

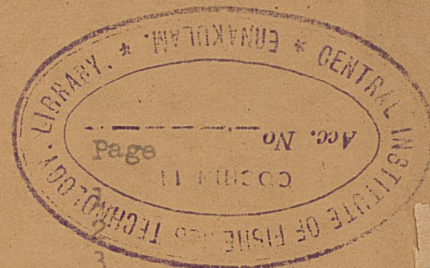
**TRAINING COURSE IN FISHING BOAT DESIGN & CONSTRUCTION**

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# C O N T E N T S



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Ship Geometry and Hydrostatics

Basic Drawings: Lines plan

Required information for all boats:

$\nabla, \Delta$ , Displacement to a number of waterlines (in volume and weight)  
LCB Movement of longitudinal centre of buoyancy

Desirable information (always required for big vessels):

LCP Longitudinal centre of flotation for each waterline  
KB Height of centre of buoyancy from fixed reference line (usually rabbet line at midship section)  
BM Distance between centre of buoyancy and transverse metacentre  
KG Height of ship's centre of gravity at various loading conditions  
GM Transverse metacentric radius for same loading conditions  
MTI Moment to trim one inch (or in metric system: to trim 5 cm.)  
Bonjean Curves Curves of section areas with increasing immersion

✓ Basic law: Archimedes Principle: The weight of a floating body at rest is at all times equal to the weight of the displaced water.

NOTE: A boat will only float at a predetermined waterline if its weight is equal to the weight of the water volume represented by the boat's displacement, or underwater volume of the boat.

✓ Basic condition: The centre of buoyancy (volume centre of displacement, or centre of gravity of displaced water) must lie on a vertical line below or above the ship's centre of gravity for even trim.

The volume up to the design waterline DWL thus defines the weight the ship must have in a certain loading condition to float at that line. This condition is often taken as including all weights on board, but no payload and no ice in the hold.

The LCB at this DWL must lie vertically below the ship's centre of gravity CG in this condition; otherwise the vessel will trim.

NOTE: Excessive displacement at DWL means the ship will float high (with possible dangerous effect on the static stability).  
Deficient displacement at DWL means the ship will float low.  
LCB in front of CG will result in trim by the stern, while  
LCB astern of CG will result in trim by the head.

Convenient tools for the calculation of hydrostatic properties are:

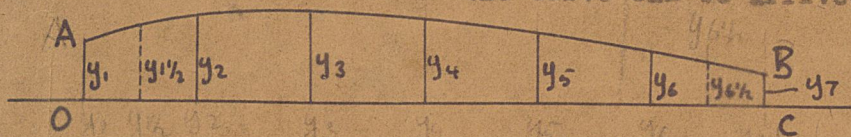
Planimeter, scales, slide rule and Simpson's rules.

For guide to calculations see sheet 2.



Simpson's first rule: For a curve OABC, the area under the curve can be arrived at thus:

2)



$$A = \frac{h}{3} (y_1 + 4y_2 + 2y_3 + 4y_4 + 2y_5 + 4y_6 + y_7) = \frac{h}{3} \times \sum f(o)$$

where  $f(o)$  is the function of ordinates. With half sections  $y_{1/2}, y_{6/2}$ :

$$A = \frac{h}{3} (\frac{1}{2}y_1 + 2y_{1/2} + 1\frac{1}{2}y_2 + 4y_3 + 2y_4 + 4y_5 + 1\frac{1}{2}y_6 + 2y_{6/2} + \frac{1}{2}y_7)$$

Simpson's second rule:

$$A = \frac{3h}{8} (y_1 + 3y_2 + 3y_3 + 2y_4 + 3y_5 + 3y_6 + y_7)$$

Simpson's third rule:

$$A = \frac{h}{12} (5y_1 + 8y_2 - y_3)$$

Rule for six ordinates:

$$A = \frac{5h}{288} (19y_1 + 75y_2 + 50y_3 + 50y_4 + 75y_5 + 19y_6)$$

In all rules: The sum of the multipliers divided by the number of spaces is equal to the division of  $h$ .

Application of the different rules: First rule is valid for 3, 5, 7, 9, 11 etc. ordinates. First and last ordinate always counted. Curve must start and finish with an ordinate.

Second rule: 4, 7, 10, 13, 16 etc. ordinates.

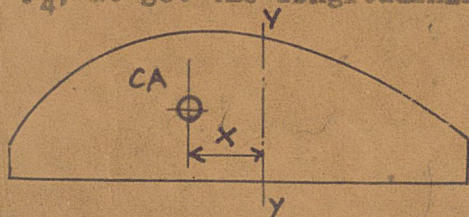
Third rule: 2 ordinates, in conjunction with a third, not bordering the area.

The principle of moments:

The moment of an area about an axis is equal to the area multiplied with the distance of its centre of gravity from the axis.

Conversely the position of the centre of the area can be found in relation to any axis by dividing the moment of the area about this axis by the area.

For the area shown under Simpson's first rule and the reference axis through  $y_4$ , we get the longitudinal position of the centre of the area:



$$x = \frac{\text{Moment of Area}(y-y)}{A}; \quad M(A) = h \times \frac{h}{3} \sum n f(o)$$

where  $n$  is the number of spaces to each ordinate from the reference axis.

Calculation of displacement and LCB:

If the area of each section in the body plan is drawn as an ordinate, a curve is obtained as above. The area under which is equal to the displacement at the waterline investigated. The centre of gravity of the area is the LCB.



The calculation is best done in tabulated form like this:

| Section<br>(ordinate) | ①<br>$\frac{1}{2}$ or full<br>area<br>(ordinate) | ②<br>SM        | ③<br>$f(V)$   | ④<br>$L(u)$ | ⑤<br>$f(M_v)$     |             |
|-----------------------|--------------------------------------------------|----------------|---------------|-------------|-------------------|-------------|
| 0                     |                                                  | $\frac{1}{2}$  |               | -5          |                   | ③ = ① × ②   |
| $\frac{1}{2}$         |                                                  | 2              |               | -4.5        |                   |             |
| 1                     |                                                  | $1\frac{1}{2}$ |               | -4          |                   | ⑤ = ③ × ④   |
| 2                     |                                                  | 4              |               | -3          |                   |             |
| 3                     |                                                  | 2              |               | -2          |                   | = ① × ② × ④ |
| ⋮                     |                                                  | ⋮              |               |             |                   |             |
| 7                     |                                                  | 2              |               | +2          |                   |             |
| 8                     |                                                  | 4              |               | +3          |                   |             |
| 9                     |                                                  | $1\frac{1}{2}$ |               | +4          |                   |             |
| $9\frac{1}{2}$        |                                                  | 2              |               | +4.5        |                   |             |
| 10                    |                                                  | $\frac{1}{2}$  |               | +5          |                   |             |
|                       |                                                  |                | $\Sigma f(V)$ |             | $\Sigma - f(M_v)$ |             |
|                       |                                                  |                |               |             | $\Sigma + f(M_v)$ |             |

$$\nabla = \frac{h}{3} \times \Sigma f(V)$$

(for full areas)

$$\nabla = \frac{2h}{3} \times \Sigma f(V)$$

(for half areas)

$$LCB = \frac{\{ + \Sigma f(M_v) - \Sigma f(M_v) \} \times h}{\Sigma f(V)}$$

This result will be relative to the midship-ordinate, + or -, + indicating forward of midships, - indicating aft.

The table and the coefficients are given here for Simpson's first rule and 11 ordinates (or 10 spaces). For any of the other rules adapt accordingly.

This process will be repeated for a number of waterlines. The section areas required will usually be found with the help of a Planimeter. Make sure to set it correctly for the scale employed or alternatively establish the proper constant before measuring up.

It is not always essential to draw curves of section areas; these will however afford a good check on the fairness of the lines, as they show humps and shoulders if there is any inconsistency in the shape of the hull.

**NOTE:** As waterline below and above DWL are measured up, it will sometimes be necessary to switch from one rule to another. Also any given rule may not necessarily give the displacement for the full length of any given waterline (in case a waterline finishes between sections). In this case the addition to the displacement should be guessed or separately calculated if it is thought that it makes an appreciable difference. Very often it could also be neglected.

#### Calculation of LCF:

LCF is the longitudinal centre of flotation at each waterline. It is simply the centre of the area bordered by the waterline (waterplane) and is found as above by simpsonising half breadth of the waterline:

| St. | $\frac{1}{2}$ breadth | SM | $f(A)$        | $L(u)$ | $f(M_A)$          | net $M(A) = \pm$ difference<br>between fore- and<br>aft moments. |
|-----|-----------------------|----|---------------|--------|-------------------|------------------------------------------------------------------|
| 0   |                       | 1  |               | -5     |                   |                                                                  |
| 1   |                       | 4  |               | -4     |                   |                                                                  |
| 2   |                       | 2  |               | -3     |                   |                                                                  |
| ⋮   |                       | ⋮  |               | ⋮      |                   |                                                                  |
| 10  |                       | 1  |               | +5     |                   |                                                                  |
|     |                       |    | $\Sigma f(A)$ |        | $\Sigma - f(M_A)$ |                                                                  |
|     |                       |    |               |        | $\Sigma + f(M_A)$ |                                                                  |

$$LCF = \frac{\{ + \Sigma f(M_A) - \Sigma f(M_A) \} \times h}{\Sigma f(A)}$$

$$A = \frac{2h}{3} \times \Sigma f(A)$$

( $\frac{2h}{3}$ , because usually  $\frac{1}{2}$  breadth are used)



The area of the waterplanes can also be used to approximate the weight that is necessary to immerse the ship 1 inch more at this particular waterline. The value for this is called: Tons per inch immersion, or TPI:

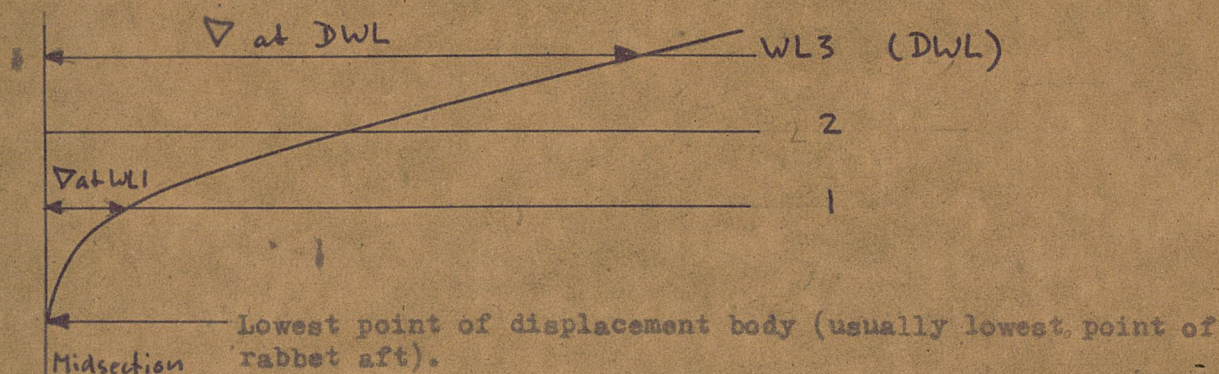
$$TPI = \frac{A_{wl} \text{ ft}^2}{12 \times 35} = \frac{A}{420} \text{ tons}$$

KB - Height of the centre of buoyancy:

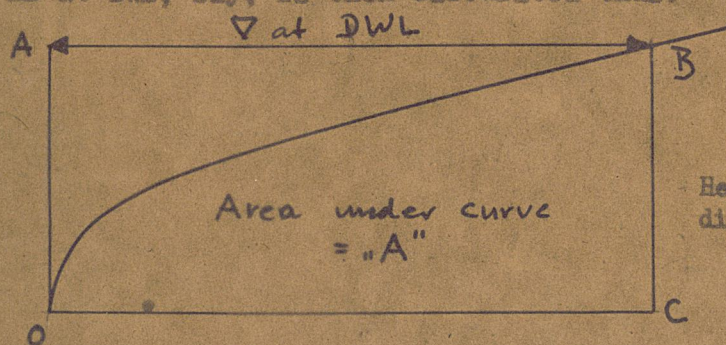
For the subsequent investigation of the statical stability of the vessel, it is necessary to know the vertical position of the centre of buoyancy at a number of displacements or loading conditions.

This is most conveniently calculated on the bases of a complete displacement curve.

The calculated displacement for each waterline is plotted as a linear measurement in convenient scale (e.g. 1" is 100 ft<sup>3</sup> or similar) at the level of the waterlines. This plot is made from a convenient vertical reference line, usually the midship section in the profile plan.



KB at DWL, say, is then calculated thus:



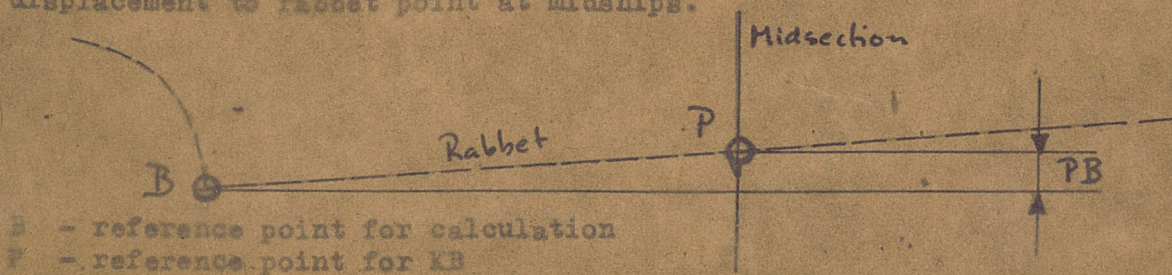
Height of CB from lowest point of displacement body is

$$HCB = \frac{A}{\Delta}$$

If planimeter readings are used only, the height of CB becomes:

$$HCB = \frac{\text{reading OBC}}{\text{reading OABC}} \times \text{distance OA}$$

KB then is: Calculated height of CB - vertical distance from lowest point of displacement to rabbet point at midships.



B - reference point for calculation

P - reference point for KB

$$KB = \frac{A}{\Delta} - PB$$

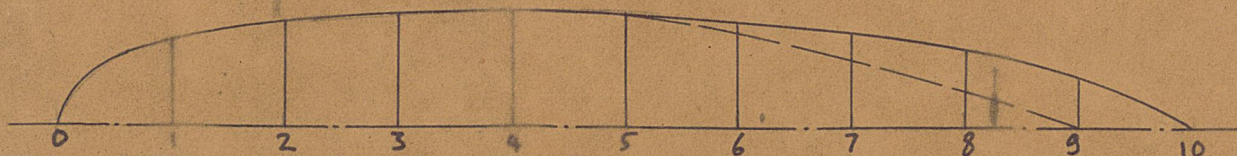


Calculation of BM:

BM is calculated for a few waterlines only normally, one or two below, and one or two above DWL. It is only dependent of the waterline shape and the displacement volume, and thus can be termed a form parameter.

where  $I_T$  is the transverse moment of inertia of the waterplane under consideration.

For a waterplane:



the transverse moment of inertia is calculated like this:

$$I_T = \frac{2h}{3} (y_0^3 + 4y_1^3 + 2y_2^3 + 4y_3^3 + 2y_4^3 + 4y_5^3 + 2y_6^3 + 4y_7^3 + 2y_8^3 + 4y_9^3 + y_{10}^3)$$

for Simpson's first rule and half breadths as ordinates, and

$$I_T = \frac{h}{4} (y_0^3 + 3y_1^3 + 3y_2^3 + 2y_3^3 + 3y_4^3 + 3y_5^3 + 2y_6^3 + 3y_7^3 + 3y_8^3 + y_9^3)$$

for Simpson's second rule. (--- waterline)

In tabulated form:

| St. | $\frac{1}{2}$ breadth ( $\frac{1}{2}B$ ) | $(\frac{1}{2}B)^3$ | SM | $f(I_T)$      | $I_T = \frac{2h}{3} \times \sum f(I_T)$      |
|-----|------------------------------------------|--------------------|----|---------------|----------------------------------------------|
| 0   |                                          |                    | 1  |               | or<br>$I_T = \frac{h}{4} \times \sum f(I_T)$ |
| 1   |                                          |                    | 4  |               |                                              |
| 2   |                                          |                    | 1  |               |                                              |
|     |                                          |                    |    | $\sum f(I_T)$ |                                              |

Position of the centre of gravity CG - longitudinally: LCG  
vertically: KG

In the initial design stage, the correct position of the CG is impossible to assess. It must however be approximated as good as possible both longitudinally (to get a check on the possible trim of the vessel), and vertically, to assess the statical stability with some degree of accuracy.

If any boats similar to the one being designed are well known, the CG for the new boat can be approximated from examining the known, built boats. Only very similar boats may be thus treated, and allowance must be made for all differences, such as different position of engine, different make and weight of engine, difference in major equipment items etc.

A check on the first CG approximation will be afforded by a careful weight calculation later on.

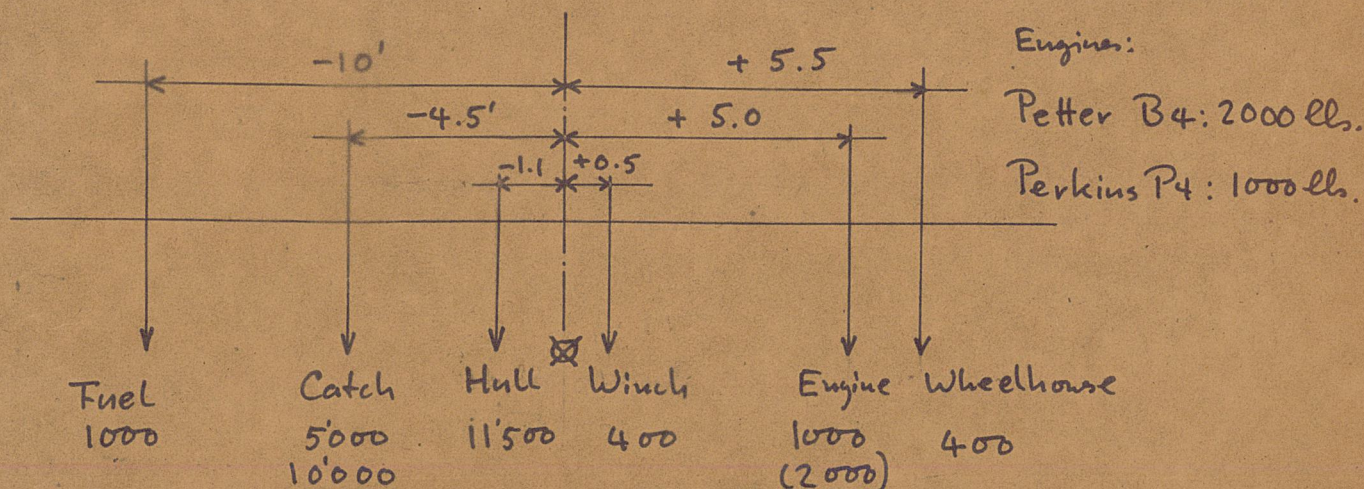
All built boats should be carefully tested and trimmed by the designer, and the ensuing correct position of CG entered in the original drawings for future reference.

For even trim (i.e. floating at, or parallel to DWL) LCG and LCB must be on a vertical line. A check should be made for a number of loading conditions by simple moment calculation. Any large deviation in the position of LCB and LCG, particularly a deviation resulting in a heel trim, must be carefully investigated. It must be seen whether a simple shifting of weights or a change in layout will be sufficient to bring the two centres in line, or whether it will be necessary to change the line, to adjust LCB to the required position.



Resulting trims of 3" and less either side can normally be neglected.

Such a loading and trim check will look somewhat as follows (example from actual calculations for 32 ft. shrimp trawler):



| Cond. I: empty: | Moments aft | Moments fwd.          | Weight       |
|-----------------|-------------|-----------------------|--------------|
| Fuel            | 10'000      | Winch -200            | 1000         |
| Hull            | 12'650      | Engine 5'000 (10'000) | 11500        |
|                 |             | Wheelh's. 2'200       | 400          |
|                 |             |                       | 1000 (2000)  |
|                 |             |                       | 400          |
|                 | 22'650      | 7'400                 | 14300        |
|                 | 7400        | (12'400)              | (15300) lbs. |
|                 | 15'250      |                       |              |
|                 | (10'250)    |                       |              |

$$LCG = \frac{-15'250}{14300} = -1.065' \text{ with Perkins P4}$$

$$= \frac{-10'250}{15'300} = -0.67' \text{ with Petter B4}$$

Cond. II: 5'000 lbs. catch and ice:

$$LCG = \frac{-37'750}{19300} = -1.95' (P4)$$

$$= \frac{-32'750}{20300} = -1.62' (B4)$$

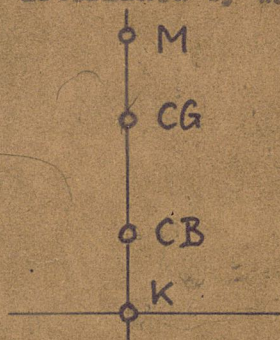
Cond. III: 10000 lbs. catch and ice:  $LCG = -2.48' (P4)$   
 $= -2.18'$

These conditions can fairly easily be met as far as LCB is concerned. At worst a light trim by the stern might be expected under full load.



### GM - The Metacentric Radius:

The vertical distance between the CG of the ship and the transverse meta-centre is an important factor when investigating the ship's initial, static stability. It is readily seen that GM depends on two major factors, vertical weight distribution and form of the underwater body. The latter has a great influence on GM, and its influence again is largely determined by waterline shape and area.



From this it is seen that:

$$KM = KB + BM$$

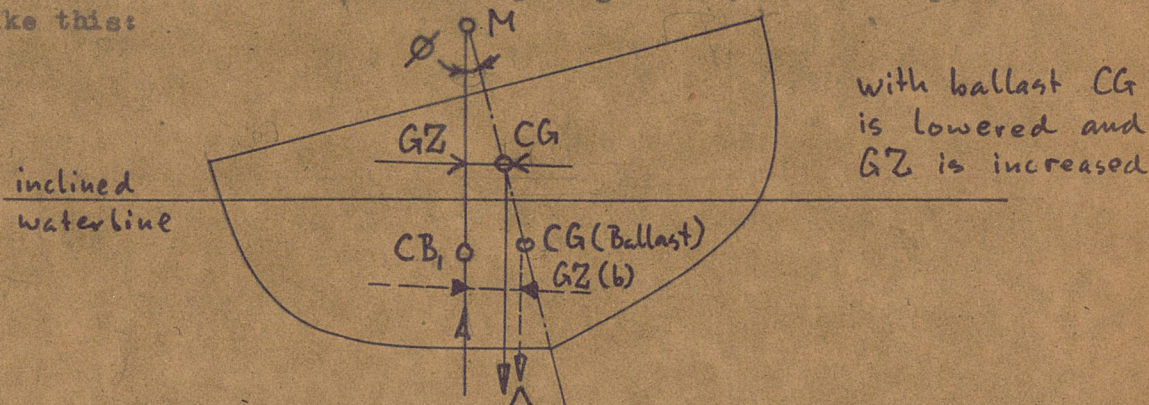
$$GM = KM - KG = KB + BM - KG$$

Both KB and BM are only dependent of form, while KG is dependent on vertical weight distribution.

The relative positions of CB and CG determine largely the amount and nature of stability. Form factors are of increasing importance with a CG above CB (normal in fishing boat work), while weight factors are overwhelming with a CG below CB. This latter is the case in sailing yachts, where the CG is brought down artificially by adding large amounts of outside ballast low down.

Many fishing boats have so little form stability that it becomes necessary to increase GM by lowering KG with ballast. However, this is undesirable in small boats, as it has been established that an increase in beam of the waterline (making for an increase in BM) has less adverse effects on resistance properties than the added displacement due to ballast. Ballast is dead weight which increases the necessary power to sustain a given speed!

The influence of size of GM on the righting ability of the ship is best seen like this:



The righting moment for the heeled ship is:  $M = \Delta \times GZ = \Delta \times GM \times \sin \phi$  where  $\phi$  is the angle of heel.

Initial GM becomes larger with an increase in waterplane moment of inertia, for a given displacement, or, if CG is brought down by adding ballast or redistributing the weights vertically. In a heeled condition, GM increases with increased movement of CB to the heeled side from its position at rest. This implies that buoyancy should be concentrated in the wings of the sections, coupled with large beam.

### Shape of midship section:

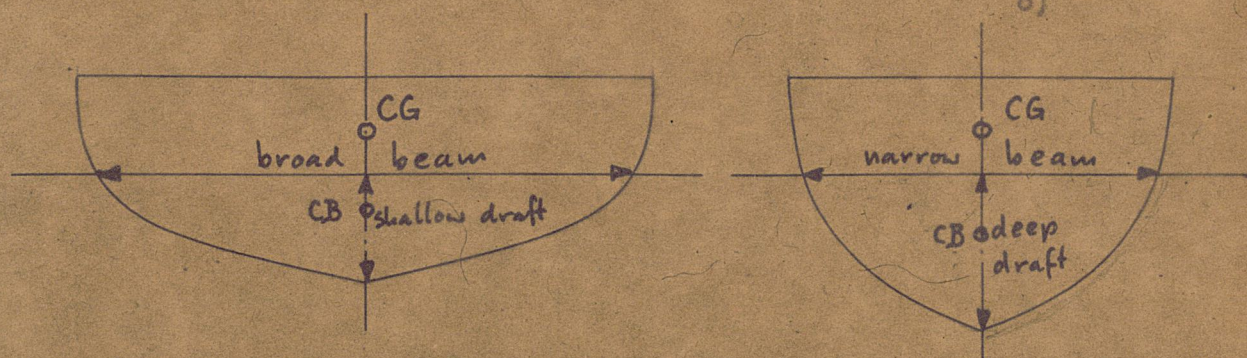
This will be a "formstable" ship

This will be a ship depending to a large extent on weight stability, or extremely low position of CG

Figures are meant for two ships of equal length and displacement.

(see next sheet)

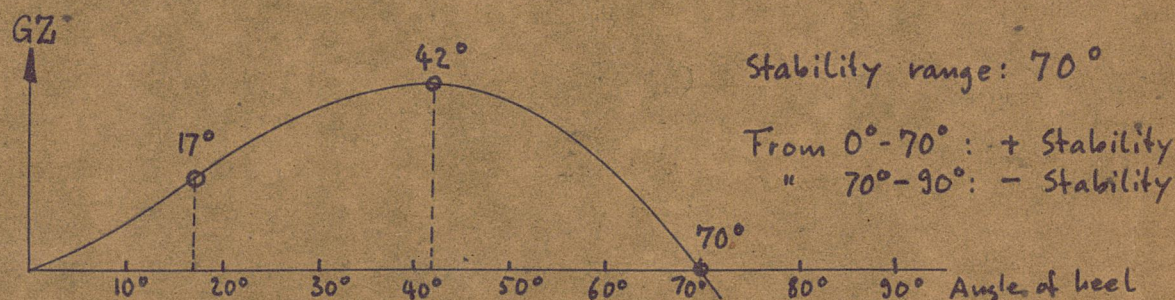




All the aforesaid is mainly connected with initial stability. Stability at large angles of heel follows somewhat more complicated criteria. It is obvious that a serious appreciation of a ship's stability should investigate the phenomena at large angles mainly, as the infinitesimal angles assumed for initial stability are of no great importance with regard to the safety of the ship. Unfortunately there is hardly ever the time available to make complete stability calculations for small or medium fishing vessels, although it is always done for large trawlers etc., obviously due to the risk and financial implications when building such a complex ship without reasonable security as to its stability and behaviour in adverse conditions.

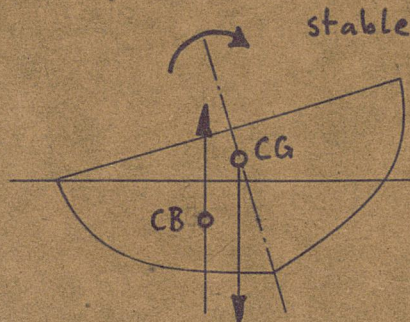
The important criterion at large angles of heel is the righting moment of the ship and not so much  $GM$  as such.

It is usual to investigate this with a number of curves of  $GZ$ , each for a definite loading condition.

