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## Some Ship Design Methods

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**SUMMARY:** Fourteen years have elapsed since the publication of an earlier paper<sup>(1)</sup> by one of the authors on 'Estimating Preliminary Dimensions in Ship Design'. During this period there have been, almost certainly, greater changes in ships than in any previous period of the same duration.

There have also been substantial changes in ship design methods with the development of computer technology. The present paper reviews the design methods presented in 1962, considers to what extent these have stood the test of time and suggests some further developments in them. It considers how the relationships between dimensions, the coefficients and approximate formulae quoted have changed and why.

Finally, the scope of the paper is extended to consider some other aspects of design.

### 1. INTRODUCTION

Since 1962 there have been four significant changes (and a great many minor ones) affecting, to varying degrees, the data and the methods presented at that date by Watson<sup>(1)</sup>.

Firstly, there has been the enormous growth in world tonnage of ships from 145 million tons in mid 1963 to a figure of 340 million tons gross in mid 1975 accompanied by striking changes in the composition of the fleet, the development of a number of totally new types of ship and a step change in both the maximum and average size of ships of a number of the traditional types.

There have been the changes in ships machinery to enable the power required for these new and developed ship types to be attained. The output of both slow and medium speed diesels has been increased by a factor of 2 or more. Steam turbines have received a new lease of life, gas turbines have entered the picture and nuclear propulsion may now be approaching commercial application.

There has been a striking growth in the availability and capacity of computers. In 1962 there were only about 20 computers of any significant size in the United Kingdom, and four years were to elapse before BSRA started their committee on the use of computers in ship design in 1966. Slide rules were therefore the normal tool in ship design offices and the 1962 paper bears abundant evidence of this, although some of the more far-sighted contributors to the discussion saw the methods presented in the paper as preparing the way, to some extent, for the use of computers.

Finally, in 1962 we were still firmly in the era of British imperial units of feet, inches, tons and horsepower. This change alone necessitated an updating of the 1962 paper if it was to continue to be of use.

Before seeing how the data and methods presented have stood the test of time, it is worth spending a moment recalling the state of the art as it was in 1962 and looking at the changes which have taken place in the intervening years.

#### 1.1 Changes in Ships and the Shipping Fleet

The tanker fleet has expanded enormously to meet the steadily expanding demand for oil which grew at 5% per annum until the OPEC price increases in 1973/1974. The tendency to site refineries in consuming countries in the years since World War II led to the development of crude oil tankers as a specialised class which has led the growth in ship size as the economies of scale have become more apparent, aided by such events as the closure of the Suez Canal. With crude oil being carried by ever larger ships, the development of a second class of specialist tanker, the products carrier, became necessary and a large number of these vessels has also been built.

Bulk carriers, in their infancy in 1962, have grown both in numbers and size, taking over the role of the tramp ship in the ore, coal and grain trades and now constitute the second largest group, by tonnage, in the world fleet.

Container ships, a class which in 1962 existed only as a few conversions by the Matson and Sea-Land companies, have taken over the role of the cargo liner on many of the world's principal trade routes. Sophisticated cargo liners represented in 1962 by the recently completed BEN LOYAL achieved in the next few years a peak of perfection in such ships as Ocean Fleets PRIAM class and P & O's STRATHARDLE class before container ships took over the cream of the general cargo trade.

Although it was not realised at the time, passenger liners typified in 1962 by the newly completed ORIANA, CANBERRA and TRANSVAAL CASTLE were already becoming uneconomic in the world of the jet aircraft and only the comparatively recent growth in the numbers of cruise liners and cross channel ferries has sustained naval architects' ability to design these most interesting ships.

Development has continued in the numbers, size and sophistication of Ro-Ro ships.

In 1962 the carriage of liquefied natural gas was represented by the METHANE PRINCESS and METHANE PIONEER. There is now a considerable fleet of these vessels and several entirely different design concepts.

Other ship types which have developed enormously in numbers, in size, and in sophistication are those associated with

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TABLE I Changes in World Shipping Fleet 1963-1975

Class of Ship	Mid 1963			Mid 1975		
	No of Ships	Aggregate Gross Tonnage Millions	Largest Ship Dwt Ton	No of Ships	Aggregate Gross Tonnage Millions	Largest Ship Dwt Ton
Tankers	4,984	47	137,000	7,461	151	483,664 (GLOBTIK TOKYO)
LNG Carrier	2	Less than 0.1	24,608 (METHANE PIONEER)	421	3	64,749 (EL PASO PAUL KAYSER)
Bulk & OBO Ships	200 (approx)	7	75,000	3,711	85	278,000 (ZVEALAND)
Container Ships	A few conversions	Less than 0.1	—	419	6	48,542 (LIVERPOOL BAY)
General Cargo Ships	34,565	91		21,560	71	97
Passenger Ships				2,710	7	
Fishing Vessels				18,490	11	
Others				8,952	8	
World Totals	39,751	145		63,724	342	

Sources: Lloyd's Statistical Tables  
Fearnley & Egers Annual Reviews

offshore exploration work including supply vessels and drilling ships.

A summary of the changes in shipping is presented in Table I.

## 1.2 The Changes in Machinery

At the same time as this development was taking place in ships, there was a parallel development in the field of marine engineering.

In 1962 turbo-charging of diesel engines had only recently been introduced and the largest engine which was in service in any substantial numbers had a cylinder bore of 760 mm and developed a power of 1500 BHP per cylinder. By contrast today's diesel engines have bores of up to 1050 mm and can provide a maximum continuous power of about 4600 BHP per cylinder.

Pielstick had started their successful run of medium speed engines and the biggest medium speed engine at sea developed about 5000 BHP. Today several major manufacturers offer well proven installations with powers of up to 27000 HP.

In 1962, steam turbine installations were confined to the largest tankers of that era which required a power close to, or beyond, the limit which could be obtained from a single screw diesel installation and where advantage could be taken of having steam available for tank heating and cleaning, and to passenger liners where the power required was beyond that which could be obtained from a twin screw diesel installation and where the advantage which the steam turbine has in respect of vibration and noise was of particular value. In 1962 it appeared likely that the slow decline of the turbine relative to the diesel would continue. However, the advent of even larger tankers and of container ships whose reduced port turn round time justified higher sea speeds led to a demand for higher powered installations in the years from 1965 onwards. This demand, which could only be met at that time, in terms of proven technology, by the steam turbine, led to a strong revival of the turbine, which lasted until the massive

increase in oil prices which followed the Arab/Israeli conflict of 1973, reinforced the value of the fuel economy of the diesel engine.

New contenders to the propulsion machinery scene appeared in 1962 with the nuclear ship SAVANAH and in 1967 with the gas turbine ship ADMIRAL CALLAGHAN.

Nuclear power has only been fitted to three merchant ships so far, without apparent success, but recent developments suggest that an economic case might be made for its application for high powered vessels if political objections, environmental worries and energy priorities can be resolved.

Gas turbines have moved very rapidly to a dominant position in warship machinery but so far only a comparatively small number of merchant ships have this type of machinery. The gas generator part of marine gas turbine installations have been developed from both aircraft engines and from industrial machines. Characteristics of the former are lightness, compactness, limited life and repair by replacement. Industrial derivatives are heavier and more robust for longer life and can run on high viscosity, cheaper fuels.

Power turbines in both cases are specially developed marine units. The large increase in fuel costs of recent years appears likely to delay the use of gas turbine propulsion generally for merchant ships.

## 1.3 Design Starting Point

One of the more fundamental changes which has occurred in the thinking of naval architects during this period is concerned with the starting point of his work. The 1962 paper's opening statement that 'The first problem that a naval architect faces when he starts to design a ship is the selection of main dimensions suitable for the development of a design meeting all the specified requirements' was true in one sense but now appears somewhat superficial. The requirements of deadweight or capacity, of speed and range, of cargo handling facilities and of dimensional limitations have to be stated. From the view point of a shipyard's naval architect these may come as specified requirements, but for an

owner's naval architect or for a consultant, establishing these is the first stage in design. However well a design meets a set of requirements it may result in an unsuccessful ship if these have not been well selected. The transportation study to determine these should consider the economics of a number of solutions involving variations in ship numbers, ship sizes and speeds, against a scenario of changing freight rates, load factors and operating costs. The ship requirements finally selected should be a compromise between those which would maximise profits in the fat years and those which would result in the smallest losses in the lean years, the weighting between these depending on an assessment of which regime would predominate in the years of the ship's life which discounted cash flow methods show to be most important.

The large number of outline ship designs, capital and operating cost estimates required for such a study makes this a suitable subject for computer modelling. Indeed, whilst it could, in principle, be done by hand calculation methods, it was not until the advent of computers that the shipowner's hunch, based on operating experience with his existing ships, gave way to a rational approach.

#### 1.4 Design Methods

Many excellent papers on preliminary design have appeared in the last few years and some of these are given in the References. Preliminary design by its very nature is perhaps the most subjective aspect of naval architecture relying as it does on the accumulated experience and data of each practitioner. Whatever means are used to make these calculations, the methods on which they are based must be of sound principle, and must reflect established characteristics for the type of vessel which is being investigated. One of the aims of this paper is to restate these principles. The methods presented therefore are generally suitable for use either with the slide rule or calculators, or in computer programs, and we do not wish to argue the case for one method or the other at this stage. Both have their place with the balance of advantage lying with the computer when frequent repetition of a type of design is likely and with slide-rule when the design required has a high degree of novelty necessitating the exercise of judgement in choosing relationships and approximate formulae. However, as we shall suggest later, the use of the computer to make a large number of repetitive calculations may not be the best way to use this valuable tool.

## 2. THE THREE SHIP DESIGN CATEGORIES

From the aspect of choosing appropriate main dimensions, ships divide into three main categories:

- (i) The deadweight carrier
- (ii) The capacity carrier
- and (iii) The linear dimension ship

### 2.1 The Deadweight Carrier

The deadweight carrier is distinguished by the fact that its dimensions are determined by the equation:

$$\Delta = C_b L B T \times 1.025 (1 + s) = W_D + W_L \quad (1)$$

where	L	=	Length BP in metres
	B	=	Breadth mld. in metres
	T	=	Load draught in metres
	C <sub>b</sub>	=	Moulded block coefficient at draught T on Length BP
	Δ	=	Full displacement in tonnes
	s	=	Shell, stern and appendages displacement expressed as a fraction of the moulded displacement
	W <sub>D</sub>	=	full deadweight in tonnes
	W <sub>L</sub>	=	lightship weight in tonnes

In the case of a deadweight carrier T is the maximum draught permitted by the geometric freeboard for the ship's dimensions and construction. It is noteworthy that the equation does not involve the depth of the ship, except in so far as it is implicit in the draught. A small increase in scantlings, a small reduction in bulkhead spacing and a few alterations in construction details may be sufficient to enable a particular value of T to be obtained with a reduced depth D with the resulting design having a reduced cargo capacity and stowage rate.

### 2.2 The Capacity Carrier

For the volume carrier the dimensions are determined by the equations:

$$V_h = C_{bD} L B D^1 = \frac{(V_r - V_u)}{(1 - S)} + V_m \quad (2)$$

where

- D<sup>1</sup> = Capacity Depth in metres
- D<sup>1</sup> = D + c<sub>m</sub> + s<sub>m</sub>
- D = Depth moulded in metres
- c<sub>m</sub> = Mean camber in metres = 2/3c for parabolic camber
- s<sub>m</sub> = Mean sheer in metres = 1/6 (s<sub>f</sub> + s<sub>a</sub>) for parabolic sheer
- C<sub>bD</sub> = Block coefficient at the moulded depth
- V<sub>h</sub> = total volume in m<sup>3</sup> of the ship below the upper deck, and between perpendiculars.
- V<sub>r</sub> = Total cargo capacity (m<sup>3</sup>) required.
- V<sub>u</sub> = Cargo capacity (m<sup>3</sup>) available above the upper deck
- S = Deduction for structure in cargo space expressed as a proportion of the moulded volume of these spaces.
- V<sub>m</sub> = Volume required for machinery, tanks etc. within the volume V<sub>h</sub>

In this equation, it is significant to note the absence of the draught T as a factor, although it is implicit as a second order term in the difference between the value of C<sub>bD</sub> and the value of C<sub>b</sub> at draught T which is established by the form required to suit the speed length ratio of the ship.

### 2.3 The Linear Dimension Ship

The linear dimension ship is distinguished by the fact that its dimensions are primarily fixed by considerations other than those of deadweight or of volume.

An example is the St. Lawrence Seaway ship where the beam limit of 22.86 m can lead to a very long slim ship with a high L/D value and for which the economic advantages of carrying a large deadweight or capacity of cargo through the canal offsets the penalties resulting from constructing a ship whose proportions are not economic for other services. The Panama Canal exercises a similar influence with a beam limit of about 32.2 m and a draught limit of about 13 m depending on season. The distortion from normal ship proportions has not been as great as that caused by the St. Lawrence Seaway locks, but there is the same trend.

For the largest tankers, the depth of the ocean itself in some of its shallower areas such as the Dover and Malacca Straits limits the draught of such vessels to about 23 m resulting in lower L/B ratios than would have been considered if this limitation had not applied.

In addition to ships influenced by external factors, there are a number of ship types whose dimensions are determined primarily by the unit size of the cargo they carry. Container ships are probably the most obvious example. For this type

of ship the beam and depth are the first dimensions to be fixed, determining the number of containers which can be carried in the midship section of the ship, and the length of the ship is then adjusted to accommodate the total numbers. As there is within limits, an optimum length/beam relationship, steps develop in the numbers of containers for which optimum ships can be designed.

The breadths of car ferries and of train ferries are similarly tailored to accommodate a number of lanes of vehicles, with the result that there are fairly distinct steps in the beam of ships of those types. As each beam value is associated with appropriate values of depth and length there tends to be optimum and non-optimum car numbers.

## 2.4 Solution of Cubic Equations

Equation (1) which involves three dimensions and the block coefficient  $C_b$  (which has a complex relationship with the speed and length of the ship) is readily solved by assuming three ship lengths and associating with each of these an appropriate beam, draught and block coefficient to obtain a displacement. If the lightship weight is then calculated for each ship and subtracted from the displacement, three values of deadweight are obtained. If these are then plotted on a base of length, the required ship's length can be read against the specified deadweight.

A solution of equation (2) can be obtained in a similar manner, with capacity depth replacing draught, the volume required for machinery, tanks etc. replacing the lightship weight and the required cargo capacity replacing deadweight.

Although both methods of calculation are extremely flexible and permit allowances to be made for special features required in the ship such as an unusual weight in a deadweight carrier or an unusual space requirement in a volume carrier, they do involve designing three ships in order to arrive at the dimensions of the required vessel. If this process is to be accomplished quickly the designer must have available, in a well marshalled format, all the data required for the calculation.

Before taking the solution of these equations any further we must now consider the ways in which all the required data can best be presented and to consider how the various relationships presented in 1962 must be amended, if required, to meet the various developments described previously.

## 3. DIMENSIONS, DISPLACEMENT AND FORM

### 3.1 The Dimensional Relationships

There are six dimensional relationships linking the four main ship dimensions of  $L$ ,  $B$ ,  $D$  and  $T$ , and it is necessary to use three of these in order to solve equations (1) or (2).

