Mass Estimation and Hull Form Design

Contents

2.1 Lightship Mass Estimation

2.1.1 Introduction

Lightship estimation is important in the ship design process; integral to the design of deadweight carriers to ensure that the required payload can be carried so that the available displacement balances the deadweight and lightship for the selected dimensions and form. For capacity carriers it is still vital that for any proposed design that this balance is still achieved at the design waterline.

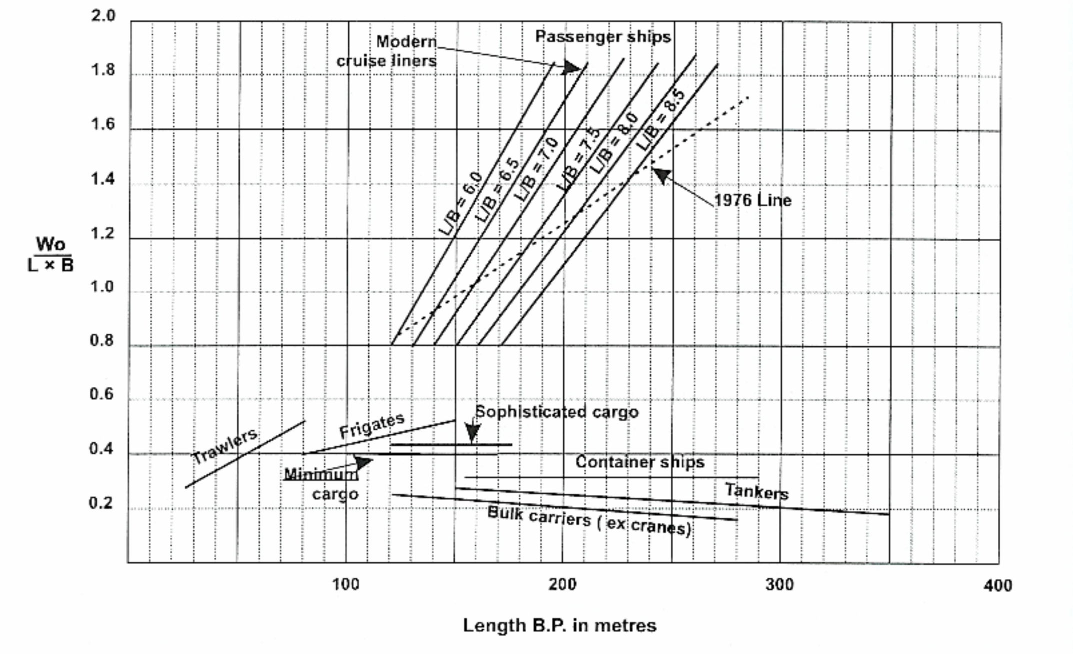
2.1.2 Outfit Mass

Outfit can be considered to include: Hatch covers, Cargo handling equipment, Equipment and facilities in the living quarters (such as furniture, galley equipment, heating, ventilation & air conditioning, doors, windows & sidelights, sanitary installations, deck, bulkhead & deckhead coverings & insulation and non-steel compartment boundaries) and Miscellaneous items (such as anchoring & mooring equipment, steering gear, bridge consoles, refrigerating plant, paint, lifesaving equipment, fire fighting equipment, hold ventilation and radio & radar equipment).

The majority of outfit weight items can be considered to be proportioned between similar ships on the basis of Deck Area i.e. using a square number approach where outfit weight is assumed proportional to the product of L and B. The square number method is applied as follows, having established a suitable value of $W/(LB)$ from basis data, such as the Watson chart.

$$W_{DESIGN} = W \frac{L_D B_D}{LB}$$

The estimate can be further refined by considering whether known weights are common to both ships, or are present only in the basis ship and not in the new design (e.g. cranes). The value of W for the basis can be corrected accordingly before W_{DESIGN} is calculated and the known item added back if necessary. Once a more detailed breakdown of the outfit weight of the basis ship is available then more refined methods can be applied to each part.



[1]

2.1.3 Influences on Outfit Mass

Comparing the original Wood and Outfit chart and the revised chart (above) compiled by Watson allows certain observations to be made:

- For some vessels *W/(LB)* remains constant (e.g. general cargo and container ships).
- For other vessels *W/(LB)* decreases with length (e.g. tankers and bulkers) as outfit such as accommodation only varies slightly with size.
- The gradient for modern passenger ships is higher than originally observed for such vessels as there has been an increase in the number of decks for given length.
- The increase in decks tends to facilitated by a corresponding increase in beam. The gradient is steepest for lower *L/B* (~ 6). For higher *L/B* the gradient is still steeper than the original Watson plot.
- Similarly the line for warships and trawlers also exhibit a higher gradient.

Other useful methods include:

Henschlde's Method (1965), appropriate for dry cargo vessels and coastal motor ships for stowage rate range of 1.2 to 2.4 $m^3/tonne$.

$$W_O = \frac{0.07(2.4 - \nabla_{CARGO}/W_{PAYLOAD})^3 + 0.15\nabla_{CARGO}}{-1 + \log\nabla_{CARGO}}$$

Where ∇_{CARGO} = total cargo space volume (m^3) and $\nabla_{CARGO}/W_{PAYLOAD}$ is stowage rate.

Weight of cranes should be accounted for separately.

Coefficients (Schneekluth) Method (1987)

$$W_O = K(L \times B)$$

K values are given for a range of different ship types or can be calculated from a suitable basis, e.g. container ships $K \approx 0.34 \sim 0.38\, tonnes/m^3$.

2.1.4 Machinery Weight

Machinery weight comprises; main engine(s), gearbox (if fitted), bearings, shafting, propeller(s), generators, switchboards, cabling, pumps, valves, piping etc. The fundamental parameter by which machinery weight can be proportioned is the installed power of the main machinery, conventionally taken as Shaft Power, P_S. A first estimate of P_S from a basis ship of similar dimensions and speed can be found from:

$P_S \propto \Delta^{2/3} V^3$ or $P_S = K\Delta^{2/3} V^3$ where K is found from the basis vessel.

Having established P_S for the design by approximate or more detailed calculation, it is possible to assume, given main engines mass is the biggest proportion of total machinery mass, that:

$$W_{M/C} \propto P_S^{2/3}$$

Watson & Gilfillan analyzed a simple two-group breakdown of machinery weight for a range of vessel types:

- The main engine;
- The remainder of the machinery installation.

By studying engine manufacturers' data over a wide range of engine type they could express the bare weight in the form:

$$W_{ENGINE} = 9.38\left[\frac{MCR}{RPM}\right]^{0.84}$$

Where MCR = Maximum continuous rating (metric horse power) and RPM is the revolutions at the quoted MCR.

For more than one engine the result for each is simply summed (n = number of engines) i.e.

$$W_{ENGINE} = \sum_{i=1}^{n} 9.38\left[\frac{MCR}{RBM}\right]^{0.84}$$

For a given MCR the higher the RPM then the lower the torque the engine must produce. The lower the torque, the smaller are the forces produced inside the engine and hence smaller are the components engine and the lower is the engine weight.

The weight of the remainder of the machinery is given by:

$$W_{REMAINDER} = K(MCR)^{0.7}$$

Typically K= 0.56 for bulk carriers and general cargo ships, 0.59 for tankers and 0.65 for passenger ships and ferries. Note K increases due to additional weight for pumps and the demand for hotel services respectively.

For diesel electric plants with a central station concept producing all needed electrical power

$$W_{M/C} = W_{ENGINE} + W_{REMAINDER} = 0.72(MCR)^{0.78}$$

where MCR is total capacity of all generators in kW.