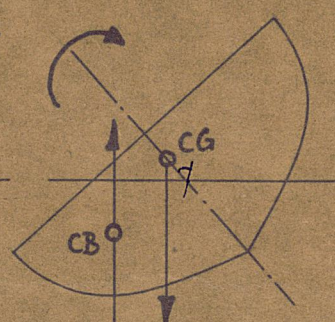


The ship in question would have a maximum heeling angle of  $70^\circ$ . Further heeling would reverse the righting moment into a heeling moment and the ship would capsize. The deck edge becomes immersed at an angle of heel of about  $17^\circ$ , indicated on the curve by the point of inversion of the curvature. The maximum righting moment would occur at an angle of  $42^\circ$ .

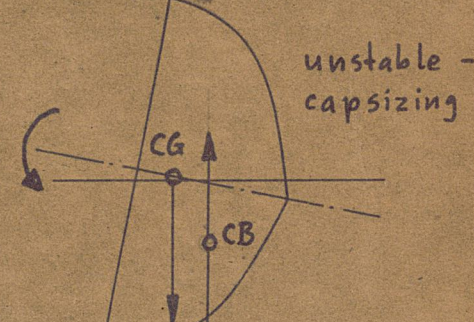
At  $17^\circ$



At  $42^\circ$



At  $80^\circ$



While rather accurate stability criteria for large ships are known, the small boat designer is somewhat at a loss as yet to determine the minimum  $GM$  he must give his boat. There are a number of suggestions as to how to assess the stability of small fishing vessels, but the lack of experimental and trial data in the very small size group ( $25' - 50'$ ) makes it somewhat doubtful whether these should be blindly followed.



Apart from initial stability, GM is of utmost importance with relation to the motion of the ship.

It is one of the hardly plausible facts of Naval Architecture, that a boat with large GM (large initial stability) will appear to be unstable in a seaway, while a boat with low GM will appear to be very steady. In terms of comfort this is expressed by rocky, fast motions with large GM and slow soft motions with low GM.

When examining the rolling characteristics of a ship, the natural period of roll is very much linked with GM.

The period of roll is the time that elapses between one complete roll over and back. The relation with GM is in the form of a relation with the transverse radius of gyration, and can be written in simplified form:

$$t = \frac{2 m B}{\sqrt{GM}}$$

where  $t$  is the period of roll;  $B$  the waterline beam in meters and  $GM$  in m.  $m$  is a factor which varies somewhat with the loading condition of the vessel and also would appear to have a different range for different types of ships.

For fishing boats in the 30 - 35 ft. range it appears to be varying from about 0.37 - 0.39, the lower figure applicable to a ballasted boat, or a boat with extremely low CG and the higher figure to the light boat. For boats in the 40 ft. range the values will have to be established yet when such boats are built and can be tested under operation.

The reverse relation of the formula gives:

$$GM = \left( \frac{2 m B}{t} \right)^2$$

This is conveniently used to estimate the GM of any finished boat in the water, by simply measuring the period of roll and the waterline beam. It is often a good way of convincing an owner that his boat may be over- or understable. Some ballast, put either low down, or high up, will result in a very noticeable change in rolling period.

$$t = \frac{2 m B}{\sqrt{GM}}$$

is a useful formula to assess the probable period of roll of a design after the GM has been calculated as accurately as possible.

For a given beam, a long period of roll would indicate little initial stability, while a short period would indicate sufficient or excess stability. A convenient rule of thumb says that the rolling period in seconds should be about the waterline beam in meters (or 0.305 times the beam in ft.), for a boat of normal proportions (see below).

This would obviously not hold for a boat with abnormally large beam for a given displacement, when GM would be very large and  $t$  would become less than  $B$  sec., while for a boat of equal displacement but unusually narrow beam,  $t$  would almost certainly be more than  $B$ .

For the foot - lbs. system, the corresponding formula in ft. is:

$$t = \frac{1.108 \cdot m B}{\sqrt{GM}} \quad ; \quad GM = \left( \frac{1.108}{t} \right)^2$$

NOTE: It would be advisable to measure the rolling period of as many boats as possible; the GM should be calculated accurately for the same boats with the help of the hydrostatic curves and an inclining test (see below). This would give a better idea of the influence on stability of various form factors, as well as tying down the range of the factor  $m$  for further reference.



Warning: It would seem advisable to avoid designing overstable fishing vessels. A fishing boat is essentially a working platform, and any excess of stability might mean that it could not be used as such any more in heavy weather, due to excessively accelerated motions. This would indicate that the use of excessive ballast should be discouraged. It is very difficult, if at all possible, to prescribe any rules for minimum GM that would still guarantee a safe ship over a wide range of heeling. It is supposed that GM of from 18" - 20" would be sufficient for very small boats (although in the case of open boats it would always tend to be more), while 20" - 24" should be regarded as sufficient for larger boats, say up to 100 ft. Much work is at present being done to bring more light into these relations; it should however never be forgotten that over and above the criterion for minimum initial stability, the range of stability is what matters most with regard to the safety of a ship. It is actually possible to have a safe ship with negative initial stability!

A further cautious note must deal with freeboard: For two given boats with identical displacement and waterline shape, the one with the higher freeboard will have less GM due to the higher KG. Thus, too much freeboard in small vessels may well result in deficient initial stability making for a tender boat. In this case, addition of ballast might become necessary to lower the CG again.

#### MTI - Moment to trim one inch:

To determine the change in trim brought about by a change in load distribution, added weights or an initial divergence of LCB and LCG, it is useful to know the moment necessary to change the trim by one inch.

The trimming moment at any moment is equal to the total weight of the ship times the distance between the two centres:

$$M = \Delta_{\text{total}} (LCB - LCG)$$

However, the ship will not trim about LCB or LCG, but always about the momentary LCF.

Thus, if a weight is added somewhere on the ship, the resulting trimming moment will be the added weight times the distance of its centre of gravity from the LCF:

$$m = w (cg - LCF)$$

w, cg standing for the added weight.

This moment would obviously have to be equal to the product of total weight of the ship (including added weight) and distance between the new CG and CB.

Trimming can be regarded as a similar feature to heeling, but in a longitudinal direction, and the calculation of MTI becomes then similar to the calculation of heeling moments. However, longitudinal moment of inertia of the waterplane and longitudinal GM,  $GM_L$ , is introduced now.

The longitudinal moment of inertia of a waterplane about any transverse axes:

$$I_L = \frac{2h}{3} (x_1^2 y_1 + 4x_2^2 y_2 + 2x_3^2 y_3 + 4x_4^2 y_4 + x_5^2 y_5) = \frac{2h}{3} \sum f(x^2 y)$$

or whatever number of ordinates there may be used. The value "x" stands for the distance of each ordinate from the axes; in shipwork usually the midship section.



The longitudinal moment of inertia about the centre of the waterplane (LCF) is then:

$$I_{LCF} = I_L - (LCF)^2 A$$

where A is the area of the waterplane

Heeling is largely dependent on BM as we have seen; similarly, trimming is largely dependent on the longitudinal BM, or  $BM_L$  as it is customarily written.

$$BM_L = \frac{I_{LCF}}{\nabla} = \frac{I_L - (LCF)^2 A}{\nabla}$$

Equilibrium in trimming will be established when:  $M = \Delta \times GM \times \tan \phi$

$$\tan \phi = \frac{t}{L}$$

$$M = \frac{\Delta GM t}{L}$$

where  $\phi$  is the angle of trim.

where t is the total trim and L the waterline length.

for "t" equal to 1 inch, we get:  $MTI = \frac{\Delta GM}{12L}$

the Moment to trim one inch.

$I_{LCF}$  is normally a very large figure and so is  $BM_L$ . BG is comparatively small and we may assume that

$$BM_L \approx GM$$

Using this relation we may write:  $MTI = \frac{\Delta BM_L}{12L} = \frac{\Delta \times I_{LCF}}{35\Delta 12L}$  or

$$MTI = \frac{I_{LCF}}{420L}$$

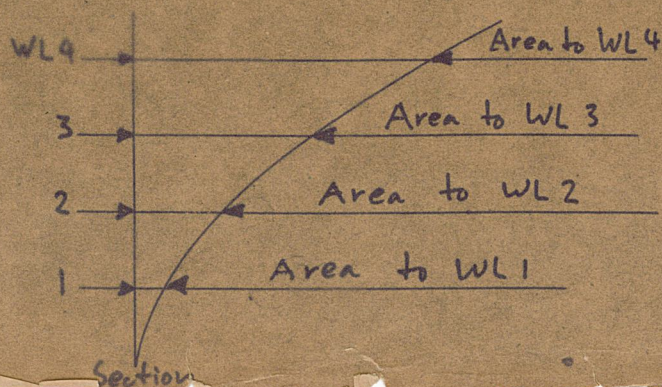
where  $\Delta$  is tons

If the vertical position of CG (KG) is not known accurately, this approximation will give useful results when examining the occurring trims for various load conditions of a new design.

**NOTE:** The method again is only of acceptable accuracy for very small trims! For large resulting trims, a trial and error method must normally be adopted, calculating displacements and LCBs for various inclined waterlines until the correct one is found. For this purpose the so called Bonjean curves will be useful helps.

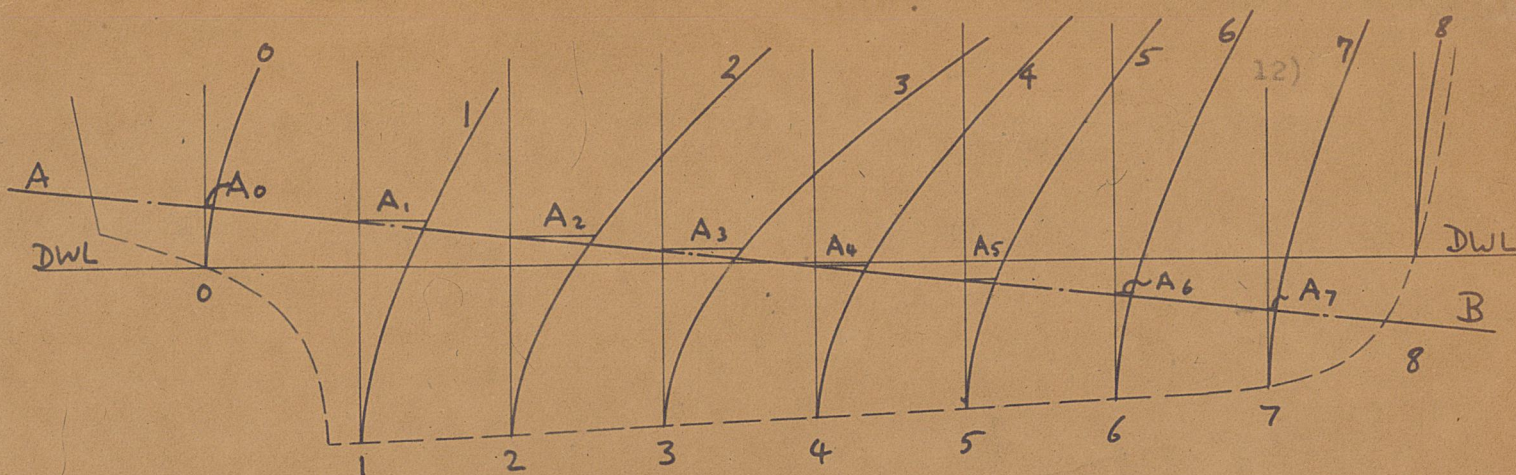
### BONJEAN Curves

As the section area for each section grows with increasing draft, it can be plotted as a curve where the ordinate is draft and the abscissa the section area up to this draft.



At each waterline level the corresponding section area is plotted horizontally as a linear measure in convenient scale. If this is done for each section in the length of the ship, a plot like below is obtained:





This is often done on a drawing where the length scale is reduced and the height scale (draft scale) is retained as in the lines plan. Thus with a lines plan scale of 1:12 and a ship's length of 45 ft., it would be useful to plot Bonjean curves on a length scale of 1:48 and the drafts again as 1:12.

To find the displacement and LCB for a waterline AB, parallel lines to the DWL are drawn through the intersection points of the section lines and the line AB. On these horizontal lines, the section area at each section is picked off. Displacement and LCB are then obtained by the usual process of applying Simpson's appropriate rule.

The same set of curves would be used to study the movement of LCB when a wave passes the ship. For this purpose a wave profile would be superimposed on the Bonjean plot and again proceeded as above.

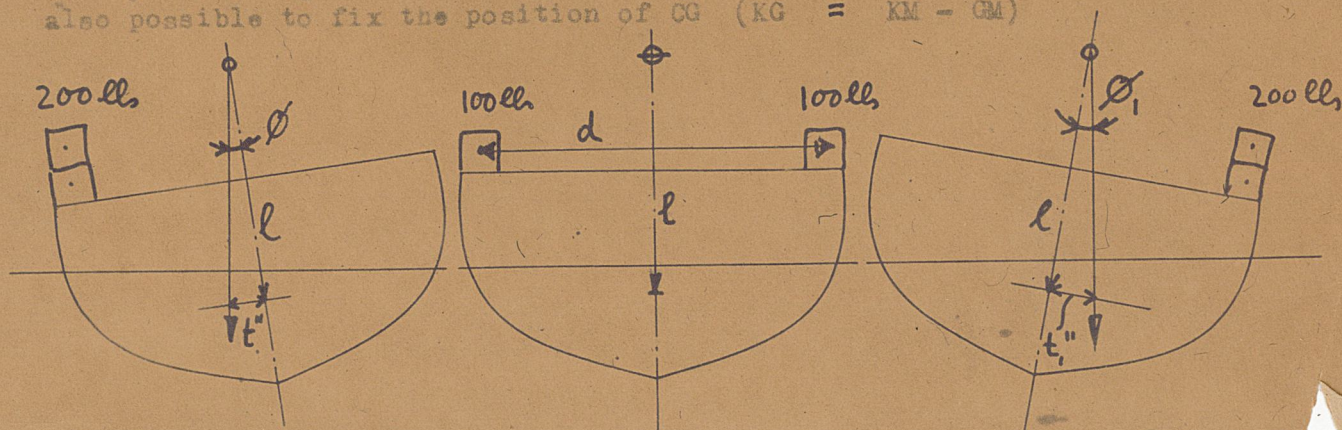
As the section areas up to the individual waterlines are normally taken to construct a full displacement curve, it is simple to expand these values into Bonjean curves. No additional work is involved, except preparing the plot.

### The INCLINING TEST

This is used to establish the GM and KG of a ship after construction. In the case of large ships it is compulsory to conduct an inclining test before delivery is taken from the yard, to establish beyond doubt that existing criteria for minimum GM are met.

It should be done also for all other vessels, if only to compile a comprehensive collection of correct GM and KG values for a variety of vessels. This data will be of tremendous value in future designs and can help considerably to reduce working time in such designs, by allowing the designer to make shortcuts and assumptions as would otherwise not be justified to make.

The principle is simple: Some known weights are moved from one side of the ship to the other, and the angle of heel is measured. Knowing the total weight of the ship, the weight of the movable mass, the distance through which it was moved and the angle of heel, it becomes simple to calculate GM. By obtaining KB and BM from hydrostatic curves or lines plan it is then also possible to fix the position of CG ( $KG = KM - GM$ ).





If the 100 lbs. from starboard are lifted on top the 100 lbs. on the port side, the ship will heel  $\theta$  degrees. If then the 200 lbs. are shifted back to starboard, the heeling angle will be  $\theta_1$  degrees to the other side. In both cases the weights were shifted through a distance  $d$ , while the pendulum deflections are read as  $t''$  and  $t_1''$  respectively, with a pendulum length of  $l$ .

Care must be taken to avoid large angles of heel, as then we would no longer be concerned with initial GM but momentary GM at a large angle.

The heeling moment is:  $M_h = w \times d$  where  $w$  = shifted weight

while the resulting righting moment is:  $M_r = GZ \times \Delta_{total}$  or

$$M_r = GM \times \sin \theta \times \Delta$$

Due to the small angles of heel,  $\sin \theta$  can be assumed to be equal to the circular measure or  $\tan \theta$ . This, by definition, is  $\frac{t}{l}$  (the pendulum length) divided by (the mean deflection read)  $\Delta$ .

$$\text{Since } GM \times \tan \theta \times \Delta = w \times d \rightarrow GM = \frac{w \times d}{\tan \theta \times \Delta}$$

$$\tan \theta = \frac{t}{l}$$

If  $l$  and  $d$ , as well as  $t$  are measured in inches,  $w$  and total weight in lbs., the GM in inches becomes:

$$GM = \frac{w \times d \times l}{\Delta \times t}$$

The experiment should be carried out a number of times to either side to allow the use of a mean value for deflection.

A number of precautions have to be taken, to assure reasonable accuracy in the result:

- a) No bilge water is permissible due to the reduction in GM with free liquid surfaces.
- b) Fuel tanks must be either completely empty or topped up full for the same reason.
- c) Tests must be done in absolutely calm water with no tide running and as little wind as possible.
- d) Slack mooring lines are essential to allow free movement of the boat.
- e) To facilitate calculations with the help of lines and hydrostatic curves, it is advisable to bring the vessel to float at a waterline parallel to DWL. To ensure this small ballast may be added forward or aft. This ballast must be kept in mind when using the results when establishing GM and KG at different loading conditions.
- f) No initial heel is admissible. To check on this, measure freeboard not only at stem and stern, but also amidships, where it obviously must be equal on either side.
- g) Only one man should be on board when testing small vessels. He should keep very still when reading pendulum deflections and should do so always in the same position.
- h) The exact drafts of the vessel have to be obtained in the testing condition (with weights and man on board) to establish the correct total weight.
- i) The pendulum should be at least 6' long to give good deflections. To dampen its swing, the lead may be hung into oil or water.

As many boats as possible should be subjected to an inclining test to gain a good knowledge of the CGs of different types and their initial GM for future reference work.



## Rudder Design

The action of the rudder is determined largely by 3 main points:

- a) Area
- b) Aspect Ratio
- c) Rudder angle

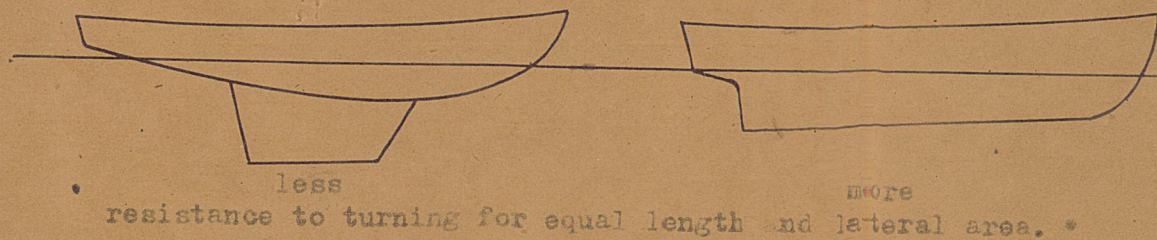
a) is usually expressed as a percentage of the lateral area of the boat.  
Suggested relations are:

| Lateral Area<br>ft <sup>2</sup> | % Rudder Area |              |                           |
|---------------------------------|---------------|--------------|---------------------------|
| 50                              | 3.5           | ( 1 / 28.5 ) | suggested by Philips-Burt |
| 100                             | 3.2           | ( 1 / 31.3 ) |                           |
| 500                             | 2.4           | ( 1 / 41.7 ) |                           |
| 1000                            | 2.15          | ( 1 / 46.5 ) |                           |

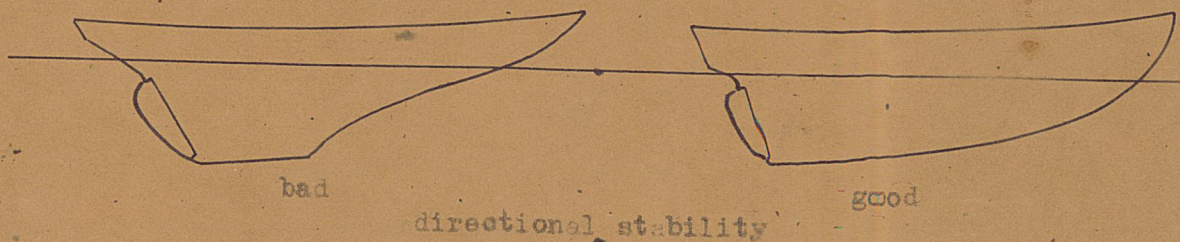
K.C. Barnaby suggests: For small craft - 1 / 30  
for liners - 1 / 70

Rudder area changes with lateral area rather than with type of vessel.

However, the disposition of the lateral area is of importance to the resistance to turning:



Good steering refers to small rudder pressures and good directional stability; these are again two requirements that can only be met by compromise:



If a large fin or skeg is fitted aft, astern steering becomes very difficult. (See FAO Beach Boats!). If good stern steering is required, a bow rudder must be fitted.

b) Aspect Ratio<sub>2</sub> is a term taken over from aircraft wing design. It denotes the ratio: Depth / Area, or, for a rectangle, Depth / Width. The ratio determines largely the proportions of lift and drag on a rudder blade.

The lift / drag ratio is an indication of the rudder force or turning moment at a given angle.

| Aspect ratio | Flat plate | Lift / Drag | Streamlined plate |
|--------------|------------|-------------|-------------------|
| 0.5          | 6          |             | 6                 |
| 1.0          | 7          |             | 12.5              |
| 2.0          | 9.5        |             | 16.0              |
| 3.0          | 12.0       |             | 19.5              |



High aspect ratio indicates higher lift / drag ratio, or smaller rudder angle for a given force.

High aspect ratio is achieved by a deep, narrow rudder.

- Limitations:
- 1) The ship's draft normally restricts the aspect ratio for a given area.
  - 2) A rudder of very high Ar becomes less effective at higher rudder angles - this phenomenon is related to the "stalling angle" of aerofoils - the angle of climb, above which the wing of the aircraft does no longer "bite".

| Aspect ratio | Rudder pressures for rudder angles of |      |      |      |
|--------------|---------------------------------------|------|------|------|
|              | 10°                                   | 15°  | 20°  | 30°  |
| 2.0          | 0.5                                   | 0.65 | 0.58 | 0.55 |
| 1.0          | 0.3                                   | 0.5  | 0.64 | 0.66 |
| 0.5          | 0.2                                   | 0.33 | 0.47 | 0.78 |

These values are valid for spade rudders. For rudders hung directly on a fin or deadwood, higher stalling angles are obtained, and the higher aspect ratio rudder will be superior up to rudder angles of 30° - 35°. (This is again reminiscent of flaps on aerofoils, used for landing and take off to push out the stalling angle).

Must rudders exceed their stalling angle at 35° at the most. That is the reason why rudder action at large angles is less effective than at smaller ones.

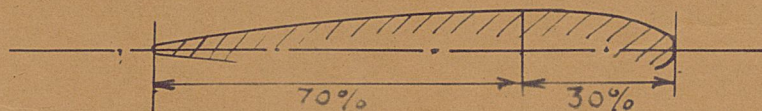
For the average fishing boat it would seem that aspect ratios of 1.5 - 2.0 would give best results. Higher ratios would seldom be possible without reducing the area proper of the rudder, and they would also be dependent on hanging the rudders on effective fins.

#### Rudder shape

When area and maximum aspect ratio are fixed, it is seldom possible to fiddle much with outline shape; rectangular rudders normally give best results.

However, the section of the rudder blade is of considerable importance, as the lift / drag ratios at the bottom of page 14) indicate: The superiority of streamlined sections is apparent.

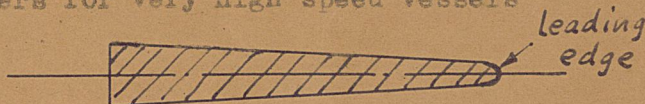
Streamlined sections normally have the shape of an aerofoil, i.e. the maximum thickness occurs at about 30% of the width from the leading edge.



Under no account should a rudder blade be thicker than the deadwood or fin in front of it!

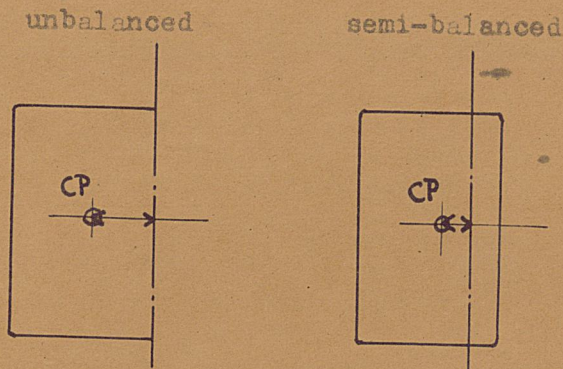
The rudder thickness may be about 0.06 - 0.13 of its width; thus a blade 2 ft. wide may be between 1 1/2" - 3" thick. 2" - 2 1/2" would probably be correct for a wooden rudder, while a built up metal rudder could be less.

It appears from recent research that rudders for very high speed vessels could profitably have sections like this:



It is however outside the scope of these notes to go into this question any further; very high speed design problems do in any case not normally pop up when designing fishing boats.



Balanced Rudders

With a semi-balanced rudder, the centre of pressure moves nearer to the rudder stock, thus reducing torsional stresses.

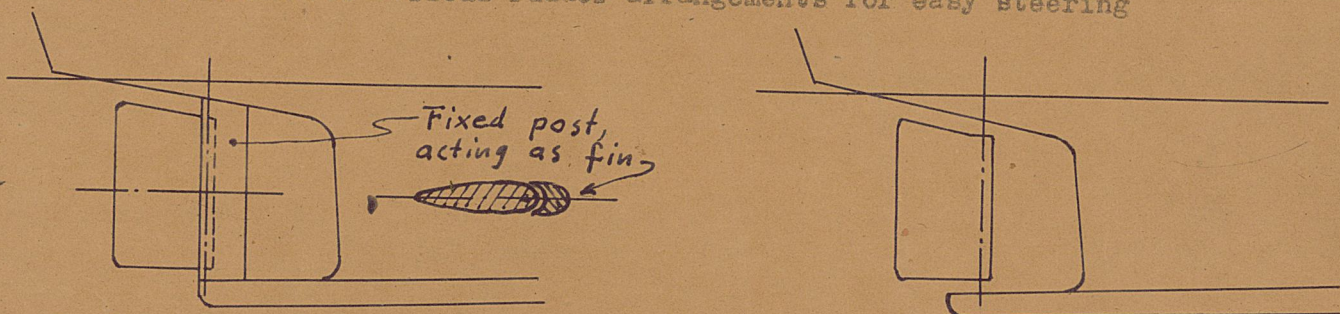
It is important to insure that the centre of pressure does not move forward of the rudder stock at any angle; this would result in locking or of the rudder - a occurrence that can have dangerous results in boats with hand steering.

Less than 20% balance area is indicated in these cases; steering leads already tend to be over sensitive and must be of very positive action type.

Semi balanced rudders would require approximately 10% more total area for rudders associated with small lateral areas, and about 5% for those acting with large lateral area.

Rudders for planing craft may have less balance only to prevent locking over; this phenomenon is not well studied as yet, but it is understood that the suggested wedge type rudder for very high speeds has something to do with it.

Relative merits of various rudder arrangements for easy steering



Good: effective shaping and sharpening of posts essential for good propeller performance.

Not so good - but the usual solution! Shaping of rudder blade not so essential.

Steering and steering leads

The simplest and best steering arrangement for very small boats is without doubt the stern hung rudder with a direct action tiller. Tiller steering can also be used if the rudder is fitted with a stock through the horn timber as on the 25 ft. fishing launch.

If steering of the boat is not possible aft due to some operational requirements, resort is normally taken to wheel steering. This may be arranged in many different ways - as simple tiller and lead (chain or wire), quadrant and lead or in bigger vessels with steering gears and motors.

Steering leads should be very carefully laid. It is essential that they are not too tight - but also not too loose. Rigging screws may be used at shackle joints to keep the tension on the system constant. Leads should be passed over suitable rollers and sheaves and should be held in place by these and additional clips if necessary. It is good practice to pass leads through pipes, at least where they have to pass through bulkheads or decks.



Important note: Where quadrants are situated under decks, arrangements must be made for an emergency tiller to be attached to the rudder stock and operated above deck, in case of breakage of steering leads.

### Rudder Stocks

A number of formulae are in use to determine the diameter of stocks; none of these is absolutely accurate, and it will usually pay to use a stock of slightly higher diameter than calculated.

It must be borne in mind that rudders of low aspect ratio impose large twisting forces on the stock, while those of high aspect ratio impose large bending moments, especially with balanced rudders, as these can only be held in two bearings on top and bottom.