The relationships are:

$$\begin{aligned} B &= f(L) & D &= f(L) \\ D &= f(B) & T &= f(L) \\ T &= f(D) & & \\ & & T &= f(B) \end{aligned}$$

Essentially a ship is a container, and as the straight-sided container which has the least surface area for a given volume is a cube, it appears that for economy of construction a ship should approach this shape as closely as the other considerations involved in ship design permit. An approach to a cubic shape requires that draught, the smallest of the dimensions, should be the maximum permitted by  $L$ ,  $B$  and  $D$ ; that depth the next smallest dimension, should be the maximum permitted to  $L$  and  $B$ , that the breadth should be the maximum permitted by  $L$ , and finally that  $C_b$  should be as full as possible.

The meanings of each of these relationships can now be considered.

### 3.2 The Beam/Length Relationship $B = f(L)$

There has been a steady decrease in the ratio  $L/B$  over the years as the pressure to reduce the capital cost of ships has increased and as tank testing has led to the development of

lines which enable these reductions in hull cost to be obtained with only a small acceptable penalty in powering.

However the extent to which beam can be increased for a given length of ship is still limited to ensure that the ship does not require excessive horsepower in relation to its displacement and speed, and also to ensure that the ship is directionally stable.

Fig. 1 shows a plot of beam against length for recent ships of a variety of types.

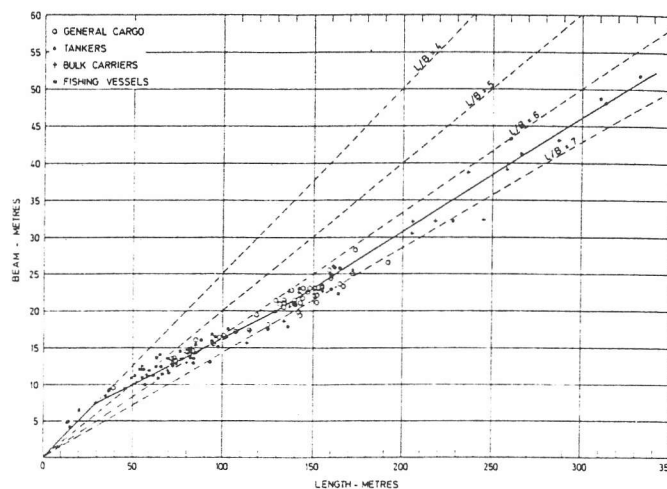


Fig. 1

In previous papers, these data have often been presented as series of different relationships for passenger ships, cargo ships and tankers, all of the form  $B = mL + c$ . We now see little basis for a formula of this type and a number of advantages in thinking in terms of  $L/B$  ratios.

There does not seem to be much reason for this relationship differing from one ship type to another except because different ship types tend to be concentrated in groups of different sizes and speeds. The values of  $L/B$  in 1962 varied between 6.6 and 7.3. Most recent practice shows a tendency to use an  $L/B$  value of about 6.5 for ships in excess of about 130 m in length and an  $L/B$  value of 4 for small craft such as fishing boats of up to 30 m in length. For vessels with lengths between 30 m and 130 m, which covers coasters and many general cargo ships,  $L/B$  varies according to the formula:

$$L/B = 4 + 0.025 (L - 30) \quad (3)$$

These figures would appear to indicate that an  $L/B$  value of about 6.5 is compatible in today's experience with the design of efficient lines and that in small ships where the installed power is in any case low it is found desirable to pay more for machinery and fuel to obtain the advantage of small dimensions to reduce hull cost and possibly to enable the ship to operate in restricted ports.

### 3.3 Depth/Beam Relationship $D = f(B)$

This relationship is primarily one which governs stability since  $KG$  is a function of depth and  $KM$  is largely a function of beam.

Fig. 2 shows a plot of depth against beam for a number of types of ships and indicates that there are two distinct groupings in the relationship between these two dimensions.

The first group which consists of volume carriers, comprising fishing vessels and cargo ships whose depth is limited by stability requirements, has a  $B/D$  ratio of about 1.65.

The second group which consists of deadweight carriers, comprising coasters, tankers and bulk carriers generally with stability well in excess of minimum requirements and



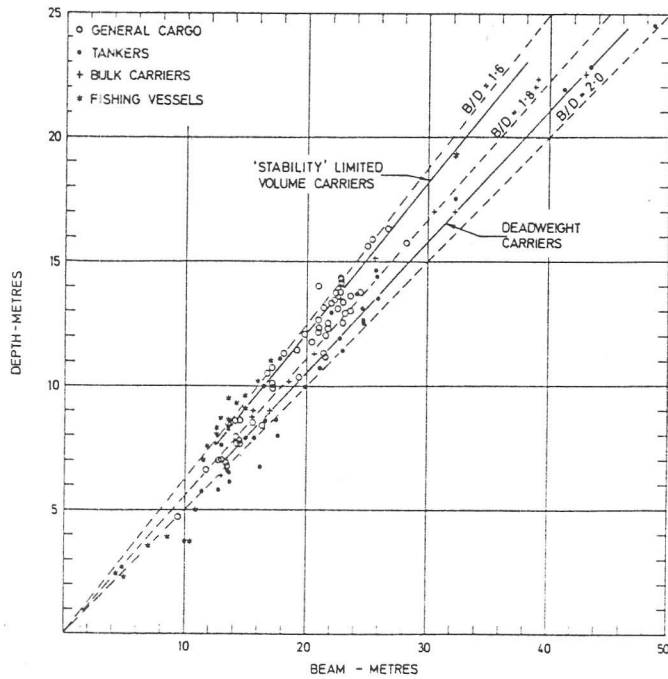


Fig. 2

depth determined by hull deflections has a B/D ratio of about 1.90.

In 1962 a formulae of the  $D = mB + C$  type was proposed for general cargo ships. Whilst formulae of this type have some theoretical justification in view of the two components KB and BM which make up KM, there are advantages in working with a B/D ratio, provided the value selected is an appropriate one for the type and size of ship under consideration.

For comparison with the current values of B/D it may be noted that the value of B/D presented in the 1962 paper varied from 1.5 for a large ship with 'moderate' stability to 1.8 for a small vessel with 'good' stability.

We do not have sufficient data on the ships used for the present plot to draw a distinction between 'moderate' and 'good' stability, but as all vessels are now required to meet a standard of stability which equates approximately with that regarded as good stability in 1962, this distinction has become academic.

The reduction in depth for a given beam implicit in the increased B/D ratios now used compared with 1962 practice will be noted; as should also the disappearance of the practice of ballasting fuel tanks which was one of the attributes of a ship of 'moderate' stability.

Factors which have brought about the changes in the value of B/D since 1962 and which should be considered when selecting the B/D for a new design are shown in Table II.

TABLE II

Requiring an increase in the ratio B/D	Permitting a reduction in the ratio B/D
<p>Faster speeds and finer lines resulting in reduced KM value for a given beam.</p> <p>Higher standards of stability.</p> <p>Reductions in main hull weight and in machinery weight.</p> <p>The carriage of deck cargo.</p>	<p>Lines which aim at a particularly high KM value.</p> <p>Reduction in sheer and camber.</p> <p>Reductions in the weight of superstructure and of cargo gear.</p> <p>Designs which aim at a high underdeck cubic with little or no deck cargo.</p> <p>Large ballast capacity in the double bottom.</p>

### 3.4 The Draught/Depth relationship $T = f(D)$

This relationship, which is the embodiment of the freeboard rules, has changed, primarily as a result of the 1966 Freeboard Convention, and secondly, as a result of the changes in length, block coefficient, sheer, camber and extent of erections which are now associated with a particular depth of ship.

In the new rules 'A' type freeboard replaces the old tanker freeboard, generally giving more draught for a given depth, and 'B' type freeboard represents the continuation of the old cargo ship freeboard with the bulk carrier being given the benefit of a deeper draught under the 'B-60' freeboard provided certain requirements which increase safety are met.

Under a dispensation of the freeboard rules, dredgers with hopper doors which can speedily dump their cargo in the event of an emergency are permitted to operate with a reduced freeboard. The values of these reduced freeboards are agreed for each case by National Administrations, taking into account the sea conditions in which the ship will be operating.

Fig. 3 comprises a plot of draught against depth for a number of ship types.

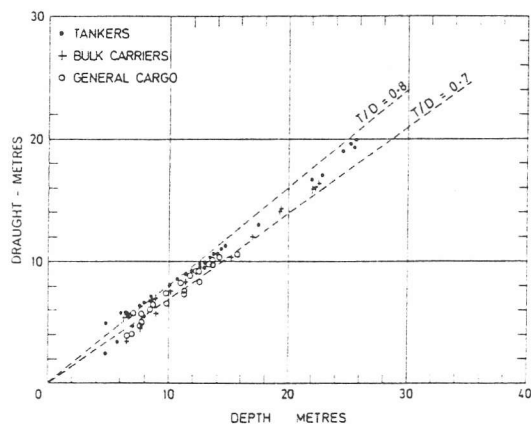


Fig. 3

A wise provision of the new tonnage convention brought an end to the old open shelter deck class of vessel with its undesirable features, but ships which would in the past have been of this type can still have a reduced tonnage, provided an increased freeboard, reduced draught and deadweight are accepted, by designating the second deck as the tonnage deck.

### 3.5 The Depth/Length relationship $D = f(L)$

It was shown in Fig. 2 that deadweight carriers have a higher B/D ratio than capacity carriers. This is because in these ships stability is greatly in excess of requirements and

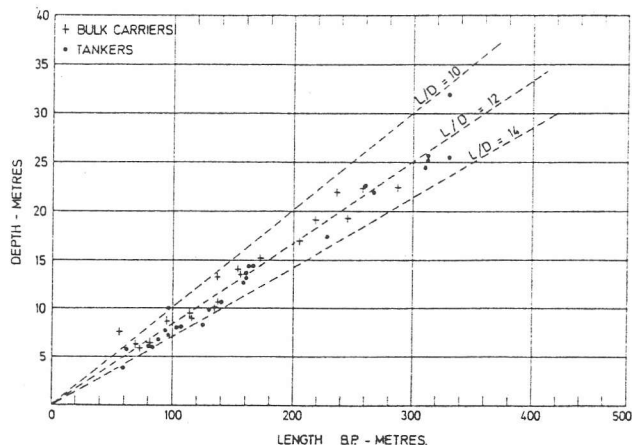


Fig. 4

depth and beam are therefore independent variables. For these ships, control of the value of  $D$  is exercised by the ratio  $L/D$  which is significant in relation to the structural strength of the ship and particularly to the deflection of the hull girder under the bending moments imposed by waves and cargo distribution. The largest  $L/D$  values are used on tankers which of all ships have the most favourable structural arrangements—longitudinal framing on bottom deck, ship sides and longitudinal bulkheads and the minimum of hatch openings.

When higher tensile steel is used to save weight, it is generally desirable to use a smaller  $L/D$  value in order to limit the deflection of the hull girder.

Fig. 4 shows  $L/D$  ratios for a variety of ship types.

### 3.6 The Draught/Length relationship $T = f(L)$

This is essentially a secondary relationship resulting from either of the following combinations of relationships:

$$\begin{array}{l} T = f(D) \} \\ \text{and } D = f(L) \} \end{array} \quad \text{or} \quad \begin{array}{l} T = f(D) \} \\ D = f(B) \} \\ \text{and } B = f(L) \} \end{array}$$

### 3.7 The Draught/Beam relationship $T = f(B)$

Again a secondary relationship, resulting in this case from either of the following combinations of relationships:

$$\begin{array}{l} T = f(D) \} \\ \text{and } D = f(B) \} \end{array} \quad \text{or} \quad \begin{array}{l} T = f(D) \} \\ B = f(L) \} \\ \text{and } D = f(L) \} \end{array}$$

The numerical values of these relationships are different for different types of ship, for a number of reasons, some of which have already been mentioned, and some of which will become apparent later.

### 3.8 Block coefficient

The only remaining factor required to obtain the relationship between dimensions and displacement is the block coefficient which has a complex relationship primarily with length and speed and also with beam and draught.

In the 1962 paper the block coefficient was obtained from the Alexander relationship of the form:

$$C_b = K - 0.5 V/\sqrt{L_f}$$

with  $K$  varying from 1.12 to 1.03 depending on  $V/\sqrt{L_f}$ .

In the discussion of that paper, Conn suggested a number of alternative formulae which appeared to have merit—notably Telfer's proposal which brought in  $L/B$  as a variable and Troost's which made a useful distinction between single and twin screw ships.

The recent significant reduction in  $L/B$  ratio together with the increase in the average size of ships seems, however, to make a new approach to the block coefficient relationship desirable. It was therefore with great interest that we studied the ideas presented by Katsoulis<sup>(2)</sup>.

Katsoulis suggested that  $C_b$ , as well as being a function of  $V/\sqrt{L}$ , should also be a function of  $L/B$  and of  $B/T$ , since both of these affect the resistance of the ship and the flow of water to the propeller (and hence both the QPC and the likelihood of avoiding propeller induced vibrations). He suggests an exponential formula for  $C_b$  of the form:

$$C_b = K f L^a B^b T^c v^d \quad (5)$$

where

$K$  = constant

$f$  = correction factor for a particular ship type.

He then shows that this can be transformed into:

$$C_b = k f (V/\sqrt{L})^d \left[ \frac{L}{B} \right]^{-b-c} \left[ \frac{B}{T} \right]^{-c} L^{a+b+c+d/2} \quad (6)$$

From a regression analysis Katsoulis deduced values of the constants in the equation. Unfortunately when we used

Katsoulis' equation and constants to calculate the block coefficients of a wide variety of ships for which we had good data, we did not obtain satisfactory agreement with the actual block coefficients, particularly for the ships at the extreme ends of the range of dimensions (crude carriers of over 250 m and ships of 100 m or less).

We therefore decided to plot the block coefficients of as many ships for which we could obtain the data—against a suitable base—the obvious one being  $F_n$  or  $V/\sqrt{L}$ , (Fig. 5(a)). This presupposes that the many naval architects concerned with these designs managed somehow or other to fix values of block coefficient for their designs which were not too far away from optimum. With a few exceptions all the values were found to be within a band of  $\pm 0.025$  from the main  $C_b$  line whilst a majority of the points lie within much closer limits.

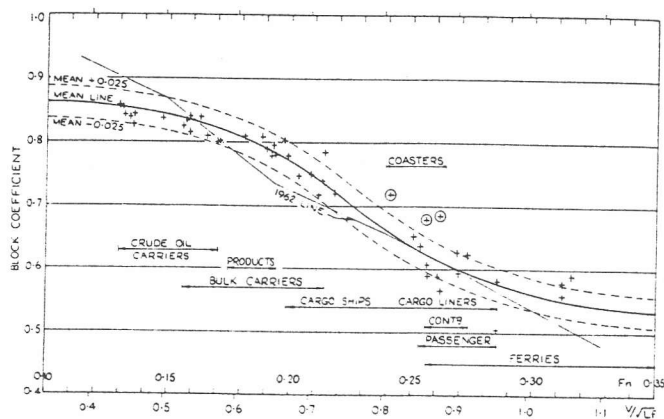


Fig. 5(a)

We were disappointed to find that we could not detect a significant effect of  $L/B$  and  $B/T$ , although it must be noted that different types of vessels, each of which generally has its own particular range of  $L/B$  and  $B/T$ , tend to be concentrated at different parts of the  $V/\sqrt{L}$  range. The same applies to twin screw propulsion which is generally confined to high speed cargo liners, passenger ships and ferries.

