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Editorial Note

Cartagena de Indias, 21st January 2014.

2013 came to an end with important projects and initiatives related to the administration of science, technology, and innovation; additionally, this was a year of productivity for COTECMAR, which is why efforts emanating from science and technology were aimed at waging on the development of projects focused not only on improving the Corporation's internal productivity levels, but also those of the naval, maritime, and riverine industry. For this purpose, competitive enhancement was carried out with the micro and small enterprises that make up the shipyard productive chain in the region of Meta, strengthening 10 companies in the zone in managerial skills, association strategies, and good practices regarding ship construction and repair.

Undoubtedly, 2013 was a period of interrelations where the collective construction of knowledge prevailed with national and international entities. A technological transference process was developed with the company STX from South Korea within the framework of the Project for the construction of the Coastal Patrol Vessel (CPV). The project on design and construction of the Amazon Patrol Vessel was consolidated with ENGEPROM from Brazil. We feel pride in seeing reflected in our products and services the result of a collective effort through which we contribute to the generation of capacities of a Strategic Industry for the nation's security.

This edition is structured with articles related to isotropic modeling, stability, optimization of hull design, noise and vibration assessment, multi-influence measures to characterize global signature, design model of a hydrodynamic test tank, and from the perspective of competitiveness, an article related to the action plan to restructure technologies in a shipyard.

We thank our authors and readers for their interest, commitment and contribution, which permit our delivering quality work for the dissemination and generation of new knowledge in the scientific community of naval design, architecture, and engineering.

Captain OSCAR DARÍO TASCÓN MUÑOZ
Editor of the Ship Science and Technology Journal

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Nota Editorial

Cartagena de Indias, 21 de Enero de 2014.

Finalizó el año 2013 con importantes proyectos e iniciativas relacionadas con la gestión de la ciencia, la tecnología y la innovación, además fue el año de la Productividad para Cotecmar, razón por la cual los esfuerzos desde la ciencia y la tecnología estuvieron encaminados a apostarle al desarrollo de proyectos enfocados no sólo en mejorar los niveles de productividad internos de la Corporación sino también los de la industria naval, marítima y fluvial, para ello se llevó a cabo el fortalecimiento competitivo de las micro y pequeñas empresas que componen la cadena productiva astillera en la región del Meta, fortaleciendo a 10 empresas de la zona en habilidades gerenciales, estrategias de asociación y buenas prácticas en materia de construcción y reparación de embarcaciones.

El 2013 fue sin duda un periodo de interrelaciones donde predominó la construcción colectiva de Conocimiento con entes nacionales e internacionales, se desarrolló un proceso de transferencia tecnológica con la empresa STX de Corea del Sur en el marco del proyecto de la construcción de la Coastal Patrol Vessel – CPV, se consolidó el proyecto del diseño y construcción de la Patrullera Amazónica con Engeprom de Brasil. Nos hace sentir orgullosos, al poder ver reflejados en nuestros productos y servicios el resultado de un esfuerzo colectivo a través del cual contribuimos a la generación de capacidades de una Industria Estratégica para la seguridad del país.

Esta edición se estructura con artículos relacionados con modelación isotrópica, estabilidad, optimización del diseño de casco, evaluación de ruidos y vibraciones, medidas multi-influencia para caracterizar firma global, modelo de diseño de un canal de ensayos hidrodinámicos, y desde la perspectiva de competitividad, un artículo relacionado con el plan de acción para reestructuración de las tecnologías en un astillero.

Agradecemos a nuestros autores y lectores por su interés, compromiso y aporte que nos permite la entrega de trabajos de calidad para la difusión y generación de nuevo conocimiento en la comunidad científica del diseño, la arquitectura e ingeniería naval.

Capitán de Navío OSCAR DARÍO TASCÓN MUÑOZ
Editor Revista Ciencia y Tecnología de Buques

Isotropic Modeling of a Composite Panel of the Stern of a Fiberglass Boat Propelled by an Outboard Motor

Modelación isotrópica de los esfuerzos del espejo de un bote de fibra impulsado por motor fuera de borda

Patrick Townsend Valencia [1]

Abstract

We performed a theoretical and experimental study to define the best way to model the finite element sandwich structure aft of a fiberglass boat less than 15 meters in length, using an isotropic linear mathematical model that fits anisotropic material conditions. This is done by defining the properties of the ship's fiberglass resin structure, which is representative of the influence of the forces acting during the glide on the geometry of the entire vessel. Formulation of the Finite Elements Method is presented, which works on the mathematical model to define the limitations of the results obtained. Isotropic material adjustment is calculated using Halpin-Tsai laws, developing its mathematical formulation for restrictions of modulus data entered as the finite element program experimentally calculated for each of the sandwich materials. The best-fit mathematical presentation to the modulus of the composite tool justifies the calculation thereof.

Key words: Sandwich, module, finite, composite resin

Resumen

Se presenta el estudio teórico para modelar en elementos finitos el sándwich del espejo de un bote de fibra menor a 15 metros de eslora, empleando un modelo matemático lineal isotrópico que se ajuste a las condiciones anisotrópicas del material. Esto se logra al definir las propiedades de la resina de fibra de vidrio empleada que sea representativo de la influencia de las fuerzas actuantes durante el planeo sobre la geometría de toda la embarcación. Se presenta la formulación del Método de Elementos Finitos sobre la cual trabaja el modelo matemático para definir las limitaciones de los resultados obtenidos. El ajuste isotrópico del material se calcula empleando las leyes de Halpin-Tsai, desarrollando su formulación matemática para conocer las restricciones del módulo de elasticidad que se ingresa como dato al programa de Elementos Finitos el cual es calculado de forma experimental para cada uno de los materiales del sándwich del espejo.

Palabras claves: sándwich, módulo, finitos, compuesto, resina

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Introduction

Composite materials applied to structures of vessels have undergone notable technological development in recent years. Their application is common in sports, recreational, or boats for military use.

The favorable weight/structural resistance ratio makes composite materials an increasingly attractive alternative when requiring a low-weight hull able to withstand rigorous loads from the marine environment.

In numerous repairs to sports vessels less than 15m in length, I have noted that cracks appear in the outboard motor coupling zone. This zone is known as the stern; it withstands the thrust or the motor's HP strength to make the vessel glide at the design speed and it corresponds to the transversal part or stern bulkhead or extreme of the vessel where the outboard motors of small gliding boats are installed.

The motors are placed on double steel plates on each side on the stern, for the pressure to distribute uniform stress on the composite material.

Fig. 1. View of boat stern (Photograph by Patrick Townsend, 07 April 2012)

Assembly configurations exist, combining fiber cloth with resin and marine plywood with which the composite panel or sandwich is put together, obtaining the necessary rigidity for the work expected.

The sum of all the hydrodynamic loads, the action of the wind over the bow, the vessel's effective thrust through the action of the outboard motor, and other stress on the plate permit the vessel to glide at high speeds.

Regarding the material, the most-often used resin for this type of yacht construction is that of unsaturated polyester. When a catalyst is added to the resin, this combination creates a series of free radicals that cause the resin's chemical elements to bond, forming an increasingly dense network that, in a first phase, makes it turn to gel and, finally, harden into a vitreous state. With this, the composite material becomes a crosslinked, thermostable polymer, without rotation or movement of chains and which when solidified acquires a firm rigidity that gives it structural characteristics

One of the most interesting mechanical properties of the unsaturated polyester resin is that of working under tension or compression only in the plastic or linear zone, going directly to the rupture or failure, without passing through the plastic zone.

This material is anisotropic and the model presented will fit mathematically its properties to isotropic properties to obtain values of elasticity modulus equivalent to the composites.

The Finite Elements method is a powerful tool in the analysis of structures, and this work will develop the best-fit model to study the appearance of cracks on sterns through motor action.

Materials and Methods

To calculate the loads acting on the boat at the moment of gliding when it develops its maximum power, it is necessary to define the acting forces, corresponding to the air pressure on the boat, which is due to the speed, resistance of the vessel to advancing, which is a cause of the vessel's velocity, V, and the hydrodynamic pressure. During gliding and thrust, which is the force exerted by the propeller, transmitted to the vessel's hull through the propulsion system, in such a manner that the force acting on the ship's stern is the sum of all forces, given their action-reaction condition.

Regarding the material, the most-often used resin for the construction of these types of yachts is unsaturated polyester. When a catalyst is added to the resin, this combination creates a series of free radicals that cause the resin's chemical elements to bond, forming an increasingly dense network that, in a first phase, makes it turn to gel and, finally, harden into a vitreous state. With this, the composite material becomes a crosslinked, thermostable polymer, without rotation or movement of chains and which when solidified acquires a firm rigidity that gives it structural characteristics.

When using non-fiber nuclei, such as plywood materials, marine plywood, wood, cardboard, and others susceptible to making a structural contribution, such laminate is denominated sandwich.

Its anisotropy depends on that formed at the moment of manufacture. This means that in a sandwich panel used for yachts, its properties may depend on the orientation of the fibers and of the nuclei used with orientations only at 0 and 90°, depending on the assembly order assigning orthotropic properties to the material.

To learn how the vessel's stern works against acting loads, the Finite Elements method (FEM) will be used to estimate the level of stress concentration upon the stern under acting loads. The use of the SAP2000 software, generally, has mathematical and structural limitations, which must be considered to have a good approximation of the results, given that the membrane and plate effects do not interact in a second order.

In its calculations, this program considers for a discretized surface, $\underline{S}$, that the strain energy generated by the unit of volume, $\underline{U}^*$, is governed by the expression in terms of the unitary strain, $\underline{\sigma}$, and the stress, $\underline{\varepsilon}$.

$$U^* =+\frac{\sigma\varepsilon}{2} \qquad \sigma_0\,\varepsilon \tag{1}$$

Given that the program will consider that the sandwich material is isotropic, that is, solid throughout its entire thickness, the ratio of the matrix of stress elements $\{\sigma\}$ and of the unitary strain $\{\varepsilon\}$ would be governed by equation 2.

$$\begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ \varepsilon_{xy} \\ \varepsilon_{xz} \\ \varepsilon_{yz} \end{Bmatrix} = \begin{bmatrix} C_{11} & C_{12} & C_{13} & 0 & 0 & 0 \\ C_{21} & C_{22} & C_{23} & 0 & 0 & 0 \\ C_{31} & C_{32} & C_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & C_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & C_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & C_{66} \end{bmatrix} \begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ \tau_{xy} \\ \tau_{xz} \\ \tau_{yz} \end{Bmatrix} \tag{2}$$

By definition, the elastic modulus will be the following in terms of the C_{ik} coefficients indicated in expressions (3) and which correspond to the main directions of the elasticity modulus

$$C_{11} = \frac{1}{E_x}\ , \quad C_{12} = \frac{\nu_{xy}}{E_x}\ , \quad C_{22} = \frac{1}{E_y} \tag{3}$$

By replacing equation 2 in 1, the strain energy in the volume, $\underline{V}$, domain of finite elements, equation 4 is obtained.

$$\begin{aligned} &\int_V^{\square} \left(\{du\}^T\{\bar{p}\} + \frac{1}{2}\{du\}^T\{d\bar{p}\} \right) dV \\ &+ \int_S^{\square} \left(\{du\}^T\{\bar{q}\} + \frac{1}{2}\{du\}^T\{d\bar{q}\} \right) dS = \\ &= \int_V^{\square} \left(\{d\varepsilon\}^T(\{\sigma\} + \{\sigma_0\}) + \frac{1}{2}\{d\varepsilon\}^T\{d\sigma\} \right) dV \end{aligned} \tag{4}$$

Upon solving, being the first-order analysis, dual differential products like $du\ d\bar{p}$, $du\ d\bar{q}$, and $d\varepsilon\ d\sigma$ are neglected so that equation 4 remains as:

$$\begin{aligned} &\int_V^{\square} \{du\}^T\{\bar{p}\}\, dV + \int_S^{\square} \{du\}^T\{\bar{q}\}\, dS \\ &= \int_V^{\square} \left(\{d\varepsilon\}^T(\{\sigma\} + \{\sigma_0\}) dV \right. \end{aligned} \tag{5}$$

This equation is the principle of the virtual strains and assumes that stress remains constant during deformation. The Rayleigh-Ritz method solution is applied to this strain model by using the condition of the minimum total potential energy[1] .

[1] Basic theory of the Finite Elements method presented by C. Zienkiewics in the text "The finite element method", McGraw Hill company, Third Edition, Berkhsire, England, 2007. And by H. Lima in "Finite elements method Em Análise de estruturas", Universidad de Sao Paulo, USP, Sao Paulo, Brasil. 2002.

Based on this, consider for the n layers of the laminate general expression (6), which relates the membrane stress (tensions), flexion (due to moments), and shear as $\hat{\sigma}_{mem}$, $\hat{\sigma}_{mto}$, $\hat{\tau}$, respectively, which are obtained through integration, z, over thickness, t, at a distance, k, from the mean surface of each layer.

$$\begin{Bmatrix} \hat{\sigma}_{mem} \\ \hat{\sigma}_{mto} \\ \hat{\tau} \end{Bmatrix} = \begin{bmatrix} E_{mem} & E_{mem\text{-}mto} & 0 \\ E_{nem\text{-}mto} & E_{mto} & 0 \\ 0 & 0 & G \end{bmatrix} \begin{Bmatrix} \hat{\varepsilon}_{mem} \\ \hat{\varepsilon}_{mto} \\ \hat{\gamma} \end{Bmatrix} \tag{6}$$

By solving, we obtain the ratios for the different stresses, which are expressed in equations 7, 8, and 9. [2]:

$$\hat{\sigma}_{mem} = \sum_{k=1}^{n} \int_{t_{k-1}}^{t_{k+1}} [\sigma_x \quad \sigma_y \quad \tau_{xy}]_k^T \, dz \tag{7}$$

$$\hat{\sigma}_{mem} = \sum_{k=1}^{n} \int_{t_{k-1}}^{t_{k+1}} z\,[\sigma_x \quad \sigma_y \quad \tau_{xy}]_k^T \, dz \tag{8}$$

$$\hat{\tau} = \sum_{k=1}^{n} \int_{t_{k-1}}^{t_{k+1}} z\,[\tau_{xz} \quad \tau_{yz}]_k^T \, dz \tag{9}$$

By relating the previous expressions to the Halpin-Tsai[45] equation, which groups the modules in terms of ξ so that the behavior of the elasticity modulus of a composite material depends on the characteristics of form, packing regularity, load conditions, and aspect of the fibers, as noted in equation 10. The value of η depends on the Poisson constant and predicts the modulus according to the orientation of the fibers, f, and the volume of resin, r, defined by ϕ_f.

$$E_{composite} = \frac{E_r\,(1+\zeta\eta\phi_f)}{1-\zeta\eta\phi_f} \tag{10}$$

Where:

$$\eta = \frac{\left(\frac{E_f}{E_r} - 1\right)}{\left(\frac{E_f}{E_r} + \zeta\right)} \tag{11}$$

So that if:

$\frac{E_f}{E_r} = 1$, *it implies that* $\eta = 0$ *is a homogeneous medium.*

$\frac{E_f}{E_r} \to \infty$, *it implies that* $\eta = 1$ *has rigid inclusions.*

$\frac{E_f}{E} \to 0$, *it implies that* $\eta = -\frac{1}{\zeta}$ *which has holes.*

Rigidity is calculated in both directions with the generalized expression corresponding to the ratio between the elasticity modulus Poisson's coefficient from which equation 12 is deducted, which fits that developed with Halpin-Tsai for rigidity.

$$G_{composite} = \frac{G_r\left(\frac{G_f}{G_r} + \xi + \phi_f\left(\frac{G_f}{G_r} - 1\right)\right)}{\frac{G_f}{G_r} + \xi - \phi_f\left(\frac{G_f}{G_r} - 1\right)} \tag{12}$$

This development permits fitting the sandwich's material anisotropy to an isotropic plate material.

Given that the membrane strength, according to the equations to be used in the Finite Elements method is linear, that is, the membrane strength, E_{mem}, of the elements does not produce flexion, E_{mto}, and vice versa, so that the value of $E_{nem\text{-}mto}=0$, leaves equation 13.

$$\begin{Bmatrix} \hat{\sigma}_{mem} \\ \hat{\sigma}_{mto} \\ \hat{\tau} \end{Bmatrix} = \begin{bmatrix} E_{mem} & 0 & 0 \\ 0 & E_{mto} & 0 \\ 0 & 0 & G \end{bmatrix} \begin{Bmatrix} \hat{\varepsilon}_{mem} \\ \hat{\varepsilon}_{mto} \\ \hat{\gamma} \end{Bmatrix} \tag{13}$$

Where:

$$E_{mem} = \sum_{k=1}^{n} \int_{h_{k-1}}^{h_{k+1}} E_k^T \, dz \tag{14}$$

For the type of analysis, it may be concluded that $E_{mem}=E_{mto}=E_T$, where E_T is the total elasticity modulus of the sandwich. An approximate solution with series of equation 14 is that presented in equation 15[3].

[2] Theory presented by E. Car, S. Oller, E. Oñate on page 47 of "Tratamiento numérico de los materiales compuestos".

[3] J Tegedor, in the text "Construcción de buques de pesca en poliéster reforzado con fibra de vidrio" from Librería Naútica, Madrid, 2001 presents this solution as a good mathematical approach by using "superposition of various layers". Page 66.

$$E_T = \sum_{i=1}^{n} E_i t_i \Big/ \sum_{i=1}^{n} t_i \quad (15)$$

This permits calculating the linear elasticity modulus through the sum of all the sandwich modules, and apply it to the consistent finite elements model, given that the material's anisotropic properties were fitted to isotropic properties.

The value of the elasticity modulus is, thus, obtained with the uniaxial laboratory traction test, governed by the ASTM norm to standardize the dimensions of samples and test conditions like ambient temperature and rate of deformation (strain) applied by the team. Samples were brought to rupture, as noted in Fig. 2. The modulus will not have a constant value, given that the chains of the macromolecule reorganize due to the stress effect being that they are crosslinked polymers.

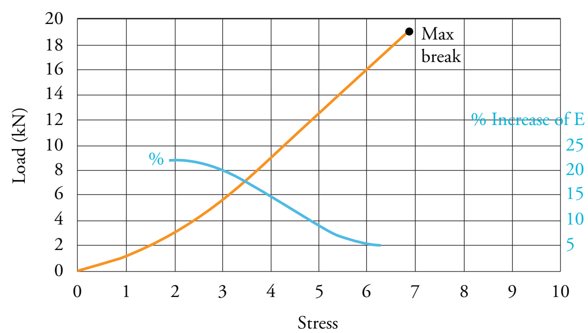
Fig. 2. Sandwich material after traction

Results

Elasticity modulus increases with traction because the chains of the macromolecule increase their resistance to it. This greater rigidity is propitiated by the growing presence of fibers on the plastic layer, which restricts the reorganization of the macromolecules.

Fig. 3 represents the already discretized bow-stern Finite Elements model of a 15-m long speed boat made from the previously described materials. The conditions of symmetry and the necessary restrictions for a Finite Elements model have been applied, plus the values of the motor loads previously calculated on the stern. Also, note the model's deformation profile.

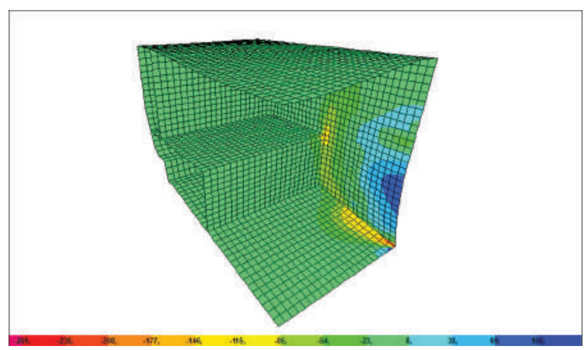
Fig. 3. Variation of E with tension



The ends of the plate produce peaks on the stress, with the minimum bending stress being greater on the lower tip of the plate. If the installation of the plate exceeds the stress limit of the sandwich, the rupture should start at this lower point.

Fig. 4 analyzes the maximum bending stress on the model, having his level.

Fig. 4. Results of stress in kg/cm² of the model



This can be explained by the model configuration, given that the transversal bottom surface with its curvature, union with the keel and side reinforcement, tends to be more rigid and bend less. This is why it is noted in the figure that the other surfaces withstand less stress on the Shell elements through the action of the stern load upon them. This is the case of the deck, which has a given flexibility because of its thickness and configuration, which does not contribute significantly to the model's rigidity, as achieved by the stern-bottom union.

We must highlight the existence of a stress peak toward the center of the model on the keel's symmetrical attachment.

Conclusions

- The results are reliable because the analysis prior to inputting the data onto the program, associated all the considerations of mathematical anisotropic approximations to isotropic material, permitting for the solution to be consistent.

- Due to the numerical results, the failure produced in boats constructed must be related to the phenomenon of material fatigue. Because of the resin action, the sandwich is completely reconstituted in its length, as observed in the tests; this only gives way to cracks given that the material returns to its initial position without it continuing to crumble.

- The effect indicated in g) is, then, a big advantage of using the fiber on other materials in naval construction for these types of vessels.

- Because the stress level remains within the experimental linear range or elastic zone, the analysis with the SAP model yields a good representation of what actually happens in the boat, given that after applying the load the structure recovers to its original position.

References

A MIRAVETE, "Materiales Compuestos I". Editorial Reverté, Barcelona España, 2007.

A. HORTA, "Macromoléculas". Librería UNED, Tercera Edición. 2002.

ALEJANDRO BESEDNJAK, "Materiales Compuestos: procesos de fabricación de embarcaciones". Books. Google.com. 1995.

ASTM, "Standard Test Method for Tensile properties of polymer Matrix Composite Materials". American Society for Testing and Material. USA, 2002.

B. MAC DONALD, "Practical Stress Analysis with Finite Elements". Page 215. Dublin, Ireland. 2007.

C. MARISCAL, "Formulación y Evaluación de Proyectos". Escuela Superior Politécnica del Litoral, ESPOL, 109 páginas. Guayaquil, Ecuador. 2001.

C. KASSAPOGLOU, "Desing and Analysis of Composite Structures". First Edition, The Atrium & Southern Gate. London, UK. 2010.

D. CHUNG, "University Notes: Science and Applications: Composite Material". Second Edition, British Library Cataloguing in Publication Data. State University of New York, Buffalo, USA. 2010.

D. FELDMAN, A. BARBALAT, "Synthetic polymers: technology, properties, applications". London, United Kingdom.1996.

E. CAR, S. OLLER, E. OÑATE, "Tratamiento numérico de los materiales compuestos". Centro Internacional de Métodos Numérico de Ingeniería. Madrid, España. 2000.

F. BILLMAYER, "Science of Polymers". New York, USA 1975.

F. SINGER, A. PYTEL, "Resistencia de Materiales". Impreso en México Tercera edición, 560 páginas. D.F. México. 1982.

G. GILI, "Arquitectura Naval: Teoría del buque y sus aplicaciones". 574 páginas. España 1956.

Stability of Ships with a Single Stranding Point

Estabilidad de embarcaciones con varamiento de punto único

Paula C. de Sousa Bastos [1]
Marta C. Tapia Reyes [2]

Abstract

During rescue operations of stranded vessels, it is essential to make immediate and reliable decisions to optimize the successful salvage potential and minimize risks of environmental damages and cost impacts. Pursuant to this scenario, the need arises for a numerical tool, which can more accurately forecast the stability conditions experienced by a vessel after running aground and help in the refloating operations of the unit. This study seeks to develop an adequate calculus systematization, which provides analytical capabilities for operational situations in case of stranding, thereby, supporting the decision-making process in these risk situations.

Key words: stranding, single point, stability, algorithm, refloating

Resumen

Durante operaciones de rescate de embarcaciones varadas, es esencial tomar decisiones inmediatas y confiables para optimizar el potencial de salvamento exitoso y minimizar el riesgo de daños ambientales e impacto de costos. De acuerdo con este escenario, surge la necesidad para una herramienta numérica, que pueda predecir de manera más precisa las condiciones de estabilidad que esté experimentando la embarcación luego de encallar y ayudar en las operaciones de reflotación de la unidad. Este estudio busca desarrollar una adecuada sistematización de cálculo, que brinde capacidades analíticas para situaciones operativas en caso de encallar, así, apoyar el proceso de toma de decisiones durante estas situaciones de riesgo.

Palabras claves: varamiento, punto único, estabilidad, algoritmo, reflotación

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Introduction

Damage sustained by vessels generates numerous losses, those caused by maritime accidents to ships or their cargo, as well as the costs to repair these damages, which include the vessel's integrity and the environmental context. With the aim to minimize these expenses, it is mandatory to ensure agility and dependability of the decisions made during risk situations, such as the vessel's stranding, a quick check of compliance with requirements for stability and the maximum allowed effort for a given loading condition.

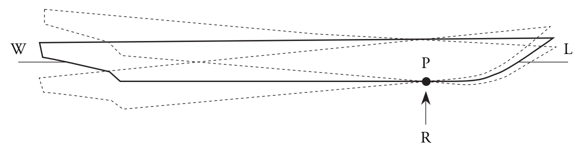
To achieve that, numerical tools should be used, which are capable of accurately forecasting the impact of the action plan to be followed. Even if the operational expertise enables rescuing the vessel, it is important to have a duly validated support tool in place.

Within this context, this paper seeks to develop a calculus algorithm that enables outputting replies about a vessel's stability in case it runs aground, taking into account the circumstances endured by the vessel, which acts as an aid to decision-making and maximizes the reliability of refloating operations. The vessel's integrity is, thus, ensured along with environmental conservation.

Stranding

A stranded ship does not respond in the same way as when it is floating freely. Part of its weight is supported by the seabed, and the ground produces a reaction force. Under these different conditions, the static stability behavior undergoes changes, making the ship more vulnerable to damage.

Fig. 1. Ship supported by single point

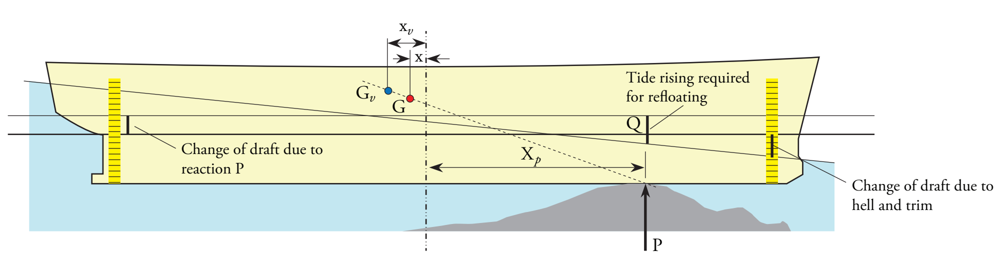


This paper develops a study about the theoretical analysis of static stability in a stranded ship resting on a single support point.

In a stranded ship resting on a single support point, the reaction is applied on point 'P' and the ship may rotate around the three coordinate axes passing through this point. When this happens, there are displacement variations and, if the weight remains unchanged, the reaction force will also vary. However, the weight added or removed from the vertical axis going through the support point does not cause rotation because it does not cause moment; therefore, it does not cause any variations in displacement. In this case, the variation in reaction force equals the variation in weight. When it is possible to take the draft measurements while the ship is stranded, both buoyancy and its center may be calculated by integrating the submerged part of the hull. The center of gravity may be defined by the load conditions. The reaction point may be determined by the balanced weight, displacement, and moment of the reaction force.

After stranding, the position of the center of gravity does not change. However, under static balance conditions, a partially floating (stranded) ship behaves like a ship from which a weight had been removed, equaling the ground reaction at a given position. That is why the virtual change in the center of gravity is considered.

Fig. 2. Variation of trim due to stranding



Where:

G = Position of the center of gravity before stranding
G_v = Virtual elevation of the center of gravity
P = Application point for the reaction force
x = Distance between the center of gravity, G, and the midship section
x_v = Distance between the application point for the reaction force and the midship section
Q = Tide height necessary to refloat the ship

Modeling the Stranding Event

Seldom are the conditions prevailing during a stranding completely identifiable, which makes rescuing a ship a critical procedure.

Some methods exist to refloat grounded ships, but none is absolutely exact due to the different circumstances to which the vessel may be subjected, given that no circumstance is absolutely predictable by numerical simulations.

In the case of stranding events, one should consider the ship's characteristics, loading conditions, the ship's fore and aft drafts, balance and stability, conditions, height of the waterline, type of ground, and tide range in the location where the incident took place.

This analysis will enable establishing the reaction force and the position where the latter is applied to the ship. As soon as this information is obtained, one may check the vessel's stability by calculating the height of the center of gravity and the metacentric height.

The effects of tide range and weight movement onboard the ship on stability will also be examined, as well as the ship refloating process.

All appraisals will be verified through a stranding simulation to validate the study.

Reaction force (R)

The value of the reaction force (R) is obtained by the difference between the ship's displacement before and after the stranding has taken place, and can be calculated after determining the fore and aft drafts in a single point, without any damage to the bottom compartments. After reading the value of these drafts, it is possible to establish the equivalent draft. Obtaining the equivalent draft for a vessel that has run aground with a maximum degree of accuracy entails some complexity because the ship may be in a state that includes some heel and/or trim. However, it is important to establish this parameter because it is necessary to calculate the ground reaction force, which is applied to the grounded ship.

The method suggested in the *U.S. Navy Salvage Manual – Volume 1 – Strandings and Harbor Clearance* [1] has been employed to calculate the equivalent draft (Eq. 1), where the draft is established by the fore, midship, and aft draft measurements on both sides. The average between the average port and starboard drafts indicates the draft known as T_{m1}.

$$T_{eq} = \frac{Tm_{zm} + T_{n}}{2} \tag{1}$$

Where:

T_{mn}= average of midship drafts taken on the port and starboard sides
T_{m2}= average of T_{m1} and T_{mn} drafts

This equivalent draft value is very close to the actual value. The error between the calculated value and the actual value may be due to the ship's hogging or sagging.

Position of the Reaction Force

The coordinates of the point where the reaction force is applied may be calculated based on displacement and the state of the ship after having run aground. The virtual movement of the vessel's center of gravity is also taken into account.

Longitudinal position of the reaction force (LCR)
In order to establish the longitudinal position where the reaction force (LCR) is applied, one employs:

$$LCR = \frac{\Delta*[(LCG*\cos\beta)+(KG*\text{sen}\beta)] \\ -[(LCB_T*\cos\beta)+(KB_1*\text{sen}\beta)]}{2} \quad (2)$$

Where:

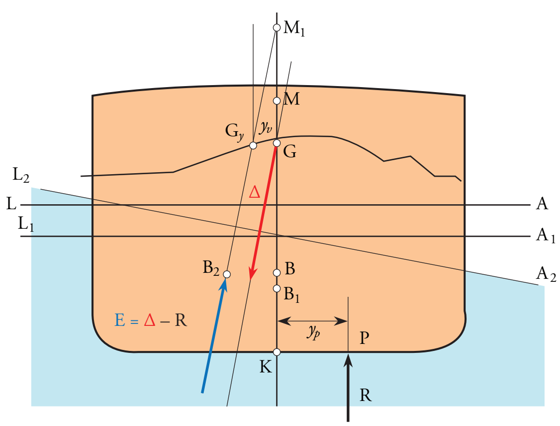
Δ = Ship's displacement before running aground [ton]
LCG = Longitudinal position of the center of gravity before running aground [m]
β = Trim angle [°]
KG = Vertical position of the center of gravity before running aground [m]
LCB= Longitudinal center of buoyancy [m]
LCB_T = LCB position after running aground corrected for trim [m]
KB_1 = Vertical position of the center of buoyancy after running aground [m]

The KB_1 values are obtained by the hydrostatic curves corresponding to the equivalent draft after running aground.

Transversal position of the reaction force (TCR)

The effect of the ground reaction away from the diametral plane is equivalent to a virtual movement of the center of gravity transversally equal to y_v.

Fig. 3. Ground reaction applied outside of center line



M = Metacenter height before running aground
$M1$= Metacenter height variation after running aground
G = Position of the center of gravity before running aground
G_v= Virtual elevation of the center of gravity
B = Position of the center of buoyancy before running aground
$B1$= Position of the center of buoyancy after running aground, in case there is no heel
$B2$= Position of the center of buoyancy after running aground, in case there is some heel
P = Application point of the reaction force
y_v= Distance between the virtual position of the center of gravity, Gv, and the center line
y_p= Distance between the application point of the reaction force and the center line. If the force of the application takes place along the central plane, yp equals zero.
Δ = Ship's displacement before running aground
R = Reaction force
E = Buoyancy
$\overline{LA}$ = Waterline before running aground
$\overline{L_1A_1}$ = Waterline after running aground, in case there is no heel
$\overline{L_2A_2}$ = Waterline after running aground, in case there is some heel

Applying the reaction force to a different point along the center line causes a heel condition. The heel angle may be measured by the draft marks on port (T_{BB}) and starboard (T_{BE}), which correspond to the longitudinal center of flotation (LCF).

$$\theta = 90° - \cos^{-1}\left(\frac{|T_{BB} - T_{BE}|}{B}\right) \quad (3)$$

In order to establish the transversal position where the reaction force is applied, one employs:

$$TCR = \frac{\Delta*[(TCG*\cos\theta)+(KG*\text{sen}\theta)] \\ -\Delta_1*[(G_vM_1*\text{sen}\theta]}{R*\cos\theta} \quad (4)$$

Where:

TCG = Transversal position of the center of gravity before running aground [m]
θ = Heel angle [°]
KG = Vertical position of the center of gravity before running aground [m]
$\Delta 1$= Ship's displacement after running aground [ton]

$GvM1$ = Virtual variation in the metacentric height after running aground [m]

Vertical position of the reaction force (VCR)
The reaction force is applied onto the ship's bottom. Taking the baseline as reference, the vertical position where the reaction force (VCR) is applied equals zero.

Stability

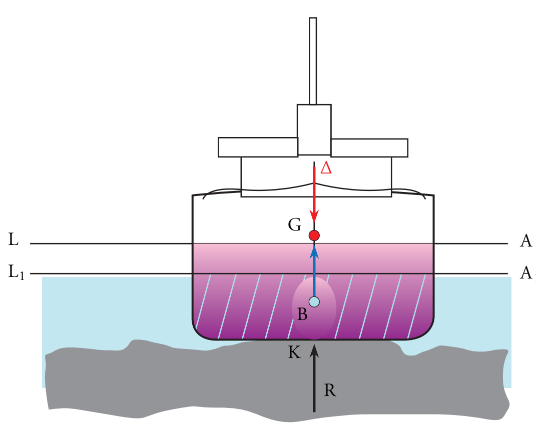
If a ship is stranded on a point, it can tilt freely in one or both directions, and may run into danger. A wide tide range may cause changes in the ground reaction, which may compound the stability issue.

To perform the stability analysis, it is necessary to establish:

- The effective or virtual increase in the center of gravity height (*KG/VCG*)
- The variation in metacentric height (*GM*) after the ship has run aground

Center of gravity height (KG1)
Changes to the position of the center of gravity are virtual changes; the center of gravity does not shift given that the weight remains unchanged. However, the ship behaves as if this position had shifted. There are two options to calculate with the reaction force applied either to the central plane or outside this central plane.

Fig. 4. Ground reaction applied in diametral plane



Reaction force applied to central plane
If the reaction force is applied to the central plane, the vertical position of the center of gravity may be established by the summation of static moments in relation to *K*.

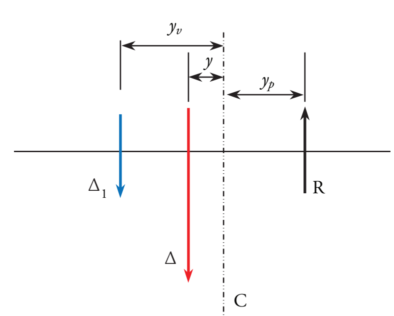
$$KG_1 = \frac{KG * \Delta}{\Delta - R} \tag{5}$$

Reaction force applied outside the central plane
In this case, the calculation of the center of gravity height depends on the heel angle. When the ship's heel is high (> 8°), the approximation to *GZ* cannot be taken into consideration; therefore, one considers two possibilities for low heel and high heel values.

Low Heel

If the reaction force is not applied to the central plane, the ship's heel will depend on the heeling moment. Considering a case where the ship's heel is low (8°), the starboard distances are positive and the port distances are negative, the resulting moment equals the sum of the component moments.

Fig. 5. Consequence of ground reaction application outside of central plane



$$y_p = \frac{\Delta * y - \Delta_1 * y_v}{\Delta - \Delta_1} \tag{6}$$

If a stable ship remains balanced while heeling, a righting moment attempts to return it to the balanced position without heel. In the case of stable ships with usual shapes, the metacenter position for angles up to 8° may be considered constant. Under these circumstances, one may consider the righting arm, *GZ*, as being:

$$GZ = GM * \text{sen}\,\theta \tag{7}$$

The restoration moment for any inclination angle represents the vessel's ability to recover its original position. This value is calculated when the vessel is on quiet waters and momentarily at rest, *i.e.*, acceleration forces due to motion may be dismissed. Thus, the recovery conjoint (*CR*) is obtained by:

$$CR = \Delta_1 * G_v M_1 * \text{sen}\,\theta \tag{8}$$

$$G_v M_1 = KM_1 - KG_1 \tag{9}$$

For the ship to retain its heel angle there must be a moment equal to the module and which is opposite of the recovery conjoint. This other conjoint is called heeling conjoint (*CE*).

$$CE = (\Delta - \Delta_1) * y_p * \cos\theta \tag{10}$$

$$CR = CE \tag{11}$$

$$\Delta_1 * G_v M_1 * \text{sen}\,\theta = (\Delta - \Delta_1) * y_p * \cos\theta \tag{12}$$

$$tg\theta = \frac{(\Delta - \Delta_1) * y_p}{\Delta_1 * G_v M_1} \tag{13}$$

$$tg\theta = \frac{\Delta * y - \Delta_1 * y_v}{\Delta_1 * G_v M_1} \tag{14}$$

$$KG_1 = KM_1 - \left(\frac{\Delta * y - \Delta_1 * y_v}{\Delta_1 * tg\theta}\right) \tag{15}$$

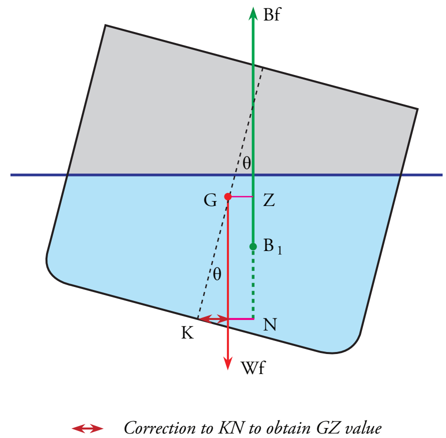
High heel

When the ship's heel angle is above 8°, the approximation to *GZ* is no longer valid because the metacenter does not have a 'fixed position'; the variation of the submerged shape is higher, so that the metacenter also varies according to the transversal inclination angle.

