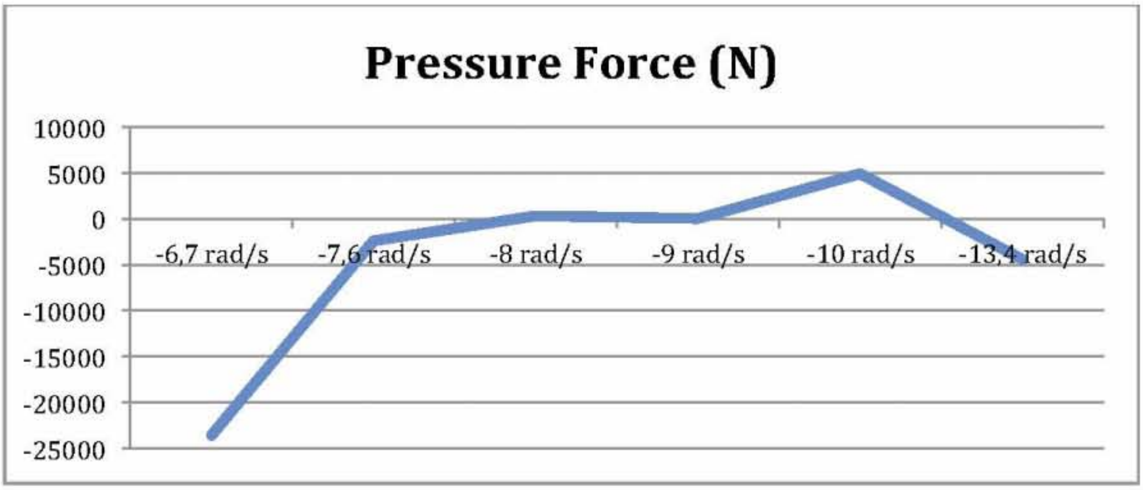
As already explained in chapter 2, the Voith Schneider Propeller not just gets great stability, because it decrease the centre of gravity, but also due to the propeller blades requiring a high orbit. The VSP itself with its vertical axis has a strong stabilising effect, because the VSP blades have a high lever relative to the rolling centre of the tug and therefore they are efficient even without active anti-roll steering.

Moreover, as we see in the next table, the blades produce the same force in both directions y and –y when the boat is going straight. The negative sign of the Pressure Force of the direction Y (-70891,399 N) just indicates the direction of the force, therefore the force of the direction –Y goes in the right opposite direction, being that way both forces cancelled by each other and giving stability to the tug.

Table 11 - Pressure Forces at -8 rad/s at Y and -Y directions

Component	Pressure Force (N)	Viscose Force (N)	Total Force (N)	Normal Torque	Viscoso Torque	Total Torque	
Hull	3.019,1946	187,577	3206,772	4929,297	306,248	5235,545	
Blades	-73910,594	-269,391	-74179,984	-120670,36	-439,822	-121110,18	direcc Y
Cover Prop	0	-181,407	-181,407	0	-296,175	-296,175	
Total	-70891,399	-263,221	-71154,62	-115741,06	-429,748	-116170,81	
Component	Pressure Force (N)	Viscose Force (N)	Total Force (N)	Normal Torque	Viscoso Torque	Total Torque	
Hull	-3.019,1946	-187,577	-3206,772	-4929,297	-306,248	-5235,545	
Blades	73910,594	269,391	74179,984	120670,36	439,822	121110,18	direcc -Y
Cover Prop	0	181,407	181,407	0	296,175	296,175	
Total	70891,399	263,221	71154,62	115741,06	429,748	116170,81	

Another Pressure Force direction interesting to check is the force in the Z-axis, which is, at different speeds, in the next graphic shown:



Graphic 6 - Pressure Force on Z-axis

That direction of the force it is a little complex to analyse, because if the force it is positive, the VSP is actually pushing the tug under water, and, if it is negative, it is pushing the tug to the air. Neither of both is the actual porpoise of the propeller, that is why I am going to understand that results as a bad influence factors on some simulations. As we already have seen earlier, with a slow rotational speed the results of the thrust were not matching with the real ones, and, at this same low speeds it is where the pressure force in the Z-axis is giving not logical results. Therefore, I will look at the stable logical results of Z-axis force, which are between 8 and 9 rad/s where it is close to 0. In the upper forces report table, we can see that the Z-axis force is 315 N for 8 rad/s, which makes sense for two reasons. The first reason it is that 315 N for a Tug it is a really small force, therefore it does not make a big influence to the thrust (which is a force of a 100.000 N order). And the second reason it is, that with 5 vertical blades rotating it is normal that produces an effect to the surrounding water in all directions.

And, to conclude, we can see how after 9 rad/s the propeller starts producing a negative force in the Z-axis, which would pull the tug up to the air. That is one of the reasons why this propeller is design to work at low speed, because with more speed it produces more thrust (as it may seem if we would just check the

X-axis), but also it produces more forces in other directions which interrupt the good functioning of the propeller.

Finally, we can here see again that 8 rad/s it is a good speed for this model and will give the more accurate results (closer to the reality) in the bollard pull test study that follows in this project.

8.7 Conclusion

The simulations model that is being used produces the similar thrust as in the real proves depending on the speed at which the propeller rotates. The right amount of speed is to be found between 8 and 10 rad/s. There is no interest to find the exact speed that gives the same thrust as the boats that I made the comparison with, because there is many factors that influence those results, as already explained, so it would not be the real rotation speed that the real tug used to give that concrete thrust. Between these factors we find the current, the wind, the obstacles in the test place…

On the other hand, ones we know the error in the moment to make a comparison with a real tugboat bollard pull test, we can still doing the next study with absolute numbers. The interest of this study is to check the effect that produces certain factors to the bollard pull test, not to use this concrete model to anticipate the thrust in the design phase. Therefore there is no need to obtain the same thrust that in the real proves, it is just important to see that the model simulations thrust results are close to the reality, which means that the model is well designed and the program conditions are well programed.

So ones decided the operational and the maximum rotational speeds of the propeller, those are the ones that will be used in the rest of the study.

9. Simulation studies of different bollard pull test zones

The simulations that will be carried out in this project will study the behaviour of the tug and the Voith Schneider Propeller in different conditions for the bollard pull test. This propeller it is not like a traditional helix, therefore, with the results of those simulations, the normative of the bollard pull test will be questioned. The traditional conditions of the bollard pull test were designed for a regular propeller and may not be appropriate for this propeller. To achieve that I will run several simulations in which different parameters will remain constant or vary depending on which parameter/s I am studding in each.

Anyway, the following parameters remain constant in all simulations:

- Mesh configuration.
- The geometry of the hull, skate, keel and propeller.
- The boundary conditions of the hull, skate, keel and propeller.
- The size and position of the rotational domains.
- Simulation methods.

And below there are the parameters that can vary from one simulation to another:

- Speed of the rotational domains.
- Boundary condition limits of the stationary domain.
- The dimensions of the geometry representing the stretch of sea where the test was performed.

So that, in the following chapters the followings parameters will be studied through simulations:

- Deep of the sea in the bollard pull test zone
- Free distance on the sides of the boat
- Distance between the harbour and the stern
- % Of error of the test when there is current
-

I performed a study in order to analyse the behaviour of the thrust force generated by a vessel in different locations where space does not have the required dimensions, in order to achieve the best results possible from the bollard pull test. The study also takes into account the velocity od the propeller and the effect produced by water currents on the performance of the test.

After these evaluations, the actual bollard pull test conditions will also be simulated with several variations on the boundary conditions of the sea-box.

From all those results I am going to obtain conclusions and prove if the actual Classification Societies bollard pull test rules fit with the Voith Schneider Propeller. If those rules do not work as well for this kind of propeller, I will propose new conditions to perform that test with a VSP.

Distances will be taken from a reference point defined in the tugboat, which will be the same as that used for CAD modelling of the ship. In our case this point is located at the intersection of the centreline, the baseline and the frame 0.

The following section names are defined to be able to study the forces exerted on each element of the tugboat:

- Hull -> Surface of the hull, skate and aft keel.
- Blades -> rotatory domain inside where there are the blades of the VSP.
- Cover Prop. -> Superior part of the VSP, which is attached to be hull and do not rotate.
- Bottom -> seabed.
- Laterals -> lateral walls of the port.
- Stern -> wall of the port aft of the boat.
- Bow -> natural or artificial interruption of the free sea fore of the boat.

9.1. Depth of the seabed study

The boundary conditions employed for the study of the effects of the depth below the vessel are *pressure-outlet* to all the boundaries except the bottom, were I assigned the condition *wall*.

This first simulation it is made with the double of the tug draft, so 10,8 meters (because it is the minimum that would accept the Classification Societies Rules), the second one is performed with a 15 meters deep seabed, the third one with 20 meters deep (the depth recommended by the Classification Societies), and afterwards there is a simulation each 10 meters till 60 meters of depth.

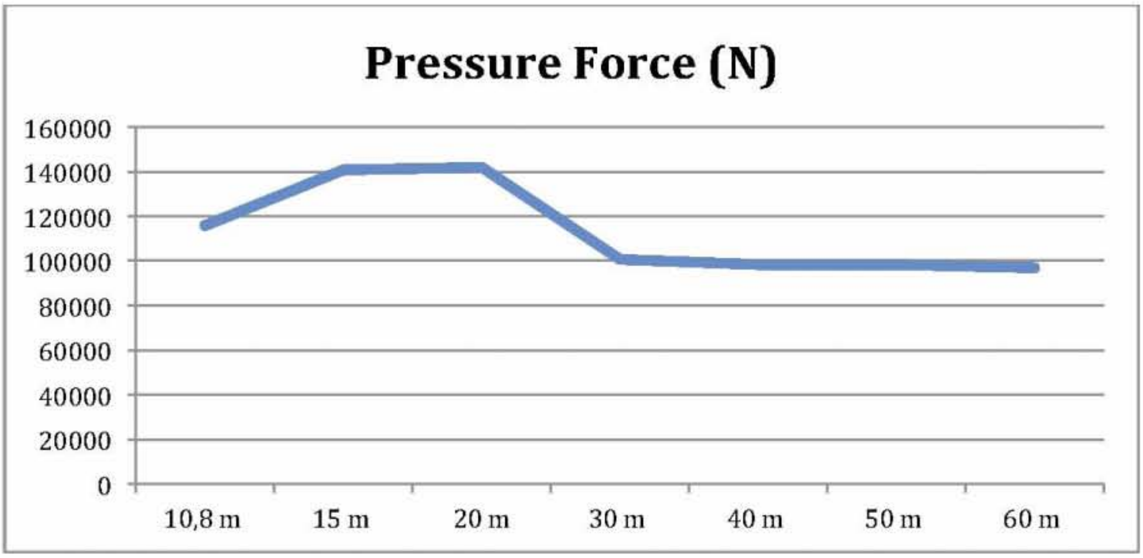
Table 12 - Force Reports with different depth

Deep Study : Bottom -> 10,8m -> wall	**Forces Report**			**Pressure Coefficients**			
Component	**Pressure Force (N)**	**Viscose Force (N)**	**Total Force (N)**	**Normal Torque**	**Viscoso Torque**	**Total Torque**	
Bottom	0	-707,285	-707,285	0	-1154,751	-1154,751	
Hull	-6.186,803	-1.554,929	-7741,732	-10100,903	-2538,659	-12639,562	
Blades	122019,3	-412,29	121607,01	199215,19	-673,127	198542,06	**direcc X**
Cover Prop	0	-208,552	-208,552	0	-340,493	-340,493	
Total	115832,5	-2883,056	112949,45	189114,29	-4707,029	184407,26	
Component	**Pressure Force (N)**	**Viscose Force (N)**	**Total Force (N)**	**Normal Torque**	**Viscoso Torque**	**Total Torque**	
Bottom	0	98,043	98,043	0	160,069	160,069	
Hull	2.487,7860	356,616	2844,402	4061,691	582,23	4643,921	
Blades	-66476,727	-262,029	-66738,756	-108533,43	-427,803	-108961,23	**direcc Y**
Cover Prop	0	-155,789	-155,789	0	-254,349	254,349	
Total	-63988,941	36,84	-63952,101	-104471,74	60,147	-104411,59	
Component	**Pressure Force (N)**	**Viscose Force (N)**	**Total Force (N)**	**Normal Torque**	**Viscoso Torque**	**Total Torque**	
Bottom	52455,18	0	52455,18	85641,11	0	85641,11	
Hull	76.243,547	174,074	76417,621	124479,26	284,202	124763,46	
Blades	5789,902	138,747	5928,649	9452,902	226,526	9679,428	**direcc Z**
Cover Prop	-93807,359	0	-93807,349	-153154,87	0	-153154,87	
Total	40681,27	312,821	40994,091	66418,399	510,728	66929,128	
Deep Study : Bottom -> 15 m -> wall	**Forces Report**			**Pressure Coefficients**			
Component	**Pressure Force (N)**	**Viscose Force (N)**	**Total Force (N)**	**Normal Torque**	**Viscoso Torque**	**Total Torque**	
Bottom	0	-878,511	-878,511	0	1434,304	-1434,304	
Hull	1.583,177	-1.409,811	173,366	2584,779	2301,732	283,047	
Blades	139235,03	-142,513	139092,52	227322,5	-232,675	227089,83	**direcc X**
Cover Prop	0	-20,898	-20,898	0	-34,119	-34,119	
Total	140818,21	-2451,734	138366,47	229907,28	-4002,831	225904,45	
Component	**Pressure Force (N)**	**Viscose Force (N)**	**Total Force (N)**	**Normal Torque**	**Viscoso Torque**	**Total Torque**	
Bottom	0	-98,375	-98,375	-98,375	0	-160,612	
Hull	-9.907,7670	48,254	48,254	-9859,512	-16175,945	-16097,163	
Blades	-90528,164	-199,831	-199,831	-90727,995	-147801,08	-148127,34	**direcc Y**
Cover Prop	0	-58,196	-58,196	-58,196	0	-95,014	
Total	-100435,93	-308,148	-308,148	-100,744	-163977,03	-164480,13	
Component	**Pressure Force (N)**	**Viscose Force (N)**	**Total Force (N)**	**Normal Torque**	**Viscoso Torque**	**Total Torque**	
Bottom	76653,266	0	76653,266	125148,19	0	125148,19	
Hull	11.199,219	664,565	11863,784	18284,439	1085,004	19369,443	
Blades	4131,101	192,905	4324,006	6744,655	314,947	7059,601	**direcc Z**
Cover Prop	-42779,176	0	-42779,176	-69843,552	0	-69843,552	
Total	49204,41	857,47	50061,879	80333,73	1399,951	81733,681	
Deep Study : Bottom -> 20m -> wall	**Forces Report**			**Pressure Coefficients**			
Component	**Pressure Force (N)**	**Viscose Force (N)**	**Total Force (N)**	**Normal Torque**	**Viscoso Torque**	**Total Torque**	
Bottom	0	-626,998	-626,998	0	-1023,671	-1023,671	
Hull	1.441,956	-1.406,080	35,877	2354,2164	-2295,641	58,575	
Blades	140450,08	-140,967	140309,11	229306,25	-230,15	229076,1	**direcc X**
Cover Prop	0	-23,707	-23,707	0	-38,706	-38,706	
Total	141892,04	-2197,753	139694,28	231660,47	-3588,168	228072,3	
Component	**Pressure Force (N)**	**Viscose Force (N)**	**Total Force (N)**	**Normal Torque**	**Viscoso Torque**	**Total Torque**	
Bottom	0	-74,011	-74,011	0	-120,835	-120,835	
Hull	-10.622,3200	34,255	-10588,065	-17342,564	55,926	-17286,637	
Blades	-92117,578	-204,862	-92322,44	-150396,05	-334,469	-150730,51	**direcc Y**
Cover Prop	0	-56,695	-56,695	0	-92,563	-92,5625	
Total	-102739,9	-301,313	-103041,21	-167738,61	-491,939	-168230,55	
Component	**Pressure Force (N)**	**Viscose Force (N)**	**Total Force (N)**	**Normal Torque**	**Viscoso Torque**	**Total Torque**	
Bottom	59917,977	0	59917,977	97825,268	0	97825,268	
Hull	8.870,475	655,855	9526,329	14482,408	1070,784	15553,191	
Blades	4190,356	196,798	4387,1547	6841,398	321,303	7162,702	**direcc Z**
Cover Prop	-41436,328	0	-41436,328	-67651,148	0	-67651,148	
Total	31542,479	852,653	32395,133	51497,926	1392,087	52890,013	
Deep Study : Bottom -> 30m -> wall	**Forces Report**			**Pressure Coefficients**			
Component	**Pressure Force (N)**	**Viscose Force (N)**	**Total Force (N)**	**Normal Torque**	**Viscoso Torque**	**Total Torque**	
Bottom	0	-271,227	-271,227	0	-442,819	-442,819	
Hull	-5.739,540	-1.289,273	-7028,813	-9370,678	-2104,935	-11475,612	
Blades	106292,73	-243,932	106048,8	173539,16	-398,256	173140,9	**direcc X**
Cover Prop	0	-51,201	-51,201	0	-83,593	-83,593	
Total	100553,19	-1855,632	-98697,562	164168,48	-3029,604	161138,88	
Component	**Pressure Force (N)**	**Viscose Force (N)**	**Total Force (N)**	**Normal Torque**	**Viscoso Torque**	**Total Torque**	
Bottom	0	12,203	12,203	0	19,923	19,923	
Hull	3.736,5480	117,956	3854,504	6100,486	192,581	6293,067	
Blades	-53286,781	-263,537	-53550,318	-86998,827	-430,265	-87429,091	**direcc Y**
Cover Prop	0	-130,202	-130,202	0	-212,574	-212,574	
Total	-49550,233	-263,58	-49813,813	-80898,34	-430,335	-81328,675	
Component	**Pressure Force (N)**	**Viscose Force (N)**	**Total Force (N)**	**Normal Torque**	**Viscoso Torque**	**Total Torque**	
Bottom	16476,465	0	16476,465	26900,351	0	26900,351	
Hull	64.145,250	142,594	64287,844	104726,94	232,806	104959,75	
Blades	4761,356	150,563	4911,919	7773,642	245,817	8019,459	**direcc Z**
Cover Prop	-98596,703	0	-98596,703	-160974,21	0	-160974,21	
Total	-13213,632	293,157	-12920,475	-21573,277	478,623	-21094,654	