The types of ships used in the plots are indicated showing the areas in which they predominate. We believe ship type has some significance in relation to selection of  $C_b$  because of the variation in practice relating to service speed, margins and engine derating adopted in different classes of ships.

For bulk carriers for example it is usual to quote a service speed based on the maximum continuous power which the machinery develops with only a small margin for weather and fouling. Cargo liner owners, however, take a much more conservative view and quote speeds which can be obtained with a quite large margin for weather and fouling and with the power limited to a service rating which may be only 85% or 90% MCR.

The line corresponding to the 1962 variant of the Alexander formula (equation(4)) is shown on Fig. 5(a) for comparison purposes and indicates the extent to which bulk carriers have fuller block coefficients than were anticipated at that time, whereas the change for other ship types has been less striking. Whilst part of the explanation of this no doubt lies in the considerable development of tank tested forms in the 0.75 to 0.80  $C_b$  range, it is suggested that the different attitude to speed on the part of bulk carrier owners is probably at least as significant a factor.

### 3.9 Displacement

In order to obtain the full displacement at the desired draught, it is necessary to make a small correction to the moulded displacement to allow for shell and appendages.

Whilst this is a comparatively small factor in the displacement calculation, it can be important in ships where the deadweight is small and margins are tight to have a good approximation for these items, at least in the later stages

when the design is being refined. If this can be done easily, there seems every reason to use these same approximations in the preliminary design stage.

For a single screw ship with an all welded shell, the simplest approximation is  $\frac{1}{2}\%$  of the moulded displacement.

Again if draught is limited then keel thickness should be allowed for in any comparison between the derived moulded draught and that permitted for the design.

### 3.10 Appendages

If a more exact estimate of appendage displacement is required, the various appendages should be considered individually.

#### (i) Shell displacement

$$= \frac{t}{380} \sqrt{\Delta L}$$

$t$  = mean shell thickness (mm)

#### (ii) Stern displacement

$$= \left[ \left( \frac{T}{H} \right)^x - 1 \right] \frac{\Delta}{1000}$$

where  $x = 2.5$  for 'fine' sterns

$x = 3.5$  for 'full' sterns

$H$  = height of counter

#### (iii) Twin screw bossing displacement

$$= 1.10d^3$$

where  $d$  = propeller diameter

Constant can vary from 0.7 for fine bossings to 1.4 for very full bossings.

#### (iv) Rudder displacement

$$= 0.13 \text{ area}^{3/2}$$

#### (v) Propeller displacement

$$= 0.01d^3$$

Other items which may affect the displacement are bow and stern thrust tunnels, the lost buoyancy in stabiliser fin stowage recesses and in the recesses for dredge pipe trunnion slides. All of these are, however, in the authors' opinion, better considered as added weights in the design stage, although for 'as fitted' documentation a 'lost buoyancy' treatment is usually advisable.

### 3.11 Longitudinal Centre of Buoyancy

Before going on to discuss preliminary power estimates, it is worth disgressing to consider the longitudinal position of the centre of buoyancy which is closely associated with block coefficient in the determination of the power required to drive a ship of given dimensions and displacement at a given speed. Many naval architects and some hydrodynamicists think of this in terms of figures and ignore its physical significance in determining the shape of ships' lines.

Fig. 5(b) shows how the LCB moves as the  $C_b$  changes from 1.0 to zero. For the fullest 'ship-shaped' ship, most of the fining takes place at the aft end to ensure flow to the propeller, the fore end remaining relatively full and the LCB in consequence being well forward. Once the run is such that it provides a satisfactory flow to the propeller, it is only necessary to fine it very gradually as the block coefficient is further reduced for ships with higher speeds and higher powers. The forebody, on the other hand, changes from being fuller than the afterbody to being markedly finer, and the LCB therefore progressively shifts to a position well aft of amidships. For very fine lined ships there is finally a tendency for the LCB to return to amidships.

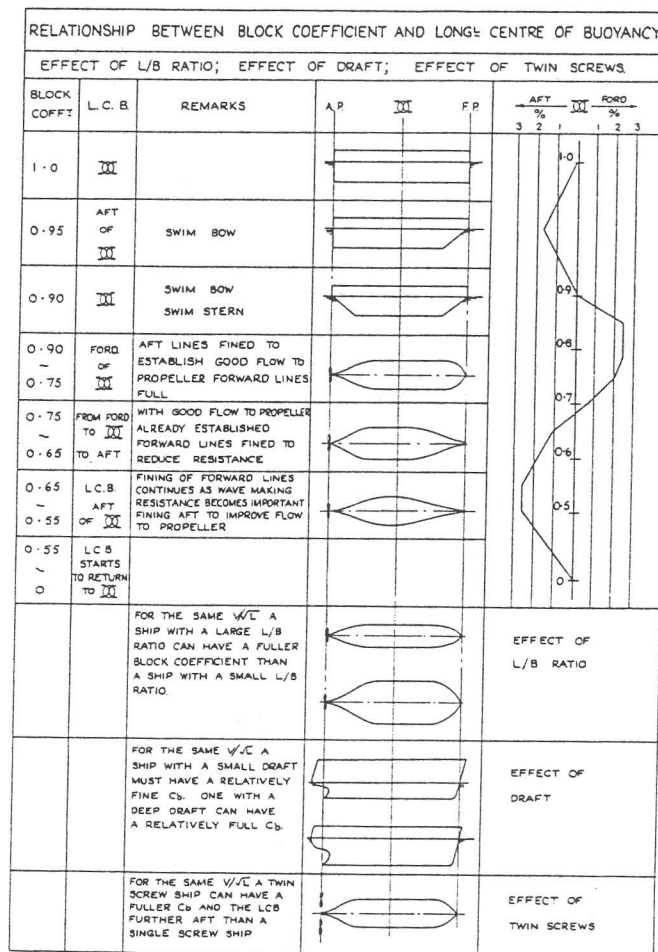


Fig. 5(b)

## 4. POWERING

There is now so much excellent data available, and so many specialist papers on the basic part of powering which leads to an estimate of the effective horsepower, that it would be wrong to attempt to deal with this subject in more than brief outline in a paper ranging over the whole field of ship design. We shall therefore confine our comments to discussion of the principal factors which affect each of the components which make up the total power estimate.

### 4.1 Effective horsepower $P_E$

Since 1962 model testing has resulted in improvements in ship's lines giving considerable reductions in the value of  $C$  which can be obtained for a given speed, dimensions and displacement. Each practitioner will have his own library of tank tests from which he can obtain  $C$  values; some particularly useful published data is, however, given in the papers mentioned in the bibliography (3,4,5,6).

In the 1962 powering formula, no correction was made for departures from standard proportions of  $L$ ,  $B$ ,  $T$ . More recent developments in ship design have led to ships of quite extreme proportions and it has become important to correct for these. One simple, convenient and relatively accurate method of correction is by the use of Mumford Indices; values of which are given in some of the papers recommended.

Caution however should be exercised when applying this method to designs with beams and draughts varying more than 15% and 10% respectively from those for the basis vessel.

The value of  $C$  must be corrected for the difference in length between the basis ship and the design under consideration. In the Froude notation, the following formula which is

simpler than that normally used, but is substantially accurate, can be applied.

$$\odot_{L_1} - \odot_{L_2} = 4 (L_2 - L_1) 10^{-4} \quad (12)$$

#### 4.2 Adoption of ITTC Notation

A change not yet universally adopted is the formulation of  $\odot$  and through it  $P_E$  and  $P_S$  on an ITTC basis in lieu of Froude basis. Whilst the ITTC basis has, in the opinion of the experts, a better scientific base and its use is expected eventually to lead to better power predictions, there is no doubt that a far greater amount of available data is in the old Froude notation. We have therefore allowed in our calculation sheets for its continued use if desired.

#### 4.3 Appendage Resistance

Additional allowances, usually expressed as a percentage of the naked  $P_E$  must be added for appendages, such as twin screw bossings or 'A' brackets, twin rudders, bow rudder, bow or stern thrusters, stabiliser fins or recesses. Each of these needs to be carefully assessed in relation to its design and the extent to which it is faired into the lines of the ship. The allowances which we would use for well designed appendages are:

Twin screw bossings	8-10%
'A' brackets	5%
Twin rudders	3%
Bow thruster	2-5%
Ice knife	0.5%

Experiment tanks differ in their treatment of the appendage resistance as measured on models, some applying the whole resistance as measured (NPL) others believing that it should be reduced by 50% to allow for a scale effect. It is important that the ship model correlation factor  $(1+x)$  used is consistent with the treatment of appendage resistance used. In this paper we have worked to NPL practice in both cases.

#### 4.4 Correlation Factors

In 1962 shell plating construction varied from flush welded to all riveted construction with many ships using an intermediate form of construction with riveted seams and welded butts and frames. The ship model correlation factors used were designed to take account of these. General adoption of flush welded shell did not lead to the simplification which might have been expected, since it was found that the ship model correlation factors derived using the Froude method, were very much lower for the larger ships that were being introduced in this period and that the classical predictions methods left something to be desired. Discussions of the problem between the principal ship model experiment tanks led to the general adoption of the ITTC extrapolator which has substantially reduced this length effect. However as has already been stated, naval architects continue to have much more data available in the Froude notation and it is therefore necessary to allow for correlation factors using both methods.

Values of  $(1+x)$  in ITTC notation are given in Refs. 7 and 8 and for Froude notation are given in Ref. 9. The use of correlation factors from the last reference will result in a more pessimistic estimate of shaft horsepower than would be obtained using ITTC factors, but this difference can be absorbed in the margin used to evaluate trial horsepower.

#### 4.5 Quasi Propulsion Coefficient

Improvement in propeller efficiency has been less notable and the Emerson formula remains a quick and reliable method of estimating QPC. In metric units it can be written as:

$$QPC = \eta_D = \frac{K - N\sqrt{L}}{10,000} \quad (13)$$

As the answers from the original formula are a little low for modern propeller designs, we have changed the constant

K to 0.84. This formula has frequently been found by the authors to give as good an estimate of the QPC as would be obtained from the synthesis of the various components. As a result of this experience we have often extended the use of the formula for length of ship and propeller RPM substantially beyond the range of these parameters used in its original derivation<sup>(10)</sup>; we would also extend its use to twin screw ships after adjusting the value of constant K from suitable tank test data. For controllable pitch propellers a reduction in  $\eta_D$  of about 0.02 appears appropriate.

The Emerson formula incidentally shows quite clearly that the best way to improve the propulsive efficiency is to reduce the RPM and it is therefore somewhat surprising that it has taken so long for this idea to be taken to the logical conclusion recently proposed by Burmeister and Wain and others.

#### 4.6 Transmission Efficiency

The reduction in the length of shafting, and therefore in the number of bearings with the change from machinery amidships to machinery  $\frac{3}{4}$  aft or all aft, together with the improvement in stern tube lubrication has reduced the frictional losses and we would now use a figure of 1% for aft end installations and 2% for others, compared with a value of 3% assumed in 1962. When appropriate, a further 3-4% would be allowed for gearing losses in medium speed diesel installations.

#### 4.7 Trial and Service Margins

The first stage in a powering calculation leads to a technical estimate of the power required on trial. If there are penalties on the attainment of a trial speed it is usually wise to provide a margin of power over this, 5% being a usual figure.

More significant, however, is the margin which must be provided over the power required for a specified speed in ideal trial conditions to allow for the same speed to be obtained in service conditions of fouling and weather.

The percentage to be allowed for this is dependent on the paint system used, whether cathodic protection is fitted, the interval between dry dockings, the voyage pattern and time spent in port particularly in the tropics, the weather conditions experienced on the trade route, and the importance of maintaining a particular speed or schedule. With all these factors involved, there are clearly significant differences in the appropriate 'service allowance'. This is a matter which an owner must specify if he wishes any increase from the usual practice adopted by shipyards of allowing a service margin of from 15% to 20% in their calculations of the required Continuous Service Power.

#### 4.8 Engine De-rating

The last item to be considered before selecting an engine is the vexed question of de-rating. Manufacturers of marine diesel engines in general quote a power rating for their engines which they call the Maximum Continuous Rating (MCR). This, as the name implies, is a power which the engine can develop continuously over considerable periods. However, experience has shown that the maintenance costs of many, if not all, engines can be significantly reduced if the engines are never operated above a certain percentage of these MCR values. Views amongst marine engineers vary on what percentage should apply to each make of engine, but the figures of 90%, or even 85%, are commonly accepted as being good practice.

### 5. LIGHTSHIP WEIGHT

The lightship weight  $W_L$  is composed of steel weight + outfit weight + machinery weight + margin.

The following paragraphs deal with methods of estimating each of these.



### 5.1 Steelweight— $W_s$

In the 1962 paper the use of Lloyd's equipment numeral was advocated as a basis for a graph of steelweight in preference to the numerals  $L \times B \times D$  or  $L \times (B + D)$  in more common use at that time. The reasons given for this were that the equipment number introduced allowances of approximately the correct order for changes in draught and in the extent of erections, and avoided the choice of the deck to which 'D' was measured being critical as it was in the other numerals.

The Lloyd's Equipment numeral of 1962 no longer takes any part in the determination of ships anchors and cables, hawsers and warps having been replaced for this function in 1965 by a new numeral, which was agreed by the Classification Societies to be a more rational measure of the wind, wave and current forces which might act on a vessel at anchor. The new numeral may have merit for its primary role, but it is not a suitable parameter against which to plot ship steelweights.

Since 1962 several numerals have been suggested as a basis for calculations of steelweight. Some of these numerals have a scientific basis and give good results for the ship types for which they were developed, particularly if they are used as a proportioning parameter applied to the known steelweight of a basis ship. None of these formulae, however, seem to be as suitable as  $E$  as a parameter applicable to a wide range of ship types.

$$E = L(B + T) + 0.85 L(D - T) + 0.85 \Sigma l_1 h_1 + 0.75 \Sigma l_2 h_2 \quad (14)$$

where  $l_1$  and  $h_1$  = length and height of full width erections

where  $l_2$  and  $h_2$  = length and height of houses

For ordinary cargo ships an allowance of 200-300 can be used for the erections, if the extent of these is not yet known (metric units).

If we had been devising a numeral specifically for this purpose it is probable that we would have chosen slightly different constants, but having collected data in the  $E$  form for many years, we have decided to continue with it in its original form.

The question of whether it is better to plot invoiced or net steel weights is a matter worthy of some debate. The net weight is the weight which is initially arrived at by detailed calculations, based on ships plans, and it is the weight which is required for the deadweight calculation. The invoiced steel weight is the weight recorded in the shipyards steel order books and the one used for cost estimates. In 1962 the invoiced steelweight was the one that was known more accurately and was therefore presented at that date. As it is current practice in many shipyards to weigh each unit before erection on the berth, it is often equally possible to obtain an accurate estimate of the net steelweight. As consultants, it is net steelweight information that we most commonly receive or calculate and we have therefore used it in our steelweight graph.

Since the  $E$  parameter attaches no significance to the fullness of the ship, which clearly has an appreciable effect on the steelweight, all steelweights are corrected to a standard fullness before plotting.

In a similar manner steelweights read from the graph must be corrected from the standard fullness to the desired block coefficient.