When a ship tilts, the center of buoyancy (*B*) moves constantly and its transversal position depends on the heel angle and on the ship's displacement.

GZ depends, foremost, on the ship's *KG*. Because *G* may take up numerous positions, it is convenient to consider the value of *GZ*, which would exist if *G* were on the keel (*KN*) and then correct it to *G*'s actual height.

Fig. 6. Determination of righting arm (GZ) to high heel



$$\text{sen}\,\theta = \frac{CorreçãoKN}{KG} \tag{16}$$

$$GZ = KN - (KG_1 * \text{sen}\,\theta) \tag{17}$$

$$\text{sen}\,\theta = \frac{KN}{KM_1} \tag{18}$$

$$KN = KM_1 * \text{sen}\,\theta \tag{19}$$

Therefore,

$$GZ = KM_1 * \text{sen}\,\theta - (KG_1 * \text{sen}\,\theta) \tag{20}$$

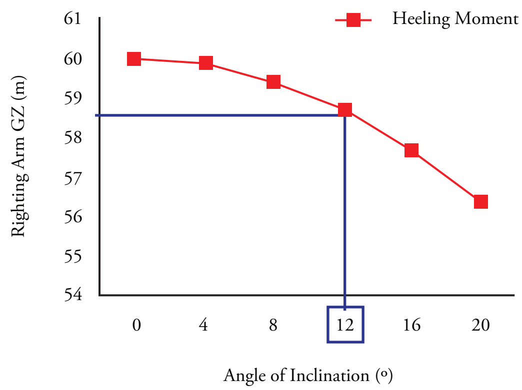
The heel angle and *KM*1 are known. The latter is drawn from the hydrostatic table for displacement after the ship's stranding.

The stability curve, which represents the heeling moment, is drawn and it is obtained by the following equation:

$$CE = (\Delta - \Delta_1) * y_p * \cos\theta \tag{21}$$

The intersection of the curve with the heel angle with which the ship partially floats defines the vessel's *GZ* under these circumstances. If *GZ* is known, one may be able to establish KG_1.

Fig. 7. Determination of GZ by Stability Curves



Virtual variation in the metacentric height (GvM1)

After having run aground, the ship has a new waterline, which is different from the former (prior to stranding).

The KM_1 value may be determined by the hydrostatic curve through the actual draft after stranding. The new metacentric height may be determined by:

$$G_vM_1 = KM_1 - KG_1 \tag{22}$$

Effect of changing weights onboard

Weight control is widely used to reduce ground reaction in stranding situations. One should always pay attention to the effects of this procedure on stability and balance, for if the position of the vessel's center of gravity is not controlled; the ship may become unstable or be brought into danger when it is refloated.

The most evident effect when shifting weights onboard a ship is the variation in the ship's center of gravity and displacement, which may cause changes to the bottom shapes, involving shifts in the vertical and longitudinal position of the center of buoyancy, possible changes in the transversal position of the center of buoyancy, and changes in the longitudinal and transversal metacentric radii. Therefore, there will be changes in the transversal and longitudinal metacentric heights. Shifting weights in the ship may cause variations in the heel and in the drafts and, as a result, there will be changes to the vessel's trim.

Variation in the ship's center of gravity

Variation in the ship's longitudinal center of gravity (LCG_2). Variation in the LCG will occur according to the variation in the center of gravity of the weight removed.

$$LCG_2 = \frac{\Delta_1 * LCG_1 - w * lcg}{\Delta_1 - w} \tag{23}$$

Where:

LCG_1 = Position of the ship's longitudinal center of gravity before weight removal [m]
w = Weight removed [ton]
lcg = Longitudinal position of the center of gravity of the weight removed [m]

Variation in the ship's transversal center of gravity (TCG_2). Variation in the ship's transversal center of gravity due to variations in weight, calculated for the weight moment, is determined by:

$$TCG_2 = \frac{\Delta_1 * TCG_1 - w * tcg}{\Delta_1 - w} \tag{24}$$

Where:

TCG_1= Position of the ship's transversal center of gravity before weight removal [m]
tcg = Transversal position of the center of gravity of the weight removed [m]

Variation in the ship's vertical center of gravity (VCG_2). Variation in the ship's vertical center of gravity (KG) due to variations in weight, calculated for the weight moment, is determined by:

$$VCG_2 = \frac{\Delta_1 * VCG_1 - w * vcg}{\Delta_1 - w} \tag{25}$$

Where:

VCG_1= Position of the ship's vertical center of gravity before weight removal [m]
vcg = Vertical position of the center of gravity of the weight removed [m]

Variation in the metacentric height (GMef)

The virtual changes to the center of gravity due to the ground reaction will decrease because the reaction force has also been reduced; therefore:

$$GG_{1reaction} = \frac{KG * R}{\Delta - R} \quad (26)$$

Regarding the removal of liquid cargo, if the compartment is partially filled, one should consider the effect of free surfaces in these tanks. The virtual change to the center of gravity due to the effect of free surfaces is expressed by:

$$GG_{1freesurf} = \frac{\Sigma I}{\nabla} = \frac{\Sigma\,(b^3 * l)}{12\nabla} \quad (27)$$

Where:

∇ = Volume displaced by the ship, taking into account the weight that has been removed [m³]

The effective KG value will be:

$$KG_{ef} = KG_1 + GG_{1reaction} + GG_{1freesurf} \quad (28)$$

Therefore, the effective metacentric height will be:

$$GM_{ef} = KG_2 - KG_{ef} \quad (29)$$

The KM_2 value is obtained through hydrostatic curves for the ship's new displacement corresponding to this state of removed weight.

Changes in trim

If the weight reduction onboard changes the ship's center of gravity, the center of buoyancy ceases to be in the same vertical line as the center of gravity. Hence, the vessel will display variations in trim, thus, displacing the center of buoyancy until the two occupy the same vertical line. This change in weight, w, introduces a trim moment. To quantify the resulting trim arising from the shift in the center of gravity one employs:

$$\text{Trim moment} = t * \frac{MT\,1}{100} \quad (30)$$

$$\text{Trim moment} = \Delta 2 * (LCG\,2 - LCB\,1) * \cos\beta \quad (31)$$

Where:

LCB_1 = Longitudinal center of buoyancy before weight removal [m]

Δ_2 = Vessel's displacement after weight removal [ton]

For small trim angles, one may consider cos β as being equal to 1. Therefore:

$$\Delta t * \frac{MT\,1}{100} = \Delta_2 * (LCG_2 - LCB_1) \quad (32)$$

$$\Delta t = \frac{\Delta_2 * (LCG_2 - LCB_1) * 100}{MT\,1} \quad (33)$$

To establish the trim associated with the ship's new state, one simply subtracts the value found for trim variation from the initial trim value:

$$t_2 = t_1 - \Delta t \quad (34)$$

Heel changes

Removing weight onboard may cause changes to the heel angle. The resulting transversal tilt may be deduced from the static stability curve.

In this case, as long as the stranded values for the center of gravity height and displacement are known, in an analogous fashion to the procedure described on the item on high heel values, a curve is drawn against the static stability curve, which corresponds to the heeling moment. The intersection between these two curves reveals the new heel angle in which the vessel may be floating freely or partially.

Fore and aft drafts

Changes to fore and aft drafts after the weight onboard has been shifted may be obtained by the ratio between buoyancy variation and the changes to the midship draft.

$$\delta T_m = \frac{\delta\Delta}{TPI} \quad (35)$$

$$\delta T_{ar} = \delta T_m * \frac{d_{ar} + d_m}{d_r} \quad (36)$$

$$\delta T_{av} = \delta T_m * \left(\frac{-d_{av}}{d_r} \right) \quad (37)$$

$$\delta B = w - \delta R \quad (38)$$

Where:

$\delta\Delta$ = Displacement variation [ton]
δT_m = Midship draft variation [m]
δT_{ar} = Aft draft variation [m]
δT_{av} = Fore draft variation [m]
d_{av} = Distance between LCF and PR [m]
d_r = Distance between LCF and the reaction center [m]
d_{ar} = Distance between the reaction center and PV [m]

Validation of the Proposed Model

To validate the proposed formulation, a barge has been modeled on the SSTAB and Hecsalv software applications, with the following characteristics:

- Length: 120 m
- Breadth: 24 m
- Depth: 12 m
- Total loading capacity: 20,930.40 ton
- Total ballast capacity: 10,184.40 ton

The adopted coordinate system complies with the following conventions:

- X: midship origin (fore, positive; aft, negative).
- Y: center line origin (port, positive; starboard, negative).
- Z: baseline origin (up, positive; down, negative).

Fig. 8. Modeled barge for validation –SSTAB

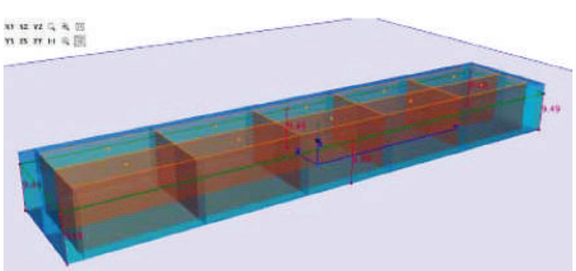
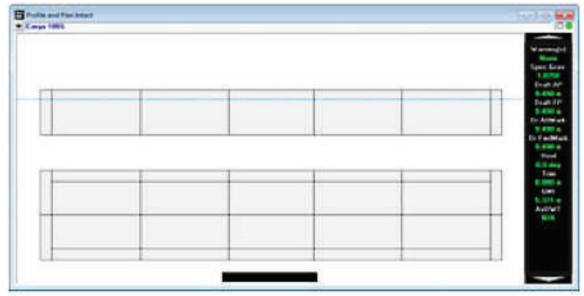


Fig. 9. Modeled barge for validation – HECSALV



Stranding Simulations

The loading conditions chosen for the simulation of the hypothetical stranding situations reflect a total loading capacity in which all cargo tanks are totally filled with a product whose specific mass is 0.85 ton/m³. Under these conditions, the barge has a parallel draft of 9.49 m.

Three hypothetical stranding situations were set up on SSTAB, the first through the application of an upward vertical force to simulate a stranding event causing trim; the second in which heel takes place; and the third in which concomitant longitudinal and transversal tilt takes place. The situations have the following characteristics:

Table 1. Hypothetical stranding situations

Condition	Ground Reaction (ton)	LCG (m)	TCG (m)	VCG (m)	Objective
1	2000	30	0	0	trim
2	2000	0	3	0	heel
3	2000	30	3	0	trim and heel

For each situation, SSTAB calculates the balance position, providing readings for six drafts in the following positions:

Table 2. Measurement of draft positions

Draft	X (m)	Y (m)
R BB	-60	12
R BE	-60	-12

MN BB	0	12
MN BE	0	-12
V BB	60	12
V BE	60	-12

The procedure to validate the proposed formulation consists of inputting draft and heel angle values onto the HECSALV software, which should estimate the value of the stranding force with its corresponding application position. The results obtained will be compared with the values inputted onto SSTAB.

Stranding situation 1

This situation has the following values for draft and heel angle:

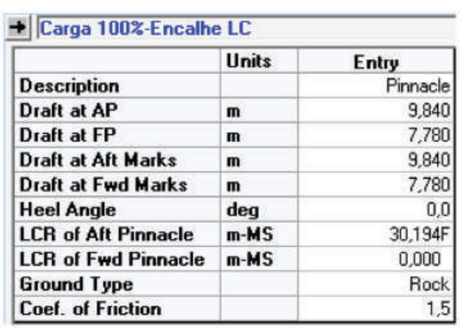
Tav = 7.78m

Tar = 9.84m

$\theta = 0°$

These input parameters are inputted onto HECSALV, as outlined on Fig. 10.

Fig. 10. Data input onto HECSALV software for stranding condition 1



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	Units	Entry
Description		Pinnacle
Draft at AP	m	9,840
Draft at FP	m	7,780
Draft at Aft Marks	m	9,840
Draft at Fwd Marks	m	7,780
Heel Angle	deg	0,0
LCR of Aft Pinnacle	m-MS	30,194F
LCR of Fwd Pinnacle	m-MS	0,000
Ground Type		Rock
Coef. of Friction		1,5

The following table shows a comparison between, on the one hand, the stranding force and the application position created by SSTAB and, on the other hand, the estimated values of the proposed formulation, as output by the HECSALV software.

A slight difference is noticed between the results obtained by the HECSALV and SSTAB software applications. This difference is due to the fact that HECSALV does not consider the virtual change in the center of gravity due to the application of the stranding force. In the proposed formulation, this consideration is made, bringing this result closer to the actual result.

Table 3. Comparison among results obtained by SSTAB and HECSALV software and the formulation

	Ground Reaction (ton)	LCR (m)	TCR (m)	VCR (m)
SSTAB	2,000	30.00	0	0
HECSALV	2,008	30.19	0	0
Formulation	2,000	30.12	0	0

Stranding situation 2

This situation has the following values for draft and heel angle:

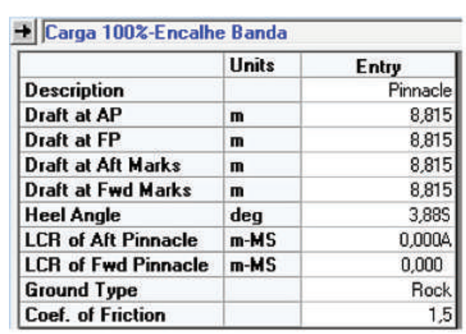
Table 4. Draft measurements

	Draft (m)	Mean Draft (m)
AR BB	8.00	8.82
AR BE	9.63	
MN BB	8.00	8.82
MN BE	9.63	
AV BB	8.00	8.82
AV BE	9.63	

$\theta = 3.876°$

These input parameters are inputted onto HECSALV, as outlined on Fig. 11.

Fig. 11. Data input onto HECSALV software for stranding condition 2



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	Units	Entry
Description		Pinnacle
Draft at AP	m	8,815
Draft at FP	m	8,815
Draft at Aft Marks	m	8,815
Draft at Fwd Marks	m	8,815
Heel Angle	deg	3,88S
LCR of Aft Pinnacle	m-MS	0,000A
LCR of Fwd Pinnacle	m-MS	0,000
Ground Type		Rock
Coef. of Friction		1,5

Table 5 shows a comparison between, on the one hand, the stranding force and the application position created by SSTAB and, on the other hand, the estimated values of the proposed formulation, as outputted by the HECSALV software.

Table 5. Comparison among results obtained by SSTAB and HECSALV software and the formulation

	Ground Reaction (ton)	LCR (m)	TCR (m)	VCR (m)
SSTAB	2,000	0	3.00	0
HECSALV	1,993	0	4.46	0
Formulation	2,000	0	2.67	0

As mentioned, the HECSALV software does not consider the virtual rise in the center of gravity due to the stranding event. In this case, where the reaction force is applied outside the center line, it is possible to see that this lack of consideration produces a larger difference than in the case involving longitudinal tilt.

Stranding situation 3

This situation has the following values for draft and heel angle:

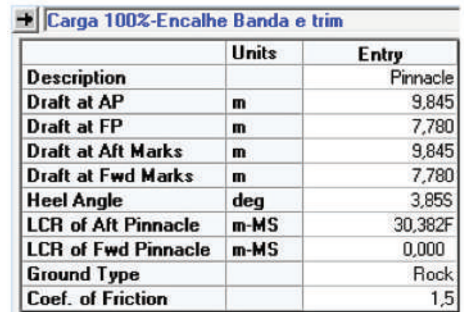
Table 6. Draft measurements

	Draft (m)	Mean Draft (m)
R BB	9.04	9.85
R BE	10.65	
MN BB	8.00	8.81
MN BE	9.62	
V BB	6.97	7.78
V BE	8.59	

$\theta = 3.853$

These input parameters are inputted onto HECSALV, as outlined on Figure 12.

Fig. 12. Data input onto HECSALV software for stranding condition 3


Carga 100%-Encalhe Banda e trim

	Units	Entry
Description		Pinnacle
Draft at AP	m	9,845
Draft at FP	m	7,780
Draft at Aft Marks	m	9,845
Draft at Fwd Marks	m	7,780
Heel Angle	deg	3,85S
LCR of Aft Pinnacle	m-MS	30,382F
LCR of Fwd Pinnacle	m-MS	0,000
Ground Type		Rock
Coef. of Friction		1,5

The following table shows a comparison between, on the one hand, the stranding force and the application position created by SSTAB and, on the other hand, the estimated values of the proposed formulation, as outputted by the HECSALV software.

Table 7. Comparison among results obtained by SSTAB and HECSALV software and the formulation

	Ground Reaction (ton)	LCR (m)	TCR (m)	VCR (m)
SSTAB	2,000	30.00	3.00	0
HECSALV	2,001	30.38	4.47	0
Formulation	2,000	30.12	2.64	0

Weight shifting in the event of stranding

Upon considering the same model used in the stranding simulation, which included trim and heel, one will notice the unit's behavior on the SSTAB software after shifting the cargo and a comparison should be possible with the results obtained by the proposed formulation of the present study. The stranding situation is as follows:

Table 8. Stranding conditions

Condition	Ground Reaction (ton)	LCG (m)	TCG (m)	VCG (m)	Objective
3	2,000	30	3	0	trim and heel

In the situation faced by the stranded ship, some weight aboard will be removed, which has the following characteristics:

Table 9. Characteristics of weight removed

Weight (ton)	LCG (m)	TCG (m)	VCG (m)
1,000	-30	-2.50	0

The formulation developed herein and the SSTAB simulated modeling produced the following results:

Table 10. Comparison among results obtained by SSTAB and HECSALV software and the formulation

	LCG (m)	TCG (m)	VCG (m)	GM (m)	Trim (m)	Heel (°)
SSTAB	-1.20	-0.14	6.72	3.20	-1.73	1.40
Formulation	-1.20	-0.14	6.72	2.80	-1.74	1.44

It is possible to see that the values are very close together, and the largest difference found pertains to the value of *GM*. That is due to the fact that the software does not consider the change in the center of gravity due to the reaction force. It was, therefore, found that the formulation is consistent, given that we observed a lower result, when taking this factor into account.

Conclusion and final considerations

This paper sought to create a calculation algorithm to reliably facilitate and optimize the salvage potential of a stranded vessel.

Re-flotation procedures for a ship that has run aground cannot be wholly predicted due to various conditions, which the vessel may be faced with on the occasion. However, for the cases analyzed here, the theoretical background developed in this study proved adequate. After carrying out the check tests, the results were satisfactory because there was only a slight difference between the results produced by the simulated model and the formulation after the necessary considerations were made.

As a follow-up to the work carried out, it would be interesting to devise a future study about the portrayals of a stranding event on a rigid plane and on a viscous plane, given that the behavior displayed by the ship in these cases would be different to the conditions proposed by the present study. For these kinds of occurrences, one should take into account the limits of the stranding plane and some physical factors of the soil where the unit has grounded.

After considering these static stranding cases, it would also be convenient to carry out a dynamic analysis of a grounded ship, taking into consideration the effects of waves and currents. It would, thus, be possible to represent the occasion closest to the real scenario, providing more accuracy and safety to the ship's refloating operation.

References

U.S. Navy Salvage Manual - Volume 1 - Strandings And Harbor Clearance - Published by Direction of Commander, Naval Sea Systems Command.

Mecânica do Navio, Estática – Parte V - Equilíbrio De Corpos Parcialmente Flutuantes.

Ship Structure Committee 2003 - Predicting Motion and Structural Loads in Stranded Ships, Phase 1.

HECSALV Salvage / Emergency Response Software – Developed and Supported by Herbert Software Solutions, Inc.

SSTAB Software – Desenvolvido por CENPES / Petrobras e Grupo de Tecnologia em Computação Gráfica (TeCGraf) / PUC-Rio de Janeiro.

Multiobjective Optimization of a Submarine Hull Design

Optimización multiobjetivo del diseño de un casco de submarino

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Abstract

A synthesis model for the concept design of a submarine is developed consisting of a parametric definition of the hull geometry, a maneuverability model based on slender-body theory, and a resistance formulation. This coupled model is suitable to be treated by a metaheuristic multiobjective optimization technique (a genetic algorithm) to find a set of design options that satisfy the need to minimize simultaneously the turning diameter and the resistance generated. According to typical data found in submarines like the one analyzed herein, the boundaries and some constraints are set for the design variables. Finally, some solutions for this design case are obtained considering the criteria adopted in this study.

Key words: submarine design, maneuverability, slender-body theory, genetic algorithm, multiobjective optimization

Resumen

Se desarrolla un modelo de síntesis para el diseño conceptual del casco de un submarino teniendo en cuenta una definición paramétrica de la geometría del casco, un modelo de maniobrabilidad basado en teoría de cuerpo esbelto y una formulación de resistencia al avance. Este modelo es incorporado a una técnica de optimización multiobjetivo metaheurística (un algoritmo genético) con el fin de encontrar un conjunto de opciones de diseño que satisfagan la necesidad de minimizar simultáneamente el diámetro de giro y la fuerza de resistencia generada. Considerando algunos valores comunes en el tipo de diseño aquí analizado, se establecen los límites de las variables de diseño, así como algunas restricciones. Finalmente, se presentan algunas soluciones para este caso de diseño contando con el desempeño obtenido para los dos criterios aquí estudiados.

Palabras claves: maniobrabilidad, diseño de submarinos, teoría de cuerpo esbelto, algoritmo genético, optimización multiobjetivo

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Introduction

Within submarine design topics, much has been developed concerning the definition of the hull shape, given that this primary feature is most important in all phases of the concept design, as implied by Jackson (1992). The use of a body of revolution is a typical design choice, comprising a smooth curved profile in the aft and the forward sections of the boat and a parallel middle body. Complete knowledge of the submarine's external geometry allows proper analysis of the hydrodynamics surrounding the body when operating underwater.

Analysis of the motion of a ship as a rigid body, including the response to a control force variation, leads to a maneuverability model, as described by Fossen (2011). These models are usually expressed as ordinary differential equations in time, whose terms are a function of some state variables like the vessel's velocity components. The maneuverability modeling for vessels is simplified in some cases to a linear system (only linear dependence on velocities), e.g., the ones considered by Clarke (1982) and Inoue (1981), from which some ship properties like dynamic stability and turning performance are suitable to be analytically assessed if the corresponding linear hydrodynamic coefficients are known; said procedures are shown by Fossen (2011). Other models allow non-linearities and, thereby, the analytical processing to determine dynamic properties is not straight-forward. In such cases, a numerical simulation may be necessary to estimate the behavior of the vessel in motion, for instance, to evaluate the diameter of a turning circle. However, this procedure may either directly or indirectly require knowing or computing the vessel's hydrodynamic coefficients. A good amount of work is available on several methods to measure or compute such coefficients, among which there are full-scale trials, scale-model experiments like the Pixel Mapping Method (PMM), published by Wagner-Smitt (1971), semi-empirical formulations and analytical techniques that arise from assuming the validity of certain theories, many of which are overseen by ITTC (2005). For some types of vessels, a slender-body theory is applied, as pointed out by Hooft (1982) and, thus, some coefficients for acceleration and velocity are evaluated. Several authors have used this theory and proposed models for the dynamics of surface ships and submarines, which may be found in Bertram (2000) and Bohlmann (1990).

Furthermore, hull resistance when moving underwater is another major topic to consider in its design because the propulsion and power supply systems are defined to overcome said force. Resistance is directly related to the geometry as its most common formulation, which for ITTC (1978) takes into account the geometric particulars of the ship.

Pursuing one of the purposes of the Colombian Ministry of Defense in the field of naval science and technology, focused on the development of vessel simulators for training, an initial effort is being dedicated to the formulation of physics-based simulation methods of diverse types of boats. One of those vessel types is the submarine and the development of a reliable simulation method not only becomes a component of a training system, but also a design tool because it estimates the behavior of the vehicle for certain maneuvers conducted when operating in a real scenario.

This paper explores the coupling of a submarine dynamics simulation method, derived from slender-body theory, a hull geometry parametrization, and a resistance model with an optimization technique, taking into account some constraints and choosing the appropriate objectives. The design variables are those that suffice to define the geometry of the hull and the control surfaces. As a starting point, the report by Zalek & Tascon (2004) on submarine hull optimization is studied and many features of it are kept for the implementation of the model proposed herein, remarking as the main difference the consideration of a non-linear maneuverability model in this case; whereas in the document referred turning performance was evaluated by means of a linear theory. The first part of the article shows the fundamental concepts of the three topics that build up the synthesis model.

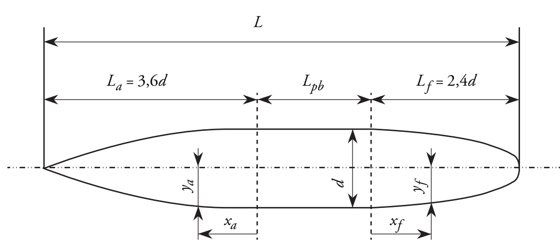
After this, optimization settings were defined and the results of this process are presented and discussed so that the potential and effectiveness of the method proposed can be assessed.

Submarine synthesis model

Parametric hull geometry definition

The submarine hull may be considered a body of revolution with some appendages, namely, a sail and stern rudders and planes. The radius of the body of revolution is given analytically for the aft and forward zones and in between there is a cylindrical or parallel middle body. Jackson (1992) proposed a parametric definition of the hull profile, which is shown in Fig. 1. The ship's length is denoted L, the hull diameter d, and the parallel middle body length is L_{pb}. The aft zone has a length of L_a and its radius is given by y_a. In the forward zone, the corresponding variables are L_f and y_f.

Fig. 1. Studied submarine hull geometry



$$y_a = \frac{d}{2}\left[1-\left(\frac{x_a}{L_a}\right)^{n_a}\right] \quad (1)$$

$$y_f = \frac{d}{2}\left[1-\left(\frac{x_f}{L_f}\right)^{n_f}\right]^{1/n_f} \quad (2)$$

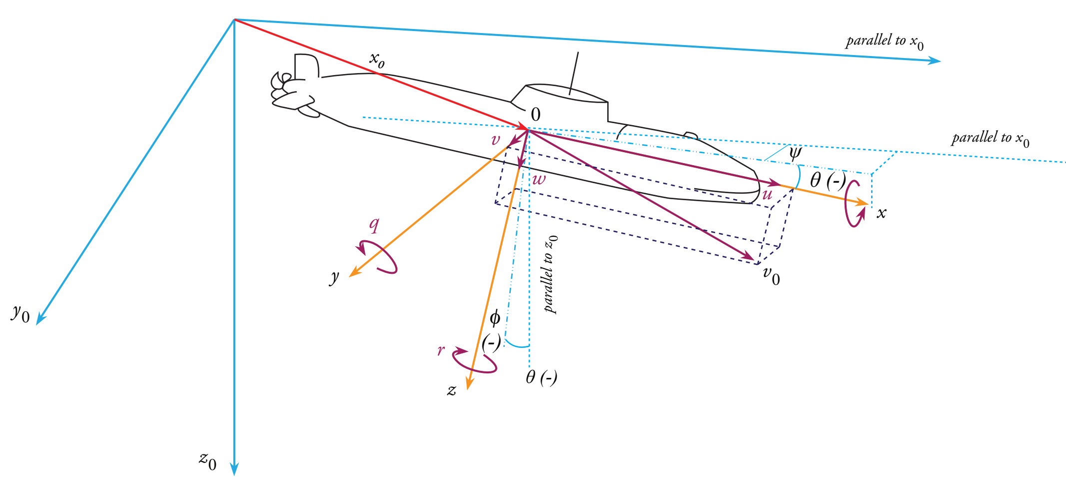
x_a is the aft longitudinal coordinate, being a value between *0* and L_a; for the forward zone, the coordinate is x_f (between *0* and L_f); exponents n_a and n_f are positive numbers that define the geometry shown in Fig. 1.

Submarine kinematics and maneuverability

To model a submarine motion, all six degrees of freedom are significant. Consequently, in translational motion, the surge (x), sway (y), and heave (z) components must be considered, while for rotational motion the roll (around x), pitch (around y), and yaw (around z) components are considered. As usual, two coordinate systems are defined, an inertial one (or fixed on earth x_0-y_0-z_0) and a moving one (or fixed on body x-y-z), the latter being aligned with the hull symmetry axis and the waterplane, having as a basis the NED convention (North/bow-East/port-Down) for the x-y-z directions, following Fossen (2011), and setting x = 0 at the midship. Fig. 2 shows the fixed and moving reference systems and the sign conventions for translational and rotational coordinates and velocities.

The unknown variables in this six-degrees-of-freedom model are the linear velocity of point, O

Fig. 2. Systems of reference for submarine kinematics



v_o, and rotational (or angular) velocity of the body, ω. These two vectors are defined, thus:

$$v_0 = \begin{pmatrix} u \\ v \\ w \end{pmatrix}, \quad \omega = \begin{pmatrix} p \\ q \\ r \end{pmatrix} \tag{3}$$

where u, v, and w are the surge, sway, and heave velocities, respectively; p, q, and r are the angular velocities of roll, pitch, and yaw, respectively. The position with respect to the fixed coordinate system is denoted x_0 and the submarine attitude is given according to the turning system, ZYX, and stored in vector ϑ, having three Euler angles necessary to define it.

$$x_0 = \begin{pmatrix} x_0 \\ y_0 \\ z_0 \end{pmatrix}, \quad \vartheta = \begin{pmatrix} \varphi \\ \theta \\ \psi \end{pmatrix} \tag{4}$$

Kinetics of the problem

The problem arises as a rigid body kinetics problem in which the submarine is a body subjected to a group of external forces and a three-dimensional motion is produced by their action. The set of forces mentioned comprises these sources: forces on the hull, F_H, propulsion, F_P, forces on the sail and control surfaces (rudder), F_A, that is,

$$\Sigma F = F_H + F_P + F_A \tag{5}$$

Hull forces include hydrodynamic forces, F_{HD}, hydrostatic forces, F_{HS} (buoyancy), and the ship's weight, W.

$$F_H = F_{HD} + F_{HS} + W \tag{6}$$

The final equations of rigid body motion are those given by Fossen (2011) as Eqs. (3.41), having yg = 0 (for symmetry).

Evaluation of forces acting in a maneuver

Hull Forces

Within the hydrodynamic forces (F_{HD}) some terms associated with the surge added mass exist; those found by the slender-body theory and those given by the viscous effect (axial resistance and cross-flow drag). The slender-body theory forces ($F_{HD,slender}$) are derived from the integration of the material derivative of the added mass momentum:

$$F_{HD\,slender} = -\int_L \frac{D}{D_t} \left\{ \begin{pmatrix} m_x^1 & 0 & 0 \\ 0 & m_y^1 & 0 \\ 0 & 0 & m_z^1 \end{pmatrix} \left[\begin{pmatrix} u \\ v \\ w \end{pmatrix} + \begin{pmatrix} p \\ q \\ r \end{pmatrix} \times \begin{pmatrix} x \\ 0 \\ z_s(x) \end{pmatrix} \right] \right\} d_x \tag{7}$$

The material derivative in this model only considers the convective effect caused by the surge component:

$$\frac{D}{D_t} = \frac{\partial}{\partial_t} - u\,\frac{\partial}{\partial_x} \tag{8}$$

A necessary assumption is that the submarine moves at a depth greater than five times the diameter of its hull, as mentioned by Allmendinger (1990) and, hence, the effect of the surface and wave interactions can be neglected. The rotational degrees of freedom can be treated similarly so that a final formulation for slender-body theory moments is obtained. The algebraic expansion and the differential and integral treatment of the above definition yields:

$$F_{HD} = -\,m_x \dot{u} + \frac{p}{2} L^2 X_{uu} u^2 \tag{9}$$

$$\begin{aligned} Y_{HD} = {} & -m_x ur - \int_L m_y^1 (\dot{v} + x\dot{r} - z_s(x)\dot{p})\, dx \\ & -u\left(m_{ym}^1 (v + x_{ym} r - z_s(x_{ym}) p)\right) \\ & -\int_{x_a}^{x_{ym}} \left(r - p\,\frac{\partial z_s(x)}{\partial x} \right) m_y^1\, dx \\ & -\frac{p}{2} \int_L h(x)\, C_{Dy} (v + rx)\, v_{cross}\, dx \end{aligned} \tag{10}$$

$$\begin{aligned} Z_{HD} = {} & -m_x uq - \int_L m_z^1 (\dot{w} - x\dot{q})\, dx \\ & -u\left(m_{zm}^1 (w - qx_{zm}) - q \int_{x_a}^{x_{ym}} m_z^1\, dx \right) \\ & -\frac{p}{2} \int_L b(x)\, C_{Dz} (w - xq)\, v_{cross}\, dx \end{aligned} \tag{11}$$

$$
\begin{aligned}
K_{HD} = & \int_L m_y^1 (\dot{v} + x\dot{r} - z_s(x)\dot{p})\, dx + u z_s(x_{ym}) \\
& m_{ym}^1 (v + x_{ym} r - z_s(x_{ym}) p) - u \int_{x_a}^{x_{ym}} \\
& z_s(x) \left(r - 2p \frac{\partial z_s(x)}{\partial x} \right) m_y^1\, dx \\
& - \dot{p} \int_L J_{xx}^1\, dx - \frac{p}{2} \int_L Z_p(x)\, h(x)\, C_{Dy}(v + rx)\, v_{cross}\, dx
\end{aligned} \tag{12}
$$

$$
\begin{aligned}
M_{HD} = & -m_x uw + \int_L m_z^1 (\dot{w} - x\dot{q})\, x dx - u \Big(x_{zm} m_{zm}^1 (w - q x_{zm}) \\
& - q \int_{x_{zm}}^{x_f} m_z^1 (w - xq)\, dx \Big) \\
& - \frac{p}{2} \int_L b(x)\, C_{Dz} (w - xq)\, v_{cross}\, dx + \frac{p}{2} L^2 X_{uu} u^2 Z_R
\end{aligned} \tag{13}
$$

$$
\begin{aligned}
N_{HD} = & \; m_x uw - \int_L m_y^1 (\dot{v} + x\dot{r} - z_s(x)\dot{p})\, x dx \\
& - u \int_{x_{zm}}^{x_f} m_y^1 (v + xr - z_s(x) p)\, x dx \\
& - u \Big(x_{ym} m_{ym}^1 (v + x_{ym} r - z_s(x_{ym}) p) \\
& - \int_{x_a}^{x_{ym}} \left(r - p \frac{\partial z_s(x)}{\partial x} \right) m_y^1\, x dx \Big) - \frac{p}{2} \\
& \int_L h(x)\, C_{Dy} (v + rx)\, v_{cross}\, x dx
\end{aligned} \tag{14}
$$

X_{uu} stands for a hydrodynamic coefficient associated with hull resistance (see section below). The cross-flow drag coefficients (C_{Dy}, C_{Dz}) are assumed equal to 0.61, according to the averages reported by Bohlmann (1991). Functions $h(x)$ and $b(x)$ denote local height and beam of the hull; $z_s(x)$ is the z-coordinate of the center of mass of the sectional added mass; $z_p(x)$ is the z-coordinate of the center of pressure of the cross-flow drag; the local cross-flow velocity is given by

$$
v_{cross}(x) = \sqrt{(v + xr)^2 + (w - xq)^2} \tag{15}
$$

Functions m'_y , m'_z and J'_{xx} are the sectional added masses for sway, heave, and roll motion (added moment of inertia). For elliptic sections, the sectional added mass and moment of inertia are given by Sorotkin (2009):

$$
m'_y = \frac{\pi}{4} \rho\, [\, h(x)]^2 \tag{16}
$$

$$
m'_z = \frac{\pi}{4} \rho\, [\, b(x)]^2 \tag{17}
$$

$$
J_{xx}^1 = \frac{\pi}{128} \quad \rho\, [\, b(x)^2 - h(x)^2]^2 \tag{18}
$$

Note that for circular sections $m'_y = m'_z$, and $J'_{xx} = 0$. Other zones of the vessel, whose cross-section is a circle with one or more ribs attached to it, have a different formula for their added mass and $J'_{xx} \neq 0$ (see Sorotkin, 2009). Longitudinal positions x_{ym} and x_{zm} are the locations of the maximum sectional added mass in sway (m'_{ym}) and heave (m'_{zm}), respectively. These values appear due to the flow separation premise of the slender-body formulation presented by Bertram (2000). The added mass in x, used in this work, is estimated as $m_x = 0{,}1m$, according to Fossen (2011).

