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RESISTANCE AND POWERING PREDICTION USING AUTOPOWER MODULE OF AUTOSHIP SOFTWARE CASE STUDY: OCEANOGRAPHIC RESEARCH ROBOT

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ABSTRACT

In the preliminary design stage of displacement ships, assessing total resistance and propulsion power demand is critical, since variations in hull form parameters significantly influence propulsion efficiency and engine sizing. In this study, the AutoPower module of the AutoShip software suite is used for a monohull vessel and a catamaran with a symmetric hull, both having similar main dimensions. Resistance and effective power were computed for a speed range between 0 and 22 knots using several empirical methods implemented in AutoPower, including Andersen/Guldhammer, Holtrop, van Oortmerssen, Digernes/Cheng, and Fung. The results highlight significant variations between the methods, with effective power predictions at the service speed of 18 knots ranging from 3.5 kW (Holtrop) to over 20 kW (Digernes/Cheng). The originality of the paper lies in emphasizing both the usefulness and the limitations of empirical formulations in the preliminary design of ships. The study concludes that, although AutoPower provides a fast and practical framework for estimating power requirements, the choice of prediction method must be correlated with the ship type and hull characteristics, while validation through model tests in hydrodynamic towing tanks or CFD simulations remains essential to ensure reliability.

Keywords: AutoShip, AutoPower, total ship resistance, effective power, oceanographic research robot, Mono hull, Catamaran hull

1. Introduction

The calculation of ship resistance is a central pillar in the design of displacement regime vessels ($Fn_V < 1$) [1], establishing the basis for sizing the propulsion system, estimating fuel consumption, and defining operational efficiency. The total resistance results from three main components [1], [2], [3], [4], [5] the resistance formed by frictional and residual resistance, the resistance formed by pressure and frictional resistance of the hull, and the third, the resistance formed by wave-making and viscous resistance – all directly influencing the required power, speed, and the vessel's environmental impact. [6]

In the ship design process, resistance prediction is introduced as early as the preliminary design phase (conceptual design). It is one of the earliest and most critical stages that guide decisions regarding hull form, engine selection, and the general configuration of the vessel. Quick estimations — whether through empirical methods, model tests (towing tank simulations), or CFD simulations — are essential for guiding subsequent optimization, as well as for reducing both costs and the overall design cycle time. [7]

In recent years, ship resistance prediction has experienced significant progress due to the integration of artificial intelligence methods and the increasing availability of experimental and numerical databases. These developments provide designers with faster and more accurate tools, reducing the exclusive reliance on traditional empirical methods or costly towing tank tests.

One major area of development is represented by the applications of machine learning. Recent studies have shown that techniques such as "stacking ensemble learning" and "transfer learning" can outperform classical methods (e.g., Holtrop–Mennen, Guldhammer–Harvald) in predicting the resistance of containerships. *Yang et al. (2024)* demonstrated that these methods ensure high accuracy even when datasets are limited, making them attractive for preliminary design stages. [8] In the same direction, *Saha et al. (2024)* proposed a "surrogate model" based on neural networks for the *AMECRC (Australian Maritime Engineering Cooperative Research Centre hull)* series, capable of accurately reproducing resistance components while significantly reducing computation time compared to CFD. [9]

Another major direction is the prediction of added resistance in waves, a critical component for evaluating performance under real operating conditions. *Zhang (2025)* employed Gaussian Process Regression (GPR) trained on extensive experimental databases, obtaining a fast and robust method for estimating the additional resistance generated by head waves. [10] In parallel, the DTU team *(Amini-Afshar et al., 2025)* developed "deep learning-based models" trained on data generated through strip theory, capable of providing generalizable predictions for various ship types and sea states. Validation using operational data confirmed the usefulness of these tools for real-time prediction. [11]

Regarding hull form optimization, surrogate models have become increasingly important. *Zhu et al. (2024)* applied evolutionary algorithms assisted by surrogate models to reduce resistance and energy consumption in the design of "green ship design", demonstrating the potential of these techniques for developing energy-efficient hulls. [12] Likewise, *Loft, Schwarz, and Rung (2025)* introduced a model for predicting three-dimensional pressure fields using convolutional autoencoder networks, which enables fast estimation of added resistance under various sea state conditions and opens new possibilities for optimized ship routing. [13]

An emerging trend is represented by interpretable hybrid models that combine physical fundamentals with data-driven methods. *Papandreou et al. (2025)* proposed a model that integrates three degrees of freedom (3DoF) motion equations with parameter optimization based on experimental and operational data. This approach ensures more robust predictions while preserving physical interpretability, an essential aspect for practical application in ship design and operation. [14]

Overall, these developments confirm a transition from purely empirical methods toward an integration of hybrid models and artificial intelligence, with clear benefits in terms of computational speed, prediction accuracy, and flexibility across various operational scenarios. However, high-fidelity CFD validation and experimental testing remain indispensable for ensuring the reliability of the results.

