POWER AND PROPELLER REQUIREMENTS FOR HARD CHINE PLANING CRAFTS

OVERVIEW

The aim of the present Part 1 compendium is to describe and synthesized the various basic elements in the evaluation of resistance and power performance of a hard chine planing craft, when propelled by conventional marine propeller(s) driven by an inclined shaft.
All relevant reference documents have been herewith highlighted as well as the various equations, extracted from the same papers herewith quoted as a reference, nowadays being considered as “the bible” in determining the magnitude, location, direction and final equilibrium of the various hydrodynamic forces involved.
The assessment of the EHP (bare hull only in this first section) for a planing boat design could, in such a way, be conveniently computed so as to offer a general but comprehensive summary form. Nothing new but at least all necessary informations are herein gathered in a useful and convenient manner for anyone interested in the subject. Where ever possible, shortcuts, reference diagrams, tables, charts, formulae and so and so forth, will also be included so as to enhance information on the issue.

SECTION A

1. BASIC ELEMENTS & BEAR HULL RESISTANCE APPRAISAL

The term “High Speed”, as stated by Prof. Emeritus D Savitsky, identifies a key parameter for operation which is the Speed Length Ratio (SLR) which, to be consistent with a hard-chine planing hull form geometry, should be not less then 3 (SLR > 3.0) being:

$$SLR = Vs * LWL^{0,5}$$

(where Vs is boat speed in knots; LWL under static condition in feet); also defined as Taylor Quotient (TQ).
This basic index, associated to calm water resistance considerations, influences the geometric configuration of the hull form related to the hydrodynamic phenomenon of planing.
To be more accurate (as the LWL of a planing boat is not easy to determine under dynamic conditions) a planing hull is better defined by the minimum acceptable value of Volumetric Froude Number identified as

$F_{N\nabla}$ » > 2,5 - non-dimensional - which is given by the relation:

$$F_{N\nabla} = V_S / [g*\nabla^{1/3}]^{0,5}$$

where Vs is boat speed in m/s; g 9,8 m/s^2 (gravitational constant); ∇ is the displ. Volume in m^3 – see Nomogram in fig.2 for determining **$F_{N\nabla}$** graphically.
The typical body plan belonging to a hard chine high speed planning hull configuration is shown on Fig. 3. Chines are essential to cause flow-separation thus producing the steady lift forces essential for trim, the hull must have then a positive trim angle and a transom stern and also, further improvements in the design will present spray-rails running bow to stern.
The body plan is showing concave transverse sections in the bow area instead of convex. However, if they are confined to the front portion of the hull, which is usually out of water due to trim on planing speeds, concave transverse sections are considered not to be detrimental to the hull resistance. In short, **basic hull design features** (as stated on many reference papers) for a planing craft, should include the following:

a. Avoidance of longitudinal convex curvature in the aft portions of the hull in order to prevent the generation of negative pressure in this area, hence, straight horizontal buttock lines on the aft should be envisaged, meaning constant deadrise and straight sections in the after body to take advantage of the extra dynamic lift available at higher speeds;
b. Chine width at the transom should not be greater then the maximum chine width in order to avoid reattachment of the separated flow developed at the forward extent of the planing area
c. deeply submerged wide transom holding a sharp trailing edge (meaning a bilge knuckle running from port to starboard), to induce early and complete flow separations at the stern in order to reduce the hull form drag;
d. sharp edge chines to induce flow separation along the side of the hull, hence, avoid a drag increase which would otherwise develop from side wetting of the hull;
e. V-bottom transverse section with a deadrise design into the hull bottom.

Typical body plan of a prismatic planing hull

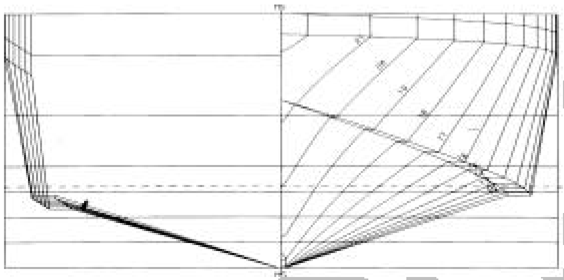
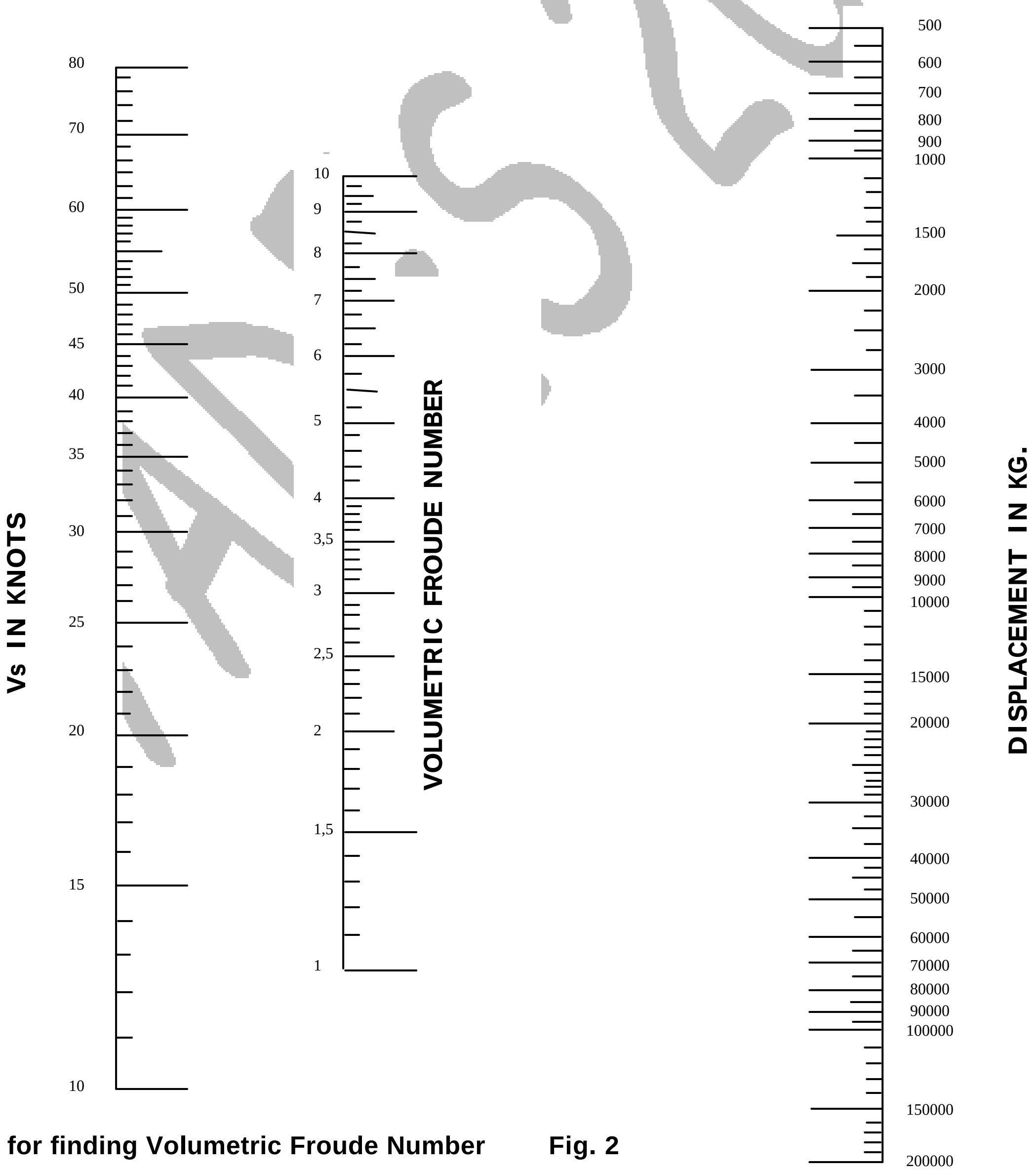


Fig. 1



Nomogram for finding Volumetric Froude Number **Fig. 2**

In **evaluating Resistance**, the main parameters and ratios involved, as emphasized by E. P. Clement ref. [2] are the following:

- Length to Beam ratio
- Size-Displacement coefficient
- Longitudinal location of CG
- Deadrise angle
- Longitudinal curvature
- Shape of chine line in plan view
- Type of section

To be further specific…..:

I) **Length to Beam ratio** is defined as **Lp/Bp_a** (projected chine length) / (beam average over chines -external spray strips excluded- in other words the aspect ratio of the projected Water Plane Area or planing bottom area).