Propulsion force

Regarding the thrust caused by the propulsion system, some data from the propeller (including its diameter D_p) and the flow incidence angle, Θ , are needed. The longitudinal force due to propulsion, X_p, and the pitch moment produced by it, M_p, (here this moment is null as $z_p = 0$).

$$
X_p = \rho n^2 D_p^4 (1 - t_p) K_t (J_m) \tag{19}
$$

$$
M_{p,0} = X_p z_p \tag{20}
$$

Flow incidence angle is defined as:

$$
\Theta = tan^{-1} \left(\frac{\sqrt{v^2 + w^2}}{u} \right) \tag{21}
$$

Propeller speed (rev. per second) is given by:

$$
n = \frac{u_c}{J_b D_p} \tag{22}
$$

J_b is a constant parameter that acts as an initial advance coefficient set to determine n as a function of D_p and a desired speed, u_c, which in this case is equal to the approach speed. An adjusted advance coefficient is proposed by Bettle *et al*., (2009) which is computed as follows:

$$J_m = \frac{u(1-w_f\ \exp(-(k\Theta)^y)}{nD_p(1-w_f)} \tag{23}$$

Hence, the thrust coefficient, K_T, can be computed with a formula corresponding to a submarine modeled by Watt (2007), that is:

$$\begin{aligned} K_T(J_m) = {} & 0{,}410758 - 0{,}115654 J_m - 0{,}107836 J_m^2 \\ & + 0{,}0713396 J_m^3 - 0{,}00620451 J_m^4 \\ & + 0{,}0127538 J_m^5 + 0{,}00487893 J_m^6 \\ & - 0{,}000678484 J_m^7 + 0{,}0000333463 J_m^6 \end{aligned} \tag{24}$$

The formulae for thrust deduction and wake fraction are extracted by regression from Jackson (1992) for *DP/d = 0.5*

$$\begin{aligned} 1-t_p &= 0{,}0001s^3 - 0{,}0039s^2 + 0{,}052s + 1{,}0661 \\ 1-w_f &= -0{,}0003s^3 + 0{,}0094s^2 - 0{,}1126s + 1{,}1201 \end{aligned} \tag{25}$$

where

$$s = L/d - K_2 \tag{26}$$

Forces on appendages

The rudder produces the following surge force, sway force, and yaw moment:

$$X_{rudder} = \frac{\rho}{2} L^2 X'_{\delta\delta} \delta^2 u^2 \tag{27}$$

$$Y_{rudder} = \frac{\rho}{2} L^2 U^2 Y'_\delta \delta \tag{28}$$

$$N_{rudder} = \frac{\rho}{2} L^3 U^2 N'_\delta \delta \tag{29}$$

$X'_{\delta\delta}$ is considered equal to 0.0208151, assuming the same condition studied in Mackay (2003). According to Spyrou (2003), the hydrodynamic coefficient associated to the rudder angle is computed as follows:

$$Y'_\delta = 3A_R / L^2 \tag{30}$$

$$N'_\delta = Y'_\delta(-1/2 + c_r/L) \tag{31}$$

Another main appendage is the sail, whose resistance is analyzed as stated below, besides its trimming moment:

$$X_{sail} = -\frac{\rho}{2} S_{ws} u^2 (C_{fs} + \Delta C_{fs} + C_{rs}) \tag{32}$$

$$C_{fs} = \frac{0{,}075}{(\log Re_s - 2)^2} \tag{33}$$

$$M_{sail} = -X_{sail} h_{SD} \tag{34}$$

where C_{fS} is the coefficient of friction of the sail, ΔC_{fS} is the roughness coefficient, C_{rS} is the residual resistance coefficient, Re_S is the Reynolds number associated with the sail, and h_{SD} is the z-coordinate of the sail's drag center of pressure. Regarding the lift effect on the sail, the sway force and yaw moment produced is:

$$Y_{sail} = \frac{\rho}{2}(u^2 + v^2) C_{L,sail} A_{sail} \tag{35}$$

$$N_{sail} = -Y_A x_{sl} \tag{36}$$

$C_{L,sail}$ is the lift coefficient of the sail and it is computed through the following expression by Whicker & Fehlner (1958):

$$C_{L,sail} = \frac{1{,}8\pi\alpha_e}{\sqrt{\alpha_e^2 + 4} + 1{,}8} \alpha_{sail} + \frac{C_{DC}}{\alpha_e} \alpha_{sail} |\alpha_{sail}| \tag{37}$$

where the sail attack angle is:

$$\alpha_{sail} = tan^{-1} \frac{-v}{u} \tag{38}$$

Sail area is $A_{sail} = L_{sl} h_{sl}$ and α_e and C_{DC} are parameters specified below.

Summing up the terms above, the appendages' contribution to the forces is:

$$X_A = X_{rudder} + X_{sail} \tag{39}$$

$$Y_A = Y_{rudder} + Y_{sail} \tag{40}$$

$$N_A = N_{rudder} + N_{sail} \tag{41}$$

$$M_A = M_{sail} \tag{42}$$

Resistance model

Hull resistance is evaluated by means of a formulation related to Reynolds number, *Re*, the wetted surface, S_w, and other geometric parameters (B_{max} and *L*), according to ITTC (1978). Hull resistance plus the contribution from the appendages yield the total resistance of the submarine. This resistance is evaluated at the maximum speed, u_{max}.

$$R_T = -\frac{\rho}{2} L^2 X_{uu} u_{max}^2 - X_A \big|_{u=u_{max}} \tag{43}$$

Hydrodynamic coefficient, X_{uu}, is defined as follows *(Bohlmann, 1991)*:

$$X_{uu} = -C_f \frac{S_w}{L^2} - \frac{\pi}{4} C_r \frac{h_{max} B_{max}}{L^2} \tag{44}$$

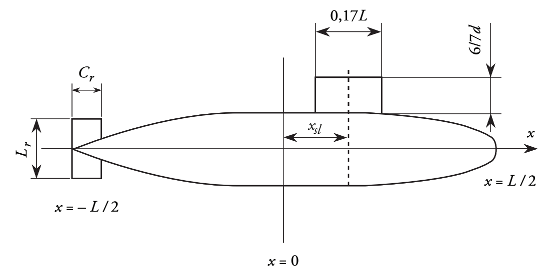
$$C_f = \frac{0{,}075}{(\log Re - 2)^2} + 0{,}00025 \tag{45}$$

Optimization Problem Definition

Design variables

According to the parametric shape of the hull introduced above, the variables to set for the optimal design are described in Table 1, including the bounds of the valid interval for each variable. Some design variables are shown in Fig. 3.

Fig. 3. Parametric definition of the submarine shape design



Rudder dimensions l_r and c_r (in Fig. 3), and design variables A_R and Λ, are related by the following expressions:

$$l_r = \sqrt{A_R \Lambda} \quad , \quad C_r = A_R / l_r \tag{46}$$

Objective functions

The goal of the optimization proposed herein was to find design options from which the best performance can be attained by taking into account different criteria. In this case, the functions of the submarine operation are maneuverability and hydrodynamic resistance so that the problem may be stated thus:

- **MINIMIZE** Non-dimensional steady turning diameter: *D'st* (computed through simulation of a turning circle)
- **MINIMIZE** Resistance at the maximum speed, *RT*

Table 1. Design variables and their respective bounds

Variable name	Symbol	Lower bound	Upper bound	Unit
Hull length	L	42	100	m
Hull diameter	d	7	10	m
Exponent of the radius function for the aft zone	n_a	1.5	5	-
Exponent of the radius function for the forward zone	n_f	1.5	5	-
Sail longitudinal location	x_{sl}	6.3	20	m
Total rudder area	A_R	6	20	m2
Rudder aspect ratio	Λ	0.6	1.3	-

In order to perform the turning circle simulation, the maneuverability model explained above is implemented and a rudder angle is applied so that the maneuver's resulting velocities and position may be estimated and, thus, the turning diameter.

Resistance is evaluated at the maximum speed, which matches the approach speed of the turning circle, and the vessel is considered to be in pure surge.

Constraints

- Jackson (1992) implied that the parallel middle body length is greater than or equals zero, which in terms of L and d is:

$$l_{pb} \geq 0 \;\rightarrow\; L \geq 6d \tag{47}$$

- The location of the sail is usually between 15% and 20% of the submarine length, stated by Zalek & Tascon (2004):

$$0{,}5L - x_{sl} \leq 0 \;,\; -0{,}2L + x_{sl} \leq 0 \tag{48}$$

- Zalek & Tascon (2004) limit the rudder area to be at least equal to a proportion of the product, Ld, (3% as for surface ships the recommended proportion is 2%), and at the most as a function of the envelope volume,∇:

$$\begin{aligned} A_R &\geq 0{,}03L = A_{R,bot} \;, \\ A_R &\leq 2\,(0{,}07\nabla^{\frac{2}{3}}) = A_{R,top} \end{aligned} \tag{49}$$

- The volume has to overtake a minimal capacity, ∇_{min}:

$$\nabla \geq \nabla_{min} \tag{50}$$

- The deck area must be greater than or at least equal to a given value $A_{deck,min}$:

$$\begin{aligned} A_{deck} &= 2\int_0^{L_a} y_a\,dx_a + L_{pb}\,d \\ &+ 2\int_0^{L_f} y_f\,dx_f \geq A_{deck,\,min} \end{aligned} \tag{51}$$

- The rudder has to satisfy a geometric constraint of not spanning beyond the hull diameter, as proposed by Zalek & Tascon (2004):

$$\frac{l_r}{2} \leq \frac{d}{2} - y_r \tag{52}$$

where

$$y_r = y_a(x_r) \;,\; x_r = L_a - c_r \tag{53}$$

Parameters

The parameters of the model are set as indicated in Table 2. These were defined by decision of the authors in some cases, but these have been mainly justified with the definitions given in several references, which are specified in Table 2. Seawater physical properties in the table correspond to a temperature of 20 °C and salinity of 35 g/kg. Most of these parameters are constant and a few are a factor of d or L. The only varying parameter used is the rudder angle, δ , which varies between 20 and 30°.

Implementation and Results

Multiobjective optimization

In order to obtain a set of non-dominated designs (Pareto front), a genetic algorithm included in ModelCenter® software was used. The problem specifications consist of a population size of 100 and a maximum of 100 generations, with a stopping criterion of 8 generations without improving. Other parameters of the genetic algorithm are automatically fixed by ModelCenter®. Objective functions are implemented in MATLAB® files, which are called by the optimization tool. Both objectives are scaled. This is to guarantee handling an equal order of magnitude in both criteria. A [0,1] range was mapped from a resistance range of [100000N, 200000N] and from a non-dimensional turning diameter of [0.7,1.4].

As stated above, two rudder angles were implemented. The case of 20° stopped after 65

Table 2. Definition of model parameters

Parameter name	Symbol	Value	Source
Center of gravity	(x_g, y_g, z_g)	(0,0,0)	Current work
Center of buoyancy	(x_{by}, y_{by}, z_{by})	(0,0,0)	Current work
Approach speed	U_0	20 knots	Current work
Minimum deck area	$A_{deck,\ min}$	100 m^2	Current work
Minimum volume	∇_{min}	800 m^3	Current work
Water density	ρ	1024 kg/ m^3	Sharqawy et al. (2010)
Water viscosity	v	$1{,}05\times10^{-6}$ m^2/s	Sharqawy et al. (2010)
Residual hull resistance coefficient	C_r	0.013	Jackson (1992)
Residual sail resistance coefficient	C_{rS}	0.005	Zalek & Tascon (2004)
Sail roughness coefficient	ΔC_{fS}	0.0004	Zalek & Tascon (2004)
Center of pressure of sail's drag	h_{SD}	$38d/21\pi$	Watt (2007)
Initial advance coefficient	J_b	1.11	Mackay (2003)
Constant to determine advance coefficient	k	3.4	Mackay (2003)
Constant to determine advance coefficient	γ	1.18	Mackay (2003)
Propeller diameter	D_p	d/2	Current work
Airfoil effective aspect ratio	a_e	0.57	Whicker & Fehlner (1958)
Airfoil cross-flow drag coefficient	C_{DC}	0.80	Whicker & Fehlner (1958)
Sail length	L_{sl}	$0.17L$	Current work
Sail height	h_{sl}	$6d/7$	Current work

generations and gave 16 designs while the 30° case ran all 100 generations with a final number of 11 designs. Top designs collected in the last generation of each case are specified in Table 3 and Table 4. In these tables, the shaded cells indicate the cases where the value of the variable was on or close to one of its bounds.

What is remarkable in Tables 3 and 4 is that variables L, d, and n_a tend to be close to their lower bounds for both rudder angles, while exponent n_f approaches its lower bound a few times. The other variable reaching its bounds is the rudder aspect ratio, Λ, which achieves its upper limit in the minimal diameter solutions for δ = 20°, but being close to the lower one in almost all the designs for δ = 30°.

Though all of the designs belong to a Pareto front, the choice of one of these designs has to be made. One feasible criterion for decision-making is the evaluation of the proximity to the Utopian point. Nonetheless, more designs can be viewed as they appear to be well located in the front because they are located close to the minimal value in one objective and have a significantly better value in the other objective than that of the extreme case. In Fig. 4, the resulting optimal designs are plotted onto the objective space after being normalized to a [0,1] scale according with their maxima and minima. Objective 1 stands for the diameter and Objective 2 denotes the resistance. The origin of coordinates on such plots is considered an estimate of the Utopian point and, thus, a distance from it to every design of the front can be computed. The distance of every Pareto point to its corresponding Utopian point is given in Tables and 6. The designs of interest in the decision-making process are underlined in those tables and circled on each plot of Fig. 7. The corresponding submarine shapes are displayed in Table 7, comprising the extreme cases of minimal resistance and minimal diameter along with two compromise solutions.

Table 3. Optimized designs for δ=20°

		Design variables							**Objectives**	
		L	d	n_a	n_f	x_{sl}	A_R	Λ	D'_{st}	R_T
	Unit	**m**	**m**	**-**	**-**	**M**	**m²**	**-**	**-**	**N**
PARETO FRONT DESIGNS	1	47.014	7.569	1.514	2.465	7.797	11.710	1.291	0.805	141987
	2	46.418	7.637	1.516	2.533	7.668	11.880	1.295	0.820	141364
	3	45.307	7.293	1.611	2.608	7.646	11.710	0.663	0.860	133541
	4	45.088	7.195	1.699	2.409	7.568	11.410	0.620	0.868	131343
	5	44.873	7.072	1.501	2.904	7.607	10.070	0.754	0.868	128642
	6	44.568	7.169	1.692	2.096	7.473	12.240	0.772	0.893	127893
	7	44.479	7.078	1.652	2.443	7.550	12.090	0.848	0.907	127421
	8	44.674	7.116	1.670	2.027	7.575	12.590	0.947	0.931	126911
	9	43.523	7.093	1.648	2.588	7.365	13.010	0.873	0.934	125154
	10	43.363	7.119	1.832	1.797	7.265	12.440	0.788	0.935	122669
	11	42.652	7.038	1.889	1.554	6.551	10.650	0.836	0.943	117829
	12	42.804	7.000	1.913	1.521	6.782	10.610	0.829	0.952	117631
	13	42.088	7.005	1.848	1.836	8.166	12.890	0.791	1.144	117499
	14	42.000	7.007	1.881	1.824	7.693	12.570	0.797	1.146	117403
	15	42.078	7.007	1.836	1.627	8.107	12.320	0.803	1.147	115937
	16	42.000	7.000	1.844	1.605	8.038	12.420	0.805	1.149	115472

Table 4. Optimized designs for δ=30°

		Design variables							**Objectives**	
		L	d	n_a	n_f	x_{sl}	A_R	Λ	D'_{st}	R_T
	Unit	**m**	**m**	**-**	**-**	**M**	**m²**	**-**	**-**	**N**
PARETO FRONT DESIGNS	1	42	7	1.5	2.134	6.854	9.51	0.6	0.810	116220
	2	42	7	1.5	2.104	6.757	9.57	0.6	0.811	116079

PARETO FRONT DESIGNS	3	42	7.003	1.517	1.968	6.912	10.13	0.6	0.815	115570
	4	42.100	7	1.512	1.897	6.860	10.01	0.601	0.818	115380
	5	42	7.003	1.553	1.872	6.741	10.11	0.6	0.821	115308
	6	42.026	7.003	1.509	1.856	6.842	10.22	0.628	0.822	114909
	7	42.075	7	1.5	1.766	6.807	10.3	0.630	0.824	114367
	8	42	7	1.515	1.690	6.803	10.26	0.629	0.826	113728
	9	42.306	7.006	1.505	1.5	6.910	9.66	0.6	0.826	113078
	10	42.238	7.001	1.505	1.5	6.846	10.5	0.6	0.828	112810
	11	42	7	1.535	1.506	6.687	10.65	0.736	0.854	112377

Fig. 4. Pareto front normalized objective space δ = 20° (left) and δ = 30° (right)

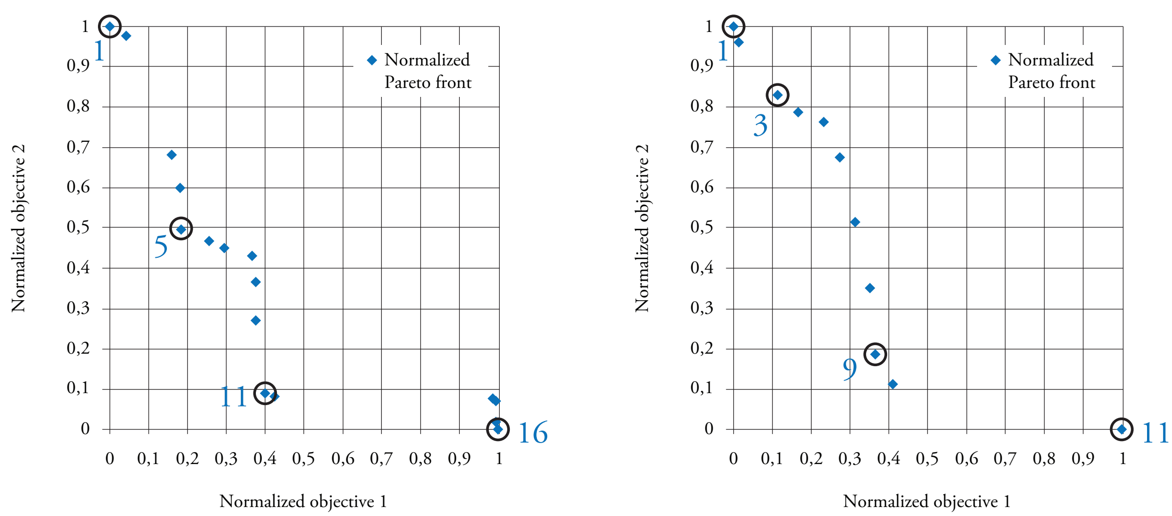


Table 5. Normalized objectives of Pareto front for δ = 20°

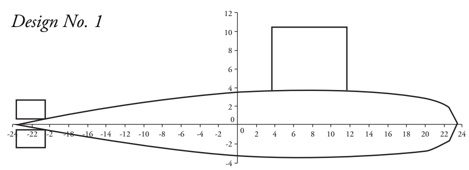
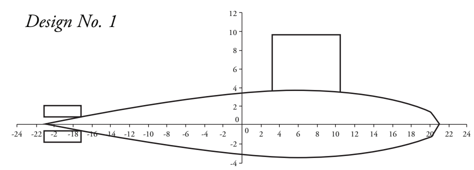
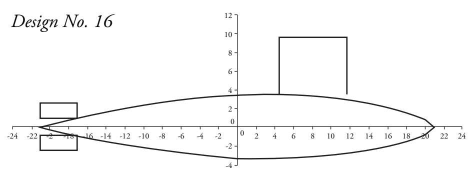
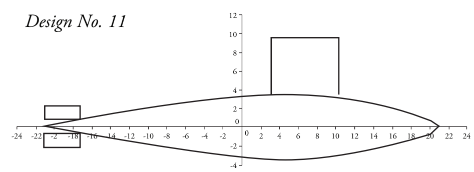
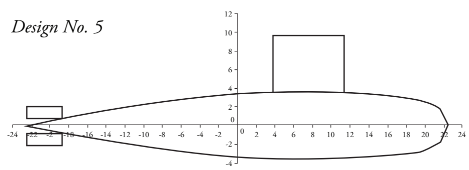
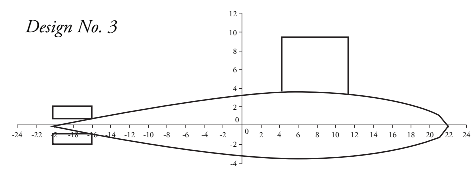
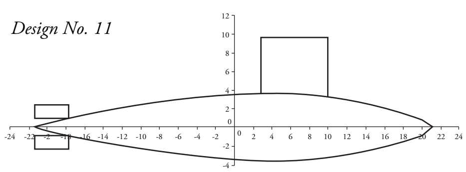
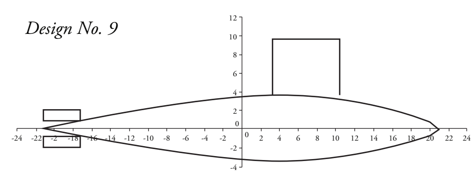
Design No.	1	2	3	4	5	6	7	8
Normalized objective 1	0	0.043	0.158	0.181	0.183	0.255	0.296	0.366
Normalized objective 2	1	0.977	0.681	0.599	0.497	0.468	0.451	0.431
Distance to Utopian	1	0.977	0.699	0.625	0.529	0.533	0.539	0.566

Design No.	9	10	11	12	13	14	15	16
Normalized objective 1	0.375	0.378	0.399	0.426	0.987	0.992	0.995	1
Normalized objective 2	0.365	0.271	0.089	0.081	0.076	0.073	0.018	0
Distance to Utopian	0.523	0.465	0.409	0.434	0.989	0.995	0.995	1

Table 6. Normalized objectives of Pareto front for δ = 30°

Design No.	1	2	3	4	5	6	7	8	9	10	11
Normalized objective 1	0	0.015	0.114	0.169	0.240	0.264	0.314	0.352	0.362	0.406	1.000
Normalized objective 2	1	0.963	0.831	0.781	0.763	0.659	0.518	0.351	0.182	0.113	0.000
Distance to Utopian	1	0.963	0.839	0.799	0.800	0.710	0.606	0.497	0.405	0.422	1.000

Table. 7. Sample of designs obtained in optimization

Main characteristic	Shape obtained by optimization with δ = 20°	Shape obtained by optimization with δ = 30°
Minimal turning diameter	*Design No. 1* 	*Design No. 1* 
Minimal resistance	*Design No. 16* 	*Design No. 11* 
1st compromise solution	*Design No. 5* 	*Design No. 3* 
2nd compromise solution	*Design No. 11* 	*Design No. 9* 

Discussion of Results

By looking at the geometries in Table 7, some features are graphically identified from the solutions obtained through the genetic algorithm. For a minimal turning diameter, a bulkier forward zone of the hull is seen. Besides, a very slim aft zone is noticed in all designs. On the other hand, if the resistance is the minimization objective, a slim forward body is obtained.

The former observation is associated with how the maneuverability model was formulated. The integrals that compose the hydrodynamic forces equations and were derived from the slender-body theory are mainly evaluated between the section of greatest added mass and the bow due to the assumption of validity of flow separation, suggested by Bertram (2000) even though other sources, like Toxopeus (2010), state that such an assumption may not be totally accurate. Then, as seen in

Bohlmann (1990) or Hooft (1982), hydrodynamic derivatives can be extracted from those integrals and, thereby, the integral limits cause an important effect on the ship's stability and maneuverability. Given that the aft body shape has a smaller effect on the maneuvering coefficients, the forward part of the hull is the one that changes the most to optimize the design under the specified criteria.

In most cases, the length and hull diameter were at or close to their lower bounds. Because these variables are almost constant for every solution obtained, and the aft zone is thin, the shape of the forward body affects the wetted surface and its volume. By recalling the formula for hull resistance, it is clear that a greater wetted surface area increases resistance, so it is noticeable that this result is consistent with the expected solution.

For the 20° rudder angle, it is remarkable that design No. 5 can play the role of a compromise solution, given that when compared to both extreme cases this shows an intermediate geometry, while design No. 11 looks more like the minimal resistance design than that with the minimal turning diameter.

Regarding the second rudder angle, extreme cases are more alike than in the previous case, yet a more compromise-like choice can be seen in design No. 3, rather than in design No. 9 that is closer to the solution with optimized resistance. Nonetheless, this assessment is performed only visually, but if the decision-maker prefers another criterion, other points in the Pareto front might be chosen as the optimal design.

Concerning the constraint action, Tables 8 and 9 contain the data of the closeness to each constraint for the top designs obtained after optimization. The shaded cells show the cases where the constraint was active and the value was exactly on the limit or marginally (even fairly out of the boundary). It is observed that mainly all values are away from the limits, except for the length of the parallel middle length, which is zero or very small. While in the solutions where the turning diameter is closer to a minimal value the volume is not near the minimal required capacity, the volume constraint turns active when the chosen design is more focused on minimal resistance.

Table 8. Active constraints in optimization for $\delta = 20°$

Design	1	2	3	4	5	6	7	8
L_{pb}(m)	1.60	0.60	1.55	1.91	2.44	1.55	2.01	1.98
x_{sl}/L	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17
(m^3)	1236	1231	1139	1107	1077	1044	1054	1025
A_{deck} (m^2)	210	207	194	187	194	178	183	178
A_R-$A_{R,bot}$ (m^2)	1.03	1.25	1.80	1.68	0.55	2.65	2.64	3.05
A_R-$A_{R,top}$ (m^2)	-4.42	-4.20	-3.57	-3.58	-4.64	-2.17	-2.42	-1.65
$l_r/2-(d/2-y_r)$ (m)	-1.23	-1.24	-1.36	-1.32	-1.42	-1.16	-1.11	-1.03

Design	9	10	11	12	13	14	15	16
L_{pb}(m)	0.96	0.65	0.43	0.80	0.06	-0.04	0.04	0.00
x_{sl}/L	0.17	0.17	0.15	0.16	0.19	0.18	0.19	0.19
(m^3)	1032	966	890	887	907	907	870	862
A_{deck} (m^2)	178	160	147	147	151	150	147	145
A_R-$A_{R,bot}$ (m^2)	3.75	3.18	1.64	1.62	4.05	3.74	3.47	3.60
A_R-$A_{R,top}$ (m^2)	-1.29	-1.25	-2.31	-2.32	-0.23	-0.55	-0.44	-0.27
$l_r/2-(d/2-y_r)$ (m)	-1.02	-1.05	-1.15	-1.13	-0.94	-0.96	-1.00	-0.98

Table 9. Active constraints in optimization for δ = 30°

Design	1	2	3	4	5	6	7	8	9	10	11
L_{pb}(m)	0	0	-0.02	0.100	-0.02	0.011	0.075	0	0.271	0.229	0
x_{sl}/L	0.163	0.161	0.165	0.163	0.161	0.163	0.162	0.162	0.163	0.162	0.159
(m³)	884	881	867	859	860	850	836	823	797	794	792
A_{deck} (m²)	165	165	162	161	159	160	159	156	153	153	150
A_R-$A_{R,bot}$ (m²)	0.69	0.75	1.31	1.17	1.29	1.39	1.46	1.44	0.77	1.63	1.83
A_R-$A_{R,top}$ (m²)	-3.39	-3.29	-2.60	-2.64	-2.55	-2.34	-2.12	-2.04	-2.38	-1.51	-1.33
$l_r/2-(d/2-y_r)$ (m)	-1.51	-1.50	-1.44	-1.45	-1.43	-1.42	-1.42	-1.42	-1.49	-1.41	-1.32

Design considerations presented by Allmendinger (1990) include some alternatives to reduce the drag of the submarine, which consist in increasing the length, reducing the wetted surface, or increasing the length-to-diameter ratio. Regarding the first statement, although the length in all designs was about its lower bound, in one of the cases (rudder angle of 20°) it is seen that greater lengths are achieved for minimal turning diameter rather than for minimal resistance and, thence, the first alternative cannot be proven here. Nevertheless, resistance is minimized with a decreasing wetted surface (because this value is strongly related, see variation of the displacement in Tables 8 and 9) and, therefore, the second alternative proposed is verified, which agrees with the comment made above. A likely explanation to this is that the wetted surface (which is highly affected by the length) has a bigger influence on the ship's resistance and the optimization process first tends to get to solutions with a more reduced length and later it makes wetted surface decrease by modifying the exponents of the shape functions. Thus, the third statement is proven as the highest length-hull ratio is attained only with the smallest hull diameter because the length has already been set about its lower bound.

Allmendinger (1990) also mentions that a desirable geometry for low drag is a long tapered hull form. This is easily checked on the designs obtained, where the aft body is very slim so that a tapered form is found.

Conclusions

A mathematical model for submarine motion was stated and implemented for parametrically defined hull shapes. Such a model may be used to simulate maneuvers in a virtual environment, as well as a design tool by means of an optimization technique that allows evaluating the best performance for a specified set of criteria.

Upon identifying two objectives, several design constraints were included to define a feasible space over which an optimization technique could be applied.

The resulting geometries showed consistency regarding the relationship between the hulls' wetted surface and resistance. Furthermore, as expected, the minimal required capacity is fairly obtained in the designs of minimal resistance and is widely accomplished in the solutions with the smallest turning diameters.

The way optimization enhances the turning ability lies on the variation of the forward zone of the submarine's hull because this part of the body affects the hydrodynamic derivatives the most and, therefore, the vessel's maneuverability.

Some design considerations were observed to check if the resulting optimized solutions agreed with them. Due to this, a possible explanation was provided on how the optimization process led to the final set of designs.

There was a graphical identification of a compromise solution for both rudder angles considered in the optimization, but the data provided can be used by the decision-maker if a new criterion is preferred.
As a prospective future work topic, implementation of a more complete model and other maneuvers is sought so that more reliable resulting designs can be obtained. That model could enhance the way appendages and propulsion forces are assessed, though the number of variables may increase if a higher complexity of the parametric geometry is present.

A deeper treatment of the slender-body theorem equations employed in this model is another purpose to be pursued. Taking into account interactions between the rudder and the hull and other effects not considered so far, a set of formulae for the hydrodynamic coefficients can be achieved, as done by Bohlmann (1990). With those coefficients explicitly computed, an analysis of stability can be performed, which can be incorporated to the optimization model as a new criterion or constraint.

References

ALLMENDINGER, E. E. 1990. Submersible Vehicle Systems Design. SNAME.

BERTRAM, V. 2000. Practical Ship Hydrodynamics. Oxford.

BETTLE, M. C.; GERBER, A. G; WATT, G.D. 2009. Unsteady analysis of the six DOF motion of a buoyantly rising submarine. Computers & Fluids 38.

BOHLMANN, H. J. 1991. An analytical method for the prediction of submarine maneuverability. RINA Warship '91 Symposium on Naval Submarines 3, London.

BOHLMANN, H. J. 1990. Berechnung hydrodynamischer Koeffizienten von Ubooten zur Vorhersage des Bewegungsverhaltens. Dissertation. Institut für Schiffbau, Universität Hamburg.

CLARKE, D., GEDLING, P., & HINE, G. 1982. The Application of Manoeuvring Criteria in Hull Using Linear Theory. London, UK: The Royal Institution of Naval Architects.

FOSSEN, T. I. 2.011. Handbook of Marine Craft Hydrodynamics and Motion Control. Chichester, U.K.: John Wiley & Sons.

GABLER, U. 2000. Submarine Design. Bernard & Graefe Verlag, Bonn.

HOOFT, J. P. 1982. Advanced dynamics of marine structures. Wiley.

INOUE, S., HIRANO, M., & KIJIMA, K. 1981. Hydrodynamic Derivatives on Ship Manoeuvring. International Shipbuilding Progress , 28 (321), 112-125.

ITTC – Recommended Procedures. Performance prediction method, 1978, revised in 1987. Edited by the 22nd ITTC QS Group 1999.

ITTC. 2005. Final report and recommendations to the 24th ITTC, The manoeuvring committee. Proceedings of the 24th ITTC , I , 137-198. Edinburgh, U.K.

ACKSON, H. A. 1992. Fundamentals of Submarine Concept Design. SNAME Transactions 100, pp. 419-448.

KOROTKIN A. I. 2009. Added masses of ship structures. Springer Science.

MACKAY, M. 2003. The standard submarine model: a survey of static hydrodynamic experiments and semi-empirical predictions. Technical Report DRDC Atlantic TR 2003-079.

NEWMAN, J. N. 1977. Marine Hydrodynamics. The M.I.T. Press.

SHARQAWY, M. H.; LIENHARD, J. H.; ZUBAIR, S. M. 2010. Thermo-physical Properties of Seawater: A Review of Existing Correlations and Data, Desalination and Water Treatment.

SPYROU, K. 2003. Ship manoeuvrability, Lectures Notes. Glasgow: Glasgow University.

TOXOPEUS, S. 2010. Validation of slender-body method for prediction of linear manoeuvring coefficients using experiments and and viscous-flow calculations. Proceedings of the 9th International Conference on Hydrodynamics (ICHD 2010). China Ocean Press.

WAGNER-SMITT, L. 1971. Steering and Maneuvering of Ships – Full Scale and Model Tests. European Shipbuilding, Vol. 19, No. 6, and Vol. 20, No.1.

WATT, G. 2007. Modeling and simulating unsteady six degrees-of-freedom submarine rising maneuvers. Technical Report DRDC Atlantic TR 2007-0089.

WHICKER, L. F.; FEHLNER, L. F. 1958. Free-stream characteristics of a family of low-aspect-ratio. All-movable control surfaces for application to ship design. David Taylor Model Basin, Report 933.

ZALEK, S.; TASCON, O. 2004. ME555-Design Optimization - Term Project: Submarine Hull Optimization. University of Michigan.

The Most Recent Noise & Vibration Assessment of the European Fleet, within the Framework of the "SILENV Project"

La evaluación de ruido y vibraciones más reciente de la flota europea dentro del marco de trabajo del "Proyecto SILENV"

Publio Beltrán Palomo [1]

Abstract

The EU's new "Green Policy" to reduce the environmental impact of all types of vessels is generating new Directives that shall affect the Shipbuilding Industry. As a direct consequence, an intense debate has been opened with the participation of all the players involved: the European Commission, Marine Institutions, Shipbuilding Industry, Marine Sector, and Scientific Community. Participation of the authors in the SILENV, BESST, and AQUO projects within the FP7 has permitted noticing that for the complete assessment of the ships' environmental impact it is essential to introduce the new so-called Noise & Vibration – Full Signature indicator. In addition to the well-known topics of Noise & Vibration (N&V) on board, it includes new ones: Noise Radiated to Harbour and Underwater Radiated Noise (URN) by the ships. Both, but especially the latter (URN) became the most outstanding novelty and the biggest challenge to be technically solved. In this sense, it is essential to know the "Starting Point": How far is the current European Fleet from the Standard Regulations and from other Directives that will soon be compulsory? And which are the technical reasons and root causes of these deviations? To address these issues within the framework of the SILENV Project, the most recent N&V database compiled with 171 ships from the European Fleet and 12,000 N&V experimental data have been assessed. It became the largest ever N&V database in the marine sector, as well as a complete novelty. Therefore, the results, conclusions and recommendations obtained from it are of paramount importance to support current policies of the EC and other Marine Institutions that focus their efforts on combating the environmental impact of the ships.

Key words: Noise, Vibration, Noise radiated to harbour, Noise radiated to the water, Underwater footprint

Resumen

La nueva Política Verde de la Unión Europea de reducir el impacto ambiental de todo tipo de buques, está generando nuevas directrices que afectarán a la industria de la construcción naval. Como consecuencia, un intenso debate se ha abierto con la participación de todos los actores: la Comisión Europea (EC), Instituciones Marítimas: ILO e IMO, la Industria de la Construcción Naval Astilleros, el Sector Marítimo y la Comunidad Científica. La destacada participación del autor en los Proyectos de Investigación BESST, SILENV y AQUO dentro del 7º Programa Marco (FP7) ha puesto de manifiesto que la evaluación del "impacto ambiental" de un buque requiere la introducción de un nuevo "indicador" llamado "Firma Acústica del Buque". Dicho indicador incluye además de los ya conocidos Ruidos y Vibraciones a bordo, los nuevos aspectos como son: el Ruido Radiado al Puerto y el Ruido Radiado al Agua por el buque. Estos dos últimos aspectos, pero especialmente el último, se ha convertido no sólo en la más destacable novedad, sino en un gran reto técnico que se debe resolver. En este sentido, es esencial conocer el "punto de partida": ¿Cómo se encuentra la Flota Europea con respecto a las Regulaciones Estándar y a las nuevas Directrices que, de forma inmediata, serán obligatorias?, y ¿Cuáles son las razones técnicas y causas originarias de estas desviaciones? Para responder a estas inquietudes, dentro del marco del proyecto SILENV, la Base de Datos más reciente integrada por más de 171 buques de diferentes tipos de la flota europea y 12.000 datos experimentales, ha sido evaluada. Ello la convierte en la base de datos experimental más extensa dentro del sector marítimo además de una reciente novedad. Los resultados y conclusiones obtenidas se revelan de gran importancia para soportar las políticas actuales de la EC y otras instituciones marítimas en el reto de reducir el impacto ambiental de los buques.

Palabras claves: Ruido, Vibración, Ruido radiado a puerto, Ruido radiado al agua, Firma acústica submarina

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Nomenclature

N&V: Noise and Vibration
IMO: International Maritime Organization
ISO: International Standard Organization
EC: European Commission
NRH: Noise radiated by the vessels to harbour
URN: Underwater Radiated Noise by the vessels
FP7: Seventh Frame Programme on Research and Development
N&V-FS: N&V full signature of the vessel; includes N&V on board, NRH, and URN for each vessel
BESST project: Breakthrough in European Ship and Shipbuilding Technologies
SILENV project: Ship-oriented Innovative solutions to reduce Noise & Vibrations
PSL: Preliminary SILENV target limits

Introduction

Noise and vibrations (N&V) induced by different N&V sources on board vessels have been among the most recurrent issues in the shipbuilding industry. Its trend over time and its presence and request in the contract specifications, always related to the state-of-the-art, has been widely changed. Thus, in the contract specifications of the 70s, references to vibration were few and consisted of ambiguous quotes such as *"the shipyard will do its best to reduce noise and vibration on board"*. At this time, this ambiguity, as well as the absence of clear and well-established limits for both topics (noise and vibration), generated different opinions: from the owner's point of view, the vessels had high vibration levels, but from the shipyard's point of view these were barely perceptible. Reaching agreement was, therefore, difficult. It is important to mention that at that moment the risk of having high vibration on board was increasingly higher due to several aspects such as the optimization of the scantling of the vessels, their larger size, and more powerful engines installed.