The **Andersen / Guldhammer – Harvald method**, developed in the **1960s–1970s** based on an extensive database of model tests, provides empirical formulas for determining the residuary resistance coefficient as a function of the Froude number, prismatic coefficient, and the length to displacement ratio. Later, Andersen introduced corrections for hull form and bulbous bow, which extended the applicability of the method to modern commercial vessels as well. Due to its simplicity and generality, the method remains a benchmark for preliminary resistance estimations. [8]

Published in 1982 and complemented in 1984, the **Holtrop – Mennen method** is today the most widely used empirical approach for commercial ships. It decomposes the total resistance into its components: frictional resistance (according to ITTC-1957), form factor, appendage resistance, wave-making resistance, bulbous bow correction, and transom stern resistance. Due to its flexibility and accuracy within the displacement ship domain, this method is extensively used in commercial software and preliminary ship design. [15]

Developed in the **early 1970s**, the **van Oortmerssen method** is particularly intended for small displacement vessels, such as tugs, and fishing boats. It is based on regressions derived from systematic series of model tests and provides simple formulas for resistance prediction at low speeds. Although it is not recommended for large transport ships, it remains useful for robust vessels with high block coefficient values. [16]

The Digernes – Cheng approach was developed through regressions based on datasets obtained from model tests of commercial vessels. The method focuses on estimating the residual resistance and scaling it to full scale. It is often used in combination with other empirical methods to refine predictions and is also included in specialized software packages for comparative resistance analysis of the results. [8]

Proposed in the **1990s**, **the Fung method** is based on an extensive database of more than 10,000 experimental data points obtained from tests on 739 ship models at the *David Taylor Model Basin*. It is particularly recommended for ships with transom sterns, such as tankers or bulk carriers, and is implemented in software packages like *NavCad*, and *HullSpeed*. The main advantage of the method lies in its dedicated applicability to hull forms where transom resistance plays a major role. [17]

AutoPower is one of the specialized modules of the *AutoShip Systems software* suite, designed for the analysis of ship resistance and the power required for propulsion. Its main role is to provide designers with a fast and comparative estimation of the required power, based on the desired or modelled hull form in *AutoShip*, as wells as on the calculated geometric and hydrodynamic characteristics.

One of the advantages of the *AutoPower module* is the integration of several well-established empirical methods for calculating ship resistance, thus allowing a critical evaluation of the results through comparative prediction. Among the implemented methods are Andersen/Guldhammer–Harvald, Holtrop & Mennen, van Oortmerssen, Digernes/Cheng, and Fung. Each method has its own domain of applicability — ranging from large commercial vessels to small ships or hulls with transom stern hull — which enables the user to select or compare the most appropriate option according to the type of vessel under design.

For example, the Holtrop & Mennen method is frequently used in the preliminary design of transport ships due to its general applicability and detailed decomposition of resistance, often being considered a de facto standard in preliminary analysis. In contrast, the van Oortmerssen method is more suitable for small vessels, such as tugs or fishing boats, where the hull form coefficients differ significantly from those of large commercial ships. *AutoPower* therefore provides the possibility to test both scenarios and observe the variations between results.

The Fung method is particularly relevant for ships with a transom stern, such as tankers or bulk carriers, where energy losses in the stern flow region are significant. On the other hand, the Digernes/Cheng method allows for refined predictions for large commercial vessels, while the Andersen/Guldhammer–Harvald method remains useful for a quick assessment based on the main geometric parameters.

What differentiates this work from other studies in the specialized literature is that, through the generation of comparative graphs (resistance–speed, effective power–speed, propeller efficiency curves), the *AutoPower module* not only supports the selection of the propeller and engine but also highlights the differences between methods, emphasizing both their usefulness and their limitations. This multi-method approach makes it a valuable tool for the preliminary design stage, where time is critical, and the level of available detail is still limited.

