PRELIMINARY CALCULATION METHOD FOR THE RESISTANCES OF A SHIP

Dragos Filimon
The University Politehnica of Bucharest
dragos_f@yahoo.com

Abstract :

The initial phases of constructing a ship are anchored in detailed modeling and the expertise of project design engineers. In the contemporary maritime landscape, there's a pressing need to pivot towards sustainability in shipping. This shift isn't just a matter of retrofitting existing vessels; it must be deeply ingrained right from the inception of the ship design.
This article introduces a numerical method dedicated to analyzing the forces exerted on the hull of a salvage tug. This methodology provides an intricate framework detailing the hydrodynamic forces acting on the ship as it moves in the direction of its journey. As this approach is rooted in statistical analysis, it holds immense promise not just for salvage tugs but can be seamlessly integrated into the preliminary design stages of various vessel categories. By implementing such methods early on, the maritime industry can ensure that ships are not only structurally sound but also aligned with the ever-evolving standards of sustainability.

Keywords*: ship design, hydrodynamic model,, Fuel Saving, ship preliminary calculations,*

1. INTRODUCTION

Ship construction has traditionally been rooted in engineering precision. The detailed modeling, coupled with the expertise of project design engineers, ensures ships can navigate the vast oceans safely. However, the maritime world is at an inflection point, where the cry for sustainability cannot be ignored. Traditionally, the primary focus during ship design was on factors such as speed, load-bearing capacity, and operational efficiency. Environmental concerns, if any, were often secondary.

Today's maritime landscape is vastly different. There's a resounding call for sustainable practices in every industry, and maritime transport is no exception. This beckoning isn't just about retrofitting older ships with new technologies; it's about reimagining the entire shipbuilding paradigm.

1.1 Main vessel data

The vessel is a salvage tug operating in the Black Sea and performs salvage, towing, water and bunkering and firefighting operations.

Maximum length.. L_{max} =39.7 m
Length between perpendiculars..............................L_{PP} =37.8 m
Length at waterline...L_{WL} =39.01 m
Vessel width..B=11 m
Construction height 2100D=4,92 m
Average draught...T=3,8 m
Displacement .... Δ=1026 t
Area of wetted surface of bare hull.............................S=537 m^2
Speed...v=14 Nd
Block coefficient..C_B =0.614
Area coefficient of the master section..............................C_M =0.822
Coefficient of the area of the floating surface.............................C_W =0.889
Range..A=2200 Mm
Number e of crew members........................ 15 persons
Pushing at fixed point ....20 t.

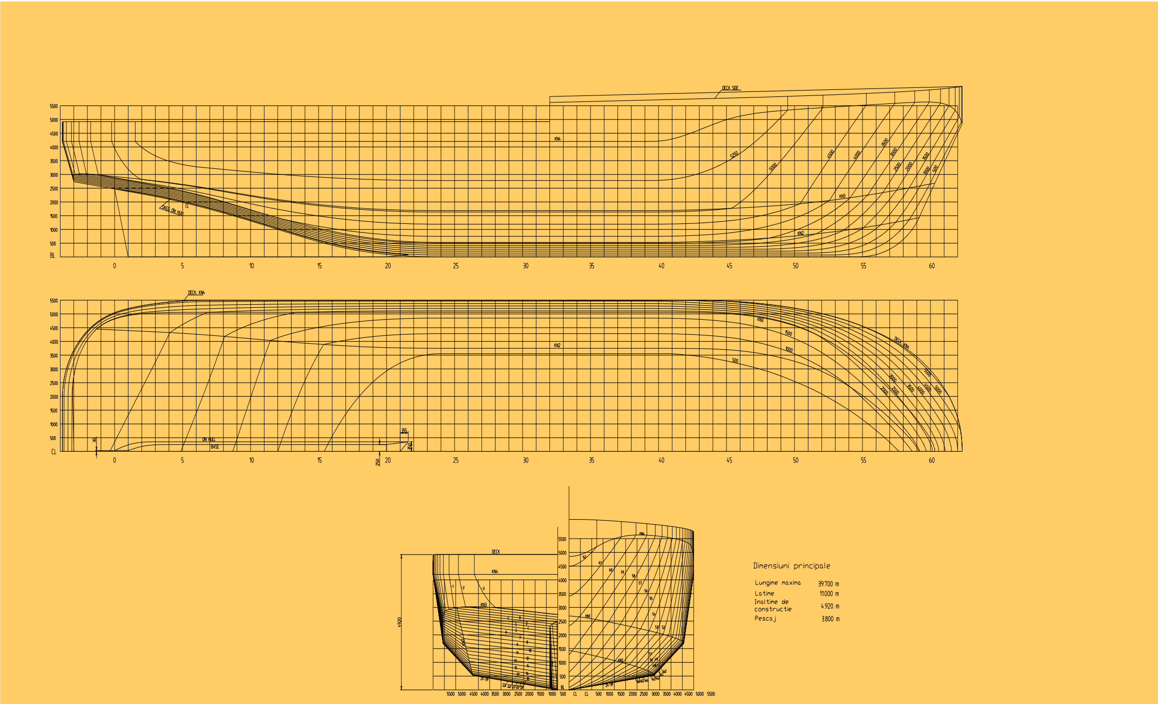

Fig. 1.1 Shape plan of the vessel [Source: Author]

2. PRELIMINARY PERFORMANCE ANALYSIS OF FORWARD STRENGTH

2.1 Components of resistance in calm water

On a ship moving at constant speed, v, acts the resultant of hydro-aerodynamic forces, R_T . The drag of the ship is the projection of the resultant of the hydro-aerodynamic forces acting on the ship in the direction of the speed of travel [1].Larsson and Baba provided a general scheme of the decomposition of the hydrodynamic drag components (Fig. 2.1) for the bare hull (without appendages) ([2], [3]).

