A COLLECTION OF SIMPLIFIED FIELD EQUATIONS FOR SES DESIGN

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ABSTRACT

Over the years I have collected a number of useful equations for the design of SES. These equations may be used either to permit the rapid sizing of an SES, or to evaluate another's design. These are "journeyman" level tools, not theoretical tools. As such, they are suitable to the smaller design office or to the naval architect working alone.

I have incorporated a number of these equations into a commercially available spreadsheet which I use for first-cut sizing of ships. As the ship matures I modify the equations or eventually replace them with calculated "real" values, but having that first set of terms allows me to quickly determine if it's "bigger than a breadbox."

In the paper I present all of the equations, and explain their derivation. I admit that in some cases I do not know the complete history of the derivation. I attempt to give credit where due and to indicate who knows more about the equations than I.

Further, since some of the equations have been drawn from commercial experience, I have had to zero some of the coefficients for this audience. However the equations themselves still represent valid laws of behavior, and appropriate values for the missing coefficients can be supplied by analysis of the reader's own database.

INTRODUCTION

I shall begin with an explanation of the purpose of this paper. I will do this by proposing a scenario in which rapid solution of the ship sizing problem is necessary.

In the remaining part of the paper I will present the field equations I have developed, which permit such a rapid ship sizing. The paper is organized topically, treating resistance and propulsion, SES lift systems, and weight estimating.

PURPOSE

The task of initial ship sizing is one of the most important, and sometimes one of the most difficult, of the tasks of the ship designer. It is also a task that must be performed rapidly.

A customer may walk in and say that he wants to carry 150 passengers and 50 cars at 40 knots for 100 miles. How big is the SES that does the job? As a designer in this situation we need the ability to make a rapid, yet reasonable, estimate of the ship's major characteristics. This initial sizing helps the customer refine his requirements, and provides a starting point for the comprehensive ship design that follows.

Using simplified relationships it is possible to estimate a craft's characteristics within acceptable accuracy in a few hours. If the craft is not too unusual I can use a spreadsheet program to get reasonable results in minutes, perhaps even while the client is on the phone.

Alternatively, suppose I have received a design from somewhere, and am asked to evaluate it. I could go through a lengthy "reverse engineering" process to check the design values. Or I could try, but I'd probably find I didn't have enough information to start with.

Instead I use my simplified equations. Knowing only cushion pressure and craft displacement I can make a quick estimate of resistance, and can see whether the power levels are reasonable. I look at the relation between lift power and displacement and I wonder whether there's enough air flow. Or perhaps that all "checks out" up until I calculate weights, when I see that the original weight estimate is unreasonably low.

Those two examples suggest the utility of this collection of equations. I offer this collection to the rest of the community in the hope that together we can better serve our customers and our world.

FIELD EQUATIONS

Resistance

This section presents formulæ for predicting resistance. It begins with the resistance of displacement catamarans (as these are SES off-cushion) and subsequently includes three simple methods for prediction of SES on-cushion. These three methods vary in complexity and, as a result, in reliability. Comments in the text indicate the limits of reliability for each technique.

Also included is an estimator for the added drag due to waves.

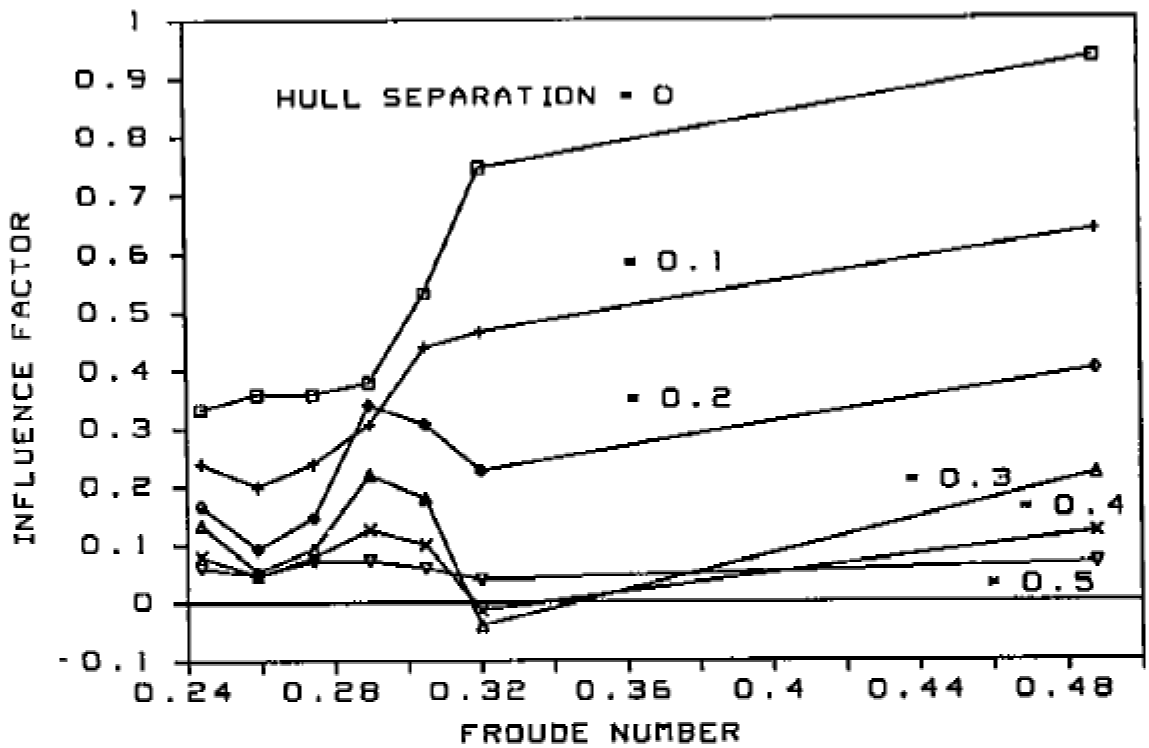

Figure 1 This figure reproduces the data presented by Turner & Taplin in 1968. "Influence Factor" is the drag augment due to hull spacing. "Hull Separation" is the clear distance between hulls expressed as a fraction of length.

Displacement Catamarans / SES Off-Cushion

I have a very simple Lotus-1.2.3 application spreadsheet which I use to predict the resistance of displacement catamarans. This simple spreadsheet is based on the following formulation:

$$C_T = 2 \cdot ((1+IF) \cdot C_R + C_f) \quad (1)$$

where IF is a catamaran interference factor derived from Turner & Taplin, 1968. This factor is plotted in Figure 1 as a function of hull spacing and Froude number. Hull separation is the clear distance between the hulls at the waterline divided by the length.