The standard fullness is set at  $C_b = 0.70$  measured at  $0.8D$ . Corrections to the steelweight for variation in  $C_b$  from  $0.70$  are made using the following relationship:

$$W_s = W_{s7} [1 + 0.5(C_b^1 - 0.70)] \quad (15)$$

where

$W_s$  = steelweight for actual  $C_b^1$  at  $0.8D$ .

$W_{s7}$  = steelweight at  $C_b^1$  of  $0.7$  as lifted from graph.

The calculation of  $C_b^1$  at  $0.8D$  from the known value of the load draught may be made using the empirical formula:

$$C_b^1 = C_b + (1 - C_b) \frac{(0.8D - T)}{3T} \quad (16)$$

There is some ambiguity in our treatment of block coefficient in various sections of this paper. Previously we have used a  $C_{bD}$  which is measured at the moulded depth of the ship and had intended to use it throughout the paper, both for uniformity and because it has a better theoretical basis. However, we found ourselves committed to the use of  $C_b$  at  $0.8D$  in this section because our accumulated data was on this basis.

The great increase in size which has taken place in VLCCs has necessitated the extension of the steel weight graph to  $E$  values almost 3 times greater than those plotted in 1962. At the same time, we have found our interest extending to ships much smaller than were dealt with in the earlier paper. A convenient solution to the problem of achieving reasonable accuracy in the figures for small ships, whilst at the same time accommodating the largest vessels in the one graph, is provided by the use of a log-log scale, as shown in Fig. 6.

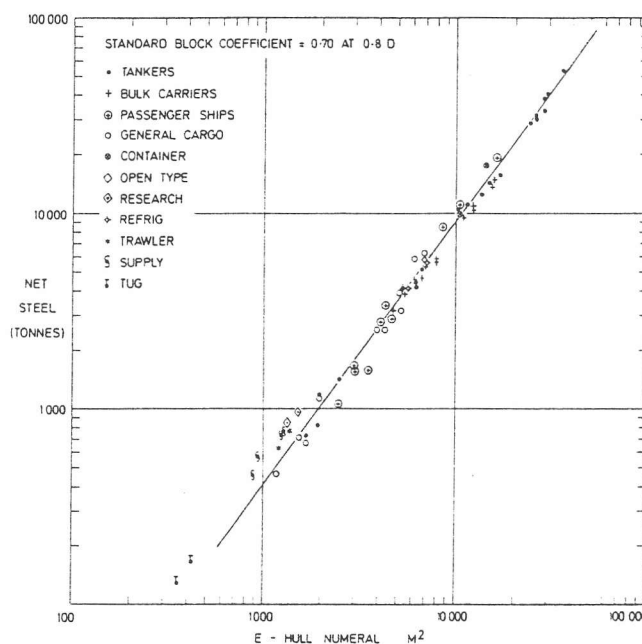


Fig. 6

The accuracy with which a steelweight can be read off this graph is limited by the scale at which it can be drawn and by the difficulty in showing the number of fairly closely spaced lines which apply to different types of ships. Formulae for various ship types, and the limits within which these are considered to be valid are as follows:

$$W_{s7} = K \cdot E^{1.36} \quad (17)$$

where values of  $K$  are given in Table III.

The change from tons to tonnes, the metrication of  $E$  and the altered presentation, means that the reduction in steelweight for the same equipment numeral since 1962 is not immediately apparent. As this is some measure of the science's advance during the period, it may be of interest to note that we have evaluated it as being of the order 15-20%.

As nearly all the data in the 1962 paper related to all welded ships, the use of welding accounts for only a small part of this reduction. The factors which in our view contribute to the change are, (not necessarily in order of importance):

The changes in the ratios  $L/B$ ,  $B/T$ ,  $D/B$ , which have occurred mean that a modern ship will have a shorter

length, but larger beam and depth than a 1962 ship with the same E.

The reduction in the extent to which 'Owners Extras' are specified for modern ships.

The reduction and simplification of internal structure in modern ships—fewer decks in cargo ships, fewer bulkheads in tankers.

The simplification of superstructure, resulting in the elimination of the overhanging decks which were a regular feature in the ships of fifteen years ago.

The rationalisation of Classification Society Rules and the reduction in scantlings which have followed.

Changes in demarcation of work. Patent steel hatch covers were at one time commonly manufactured in shipyards and therefore included in the steelweight. In more recent years they have invariably been manufactured by specialists and as 'bought' items are now in shipyard outfit.

Mention has been made of alternative steelweight estimating procedures and it may be appropriate to comment on some of these which are usefully summarized by Fisher<sup>(11)</sup>.

Most of the formulae quoted appear to have been derived by regression analysis techniques and the indices allotted to the various dimensions of L, B, D and  $C_b$  vary widely. In many cases the resultant figures appear to have little physical significance.

TABLE III

Type	Value of K	for	No. of ships in sample
Tankers	0.029-0.035	1500 < E < 40000	15
Chemical tanker	0.036-0.037	1900 < E < 2500	2
Bulk carrier	0.029-0.032	3000 < E < 15000	13
Open type bulk Container	0.033-0.040	6000 < E < 13000	3
Cargo	0.029-0.037	2000 < E < 7000	6
Refrig.	0.032-0.035	E ÷ 5000	3
Coasters	0.027-0.032	1000 < E < 2000	6
Offshore supply	0.041-0.051	800 < E < 1300	5
Tugs	0.044	350 < E < 450	2
Trawler	0.041-0.042	250 < E < 1300	2
Research vessels	0.045-0.046	1350 < E < 1500	2
Ferries	0.024-0.037	2000 < E < 5000	7
Passenger	0.037-0.038	5000 < E < 15000	4

If one may generalise on weight estimating methods, it appears that these fall into two main categories—a method based on volume and a method based on beam analogy. The truth appears to lie somewhere between—with part of the weight being volume dependent and part modulus dependent—a concept recognised by Eames and Drummond<sup>(12)</sup> and by Sato<sup>(13)</sup>. Both of the authors of the present paper had at various times investigated methods for tankers and bulk carriers, and we now decided to look into this concept more closely.

Fig. 7 compares the midship section of a ship with the cross section of an I beam.

The hull steel weight per metre =  $\rho(A_D + A_B + A_S + A_T)$  (18) where

$A_D$  = Area of deck plating + deck longls + other longl matl above 0.9D

$A_B$  = Area of bottom plating + bottom longls + other longl matl below 0.1D

$A_S$  = Area of shell + longls plus area of longl blds + longls between 0.1D and 0.9D

$A_T$  = Wt of transverse material per metre of ship's length

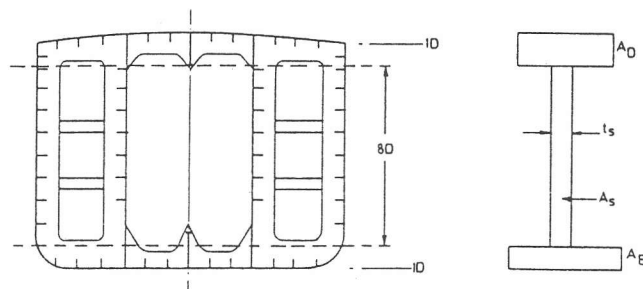


Fig. 7

The deck modulus of the hull girder is:

$$Z = \frac{A_D h_d^2 + A_B h_b^2 + 1/12 t_s (0.80D)^3}{h_d} \quad (19)$$

let  $h_d = k_d D$ ;  $h_b = k_b D$

Then

$$A_d + \left(\frac{k_b}{k_d}\right)^2 A_b + A_s \left(\frac{1}{4.35k_d}\right)^2 = \frac{Z}{k_d D} \quad (20)$$

If we approximate  $k_b = k_d = 0.50$

then:

$$A_D + A_B + \frac{A_S}{5} = K \left[ \frac{Z}{D} \right] \quad (21)$$

$$\text{Hence steel weight per metre} = \rho \left( K \left[ \frac{Z}{D} \right] + \frac{4}{5} A_S + A_T \right) \quad (22)$$

$$\text{Hence steel weight of hull} = \rho I_f L \left[ K \left[ \frac{Z}{D} \right] + \frac{4}{5} A_S + A_T \right] \quad (23)$$

where  $I_f$  = integration factor which is a function of  $C_b$

$$\text{and } Z = C_1 L^2 B (C_b + 0.7) \text{ cm}^3 \quad (24)$$

(to Lloyds 1976 rules for ships with a still water bending moment not exceeding approximately 70% of the wave bending moment).

In this formula  $C_1$  is not a constant but varies from 7.84 to 10.75 as L changes from 90 m to 300 m, at which point it becomes substantially constant. For present purposes we intend to treat it as a constant, but it could provide the explanation for the slightly higher index of L which Sato suggests in his formula.

$$A_S = 0.80 t_s D \text{ and } t_s = f(L) \text{ from which } A_S = K(L \times D)$$

$$A_T = f(B \times D) \text{ or possibly } = f(B + D)$$

$$\text{Hence steel weight of hull} = \rho f(C_b) L \left[ m_1 C_b L^2 \frac{B}{D} + m_2 L \times D + m_3 (B \times D) \right] \quad (25)$$

To obtain the total steel weight it is necessary to add three more items, (i) the weight of bulkheads and casings (ii) the weight of platform decks and flats and (iii) the weight of superstructure, masts and deck fittings. The most rational expression for these appear to be:

$$(i) \text{ bulkheads } m_4 \times C_b \times L \times B \times D$$

$$(ii) \text{ platform decks } m_5 C_b L^2 B$$

(iii) superstructure  $m_6 (V)$  or  $m_6 B^2 L$

where  $V$  = volume of superstructure

An expression for hull weight may be deduced as:

$$\begin{aligned} & \left[ \begin{array}{l} \text{Modulus} \\ \text{related} \end{array} \right] + \left[ \begin{array}{l} \text{Side shell} \\ \text{and} \\ \text{Longitudinal} \\ \text{Bulkheads} \end{array} \right] + \left[ \begin{array}{l} \text{Transverse} \\ \text{Frames} \\ \text{Beams} \\ \text{Bulkheads} \end{array} \right] \\ & + \left[ \begin{array}{l} \text{Platform} \\ \text{Decks} \\ \text{and} \\ \text{Flats} \end{array} \right] + \left[ \begin{array}{l} \text{Superstructure} \\ \text{and} \\ \text{deck fittings} \end{array} \right] \end{aligned}$$

$$W_s = n_1 \frac{L^3 B}{D} (C_b)^x + n_2 L^2 D (C_b)^y + n_3 L B D (C_b)^y + n_4 L^2 B (C_b)^z + n_5 (v) \text{ or } n_5 B^2 L \quad (26)$$

In this expression the indices of  $C_b$  in the various terms have been left as alphabetical symbols. It would appear from inspection that 'x' might have a value close to unity as it has components both from the integration factor and from Lloyd's modulus formula, both 'y' and 'z' are clearly fractional indices.

An extrapolation on log-log paper of information on integration factors available to us as an extension of the data given in Fig. 20 indicated that overall the steelweight is proportional to the square root of the block coefficient. If it is accepted that for one type of ship the dimensions  $L$ ,  $B$  and  $D$  are related, this formula can be simplified to:

$$W_s = C_b^{1/2} L B \left[ K_1 L \cdot \frac{L}{D} + K_2 D \right] \quad (27)$$

which has one modulus related and one volume related term. The similarity which this bears to Sato's expression will be noted:

$$W_s = (C_b)^{1/3} \left[ w_1 L^3 \cdot \frac{B}{D} + w_2 L^2 (B + D)^2 \right] \quad (28)$$

We have not yet been able to determine values of  $K_1$  and  $K_2$  for various types of ship, but believe an investigation into this will lead to more accurate steel weight estimation.

## 5.2 Scrap

Although we have now presented our steelweight data as net weights, it is still necessary to consider the scrap allowance required to produce the invoiced weight used in estimating the cost.

In 1962, 12% of invoiced steel was suggested as a suitable scrap figure. For the wider range of ship sizes and types now being considered, a single scrap figure is no longer sensible.

The factors which affect the scrap deduction include:

Shipyard ordering methods—the use of standard plates, the necessity of ordering sections for stock to ensure supply when required.

Shipyard constructional methods—the allowance of overlaps on prefabricated units to cut at the ship to ensure a good fit; the use of optical and numerical methods involving nesting procedures. Extra lengths on sections to suit the operation of cold frame benders.

The effect of the increased cost of steel in enforcing economy in its use.

The skill of draughtsmen in utilising material, particularly in nesting of plates.

The accuracies of the calculations or the weighing methods employed to assess both invoiced and net weights.

The type of ships constructed and, in particular, their fullness of form.

An investigation showed block coefficient to be the main determining factor, with small ships and those with complicated structure showing an increase above the average. A plot is given in Fig. 8.

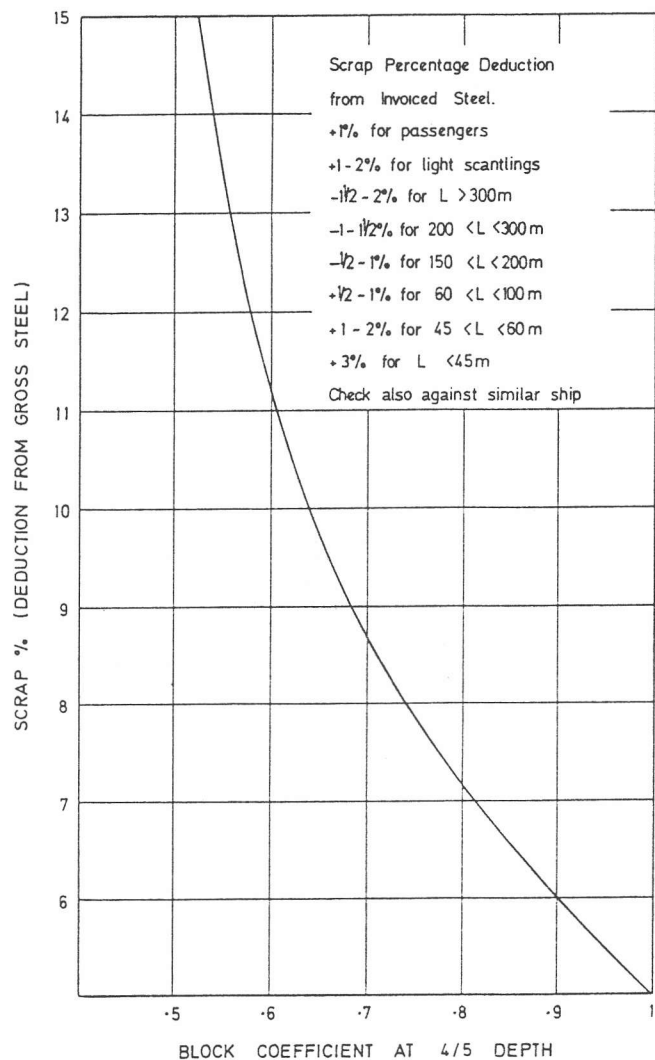


Fig. 8

In making up the lightship weight an addition of 1% should be made to this net steel weight to allow for weld metal deposited and the rolling margin on the steel.