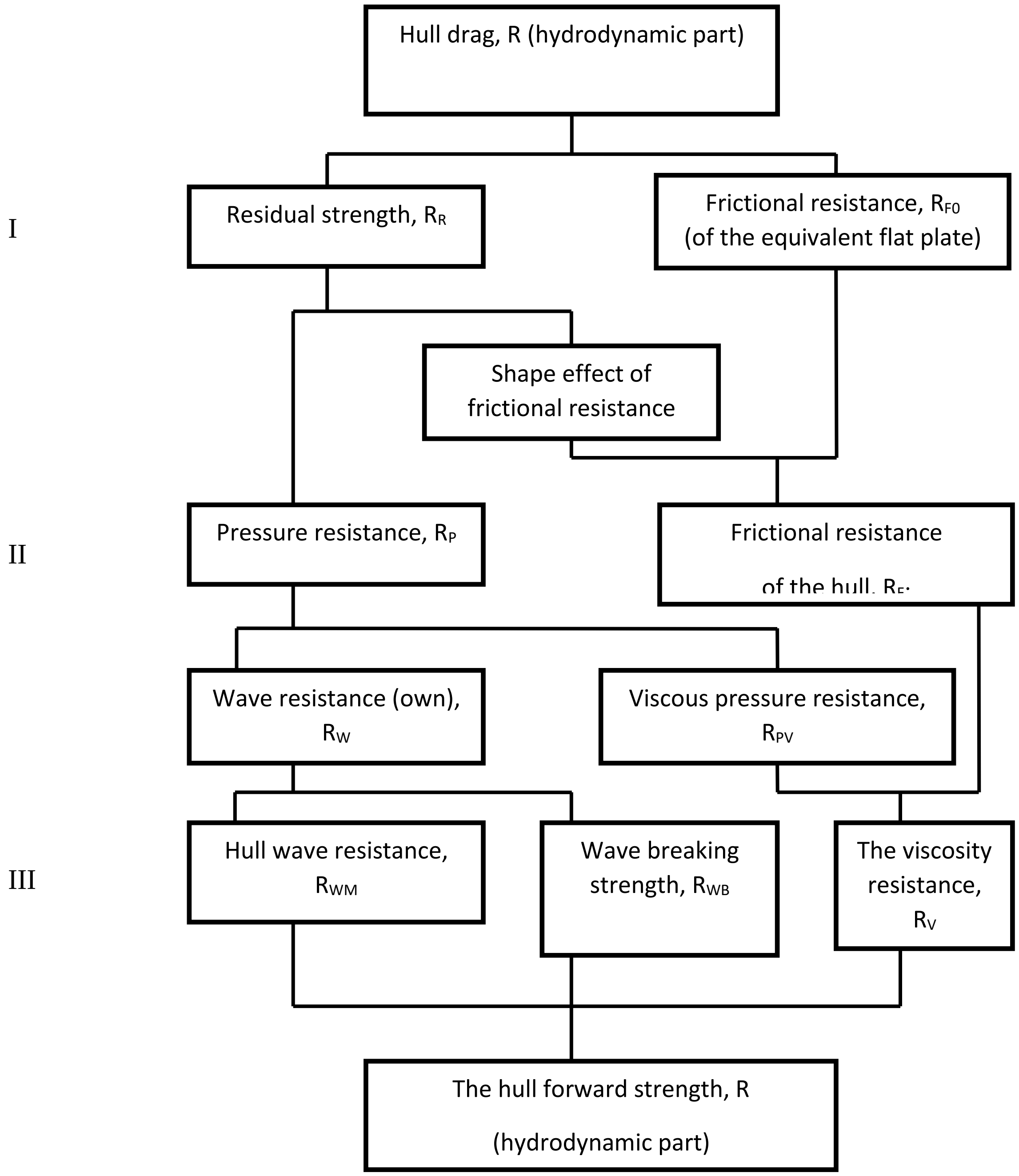

Fig. 2.1 Components of drag in calm water for bareboat hull [Source: Author]

On the first level (I) the Froude hypothesis is treated, considering that the (hydrodynamic) drag resistance is the sum of the equivalent plane plate frictional resistance, R_{F0} and the residual resistance, R_r [3]:

$$R = R_{F0} + R_R$$

In the residual strength all the components that depend on the Froude number, Fn, are included (including the shape component of the frictional resistance which actually depends on the Reynolds number, Re).

Within level II of the scheme presented, the hydrodynamic drag, is composed of the pressure drag and the hull friction drag, which also incorporates the shape effect [3].

Frictional resistance arises due to the influence of the viscosity of the fluid which conditions the adhesion of its particles to the hull surface and the occurrence of tangential frictional stresses.