2. Hull parameters for AutoPower

AutoPower has two Hull parameters grids (for displacement and catamaran hulls, and for planing and semi-displacement hulls). These play an important role in all *AutoPower* work sessions (figure 2.1.). All hull parameters must be entered, with "Figure 2.1." showing the program's main screen. When a hull parameter is selected, a brief description of it appears in an information box at the bottom of the main screen, also indicating whether the parameter can be entered manually or estimated automatically by the program, if applicable.

Table. 2.1. Hull characteristics for the Mono Hull and Catamaran Hull used for the oceanographic research vessel

Parameters (name, symbol & units)	Mono Hull	Catamaran Hull
Length on waterline $L_{WL}[m]$	33.5	
Breadth $B[m]$	8.20	11.23
Draft $T_{For}[m]$, $T_{Aft}[m]$	2.30	1.80
Displacement $\Delta[t]$	170	
Longitudinal centre of buoyancy measured percentage of LBP from midship $LCB[\%]$	-3	-7.45
Waterplane area coefficient $C_{WP}[-]$	0.850	0.760
Midship section coefficient $C_M[-]$	0.98	0.79
Form factor $(1 + k)$	1	
Wetted hull surface area, excluding appendages $S[m^2]$	251	290
Half Angle of Entrance / Of Run / Angle at ¼ Buttock	30°	
Hull form type forward / aft of midship (*N – normal shape, U – U shape, V – V shape*)	N/N, U/N, U/U, V/N, V/U	
Effective form factor $[1 + k_2]$ for appendages (*required for Holtrop method*)	1	-
Block coefficient - $C_B[-]$ (*calculus by AutoPower*)	0.262	0.245
Longitudinal prismatic coefficient $C_{LP}[-]$ (*calculus by AutoPower*)	0.268	0.310
Service speed $u[knots]$	18	

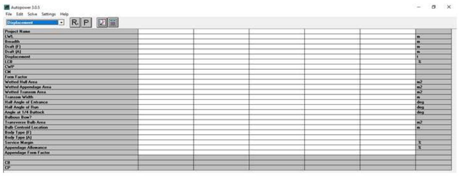

Figure 2.1. Main Screen of the AutoPower module of AutoShip software

Table 2.1 presents the parameters for the Mono Hull and Catamaran Hull of the shapes that are tested.

The equations used for the resistance calculation, depending on the method applied, are presented below.

The Andersen & Guldhammer method, presents limitations regarding the maximum value of the Froude number, around the value of 0.33. Equation (1) shows the working principle of this method, which is based on the preliminary design procedure used by the *Technical University of Denmark*. In Equation (1), C_T represents the total resistance coefficient, C_R the residuary resistance coefficient, C_F the frictional resistance coefficient, C_A the additional resistance coefficient, and D_C the coefficient of resistance due to air and rudder shape.

$$C_T = C_R + C_F + C_A + D_C \qquad (1)$$

The van Oortmerssen method is generally used for small vessels and employs the ITTC 1957 friction lines to determine the frictional resistance. The form of the resistance equation is presented in Equation (2), where the terms C_i are represented by tabulated coefficients of the method, and X_i are parameters depending on $f(C_P, Fn)$.

$$R_R = \sum_{i=1}^{4} C_i X_i \qquad (2)$$

The Digernes and Cheng method, expressed by Equation (3), analyses characteristics similar to those of 34 Norwegian ships and 20 Danish ships. In this equation, a, b, c, ∂ and β are constants, according to reference [18], [19], and V is the displacement of the ship.

$$R_T = a(LB)^b (BT)^c V^{\partial} exp(\beta Fn) \qquad (3)$$

The Jin, Su, and Tan method is generally used for ships with full forms and for high-speed craft. In Equation (4), the B_i values are represented by tabulated values mentioned in reference [20], and X_i terms are parameters depending on the Froude number and the geometry of the ship (prismatic coefficient, stern area ratio, entrance angle, etc.).

$$C_R = \sum_{i=1}^{54} B_i X_i \qquad (4)$$

The Fung method, represented by Equation (5), is a regression-based approach that analyses 426 types of ships, covering a range of Froude numbers from 0.18 to 0.51. The characteristic elements of the equation are C_R, the residuary resistance coefficient, which is the sum of several body parameters C_{Ri} corresponding to the Froude range covered by this method, as extracted from reference [21].