## 5.3 Outfit Weight

The factors which have affected outfit since 1962 are:

### Leading to increases in weight

Higher standards of crew accommodation	All ships
Fitting of air conditioning, sewage systems	Most ships
Fitting of more sophisticated cargo gear	Cargo ships
Stabilisers, bow thrusters	Passenger ships
Patent steel hatch covers now in outfit	General cargo ships and bulk carriers

### Leading to reduction in weight

Reduction in weights of most deck machinery for same duty	All ships
Reduction in weight of deck coverings, elimination of wood decking, ceiling and most sparring	General cargo

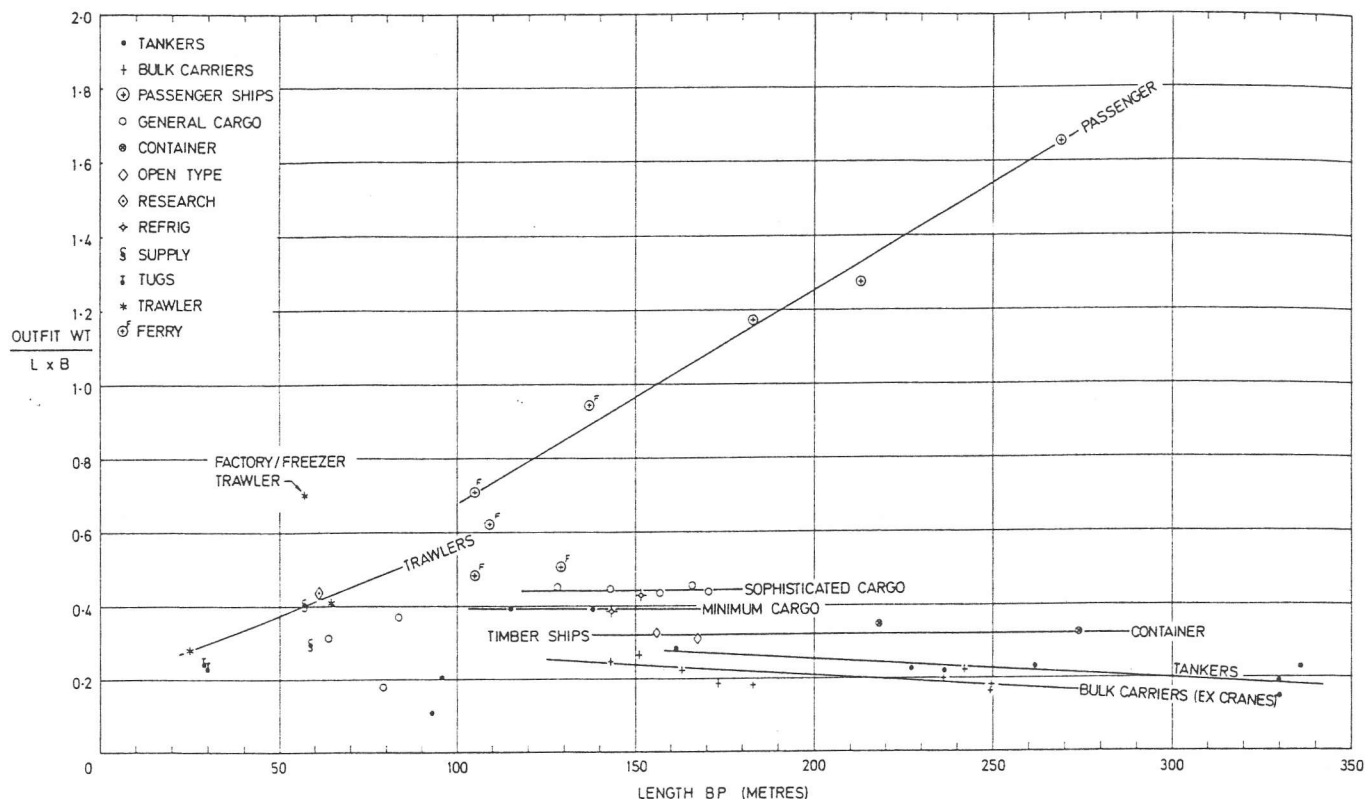


Fig. 9

The choice of method for calculating the outfit weight in the preliminary design stage depends on the information available for the basis ship and the relative importance of the outfit weight in the total weight of the lightship.

In the absence of detailed information the best method remains that of selecting a suitable basis ship and proportioning its outfit weight in relation to the square number ( $L \times B$ ).

In using this method, any known unusual features, such as insulation and refrigerating machinery should be subtracted from the basis ship before proportioning and/or added to the estimate after proportioning.

The ratio between outfit weight and square numbers ( $L \times B$ ) varies with ship type and size (Fig. 9). For volume carriers such as passenger vessels, etc. for which the outfit weight is spread more or less homogeneously throughout the ship and is therefore proportional to volume, the ratio increases linearly with ship's length. For deadweight carriers such as tankers and bulk carriers, for which some items of outfit such as accommodation weight, vary only slightly with ship size, the ratio reduces slowly with increase in ship length. The ratio for general cargo ships is about 0.39 and this corresponds to a value in imperial units of about 0.036 which when compared with the ratio of 0.033 quoted in the 1962 paper, represents an increase of about 10% in outfit weight since that date.

When the outfit weight is a significant proportion of the total weight of the lightship, it is preferable to make a more detailed estimate and this need not be a lengthy process if data are zealously gathered, carefully filed and when appropriate, plotted against suitable parameters.

It is useful to have a standard grouping of outfit weights which can be used for more detailed assessments. One such grouping comprising 32 items is given in Table IV.

#### 5.4 Machinery Weight

An inspection of the formulae and graphs presented in 1962 paper shows how many changes there have been in machinery since that date. The maximum power shown was 15,000 SHP, a figure which at that date had only been exceeded on a

limited number of passenger ships. The line entitled 'high speed diesels' (today's medium speed engines) tailed off at 3,500 SHP.

Turbocharging was so novel that the machinery weights for this type of engine were plotted on a base of the power they would have developed if not turbocharged. Machinery was usually fitted amidships and a reduction in weight was suggested for the cases where the machinery was fitted aft.

Although metrication strictly requires the abandonment of horsepower in favour of kilowatts, we find this rather pedantic and have opted for metric horsepower which is still the usual power figure quoted by engine manufacturers.

The formulae for machinery weights quoted in 1962 for diesels and turbine machinery were both of the  $y = mx + c$  type; whilst this gives the decrease in specific weight per unit of power as the power increases which one would expect, it is a type of formula which necessarily has a limited range and we now prefer a formula of the type  $y = mx^n$ .

The various types of engine which have to be considered include:

- (i) Direct drive slow speeds diesels
- (ii) Geared medium speed diesels
- (iii) Geared steam turbines
- (iv) Diesel electric installations
- (v) Turbo electric installations
- (vi) Geared gas turbines:
  - (a) Aero type
  - (b) Industrial type
- (vii) Gas turbo electric installations
- (viii) Nuclear power.

After the choice of the main engine, the three factors which appear to come next in importance in their effect on machinery weight are:

- (a) The type of ship and cargo carried, which determine to a large extent the auxiliaries fitted; passenger ships and refrigerated cargo ships



generally having additional generating capacity and refrigeration machinery, whilst oil tankers have boilers to provide steam for oil heating and tank cleaning.

- (b) The number of propellers—single or twin screw.
- (c) The position of the engine room in the ship—amid-ship,  $\frac{3}{4}$  aft, or 'all' aft.

As with wood and outfit weights, accurate machinery weights are best obtained by a synthesis from a number of group weights and a suggested system for this is included in Table IV.

A simplified treatment divides the machinery weight into two groups—the main engine which for diesels and gas turbines at all events, can be obtained from a manufacturer's catalogue and a remainder, which can be proportioned on a suitable parameter from the weight of this portion of the machinery weight of a similar installation.

The selection of a suitable base against which to plot main engine weight proved reasonably simple, as we found that the weight was a function of maximum torque rating, represented in this case by MCR/RPM. What was somewhat less expected was the closeness with which most current engine types conformed to the pattern. Only two makes of medium speed engines did not conform closely to the mean line, one being heavier because of maximum use of castings and the other being lighter because of maximum use of welded construction.

Fig. 10 shows a plot of main engine weights against MCR/RPM for a large number of engines in current production. From this the main engine weight can be determined in the first instance. A similar plot of engine weights from 1962 shows that there has been a reduction in weight of approximately 14% for a given power since that date. From Fig. 10 the following equation was derived:

$$\text{Dry weight of main engine (diesel)} = 9.38 \left[ \frac{\text{MCR}}{\text{RPM}} \right]^{0.84} \quad (29)$$

This equation gives a weight which is 5% higher than that represented by the line through the data spots to allow for the fact that the graph really ought to be a stepped line corresponding to cylinder numbers with approximately 10% weight steps for the addition of each cylinder.

We then tried to find an equally satisfactory plot for the remaining component of machinery weight.

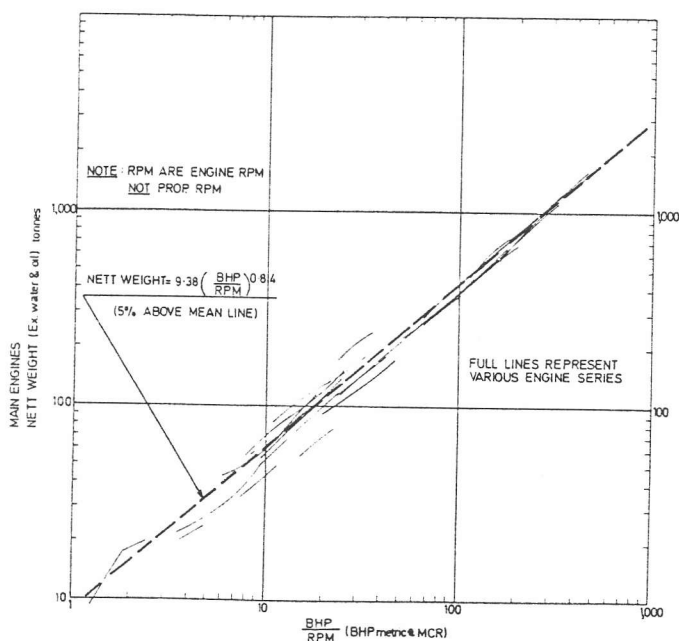


Fig. 10

Two possible abscissae occurred to us:

- the Maximum Continuous Rating of the main engines, and
- once again the quotient, MCR/RPM

The argument for the first of these parameters is that cooling water and lub. oil piping and auxiliaries, exhaust gas boilers, uptakes, shafting and propellers should be related to MCR. The argument for the second parameter is that a medium speed engine will require a much smaller engine room with corresponding reductions in the weight of piping, floorplates, ladders and gratings, and spare gear.

We tried both plots, together with a compromise abscissa of  $\frac{\text{MCR}}{4} \left[ 3 + \frac{120}{\text{RPM}} \right]$  before deciding that the lines which could be drawn on a base of MCR gave the best 'fit' to our data. We found that it was possible to identify separate lines for such different ship types as: Cargo Ships and Bulk Carriers; Tankers; Passenger Ships.

Although Fig. 11 shows a nice series of lines, it must be recorded that there was a considerable scatter in the data points from which these lines were derived. This scatter reflects wide divergence in machinery superintendents' views on desirable installations and on various manufacturers' design and construction techniques.

To make best use of Fig. 11 we recommend the use of a line parallel to those reproduced, through a data spot which the user knows to represent the standard of machinery fit specified. This technique can incidentally be applied with advantage to all of the graphs given in this paper.

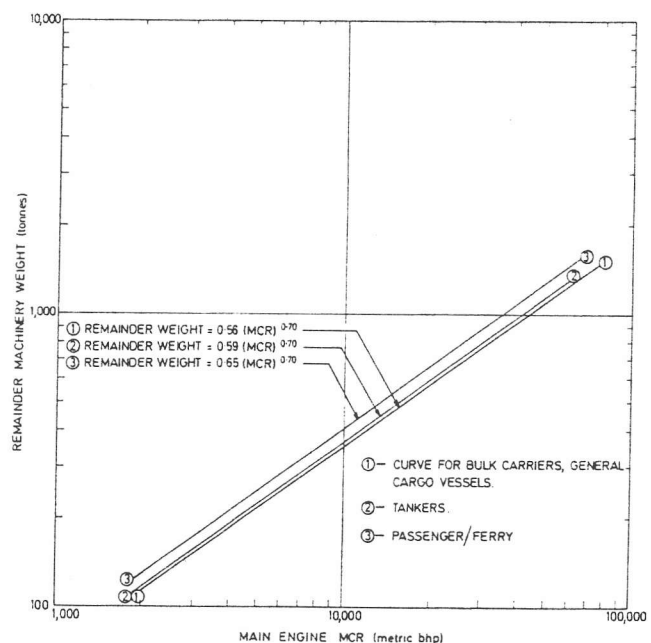


Fig. 11

The simplified two group treatment can also be used for steam turbine ships with the main turbines(s), gearing boiler and condenser weights constituting one group and the remainder a separate group. However, the weights of turbines, gearing, boilers and condensers are much less readily obtainable than are the weights of diesel engines, and the procedure is not, therefore, such a convenient one to use. Accordingly, in Fig. 12 we have plotted the *total* machinery weight against shaft horsepower. We have not attempted in this case to draw a series of curves for the various ship types because the data available to us were not sufficiently extensive to enable us to distinguish between the factors involved.

Although we have dealt only with three out of the eight machinery types listed this probably covers 98% of all machinery installations built. We regret that lack of data has inhibited us from making similar generalisations for the other machinery types, and can only say that if anyone would like to let us have more data we will gladly analyse them!

The methods of estimating machinery weights outlined above are thought to be reasonably accurate for the more usual ship/machinery types. However, care should always be ex-

ercised when estimating the machinery weight and in particular when the estimate is for a more specialised vessel detailed group weights should be established at the earliest possible opportunity.

### 5.5 Margin

The final item required to make up the lightship is a margin. The purpose of a margin is to ensure the attainment of a specified deadweight even if there has been an underestimate of the light weight or an over-estimate of the load displacement. The size of the margin must reflect both the uncertainty in the designer's mind in relation to these and the severity of the penalties which may be exacted for non-compliance. Whether it is a sign of increasing caution or of the greater variety that there is today in ships and their equipment, we are not sure, but we would today recommend a rather larger margin than was suggested in the 1962 paper and our practice is now to allow 2% of the lightweight, if possible.

## 6. STANDARD CALCULATION SHEET

Any design method is most conveniently carried out on a standard sheet, which ensures that all significant items are remembered in the hurry in which designs frequently have to be prepared. Such a standard sheet is presented in Table IV. If the 'three trial ships' method is used, each ship can be designed on a page of this type, or, alternatively, a revised version of the sheet with three or four columns can be used. This table is also used to confirm the weight, displacement and powering aspects of volume dependent designs in conjunction with the standard calculation sheets presented in the next section.