The pressure resistance is due to the change in the pressure field distribution along the hull due to the existence of the boundary layer and its detachment phenomenon, accompanied by the appearance of strong vortex systems. The sum of the hull frictional resistance and the viscous part of the pressure resistance, R_{PV} , forms the viscous resistance, R_V , and what remains of the pressure resistance is called the (proper) wave resistance, R_W .

Accordingly, in the Hughes hypothesis presented on Level III, the (hydrodynamic) forward resistance is the sum of the (proper) wave resistance, R_W and the viscosity resistance, R_V [3]:

$$R = R_W + R_V = R_W + R_{PV} + R_F$$

The viscous pressure resistance depends on the shape of the vessel. Vessels with a full shape will have a higher viscous pressure resistance than vessels with thin, elongated shapes.

The (self) wave resistance is a very important component of the pressure resistance and represents the energy consumed by the vessel to generate and maintain the self wave system when moving in the fluid medium.

In the fore end of full-formed vessels, it is possible that the phenomenon of self-wave breaking occurs, leading to the identification of turbulent flow areas in the hydrodynamic wake near the hull.

The wave resistance generated at hull displacement (R_{WM}) and the wave breaking resistance (R_{WB}) form the wave resistance (R_W).

In the hydrodynamic analysis of the components of the forward resistance, carried out above, the case of smooth hull (no roughness) without appendages and superstructures was considered.

In the following, three additional components of the swimming resistance î will be considered: appendage resistance R_{APP} , aerodynamic resistance R_{AA} and additional resistance on incident waves R_{AW} . The total drag resistance R_T is determined with the relation

$$R_T = R + R_{APP} + R_{AA} + R_{AW} \quad .$$

Appendage strength is due to rudders, rigging, active wings, roll keels, hull openings, etc. Appendage strength can lead to a significant increase in forward strength depending on the relative development ă of the appendages.

The aerodynamic drag of the above-ground part of a ship travelling at regime speed, in calm atmospheric conditions (zero wind speed), may be between 2% and 4% of the

hydrodynamic drag. Consideration of wind speed can lead to a significant increase in the aerodynamic component of drag.
Additional resistance on incident waves is a very important additional component, which depends on the sea state (height of incident waves).

2.2. Preliminary determination of the forward strength with the Holtrop - Mennen

The Holtrop - Mennen method is based on regression analysis of the results of systematic experimental tests on model series as well as nature measurement data from the Dutch Wageningen Basin ([1], [3], [4], [5]).

Being a statistical method, it can be used in the preliminary design phase of the following types of displacement vessels:

- oil tankers, bulk carriers (Fn≤ 0.24; 0.73≤ Cp≤ 0.85; 5.1≤ L /B_{WL}≤ 7.1; 2.4≤ B/T≤ 3.2);
- container carriers, destroyers (Fn≤ 0.45; 0.55 Cp≤≤ 0.67; 6.0 L /B≤$_{WL}$≤ 9.5; 3.0 B/T≤≤ 4.0);
- trawlers, coastal, tugs (Fn≤ 0.38; 0.55 Cp≤≤ 0.65; 3.9 L /B≤$_{WL}$≤ 6.3; 2.1≤ B/T≤ 3.0).

According to this method, the total forward resistance of the ship is determined with the relation [3]:

$$R_t = R_F(1+k_1) + R_{APP} + R_W + R_B + R_{TR} + R_A$$

where R_F is the frictional resistance calculated according to the ITTC-1957 formula, ($1+k_1$) is the shape factor of the hull without appendages, R_{APP} is the appendage resistance, R_W is the self wave resistance, R_B is the additional pressure resistance of the bow bulb near the free surface, R_{TR} is the additional pressure resistance of the immersed mirror stern and R_A is the correlation resistance between model and ship.

The frictional resistance, according to the ITTC-1957 formula, can be written as [3]:

$$R_F = C_F \cdot \frac{1}{2} \cdot \rho v^2 \cdot S$$

where C_F is the coefficient of frictional resistance,ρ is the water density, v is the vessel velocity, and S is the wetted surface area of the hull without appendages. The coefficient of frictional resistance of the equivalent flat plate is calculated according to the ITTC - 1957 formula [3]:

$$C_F = \frac{0{,}075}{(\log Re - 2)^2}$$

where Re is the Reynolds number, which is determined from the length at float, L_{WL} , with the expression :

$$Re = v \cdot L_{WL} / \nu$$

whereν is the kinematic viscosity of the water and v is the velocity of the ship. Its values are :

$$\nu = \begin{cases} 1{,}191 \cdot 10^{-6}\ \mathrm{m}^2/\mathrm{s} \text{ for seawater;} \\ 1{,}141 \cdot 10^{-6}\ \mathrm{m}^2/\mathrm{s} \quad \text{for fresh water.} \end{cases}$$

If the density ρ is expressed in [t/m^3], the velocity v in [m/s] and the area S in [m^2], then the drag is given in [kN].

The shape factor of the body of the appendages is calculated with relation [3]:

$$1 + k_1 = 0{,}93 + 0{,}487118 \cdot c_{14} \cdot (B/L_{WL})^{1{,}06806} \cdot (T/L_{WL})^{0{,}46106} \cdot \\ \cdot (L_{WL}/L_R)^{0{,}121563} \cdot (L_{WL}^3/\nabla)^{0{,}36486} \cdot (1 - C_p)^{-0{,}604247} \quad ($$

where the longitudinal prismatic coefficient C_P is calculated from the length at float, L_{WL} , B is the width of the vessel, T is the mean draught, Δ is the displacement of the vessel, and L_R is the distance from the perpendicular aft to the area from which the cylindrical part of the vessel begins.