II) **Size-Displacement coefficient (or loading coefficient)**
The relationship between hull size and the gross weight of a boat can be expressed in a convenient dimensionless form by the ratio **$Ap/\nabla^{2/3}$** where Ap is the projected planing bottom area and ∇ is the displacement volume in static condition); since this coefficient is dimensionless it yields the same value for geometrically similar boats of different size but of corresponding loading; allegedly, it also yields the same value for two boats which have different length-beam ratios but the same area, Ap, and the same displacement volume.
N.B. on these basis it does not appear possible to make as plausible a case for the other coefficients which have been used to characterize the size-displacement relationship of planing boat - i.e. $\Delta/(L/100)^3$ and the load coefficient $\Delta/\rho*Bp_x^3$, otherwise most frequently used.

III) **Longitudinal location of CG** It is considered as the ratio **LCG/Lp,** however it is also regarded appropriate to define longitudinal CG location as the distance of the LCG from the centroid of the area, Ap, expressed as a percentage of the length Lp; LCG typical values are considered being in the range of **40% to 45%** approx. of Lp, measured from transom.

IV) **Deadrise angle** in order that hull bottom impact loads be kept within acceptable limits an average deadrise angle should be around not less then **10° to 15°** from amidships to the transom and obviously with a much larger angle towards the bow.

V) **Longitudinal curvature** Being the longitudinal curvature of the hull bottom shown by the shape of the buttock lines, for the purposes of comparison and analysis, it is desirable to define an average, or mean, buttock. This can be conveniently done by intersecting the straight line approximations to the body plan sections by a buttock plane spaced at b/4 from the centreline plane. The mean buttock lines reflect the general practice to have straight buttock lines in the after portion of planing hull bottoms.

VI) **Shape of chine line in plan view** The significant features which are indicated by the shape of the chine line in plan view are the length/beam ratio of the boat and the of breadth and of bottom area. Length/beam ratio has already been defined as the ratio **Lp/Bp_a**.; however, to compare relative fore-and-aft distribution of bottom area, it was appropriate to reduce the plan view of the chine line to a form independent of length/beam ratio, to allow using several dimensionless ratios indicative of the relative fore-and-aft distribution of breadth such as:
- the location of point of max chine breadth, as a percentage of hull length from transom;
- ratios of maximum breadth and of transom breadth to the mean breadth Bp_a;
- the location of the centroid of the plan-view bottom area Ap as a percentage of Lp

VII) **Type of section** As said previously the use of convex transverse sections in a body plan is nowadays a must. And the use of developable surfaces will generally result in this type of section.

The fundamental hydrodynamic characteristics of prismatic planing hulls and the empirical planing equations involved are thoroughly described in Dr. D. Savitsky paper - ref .[1].
The equations given, describe the lift, drag, wetted area, centre of pressure and porpoising stability limits of the planing hull, as a function of speed, trim angle, deadrise angle and loading.
These empirical planing equations are combined so as to formulate a computational procedure in the form of a table where forces and moments acting on a planing hull are considered, in order to determine an equilibrium trim and, are applicable only to the bottom pressure area aft of the leading edge stagnation line. On that basis a prediction can be made on resistance, effective power, running trim, draft and porpoising stability of a prismatic planing hull.

This tabulated procedure is commonly used by naval architects today and can easily been implemented by means of a computer program which, via calculations and use of iterative methods, allows a graphical interpolation of the results thus obtained.

Furthermore, there are several reliable sources for hull resistance data.
The most well known procedures for the analysis of the resistance and the prediction of the propulsive system comes from the studies of systematic hull series where an extensive and fundamental investigation on planing hull forms has been carried out on systematic series DTMB Series 62 and Series 65 (for hard chine planing crafts power prediction) conducted by *Clement- Blount* and *Hubble* in the mid-sixties and still valuable.
Subsequently, it has been developed a practical method for predicting the power performance of a planing craft when propelled by conventional marine propeller and driven by on inclined shaft.
The tests incorporated systematic variations of chine planform, length to beam ratio and deadrise angle and resistance to displacement weight ratios and trim angles, derived from model experimental data at comparable speed and loading conditions, as a function of the LCG.
When using this data it is important to make sure that your hull form is within the *envelope* of the series.
For instance..... “is the hull falling within the range of Lp/Bp_a ratios or $F_{N\nabla}$ numbers tested?”

Two other major resistance contributors are appendage drag, and wind resistance in the case of very high speed craft.
There is no standard for how each of the systematic series treats these drag producers.
Therefore, it is important that the designer verify that these are accounted for properly.

Further investigations, made over the last few years, have shown that for resistance calculation some of the latest regression methods are sufficiently accurate over the speed range for which they had been developed; while some others, have been considered inadequate.
Of the former ones, is worth mentioning that of D. Radojcic (1985) ref. [12].

Moreover, it goes without saying that, the most powerful tool in the hand of a yacht designer is undoubtedly model testing. This can be a fairly expensive and time consuming undertaking.
As such, its value to the overall project should be discussed with the client.
It is common practice that, unless the design is particularly unusual or there is a strict contract speed, a combination of in-house data, experience, and/or existing data in the public domain, is sufficient to predict speeds within **5%** approximation values.
For preliminary design purposes, tank test results or full size speed trial results from similar parent vessels, can be used.
The so called “model correlation technique”, meaning the comparison between calculated numerical results and model test (or sea trial) of a similar *parent vessel* can be used.
The information thus obtained can be adjusted according to the traditional hydrodynamic methods, to account for frictional and viscous drag.

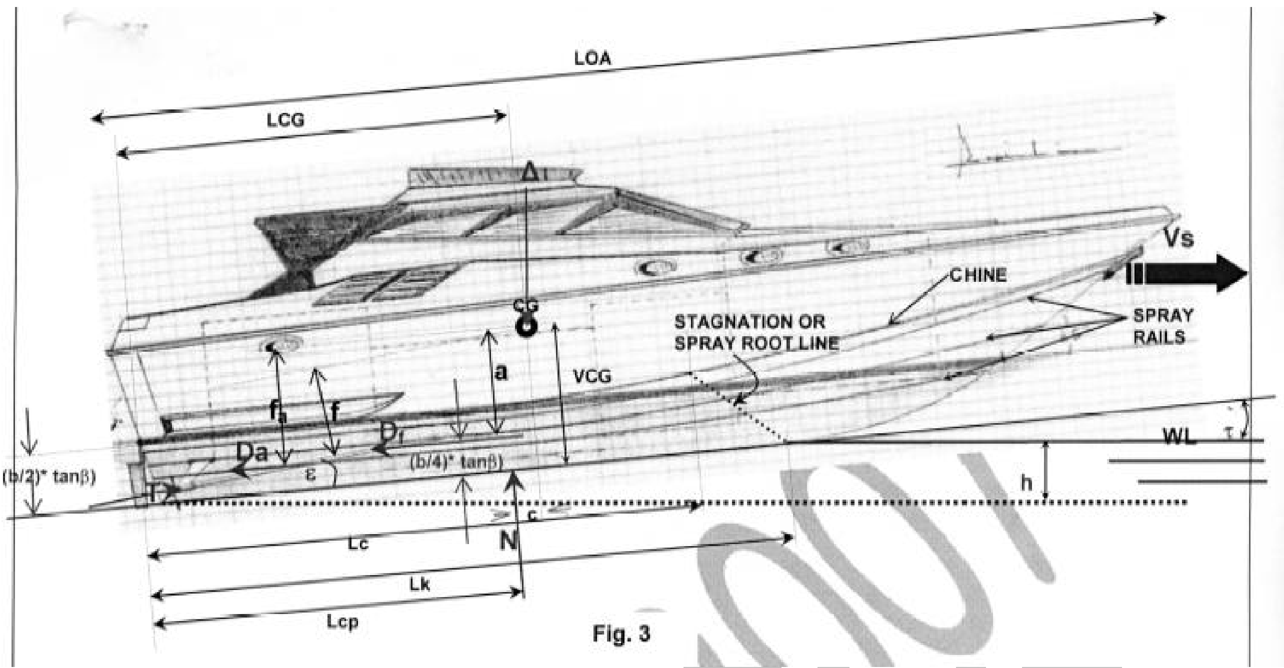


Fig. 3

Main parameters involved in the computational procedure for resistance prediction – see ref. [1]