$$C_R = \sum_{i=1}^{10} C_{Ri} \qquad (5)$$

The Holtrop method is one of the best-known approaches and represents one of the fundamental methods for calculating ship resistance, which every naval officer, captain (according to STCW), or naval architect should be familiar with. This method is characterized by Equation (6), where R_W is the wave resistance, R_{Ap} is the appendage resistance, and R_A is the correlation allowance. The analysis is based on regression of data obtained from both full-scale trials and model tests, covering a total of 334 models at *NSMB* (*Netherlands Ship Model Basin*, known today as MARIN). [22], [23], [24].

$$R_T = R_F(1 + k) + R_W + R_{Ap} + R_A \qquad (6)$$

The FAO method is based on Japanese and European data, formulated through regression using approximately 570 model scale test results. The elements in Equation (7) are: A, B, C constants, and CR_{16} the residuary resistance coefficient for a standard 16 feet model.

$$C_R = CR_{16} - A\left(\frac{SL}{\Delta}\right)\left\{\left[log\left(\frac{BV}{L}\right)^{0.5}\right]^{-2} - \left[log\left(\frac{CV}{L}\right)^{0.5}\right]^{-2}\right\} \quad (7)$$

For predicting resistance and effective power in the case of catamarans, the *AutoPower program* uses the **Marintek Fastcat method**, based on tests conducted by MARIN (Maritime Research Institute Netherlands) on high-speed slender catamaran models, both at model scale and full scale. In this method, the frictional coefficient is determined using the ITTC 1957 line. In Equation (8), the parameters are: A is the constant, ΔC_F is the roughness allowance, R_{AA} is the air resistance allowance, S is the total wetted hull surface area, and u is the velocity. [25]

$$R_T = A(C_F + C_R + \Delta C_F)u^2 S + R_{AA} \quad (8)$$

3. Scenarios and results

Two structural configurations were analysed for an oceanographic research vessel, a monohull and a catamaran, each having the characteristics presented in Table 2.1. For both configurations, several hull form typologies (N - normal, U - rounded, and V - sharp) were selected.

Table 3.1. Limitations obtained for the analysed cases for the Mono Hull vessel type (R&P – resistance and propulsion)

R&P method	Speed $u[knots]$		Frounde number - Fn	
	Min.	Max.	Min.	Max.
M. Andersen	0.00	11.50	0.00	0.33
M. van Oortmers	0.00	17.50	0.00	0.50
M. Digerne	0.00	17.50	0.00	0.50
M. Jin	14.50	22.00	0.41	0.62
M. Fung	6.50	17.50	0.18	0.50
M. Hoptrop	0.00	22.00	0.00	0.62
M. FAO	0.00	12.50	0.00	0.35

From the initial results, limitations were observed regarding the Froude number, and consequently the speed range, for the different methods analysed in the case on Modo Hull type construction. These limitations are presented in Table 3.1.

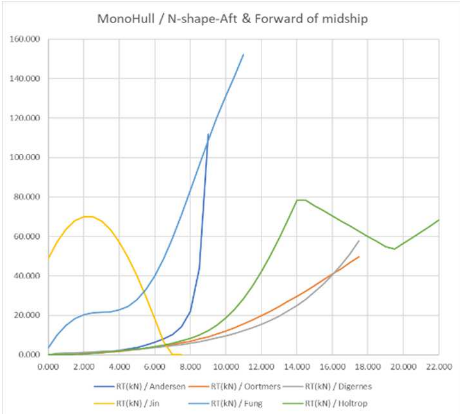

Figure 3.1.a. Total resistance $R_T[kN]$ for MonoHull N-shape aft & fore of midship section – for all methods

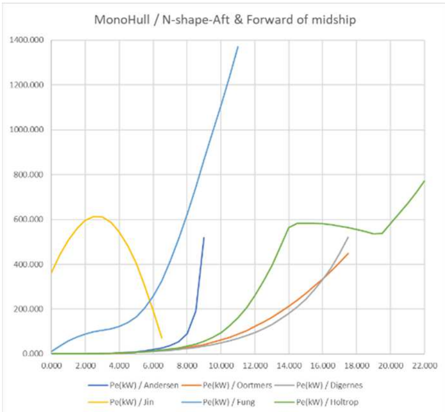

Figure 3.1.b. Effective power $P_E[kW]$ for MonoHull N-shape aft & fore of midship section - – for all methods

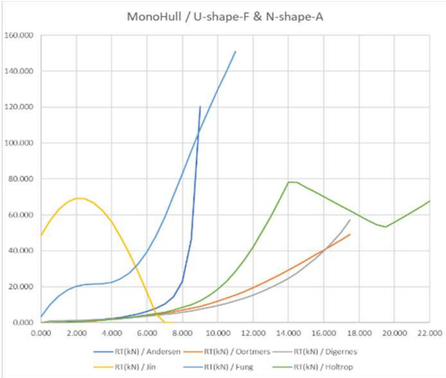

Figure 3.2.a. Total resistance $R_T[kN]$ for MonoHull U-shape fore & N-shape aft of midship section – for all methods