Table 13 - Force Reports with different depth

Deep Study : Bottom -> 40m -> wall	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscose Force (N)	Total Force (N)	Normal Torque	Viscoso Torque	Total Torque	
Bottom	0	-129,063	-129,063	0	210,714	-210,714	
Hull	-3.578,149	-1.190,929	-4769,078	-5841,876	-1944,374	-7786,25	
Blades	101972,38	-165,378	101807	166485,51	-270,004	166215,51	direcc X
Cover Prop	0	-1,325	-1,325	0	-2,164	-2,164	
Total	98394,226	-1486,695	96907,531	160643,63	-2427,256	158216,38	
Component	Pressure Force (N)	Viscose Force (N)	Total Force (N)	Normal Torque	Viscoso Torque	Total Torque	
Bottom	0	8,522	8,522	0	13,914	13,914	
Hull	4.243,4720	163,832	4407,304	6928,117	267,482	7195,599	
Blades	-50284,879	-244,697	-50529,576	-82097,761	-399,506	-82497,267	direcc Y
Cover Prop	0	-73,522	-73,522	0	-120,036	-120,036	
Total	-46041,407	-145,865	-46187,272	-75169,644	-238,146	-75407,791	
Component	Pressure Force (N)	Viscose Force (N)	Total Force (N)	Normal Torque	Viscoso Torque	Total Torque	
Bottom	11622,061	0	11633,061	18992,752	0	18992,752	
Hull	49.475,391	240,144	49715,534	80776,148	392,071	81168,219	
Blades	3940,228	154,566	4094,793	6433,025	252,352	6685,377	direcc Z
Cover Prop	-84487,516	0	-84487,516	-137938,8	0	-137938,8	
Total	-19438,837	394,709	-19044,128	-31736,877	644,423	.31092,453	
Deep Study : Bottom -> 50m -> wall	**Forces Report**			**Pressure Coefficients**			
Component	Pressure Force (N)	Viscose Force (N)	Total Force (N)	Normal Torque	Viscoso Torque	Total Torque	
Bottom	0	-62,336	-62,336	0	-101,773	-101,773	
Hull	-4.348,569	-1.592,787	-5941,356	-10365,011	-2600,468	-12965,48	
Blades	102814,3	-193,405	102620,895	176023,34	-315,763	175707,58	direcc X
Cover Prop	0	-20,796	-20,796	0	-33,953	-33,953	
Total	98465,731	-1869,323	96596,404	165658,33	-3051,956	162606,37	
Component	Pressure Force (N)	Viscose Force (N)	Total Force (N)	Normal Torque	Viscoso Torque	Total Torque	
Bottom	0	-4,227	-4,227	0	-6,902	-6,902	
Hull	2.990,1650	40,783	3030,948	4881,902	66,585	4948,486	
Blades	-54533,863	-266,054	-54799,917	-89034,879	-434.374	-89469,252	direcc Y
Cover Prop	0	-109,812	-109,812	0	-179,285	-179,285	
Total	-51543,698	-339,309	-51883,008	-84152,977	-553,975	-84706,952	
Component	Pressure Force (N)	Viscose Force (N)	Total Force (N)	Normal Torque	Viscoso Torque	Total Torque	
Bottom	8698,711	0	8698,711	14201,977	0	14201,977	
Hull	63.335,445	183,434	63518,879	103404,81	299,484	103704,29	
Blades	4305,845	130,652	4436,497	7029,951	213,309	7243,259	direcc Z
Cover Prop	-92050,617	0	-92050,617	-150286,72	0	-150286,72	
Total	-15710,616	314,086	-15396,531	-25649,986	512,793	-25137,193	
Deep Study : Bottom -> 60m -> wall	**Forces Report**			**Pressure Coefficients**			
Component	Pressure Force (N)	Viscose Force (N)	Total Force (N)	Normal Torque	Viscoso Torque	Total Torque	
Bottom	0	-40,654	-40,654	0	-66,374	-66,374	
Hull	-4.133,984	-1.256,663	-5390,647	-6749,361	-2051,695	-8801,056	
Blades	101058,77	-189,238	100869,53	164993,9	-308,959	164684,94	direcc X
Cover Prop	0	3,234	3,234	0	5,279	5,279	
Total	96924,782	-1483,321	95441,46	158244,54	-2421,749	155822,79	
Component	Pressure Force (N)	Viscose Force (N)	Total Force (N)	Normal Torque	Viscoso Torque	Total Torque	
Bottom	0	-0,321	-0,321	0	-0,524	-0,524	
Hull	4.396,9760	195,752	4592,728	7178,736	319,596	7498,331	
Blades	-50998,926	-245,729	-51244,655	-83263,552	-401	-83664,742	direcc Y
Cover Prop	0	-70,218	-70,218	0	-114,642	-114,642	
Total	-46601,95	-120,515	-46722,465	-76084,817	-196,759	-76281,576	
Component	Pressure Force (N)	Viscose Force (N)	Total Force (N)	Normal Torque	Viscoso Torque	Total Torque	
Bottom	5878,516	0	5878,516	9597,577	0	9597,577	
Hull	46.611,578	202,149	46813,727	76100,536	330,039	76430,575	
Blades	3886,773	157,894	4044,667	6345,753	257,786	6603,539	direcc Z
Cover Prop	-83958,273	0	-83958,273	-137074,73	0	-137074,73	
Total	-27581,406	360,043	-27221,363	-45030,867	587,825	-44443,042	

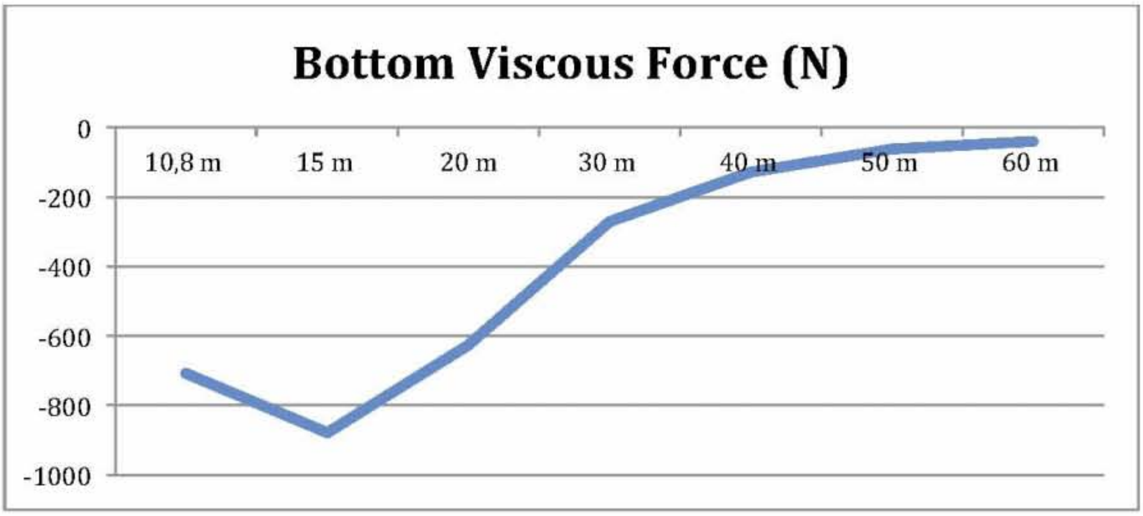
To be able to understand and study the data there is a graphic following, which shows the change of the thrust (Pressure Force (N)) depending on the depth:



Graphic 7 - Pressure Force of simulations with different depth

A curious effect that we can see in the graphic is that the supposed proper thrust (the same one as in the simulation at -8 rad/s in the chapter number 8, the called model simulation) it is achieved between 10,8-15 and 20-30 meters of depth. But of course there is an influence from the bottom producing turbulences and currents when the water with motion approaches it that interfere the thrust results. So, even due between 10,8 and 15 meters of depth the same pressure force as in the simulation model is found, there is no real stability of thrust amount till there is 30 meters of depth.

I am also going to take a look at the viscous forces in the bottom, which make a negative influence to the total force results, i.e. the thrust, to complete the depth study. The following graphic shows the viscous force from the bottom of the sea that influences the thrust at different seabed depths:



Graphic 8 - Bottom Viscous Force with different depth

It is noticeable, that with little depth there is a big influence from the viscous forces to the thrust. It is around 30 and 40 meters depth, where the viscous force values start being acceptable (closer to 0).

Therefore the simulations show that 20 meters of depth, what the Classification societies recommend for the bollard pull test performance, it is not enough for this kind of propeller. It is not till 30 meters of depth that the seabed stops making such a big influence on the thrust results. So, from this first study I am going to conclude, that 20 meters of depth in the bollard pull test zone for a tug with a VSP propeller is not enough. I would suggest that the Classification Societies could consider requiring 30 meters of depth in the bollard pull test zone.

9.2. Free distance on the sides of the boat

In this second part of the ninth chapter, I am going to study the influence of the obstacles (port walls, rocks...) on the sides of the boat. To achieve simulations that matches with those situations, the boundary conditions employed for the study are *pressure-outlet* to all the boundaries except the sides of the vessel, were I assigned the condition *wall*.

The Classification Societies require 300 meters without obstacles in a circle around the boat. To check if this rule is also applicable to the Voith Schneider propeller, I did some simulation with different distances of the obstacles on the side of the boat. I start with a first simulation studying the effects of a wall at 20 meters from the centre of the boat and afterwards 8 simulations more each one with the wall 10 meters further as the last one.

In the following tables there are the forces reports of them:

Table 14 - Force Reports of simulations with different free distances on the si

Laterals study, Y=20m, 450x50x40	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	614,074	-1.481,284	-840.209	1046,651	-2418,422	1371,771	
Laterals wall	0	-1783,979	-1783,977	0	-2912,619	-2912,619	
Blades	142376,03	-143,966	142232,06	232450,66	-235,047	232215,62	direcc X
Cover Prop	0	-21,127	-21,127	0	-34,493	-34,493	
Total	142990,10	-3.430,356	-699.782	233497,311	-5600,581	230640,279	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-5.502,561	18,000	-5484,561	-8983,773	29,388	-8954,385	
Laterals wall	-5170,124	0	-5170,124	-8441,019	0	-8441,019	
Blades	-93306,156	-190,517	-93496,674	-152336,58	-311,048	-152647,63	direcc Y
Cover Prop	0	-64,849	-64,849	0	-105,876	-105,876	
Total	-103.978,841	-237,366	-104216,208	-169761,372	-387,536	-170148,91	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	11.780,537	596,507	12377,044	19233,53	973,889	20207,419	
Laterals wall	0	-758,72	-758,721	0	-1238,728	-1238,728	
Blades	4229,017	194,847	4423,864	6904,517	318,118	7222,635	direcc Z
Cover Prop	-40587,207	0	-40587,207	-66264,828	0	-66264,828	
Total	-24.577,653	32,634	-24545,02	-40126,781	53,279	-40073,502	

Laterals study, Y=30m, 450x50x60	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-3.862,554	-1.503,905	-5.366	-6306,209	-2455,355	-8761,564	
Laterals wall	0	-1494,662	-1494,662	0	-2440,264	-2440,264	
Blades	109197,83	-208,468	108989,36	178282,17	-340,356	177941,81	direcc X
Cover Prop	0	3,227	3,227	0	5,268	5,268	
Total	105335,28	-3.203,808	102.131	171975,961	-5230,707	166745,25	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	2.112,149	186,756	2298,906	3448,407	304,908	3753,315	
Laterals wall	9051,053	0	9051,053	14777,229	0	14777,229	
Blades	-54304,012	-257,184	-54561,196	-88659,611	-419,893	-89079,503	direcc Y
Cover Prop	0	-61,656	-61,656	0	-100,663	-100,663	
Total	-43.140,810	-132,084	-43272,893	-70433,975	-215,648	-70649,622	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	40.929,711	218,269	41147,981	66824,018	356,359	67180,377	
Laterals wall	0	-450,27	-450,27	0	-735,135	-735,135	
Blades	4334,172	139,665	4473,8368	7076,199	228,024	7304,223	direcc Z
Cover Prop	-71076,336	0	-71076,336	-116043	0	-116043	
Total	-25.812,453	-92,336	-25904,7882	-42142,783	-150,752	-42293,535	

Laterals study, Y=40m, 450x50x80	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-3.985,989	-1.457,893	-5.444	-6507,737	-2380,234	-8887,971	
Laterals wall	0	-1236,417	-1236,417	0	-2018,64	-2018,64	
Blades	106327,83	-199,487	106128,33	173596,44	-325,692	173270,75	direcc X
Cover Prop	0	-0,696	-0,6959	0	-1,136	-1,136	
Total	102341,84	-2.894,493	99.447	167088,703	-4725,702	162363,003	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	2.805,613	235,542	3041,155	4580,593	384,556	4965,151	
Laterals wall	9231,969	0	9231,969	15072,602	0	15072,602	
Blades	-53738,891	-257,293	-53996,183	-87736,964	-420,069	-88157,034	direcc Y
Cover Prop	0	-67,694	-67,694	0	-110,521	-110,521	
Total	-41.701,309	-89,445	-41790,753	-68083,769	-146,034	-68229,802	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	44.912,387	235,670	45148,057	73326,346	384,767	73711,113	
Laterals wall	0	-436,346	-436,346	0	-712,403	-712,403	
Blades	4185,276	152,798	4338,074	6833,104	249,466	7082,569	direcc Z
Cover Prop	-77065,258	0	-77065,358	-125820,83	0	-125820,83	
Total	-27.967,595	-47,878	-28015,573	-45661,38	-78,17	-45739,551	