In preliminary phase, the wetted surface area of the hull without appendages can be estimated based on the relation [3]:

$$S = L_{WL} \cdot (2 \cdot T + B) \cdot \sqrt{C_M} \cdot (0{,}453 + 0{,}4425 \cdot C_B - 0{,}2862 \cdot C_M - \\ - 0{,}003467 \cdot B/T + 0{,}3696 \cdot C_W) + 2{,}38 \cdot A_{BT}/C_B$$

where A_{BT} is the cross-sectional area of the bulb, C_B is the block coefficient, C_M is the master section area coefficient, and C_W is the floating surface area coefficient.

The additional pressure resistance due to the presence of the bulb near the free surface of the water is calculated with relation [3]:

$$R_B = 0{,}11 \cdot e^{(-3 \cdot p_B^{-2})} \cdot Fn_i^3 \cdot A_{BT}^{1{,}5} \cdot \rho \cdot g / (1 + Fn_i^2)$$

where, the coefficient p_B takes into account the emersion of the evidence, and Fn_i is the Froude number based on the immersion. The two quantities are determined with the relations :

$$p_B = 0{,}56 \cdot A_{BT}^{1/2} / (T_F - 1{,}5 \cdot h_B)\ ;$$
$$Fn_i = v / [g \cdot (T_F - h_B - 0{,}25 \cdot A_{BT}^{1/2}) + 0{,}15 \cdot v^2]^{1/2}\ .$$

The coefficient C_V is the coefficient of viscous resistance:

$$C_V = C_F(1 + k) + C_A\ .$$

The hull form factor of the vessel with appendages is determined with the relation :

$$1 + k = (1 + k_1) + [(1 + k_2)_{eq} - (1 + k_1)] \cdot S_{APP} / (S + S_{APP})\ .$$

2.3. Analysis of the resistance of the rescue tug to the calm water, using the Holtrop - Mennen

For the actual calculation of the forward resistance in calm water, the POWERING program of the INITIAL DESIGN module of the AVEVA MARINE CAD-CAE system was used for the case of the salvage tug under study [6]. The ship's shape plan and the coupling plotting table were used to describe the ship's shape.

Figure 2.2 shows the description of the cross-sectional shapes of the ship, created using the BRITFAIR program in the INITIAL DESIGN module.

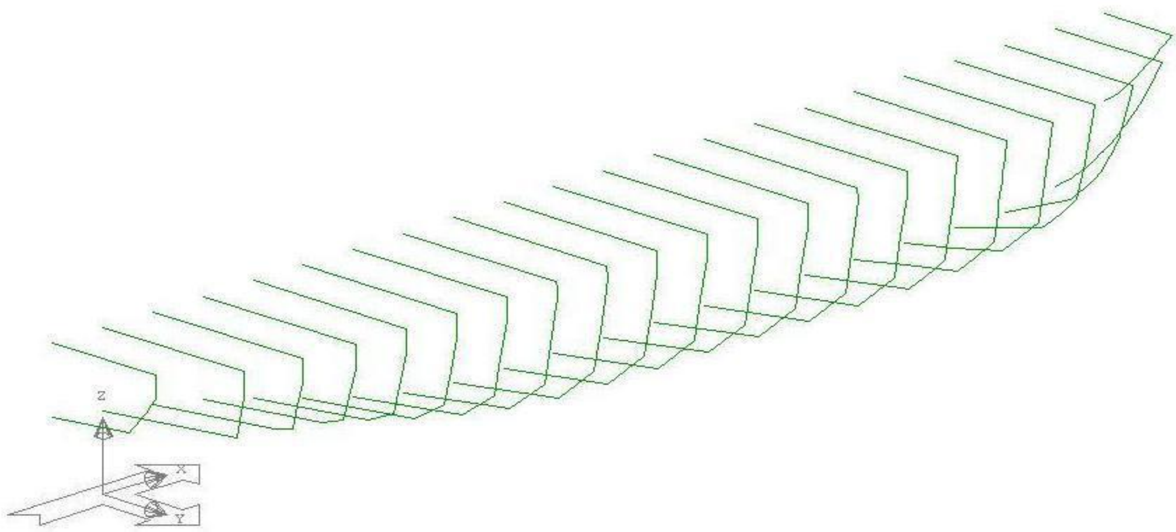

Fig. 2.2 Cross-section shapes of the salvage tug [Source: Author]

Based on the Holtrop-Mennen method, the ship's drag was calculated for a speed range of 10-16 Nd. Table 2.1 shows both the coefficients of the drag components and the total drag, R_T . Note that the ship has no bow bulb and no submerged stern mirror, and therefore the components R_B and R_{TR} are zero. Also, the strength of the appendages R_{APP} has been neglected.