At the beginning of the 80s, with the appearance of the IMO A.468 (XII) Noise Resolution [4] and ISO 6954-1984 Standard [5], related to vibration on board, said uncertain scenario changed drastically. At that moment, the noise and vibration assessment could be performed objectively and impartially and, more importantly, based on contractually defined noise and vibration limits. In any case, as can be seen in the corresponding paragraphs related to the scopes and fields of application of both standards, if the ship is under 1,600 tons of gross tonnage, it is exempted from complying with the noise standard, and if the vessel is under 100 m, it is exempted from complying with the vibration standard. This clearly means that practically all fishing vessels and most of the small merchant vessels were outside all N&V regulations. Particularly, the IMO Resolution stated that the Code is not applicable to fishing vessels, among others. Furthermore, it also explains that "the Code should apply to new ships under 1,600 tons gross tonnage, as far as reasonable and practicable, to the satisfaction of the Administration". These deliberate exceptions would have been related to the N&V state-of-the art of that moment. Also, it is important to point out that at that time there was an extended belief on the impossibility of mitigating noise in fishing vessels because they were "noise boxes". In fact, the belief was so deeply-rooted that even when the chief engineer was sleeping next door to the main diesel engine room, no actions were taken. The dramatic consequences of this absence of regulations arose quickly: several dedicated investigations have highlighted the high percentage of hearing loss and injuries among crew members and the negative effects on their health. However, and despite these absences, it is essential to highlight the excellent role played by these N&V standards on the sensitization of the shipbuilding industry and owners, which took a first step on the abatement or mitigation of N&V on board.

If we take into account the great growth that the cruise market has experienced during the last two decades (146%), and the companies' point of view (which are convinced that the passengers choose a cruise evaluating both safety and comfort), it is obvious to point out that cruise liners are the vessels that have made the biggest efforts and have moved ahead to abate N&V on board, focused on satisfying the passenger´s expectations.

The Comfort Class Notations of the different classification societies based most of them on these N&V Standards, but with much more strict N&V limits, have enabled standardizing the different degrees of comfort related to N&V on board the ships. Usually, these comfort class notations with their corresponding N&V limits are contractually required in the contract specifications signed between owners and builders. As it is a vital point for the owner companies (their reputation could be questioned), achievement of high levels of comfort, requested by the customers (passengers), is a must. In many cases, penalties are included in the contracts and in extreme cases these contracts could even be cancelled if there is a severe level of incompliance. To avoid this extreme situation and guarantee compliance with the N&V targeted limits, the application of simulation tools, such as N&V prediction calculations, during the early stages of the design project, are contractually demanded by the owners. This trend has forced builders and the rest of suppliers of the shipbuilding industry to apply the most recent state-of-the art focused on minimizing N&V on board.

This could be a summary of what has been done along these years in the field of on-board N&V regulations. All the efforts for improving these topics have centred on combating the N&V inside the vessel, but nothing has been done for reducing the outside impact. The emergence of the new EU "Green Policy" to reduce the environmental impact of all types of vessels is generating new directives, among others, Directive 2003/10/EC [1], Directive 2006/87/EC [2], and the recent Directive 2008/56/EC [3], which is establishing a new, consistent, and "global regulatory framework" focused on avoiding the negative effects of N&V not only on the health of the crew/passengers "inside the vessels" but also on the environment. When we talk about negative effects on the environment we are talking about the noise/vibration radiated by the vessels to the air and into the water. In the first case (noise radiated by the vessels to the harbours- NRH-), the negative effects of this aspect have been noted because of increased complaints from the people affected from nearby residential areas or from the island waterways. Regarding the second topic: underwater radiated noise by the vessel (URN), many publications and research, conducted by the Scientific Community, have permitted pointing out that the shipping traffic is one of the root causes of the noise increase in oceans (more than 50 dB (ref 1mPa) in the last twenty years). This fact has brought this topic into the limelight and promoted a hot debate related to the disturbance that noise produces on marine fauna.

This global approach promoted by the expected regulatory frameworks is a great challenge for the shipbuilding industry. In this sense, in order to pay the necessary attention to this concern and to move forward, the EC within the context of the 7th Frame Programme (FP7) has launched two dedicated research projects: BESST and SILENV to conduct deeper analysis and obtain knowledge about the previous topics. The first project, among other aspects, centred its attention on the big Ro-Ro and Passenger vessels; the second one was more general and analysed all types of vessels. Practical Guides for Improvement should be provided by both projects.

Simultaneous participation of the authors in the SILENV and BESST projects within the FP7 has permitted their noticing that for the complete assessment of the ships' environmental impact it is essential to introduce a new so-called N&V-Full Signature (N&V-FS) indicator of the vessel. This indicator, besides the well-known topics of N&V on board, also includes: Noise Radiated to Harbour (NRH) and Underwater Radiated Noise (URN). Both of them, but especially the latter (URN) became the most outstanding novelty and the biggest challenge to be technically solved.

Heeding previous requirements, the SILENV project proposes a holistic approach to reduce ship generated N&V pollution. So, after defining some realistic N&V levels, the next step would be to identify the noisiest sources to abate them. To carry out this activity, all the experimental data available should be assessed and analysed. This information compiles data from all types of vessels and has been collected during dedicated on-site measurement activities. Afterwards, innovative solutions will be listed and meticulously assessed from technical and economic points of view. Then, numerical models of the ship will be created. These models will be gradually improved and refined until an optimized model is found. The optimized model will be a guide to determine what improvements should be implemented on the ship. Finally, the main deliverable proposed by SILENV is a green label that includes recommended N&V levels and some design guidelines.

Thus, in the concern of the abatement of the N&V FS of the vessels, it has been considered essential to know, in a first stage, the "Starting Point", or in other words: *How far is the current European Fleet from the Standard Regulations and from other Directives that will soon be compulsory?* This assessment has been performed by done analyzing the large collection of experimental data that has been compiled in the SILENV-N&V database, and that includes information from different types of vessels representative of the current European Fleet. In a second stage, a prudent analysis of the data has enabled answering the following key question: *Which are the technical reasons and the root causes for the deviations?* Both statistical approaches, developed in the framework of the SILENV project and the results obtained from them, have been contrasted and validated with a Mapping Analysis.

This analysis was also conducted by using the experimental data collected through dedicated on-site measurement activities performed on a representative sample of different types of vessels.
The aim of the present paper is to present to all players (EC, Marine Institutions, Shipbuilding Industry, Marine Sector, and Scientific Community), involved in the N&V pollution debate, the results and conclusion of these analyses. It will allow defining the right and proper short-, medium-, and long-term, policies for achieving this goal.

This paper must be understood within the context of the dissemination activities carried out by the author's company, as leader of the WP2 "Measurement and Requirements" of the SILENV project.

The Target: The European Fleet

Before the proper N&V assessment is explained, in order to know the real scope of the matter related to the number of vessels that could be affected by the conclusions of the present paper, it has been considered important to present some figures on the current European Fleet.

On 1st January 2009, the total number of cargo and passenger vessels that made up the European Fleet (Fig. 1) was the following:

Fig. 1. Cargo and Passenger Vessels

Fleet for transport of goods and passengers	
Ships of 1000 GT and over	**Number**
Tankers	2914
Bulk carriers	2152
General cargo	3390
Passenger and passenger cargo	735
Sub-Total	11621
Ships of 300 GT and over	**Number**
Cargo passenger and Ro-Ro passenger ships	842
Passenger (not Ro-Ro)	405
Sub-Total	1247
Cruise Ships	97
TOTAL	**12965**

The average age of the fleet is estimated around 22 years. Other sources estimate that 38% of the fleet is older than 25 years, 24% older than 30, and 12% older than 35. However, and according to different sources (ISF/BINCO, ECSA, etc), the total number of seafarers employed in shipping activities for these types of vessels has been estimated around 254,119 people.

To these vessels we must add all the European Fishing Fleet, as well as all the European Research Vessel Fleet. In 2008, the EU-27 fishing fleet consisted of 86,587 vessels with an average age of 22.8 years. Some 82% of these fishing vessels are shorter than 12 m. The number of seafarers is estimated around 141,110. The European Ocean Research Fleet is composed of a total of 46 vessels, including 11 of Global class, 15 of Ocean class, and 20 of Regional class. The average age for this fleet has been estimated around 16 years (source: Paper 10 Ocean Research Working Group).

Thus, focused on the analysis that will be carried out in the following paragraphs, the summarized figures of the European Fleet are:

- The Fleet is made up of a total of 99,598 vessels
- With an average age estimated around 20 years
- The total number of seafarers that could be potentially affected by N&V on board is 395,229 people

The "SILENV-N&V Database"

A complete collection of N&V information has been compiled. This collection had to be representative of the European fleet and, therefore, it had to include all kinds of vessels. It was also the foundation upon which all the assessment was based. For this compilation, a tailor-made tool was developed: the "TSI-N&V Excel Tool". All the participants in the project had access to it and used it to finally obtain a collection that included 151 vessels; each vessel had its own "Vessel Card", specifically designed for the purpose. The "Vessel Card" contains, in addition to the N&V experimental data, all the information of what has been done on the vessel to abate N&V, including information related to the technical efforts applied during the design of the ship: use of simulation tools, N&V sources, as well as control and engineering solutions applied. The collection (151 vessels) comprises 54 Merchant vessels, 48 Ro-Pax and Passenger vessels, six FRV, five Ferry vessels, six Fishing vessels, 12 Tug vessels, and 20 vessels from different types grouped into a block labelled *Other*. For each family of vessels, data was sought from old and new ships.

None of the vessels included in the SILENV-N&V existing database reaches the N&V Full Signature grade. To fill this lack of experimental information and with the objective of enabling the N&V FS assessment of the current vessels, dedicated and extensive on-site measurement activities were developed within the frame of the WP2 of the SILENV project. The main goal of these activities was to obtain the N&V-FS of at least one representative vessel from each family or group. The tests and measurements were performed on board of 20 vessels, keeping in mind that at least one vessel from each family should be characterised and its N&V FS obtained.

Previously, a measuring procedure was defined to obtain comparable experimental data. The measurement activities were carried out on five merchant vessels, eight passenger vessels, four fishing research vessels, two fishing vessels, and one LNG. As a result, the SILENV-N&V database was obtained, which compiles 171 representative vessels from the Civil European Fleet. This database has over 12-thousand N&V values. All this information is crucial to validate modelling tools and becomes a novelty in the maritime sector. It is the only database that can enable the assessment of the real impact of current ships on the environment and on marine life, but it also permits identifying the root causes of the deviations with regard to current and future N&V requirements. The high quality of the information compiled in the SILENV-N&V database ensures the suitable and successful development of the following sub-tasks: Assessment and Mapping Analysis. Thereby, the questions previously formulated:

- *How far are current ships from the preliminary SILENV limits?, and*
- *Which are the technical reasons and lack of requirements motivating these deviations?*

may be successfully answered.

The Fleet's N&V Assessment

Background and introduction

As previously mentioned the design of noise- and vibration-free ships has been for many years and continues being one of the main concerns of owners, shipyards, and ship-operators. To meet this demand, many Classification Societies have published guides for avoiding N&V on board. The problem with these design guides is that they are based on a reduced sample of vessels from a certain family. In this regard, especial mention is required of the works by the Research and Development Division of Bureau Veritas, American Bureau of Shipping (ABS), Lloyd Register of Shipping (LR), Det Norske Veritas (DNV), and Germanischer Lloyd (GL); as well as the dedicated analysis carried out by Smogli on 41 vessels and M. Biot and F. De Lorenzo on board cruise ships.

In most of the previous cases, the analysis focused only on the N&V on board and on identifying the main sources that cause these problems. In some of these investigations, the propeller has been found as one of the main sources that cause high vibration levels on board. Despite the great utility that these documents have had for a long time, some mistakes have been identified, especially when seeking to extrapolate conclusions that were valid for the N&V on board to URN and NRH. For instance, when the propellers are identified as the main root cause of vibration on board, it is not possible to say the same for URN or NRH. These types of widely spread assertions must be backed by experimental information.

Assessment of the SILENV-N&V database that will be described in the following paragraphs is aimed at enriching these guides and providing a global picture of the N&V impact not only on crew/passenger health (N&V on board), but also on the environment (NRH and URN). The dedicated URN and NRH measurement activities carried out in two vessels can be seen in Fig. 2.

Finally, to fulfil the holistic approach committed by the SILENV project, assessment of the SILENV-N&V database will be done by following the next schedule:

At a first stage, the N&V Targeted Limits must be defined; not only related to N&V on board, but also for NRH and URN. Hereinafter, we will refer to these targeted limits as Preliminary SILENV Limits (PSL).

In a second step and once the targeted limits have been defined, the assessment is ready to be performed. One of the main goals, and so novelties of this assessment is that it is focused on providing not only a realistic consistent picture of the current fleet with regard to the N&V standards, as well as new directives, but on identifying the formal and technical absences identified as root causes of the deviation detected. Finally, through the corresponding Sensitivity Analysis, improvements and recommendations to improve the current status will be provided.

Targeted limits definition

N&V on board Preliminary Targeted Limits

The preliminary limits have been selected from the most restrictive limits among existing ones to anticipate new-coming trends (Fig. 3).

Noise Radiated to Harbour Targeted Limits

Regarding this topic, after a detailed and well-documented analysis of the current EC

Fig. 2. URN and NRH measurement

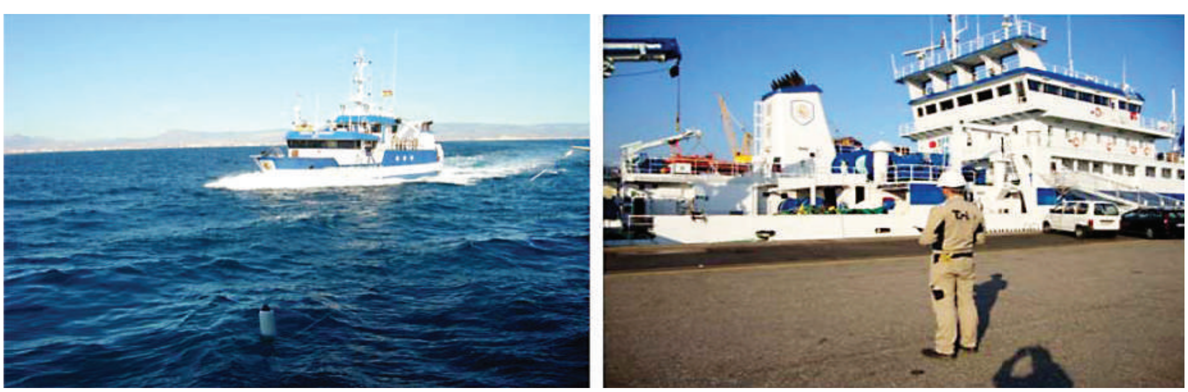
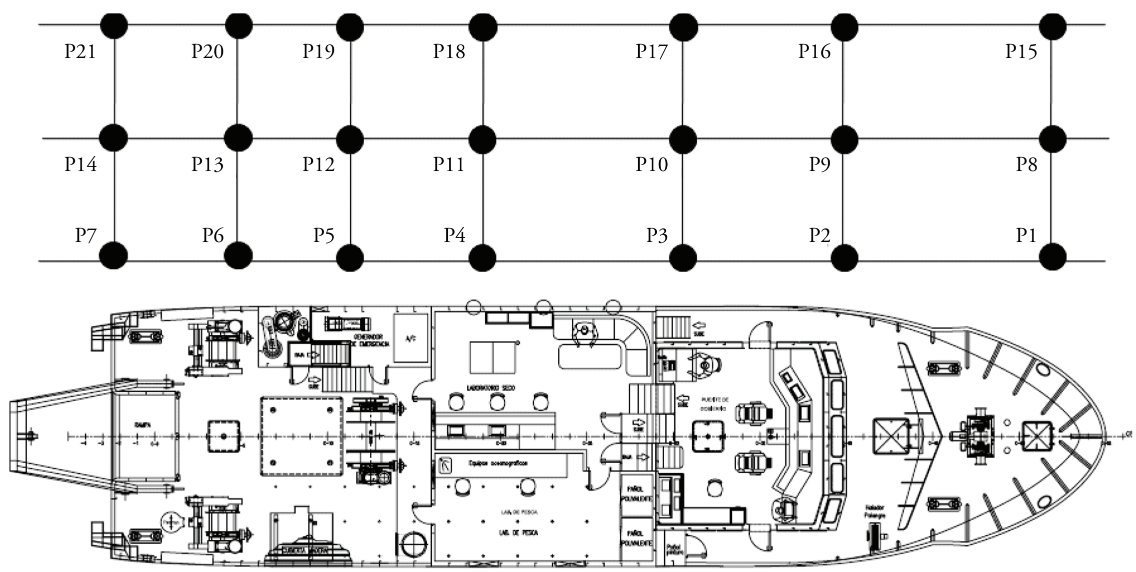


Fig. 3. URN and NRH measurement

SILENV- Noise Limits (dB(A))			
CODE	DATA BASE	SILENV SPACES (WP 1)	Limit (dBA)
I	Passenger Cabins	Passenger top level Cabins	44
		Passenger standard cabins	45
II	Crew cabins	Crew Cabins	50
		Officers Cabins	
		Day Cabins	55
III	Offices	Oficces	55
		Radar room	
		Chart rooms	
		Radio room	
IV	Public Spaces	Hospital	45
		Libraries, theatre	50
		Gymnasium	52
		Discotheque, Ballroom	
		Shops	
		Restaurant, Lounge	
		Closed public spaces	55
		Dining spaces	
		Recreation	57
		Corridors, Staircase	58
V	Crew Public Spaces	Hospital	50
		Corridors, Staircase	54
		Closed public spaces	55
		Mess room	57
		Alleways, changing room	70
VI	Outdoor Recreations Areas	Outside installation	64
		Open deck recreation	65
VII	Wheelhouse	Wheelhouse	55
		Navigation spaces	
		Listing posts, Bridge wings	65
VIII	Work Spaces	Open desk working areas	63
		Pantries	66
		Galleys	68
		Stores	70
		Laundries	75
		Workshops	80
		Cargo handling spaces/areas	
		Fan rooms	85
IX	Control Room	Engine control room	65
X	Auxiliary Engine Room	Continuously manned machinery spaces	85
		Not continuously manned machinery spaces	105
XI	Engine Room	Continuously manned machinery spaces	85
		Not continuously manned machinery spaces	105

SILENV- Vibration Limits (mm/s-rms)			
CODE	DATA BASE	WP 2.1	Limit (mm/s) RMS
I	Passenger Cabins	Passenger top level Cabins	0,8
		Passenger standard cabins	1,2
II	Crew cabins	Officer Cabins	1,2
		Crew Cabins	1,6
III	Offices	Radio room	2
		Oficces	3
IV	Public Spaces	Hospital	1,2
		Libraries, theatre	1,4
		Gymnasium	
		Discotheque, Ballroom	
		Restaurant, Lounge	
		Corridors, Staircase	2
		Recreation	
		Shops	
		Closed public spaces	3
V	Crew Public Spaces	Mess room	2
		Closed public spaces	3
		Alleways, changing room	5
VI	Outdoor Recreations Areas	Outside installation	2
		Open deck recreation	2,6
VII	Wheelhouse	Navigation spaces	2
		Listing posts, Bridge wings	2,8
VIII	Work Spaces	Galleys	2
		Pantries	2,4
		Workshops	3,2
		Stores	4
IX	Control Room	Engine control room	2,4
X	Auxiliary Engine Room	Not continuously manned machinery spaces	4,4
XI	Engine Room	Not continuously manned machinery spaces	4,4

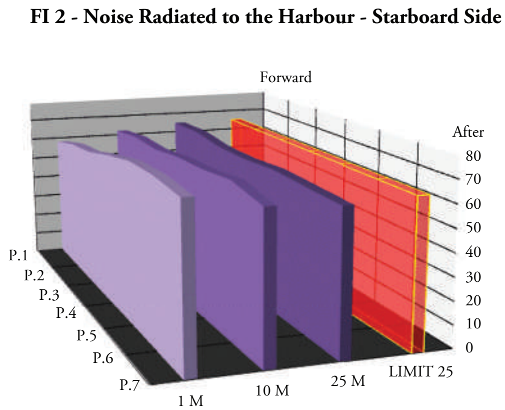
Fig. 4. Port side. Measurement grid locations

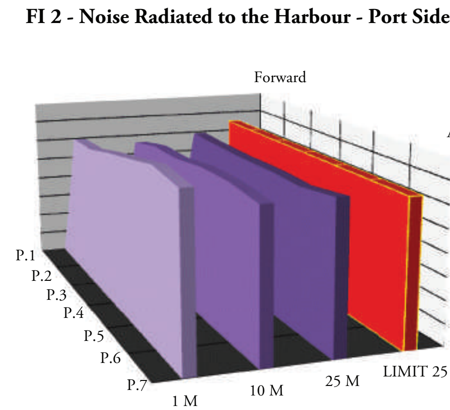


Directives, as well as of national legislation of several Member States, the preliminary targeted limits for the Noise Radiated to Harbour, for all types of vessels (in coincidence with Directive 2002/49/EC), have been the following: the noise levels (expressed as Leq db(A)) radiated by the vessel when it is berthing at a distance of 25 m from both hull sides, shall be lower than 65 db(A). To carry out these measurements, a dedicated SILENV measurement procedure has been developed. This procedure defines a grid of the measuring points at 1, 10, and 25 m, on both sides of the targeted vessel (Fig. 4).

To facilitate the corresponding assessment, the results obtained must be presented graphically in the form shown in Fig. 5.

Fig. 5. Fishing Vessels 2. URN measurement results

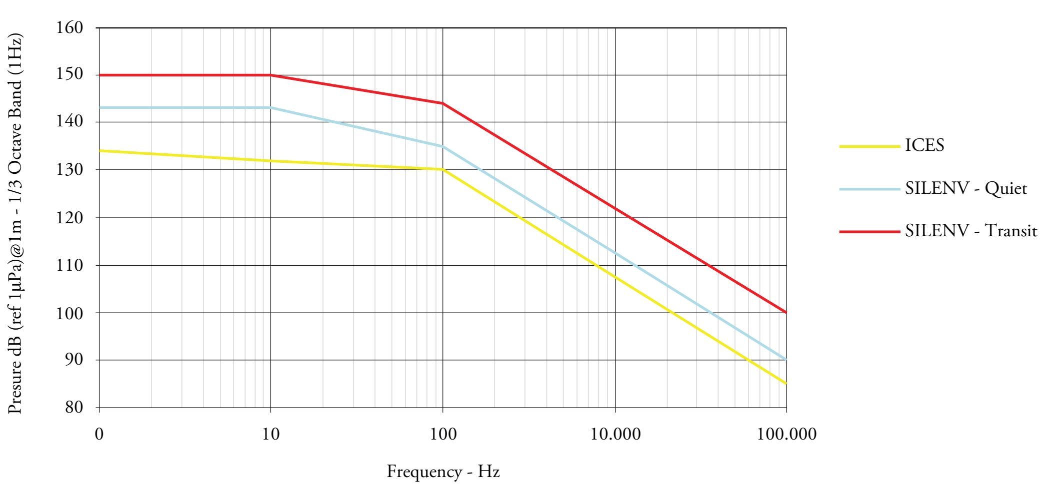




Underwater Radiated Noise Targeted Limits

As previously, after a detailed review of the state-of-the-art, a preliminary proposal for the URN targeted limits was made (Fig. 6). Regarding the URN topic, the vessels were sub-divided into two different groups: one that included Fishing Research Vessels and Fishing vessels; and another that grouped all the commercial vessels. For the first group, the URN targeted limits adopted were the well-known ICES Nº 209 Regulation limits at 11 knots. Based on experience, these targeted limits are consistent with modern FRV needs and are technically and economically feasible. For the second group of vessels, the final URN targeted limits adopted were a combination between the ICES Nº 209 and that proposed by DNV in its optional Silent Class Notation limits. In this case, two operational conditions were adopted: SILENV-Quiet at 11 knots and SILENV-Transit at normal sea-going conditions. In all the cases, the results shall be provided in terms of pressure in dB (ref 1 μPa) in 1/3 octave band (1 Hz) referred at 1 m from both hull sides of the vessel. Using as a guideline the ANSI-ASA standard, a dedicated SILENV URN measurement procedure was established to harmonize the experimental data provided by the different participants. The propagation model adopted for the transmission lost was $TL = 20 \log D_{CPA}$.

Fig. 6. Preliminary SILENV URN targeted limits



The global N&V assessment: the assessment matrix.

The Aim and Scope of the Assessment

Thus and once:

- The targeted limits related to the different topics making up the N&V-FS of a vessel have been established.
- The SILENV- N&V Database has been compiled; the assessment could now be carried out.

The main targets of the assessment are:

- *Comparison of data and measurement results with the preliminary targeted limits adopted*. This will permit answering the question of how far are the current different types of ships from the preliminary SILENV targeted limits.
- *Root cause/s identification of the deviations detected*. This second level of assessment focuses on accurately identifying not only the N&V sources that cause the N&V deviation but the technical lacks or absences that motivate the deviation observed among the different N&V topics that make up the N&V-FS, and the preliminary N&V limits.

To conduct the assessment, it has been subdivided into the three following levels or phases:

- Assessment Level 1: Comparison of ships with the preliminary targeted limits (PSL).
- Assessment Level 2. Phase 1: Effect of the N&V Control techniques applied during the design processes.
- Assessment Level 2. Phase 2: Identification of the main N&V sources as root cause/s of deviations.

Finally, this analysis has been completed with the "Sensitivity Analysis" that will provide the improvements/recommendations required to meet compliance with the targeted limits. It will consist on assessing and evaluating how the application of these blocks could contribute to building vessels that comply with the new SILENV limits, or at least as close as possible.

Thus, once the aim and scope of the assessment activities have been widely explained, the methodology used to complete this commitment will be described in the following paragraphs. In other words, *what should be done and how should we do it*, will be detailed step-by-step.

The Utilities developed

Based on the expertise of some partners and considering the arduous task entailed in assessing thousands of N&V experimental data coming from many vessels, at the beginning of the project the update of the dedicated "TSI- N&V Tool" was foreseen. Likewise and focused on automating the assessment process as far as possible, a whole package of dedicated "utilities" were foreseen and have been developed. These utilities are:

- The *"Vessel Excel Sheet"*. Its purpose is to homogenously compile the N&V experimental data, as well as the technical information from each of the vessels. This document has been generated for all the vessels.
- The *"Vessel Card"*. It summarizes the main particulars of each vessel, as well as the answers to the required questionnaires related to N&V issues.
- The *"Vessel Graphs"*. Using the "TSI-N&V Tool" the graphical reports (one for each topic) of the N&V numerical data compared with the N&V targeted limits adopted have been automatically generated for each of the vessels in the database. In Fig. 7, the *"Noise and Vibration Vessel Graphs"* of fishing vessel (FI2) (FI2 vessel is one of the many vessels compiled in the SILENV-N&V database) are reported.

Similarly, the tool organizes the information by types or families of vessels. For example, Fig. 8 shows the N&V information of the Merchant vessels.

Using this information, it is easy to conduct statistical analysis related to the distribution of the N&V levels on board regarding both the PSL and the Standard limits: IMO and ISO. This kind of analysis has been done for all the families of vessels analysed.

For the assessment of the other topics of the N&V-FS: NRH and URN, the way information has been treated was different when it came from

Fig. 7. N&V Vessel Graphs. Fishing Vessel FI-2

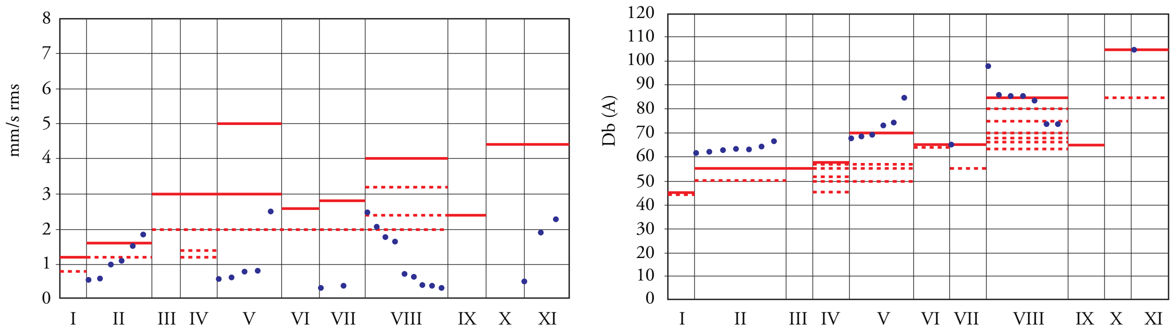
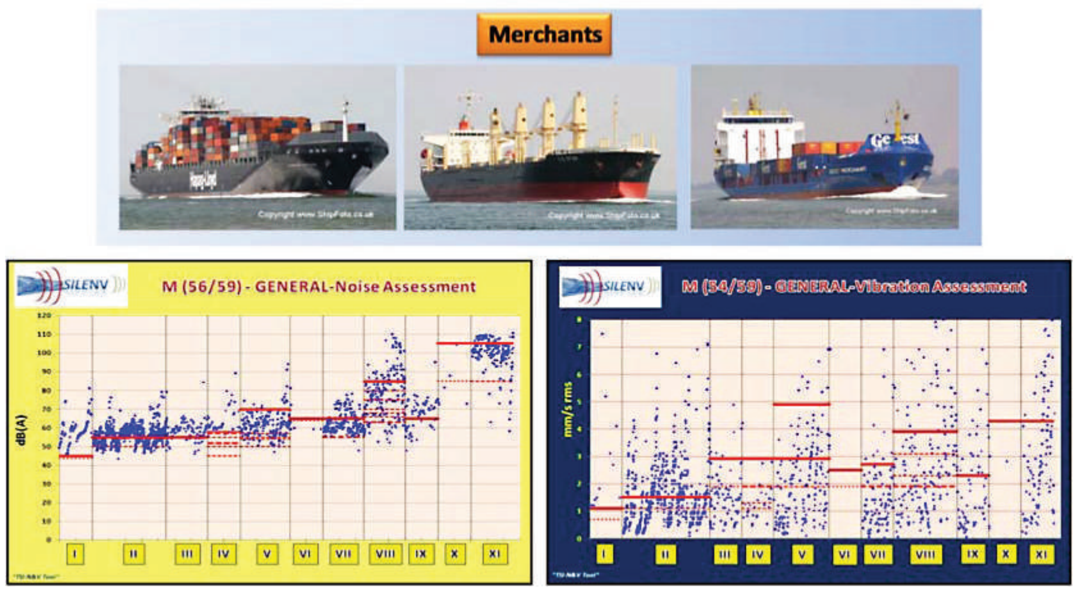


Fig. 8. N&V distribution for Merchant vessels



vessels that had previous information available or from vessels that had to be measured. In the first case (vessels that had previous information), due to the low availability of NRH and URN data, the assessment had to be done case by case. In the second case (vessels that had to be measured), the "Assessment Utility" generated a report like the one seen in Fig. 9.

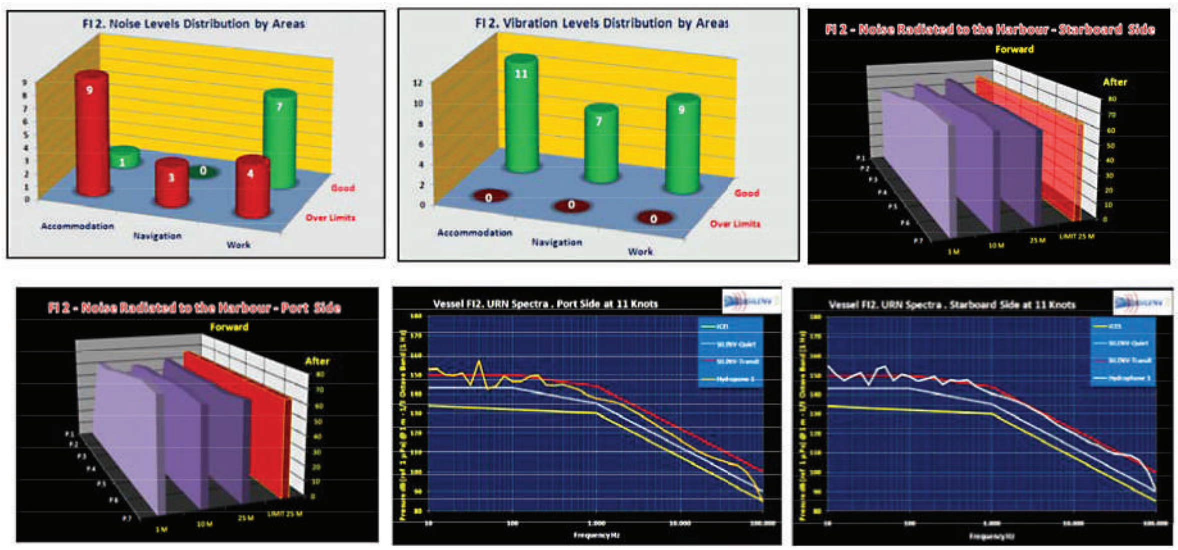
As noted, this utility that graphically summarizes all the experimental results from the corresponding "Measurement Report", permits ding – whenever possible – the complete assessment of the N&V-FS for each of the vessels comparing them with the PSL.

Thus, and using this collection of utilities, the so-called Assessment Level 1: comparison of ships with the PSL has been fulfilled.

The Assessment Tool: The Assessment Matrix.

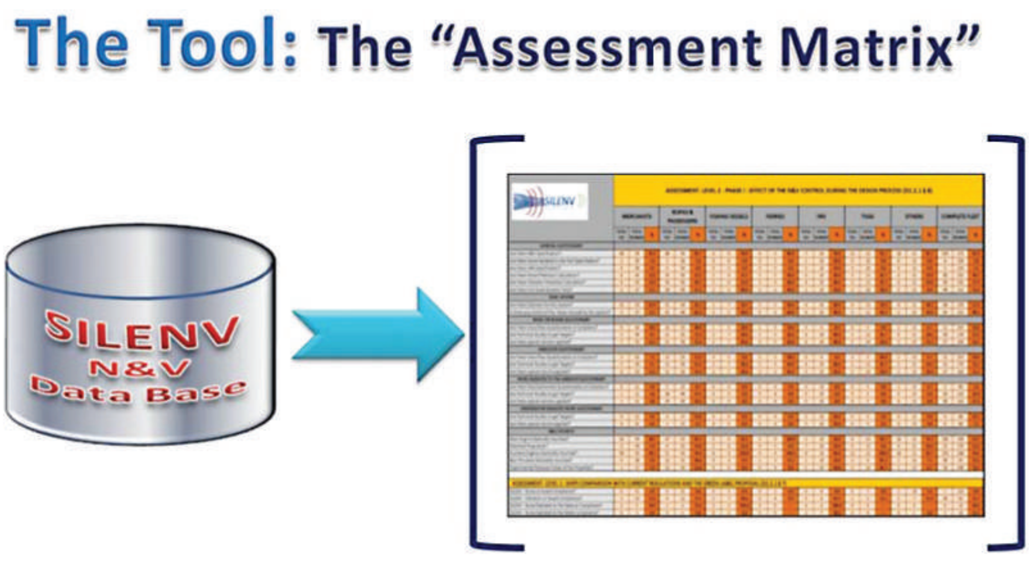
The aim of this tool was to easily summarise the results and the analyses carried out by the different partners involved in this task. Due to the large number of vessels analysed and because of the scope of the assessment, the proper and suitable use of this tool (the Assessment Matrix) has enabled extracting the general conclusions. Through these conclusions, it may be found if the vessels comply or not with the PSL. Likewise, it has enabled identifying if the application of the technical

Fig. 9. "Assessment Utility" for FI2



efforts, during the design process, could positively or negatively affect compliance. Additionally, the partners performed deeper analysis of the vessels under their scope. In these specialised analyses, the partners identified the N&V sources causing the deviations by using the corresponding spectra. The general process followed is summarized in Fig. 10.

Fig. 10. Assessment Process



To fulfil the targets, the rows of this Assessment Matrix have been subdivided into three different sub-groups. In the first column, the questionnaires that permitted studying the different topics have been placed: compliance (or not) with the limits, technical efforts applied, and identification of N&V sources. Complementarily, the columns of the matrix included all the details of the vessel: type of vessel, age, partner involved, and origin (vessels that had previous information or vessels that had to be measured). In other words, all the information available on the vessels, which was processed and compiled by the previously described "utilities tools" can be found in this Assessment Matrix.

In the Assessment Matrix, the questions were answered by using a (1) *(positive answer)* or a (0) *(negative answer)*. These answers will permit easily carrying out a statistical analysis. The columns left *"empty"* mean there is no information available related to that particular question. By comparing the results of vessels that comply with the targeted limits to those that do not fulfil them and correlating their particularities (N&V control activities during the design process, calculations performed, corrective solutions applied, etc.), it will be possible to develop a "Sensitivity Analysis". Its aim is to analyse if these preventive solutions are good enough to obtain vessels that comply with the limits, or at least if they can be close to meeting these requirements.

Finally, the third group of rows is dedicated to the answers that will enable accurate identification of the N&V source/s responsible for the N&V deviations detected. In this concern, it is remarkable that all the analyses have been carried out by each partner, using their own expertise and the N&V spectra.

It is assumed that in many cases, especially for vessels with previous information, the level of information provided could be limited by confidentially agreements. Of course, this unavailability will not permit reaching full and consistent conclusions. Anyway, in other cases, this information was achieved through other questions. These answers have permitted seeing the trend of the topic under consideration. These trends found are supported by the information compiled on each vessel and by the corresponding Mapping Analysis made.