Laterals study, Y=50m, 450x50x100	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	1.806,999	-1.524,051	282,948	2950,203	-2488,246	461,956	
Laterals wall	0	-821,855	-821,855	0	-1341,805	-1341,805	
Blades	141993,17	-149,859	141843,31	231825,59	-244,668	231580,92	direcc X
Cover Prop	0	-27,289	-27,189	0	-44,553	-44,553	
Total	143800,17	-2.523,054	141.277	234775,79	-4119,272	230656,52	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-11.065,467	23,536	-11041,931	-18066,068	38,426	-18027,643	
Laterals wall	-984,397	0	-984,397	-1607,179	0	-1607,179	
Blades	-93155,898	-207,207	-93363,105	-152091,26	-338,297	-152429,56	direcc Y
Cover Prop	0	-60,343	-60,343	0	-98,519	-98,519	
Total	-105.205,760	-244,014	-105499,78	-171764,51	-398,39	-172162,901	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	7.087,676	690,956	7778,632	11571,716	1128,092	12699,807	
Laterals wall	0	-406,373	-406,373	0	-663,466	-663,466	
Blades	4169,194	204,364	4373,559	6806,848	333,656	7140,504	direcc Z
Cover Prop	-39404,094	0	-39404,094	-64333,214	0	-64333,214	
Total	-28.147,224	488,947	-27658,276	-45954,65	798,282	-45156,369	

Table 15 - Force Reports of simulations with different free distances on the sides

Laterals study, Y=60m, 450x50x120	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	1.548,927	-1.447,239	101,688	2528,86	-2362,839	166,021	
Laterals wall	0	-665,613	-665,613	0	-1086,715	-1086,715	
Blades	142117,63	-144,102	141973,52	232028,78	-235,268	231793,51	direcc X
Cover Prop	0	-27,37	-27,37	0	-44,686	-44,686	
Total	143666,55	-2.284,324	141.382,23	234557,64	-3729,509	230828,13	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-11.039,174	13,435	-11025,739	-18023,141	21,934	-18001,207	
Laterals wall	-1224,726	0	-1224,726	-1999,552	0	-1999,552	
Blades	-93254,984	-203,415	-93458,399	-152253,04	-332,106	-152585,14	direcc Y
Cover Prop	0	-56,003	-56,003	0	-91,433	-19,433	
Total	-105.518,880	-245,983	-105764,87	-172275,73	-401,605	-172677,33	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	7.832,585	687,309	8519,895	12787,894	1122,138	13910,032	
Laterals wall	0	-345,228	-345,228	0	-563,638	-563,638	
Blades	4143,202	201,342	4344,543	6764,411	328,721	7093,132	direcc Z
Cover Prop	-40254,484	0	-40254,484	-65721,607	0	-65721,607	
Total	-28.278,698	543,423	-27735,275	-46169,302	887,221	-45282,082	

Laterals study, Y=70m, 450x50x140	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	1.229,746	-1.542,863	-313,117	2007,748	-2518,959	-511,211	
Laterals wall	0	-545,195	-545,195	0	-890,114	-890,114	
Blades	142030,03	-141,009	141889,02	231885,77	-230,218	231655,55	direcc X
Cover Prop	0	-24,933	-24,933	0	-40,707	-40,707	
Total	143259,78	-2.253,999	141.005,78	233893,51	-3679,999	230213,52	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-10.128,930	62,246	-10066,684	-16537,028	101,625	-16435,403	
Laterals wall	-1447,913	0	-1447,913	-2363,94	0	-2363,94	
Blades	-92140,516	-202,457	-92342,972	-150433,49	-330,542	-150764,04	direcc Y
Cover Prop	0	-60,005	-60,005	0	-97,967	-97,967	
Total	-103.717,360	-200,216	-103917,57	-169334,46	-326,883	-169661,35	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	8.525,171	689,242	9214,413	13918,646	1125,293	15043,939	
Laterals wall	0	-281,746	-281,746	0	-459,993	-459,993	
Blades	4149,907	202,765	4352,672	6775,358	331,044	7106,402	direcc Z
Cover Prop	-40458,816	0	-40458,816	-66055,21	0	-66055,21	
Total	-27.783,739	610,261	-27173,478	-45361,206	996,344	-44364,862	

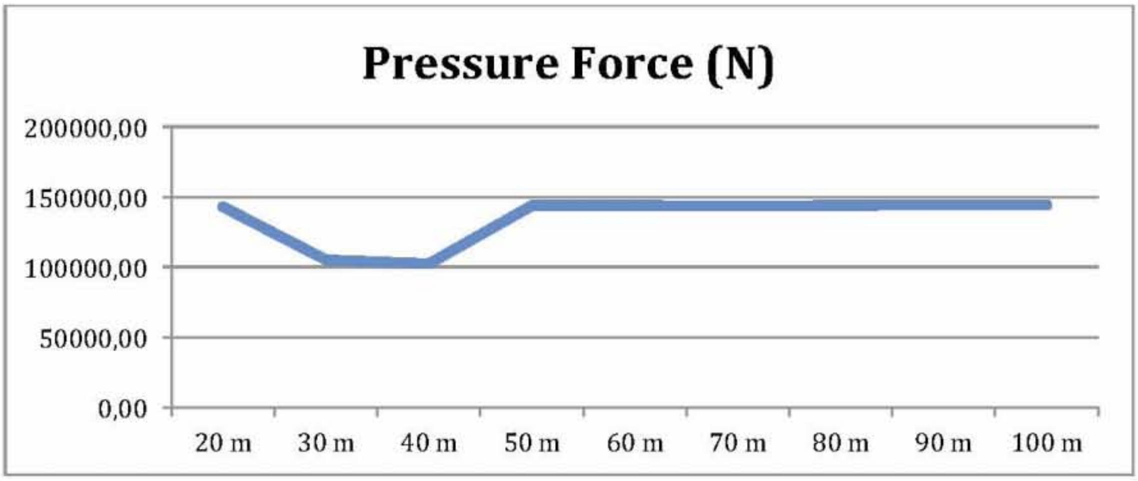
Laterals study, Y=80m, 450x50x160	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	1.598,965	-1.556,272	42,694	2610,556	-2540,852	69,704	
Laterals wall	0	-414,082	-414,082	0	-676,052	-676,052	
Blades	142172,66	-143,38	142029,28	232118,62	-234,089	231884,53	direcc X
Cover Prop	0	-29,067	-29,067	0	-47,456	-47,456	
Total	143771,62	-2.142,803	141.628,82	234729,18	-3498,449	231230,73	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-10.219,893	45,245	-10174,648	-16685,539	73,869	-16611,67	
Laterals wall	-1807,245	0	-1807,245	-2950,604	0	-2950,604	
Blades	-92218,922	-202,479	-92421,402	-150561,51	-330,579	-150892,08	direcc Y
Cover Prop	0	-58,489	-58,489	0	-95,493	-95,493	
Total	-104.246,060	-215,725	-104461,78	-170197,65	-352,203	-170549,85	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	8.396,271	685,641	9081,912	13708,197	1119,414	14827,611	
Laterals wall	0	-215,233	-215,233	0	-351,4	-351,4	
Blades	4164,115	204,318	4368,433	6798,556	333,58	7132,136	direcc Z
Cover Prop	-40731,184	0	-40731,184	-66499,892	0	-66499,892	
Total	-28.170,798	674,726	-27496,071	-45993,139	1101,594	-44891,545	

Laterals study, Y=90m, 450x50x180	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	1.685,254	-1.462,576	222,678	2751,435	-2387,879	363,556	
Laterals wall	0	-322,103	-322,103	0	-525,883	-525,883	
Blades	142438,25	-144,487	142293,76	232552,24	-235,897	232316,35	direcc X
Cover Prop	0	-27,974	-27,974	0	-45,671	-45,671	
Total	144123,50	-1.957,139	142.166,36	235303,68	-3195,33	232108,35	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-11.101,392	2,201	-11099,19	-18124,721	3,594	-18121,127	
Laterals wall	-1685,364	0	-1685,364	-2751,614	0	-2751,614	
Blades	-93896,898	-201,426	-94098,324	-153301,06	-328,858	-153629,92	direcc Y
Cover Prop	0	-57,773	-57,773	0	-94,323	-94,323	
Total	-106.683,650	-256,997	-106940,65	-174177,39	-419,587	-174596,98	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	8.266,163	676,931	8943,094	13495,776	1105,193	14600,97	
Laterals wall	0	-170,436	-170,436	0	-278,263	-278,263	
Blades	4169,752	200,892	4370,644	6807,758	327,988	7135,746	direcc Z
Cover Prop	-40231,656	0	-40231,656	-65684,337	0	-65684,337	
Total	-27.795,741	707,387	-27088,354	-45380,802	1154,918	-44225,884	

Table 16 - Force Report of the simulation with 100 meters free on the sides

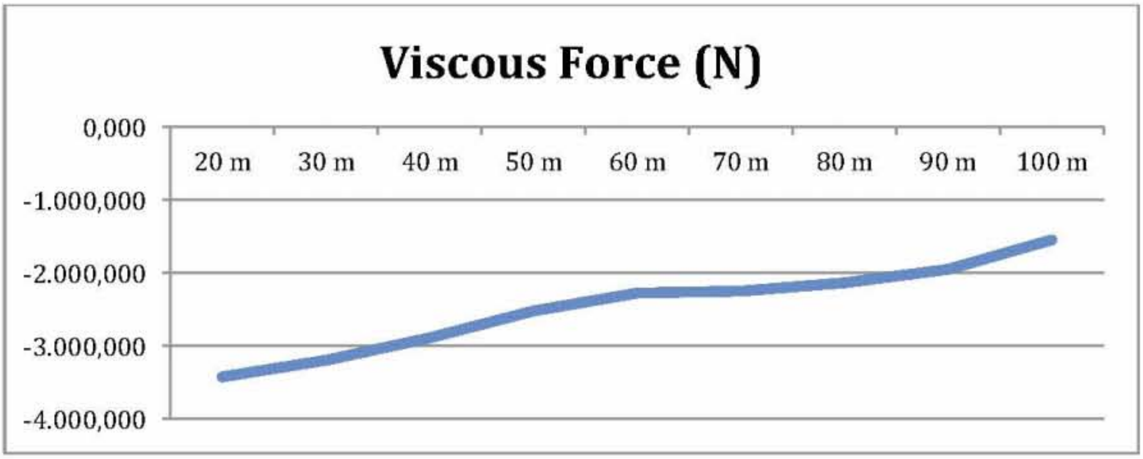
Laterals study, Y=100m, 450x50x18	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-5.777,687	-1.289,309	-7.066,996	-9432,958	-2104,994	-11537,952	
Laterals wall	0	-15,071	-15,071	0	-24,605	-24,605	
Blades	149990,47	-215,4	149775,07	171264,44	-351,674	170912,77	direcc X
Cover Prop	0	-33,475	-33,475	0	-54,653	-54,653	
Total	144212,78	-1.553,254	142.659,53	161831,48	-2535,926	159295,55	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	3.402,170	115,866	3518,037	5554,564	189,169	5743,733	
Laterals wall	-878,051	0	-878,051	-1422,552	0	-1433,552	
Blades	-53258,766	-258,574	-53517,34	-86953,087	-422,163	-87375,249	direcc Y
Cover Prop	0	-112,393	-112,393	0	-183,499	-183,499	
Total	-50.734,646	-255,101	-50.989,747	-82832,075	-416,492	-83248,567	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	70.570,930	137,188	70708,118	115217,84	223,981	115441,82	
Laterals wall	0	-4,048	-4,048	0	-6,609	-6,609	
Blades	4571,169	135,795	4706,964	7463,133	221,705	7684,838	direcc Z
Cover Prop	-99127,305	0	-99127,305	-161840,5	0	-161840,5	
Total	-23.985,206	268,934	-23716,272	-39159,52	439,077	-38720,443	

To be able to look at those force reports focusing in the study objective, I created the following graphic of all the simulations x-axis Pressure Force:



Graphic 9 - Pressure Force of simulations with different distances on the sides

In the first simulation (a wall at 20 meters) gives a high thrust result compared with the next simulations (wall at 30 and 40 meters). But from the wall at 50 meters distances on, the thrust results get stability and stay close to 150.000 N. It is logical to think that even that in the simulation with the wall at 20 meters of distance the results may seem right, it is probably caused due to the turbulences produced in such a small place to performance the bollard pull test. To confirm this suppose, I will take a quick look at the viscous forces produced due to the wall in the next graphic:



Graphic 10 - Viscous Force of simulations with different distances on the sides

As expected, in the first simulation is where the negative effect of the wall is bigger; therefore, that the thrust results are similar to the right ones is due to the turbulences effect.

Taking all that into account, it is clearly seen that between 50 and 60 the effect of the wall keeps decreasing and the thrust results stay almost equal for the next simulations.

The Classification Societies require minimum 100 meters free of walls, which of course would be better because the viscous force effect it is even smaller, but I suggest that with 50-60 meters it is enough to perform the bollard pull test of a tug with a Voith Schneider Propeller.

9.3. Distance between the harbour and the stern

The last measure that has a requirement from the Classification Societies in the bollard pull test is the distance between the port (or place where the bollard is attached) and the stern on the boat. It is the most important distance to determinate because it is the direction of the powerful thrust, (in the study represented in the X-axis direction). Therefore, the Classification Societies require three times the amount of meters for that concrete distance as in the rest of the boat directions, i.e. 300 meter.

For these situations, the boundary conditions employed for the study are *pressure-outlet* to all the boundaries except the wall aft of the tug, were I assigned the condition *wall*. In the following tables there are the report forces from Ansys of six different simulations, each one with a different distance between bollard and stern (40, 110, 180, 260, 320 and 380 meters):

Table 17 – Force Report of simulations with different distance between the bollard and stern of the tug

Lateral walls study : Stern -> 40 m -> wall	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-4.947,803	-1.525,997	-6473,799	-8078,045	-2491,423	-10569,468	
Blades	104695,17	-161,346	104533,83	170930,89	-263,423	170667,47	direcc X
Stern	-30943,213	0	-30943,213	-50519,513	0	-50519,531	
Cover Prop	0	7,871	7,871	0	12,851	12,851	
Total	68804,156	-1679,472	67124,684	112333,32	-2741,995	109591,32	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	3.298,867	63,738	3362,594	5385,905	104,045	5489,95	
Blades	-48650,641	-264,933	-48915,573	-79429,617	-432,543	-79862,161	
Stern	0	123,333	123,333	0	201,359	201,359	direcc Y
Cover Prop	0	-72,024	-72,024	0	-117,59	-117,59	
Total	-45351,774	-149,896	-149,896	-74043,713	-244,728	-74288,441	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	41.901,871	267,177	42169,049	68411,218	436,208	68847,427	
Blades	4299,632	108,812	4408,444	7019,808	177,652	7197,459	
Stern	0	-1103,829	-1103,829	0	-1802,169	-1802,169	direcc Z
Cover Prop	-69226,773	0	-69226,773	-113023,3	0	-113023,3	
Total	-23025,27	-727,839	-23753,109	-37592,278	-1188,309	-38780,587	
Lateral walls study : Stern -> 110 m -> wall	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-7.151,426	-1.700,096	-8851,522	-11675,797	-2775,667	-14451,464	
Blades	124184,99	-418,721	123766,27	202751,01	-683,626	202067,38	direcc X
Stern	-29824,525	0	-29824,525	-48693,103	0	-48693,103	
Cover Prop	0	-201,153	-201,153	0	-328,413	-328,413	
Total	87209,041	-2319,969	84889,071	142382,11	-3787,706	138594,4	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	7.891,394	191,840	8083,234	12883,908	313,208	13197,116	
Blades	-71082,984	-286,199	-71369,184	-116053,85	-467,264	-116521,12	
Stern	0	77,849	77,849	0	127,101	127,101	direcc Y
Cover Prop	0	-188,102	-188,102	0	-307,105	-307,105	
Total	-63191,591	-204,611	-63396,202	-103169,94	-334,059	-10354	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	71.943,203	157,082	72100,285	117458,29	256,459	117714,75	
Blades	5916,27	183,002	6099,272	9659,216	298,779	9957,995	
Stern	0	-658,157	-658,157	0	-1074,542	-1074,542	direcc Z
Cover Prop	-94931,406	0	-94931,406	-154990,05	0	-154990,05	
Total	-17071,933	-318,073	-17390,007	-27872,544	-519,304	-28391,847	
Lateral walls study : Stern -> 180 m -> wall	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-6.449,062	-1.597,206	-8046,268	-10529,08	-2607,683	-13136,763	
Blades	113108,61	-231,408	112877,2	184667,12	-377,809	184289,31	direcc X
Stern	-30998,35	0	-30998,35	-50609,55	0	-50609,55	
Cover Prop	0	-74,785	-74,785	0	-122,098	-122,098	
Total	75661,198	-1903,399	73757,799	123528,49	-3107,591	120420,9	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	3.135,294	89,407	3224,701	5118,848	145,97	5264,819	
Blades	-54324,467	-284,429	-54608,796	-88692,844	-464,373	-89157,218	
Stern	0	-8,048	-8,048	0	-13,139	-13,139	direcc Y
Cover Prop	0	-124,575	-124,574	0	-203,387	-203,387	
Total	-51189,073	-327,644	-51516,717	-83573,996	-534,929	-84108,926	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	66.515,375	150,279	66665,654	108596,53	245,353	108841,88	
Blades	4795,057	144,254	4939,31	7828,664	235,516	8064,18	
Stern	0	-487,661	-487,661	0	-796,181	-796,181	direcc Z
Cover Prop	-92655,188	0	-92655,188	-151273,78	0	-151273,78	
Total	-21344,756	-193,129	-21537,885	-34848,581	-315,312	-35163,893	