Speed [Nd]	Fn	Rn /10^9	C_F *10^3	C_F x k *10^3	C_W *10^3	C_A *10^3	C_T *10^3	R_T [kN]
10.000	0.263	0.169	1.934	1.010	3.248	0.674	6.910	50.3
11.000	0.289	0.186	1.908	0.997	5.582	0.674	9.205	81.0
12.000	0.316	0.203	1.886	0.985	11.013	0.674	14.601	152.9
13.000	0.342	0.220	1.865	0.974	15.778	0.674	19.334	237.6
14.000	0.368	0.236	1.846	0.964	16.936	0.674	20.463	291.6

Speed [Nd]	Fn	Rn /10^9	C_F *10^3	C_F x k *10^3	C_W *10^3	C_A *10^3	C_T *10^3	R_T [kN]
15.000	0.394	0.253	1.829	0.955	19.241	0.674	22.741	372.1
16.000	0.421	0.270	1.813	0.947	23.683	0.674	27.159	505.6

Table 2.1 Coefficients of drag components and total drag [Source: Author]

Figure 2.3 shows a plot of the total drag, R_T , as a function of ship speed. A local maximum is observed at 13 Nd, followed by a decrease in the drag gradient at the steady state speed of 14 Nd.

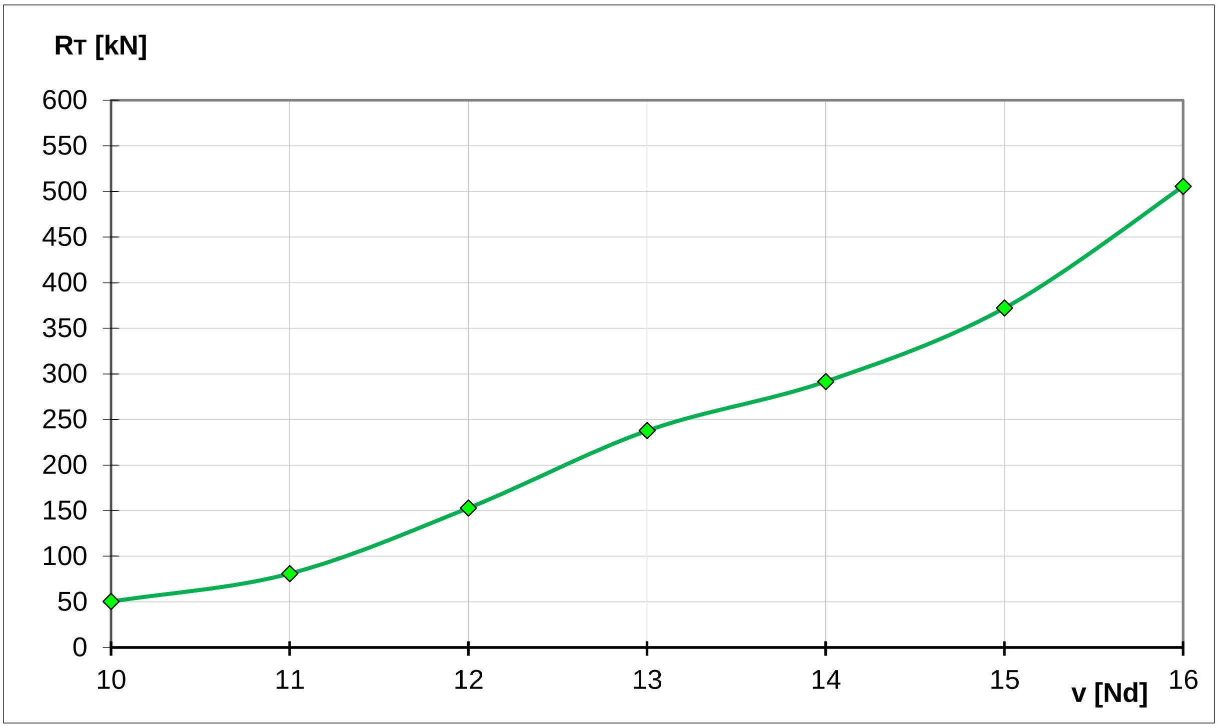

Fig. 2.3 Diagram of total forward resistance, R_T , as a function of vessel speed, v [Source: Author]

3. PRELIMINARY PERFORMANCE ANALYSIS PROPULSION

3.1. Determination of propulsion power required on board the vessel

The effective towing power is calculated with the relation [7]

$$P_E = R_T \cdot v \cdot (1 + M_D)$$

where, M_D is the design reserve, which takes into account the accuracy of the ship's forward resistance estimate R_T , and v is the ship's speed. Common values of the design reserve are [8]:

- M_D =0.01 ... 0.02, for forecasts based on self-propulsion tests with final propeller in fairing basins;
- M_D =0.03 ... 0.06, for forecasts based on self-propulsion tests with stock propellers in hull basins;
- M_D =0.07 ... 0.08, for forecasts based on forward strength tests in hull basins;
- M_D =0.10 , for preliminary theoretical forecasts (with regression methods or according to specific kine series diagrams).

The available propeller power is calculated with the expression [7]

$$P_D = \frac{P_E}{\eta_D \cdot n_p}$$

where, η_D is the quasi-propulsive coefficient, and n_p is the number of thrusters.

Engine flange power at 100% MCR is calculated with the relation [7]

$$P_B = \frac{P_D}{\eta_{ax} \cdot \eta_{red} \cdot (1 - M_S)}$$

where, η_{ax} is the axle line efficiency, η_{red} is the gearbox efficiency, and M_S is the ship's service reserve. The usual values of the yields are:

$$\eta_{ax} = 0.97...0.98$$

$$\eta_{red} = 0.97...0.98$$

The ship's service reserve M_S takes into account the excess power required during ship operation to overcome the additional resistance due ([1], [3]):

- marine deposits on the hull (algae, crustaceans, etc.);
- additional wind, wave and sea current action;
- limited bottom effects.