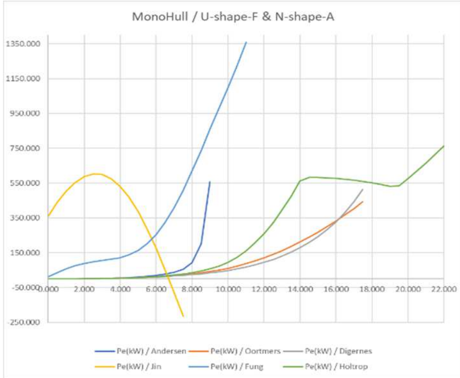

Figure 3.2.b. Effective power $P_E[kW]$ for MonoHull U-shape fore & N-shape aft of midship section – for all methods

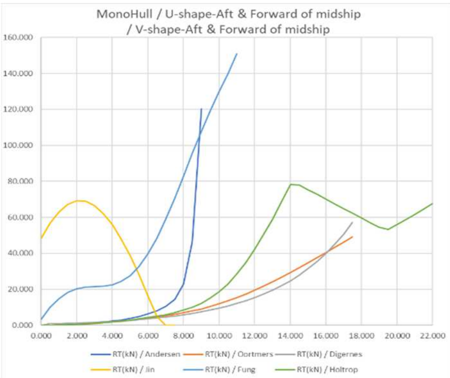

Figure 3.3.a. Total resistance $R_T[kN]$ for MonoHull V/U-shape aft & fore of midship section - – for all methods

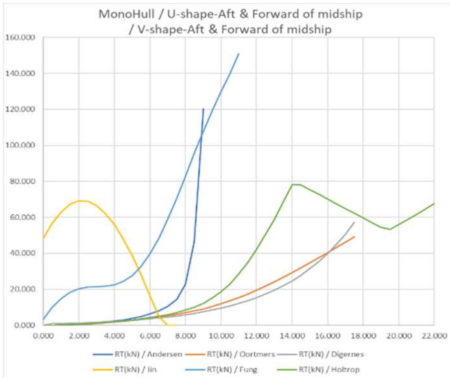

Figure 3.3.b. Effective power $P_E[kW]$ for MonoHull V/U-shape aft & fore of midship section - – for all methods

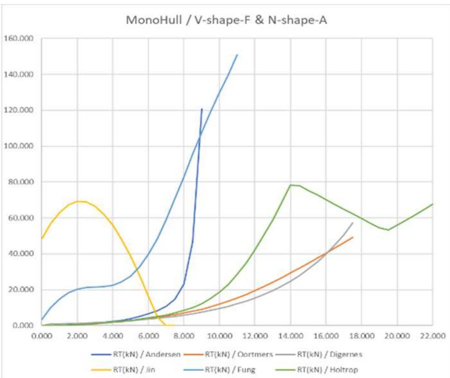

Figure 3.4.a. Total resistance $R_T[kN]$ for MonoHull V-shape fore & N-shape aft of midship section – for all methods

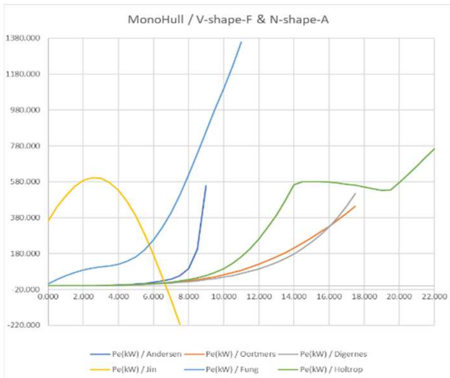

Figure 3.4.b. Effective power $P_E[kW]$ for MonoHull V-shape fore & N-shape aft of midship section – for all methods

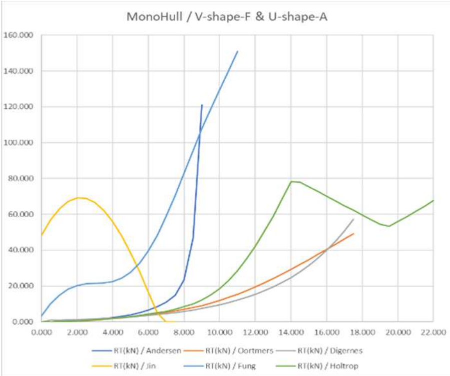

Figure 3.5.a. Total resistance $R_T[kN]$ for MonoHull V-shape fore & U-shape aft of midship section – for all methods