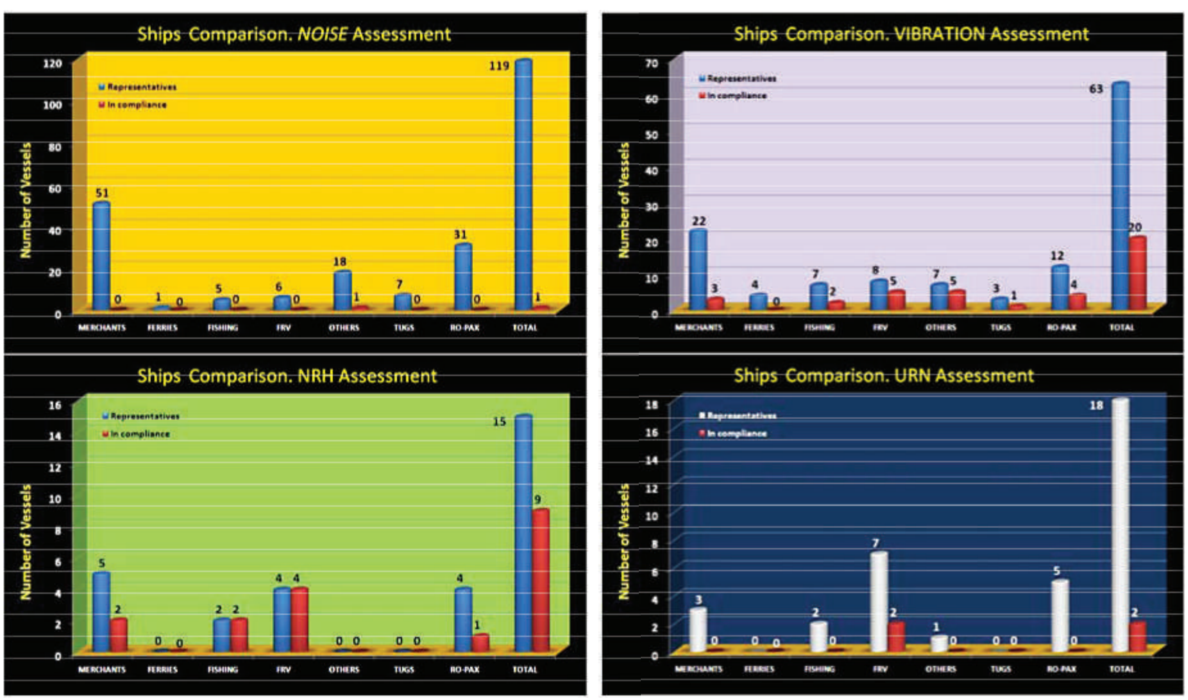
Assessment Results

Results of the Assessment will be summarized according to the different assessment levels and phases adopted (§ 4.3 a).

Comparison of ships with the preliminary SILENV limits Assessment level 1

The status of the fleet and of each of the families of vessels that comprise it with regard to the

Fig. 11. Current N&V status of the families of vessels and of the fleet related to the PSL



preliminary SILENV limits adopted has been graphically summarized in Fig. 11.

As a summary, in all the 171 vessels comprising the SILENV-N&V database none has an N&V-Full Signature in compliance with the Preliminary N&V SILENV limits adopted for the different issues that compose this N&V indicator; in most cases because it has been unfeasible to assess the vessels, given the absence of dedicated data related to some issues of the N&V-FS. Likewise, and related to the 10 vessels for which the N&V-FS has been obtained by the dedicated on-site measurement activities, none fulfils the PSL adopted. Only the Fishing Research Vessel "FRV-4-TSI" is the closest vessel to fulfil it. In fact, for this vessel the vibration on board, the noise radiated to harbour, and the underwater radiated noise are in compliance with the preliminary SILENV limits, but not the Noise on board.

Effect of N&V control during the design and building process

Assessment level 2. Phase 1

The experience has permitted confirming that the achievement of vessels that are in compliance with any N&V limits established is not a question of luck but the consequence of a suitable implementation of a whole set of preventive actions focused on achieving the N&V targeted limits. These actions shall be applied conveniently and at the right time (at the early stages of the project and during its construction). The aim of this level of assessment during phase I has focused on assessing the "current trends" related to these preventive actions or efforts being applied on the different families of vessels and, therefore, on the current fleet to tackle and control the different N&V topics included in the N&V-Full Signature of each vessel. To coherently fulfil this target, preventive actions or efforts have been grouped into the following blocks:

Block 1- General Questionnaire: Has enabled assessing the presence of N&V requirements related to the different N&V topics: N&V on board, NRH, and URN in the contract specifications, as well as the extension of the use of technical preventive actions, particularly those related to the use of simulation tools and tests focused on achieving the N&V limits defined in the contract specifications.

Block 2- Technical Preventive Actions during the Early Design and Building Process: Has enabled assessing the current technical efforts developed by the different families of vessels and by the complete fleet, during the early design and building process and is focused on reaching, separately, compliance of all the N&V topics of the N&V-FS.

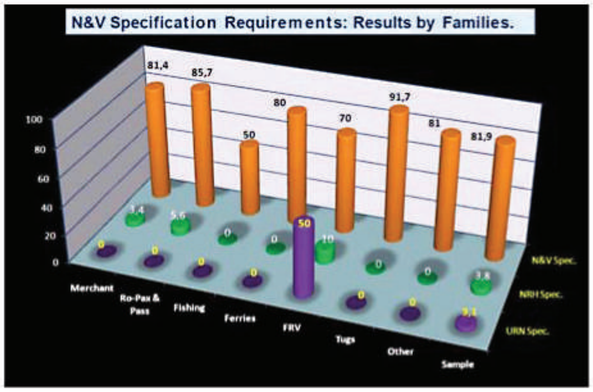
Block 3- Proper Selection of the N&V Sources: According to the basic principle of N&V Control Methodology the "proper selection" or, in other words, the selection of the most "silent" sources with regard to intensity, airborne and structural noise of the N&V sources is the cheapest and easiest technical way to achieve the targeted limits. Consequently, this block has enabled assessing, for the fleet under control, the right application of this previous principle. Finally, a dedicated and specific question related to the experimental pressure pulses fluctuations induced by the propeller is included. These pulses, obtained with experimental basin channel models or with the use of full-scale tests, have also been introduced. It has enabled assessing with a numerical indicator if there has been any kind of control of the main N&V sources' intensity. Detailed analysis of these aspects would be sufficient for a paper; thereby, in the following paragraphs the most relevant issues will be summarized.

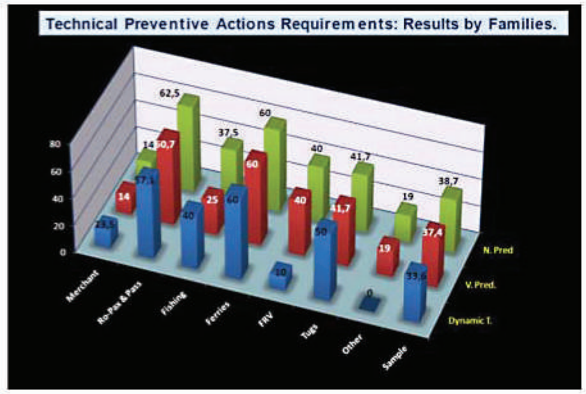
With regard to the Block 1 activities, the status found is reported in Fig. 12.

Examination of the previous figure enables highlighting the following issues:

- The requirements of Noise and Vibration on board are well extended in the current Contract Specifications with a percentage that achieves the value of 81.9% of the total sample of vessels (171). It shows that these topics directly related to human health, working conditions of the crew, and passenger comfort are an extended concern for owners and shipyards.
- On the contrary, this previous trend is not observed nowadays in the current Contract

Fig. 12. N&V Specification requirements & Technical Preventive actions





Specifications of the vessels with regard to the noise radiated by them to the outside. In fact, from the total number of vessels in the sample (159) that could be analysed, 3.8 and 3.1% of them included specific NRH and URN requirements in their Contract Specifications, respectively.

This fact, related to the low presence of NRH and URN requirements in the specifications of the current vessels, may be seen as logical if we consider that these topics are quite new. All the new environmental regulations are quite recent, but will be compulsory shortly.

- Focusing attention on the spread of the use of simulation tools: N&V Prediction Calculations, as technical preventive actions focused on achieving vessels that comply with the N&V requirements, it is an outstanding fact that only 38.7% of the vessels have a Noise Prediction calculation and only 37.4% have a Vibration Prediction calculation, which clearly shows a very poor level of spreading.

In relation to Block 2 of the technical preventive actions and, particularly, with those related to the HVAC system (identified as one of the main root causes of noise on board), the statistical analysis shows that only half the vessels from the sample have adopted preventive actions like installation of silencers or other preventive control actions on this system. It is highlighted that only in the family of Ro-Pax and Passenger vessels these preventive actions are widespread. This is a clear and consistent indicator of how concerned the owners of these types of vessels are, not only about compliance with noise limits in all the cabins but also in open spaces for recreation.

The assessment results related to the rest of the technical preventive actions included in Block 2 under the general denomination of Questionnaires with regard to each of the topics of the N&V-FS: N&V on board, NRH, and, URN show a wide variety of solutions.

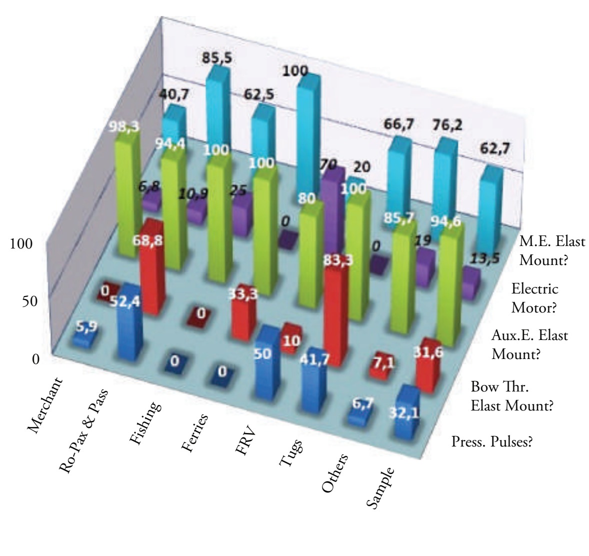
Regarding N&V on board, given that both topics are included in the same package of the contract specifications, the results obtained are quite coincident. Indeed, for the sample of vessels analysed (171) the percentage of them that employ simulation tools or technical studies focused on complying with the Noise and Vibration requirements achieves values of 32.9% for noise and 31.6% for vibration, respectively.

Regarding NRH, the assessment results obtained for a sample of 84 vessels show in, general, that vessels do not apply technical studies and technical solutions to combat their outdoor radiated noise, with the only exception (only 26.4%) of Passengers and Merchant vessels. Similarly, and related to URN, only 10.2% of the total sample of vessels analysed (88) have reported using dedicated simulation tools as, well as the application of dedicated technical solutions to achieve compliance with URN requirements (in those exceptional cases were they are required) or on minimising their underwater footprint impact. The biggest contribution to this result is made by the fishing research vessels, which is also starting to be observed in passenger vessels.

These results related to the application of dedicated techniques and solutions to combat the NRH and URN impact of the vessels is a direct consequence of the identified absence of specific NRH and URN requirements in the contract specification, already reflected in Fig. 12. They obey the basic principle of "nobody will do something if they are not forced to".

Finally, the assessment results related to the Technical Preventive Actions included in the Block 3- Proper Selection of the N&V Sources and focused on reducing or minimising the intensity of the excitation forces or the noise induced by the different main N&V sources: machinery and propeller, etc., that have been adopted during the early design of the vessels are summarized and reported in Fig. 13.

Fig. 13. Proper selection of the N&V sources



It is assumed that in most cases, until now, the selection, primarily of the main machinery and propeller, has been related to fuel consumption and to economic aspects of exploitation of the vessel. On the other hand, if vessels wish to comply with N&V requirements and especially want to abate their underwater footprint, it is possible that the previous selection could be affected and limited in some aspects. Therefore, for the propulsive and economic aspects it will be necessary to add new criteria and make a sensible balance between them. It must never be forgotten that according to the experience, nowadays the right selection of the N&V sources is the cheapest way to achieve the N&V targets. When these preventive actions are not possible to be carried out, care should be taken related to the proper insulation of the transmission paths. So, the aim of the assessment of this Block 3 of activities is to know how widespread are the preventive actions focused on minimising the incidence of the N&V sources, within the current fleet.

Examination of the previous Fig. 13 enables highlighting the following aspects:

- As noted, elastically mounting the main engines is an extended countermeasure to minimise their influence in N&V, within the current fleet, with an average value of 62.7% (169 vessels have been analysed). All the families, except the merchant and fishing research vessels, are over this average. The deviations for these two families of vessels are related to the use of two-stroke diesel engines, for the merchant vessels, and to the use of an electric motor rigidly connected to the hull, for the fishing research vessels.
- In relation with the topic of the use of electric motors, it is clear that this type of propulsion is not as widespread within the current fleet, only 13.5% of the sample of 171 vessels uses it. The vessels that use them more are those with the most restrictive URN requirements like fishing research vessels. Electrically driven propulsion is implemented in 70% of the FRV. The other families of vessels do not use this solution very much.
- With regard to the use of elastic mounts on the auxiliary engines, the high value obtained, 94.6% (168 vessels were considered), enables us to state that this preventive solution is a very well widespread standard within the current fleet.
- The bow thrusters have usually been identified among the main noise sources when they are operating, during the berthing and un-berthing transient periods. Based on the expertise of some partners, and due to the unavailability of experimental data that

enables properly characterizing this source, a double skin design is recommended. This design has proven to be a useful preventive solution to reduce their impact.

- Finally, with regard to the use of pressure pulse fluctuation measurements through basin channel models, as an approach to estimate and control the intensity of the excitation forces induced by the propeller, this measure is very poorly spread (32.1% of a sample of 106 vessels). As seen, the major contribution came from passenger, FRV and tug vessels, respectively.

Identification of the root cause/s of the deviations detected

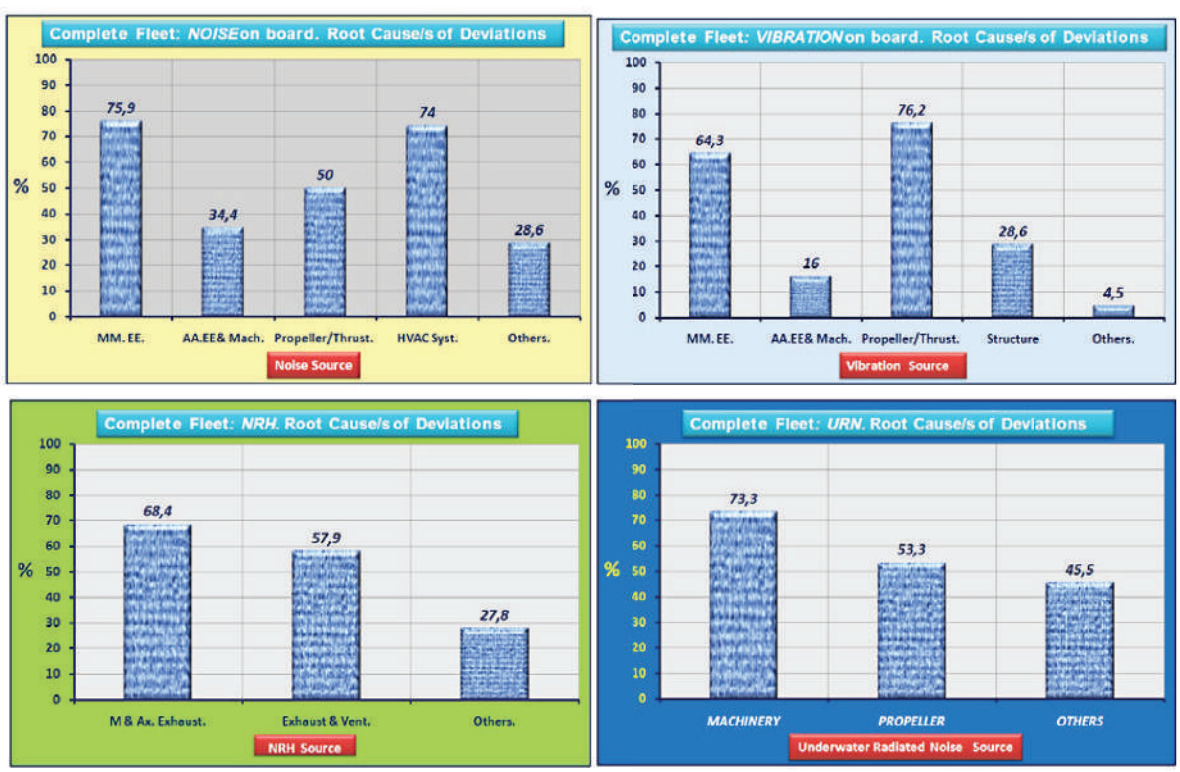
Assessment level 2. Phase 2

The numerical assessment related to identifying the root cause/s of deviation of N&V on board, NRH, and URN compared with the PSL has been reported and it has been separately done for each of the families of vessels, as well as for the entire fleet.

Finally, a general picture of the fleet with regard to the root causes of the deviations is depicted in Fig. 14. Examination of Fig. 14 and of the numerical assessment results obtained enables highlighting the following aspects and conclusions:

- With regard to noise on board, the main engine, the HVAC system, and the propeller/thrusters have been identified as the major root cause of noise deviation with regard to the PSL.
- Related to vibration on board, the propellers and the main engines appear to be the main sources of vibration that cause the deviations with the PSL. This fact is also seen in other topics, but it does not mean that it could be extended to other topics, for example URN.
- The exhaust of the main and auxiliary engines, as well as the inlet/exhaust ventilation systems, has been identified as the main factor responsible for the deviations of the outdoor radiated noise when compared with the PSL.
- The machinery and the propeller, especially when cavitation appears, have been identified and, in this order, as the major root cause/s of

Fig. 14. Entire Fleet. Root Cause/s of N&V deviations



the deviations detected in the noise radiated by the vessel into the water with regard to the preliminary URN targeted limits. This approach is consistent enough with the experience documented at Glacier Bay, mainly related to passenger vessels.

Additionally, and based on these results, experimentally well-supported and in clear contradiction with the general and well-extended assertions (but with no experimental or consistent data to back them), it would be recommendable to completely review them to find a new starting point that will enable us to look forward together (scientific community and marine sector), for solutions that could improve the current state.

Sensitivity analysis to achieve the preliminary n&v targeted limits adopted

The "Sensitivity Analysis" carried out within the framework of the SILENV project is aimed at identifying the improvements or complementary technical efforts required to achieve compliance with the preliminary SILENV targeted limits. The main steps and the strategy followed to develop this dedicated activity of "Sensitivity Analysis" were the following: in a first step, the "numerical gaps" or deviations among the experimental numerical N&V on board, NRH, and URN values measured in the different families of vessels and the preliminary SILENV targeted limits were obtained. Complementarily to this analysis, in a second step, the "numerical gap" or deviation among these values and the standard IMO and ISO limits (or any other limits) were also obtained.

Once these numerical gaps have been identified and the dedicated technical efforts done by each family of vessels, and so by the fleet, are well known, the "deficits" in terms of technical efforts will also be identified. So, based on these "deficits", it will be possible to advance, as a first approach, in the complementary efforts and technical improvement solutions that should be required to get close to the preliminary SILENV targeted limits.

Noise & Vibration statistical numerical deviations

The statistical analysis, for each family of vessels, was made by comparing the N&V numerical values available, from each main space, with the following limits: Current Regulation Limits (IMO A.468(XII) Regulation, for noise and ISO 6954 (2000), for vibration); upper and lower preliminary SILENV limits for each space.

In each graph and for each space, the targeted limits used and the "average value" of the sample of available data are reported. Likewise, the percentages of deviation for each space over the predefined limits are also reported. As explained previously, these graphs (N&V) were obtained for all the families of vessels.

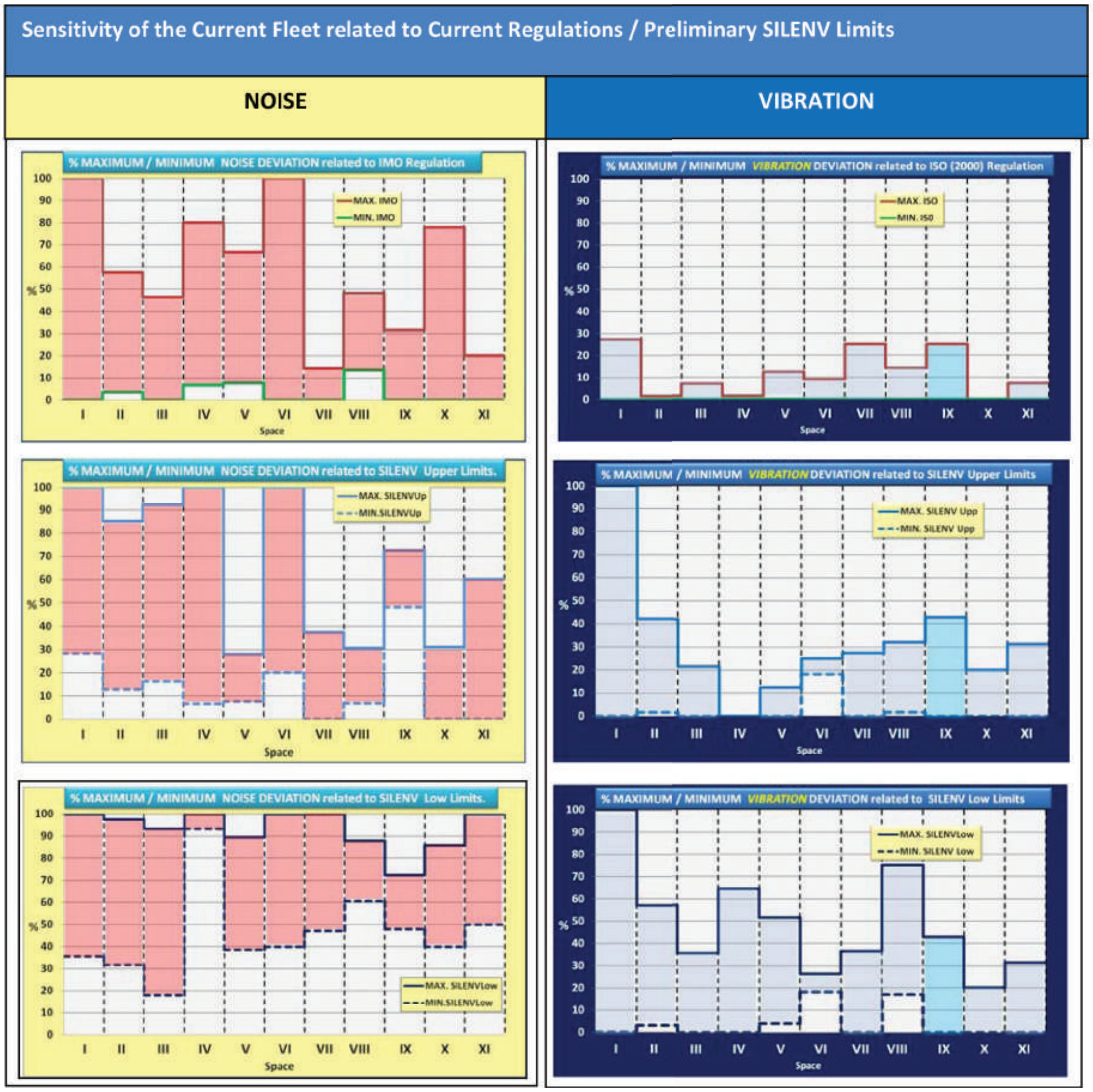
Current fleet. Sensitivity analysis of the change of N&V limits

In a first step, (to characterise the entire Fleet) and based on the previous results, the *Maximum* and *Minimum* percentage deviations of N&V regarding the current N&V limits and the upper and lower preliminary SILENV N&V limits were obtained. In fact, this information becomes a consistent indicator of the status of the current fleet with regard to these N&V limits. In fact, and as previously mentioned, the particular N&V deviations for each family of vessels is comprised in the area bordered by those *Maximums* and *Minimums*. These results are summarized in Fig. 15.

Examination of the previous Fig. 15 and analysis of the results obtained related to the Sensitivity Analysis for the current fleet with regard to the different noise limits: standard current limits (IMO) and the PSL, has enabled us to highlight the following aspects:

- For all the families of vessels analysed, in some of them and at some spaces, the general percentage trends of noise level deviations with regard to the current noise IMO A-468(XII) standard achieves values of 100%. For these families of vessels and consequently for the whole current fleet, independent of compliance

Fig. 15. Current Fleet. Max/Min. % of deviation related to standard and PSL



by some particular vessels, compliance of the noise levels with any other more restrictive noise limit like the preliminary SILENV is not feasible.

Likewise, examination of Fig. 15 and the results related to the Sensitivity Analysis of the current fleet with regard to the different vibration limits: current standard ones (ISO 6954 (2000)) and the Preliminary SILENV vibration limits, enables us to highlight the following aspects:

- In general terms, and as far as the vibration levels are concerned, the current fleet is close to complying with not only the current ISO 6954(2000) standard but also with the preliminary SILENV vibration limits. The same cannot be said for compliance with the noise levels of the IMO standard and with the preliminary SILENV noise limits.
- In fact, different families of vessels exist that not only comply with the ISO standard vibration level, but which almost comply with the preliminary SILENV vibration limits in nearly all spaces.
- Thus, compliance with the vibration limits of the ISO standard and of the PSL could be

more achievable than compliance with the noise limits.
- The results obtained from the Sensitivity Analysis with regard to the change of the vibration limits clearly show that the current situation could improve (the deviation) if the preliminary SILENV vibration limits were raised slightly. By doing this, we will have more permissive limits, but they will still be very demanding. In any case, full and strict (at all spaces and with no deviations) compliance could be achieved.

Cross comparison: efforts compared to the N&V results. Unitary model approximation.

From the authors' point of view, and taking into account their expertise (they have dedicated many years to the design of vibration-free and silent vessels [6]), the following idea has to considered: *"Compliance of a vessel with any N&V limits is the result of the proper application of a whole set of tailor made efforts and technical solutions. These must be well-pondered and applied at the right time/stage of the project".*

On the other hand, the activities developed within the framework of the assessment related to the effect of N&V control during the design process §5.2, has permitted noticing that some "deficits or lacks in the technical efforts" exist that will hinder the final objective of complying not only with the PSL but also with the IMO Standard and with the ISO Regulation.

Consequently, the aim of the new sensitivity analysis, in terms of efforts applied, has focused on answering the following two key questions:
- What has been done until now, in terms of technological efforts, in the different families of vessels of the current fleet in order to abate N&V?
- What additional efforts and technological solutions should be applied to the new constructions in order to reduce their environmental impact?

As a result of this "Sensitivity Analysis" that will be expressed in terms of Δ of Percentage (%) of Compliance with the N&V limits by unitary effort applied, the lack of efforts required to achieve compliance with the Standard Regulation could be estimated and, consequently, the additional efforts and technical improvements the new constructions need. The previous analysis was made separately for each of the different N&V topics included in the N&V-FS. The results obtained were summarized in the following way:

Noise on board

The total technical efforts applied (for the complete fleet) to abate noise on board and, therefore, achieve full compliance are 49.8%. The efforts are logically not applied in the same manner in every family. Passenger vessels and ferry vessels are the families applying the maximum efforts.
Related to this topic, the sensitivity analysis enables stating that full compliance with the IMO standard and with the upper PSL will be achievable and it would only require the promotion of the following different dedicated activities, organised by Blocks:
- *Specifications:* Making it compulsory for noise requirements to appear in the entire contract specifications.
- *Simulation Tools and Test:* Fully applying all these dedicated activities: noise prediction calculations and technical studies.
- *Noise Source Control:* Improving the activities of this block, especially the generalisation of the HVAC silencers, HVAC control, elastic mounting of the main and auxiliary engines, and installation of elastically mounted thrusters.
- *Engineering Solutions:* Taking into account the current "State–of-the-art".

Compliance of the current fleet with regard to the "strict application" of the strictest lower PSL is not achievable. The approximation made with the Unitary Model has permitted estimating that this compliance would not be reachable even if additional "improvements" were made. In fact, all the activities of the different blocks (specification, simulation tools and tests, noise sources control and engineering solutions, additional technological improvements of the solutions) should be implemented if we want to achieve those limits.

Among others, as "Technological Improvements" the following could be mentioned: replacement of noisy sources by other silent ones; for instance, main propulsion based on electric motors instead of diesel engines; special and dedicated design of elastic suspensions to minimise structural noise contribution; encapsulation of the main noise sources, etc. As previously mentioned, for all the cases, the costs/benefits ratio shall be assessed.

Vibration on board

The total technical efforts applied for combating this topic are 45.4% of the total efforts required to reach full compliance. Passenger vessels and ferry vessels are the two families applying the maximum efforts.

Similar to what happened in the case of noise on board, particularly to the efforts made to comply with the current ISO-6954 (2000) Standard and with the preliminary SILENV Upper and Lower Vibration Limits, it must be stated that they are not fulfilled.

Anyway, the high percentage values of compliance identified are a good indicator of the efficiency of the efforts currently applied by the shipbuilding industry to minimise the impact of this topic.

In general, full compliance of new vessels could be achievable if the proper corrective actions were applied. As noted through the previous analysis, achievement of the targeted vibration limits for the new constructions is conditioned by the application of the dedicated activities included in the following blocks:

- *Specifications:* Making it mandatory for vibration requirements to appear in the contract specifications of new constructions.
- *Simulation Tools and Tests:* By reinforcing and, in some cases, including it as a requirement in the specifications of the development of Vibration Prediction Calculations and Technical Studies required in achieving the targeted limit required.
- *Vibration Sources Control:* In general, reinforcement of all the activities within this block and focused on minimizing the excitation forces induced by the different vibration sources, especially the propeller.
- *Engineering Solutions:* In some families of vessels, these activities should also be reinforced to make up for the lack of available silent and competitive supplies in the current market.

Noise Radiated to Harbour (NRH)

Related to this novel N&V topic, the technical efforts currently applied by the fleet are 19.2% of the total required, which is a very low value. The low trend of technical efforts deployed by the fleet to confront this disturbing topic is directly related to the entire absence, as it has been proven, of specific requirements in the contract specifications of most of the current vessels, including the most modern ones. Only those owners that have been occasionally affected by these complaints are starting to include these specific requirements in their contract specifications. Once again, the principle *"no one does anything if they are not forced to"* is confirmed.

Thus, despite the appearance of the Directive 2006/87 EC [2], the presence of these types of requirements is still strange, if we consider that this regulation has recently become compulsory. Furthermore, from the authors' point of view, there is no harmonisation in the different limits. Moreover, its application changes from one member state to another, in fact, it can change from one harbour to another within the same country. Consequently, the corrective actions that should be implemented to improve this situation could be, among others, the following:

- The Cost/Benefit (very low ratio) of applying these kinds of solutions during the early stages of the project should be disseminated.
- Improvements related to the harmonisation of applying Directive 2006/87 EC in all the member states should be considered.
- To apply the most suitable noise management of the harbours, the current NRH signature of the current fleet should be obtained, independent of its compliance or not with the standard limits. It will enable placing the vessels in the best locations, taking into account the disturbance they produce.

Despite the small size of the sample of available vessels analysed, but based on the authors' experience and on the case studies available [7], it could be stated that compliance with the PSL could be achievable for all the families of vessels that make up the current fleet. As seen with the previous analysis, if we expect for all the new constructions to comply with the targeted NRH limits, the following activities should be applied:

- *Specifications:* Making it mandatory for NRH requirements to appear in the contract specifications of new constructions.
- *Simulation Tools and Tests:* Generalising the demand for technical studies in the contract specifications. These technical studies will provide reliable outputs only if the sources introduced are well-known; hence, it would be vital to control these sources and have their complete characterisation.
- *NRH Engineering:* Based on the results obtained through these studies, the most effective technical solutions shall be applied.

Underwater Radiated Noise (URN)

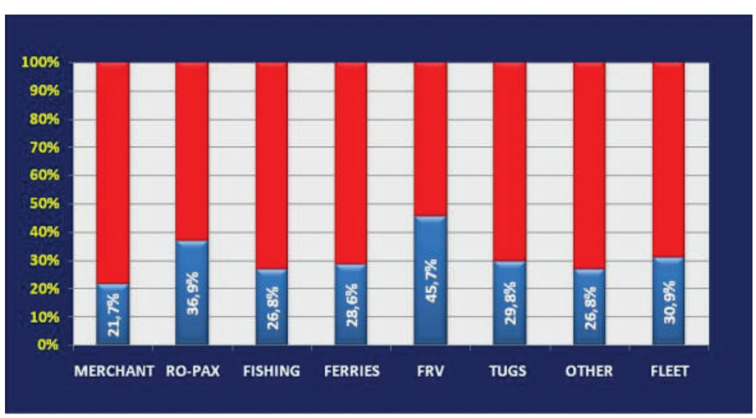
With regard to this also novel N&V topic, the figures related to the efforts currently applied by the different families of vessels and by the current fleet are reported in Fig. 16. As noted, the total technical efforts applied by the fleet to address this disturbing N&V topic achieve a value of 30.9%.

Likewise, the fishing research vessels appear as the family of vessels deploying the maximum relative efforts (45.7%) for abating their underwater footprints, followed by passenger vessels that seem to be increasingly more concerned about this topic. Related to the efforts applied by the rest of the families of vessels, these results could be described as deceitful. In most cases, these families have centred their attention on abating N&V on board, but have not done the same with the URN. Therefore, if they want to solve their underwater footprint problem, they will have to face the problem in a comprehensive way.

As explained before for the case of NRH, the disappointing results observed in the application of technical efforts on the European fleet (not considering the FRV family) could be explained because of the absence of URN requirements in the contract specifications of most of the current vessels. Even the most modern ones do not pay attention to this disturbing topic and do not include these requirements in their specifications. Another crucial fact from the authors´ point of view is the confusing scenario that exists in the marine sector.

On one side there is an increasing concern from the scientific community about the damage in marine life. On the other side, the shipbuilding and marine industry needs more information about the cost/benefit ratio that will suppose for the sector the upgrade of all the commercial vessels. Another pending matter is that, until now, no one has assessed, in terms of cost/benefit, the impact of the underwater footprint of the vessels. Furthermore, there is no agreement on how the measurement procedure should be, on which should be the new URN limits for each family of vessels (based on

Fig. 16. Unitary URN efforts by families of vessels



experimental information), or on how to apply these solutions.

Within this confusing scenario (well-described and detailed by the authors in reference [6]), if we want to move ahead, it will require establishing a well-defined policy or strategy. This policy should "convince and not only obligate". The right way for doing this would be:

- Dissemination of dedicated activities to abate this URN topic shall be launched within the marine sector and the technician community.
- Once the current URN framework has been pictured, the SILENV project has described the current situation of the fleet (in relation to its compliance with the PSL) and the impossibility to apply retrofits to all the vessels of the fleet (to improve their degree of incompliance) is accepted, potential solutions should be found:
  - In the short term, it would be necessary to significantly improve the current state. Proper Underwater Radiated Noise Management is essential, but it will require knowledge of the current fleet's URN signature, independent of its compliance or not with the standard limits. Combined with the information provided by the AIS system, it will enable evaluating if a vessel could enter or not a protected area. As a first step, a rough approach could be made based on experimental information such as the one provided by the SILENV project.
  - Harmonisation of the measurement procedure and agreement between the scientific community and the marine sector on the definition of the preliminary URN limits should be promoted. At this stage, these preliminary targeted limits may not be the perfect ones, related to the threshold of all the marine species, but they will be enough to improve the current state. It will enable the marine sector to assess the technical viability of these design changes and the economic consequences.
  - Currently, with some exceptions, the owners do not see the costs/benefit related to the investments required to reduce the URN impact of their vessels. From the authors' point of view, the EC authorities, currently worried by the increase of noise in the oceans, have established the right policy that will enable incorporating in the marine sector, hardly affected by the economic crisis, this new challenge related to the reduction of the underwater footprint of all the vessels.

From a technical point of view, full compliance of "new vessels", similar to those that make the current fleet, with the PSL could be achievable. This could be done if the proper and right actions were applied during the earlier stages of the design of the new vessels. These preventive actions could be summarized in the following blocks:

- *Specifications:* Making mandatory the presence of specific URN requirements in all contract specifications of new constructions.
- *Simulation Tools and Tests:* Generalizing the requirement of technical studies to achieve the targeted limit in the specifications. It must be taken into account that to have good results from the simulation tools, these depend on the quality of the information. Therefore, the intensity of the sources must be well known and the suppliers must provide reliable information that should be controlled by means of FAT.
- *URN Sources Control:* All the preventive actions related to N&V sources, which for most of the cases are the same as for the URN sources, and which are focused on abating the corresponding topic, noise or vibrations, respectively, are not enough to abate the URN signature of the vessel. It means that the conventional and well-extended preventive solutions have to be significantly improved to reduce the intensity of the sources and their impact on the hull of the vessels. In this sense, the use of main propulsion based on electric motors, instead of diesel propulsion, could be considered in some cases.
- *URN Engineering:* Depending on the type of vessel, and based on the previous points and on the theoretical results, the most effective technical and economic solutions should be sought.

Finally, it is important to consider that all these preventive actions grouped into blocks and recommended to address the different topics included in the N&V-FS are based on the authors' documented experience.

Indeed, there are many vessels that have applied these preventive solutions, proposed by the authors, which comply or are quite close to complying with the strict PSL. These vessels belonging to different families became a milestone and must be seen as "technological references" [6].