Other useful methods include:

Scheekluth Method

$$W_{M/C} = C \times W_{MAIN\ ENGINE}$$

C typically in range 2.2 to 3.6 and can be obtained form a suitable basis vessel. Alternatively:

$$W_{M/C} = a(P_{SHP}/1000)^{0.57}$$

a is a coefficient dependent on whether direct drive or geared.

2.1.5 Contemporary Influences on Machinery Mass Estimation

- Increasing size of slow speed diesel engines for larger installed power applications replacing twin engine and other machinery types (e.g. ultra large container carriers of 6000 to 13000 TEU).
- All electric propulsion providing central generation of all propulsive power and hotel services supply.
- Novel propulsion devices (e.g. pods) and hybrid designs.
- Replacement of hydraulic control systems with electrical ones.
- Replace with more accurate methods once more information is available, especially for innovative arrangements.

2.1.6 Steel Mass Estimation

Steel mass is considered under two categories: net and gross steel. *Gross* steel is the steel bought into the shipyard for a particular ship (invoiced steel) and *Net* steel is the steel built into the ship. The relation ship between the two is:

$$Net = Gross - Scrap$$

The scrap allowance changes with respect to ship type, fullness and the complexity of the structure, e.g. a warship would typically have a greater scrap allowance than a bulk carrier with greater fullness and less complex structure. For a warship it may be more than a 15% deduction from gross steel and for a bulk carrier only around a 6% deduction from gross steel.

Net steel mass is of importance with respect to steel mass estimation for design purposes but obviously the corresponding gross steel is required for tendering purposes.

Steel mass includes all plates and sections forming shell, outer & inner bottom, longitudinal girders, decks, bulkheads, superstructure, deckhouse, seats for equipment & auxiliaries as well as forgings & castings for stem, stern frame, rudder stock, shaft brackets etc.

There are a number of ways of calculating steel mass. The methods given here concentrate on methods suitable to estimate steel mass at the early stages of design where weight estimate is vital and before steel plans exist for the ship or even the midship section is proposed as a basis for steel mass estimation.

2.1.7 Surface and Cubic Numeral Methods

Surface Numeral takes the form:

$$SN = L(B + D) \quad .$$

This method assumes that steel mass is proportional to *SN*, i.e.

$$Steel\,mass \propto L(B + D)$$

Dimensionally *SN* is proportional to λ^2 or surface area. Therefore effectively the steel mass estimate is based on change in surface area that implies the scantlings (thickness) remains constant with changing dimensions.

This is clearly not the case and suggests such SN methods underestimate steel mass.

Cubic numeral takes the form:

$$CN = LBD \quad .$$

Here steel mass is assumed proportional to *CN*, i.e.

$$Steel\,mass \propto LBD$$

Dimensionally *CN* is proportional to λ^3. This can be considered as:

$$\lambda^2 \times \lambda \text{ or surface area} \times \text{scantlings.}$$

In this case it suggests that scantlings (thickness) changes at the same rate as changes in dimensions λ.

Again, this is clearly not the case and suggests such *CN* methods overestimate steel mass.

This simple approach suggests that a better estimate would results from a method where:

$$Steel\,mass \propto \lambda^y \text{ where } 2 < y < 3 \quad .$$

A simple approach would be to simply take the mean of the *SN* and *CN* methods such that:

$$Steel\,mass \propto LBD + L(B + D) \text{ or } Steel\,mass \propto \lambda^{2.5} \quad .$$

This can provide a better estimate but still tends to underestimate slightly.

There should be an additional correction to account for change in fullness:

$$Fullness\ correction = \frac{Steel\,mass}{2} \times \delta C_B{'}$$

where

$$C_B{'} = C_B + \frac{(1 - C_B)(0.8D - T)}{3T} \text{ for both basis and design.}$$

2.1.8 Watson's Method for Steel Mass Estimation

Steel mass for a number of ships, of similar type, are reduced to a standard form having a block coefficient of 0.7 at 0.8*D*. The data is then plotted on a base of Lloyds equipment numeral where Lloyds equipment numeral, *E*, is in the form:

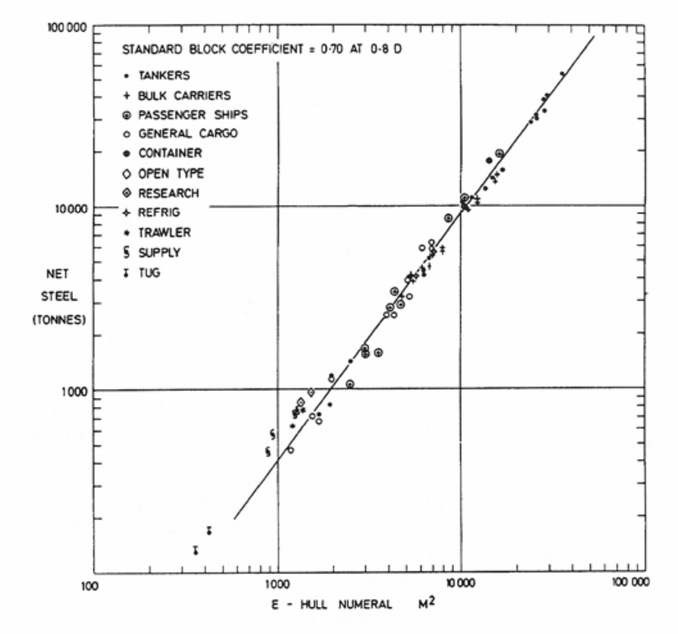
$$E = L(B+T) + 0.85L(D-T) + 0.85\sum l_1 h_1 + 0.75\sum l_2 h_2$$

The latter two terms are for superstructure and erections and can typically approximated by a value of 200 to 300 for cargo ships. This form of E was originally used to calculate a ships requirement for mooring and anchoring cables, hawsers and warps, but was superseded by a new numeral in 1965. However it is steel used in this context for steel mass estimation.

Steel mass is then calculated from:

$$\text{Steel mass} = KE^{1.36} \quad .$$

Where K is found from the calculated Lloyds equipment numeral, E, and the corresponding K value found from interpolating with respect to E for the appropriate ship type in the accompanying table.



[1]

There should be a form correction to account for variation from the standard 0.70 value for which the data is derived. The correction is:

$$\textit{Steel mass (corrected)} = \textit{Steel mass}(1 + 0.05(C_B' - 0.70))$$

And again C_B' is the block coefficient for the design at $0.8D$.

Type	K			Range of E	No. of ships in sample
	Mean value		Range		
Tankers	0.032	±	0.003	1500–40000	15
Chemical tankers	0.036	±	0.001	1900–2500	2
Bulk carriers	0.031	±	0.002	3000–15000	13
Container ships	0.036	±	0.003	6000–13000	3
Refrigerated cargo	0.034	±	0.002	4000–6000	6
Coasters	0.030	±	0.002	1000–2000	6
Offshore supply	0.045	±	0.005	800–1300	5
Tugs	0.044	±	0.002	350–450	2
Research ships	0.045	±	0.002	1300–1500	2
Ro-Ro ferries	0.031	±	0.006	2000–5000	7
Passenger ships	0.038	±	0.001	5000–15000	4
Frigates and corvettes	0.023		not known		

[1]

It is interesting to note that Lloyds equipment numeral, *E*, is in the form of a type of surface numeral, as previously discussed, and as such may be considered proportional to λ^2 . Watson's Method assumes that:

$$Steel\,mass \propto E^{1.36} \quad .$$

Therefore

$$Steel\,mass \propto (\lambda^2)^{1.36} \;\text{ or }\; Steel\,mass \propto \lambda^{2.72} \quad .$$

This is in keeping with the previous observation on the relationship between λ and steel mass and provides justification of the usefulness of Watson's Method. However it should be noted that some of the data given in Table 1 is rather dated and in some cases derived from small data sets for some ship types.