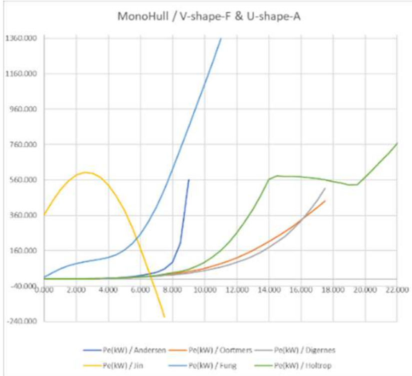

Figure 3.5.b. Effective power $P_E[kW]$ for MonoHull V-shape fore & U-shape aft of midship section – for all methods

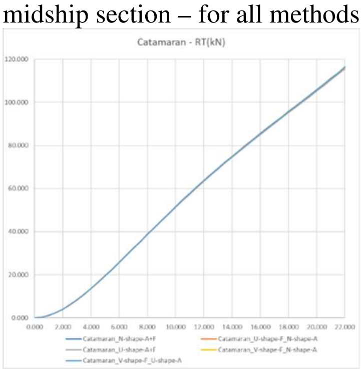

Figure 3.6.a. Total resistance $R_T[kN]$ for Catamaran Hull all methods and all shapes

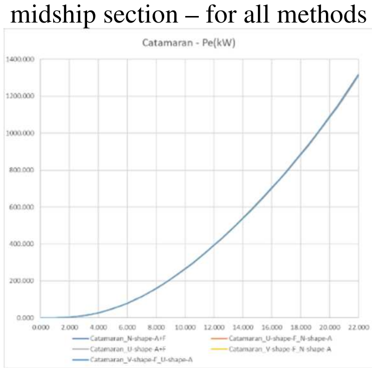

Figure 3.6.b. Effective power $P_E[kW]$ for Catamaran Hull all methods and all shapes

Figures 3.1 – 5.a., b., show the diagrams obtained for each hull form type, illustrating the results for total resistance and effective power according to the methods presented in subchapter 2. From the tabulated data, identical values were obtained for several Mono Hull configurations, specifically for U-shape-F_N-shape_A, U-shape-F_U-shape_A, V-shape-F_N-shape_A, and V-shape-F_U-shape_A constructions. Additionally, Table 3.2. presents the most significant results obtained for the various constructive scenarios of the analysed vessels, highlighting the total resistance and the effective propulsion power. The comparison was made for a service speed of 18 knots.

Figure 3.6.a., b. presents the results obtained for the total resistance and effective power corresponding to the Catamaran Hull configurations. From the graphs, it can be observed that for this constructive type, the results for each hull form model show only minor variations in the obtained values.

The calculations performed using the AutoPower module for a service speed of 18 knots highlight significant differences between the Mono Hull and Catamaran configurations. The analysis focused on determining the effective power and total resistance for several combinations of hull form types, selecting for future studies the most efficient models in each category - namely the Mono Hull V-shape fore & N-shape aft of midship section and the

Catamaran with U-shape aft & fore of midship section.

From Table 3.3 and Figures 3.1–6. a., b., specifically the graph shown in Figure 3.7., it can be observed that for the Mono Hull models, the average effective power values are approximately 564 kW, while the total resistance is around 61 kN, with variations below 2% among the different hull form combinations (normal, V shape, or U shape).

In the case of the Catamaran Hull models, the average effective power reached about 881 kW, and the total resistance approximately 95 kN, representing an increase of about 55% in power and 35% in resistance compared to the Mono Hull selected model.