Anyone using the data presented in Figure 1 is advised to also look at Turner & Taplin's curves as originally published. They include the data points on which the curves are based and this gives some feel for their accuracy. There is considerable scatter in the original data and the accuracy of this method suffers in consequence. Note also the large gap between Froude numbers of 0.32 and 0.48. Filling this gap with straight lines is obviously incorrect.

Equation (1) calculates the demihull's residuary resistance, augments it by the influence factor, adds the frictional resistance, and multiplies the whole by two to make a catamaran.

I predict C_R using a curve based on Taylor's series and published by Skene, 1942, see Figure 2. The friction term C_f is predicted using the ITTC formulation (C_f=.075/(log Rn^2-2). In the spreadsheet I use a very simple wetted surface estimator ($S=0.5L \cdot B \cdot \pi$ = the surface of a half cylinder of same length and beam) but I have a multiplying factor so that this value can be adjusted to equal the (externally calculated) exact wetted surface.

The limitation of this spreadsheet is in its calculation of the demihull residuary drag C_R. My formulation for C_R depends only on Froude number and displacement-length ratio. There is no dependance on the obviously important parameter of prismatic coefficient, or on any of the secondary parameters such as transom immersion or entrance angle, etc.

To help correct this deficiency I use a "worm curve." A "worm curve" is a vector of correction factors which are applied to the C_R. They can be derived from a known drag curve by running the drag routine with all worm values set to 1., and then adjusting the worm values as necessary to get the correct C_R.

Use of a "worm curve" lets me use one hull as the parent for another. Significant changes in hull spacing or other values are then accounted for via the interference factors or other key parameters.

SES - Extremely Simple Method

Based on analysis of a variety of built SES, I find:

$$SHP=20.25 \cdot Disp \cdot Fn^{1.45} \quad (2)$$

But results nearly as good can be obtained using the easy to remember values of $SHP=20. \cdot Disp \cdot Fn^{1.5}$. This makes a handy formula that can be kept in one's head for giving quick off-the-cuff responses to questions. Both of these curves and the data points that support them are shown in Figure 3.

Please note that use of this equation requires that the L/B and other parameters of the SES be appropriate to the design Froude number. In effect this equation predicts the resistance of a "properly designed" SES. It is up to the user to ensure that his vessel is "properly designed."

SES - Simplified Method

I have collected a number of data points which suggest that the resistance of an SES can be related to the fraction of the vessel's weight carried by the cushion. Analysis of a variety of craft model-test results yields the following set of equations:

Fn= .329	R'= 29.5	45 < %c < 80	(3a)
Fn= .374	R'= 36.	45 < %c < 80	(3b)
Fn= .549	R'= 95.72-.787%c	45 < %c < 80	(3c)
Fn= .659	R'= 113.2-.888%c	55 < %c < 80	(3d)
Fn= .768	R'= 147.1-1.23%c	55 < %c < 80	(3e)

R' is the specific resistance in kg/t displacement, %c is the percent of vessel weight carried by the cushion (a number between 45 and 80.) The limits of applicability are given in the third column. The data points and the least-squares lines are shown in Figure 4.

These equations were derived from model tests of craft with fairly different hull forms, but all of generally similar L/B (about 5). Thus they will not be accurate for craft of different L/B. However, where the designer has one or two data points for a craft of different geometry, he may use those data points to linearly modify the field equations. The modifying equation would be $R'_{REAL}=k \cdot R'_{MCKESSON}$ where k is a function of Froude number.

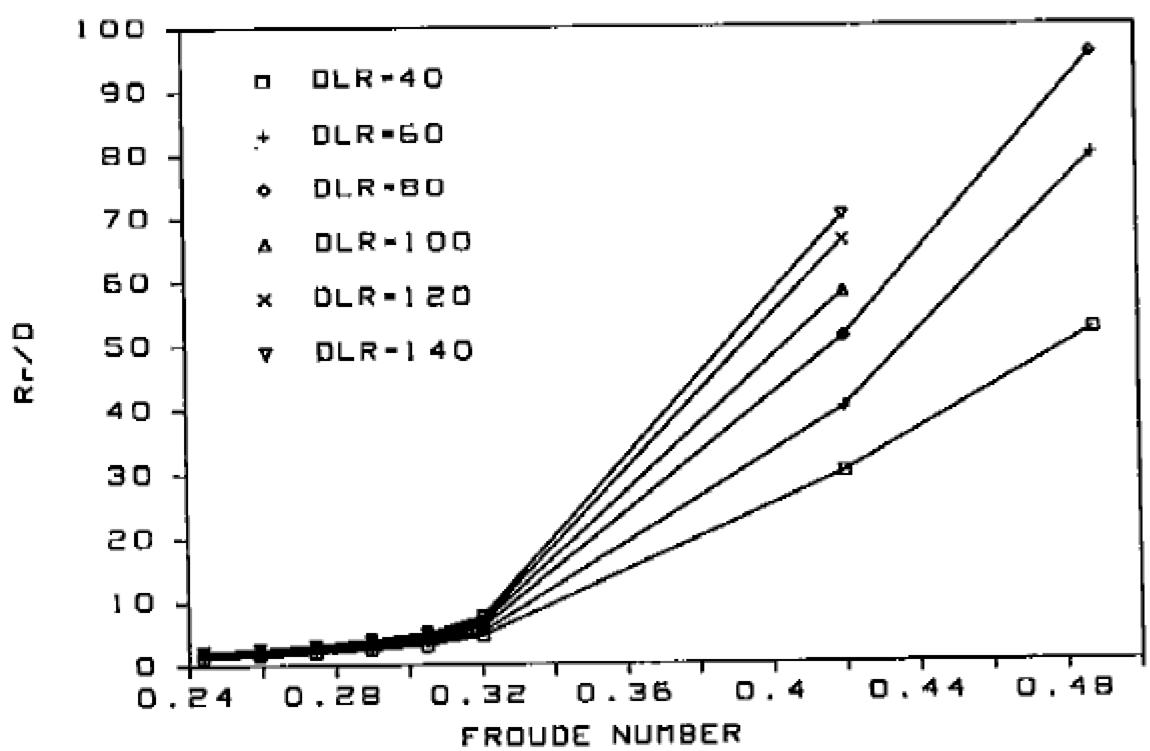

Figure 2 This figure reproduces Skene's version of the Taylor series data. DLR is the displacement/length ratio. The drag is expressed as pounds of residuary drag per ton displacement.