Other useful Methods

Harvald and Juncher Jensen Method (DTU)

Specifically based on data collected from Danish shipyards covering 1960 to 1990 but with a substantial of vessels built from 1980 to 1990.

Again steel mass is given as a function of cubic numeral, *K.*

$$K = LBD + \text{ volume of poop \& forecastle } .$$

$$Steel\,mass = C_S K$$

Where C_S is *steel coeffcient* and is found from:

$$Steel\,coefficient = C_{S_o} + 0.064\exp(-(0.50u + 0.10u^{2.45}))$$

Where C_{S_o} is dependent on ship type (e.g. 0.70 t/m^3 for cargo ships and bulk carriers and $u = \log_{10}(\Delta/100)$.

The advantage of this method is that it is easy to apply and based on empiricism derived from modern ships. However the volume of poop and forecastle needs to be estimated in early design calculations before such features have been considered.

Technical University of Aachens Method (Schneekluth (1985))

Applicable for container ships within the following constraints:

$100 < L < 250$; $4.7 < L/B < 7.63$; $1.47 < B/D < 2.38$; $2.4 < B/T < 3.9$; $0.52 < C_B < 0.716$.

$$\begin{aligned} Steel\,mass = \nabla_u\, 0.093[1 + 0.002(L - 120)^2 \times 10^{-3}][1 + 0.057(L/D - 12)][30/(D + 14)]^{1/2} \\ [1 + 0.1(B/D - 2.1)^2][1 + 0.2(T/D) - 0.85)][0.92 + (1 - C_B)^2] \end{aligned} .$$

Where ∇_u is the under deck volume including poop forecastle and hatchways and C_B is the block coefficient at the draught tangential to the main deck (i.e. at the depth).

Kerlen (1985) Method

$$Steel\,mass = 0.0832X\exp(-5.73 \cdot X \cdot 10^{-7}) \text{ where } X = (1/12)L^2 BC_B^{1/3} .$$

Note there is no dependence on D in this method.

Murray (1964) Method

$$Steel\,mass = 0.026L^{1.65}(B + D + (T/2))((0.5C_B + 0.4)/0.8)$$

Telfers (1956) Method

$$Steel\,mass = C_O)(1 + (b_O C_B{}' B/D)(DL^{5/3}/1000)$$

Where C_O is typically 17 for a dry cargo vessel. A value for b_O is found from basis data.

2.1.9 A More Rational Approach to Steel Mass Estimation

Watson suggests a more rational approach based on area of plating in both the transverse and longitudinal directions and the section modulus, this provides an expression of the form:

$$Steel\ mass = C_B^{1/2} LB[K_1 L(L/D) + K_2 D]$$

K_1 is section modulus related and K_2 is volume related. This compares favourably with Sato's expression:

$$Steel\ mass = C_B^{1/3}[w_1 L^{3.3} B/D + w_2 L^2 (B+D)^2] \quad .$$

Values for K_1 and K_2 can be derived from contemporary basis data.

More refined methods may be used if a better breakdown of the steel weight of the basis ship is available, e.g. upper deck, 'tween Deck, inner bottom, outer bottom, side shell, bulkheads , superstructure weights.

A square number approach is probably appropriate for each of the above elements of the structure, except Superstructure.

For the Upper Deck is proportional to $L \times B$ with a form correction ideally dependent on the waterplane area coefficient but practically varying with the block coefficient and a scantling correction depending on L/D ratio. The Outer Bottom could be treated in a similar way.

'Tween Deck(s) and Inner Bottom will tend to vary only with $L \times B$ and block coefficient, while Side Shell will follow $L \times D$ and block coefficient.

Bulkhead weight will tend to vary with $B \times D$, block coefficient and number of bulkheads. Superstructure(s) can be treated using their own mini cubic number $l \times b \times h$ where l ,b and h are the mean values of length, breadth and height of the superstructure.

2.1.10 Contemporary Influences on Steel Mass Estimation.

- More modern ships for the same E have reduced L but larger B and D (as previously discussed).
- Steel mass for more modern designs is less for given E as cargo vessels have less decks (e.g. no 'tween deck) and fewer bulkheads (e.g. no deep tanks etc.).
- Reduction of 'owners extras'.
- Classification rules are based on better analysis and evaluation tools allowing reduced scantlings in comparison to older ships of the same type.
- Mandatory changes to structural arrangement (e.g. double hull tankers).
- Steel mass in accommodation has reduced in keeping with smaller crews.
- Most methods are only for all steel structure. No influence of higher tensile steel or other materials (particularly in superstructure and deckhouses). Higher tensile steels with significantly higher strength properties (greater than NV40) are likely to become available.
- Other materials have been used, such as aluminum, and once certain issues are resolved, composites will be more common especially in weight sensitive designs.

- Watson suggests the following for equivalent structure:
    - 1 tonne higher tensile steel equates to 1.13 tonnes mild steel.
    - 1 tonne aluminum equates to 2.9 tonnes mild steel.
    - 1 tonne FRP equates to 2.9 tonnes mild steel.
- Replace with more accurate method once more information is available (e.g. 'midship section method).

2.1.11 Lightmass Margins

Lightship = Steel mass + Machinery mass + Outfit mass + **Margin**.

The margin is an essential part of the weight make up as it allows for errors and omissions in the remainder of the calculations. For a vessel whose lightship is a relatively small part of the full load displacement a value of about 2% to 3% of lightship is likely to be appropriate for commercial designs. The margin can be rationalised with respect to the uncertainty associated with each component of lightship to provide a more informed estimate. Where the lightship is a much greater proportion of the full load displacement and a weight over-run would be seriously embarrassing then a greater percentage may be chosen…

For naval ships there is not just this margin to account for inaccuracy in the design estimation of mass but there are also two other margins considered in UK warship rules:

- A board margin – Allowance for change in weight due to changes in the design during construction (perhaps 2% lightship);
- A growth margin – Allowance for increase in weight due to additions and alterations during subsequent refit and the weight growth as a consequence of accretion of paint, coatings etc (typically 0.5% per annum over ship life).

This has to be carefully rationalised because one tonne of margin increases the ship size and first cost…

2.1.12 Estimating Mass of Consumables

- **Fuel oil** can be found from the specific fuel consumption (g/kWh) figures supplied by engine vendors but a margin of 10% should be included for real operating conditions. If heavy fuel oil is to be specified than an additional increase in the sfc of 8% is suggested. The range, endurance at the service speed and a reserve can then be used to find the total fuel oil requirement.

- **Diesel oil** for auxiliaries and engine starting (as appropriate) can be estimated in the same way based on the sfc for the auxiliaries.

- **Lube oil** can be based on main engine consumption and other needs but is at least two orders of magnitude less than fuel oil requirements. Typically 15 tonnes in slower vessels to 40 tonnes in larger high-speed ships.

- **Fresh water** can be approximated to 200 litres per day per person for a period consistent with the endurance of the vessel and typically a three day margin. A water maker can offset the quantity required.

- **Provisions** can be found by assuming about 0.01 tonnes per day per person including associated packaging.

- **Passenger, crew and effects** can be represented by about 0.17 tonnes per person.