### Specimen rudder stock calculation

A = Area of rudder                      B = Width  
 CP = Centre of pressure, assumed at 0.375 B from leading edge  
 cp = Distance CP to rudder axis  
 y = " CP below stock bearing  
 V = Speed in knots  
 $\phi$  = Rudder angle

Pressure  $P = 3.2 \times A \times V^2 \times \sin \phi$  tons

Torque =  $P \times cp$  ft. tons

Twisting Moment =  $P \times cp \times 12$  in. tons = TM

Bending Moment =  $P \times y \times 12$  in. tons = BM

Equivalent twisting moment  $T = BM + \sqrt{BM^2 + TM^2} = 1/16 \pi f \times d^3$

where  $f$  = breaking strength of stock

$d$  = diameter

This formula is British standard practice.

Caution: For the ordinary steering arrangement with wheel, chain (or wire) and quadrant, it would seem advisable to be very cautious with balance! Steering will get very tiresome if rudder develops any tendency to lock over at small angles. 10% balance may be used (see Fishtech II).

### Astern steering

When backing, the CP moves aft and much greater torques are experienced. If the rudder is put hard over astern, it may not be possible to bring it back without breaking anything! This need not only happen at high speeds!

Barnaby offers as an explanation the fact that a rudder, when going astern, works in undisturbed water and may build up abnormally high lift coefficients before the pressure suddenly breaks down.

When going astern, rudder should be used very sparingly, and high speed should be avoided at all cost.



## Part 2: Resistance and Propulsion

Resistance is the sum of all outside effects having an influence on a moving ship, and making it necessary to equip the ship with a power plant for its propulsion. The magnitude of the resistance at a given speed is equal to the force that would be required to tow the ship at this speed.

Roughly, total resistance is made up of the following parts:

- a) Frictional Resistance -  $R_f$
  - b) Wavemaking Resistance
  - c) Head Resistance
  - d) Eddy Resistance
  - e) Wind Resistance
- } - Residuary Resistance -  $R_r$

a) and b) will be treated at some length, while c) can normally be neglected in a well designed ship (sharp stem!); d) is mainly caused by appendages, i.e. rudder, shaft brackets, bilge keels etc. and e) is caused by the above water part of the ship, mainly its superstructures, offering air resistance.

b), c) and d) together form residuary resistance, which will be treated in detail.

The diagram on page 19) shows roughly the relative distribution of  $R_f$  and  $R_r$  over a range of Speed/Length ratios. This diagram is taken from Philips-Burt, Naval Architecture of small craft. It is not meant to give accurate figures and can not be used for power calculations - it simply shows the trend!

Resistance below  $V/\sqrt{L} = 1.0$  is very low, and consists mainly of friction. A cargo ship might have about 3 - 4 lbs./ton total resistance, and 65% - 80% of this will be frictional. The ship will create little disturbance in the water as it travels along.

Around  $V/\sqrt{L} = 0.7 - 1.0$  total resistance becomes about 10 - 15 lbs./ton. Fast liners and express cargo boats (fruit carriers) operate in this range. Fric.resistance accounts for between 50% and 70% of the total.

From 1.0 - 1.25 is the range in which fast ferry boats, warships etc. operate. Total resistances might be in the range of 30 lbs./ton displacement, accounting for the tremendous power plants these ships need to sustain their speeds.

A 500 ft. cruiser, travelling at 28 knots will have a  $V/\sqrt{L} = 1.25$   
Assuming a displacement of 3000 tons and a total resistance of 30 lbs./ton, the  $R_t$  becomes 90'000 lbs.  
EHP then becomes  $0.00307 \times 90'000 \times 28 = 7'800$  h.p.

Small boats in general, and fishing vessels in particular operate in this speed/length range!

$V/\sqrt{L} = 1.0 - 1.25$  represents the following speeds for boats from 25 - 100 ft. L:

| $L_{wl}$ | 25   | 30   | 40   | 50   | 60   | 80    | 100  |
|----------|------|------|------|------|------|-------|------|
| V (1.0)  | 5    | 5.5  | 6.25 | 7    | 7.75 | 9     | 10   |
| V (1.25) | 6.25 | 6.75 | 8    | 8.75 | 9.75 | 11.25 | 12.5 |

Taking a 40 ft. boat at 8 knots, its speed is perhaps only 25% higher (relatively) than the speed of an express liner of 450 ft. length at 21 knots; its specific resistance of perhaps 60 lbs./ton however can be 300 - 500% that of the liner!



# RESISTANCE - SPEED

lbs/ton

200

$\Delta R_t$

100

0

Cargo shipping

Liners

Warships  
Fishing Boats

Limit Speed

1.0

Displacement Craft

1.0

2.0

Semi planing

2.0

3.0

Planing hulls

3.0

$V/\sqrt{L}$

Total Resistance (Frictional + Residuary)

Frictional Resistance (% of Total)

100%  $R_f$

50%

0%



At still higher  $V/\sqrt{L}$  ratios, the phenomenon of changing trim due to speed becomes increasingly noticeable.

At about 1.5, wave making resistance reaches its maximum proportion of about 85% of total resistance. Changes in hull design become necessary to prevent boats from becoming unseaworthy.

At about 1.6  $V/\sqrt{L}$ , a boat will run in what amounts to a deep groove in the water; the slightest wave could roll on board, and a heavy displacement boat would no longer be safe. Fortunately, we do not have to worry about this condition with heavy displacement fishing vessels; no amount of power could drive such a hull over a  $V/\sqrt{L}$  ratio of about 1.55!

At this point in the range, drastic changes in hull form become necessary. As the wave system created by the boat exceeds the boat's length after 1.34  $V/\sqrt{L}$ , it becomes important to ensure that this wave system is made up of shallow waves, to reduce the trim effects. Flat runs with straight buttocks and wide, flat transoms become increasingly important should the boat attain such high speeds. As can be seen from the diagram, the total resistance in this region will be somewhere near the 150 lbs./ton mark; however, the resistance curve flattens out, and at 2.0 it has about the same slope as at 1.0. At this stage sinkage due to speed as observed before will cease with the correct hull form, and at  $V/\sqrt{L}$  2.2 - 2.5 the hull will again have assumed its original static trim, with a tendency to rise with still increasing speeds. Planing motion starts!

As speeds continue to rise, the percentage of wave making resistance will decrease steadily and the main object in design of hull forms for such speeds becomes the reduction of friction resistance again.

After  $V/\sqrt{L}$  3.0 it becomes inevitable to use V-bottom, hard chine hulls of full planing type. At 4.0, wave making resistance is about the same percentage as at 1.0 - total resistance, however, has now mounted to perhaps 1400% of that at 1.0! One of the reasons for the relatively low  $R_w$  at these speeds is the fact that the hull has now risen high over its static trim.

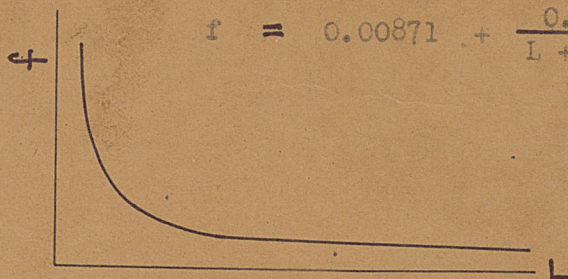
### Friction Resistance

The first to seriously investigate this was W. Froude, who found:

$$R_f = f \times S \times V^{1.825} \quad \text{where} \quad \begin{array}{ll} S & = \text{wetted surface} \\ V & = \text{speed in knots} \\ f & = \text{friction coefficient} \end{array}$$

Much work went into the determination of a suitable friction coefficient, and in 1935 it was agreed to use  $f$  values as under:

$$f = 0.00871 + \frac{0.053}{L + 8.8} \quad \text{resulting in a curve thus:}$$



$$\text{for } L = 16 \text{ ft.} \quad f = 0.01085$$

$$L = 400 \text{ ft.} \quad f = 0.00884$$

More modern practice takes into account the theory of viscous fluids, and the concept of Reynold's Number makes its appearance.

$$R_n = \frac{VL}{\nu} \quad \text{where } \nu = \text{kinematic viscosity}$$

The basic formula for friction resistance now becomes:

$$c_f = f(R_n) \quad \text{or, the friction coefficient is expressed as a function of Reynolds number.}$$





This relationship has been worked out by several people and the following results are commonly in use:

|            |                                                 |
|------------|-------------------------------------------------|
| Schoenherr | $0.242 \times c_f = \log_{10} (R_n \times c_f)$ |
| Prandtl    | $c_f = 0.455 (\log_{10} R_n)^{-2.58}$           |
| Telfer     | $\rightarrow c_f = 0.0012 + 0.34 R_n^{-1/3}$    |
| Falkner    | $c_f = 0.0306 R_n^{-1/7}$                       |

Schoenherr's notation is accepted as the most accurate, and is generally used in America. Prandtl is often used for simplicity.

In terms of Froude values, the relation becomes:

$$c_f = 2.772 \times f \times L^{0.175} \times R_n^{-0.175}$$

which is somewhat similar to Faulkner.

Friction resistance may then be written:

$$R_f = 2.824 \times c_f \times S \times V^2$$

Schoenherr coefficients are given in many books on a bases of  $R_n$ .

What now is Reynold's number? According to Prof. Reynolds, the characteristics of a flow system along a surface may be assessed by

$$\frac{VL}{\nu}$$

where  $\nu$  is the kinematic viscosity, which changes with salinity and temperature.

At 59°F (15°C) and 3.5‰ salinity,  $\nu = 0.000012817$  or  $1.2817 \times 10^{-5}$

$$R_n = \frac{V(\text{ft./sec}) \times L(\text{ft.})}{1.2817} \times 10^{+5} \quad \text{or}$$

$$R_n = \frac{V(\text{knots}) \times \frac{6080}{3600} \times L(\text{ft.})}{1.2817} \times 10^{+5} \quad \text{for normal conditions of salinity and temp.}$$

For a ship, 100 ft. long, at 7.589 knots,  $R_n = 1 \times 10^8$

This allows to simplify the formula somewhat, by writing

$$R_n = \frac{V \times L \text{ for a ship}}{V \times L \text{ for 100 ft. ship}} \times 10^8 \quad \text{or}$$

$$R_n = \frac{VL}{758.9} \times 10^8$$

for a 30 ft. boat at 6 knots,  $R_n = 2.375 \times 10^7$

for 500 ft. ship at 25 knots,  $R_n = 16.45 \times 10^8$



Two main flow characteristics have to be examined.

Laminar flow and Turbulent flow, or, as the names suggest,  
Smooth flow and Disturbed flow.

The important distinction is that laminar flow along a surface results in less friction resistance than turbulent flow. This is unfortunate, as boats and ships, with the possible exception of very well finished sailing yachts under ideal conditions, always move in turbulent flow. The main reason why there is great interest in laminar flow phenomena is its occurrence under testing conditions in ship model tanks.

The  $c_f$  values for turbulent and laminar flow show that it would be extremely valuable if a ship could be conceived that could travel in laminar flow:

|                          |                   |                           |            |
|--------------------------|-------------------|---------------------------|------------|
| At $R_n = 1 \times 10^6$ | $c_{ft} = 0.0045$ | at $R_n = 10 \times 10^6$ | $= 0.003$  |
|                          | $c_{fl} = 0.0014$ |                           | $= 0.0004$ |

As  $R_f$  is proportional to these values, the low resistance resulting from laminar flow is evident for any given ship.

### Surface Roughness

Froude studied this in his early work by using various grades of sand on the surfaces of plank which he towed along his tank. He then gave coefficients for surface finish, varying from high grade varnish to coarse sand.

| Resistance           | Surface |           |             |             |
|----------------------|---------|-----------|-------------|-------------|
| 100 ft., 20 ft. long | Varnish | fine sand | medium sand | coarse sand |
| at 5 knots           | 21 lbs. | 34        | 38          | 42          |
| Increase in %        |         | 62        | 81          | 100         |

It is now known that these results are questionable, since there was laminar flow present over a portion of the plank; furthermore Froude conducted his experiment in waters of changing kinematic viscosity. Yet the trend is obvious.

Much research work is being conducted into the question of bottom finish influences; one of the problems encountered is the classification of finishes so that they can be reproduced anywhere.

The designer will be presented with the problem to achieve a hull with lowest possible friction resistance during his search for an easy driven hull.

### How to reduce skin friction

At low and extremely high  $V/\sqrt{L}$  values, this is the only way to reduce total resistance.

From Froude's formula it becomes evident that for a given speed both  $f$  and  $S$  should be reduced to achieve lower  $R_f$ .

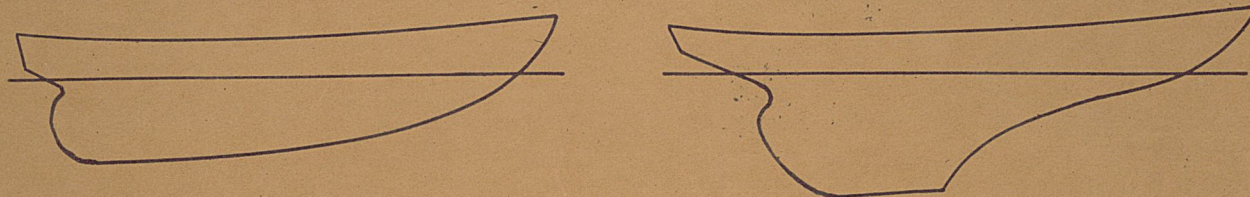
From the above it can be seen that  $f$  is very dependent on roughness, while  $S$ , of course, is a given quantity for a given hull.

When working out  $S$  from a given design, it is readily seen that small changes in hull dimensions will have very little effect on  $S$ , for a given length and displacement. This is the reason why  $S$  is normally not taken into account when fixing the main dimensions of fishing vessels.

Furthermore, as such ships are operating normally around  $V/\sqrt{L}$  values of 1.0 - 1.3, where  $R_f$  in any case only accounts about 40% - 25% of the total resistance, it is readily seen that a small reduction in  $R_f$  due to a decrease in  $S$  would bear very little fruit.



In the case of very slow or very high speed craft, the position is somewhat different, as there  $R_p$  may be anything like 70% - 90% of the total resistance! Sailing yachts with their large wetted surfaces due to their deep ballast keels have shown a marked tendency towards reduction of  $S$  since Froude in this way:



The ideal ship, the one operating under laminar flow conditions, has not been conceived yet. It is not likely ever to happen, not least because laminar flow is only possible at low Reynolds numbers. Thus only small boats at low speeds could profit from it.

It is interesting to note that there is a theory which alleges Dolphins to operate under laminar flow conditions, as otherwise they could hardly achieve speeds of 10 - 25 knots!

Fouling of ship's bottoms is a very important cause of added resistance, at least with vessels operating under  $V/\sqrt{L} = 1.0$ . Ship's bottoms should be regularly scraped and they should be protected against marine growth. This growth is especially strong in warm waters with high light intensity, and it adds roughness to the surface. As previously mentioned, skin roughness may even double frictional resistance in bad cases.

At  $V/\sqrt{L} = 1.4$ , a doubling of frictional resistance due to fouling might only mean a 20% increase in total resistance; at 1.0 it would mean most likely an increase of 50%.

### Wave - Making Resistance

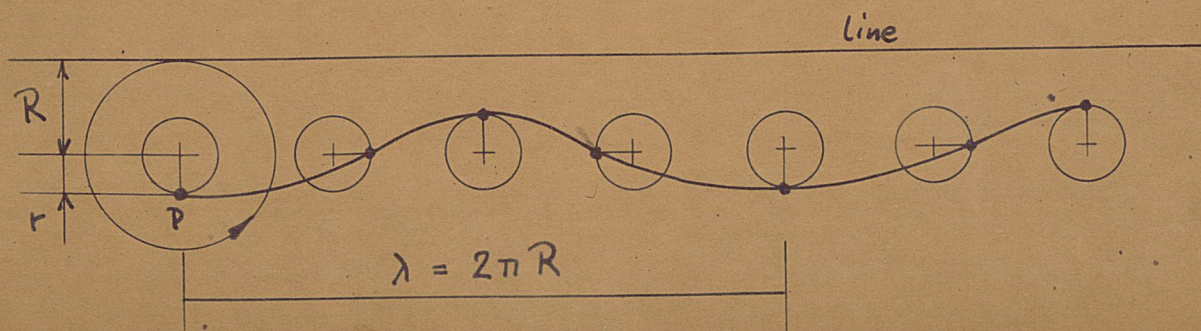
This is of utmost importance to the fishing boat designer, as fishing boats generally work in the high wave making resistance speed range.

Investigation of flow patterns around a hull in motion is a very complex problem; for the purpose of this treatise there must be great simplifications.

Generally speaking the flow is not around the hull, but rather under the hull, in the direction of buttocks or very steep diagonals. These lines on the linesplan are perhaps telling more regarding the waterflow than waterlines.

Waves are created by pressure differences at the surface; generally there are positive pressures at the ends of a hull and negative pressures in the middle, up to about  $1.3 V/\sqrt{L}$ .

Ideal waves are of Trochoid shape, i.e. they follow strict mathematical rules; the "wave" is the tracing of point  $P$  at a given distance  $r$  from centre of circle of radius  $R$  moving along line:





Wave length  $\lambda = 2\pi R$

Wave height  $h = 2r$

In actual fact the wave does not travel forward, but is caused by the orbital motion of water particles, started at the origin of the wave by a pressure difference.

This can be observed when watching driftwood in a wind induced ocean swell: At the crest of the wave, where the water particles move tangential and in the direction of the wave, the driftwood will advance a little in the direction of the wave; in the trough, where the particles again move tangential, but in the opposite direction, the driftwood will move slightly against the wave direction, the two movements creating the familiar to and fro motion of drifting objects. Between crest and trough there is obviously increasing and decreasing vertical motion.

Trochoidal waves can easily be studied and certain laws of behaviour found; the results are presented here without proof, as the mathematical implications are outside the scope of this text - the theory of trochoidal waves can be studied in a number of books on the subject of theoretical naval architecture.

The speed of the wave in knots follows the relation:

$$V^2 = 1.8 \lambda \quad \text{where } \lambda \text{ is the wave length in ft. from crest to crest.}$$

This results in  $V = 1.34\sqrt{\lambda}$  a result to be remembered!

As a wave is created by a travelling disturbance (ship), this will mean that the wave length is dependent on the speed of the ship that creates the wave.

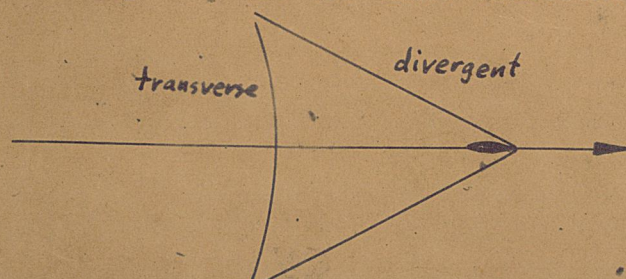
The speed / length ratio  $V/\sqrt{L}$  is derived from the trochoidal wave theory and simply shows the relation between the length of the wave created by the ship travelling at speed  $V$  and the length of the ship. At  $V/\sqrt{L} = 1.34$  these lengths are equal!

A ship travelling at a speed / length ratio of 1.34 will create a wave system of the ship's own length.

The height of the wave has no effect on the speed! However, the energy stored in the wave is dependent on the length and the square of the height. The energy stored into a created wave is produced by the ship's power! Thus the power that is lost in wave making is largely dependent on the wave length and the square of the wave height. The wave length, on the other hand, is dependent on the ship's speed, while it has been found that the wave height is largely dependent on the ship's bulkiness, or it's displacement and the distribution thereof.

Thus wave making resistance is largely governed by speed and displacement and prismatic coefficient.

A moving ship creates three different wave systems:



At low speeds the divergent systems are clearly visible, travelling at speeds equal to the ship's speed component in their direction.



At higher speeds the transverse wave system will show up clearly in the form of crests and troughs along the ship's side and as distinct waves behind the ship. Photographs taken from the air, of ships travelling at high speeds through still water, show these systems of waves very nicely.

The transverse wave system is the major source of wave making resistance!

A ship traveling at a particular speed will create THE wave system belonging to this speed:

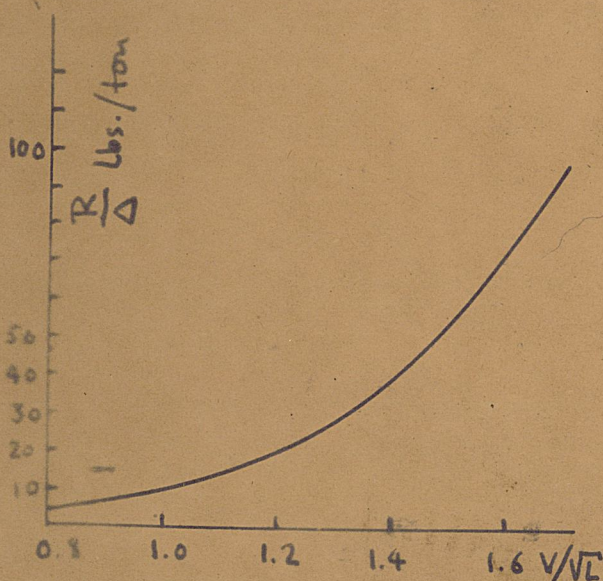
|                |    |      |      |    |      |      |      |     |
|----------------|----|------|------|----|------|------|------|-----|
| V knots        | 6  | 7    | 8    | 9  | 10   | 11   | 12   | 14  |
| $\lambda$ feet | 20 | 27.2 | 35.6 | 45 | 55.6 | 67.3 | 80.1 | 109 |

Fishing boats operating in the 6 - 10 knot range will thus create waves of 20 - 56 ft. length.

A 30 ft. boat running at 8 knots will create a wave of about 36 ft. The boat is very much overdriven, and will have to be of special hull shape to minimise the energy that is dissipated in creating the wave. Also its displacement will have to be so distributed that the wave height created is as small as possible, as otherwise the boat would trim very badly by the stern, with the aft wave crest far behind the stern.  $V/\sqrt{L}$  is 1.46 in this case, well over 1.34.

Now to the relation  $V/\sqrt{L}$  - Residuary Resistance.

$R_r$  includes such items as eddy and head resistance, but is made up overwhelmingly of wave making resistance. A curve of  $R_r$  per ton displacement to the base of  $V/\sqrt{L}$  will serve to illustrate the trend:



This is again just a trend and not an accurate curve! It is of course very much dependent on the shape of the vessel, especially its prismatic coef. However, it shows clearly that we can expect  $R_r$  at 1.4 to be 400% of  $R_r$  at 1.0.

At 1.34, when the boat is literally supported on two wave crests at the ends,  $R_r / \Delta$  may easily be around 30 - 40 lbs./ton.

From 1.2 - 1.3  $R_r / \Delta$  may easily increase by 50%.

Most small fishing boats operate in regions where resistance is most sensitive to  $V/\sqrt{L}$

Between 1.34 and 1.6,  $R_r$  may easily double again! This trend is most marked; for a heavy displacement hull the increase may be still more, while conditions would be somewhat better for a light hull.

This will lead to the immediate conclusion:

It would be advisable to spread a given displacement over as much length as possible to operate at lower  $V/\sqrt{L}$  values for given speeds.

Conversely, it is seen that a longer ship may be able to carry a much higher displacement at equal speed with equal power.



A 400 ft. ship would have about the same resistance at 30 knots as a 700 ft. one. If they were geometrically similar, the latter would have a displacement about 5.35 times bigger than the former.

A 30 footer working at  $V/\sqrt{L}$  1.0 or 5.5 knots might have a  $R/\Delta$  of 10 lbs./ton. With 6 tons displacement this would mean 60 lbs.  $R_t$ .

A 47 footer, at the same speed, but  $V/\sqrt{L}$  of 0.8 could carry a displacement of 20 tons to achieve the same  $R_t$  of 60 lbs., with  $R_t/\Delta$  of about 3 lbs./ton.

Or consider the following table:

| Displ.  | L  | B.H.P. for<br>10 knots | $V/\sqrt{L}$ | V for 200 B.H.P. | $V/\sqrt{L}$ |
|---------|----|------------------------|--------------|------------------|--------------|
| 20 tons | 40 | 185                    | 1.58         | 10.3             | 1.63         |
| 20      | 45 | 140                    | 1.49         | 11.0             | 1.635        |
| 20      | 50 | 120                    | 1.415        | 11.3             | 1.59         |

These are figures from actual performance trials and the B.H.P. include total resistance in each case; the  $V/\sqrt{L}$  operated at are however so high that the bulk of the resistance will be  $R_t$ .

The table shows that 25% increase in length can save 35% power for the same speed; or, 25% increase in length allows 10% more speed for the same power!

Furthermore, the 50 footer could carry 27 tons at 10.3 knots with the same power the 40 footer needs to carry 20 tons!

Unfortunately it is not always possible to simply increase length, not least because of the heavy penalty in price that has to be paid for length. However, the decision regarding the length to adopt for a particular boat is a delicate one, in view of the obvious advantages of great length in power and fuel economy, greater payload and easier motions in a seaway.

From trials with Fishtech I and II the following table is deduced:

| L      | Displ. | $V/\sqrt{L}$ | V knots | D.H.P. | $R_t/\Delta$ lbs./ton | $R_t$ | $R_t$ |
|--------|--------|--------------|---------|--------|-----------------------|-------|-------|
| 30 ft. | 4.6    | 1.24         | 6.5     | 10.7   | 58                    | 267   | 187   |
|        |        | 1.35         | 7.1     | 23     | 115                   | 530   | 370   |
| 32 ft. | 8.5    | 1.29         | 6.9     | 10     | 28                    | 238   | 167   |
|        |        | 1.35         | 7.2     | 36     | 96                    | 816   | 570   |

The last column is based on the assumption that at these speeds  $R_t$  will be about 70% of the total resistance.

Disregarding the difference in  $V/\sqrt{L}$  for the two slower speeds, it is seen that the 2 ft. additional length allow 85% more displacement to be carried at the same speed with even 11% less total resistance, or about 50% of the specific resistance.

For the higher speeds, the heavier boat begins to pay a penalty, but it is not very big as yet: 85% more displacement is still carried at lower specific resistance, although at 53% more total resistance.

As these boats are operating in regions of elevated wave resistance importance, not all the gains in the 32 ft. boat could be attributed to the 2 ft. more length. It must be concluded that the 32 ft. boat has superior lines for lower wave making resistance.



In view of these facts, an investigation of the relation between  $R_r$  and displacement becomes necessary.

The influence of waterline length has been shown above; it was stated that the resulting wave height depended very much on the bulk of the ship, or the distribution of it's displacement along the length. An expression of these factors are the longitudinal prismatic coefficient and the length / displacement ratio.

At  $V/\sqrt{L}$  values over unity, residuary resistance per ton displacement increases quite a bit with decreasing  $L/\nabla^{1/3}$  values. Some figures from Taylor's standard series may prove this, although they refer to cargo ships rather than fishing vessels:

| L  | V    | $\nabla_m^3$ | $L/\nabla^{1/3}$ | $R_r/\nabla$ | $R_r$ |                      |
|----|------|--------------|------------------|--------------|-------|----------------------|
| 60 | 9.25 | 21.6         | 6.6              | 18.5         | 400   |                      |
| 60 | 9.25 | 32.4         | 5.75             | 21.4         | 692   | $V/\sqrt{L} = 1.195$ |
| 60 | 9.25 | 43.6         | 5.2              | 23.2         | 1000  |                      |
| 60 | 9.25 | 64.8         | 4.56             | 25.2         | 1630  |                      |

The residuary resistance for the heaviest hull is almost four times that of the lightest with only about three times the displacement. The specific  $R_r$  has increased by about 36% !

It is evident that it is impossible to build fishing boats of conventional type having a  $L/\nabla^{1/3}$  ratio as high as 6 or more. Reasonable ratios for the light condition will be around 4.5 - 4.7 . A small variation of  $L/\nabla^{1/3}$  ratios in this region would not have too great an influence on  $R_r$ , in the above example the specific  $R_r$  grows by 8.5% for a decrease of  $L/\nabla^{1/3}$  from 5.2 - 4.56.

However, it will pay in terms of reduced  $R_r$  to build fishing boats to highest possible  $L/\nabla^{1/3}$  ratios compatible with strength and safety. This again calls for lighter scantlings and more advanced methods of construction.