## 7. CAPACITY CARRIERS

In 1962, two types of ships were considered whose dimensions were determined by volume rather than by weight,

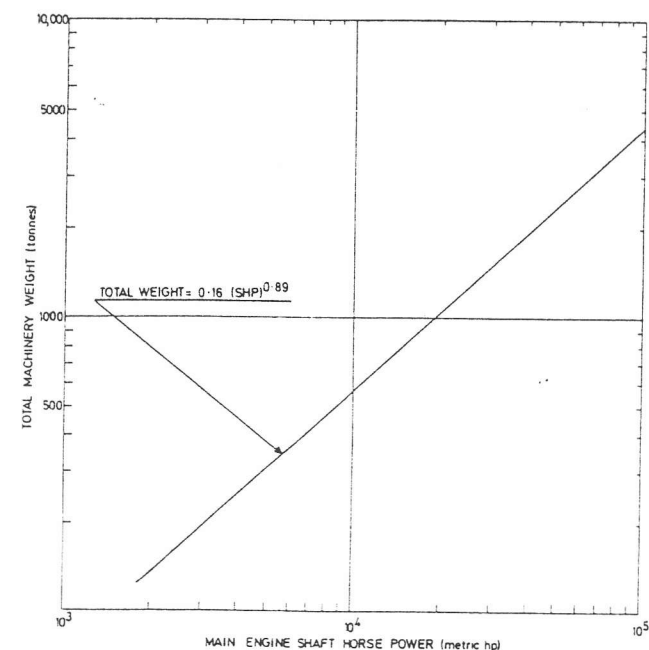


Fig. 12

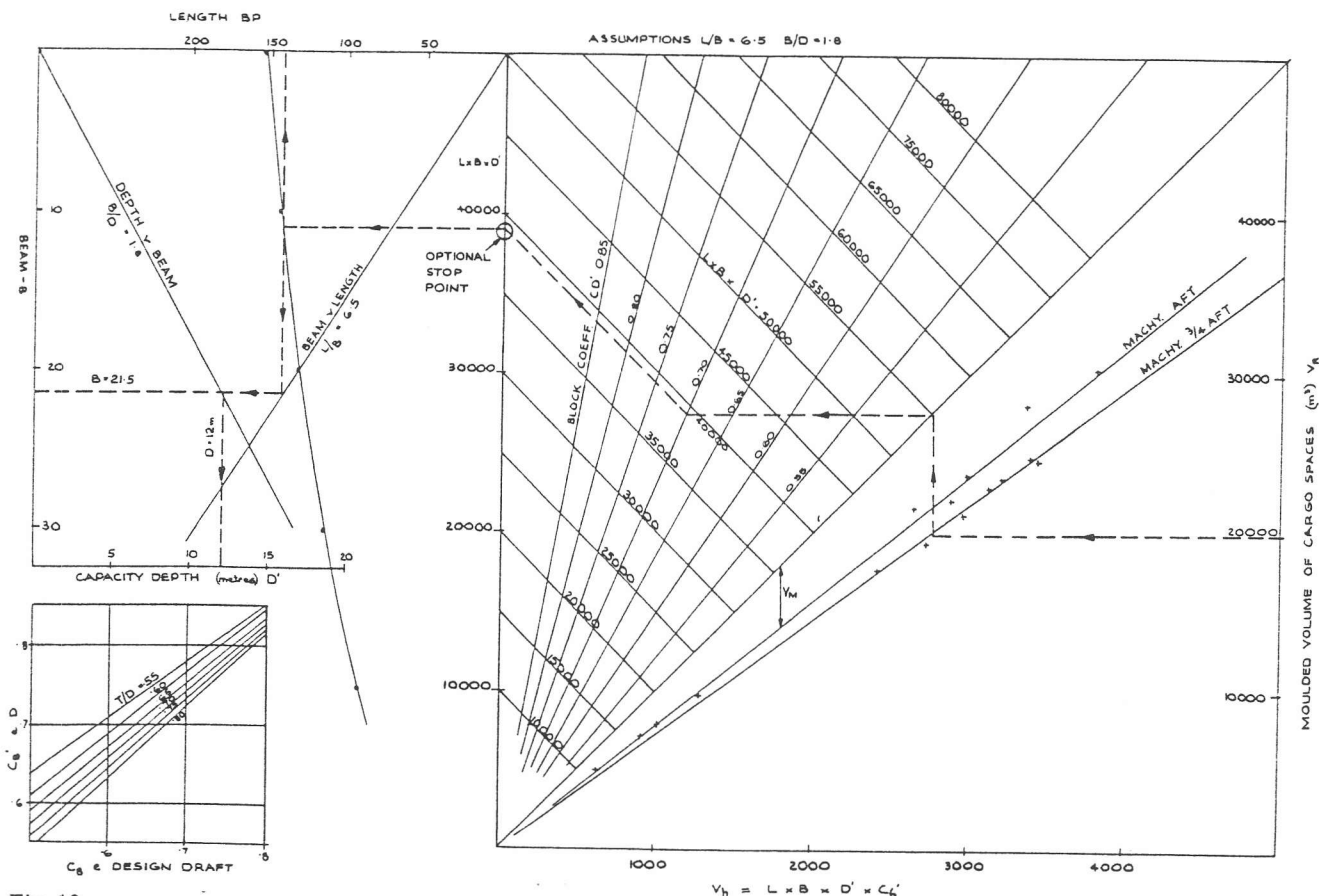


Fig. 13

TABLE IV Preliminary Design Calculation Sheet

REF. SKETCH No.		PAGE No.			
DIMENSIONS metres		STEEL		OUTFIT WEIGHT tonnes	
Length OA		$L(B + T)$		Structural Castings	
L Length BP		$0.85L(D - T)$		Small Castings	
B Beam		$f = 0.85$	$l$ $h$ $f$	Smithwork	
D Depth to		for Super-structures		Sheet Iron	
Depth to				Carpenter	
T Draft (Scantling)				Plumberwork	
Draft (Design)		for Deck		Electrical	
WEIGHTS tonnes		Houses		Paint	
Invoiced Steel		STEEL NUMERAL		Joinerwork	
Scrap ( %)	—	$0.8D - T$		Upholstery	
Net Steel		$(1 - C_B)/3T$		Decorator	
Electrodes	+	$\delta C_B$		Deck Coverings	
Steel for Lightship		$C_B @ T =$		Casing Insulation	
Outfit		$C_B' @ 0.8D$		Sidelights	
Machinery		STEEL WEIGHT tonnes		W.T.F.R. Doors	
Margin		From Graph		Firefighting	
Lightship		$1 + 0.5(C_B' - 0.7)$		Galley Gear	
DEADWEIGHT		Steel at $C_B'$		Refrig. Machinery	
Displacement		Corrections		Cargo/Stores Insul.	
Appendages	—			Ventilation A/C	
Displacement (Mld)				Steering Gear	
DRAFT				Anchors, Cables	
Block Coefficient		TOTAL STEEL WEIGHT		Mooring Machinery	
POWERING		RESISTANCE		Cargo Winches	
Trial Speed		$\odot_{122}$ Basis		Cargo Gear	
Service Speed		$B' : T'$ for $L = 122$		Rigging	
$V/\sqrt{L} : F_n$		$B'/17 : T'/8$		Canvas	
$K = C_B + 0.5V/\sqrt{L}$		Mumford Indices		Hatchcovers	
$\odot$ or $C_t$		$\odot_{122}$ Corrected		L.S.A.	
$\Delta^{2/3}$		$\delta \odot = 4(122 - L)10^{-4}$		Nautical Inst.	
$V^3$		$\odot = \odot_{122} + \delta \odot$		Stores & Sundries	
$P_E$		APPENDAGE RESIST. %		Special Items	
$(1 + a/100)$		Bossing		TOTAL OUTFIT WEIGHT	
$(1 + x)_F/(1 + x)_{ITTC}$		Thruster		DEADWEIGHT tonnes	
$QPC = - (N\sqrt{L})/10^4$		Stabiliser		Oil Fuel	
$\eta_t$		Twin Rudder, etc		Diesel Oil	
$P_s$		TOTAL APP. RESIST. (a)		Fresh Water	
Margin		$A_c = \frac{427.1 \times QPC \times \eta_t}{\odot(1 + x)(1 + a/100)}$		Engineers Tanks	
$P_s$ (Trial)				Stores	
Service Margin		MACHINERY WEIGHT tonnes		Crew & Effects	
$P_s$ (Service)		Main Engine		Passengers	
Derating		Gearing		Swimming Pools	
M.C.R.		Boiler & Condenser		Cargo	
Main Engine		Shafting & Propeller		TOTAL DEADWEIGHT	
N R.P.M.		Generators		S.W. Ballast	
Fuel/Day		Auxiliaries		CAPACITY metres <sup>3</sup>	
Range		Piping, Ladders, Gratings		Gross Volume	
Miles/Day		Funnel Uptakes		Deduction	
Days at Sea		Remainder		Net Volume	
Days in Port		TOTAL MACHY WEIGHT		Cargo Cubic ( )	

TABLE V Calculation Sheet for Design by Volume

REF. SKETCH No.				PAGE No.					
				Number of Units	Unit Area metres <sup>2</sup>	Gross Area metres <sup>2</sup>	Height metres	Volume metres <sup>3</sup>	Berths
1	Passengers'	Type 1							
2	Cabins and Private Toilets	Type 2							
3		Type 3							
4		Type 4							
5	Passages, Foyers, Entrances, Stairs			$\equiv 45\% \Sigma(1 \text{ to } 4)$					
6	Public Lavatories, Pantries, Lockers								
7	Dining Saloons								
8	Lounges, Bars								
9	Shops, Bureaux, Cinema, Gymnasium								
						Total of 1 to 9			
10	Captain's and Officers'	Type 1							
11	Cabins and Private Toilets	Type 2							
12		Type 3							
13	Offices								
14	Passages, Stairs			$\equiv 40\% \Sigma(10 \text{ to } 13)$					
15	Public Lavatories, Change Rooms								
16	Saloon, Lounge								
						Total of 10 to 16			
17	P. Os' and Crew's Cabins	Type 1							
18		Type 2							
19	Passages, Stairs			$\equiv 35\% \Sigma(17 \text{ to } 18)$					
20	Lavatories, Change Rooms								
21	Messes, Recreation Room								
						Total of 17 to 21			
22	Wheelhouse, Chartroom, Radio Room								
23	Hospital								
24	Galley								
25	Laundry								
						Total of 22 to 25			
26	Fan Rooms			$\equiv 2\frac{1}{4}\% \Sigma(1 \text{ to } 25)$					
27	Lining and Flare			$\equiv 3\frac{1}{4}\% \Sigma(1 \text{ to } 26)$					
						Total of 26 to 27			
28	General Cargo (Bale)	metres <sup>3</sup>		(	m <sup>3</sup> ) ÷ 0.88				
29	Refrigerated Cargo	metres <sup>3</sup>		(	m <sup>3</sup> ) ÷ 0.72				
30	Mails, Baggage and Passages								
						Total of 28 to 30			
31	Oil Fuel	tonnes @	SG	(	t ÷ ) ÷ 0.98				
32	Diesel Oil	tonnes @	SG	(	t ÷ ) ÷ 0.98				
33	Fresh and Feed Water	tonnes @ 1.000	SG	(	t ÷ 1.000) ÷ 0.98				
34	Water Ballast	tonnes @ 1.025	SG	(	t ÷ 1.025) ÷ 1.00				
35	Associated Cofferdams, Pipe Tunnels			$\equiv 15\% \Sigma(31 \text{ to } 34)$					
36	Solid Ballast								
						Total of 31 to 36			
37	Refrigerated Stores	metres <sup>3</sup>		(	m <sup>3</sup> ) ÷ 0.68				
38	General Stores and Stores Passages	metres <sup>3</sup>		(	m <sup>3</sup> ) ÷ 0.88				
						Total of 37 to 38			
39	Machinery Space to Crown of Engine Room								
40	Casings								
41	Shaft Tunnels								
						Total of 39 to 41			
42	Sewage Plant, Stabilisers, Thrust Units								
43	Steering Gear, Windlass & Capstan Machinery								
44	Carpenter's Shop, Workshops								
45	Switchboard Rooms, Refrigeration Machinery								
46	CO <sub>2</sub> Room, Sprinkler Plant								
47	Chain Locker								
48	Emergency Generator								
49	Swimming Pool, Trunks, etc								
						Total of 42 to 49			
						TOTAL VOLUME			



TABLE VI Calculation Sheet for Accommodation Areas

REF. SKETCH No.		PAGE No.				
Hull	L(BP) metres	B(Mld) metres	D(Mld) metres	T(Mld) metres		
Machinery	Type	Maker			Power (MCR) h.p.	
		Number of Units	Unit Area metres <sup>2</sup>	Gross Area metres <sup>2</sup>	Area Allocation	Berths
1	Owner's Suite					
2	Senior Officers' Suites					
3	Deck Officers' Cabins with Toilets					
4	Engine Officers' Cabins with Toilets					
5	Chief Steward's Cabin with Toilet					
6	Pilot's Cabin with Toilet					
7	Cadets' Cabin					
8	Passengers' Cabins with Toilets					
9	P.Os' Cabins					
10	Deck Ratings' Cabins					
11	Engine Ratings' Cabins					
12	Catering Staff Cabins					
		Total of 1 to 12				
13	Wheelhouse/Chartroom					
14	Radio Room					
15	Engs' Change Room/Officers' Toilet					
16	Officers' Laundry & Drying Rooms					
17	Offices					
18	Officers' Dining Room & Duty Mess					
19	Officers' Lounge					
20						
		Total of 13 to 20				
21	P.Os' & Crew's Messes					
22	Crew's Recreation Rooms					
23	P.Os' & Crew's Toilets					
24	Crew's Laundry & Drying Rooms					
25						
		Total of 21 to 25				
26	Galley & Pantries					
27	Hospital, Bath & Dispensary					
28	Hobbies Room					
29	Fan Rooms					
30	Emergency Generator/Battery Room					
31	Cold Rooms					
32	Dry Provision Store-room					
33	Bonded & Other Store-rooms					
34	Deck Store-rooms					
35	Lockers					
36	Refrigeration Machinery					
37	Deck Machinery Equipment Spaces					
38	Swimming Pool					
39	Engine Casings					
40						
		Total of 26 to 40				
41	Passages/Stairs	≡	%Σ (1 to 12)			
42	Outside Deck Area	≡	%Σ (1 to 41)			
		TOTAL AREA				

namely, cargo ships carrying light cargoes with a high stowage rate and passenger ships.

For cargo ships, a graph was developed which enabled the dimensions to be obtained from the required capacity and speed. A number of changes have rendered this graph obsolescent; capacities are now quoted in cubic metres, the space required for machinery has been reduced; the use of standard sheer and camber is now rare; and finally, the proportions of the main ship dimensions have changed. A revised graph, and the revised assumptions on which it is based, is given in Fig. 13.

The design method suggested for passenger ships has proved to be an effective way of designing many different ship types, including factory trawlers, research vessels, and small warships such as frigates and corvettes. An adaptation of the method has also proved useful in the design of accommodation blocks on cargo ships, bulk carriers and tankers, enabling the designer to plan the deck on which he prefers to locate each room before starting drawing. Desirable for all forms of arrangement, this becomes almost essential when the aim is a 'block of flats' type of accommodation.

At the end of the section of the 1962 paper which dealt with the areas required for each type of room, the author 'felt bound to apologise for giving a succession of figures, many of them common knowledge and all of them easily obtained from a study of ship plans', and explained that 'he was faced with alternatives of either presenting the bare idea of a volume calculation (which might well have been dismissed as impracticable) or of supporting this thesis with suitable data to prove its practicability'.

The authors feel that they do not, this time, have to prove the practicability or the value of the volume method, and it seems unnecessary to re-write several pages of text purely to metricate it and incorporate relatively minor revisions. A summary of the revised and metricated data is given in the Appendix.

As mentioned briefly in the previous section, the calculation of the required volume or area for a design is best done using standard calculation sheets. Table V is intended for a calculation of the total volume of a passenger ship, whilst Table VI is intended to help in the design of a cargo ship or tanker and deals with the allocation of accommodation between the various decks of the ship in order to produce a neat and convenient arrangement.