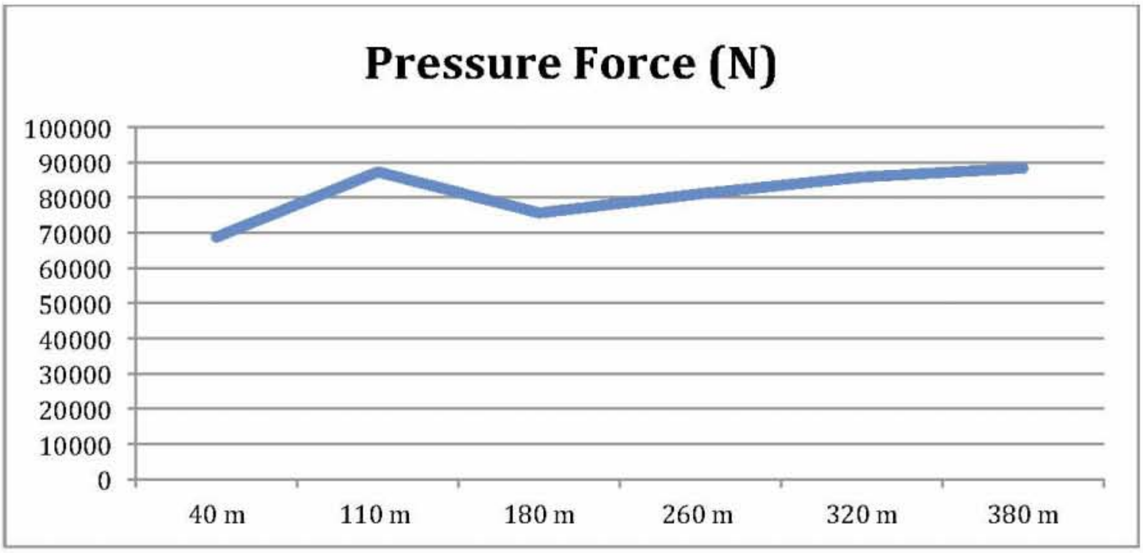
Table 18 - Force Report of simulations with different distance between the bollard and stern of the tug

Lateral walls study : Stern -> 260 m -> wall	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-5.079,623	-1.435,068	-6514,69	-8293,261	-2342,968	-10636,229	
Blades	109361,59	-154,338	109207,26	178549,54	-251,98	178297,56	direcc X
Stern	-23182,549	0	-23182,549	-37849,059	0	-37849,059	
Cover Prop	0	10,983	10,983	0	17,931	17,931	
Total	81099,422	-1578,423	79520,999	132407,22	-2577,017	129830,2	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	4.370,338	70,067	4440,405	7135,246	114,395	7249,641	
Blades	-52590,668	-261,617	-52852,285	-85862,315	-427,129	-86289,445	
Stern	0	-2,614	-2,614	0	-4,268	-4,268	direcc Y
Cover Prop	0	-87,709	-87,709	0	-143,199	-143,199	
Total	-48220,33	-281,874	-48502,204	-78727,07	-460,202	-79187,271	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	45.846,641	229,451	46076,091	74851,658	374,614	75226,272	
Blades	4992,749	88,538	5081,287	8151,427	144,551	8295,978	
Stern	0	-196,876	-196,876	0	-321,431	-321,431	direcc Z
Cover Prop	-72026,656	0	-72026,656	-117594,54	0	-117594,54	
Total	-21187,267	121,112	-21066,154	-34591,456	197,734	-34393,721	

Lateral walls study : Stern -> 320 m -> wall	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-4.525,955	-1.493,525	-6019,479	-7389,314	-2438,408	-9827,722	
Blades	101419,98	-147,635	101272,34	165583,64	-241,037	165342,6	direcc X
Stern	-11013,197	0	-11013,197	-17980,73	0	-17980,73	
Cover Prop	0	10,623	10,623	0	17,344	17,344	
Total	85880,824	-1630,537	84250,287	140213,59	-2662,101	137551,49	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	5.012,238	115,499	5127,738	8183,246	188,571	8371,817	
Blades	-44170,367	-247,188	-44417,556	-72114,885	-403,573	-72518,458	
Stern	0	-12,407	-12,407	0	-20,256	-20,256	direcc Y
Cover Prop	0	-85,265	-85,265	0	-139,208	-139,208	
Total	-39158,129	-229,361	-39387,489	-63931,639	-374,466	-64306,105	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	42.690,469	225,399	42915,867	69698,724	367,998	70066,722	
Blades	4133,731	104,277	4238,008	6748,949	170,249	6919,197	
Stern	0	-88,152	-88,152	0	-143,921	-143,921	direcc Z
Cover Prop	-70966,281	0	-70966,281	-115863,32	0	-115863,32	
Total	-24142,082	241,524	-23900,557	-39415,643	394,325	-39021,318	

Lateral walls study : Stern -> 380 m -> wall	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	X-axis
Hull	-4.273,227	-1.325,579	-5598,806	-6976,696	-2164,211	-9140,907	
Blades	104995,72	-139,942	104855,78	171421,58	-228,477	171193,1	
Stern	-12279,446	0	-12279,446	-20048,076	0	-20048,076	
Cover Prop	0	-1,832	-1,832	0	-2,991	-2,991	
Total	88443,046	-1467,354	86975,692	144396,81	-2395,679	142001,13	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	Y-axis
Hull	3.608,220	60,800	3669,02	5890,972	99,265	5990,237	
Blades	-50507,051	-246,207	-50753,258	-82460,491	-401,97	-82862,462	
Stern	0	-16,951	-16,951	0	-27,676	-27,676	
Cover Prop	0	-71,408	-71,408	0	-116,584	-116,584	
Total	-46898,831	-273,766	-47172,597	-76569,519	-446,965	-77016,484	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	Z-axis
Hull	47.507,828	214,174	47722,002	77563,801	349,672	77913,473	
Blades	4133,528	141,075	4274,603	6748,618	230,327	6978,944	
Stern	0	-106,817	-106,817	0	-174,395	-174,395	
Cover Prop	-76119,969	0	-76119,969	-124277,5	0	-124277,5	
Total	-24478,612	248,433	-24230,18	-39965,081	405,604	-39559,477	

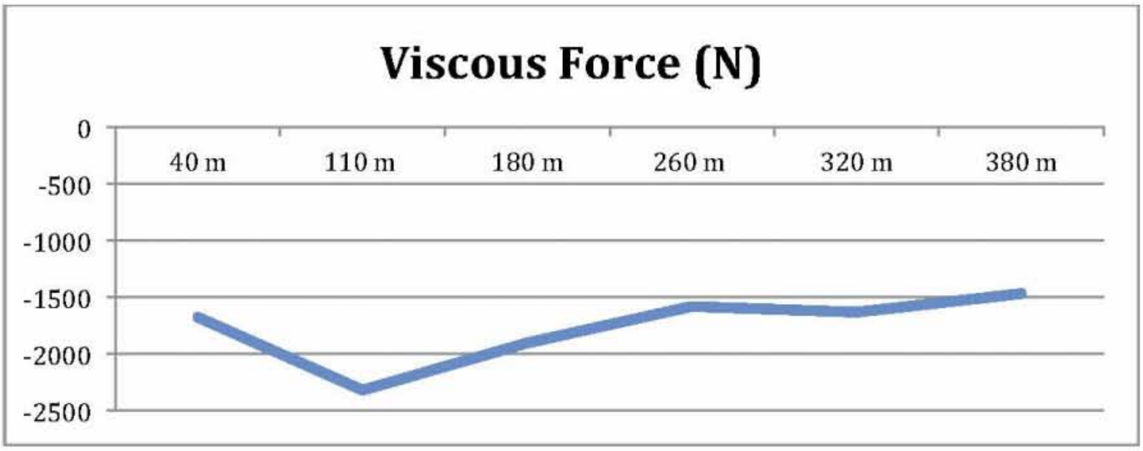
The graphic shows the change of pressure force influenced by the distance between the bollard and the stern of the boat:



Graphic 11 - Pressure Force of simulations with different distance from the bollard to stern of the tug

It is noticeable that this factor highly influences the results. From values around 140.000 N that we were working with in the other simulation studies, here the 90.000 N are not even achieved with 380 meters of distance. The Classification Societies require 300 meters of distance between the bollard and the stern of the boat, but, as we can see in the study it still making a lot of influence on the results.

To see more information that could give answer to that fact, following there is a graphic of the viscous forces produces with the different distances of the “wall” where there is the bollard situated:



Graphic 12 - Viscous Force of simulations with different distance from the bollard to stern of the tug

As in the other simulation studies, with the smallest distances to the wall appears a wear effect. In this case the pressure force it is not high as in the lasts simulations with more distance, but we see this effect in the viscous force. In the first simulation it is almost as small as in the later ones due to the turbulences. But there is no stability on the viscous force results till the 260 meters distance, where is stays around -1500 N.

So, in that case, the biggest distance would be the better for the Voith Schneider Propeller, but, since 260 some stability of the viscous forces is found and the pressure force stills so small even over 300 meters, I think that the 300 meters of minimum distance required from the Classification Societies it is an acceptable distance also for that propeller.

Anyway, as we see in both graphics, the pressure force keeps growing with the distance and the viscous pressure keep decreasing, so at some point the thrust results will be good as in the model simulation (simulation 8 rad/s from chapter 8). Therefore these 300 meters of distance should be taken as a minimum and if there is the chance to perform the bollard pull test with more meters between the bollard and the stern of the boat, do it with 400 or more meters because the test would give much better results.

9.4. % of error of the test when there is current study

Always that it is possible, the test should be conducted without current, if that is not possible, Classification Societies accept maximum one knot of current, i.e. shall be less than 0,5 meters/second. This Societies estimate, for a regular propeller, that a current of 1 knot the bow represents a loss of "Bollard Pull" of about 4%.

In this study, I am going to check this loss of thrust for the VSP case. To do that, I am going to compare the model simulation (8 rad/s from chapter 8, which have is simulated with no current) and another simulation with the same boundary conditions, same rotative speed of the propeller but with 1-knot current. In this case, because I want to achieve a percentage of error when there is current and compare it with the percentage already given by the Classification Societies, I established the same distances that they recommend, i.e. over 100 meters free in all directions, over 300 meters from the bollard to the stern and over 20 meters of depth. So the exact distances I performed for the simulation were 450x30x250m with boundary condition *wall* in all of them.

In the following tables we find the forces report results of those two simulations:

Table 19 - Force Report of the model simulation with and without current

1) test bollard pull operational power and maximum current accepted by the Classification Societies

Current study 1 Knot = 0,5144 m/s, -8 rad/s — Forces Report — Pressure Coefficients

Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom	0	93,585	93,585	0	152,793	152,793	
hull	-3752,742	-1232,233	-4984,975	-6126,925	-2011,809	-8138,734	direc X
blades	103175,73	-178,123	102997,6	168450,17	-290,812	168159,35	
stern	-17287,383	0	-17287,383	-28224,298	0	-28224,298	
cover Prop	0	4,884	4,884	0	7,974	7,974	
Total	82135,602	-1311,886	80823,716	134098,94	-2141,854	131957,09	

Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom	0	-51,555	-51,555	0	-84,172	-84,172	
hull	3546,937	151,011	3697,949	5790,918	246,549	6037,468	direc Y
blades	-48519,836	-257,366	-48777,202	-79216,059	-420,189	-79636,248	
stern	0	-24,603	-24,603	0	-40,168	-40,168	
cover Prop	0	-79,042	-79,042	0	-129,048	-129,048	
Total	-44972,898	-261,554	-45234,453	-73425,14	-427,028	-73852,168	

Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom	-5553,113	0	-5553,113	-9066,307	0	-9066,307	
hull	44494,097	225,864	44720,384	72644,114	368,758	73012,871	direc Z
blades	4344,099	123,597	4.467,696	7092,406	201,792	7294,198	
stern	0	-44,668	-44,668	0	-72,927	-72,927	
cover Prop	-72851,984	0	-72851,984	-118942,02	0	-118942,02	
Total	-29566,479	304,794	-29261,686	-48271,803	497,622	-47774,181	

2) test bollard pull operational power and conditions recommended by the Classification Societies B

Current study, 0 Knot , -8 rad/s — Forces Report — Pressure Coefficients

Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom	0	-162,382	-162,382	0	-265,114	-265,11405	
hull	-8124,115	-1639,055	-9763,169	-13263,861	-2676,009	-15939,869	direc X
blades	134129,3	-358,975	133770,32	218986,61	-586,081	218400,53	
stern	-19481,566	0	-19481,566	-31.806.639	0	-31806,639	
cover Prop	0	-285,3	-285,301	0	-465,798	-465,798	
Total	106523,62	-2445,714	104077,9	173916,11	-3993,002	169923,11	

Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom	0	151,036	151,036	0	246,589	246,589	
hull	4105,651	180,634	4286,286	6703,104	294,913	6998,017	direc Y
blades	-74374,484	-271,859	-74646,344	-121427,73	-443,853	-121871,58	
stern	0	-18,83	-18,83	0	-30,743	-30,743	
cover Prop	0	-180,255	-180,255	0	-294,295	-294,29	
Total	-70268,833	-139,275	-70408,108	-114724,63	-227,388	-114952,01	

Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom	26097,535	0	26097,535	42608,221	0	42608,221	
hull	82735,719	154,6	82890,319	135078,72	252,409	135331,13	direc Z
blades	5967,461	123,327	6090,788	9742,793	201,349	9944,143	
stern	0	-30,616	-30,616	0	-49,986	-49,986	
cover Prop	-89360,148	0	-89360,148	-145894,12	0	-145894,12	
Total	25440,566	247,31058	25687,877	41535,619	403,772	41939,391	

A curious effect that we can notice is, that even with much higher distances than the minimums of the Classification Societies, always when we set the boundary condition wall there is a big lost of thrust. In this case, if we compare the second simulation of these two, the one with no current, it gives a thrust of 106523,62 N next to the model simulation (also 8 rad/s, same distances but all the boundary condition set as *pressure outlet*) that gives 125725,41 N. Clearly showing that the thrust obtained from the bollard pull always will be higher as more distance in all directions without obstacles the boat has.

Going back to this study case, where it does not matter this last objection because both simulations that are compare have same conditions and calculating the error with a quotient this thrust difference does not influence the result of the study.

We obtain just 82135,602 N with 1-knot current next to the 106523,62 N from the simulation with no current and that results in the error that is following calculated:

$$e_{rel} = \frac{T_c - T_m}{T_m} = \frac{82135{,}602 - 106523{,}62}{106523{,}62} = -0{,}2289$$

We obtain 22,89% of error when there is 1-knot current. It is much higher than the error calculated by the Classification Societies, which means that current affects more this kind of propeller than a regular helix. Therefore, for a Voith Schneider Propeller equipped boat I would require no current at all in the bollard pull test or give a much smaller maximum current speed, for example 0,25-knot current as a maximum in the test performance moment.

9.5. Conclusions and suggestions

This chapter let us see that the Classifications Societies Rules for the Bollard Pull Test Zone are not applicable to a Voith Schneider Propeller, those rules were designed for boats with regular propellers and they should be modified for the VSP boats case.

From those last 4 studies, I conclude that:

1) The **depth** of the sea zone should be minimum **30 meters** instead of 20 meters and, of course, the exception that Classification Societies accept of 2 times the draft depth should not be accepted in any case, because the turbulences produced at less depth influence too much the thrust results.