#### Residual Resistance and the Prismatic Coefficient

For a given displacement, length and operational  $V/\sqrt{L}$ , the main variable with an influence on  $R_r$  is the prismatic coefficient. Optimum prismatic $_r$  should be used whenever possible, provided it will still give fair and even lines! The latter requirement is important; criticism has been forthcoming against the use of lower  $C_p$  values, advancing as examples boats with extremely uneven lines, who were found to be behaving very badly in a seaway.

These relations can only be studied with the help of model tests, and a wealth of material is available for large ships, while only scanty material can as yet be reviewed in the case of fishing vessels.

The trends obtained from big ship work are:

$$V/\sqrt{L} = 1.15 \quad - \quad \text{optimum } C_p = 0.52 - 0.55$$

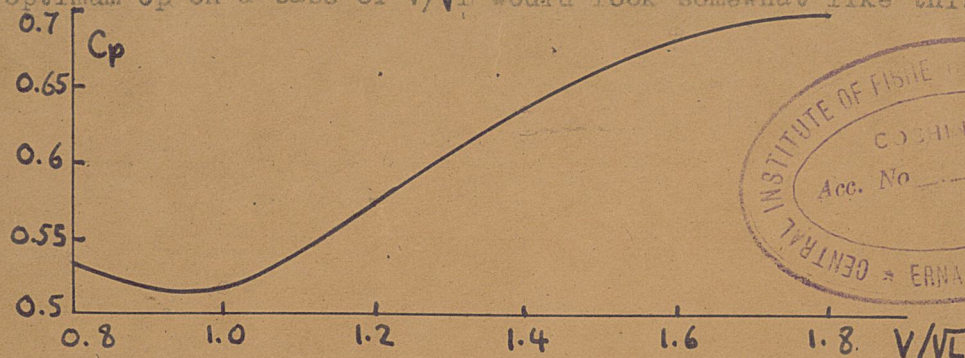
$$R_r \text{ for } C_p = 0.5 \text{ is only about } 1/4 \text{ } R_r \text{ for a } C_p \text{ of } 0.7!$$

$$V/\sqrt{L} = 1.8 - 2.0 \quad - \quad \text{optimum } C_p = 0.7$$

These trends are fully substantiated by recent model tests with a family of 90 ft. fishing boat models - the optimum  $C_p$  for smooth water was found to be in the region of 0.575 .



A curve of optimum  $C_p$  on a base of  $V/\sqrt{L}$  would look somewhat like this:



The choice of the right  $C_p$  has a very important bearing on the hull shape that can be adopted. The curve indicates that for very high speeds  $C_p$  has to be higher, which means filling the ends and very likely will mean adoption of a relatively high  $L/B$  ratio for a given displacement.

The relatively low  $C_p$  (optimum) for the fishing boat's speed range of  $V/\sqrt{L}$  from 1.0 - 1.3 can only be obtained by going to the limit of sharpening the ends and using a larger  $C_m$  (midship section area coefficient). In turn, this will mean lower  $L/B$  ratios. Because of the importance of a low angle of entrance and the longest possible entrance proper, it will usually be wise to adopt the LCB well aft of midships. This is also required to prevent excessive trims by the stern, especially at speeds exceeding 1.34  $V/\sqrt{L}$ . With stern fishing vessels of the American type, this requirement is somewhat difficult to fulfill, as the engine has to be accommodated far forward.

At higher speeds, the problem of getting a higher  $C_p$  without undue increase in angle of entrance is no less difficult to solve. A broad transom becomes a necessity, bringing LCB well aft, but allowing for a fairly sharp entrance. Some fast launches in this range have underwater shapes that look rather like an entrance half of the hull only!

The angle of entrance is of great importance at  $V/\sqrt{L}$  from 1.2 - 2.0. In this range it must be kept low, to prevent large pressures building up at the bows, resulting in a high bow wave system. The optimum angle (half waterline) appears to be about  $10^\circ$ . For low  $C_p$  values this is usually impossible to achieve without excessive hollowing out of the waterlines and lines resulting that can no longer be termed fair and even. Hollowing of waterlines near the surface should be omitted at all cost, because it adds more to resistance than a somewhat higher  $C_p$  would!

At higher speeds, it will again be difficult to achieve such low angles of entrance together with the requisite higher  $C_p$ 's. A very full stern would probably have to be adopted, with the entrance extending far aft and the transom well immersed at rest to ensure good results.

It is fortunate indeed that the tendency of fishing vessels to trim by the stern when running free at higher speeds will produce a reduction in angle of entrance. Again this signals the importance of avoiding head trims!

Very fine bows must be combined with flaring topsides to give the foreship sufficient reserve buoyancy in heavy weather. Should this be omitted, the ship will be very wet forward, and each encountered wave will be shipped with possibly dangerous results, but definitely resulting in great discomfort to the crew.

All these considerations have to be kept in mind when designing fishing vessels. It is unfortunate that it will never be possible to adopt the optimum course for each item of consideration, as one will always have to sacrifice optimum performance to cargo capacity or the to the requirements dictated by the very much changing loading conditions.







Group 2)  $V/\sqrt{L}$  1.0 - 1.34

|                                |    |      |     |     |      |      |
|--------------------------------|----|------|-----|-----|------|------|
| Length                         | 20 | 30   | 40  | 50  | 60   | 80   |
| Speed for<br>$V/\sqrt{L}$ 1.34 | 6  | 7.35 | 8.5 | 9.5 | 10.4 | 12.0 |
| Wave length<br>at 1.34         | 20 | 30   | 40  | 50  | 60   | 80   |

Number of full wave lengths in hull length: 1.8 - 1 at  $V/\sqrt{L}$  1.34

Trim of hull: From  $V/\sqrt{L}$  = 1.0 upwards, increasing squatting of stern and bodily sinkage of hull at higher speeds.

Specific resistance: About 70 lbs./ton at 1.34

Frictional resistance: 30 - 20%

Residual resistance: 70 - 80% of total

|                  |              |      |      |      |      |      |
|------------------|--------------|------|------|------|------|------|
| Ideal prismatic: | $V/\sqrt{L}$ | 1.0  | 1.1  | 1.2  | 1.3  | 1.34 |
|                  | $C_p$        | 0.52 | 0.54 | 0.58 | 0.62 | 0.63 |

Notes on hull form: The ship is still supported on two wave crests. In the higher part of the range, residuary resistance is getting very high. Increasing optimal  $C_p$  indicates the importance of filling ends for higher speeds and adopting smaller  $C_m$ . High values of displacements only acceptable in lower range while resulting in greatly increased cost at higher speeds. The change to flat aft body lines and transom stern is advisable.

Group 3)  $V/\sqrt{L}$  1.34 - 1.65

|                                |      |    |      |      |      |      |
|--------------------------------|------|----|------|------|------|------|
| Length                         | 20   | 30 | 40   | 50   | 60   | 80   |
| Speed for<br>$V/\sqrt{L}$ 1.65 | 7.4  | 9  | 10.5 | 11.7 | 12.8 | 14.7 |
| Wave length<br>at 1.65         | 30.4 | 45 | 61.8 | 76   | 91   | 120  |

Number of wave lengths in length of hull: Decreasing from 1 to 0.65

Trim of hull: Stern trim and bodily sinkage of hull at its most severe.

Specific resistance: About 150 lbs./ton at 1.65

Friction resistance: 15 - 10%

Residual resistance: 85 - 90% of total

|                  |              |      |      |      |      |      |
|------------------|--------------|------|------|------|------|------|
| Ideal prismatic: | $V/\sqrt{L}$ | 1.34 | 1.4  | 1.5  | 1.6  | 1.65 |
|                  | $C_p$        | 0.63 | 0.64 | 0.66 | 0.68 | 0.68 |

Notes on hull form: The vessel is creating a transverse wave system of greater length than its waterline! It is supported by one wave crest only with the second one abaft the stern. To reduce the height of the crest forward and the depth of the trough aft, fine angle of entrance is essential, coupled with broad, flat, immersed transom and extremely flat buttocks aft. Light displacement is essential! A very high price will have to be paid in terms of power for each extra ton.



Model tests

From the foregoing it became clear that a ship's resistance is mainly split in two groups:

- a) Frictional resistance: Largely dependant on Reynolds Number
- b) Residual resistance: Mainly a gravity problem, as it is due to pressures set up by the hull in motion.

We have to examine how these two groups affect the resistance of a model, and how this resistance can be used to deduce the resistance of the full size ship. For this purpose a short excursion into theoretical physics becomes necessary:

Laws of Dimensional Similarity and Comparison:

Resistance is a force and thus of the dimension  $MLT^{-2}$  where  $M$  = mass  
 $L$  = length  
 $T$  = time

Using these three basic dimensions,  $R_F = L^2 \times V^2 \times \rho \times \phi\left(\frac{\nu}{VL}\right)$ , where  $\phi$  is an unknown function, and  $\rho$  is the mass density.

Resistance also is  $R = L^2 \times V^2 \times \rho \times \phi\left(\frac{Lg}{V^2}\right)$  where  $g$  is the gravity constant. (This applies when  $R$  is independent of friction).

If resistance were entirely frictional, the necessary requirement for dynamical similarity would be a constant function  $\phi$ . Hence for dynamical similarity the term  $\phi\left(\frac{\nu}{VL}\right)$  must be constant. As  $\frac{VL}{\nu}$  was defined as Reynolds number, this means that model and ship frictional resistance can only be compared for constant Reynolds number, or meaning that for the same fluid (constant viscosity)

$$\frac{V_m}{V_s} = \frac{L_s}{L_m} \quad \text{or, if } \lambda \text{ is the linear ratio of length}$$

$$V_m = \lambda V_s$$

This  $V_m$  is known as the corresponding speed for constant viscosity.

If resistance were totally unaffected by friction (i.e. viscosity), similarity would be obtained for a constant function

$$\phi\left(\frac{Lg}{V^2}\right)$$

The relation then becomes  $\frac{V_s}{\sqrt{L_s g}} = \frac{V_m}{\sqrt{L_m g}}$  or, because  $g$  is constant for all practical purposes  $\frac{V_s}{\sqrt{L_s}} = \frac{V_m}{\sqrt{L_m}}$  which is the relative speed we have met before.

$$\frac{V}{\sqrt{gL}}$$

is known as Froude's number, and is fully non-dimensional.

For a comparison of model and ship resistance the following relations must then be met:

$$\frac{V_s}{\sqrt{L_s}} = \frac{V_m}{\sqrt{L_m}} \quad \text{or } V_m = V_s \frac{\sqrt{L_m}}{\sqrt{L_s}}; \quad \text{if } \frac{L_s}{L_m} = \lambda \rightarrow V_m = V_s \times \frac{1}{\sqrt{\lambda}}$$

These speeds are known as corresponding speeds for wave making.



These corresponding speeds are clearly different from those for viscous flow.

It has been seen earlier that wave making resistance varies rapidly with changing  $V/\sqrt{L}$ , whereas friction changes less with Reynolds number. Thus it becomes evident that speed / length ratio is the more important criterion for assessing a ship's resistance.

It has also been shown that total resistance combines both friction and wave making. True dynamic similarity would then only be achieved if the function

$$\phi \left\{ \left( \frac{V}{L} \right) * \left( \frac{Lg}{V} \right) \right\} \text{ were constant.}$$

This is physically impossible!

Consider a 20ft. model of a 400ft. ship at 16 knots;  $\lambda = 20$ ,  $V/\sqrt{L} = 0.8$

For constant Reynolds number the model speed would have to be a fantastic 320 knots, while for constant  $V/\sqrt{L}$  it becomes a reasonable 2.58 knot.

From this it follows implicitly that experimental model testing can only be reasonably possible if frictional and wave making resistance are independent of each other and if the resistance equation then can be written:

$$R = L^2 V^2 \rho \phi_1 \left( \frac{V}{L} \right) + L^2 V^2 \rho \phi_2 \left( \frac{Lg}{V^2} \right)$$

William Froude's experiments provided the evidence that such an equation is in principle acceptable, and with this proof Froude became truly the father of tank testing technique.

He showed that it is permissible to deduct the friction resistance of a vessel from the total, the remainder then closely following his laws of comparison. He also showed how to calculate the friction part of the total.

Froude's assumptions are of course not universally applicable; in various instances troublesome scale effects have to be considered and applied. This is however beyond the scope of this text and more a matter for consideration by scientific personnel of testing establishments.

### Testing Tanks

The original Froude tank at Torquay in England was 270 ft. long, 36 ft. wide and 10 ft. deep.

Gradually, with newer tanks, length increased because higher speeds became necessary and also because it was found that larger models gave more reliable results. Froude's models never exceeded 12 ft. in length.

This development culminated so far in the Carderock Basin in America, which has a high speed channel of 2963 ft. length and 91 ft. width, designed for speeds up to 60 knots, and a deep water basin of 3775 ft. length, 51 ft. wide and 22 ft. deep. Models of up to 32 ft. length can be tested there (FISAPRO II). The new tank at Falmouth in England can take models up to a weight of 5 tons.

Later it was found that smaller basins and models could be used with good results if a few safeguards were taken: For example, laminar flow must be avoided and turbulence stimulators have to be introduced. Newer friction formulations are used, based on Reynolds number instead of length only as with Froude.

Large tanks are always equipped with a carriage running on rails at the basin side. The models are suspended from the carriage, and it also houses all the instruments that are necessary to measure resistance. The carriage is normally self propelled by electric motors, somewhat like a railway locomotive.



Small tanks normally propel the models by pulling them on a wire accelerated by falling weights.

The main difference between the two methods is that the carriage operates at a set speed, measuring the resistance, while the drop weight system operates with a known resistance (the weights) and measures the speed.

The model size is determined by a number of considerations:

- 1) The model must run at a speed which allows the formation of trochoidal waves in the medium.
- 2) It must be small enough to allow reflected waves from the tank wall to pass astern of it.
- 3) There must be sufficient depth of water under the model to exclude increase in resistance due to shallow water effect - unless this wants to be investigated!
- 4) The volume of the model must be within certain limits. These are usually expressed as the permissible ratio between the maximum section area of the model and the cross sectional area of the tank.

#### How are Resistance results obtained

Models are run at correct displacement volumes, bringing the relation between ship and model displacement to  $\lambda^3$ .

As most tests are run in fresh water, a correction by the density ratio of sea / fresh water must be applied.

The displacement ratio then is about  $\frac{\lambda^3}{0.975}$ .

Roughly the routine of deducing resistance results then is thus:

At a given speed of the carriage the dynamometer registers the total resistance of the model  $R_{mt}$ . From this the calculated friction resistance of the model is deducted, giving the residuary model resistance:

$$R_{mr} = R_{mt} - R_{mf}$$

This is scaled up for the full size ship,  $R_{sr} = R_{mr} \times \lambda^3$  (taking into account density difference).

The friction resistance for the ship is again calculated and added -

giving  $R_{st} = R_{mr} \times \lambda^3 + R_{sf}$ . If R is in lb., and V in knots,

$$EHP = R_{st} \times V \times 0.00307$$

#### The use of model test data

Data received from a testing tank is usually grouped thus:

- a) Ship's and model's data: Dimensions, volumes, coefficients
- b) Ship's total resistance over a range of speed: Expressed in tons, lb., kgs or EHP
- c) A set of curves giving resistance results on a base of Froude number or  $V/\sqrt{L}$  or actual speed, either in  $C_r$  notation (Admiralty coefficient) or in any other accepted form such as circular notation (mainly in Britain).



The data sheets issued by FAO in volume 1 and 2 of Fishing Boat Tank Tests give the following information:

Top block: Test and model information as well as running FAO number.

Lines block: Profile and body plan with scale indicators; tested waterlines drawn in profile and plan; section area curves for tested waterlines in plan; LCB positions.

Data block: Main dimensions and coefficients for ship size for all test runs represented; dimensions and coefficients worked up for standard lengths of 20 meters and 100 ft.

Graph block: Curves of Admiralty Coefficients ( $C_A$ ) on a base of Froude Number; EHP curves on base of Froude Number; Scale of Speed / Length ratio; scale of actual speed for actual length (as indicated) and scales for actual speeds for standard lengths.

The use of these data sheets is explained briefly as under:

- a) Direct deduction: Only possible for the actually tested vessel or a completely equal ship.
  - b) Direct comparison: Only possible for ships of closely similar dimensions, shape and form parameters.
  - c) Interpolation: Comparison on the bases of selected parameters for ships of a certain group, leading to data for a projected or investigated vessel, following within the boundaries set for the group under investigation.
- 
- a) Direct Deduction: This is useful for computing resistance and EHP values of the actually tested vessel at different speeds and at slightly different loading values than those tested. It would not be applicable to different ships, unless exactly equal to the tested vessel.

Assuming a 40 ft. in the waterline vessel, at 1228 ft<sup>3</sup> ( 34.7 m<sup>3</sup>) displacement, running at Froude Number 0.30 and showing there a  $C_A$  of 330.

The 0.3 Froude number corresponds to a ship's speed of 6.38 knots (see graph).

Calculating EHP from  $C_A = 330$  it is found to be 8.36 h.p.

For slight changes in displacement, and consequently waterline length, the new EHP may be directly deduced such as:

$L = 40.5$  ft.  $\nabla = 1300$  ft<sup>3</sup> or 36.8 m<sup>3</sup>; the speed at Froude No. 0.30

now corresponds to 6.43 knots due to the slightly longer length, and the  $C_A$  at this speed can be assumed as 330 still. Working out EHP for these new values then yields the figure of 8.9 h.p. Conversely, the speed could be assumed as being equal, giving a slightly lower Froude Number for the second calculation, this in turn most likely resulting in a somewhat higher  $C_A$ . To obtain the EHP, this new  $C_A$  would have to be used in the calculation.



b) Direct Comparison: For a number of tested vessels, the available  $C_1$  curves can be simply superimposed on the bases of Froude Number, and it can then be easily deduced which of the models was the best.

If possible, such direct comparison should be made on the bases of equal length, and only for vessels with equal, or nearly so, displacement at equal length (for rough first hand reference a disparity in displacement of perhaps  $\pm 10\%$  may be accepted, as it has been seen that  $L/\nabla^{1/3}$  ratio has comparatively little influence on resistance).

By using this direct comparison cleverly, and studying the influence of variable design parameters on the curves under observation, it becomes possible to form an opinion as to the optimum value for these parameters for vessels of the type as those under comparison.

When this is done, it becomes soon evident that a sharp half angle of entrance will result in lower resistance; equally will a transom stern and LCB well aft of midships normally mean reduced EHP. It would also become obvious that beam could be increased considerably without adverse effect on resistance, thus making it possible to reduce draft for a given displacement, and to work at higher B/T ratios.

Interesting conclusions can be drawn from observing the trend of resistance curves with changing trim: Almost invariably will a trim by the head result in higher resistance than a trim by the stern for the same displacement. This is undoubtedly due to some extent to the decreased angle of entrance and the LCB being moved aft.

Similarly the influence of various parameters such as prismatic coefficient can easily be studied in this way.

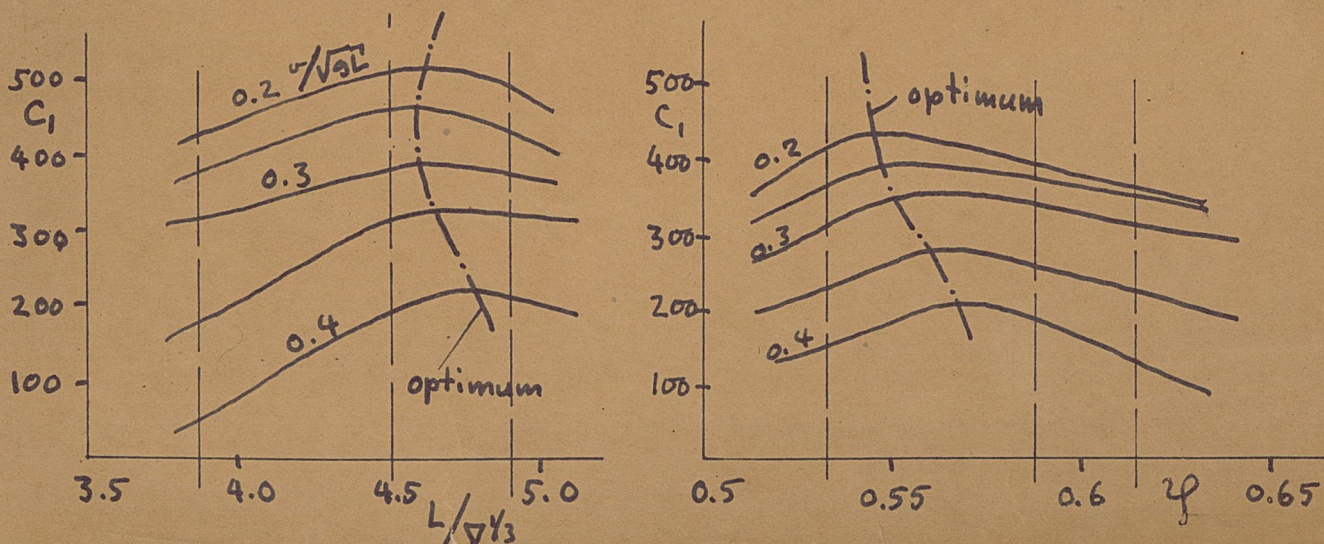
c) Interpolation: If a number of vessels are compared with a view to obtaining a set of optimum dimensions and coefficients for a new design, a rather more elaborate method becomes necessary.

It would first be important to recalculate the  $C_1$  values for equal length, to be able to draw a set of cross curves of  $C_1$  on a bases of  $L/\nabla^{1/3}$  ratios. These cross curves would be drawn for variable Froude Number, and they would allow to find the optimum  $L/\nabla^{1/3}$  ratio, for the given length and required speed.

Similar sets of cross curves could be drawn for varying LCB positions and value of prismatic, giving optimum values for these parameters as well.

In this way a whole set of dimensions could be obtained at; but it is not certain that they would be practical dimensions, and many a compromise would still have to be investigated to be sure of their workability for the new design.

Traug gives examples for such comparisons in his paper in Fishing Boats of the World I, pp. 312 - 319. Examples of cross curves appear below:





## How to use tank test data and when, and how, to order new tests

It is hoped that C.F.T.S. will be able to publish all data on tests with models of small boats conducted to date in India by about the end of 1960. The data available by then will cover boats from 25 ft. to 40 ft. in length; this size group will undoubtedly remain the most important group in India for many years to come. Thus the data published will be of great value to eventual designers of new types of boats in this class.

There will be sufficient tests available by the later part of this year, to allow for a conclusive study of resistance properties of small craft and their EHP requirements, to enable the designer to predict with a great degree of accuracy the performance of any new design. Quite contrary to the problems posed by big shipbuilding, the designer of small craft is normally not concerned too much with obtaining information as to the last % of EHP necessary for his boat to attain a certain speed. He can not normally dispose of an almost customs built engine plant for his ship, but has to rely on standard production engines off the factories of the engine making industries. Thus it will normally be rather a question of what speed one can guarantee for a new boat if a certain type of engine is installed. The test results will also give much material to interpret certain novel design features with reasonable accuracy, and they should prove to be very helpful in doing away with old, traditional ideas, in the meantime shown to be misleading and contrary to best possible performance.

If an opportunity arises to run a set of tests for a new design, the following procedure may be adopted:

A new 38 ft. boat is to be tested, for example: The designer would first set himself a reasonable goal for the performance of the boat, by interpolating test results and resistance data available for vessels in the size group 36 ft. to 40 ft. Initial tests with the new 38 ft. boat would then be run at possibly two displacement (light and  $1/2$  or  $3/4$  load) and three trims each. The results of these tests, when compared to the deductions made from earlier tests, will allow certain conclusions regarding the performance of the new vessel; the only straight answer that can be obtained being: Does it fulfill the designers expectations or does it not? If the performance is greatly unsatisfactory, further tests will have to be run with amended models to see whether the amendments contemplated are giving better performance and by what degree.

It must constantly be remembered that single tests with one model do give an accurate idea of the full sized vessel's performance at exact test conditions, and very little more! The true value of model testing will only be apparent if whole series or families of models are tested, where each new model is an amendment of another one in a strictly planned direction. In this way the influences of different form parameters can be studied very nicely indeed.

It is suggested here that the Boat Design Section of C.F.T.S. should be consulted whenever any model testing programs are being contemplated by any Department. The section is well equipped to suggest proper testing programs for any new design under study, to avoid duplication of effort and to ensure that all new tests greatly enhance the wealth of knowledge already available with regard to the performance of small boats.

**WARNING:** It must continuously borne in mind by all concerned that the mere testing of a model does not mean that the design under scrutiny is a good one. Only the proper application of tests results and their logical use when working on new designs can ensure better boats in the long run.



## Propulsion

A propeller is a device converting engine power in the form of shaft torque into thrust and ultimately into advance motion of the ship.

This brief and simple definition makes it clear that Torque, Shaft Horse Power, Thrust and Speed will be important factors when deciding on the properties of a marine propeller.

Fundamental propeller selection data are:

- a) Shaft Horse Power
- b) Revolutions per minute, at which power has to be absorbed by propeller
- c) Speed of advance of the propeller in relation to ship's speed.

a) and b) are obtained from powering calculations, or, in the case of utilising a stock engine, they are usually extracted from the information given by the manufacturers of the engine.

c) is somewhat more involved and depends largely on ship-form factors.

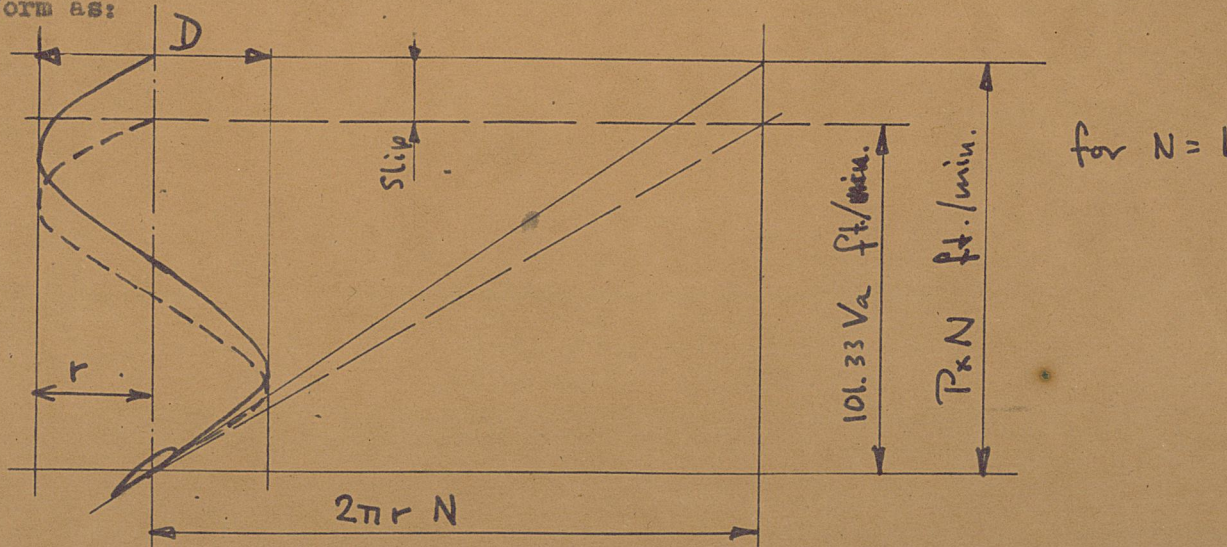
## Basic screw propeller theory

It is evident that the thrust developed by the propeller must be equal to the ship's resistance at a certain speed. Hence a good propeller is one that develops maximum thrust at highest possible efficiency, where efficiency can be defined as:

$$\eta = \frac{\text{Thrust developed by prop.}}{\text{Power supplied to prop.}} \times \text{speed of advance of prop.}$$

i.e. for high efficiency: Small power supplied, large thrust developed at high speed of advance.