Conclusions

Based on the analyses made and on the results obtained, the following conclusions could be highlighted:

- The SILENV-N&V database, compiled under the framework of the SILENV project, became the largest ever N&V database in the marine sector, as well as a complete novelty. Therefore, the results, conclusions, and recommendations obtained from its assessment, not only related to N&V on board but also with regard to outdoor noise radiated by the vessels, could be of paramount importance to support current EC policies and other marine institutions that focused their efforts on reducing the environmental impact of the ships.
- The complete assessment, as well as the sensitivity analyses carried out enable highlighting the following points:
  - Within the current fleet, there is complete and generalised absence of specific NRH or URN requirements in the contract specifications. Even the new vessels do not include them in their specifications. Only the modern fishing research vessels have specific URN requirements, but not NRH ones.
  - Likewise, and due to the average age of several families of vessels (particularly the fishing vessels), N&V on board requirements are also scarce in the contract specifications.
  - Consequently, and due to the lack of information, the complete environmental impact of the fleet is not assessable. Anyhow, due to the experimental information provided by the SILENV-N&V database, the environmental assessment of at least 20 vessels (belonging to different families) has been completed. The conclusions obtained could be consistently extrapolated to the rest of the fleet.
  - As a direct consequence of these critical absences of specific environmental requirements and also because "nobody will do something if they are not forced to", the technical efforts applied by the fleet to minimise the environmental impact have been limited. Even the vessels that have N&V on board requirements do little. Obviously, less is done when we refer to outdoor noise (to the air and into the water) radiated by the vessels.
  - On the contrary, the modern fishing research vessels that have specific URN requirements in their contract specification are the only ones using a specific strategy of application of technical efforts focused on minimising their underwater footprints. It has been noted and it is remarkable that they comply with URN requirements (ICES Nº 209), and with N&V standards (IMO and ISO); in fact, if a little more were done they would comply with the PSL.
- After analysing the previous information and using the statistical assessment made, related to N&V on board, it could be stated that strict compliance (at all spaces and with no deviation) of all the current fleet (independent of the compliance of some particular vessels) with IMO and ISO is not achievable. Likewise, if any other stricter limits were defined, higher percentages of deviations would be found.
- Based on the results of the SILENV-N&V database, it has been proven that compliance with any N&V limits established is the direct consequence of the suitable application of a whole set of preventive actions focused on achieving the targeted N&V limits. These preventive actions shall be applied conveniently and at the right time (at the early stages of the project). Indeed, these vessels that have applied them correctly have become "technological references" that could be used as "practical guides" for the construction of the future "silent vessels".
- Thus, from the authors' point of view, the strategy that should be followed includes:
  - Intense dissemination of the dedicated

activities focused on abating the environmental impact of the vessels.

- Regarding the old current fleet and considering that retrofitting to improve the current environmental impact of the old vessels is not technically and economically feasible. Only two potential alternatives could be considered: First, launch a debate related to the renewal of the fleet and, second, apply a convenient Noise Management Policy, as the one described in the paper.
- Regarding the construction of the future "silent vessels", the recommendations launched in the paper and based on the successful results obtained [6], when properly applied, should be clearly defined and included as mandatory in the contract specifications of the new vessels.

• Last but not least, and in clear coincidence with the authors' point of view, the policies launched separately by the EC related to achieving vessels with "Zero Emissions" and vessels with "Green Card" notation (with minimum N&V environmental impact) should be conveniently engaged and harmonized.

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References

[1] DIRECTIVE 2003/10/EC "Minimum health and safety requirements regarding the exposure of the workers to the risk arising from physical agents (noise)".

[2] DIRECTIVE 2006/87/EC. "Laying down technical requirements for inland waterway vessels and repealing Council Directive 82/714/EEC". 12th of December 2006.

[3] DIRECTIVE 2008/56/EC. "Establishing a framework for community action in the field of marine environmental policy (Marine Strategy Framework Directive). 17th of June 2008.

[4] IMO RESOLUTION A.468 (XII). Code on noise levels on board ships. Adopted on 19 November 1981.

[5] INTERNATIONAL STANDARD ISO 6954-1984. "Mechanical vibration and shock- Guidelines for the overall evaluation of vibration in merchant ships".

[6] PUBLIO B. /J.I. DIÁZ /R. SALINAS. "Achievement of the new underwater RADIATED NOISE requirements by the Spanish shipbuilding industry. The FRV "Ramón Margalef".

[7] PUBLIO BELTRÁN. "Shipbuilding and new requirements to reduce the environmental impact of ships: New technological challenges and business opportunities". Ship Science and Technology, Vol. 5 -nº 10. January 2012. (www.shipjournal.co).

Underwater Multi-influence Measurements as a Mean to Characterize the Overall Vessel Signature and Protect the Marine Environment

Medidas multi-influencia como medio para caracterizar la firma global de un buque y proteger el entorno marino

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Abstract

The overall signature of a vessel comprises acoustic, magnetic, electric-field, pressure, and seismic radiations. Over the past years, the international community's efforts have mainly centered on reducing the acoustic influence with the dual aim of decreasing the vessel's detectability and reducing the levels of acoustic pollution generated in the marine environment. Nowadays, the need to act not only on the acoustic radiation but against the overall set of the vessel's radiations is becoming increasingly clear, both in the military and the civilian fields, based on aspects like vessel stealthiness, security of harbor and critical infrastructures, and environmental protection. As a key element to achieve this goal, it is greatly important to have at our disposal highly modular and adaptable measurement systems covering the overall set of the vessel's radiations, with a high capacity of data transmission to base centers to have the capacity to make measurements in all kinds of marine environments.

Key words: vessel signature, multi-influence, underwater measurement, environmental protection, portable systems

Resumen

La firma global de un buque está compuesta por radiaciones: acústica, magnética, de campo eléctrico, de presión y sísmica. A lo largo de los últimos años, los esfuerzos de la comunidad internacional se han centrado principalmente en reducir la influencia acústica, con el doble objetivo de disminuir la detectabilidad del buque y reducir el nivel de contaminación acústica generada en el entorno marino. En la actualidad, se está constatando con claridad la necesidad de actuar no sólo sobre la radiación acústica, sino sobre el conjunto global de las radiaciones del buque, tanto en el ámbito militar como en el ámbito civil, basado en aspectos como: la discreción de los buques, la seguridad en puertos e infraestructuras críticas y la protección ambiental. Como elemento clave para alcanzar esta meta, es de gran importancia tener a nuestra disposición sistemas de medida adaptables y altamente modulares que cubran el conjunto total de las radiaciones de los buques, con alta capacidad de transmisión de datos a centros base, con el objeto de tener la capacidad de realizar medidas en toda clase de entornos marinos.

Palabras claves: firma global del buque, multi-influencia, medición bajo agua, protección ambiental, sistemas portátiles

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Introduction

All vessels, independent of their shape and size, emit onto the sea a set of radiations that make up the so-called vessel signature. This signature characterizes and identifies univocally the vessel the same way the fingerprint identifies human beings. The importance of this signature is well-known since early in the past century, mainly in the defense field and specially centered on the so-called acoustic and magnetic signatures. For example, detection of vessels based on their acoustic signatures was quite important in the naval field during World War II.

In parallel with technological improvements during the 20th century, and especially in the defense field, specific techniques have been developed to reduce the level of radiations emitted onto the sea by vessels. At the beginning, reduction techniques were limited to the acoustic radiation. Next, magnetic radiation was also taken into consideration and, more recently, the electric-field, pressure, and seismic radiations have also been considered of interest. The group of five radiations referred to above make up the so-called multi-influence signature of a vessel. An example of the simulated electric signature radiated by a sweep gear is shown in Fig. 1.

Monitoring the multi-influence signature is greatly important in the defense field and in the case of vessels like submarines it becomes a matter of survival. Also, it is becoming increasingly important in the civilian field related to the marine environment preservation, especially in the case of the marine fauna living in this environment, due to the influence of these radiations on their behavior. Finally, it is worth noting that detection of this signature permits us to determine the presence of threats in harbors/ports, critical infrastructures or cultural assets located on the sea floor, making it possible to implement specific actions to neutralize these threats.

This paper comprises four sections in addition to this introduction section: the first section describes the main characteristics of the radiations that make up the multi-influence signature. In the next two sections, the importance of this signature in the defense and civilian fields is analyzed. In the following section, a system especially adapted to measure multi-influence signatures is described. Finally, the paper is completed with the conclusions of the study.

Characteristics of the Multi-influence signature of a vessel

As previously stated, the multi-influence signature of a vessel comprises five types of radiation: acoustic, magnetic, electric, pressure, and seismic. Each incorporates specific characteristics. Then, a brief description of their main characteristics is presented.

Fig. 1. Simulation of the electric field radiated by a sweep gear

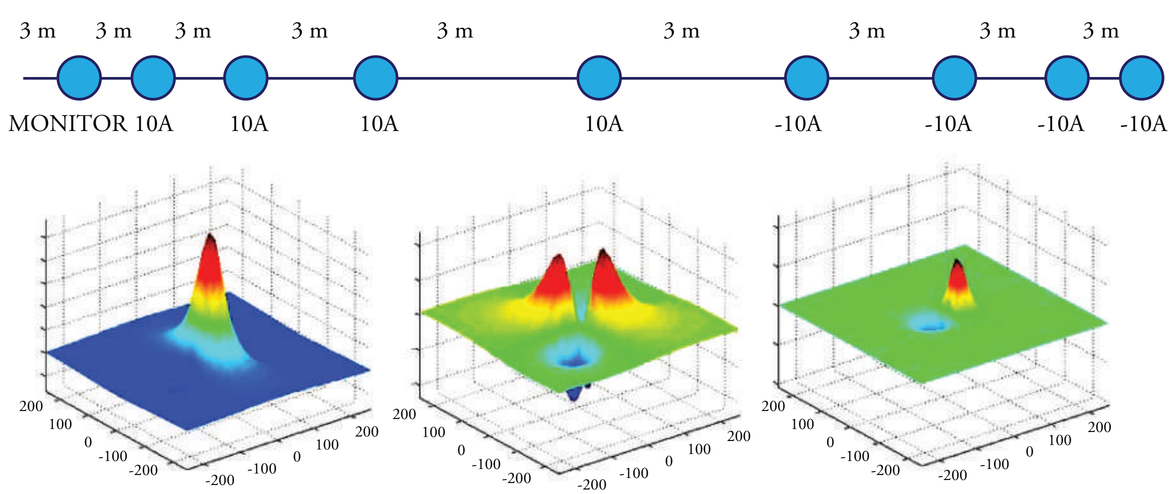
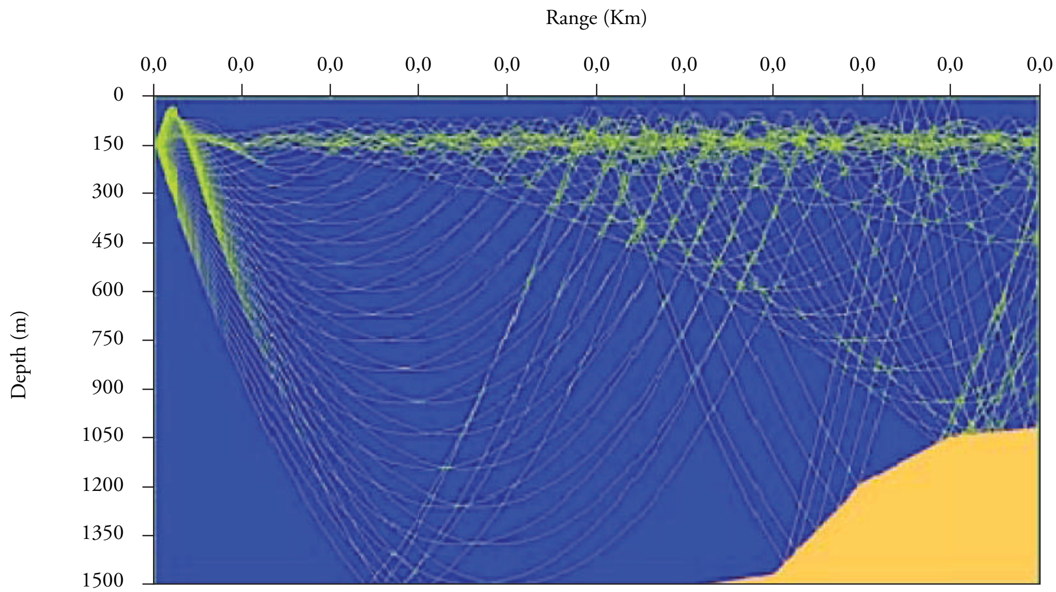


Fig. 2. Underwater sound propagation paths obtained from an acoustic propagation model



Acoustic radiation

The sound generated by a vibrating source is propagated as a wave in an elastic medium such as the sea, originating pressure changes that are susceptible of being measured.

The underwater sound propagation, characterized by its high performance, being the way radiation currently best propagates through this medium [1], reaching long distances in the case of low frequencies.

Vessels emit two types of generic signals: broadband and narrowband. The former is characterized by covering a wide spectrum of frequencies, meanwhile the latter is limited to a narrow spectrum.

There are different sound sources in vessels. The three main are: machinery noise, propeller noise, and hydrodynamic noise. A scheme of underwater sound propagation obtained from an underwater acoustic propagation model is shown in Fig. 2.

Magnetic radiation

The Ship's magnetic influence comprises two components: the static component (SM) and the alternating one (AM). The static component is generated by permanent and induced magnetic fields.

The ship's permanent magnetic field is due to the magnetization of its construction magnetic materials by the Earth's magnetic field. Besides, the Earth's magnetic field, as an external magnetic field, always contributes to the ship's magnetic signature. This component depends on the ship's course and localization of the area.

In addition to permanent and induced magnetic fields, the Corrosion Related Magnetic (CRM) field also contributes to the static magnetic component of the signature. This field is due tto the existence of corrosion currents through the sea water, which has an associated magnetic field.

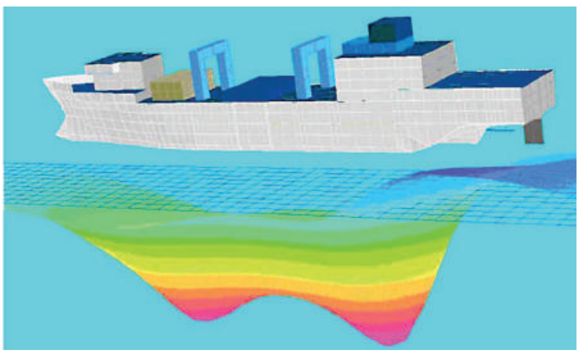
The alternating component of magnetic signature is generated by:

- Currents in the rotating coils of ship turbines. These coils perform as magnetic dipoles, which generate AC magnetic signature.
- Sea water Foucault currents induced by the magnetic dipoles. These currents are time varying and are also associated with alternating electric fields.
- Electric currents flowing through the ship's

hull due to electric equipment failures or inadequate design.
- Inherent magnetic field radiated by any rotating electric machinery in the ship.

Besides, power supply ripple generates alternating currents through the water. An example of simulation of the magnetic field generated by a vessel is shown in Fig. 3.

Fig. 3. Simulation of magnetic field generated by a vessel



Electric-field radiation

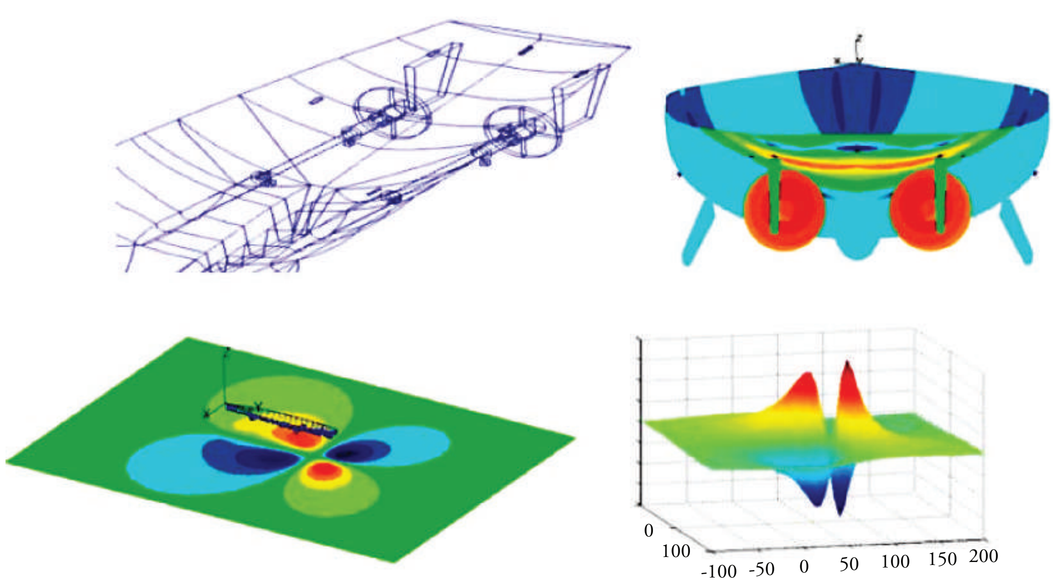
The ship's electric signature is composed by two components: the static component, which is the Underwater Electric Potential (UEP) and the alternating component, or Extremely Low Frequency Electric (ELFE).

The static UEP component represents the near field influence and its temporal variation depends on the ship's speed and size. The static electric signature of a ship is due to the electric currents generated by the galvanic corrosion process. In order to avoid this corrosion, cathodic protection systems are used. Two types of cathodic protection systems exist: passive and active or Impressed Current Cathodic Protection (ICCP). Passive systems use sacrifice anodes, whereas active systems use impressed current anodes and reference electrodes. Quite often, cathodic protection systems contribute drastically to electric signature and constitute the main generator for this influence.

The alternating ELFE component covers a bandwidth of approximately 3 kHz and represents the near and far field influence. The ELFE component is due to the following factors:

1. Modulation of the corrosion current: galvanic current is modulated due to the spin of blades and propellers.
2. Ripple in machinery power supply of the ship. A frequency tone appears, corresponding to the power supply frequency.
3. Ripple in degaussing ICCP systems, corresponding to the modulation suffered by the ICCP system current due to variations

Fig. 4. Example of simulation of the electric signature radiated by a vessel



in the resistance between the shaft and hull of the ship. Fig. 4 shows an example of simulation of the electric signature radiated by a vessel.

Pressure radiation

Hydrostatic pressure due to water depth varies slowly with atmospheric pressure changes and tide rising and falling. Besides, it can vary fast with waves and train of waves, or with a ship. Pressure variation due to a ship movement is usually very small. This small variation constitutes the ship's pressure signature and it is produced by the Bernoulli Effect of the water flowing from bow to stern. This flow originates a pressure increase at the ship's bow and stern and a decrease in the central zone (suction), whose peak is directly proportional to ship speed and its underwater shape. Fig. 5, shows this pattern in a simulation of a ship´s pressure signature. Therefore, induced pressure fluctuations by the ship are superimposed with the nominal static pressure from the bottom and natural disturbances caused by tide, waves and swells.

Seismic radiation

Seismic influence is generated by the same sources as for the acoustic influence. When an acoustic wave reaches a surface, the majority of the energy is reflected but a percentage is absorbed by the new medium. Thus, very low frequency (below 10Hz) acoustic signals propagate up to the sea floor and transmit through it as a seismic perturbance. This type of perturbance travels much faster through the sea floor than through sea water.

From this seismic point of view, it is obvious that the acoustic energy penetrating into the sea floor may sometimes contribute considerably to medium-range and long-range acoustic trans mission. One clear example that shows seismic influences behavior is the existence of a critical frequency. For all frequencies below this limit, absorption phenomena in incident waves appears, and this phenomena depends on the characteristics of the materials and layers of the sea floor.

Fig. 6. Ship seismic signature physical phenomena

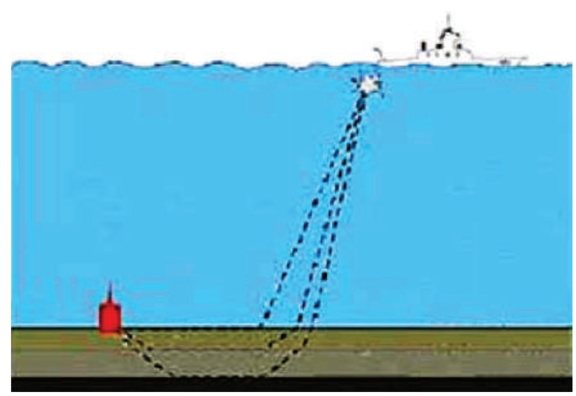
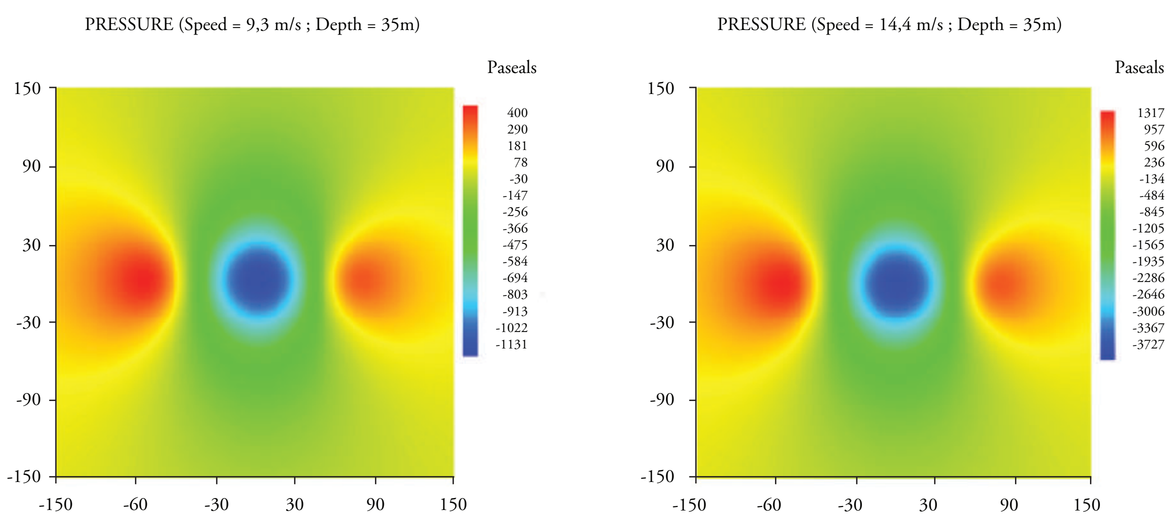


Fig. 5. EShip's Pressure Signature Simulation

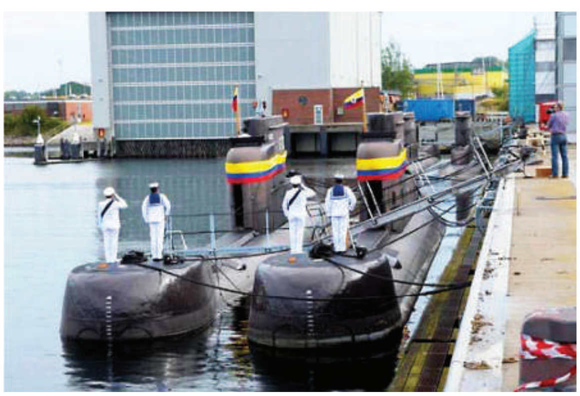


The multi-influence signature in the defense field

Intelligence databases have become elements of great importance for the navies in different countries. These databases contain, as distinctive data, the signatures of the vessels. At the beginning, they contained acoustic data and sometimes magnetic data. Currently, they also seek to incorporate data corresponding to the rest of influences. These databases permit discriminating not only the type and class of vessel, but also the specific unit of the class, providing a considerable tactical advantage.

Currently, the trend in recent decades relative to the reduction of the vessel's signature is being accentuated. This trend is particularly intense in the case of submarines. They seek to increase their level of protection by increasingly becoming more stealth vessels as a means to counter-act the development of increasingly intelligent weapons like last-generation torpedoes. Fig. 7 shows an image of U-206 type submarines.

Fig. 7. U-206 class submarines : A.R.C. Intrépido and A.R.C. Indomable



In parallel with that stated above, more sophisticated systems incorporating a wide range of sensors are now being developed. The use of these multi-influence systems permits vessels to significantly increase their detection capacities from the combination of the data provided by their suite of sensors, enabling a considerable reduction in the number of false alarms. Also, in the defense facilities protection field, the use of these multi-influence sensors has permitted significantly increasing their security. This fact is based on early and more accurate detection of threats coming from the marine environment, such as: divers moving autonomously, manned and unmanned underwater vehicles (SDV, ROV, UUV), mini-submarines, etc.

The multi-influence signature in the civilian field

During recent years, growing awareness has emerged worldwide on the need to protect the marine environment, especially from human activities like: fishing, sailing, harbor works, or seismic oil and gas explorations that convey significant increase in the level of a range of pollutants such as acoustic noise and other sources of energy, including electric and magnetic sources. This growing awareness has entailed the development of a range of national and international regulations focused on achieving effective preservation of the marine environment. Among these regulations is the Marine Strategy Framework Directive, promulgated by the European Union in 2008, which introduced a set of qualitative descriptors to determine the good environmental status. One of these descriptors states that: "the introduction of energy, including underwater noise, is at levels that do not adversely affect the marine environment". The effective application of this Directive, and of other regulations related with the marine environment protection, implies both measurement of the level of energy radiations emitted to the sea [2] and detection of marine fauna presence in specific areas on which high energy levels are detected with the aim of protecting this fauna from the potential harmful effects of the radiated energy.

This detection is mainly based on acoustic sensors, although other alternative detection means are currently being analyzed, such as detection of marine fauna based on the alteration of the underwater electric or magnetic fields originated by their presence. In Fig. 8, two kinds of cetacean species: bottlenose dolphin and sperm whale are shown; both are endangered species.

Fig. 8. Species of cetaceans: bottlenose dolphin (left) and sperm whale (right)

Just like in the defense field, use of multi-influence sensor systems provides an effective protection of harbors and critical infrastructures like oil refineries and thermal power stations against hostile intruders. This protection can also be extended to other fields of remarkable interests as is the case of marine reserves, ship wrecks, or underwater archaeological remains.

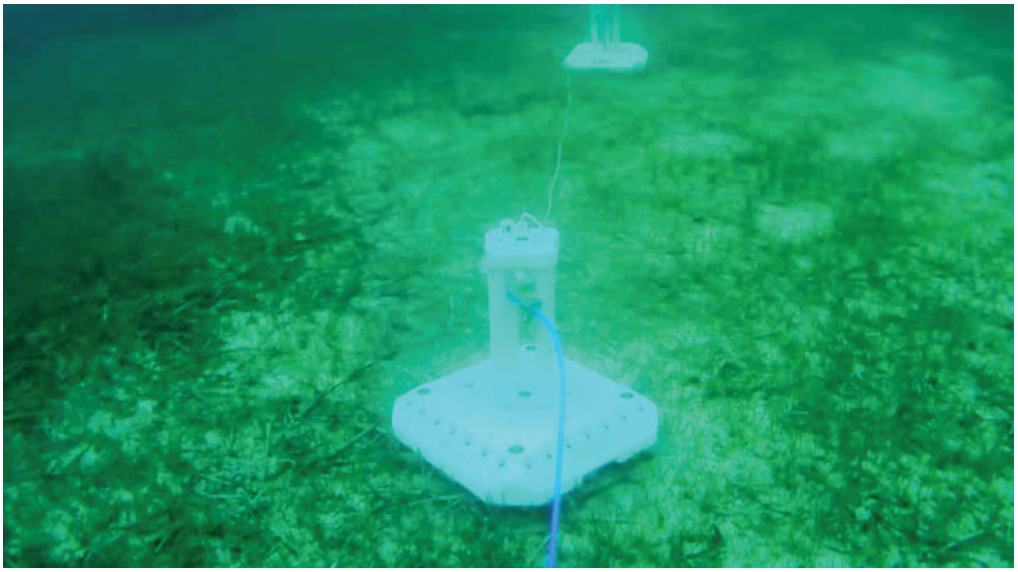
The multi-influence signature measurements

With regards to multi-influence measurements, the trend in the civilian field is to advance towards a standardization process in the procedures and parameters of the measurements. These standards have long existed in the defense field. The precursor of this process is, again, the acoustic influence for which a standard has recently (2009) been developed by the Acoustical Society or America (ASA): *"Quantities and Procedures for Description and Measurement of Underwater Sound from Ships"*. Additionally, within the European Union scope specific programs exist focused on the definition of a European standard. Measurements are usually normalized to common references in order to compare these coming from different systems, as is the case of the 1-meter from the source reference for acoustic measurements or a common reference point for all the vessels in the case of measurements of electric and magnetic fields.

The fact that multi-influence sensors provide different detection ranges permits establishing different detection layers as a function of the sensor range. Detection ranges depend on both the specific characteristics of the marine environment and those of the vessel being tested and the sensors used. In a first approach, it can be stated that both acoustic and seismic sensors provide detection distances in the range of kilometers, being the seismic sensor especially dependent on the characteristics previously referred to, electric-field and magnetic sensors in the range of hundreds of meters and pressure sensors in the range of tens of meters.

The wide range of operational environments in which multi-influence measurements are susceptible to be taken makes it highly advisable to have at our disposal modular and portable systems with small dimensions and weights. These kinds of systems permit their deployment and recovery in different marine areas within a reduced time interval and without requiring complex means. Other aspects to be stressed are: the capacity of data transmission to base centers (located on shore or onboard vessels) with an adequate bandwidth to cover the characteristics of the signatures tested and the capacity of storing and processing the measured influences, focused on providing accurate and useful information to the system operator. An example of a system that complies with the characteristics previously described is the multi-influence measuring system (MIRS) developed by SAES (see underwater units deployed on the sea bottom in Fig. 9).

Fig. 9. Underwater units of the multi-influence measurement system (MIRS) developed by SAES



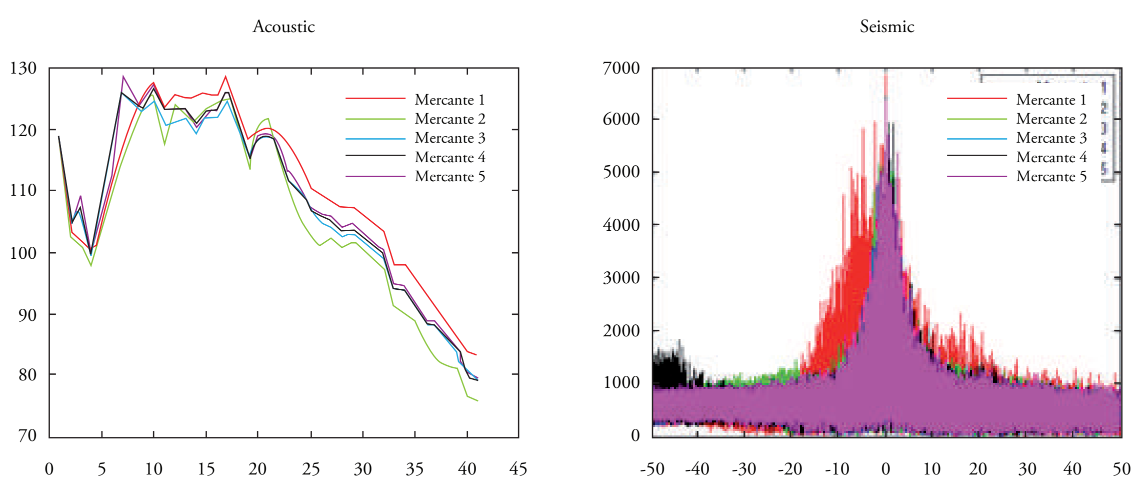
The MIRS has been tested in operational environments, showing its versatility, ease of use, and accuracy taken as reference calibrated systems. In Fig. 10, graphic outputs of the multi-influence signature of a merchant vessel are shown.

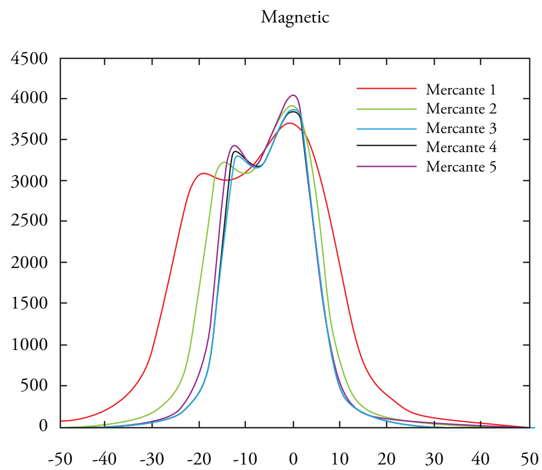
Conclusions

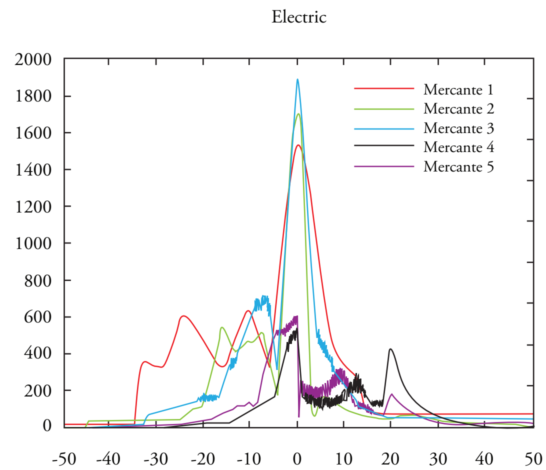
All vessels, when sailing through the sea, radiate a set of influences (acoustic, magnetic, electric, pressure, and seismic) that make up their multi-influence signature. Some of these influences (acoustic and magnetic) have been measured since decades ago to characterize vessels in the defense field and to monitor the acoustic pollution in the civilian field. Recently, interest in the international community on having at its disposal the overall signature of the vessels has emerged, seeking to globally evaluate its impact on the marine environment.

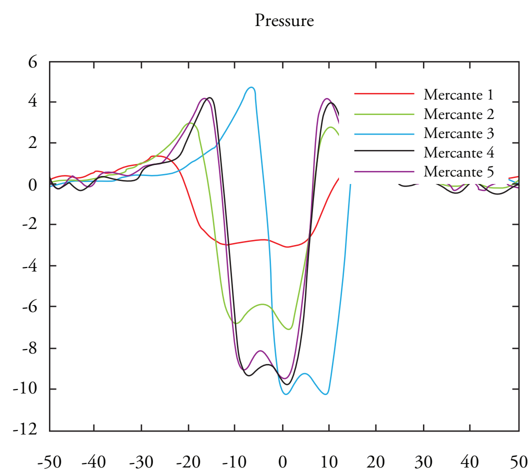
In the defense field, from the fleet's point of view to have at our disposal multi-influence data from vessels permits performing specific

Fig. 10. Set of measurements of influences of a merchant vessel taken by the MIRS developed by SAES









tasks and studies focused on reducing its own signature to decrease the probability of being detected. On the other hand, from the point of view of threats, this data permits characterizing the vessel's signature with the aim of increasing our capacity to detect them.

In the civilian field, interest is centered on the marine environment preservation, especially of its marine fauna. In the dual defense-civilian field, protection systems based on multi-influence sensors constitute a highly efficient means to detect hostile intruders.

Due to the variety of operational areas in the marine environment, it is highly advisable to have at our disposal modular systems with contrasted capacities of data transmission, recording the measurements and processing focused on providing relevant information to the system operator. The MIRS developed by SAES configures as a verified and in-service system than complies with the requirements established for the multi-influence measurement of all kinds of platforms or naval devices in the whole spectrum of operational environments.

References

[1] R. J. URICK. Principles of Underwater Sound. McGraw-Hill (1993).

[2] W. J. RICHARDSON, C. R. GREEN, C. I. MALME, D. H. THOMSON. Marine mammal and noise. Academic Press Inc. (1995).

Action Plan for Restructuring the Technology of a Medium-Sized Shipyard

Plan de Acción para la Reestructuración de la tecnología de un Astillero Mediano

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Ulrich Kothe [2]
Martin Eggert [3]

Abstract

Last years, a medium-sized shipyard has specialized in building tankers for chemicals. This is planned to increase the production of shipyard in medium or long terms for 4-6 boats. To achieve these objectives, the changes have to be made in all kinds of services shipyard. These changes have to take place in the organization, as well as in technological areas. In the first phase of analysis, there has been a single sequence analysis of part 1, where they perform six different methods that resulted in a large number of solutions that will help restructure of shipyard.

Key words: shipyard, restructuring, productivity.

Resumen

En los últimos años, un astillero de tamaño medio se ha especializado en la construcción de buques cisterna para productos químicos. Se prevé aumentar la producción del astillero a mediano y largo plazo a 4 - 6 barcos. Para alcanzar este objetivo, los cambios tienen que hacerse en todo tipo de servicios del astillero. Estos cambios deben tener lugar en la organización, así como en las áreas tecnológicas. En la primera fase de análisis, se ha producido una secuencia de análisis individual sobre la parte 1, donde llevan a cabo seis métodos diferentes que dieron como resultado un gran número de soluciones que ayudarán a la reestructuración del astillero.

Palabras claves: astillero, reestructuración, productividad.

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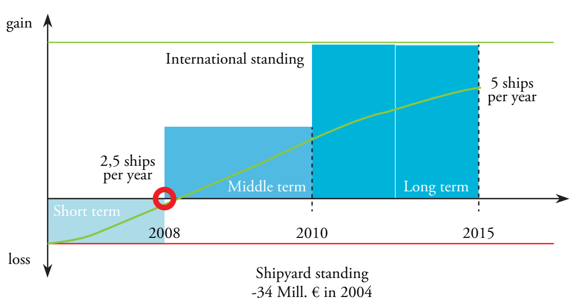
Initial Situation

Within the last few years, a medium-sized shipyard has specialized in the building of product and chemical tankers. A product arose that secured a good position in the world market. However, continuous economic development opposes a deficit of €34-million per year. To address this deficit, the shipyard has to build more ships per year. To achieve this aim, the shipyard started a multi-project in 2006. In this multi-project, the project of the development of an "Action plan for restructuring the technology and organization of the shipyard" was conducted.

As seen in Fig. 1, the requirements of the project were:

- The economic loss is compensated when 2.5 ships are built per year and this should be reached by 2008.
- Additionally, it is planed to increase the output of the shipyard in middle or long terms to 4 – 6 ships.

Fig. 1. Definition of achievable project results



To achieve these aims, changes have to be made in all types of services of the shipyard. These changes have to take place in organizational, as well as in technological areas. Basis and guideline for other service areas are the technological changes.

There has been a program including two milestones, to analyze the shipyard in detail and develop suitable actions to be undertaken.

- An analysis phase to analyze the shipyard; to show bottlenecks and their negative results towards ship production.
- A phase for development of solutions, which as a whole result in reaching the aims defined by the shipyard itself.