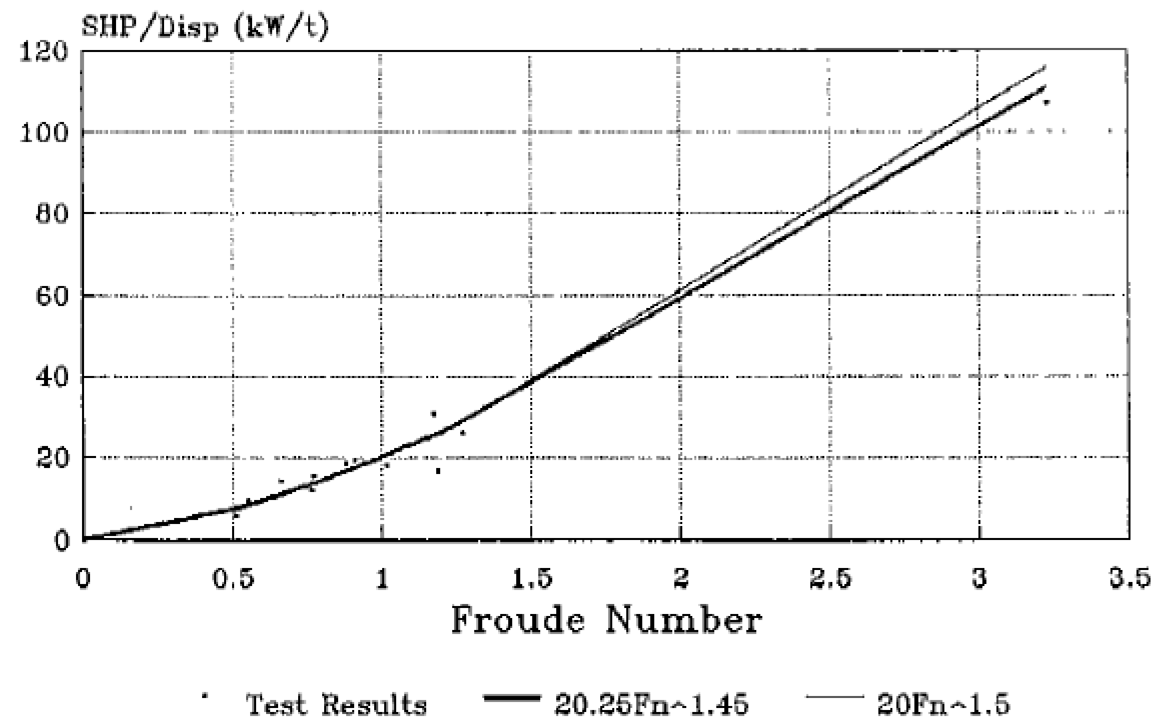

Figure 3 This figure shows the fit of Equation (2) through a database of known SES performance.

SES - Reliable Method

This method allows one to use data from one ship to estimate the resistance of another. The method accounts for variations in L/B, cushion pressure, aerodynamic area and coefficient. The technique assumes that:

$$C_T = C_{DELTA} + C_f + C_{DOCTOR} + C_{AIR} \quad (4)$$

The breakdown is depicted in Figure 5. C_T is known for the parent ship. C_f, C_{DOCTOR} and C_{AIR} can all be calculated (one must be careful what is assumed for wetted surface.) Thus for a given ship one can calculate a curve of C_{DELTA} versus Froude number.

The trick lies in knowing (or guessing) how to extrapolate the C_{DELTA} from the parent to the offspring. We know that C_{DELTA} includes, at least, sidehull wavemaking and seal friction. We do not know in what proportions these two components are present.

I use two different assumptions, and produce two different drag curves. These two curves bound the range of likely drag values.

The first assumption is that C_{DELTA} is totally sidehull wavemaking. For this case the spreadsheet then assumes that C_{DELTA} can be "lambda-cubed" to give the new ship's value. It calculates new values of the other three components and adds them on.

The second assumption is that some part of the C_{DELTA} is frictional in nature. Its handling is more complex:

The seal friction itself is handled by augmenting the wetted surface of the final hull by some factor and getting a thus-increased frictional drag. I estimate the necessary augment by the following logic: First, I assume that the augment is not a function of speed, but is a fixed number of square feet of wetted surface.

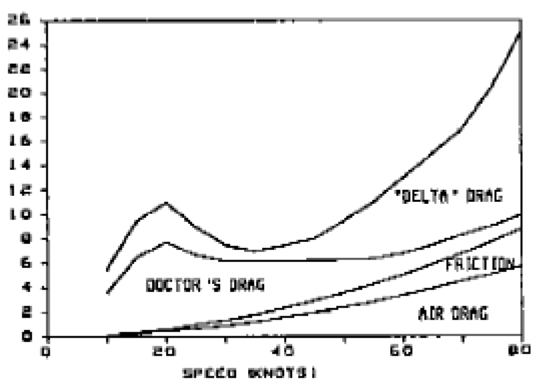

Figure 4 This plot graphically depicts the drag breakdown used in the drag prediction method called "reliable." The key to this technique is the definition and scaling of the component called "Delta" drag.