The action of a propeller blade, when rotating, may be shown in simplified form as:



where  $N = \text{r.p.m.}$ ;  $V = \text{speed of ship in knots}$ ;  $V_a = \text{speed of advance of propeller through the wakestream.}$

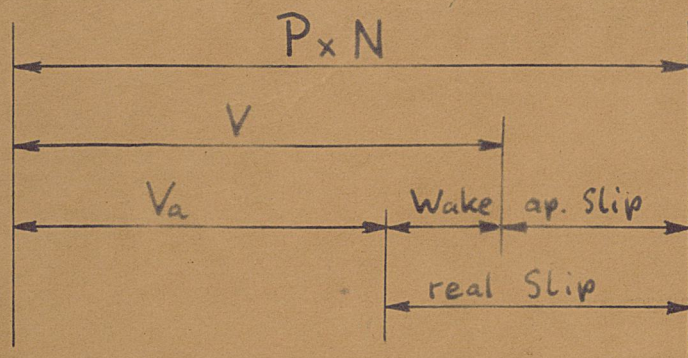
$P = \text{pitch}$ ;  $D = \text{diameter}$ ;  $P/D = \text{pitch ratio}$

$P \times N$  would indicate the advance of the screw of pitch  $P$  in a solid body per minute, while  $V_a$  will mean the actual advance of the blade in a fluid.

$V_a = P \times N - \text{Slip.}$  (The factor 101.33 comes in to convert knots into ft./min.)



It becomes necessary at this stage to introduce the phenomenon of "wake":  
This is best done graphically at first -



$$V_a = PN - \text{Slip or,}$$

$$V_a = PN - (\text{wake} + \text{ap. slip})$$

When slip is mentioned, it is always real slip, i.e. wake plus apparent slip. Apparent slip is always termed correctly to avoid confusion.

To better explain the terms slip and wake, an important assumption has to be made:

It is assumed that a propeller creates a "slipstream" of the cross sectional area of its disc, i.e.  $\frac{\pi D^2}{4}$  with a linear speed equal to the linear speed of the propeller blades in the direction of motion of the ship.

This is a working assumption and not a scientific fact! However, it allows the phenomenon of slip to be explained and understood as a simple geometric property, although, in actual fact, it is a very complicated thing.

Apparent slip then becomes  $\frac{PN - 101.33 V}{PN - 101.33 V}$  or expressed as a percentage of PN

$$\text{app. slip} = \frac{PN - 101.33 V}{PN} \times 100\%$$

**Wake:** The wakespeed is the difference between the ship speed  $V$  and the intake velocity of the water into the propeller disc in the absence of a propeller.

It is largely determined by length and fullness of the hull.

The main component of the wake is the frictional wake, i.e. the water that follows the ship aft due to an advance motion imparted to it by the passing ship through skin friction.

A large merchant ship with considerable skin friction might have water passing the propeller at only about 65% of the ship's speed, while the two speeds could be almost equal for a fast planing hull.

Between these extremes, allowances have to be made for the influence of wake in reducing the speed of advance of the propeller,  $V_a$ .

Wake speed itself can be expressed as a percentage of either  $V$  (ship speed) or  $V_a$  (speed of advance). Froude did it in the second way, while Admiral Taylor choose the first, which is normally now used as it is found more practical in view of the fact that  $V$  is usually known whereas  $V_a$  is an unknown quantity.

The Taylor wake fraction is expressed thus:

$$w_t = \frac{V - V_a}{V} \quad \text{giving}$$

$$V_a = V (1 - w_t) \quad \text{and} \quad V = \frac{V_a}{(1 - w_t)}$$



In the Froude notation:  $w_f = \frac{V - V_a}{V_a}$  resulting in

$$V_a = \frac{V}{(1 + w_f)} \quad \text{and} \quad V = V_a (1 + w_f)$$

It may also be noted that  $w_f = \frac{w_t}{(1 - w_t)}$  and  $w_t = \frac{w_f}{(1 + w_f)}$

If either wake fraction is known or used, the other can easily be established.

Sometimes wake is designated by the fraction  $\frac{V_a}{V} = (1 - w_t)$ , and Phillips-

Birt gives a Nomogram on this bases which might be useful (see appended curve sheets). The Nomogram has been extrapolated for the somewhat lower speeds normally found with fishing vessels. For a 20 ft. boat at 18 knots, the nomogram yields  $V_a/V = 0.9 = (1 - w_t)$ . This means a  $w_t$  of 10%. Similarly a 30 ft. boat at 8 knots is indicated by  $V_a/V = 0.835$ , giving a  $w_t$  of 16.5%. The nomogram seems useful for medium sized, medium speed craft.

Another way of obtaining good values for wake fraction will be treated below.

The only way to obtain a correct value for the wake fraction of a hull is by running careful self propulsion tests in a tank!

Real slip can now be appraised as  $\frac{PN - 101.33 V_a}{PN}$  or

$$\frac{PN - 101.33 \{V(1 - w_t)\}}{PN}$$

### Propeller Notations

Propeller data is usually presented in two forms:

For design purposes: In terms of  $\delta$  and  $B_p$  values

Propeller model data: In terms of  $J$ ,  $K_t$  and  $K_Q$  values.

The latter are:  $J = \text{Speed coefficient} = \frac{V_a}{nD}$  where  $V_a = \text{speed of advance in unity system (ft./sec.; m./sec.)}$  and  $n = \text{revs. per second}$

$K_t = \text{Thrust coefficient} = \frac{T}{\rho n^2 D^4}$  where  $T = \text{thrust}$   
 $D = \text{diameter}$   
 $\rho = \text{mass density}$

$K_Q = \text{Torque coefficient} = \frac{Q}{\rho n^2 D^5}$  where  $Q = \text{Torque}$

Efficiency then becomes  $\frac{\text{work done}}{\text{work supplied}}$  or  $\eta = \frac{T V_a}{2\pi n Q}$  or  $\frac{K_t J}{2\pi K_Q}$

This is applicable in all unit systems!



Design or selection data:

$B_p$ , the power absorption or loading constant was introduced by Taylor

$$B_p = \frac{\sqrt{SHP} \times N}{V_a^{2.5}}$$

$\delta$ , the diameter constant =  $\frac{ND}{V_a}$ . Observe that  $\delta$  is the inverse of J

and  $J = \frac{101.33}{\delta}$

When selecting a screw, one normally knows the power to be absorbed, the propeller rpm and the speed desired; further details, such as blade thickness, blade area ratio, and sometimes even number of blades are generally not known at the outset. To help the boat designer in his task of selecting the right screw, Taylor has presented a large number of propeller series in the above data; the more well known Troost propellers are also presented in the Taylor notation (see graph sheet appended).

All these curves are drawn on a base of  $B_p$  values, to an ordinate of pitch ratios and for variable  $\delta$  values and  $P$  efficiencies.

Tables of the Troost series are available for 2 - 5 bladed screws and disc area ratios (DAR) of 0.3 - 0.7.

Disc area ratio is simply 
$$\frac{\text{developed blade area}}{\text{disc area or}} = \frac{DA}{\frac{\pi D^2}{4}}$$

The most useful Caterpillar propeller calculator is based on the Troost 3 bladed screw of 0.5 DAR, the B 3-50 propeller series.

Barnaby gives a table of useful propeller data in his book, the table being appended to these notes.

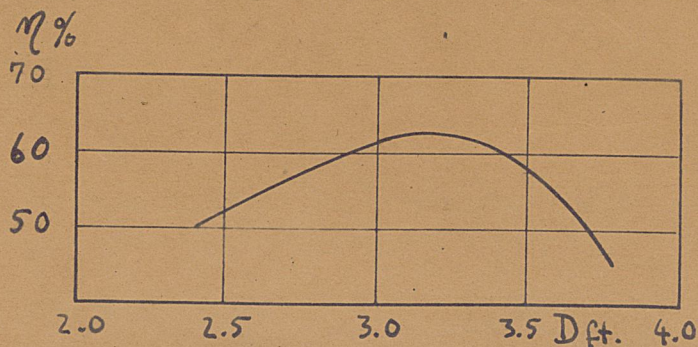
Full propeller data on the Troost series is available in van Lammeren's book: Resistance and Propulsion, and in a number of other works dealing with the theories of resistance and propulsion of ships.

The requirements for a good fishing boat screw are now examined a bit more in detail.

From the earlier assumption of slipstream (being a column of water of the propeller disc area cross section) it becomes evident that the bigger the propeller diameter, the bigger the volume of water in the slipstream for a given r.p.m.

We can therefore tentatively say that as large a propeller as possible should be fitted. There is, of course, a limit propeller diameter for optimum efficiency, but it is generally true to say that a large diameter would increase efficiency, definitely so if accompanied by reduced r.p.m.

In practice the optimum diameter can seldom be utilised due to restrictions in propeller aperture; the above indicates one of the main reasons why trawlers should be deep aft - to accommodate the optimum propeller diameter! The variation of propeller efficiency with diameter may be seen from the curve:

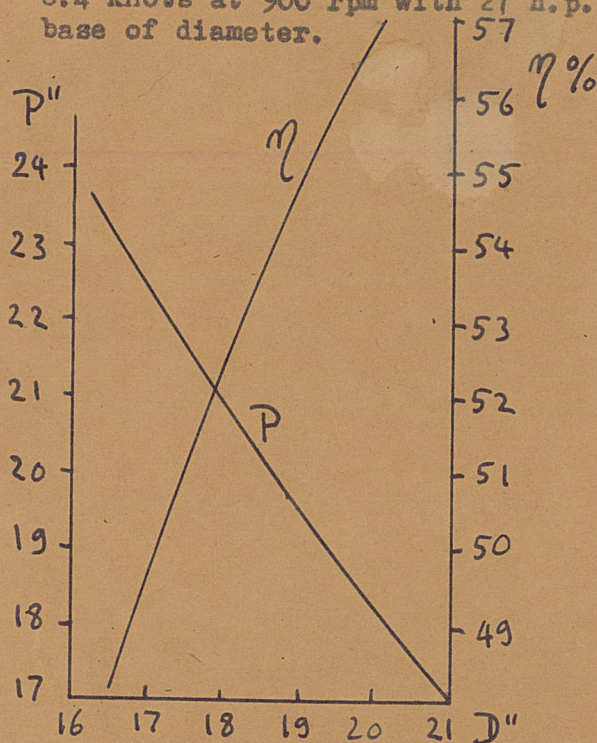


BHP = 83  
rpm = 500  
 $V_a = 9.0$



For a speed of advance of 9 knots, with 83 h.p. and the prop. turning at 500 rpm, optimum efficiency would be obtained with a diameter of 3.175 ft. Reduction in D would mean lower efficiency, just as well as bigger D would. If it is realised that this graph could apply to a boat in the 38 - 40 ft. range, it is readily seen that it would be difficult to fit a propeller of even optimum diameter (38"). In practice it will be found that propellers are usually not big enough for optimum efficiency.

The pitch of the screw is also governing the motion of water in the slipstream, and it is obvious that the thrust developed is largely governed by the amount of pitch for given diameter and rpm. The volume of water moved may be kept constant by giving a small diameter to a high pitch screw, while a large diameter screw would get a low pitch. This tendency is shown in a graph, showing a propeller designed to give 8.4 knots at 900 rpm with 27 h.p. Both pitch and efficiency are drawn on the base of diameter.



A propeller of  $D = 20"$  and  $P = 18"$  would produce the thrust and absorb the power at an efficiency of about 57%.

With  $D = 18"$  and  $P = 21"$ , the efficiency would be only 52%.

Reduced diameter and increased pitch tend to lower the efficiency appreciably.

Turning to the slip equation, it will be seen that a propeller with a high pitch will work at a higher slip percentage than one with low pitch:

$$\text{Slip} = \frac{PN - 101.33 V_a}{PN}$$

At  $V_a$  of 10 knots, a prop. with 24" pitch at 700 rpm will have a slip of 27.5%. If a larger diameter could be used, allowing a lower pitch of 21", the slip for the same speed of advance and rpm would be only 17.5%.

Thus it is seen that for a given diameter thrust increases with pitch and therefore with slip.

If a bollard test is made, i.e. with  $V_a$  of 0 knots, the propeller will develop its highest thrust at 100% slip, while if the prop were working in a block of wood like an ordinary wood screw, slip would be 0, but so would thrust.

This looks exceedingly fine - but unfortunately it is not so at all!!

A screw running at very high slip ratios is producing its thrust at very low efficiency, i.e. one needs to put in a lot more power than one gets out. A trawler operating at low  $V_a$  when trawling, hence relatively high slip, needs practically full engine power to produce the thrust needed, i.e. the propeller works at low efficiency.



What is really required of a good propeller is high thrust at high efficiency, which means high pitch at low slip! It is possible to get exceptionally high efficiencies with large P/D ratios, but only with very low slip.

For vessels of moderate speed this would mean to have a large D at relatively large P/D and very low rpm. Normally it will not be possible to get the extremely low rpm required, especially in diesel craft, as the reduction gearing would have to be of the order of 8:1 or more. Such gears are not only not available in the market, they would also be extremely bulky and heavy if they were.

For prop. speeds of 2000 rpm about, an efficiency of up to 50% may be expected; with 2:1 reduction gearing this could be expected to go up to 60% !

The advantage of fitting low reduction gearing to workboats (except high speed craft) is apparent; caution must however be exercised to see that the optimum propeller diameter can then be installed. Reduction of optimum diameter results in larger decrease of efficiency than using somewhat higher rpm and optimum diameter.

The tendency is shown by an example:

| Speed | Gearing | D ft. | $\eta$ % | engine rpm constant at 3150 |
|-------|---------|-------|----------|-----------------------------|
| 10    | 5:1     | 2.5   | 70       |                             |
| 10    | 3:1     | 2.0   | 65       |                             |
| 10    | 2:1     | 1.6   | 58       |                             |
| 10    | direct  | 1.1   | 45       |                             |

Fortunately for the designer of very high speed craft (semi planing and planing) this vexing thing gets reversed!

It would be physically impossible to fit large, slow turning screws to very light hulls of usually shallow draft. The large power transmitted would necessitate tremendous shafts and bearings, and it is quite possible that the torque produced by the large screw would turn the boat around instead of the propeller.

Because of the very high speeds at which such boats operate, it is possible to fit small diameter screws running at very high speeds, without slip getting too large even as very high pitches are used. This becomes understandable when looking at the slip equation!

One of the world's fastest boats with 1,200 h.p. engine running at 3,200 rpm uses actually a multiplication gear of 3:1 and the propeller of 13.5" diameter and 25" pitch turns at something like 9,000 rpm!

|              | Boat A | B      | Blue Bird I    |
|--------------|--------|--------|----------------|
| Engine power | 100    | 100    | 1950           |
| rpm          | 3,000  | 3,000  | 3,150          |
| Gear         | 2:1    | direct | 1:2.85 (mult.) |
| D            | 1.7    | 1.4    | 1.17           |
| P            | 1.42   | 2.2    | 1.71           |
| P/D          | 0.835  | 1.57   | 1.46           |
| V            | 16.2   | 52     | 141.7          |
| Slip %       | 22     | 19     | 7 !            |
| $\eta$ %     | 65     | 70     | 72             |

For A and B is seen that the increase in speed for a given power lowers the slip, even considering the greatly decreased D and increased P and higher prop. rpm. The matter is even more striking for Blue Bird, operating at the very low slip value of 7%.

It must be borne in mind that all this only works at the top speeds indicated; at lower speeds, both B and Blue Bird would of course operate at shockingly low efficiencies.



The requirements for a good propeller for a work boat are thus:

- 1) Big Diameter
- 2) Low rpm
- 3) Relatively large Pitch
- 4) Low Slip

High pitch ratios of up to 2.0 are only practical with extremely low rpm screws or for very high speed vessels.

Practical fishing boat P/D ratios would be between 0.6 and 0.8, the former for optimum or near optimum diameter and the latter for smaller diameters in conjunction with lower reduction gearing.

Note: Optimum propeller diameter should always be used where aperture size allows it; if aperture size is too small, it will usually be advisable to have the propeller run somewhat faster and use optimum size for the new speed, rather than just cut down on diameter and increase pitch a little.

Thus if a 3:1 gear box would turn the propeller at 400 rpm and calculations show an optimum size of 38" with P/D = 0.6, aperture, however, can only take a maximum of 35" diameter, it would be best to go in for 2.5:1 reduction if available, or even 2:1, giving perhaps a propeller of 34" diameter at 0.6 P/D, but turning at 600 rpm. This would entail less drop in efficiency than simply using a 35" prop at 400 rpm with a correspondingly lower pitch!

#### Propeller area and thrust loading

For heavily loaded screws it is of importance to determine the thrust load in order to be certain that the chosen blade number and blade area is sufficient to sustain the thrust developed.

The thrust developed by the propeller, measured at the thrust bearing, is

$$T = \frac{\text{SHP} \times 33,000 \times \eta}{V_a \times 101.33} \text{ lbs.} \quad \text{or} \quad = \frac{325.67 \times \text{SHP} \times \eta}{V_a} \text{ lbs.}$$

SHP (shaft horse power) is often not known from engine information, and at times it has to be determined by making overall allowances for gearing losses on BHP. Some engine makers give the continuous output for their engines, measured at the gearbox output flange. This value can be used, but it might be useful to make an allowance of about 5% on it for bearing and shafting losses. Also, a further allowance of perhaps 15% might be made to account for increased resistance in heavy weather for sustaining the selected speed.

The intensity of thrust loading on the blades is then given by:

$$T \text{ lbs./sq.in.} = \frac{325.67 \times \text{SHP (corrected)} \times \eta}{V_a \times \text{developed blade area in sq.in.}} \quad \text{or}$$

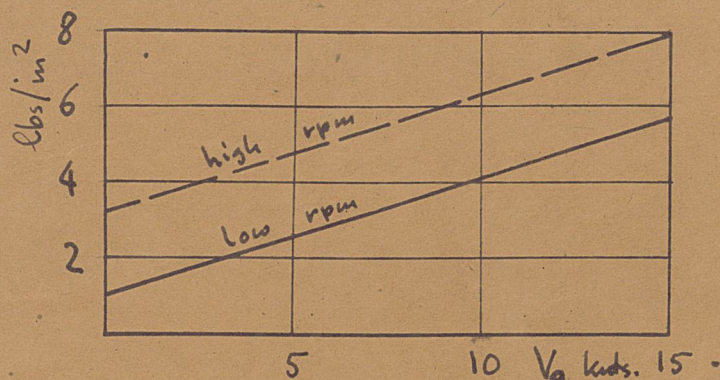
$$\frac{325.67 \times \text{SHP} \times \eta}{V_a \times \text{DAR} \times \text{Disc area}} \quad \text{where DAR} = \frac{\text{dev. area}}{\frac{\pi D^2}{4}}$$

This is a most important criterion, as it has to do with the phenomenon of cavitation. If the loading of the blades gets too high, the thrust may collapse on the blade and the screw will work at greatly increased slip and much lower efficiency. This might happen at very high  $B_p$  values.

Cavitation problems as such are outside the scope of this treatise.



Barnaby suggests the following limit criterion for thrust loading to be outside cavitation danger:



To establish safe limits of thrust load for trawlers, it is suggested to establish the values for static thrust loads, i.e. the loads imposed on the blade when running at zero speed of advance. This would ensure safe loading figures for trawling at low speeds.

Barnaby also suggest a formula for the establishment of static thrust, viz.

$$T_s \text{ in tons} = 0.028 ( \text{SHP} \times \text{Dft.} )^{2/3}$$

SHP to be used here is of course the power actually developed at static thrust conditions. Normally this will be the power developed at perhaps 80% maximum engine rpm, in case of a fixed pitch propeller installation.

In such a case only a reduction in pitch could ensure full power output and absorption, but this would mean lower efficiency at higher speeds.

The controllable pitch propeller solves this difficulty, by allowing the blade pitch to be adjusted to the operating condition, i.e. at constant rpm, the pitch would be high for free running, while it must be reduced for towing work. At the same time the controllable pitch propeller is an excellent helper in manoeuvring the vessel. It becomes increasingly popular now in trawlers of even large size.

The Caterpillar propeller calculator suggests a safe blade loading of 8 lbs./in<sup>2</sup>, any higher value should be corrected by an increase in blade area. Comparing this with Barnaby's curve, it is somewhat on the high side, except for fast vessels. (American fishing vessels are normally faster than their European counterparts, due not least to the craze for very high power engine installations in the U.S.A.).

If the thrust loading figure suggests a critical condition, the blade area of the propeller has to be increased. This is best done by increasing the diameter, if possible, in view of the above. Under special circumstances it might be necessary to use wider blades (higher DAR) or even bigger number of blades (3 instead of 2, or even 4 or 5 instead of 3).

More detailed information on the theory of screw propellers will be found in any standard text book on naval architecture, for the purpose of small craft design work the books suggested in the appended list of reference are recommended most. Traung's article on Selection of Propellers for Fishing Vessels, as distributed, should be carefully read and digested, as it gives a very useful procedure to follow when selecting props for trawlers and other heavy duty vessels.



## Interaction between Ship and Propeller

So far hull resistance and propeller performance were considered entirely separat, i.e. the hull without propeller, and the propeller without acting on a hull, or, as the terminus technicus goes, in open water condition.

Unfortunately the interaction of propeller and hull is of an extremely complex nature and only the rough principles can be treated here.

The propeller affects the hull resistance mainly by its suction; resistance is also affected of course by the rudder and other appendages.

The hull bears on the performance of the screw mainly due to its form and the resulting wakestream.

With a hull is towed in a tank without propeller at  $V$  knots, the total resistance will be  $R$ . If the same hull travels at  $V$  knots self propelled (i.e. with propeller working behind the hull), the resulting thrust of the propeller will not equal  $R$ , but will be somewhat larger, the difference  $T - R$  being the additional resistance caused by the propeller action. This introduces a new factor into the play, the thrust deduction factor.

$$T - R = t \times T \quad \text{or} \quad R = T (1 - t)$$

$(1 - t)$  is known as the thrust deduction factor, and  $t$  as the thrust deduction coefficient.

Thus the resistance  $R$  of a towed hull must be multiplied by  $T/R$  or divided by  $(1 - t)$  to obtain the total resistance of the self propelled hull, or the thrust needed to travel at speed  $V$  knots.

As the propeller is working at a speed of  $V$  knots behind the ship as if working in open water at  $V_a$  knots (provided<sup>a</sup> the wake is assumed to be constant), thrust, torque and Power absorbed will be as for a screw operating in open water at  $V_a$  knots.  $V_a$  follows the relation  $V_a = V (1 - w_t)$  as seen previously.

For thrust  $T$  and  $P_o$  power absorbed at  $V_a$  knots, the propeller efficiency can be written as

$$\eta_o = \frac{T \times V_a}{P_o} ; \text{ this is known as the open water eff.}$$

$T \times V_a$  is the useful work done behind the vessel or in open water and is termed Thrust Horse Power, or THP.

The thrust  $T$  produced by the propeller is applied to the ship at the thrust bearing at a speed of  $V$  and not  $V_a$ . Thus the work done at the thrust bearing is  $T \times V$ , and the ratio  $V/V_a$  then represents the improvement on output power over input power due to the wakestream. It is known as wake gain and means the work done by the wake on the propeller.

The wake is normally not uniform over the whole disc as assumed, but varies considerably. The variation factor to indicate the power required due to this wake variation in the troubled stern waters is termed relative rotative efficiency  $\eta_r$  or  $P_o/P$ . This is normally very near unity and can be neglected.



Three major effects of interaction have been considered so far:  
Thrust Deduction, Wake Gain and Relative Rotative Efficiency.

They can be combined into one equation, representing the whole propulsion complex of the ship under consideration:

$$\text{EHP} = \eta_o \times P_o \times R/T \times V/V_a \times P_o/P \quad \text{out of this the following are extracted:}$$

$R/T \times V/V_a$  is termed Hull Efficiency  $\eta_h$  and can also be written

$$\eta_h = \frac{(1 - t)}{(1 - w_t)}$$

the equation then becomes:

$$\text{EHP} = \eta_o \times P_o \times \eta_h \times \eta_r$$

$\eta_o$  = open water efficiency  
 $\eta_h$  = hull efficiency  
 $\eta_r$  = relative rotative efficiency

$\eta_r$  can normally be neglected, and  $\eta_h$  is then also the ratio  $\frac{\text{EHP}}{\text{THP}}$  if both refer to the same condition of the hull tested.

Model tests are normally run for both without appendages due to the difficulty with boundary layers and scale effects if appendages are introduced.

#### The Propulsive Coefficient

For a diesel powered vessel this can now be written as  $\frac{\text{EHP}}{\text{BHP}}$  or

$$\text{PC} = \frac{\eta_o P_o \times \frac{\text{EHP}}{\text{THP}}}{\text{BHP}}$$

It will be observed that there is no allowance for appendage resistance, air resistance and transmission losses in this expression. If all these are taken into consideration, the Quasi Propulsive Coefficient, or QPC is obtained:

$$\text{QPC} = \frac{\text{Bare hull EHP} + \text{Appendage EHP} + \text{Air HP}}{\text{Delivered HP}}$$

#### Propeller selection for small fishing craft

The small craft designer is normally faced with the problem of selecting a propeller for use with a definite engine, giving a set amount of h.p. at a set rpm.

As the power output figures for small diesels are not normally absolutely trustworthy, a number of allowances and assumptions have to be made by the designer based on his experience, and it is not surprising under these circumstances, that even the most elaborate computation of propeller size does not absolutely safeguard against surprises! A great element of doubt is brought into the calculations as outlined above, and it is usually permissible to use shortcuts and simple calculations to arrive at a suitable propeller. The theory above is given more to assure an understanding of the issues involved than to be meticulously applied when calculating propeller diameters and pitch for a given installation.



While the small craft designer is normally tied to a certain power and rpm of his engine, and will foremost want to establish diameter and pitch of the best suited propeller, the designer of big merchant ships is often faced with the problem the other way round. He may only be given the max. possible propeller diameter and the desired service speed, and he will have to establish the best rpm and the power to be used. With turbine machinery this is not so difficult as it looks, as practically any number of revolutions can be had by installing the proper gearing.

There are a number of aids available for the establishment of best suitable propeller dimensions. The best know of these are the tables and graphs given for the various series of standard propellers in the B and system, Barnaby's table of propeller data, and the Caterpillar<sup>P</sup> calculator.

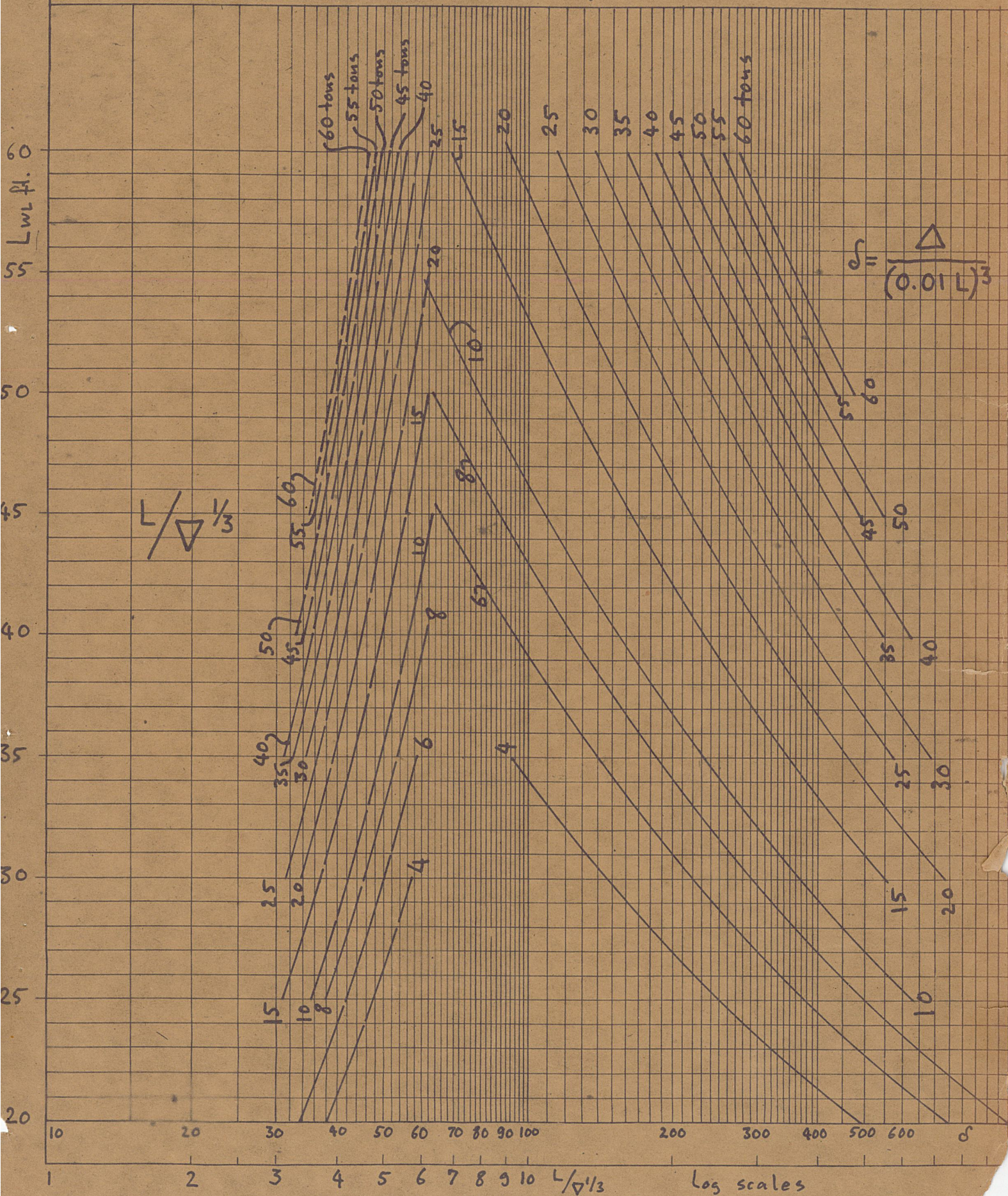
When using these aids, great care has to be taken to assess the power available as correctly as possible, i.e. to apply useful allowances for losses in gears and stern tubes or intermediate bearings.