Table 3.2. Values of Total Resistance$R_T[kN]$ and Effective power $P_E[kW]$ for the verified models (1. MonoHull N-shape aft & fore of midship section, 2. MonoHull U-shape fore & N-shape aft of midship section, 3. MonoHull V/U-shape aft & fore of midship section, 4. MonoHull V-shape fore & N-shape aft of midship section, 5. MonoHull V-shape fore & U-shape aft of midship section, 6. Catamaran U-shape fore & N-shape aft of midship section, 7. Catamaran V-shape fore & N-shape aft of midship section, 8. Catamaran V-shape fore & u-shape aft of midship section, 9. Catamaran N-shape aft & fore of midship section, 10. Catamaran U-shape aft & fore of midship section) for all methods

<table>
<tr><th rowspan="2">Ship model</th><th colspan="2">$R_T[kN]$</th><th colspan="2">$P_E[kW]$</th></tr>
<tr><th>Min.</th><th>Max.</th><th>Min.</th><th>Max.</th></tr>
<tr><td>1.</td><td>60.5</td><td>63.1</td><td>560</td><td>583</td></tr>
<tr><td>2.</td><td>59.6</td><td>62.2</td><td>553</td><td>576</td></tr>
<tr><td>3.</td><td>59.6</td><td>62.2</td><td>553</td><td>576</td></tr>
<tr><td>4.</td><td>59.6</td><td>62.2</td><td>552</td><td>576</td></tr>
<tr><td>5.</td><td>59.6</td><td>62.2</td><td>553</td><td>576</td></tr>
<tr><td>6.</td><td colspan="2" rowspan="4">95.7</td><td colspan="2" rowspan="4">886</td></tr>
<tr><td>7.</td></tr>
<tr><td>8.</td></tr>
<tr><td>9.</td></tr>
<tr><td>10.</td><td colspan="2">95.2</td><td colspan="2">881</td></tr>
</table>

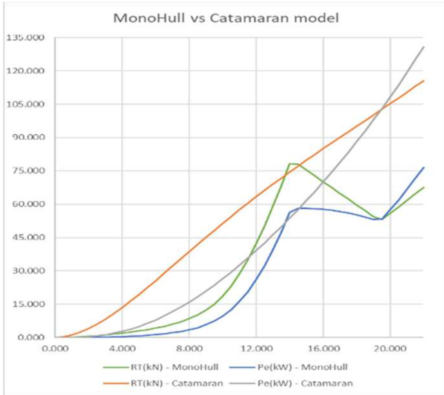

Figure. 3.7. Comparative Total resistance $R_T[kN]$ and Effective power $P_E[kW]$ for the chosen models
Mono Hull and Catamaran