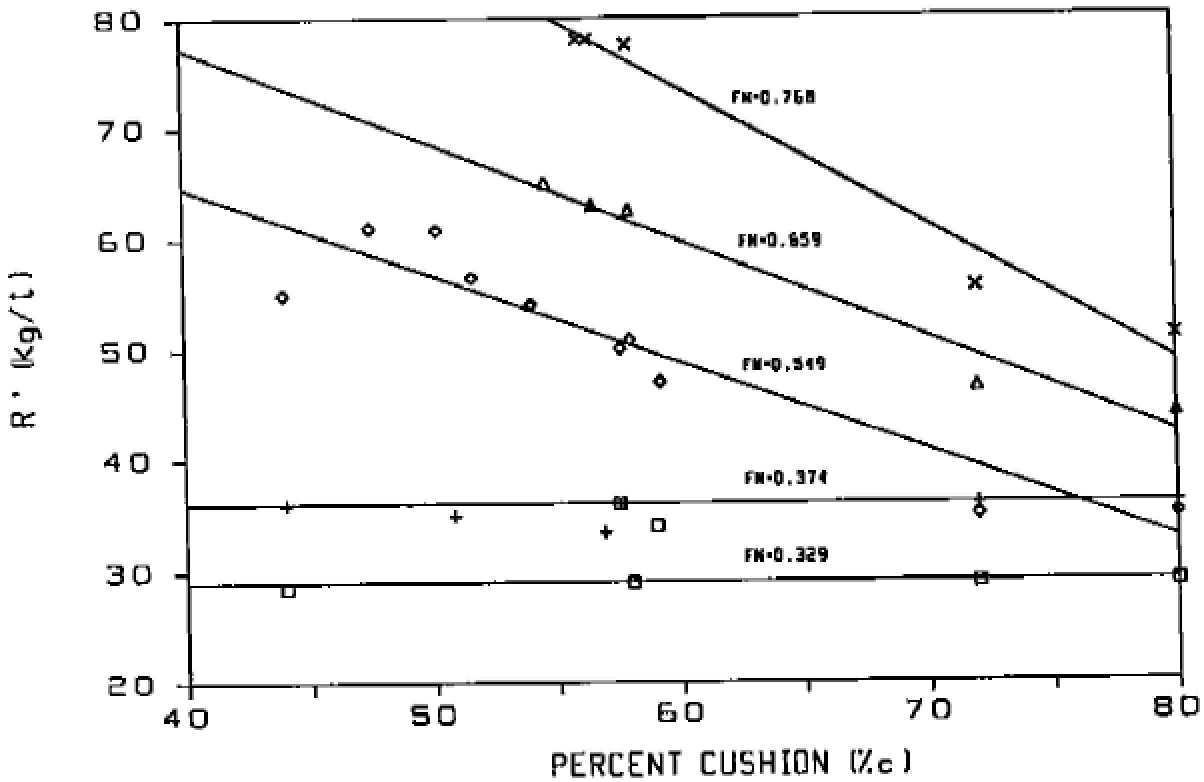

Figure 5 The Simplified Method (Eq'n 3a-e) for SES drag assumes that resistance can be related to the percent of the craft's weight that is carried by the cushion. The lines are fit through experimental data. The equations are in the text.

Since we know that the sidehull wavemaking drag can never be less than zero, this gives us a maximum value for the wetted surface augment. This logic is depicted in Figure 6. At each point in the C_{DELTA} curve I calculate the friction augment that would completely consume the C_{DELTA}. The *lowest* of these friction augments is the *largest* friction augment that complies with my assumptions. I merely select it and use it.

The result of this method is a drag curve presented as two lines, an upper and a lower bound, which gives a good workable drag prediction and a measure of the confidence in that drag prediction at the same time. (See Figure 7.)

SES - Added Drag Due to Waves

Following is an equation for the added resistance for an SES operating in waves. I don't know where it came from. It works only in english units.

$$R_W = 15.3 \cdot 2240 \cdot Disp \cdot SSD \cdot H_W \cdot CSAR/360 \qquad (5)$$

where: $SSD = .4138 \cdot V_F - .0067 \cdot V_F^2 + .4655$

$$V_F = V_C \cdot (5.099/(L_C \cdot B_C)^{.5})^{-1}$$

Propulsion

In this section I address only equations for predicting propulsive efficiencies or machinery sizes. Propulsion plant weight is given in the weight section of the paper.

Propulsive Efficiency

In 1989 Mr. Svensson published a collection of data points for water jet propulsion efficiency. I have taken the simple step of fitting a line through these points and developing an equation for it. The result is:

$\eta_{WJ} = .00681 \cdot V_k + .41076$

Comparing this formula with the points published by Svensson suggests that the formula is accurate within ±6% at 40 knots and ±8% at 18 knots. Svensson's figure is reproduced, with this line on it, as Figure 8.

Engine Geometry

I needed a set of equations for predicting the size of a diesel engine, so that I could see whether it would fit in the ship's sidehulls. I performed a very simple analysis of the 1990 MTU catalog and established the relationships given here. The data is presented in Figures 9, 10, and 11.

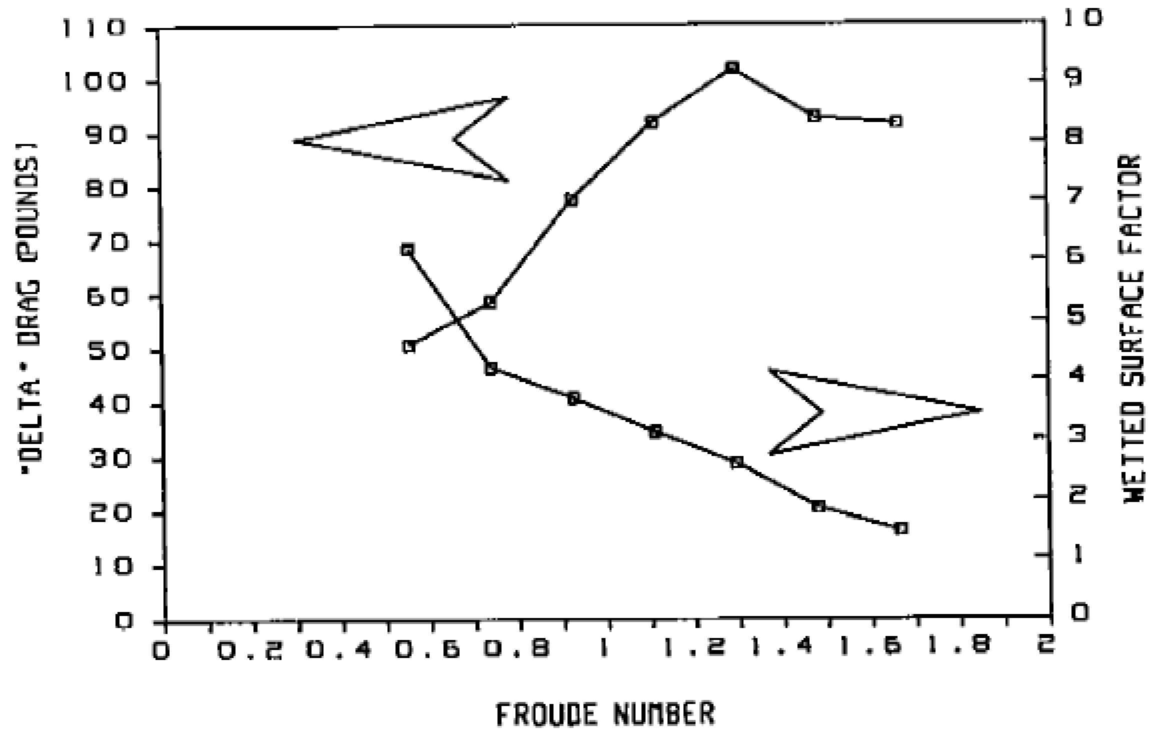

Figure 6 The ascending curve shows the variation of the "DELTA" drag with speed (see Eq'n 4). The descending curve shows the wetted surface augment that would produce the same drag.