2) On the other hand, from the "**Free distance on the sides of the boat**" study, we discover that for the VSP it is not necessary that much free space. The Classification Societies require a 300 meters radius free distance of obstacles under the water from the boat. The study proves that for the Voith Schneider Propeller **between 50 and 60 meters radius from the boat** it is enough. That makes the performance of the test easier for many shipyards.

3) In this third study, the simulations show that the 300 meters required from the stern to the bollard from the Classification Societies are not enough. At 300 meters the turbulences keep producing too much different directions current that influences the thrust results of the simulations. So, for a Voith Schneider Propeller would be more appropriated to require **400 meters or more between the stern and the bollard** for the bollard pull test performance.

4) In the last study of this chapter we calculate the lost of thrust produced by the current in the test zone. The Classification Societies do not allow to perform the test if there is more than 1-kn current, and, in this case, a lost of 4% of the thrust is to be taken into account. For the Voith Schneider Propeller, 1-kn current produces 22,89% thrust lost. For this reason, 1-kn current in the test zone for a boat with a VSP should not be accepted. Therefore, it is suggested to require **equal or less than ¼-kn current** for this propeller case.

10. Bollard Pull Tests and Classification Societies Rules

In all this next simulations, I used recommended sides, deep and stern distances recommended from the Classification Societies, i.e. not the minimum. And the measures of the piece of sea, which is going to represent different port situations, are: 30 meters of deep, 300 meters from stern to port wall, 150 meters from bow to infinite sea direction and 125 meters on the sides. So it is not minimum conditions of the Classification Societies in any of the directions. This study is made to check the thrust with operational and maximum rotational speed and Classification Societies zones recommendations, and also, to study (without changing "sea" sizes) the influence of a wall in the bottom, stern or sides, two of those together or a wall in all of the sides. Therefore bigger distances as the minimum are taken, not to interrupt our focus of attention with other possible perturbations of turbulences and so on.

The results of the force exerted during the simulations in each element of the tugboat are collected in the following tables:

10.1. Open Water

10.1.1. Maximum power

Table 20 - Report Force of the simulation with maximum power in open water

1) Bollard pull test at maximum power in open water				sea = 450x30x250m			
Stationary state simulation	-10 rad/s			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-11.334	-2.239	-13573,504	-18504,601	-3656,222	-22160,824	
Blades	209483,39	-562,052	208921,34	342013,7	-917,637	341096,06	direcc X
Cover Prop	0	-481,242	-481,242	0	-785,702	-785,702	
Total	198149,32	-3282,731	194866,59	323509,1	-5359,561	318149,54	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	1.620,3481	71,918	1692,266	2645,466	117,417	2762,883	
Blades	-128320,08	-425,842	-128745,92	-209502,17	-695,252	-210197,42	direcc Y
Cover Prop	0	-304,766	-304,766	0	-497,578	-497,578	
Total	-126699,73	-658,69	-127358,42	-206856,7	-1075,413	-207932,11	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	129.965,760	240,358	130206,12	212188,99	392,42	212581,41	
Blades	9597,024	214,992	9812,017	15668,611	351,007	16019,619	direcc Z
Cover Prop	-134632,22	0	-134632,22	-219807,7	0	-219807,7	
Total	4930,564	455,3497	5385,913	8049,899	743,428	8793,328	

In this case the force exerted by the tug in testing bollard pull will be discussed in open water, which means there should not be any different element than the tug influencing the results. Besides considering a draft of 30 meters, which is supposed enough not to influence the results. Moreover I set all boundary conditions in the mode *pressure-outlet*. Since the simulation is at full power, the revolutions of the propeller is set to -10 rad/s, the force generated to the theoretical speed of **198149,32 N** or 20,2056 Newton Force Tons.

10.1.2 Operating power

Table 21 - Force Report of simulation with operational power in open water

2) Bollard pull test to operating power in open water							
Simulación en estado estacionario	-8 rad/s			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-7.941	-1.578	-9518,762	-12964,363	-2576,4732	-15540,836	
Blades	133666,08	-362,539	133303,54	218230,33	-591,901	217638,43	direcc X
Cover Prop	0	-277,816	-277,8158	0	-453,577	-453,577	
Total	125725,41	-2218,445	123506,96	205265,97	-3621,951	201644,02	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	3.019,1946	187,577	3206,772	4929,297	306,248	5235,545	
Blades	-73910,594	-269,391	-74179,984	-120670,36	-439,822	-121110,18	direcc Y
Cover Prop	0	-181,407	-181,407	0	-296,175	-296,175	
Total	-70891,399	-263,221	-71154,62	-115741,06	-429,748	-116170,81	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	84.977,867	154,043	85131,911	138739,38	251,5	138990,88	
Blades	5954,691	125,913	6080,604	9721,9444	205,573	9927,517	direcc Z
Cover Prop	-90617,531	0	-90617,531	-147946,99	0	-147946,99	
Total	315,027	279,957	594,984	514,32956	457,073	971,403	

This simulation is the same as before, but the rotation speed of the blades is varied. As in the previous case, no obstacle interferes and all boundary conditions are set as *pressure-outlet*. In this case the thrust obtained is **125725,41 N** (the same as the model simulation because it is simulated with the same parameters conditions).

10.2. Classification Societies Rules

10.2.1. Maximum power and recommended conditions

Table 22 - Force Reports of simulation made at maximum power and recommended conditions

3) test bollard pull maximuml power and conditions recommended by the Classification Societies							
Simulación en estado estacionario	-10 rad/s	Bottom -> 30m -> wall		Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	-8.373,097	-2.576,801	-10949,88	-13670,333	-4207,022	-17877,355	
Blades	179375,13	-428,759	178946,37	292857,35	-700,015	292157,33	direcc X
Cover Prop	0	-149,829	-149,829	0	-244,618	-244,618	
Total	171002,05	-3155,3885	167846,66	279187,01	-5151,655	274035,36	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	4.201,2070	327,267	4528,474	6859,114	534,313	7393,427	
Blades	-94793,828	-423,042	-95216,87	-154765,43	-690,681	-155456,11	direcc Y
Cover Prop	0	-234,893	-234,893	0	-383,499	-383,499	
Total	-90592,621	-330,669	-90923,29	-147906,32	-539,868	-148446,19	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Hull	99.552,852	220,746	99773,598	162535,27	360,402	162895,67	
Blades	7760,356	229,056	7989,413	12669,97	373,969	13043,939	direcc Z
Cover Prop	-142257,56	0	-142257,56	-232257,24	0	-232257,24	
Total	34944,354	449,803	-34494,552	-57052,007	734,372	-56317,636	

This last results table is simulated with a wall in the 30 meters depth bottom and with the rotational speed of 10 rad/s. This simulation is made with a single *wall* boundary condition in the bottom and not in the other "sea" ending faces because it is complicated to find a place close to the shore with even more depth than 30 meters, i.e. it is an intend to get closer to the real performances.

The total thrust generated is **171002,05 N.** In the following simulations, wall are going to be simulated in different directions to make a comparison between all the simulations in this chapter and determinate in which direction is better or worse (in the sense of % of influence in the results) to have a wall in the performance of the Bollard Pull Test.

10.2.2 Operational power and recommended conditions

Table 23 - Force Report simulation made with operational power and recommended conditions

4) test bollard pull operational power and conditions recommended by the Classification Societies							
Simulación en estado estacionario	-8 rad/s	Bottom -> 30m -> wall		Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Bottom	0	-285,943	-285,943	0	-466,8452	-466,845	
Hull	-7.869,202	-1.556,460	-9425,663	-12847,677	-2541,16	-15388,837	
Blades	133339,88	-374,312	132965,56	217697,76	-611,122	217086,63	direcc X
Cover Prop	0	-281,291	-281,291	0	-459,251	-459,251	
Total	125470,67	-2498,007	122972,67	204850,08	-4078,379	200771,7	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Bottom	0	87,946	87,946	0	143,586	143,586	
Hull	2.666,1820	158,683	2824,865	4352,9496	259,075	4612,0243	
Blades	-74732,875	-269,882	-75002,757	-122012,86	-440,624	-122453,48	direcc Y
Cover Prop	0	-191,469	-191,469	0	-312,603	-312,6032	
Total	-72066,693	-214,722	-72281,416	-117659,91	-350,567	-118010,47	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
Bottom	35257,645	0	35257,645	57563,501	0	57563,501	
Hull	83.937,344	151,593	84088,936	137040,56	247,498	137288,06	
Blades	5918,189	122,147	6040,338	9662,351	199,425	9861,776	direcc Z
Cover Prop	-90913,453	0	-90913,453	-148430,13	0	-148430,13	
Total	34199,725	273,74	34473,465	55836,286	446,923	56283,209	

This simulation performs the same conditions as last one except for the rotational speed, which is the operational speed, 8 rad/s, instead of the maximum speed and produces a thrust of **125470,67 N.** If we compared this result with the results of the simulation 14.2, where instead of a wall at the bottom we have the boundary condition "pressure outlet", i.e. the sea follows equal in that concrete face boundary of the "sea", a loss of 254,74 N of thrust is appreciated.

10.2.3. Operational power and minimum conditions

Table 24 - Force Report of simulation made with operational power and minimum conditions

9) test bollard pull operational power and general minimum conditions recommended by the Classification Societies								
Simulación en estado estacionario		-8 rad/s		Bottom-20,popa-300,laterales100	Pressure Coefficients			
	Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom		0	-46,295	-46,295	0	-75,584	-75,584	
hull		-6796,046	-1651,984	-8448,03	-11095,586	-2697,116	-13792,702	
laterals		0	285,984	285,917	0	466,803	466,803	direc X
blades		121575,05	285,917	121258,12	198489,87	-517,432	197972,44	
stern		-23454,225	0	-23454,225	-38292,612	0	-38292,612	
cover Prop		0	-109,083	-109,083	0	-178,095	-178,095	
Total		91324,776	-1838,372	89486,404	149101,67	-3001,424	146100,25	
	Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom		0	-139,139	-139,139	0	-227,167	-227,167	
hull		7505,802	409,756	7915,558	12254,37	668,989	12923,359	
laterals		905,767	0	905,767	1478,803	0	1478,803	direc Y
blades		-60966,742	-281,579	-61248,321	-99537,538	-459,721	-99997,259	
stern		0	50,691	50,691	0	82,76	82,76	
cover Prop		0	-163,611	-163,611	0	-267,119	-267,119	
Total		-52555,174	-123,883	-52679,057	-85804,365	-461,669	-86006,623	
						-202,258		
	Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom		321516,34	0	321516,34	524924,64	0	524924,64	
hull		59448,805	216,179	59664,984	97059,273	352,946	97412,219	
laterals		0	-1,213	-1,213	0	-1,98	-1,98	direc Z
blades		5303,625	185,325	5.488,949	8658,979	302,572	8961,551	
stern		0	-6,521	-6,521	0	-10,647	-10,647	
cover Prop		-93676,68	0	-93676,68	-152941,52	0	-152941,52	
Total		292592,09	393,771	292985,86	477701,38	642,891	478344,27	

Minimum conditions required by the Classification Societies are here applied with the option wall in all boundaries (except the surface of the "sea" of course) and the thrust result is, as expected, really low, i.e. **91324,776 N**. That means a difference of 34400,634 N.

10.2.4. Operational power and extreme minimum conditions

Table 25 - Force Report of simulation made with operational power and extreme minimum conditions

10) test bollard pull operational power and extrem minimum conditions recommended by the Classification Societies								
Simulación en estado estacionario		-8 rad/s		Bottom-10,8,popa-60,laterales10(	Pressure Coefficients			
	Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom		0	378,211	378,211	0	617,487	617,487	
hull		-7821,059	-1746,321	-9567,38	-12769,077	-2851,136	-15620,213	
laterals		0	61,505	61,505	0	100,416	100,416	direc X
blades		103686,95	-220,633	103466,32	169284,82	-360,218	168924,6	
stern		-122322,39	0	-122322,39	-199710,03	0	-199710,03	
bow		92047,617	0	92047,617	150281,82	0	150281,82	
cover Prop		0	-38,691	-38,691	0	-63,169	-63,169	
Total		65591,12	-1565,929	64025,191	107087,54	-2556,619	104530,92	
	Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom		0	318,552	318,552	0	520,086	520,086	
hull		8417,304	93,934	8511,238	13742,537	153,362	13895,899	
laterals		-3008,703	0	-3008,703	-4912,168	0	-4912,168	direc Y
blades		-46106,094	-269,139	-46375,233	-75275,255	-439,411	-75714,666	
stern		0	131,362	131,362	0	214,469	214,469	
bow		0	-145,785	-145,785	0	-238,016	-238,016	
cover Prop		0	-109,388	-109,388	0	-178,593	-178,593	
Total		-40697,493	19,537	-40677,957	-66444,887	31,897	-66412,99	
	Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom		-929993,31	0	-929993,31	-1518356,4	0	-1518356,4	
hull		53539,047	174,769	53713,816	87410,689	285,338	87696,027	
laterals		0	-6,037	-6,037	0	-9,856	-9,856	direc Z
blades		4455,833	156,202	4.612,035	7274,829	255,024	7529,854	
stern		0	-7,633	-7,633	0	-12,462	-12,462	
bow		0	-2,539	-2,539	0	-4,145	-4,145	
cover Prop		-88064,375	0	-88064,375	-143778,57	0	-143778,57	
Total		-960062,81	314,763	-959748,04	-1567449,5	513,899	-1566935,6	

And here, the extreme minimum conditions are simulated, i.e. twice the draft large depth (10,8 m) and twice the boat water line large from the stern to the bollard (60 m). To perform the Bollard Pull Test with this zone conditions, the owner of the vessel have to be aware of the conditions and their influences on the thrust results, and he has to agree to perform the test that way. As logical, with this zone conditions we loss even more thrust in comparison with the model simulation, 60134,29 N, achieving just a thrust of **65591,12 N**.

10.3. Wall probes with Classification Societies Distances

10.3.1 Bottom, stern and laterals

Table 26 - Force Report of simulation with "wall" condition at bottom, stern and laterals

5) test bollard pull operational power and conditions recommended by the Classification Societies A							
Simulación en estado estacionario	-8 rad/s	Bottom,popa,laterales -> wall		Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom	0	441,1093	441,109	0	671,199	671,199	
hull	-8250,876	-1638,1752	-9889,051	-13470,818	-2674,572	-16145,39	
laterals	0	315,5535	315,5535	0	515,189	515,189	direc X
blades	132901,44	-359,8621	132541,58	216981,94	-587,53	216394,41	
stern	-20800,285	0	-20800,285	-33959,649	0	-33959,649	
cover Prop	0	-268,156	-268,1556	0	-437,805	-437,805	
Total	103850,28	-1539,53	102310,75	169551,47	-2513,518	167037,95	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom	0	16,169	16,169	0	26,399	26,399	
hull	2725,447	148,709	2874,156	4449,71	242,79	4692,5	
laterals	-6253,965	0	-6253,9653	-10210,556	0	-10210,556	direc Y
blades	-74173,563	-272,275	-74445,837	-12099,69	-444,53	-121544,22	
stern	0	16,567	16,566618	0	27,048	27,048	
cover Prop	0	-191,943	-191,943	0	-313,376	-313,376	
Total	-77702,08	-282,773	-77984,853	-126860,54	-461,669	-127322,21	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom	285828,69	0	285828,69	466659,08	0	466659,08	
hull	80170,016	154,657	80324,672	130889,82	252,5	131142,32	
laterals	0	-47,588	-47,588	0	-77,695	-77,695	direc Z
blades	5950,2036	118,809	6.069,013	9714,618	193,974	9908,593	
stern	0	0,251	0,251	0	0,409	0,409	
cover Prop	-90385,844	0	-90385,844	-147568,72	0	-147568,72	
Total	281563,06	226,128	281789,19	459694,8	369,189	460063,99	

Here, I performed a zone with walls at the bottom, sides and stern directions. The lost of thrust in this case is much bigger, falling from 125725,41 N of the simulation model to the **103850,28 N** of thrust in this simulation 14.5. So there is a exact lost of 21875,13 N due to the walls at the recommended distances by the Classification Societies, which, on my eyes, it is a too big lost of thrust to accept this distances as valid for the test performances of boats with a VSP.