2.2 Bow and Stern Design Considerations

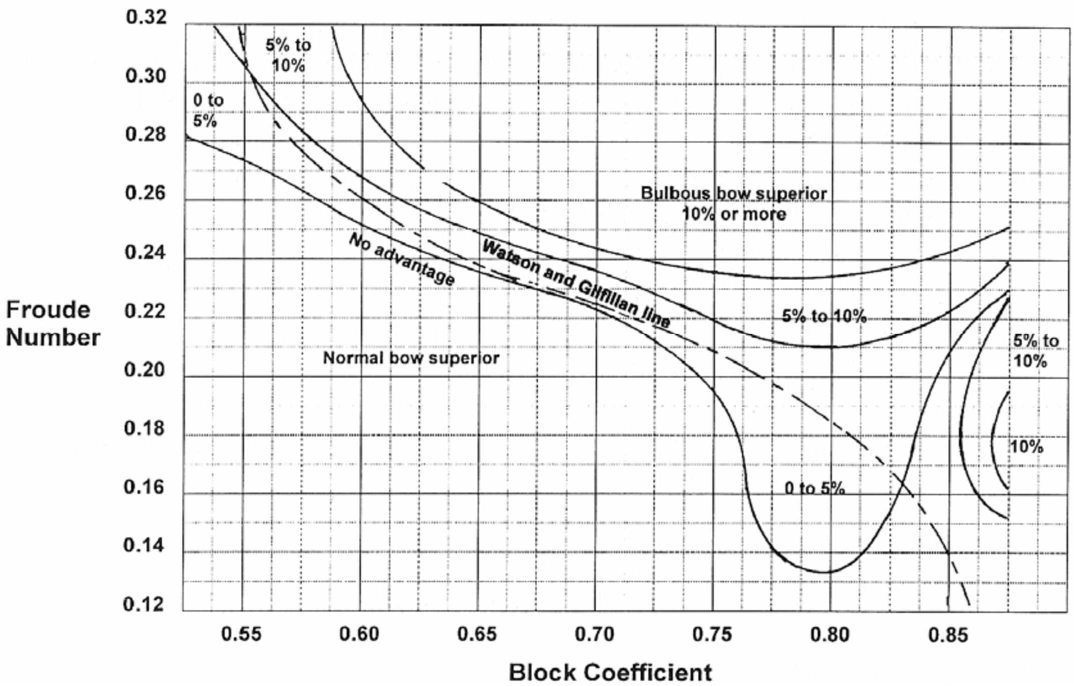
2.2.1 Bow Design

The first choice is normal bow v bulbous bow. Bulbous bow can be generally advantageous in the range $0.17 < F_N < 0.7$. Benefits of up to a 20% reduction in R_T can be possible. Other benefits include added resistance in a seaway, seakeeping, maneuverability, trim, housing for bow thruster etc. Care has to be taken regarding build cost and anchor handling to avoid anchor clashing with bulb form.

The following chart is for the load draft condition. The biggest benefit of a bulbous bow tends to be exhibited in the ballast condition, especially for full ships with $C_B >$ o.75. For vessels making extended passages in ballast at the service speed this is of obvious benefit, however this argument for a bulbous bow is less significant if, for general fuel economy, the speed in the ballast condition is reduced.

Bulbous bows are defined using the following form characteristics.

- Shape of section.
- Side-view.
- Length of projection beyond perpendicular.
- Position of axis.
- Area ratio ($A_{Bulb\ Section\ at\ FP}$ / $A_{Midship}$)
- Transition and fairing into hull lines.

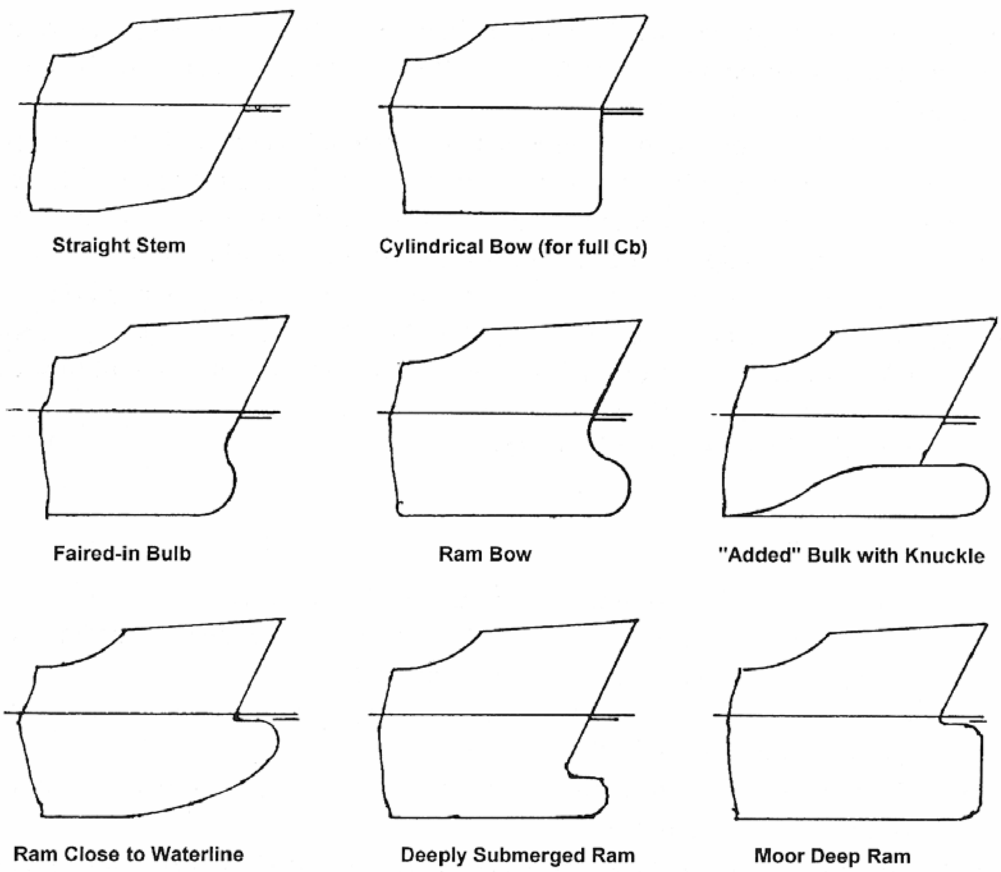


[1]

'*V*' section bulbs (low area ration) are preferred as they reduce slamming, increase the water plane shape forward in the ballast condition and allow for easier anchor handling . They are easily blended into finer 'V' shaped forward hull sections. However they are more complex and expensive to build.

'*O*' section bulbs (high area ratio) are cheap to build and can be faired more easily into '*U*' shaped forward hull sections but are generally accepted to be less effective.

The bulb never extends beyond the extreme forward point of the stem. The length of projection forward of the FP can be approximated to 20% *B*. Bulbs that extend significantly forward of the FP can be referred to as 'Ram' bows.

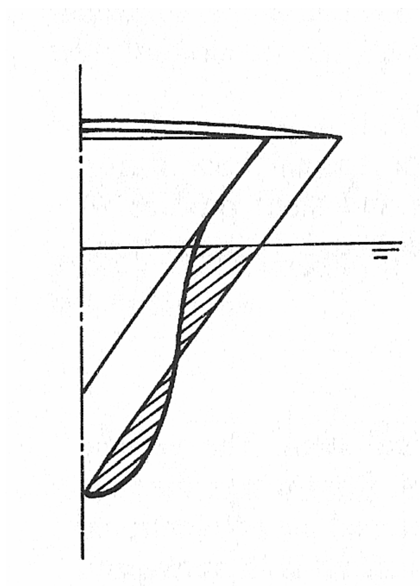


[1]

For very full ships with full '*U*' forward sections a 'Parabolic bow' can avoid shoulders that would result from attempting a finer bow. Typically for $C_B > 0.8$, $F_N > 0.18$ and high B / T .