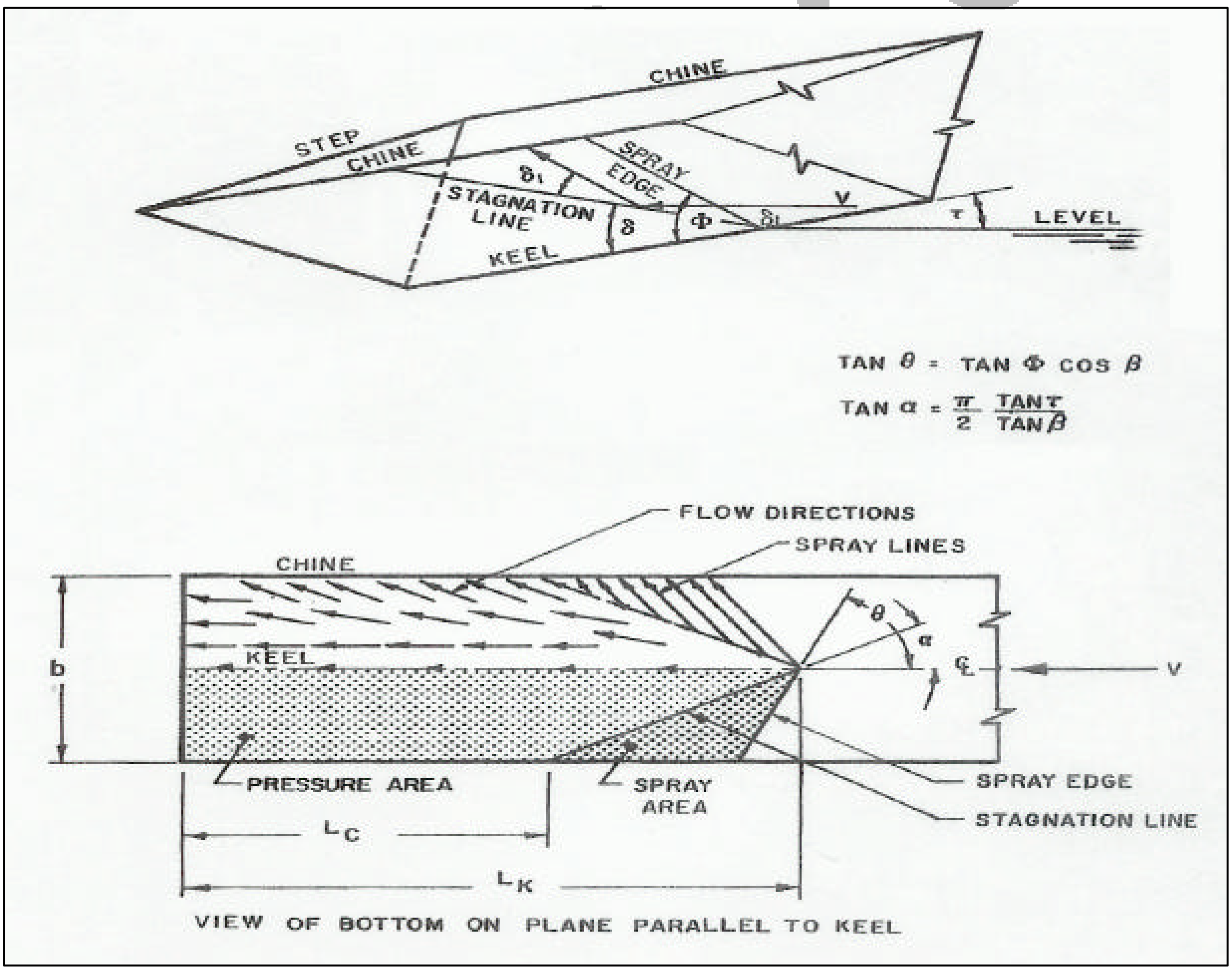


Fig. 4

3 REFERENCE SYMBOLS COMMONLY USED

Planing hull geometry		
LOA	length over all	m
LWL	length water line on (hydro)static condition	m
b	or **B_{Pa};** chine beam (average) or beam of planing surface measured between the chines	m
Ä	displacement mass or vertical load on water (gross weight – i.e. static condition)	kg
Ñ	displacement volume (or volume of water displaced @ rest – i.e. static condition)	m^3
h	vertical depth of trailing edge of boat (at keel) below level water surface or depth of keel @ transom	m
LCG	location of the longitudinal centre of gravity , forward of transom	m
VCG	vertical centre of gravity, above baseline	m
p	number of propellers	
r	number of rudders	
V_s	boat forward planing velocity or horizontal velocity of planing surface	m/s
b	deadrise angle (degrees) - average, usually taken @ 0,5 **Lp**	deg
e	propeller shaft line inclination relative to the baseline (or keel line)	deg
t	trim (angle between planing bottom and horizontal)	deg
C_V	Speed coefficient **[Vs/(g*b)$^{0.5}$]**	
$F_{NÑ}$	volume Froude number, $V_S \Big/ \sqrt{g\nabla^{1/3}}$ but also	
Planing surface hydrodynamics		
C_f	Schoenherr frictional drag coefficient based on Raynolds number	
ÄC_f=C_A	Friction coefficient allowance for roughness of planing surface or Correlation Allowance, Savitsky used **C_A = 0.0004** & ITTC recommends C_A = 0.003 for this friction line.	
C_{L0}	Lift coefficient @ zero deadrise; $\dfrac{\Delta}{0.5*\rho*V^2*b^2}$	
C_{Lb}	lift coefficient with deadrise surface; $C_{L\beta} = C_{LO} - 0.0065*\beta*C_{LO}{}^{0,6}$	
L_P	**or L;** projected length of chine from transom to bow profile	m
L_K	Projected wetted keel length	m
L_C	Projected wetted chine length measured from transom to spray root (stagnation line) intersection with chine (excluding spray)	m
L_M	Mean wetted length of pressure Area	
B_X	beam max	m
l	**or L_M/b**; mean wetted length / beam ratio: **[(L_K + L_C)/ 2b]** or elsewhere **[L_P / B_{Pa}]** ca.	
Äl	Effective increase in friction area length beam ratio due to spray contribution to drag	
L_{CP}	longitudinal location of centre of pressure from trailing edge (i.e.transom)	
C_P	Centre of pressure C_P = **L_{CP}/ L_M**	
N	Resultant of pressure (hydrodynamic) and buoyancy (hydrostatic) forces assumed acting normal to hull bottom	N
A_P	projected planing bottom area (excluding external spray strips) or total bottom pressure area	m^2
S_W	Principal wetted surface area (bounded by trailing edge, chines and heavy spray line)	m^2
S_S	Area wetted by spray	
f	perpendicular distance off shaftline to Centre of Gravity (CG)	m
a	Also equal to f_f in the reference papers, is the perpendicular distance between frictional drag-force component D_f and CG; **a** = **[VCG - (b/4)* tanb)]**	m
f_a	Distance between Appendage Drag **D_a** (assumed as acting parallel to keel line) and CG , measured normal to **D_a**	
g	acceleration due to gravity (or gravitational constant) = **9.81**	m/s^2
R_N	Reynold's Number, $\dfrac{V*L}{\nu} = \dfrac{V_M*L_M}{\nu} = \dfrac{b*\lambda*V_S}{\nu}*\left(\dfrac{V_M}{V_S}\right) = \dfrac{V_M*b*\lambda}{\nu}$	