For all engines in "Application Group 1DS":

L_g= 1.842m + .000613 m/kW (7a)
W_g= 1.441m + .000053 m/kW (7b)
H_g= 1.348m + .000327 m/kW (7c)
Wt_g= -.835t + 3.79 kg/kW (7d)

The smaller engines don't fit the trends as well, so I broke them out separately. This is particularly important as these are likely to be the lift engines.

For all "1DS" Engines less than 2 MW:

L_g= 1.284m + .001016 m/kW (8a)
W_g= 1.4m + .000060 m/kW (8b)
H_n= 1.m + .000517 m/kW (8c)
Wt_g= .842t + 2.242 kg/kW (8d)

SES Lift Systems

Definitions

The first problem in design of the lift system is what coefficient system one should use. There are a variety of different formulations for fan coefficients. I use those used by the French fan manufacturer NEU:

Cp ≡ Pressure coefficient = P/U^2
Cd ≡ Flow coefficient = Q/U·Su

where:

U ≡ tangential velocity
Su ≡ $\pi \cdot D_p^2/4$

These terms can be re-arranged to give the following relationships:

$U = \sqrt{(P/Cp)}$ (9)
$Su = U \cdot Cd/Q$ (10)
$D_p = \sqrt{(4 \cdot Su/\pi)} = \sqrt{((4 \cdot U \cdot Cd)/(\pi \cdot Q))}$ (11)
$N = U/D_p$ (in rps) (12)

Those, in turn, give us a set of fan extrapolation laws:

$Q' = Q(N'/N)$ (13)
$P' = P\sqrt{(N'/N)}$ (14)

Required Air Flow

The two key characteristics of an SES lift system are the cushion pressure and the air flow. The cushion pressure is easy to estimate. The air flow is less so.

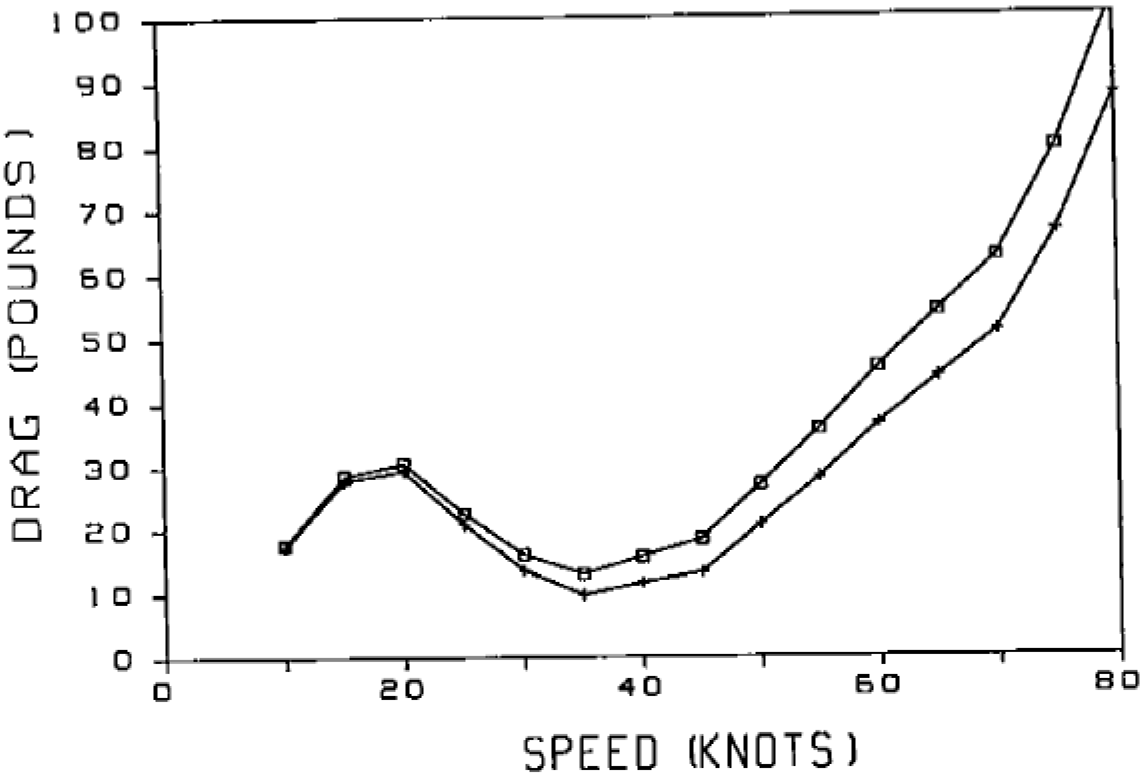

Figure 7 This figure shows the upper and lower bounds on drag produced by the "DELTA-Drag" scaling technique. The true drag of the craft lies somewhere between the two lines.

I have three methods which I use, using each to check the others. In each case the equations are presented with a coefficient "k" which you will have to determine for yourself.

The first, and simplest, is to extrapolate from a known operating point according to the change in pressure and in cushion beam. This assumes that the air flow depends linearly on the product of cushion beam (which is the escape area) and pressure to the 1.5 power (which is the energy behind the air's escape.)

Thus, to extrapolate from a vessel with known Q, P, and B, one says:

$$Q'=Q\cdot B'_C\cdot P'^{1.5}_C/(B_C\cdot P_C^{1.5}) \tag{15}$$

There is a second formulation which was derived from hovercraft work. I believe a variant of this formula to be in use at British Hovercraft Corporation. This formula allows for some leakage under the sidehulls (consistent with its hovercraft origins), as will be seen:

$$Q'=k\cdot Q\cdot B_C\cdot P_C^{0.5}\cdot S_C^{0.5} \tag{16}$$

The third method is the wave-pumping method. This formula considers the ocean's waves as pistons which "pump" the cushion and determine the air demand. Note that according to this method no airflow is required in calm water.

$$Q=k\cdot B_C\cdot H_W\cdot(V_C+V_W) \tag{17}$$

V_W=wave speed=$(g\cdot T_0)/2\pi$
V_C=craft speed in m/s (or ft/sec)
H_W=wave height (m or ft)
T_0=wave period

For the necessary wave characteristics one may use the NATO sea state definitions (see Table 1) and I assume $T_0=\sqrt{2}\cdot T_{MODAL}$.