For more modest C_B forms the bow profile will depend on the shape of forward sections. More '*U*' shaped bow sections will generally result in a more vertical stem profile whereas with '*V*' shaped sections there is the possibility of a raked stem profile.

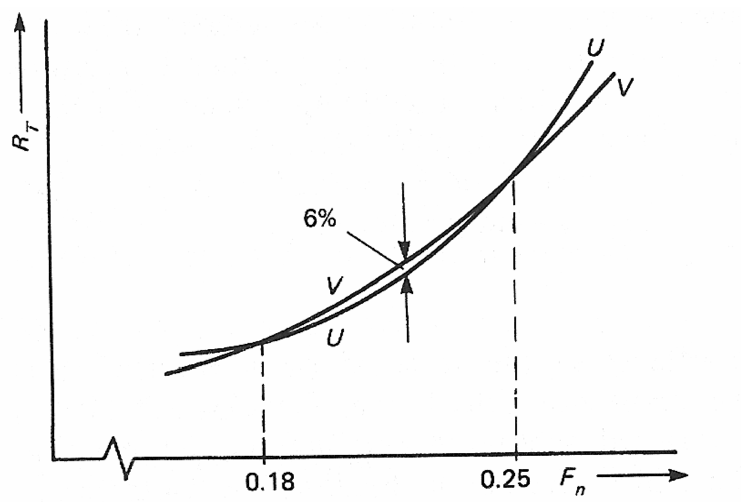
The use of equivalent '*V*' shaped sections to replace '*U*' sections can have advantage.



'*V*' sections offer:

- More reserve buoyancy and internal volume.
- Increase second moment of the water plane and hence *KM*.
- Less wetted surface with benefit to R_F.
- Lower steel weight and inherent work content through reduction of curvature.
- Increased deck area with benefits for cargo stowage and access.
- Benefits to R_T can be achieved in region $F_N < 0.18$ and $F_N > 0.25$ for $B/T > 3.5$.

[4]



Above the waterline the stem profile tends to be raked forward to conform with the flare of adjacent sections. Flare and rake are intended to provide reserve buoyancy forward to help reduce large amplitude pitching and the amount of water shipped on the foredeck.

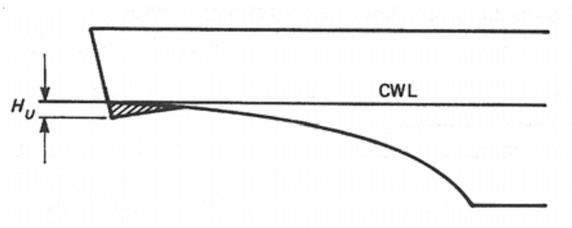
'*V*' shaped forward section inherently have flare. '*U*' sections tend to have flare introduced above the waterline as required. Flare tends to throw seas clear and reduces the occurrence of green seas on deck when relative motion is large enough for freeboard to be exceeded. Too much flare should be avoided as in heavy weather the pressure experienced can be very significant resulting in damage. If the pressure distribution is not symmetric it can also cause torsional loading of the bow structure. Bow structure with extensive flare needs to be strengthened accordingly as the hydrodynamic pressures can be very significant.

2.2.2 Stern Design

The requirements in stern design are to:

- Minimize flow separation and therefore resistance;
- Provide adequate propeller clearance to avoid p.e.v. problems;
- Ensure high propulsive efficiency by attempting uniform inflow into the propeller disc and good relationship of thrust deduction to wake (η_H);
- The provision of good flow into the rudder(s) to ensure good control and course stability;
- Aft end structure to support propeller(s) and rudder(s) with sufficient space and clearance for related internal systems.

In terms of stern lines above the propeller, the transom stern has evolved as the most common arrangement replacing the elliptical and cruiser stern. The transom stern, effectively a 'sawn off' cruiser stern, is simpler and cheaper to produce as well as providing resistance advantages at higher speeds by giving a less turbulent wake. The extra deck area afforded is also beneficial for mooring equipment, stowage of containers aft or moving the accommodation further aft.



To counter stern trim and reduce resistance by reducing the high stern wave that can build up, there is the possibility of incorporating a stern wedge faired into the stern or the fitting of stern flaps.

Recommendations for stern design are given as:

$F_n < 0.3$	Stern above CWL. Some stern submergence during operation.
$F_n \approx 0.3$	Small stern—only slightly below CWL.
$F_n \approx 0.5$	Deeper submerging stern with average wedge.
	Submergence $t = 10–15\%T$.
$F_n > 0.5$	Deep submerging stern with wedge having approximately width of ship.
	Submergence $t = 15–20\%T$.

[4]

The underwater lines need to be considered to reduce flow separation. Sharp shoulders at the stern and lines exceeding a critical angle relative to the flow should be avoided. With an angle of run of over 20 degrees in waterlines aft separation is considered inevitable and is considered to start at around 15 degrees. The development of forms adhering to this becomes more difficult with increasing fullness. However if the flow follows the buttock lines then separation is unlikely even if the run aft is not very fine. This can be the case with tern bulbs and pram type sterns.

For a single screw ship where the ratio of diameter to draft is in the order of 0.75, to ensure good flow with respect to the propeller disc, the angle of run aft should not exceed about 30 degrees. For a full form this forces the LCB forward as mentioned previously. Lloyds recommended minimum clearances as a fraction of the diameter for a four bladed propeller are:

Tip to sternframe arch	$= 1.00\ K$
Sternframe to leading edge at $0.7\ R$	$= 1.50\ K$
Trailing edge to rudder at $0.7\ R$	$= 0.12$
Tip to top of sole piece	$= 0.03$

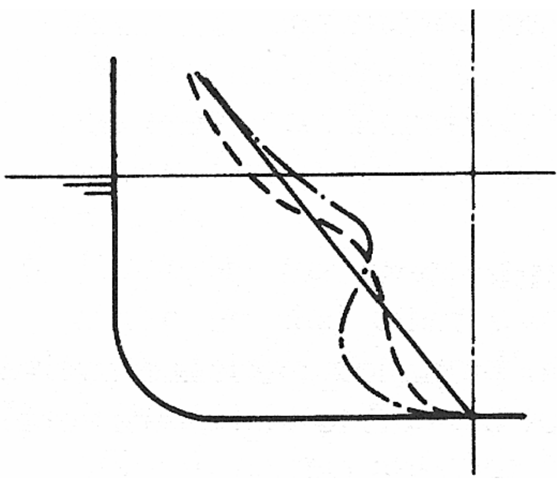
$$K = \left(0.1 + \frac{L}{3050}\right)\left(\frac{2.56C_b \cdot P}{L^2} + 0.3\right)$$

where P = power in kW.

[1]

The choice of stern sections can be considered as:

- '*V*' section;
- '*U*' section;
- Bulbous stern.



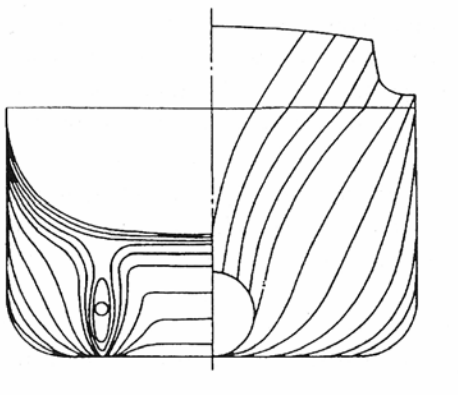
These three types of sections have a different influence on resistance and propulsive efficiency.