10.3.2 Stern and laterals

Table 27 - Force Report of simulation with "wall" condition at stern and laterals

6) test bollard pull operational power and conditions recommended by the Classification Societies B							
Simulación en estado estacionario	-8 rad/s		popa,laterales -> wall	Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
hull	-7899,81	-1543,098	-9442,908	-12897,649	-2519,344	-15416,993	
laterals	0	-143,187	-143,187	0	-233,775	-233,774	direc X
blades	132988,23	-358,596	132629,64	217123,65	-585,463	216538,19	
stern	-14384,34	0	-14384,34	-23484,636	0	-23484,636	
cover Prop	0	-286,852	-286,852	0	-468,33	-468,33	
Total	110704,08	-2331,733	108372,35	180741,36	-3806,911	176934,45	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
hull	3185,203	176,156	3361,359	5200,331	287,601	5487,933	
laterals	4486,444	0	4486,444	7324,807	0	7324,807	direc Y
blades	-74705,188	-274,644	-74979,831	-121967,65	-448,398	-122416,05	
stern	0	23,449	23,449	0	38,284	38,284	
cover Prop	0	-177,597	-177,597	0	-289,954	-289,954	
Total	-67033,54	-252,636	-67286,176	-109442,51	-412,467	-109854,98	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
hull	85954,547	147,403	86101,95	140333,95	240,657	140574,61	
laterals	0	-158,237	-158,237	0	-258,346	-258,346	direc Z
blades	5891,009	126,856	6017,865	9617,974	207,111	9825,085	
stern	0	-91,119	-91,119	0	-148,766	-148,766	
cover Prop	-89886,313	0	-89886,313	-146753,16	0	-146753,16	
Total	1959,2432	24,903	1984,146	3198,764	40,657	3239,422	

In this 10.3.2 simulation, a wall stern (a popa) and at the laterals of the boat is recreated with the program and the resultant thrust is **110704,08 N** with 15021,33 N lost.

10.3.3 Bottom and stern

Table 28 - Force Report with simulation with "wall" condition at bottom and stern

7) test bollard pull operational power and conditions recommended by the Classification Societies B							
Simulación en estado estacionario	-8 rad/s		bottom,popa -> wall	Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom	0	-162,382	-162,382	0	-265,114	-265,11405	
hull	-8124,115	-1639,055	-9763,169	-13263,861	-2676,009	-15939,869	direc X
blades	134129,3	-358,975	133770,32	218986,61	-586,081	218400,53	
stern	-19481,566	0	-19481,566	-31.806.639	0	-31806,639	
cover Prop	0	-285,3	-285,301	0	-465,798	-465,798	
Total	106523,62	-2445,714	104077,9	173916,11	-3993,002	169923,11	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom	0	151,036	151,036	0	246,589	246,589	
hull	4105,651	180,634	4286,286	6703,104	294,913	6998,017	direc Y
blades	-74374,484	-271,859	-74646,344	-121427,73	-443,853	-121871,58	
stern	0	-18,83	-18,83	0	-30,743	-30,743	
cover Prop	0	-180,255	-180,255	0	-294,295	-294,29	
Total	-70268,833	-139,275	-70408,108	-114724,63	-227,388	-114952,01	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom	26097,535	0	26097,535	42608,221	0	42608,221	
hull	82735,719	154,6	82890,319	135078,72	252,409	135331,13	direc Z
blades	5967,461	123,327	6090,788	9742,793	201,349	9944,143	
stern	0	-30,616	-30,616	0	-49,986	-49,986	
cover Prop	-89360,148	0	-89360,148	-145894,12	0	-145894,12	
Total	25440,566	247,31058	25687,877	41535,619	403,772	41939,391	

When the wall boundary condition is simulated at the bottom of the “sea” and (a popa) stern of the boat, the obtained thrust is **106523,62 N**, and the loss compared with the simulation model is 19201,79 N.

10.3.4 Bottom and laterals

Table 29 - Force Report of simulation with "wall" condition at bottom and laterals

8) test bollard pull operational power and conditions recommended by the Classification Societies C							
Simulación en estado estacionario	-8 rad/s	Bottom,laterales -> wall		Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom	0	-1020,339	-1020,339	0	-1665,859	-1665,859	
hull	-7748,212	-1672,901	-9421,113	-12650,143	-2731,266	-15381,409	
laterals	0	-151,849	-151,85	0	-247,918	-247,918	direc X
blades	134441,25	-360,091	134081,16	219495,92	-587,904	218908,01	
cover Prop	0	-276,7	-276,7	0	-451,756	-451,756	
Total	126693,04	-3481,881	123211,16	206845,78	-5684,703	201161,07	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom	0	-149,396	-149,396	0	-243,911	-243,912	
hull	3536,823	192,105	3728,928	5774,4041	313,642	6088,046	
laterals	13102,595	0	13102,595	21391,991	0	21391,991	direc Y
blades	-73973,523	-271,037	-74244,56	21391,991	-442,509	-121215,61	
cover Prop	0	-186,099	-186,099	0	-303,836	-303,836	
Total	-57334,106	-414,426	-57748,532	-93606,704	-676,614	-94283,318	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	
bottom	285450,56	0	285450,56	466041,73	0	466041,73	
hull	80782,195	165,811	80948,006	131889,3	270,712	132160,01	
laterals	0	1,322	1,322	0	2,159	2,158	direc Z
blades	5936,791	117,551	6.054,342	9692,72	191,919	9884,639	
cover Prop	-88945,984	0	-88945,984	-145217,93	0	-145217,93	
Total	283223,56	284,684	283508,25	462405,82	464,789	462870,61	

In the last simulation of this chapter, the test zone of the bollard pull test is simulated with walls at the sides of the boat and at the bottom of the "sea". In this result table we can see a much smaller lost than in the lasts simulations, actually a negative lost, i.e. -967,63 N, achieving a thrust of **126693,04 N**. That means that with the walls situated at the Classification Societies required distances, which in the last study we saw that were not appropriated for this propeller, we achieve a higher thrust than with free sea (boundary condition "pressure outlet") in all directions. Obviously, this is impossible; therefore it is logical to think that this result it is been influenced by the turbulences and currents produced by the water crashing with the walls. Therefore, the thrust result of this simulation cannot be accepted for future comparisons and studies.

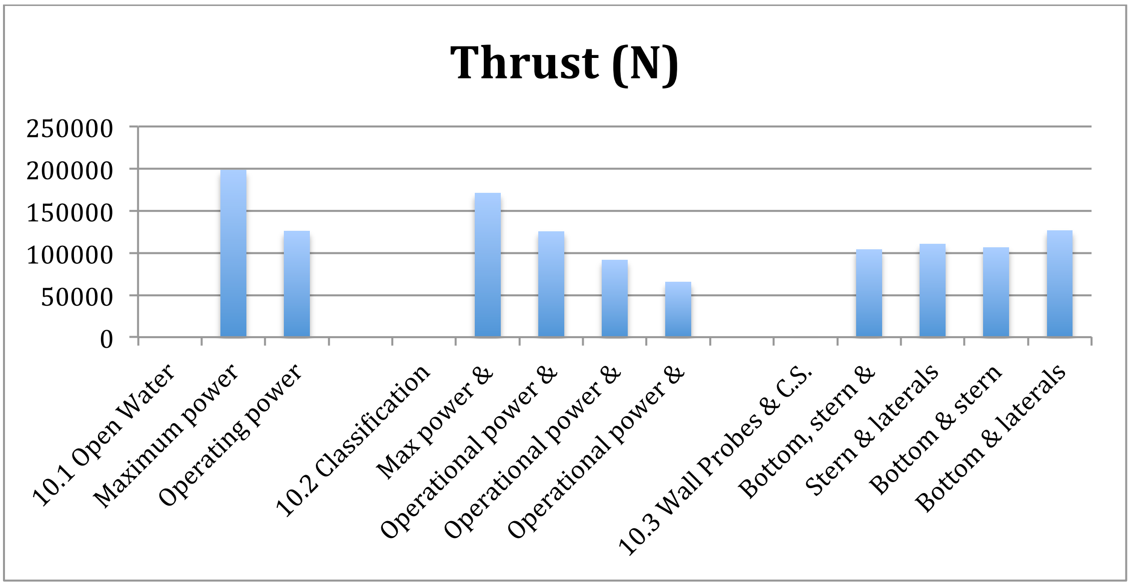
10.4 Comparing table between Bollard pull simulated tests

In the next table I summed up the important thrusts of this chapter to be able to make comparisons between them:

Bollard Pull & Classification Societies	
10.1 Open Water	**Thrust (N)**
Maximum power	198149,32
Operating power	125725,41
10.2 Classification Societies Rules	
Max power & recommended conditions	171002,05
Operational power & recommended condition:	125470,67
Operational power & minimum conditions	91324,776
Operational power & extreme min conditions	65591,12
10.3 Wall Probes & C.S. Distances	
Bottom, stern & laterals	103850,28
Stern & laterals	110704,08
Bottom & stern	106523,62
Bottom & laterals	126693,04

Table 30 - Comparing simulations of Classification Societies distances requirements

Of course, in the fist part (Open Water) we find the higher thrust results, because there is nothing under water affecting it negatively. It is also noticeable, that both the Operating power (125725,41 N) and the Maximum power (198149,32 N) are higher that the thrust of the next simulations in the 10.2 Classification Societies Rules. That is toe to the distances on the bottom, sides and stern at different distances. Before calculating the error of the 10.2 simulations compared with the 10.1 proves, we find the next graphic, where the same information it is easier to evaluate:



Graphic 13 - Thrust comparison between simulations with Classification Societies free distances requirements

- With **recommended conditions**:

Which are 350 meters from stern to the bollard, 125 meters free to each side and 30 meters deep, quite more that what require Classification Societies:

- Maximum power

$$e_{rel} = \frac{T_c - T_m}{T_m} = \frac{171002{,}05 \ - 198149{,}32}{198149{,}32} = -0{,}137$$

There is a 13,7% error when compare it with the Open Water thrust results.

- Operational power

$$e_{rel} = \frac{T_c - T_m}{T_m} = \frac{125470{,}67 \ - 125725{,}41}{125725{,}41} = -0{,}002$$

There is a 0,2% error when compare it with the Open Water thrust results, which is good and could be an indicator for the new suggestions and the respective simulations.

- With **minimum conditions**

Which are the required by the Classification Societies specified in chapter 4.

$$e_{rel} = \frac{T_c - T_m}{T_m} = \frac{91324{,}776 \ - 125725{,}41}{125725{,}41} = -0{,}274$$

As we can see, if we use the distances required by the Classifications Societies for this propeller we obtain a 27,4% thrust loss, which is too high to be accepted.

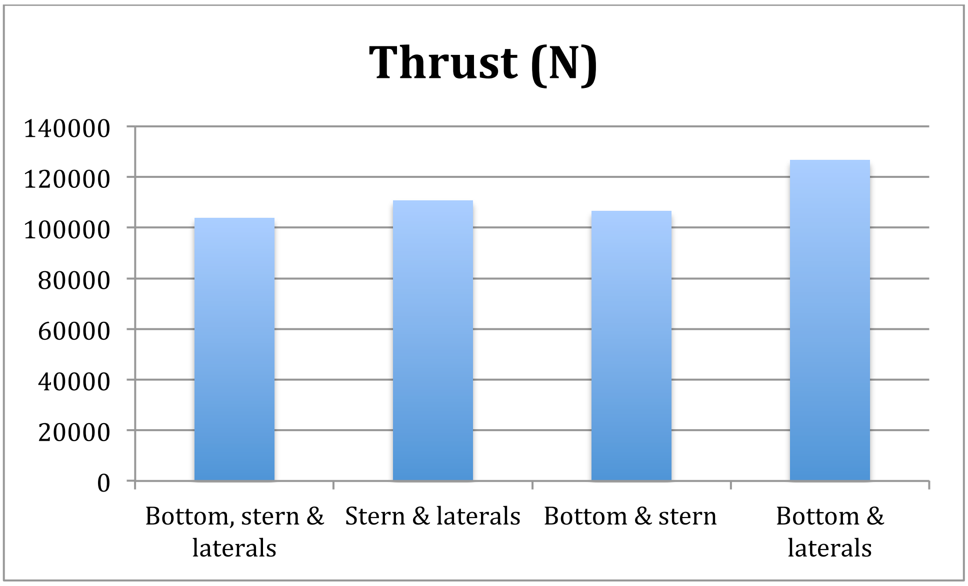
- With **extreme minimum conditions**

In this case, for the normal propellers with helix and rudder, the Classification Societies also say that there is a high loss and that the owner of the boat should be adverted and should agree to do the test in this conditions. Those distances for special cases are two times the draft depth and two times the waterline length between the stern and the bollard. Following we calculate the relative error:

$$e_{rel} = \frac{T_c - T_m}{T_m} = \frac{65591{,}12 \ - 125725{,}41}{125725{,}41} = -0{,}478$$

But, in any case, that thrust loss of the regular propellers reach even a third of the total, because it would not be accepted. As we just calculated, for this propeller and these conditions of the test there is almost a 50% thrust loss, which is totally unacceptable.

The last 4 simulations studied in the 10.3 Wall Probes and Classification Societies Distances, as it is shown in the last graphic, we find the lowest thrust results for operational power and recommended distances. This is due to the combination of more than one wall. In this case, we want to see in which direction it is more important not to have a wall; i.e. obstruction of the current direction. The following graphic is the same as earlier but zoomed in to pay attention to this 4 last simulations.



Graphic 14 - Thrust comparison between chapter 10.3 simulations "Wall proves with Classification Societies Distances"

It is noticeable, that when we set the condition “wall” in the three studied directions we obtain the lowest thrust value, as expected. But it is also easy to see, that when the stern current direction it is not disturbed, we obtain the highest results. Therefore, if there is no the chance than interrupt one or two directions distances and there is the chance to choose which one, we will **always save the largest free distance for the stern-bollard distance**.

11. Conclusions and suggestions to the Bollard Pull test for the VSP

Summing up the results of last simulations, for a Voith Schneider Propeller should be required 30 meters of depth, 60 meters free to port and starboard and minimum 400 meters free from stern to the bollard. In this chapter, two more simulation studies have been done, to check if those distances also work together. It is easy to imagine, that when the study is done just for one parameter and all the other boundary conditions are "pressure outlet" there is not any other turbulence or current that perturbation the results and the obtained results may not be appropriate when using all the suggestions in the same simulation.

11.1. Simulation of the suggested Bollard Pull for the VSP

In the following table are disposed the results for the suggested parameters simulated all together (i.e. boundary condition "wall" in all directions) for the bollard pull test with 8 rad/s rotational speed, the operational speed:

Table 31 - Force Report of simulation with suggested distances at operational speed

1) Suggested Bollard Pull Test; -8 rad/s	Y=60, Z=30,-X=400 --> walls						
Stationary state simulation	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	X-axis
bottom	0	570,179	570,179	0	930,905	930,905	
hull	1690,849	-1550,386	140,463	2760,571	-2531,243	229,328	
laterals	0	239,406	239,406	0	390,867	390,867	
blades	141970,34	-147,851	141822,49	231788,32	-241,289	231546,93	
stern	-28778,809	0	-28778,809	-46985,81	0	-46985,81	
cover Prop	0	-29,124	-29,1244	0	-47,55	-47,55	
Total	114882,38	-917,776	113964,61	187563,08	-1498,41	186064,67	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	Y-axis
bottom	0	63,608	63,608	0	103,85	103,85	
hull	-11308,518	32,331	-11276,186	-18462,886	52,786	-18410,1	
laterals	-47,905	0	-47,905	-78,213	0	-78,213	
blades	-92189	-203,471	-92392,471	-150512,65	-332,197	-150844,85	
stern	0	0,681	0,681	0	1,113	1,113	
cover Prop	0	-59,174	-59,174	0	-96,611	-96,611	
Total	-103545,42	-166,024	-103711,45	-169053,75	-271,059	-169324,81	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	Z-axis
bottom	-403902,47	0	-403902,47	-659432,6	0	-659432,6	
hull	3438,841	699,959	4138,799	5614,434	1142,79	6757,224	
laterals	0	-16,756	-16,756	0	-27,357	-27,357	
blades	4170,721	206,981	4.377,702	6809,34	337,929	7147,269	
stern	0	0,007	0,007	0	0,011	0,011	
cover Prop	-39729,602	0	-39729,602	-64864,656	0	-64864,656	
Total	-436022,51	890,191	-435132,32	-711873,48	1453,373	-710420,11	

The results give **114882,38 N** of pressure force, which it is not far from the model simulation. Exactly there is the following error:

$$e_{rel} = \frac{T_c - T_m}{T_m} = \frac{114882{,}38 - 125725{,}41}{125725{,}41} = -0{,}086$$

There is 8,6% of error. Which compared with the error from the simulations made with the Classification Society distances (also boundary condition “wall” at stern, sides and bottom; simulation 10.3.1.) which gives 103850,28 N of pressure force is:

$$e_{rel} = \frac{T_c - T_m}{T_m} = \frac{103850{,}28 - 125725{,}41}{125725{,}41} = -0{,}174$$

There is 17,4% of error, over the double of the error of the simulation with the suggested distances.