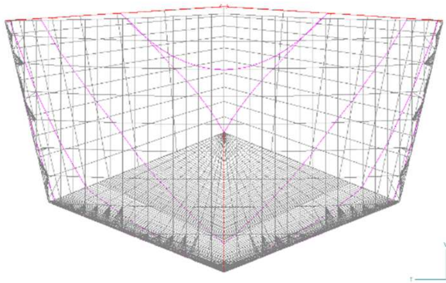

Fig. 3.8.a. Mono Hull Front view

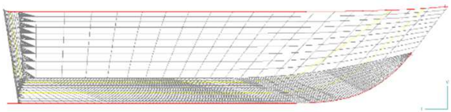

Fig. 3.8.b. Mono Hull Longitudinal view

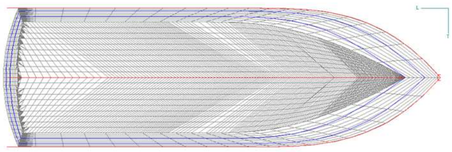

Fig. 3.8.c. Mono Hull Horizontal view

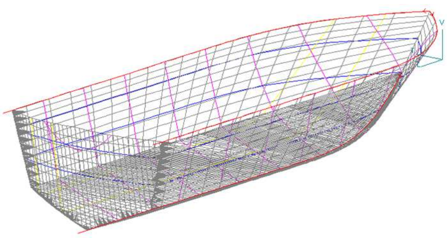

Fig. 3.8.d. Mono Hull 3D view

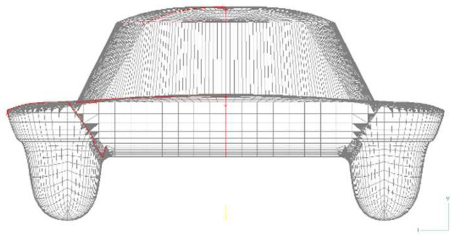

Fig. 3.9.a. Catamaran Front view

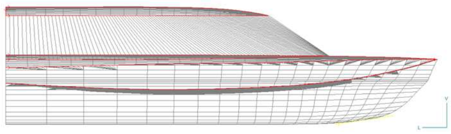

Fig. 3.9.b. Catamaran Longitudinal view

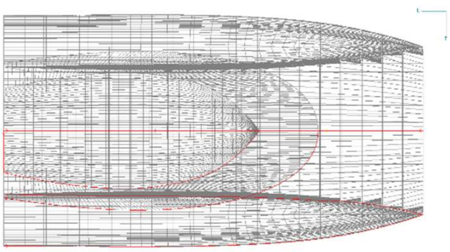

Fig. 3.9.c. Catamaran Horizontal view

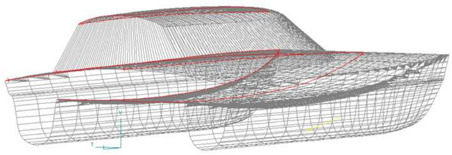

Fig. 3.9.d. Catamaran 3D view

In particular, the selected Mono Hull model (V-F / N-A) ranked equally with the best Mono Hull alternatives, demonstrating superior hydrodynamic efficiency for the considered speed regime. The chosen Catamaran Hull model (U – F+A) proved to be the most efficient among the catamarans studied, achieving the lowest values of $P_E[kW]$ and $R_T[kN]$ within this category; however, it remains significantly more energy-demanding compared to the Mono Hull.

These results confirm that, for service speeds around 18 knots, the Mono Hull configuration remains more energy-efficient, requiring a lower installed power and resulting in reduced fuel consumption and emissions. However, the selected Catamaran remains a viable solution where criteria such as high transverse stability, increased usable deck area, or safety redundancy are prioritized – even at the cost of higher energy consumption.

Figures 3.8.a–d and 3.9.a–d present the models created in *AutoShip* for the two selected structural configurations: the Mono Hull with a V-shaped bow and normal stern, and the Catamaran with a U-shaped bow and stern. These models were generated using the hull form modelling modules in *AutoShip*.

The 3D models were generated in *AutoShip* based on the hull form parameters validated by *AutoPower*, ensuring a complete geometric description (waterlines, transverse sections, and wetted hull surface) necessary for the subsequent stages of the design process.

These models will be used in further analyses within the preliminary design stage, including:

- studies on the ship's basic seakeeping qualities (buoyancy, stability, motions, resistance to forward movement);
- assessments of intact and damaged stability;
- evaluation of the operational capability of the constructive models under different loads.

This sequence – preliminary analysis in *AutoPower* → 3D modelling in *AutoShip* → hydrodynamic and stability calculations – ensures continuity and accuracy throughout the entire design process, reducing the risk of errors and allowing for the rapid optimization of the constructive solutions.

This integrated workflow validates the selection of hull forms and provides a solid foundation for the subsequent design stages, providing robustness and coherence throughout the entire project.

4. Concluding remarks

The *AutoPower* module of the *AutoShip* suite provides a versatile and accessible framework for predicting ship resistance and propulsion requirements across a wide range of hull types, including displacement, semi-displacement, planning, and catamaran hulls. Its main advantage lies in the integration of multiple empirical methods (e.g., for displacement hulls - Andersen/Guldhammer, Holtrop, van Oortmerssen, Digernes/Cheng, Fung, Jing, Su and Tan, FAO; for semi-displacement hulls – Compton; planning hulls - Savitsky, Radojcic; and for catamarans - Marintek FastCat), each with its own applicability limits depending on the Froude number. This allows designers to perform comparative evaluations and gain a better understanding of the sensitivity of results to different regression-based formulations.

The program's user-friendly interface and structured hull parameter grids make it suitable for the conceptual and preliminary design stages, where rapid estimation and iteration are critical. The ability to link directly with *AutoShip*'s hull modelling environment further reduces data entry errors and accelerates the workflow. Output in the form of tables, graphs, and Word / Excel - compatible reports supports both technical analysis and communication with stakeholders.

Nevertheless, the software also exhibits inherent limitations. *AutoPower* calculations are restricted to the assumptions embedded in the empirical formulations. For example, catamaran propulsion is not directly modelled: the program computes only the effective power of a demi-hull, which must be doubled to approximate total requirements. Similarly, certain algorithms are valid only within narrow ranges of Froude numbers, coefficients, or hull proportions, and their extrapolation may lead to significant inaccuracies. The program does not natively account for added resistance in waves or complex flow phenomena, which necessitates validation by CFD simulations or model testing in advanced design stages.

In conclusion, *AutoPower* should be regarded as a decision-support tool rather than a definitive predictor of propulsion requirements. It enables designers to quickly explore alternative hull forms, assess trends across different methods, and establish the initial propulsion requirements. However, for reliable performance predictions – especially in the case of novel hull forms or operational conditions outside empirical datasets - its results must be supplemented with higher-fidelity approaches.

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