n	Kinematic viscosity of fluid (salt water @ 20° = $1*10^{-6}$)	m^2/s

Planing surface hydrodynamics/…cont.d		
V_M	Average water bottom velocity over the pressure area (see ref. [1] pag 83)	
q	Angle between the keel (centerline) and the outer edge of spray area measured in plane of bottom	deg
ñ	specific weight of water (or mass density of water)	kg/m^3
D_f	Frictional Drag-force component along bottom of surface	N
D_a	Appendage Drag (assumed as acting parallel to keel line)	N
T	Propeller thrust along shaft line	N
d	Diameter of shaft or bossings	m
c	$c = L_{CG} - L_{CP}$; distance between N (pressure force applied to centre of pressure) and CG measured longitudinally from transom stern and normal to N	m

3. EMPIRICAL EQUATIONS AND DIAGRAMS

For Resistance and EHP computations of planing hulls in general, Savitsky methods involves the following empirical equations, amply described in ref. [1]; [3]; [12]:

$$C_V = \frac{V_S}{\sqrt{g * b}}$$ speed coefficient computed from given data (or better saying "Breadth-Froude-Number"**(1)**

used for planing analysis instead of length – SLR –, as previously mentioned).

The lift coefficient **for flat** planing surfaces, developed from given data, is given by the following equation:

$$C_{LO} = \frac{\Delta}{0,5 * \rho * V_S^{\,2} * b^2}$$ (**N.B.** for a hull with a deadrise it will give $C_{L\beta}$ - see ref. [18] pag. 192) **(2)**

Lift for flat planing surfaces, solved for τ values is also given from the equation:

$$C_{LO} = \tau^{1.1} * \left(0.012 * \lambda^{0,5} + 0.0055 * \frac{\lambda^{2.5}}{C_V^{\,2}}\right)$$ **(3)**

C_{LO} can be readily find for a given value of τ from diagram of fig. 5, extracted from Savitsky paper and, as stated in the paper, for convenience in use, the lift coefficient equation for C_{LO} is plotted in the form $\left(C_{LO} / \tau^{1,1}\right)$ versus λ for different values of C_V.

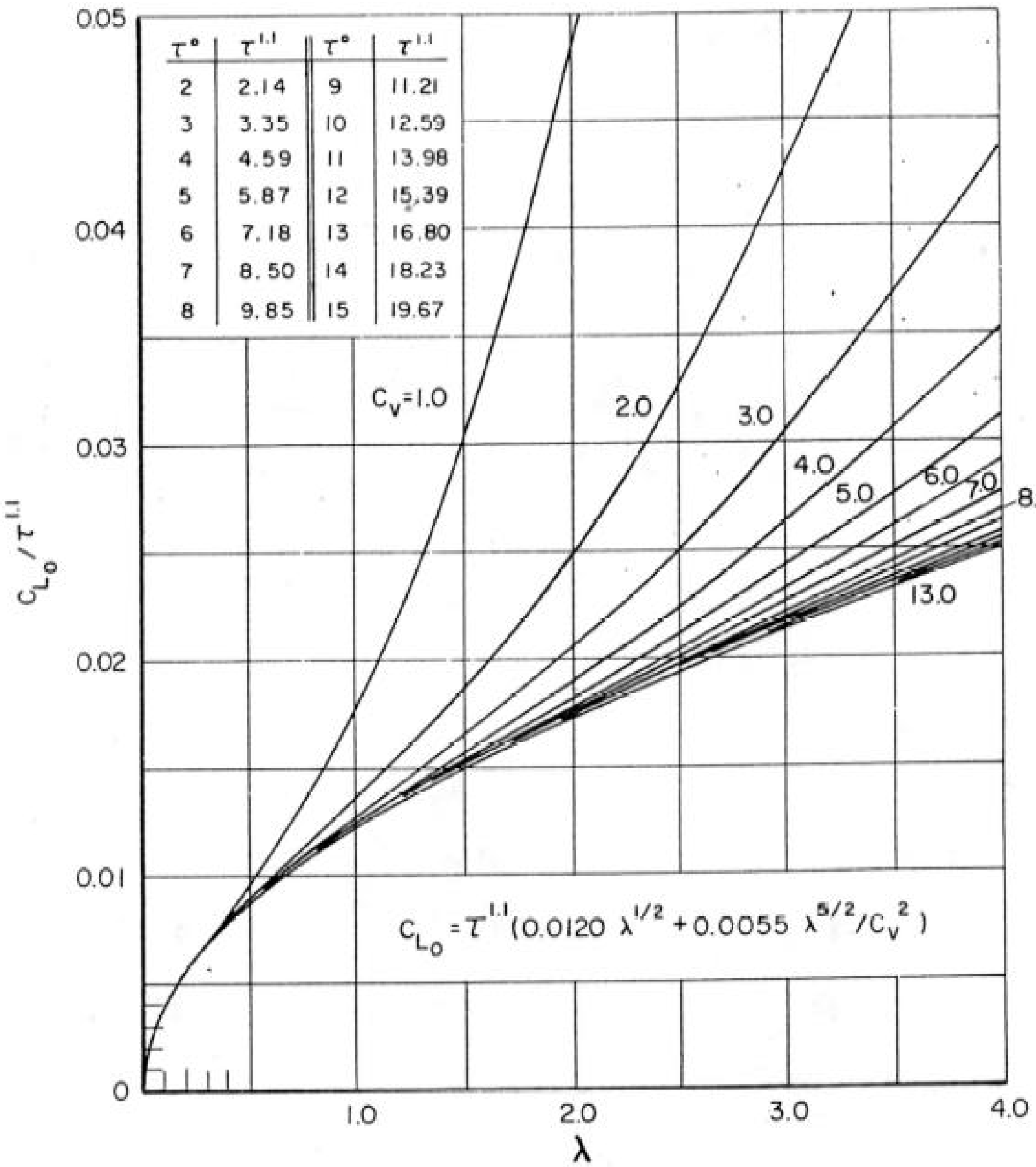


Fig. 5

Lift coefficient of a flat planing surface at 0°deadrise (β) - Ref. [1] Savitsky 1964

When deadrise is introduced, this tends to reduce the planing lift, so a larger wetted surface or trim angle is required, which both increase the resistance.
Savitsky has also developed an empirical equation for use in predicting the lift of a so called prismatic hull, which corrects the lift of the flat planing surface as follows:

$$\boxed{C_{L\beta} = C_{LO} - 0.0065 * \beta * C_{LO}^{0,6}} \quad (4)$$

For quick reference corresponding value of C_{LO} can be found for a given $C_{L\beta}$ at corresponding deadrise angle β, from figure 6 – see also ref. [1]