### 7.1 Crew Numbers

A part of the volume design section of the 1962 paper dealt with crew numbers for passenger ships, but we would now like to deal rather more generally with manning requirements.

For passenger ships, the addition of data for recent ships to those presented in 1962 shows that the passenger/crew ratio has changed little. We found this somewhat surprising when considered against the significant reduction in the crews of cargo vessels which has occurred in this period. The explanation may lie in two factors:

- (i) The higher standard of hotel services now being provided.
- and (ii) The fact that passenger shipping, unlike cargo shipping, has been declining, so that the reduction in manning which has proven politically and socially acceptable in cargo shipping has met with resistance in passenger shipping.

Modern ships appear to group into passenger/crew ratios of about 1.7 to 2.2 for ships aiming for the upper end of the cruise trade with ratios of 2.5 to 3.0 applying to ships catering for the more popular section of the trade. In both cases the lower figures apply to the smaller ships, and higher ones to the larger ships.

Although total crew numbers may not have changed the distribution by departments has altered with reduction in deck and engine departments corresponding generally to those made in cargo ships being offset by increases in the hotel service department.

The change in the manning of cargo ships since 1962 appears to have come about as a result of a felicitous conjunction of motive and means—the growing pressure for cost reduction and a mass of new technology, respectively.

Tables VII and VIII list some of these factors.

TABLE VII

#### Cost reduction motives

Competition from aeroplanes to passenger ships

Competition from land routes to container ships

Competition between shipping companies as many new nations enter the field

All of these leading to relatively if not, actually, lower freight rates

Better job opportunities ashore leading to the necessity of paying higher wages and providing better conditions for seagoing personnel

The enormous growth in shipping making the acceptance of reduced manning politically acceptable

TABLE VIII

#### Cost reduction means

Improved machinery, requiring less attention, less maintenance

Automation of machinery

Use of self-lubricating fittings

Cargo gear requiring less attention

Patent hatch covers with push-button operation

Self-tensioning winches, universal fairleads, thrust units

Modern paint systems, modern plastic accommodation linings

Electric galley gear

The use of work study

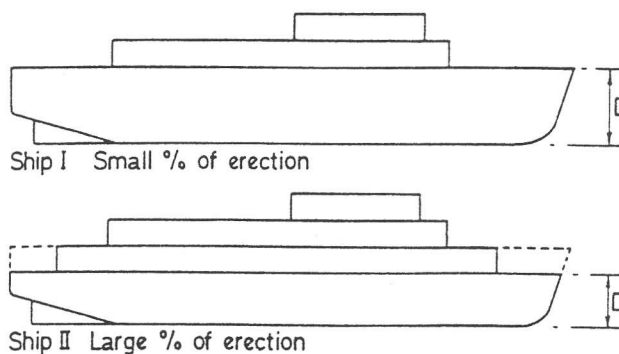
The use of general purpose crews

The effect of these changes on the manning of some typical ships is shown in Table IX.

TABLE IX

Ship Type	1962 Typical	1976 Typical	Automated	Future* Automated
General Cargo or Bulk Carrier	36	30	26	11
Sophisticated Cargo Liner or Container Ship	50	36	28	
Tanker	45	36	26	9

\* Figures for possible future crew numbers are taken from Ref. 14.



With the exception of the dotted part, ships I and II are identical but  $D$  and the proportion of erection volume to total volume are obviously widely different.

Fig. 14

Increasingly national and international agreements are fixing officer and crew numbers. Table X has been constructed from a study of Ref. 15, and represents a simplification of much more complex regulations, but its use will in most cases enable a synthesis of required crew numbers to be made to a reasonable degree of accuracy. If the design and specification of equipment and systems is directed towards labour saving, a reduction in crew numbers can generally be negotiated.

TABLE X

Deck Officers including Master		Engineering Officers including Electrical	
<u>Coasters</u>		<u>Coasters</u>	
up to 200 tons gross	1	up to 500 BHP	1
200-700 tons	2	over 500 BHP	2
700-1600 tons	3	Foreign Trade	
over 1600 tons	4	up to 5000 BHP	3
Additional 1 or 2 cadets carried in larger vessels.		over 5000 BHP	4
		Additionally 1 or 2 junior engineers carried in higher powered vessels.	
<u>Radio Officers</u>		<u>Refrig. Engineers</u>	
Up to 500 tons	0	A specialist refrig. engineer	
Over 500 tons	1	is usually carried on ships with a large refrig. capacity.	
<u>Deck Ratings including P.O.'s</u>		<u>Engine Ratings including P.O.'s</u>	
700-2500 tons gross	6	Coasters	2
2500-5500 tons gross	7	Foreign Going	4-5
5500-15,000 tons gross	8	(automated)	3-4
over 15,000 tons gross	10		
<u>Catering</u>		<u>Stewards</u>	
For total crew up to 45	2	For 6 officers	2
For total crew up to 60	3	For 7-9 officers	3
		For 10-12 officers	4

Numbers of crews recruited from some Asian and African countries may require to be 20% to 50% higher but the area required for accommodation will remain substantially the same.

## 7.2 Dimensions of Volume Carriers

To arrive at the main dimensions, it is necessary to divide the total volume which has been calculated into main hull and a superstructure volume. This seems best done by assuming that the superstructure volume is a certain percentage of the total. That this percentage can vary considerably is illustrated by the diagram in Fig. 14. The use of a percentage derived from a suitable basis ship seems the best procedure but if these data are not available it may be reasonable to assume that 25% of the volume will be provided by the erections. This will give a ship with a relatively high uppermost continuous deck and the minimum amount of erections.

With the volume of the main hull known, the dimensions can be obtained by calculating the volume for three trial ships in a similar manner to that described for the deadweight calculations. For this calculation a beam/depth ratio of 1.55 can be used.

With the main dimensions and the volume of erections known, a preliminary profile can be drawn, but before this is done it will usually be desirable to modify the depth  $D$  to provide double bottom, holds and tween decks of suitable height. This modification should take the form of reducing the depth and adding the volume subtracted in this way from the main hull to the volume of the erections so that the ship changes from Type I towards Type II.

The weight, displacement and powering for the design can then be checked using the standard calculation sheet given in Table IV.

## 8. THE 'LINEAR DIMENSION' SHIPS

We mentioned earlier that there are now a number of ship types in which the design process proceeds directly from the linear dimensions of the cargo, an item or items of equipment, or from constrictions set by canals, ports, etc. and for which the deadweight, volume and sometimes the speed are determined by the design instead of being the main factors which determine it.

The design processes for these ships are essentially non-standard and give the naval architect a chance to exercise his ingenuity.

### 8.1 Container Ships

As the design deadweight of most container ships can be obtained at a draught less than that obtainable with a Type B freeboard, deadweight cannot be used directly to determine the main dimensions.

As container ships usually carry a substantial percentage of their cargo on deck, it is not possible to base the design on the required cargo volume as this is indeterminate. In these circumstances, stability considerations take over the primary role in the determination of the main dimensions.

For maximum economy in the design of any container ship, containers will be stacked up in tiers to the limit permitted by stability. To maximise the numbers, the upper tiers are reserved for relatively lightly loaded (or even empty) containers, whilst heavier containers are directed to the lowest levels, and ballast, either water or permanent or both, may be carried even in the load departure condition.

For each number of tiers of containers carried there is an associated breadth of ship which will provide the KM necessary to ensure adequate stability. Whether the tiers are enclosed below deck or carried on deck is a second order effect.

Longitudinal and torsional strength considerations then require a proportion of the breadth of the ship thus determined to be devoted to structural decks, the balance of the 'open' ship providing space for a number of container cells with their guides. Thus the number of container tiers determines the number of container rows in the breadth.

The length of ship, and very largely the number of container rows in the length, is then determined by the economically and technically desirable length/beam ratio.

Of course, speed affects these numbers, both because of its influence on the block coefficient and its influence on machinery power and thus on the engine room dimensions, but these may also be regarded as second order effects. Fig. 15 shows container numbers which give economic container ships for various speeds. It also shows the tier  $\times$  row numbers for which the midship section should be arranged.

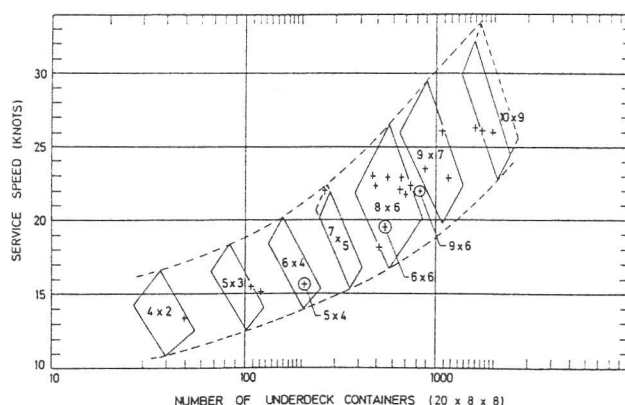


Fig. 15

## 8.2 Multi Purpose Ships

The design process which led to the development of the CLYDE class, by the ill-fated Upper Clyde Shipbuilders, is an interesting variation on this theme. It was decided to design a ship which would be capable of carrying a wide range of cargoes—general break bulk cargoes, bulk cargoes such as grain or ore, timber and containers.

It was realised that, although general cargo and the smaller bulk cargoes would be those most frequently carried, the cargo which should be given most consideration in the determination of the design should be the only one which always came in the same large unit size, namely, containers. To maximise the container numbers, it was decided to fit twin hatches and make these capable of taking three containers in the width. This necessitated a breadth of about 22 m and it was decided to increase this to 22.86 m, the St. Lawrence Seaway limit. With this beam it was possible to provide a depth of 13.72 m enabling 5 tiers of containers to be accommodated below deck, and two tiers of containers to be carried on deck. The length of each main hatchway was then arranged to accommodate three rows of containers in the length.

It was decided to have three main holds and a short No. 1 hold. The stowage space required for hatch covers and the space required for the cargo handling gear then determined the length of the cargo spaces.

Up to this stage in the design, no decision had been taken on the deadweight or the speed of the ship.

To complete the dimensions, a fore peak, engine room and aft peak of approximately correct lengths were added. Various alternative speeds were then considered, leading to a variety of block coefficients, deadweights, cargo capacities, powers and machinery fits—all around the same basic cargo arrangement. These possibilities were then examined against market research indications before fixing the final specification. Looking back now, the design can be faulted (advanced though it seemed at the time) for not being bold enough. The concept even then probably merited development to a bigger and faster ship, such as the type now being built by Govan Shipbuilders for Kuwait Shipping. Fig. 16 shows the various cargo alternatives for which the CLYDE class was designed.

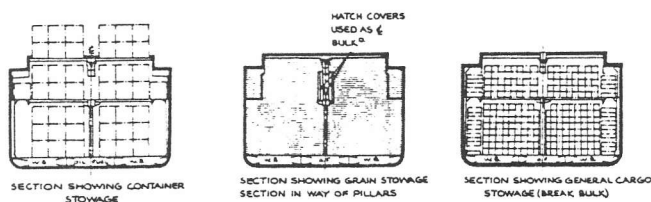


Fig. 16

## 9. STABILITY AND TRIM

Having made a first estimate of the dimensions, weights and power requirements for the design it is then necessary to make a preliminary assessment of the trim and stability before proceeding with any further development. Generally at this stage the stability check can be limited to the calculation of the metacentric height GM, which requires a knowledge of the heights above base of the metacentre and centre of gravity. Eventually the former will be obtained from hydrostatic calculations and the latter from detailed analysis of weights and centres using the detailed plans of the vessel, neither of which are available in the preliminary design stage. It is therefore necessary to develop procedures which can be easily applied, but which also reflect the wider range of designs which have to be considered.

In the design office, preliminary hydrostatics can be obtained very quickly from forms generated by computer programs which require only the main dimensions, block coefficient and

LCB as input. Alternatively, approximate values of KM can be obtained from the diagrams given in Ref. 16.

Again the capacity of holds and tanks can be obtained very easily from bonjean data produced by an extension of the computer program systems mentioned above. Alternatively an estimate of capacity can be made from Fig. 13 or from data for particular ship types, such as those given for bulk carriers in a paper by Gilfillan<sup>(17)</sup>. At this stage it is assumed that the designer has made a rough sketch of his design from which the centres of the major items of deadweight can be estimated.

The only item remaining for which the centre of gravity is to be estimated is the lightship, and unfortunately this is the area of greatest uncertainty.

### 9.1 Volume Density Method for Lightship KG

In 1962 a method of analysing and calculating the lightship KG—the Profile Stability Method—was described, and this has proven to be satisfactory in practice provided that a generally similar design to that under consideration is used as a basis. A logical development of this method, and one which would be less sensitive to choice of basis ship, would be to incorporate the relative densities and difference between centres of volume and gravity of each component of the volume calculation.

If the volume of each individual space is multiplied by the appropriate density factor and the height of its centre of volume is corrected by an appropriate factor which relates the centroid to its VCG, the calculations would, in fact, become a weight calculation, which would be completely accurate—if the factors used were accurate. So far as the accuracy of the final centre of gravity is concerned, it is the accuracy of the relative values of the density factors which matters, not their absolute values.

Notes on this type of calculation—the Volume Density method—and a standard sheet for it are presented to Table XI. The accuracy of the calculation depends on the correctness of the value of  $\rho_S$ ,  $K_S$ ,  $\rho_H$ ,  $K_H$  used.

$\rho_S$  for accommodation constructed in steel and fitted out to normal cargo ship standards appears to have a fairly consistent value of about 0.13 tonnes/m<sup>3</sup>.

$K_S$  for accommodation of the same type generally has a value of 0.6.

$\rho_H$  varies not only with ship type, but also with ship size and should be assessed with care.

Where the hull below No. 1 deck and the superstructure both contain accommodation as on a passenger ship, it may be reasonable to make  $\rho_S = \rho_H = 1.00$  at lines 1 and 8 and apply a correction factor to the weight obtained at line 10 to give the 'corrected hull weight' at line 12.

With  $\rho_S$ ,  $K_S$  and  $\rho_H$  known the value of  $K_H$  for a suitable basis ship can be determined by analysis starting at both top and bottom of the table.

Using a suitable value of  $K_H$  from a good basis ship the light ship VCG of a new design can be calculated.

The method can also be used for the calculation of the LCG, with the LCG of the hull to No. 1 deck being established through the relationship which it bears to the LCB at, of course, the moulded depth D, (or a fixed proportion of this dimension).