The judgment as to power required for a given boat to attain a certain speed is not easy; this will be treated to some extent in Part III under the heading Powering.

Conclusion: It will always be worth while to check on the propeller diameter for a boat under consideration, to ascertain whether a standard propeller offered by the engine manufacturers might be useful. Generally speaking it must be remembered that standard propellers are offered for use with medium to light craft operating at medium to high speeds, thus making them unsuitable propellers for heavy work with medium to heavy boats at lower speeds.

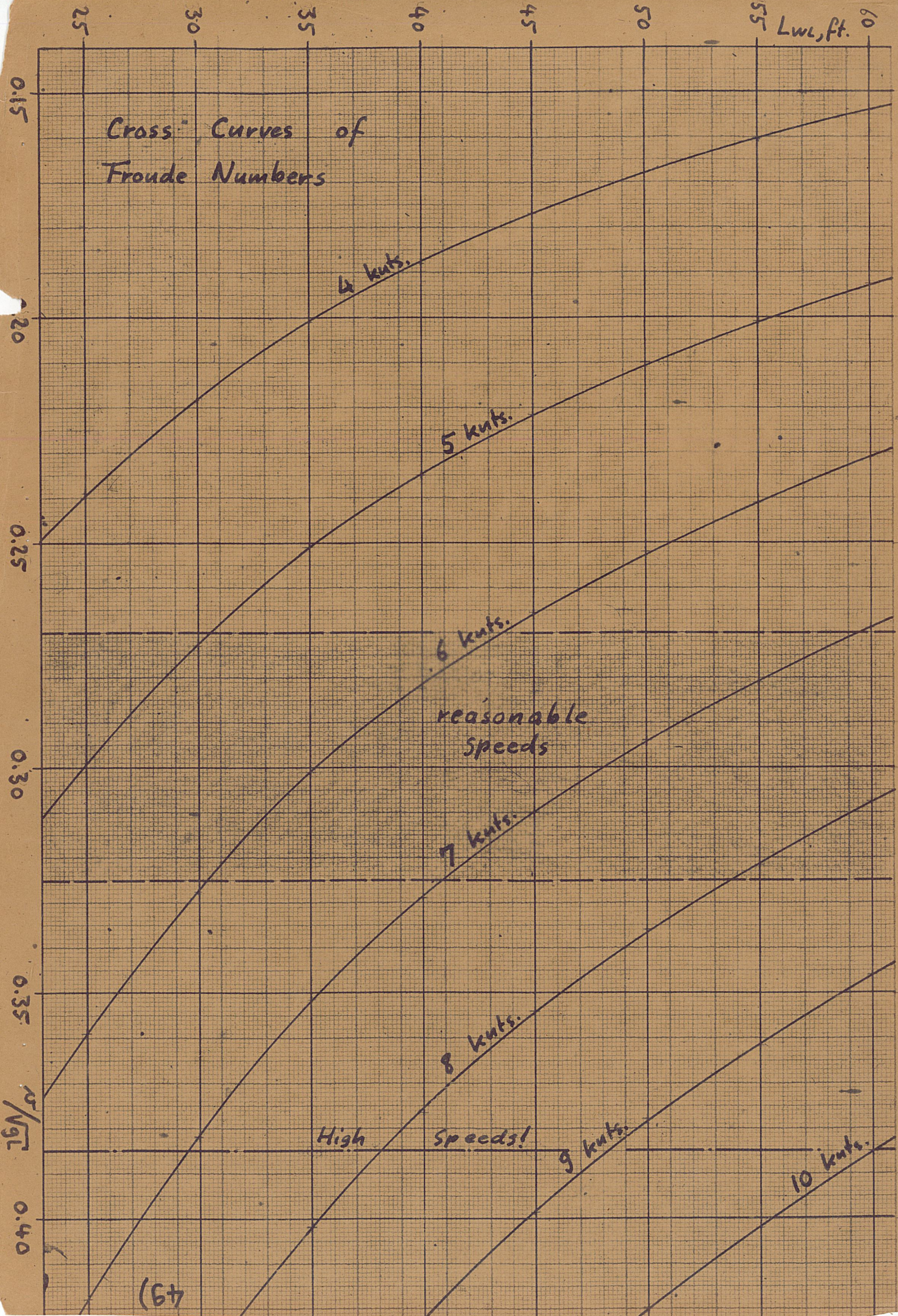


Relation between two common Length/Displacement notations:  $\delta = \frac{\Delta}{(0.01L)^3}$  and  $\frac{L}{\Delta^{1/3}}$





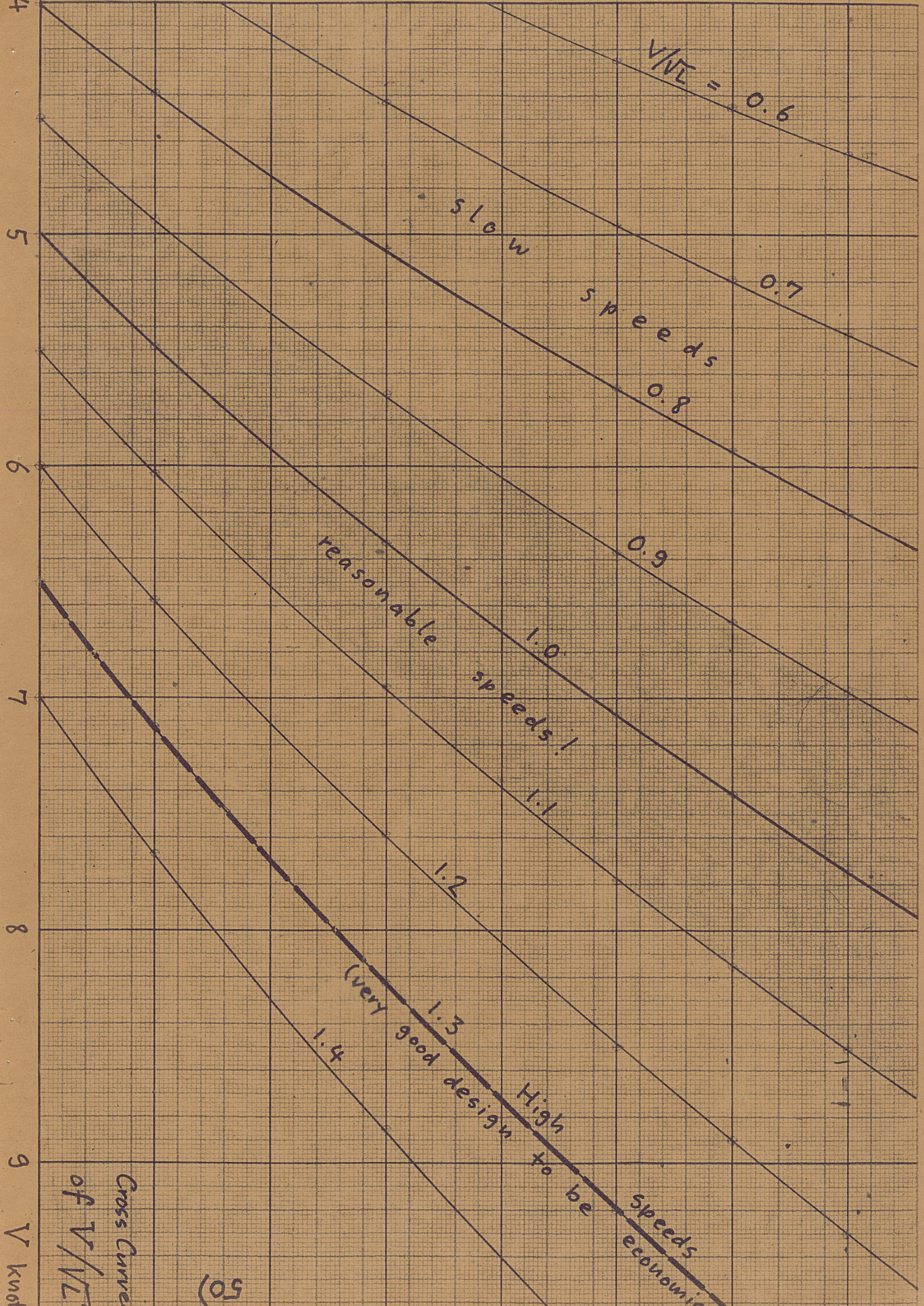
# Cross Curves of Froude Numbers



(67)



60 55 50 45 40 35 30 25



Cross Curves  
of  $V/V_L$

(50)

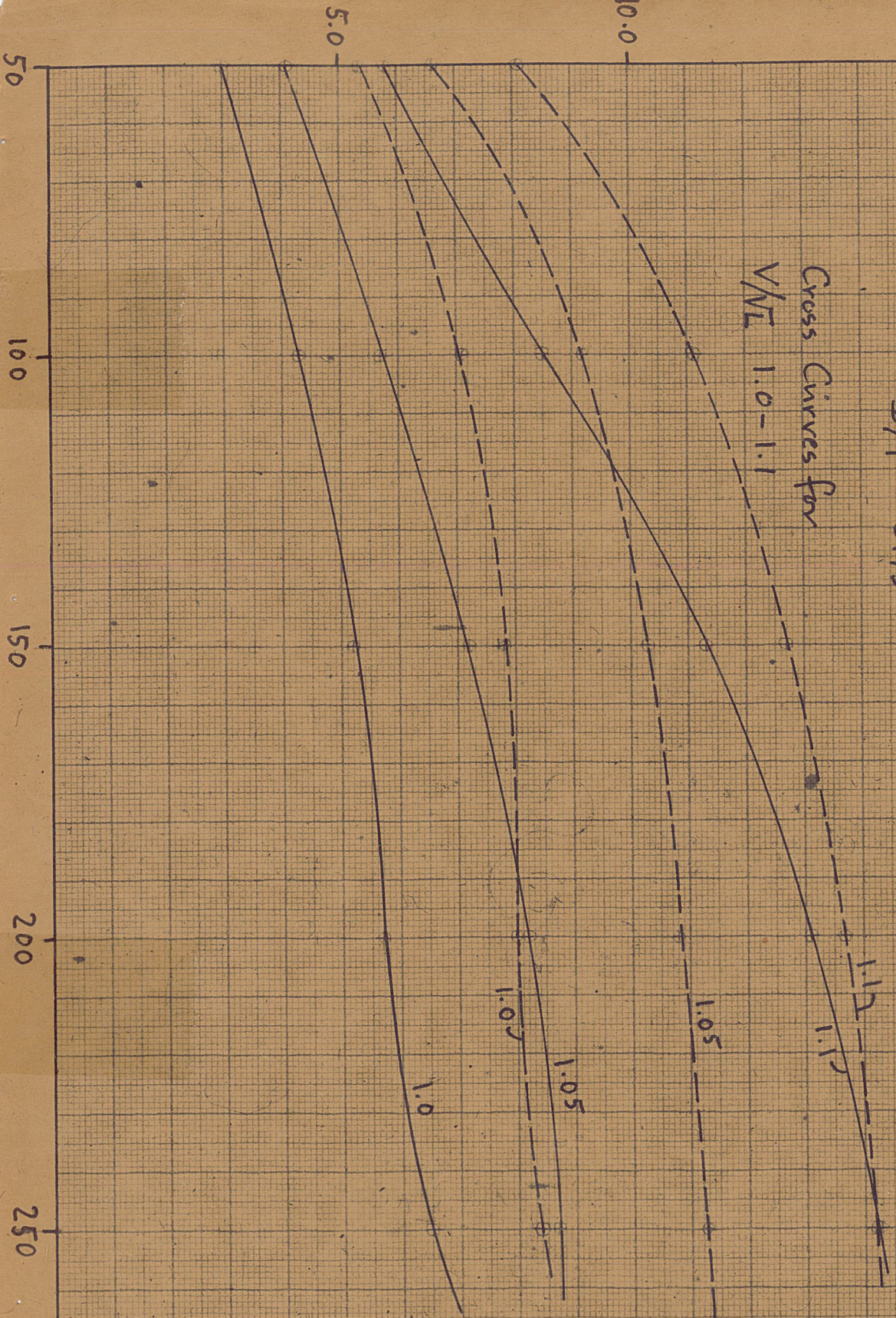


$\frac{R_r}{\Delta}$  lb/ton

Residual Resistance per ton  
Taylor's Standard Series

$B/T = 3.75$

Cross Curves for  
 $V/\sqrt{L}$  1.0-1.1



$C_p(\varphi) = 0.5$

$C_p(\varphi) = 0.60$



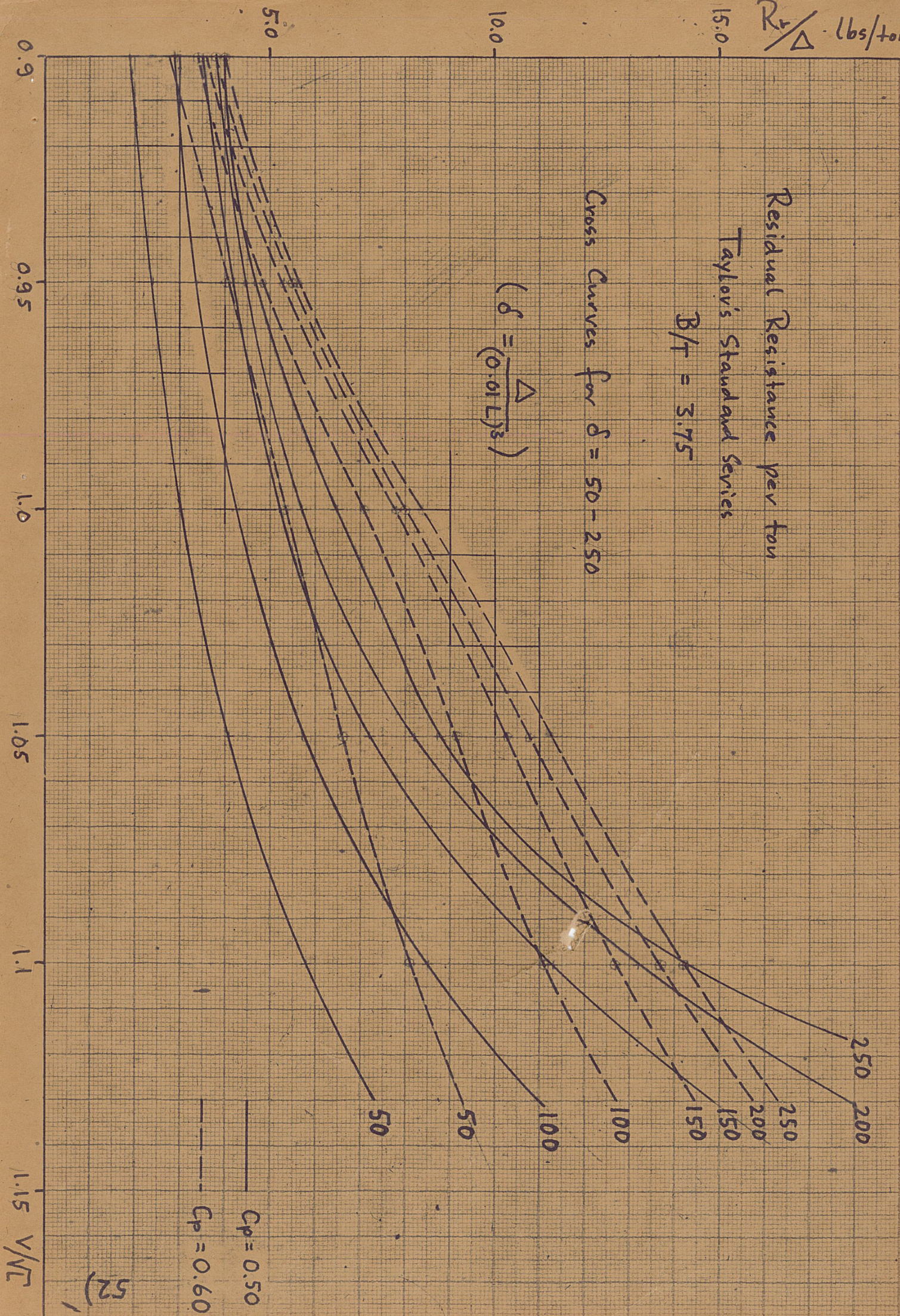
$R/\Delta$  lbs/ton

Residual Resistance per ton  
Taylor's Standard Series

$B/T = 3.75$

Cross Curves for  $\delta = 50-250$

$$\left(\delta = \frac{\Delta}{(0.01L)^3}\right)$$



$C_p = 0.50$

$C_p = 0.60$



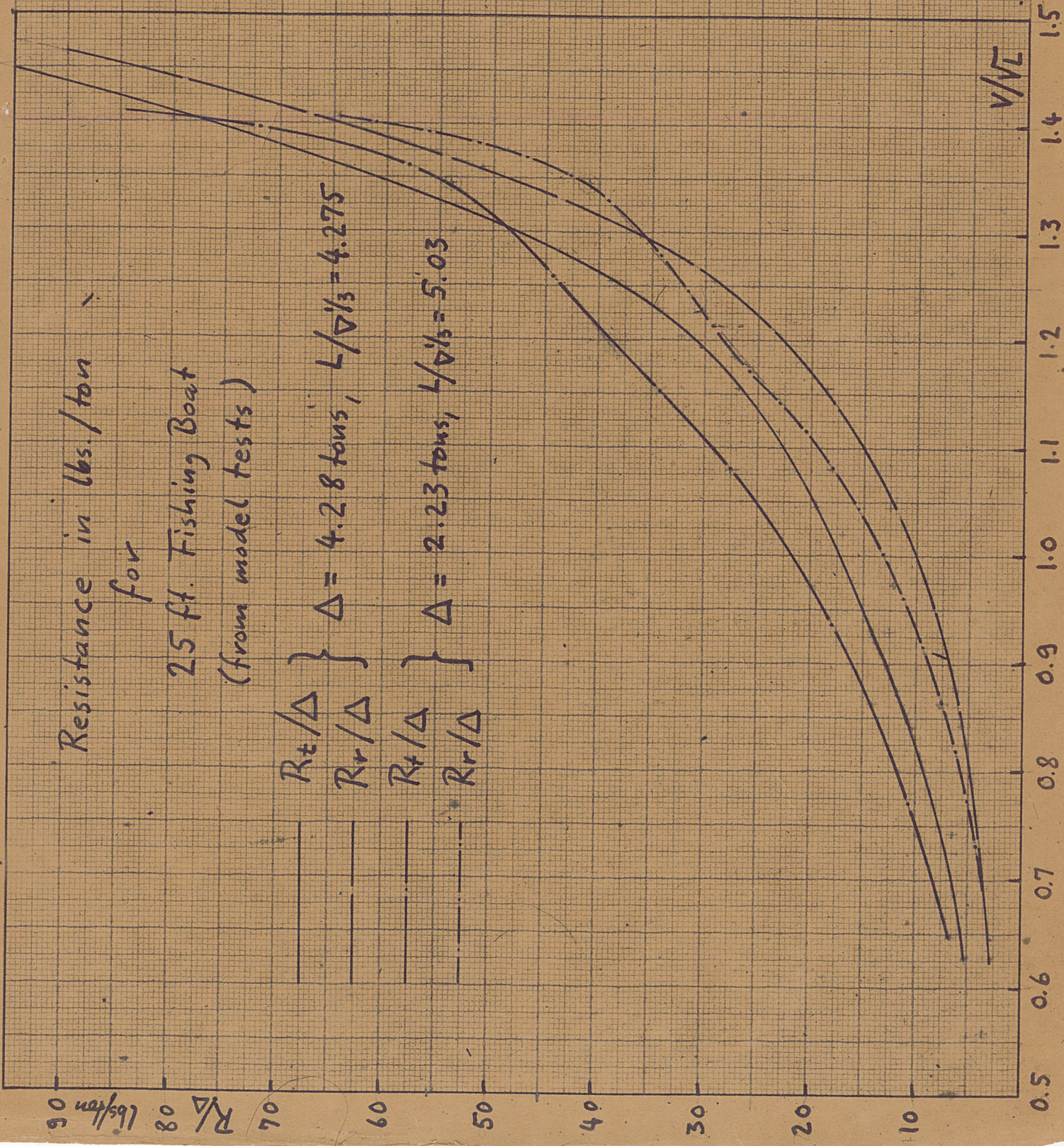
Resistance in lbs./ton

for

25 ft. Fishing Boat

(from model tests)

$R_t/\Delta$  }  $\Delta = 4.28 \text{ tons}, L/\Delta^{1/3} = 4.275$   
 $R_r/\Delta$  }  
 $R_t/\Delta$  }  $\Delta = 2.23 \text{ tons}, L/\Delta^{1/3} = 5.03$   
 $R_r/\Delta$  }

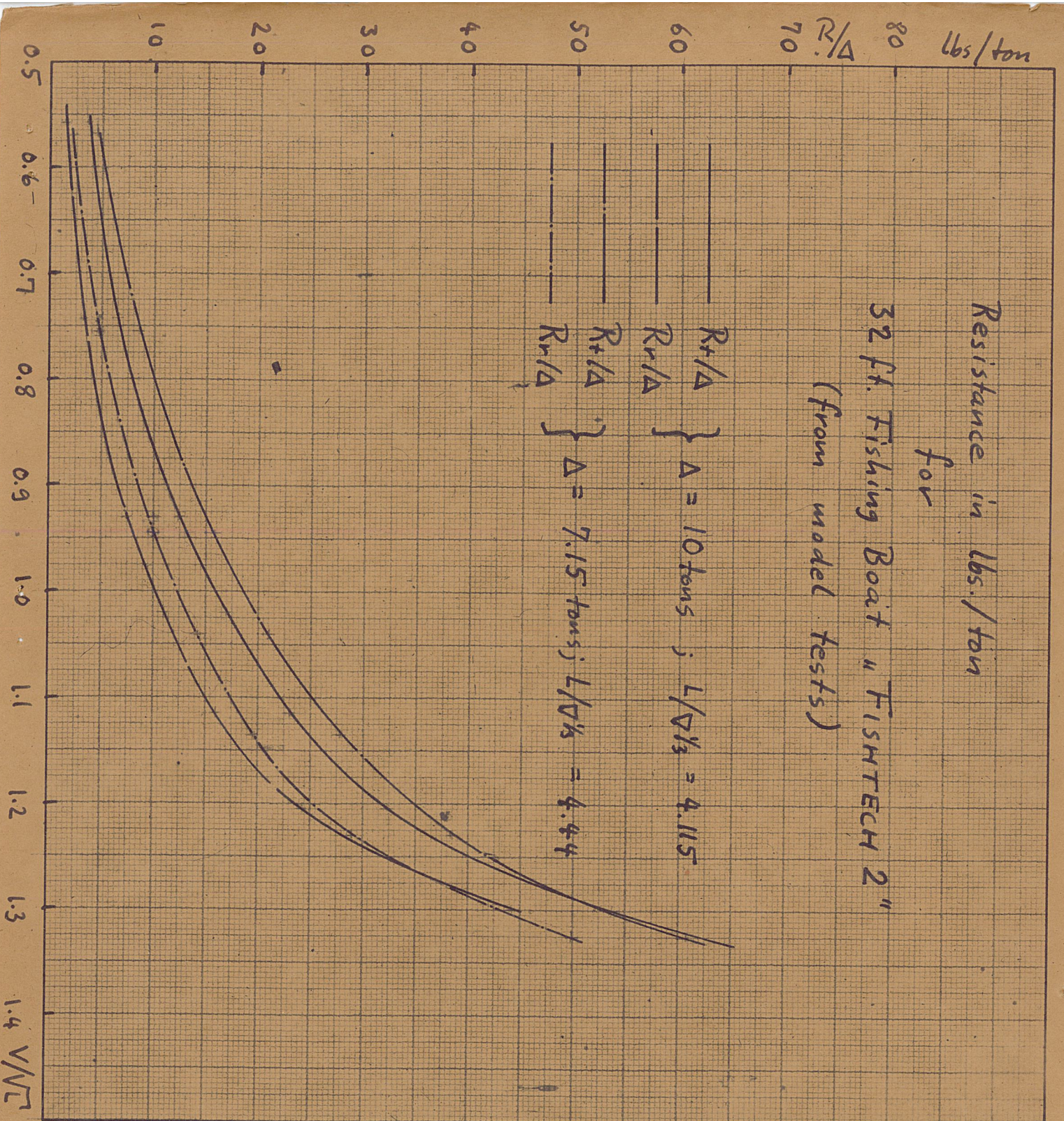


While the total resistance for the heavy displacement is obviously much bigger than for the light, the figures for specific resistance ( $R_t/\Delta$ ) indicate that displacement, or rather  $L/\Delta^{1/3}$ , is of secondary importance only. Specific resistance in low speed range is actually bigger for the light displacement!