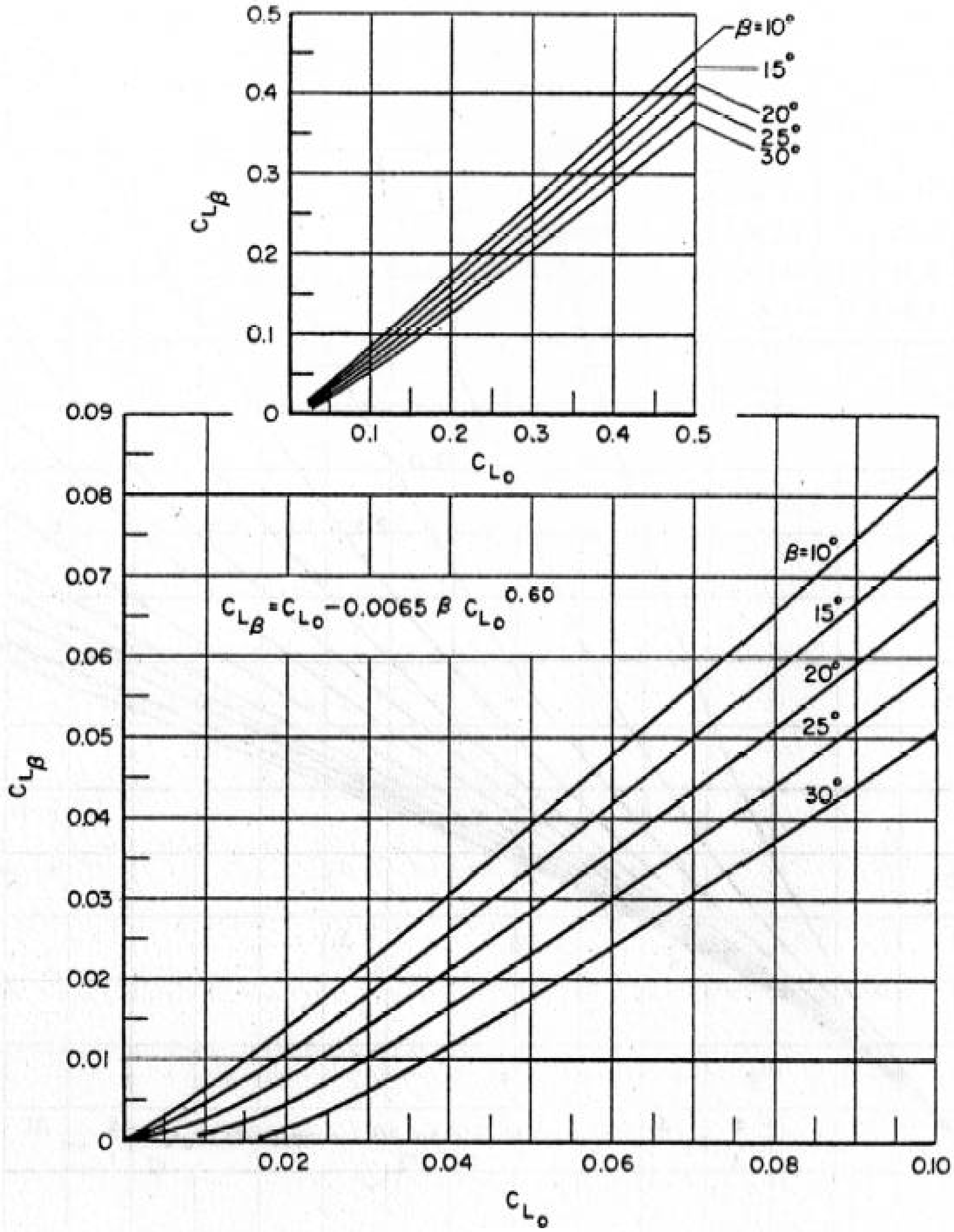


Fig. 6

Lift coefficient for a deadrise planing hull

Centre of pressure for **flat** planing surfaces - see ref. [12] - is given, as a fraction of Mean Wetted Length L_M, as follows:

$$C_P = \frac{L_{CP}}{L_M} = 0.75 - \frac{1}{5.21 * \frac{C_V^{\,2}}{\lambda^2} + 2.39}$$ the equation is solved for values of ı.. **(5)**

<u>**N.B.** all above equations are only valid within a limited range of parameters as follows:</u>
For Equations (3) & (5) they are applicable within the following values:

$0.60 \le C_V \le 25$

2° ≤ ô ≤ 15°

ë ≤ 4

whilst equation (4) is applicable for:

$$10° \le \beta \le 30°$$
$$1.0 \le \lambda \le 4.0$$

In addition, the best results are attained <u>for beam equal to B_{Pa}</u> and <u>deadrise at **0.5** L_P</u> – see ref. [12]

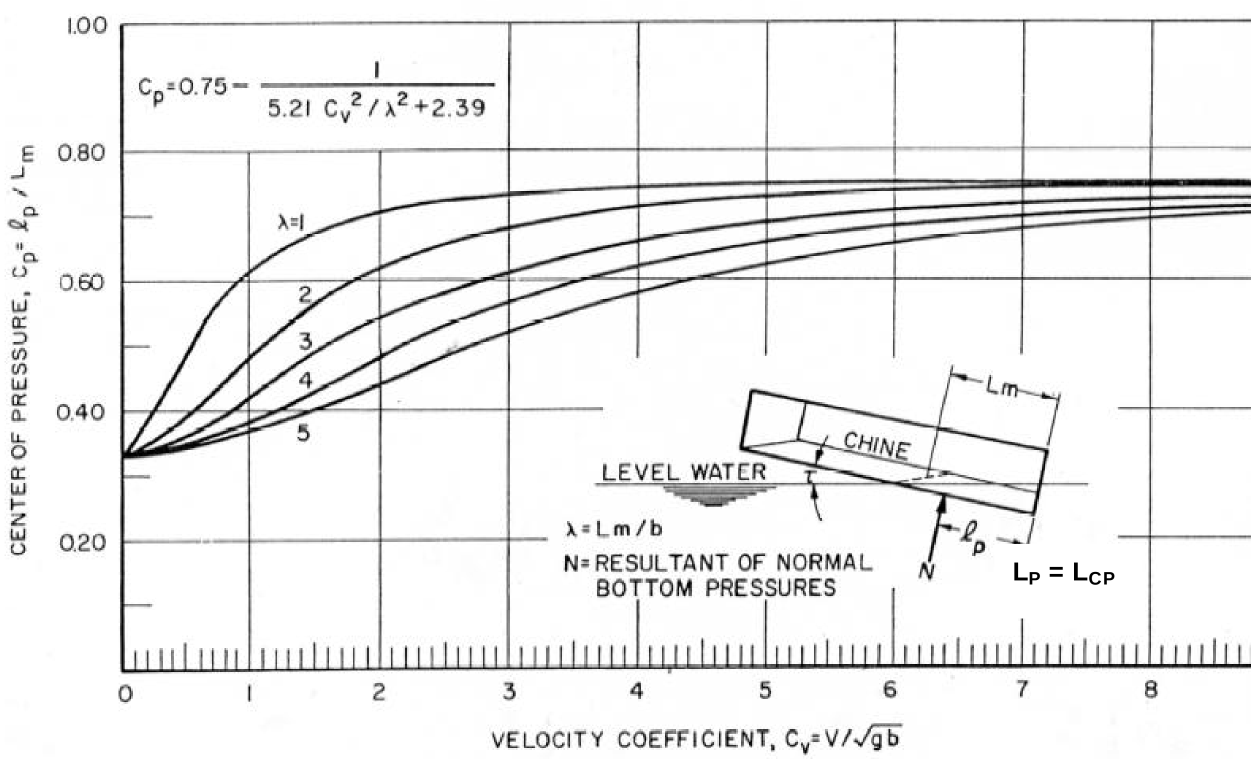


Fig. 7 **Centre of pressure of planing surfaces for given ı and C_V**

Furthermore, the **wetted keel length** measure is given by:

$$L_K = L_M + \frac{b * \tan\beta}{2\pi * \tan\tau}$$ however, nowadays, is rear to find a hard-chine planing hull fitted with a keel. For a hull without keel, the above equation can be omitted from the resistance calculation.