'*V*' sections have the lowest resistance at all Froude numbers. The '*U*' section has higher resistance and the bulbous stern generally the highest resistance, although a well designed bulbous stern will be comparable to the '*U*' section case. Conversely '*V*' sections result in the most non-uniform wake distribution and bulbous sterns the most uniform, with '*U*' sections between these two extremes.

For single screw ships, '*U*' sections or bulbous sections are therefore preferred to give higher propulsive efficiency, less torque and thrust variation and reduced chance of propeller excited vibration. The bulbous form is significantly more expensive due to the more complex curvature and inherent work content. However the improvement in propulsive efficiency can offset the resistance and cost penalties.

For twin screw ships '*V*' sections are generally preferred as there is the benefit of better resistance and both propellers still benefiting from uniform wake at their positions off the centreline.

For twin-screw arrangements there is an appendage resistance penalty due to bossings and shaft brackets. Also there is the possibility of shaft 'wirling' vibration between supports. Fully enclosed bossings provide more support but can increase the naked hull resistance in the order of 10%. To reduce this resistance penalty, less extent of bossings and supporting the shaft on more 'A' brackets is beneficial. A more recent innovation to provide the advantages of the greater shaft support afforded by extended bossings and the reduction of vibration is the 'twin skeg form' where the bossings become effectively part of the main hull. However there is debate over the influence on resistance.

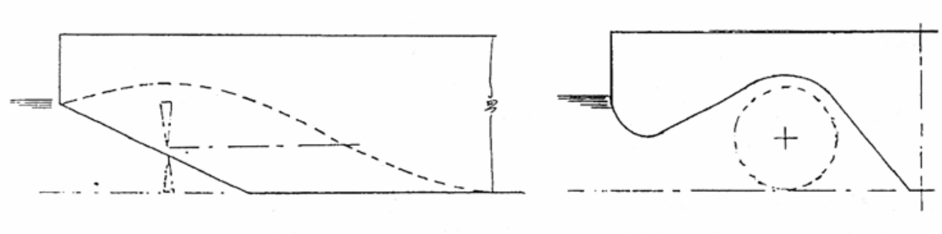


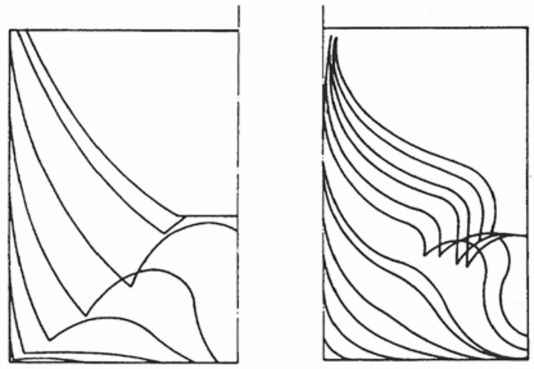
[1]

In order to benefit from an increased propeller diameter there are a number of possibilities:

- Allow the propeller tip to be below the keel line. Although common practice for warships, the increased likelihood of damage has stopped this practice being adopted for merchant ships.

- Designing to operate at a stern trim or to have a raked keel. For draft limited merchant ships this has obvious disadvantage in operation.
- A Mariner type rudder allows for a modest increase in diameter through the omission of a stern frame sole piece.
- A tunnel stern can be adopted. This has the most impact on the stern lines, however it provides the possibility of increased propeller diameter for both single and twin screw forms, especially where draft is restricted.

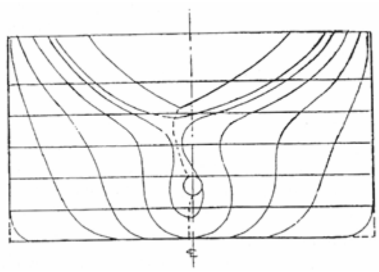




Great Laker "Canadian Enterprise" Ocean-going ship "Canadian Pioneer"

[1]

The propeller tip can come right up to the static waterline and still remain fully immersed with this arrangement. Tunnelled forms can provide a further benefit of increased deadweight by providing more displacement. In keeping with the fitting of devices to improve propulsive efficiency, such as the Mitsui duct, Grothues spoilers, Grim vane wheel, contra-rotating propellers etc., the stern lines can be modified with the same intent of reducing loses to rotational energy. An asymmetric stern can 'pre-rotate' the flow into the propeller in the opposite direction to the rotation imparted by the propeller. This approach has proven to be effective, incurs minimal extra hull construction costs and does not penalise resistance.



2.3 Hull Form Distortion

2.3.1 Introduction

As an adaptive design problem, the generation of a hull form to meet the design dimensions and form determined requires the modification of a basis form. Such a basis vessel may be representative of a particular design office's practice, a previous solution to a similar vessel or determined from appropriate series or published data. Such hull form distortion is likely to be done using hull form development and design software such as Tribon or NAPA. The methods to achieve distortion of principal dimensions are straightforward in their implementation but the modification of form characteristics is more involved. The particular method discussed here is that of Lackenby. This is the method used in the software mentioned.

The Forward and Inverse Analysis Approaches introduced in a subsequent unit also require systematic control of principal dimensions and form characteristics. The particular requirements and means of doing this are also discussed here.

2.3.2 Variation of Principal Dimensions

To modify length, beam, or draught, the hull offsets may simply be multiplied by corresponding constant expansion or contraction factors.

$$\text{Station spacing design} = \text{Basis station spacing} \times \frac{L_D}{L_B}$$

$$\text{Waterline spacing design} = \text{Waterline spacing basis} \times \frac{T_D}{T_B}$$

$$\text{Offsets deign} = \text{Offsets basis} \times \frac{B_D}{B_B}$$

In a later unit there is the need, as a special case, to be able to alter length while maintaining the same displacement In order to change length, while the displacement is fixed, the midship section area is altered in inverse ratio to the length. The breadth to draught ratio remains constant as well as the displacement, block coefficient and other form parameters associated with the hull form. The new main hull form dimensions become:

$$L' = (1+\delta L)L\,,\ B' = B\Big/\sqrt{(1+\delta L)}\ \text{ and }\ T' = T\Big/\sqrt{(1+\delta L)}\ .$$

Since B'/B and T'/T are equal and L'/L is specified, the waterlines and offsets of corresponding stations can be found using the above simple procedures.

Similarly, with L fixed and B/T changing by a factor of $1+\delta x$, then changes in B and T now correspond to:

$$B' = B\Big/\sqrt{(1+\delta x)} \text{ and } T' = T\Big/\sqrt{(1+\delta x)} \quad .$$

The corresponding changes in offsets and waterlines for each demihull are again undertaken in the usual manner.

2.3.3 Variation of Form Parameters

A Common technique to derive an alternative design with the same main dimensions but different C_P or *LCB* is to change the location of stations at which the offsets are given. That is, the shape of the sections remain the same as in the parent hull, but they are moved forward or aft in some manner so that the curve of sectional area changes in the desired manner.