Table 1 - NATO Standard Sea State Definitions

Sea State Number	Significant Wave Height Range	Mean	Modal Wave Period Range	Most Probable	T_0
0 - 1	0-0.1	0.05			
2	0.1-0.5	0.3	3.3-12.8	7.5	10.6
3	0.5-1.25	0.88	5.0-14.8	7.5	10.6
4	1.25-2.5	1.88	6.1-15.2	8.8	12.5
5	2.5-4	3.25	8.3-15.5	9.7	13.7
6	4-6	5	9.8-16.2	12.4	17.5
7	6-9	7.5	11.8-18.5	15.0	21.2
8	9-14	11.5	14.2-18.6	16.4	23.2

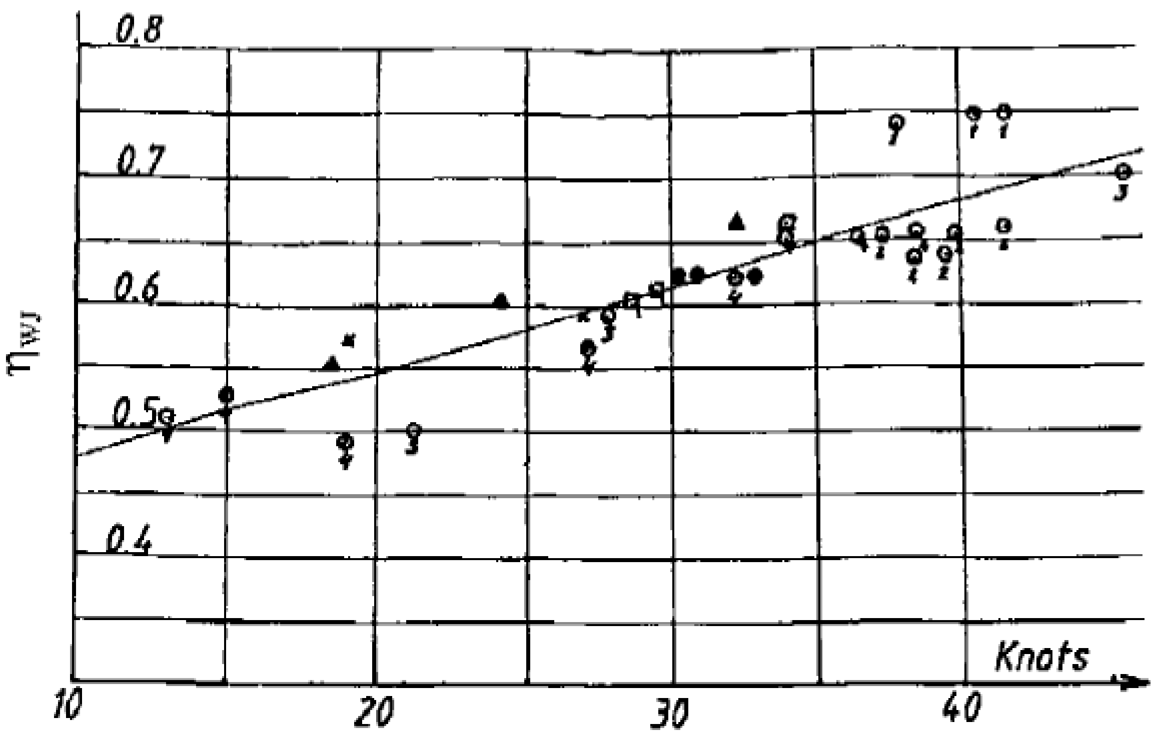

Figure 8 Svensson's data, from his 1989 paper, with a straight line least-squares equivalent.

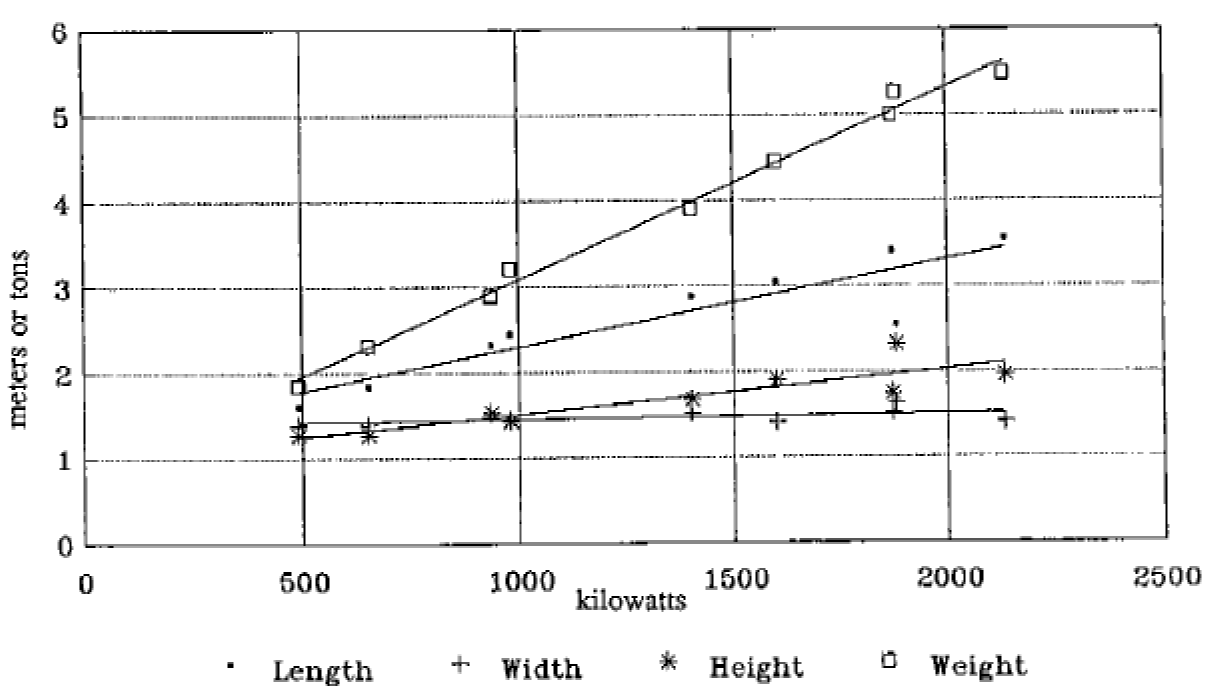

Figure 9 The principal characteristics of the smaller MTU diesels (under 2 MW).