Instead, when the speed coefficient (C_v) is **grater then 2**, it is applicable for all deadrise and trim values the **wetted chine length** given as follows:

$$L_C = L_M - \frac{b * \tan\beta}{2\pi * \tan\tau} \qquad \textbf{(6)}$$

From ref. [20], S_S (area wetted by spray) is expressed by the following relation:

$\boxed{S_S * \cos\vartheta = \frac{(\Delta\lambda) * b^2}{\cos\beta}}$ being **S (wetted surface)** equal to $\boxed{S = \frac{\lambda * b^2}{\cos\beta}}$ where $(\lambda * b^2)$ is the **bottom pressure area.**

The **main wetted surface area** parameter is then given by the following equation:

$$S_w = \frac{L_M}{b} * b * \frac{b}{\cos\beta} = (\lambda + \Delta\lambda) * \frac{b^2}{\cos\beta} \qquad \textbf{(7)}$$

The hydrodynamic drag of a planing surface is composed of the pressure drag acting normal to the inclined surface and the viscous drag acting tangential to the surface (assuming no side wetting of the hull). The viscous drag forces can be expressed as the sum of two components, the wetted surface drag and the viscous component of the spray drag, as the spray from a bottom with deadrise, normally, increases the frictional resistance since most of the spray actually goes backwards – see ref. [18].

Thus, the viscous drag force in the direction of the planing surface maybe simply expressed by the following equation: $\boxed{D_f = 0.5 * \rho * S_W * V_S^{\,2} * C_f}$

Substituting in the above formula the S_W value as outlined in equation (7), the viscous force in the direction of V_M, can be simply expressed as follows

$$D_f = \frac{\rho}{2} * \frac{b^2 * V_S^{\,2}}{\cos\beta} * (C_f + \Delta C_f) * \left[\left(\frac{V_M}{V_S} \right)^2 * \lambda + (\Delta\lambda) \right] \qquad \textbf{(8)}$$

Next, we can compute the Friction Coefficient C_f, adopted as the "I. T. T. C. 1957 Model-Ship Correlation Line" or Schoenherr's turbulent- skin friction drag coefficient, which is a function of Reynold's Number. Hence, Schoenherr frictional drag coefficient C_f is found by applying the following formula:

$$C_f = 0.075 / (\log_{10} R_N - 2)^2 \qquad \textbf{(9)}$$

The Reynold's Number R_N can be solved by applying the formula:

$$R_N = \frac{V_M * L_M}{\nu} = \frac{\lambda * b * V_S}{\nu} * \left(\frac{V_M}{V_S} \right) = \frac{V_M * b * \lambda}{\nu} \qquad \textbf{(10)}$$

Where, to solve equation (8), the values for $(\Delta\lambda)$ and V_M have been expressed as functions of the geometry and load characteristics of the planing surface. Savitsky and Ross in ref. [19] have developed these functional relationships in terms of the **trim** and **deadrise** of the hull.

The results have been presented in the form of simple diagrams as shown on ref. [3] (see fig. 8) from which the magnitude of average bottom velocity V_M for a planing surface is easily developed for a given vessel speed (V_S)

The equation involved for extracting the diagrams as outlined in ref. [3] is the following:

$$\frac{V_M}{V_S} = \left[1 - \frac{0.012 * \lambda^{0.5} * \tau^{1.1} - 0.0065 * \beta * \left(0.012 * \lambda^{0.5} * \tau^{1.1}\right)^{0.6}}{\lambda * \cos\tau} \right]^{0,5} \qquad \textbf{(11)}$$

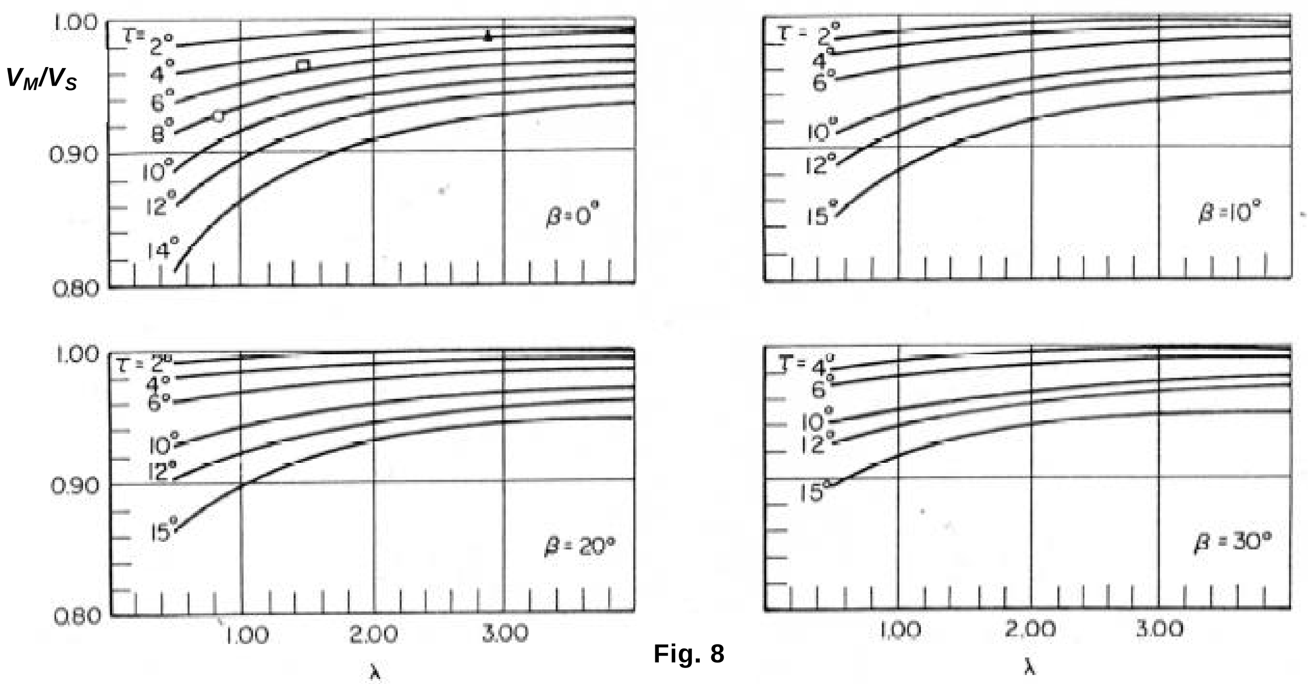


Fig. 8

Magnitude of average bottom velocity for a planing hull expressed as V_M / V_S

The correction equation due to increase in friction area length beam ratio ($\Delta\lambda$) due to spray contribution to drag, is given by the following equation (12). For quick reference, values of ($\Delta\lambda$) for different deadrise **b** and trim angles **t** can be extracted from diagram of fig. 9 (see ref. [18] pag. 189 & ref . [3] pag. 9).

$$\Delta\lambda = \frac{1}{2} * \left(\frac{\tan\beta}{\pi * \tan\tau} - \frac{1}{2 * \tan\theta}\right) * \cos\theta \qquad \textbf{(12)}$$

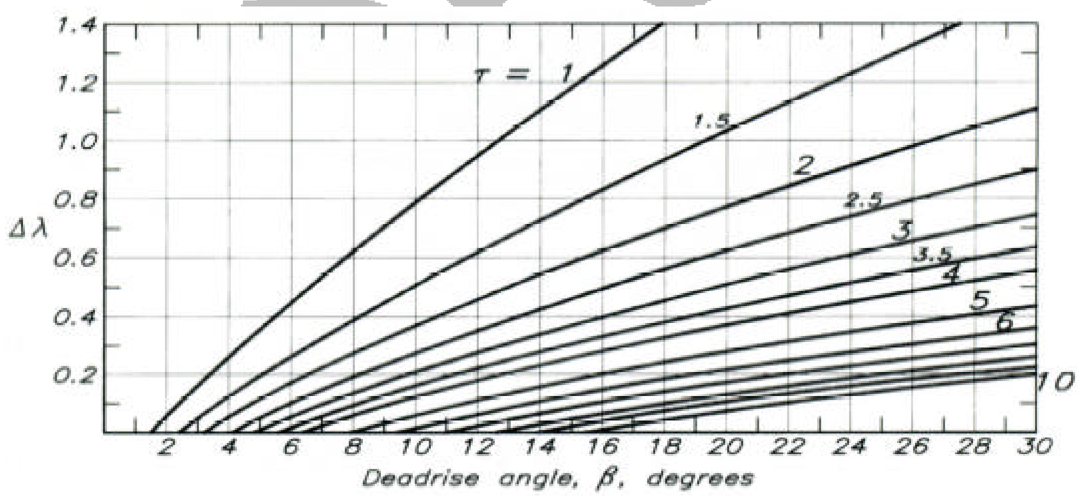


Fig. 9

Diagram for finding a ready-to-use value for $\Delta\lambda$

On this basis the final empirical equation for the Resistance Frictional component, can be written as follows:

$$D_f = C_f * 0.5 * \rho * V_S^{2} * \left(\frac{L_M}{b} + \Delta\lambda\right) * \frac{b^2}{\cos\beta} \qquad \textbf{(13)}$$