Resistance in lbs./ton  
for

32 ft. Fishing Boat "FISHTECH 2"  
(from model tests)



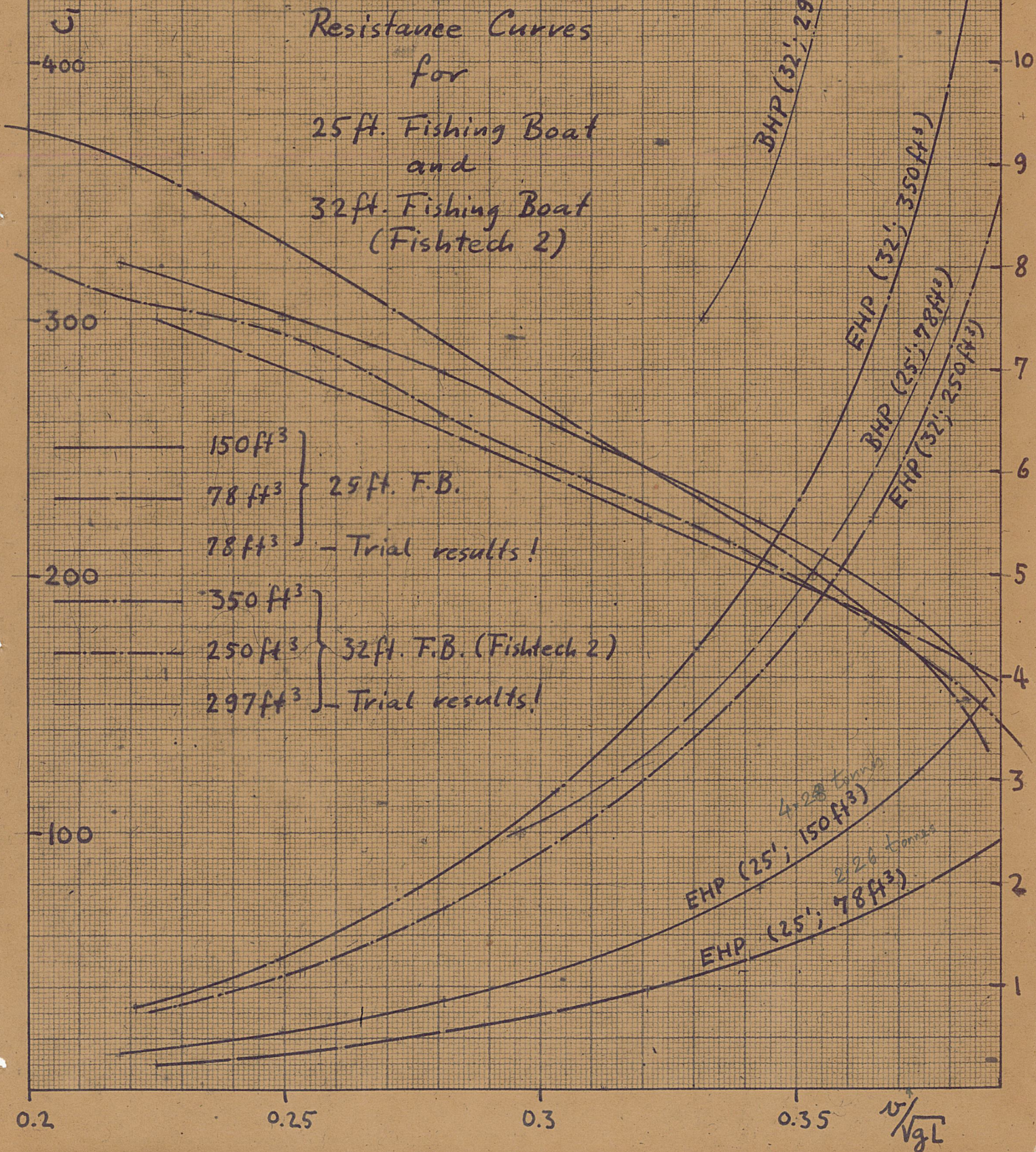
The curves show similar characteristics as those for 25' boat. The curves for the two displacements are nearer due to the lesser difference in  $L/V_t^3$ . The somewhat lower  $R_t/\Delta$  is due to lower  $R_F$  only, with  $R_r/\Delta$  figures being almost identical at speeds up to 1.15 for both boats.



|     |      |     |        |     |                             |     |              |     |     |      |
|-----|------|-----|--------|-----|-----------------------------|-----|--------------|-----|-----|------|
| 3.5 | 0.25 | 4   | 25 ft. | 4.5 | $\nabla = 150 \text{ ft}^3$ | 5   | $L = 22.75'$ | 5.5 | 6   | kts. |
| 3.5 |      | 4   | 25 ft. | 4.5 | $\nabla = 78 \text{ ft}^3$  | 5   | $L = 21.5'$  | 5.5 | 6   |      |
| 4   |      | 4.5 | 32 ft. | 5   | $\nabla = 350 \text{ ft}^3$ | 5.5 | $L = 29'$    | 6   | 6.5 | kts. |
| 4   |      | 4.5 | 32 ft. | 5   | $\nabla = 250 \text{ ft}^3$ | 5.5 | $L = 28'$    | 6   | 6.5 |      |

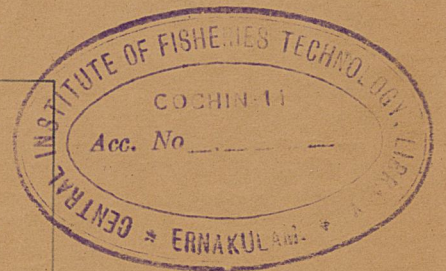
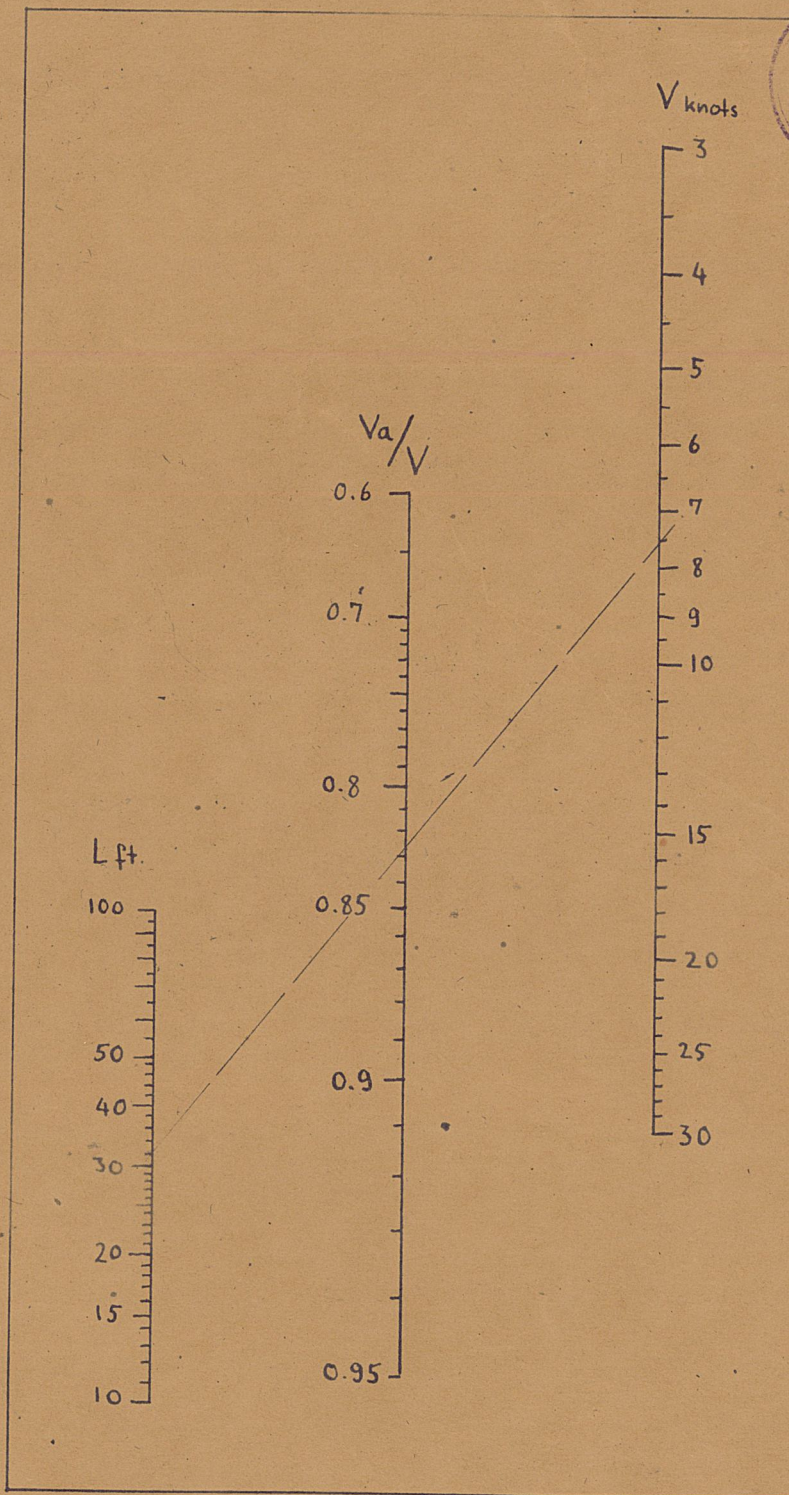
### Resistance Curves for

25 ft. Fishing Boat  
and  
32 ft. Fishing Boat  
(Fishtech 2)





P.G. Tomlin's Wake Nomogram (expanded for low speeds)  
from D. Phillips-Burt: Nav. Arch. of small craft.



For use: Connect correct  $L_{wl}$  and ships. speed  $V$   
with straight line and read off fraction  
 $V_a/V$ . (Example: 32 ft.  $L_{wl}$ , at 7.5 knots  $V$ ,  
 $\rightarrow V_a/V = 0.825 \rightarrow w_t = 17.5\%$ )



Propeller Constants due to K.C. Barnaby;  $B_p$  and  $\delta$  notation:  $B_p = \frac{\sqrt{SHP} \times rpm}{V_a^{3.5}}$   
 optimum pitch ratios between thick lines!  $\delta = \frac{ND}{V_a}$

| Pitch ratios | 8                |        | 10               |        | 12               |        | 14               |        | 16               |        | 20               |        | 26               |        | 32               |        | 40               |        | 50               |        | 60               |        |
|--------------|------------------|--------|------------------|--------|------------------|--------|------------------|--------|------------------|--------|------------------|--------|------------------|--------|------------------|--------|------------------|--------|------------------|--------|------------------|--------|
|              | $\frac{ND}{V_a}$ | $\eta$ | $\frac{ND}{V_a}$ | $\eta$ | $\frac{ND}{V_a}$ | $\eta$ | $\frac{ND}{V_a}$ | $\eta$ | $\frac{ND}{V_a}$ | $\eta$ | $\frac{ND}{V_a}$ | $\eta$ | $\frac{ND}{V_a}$ | $\eta$ | $\frac{ND}{V_a}$ | $\eta$ | $\frac{ND}{V_a}$ | $\eta$ | $\frac{ND}{V_a}$ | $\eta$ | $\frac{ND}{V_a}$ | $\eta$ |
| 0.5          |                  |        |                  |        |                  |        |                  |        |                  |        | 218              | .47    | 236              | .475   | 253              | .476   | 271              | .47    | 293              | .455   | 310              | .440   |
| 0.55         |                  |        |                  |        |                  |        |                  |        |                  |        | 210              | .515   | 227              | .52    | 245              | .512   | 264              | .498   | 285              | .482   | 301              | .465   |
| 0.6          |                  |        |                  |        | 172              | .475   | 181              | .505   | 189              | .528   | 204              | .549   | 221              | .550   | 239              | .54    | 257              | .525   | 277              | .506   | 293              | .483   |
| 0.65         |                  |        | 159              | .50    | 167              | .546   | 175              | .565   | 183              | .575   | 194              | .58    | 214              | .57    | 231              | .554   | 250              | .533   | 270              | .508   | 286              | .484   |
| 0.7          | 146              | .48    | 154              | .56    | 162              | .59    | 170              | .60    | 177              | .605   | 190              | .60    | 207              | .582   | 224              | .56    | 242              | .535   | 263              | .506   | 279              | .483   |
| 0.75         | 141              | .555   | 149              | .601   | 157              | .619   | 165              | .623   | 172              | .623   | 184              | .610   | 201              | .585   | 217              | .56    | 235              | .53    | 255              | .501   | 272              | .48    |
| 0.8          | 137              | .604   | 145              | .631   | 153              | .638   | 160              | .637   | 167              | .632   | 179              | .617   | 196              | .585   | 212              | .556   | 229              | .523   | 249              | .495   | 265              | .474   |
| 0.85         | 133              | .64    | 141              | .653   | 148              | .65    | 155              | .646   | 162              | .638   | 174              | .618   | 191              | .58    | 206              | .55    | 223              | .515   | 243              | .488   | 259              | .466   |
| 0.9          | 129              | .662   | 137              | .668   | 144              | .66    | 151              | .65    | 158              | .638   | 170              | .616   | 187              | .573   | 201              | .541   | 217              | .507   | 237              | .48    | 253              | .458   |
| 0.95         | 125              | .68    | 133              | .675   | 140              | .664   | 147              | .65    | 154              | .634   | 166              | .61    | 182              | .566   | 196              | .533   | 212              | .498   | 232              | .47    | 247              | .45    |
| 1.0          | 121              | .689   | 130              | .68    | 137              | .666   | 144              | .649   | 151              | .63    | 163              | .601   | 178              | .559   | 192              | .526   | 208              | .489   | 226              | .46    | 241              | .441   |
| 1.05         | 118              | .695   | 127              | .68    | 134              | .663   | 141              | .644   | 148              | .625   | 160              | .591   | 174              | .549   | 187              | .517   | 203              | .48    | 221              | .452   | 235              | .438   |
| 1.1          | 116              | .698   | 124              | .678   | 131              | .659   | 139              | .639   | 145              | .618   | 156              | .582   | 170              | .540   | 183              | .508   | 199              | .470   | 217              | .444   | 230              | .427   |
| 1.15         | 113              | .698   | 121              | .674   | 129              | .653   | 136              | .631   | 142              | .61    | 153              | .573   | 167              | .53    | 180              | .50    | 195              | .46    | 212              | .436   |                  |        |
| 1.2          | 111              | .695   | 119              | .67    | 126              | .646   | 133              | .624   | 139              | .602   | 150              | .563   | 164              | .520   | 177              | .49    | 191              | .452   |                  |        |                  |        |



L/B : Waterline length divided by waterline beam

This ratio has a considerable bearing on the statical stability of the vessel. Very low ratios would indicate a boat fully form stable, while high values would mean that ballast would probably have to be added.

In terms of influence on displacement, L/B would generally be lower for shallow draft boats and higher for deep boats.

For very small, fully decked boats, it is found by experience that the rise in CG with low drafts often offsets the gain in BM due to the large waterline area - ballasting becomes necessary there.

Useful ranges of values of L/B are:

| Length group | Shallow draft boats | Deep boats |
|--------------|---------------------|------------|
| 10' - 25'    | 2.6 - 2.9           | 3.0 - 3.2  |
| 25' - 35'    | 2.6 - 3.0           | 3.0 - 3.4  |
| 35' - 50'    | 2.9 - 3.2           | 3.2 - 3.6  |
| 50' - 70'    | 3.0 - 3.5           | 3.5 - 3.8  |
| 70' - 90'    | 3.4 - 3.7           | 3.7 - 4.2  |

Caution: These values are only guide lines! It is impossible to demarcate shallow draft and deep boats completely, and the range of values merely shows where a majority of tried boats would be placed.

B/T : Waterline beam divided by mean draft (draft amidships to rabbet line)

This ratio is of some influence in determining the range of displacement for a boat with determined L/B ratio. It is not easily possible to give a table of ranges for this ratio, as almost any reasonable figure between 2 and 4 is possible, depending largely on the type of vessel, the displacement wanted and the prismatic coefficient and midship area adopted.

From a number of model tests it becomes evident that a high B/T is required (over 3.0) to assure low resistance, i.e. low draft is preferable. This requirement is in conflict with the need to accommodate large diameter propellers, and one compromise resulting is often very extensive rake to the keel, this also resulting in the desirable feature of bringing LCB aft of midships.

$L/\nabla^{1/3}$  : Waterline length divided by the cube root of the displacement volume

This ratio is very indicative of the relative lightness of the boat for a given length. Generally speaking, low ratios mean heavy boats, while high ratios stand for light boats.

As has been seen in Part II,  $L/\nabla^{1/3}$  is a ratio of some importance in model testing work and when comparing model results.

An attempt is made here to show ranges of  $L/\nabla^{1/3}$  ratio for various length of boat and relative heaviness:

| Length(WL) | Light boat | Medium boat | Heavy boat |
|------------|------------|-------------|------------|
| 20' - 30'  | 5.5 - 6.5  | 4.5 - 5.5   | 4.3 - 4.8  |
| 30' - 50'  | 5.5 - 6    | 4.5 - 5     | 4.1 - 4.5  |
| 50' - 70'  | 4.5 - 5.5  | 4 - 4.5     | 3.7 - 4    |
| 70' - 90'  | 4.5 - 5    | 3.8 - 4.3   | 3.3 - 3.8  |



It should be borne in mind that  $L/\nabla^{1/3}$  ratio can vary considerably for one boat, when investigating light and loaded conditions!

These 3 ratios must be used in close connection with each other. When working out main dimensions for new boats, it will be seen that they interdepend very much. The graph on page 61) gives a range of values for all three, based on length and draft and to ordinates of beam and displacement volume.

The following tables contain ratios for a number of boats built and projected.

#### Built boats

| Type        | Lwl   | Bwl  | $\nabla$ | Tm   | L/B  | B/T  | $L/\nabla^{1/3}$ |
|-------------|-------|------|----------|------|------|------|------------------|
| Pablo       | 21.6  | 6.3  | 91       | 1.7  | 3.43 | 3.71 | 4.8              |
| 25 ft.      | 23.0  | 7.0  | 95       | 1.67 | 3.29 | 4.19 | 5.04             |
| 30 ft.      | 28.0  | 7.4  | 161      | 2.0  | 3.78 | 3.7  | 5.15             |
| 31' - 9"    | 28.13 | 8.0  | 205      | 2.3  | 3.51 | 3.48 | 4.77             |
| Fishtech II | 28.6  | 8.67 | 297      | 2.9  | 3.3  | 2.99 | 4.28 (actual)    |

#### Projected boats

| Type            | Lwl  | Bwl  | $\nabla$ | Tm   | L/B  | B/T  | $L/\nabla^{1/3}$ |
|-----------------|------|------|----------|------|------|------|------------------|
| 32 ft.          | 28.0 | 9.25 | 233      | 2.35 | 3.03 | 3.93 | 4.54 (light)     |
| 36 ft. Trawlers | 33.4 | 9.92 | 375      | 2.85 | 3.37 | 3.48 | 4.64 "           |
| 49 ft. Bombay   | 45.0 | 13.2 | 1065     | 4.3  | 3.4  | 3.07 | 4.41             |

#### The Prismatic Coefficient $\psi$ and the Midship Section Area Coefficient $C_m$

In part II it was seen that the prismatic has considerable, if not decisive, influence on the resistance properties of a fishing vessel. It would seem to be good policy during the initial design stage to adopt the optimum prismatic, i.e. a value in the neighbourhood of 0.575; other considerations than those dictated by minimum resistance, such as increased carrying capacity, might make it necessary to increase or decrease the figure later. It will hardly be necessary to decrease from the optimum!

Once  $L/B$ ,  $B/T$  and  $L/\nabla^{1/3}$  are fixed, the prismatic becomes the decisive factor in fixing the midship section area. From the definition of prismatic coefficient in Part I, we know that it is displacement divided by the product of waterline length and maximum section area. Maximum section area normally occurs at or near midships (for fishing vessels almost always somewhat aft of midships), and is usually very nearly equal to midship section area. To show the interdependence between prismatic and section area, the midship section area coefficient  $C_m$  is used here:

$L/B$  fixes  $B$  —  $B/T$  fixes mean draft  $T_m$  — and  $L/\nabla^{1/3}$  fixes displacement volume.

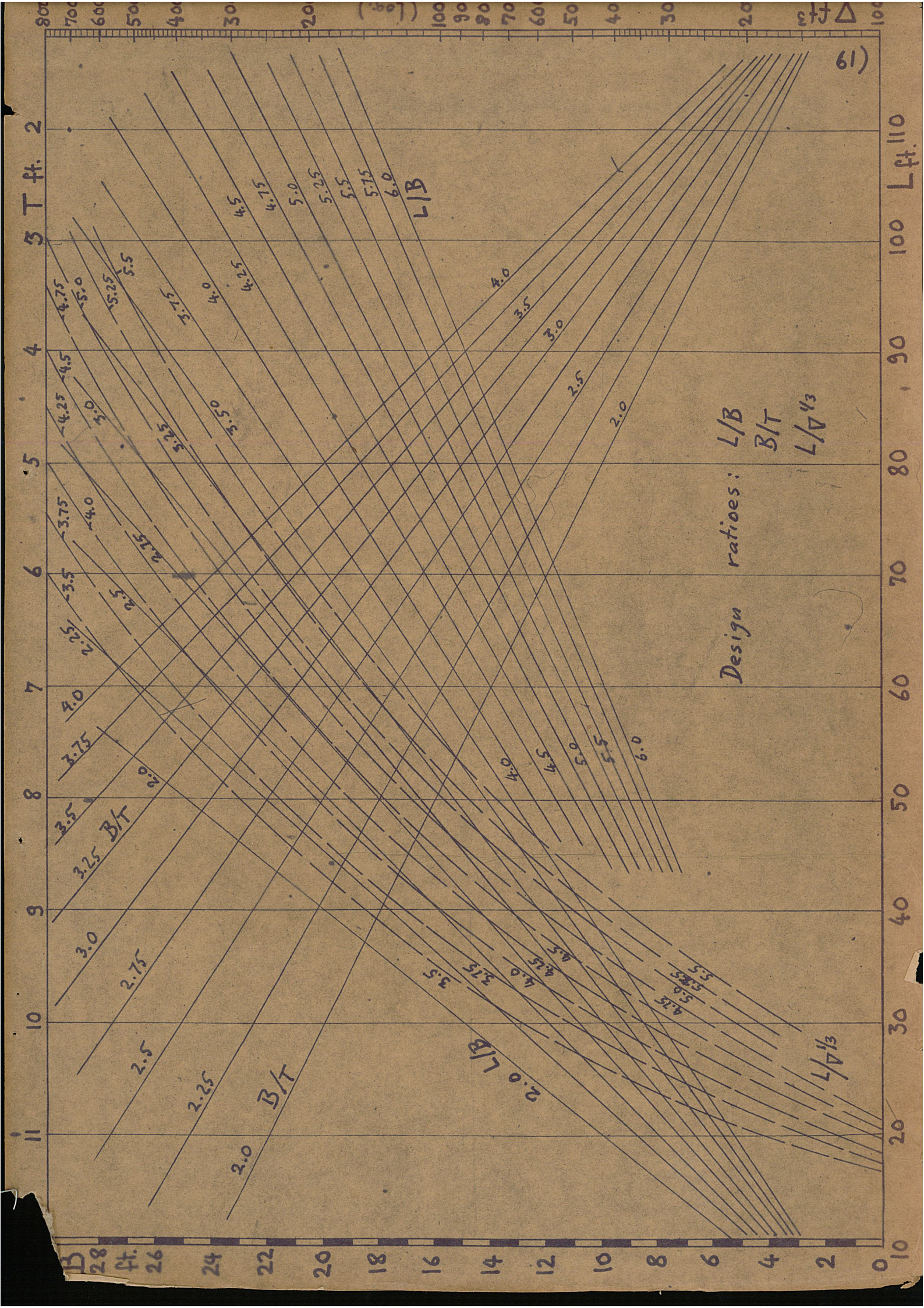
For a midship section area of  $A_m$ ,  $C_m = \frac{A_m}{B \times T_m}$

$$\psi = \frac{\nabla}{L \times A_m} = \frac{\nabla}{L \times B \times T \times C_m}$$

$$C_m = \frac{\nabla}{L \times B \times T \times \psi}$$

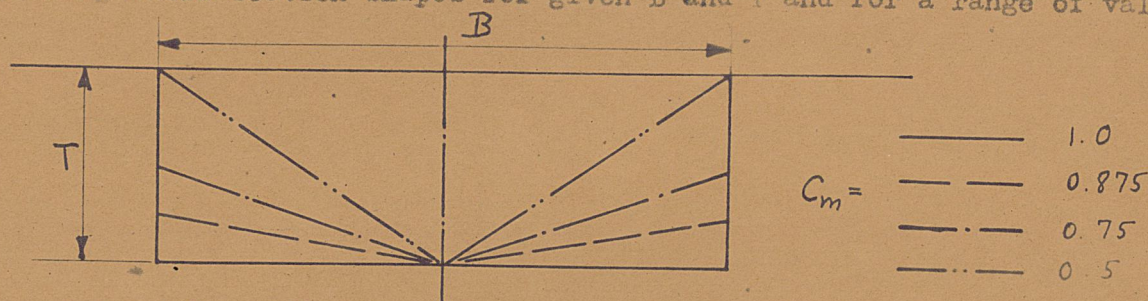
or, as  $\frac{\nabla}{LBT} = \delta$  or Block coefficient,  $C_m = \frac{\delta}{\psi}$







Thus  $C_m$  is fixed also, if all the other ratios are fixed! It is well to remember that  $C_m$  has a decisive influence on the underwater shape of the vessel and as such is also an important factor in considerations of more or less wave making resistance. It was earlier said that wave making resistance is due to the ship bodily moving away water; from this it could be easily followed that small  $C_m$  would be associated with relatively lower  $R_w$ , while high  $C_m$  would also account for somewhat larger  $R_w$ . As a guide to its importance, it is well to think of  $C_m$  as representing the following basic section shapes for given  $B$  and  $T$  and for a range of values:



Rounding of the bilge will of course somewhat lower these values, but they are well worth to be kept in mind when settling on a midship section in the initial design stage.

The table on page 63) was compiled in an attempt to show how changing ratios affect the final vessel, and it will also serve to show that by no means any combination of ratios is going to produce a useful hull.

In the table, a boat of  $L_{wl}$  40 ft. is considered, with  $L/B$  ratios of 2.75, 3.0, 3.25 and 3.5;  $B/T$  ratios of 2.5, 3.0 and 3.25, at  $L/\nabla^{1/3}$  ratios of 3.75, 4.25 and 4.75, each at a prismatic of 0.57, 0.60 and 0.63.

With ratios and coefficients fixed, keeping in mind the physical properties required of the boat, the first lines plan has to be attempted.

### First Draft Lines

These may be drawn with only a few sections and waterlines, to give a reasonably accurate idea of the shapes and the volumes and volume centres. These lines will serve well to check on the displacement of the vessel as compared to the weight arrived at in the preliminary weight check, as well as giving an indication of the statical stability properties. It is very easy, and quickly done, to calculate a series of BM values for these first lines, to assume various positions of VCG and to check on the GM resulting. It is not possible here to give any ruling on the GM minimum that has to be maintained, but it would seem that 20" should be sufficient for boats up to about 35 ft., and 24" - 30" for larger boats up to about 75 ft.

For the purpose of the first GM check, the KG can be taken at approximately 0.775 D for boats with no or little superstructures, and at about 0.825 D for those with full decks and deckhouses. D in this case is measured from the side of the deck amidships to the rabbet line (or from top of topsides in case of open vessels).

The first draft lines will also serve as a base for a draft general arrangement (in the case of bigger, more complex vessels) and a draft construction drawing. It will be well worth while to assess the approximate power requirements of the vessel at this stage. This can be done in numerous ways, and the table on page 64) will be useful for the purpose, although perhaps giving values slightly on the low side for heavy fishing vessels. If any model results of similar vessels are available, full use of these should be made at this stage.