The ship's service reserve may have usual values in the range [7]

$$M_S = 0.15...0.25$$

The engine flange power for the specific service speed SR is calculated with the relation [7]

$$P_B^{SR} = \frac{P_B}{SR}$$

where, P_B is the power at the engine flange at 100% MCR. The SR service speed has usual values in the range [7]

$$SR = 0.85...0.95,$$

values corresponding to engine load between (85%-95%) MCR.

3.2 Rescue tug propulsion power analysis

Table 3.1 shows the results of the calculation of the ship's propulsion coefficients ţ using the expressions of the Holtrop-Mennen method (wake coefficient w, suction coefficient t, relative rotational efficiency and η_R), open water propeller efficiency η_0 , quasi-propulsive coefficientη_D and propeller speed RPM, obtained with the POWERING program in the INITIAL DESIGN module of the AVEVA MARINE CAD-CAE system [6]. The propeller design parameters are:

Propeller diameter.. D_e =2.5 m
Step ratio..P/D_e =0.994
Disc ratio..BAR=0.9
Number of blades ....Z=4 blades.

Speed [Nd]	t	w	η_R	η_0	η_D	RPM [rpm]
10.000	0.127	0.123	1.003	0.608	0.608	144.10
11.000	0.127	0.123	1.003	0.591	0.591	169.19
12.000	0.127	0.123	1.003	0.548	0.548	208.08
13.000	0.127	0.123	1.003	0.515	0.514	245.09
14.000	0.127	0.122	1.003	0.508	0.507	268.57
15.000	0.127	0.122	1.003	0.495	0.494	297.37
16.000	0.127	0.122	1.003	0.471	0.470	336.68

Table 3.1 Calculation of propulsion coefficients [Source: Author]

In Table 3.2 are presented the results of the calculation of the propulsion power required on board the ship, obtained by using the PROPELLING POWER program of the CAD-CAE PHP (Preliminary Hydrodynamics Performances) system [7], for the following values of the coefficients of the relations presented in the previous paragraph: M_D =0.10; M_S =0.15; n_p =2 thrusters; $= \eta_{ax}\ \eta_{red}$ =0.97; SR=100%.

Speed [Nd]	P_E [kW]	P_D [kW]	P_B [kW]
10.000	284.4	233.9	292.4
11.000	503.8	426.2	532.8
12.000	1037.4	946.5	1183.1
13.000	1746.4	1698.8	2123.5
14.000	2098.4	2069.4	2586.8
15.000	3155.8	3194.1	3992.6
16.000	4573.9	4865.9	6082.4

Table 3.2 Calculation of the propulsion power chain [Source: Author]

In fig. 3.1 shows the engine flange power diagram at 100% MCR, P_B , as a function of ship speed.

A local maximum is observed at 13 Nd, followed by a decrease in the power gradient at the engine flange at the engine speed of 14 Nd.

The flange power of a single engine at 14 Nd engine speed at 100% MCR is 2586.8 kW and the total power e of the two engines is 5173.6 kW.