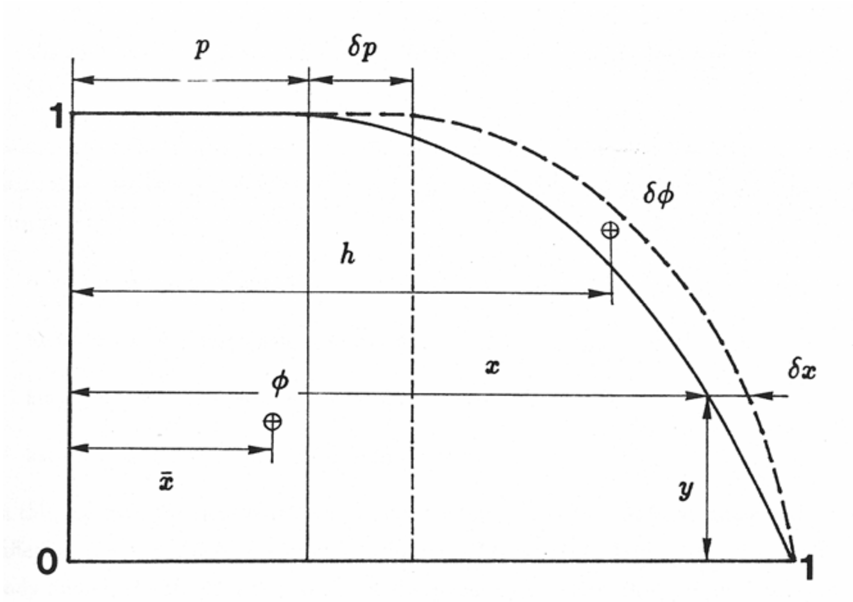
In the figure below, *F(x)* represents the dimensionless sectional area curve (SAC) of one half-body of the ship. It is convenient to consider this half-body as being one unit long and the maximum ordinate of the sectional area curve also equal to unity. All horizontal dimensions are therefore fractions of the half length and the area under the curve *F(x)* is numerically equal to the prismatic coefficient of the half-body.

A simple method that might be adopted is the 'one-minus prismatic' method where the new spacing of the sections from the end of the body is made proportional to the difference between the respective prismatics and unity

$$\delta x = \frac{\delta\phi}{1-\phi}(1-x) \quad .$$

Here ϕ is the prismatic coefficient of the half body, $\delta\phi$ is the required change in prismatic coefficient of the half-body, x is the fractional distance of any transverse section from midships, and δx is the necessary longitudinal shift of the section at x to produce the required change in prismatic coefficient. The change in the parallel middle body δp will equal δx at $x = p$, therefore

$$\delta p = \frac{\delta\phi}{1-\phi}(1-p) \quad .$$



Although, the 'one-minus prismatic' method is useful and simple to apply, it has major disadvantages:

- For a given change in fullness the longitudinal distribution of the displacement added (or removed) cannot be controlled, and
- There is no control over the extent of the parallel middle body (p.m.b.) in the derived form.
- For forms with no p.m.b. a reduction in fullness cannot be made and an increase in fullness cannot be achieved without introducing it.

Lackenby (1950) describes a simple method for deriving the amount by which each offset station should be moved to generate given changes in the following parameters:

- Total prismatic coefficient, ϕ_t.
- Forward and aft prismatic coefficients, ϕ_f and ϕ_a.
- Longitudinal centre of buoyancy, $\bar{z}$.
- Length of parallel middle body, *p*.

In this method, for the transformation of a parent form to a desired ship form with different prismatic coefficient and *LCB* position, the after body and the fore body are transformed separately as a function of the desired change in prismatic of the two bodies. This approach can be used in the distortion of the sectional are curve and design waterline independently to change fullness, location of centroid and the length and location of the parallel middle body for each curve. Application of the method to the design waterline therefore provides control of water plane area coefficient and *LCF*.
Lackenby assumes an expression for δx of the form

$$\delta x = c(1-x)(x+d) \quad ,$$

where c and d are constants in a given case. Terminal conditions are $\delta x = 0$ at $x = 1$ and $\delta x = \delta p$ at $x = p$. These conditions and $\delta\phi = \int_0^1 \delta x\, dy$ yields

$$c = \frac{\delta\phi - \delta p((1-\phi)/(1-p))}{\phi(1-2\bar{x}) - p(1-\phi)}$$

and

$$d = \frac{\delta p}{c(1-p)} - p \quad .$$

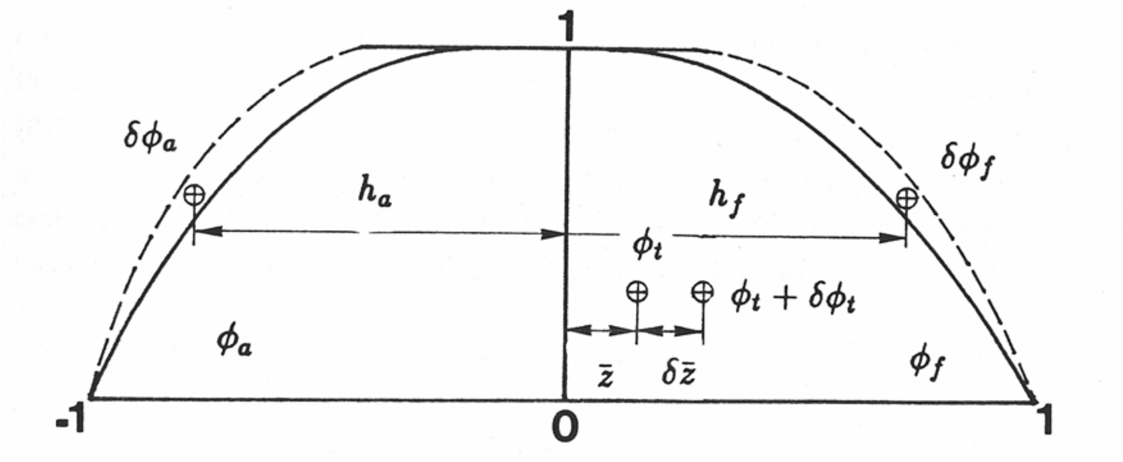
Hence

$$\delta x = (1-x)\left\{\left[\delta\phi - \delta p\frac{1-\phi}{1-p}\right]\frac{x-p}{A} + \frac{\delta p}{1-p}\right\}$$

where

$$A = \phi(1-2\bar{x}) - p(1-\phi) \quad .$$

In these expression all the terms are related to the properties of one half body, i.e. here ϕ is either ϕ_f or ϕ_a depending on the body under consideration. Both bodies need to be considered together in order to determine the effect of the variations discussed on the total prismatic coefficient and the position of the LCB.



In the above figure, where each half-body is one unit long and one unit maximum ordinate, to change the total prismatic by an amount $\delta\phi_t$ and the change the position of the centroid by $\delta\bar{z}$, the amount of change in prismatic coefficient forward, $\delta\phi_f$ and prismatic coefficient aft, $\delta\phi_a$, can be calculated using the following simple relationships;

$$\delta\phi_t = \frac{\delta\phi_f + \delta\phi_a}{2}$$

and

$$\delta\phi_f(h_f - \bar{z}) + (\phi_t + \delta\phi_t)\delta\bar{z} = \delta\phi_a(h_a + \bar{z}) \quad .$$

Hence

$$\delta\phi_f = \frac{2[\delta\phi_t(h_a - \bar{z}) - \delta\bar{z}(\phi_t + \delta\phi_t)]}{h_f + h_a}$$

and

$$\delta\phi_a = \frac{2[\delta\phi_t(h_f + \bar{z}) + \delta\bar{z}(\phi_t + \delta\phi_t)]}{h_f + h_a}$$

where $\bar{z}$ is the distance of the *LCB* in the basis ship from midships expressed as a fraction of the unit half length.