In the first analysis phase, there has been a sequence of individual analysis concerning the following topics:

1. Application of lean principles
2. Definition of bottlenecks
3. Crane capacities and technological workflow
4. Communication and information flow
5. Core competencies
6. Micropanel line
7. New outfitting place
8. Organizational workflow
9. Performance of outfitting
10. Payment methods
11. Process orientation
12. Second berth
13. Spatial structures and material flow
14. Welding speed and quality

The whole project was executed successfully and on schedule from the end of February to the end of August 2006 in Trogir, Coratia.

Analysis phase

The 14 analyses from part 1 where accomplished by these six different methods:

- Simulation
- Input-Output relation
- Benchmark
- Planning table
- Value stream analysis and
- Questionnaire.

How these methods were applied in the individual analysis is described in this final report.

Productivity analysis per CGT

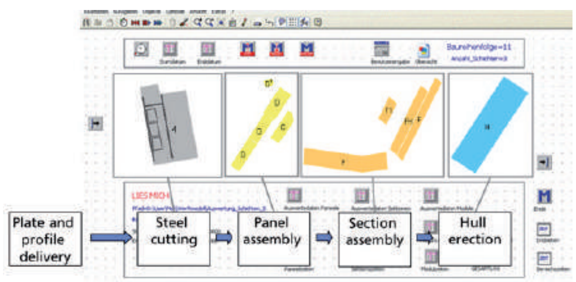
An essential conclusion of the analysis is a deficiency in all areas of production in the whole shipyard. This problem was examined closely in the main processes of steel manufacture. With a steel throughput of 10,464 tons per ship comes a 41.3 man-hours manufacturing input per ton of steel. That is 21.3 man-hours per ton of steel based on CGT. Including an outfitting expenditure of 40%, the manufacturing expenses are 35.5

man-hours per CGT and, therefore, 2.5 times the international standard (best practise 14 man-hours/ton CGT).

Simulation

Workflow simulation is an introduced and applied method to analyse lead times in complex manufacturing systems. In this project, the simulation led a distinct model (Fig. 2) of the manufacturing of the ship's hull. With the help of this model, the manufacturing workflow can be clearly described. Therefore, it was possible to show and analyse bottlenecks in manufacturing.

Fig. 2. Introduction into shipyard´s simulation model



Based on this model, future alternative manufacturing concepts can be verified and integrated into the ERP landscape to be developed.

In this simulation, we concentrated our attention on a defined number of representative types, which give a good mapping onto the real manufacturing.

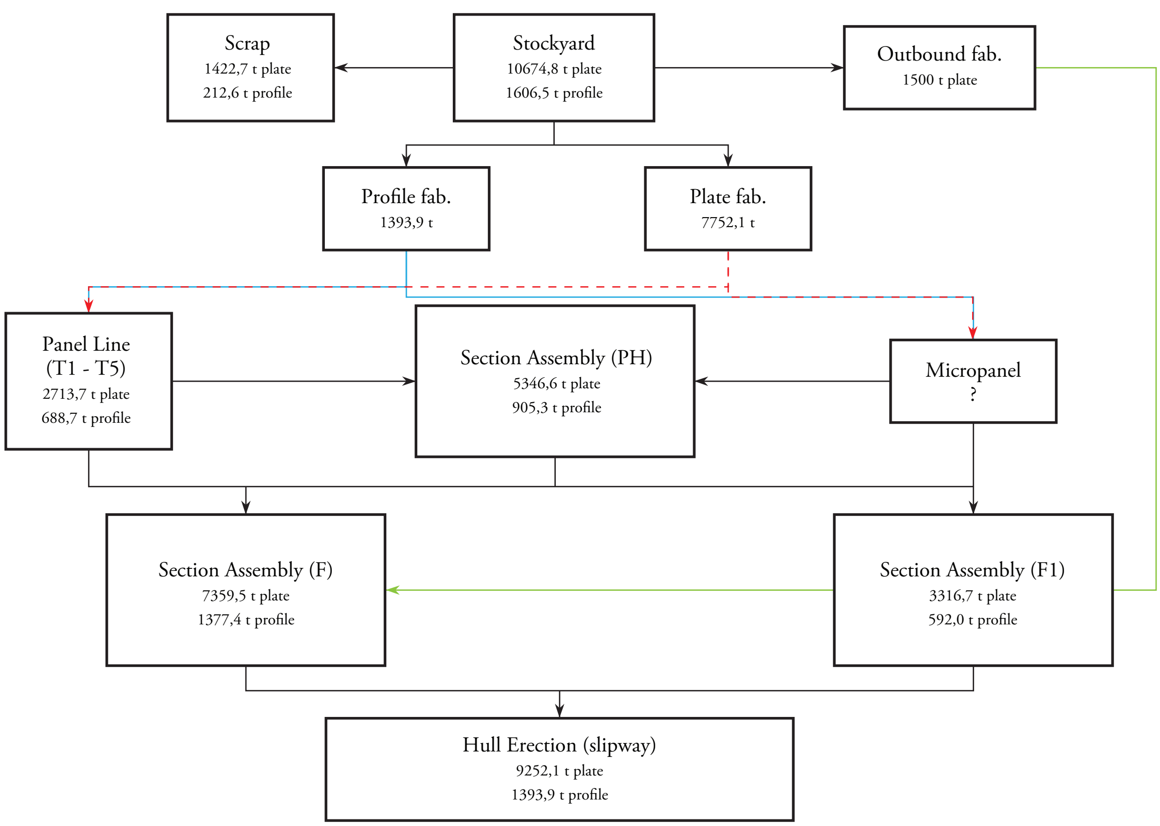
Input-Output relation

With the help of actual input-output relations of the different workshops and the expected steel throughput, at the aimed increased production rate, it can be analysed if the capacity of all workshops meets the increasing requirements.

Benchmark

With the help of the benchmark we could find if the existing welding technology and quality control

Fig. 3. Steel input and output



matches the increasing requirements. In comparison to the international ship building industry, it is clear that the welding and quality control areas are far below the international standard.

Also, with one benchmark the situation of outfitting was researched. The rates of pre-outfitting and pre-assembling in the outfitting section are less than equal to the international state-of-the-art either.

Planning table

With the Fraunhofer developed analysis tool "planning table", the spatial structure of the shipyard was analysed and checked if it could meet the requirements caused by future layout changes. The, hereby developed, spatial model can be used either as basis for alternative manufacturing concepts or as basis for the arrangement of the manufacturing workflow in workflow simulations and in ERP systems.

Value stream analysis

The question for this analysis method is, if the organizational structure of the shipyard was open to integrate lean principles. The structure was determined as not sufficiently clear to allow the integration of lean principles. This becomes apparent when looking at the material flow, which is organized by push principle.

Additionally, more deficiencies were found that interfere with a structured workflow, while applying the value stream analysis. The shipyard's internal project "Propellar" was founded to eliminate those deficiencies and first prosperities were made.

Questionnaire

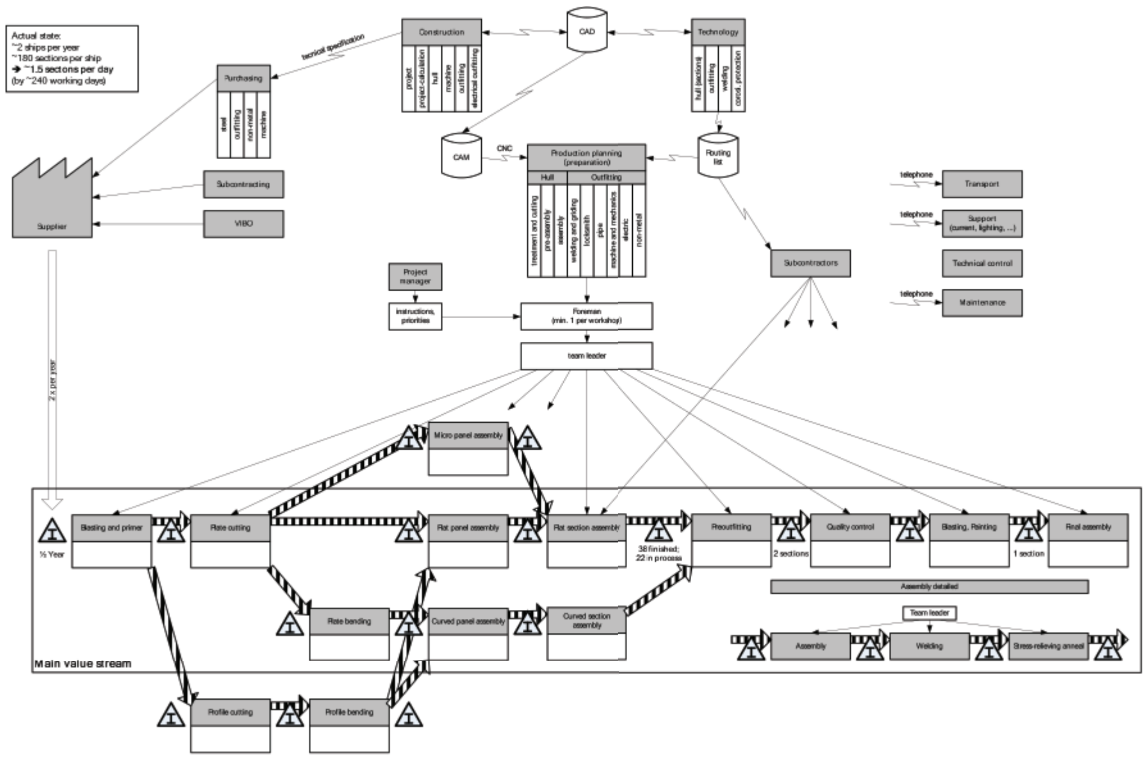
With the help of questionnaires and interview questions, which arose throughout the previously mentioned parts of our work and which could not be answered with any other analysis method were dissolved. The substantiated state of knowledge and willingness to cooperate by employees was very helpful for this project.

Solutions

Solution 1: Accuracy Control and Shrinkage Management

A basic problem at the shipyard, as well as at many other shipyards, is a lack of awareness of quality-

Fig. 4. Value Stream Model of the Shipyard



oriented manufacturing and its effect on costs in the manufacturing process. To reduce these costs, a stepwise procedure has to take place.

In the first step, awareness of the lack of quality and its effect on costs has to be created. This can only be done through consistent quality control. The tools used for this approach are described in detail ahead. To execute quality control, a team of experts has to be trained to meets international standards in quality control. The procedure is detailed ahead in this paper.

After introducing and executing the above previously measures, the shipyard can find and analyse deficiencies in the manufacturing process. These deficiencies have to be eliminated. A first step is to introduce a shrinkage manager for the shipyard. This is a software tool, which is able to deform the cutting plates so the thermal deformation due to the welding process is equalized. The tool collaborates with a CAD System and requires corresponding nesting. The introduction of such a tool requires implementation of a project-specific measuring program and external help from specialists in this topic.

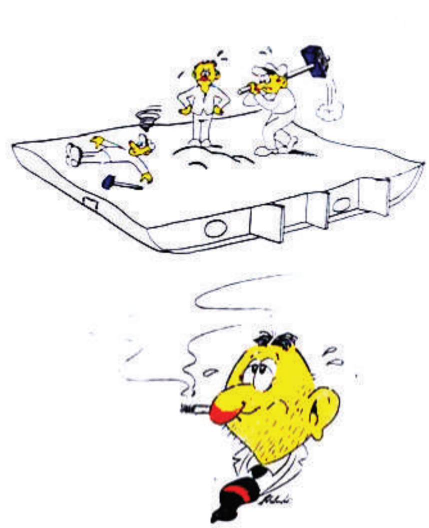
Solutions 2 and 3: Partial, panel and closed section fabrication

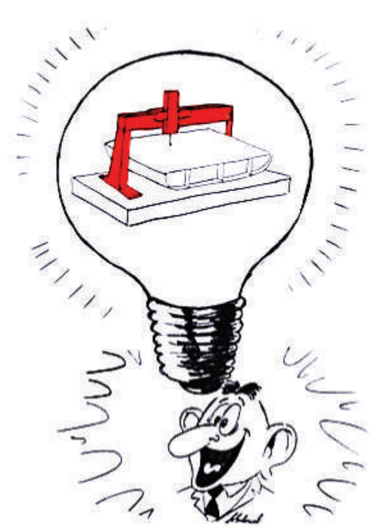
The existing buildings for the partial fabrication including plate and profile cutting and bending are in bad shape. This affects especially the foundations of the existing gantry cranes. An additional installation of new cranes with a capacity above 30 tons, which is recommended, is impossible due to the building's static. Also, the material flow is unsteady due to the existing low capacities of the cranes and transport facilities.

In panel manufacturing, similar problems are faced. This building was designed for section fabrication and the crane system installed does not reach every workplace of the panel line and assembly area. Additionally, the ways of transport are very long for plates and profiles. In general, these transport ways are too long, less developed and, hence, hard to organize. The outdoor micro-panel manufacturing depends heavily on the right light and climate conditions.

Improving the part and panel fabrication would increase steel throughput and decrease costs. Therefore, it would be necessary to relocate the panel line closer to the part fabrication.

The space given for panel and section assembly is too small to produce four ships. Outputs in the micropanel and flat panel fabrication, as well as productivity in the section assembly have to be increased in coherence with a closed production. Production in three shifts maximum is highly recommended for this reason.





The relocation of the part fabrication includes the plate and profile warehouse and the working facilities cutting and bending should be conducted in a fixed procedure. In addition to the relocation of the part fabrication, the panel fabrication including the micropanel and flat panel fabrication and closed section assembly should also be relocated into the part fabrication hall. It is important to not disrupt ongoing production while constructing the new building for part and panel fabrication with a minimal investment. The new building should provide optimal material flow and the possibility to implement future technologies.

Building the new fabrication hall

The first task is to dismantle the old plate and profile warehouse and rebuild it at the new location. Relocate the old crane at the new location, as well. Thereafter, construct the new conservation hall and pull down the old one. Afterwards, part one of the new fabrication hall will be constructed where plasma and profile cutting and profile bending will take place. In the second part of the new hall the panel and micropanel line will be located. The last step is the implementation of the additional fourth and fifth workplace into the panel line.

Technological changes

In the preparation phase, two to three plasma cutting machines should be available to raise the shipyard's output to four to five ships per year. The existing bending machines have to be refurbished and modernized. To bend the profiles, an automated bending machine has to be considered.

To transport the plates from the conservation to the cutting machines, roller conveyors are to be installed. The transition from the roller conveyors to the cutting machines is done by the crane system. This roller conveyor also takes the cut and bent plates and profiles into the sorting workplace where they are sorted into comb pallets. These comb pallets are especially produced in the shipyard; about 10 of these pallets are needed. After the sorting procedure, the plates and profiles are transported directly to the micropanel line or to the outside workplaces. The route to the out-side workplaces is done by fork lifts and transport to the micropanel line is done by the installed cranes and roller conveyors. All bent plates and profiles are placed horizontally on those pallets.

The micro-panel line

The manual manufacturing process of micro-panels should be mechanized to minimize the space used by it. Therefore, for example, four work stations could be created:

- positioning of the plates on the building grid
- placement and tacking of the plates
- welding of the profiles and
- adjustment and rework

Transport of the plates and profiles is done by cranes and roller conveyors already mentioned. Next, is the transport of the finished micro-panels on pallets to the manufacture of the frame beams and intercostals. There should be as much prefabrica-tion as possible to build the frame beams to increase the productivity of the panel line.

The panel line

There should be a dry plasma cutting machine integrated able to cut plates that are 12 by 3 m big. The transport of the primered plates is again carried out by a roller conveyor reaching the cutting machines and the crane system for the route to the panel line. The plates produced should not exceed a size of 12.7 by 11.5 m. There are five working steps needed:

- The first step is to cut, bend, and mark the plates
- Followed by the placement and tacking of the plates
- The third step is to weld the profiles
- Part four and five are to place the frame beams setting, tacking, and welding

In the long term, the capacity in areas four and five has to be doubled to increase productivity and the degree of space allocation. Finally, the transport of the sections to the closed section assembly takes place.

In the closed section assembly, respectively, the panels from sectors three and five are brought together. That means the panels with attached frame beams and without them. They are placed opposite of each other, tacked and welded together.

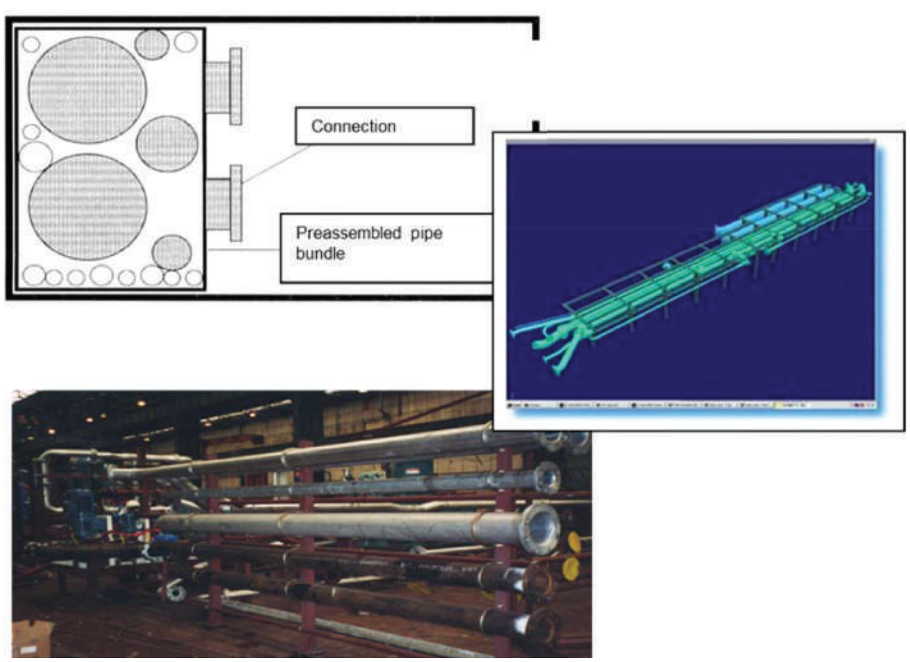
Fig. 6. Installation of a new hall over existing structures

Solution 4: Outfitting

Introduction of new designs and modern pipe connections

A lot of work force is currently used to fit and rework pipes. The introduction of new design principles would lead to decreasing rework. Therefore, it is necessary to de-velop a bidirectional interface between the CAD system for shipbuilding and the CAD system for piping. With these systems connected, the fitting points for the pipes and the ship hull structure can be easily defined to prepare the adjusting pipes between the pipe traces. The gauges of the hull

Fig. 7. Pre-assembled outfitting systems (modules)



structure define the gauges of the pipes. Along with the use of modern pipe connection systems, the cost of fitting and reworking pipes can be reduced by up to 20%.

Increasing the documentation of preoutfitting

Installation of pipe systems is a core competence of a shipyard specialized in tankers. To bundle the installation of pipe systems it is necessary to work with a good and fast documentation system to establish simultaneous engineering. The documentation system should be "open source" to include subconstructors CAD data (e.g. Integration pipe modules, pump modules, etc.). The documentation permits meeting customer's demands throughout order fulfilment. Today, changes can only be respected until the start of manufacturing.

Installation of a new 35-ton crane

The shipyard builds a new outfitting area along the pipe fabrication building. The actual crane capacity is not able to fulfil all tasks in this area. A good solution would be a crane with a 35-ton capacity and a 50-m range. With this crane, it is possible to transport pre-assembled outfitting modules into the ship. Additional flexibility could be achieved with the existing maximum range of 50 m. It is then possible to outfit two ships parallel at the same time and this leads to independencies of delivery delays by subcontractors.

Sub constructing pipe fabrication

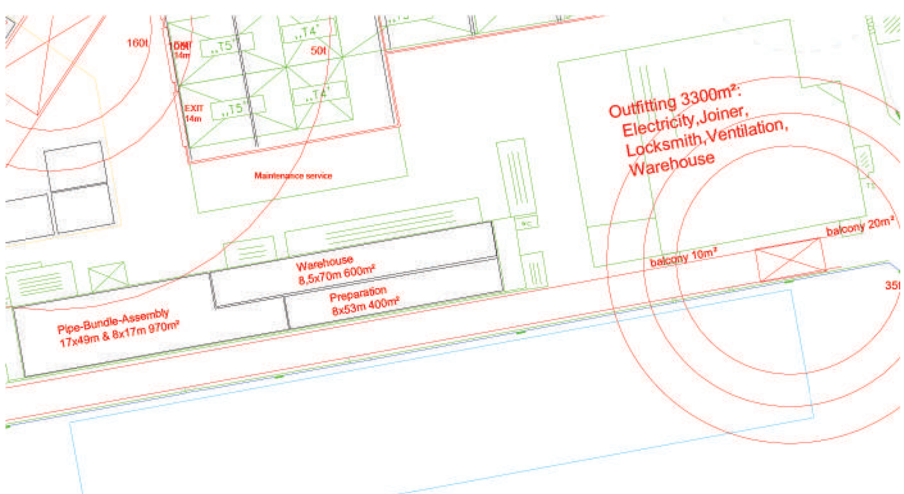
The core competence of this shipyard is the development and installation of pipe systems, as already mentioned. Preparation of these pipes can be done much more efficiently by special pipe manufacturers. Therefore, preparation of pipes should be outsourced to reduce costs as a mid-term solution. The free workers will be relocated to the pre-outfitting of sections and rings to decrease the lead time of pre-outfitting.

A 600-m² warehouse is enough to buffer the pipes on stackable pallets and deliver them "just-in-time". After these adjustments, the shipyard is responsible only for the assembly of the pipe traces and the preparation of adjusting pipes.

Relocation and elimination of outfitting processes

Along with the concentration of the core competencies the number of outfitting workshops has to be reduced and the employees relocated. The present halls and buildings are no longer necessary to full extent for the outfitting of new ships. Only the two halls closest to the outfitting quayside are needed for the new outfitting structures. Corresponding to the strategic plan of the shipyard to install a new ship repair division, the two spare buildings and the workers should be part of the new repair division. This will decrease the number of workers and areas for the shipbuilding division

Fig. 8. Relocation of outfitting processes



and increase productivity. The shipyard should look for a company that takes over parts of the outfitting and, thereby, reduces the number of outfitting procedures. A benchmark has shown that a comparable shipyard will require approx. 6000 m² for all outfitting workshops including warehouse.

Solution 5: Section assembly

With the current production, the shipyard could assemble sections for 2.5 – 3 ships maximum. The section assembly area is a bottleneck in the middle term. With a roofed-over section and module assembly area sections, more than five ships could be built. In the long term, a new hall for section fabrication has to be built.

The transport ways of pallets with profiles, plates, micro-panels, and flat panels have to be directed to the assembly area inside the new building. The area becomes assembly area for curved sections, small sections, and pre-assembled sections with a maximum weight of 60 tons. The erection of new sections takes place either on the ground or on jig pillars. The new sections will be transported either by the heavy transport vehicle or with the new transport crane.

Module assembly

The new module assembly building should be placed near the existing section assembly building and the final conservation area. The dimensions of the new building should be 56 by 32 m and it should have approximately seven assembly work-places for large sections and modules with section dimensions of approximately 12 by 12 m. There should also be a direct connection to the sections assembly building with a 60-ton crane or a roller conveyor. Inside the new hall a new 160-ton crane should be installed. Furthermore, transport of profiles, plates, micropanels, and flat panels would be secured by a predefined material flow.

The advantages of the construction of new assembly halls are the independence of light and climate conditions, increasing efficiency and capacity of supporting facilities like cranes and power supplies and production in three shifts is possible.

Solution 6: Ring and final assembly

Enlargement of the launching berth is basic for all further measures to improve out-put of the final assembly. The enlargement program includes the following steps:

- Enlargement of the existing launching berth to a width of 70 m and a length of 200 m
  - Launching berth 1 is located on the existing launching berth
  - Possibility to install new cranes with higher capacity in a long-term solution

- Steps to implement the facilities without interruption of the shipbuilding production:

1. Integration of the 160-t crane and relocation of the 50-t crane
2. Preparation of the new launching berth 2
   a. Preparation of the ground and sides
   b. Placement of the sliding rails
   c. Placement of the diagonal transport rails
3. Preparation of the old launching berth 1
   a. Preparation of the ground
   b. Relocation of the sliding rails
   c. Integration of the diagonal transport rails

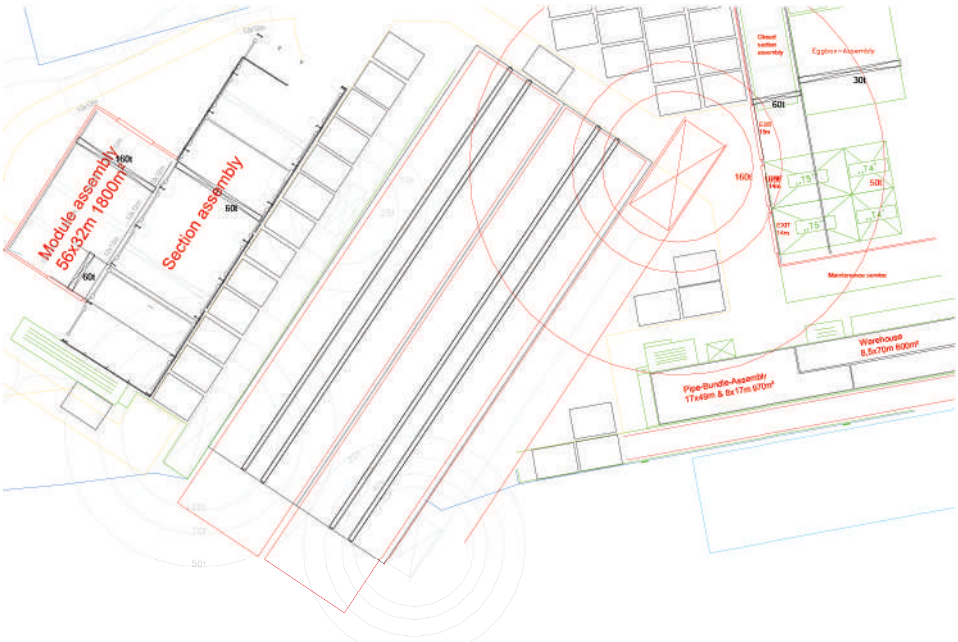
During the enlargement of the berth it is necessary to use the full capacity of both berths along the following plan to achieve the usual output rate:

1. Using the existing berth for the hull erection
2. Parallel preparation of launching berth 2
3. Using the launching berth 2 after finishing the preparation
   a. Erection of separate stern and middle/ bow ship (tandem assembly)

→ Erection of the hull on three different assembly areas on the berth

- Setting the assembly starting position
- Starting with the double bottom and pyramidal erection of the ship middle
- Hull erection with the 160-t crane for ship middle and bow
- Erection of the ship stern with the 50-t crane

Fig. 9. Initial situation in final assembly



- Connecting ship middle and stern with an adjusting cut (achievable accuracy of 60 mm)
- Improve pre-outfitting of sections with pipes and ventilation systems, especially in the ship stern on the final assembly area

Preparation of infrastructure – cranes and ways of transport

Transport capacity to the launching berth is a main bottleneck in the shipyard. Therefore, a crane shall be constructed at the berth. This is followed by the necessity to build a rail system for the cranes to move on. For the 160-ton crane, two rails with an entire length of 320 m have to be placed. The existing 50-ton crane will be relocated and installed onto two rails 290-m long.

In the berth, a rail-based transport system will be installed to allow sections being transported lengthwise and crossways. At five places, there has to be a crosslink between the two berths to allow cross transport. The allowed load of the transport system has to be up to 1800 t. The length of the whole rail system lengthwise and crossways is about 1140 m. Parallel to the transportation system, a positioning system is planned to align the ring sections with the ship. This results in a decrease of crane demands. Furthermore, sliding bars have to be installed for launching; they are 640 m cumulated. Possible workflows on the launching berth after the rebuilding are shown in the following alternatives.

Ship hull erection

Different alternatives exist for hull fabrication and assembly. Fraunhofer prefers alternative 2 and alternative 1 as an interim solution until all systems are implemented into the launching berth. Alternative 1a is merely mentioned for the sake of completeness but should be avoided.

Alternative 1

Two ships are being erected in parallel with today's methods. On the first berth, mainly the 100-ton and the 25-ton cranes and on the second berth the newly installed 160-ton and 50-ton cranes are used. Using the tandem assembly method, the ship stern and middle are erected at the same time. Therefore, the ship stern is being positioned opposite of the ship middle by the new installed transportation and posi-tioning system. The assembly procedure on the launching berth could be supported with three cheap and profitable tower cranes. An additional investment for a new 160-ton crane for the launching berth 1 should be taken into account.

Alternative 1a

Both launching berths 1 and 2 with their transport and positioning systems will be used for hull erection. On the free spaces of the launching berths the rings will be erected in a defined tact. The degree of completion will be less with each ring. This alternative recommends another crane on the second berth. The erection procedure on the launching berth could be supported with three cheap and profitable tower cranes.

Alternative 2

With this alternative, the ship is erected in the first berth and the ring section assembly takes place in the second berth. It starts with the erection of the ship stern sections, which are transported to the first berth directly after completion. Afterwards, the ring sections are assembled to be pushed onto the ship stern. Five additional workplaces will be available. One of them for the erection of the ship stern and four for the assembly of the ring sections of the middle ship. All together, the ship with a length of 180 m consists of six ring sections and, respectively, one ship stern and one ship bow. Transport and positioning are managed by the rail system installed.

Ring assembly workstation on the launching berth

The area at the end of the second berth is designated for ship stern ship erection. The 50-ton crane does most of the work on this berth. The erection takes place on so called keel block pillars, while the stern is put together from all the small sections. The superstructure on the ship stern should be assembled as one module. The weight of the whole ship stern structure with the superstructure on top is about 1800 t.

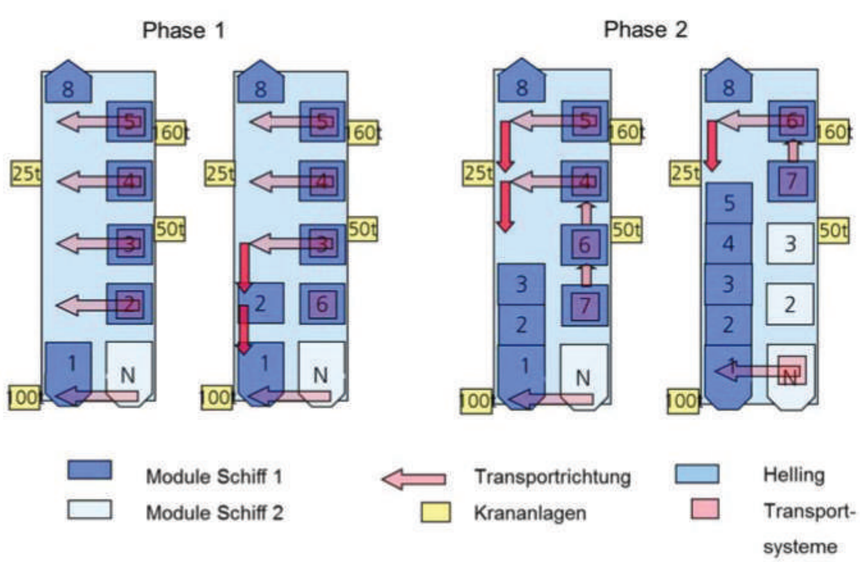
Transport to berth 1 is managed with the use of keel block pillars. Placement of the superstructure and the main engine will be done by a floating crane.

In the area above the ship stern erection, the ring sections of the ship middle are assembled. The 160-t crane supports the erection. The ring sections will be approximately 23 m long and have a weight about 1200 t. At these workplaces, pre-outfitting of the sections also takes place. The sections have to be assembled according to the defined erection sequence.

The area at the top of the launching berth 1 is allocated to the ship bow erection. The erection is supported by the 100 and 25-t cranes. The ship bow will be transported to the rest of the ship. After welding the sections, the hull is complete and the ship can be launched.

Because this shipyard did not introduce the assembly of ring sections, it was necessary to

Fig. 10. Concept of the new final assembly



analyze comparable shipyards with this erection method. Through these comparisons, a possible increase of productivity caused by the change of methods can be previously calculated. As basis, analysis from German shipyard activity sampling is taken. In general, it can be said that enclosure of the assembly area into closed buildings could increase productivity by about 20% just through light and climate conditions. Use of mechanical welding could increase this productivity by another 25%.

Another advantage of mechanization is that employees are free to work in second and third shifts.

Solution 7: Organization

Production planning and control and simulation

To introduce manufacturing simulation, the simulation model developed by Fraunhofer should be used. Here, reference sections are defined and their lead times in the main production and in bottlenecks are acquired. These sections would be the part fabrication, panel assembly, section assembly, and hull erection. Simulation can be started with this data. In a following step, this simulation can be upgraded with data from PPC systems, NC systems and CAD systems. A database reference to classify the building parts and groups is created. By implementing a specific IT structure, the simulation can be done in real time.

Multi-sampling

Multi-sampling can be used in the ship building industry, but due to the high complexity of the workflow and the assembled structures, modifications in preparation, execution, and analysis have to be undertaken. Thereby, this increases the evaluation possibilities and creates a tool to analyse and control the lead times.

The aim is to find the production expenses and develop comparable figures for effort estimation and controlling. The procedure to enforce activity sampling in the shipyard is very complex. Therefore, an expert team is needed to generate it in the shipyard, which will be trained by external specialists, *i.e.*, Fraunhofer AGP.

Solution 8: Design requirements

The main fraction of the costs is generated during the planning and constructing phase due to mistakes that delay the assembly or lead to reworks. By rationalizing in this area and introducing "FMEA", costs could be reduced significantly. It is highly recommended to change design methods to design for manufacturing and design for assembly.

Investment plan

The investment plan is based on the concept developed and current possibilities at the shipyard.

Fig. 11. Implementation of shop floor simulation: Future scenario

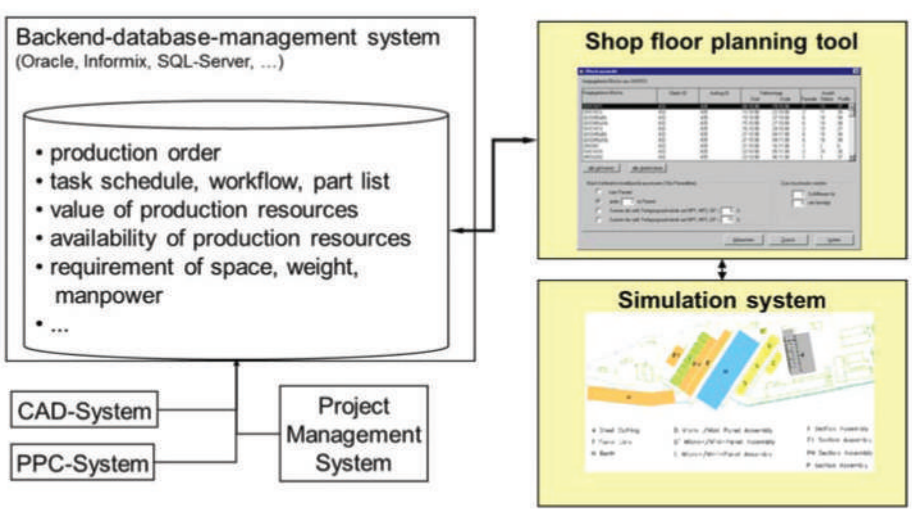
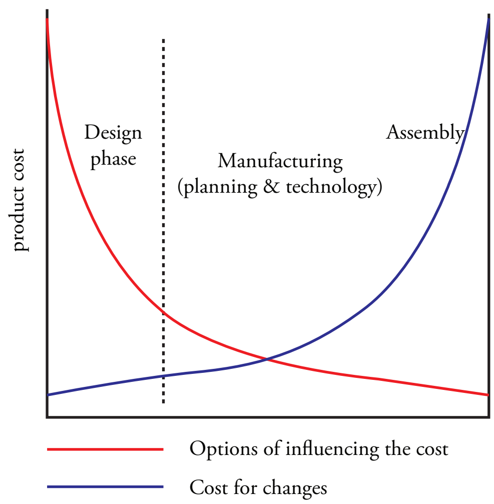


Fig. 12. Influence of design requirements



The costs for investments depend on prices estimated from Fraunhofer and the shipyard and on market prices. The investment plan is divided into three parts and should be followed consecutively.

The first package includes new section and steel conservation buildings, the new part and panel fabrication hall, as well as IT infrastructure. The investment volume is about 16.1-million Euros. The second package includes the new cranes, the berth's extension, and the rail and transport system, as well as pipe conservation, welding robots and the new positioning of the outfitting. The volume of this investment is about 15.4-million Euros. The third package includes the rail system for the ring as-sembly, the keel block pillars, the ring positioning system, and the web mounting gantry.

Additionally, the new building for the module assembly with the correspond-ing cranes and a profile bending machine complete the package. Its investment vol-ume is about 8.3-million Euros.

The complete investment volume is 39.8-million Euros. The actions combined in these packages should be carried out as a whole to equalize the output of all work-ing areas and facilities. Should any points of this investment plan not be arranged, this would have negative consequences for the shipyard.

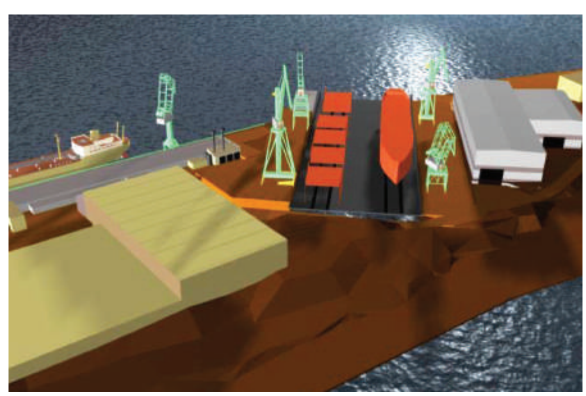
Productivity analysis

By implementing the measures introduced, the shipyard could increase the actual production from 2 ships per year up to 5 ships per year with the same number of employees. Currently, the productivity is about 18.5 tons of steel per person per year. The launching berth and the cutting station turned out to be the bottlenecks in the manufacture process. By arranging the investments mentioned in package 1, organisational changes and the use of the tandem assembly method, approximately three ships per year could be manufactured; this corresponds to 50% productivity increase.