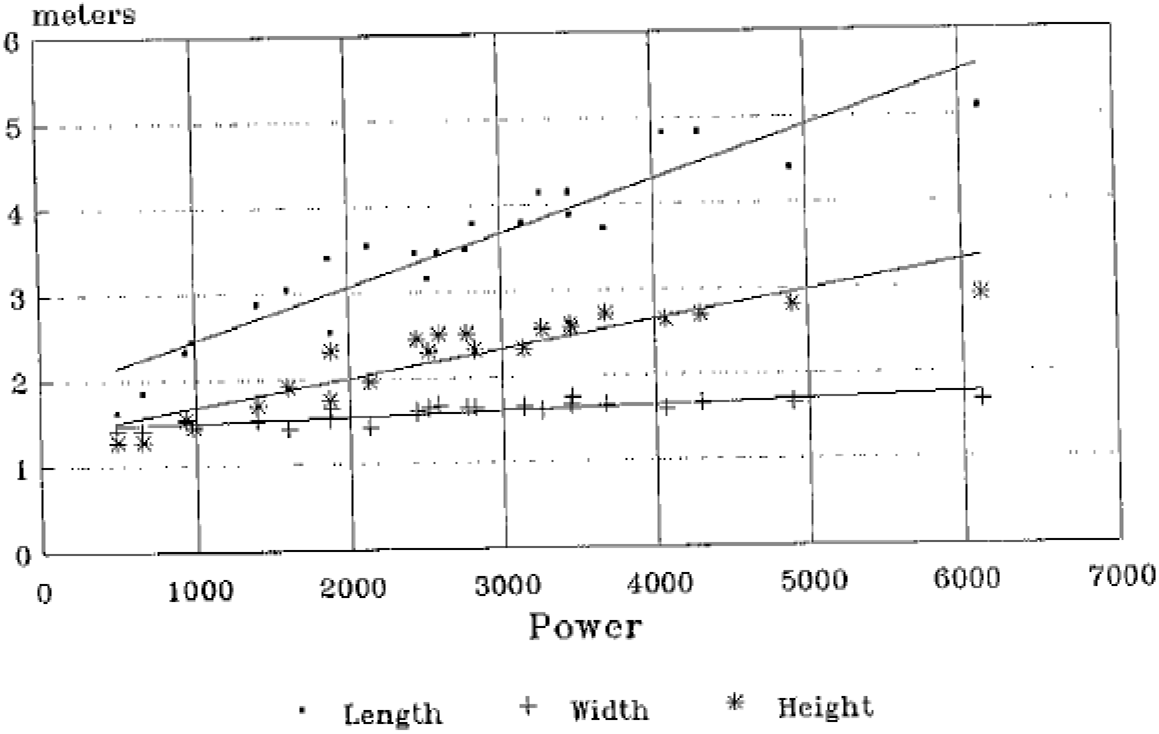

Figure 10 The principal characteristics of the full range of MTU diesels.

Weight

Structure

I use two main parameters for establishing first-cut structural weights. Courtesy of the experience of Dr. Colen Kennell I am convinced of the validity of estimating structural weight based on volume. The underlying assumption is that similar ships (not necessarily of similar size) will have similar structural densities. This, in turn, implies that as one scales up the ship in size one increases plating thickness linearly. An argument can be made that this is not correct, and that plating thickness should vary either faster or slower than the displacement. The general form of this equation would then be:

$$W_{100} = k \cdot Vol^{x} \qquad (18a)$$

In Eq. 18a the subscript "100" denotes the structural weight group. I am accustomed to using the US Navy's SWBS weight groups, and my subscripts correspond thereto.

I have analyzed my database of structural weights and I find that the value of "x" is remarkably close to 1. Thus, while acknowledging that I am giving up a little accuracy, I prefer the much simpler formulation:

$$W_{100} = k \cdot Vol \qquad (18b)$$

The next question is "what is Vol?" Ideally one would like to use the total enclosed ship volume, and this has been Dr. Kennell's practice. However for first-cut estimations I prefer to take a page from Harry Benford's work and use a pseudo-volume based on cubic number.

In monohull parlance the cubic number is LBD/100. For an SES this is inadmissable as it ignores the cushion. I use:

$$Vol = LBD - LB_C D_C \qquad (19)$$

Where L, B, and D, are length, beam and depth, and the subscript "C" means that I am taking the cushion dimensions. Thus "Vol" is the volume of a prism circumscribing the hull, minus the volume of a prism circumscribing the cushion, see Figure 12.

I take D as to the uppermost continuous deck, and I accept the cushion nominal dimensions, measured at the keel plane. Note that I don't use the cushion length. I assume the cushion extends the full length of the ship.

Using all of these definitions I arrive at a structural density of 40 kg/m^3 for aluminum structure. I use the same value for GRP sandwich structure, and multiply by 1.5 for steel structure.

It is interesting to try to make physical sense of this value of 40 kg/m^3. This implies that a one-meter aluminum cube will weigh 40kg. To do so it would have to be made of material 2.5mm thick (a two-meter cube ends up 5mm thick). These values seem reasonable and consistent with good lightweight practice.

There is an alternative formulation that was in use by Mr. David R. Lavis of Band, Lavis and Associates, Inc. Given that obtaining even rudimentary values of cushion dimensions or ship depth is often difficult, David has found good results can be