Now it's important that the planing hull achieves equilibrium at a reasonable trim angle and therefore the above twelve equations, should be enclosed in a computational procedure, for calculating trim, resistance and predicting power requirements where equilibrium equations are taken into consideration.
Hence, taking into account all the afore-mentioned forces, three static equilibrium equations are considered for Vertical, Horizontal Forces and Pitching Moments. To simplify, the moment to satisfy the equilibrium condition must equal zero. In other words, it is seen that the pressure force **N**, the friction drag force $\mathbf{D_f}$ and the resistance generated by the appendages $\mathbf{D_a}$, create a moment to trim the vessel by the bow and that their respective lever arms are **c**, **a** and $\mathbf{f_a}$. . The propeller thrust **T**, on the contrary, creates a bow-up moment with the arm **f**. The hull has then to attain a trim angle where the moments cancel, meaning that the net algebrical summation of moments should equal zero.
The so called bow-down moment (M_{BH}) for bear hull is found by linear interpolation between two computed moments at two trims angles, giving zero moment for equilibrium condition, as, most likely, the computed moment will be different from zero and so the trim angle has to be changed to obtain balance.
The equation derived in Hadler's paper (see ref [3] for more detailed explanation) considering the horizontal and vertical force balance is (for <u>bare hull moment only)</u> given by the following equation where the Moment Due to Displacement and the Moment Due to Drag Force are considered as follows :

$$M_{BH} = \Delta\left[\frac{c*\cos(\tau+\varepsilon)}{\cos\varepsilon} - \frac{f*\sin\tau}{\cos\varepsilon}\right] + D_f\left[a - c*\tan\varepsilon - \frac{f}{\cos\varepsilon}\right] = 0 \qquad \textbf{(14)}$$

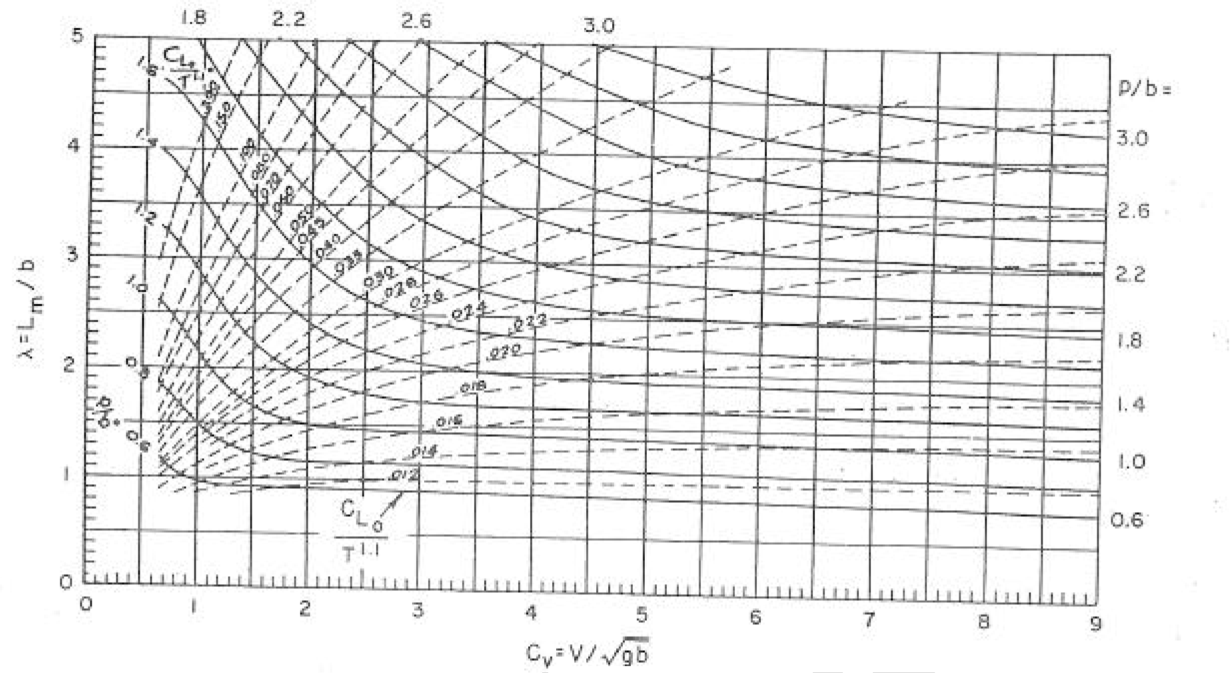
N.B. <u>The Appendages Drag ($\mathbf{D_a}$) and the relative moment generated, has not been considered, at this stage, as the aim of the computational procedure is the **Bear Hull Resistance** calculation only.</u>

In ref. [20] a simplification of the procedures proposed by Hadler (SNAME Nov. 1966 – see ref [3]), based on Savitsky's 1964 paper), has been schematically introduced.
Although Hadler's work is quite dated, it still seems to be (…..*vox populi*…..) the most accepted procedure for planing hull predictions.
For sake of clarity, the computational procedure referred to the foregoing equations listed, is here-in-after been outlined, based on Dr. Savitsky's **<u>General Case</u>** **<u>(≡ meaning for all forces NOT passing through CG)</u>** as follows: (see ref. [1], [3] & [18]):

COMPUTATIONAL TABLE FOR EQUILIBRIUM TRIM ANGLE AND CORRESPONDING <u>BEAR HULL</u> RESISTANCE AND POWER CALCULATION – TABLE 1		
step	**Description of the computational procedure and corresponding equations involved (see ref [1], [3], [18])**	**Reference**
	Given quantities:	
	Ä = displacement mass	kg
	LCG = longitudinal centre of gravity forward of transom	m
	VCG = vertical centre of gravity above keel line (or base line)	m
	b = beam of planing surface	m
	ε = propeller shaft inclination relative to base line	deg
	β = deadrise angle	deg
	f = distance between shaftline and centre of gravity	m
	Vs = hull speed	m/s
	$\mathbf{ÄC_f}$ = roughness or correlation allowance	
	R_N = Reynold's Number	
	ν = Kinematic Viscosity coefficient	
1	C_V = speed coefficient computation from given data	**(equation 1)**
2	C_{LO} = lift coefficient computation for flat planing surface = $\mathbf{C_{L\beta}}$ for a hull with deadrise (see ref [18] p. 192)	**(eq. 2)**
3	For $C_{L\beta}$ found on step 2, solve for corresponding value of $\mathbf{C_{LO}}$ by trial and error, meaning Iterations & Interpolations (for quick ref. see fig. 6)	**(eq. 4)**
4	For C_{LO} found on step 3, solve for corresponding value of λ (mean wetted length / beam ratio) assuming trim values for 2°, 3° & 4° (for quick ref. see fig. 5); where for convenience, the lift coefficient equation is plotted in the form ($\mathbf{C_{LO}}/\tau^{1.1}$)	**(eq. 3)**
5	For λ values found on step 4 compute the Mean Wetted Length $L_M = \lambda * b$ **N.B.** λ is also equal to $\mathbf{[(L_K + L_C)/ 2b]}$ where L_K and L_C can be obtained from **equation 6**	