After this phase, the berth with its cranes will be the new bottleneck. After applying the second package of investments, productivity will increase by another 26.7% and the shipyard could now build 3.8 ships per year. Again, there would be a shift of the bottleneck to the transport capacity at the manufacture section. After completing the investment plan's third package, another increase of 31.6% in production is possible. The shipyard would now be enabled to produce five ships. The capacities would be balanced. The assembly section would be bounded by the closed fabrication section and the fitting durations at the ring assembly to about five ships. The area on the launching berth is also bounded by the fitting duration of the ring sections.

With optimal process control and discarding of more measures even the manufacture of six ships per year is possible.

Fig. 13. The new shipyard concept



Conclusion

The targeted aim to produce 4 ships is achievable if the previously mentioned measures are carried out. Due to upcoming crisis in shipbuilding, the shipyard did not find a new private owner until the summer of 2012. The conversation process did not start until this time.

Bibliography

http://www.fraunhofer.de/en.html

Design Model of a Hydrodynamic Towing Tank for Colombia

Modelo de diseño de un canal de ensayos hidrodinámicos para Colombia

Wilson Toncel Zuleta [1]
Jairo Cabrera T. [2]

Abstract

This paper develops a methodology for the design of a hydrodynamic ship tank model and its main components considering the current and projected needs of the maritime and river industry of Colombia. Besides allowing resistance, seakeeping, and maneuvering model tests with international standards.

Key words: Ship model hydrodynamic tank, wave generation, PMM, ship model scale

Resumen

En el presente trabajo se desarrolla una metodología de dimensionamiento para un canal de ensayos hidrodinámicos y sus principales componentes considerando las necesidades actuales y proyecciones del sector marítimo y fluvial de Colombia. Además de permitir realizar ensayos de resistencia al avance, comportamiento en olas y de maniobras con estándares de confiabilidad internacional.

Palabras claves: Canal de ensayos, hidrodinámica, generación y amortiguación de olas, ensayos modelos a escala de buques

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Introduction

Colombia has broad coastlines and hydrographic basins that have granted it great diversity and hydric richness; its seacoasts and rivers are potential commercial and transportation paths currently unappreciated, perhaps without knowing that using this mode of transportation may significantly reduce logistics costs. In spite of having coastlines on two oceans, the development of the naval industry in Colombia has been limited due to lack of ideas and proposals that push an effective start to this field.

Colombia's strategic location offers the advantage of availing of the richness of riverine and maritime resources, only requiring an initiative to adequately use said geographically prized richness.

Appropriate development of the naval industry in Colombia will permit addressing the growing demand for mobility presented in these moments of industrial expansion.

Due to the aforementioned, it becomes important for the development of the naval area in Colombia to have a Hydrodynamic test tank and which can be defined as a physical laboratory in which tests are developed with models of scaled-down geometrically similar maritime systems, to predict the behavior of said designs under specific conditions in the different situations at sea.

A broad variety of channels for hydrodynamic tests exists, depending mainly on the purpose or types of tests destined for execution.

Among the channels with captive models there are those aimed at evaluating pure drag and having sufficient breadth may, likewise, run maneuverability tests for which a planar motion mechanism (PMM) is required.

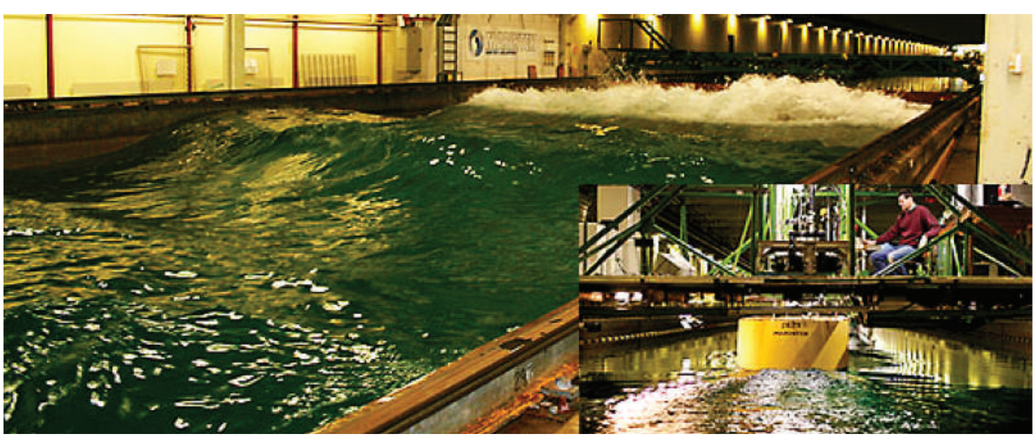
Other channels are destined to evaluating the behavior against the environmental conditions of vessels and/or maritime platforms for which specific components are necessary as wave generators and dampers, besides equipment associated with the acquisition and treatment of data. Fig. 1 shows the hydrodynamic test tank in MARINTEK (Norway).

This work is aimed at the development of a towing tank sizing model, which – likewise – permits analyzing behavior in waves and of maneuverability, considering the current and projected needs of the Colombian naval and riverine industry.

Use of test tanks in ship design

During its early days, naval design was predominated by empirical tendencies; it may be said that the first studies using scale models were by Leonardo Da Vinci *(Alaez J, 1953)*, after him, followed in the progress of said field, researchers like D'Alembert, John Bernoulli *(Alaez & Roberston, 1965)*, and John Smeaton

Fig. 1. Test tank at MARINTEK



(Dos Santos & Michima, 2007) among many others.

Since then, naval design has, thus, evolved permitting greater efficiency in each of the processes related to such. Naval design currently has numerous computer tools that, through good management, could provide an approach of the behavior expected in the vessels; however, much is still needed for these procedures to be completely reliable, which is why tanks are used to perform different tests.

From these tanks diverse maneuvers may be executed with which we can obtain values and interpret such to predict the behavior of a particular design. The difference with the computational models lies in the approximation to the real behavior, carrying out good practices with adequate equipment.

The sizing model proposed for the test tank presents the general and specific characteristics of a tow-type tank, as well as all its necessary component equipment, according to the tow test requirements, behavior on waves, and maneuverability.

Naval construction in Colombia

According to a recent study of the shipyard sector in Colombia conducted by Universidad Tecnologica de Bolivar, Universidad del Norte, Universidad del Rosario, and COTECMAR in 2012, this activity is concentrated thus: Caribbean region (Cartagena, Barranquilla, and Santa Marta) with 55% of the total; Antioquia (Medellín, Envigado, and Turbo) participating with 18%, the Pacific Coast (Cali, Buenaventura, and Bahía Solano) participating with 15%; followed by Cundinamarca with 12% of the total.

The same study reveals that the vessels with the greatest demand are:

- Inflatable Pleasure Boats
- Boats
- Canoes
- Skiffs
- Water Bikes
- Barges
- Tugboats
- Fast Patrol Boats
- Heavy Patrol Boats
- Patrol-Type Ship

COTECMAR[1] is among the most highlighted companies in the sector, being the main shipyard in Colombia and which has been waging on the innovation management process, promoting the culture of naval design and construction, achieving the development of patrol-type boats, landing ships, and fast boats, among others.

Currently, COTECMAR is working on the development of a strategic surface platform (SSP) that will substitute the current fleet of frigates, major defense ships in the country.

Development of the hydrodynamic tank proposed in this work will permit optimizing disciplines of the design process, especially those related with the calculation of drag, behavior on waves, and maneuverability with sufficient capacity to meet the current and projected demand of the SSP project and other medium and small-sized ships requiring design in the country.

Sizing model

Determination of the adequate dimensions for a hydrodynamic test tank is closely related to the geometric characteristics of its components and to the dimensions of the scale models to be evaluated.

The ITTC[2] is the regulatory entity of the practices mentioned, accrediting institutions that comply with the minimum requirements for the reliable execution of tests and standardizing the procedures to be followed in each of the practices.

The elements participating in the execution of tests for the hydrodynamic tank proposed are:

- Scale models
- Tow truck - PMM

[1] Science and technology corporation for the development of the naval, maritime, and riverine industry.
[2] International Towing Tank Conference.

- Wave generator
- Wave absorber

Of the previous components, the scale models play the most relevant role in the whole tank sizing process; based on these, the following will be the defined characteristics of wave generation, dimensions of wave absorption beaches, and PMM tow rate and acceleration.

From the dimensions of the models, the reference scales are defined in the different tests, thus, the main dimensions of the tank have a direct relationship with the geometric characteristics of the models to evaluate.

In carrying out the sizing model of the test tank a data bank was required, contemplating the characteristics of the types of ships to be tested.

Fig. 2. Sizing model of the hydrodynamic test tank

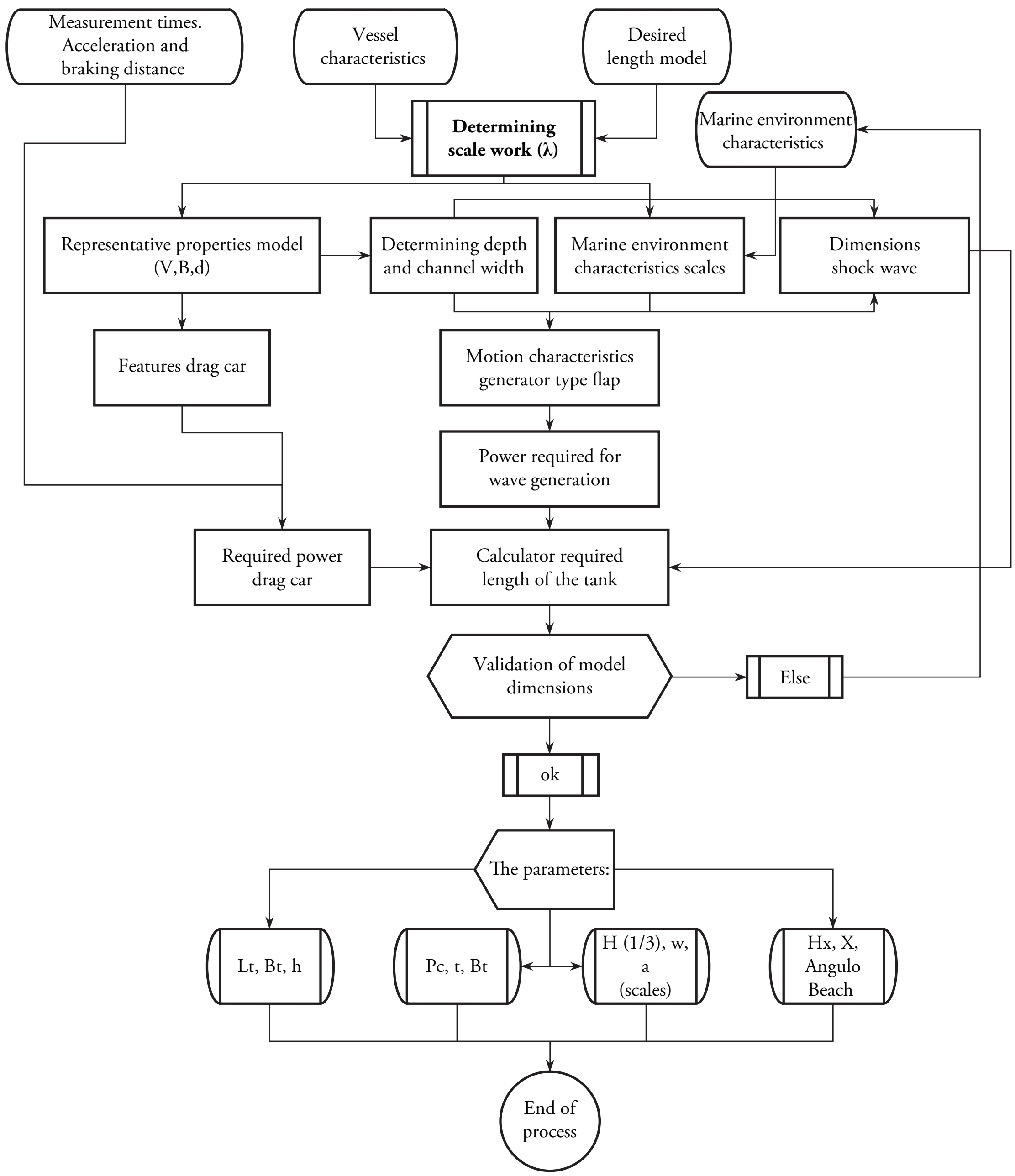


Fig. 2 shows the sizing model of the tank and its different components. As input values of the model, the geometric characteristics of the scale model, conditions at sea to recreate, and the parameters of test execution are necessary, thus, obtaining the dimensional value of each of its basic components.

Taking as reference the parameters of the tank's constitutive elements, we managed to determine from equations 1 *(Ueno & Nagamatsu, 1971)* and 2 (citation VID12 \l 9226 | *(Vidal Bosch & Barberá Fernández, 2012)* its appropriate dimensions: length, depth, and breadth.

$$B_T > \frac{3}{2} L_M \quad (1)$$

$$h_T > \frac{10{,}72 \cdot T \cdot V}{\sqrt{L_{pp}}} \quad (2)$$

Where:

L_{pp}: Length between perpendiculars
B_T: Tank width
L_M: Length of model
h_T: Tank depth
T: Draft of model
V: Tow rate

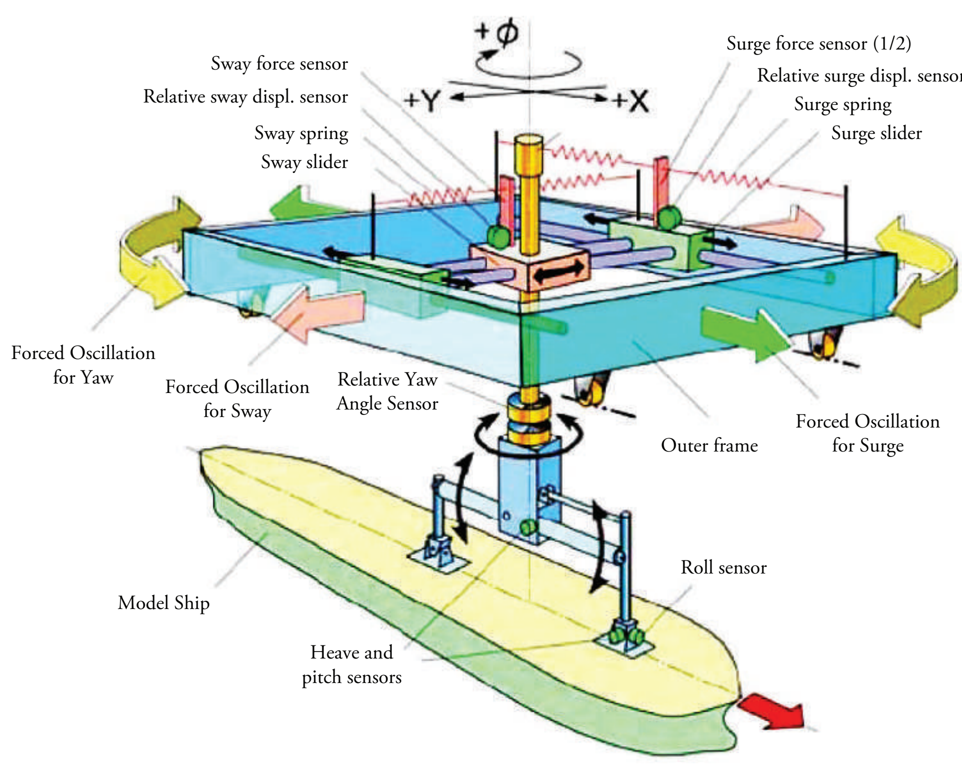
Scale model

As previously mentioned, scale models represent parameters of special importance to size the tank and its components, as observed in Fig. 2.

For tank sizing, we must have the initial value of this parameter to start the process; at the end we validate if the values obtained, with the scales worked, comply with the functional restrictions for each test proposed; otherwise, the necessary iterations must be conducted to achieve the respective adjustments.

Determination of the magnitudes of the models leads to the use of similarity laws; for this case it will be geometric similarity, although there are also kinetic and dynamic similarities.

Fig. 3. Tow truck or PMM (ITTC, 2011)



The geometric similarity proposes that the relationship between the "length" (length, breadth, draft, etc.,), L_g, of a ship at real scale and the "length" of the scale model, L_m, must be constant; this relationship is called scale factor and it is designated with the letter λ, according to equation 3.

$$L_s = \lambda \cdot L_m \tag{3}$$

By using λ we will manage to maintain similarities regarding conditions at sea and velocities.

PMM Sizing

The sizing process of the Planar Motion Mechanism (PMM) or tow truck refers to the calculation of the power required to propel the structure under the parameters desired.

Fig. 3 shows a sample PMM structure. This component is closely related to tow, vertical motion, and maneuverability tests. Its calculation is made based on the nature of the motion in each of its stages, as noted in Fig. 4. The process starts with the determination of the forces present for the zones of constant acceleration, calculated according to equation 4.

$$\Sigma F_a = F_{ra} + E_f + F_{IM} + m_{ca} a \tag{4}$$

Where:

F_a: forces in the acceleration zone (N)
F_{ra}: air resistance force:

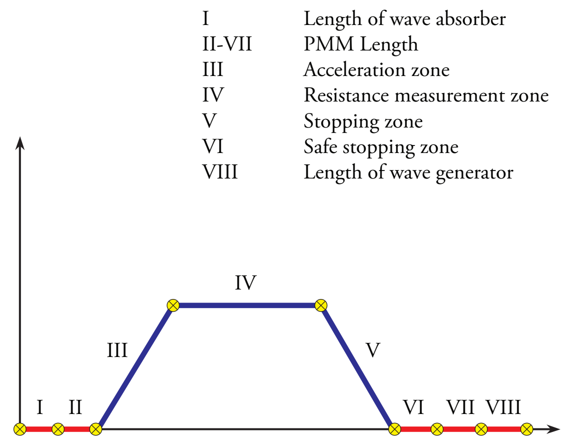
$$\left[\frac{1}{2} C_{aa} \rho_{aa} V^2 A_{TC} \right] \tag{5}$$

Where:

F_f: frictional force on the four wheels
F_{IM}: drag force induced by the model
m_{ca}: estimated mass of the tow truck (Kg)
a: acceleration (m/s^2)

The determination of power results from the multiplication of the sum of forces and the velocity to be reached. In the constant velocity zone, the forces would be evaluated through equation 6.

Fig. 4. Tow rate and tank length ratio (Saldarriaga Muñoz, 2011)



$$F_{vc} = F_{ra} + F_F + F_{IM} \tag{6}$$

Upon calculating the power required in the constant velocity zone, we determine the required nominal power as the mathematical sum of the previously mentioned. Fig. 5 details the complete methodology of the calculation.

Wave Generator

The wave generator does not have the simple purpose of creating arbitrary disturbances in the work fluid; rather, its final purpose is that of reproducing the sea state under different conditions, from the most passive to those of greater magnitude.

To design a wave generator, we must consider the relationship between wave height and the displacement of the wave generator actuator; this ratio is given by the following equation *(Dalrymple & Dean, 1984)*:

$$\frac{H_{\frac{1}{2}}}{s} = 4 \left(\frac{sinhK_v h}{K_v h} \right) \frac{K_p h_0\, sinhK_v h + coshK_p(\Delta V) - coshK_p h}{2\, K_p h + sinh2K_p h} \tag{7}$$

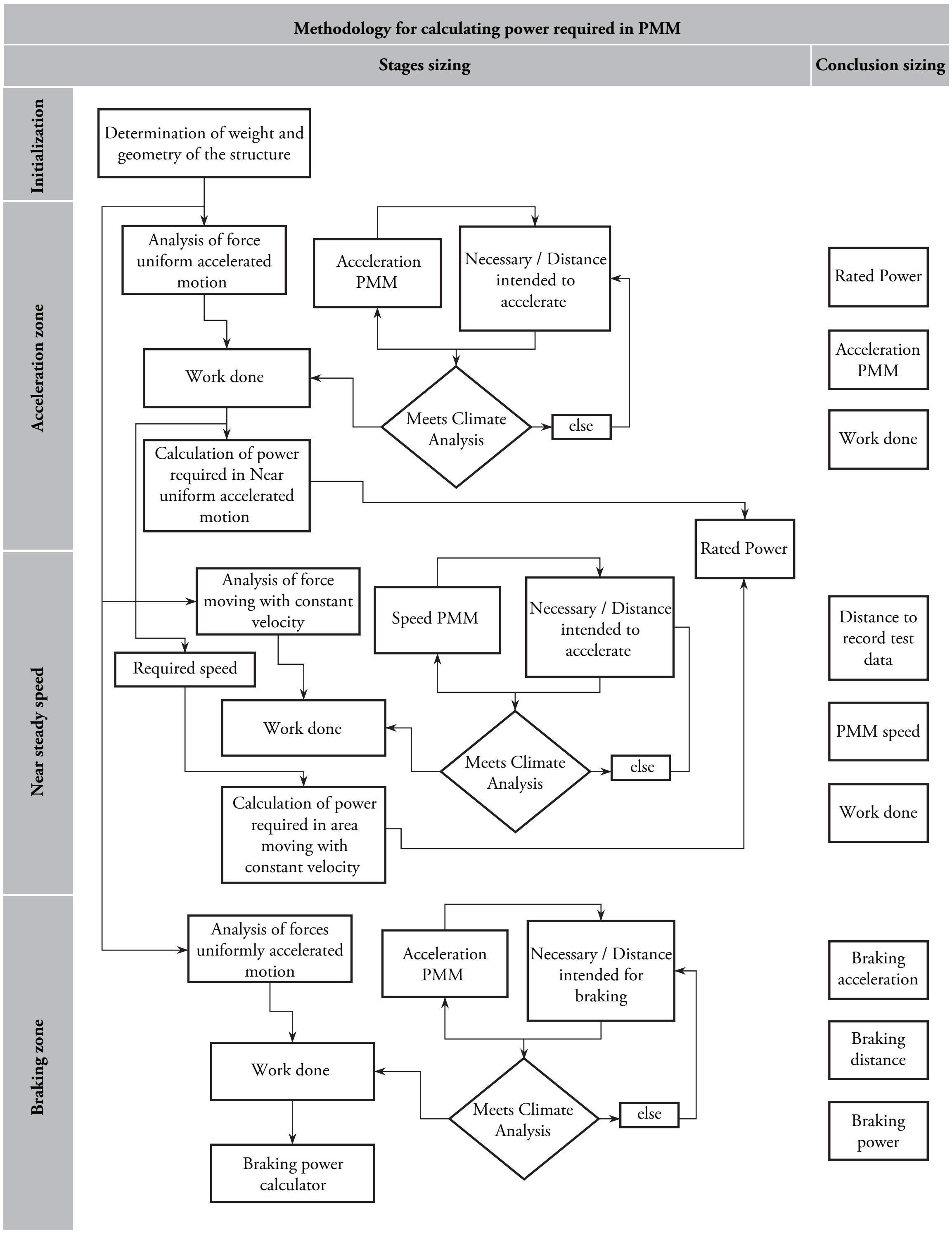
Where:

$H_{1/3}$: wave height
S: Course or "stroke"

K_p:K: wave number (2π/Lw)
L_w: wave length
h: total depth of tank

By calculating this equation, we can determine the stroke the generation plate must have, becoming only a geometric problem the determination of the

Fig. 5. PMM sizing model



angles of motion, the minimum length required for the tilting plate, see Fig. 6.

The resistive forces (FRR) are obtained through calculating the momentum on the rotation axis *(Gonzalez Alvarez Campana, 1988)*, which varies in function of the plate height and its motion according to equation 8.

$$FRR(h-\Delta v) = \frac{\rho g a}{K \cdot cosh(K \cdot h)} \left[sinh(K \cdot h) - \frac{1}{K(h-\Delta v)} \cdot (cosh(K \cdot h) - cosh(K \Delta v)) \right] \quad (8)$$

Power was determined according to equation 9.

$$P = \frac{1}{2}\omega \cdot s \cdot FRR = \frac{\pi \cdot s \cdot FRR}{T} \quad (9)$$

Wave absorption beach

The wave absorption beach is a damping element of the waves present in the tank. Its function is to dissipate the energy produced by waves, avoiding disturbance of the simulated environment by the reflection of the waves and interference in recording the movements of the model, see Fig. 7.

Fig. 6. Sizing model of wave generator

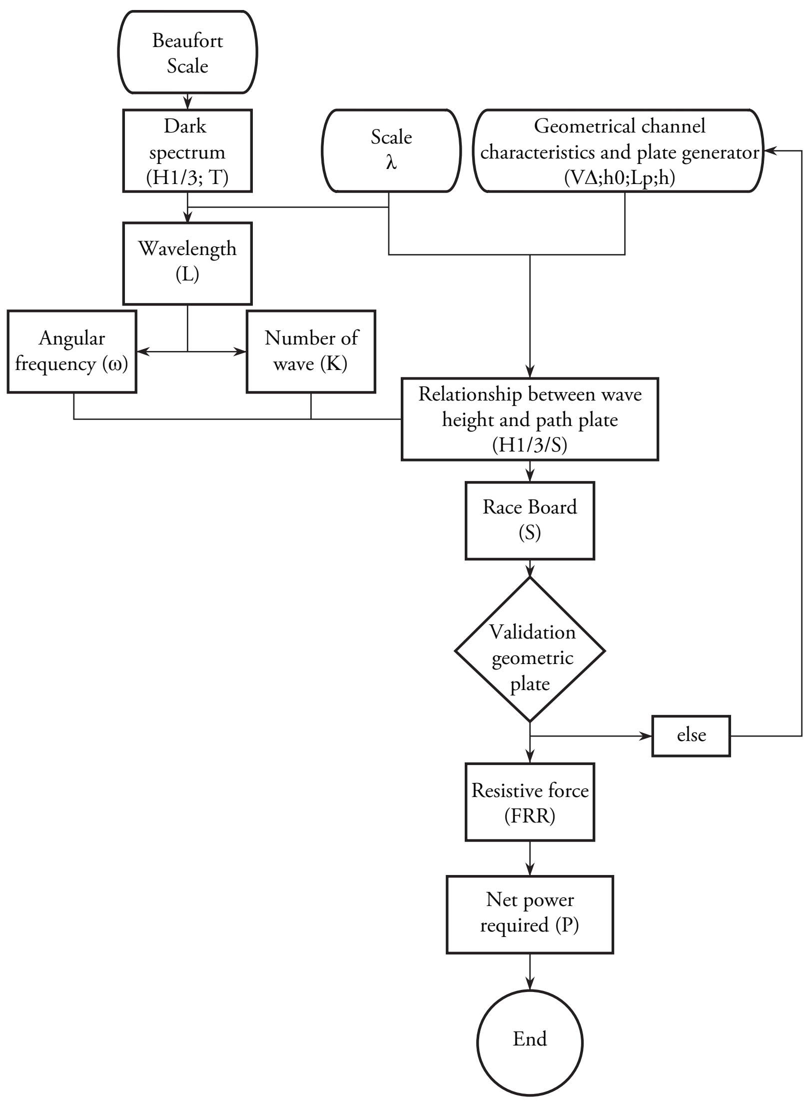
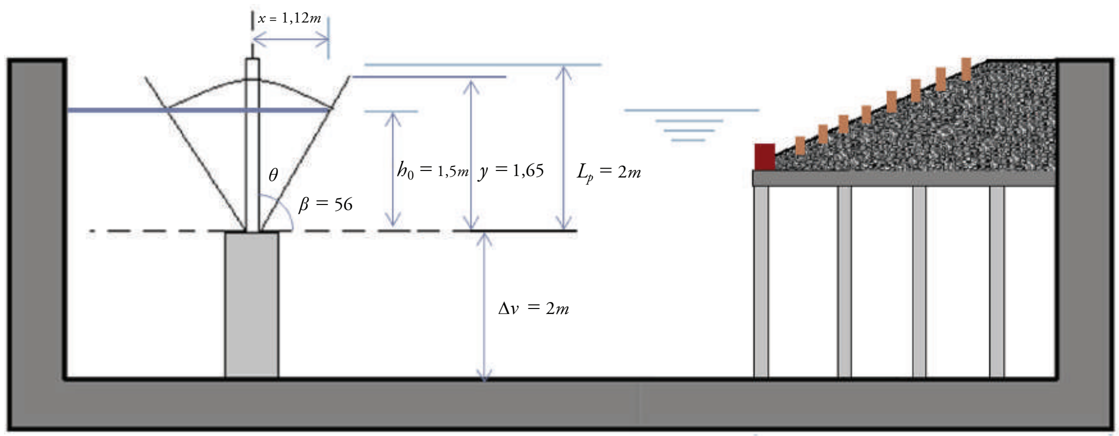


Fig. 7. Diagram of wave generator and damper



Sizing of this component, its length (x) and height (h_x) may be evaluated from Svendsen and Jonsson's equation, where (h_T) is the tank's depth and (L_w) the length of the wave *(Saldarriaga Muñoz, 2011)*:

$$\frac{h_x L_w}{h_T} \approx 1 \quad \rightarrow \quad h_x = \frac{h_T}{L_w} \tag{10}$$

Based on the Irribaren ratios *(Saldarriaga Muñoz, 2011)*, we may calculate the angle of the absorption beach:

$$\theta = tan^{-1}\left(lr \cdot \sqrt{\frac{H_{\frac{1}{3}}}{L_w}} \right) \tag{11}$$

And through simple geometry, calculate the beach length required:

$$x = \frac{h_x}{tan(\theta)} \tag{12}$$

Application of the model

analysis of the global tendencies for the size of the model was considered as a starting point; a value of up to 4 m is considered and a CPV[3] -type vessel is selected, which serves as reference for the development of the model.

By following the stages of the model proposed, we achieve the sizing of the tank and its components, obtaining the data presented by the following:

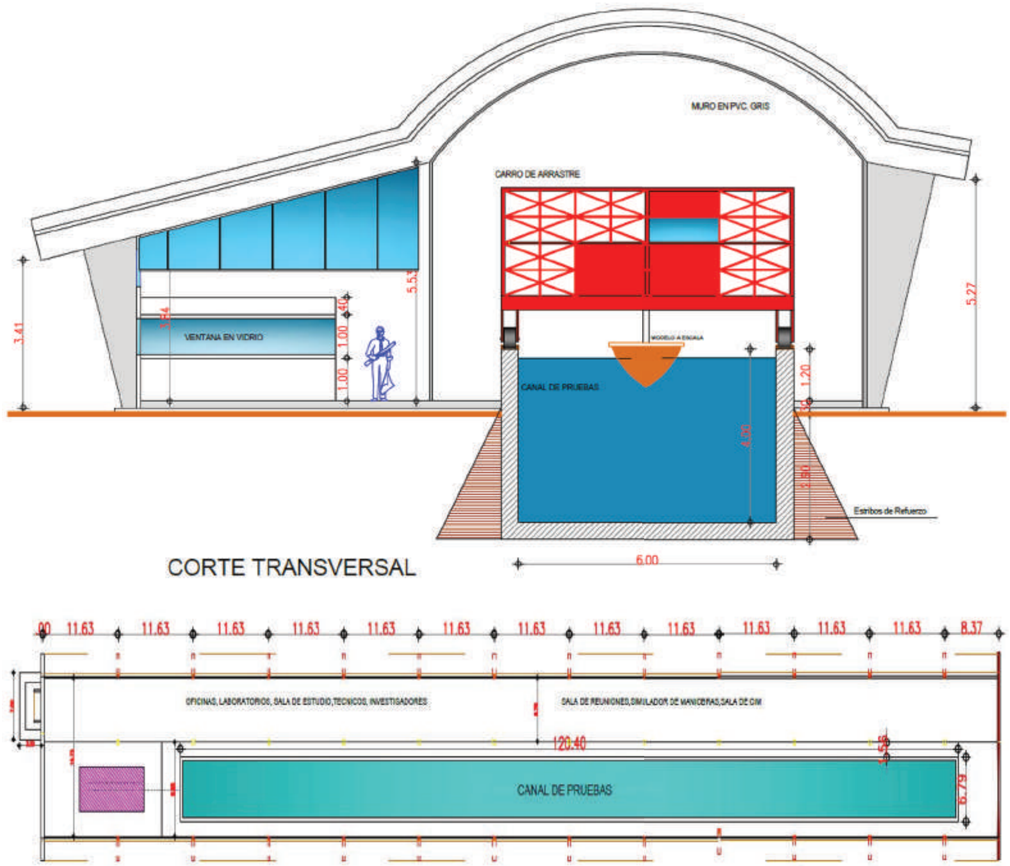
TANK	Tank dimensions will be lxbxh: 120x6x4 (m)
DYNAMOMETRIC TRUCK	Required impulse power 162.3 hp Stopping power: 70 hp
WAVE GENERATOR	Required power: 7.26 hp The frequency of agitation will depend on the sea state to reproduce.
WAVE ABSORBER	Dimensions required will be lxh:10.3x2 (m); these will provide adequate absorption under the different wave parameters worked
MODEL	Sizes recommended for the models could vary between 1m<L_m<4.6m

The test tank, product of the application of the model, can reliably run tests for a broad range of small ships, including maritime and riverine patrol boats, multi-purpose tugboats, and even frigate-type ships from the Strategic Surface Platform project.

Fig. 8 shows the preliminary architectural project of the hydrodynamic test tank proposed for Colombia.

[3] Coast Patrol Vessel

Fig. 8. Architectural plans of the hydrodynamic test tank building



Conclusions

After analyzing the critical variables in sizing a hydrodynamic tank, equally covering current and projected needs of the naval industry in Colombia, it was possible to methodologically establish the preliminary dimensions of a tow test tank complying with the different restrictions imposed by the authors.

The facilities were designed to have the capacity to execute tow, seakeeping, and maneuverability tests according to the recommendations established by the *International Towing Tank Conference* – ITTC.

Bibliography

ALAEZ J, T. L. (1953). Leonardo Da Vinci and Problems of the navegation and naval design. International Journal of Marine Design, vol 95.

ALAEZ, J., & ROBERSTON, J. (1965). Hidrodinamics in theory and applications. Prentice-Hall.

DALRYMPLE, R. A., & DEAN, R. G. (1984). Water Wave Mechanics for Engineers & Scientists (Vol. II). Singapore: World Scientific.

DOS SANTOS, P., & MICHIMA, S. (2007). Estudio de un tanque amortiguador de onda para la enseñanza activa. Disertación, Universidad De Sao Paulo, São Paulo.

GONZALEZ ALVAREZ CAMPANA , j. (1988). Nuevos metodos de prediccion de calidad de las olas generadas en laboratorios. tesis doctoral.

ITTC. (2008). Recommended procedures and guidelines. ITTC, Maneuvreing Committee.

ITTC. (2011). Final report and recommendations to the 26th ITTC. Technical Report, The Maneuvring Committee.

SALDARRIAGA MUÑOZ, J. M. (2011). Estudo de uma metodologia para o dimensionamento de um tanque de provas do tipo reboque. Tesis de Maestria en Ingenieria, Universidade de São Paulo, Escola Politécnica, São Paulo.

UENO, K., & NAGAMATSU, T. (1971, March). Effect of restricted water on wavemaking resistance. Journal of Seibu Zosen Kai-The Society of Naval Architects of West Japan(41).

Universidad Tecnológica de Bolívar, Universidad del Norte, Universidad del Rosario, & COTECMAR. (2012). Sistema Sectorial de innovacion de la industria astillera de Colombia.

VIDAL BOSCH, J. R., & BARBERÁ FERNÁNDEZ, A. (24 de 01 de 2012). Enavales. Recuperado el 10 de 10 de 2012, de http://www.enavales.com/index.php?option=com_phocadownload&view=category&id=14:n-hidrodinamica-resistencia-y-propulsion-&Itemid=74#

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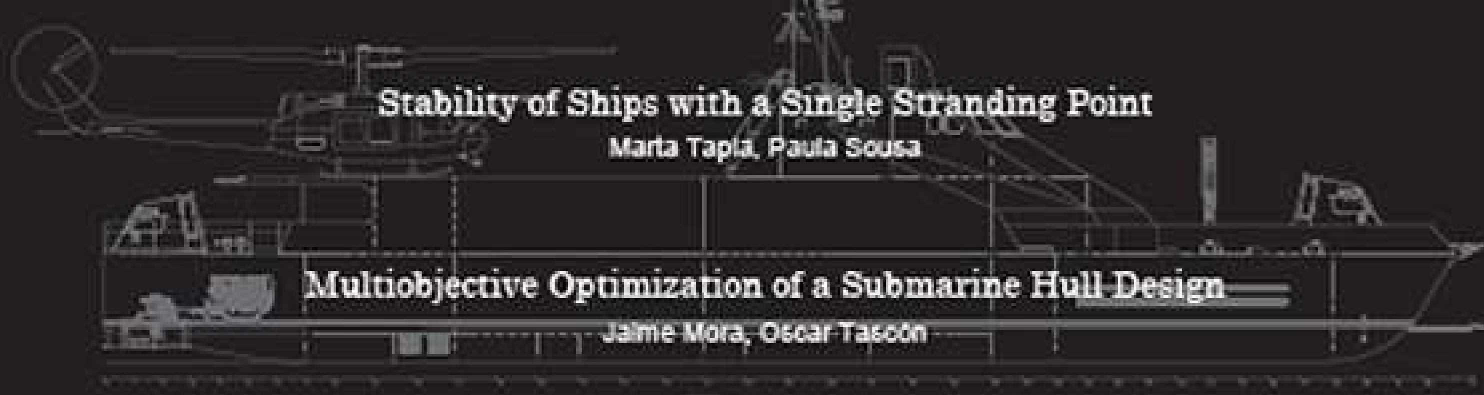

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