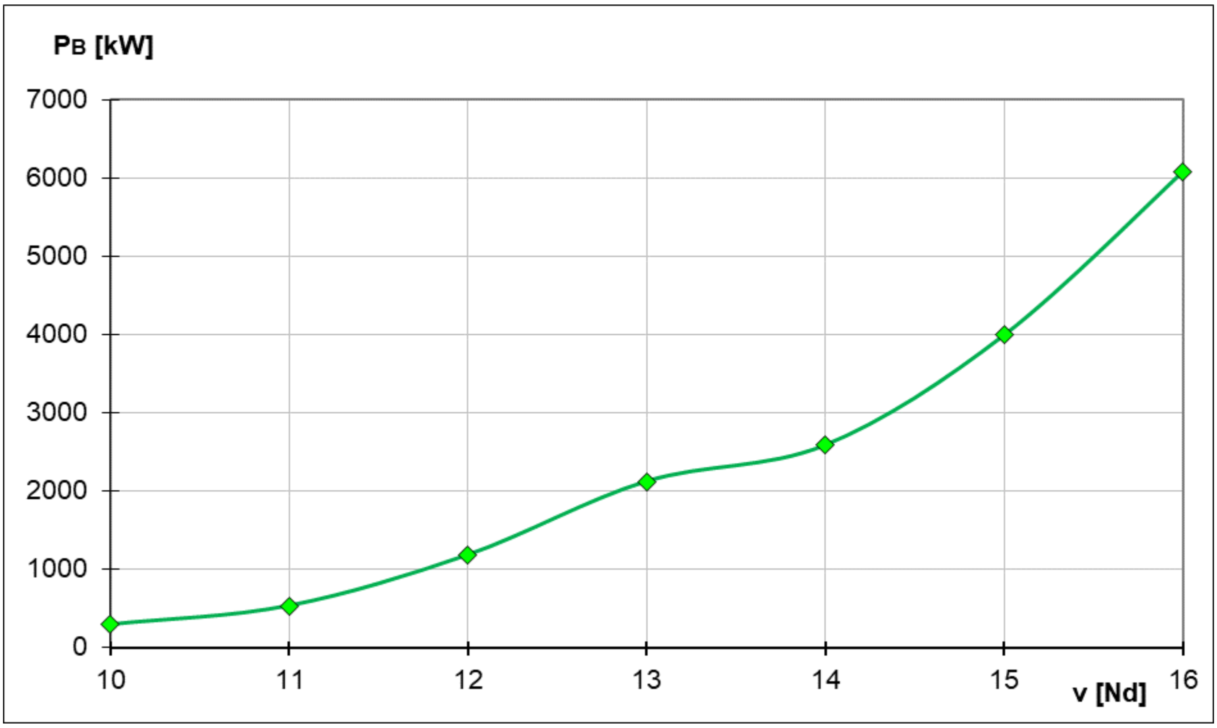

Figure 3.1 Engine flange power diagram at 100% MCR, P_B [Source: Author]

4. PRELIMINARY PERFORMANCE ANALYSIS

INCIDENT WAVE PROPULSION

4.1. Determination of propulsion power required on board the vessel on crashing waves

For the determination of the propulsive power on incident waves, the methodology presented ă in paragraph 3.1. was used, with the specification that the total resistance to forward motion on calm water ă was replaced by the total resistance to forward motion on incident waves, R_{TW} obtained by adding the R.M.S value to the additional resistance on incident waves, $R.M.S._{RAW}$, to the total resistance to calm water forward, R_T :

$$R = R_{TWT} + R.M.S._{RAW}$$

The above relation is possible under the assumption of a linear hydrodynamic model, which allows the superposition of hydrodynamic effects, an assumption used by the SEAKEEPING program within the INITIAL DESIGN module of the AVEVA MARINE CAD-CAE system.

In this sense, the effective towing force on incident waves, P_{EW}, is calculated with the relation

$$P_{EW} = R_{TW} \cdot v \cdot (1 + M_D)$$

The available propeller power on incident waves is determined with the expression

$$P_{DW} = \frac{P_{EW}}{\eta_D \cdot n_p}$$

Engine flange power at 100% MCR is calculated with the relation

$$P_{BW} = \frac{P_{DW}}{\eta_{ax} \cdot \eta_{red} \cdot (1 - M_S)}$$

4.2 Analysis of the propulsion power of the rescue tug on incident waves from the bow

Table 4.1 shows the results of the calculation of the total forward resistance on incident waves from the bow, R_{TW}, for a range of velocities between 10-14 Nd).

v [Nd]	R_{TW} [kN]		
	$h_{1/3}$ =3 m	$h_{1/3}$ =4 m	$h_{1/3}$ =5 m
10	91.3	99.3	106.3
12	199.9	211.9	221.9
14	345.6	360.6	373.6

Table 4.1 Total forward resistance on incident waves from the bow[Source: Author]

For the calculation of the power ,launch the values of the coefficients in the relationships presented in paragraph 4.1 are the same as those used in the calculation of the still water power chain: M_D =0.10; M_S =0.15; n_p =2 thrusters; $= \eta_{ax}\ \eta_{red}$ =0.97; SR=100%.

Table 4.2 shows the results of the calculation of the effective towing power on incident waves from the bow, P_{EW} , for a speed range of 10-14 Nd.

Under the assumption of keeping the values of the quasi-pulsive coefficient η_D used in the calculation of the available propeller power in calm water (shown in Table 3.1), the available propeller power was determined when sailing on incident waves from the bow .

Table 4.3 shows the results of the calculation of the available propeller power on incident bow waves, P_{DW} , for a speed range of 10-14 Nd.

Table 4.4 shows the results of the calculation of engine flange power at 100% MCR on incident bow waves, P_{BW} , for a speed range of 10-14 Nd.

v [Nd]	P_{EW} [kW]		
	$h_{1/3}$ =3 m	$h_{1/3}$ =4 m	$h_{1/3}$ =5 m
10	516.2	561.4	601.0
12	1356.3	1437.7	1505.6
14	2735.6	2854.5	2957.4

Table 4.2 Effective towing power, per incident wave from bow [Source: Author]

v [Nd]	P_{DW} [kW]		
	$h_{1/3}$ =3 m	$h_{1/3}$ =4 m	$h_{1/3}$ =5 m
10	424.5	461.7	494.2
12	1237.5	1311.8	1373.7
14	2697.8	2815.1	2916.6

Table 4.3 Available propeller power on incident waves from the bow [Source: Author]

v [Nd]	P_{BW} [kW]		
	$h_{1/3}$ =3 m	$h_{1/3}$ =4 m	$h_{1/3}$ =5 m
10	530.6	577.1	617.8
12	1546.9	1639.8	1717.1
14	3372.3	3518.9	3645.8

Table 4.4 Engine power at and flank, at 100% MCR, on incident waves from the bow [Source: Author]

Figure 4.1 shows the engine flange power diagram at 100% MCR, on waves incident from the bow, P_{BW} , as a function of vessel speed and and actual sea state (waves with significant height of 3 m, 4 m and 5 m). Also plotted is the engine flange power characteristic curve at 100% MCR in calm water, P_B .