A quick propeller calculation for an optimum screw at full power and maximum speed, with the help of the Caterpillar calculator or a set of  $R_p$ ,  $\delta$  curves



# Interdependability of ratios and coefficients for 40 ft. Lwl boat.

| L/B    | 2.75                                                                                                        |                                                                                            |      | 3.0   |                                                                                   |      | 3.25 |                                                                                                                    |       | 3.5   |                                                                                                                          |      |      |
|--------|-------------------------------------------------------------------------------------------------------------|--------------------------------------------------------------------------------------------|------|-------|-----------------------------------------------------------------------------------|------|------|--------------------------------------------------------------------------------------------------------------------|-------|-------|--------------------------------------------------------------------------------------------------------------------------|------|------|
| Bft.   | 14.5                                                                                                        |                                                                                            |      | 13.35 |                                                                                   |      | 12.3 |                                                                                                                    |       | 11.42 |                                                                                                                          |      |      |
| B/T    | 2.5                                                                                                         | 3.0                                                                                        | 3.25 | 2.5   | 3.0                                                                               | 3.25 | 2.5  | 3.0                                                                                                                | 3.25  | 2.5   | 3.0                                                                                                                      | 3.25 |      |
| Tft.   | 5.8                                                                                                         | 4.83                                                                                       | 4.46 | 5.35  | 4.45                                                                              | 4.11 | 4.92 | 4.1                                                                                                                | 3.78  | 4.57  | 3.81                                                                                                                     | 3.52 |      |
| $\psi$ | $L/\nabla^{1/3} = 3.75 \rightarrow \nabla = 1215 \text{ ft}^3 \rightarrow \Delta = 34.7 \text{ tons}$ Heavy |                                                                                            |      |       |                                                                                   |      |      |                                                                                                                    |       |       |                                                                                                                          |      |      |
| 0.57   | $C_m$                                                                                                       | .634                                                                                       | .759 | .822  | .746                                                                              | .885 | .968 | .88                                                                                                                | 1.051 | -     | more than 1                                                                                                              |      |      |
| 0.60   |                                                                                                             | .601                                                                                       | .721 | .780  | .708                                                                              | .85  | .92  | .836                                                                                                               | 1.005 | -     | 0.966                                                                                                                    | -    | -    |
| 0.63   |                                                                                                             | .572                                                                                       | .686 | .744  | .674                                                                              | .81  | .876 | .796                                                                                                               | .956  | -     | 0.922                                                                                                                    | -    | -    |
| $\psi$ | $L/\nabla^{1/3} = 4.25 \rightarrow \nabla = 835 \text{ ft}^3 \rightarrow \Delta = 23.9 \text{ tons}$ Medium |                                                                                            |      |       |                                                                                   |      |      |                                                                                                                    |       |       |                                                                                                                          |      |      |
| 0.57   | $C_m$                                                                                                       | .435                                                                                       | .522 | .566  | .516                                                                              | .621 | .672 | .606                                                                                                               | .727  | .789  | .70                                                                                                                      | .84  | .908 |
| 0.60   |                                                                                                             | .413                                                                                       | .496 | .538  | .490                                                                              | .590 | .639 | .576                                                                                                               | .690  | .749  | .666                                                                                                                     | .798 | .864 |
| 0.63   |                                                                                                             | .394                                                                                       | .472 | .512  | .467                                                                              | .561 | .608 | .549                                                                                                               | .657  | .714  | .634                                                                                                                     | .76  | .822 |
| $\psi$ | $L/\nabla^{1/3} = 4.75 \rightarrow \nabla = 600 \text{ ft}^3 \rightarrow \Delta = 17.15 \text{ tons}$ Light |                                                                                            |      |       |                                                                                   |      |      |                                                                                                                    |       |       |                                                                                                                          |      |      |
| 0.57   | $C_m$                                                                                                       | .313                                                                                       | .377 | .407  | .368                                                                              | .443 | .480 | .435                                                                                                               | .522  | .566  | .505                                                                                                                     | .604 | .655 |
| 0.60   |                                                                                                             | .297                                                                                       | .358 | .387  | .35                                                                               | .421 | .455 | .413                                                                                                               | .496  | .538  | .479                                                                                                                     | .575 | .621 |
| 0.63   |                                                                                                             | .283                                                                                       | .341 | .368  | .333                                                                              | .40  | .434 | .394                                                                                                               | .472  | .512  | .456                                                                                                                     | .547 | .592 |
|        |                                                                                                             | Very beamy!<br>requires low<br>draft to be<br>reasonable.<br>Not realistic<br>for light V! |      |       | Beamy!<br>Medium draft<br>to be reason-<br>able.<br>Not realistic<br>for light V! |      |      | Normal!<br>Medium draft<br>required.<br>Not realistic<br>for heavy V!<br>With lowest<br>draft only<br>for light V! |       |       | Narrow beam!<br>High draft<br>required for<br>medium V!<br>Not realistic<br>for heavy V!<br>Lowest draft<br>for light V! |      |      |



# Barnaby's Powering Table (from D. Phillips-Burt)

corresponding to curves in K.C. Barnaby: Basic N.A., pp. 132-33

## B.H.P.

| Recommended<br>type of stern |      | Canoe &<br>Cruiser pos. |      | Transom and<br>flat run |      |      |     |     | Immersed transom;<br>very flat run |     |     |     |     | Chine<br>hull |  |
|------------------------------|------|-------------------------|------|-------------------------|------|------|-----|-----|------------------------------------|-----|-----|-----|-----|---------------|--|
| Length ft. Δ, tons           |      | Speed in knots          |      |                         |      |      |     |     |                                    |     |     |     |     |               |  |
|                              |      | 5                       | 6    | 7                       | 8    | 9    | 10  | 11  | 12                                 | 13  | 14  | 15  | 16  | 17            |  |
| 20                           | 0.5  | 1.0                     | 1.7  | 2.9                     | 4.7  | 7.2  | 10  | 12  | 14                                 | 17  | 19  | 22  |     |               |  |
|                              | 1.0  | 1.8                     | 3.6  | 6.6                     | 10.8 | 16   | 20  | 24  | 28                                 | 33  | 39  | 44  |     |               |  |
|                              | 1.5  | 2.6                     | 5.7  | 11                      | 17   | 24   | 30  | 36  | 43                                 | 50  | 58  | 67  |     |               |  |
|                              | 2.0  | 3.1                     | 8.0  | 15                      | 22   | 32   | 40  | 48  | 57                                 | 67  | 77  | 89  |     |               |  |
|                              | 3.0  | 3.7                     | 12.0 | 24                      | 33   | 48   | 59  | 72  | 85                                 | 100 | 116 | 134 |     |               |  |
| 25                           | 2.0  | 2.4                     | 5.0  | 10                      | 17   | 25   | 34  | 42  | 50                                 | 59  | 68  | 78  |     |               |  |
|                              | 3.0  | 3.0                     | 6.5  | 15                      | 26   | 37   | 48  | 61  | 74                                 | 88  | 102 | 115 |     |               |  |
|                              | 4.0  | 4.0                     | 8.7  | 22                      | 36   | 50   | 64  | 84  | 100                                | 117 | 136 | 155 |     |               |  |
|                              | 5.0  | 5.0                     | 12   | 28                      | 46   | 65   | 85  | 105 | 125                                | 146 | 170 | 196 |     |               |  |
| 30                           | 1.5  | 1.6                     | 2.9  | 4.9                     | 7.4  | 11   | 15  | 23  | 31                                 | 37  | 43  | 50  | 57  |               |  |
|                              | 2.0  | 1.9                     | 3.6  | 6.4                     | 10.4 | 15   | 22  | 32  | 42                                 | 50  | 58  | 67  | 76  |               |  |
|                              | 3.0  | 2.5                     | 5.0  | 9.7                     | 17   | 26   | 36  | 48  | 62                                 | 75  | 87  | 100 | 114 |               |  |
|                              | 4.0  | 3.0                     | 6.4  | 13                      | 26   | 37   | 51  | 64  | 83                                 | 100 | 116 | 133 | 152 |               |  |
|                              | 5.0  | 3.3                     | 7.7  | 16                      | 32   | 46   | 66  | 80  | 104                                | 125 | 145 | 167 | 190 |               |  |
|                              | 6.0  | 3.5                     | 8.8  | 19                      | 39   | 56   | 79  | 96  | 125                                | 150 | 174 | 200 | 227 |               |  |
|                              | 7.0  | 3.7                     | 9.9  | 22.5                    | 45   | 65   | 92  | 112 | 146                                | 175 | 203 | 234 | 265 |               |  |
|                              | 8.0  | 4.0                     | 11   | 26                      | 51   | 74   | 105 | 128 | 166                                | 200 | 232 | 267 | 303 |               |  |
| 40                           | 8.0  | 4.0                     | 8.4  | 15                      | 26   | 47   | 73  | 94  | 119                                | 145 | 168 | 193 | 219 | 248           |  |
|                              | 9.0  | 4.2                     | 9.2  | 16.5                    | 30   | 54.5 | 83  | 109 | 135                                | 163 | 190 | 218 | 247 | 279           |  |
|                              | 10.0 | 4.4                     | 9.9  | 18                      | 33   | 61   | 92  | 122 | 149                                | 180 | 210 | 242 | 274 | 310           |  |
|                              | 12.0 | 4.6                     | 11   | 21                      | 40   | 75   | 110 | 146 | 179                                | 217 | 252 | 290 | 329 | 372           |  |
|                              | 14.0 | 5.0                     | 12   | 24                      | 46   | 87   | 128 | 170 | 208                                | 253 | 294 | 338 | 384 | 434           |  |
|                              | 16.0 | 5.2                     | 13   | 27                      | 53   | 100  | 147 | 195 | 238                                | 289 | 336 | 387 | 439 | 495           |  |
|                              | 18.0 | 5.6                     | 14   | 30                      | 59   | 112  | 165 | 219 | 268                                | 325 | 378 | 435 | 494 | 558           |  |
|                              | 20.0 | 5.9                     | 15   | 33                      | 66   | 125  | 183 | 244 | 298                                | 361 | 420 | 484 | 548 | 620           |  |
| 50                           | 12.0 | 5.0                     | 8.8  | 17                      | 27   | 42   | 66  | 96  | 122                                | 164 | 199 | 243 | 283 | 309           |  |
|                              | 14.0 | 5.3                     | 9.6  | 20                      | 30   | 49   | 82  | 116 | 155                                | 198 | 243 | 286 | 330 | 360           |  |
|                              | 16.0 | 5.6                     | 10   | 11                      | 34   | 56   | 98  | 137 | 183                                | 234 | 278 | 327 | 376 | 412           |  |
|                              | 18.0 | 5.8                     | 11   | 23                      | 38   | 63   | 112 | 168 | 212                                | 270 | 313 | 368 | 423 | 463           |  |
|                              | 20.0 | 6.0                     | 12   | 25                      | 41   | 70   | 128 | 192 | 248                                | 300 | 348 | 408 | 470 | 515           |  |
|                              | 25.0 | 6.5                     | 13   | 30                      | 50   | 87   | 164 | 240 | 312                                | 375 | 435 | 510 | 588 | 643           |  |
|                              | 30.0 | 7.0                     | 14   | 34                      | 57   | 105  | 197 | 288 | 374                                | 450 | 522 | 612 | 705 | 775           |  |
|                              | 35.0 | 8.0                     | 15   | 37                      | 66   | 123  | 230 | 336 | 437                                | 525 | 609 | 715 | 823 | 900           |  |

These are mean values for hulls of correct  $\psi$  for the speed and prop. efficiencies of approx. 58%!



will show whether the draft aft contemplated will accommodate an optimum diameter propeller; it might become necessary to revise the draft drastically as a result of this. The whole balance of ratios might come loose and new main dimensions might have to be fixed.

The first powering estimate and the rough propeller calculation will also throw some light on the question of which reduction gear to adopt for the boat's engine.

It is on these first lines that such important details as freeboard, deadrise, flare of forebody sections, shape of stem and stern and rake of keel are decided upon.

While it is not the purpose of these notes to give detailed accounts of how to actually draw lines and other plans, it is nevertheless important to remind the reader of the importance of accurate draughtsmanship throughout. The success of the new vessel depends to no small extent on the clever choice of dimensions and ratios and their accurate interpretation in the drawings.

#### First checks with draft lines

Possibly the most important check to be effected with the help of draft lines and a draft general arrangement is the balancing of weights.

It becomes very important to check and recheck throughout the design whether the contemplated arrangement will result in a balanced hull or whether substantial trims will occur. To do this, it is suggested to draw up schematic balancing diagrams as shown on page 6) for a 32 ft. vessel.

A number of important assumptions have to be made here, such as fixing the position of the LCG of the hull weight, and the hull weight itself.

While the CG of a wooden hull of medium size is normally very near to its midlength (perhaps 2.5% forward or aft of it, mainly depending on the position of superstructures, and the existence of excessive overhangs), the estimation of the hull weight is not so easy for the beginner. The best chances for hitting a reasonably accurate figure are obtained when accurate hull weights for a number of built vessels in the same size group are known.

It is usually possible to assess the weights of major items of equipment correctly at this stage already.

If the balancing calculation is done with the maximum of care and accuracy, it will be seen that only minor adjustment will become necessary at a later stage.

Note Once the arrangement of main weights is fixed, the LCG position relative to various load conditions determines largely the position of LCB for the corresponding displacements.

If no equitable compromise establishes identity for the two centres, major weights will have to be shifted, the lines adjusted or ballast introduced to minimise trims.

In the worst cases a radical change of the underwater shape might become necessary.

Another check that should be made here regards Tonnage Regulations. At present there are no tonnage rules in force for fishing vessels in India, but it is likely that the future will bring the recognition of the 12 corrected Gross Tons regulation for fishing vessels operating within 7 miles of the shore. The regulations and the procedure for calculating the corrected G.T. of a new fishing vessel can be obtained from the Mercantile Marine and the resident ship surveyors of Government of India.

#### Construction Drawings

The next step is the preparation of detailed construction and arrangement drawings. It is suggested to commence this by first tabling the main scantlings



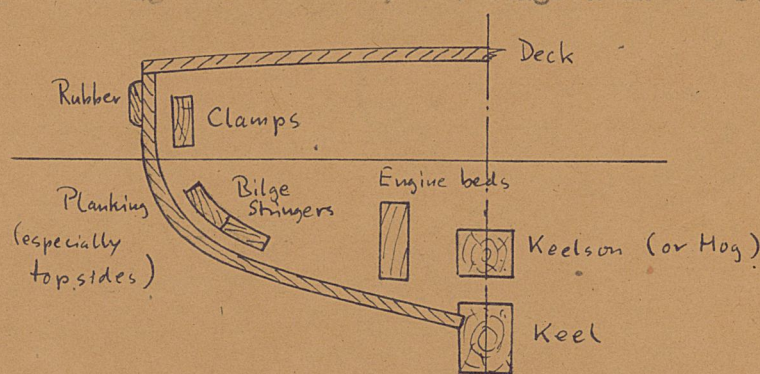
of the vessel and then drawing a large scale section or two, showing all scantlings to scale.

In the absence of any universally applicable system of computing scantlings for wooden vessels, much is left to tradition and usage when deciding on the timber sizes to be built into the vessel. A lot of information regarding scantling of wooden boats is available in Fishing Boats of the World I and II (to be issued soon), and it will be well worth to study the various tables given there. However, it must be borne in mind that almost all this information refers to timbers not commonly in use in India. The hardwoods used here make it often possible to use somewhat lighter scantlings without sacrificing strength of the vessel. Another point to be kept in mind is the fact that many of the tabulated scantlings refer to vessels operating under very severe weather conditions in the North Atlantic or North Sea. The boats built in India will not have to sustain such heavy weather, although those on the East Coast especially have to put up with fairly heavy seas at times.

A few basic points should be kept in mind when fixing scantlings and drawing construction plans:

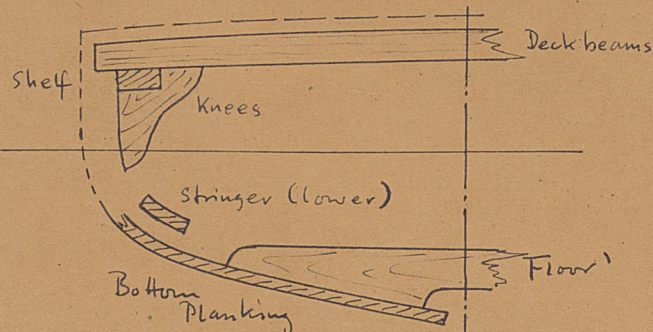
Longitudinal strength: This is given to the boat mainly by members such as: Keel, Hog, Keelson, Bilge Stringers, Clamps, Rubbing Strakes, Planking, Decking and Engine Stringers or Beds.

Deep members will give the boat greater longitudinal strength than flat members. In a typical fishing boat section, the longitudinal strength members are thus:



From the drawing it becomes obvious that members of this section are more contributory to strength than these

Transverse strength: This is given mainly by Floor Timbers, Deckbeams, Horizontal and vertical knees, Shelves, Bilge Stringers (to a certain extent), Bottom Planking and Ceiling.



In open boats a large amount of transverse strength is afforded by thwarts and cross ties under the floor boards.

Note Contrary to popular belief, frames are not to be considered strength members! They serve more as "tie rods", keeping the planking together, and helping to achieve the desired shape of the hull. A boat with heavy grown or sawn frames need not be stronger than one with steam bent frames; the opposite will often be the case!



Whether or not to use bent frames is a question answered often only by the boat building skills available. In India, where boats are often built with primitive means by unskilled carpenters, the use of sawn or grown frames has many points in its favour. This applies more particularly to bigger boats. Also it may often be possible to buy crooked timbers very cheaply, thus offering an economic advantage over bent frame construction.

However, bent frames are greatly favoured on the Malabar Coast now, while most boats built on the Konkan Coast are built over sawn frames.

Both methods are used on the Koromandel Coast and the North Circars.

It is unquestionable that a boat built on bent frames will be somewhat lighter and will also have somewhat larger inside volume than one built on sawn frames.

#### Fastenings:

Black iron fastenings should be avoided whenever possible unless they are completely sunk in the wood and effectively protected against contact with sea water.

The safest fastenings for tropical waters are no doubt those of copper and alloys like Monel. Brass, unless of very superior quality, should also be avoided as it tends to become spongy and brittle.

Hot dipped galvanised iron fastenings should be progressively used, especially inside the hull where there is little danger of direct contact with sea water. They would result in considerable savings in cost over copper rod.

Note It appears from experience that galv. iron fastenings deteriorate in Aini wood! They are very sound in Teak, however. This phenomenon is not sufficiently well studied yet to reach any conclusions as to the possible reasons.

This compilation of notes not being a compendium on boat building practices, this may suffice; it may perhaps be possible at a later date to issue further parts, dealing with detail questions of boat building practice.

It is usual to show the constructional details both in profile and plan view. The profile drawing is most conveniently arranged as a longitudinal section through the centre of the boat, while the plan may be arranged in two halves, one showing the under deck construction and the other the deck beams and carlins.

A number of sections transversely may be drawn, especially with bigger boats with complex internal layouts, to assist in the efficient planning of arrangements, cabins, engine rooms and holds.

#### Weight Calculation

After completing the construction drawing and drawing up the arrangement plan, a careful weight estimate has to be made before finalising the lines.

It is most convenient to do this in groups and in a tabulated form, to obtain both the longitudinal and vertical position of the CG of the ship.

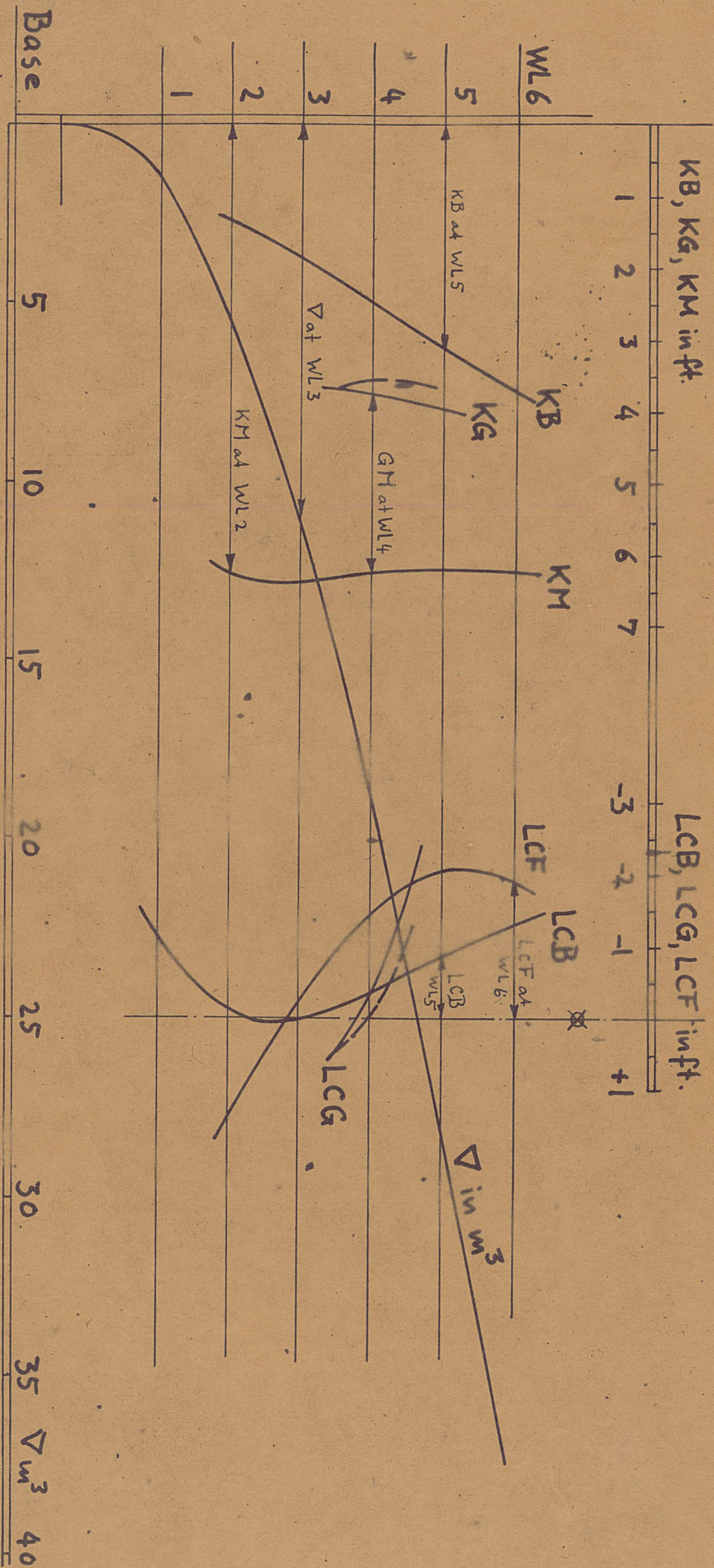
For vertical moments the baseline is a convenient reference line, while longitudinal moments are normally computed with reference to midships.

All items going into the construction, and all pieces of equipment have to be included in the weight estimate.

A convenient table for this purpose is laid out on the next page, also showing the main groups to be treated.

When all weights are grouped and summed up, it will be useful to draw up a weight summary for a number of loading conditions to assist in determining the static stability for each condition.





Hydrostatic Curves for 40ft. Trawler

— Purse Seining  
 --- Trawling



Useful conditions to investigate would be:

- I) Empty Ship: All equipment, on board, no liquids.
- II) Ready to go to sea: All tanks full, fishing gear on board, ice in hold, crew and effects and stores.
- III) Fishing: Tanks half full, hold filled to half capacity, wet gear on board.
- IV) Returning to port: Tanks  $3/4$  empty, hold  $3/4$  full, ice 25% used, wet gear on board, stores spent.
- V) Unloading: Tanks  $3/4$  empty, hold empty, 20% ice capacity on deck, wet gear hanging or on deck.

Any condition constituting a particular operational instance for the ship may be treated like this. The summary presentation will give a good idea of the shifts in LCG and VCG to be expected, reflecting on changes in trim and giving a clue to the stability properties to be met.

In this condition it must be well remembered again that any condition treated represents one, only one, displacement and a set of hydrostatic particulars, such as LCB, KM etc.

### Final Lines

Final lines are prepared when the designer is satisfied that the ship will behave according to his predictions. An effort must be made to obtain centres corresponding to the ones required by the various weight summaries. Should this not be possible in the first instance, it must be seen whether minor changes in the lines will bring the necessary agreement.

Small changes in fullness either forward or aft, change of keel rake, change of position of maximum area section can have considerable influence on the hydrostatic properties of the ship.

These changes do not normally affect CM, unless they are large, or if the waterline area is substantially changed locally.

When the lines are finalised, it is good practice to draw up a set of hydrostatic curves immediately for ready reference. These may comprise a displacement curve, LCB, KB and BM curves for small boats, and such items as  $\Delta$ , BM and MTI curves for larger vessels. Trims and CM values for any weight condition can easily be checked with these curves.

Note Whenever a boat is built, the hydrostatic curves should be corrected and completed by inserting the values obtained from full scale trials and tests. Especially inclusion of actual CG positions will be very helpful in future designs of similar ships.

Sample hydrostatic curves for a small vessel appear on page 70).

The next important step is to prepare an accurate table of offsets for the boat. The offsets should fix all points in the lines plan vertically and transversally, while measurements on the lines plan, or a special measurement drawing should fix them longitudinally. All these measurements are very important, as the builder uses them to prepare his moldloft drawing of the boat in scale 1:1. Suitable reference lines for the offsets are the baseline or DWL for heights and the Centre Line for half breadths. All measurements such as spacing of sections and waterlines, overall length, waterline length, max. beam, length of overhangs, spacing of buttocks, fixation of diagonals etc. should be available from one of the drawings. A sample offset table for a 25 ft. boat appears on page 71).



Table for Weight Calculation

| Item | Spec.w. | Volume | Weight | cg+ | Mt. f | cg- | Mt. a | cg up | Mt. up |
|------|---------|--------|--------|-----|-------|-----|-------|-------|--------|
|      |         |        |        |     |       |     |       |       |        |

Convenient groups for wooden vessels:

Keel, Stem, Stern assembly

Framing, Planking, Bulwarks, Ceilings, Floorings

Beams, Decking

Floors

Stringers, Clamps, Shelves

Engine Stringers, Beds

Bulkheads

Knees, Breasthooks, Carlins, Coamings, Skylights, Hatches

Deckhouses

Cabin interiors, Galley, Wheelhouse equipment, Tanks

Hold insulation

Engine Room: Main Engine, Auxiliaries, Exhausts, Spares, Tools, Shaft, Propeller

Rudder Assembly

Deck Machinery

Rigging

Fishing Gear

Ice

Fuel, Fresh Water

Catch

Stores, Crew effects, Crew



# Offsets for 25ft. Fishing Boat

| Sections                | T         | AP   | 1/2  | 1    | 1 1/2 | 2        | 3    | 4    | 5        | 6    | 7    | 8    | 9    | 9 1/2 | FP   |
|-------------------------|-----------|------|------|------|-------|----------|------|------|----------|------|------|------|------|-------|------|
| Half Breadths           | WL 1      |      |      |      |       | 3.5      | 8.1  | 11.2 | 8.6      | 4.0  |      |      |      |       |      |
|                         | 2         |      |      |      | 4.0   | 10.1     | 22.6 | 28.0 | 26.1     | 20.3 | 14.6 | 8.1  | 3.0  |       |      |
|                         | 3         |      |      | 7.0  | 16.5  | 25.5     | 35.7 | 38.6 | 37.0     | 29.7 | 25.1 | 16.4 | 7.4  | 3.2   |      |
|                         | 4         |      |      |      |       |          |      |      |          |      |      |      |      |       |      |
|                         | 5         | 24.3 | 25.4 | 30.3 | 34.3  | 36.4     | 36.0 | 40.3 | 42.0     | 40.6 | 37.0 | 30.4 | 21.5 | 11.4  | 6.0  |
|                         | 6         | 29.0 | 30.2 | 33.5 | 36.5  | 38.1     | 39.7 | 41.7 | 42.5     | 42.1 | 39.2 | 33.4 | 25.2 | 15.0  | 8.6  |
|                         | 7         |      |      |      |       |          |      |      | 42.5     | 42.0 | 39.5 | 34.0 | 24.3 | 16.7  | 7.1  |
| Heights above Base Line | Sheer     | 33.0 | 34.5 | 36.6 | 38.5  | 40.1     | 41.2 | 42.3 | 43.0     | 43.2 | 42.4 | 40.5 | 36.6 | 29.4  | 23.3 |
|                         | Keel      |      | 0.0  | →    |       | Straight | line | →    | St. 8    | →    | 11.0 | 13.5 | 17.4 | 30.0  |      |
|                         | Rabbit    | 30.4 | 30.3 | 26.4 | 21.2  | 11.0     | 5.4  | →    | Straight | line | →    | 14.0 | 16.5 | 20.7  | 36.0 |
|                         | Buttock I | 34.3 | 34.0 | 31.0 | 28.1  | 25.2     | 22.2 | 17.1 | 15.2     | 15.7 | 18.1 | 21.0 | 29.0 | 47.0  | 63.0 |
|                         | II        | 32.2 | 31.7 | 28.4 | 25.1  | 21.5     | 18.1 | 13.2 | 11.6     | 12.5 | 14.1 | 16.0 | 19.3 | 28.2  | 39.7 |
|                         | Sheer     | 58.0 | 56.7 | 55.0 | 53.3  | 52.3     | 51.3 | 50.5 | 51.1     | 52.5 | 54.7 | 57.4 | 61.2 | 65.5  | 70.5 |

Offsets in inches and eighths.



Note Offset tables should be corrected when the first mold loft drawing has been prepared to take care of any small errors that might have crept in!

### Sundry Drawings

To complete the design of a new vessel, it is good practice to make a detail drawing of the keel, stem and stern assembly for the guidance of the builder. This drawing can often conveniently be combined with the above mentioned measurement drawing and the offset table.

To give a better impression of the boat to laymen, it is desirable to draw a small scale profile and deck plan drawing, giving all the details of deck gear and rigging.

### Specifications

These are an important part of any new design as they form the base for costing calculations and contracts.

They should contain a complete list of scantlings with all information that can be given regarding timbers to use, fastenings etc.

They should contain a description of the constructional details of the vessel, as well as giving full details regarding all equipment to be built in.

It is especially important to specify clearly what equipment will be supplied to the builder by the owner and what will have to be supplied by the builder.

Specifications and drawings are complementary - however, it is better to show things both in drawings and specifications than not at all.

### Conclusion

The main parts of these notes are Part I and II, dealing with the theories of Naval Architecture.

Part III is in no way complete as a guide to design work, but if followed with common sense and if forming the bases of careful design work, it should prove to be helpful for each reader to establish a certain personal design procedure. It is hoped that a more complete treatise on practical drawing work and boat building practices can follow this at a later date.

It must be stressed again here that unless otherwise stated, all graphs and curves contained in these notes are merely to illustrate trends rather than form the bases for calculations on individual designs.

Much complementary information on all subjects treated here can be found in the books listed in the publications list distributed during the course to each participant.