Following, I performed the same simulation with the maximal power and this following table contain the forces reports:

Table 32 - Force Reports of simulation with suggested distances with maximum power

1) Suggested Bollard Pull Test; -10 rad/s	Y=60, Z=30,-X=400 --> walls						
Stationary state simulation	Forces Report			Pressure Coefficients			
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	X-axis
bottom	0	548,186	548,186	0	894,997	894,997	
hull	4769,452	-2307,113	2462,339	7786,859	-3766,715	4020,145	
laterals	0	98,119	98,119	0	160,194	160,194	
blades	221816,03	-223,006	221593,03	362148,62	-364,091	361784,53	
stern	-54796,16	0	-54796,16	-89463,119	0	-89463,119	
cover Prop	0	-45,529	-45,529	0	-74,334	-74,334	
Total	171789,32	-1929,344	169859,98	280472,36	-3149,949	277322,42	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	Y-axis
bottom	0	171,52	171,52	0	280,033	280,033	
hull	-16190,852	71,325	-16119,527	-26434,043	116,449	-26317,595	
laterals	-2891,201	0	-2891,201	-4720,328	0	-4720,328	
blades	-142927,14	-314,442	-143241,58	-233350,43	-513,375	-233863,81	
stern	0	0,15	0,15	0	0,246	0,246	
cover Prop	0	-89,375	-89,375	0	-145,919	-145,919	
Total	-162009,19	-160,822	-162170,02	-264504,81	-262,566	-264767,37	
Component	Pressure Force (N)	Viscous Force (N)	Total Force (N)	Normal Torque	Viscous Torque	Total Torque	Z-axis
bottom	-461964,38	0	-461964,38	-754227,55	0	-754227,55	
hull	8175,982	1068,711	9244,694	13348,543	1744,835	15093,377	
laterals	0	-212,203	-212,204	0	-346,455	-346,455	
blades	6655,581	304,061	6.959,642	10866,254	496,426	11362,68	
stern	0	-0,073	-0,073	0	-0,119	-0,119	
cover Prop	-64909,59	0	-64909,59	-105974,84	0	-105974,84	
Total	-512042,4	1160,496	-510881,91	-835987,59	1894,687	-834092,91	

In this case, the simulation pressure force is **171789,32 N,**

$$e_{rel} = \frac{T_c - T_m}{T_m} = \frac{171789{,}32 \; - \; 198149{,}32}{198149{,}32} = -0{,}133$$

Which represents a lost of 13,3 next to the model simulation. It is a much higher error than in the last simulation with operational power. But, in any case, there is not even the need to proceed making the simulation with all boundary conditions as a "wall" with the Classification Societies distances because if we compare it with the simulation 10.2.1 it already gives less Newton of pressure force. The simulation 10.2.1 is performed just with the sea bottom boundary condition set as "wall" and reports 171002,05 Newton. Even if the forces difference between both is not so big, that means that the Classification Societies distances, with maximum power and all boundary conditions set as a “wall” would give a much higher error.

Therefore, even if there is still an error when comparing the model simulation (with operational and maximum speed) with the suggested distances simulations, by the moment they give better results than the simulations made with the actual bollard pull distances regulation.

In this chapter is being proved, that the actual requirements that the Classification Societies apply to all bollard pull should not be applied for boats with a Voith Schneider Propeller. In this case, till there is not a more accurate study validated with results of real bollard pull tests cases (which were not available for this project), the mention suggestions of distance requirements for the bollard pull test area could be used.

12. Budget

It is complicated to quantify the hours of work till you can have a 100% reliable model that enables the boats designer and builders to directly put out the bollard pull test. Basically it is impossible to know how many models have to be done till you find the way of simulating that gives almost no error in the moment to compare the thrust result with the thrust obtained in the bollard pull test of different boats. The goal is to find a method that directly imports the boat drawings to the CDF program and all the sets are already proved and worked. With that methodology, many more bollard pull tests have to be done and the results will be compared with the simulations results. Obviously this process shall be carried in a automatized shipyard, where all the drafts match perfectly with the reality, so there is no error before importing them and creating the 3D model in the CFD program. Ones the method had been proved and there is minimum error in all the different boat comparisons, the model can be accepted and the error can be attributed to the sea state and wind.

Therefore I can only do a simple approach:

a) Ansys CDF Workbench

The cost to the customer of commercial CFD software varies from zero to a sort of prices depending on the costs and likely benefits to the CFD company of the customer using the code. That means that the price for the program licence it is not fix, anyway, an indicative price is:

- Cost: 17.170 euro
- Annual licence & updates: 3.260 euro
- Introduction curse: 1.500 euro

b) Working engineer price in Spain

It is also a complicated matter to estimate. In the supposed case that 40.000 euro brute per year is the standard salary of an engineer in Spain, that is 3333,33 euro per month for the company. It is also interesting that more than one engineer is inside the project to avoid personal mistakes.

One study should be done for each boat build in the shipyard till there is enough to prove that the develop methodology is right, therefore it is not a continued work because not all boats in a shipyard are done at the same time. Anyway, adding all the separated times I would estimate two month of work to develop the methodology, which means 13333,32 euro.

c) Bollard Pull Test

I cannot estimate the price of the bollard pull test performing because only the shipyards know the price of it for each vessel. Moreover it is not a cost of that project because the shipyard would do the bollard pull test anyway. But it is interesting to take into account (supposing that it is expensive to perform that test) the things and procedures that the shipyard would save:

- Procedure of the test (time, attending personal, gas of the boat…)
- Equipment (rope, dynamometer…)
- Certification (price of the certification, bureaucracy…)

Table 33 - Prices estimation

CONCEPT	**PRICES**		
ANSYS			
	Cost	17170	
	Annual Licence & Updates	3260	
	Introduction Curse	1500	
		21930	
Workers			
	euro/month	3333,33	
	2 month	2	
	2 engineers	2	
		13333,32	
Total methodology price		**35263,32**	**euro**
Further annual costs		**3260**	**euro**

To conclude, a possible estimation of the price is a cost of 35.263,32 euros to make models, compare it with results from the real test and validate the methodology for achieving the thrust force since the design phase. And, from then on, would be a cost of 3260 euros per year for the program licence and updates.

13. Final Conclusions

To solve complicated, costly and dangerous processes, the importance of CFD models is in this project highlighted. In this case a methodology to achieve an accurate thrust value already in the design phase of the ship design is created. As already said, the full project studies different propeller types and sizes and different zone recreation, and in this concrete part of the investigation solutions of a model with a VSP are found. A methodology of analysis has been recreate, through CFD simulations, for testing the bollard pull test of a tug with a Voith Schneider Propeller obtaining similar results to the reality. The model has been developed using the program ANSYS FLUENT finite volumes. The effectiveness of the method was checked validating the results of the simulations with actual information of really similar tugs with the same propeller. Of course one model it is not enough to fully validate a real methodology. That's why, as earlier explained in the project, this is just a part of the LEPUM investigation project to create an actual methodology for the bollard pull test, which does not yet exist in the world. If the LEPUM project would finally have enough validated models with the same results to ensure that the discovered methodology functions, after Classification Societies would check it, there would be no more wrong estimations in the design phase, saving that way huge amounts of money. Because the boat and propeller shape would be inserted in the computer and the program, following the discovered methodology, would run the simulation and give the thrust results.

After verifying the validity of this part of the project methodology, not as good results as in the helix studies were found. That is because the thrust and torque coefficient graphics are confidential data from the Voith Company so it is not possible to find a theoretical result. Moreover the tugs with almost the same shape hull bring two VSP, so there is no simulation result of the effect of the second propeller, which it is evident that it makes a difference. Nonetheless the model created, in this part of the investigation, gives similar thrust results to the already built really similar tugs with VSP.

Ones the quantity of thrust and some other parameters were studied, the influence of the characteristics of the area (where tests are performed) on the test results are studied. These simulations show, that a major thrust will be obtained if the test is conducted in areas of greater depth and with greater clearance around the tug and to be the most critical distance the one from the bollard to the stern. Additionally, the reduction of the pressure force caused by 1-kn current was also studied.

Following all the area characteristics possibilities accepted by Classification Societies were studied to compare the results with simulations of the previous chapter.

After analysing all the simulations, some requirements suggestions for the test area for bollard pull tests with Voith Schneider Propellers are proposed. With these distances proposals, simulations were also run, thus demonstrating that the combination of the suggested distances (depth, bollard-stern and to the sides) give better thrust results, i.e. more similar to the reality, than the current simulations with the distances of the Classification Societies (as long as the test is performed with a Voith Schneider Propeller). Because, as seen in Joshua's thesis work, the simulations studies of the bollard pull test distances with conventional propellers demonstrate that the Classification Societies distances are the most appropriated to perform the test for this kind of propellers. Therefore would be very interesting to recreate the distances study made in this final grade project to really validate it and, in the case this simulations are right, avoid mistakes for the next bollard pull test with embarkations using a VSP.

The problem, obbiusly, have to keep being studied and validated with the results from actual proves (which were also not available for this project) and after many proves where the error of the simulation is almost zero, the method could be officially approved not just as a indication of distances in the bollard pull test zone, but also as a method to assure a thrust value before the boat is build, giving solution to the exposed problem in the introduction. If the test would be perfectly reconstructed by CFD, there would be no more need to lose time and money performing the bollard pull test and this is the LEPUM objective.

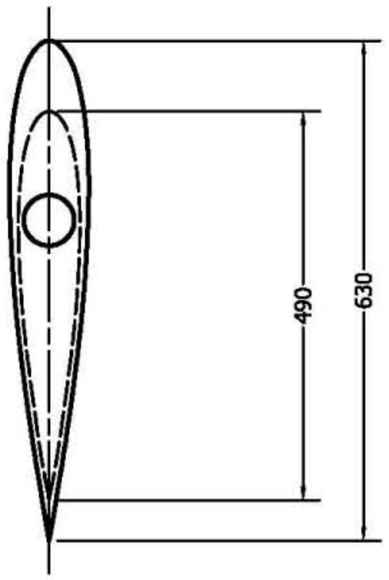
In this fourth part of the global project there is not the end, now another student (Carlos Vega) will start modifying the present model to simulate a turning tug. That will be achieved by designing a model where the blades inside the rotary domain also rotate as explained in chapter 6. For further future projects, conduct studies on boats with multiple propellers would be a great idea to analyse the interaction in this type of configuration.

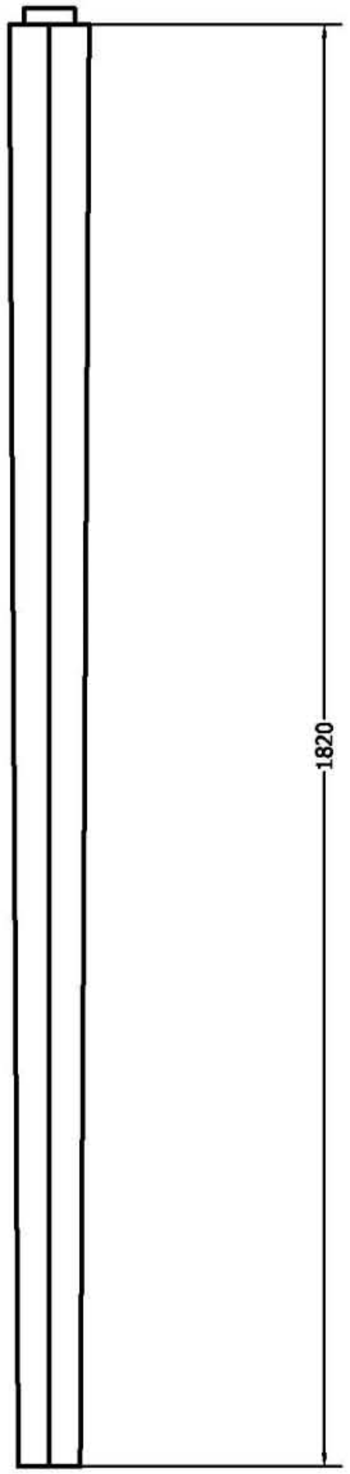
14. Annexes

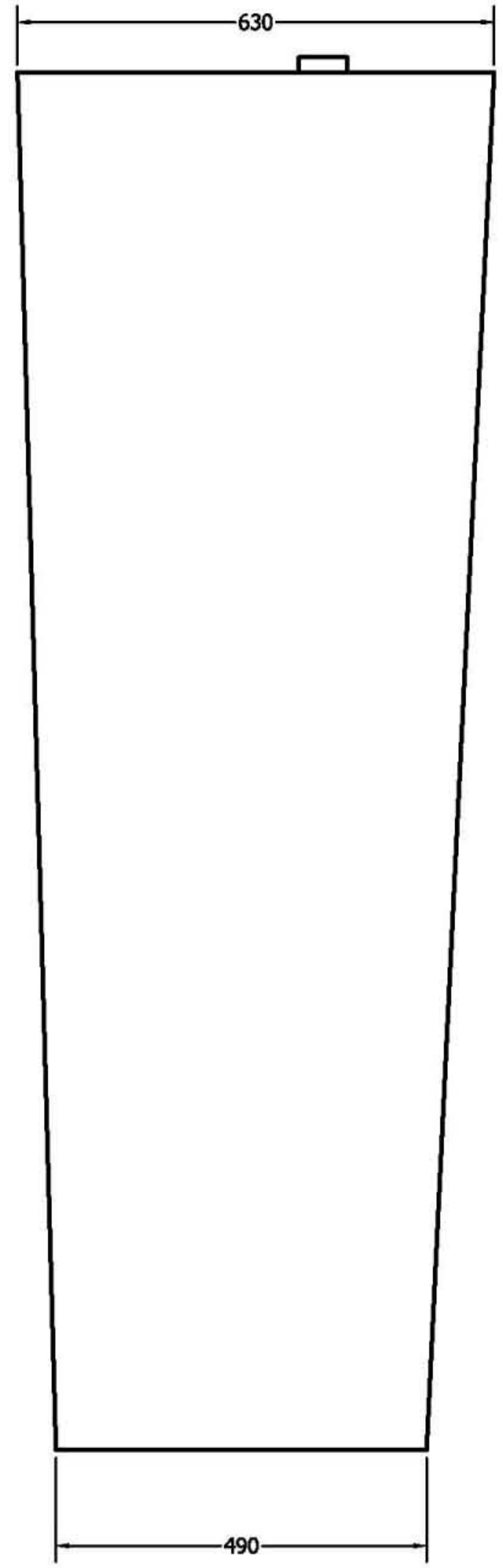
In this chapter the drawings of the hull, propeller and blades remade for this project are in A3 pages following displayed.