COMPUTATIONAL TABLE FOR EQUILIBRIUM TRIM ANGLE…………………….. – TABLE 1………./cont'd		
step	**Description of the computational procedure and corresponding equations involved (see ref [1], [3], [18])**	**Reference**
6	From diagram of fig. 8 find values for V_M/V_S for a given deadrise β and λ, then $V_M = (V_M/V_S)\,V_S$ @ given hull speed and	
7	Calculate Reynold's Number from the equation $R_N = \frac{V_M * L_M}{\nu}$ (where ν is the Kinematic viscosity of fluid – in salt water - @ 20° = $1*10^{-6}$ m^2/s)	**(eq. 10)**
8	Compute Schoenherr's turbulent- skin friction drag coefficient C_f adopting the "I. T. T. C. 1957 Model-Ship Correlation Line" formula	**(eq. 9)**
9	Add ATTC Standard Roughness, where ΔC_f = C_A (see ref. symbols) = 0,0004	
10	Find increase in λ due to spray ($\Delta\lambda$) for trim angle of 2°, 3° and 4°- (for quick ref. see fig.9)	**(eq. 12)**
11	Compute Frictional (or Viscous) Drag-Force Component D_f along bottom of surface	**(eq. 8)**
12	Compute the lever arm (a) - see reference symbols - for D_f, relative to the centre of gravity CG; **a = [VCG - (b/4)* tanβ)]**	
13	Calculate the Centre of Pressure coefficient (C_P) for values of λ.	**(eq. 5)**
14	Calculate the distance (L_{CP}) to Centre of Pressure from the transom where $L_{CP} = (C_P * \lambda * b) = C_P * L_M$ (for quick ref. see fig. 7)	
15	Calculate $[c = (L_{CG} - L_{CP})]$ ….(N.B. at equilibrium trim, the value should equal zero – i.e. $L_{CP}=L_{CG}$)	
16	Find Moment due to Displacement $M_\Delta = \Delta\left[\frac{c * \cos(\tau + \varepsilon)}{\cos\varepsilon} - \frac{f * \sin\tau}{\cos\varepsilon}\right]$ in Newton/m	
17	Find Moment due to Drag Force $M_{Df} = D_f\left[a - c * \tan\varepsilon - \frac{f}{\cos\varepsilon}\right]$ in Newton/m	
18	Find total moment Bare Hull $M_{BH} = M_\Delta + M_{Df}$ in Newton/m	
19	Find by linear interpolation between two computed moment M_{BH1} and M_{BH2} the equilibrium trim angle τ_0 for zero moment; where $\tau_0 = \tau_1 - \frac{M_{BH1} * (\tau_2 - \tau_1)}{M_{BH2} - M_{BH1}}$	
20	Find by linear interpolation between two computed frictional resistance values D_{f1} and D_{f2} ,the equilibrium Frictional Drag-force component along bottom of surface D_{f0} occurring at equilibrium trim angle τ_0; where $D_{f0} = D_{f1} + \frac{D_{f2} - D_{f1}}{\tau_2 - \tau_1} * (\tau_0 - \tau_1)$	
21	Tot. Resistance at equilibrium trim angle is found where $R_{BH} = \frac{\cos(\tau_0 + \varepsilon)}{\cos\varepsilon} * [\Delta \sin\tau_0 + D_{f0}]$ in Newton	
22	**The Effective Horse Power Bare Hull is given by** $EHP_{BH} = (R_{BH} * V_S) * 1{,}34$	

The computational procedure referred to the foregoing equations and used when **all forces are passing through CG)**, hence assuming in this condition that the distances a = f = c = ε =0 is defining a relative simple case where the empirical equations for planing lift, wetted area and center of pressure can be combined into one summary plot as per the design nomogram – (see details in fig. 10 and ref. [1], [3] & [11]).



LCG = $C_p b \lambda$

Fig. 10

Case for all forces passing through CG

The computational procedure as outlined in ref. [1] is as per the following table 2:

COMPUTATIONAL TABLE FOR EQUILIBRIUM TRIM ANGLE AND CORRESPONDING <u>BEAR HULL</u> RESISTANCE AND POWER CALCULATION – TABLE 2		
step	**Description of the computational procedure and corresponding equations involved (see ref [1], [3], [18])**	**Reference**
1	Compute C_{L_O} from given values (for quick ref see fig. 6)	**(eq. 4)**
2	Calculate $\left(\frac{L_P}{b}\right)$ from given quantities	
3	Compute C_V from given quantities	**(eq. 1)**
4	Enter C_V and $\left(\frac{L_P}{b}\right)$ in fig. 10 and read across values for $\left(\frac{C_{L_O}}{\tau^{1.1}}\right)$ and $\left(\lambda = \frac{L_M}{b}\right)$ for given $\left(\frac{L_P}{b}\right)_i$	
5	Compute the value of τ	
6	Determine the mean wetted length $L_M = \lambda * b$	
7	From diagram of fig. 8 find values for V_M/V_S for a given deadrise β and τ, then $V_M = (V_M/V_S)\, V_S$ @ given hull speed.	
8	Calculate Reynold's Number from the equation $R_N = \frac{V_M * L_M}{\nu}$ (where ν is the Kinematic viscosity of fluid – in salt water - @ 20° = $1*10^{-6}$ m^2/s)	**(eq. 10)**
9	Compute Schoenherr's turbulent- skin friction drag coefficient C_f adopting the "I. T. T. C. 1957 Model-Ship Correlation Line" formula	**(eq. 9)**

COMPUTATIONAL TABLE FOR EQUILIBRIUM TRIM ANGLE…………………….. – TABLE 2………./cont'd		
10	Add ATTC Standard Roughness, where **$\Delta C_f = C_A$** (see ref. symbols) = 0,0004 approx.	
12	Compute Frictional (or Viscous) Drag-Force Component ***D_f*** along bottom of surface	**(eq. 8)**
9	Compute the quantity obtained from the following product ($\Delta * \tan\tau$)	
10	Compute the quantity obtained from the following division $\frac{D_f}{\cos\tau}$	
11	The total **Bear Hull Resistance** $R_{BH} = (step9) + (step10) = (\Delta * \tan\tau) + \left(\frac{D_f}{\cos\tau}\right)$	
12	**The Effective Horse Power Bare Hull is given by** $EHP_{BH} = (R_{BH} * V_S) * 1{,}34$	

N.B.
A metric horsepower (PS) is 98.6% of an imperial horsepower. To convert a horsepower rating (HP) into Kilowatts (kW) simply multiply the horsepower by 0.746, or, vice-versa, to convert Kilowatts (KW) into horsepower (HP) multiply kW by 1,34. However, a kilowatt is a kilowatt, so, as to avoid the difference between imperial and metric horsepower, it is wiser to express everything in terms of kilowatts.
It should also be considered that a 1000 HP (746 KW) engine operating in 95°F (35°C), and 80% humidity would produce 951 HP (709 KW) bearing **a 4.9%** reduction in output.

2. MISCELLANEOUS TO SECTION A

Establishing the effective beam as the maximum chine beam, and the effective deadrise as the deadrise at mid-chine length, allows the development of an 'engineering factor' used for modifying Savitsky's prediction method. The 'modification factor' (M) for non-prismatic hulls is a resistance multiplying factor which enables more accurate resistance prediction in the **pre-planing range**, for **non-prismatic planing hulls**, and is most suited to heavier hulls such as to be expected for normal commercial or military loading:

Modifying Factor: $$M = 0.98 + 2\left(\frac{LCG}{B_{PX}}\right)^{1.45} * e^{-2(F_{N\nabla}-0.85)} - 3\left(\frac{LCG}{B_{PX}}\right) * e^{-3(F_{N\nabla}-0.85)}$$

The limits of applicability of this equation are: $F_{N\nabla} \geq 1.0$ and $\frac{LCG}{L_P} \leq 0.46$

3. PREVIEW TO SECTION B

Section B will deal with calculation of Appendage Drag (to be added on Bear Hull Drag computation), purpoising check and finally the computational procedure involved in propeller calculation.
It will be also mentioned the Law of Similarity and the major performance factors used.

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