The power at the engine flange was determined, at 14 Nd operating speed, at 100% MCR, assuming calm water, power equal e to 2586.8 kW. Taking this value into account, the loss of speed of the vessel is determined, on frontal incident waves, by drawing the curve with constant power of 2586.The values of the ship's speed on frontal incident waves, v_w , at flange power equal to 2586.8 kW are shown in Table 4.5. A significant reduction of the ship's speed on frontal incident waves compared to the regime speed on calm water (14 Nd) is observed for the three sea states studied. With increasing significant wave height, the loss of speed increases.

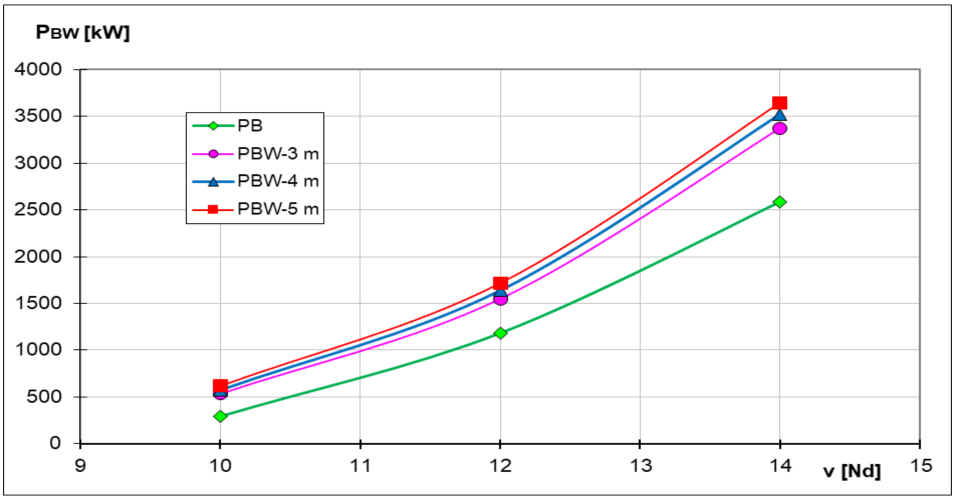

Figure 4.1 Engine flange power plot at 100% MCR, on incident waves from the bow, P_{BW} , as a function of vessel speed, v and actual sea state (significant wave heights of 3 m, 4 m and 5 m) [Source: Author]

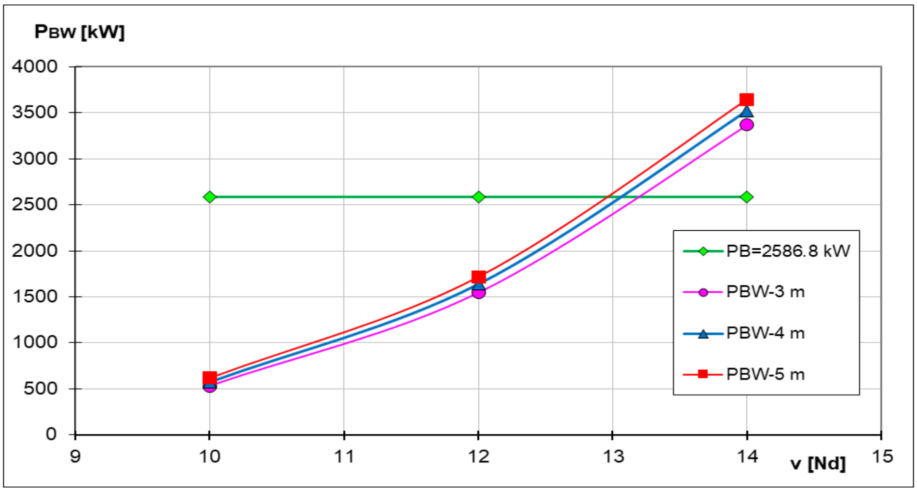

Figure 4.2 Determination of the ship's speed, on frontal incident waves, at flange power equal to 2586.8 kW [Source: Author]

v_W [Nd]		
$h_{1/3}$ =3 m	$h_{1/3}$ =4 m	$h_{1/3}$ =5 m
13.2	13.1	12.95

Table 4.5 Vessel speed on incident waves from the bow, v_w , at engine flange power at 100% MCR, equal to 2586.8 kW [Source: Author]

BIBLIOGRAPHICAL REFERENCES

[1] **Obreja C.D., Manolache L., Popescu G.**, *Bazele progettazione preliminare a nave*, Editura ACADEMICA, Galați, 2003.

[2] **Larson L., Baba E.**, *Ship resistance and flow computations*, Advances in Marine Hydrodynamics, p.1-75, 1996.

[3] **Obreja C.D.,** *Ship theory. Concepts and methods of analysis of hydrodynamic performance*, Didactic and Pedagogical Publishing House, Bucharest, 2005.

[4] **Holtrop J., Mennen G. J.**, *An Approximate Power Prediction Method*, International Shipbuilding Progress, 1982.

[5] **Holtrop J.**, *A Statistical Re-Analysis of Resistance and Propulsion Data*, International Shipbuilding Progress, 1984.

[6] **Obreja C.D.,** *CAD-CAE Tools for Initial Design*, Lectures notes, "Dunarea de Jos" University of Galati, Naval Architecture Faculty, 2014.

[7] **Obreja C.D.**, **Marcu O.**, *Preliminary ship design. A guide to computer-assisted works*, Publishing house of the "Dunărea de Jos" University Foundation of Galati, 2016.

[8] **Parsons M.G.**, *Parametric design*, University of Michigan, 2002.