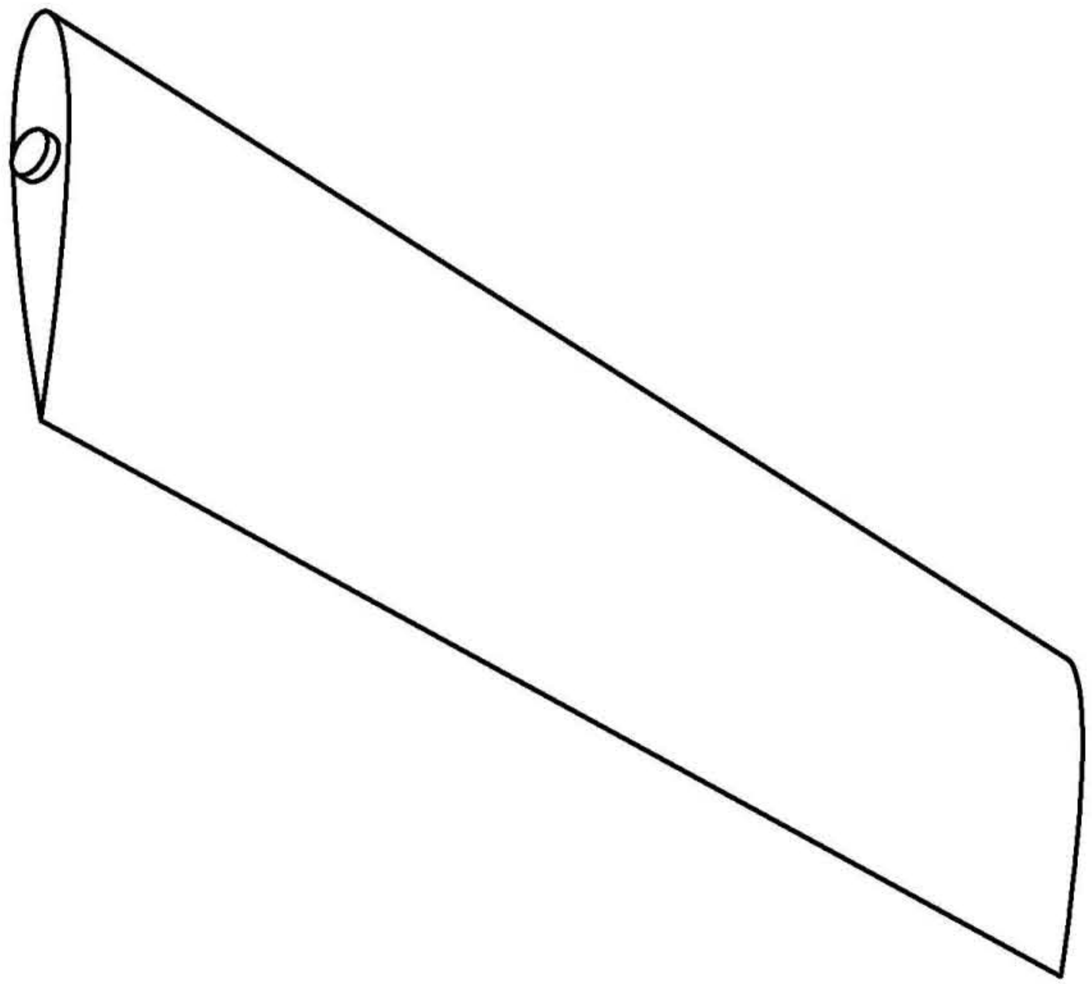
A.1. Blades

Ones one blade was created, it was multiplied and the five resultants were properly placed each with its angle of attack.









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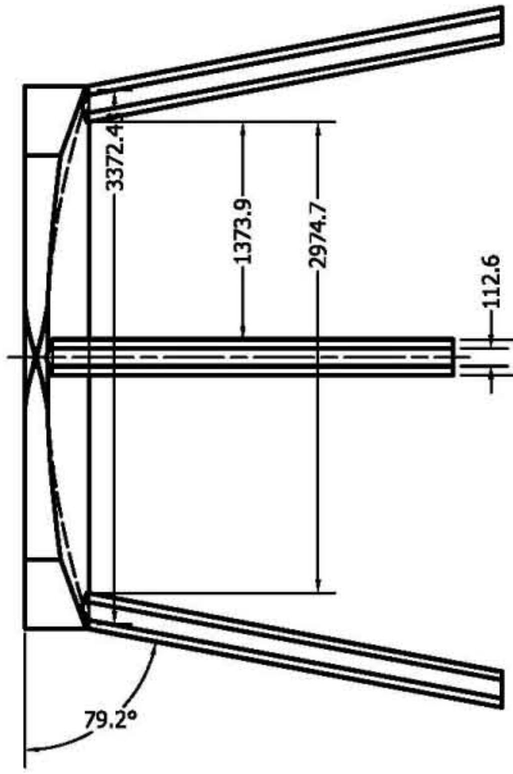
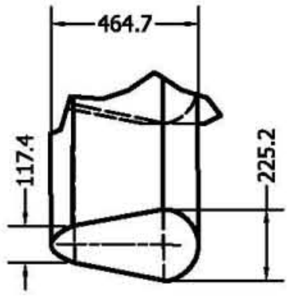
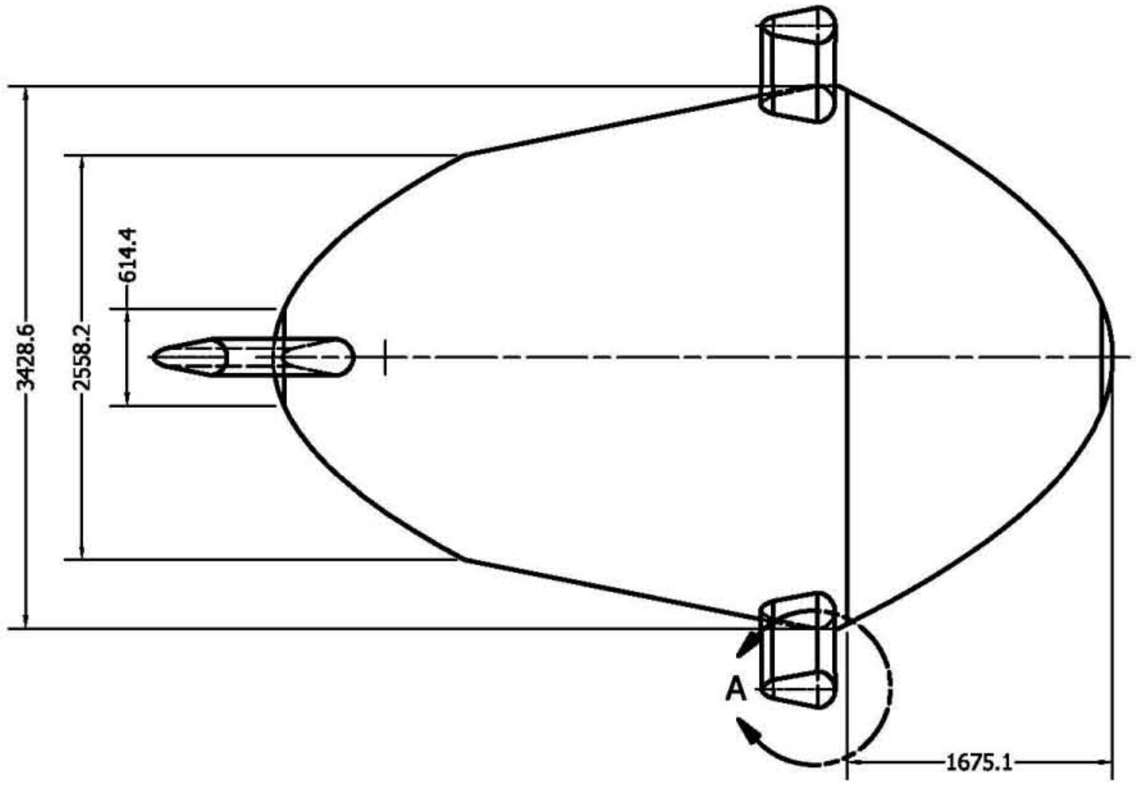
Maria Massó Duxans	
Drawn 4.7.2015	Scale 95,8:1 — A3
BLADE PROFILE	PROYECTO FINAL DE GRADO

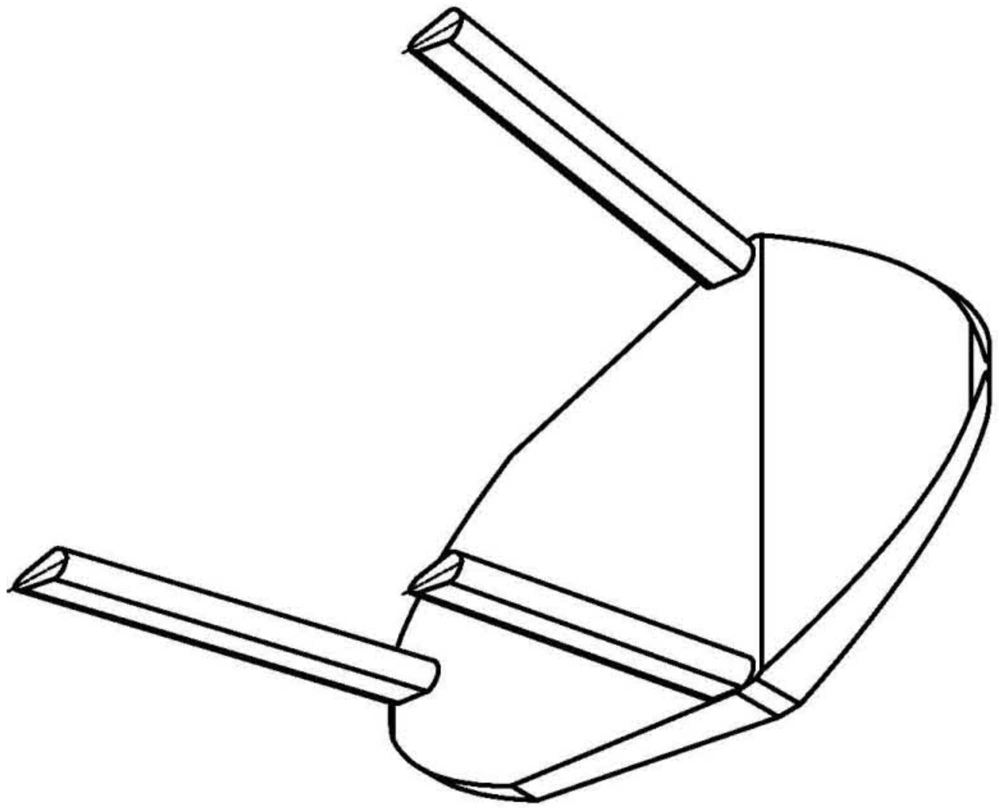
A.2. Skate

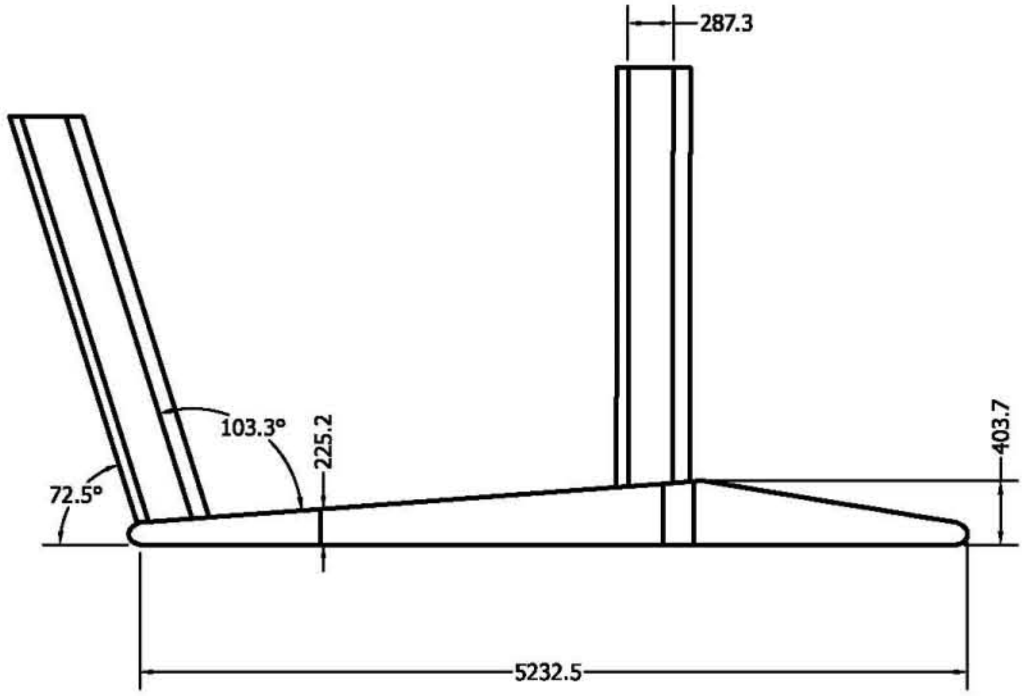




DETAIL A
SCALE 0,06 : 1







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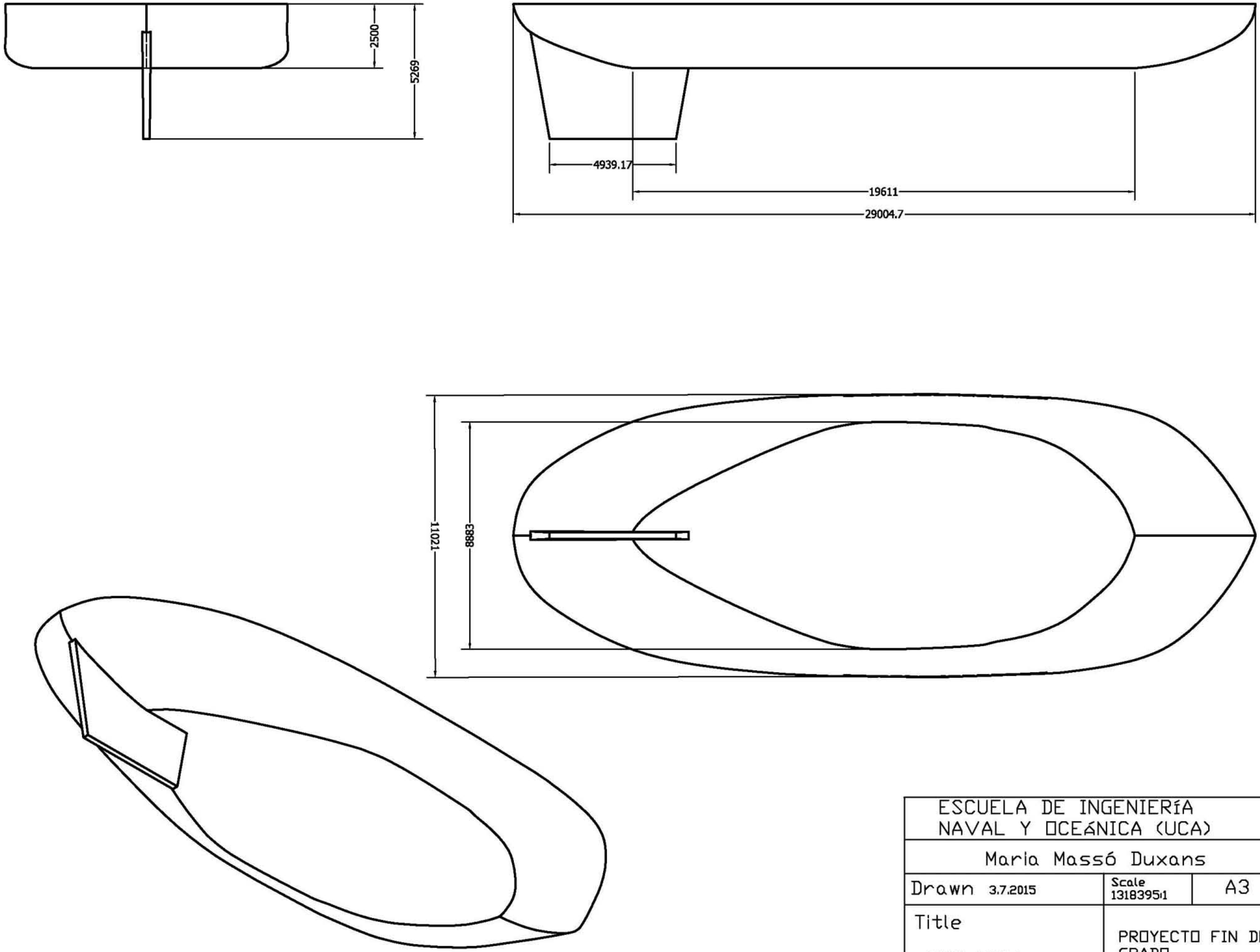
Drawn 30.6.2015	Scale 95,8:1	A3
Title SKATE	PROYECTO FIN DE GRADO	

A.3. Tug



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Maria Massó Duxans

Drawn 3.7.2015	Scale 1318395:1	A3
Title TUG HULL	PROYECTO FIN DE GRADO	

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