The moment of the added ‘sliver’ of area about midships is given by

$$\delta\phi\, h = \int_0^1 \delta x\, x\, dy \quad .$$

From the original expression used by Lackenby and the expressions for *c* and *d* previously given, the moment of area added about midships becomes

$$\begin{aligned}
\delta\phi\, h &= \int_0^1 (x^2 - x^3 - px + px^2)\, dy + \frac{\delta p}{1-p}\int_0^1 (x - x^2) dy \\
&= c(2\phi\bar{x} - 3\phi k^2 - p\phi + 2p\phi\bar{x}) + \frac{\delta p}{1-p}(\phi - 2\phi\bar{x}) \\
&= c\phi[2\bar{x} - 3k^2 - p(1-2\bar{x})] + \frac{\delta p\phi}{1-p}(1-2\bar{x}) \quad .
\end{aligned}$$

Substituting for *c* and dividing by $\delta\phi$ provides the lever, *h*, as

$$\begin{aligned}
h &= \phi\left\{\left(1 - \frac{\delta p(1-\phi)}{\delta\phi(1-p)}\right)\left(\frac{1}{\phi(1-2\bar{x}) - p(1-\phi)}\right)[2\bar{x} - 3k^2 - p(1-2\bar{x})] + \frac{\delta p}{\delta\phi}\frac{1-2\bar{x}}{1-p}\right\} \\
&= \phi\left\{\frac{B}{\phi}\left(1 - \frac{\delta p}{\delta\phi}\frac{1-\phi}{1-p}\right) + \frac{\delta p}{\delta\phi}\frac{1-2\bar{x}}{1-p}\right\}
\end{aligned}$$

where *B* is a constant depending on the geometrical properties of the basis form,

$$B = \frac{\phi[2\bar{x} - 3k^2 - p(1-2\bar{x})]}{A} \quad ,$$

and k is the radius of gyration (or lever of the second moment) of the original curve about midships. Therefore the levers for the fore and aft bodies can be written as

$$h_f = \phi_f \left\{ \frac{B_f}{\phi_f} \left(1 - \frac{\delta p_f}{\delta \phi_f} \frac{1-\phi_f}{1-p_f} \right) + \frac{\delta p_f}{\delta \phi_f} \frac{1-2\bar{x}_f}{1-p_f} \right\}$$

and

$$h_a = \phi_a \left\{ \frac{B_a}{\phi_a} \left(1 - \frac{\delta p_a}{\delta \phi_a} \frac{1-\phi_a}{1-p_a} \right) + \frac{\delta p_a}{\delta \phi_a} \frac{1-2\bar{x}_a}{1-p_a} \right\} \quad .$$

Having calculated h_f and h_a for the fore and aft bodies the required adjustments to the fore and after body prismatic coefficients to give any desired change in *LCB* position and total prismatic coefficient can be determined from the previous equations for $\delta\phi_f$ and $\delta\phi_a$. From these the revised spacing of the transverse sections can then be calculated using the appropriate expressions for δx given.

In the general case where $\delta p \neq 0$, the relations for h_f and h_a themselves involve the required changes in fineness $\delta\phi_f$ and $\delta\phi_a$. This difficulty can be overcome by substituting the expressions for h_f and h_a and solving for $\delta\phi_f$ and $\delta\phi_a$. These general expressions are

$$\delta\phi_f = \frac{2[\delta\phi_t(B_a + \bar{z}) + \delta\bar{z}(\phi_t + \delta\phi_t)] + C_f \delta p_f - C_a \delta p_a}{B_f + B_a}$$

and

$$\delta\phi_a = \frac{2[\delta\phi_t(B_f - \bar{z}) + \delta\bar{z}(\phi_t + \delta\phi_t)] + C_f \delta p_f + C_a \delta p_a}{B_f + B_a}$$

where C is a constant for the parent form given by

$$C = \frac{B(1-\phi) - \phi(1-2\bar{x})}{1-p} \quad .$$

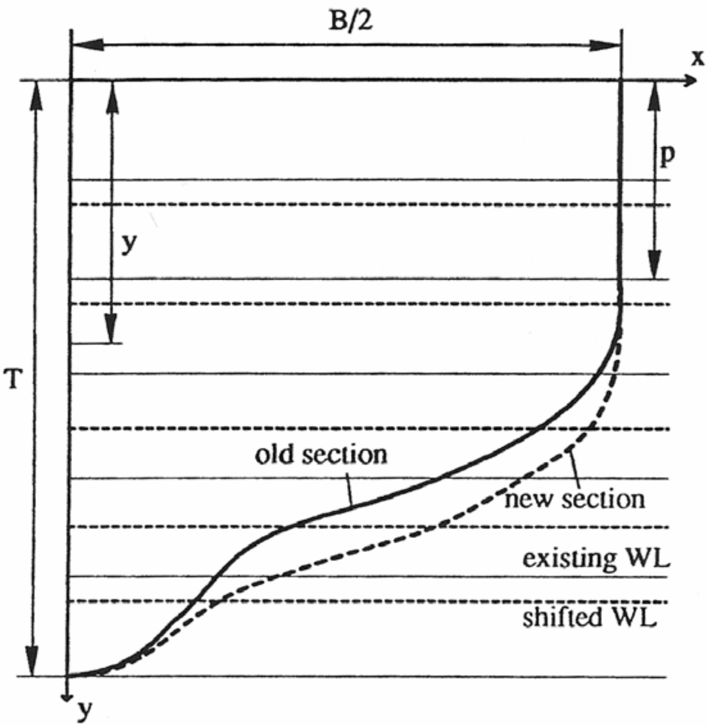
There are definite limits to which the fullness of a given form can be varied for this method. **Lackenby (1950)** provides the absolute limit for changes in fullness in Appendix II of the paper as

$$\delta\phi = \frac{\delta p(1-\phi) \pm A\left(1 - \frac{\delta p}{1-p}\right)}{1-p} \quad .$$

Orthodox methods of geometric variation in which transverse sections of a parent design are moved from their position on the parent sectional area curve to the position with the same ordinate on a new curve, as described above, do not allow independent variation of block coefficient and longitudinal position of the centre of buoyancy on the one hand, and of water plane area coefficient and longitudinal position of the centre of flotation on the other. The Forward Analysis and Inverse Analysis approaches require systematic variations where one of the principles is to change each form parameter independent from each other

Having established the use of Lackenby's method to obtain new sectional area and beam distributions for required changes in C_P, *LCB*, C_{WP} and *LCF* values, it can also be used as the basis for the linear distortion of transverse sections. In order to generate new section forms, as with the longitudinal shifting of sections, a similar procedure based on Lackenby's linear distortion methods, can be applied to shift the waterlines vertically. Given the new values of sectional area and beam for each section, two different types of variation procedures are needed:

- A change in sectional beam by multiplying offsets by the ratio of (new sectional beam)/(initial sectional beam) to satisfy the required C_{WP} and *LCF* variations.
- A change in sectional area while the sectional beam is fixed to satisfy the required *LCB* and C_P variation or just *LCB* variation if C_P is held invariant.



For each section, as shown in the figure, the baseline and the design waterline maintain their positions. The intermediate waterlines are displaced using the linear distortion procedure that gives the amount of change in the position of each waterline as

$$\delta y = \frac{\delta\phi(1-y)(y-p)}{\phi(1-2\bar{y})-p(1-\phi)} \quad .$$

where ϕ is now the sectional area coefficient, $\delta\phi$ the required change in sectional fullness, p the parallel length of the section (for HSRB forms p = 0) and $\bar{y}$ is the vertical position of the section centroid.

The whole lines distortion based variation procedure applied in the Forward and Inverse Analysis Approaches (to be discussed in the next units) is shown below:

Key References

[1] Watson , D. G. M., ‘Practical Ship Design’, Elsevier, 1998.
[2] Watson , D. G. M., ‘Designing Ships for Fuel Economy’, YARD, 1988.
[3] Fisher, K. W., ‘The Relative Cost of Ship Design Parameters’, Trans RINA, 1973.
[4] Schneekluth, H., and Bertram, V., ‘Ship Design for Efficiency and Economy’, Butterworth Heinemann, 1998.