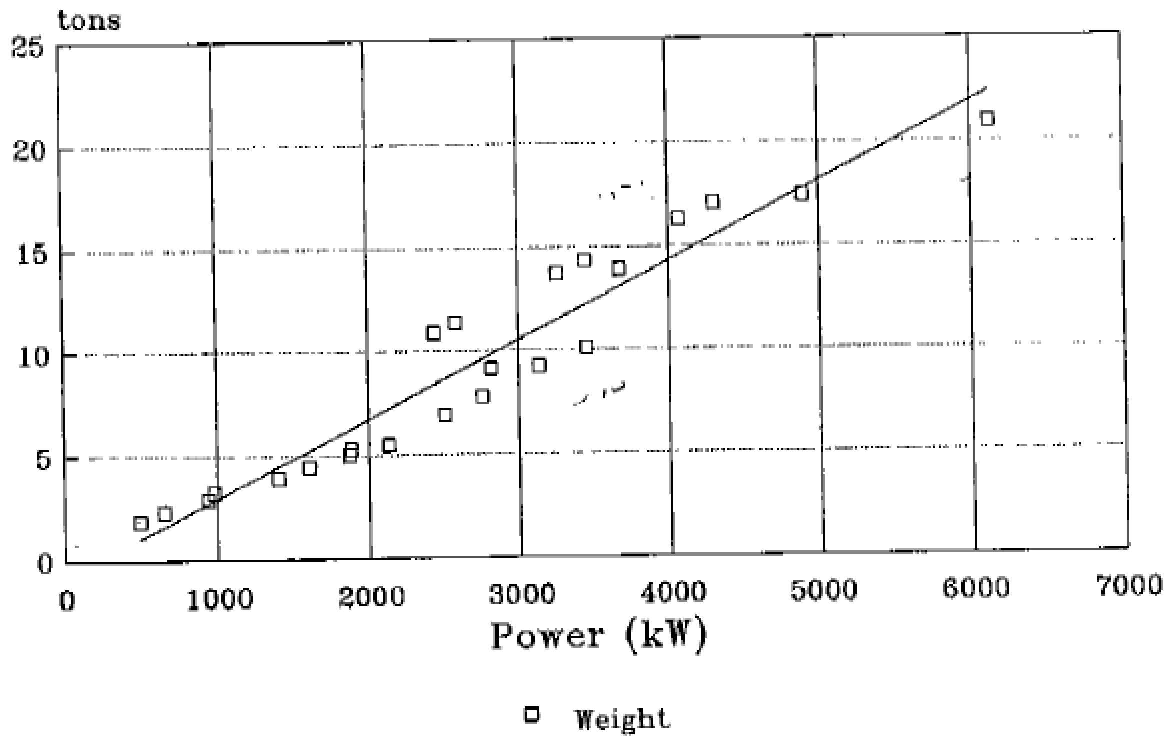

Figure 11 Power versus weight for the range of MTU diesels.

obtained from a volume which is merely $(L_{OA}B_{OA})^{1.5}$. Mr. Lavis has a collection of data points and trend lines based on this type of analysis, but I shall not present them here due to the proprietary nature of the information. I can, however, translate my value of 40 kg/m³ into his scale of measurement and tell you that my database yields 5.36 kg/m³ *à la* Lavis.

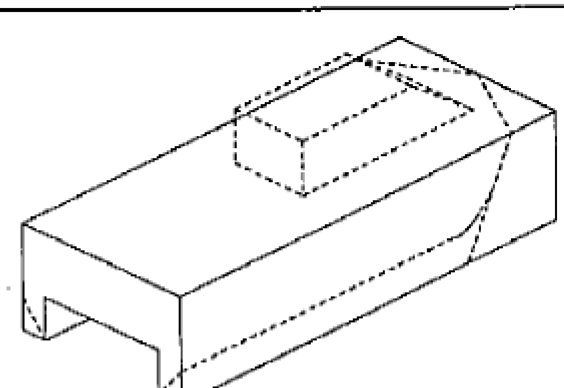

Figure 12 This figure shows the definition of "SES Cubic Number." This pseudo-volume is the volume of a rectangular prism that circumscribes the hull, minus the volume of a prism circumscribing the cushion.

Propulsion

For a lift and propulsion system based on MTU diesels and KaMeWa waterjets, I have found the following equations to give dependable results:

Propulsion Engines = 5.69 kg/kW (20)
Includes Engine, intakes, exhausts, lube oil system, cooling water system, and fuel supply system.

Transmission = 5.11 kg/kW (21)
Includes reduction gears, couplings, shafts, propulsors, machinery space ventilation, shaft lubrication systems, auxiliary cooling water circuits, machinery control systems, propulsion control electric cable.

Waterjets = 0.0182 kg/(kW$^{1.5}$) (22)
(This weight is included in the 5.11 kg/kW listed above, for first approximations. For more detailed analysis I prefer the formulation given here. I use power to the 1.5 based on the following logic: Weight varies as pump diameter cubed. Pump diameter varies as pump disc area to the one-half. Pump disc area varies as power for a constant speed. Thus weight varies as power to the 1.5.)

Conclusions

I have presented in this paper a set of field equations for SES design. While these equations are only approximations, I have found them to be useful tools for rapidly iterating a design. I offer them to the community in the hope that this will encourage non-AMV naval architects to include AMVs in their trade-off studies. I believe that such inclusion will directly lead to the selection of the SES form for an increasing range of missions.

ACKNOWLEDGEMENTS

Amelio d'Arcangelo	Who first introduced me to parametrics.
Harry Benford	Who introduced me to the cubic number.
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STCAN	Who provided the opportunity to develop and check these relationships.
BLA	Who have provided me some of the formulations.
Mike Yermakov	Who developed some of the Lotus-1.2.3 applications.

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NOMENCLATURE

B	Beam (at hullborne waterline)
B_C	Cushion beam
B_{OA}	Beam over all
C_{AIR}	Air resistance coefficient $R_{AIR}/(.5\rho_W S V_C)$
Cd	Fan Flow coefficient=Q/U·Su
C_{DELTA}	An auxiliary drag coefficient defined in the paper.
C_{DOCTOR}	Doctor's wave drag coefficient for drag of an air cushion moving over a free surface.
C_f	Frictional resistance coefficient= $R_f/(.5\rho_W S V_C)$ May be taken as equal to: $0.075/(\log Rn^2-2)$
C_p	Fan Pressure coefficient=P/U^2
C_R	Residuary resistance coefficient= $R_R/(.5\rho_W S V_C)$
C_T	Total resistance coefficient= $R_T/(.5\rho_W S V_C)$
D_F	Fan diameter
D	Depth - measured from keel to uppermost deck that continues over 50% of ship length.
D_C	Cushion depth
Disp	Craft displacement in metric tons
Fn	Froude number based on cushion length.
H_E	Engine height
H_W	wave height (m or ft)
IF	Catamaran influence factor. Represents the increase in resistance due to 'tween-hull interference as compared with a ship of infinite hull spacing.
kg	kilograms
L	Length between perpendiculars
L_C	Cushion length
L_E	Engine length
L_{OA}	Length over all
N	Fan rotation rate (in rps)
P	Pressure produced by a fan
Q	Air flow rate, m^3/sec
R'	Specific resistance: Total hydrodynamic drag divided by ship displacement. Kilograms per ton (kg/ton)
Rn	Reynolds number, generally based on waterline length or cushion length.
S	Wetted surface
Su	A fan reference area equal to $\pi \cdot D^2/4$
SHP	Shaft power in Kilowatts
t	tons (metric)
T_0	wave period $=\sqrt{2} \cdot T_{MODAL}$.
T_{MODAL}	wave modal period.
U	Fan rotor tangential velocity
Vol	LBD-LcBcDc
V_C	Ship speed in m/s (or ft/sec)
V_K	Ship speed in knots
V_W	wave speed=$(g \cdot T_0)/2\pi$
W_E	Engine width
Wt_E	Engine weight, as given in manufacturer's catalogue.
%c	Percentage of craft weight carried by the cushion.
η_{WJ}	Propulsive efficiency of a water jet. Does not include mechanical efficiency of transmission system. Does include hull interactions and other hydrodynamic effects.
ρ_W	Density of water.
ρ_A	Density of air.