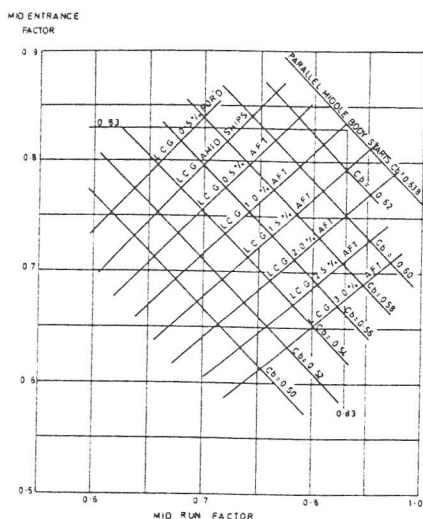
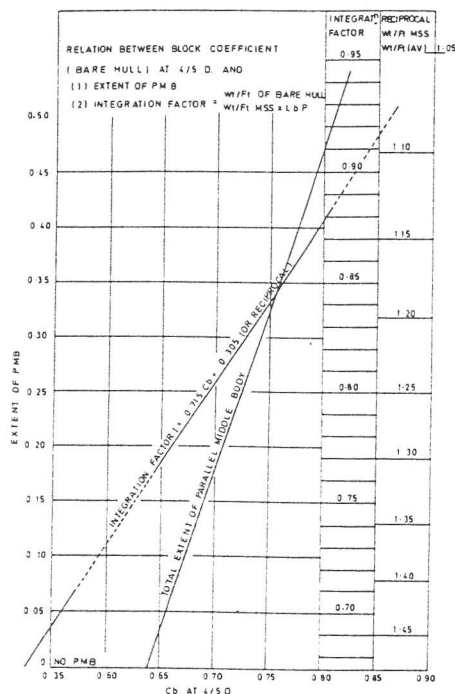
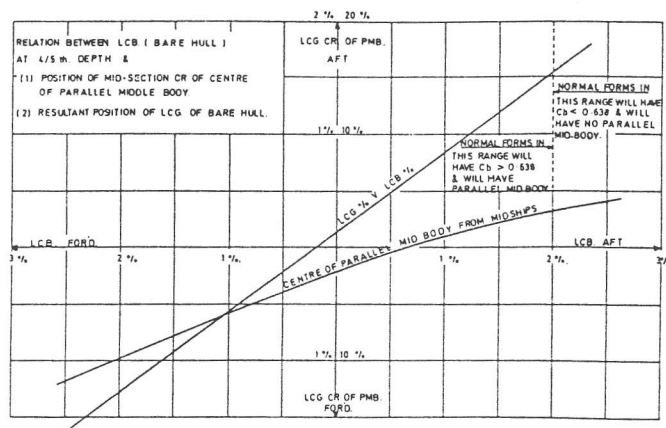
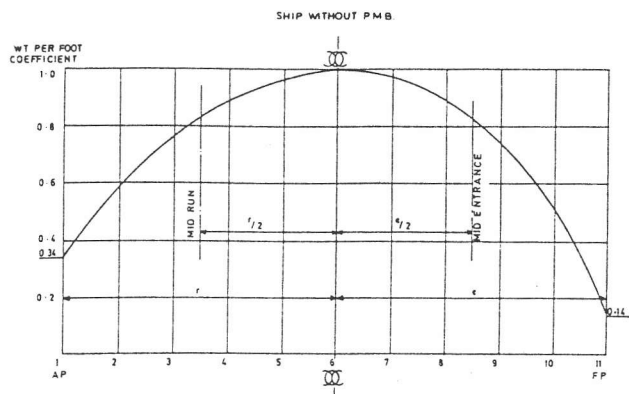
This relationship is discussed in the next section and is shown in Fig. 22 for the bare steel hull. The hull to No. 1 deck used in the volume/density method also includes outfit within the hull and deck machinery and gear mounted on No. 1 deck. This may modify the relationship LCG to LCB somewhat from Fig. 22 but the trend that the LCG will follow the LCB, but stay somewhat nearer amidship as indicated by Fig. 20 seems highly probable.

### 9.2 Weight Distribution and Centre of Gravity

In an earlier section we digressed to discuss ships' lines and the position of the longitudinal centre of buoyancy.







Weight distribution diagrams derived by these methods give very close approximations for a wide range of ship type, fullness and form and are markedly better than any of the 'coffin' diagrams generally used for strength calculations.

## 10. APPLICATION OF COMPUTERS IN PRELIMINARY DESIGN

We mentioned earlier in the paper the contribution which the computer can make at the preliminary design stage and it is now perhaps useful to consider the reasons why these programs are not more widely used in spite of the numerous papers on the subject. We also wish to present our own ideas on the type of programs which we consider can and should be developed.

The early development of computer design programs has been described in papers by Murphy, Sabat and Taylor<sup>(19)</sup>, Mandel and Leopold<sup>(20)</sup>, and by one of authors<sup>(21)</sup>, and more recently by Fisher<sup>(22)</sup>, and Eames and Drummond<sup>(12)</sup>. Most of these programs were written for batch computers in which the data were input on cards or on paper tape, the calculations made using fixed formulae and a mountain of answers printed out at high speed on a line printer. The main effort was in the gathering and preparation of data or in the analysis and confirmation of the sense of the results produced. In hindsight we would now suggest that this type of study was not particularly suitable for the batch machine in which the whole process is highly impersonal.

Preliminary design is not like that at all; it is a process in which the designer, through his experience, is personally involved in the development of the design, in terms of concept and methods used and also in the reasonableness of answers obtained.

One reason, therefore, why these programs have not been widely adopted, results from the impersonal nature of the machines for which they are written. The programs offered by BSRA on their remote access systems, go some way but not far enough to remove this objection.

Whatever means of communication are devised to overcome this problem, the influences which dimensions, block coefficient etc. have on displacement, powering, the weight and cost of steel, outfit and machinery, fuel consumption etc., must be correctly expressed if an optimum design in which the designer can have confidence, is to be obtained. If, however, the influences are not correctly expressed and there is reason to be sceptical about some of the approximate formulae which are used in some computer design programs—the apparent optimum may not in fact be a true optimum. In most ship designs however the curve expressing value, whether as minimum required freight rate or maximum present value tends to have a fairly flat optimum so the lack of sophistication which exists in some of these relationships has only a small effect on the final answer.

Of the large number of alternative dimensions and coefficients which would be considered when using a computer, many could, and in fact, often are eliminated as impracticable or obviously uneconomic on the basis of relationships established by previous design.

If this process is carried further so that each of the basic relationships used is an optimum one, it becomes possible to produce a design using a slide rule or calculator as quickly as by computer. If the facility for exercising judgement implicit in slide rule technology is exercised with skill it can be argued that the slide rule design may be closer to the true optimum than that produced by the computer.

It is however also fair to say that the number of designs that can be produced and fully tested by slide rule techniques is extremely limited, and there is always the danger that the designer is not working in the region of the optimum combination of dimensions.

What is needed therefore is a comparatively simple framework of generalised interactive programs, which make all the necessary calculations very rapidly, but also allows the designer to choose the equations to be used, the level of detail and data required and to generally guide the development of his design towards the optimum.

Instead of writing specialised preliminary design programs for specific ship types and or with particular operations in mind, it is suggested that such a program should concentrate on broad methods leaving the designer to make his run specific by supplying the appropriate specialised data. It should be possible using the sophisticated features of modern computer programming languages, to prepare a conversational type program which permits an almost infinite permutation of specialised blocks to be called as required to meet each specified case.

The procedures for calculating dimensions, checking displacement and stability and trim, provide a logical basis for computer programs for the design of both deadweight and capacity carriers. The logical next step in this process is the preparation of the outline general arrangement plan by an interactive method and a flow diagram for this has been developed and is shown in Fig. 23.

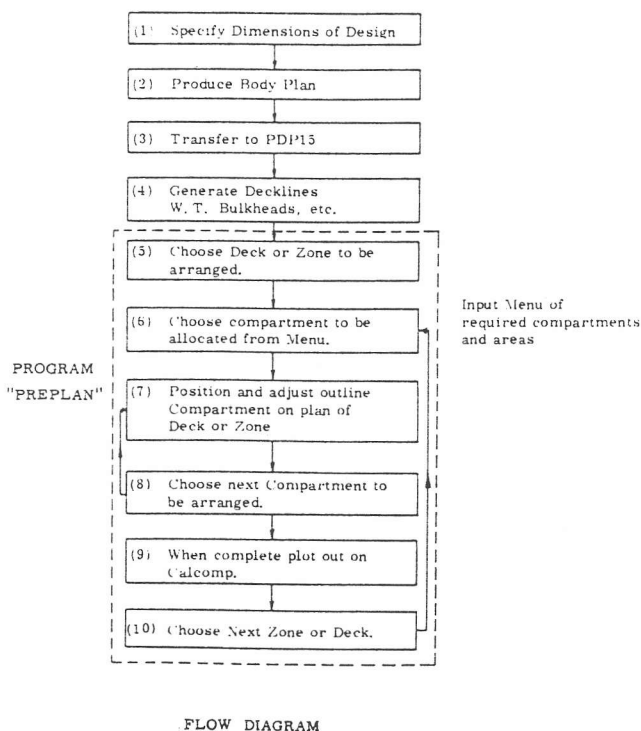


Fig. 23

## 11. FINAL REMARKS

This paper concerns itself with the naval architect's basic stock-in-trade. Much of it will therefore be familiar to experienced practitioners, and we apologise to them for some of the statements of the obvious which we have included for the sake of completing our arguments, and would explain that we have been encouraged to do this by the extent to which the 1962 paper has been adopted as a text book for students.

We would like to repeat the final word of warning given in the 1962 paper—before any of the data or approximate formulae quoted in the paper are used, they should be checked against the user's own data.

## ACKNOWLEDGEMENT

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## APPENDIX

## Areas/Volumes of Spaces

1-4. Passenger Cabins (Excluding bath or toilet) CruiseLiners

De Luxe Suites for two persons	16 m <sup>2</sup>	
1st Class Single 9m <sup>2</sup> twin	13 m <sup>2</sup>	
Tourist twin	6 m <sup>2</sup>	Three 9 m <sup>2</sup>
		Four 12 m <sup>2</sup>

Overnight Accommodation

1st Class Single 3.6 m <sup>2</sup> twin	5 m <sup>2</sup>	
Tourist twin	4 m <sup>2</sup>	Three 6.0 m <sup>2</sup>
		Four 6.6 m <sup>2</sup>

<u>Private Bathrooms &amp; Toilets.</u>	Bathroom 3.8 m <sup>2</sup>
	Toilet 2.8 m <sup>2</sup>

5. Passages, Foyers, Entrances, Stairs. About 45% of sum of 1-4.
6. Public Lavatories. To serve public rooms and passenger sections without private facilities. Space based on facilities provided. Following rates allow also for necessary access space:  
Bath 3.3 m<sup>2</sup>. Shower 1.7 m<sup>2</sup>. WCs 1.9 m<sup>2</sup>. Washbasin 1.4 m<sup>2</sup>. Urinal 1 m<sup>2</sup>. Ironing Board 1 m<sup>2</sup>. Slop Locker 1.5 m<sup>2</sup>. Deck Pantry 4.5 m<sup>2</sup>.
7. Dining Saloon. Base on number eating at one sitting.  
1st Class from 1.5 m<sup>2</sup> Large Numbers to 2.3 m<sup>2</sup> Small Numbers.  
Tourist from 1.3 m<sup>2</sup> Large Numbers to 1.6 m<sup>2</sup> Small Numbers.
8. Lounges, Bars. Base on aggregate seating required. Usually 100% in Tourist and in excess of 100% in 1st Class. Area per seat. Lounges 2 m<sup>2</sup>. Libraries 3 m<sup>2</sup>.
9. Shops, Bureau, Cinema, Gymnasium.  
Shops, Bureau 15 m<sup>2</sup>-20 m<sup>2</sup>, Cinema 20 m<sup>2</sup> Stage + 0.8 m<sup>2</sup> per seat
- 10-12. Captain and Officers Cabins (Excluding bath or toilet).  
Captain and Chief Engineer 30 m<sup>2</sup> + Bath 4 m<sup>2</sup> or Toilet 3 m<sup>2</sup>  
Chief Officer, 2nd Engineer, Chief Purser 14 m<sup>2</sup> + Toilet 3 m<sup>2</sup>  
Other Officers 8.5 m<sup>2</sup> (Sometimes + Toilet)
13. Offices. Captain, Ship, Engineers, Chief Steward each about 7.5 m<sup>2</sup>. Large ships add Chef, Provision Master, Laundryman.
14. Passages, Stairs. 40% of sum of 10-13.
15. Officers Lavatories. Number of fittings usually in excess of DoT rules. Area per fitting as in 6.
16. Dining Saloon, Lounge.  
Dining Saloon about 1.3 m<sup>2</sup> per seat  
Lounge about 1.7 m<sup>2</sup> per seat
- Dining Saloon usually seats 100% Officers although some may dine with passengers. Lounge usually seats about 60% Officers.
- 17-18. P.O.'s and Crew Cabins. Single berth cabins (usually Senior POs) 7 m<sup>2</sup>  
Two berth cabins (Junior POs, Deck and Engine Ratings) 6.5 m<sup>2</sup>  
Four berth cabins Stewards 10.5 m<sup>2</sup>
19. Passages, Stairs. 35% of 17.
20. Crew Lavatories, Change Rooms. Sanitary fittings to DoT rules. WCs 1 per 8; shower 1 per 8; washbasins 1 per 6 (if not in cabins) area per fitting as in 6.
21. Messes and Recreation Room.  
Messes for POs,  
Deck and Engine Ratings seating for 100% } 1.1 m<sup>2</sup>/seat  
Messes for Stewards seating for 40% }  
(Other Stewards eat in Saloon after passengers)  
Recreation Room for Deck and Engine Ratings—seating for 50% 1.2 m<sup>2</sup> per seat
22. Wheelhouse, Chartroom, Radio Room.  
Wheelhouse 30 m<sup>2</sup> Chartroom 15 m<sup>2</sup>  
Radio Room 8 + 2.5 m<sup>2</sup> per Radio Officer
23. Hospital.  
Number of berths all hospitals = 2 + 1 per 100 of total complement. 35% of these may be upper berths.  
Area per berth one or two tier = 6 m<sup>2</sup>
24. Galley.  
Area per person served = 0.65 m<sup>2</sup> for small numbers  
Reducing to about 0.55 m<sup>2</sup> for 1000 or more total complement.
25. Laundry including Ironing Room etc.  
(50 + 0.07 complement) m<sup>2</sup>
26. Fan Rooms. 2.5% of total ventilated volume 1-25.
27. Lining and Flare. 3.4% of total ventilated volume 1-25
- 28-30. Cargo Spaces. As specified. Convert to moulded volumes as follows; Bale ÷ 0.88; Refrig. ÷ 0.72
- 31-32. Oil Fuel, Diesel Oil. Calculated for the required endurance at specific consumption rates corresponding to engines selected. Allow for port consumption and for margin remaining on arrival at bunkering port. Allow for fuel used for heating, distillation and hotel service purposes.
33. Fresh/Feed Water. With a distillation plant generally fitted fresh and feed water storage capacity is arranged to provide for emergency resulting from break down of distillation plant—and depends on voyage route.
34. Water Ballast. Refers to tanks available only for water ballast. Consists of tanks required to maintain stability in burnt-out arrival condition plus tanks required to provide flexibility of trim to cope with all required loading conditions.  
Generally water ballast capacity will require to be between two thirds and three quarters of oil fuel plus fresh water consumption from tanks.
35. Cofferdams, Pipe Tunnels. 15% of volume of 31-34.
36. Solid Ballast. If required allow necessary stowage space.
37. Refrig. Stores. Allow 0.04m<sup>3</sup> per person per day of voyage.
38. General Stores. Allow 140m<sup>3</sup> + 0.1m<sup>3</sup> per person per day.
- 39-41. Machinery Space Volume, Casings, Shaft Tunnel.  
Having arrived at reasonably satisfactory methods for determining machinery weights in an earlier section of the paper it is suggested that these be used to determine the required volume by dividing this weight by the appropriate density. Density values for machinery spaces appear to be of the following order: