SHIP CONSTRUCTION

AND

CALCULATIONS.

WITH

NUMEROUS ILLUSTRATIONS AND EXAMPLES.

FOR THE USE OF OFFICERS OF THE MERCANTILE MARINE, SHIP SUPERINTENDENTS, DRAUGHTSMEN, ETC.

BY

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PREFACE.

THE rapid advances that have been made in recent years both in design and in the details of construction of steel ships is the writer's best apology for placing the present volume before the public. The book is intended to explain in a simple and practical manner some of the problems met with in the building and subsequent management afloat of ships, particularly cargo steamships, and while no claim to special originality is made, it is hoped that the matter presented will be found up to date. It may be mentioned that publication has been specially delayed so as to include reference to Lloyd's latest rules, which differ in certain important respects from those preceding them, and are "more readily applicable to the changing conditions of construction."

It is hoped the book will be found useful by officers of the Mercantile Marine, ship superintendents, draughtsmen, and shipyard apprentices, to all of whom a more or less intimate knowledge of naval architecture is essential. To the officer mariner the subject may now be said to be a compulsory one, in that those who wish to qualify for the certificate of extra master must pass an examination in it. But besides this, there are other good reasons why an officer should know something regarding the construction and theory of ships. For instance, it would enable him, if called upon, to act as inspector on behalf of his employers at the building of a new vessel or the repair of an old one. Or, if his vessel were to receive sudden damage, calling for immediate temporary repairs, it would give him confidence in directing his crew in the carrying out of these. In the management of his vessel afloat, a knowledge of simple theory would assist him at any time to arrive quickly at satisfactory conditions of draught, trim and stability, unattainable by mere guess-work or a system of trial and error. In other ways also such knowledge would prove useful.

The examples chosen for illustration throughout the book have been selected for their practical interest, and every effort has been exerted to make the explanations simple.

In conclusion, the author begs to thank Messrs. J. L. Thompson & Sons, Ltd., Sunderland, for their kind permission to publish diagrams and results of calculations of vessels built by them; and he also desires to acknowledge his indebtedness to Mr. W. Thompson, B.Sc., for help in reading the proof sheets, and in verifying the examples, as well as for other valuable assistance rendered while the work was passing through the press.

GLASGOW, *November*, 1909.

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SHIP CONSTRUCTION AND CALCULATIONS.

CHAPTER I.

Simple Ship Calculations.

A KNOWLEDGE of the principle of moments and of how to calculate areas of surfaces having curved boundaries may perhaps be said to be the only indispensable requisites in dealing with ordinary ship calculations.

In view of this we propose to spend a little time on these subjects, first taking up areas of surfaces, and afterwards, as may be found convenient, the principle of moments, especially as applied to ship problems.

The area of a plane surface bounded by straight or curved lines may be defined as the number of units of surface contained within its boundaries. The unit, in English measure, is usually a square foot, although it is sometimes taken as a square inch, and sometimes, although more rarely, as a square yard. In France, and on the Continent generally, the metrical system is employed, the units of surface being the square metre and square centimetre, respectively. These metrical units have many points of advantage, but as the square foot is more familiar to us, we shall make it the standard in our calculations.

The simplest figure of which we may obtain the area is a square, whose chief properties are—all sides equal, and all angles right angles. In fig. 1, $ABCD$ is a square, the length of one side being, say, 6 feet. If two adjacent sides, such as AB and AD, be divided into 6 equal parts, and lines be drawn through the points of division parallel to these sides, as shown in the figure, the square will contain 36 small squares, each of which has its 4 sides equal to a foot, and encloses one unit of area. There are, therefore, 36 units in a square having a side of 6 feet. It is obvious that to find the number of units in any square it is only necessary to multiply the length in lineal units of one side by itself.

In passing to a rectangle the rule for the area is the same as for a square, *i.e.*, the length of two adjacent sides are multiplied together, these being, however, in this case, unequal. As an example, let the adjacent sides in a rectangle be 16 feet and 8 feet long respectively. By the rule, we have—

$$\text{Area} = 16 \times 8 = 128 \text{ square feet.}$$

Fig. 1.

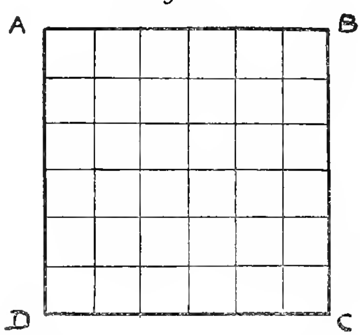
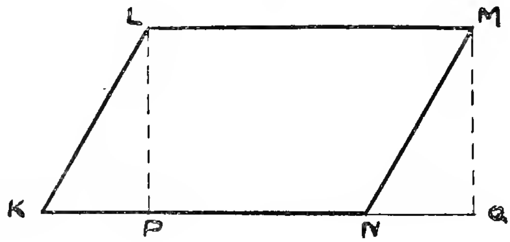
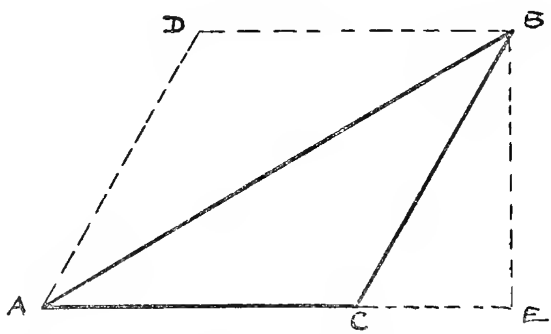


Fig. 2.



To find the area of a rhomboid, which is a figure having its opposite sides and angles equal, but none of the angles right angles, a modification of the previous rule is used. $KLMN$ (fig. 2) is such a figure. From L and M drop perpendiculars on KN, or KN produced, as shown. It is easy to prove that the rectangle $LPQM$ = rhomboid $LKNM$, since obviously the triangles LKP and MNQ are equal in area. But the area $LPQM$ =

Fig. 3.



$LM \times LP$, from which it follows that to obtain the area of a rhomboid the length of a side should be multiplied by the perpendicular distance between it and the one opposite.

This rule is specially useful in explaining the method of obtaining the area of a triangle. Let ABC (fig. 3), be any triangle. Complete the parallelogram $ACBD$, and from B drop a perpendicular on AC produced.

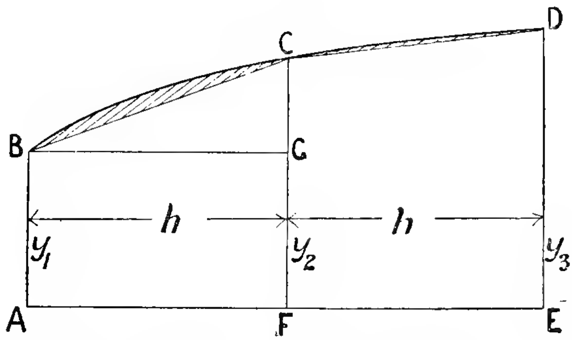
Obviously $A\,B$ bisects the parallelogram $A\,C\,B\,D$, and, as we have just seen

$$\text{Area } A\,C\,B\,D = A\,C \times B\,E,$$

Therefore, the area of the triangle $A\,B\,C$ equals $\frac{A\,C \times B\,E}{2}$. By this rule the area of any triangle may be obtained; and it is seen that we only require to know the length of one side and the vertical distance between that side and the point in which the other two sides intersect. Thus, a triangle having a base of 25 feet, and a vertical height of 22 feet, will have an area of $\frac{25 \times 22}{2} = 275$ square feet. The area of any plane figure bounded by straight lines may be found by one of the foregoing rules, or by a combination of them, and it should be noted that the earliest rules applied to the finding of the areas of ship waterplanes and sections were of this nature.

Let $A\,B\,C\,D\,E$ (fig. 4) be a portion of a ship's waterplane. Bisect $A\,E$ in F, and through F draw a line perpendicular to $A\,E$ to intersect the curve in C. $F\,C$ will be parallel to $A\,B$ and DE. Join $B\,C$ and $C\,D$ by straight lines, then $A\,B\,C\,F$ and $F\,C\,D\,E$ will be trapezoids. $A\,B$, $F\,C$, $E\,D$ are called ordinates to the curve: let these lines be represented by the letters y_1 y_2 and y_3, respectively; and let h be the common distance between consecutive ordinates. Obtain now the areas of the trapezoids $A\,B\,C\,F$ and $F\,C\,D\,E$ by ap-

Fig. 4.



plying the rules already established. Draw $B\,G$ parallel to $A\,F$, meeting $C\,F$ in G, then—

$$\text{Area } B\,C\,G = \frac{B\,G \times G\,C}{2}, \text{ and area } A\,B\,G\,F = A\,F \times A\,B.$$

Using the symbols, these may be written—

$$\text{Area } B\,C\,G = \frac{y_2 - y_1}{2} \times h, \text{ and area } A\,B\,G\,F = h \times y_1.$$

Combining we get—

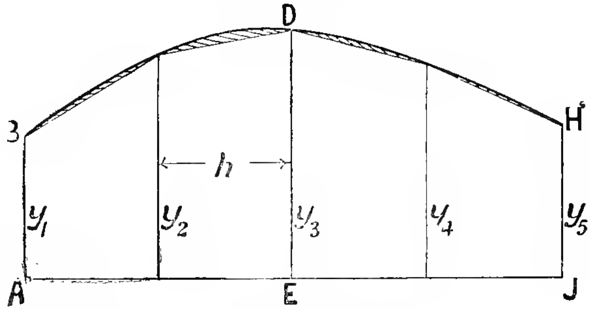
$$\text{Area } A\,B\,C\,F = \frac{h}{2}(y_1 + y_2).$$

$$\text{In the same way area } FCDE = \frac{h}{2}(y_2 + y_3)$$

$$\therefore \text{ whole area } A\,B\,C\,D\,E = \frac{h}{2}(y_1 + 2y_2 + y_3).$$

The rule may be applied to curves with any number of ordinates. For example, take one having five as in fig. 5.

Fig. 5.



$$\text{By the rule, area } ABDE = \frac{h}{2}(y_1 + 2y_2 + y_3)$$

$$\text{and, area } EDHJ = \frac{h}{2}(y_3 + 2y_4 + y_5)$$

$$\therefore \text{ the whole area, } ABDHJ = \frac{h}{2}(y_1 + 2y_2 + 2y_3 + 2y_4 + y_5)$$

$$= h\,(\tfrac{1}{2}y_1 + y_2 + y_3 + y_4 + \tfrac{1}{2}y_5).$$

This is called the trapezoidal rule for obtaining areas of surfaces bounded by curved and straight lines. It may be stated as follows:—To obtain the area of any plane surface, bounded by three straight lines and a curved line, two of the straight lines being perpendicular to the other, which is taken as the base of the figure—divide the base between the end ordinates into any number of equal parts, and through the points of division draw perpendiculars to the curved line, as in figure 5; measure the length of these ordinates, taking one foot or one inch as the unit of measurement; then to the sum of all the ordinates, except the end ordinates, add half the sum of the end ordinates; the result, multiplied by the normal distance between any two ordinates, measured in same unit, will be the area of the surface, approximately.

Example.—Let length of base = 48 feet. Let there be 5 ordinates, spaced as directed, giving a common interval of 12 feet, and let the value of the ordinates be in feet—$y_1 = 3$; $y_2 = 10$; $y_3 = 16$; $y_4 = 12$; $y_5 = 4$, respectively.

Tabulating the information, the calculation becomes—

Ordinates.	Multipliers.	Function of Ordinates.
3	$\frac{1}{2}$	1·5
10	1	10·0
16	1	16·0
12	1	12·0
4	$\frac{1}{2}$	2·0
		41·5

Area approximately = 41·5 × 12 = 498 square feet.

The area obtained as above is less than the actual area required by the small areas enclosed between the straight lines joining the extremities of each two consecutive ordinates and the curve, as indicated by the hatched spaces in fig. 5. It is clear that by taking a great number of ordinates, these hatched areas may be made extremely small, but however numerous the ordinates may be, the area obtained in this manner is always less than the actual area. It is obvious, too, that the difference is greatest when the curvature is excessive; this rule, therefore, will give most accurate results when applied to surfaces having boundaries with comparatively little curvature. In applying the rule to ordinary ship curves the ordinates should be close spaced at the points of greatest curvature, and wider spaced elsewhere.

If the curves met with in ship design were of regular form, equations to them could be deduced by means of which the correct areas of surfaces enclosed by them up to any point could be written down. Unfortunately, this is not so. No rigid equation can be applied to ordinary ship curves, but it is found that no great error is usually involved in treating them as parabolas,* and this is now the common practice.

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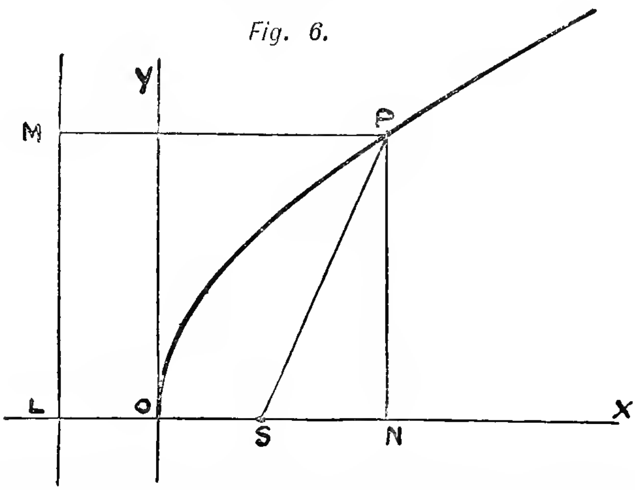
* A parabola may be described as the curve forming the line of intersection of a right cone with a plane parallel to one of its sides; it is also sometimes defined as the locus of a point which moves, so that—

$$\frac{\text{its distance from a fixed point}}{\text{its distance from a fixed straight line}} = 1.$$

Fig. 6.



The fixed point is called the focus and the fixed straight line the directrix. In fig. 6, OP is the parabola, OX and OY the co-ordinate axes, ML the directrix which is parallel to OY, and S the focus, or fixed point. If, now, any point P in the curve be taken, we have— $\frac{SP}{MP} = 1$, MP being perpendicular to the directrix. PN, the ordinate to any point P, may be expressed in terms of the abscissa and a constant, thus—

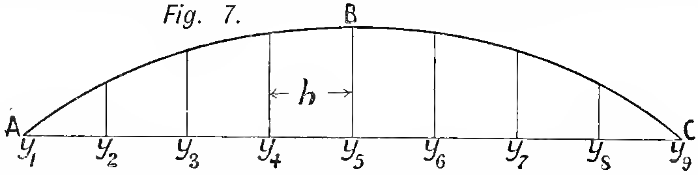
$$PN^2 = C \times ON.$$

If the value of C be varied, other parabolas will be obtained also passing through O.

By an important formula, known as Simpson's Rule, named after the inventor, the area enclosed by any parabolic curve may be obtained. The rule may be stated as follows:—SIMPSON'S FIRST RULE.—Through the extremities of any chosen base-line, draw verticals to cut the curve. Divide the distance between these verticals into an even number of spaces, and through the points of division draw ordinates to the curve; which ordinates will therefore be odd in number. Measure the length of each ordinate, and then add together four times the sum of the even ordinates and twice the sum of the odd ordinates, omitting the first and last. To this total add the sum of the first and last ordinates. Finally, multiply the result by one-third the length of the common interval between the ordinates; this will give the exact area of the surface if the curve be that of a common parabola, and a close approximation to it if the curve be an ordinary ship-shape one.

This rule is of immense value in ship calculations, and we shall proceed to take a few examples showing its application. Let $A\,B\,C$ (fig. 7), be a ship's half-waterplane of which the area is required. The base $A\,C$ is divided as indicated in the rule, and the ordinates y_1 y_2 y_3, etc., are drawn

Fig. 7.

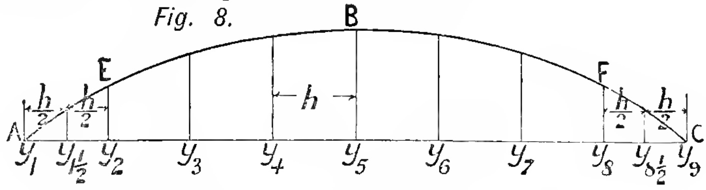


through the points of division; h is the common interval between the ordinates. We may write—

$$\text{Area } A\,B\,C = \frac{h}{3}(y_1 + 4y_2 + 2y_3 + 4y_4 + 2y_5 + 4y_6 + 2y_7 + 4y_8 + y_9).$$

It frequently happens that the curvature is greater at the ends of the waterplane, and a closer approximation to the area is attained by inserting intermediate ordinates at these places, as shown in fig. 8. The total area is now made up of three portions, as follows:—

Fig. 8.



$$\text{Area } A\,E\,D = \frac{h}{6}(y_1 + 4y_{1\frac{1}{2}} + y_2) = \frac{h}{3}(\tfrac{1}{2}y_1 + 2y_{1\frac{1}{2}} + \tfrac{1}{2}y_2).$$

$$\text{Area } D\,E\,F\,G = \frac{h}{3}(y_2 + 4y_3 + 2y_4 + 4y_5 + 2y_6 + 4y_7 + y_8).$$

$$\text{Area } F\,G\,C = \frac{h}{6}(y_8 + 4y_{8\frac{1}{2}} + y_9) = \frac{h}{3}(\tfrac{1}{2}y_8 + 2y_{8\frac{1}{2}} + \tfrac{1}{2}y_9).$$

Combining these portions, we get—

$$\text{Area } A\,B\,C = \frac{h}{3}(\tfrac{1}{2}y_1 + 2y_{1\frac{1}{2}} + 1\tfrac{1}{2}y_2 + 4y_3 + 2y_4 + 4y_5 + 2y_6 + 4y_7 + 1\tfrac{1}{2}y_8 + 2y_9 + \tfrac{1}{2}y_9). \quad (1)$$

Suppose now that the ordinates have certain definite values beginning with y_1, as follows:—·1, 5, 11·6, 15·4, 16·8, 17·0, 16·9, 16·4, 14·5, 9·4, ·1, and that the total length of the plane is 200 feet: we could find the area by filling in the values in equation (1), but it is more convenient to tabulate the figures, as follows:—

Nos. of Ordinates.	½ Ordinates.	S.M.	Function of Ordinates.
1	·1	½	·05
1½	5·0	2	10·00
2	11·6	1½	17·4
3	15·4	4	61·6
4	16·8	2	33·6
5	17·0	4	68·0
6	16·9	2	33·8
7	16·4	4	65·6
8	14·5	1½	21·75
8½	9·4	2	18·8
9	·1	½	·05
			330·65

$$\text{Area } ABC = 330{\cdot}65 \times \frac{25}{3} = 2755{\cdot}4 \text{ square feet.}$$

This result, of course, must be multiplied by 2 for the area of the complete waterplane.

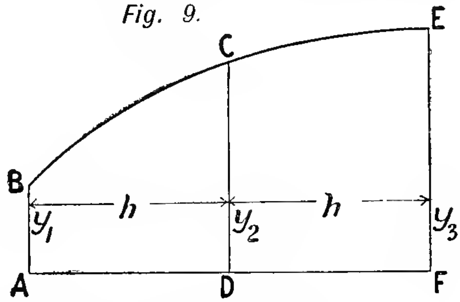
As showing the application of the rule to transverse sections, take the following:—The half ordinates in feet of the midship section of a vessel are—12·5, 12·8, 13, 13, 12·8, 12·4, 11·8, 10·4, 6·8 and ·5 respectively; the common interval is 2 feet; between the two bottom ordinates an ordinate at half interval is taken, its value being 4 feet; find the area of complete section. Arranging the work as before, the calculation becomes—

Nos. of Ordinates.	½ Ordinates.	S.M.	Function of Ordinates.
1	·5	½	·25
1½	4·0	2	8·0
2	6·8	1½	10·2
3	10·4	4	41·6
4	11·8	2	23·6
5	12·4	4	49·6
6	12·8	2	25·6
7	13.0	4	52·0
8	13·0	2	26·0
9	12·8	4	51·2
10	12·5	1	12·5
			300·55

$$\text{Whole area} = 300{\cdot}55 \times \frac{2}{3} \times 2$$
$$= 400{\cdot}72 \text{ square feet.}$$

We have seen that to apply Simpson's First Rule to finding the area of any surface having a curved boundary, there must be an odd number of ordinates, and not less than three. It is, however, sometimes necessary to find the area between two consecutive ordinates, as, for instance, between y_1 and y_2 in fig. 9. To do this we employ another rule known as the

Fig. 9.



Five Eight Rule, which may be stated thus:—

Five Eight Rule.—Three ordinates being given, to obtain the area between any two multiply the middle ordinate by 8, the ordinate forming the other boundary to the space whose area we are finding by 5, and the remaining ordinate by -1; the algebraic sum of these products, when multiplied by $\frac{1}{12}$ the common interval between the ordinates, will give the area required. For example—

$$\text{Area } ABCD \text{ (fig. 9)} = \frac{h}{12}(5y_1 + 8y_2 - y_3)$$

$$\text{and area } DCEF = \frac{h}{12}(5y_3 + 8y_2 - y_1).$$

If these be added we get, after re-arrangement—

$$\text{Total area } ABEF = \frac{h}{3}(y_1 + 4y_2 + y_3),$$

which shows that the Five Eight Rule is based on the same assumption as the first rule, namely, that the curve is that of a common parabola.

Take a practical example.—The length of the half ordinates of a portion of a ship's waterplane are in feet, 6, 7·6 and 8, respectively, the common interval being 9 feet. Find the area of the portion of the full waterplane between the first and second ordinates. Tabulating, we get—

½ Ordinates.	S.M.	Function of Ordinates.
6·0	5	30·0
7·6	8	60·8
8·0	-1	-8·0
		82·8

$$\text{Area} = 82{\cdot}8 \times \frac{9}{12} \times 2 = 124{\cdot}2 \text{ square feet.}$$

If the area between the second and third ordinates were required, the calculation would be—

½ Ordinates.	S.M.	Function of Ordinates.
6·0	·1	6·0
7·6	8	60.8
8·0	5	40·0
		94·8

$$\text{Area} = 94{\cdot}8 \times \frac{9}{12} \times 2 = 142{\cdot}2 \text{ square feet.}$$

Besides the First Rule, which requires an odd number of ordinates for its application, Simpson introduced another one specially adapted for figures, having 4, 7, 10, 13, etc., ordinates, the number of ordinates to which the rule applies being given by a general expression. Obviously, this rule will apply in cases where the first rule would fail, and therein lies its importance. The statement of the rule is as follows:—

Simpson's Second Rule.—Choose any base line in the surface, and, through its extremities, draw ordinates to the curve. Divide the space between these limits into equal parts so as to obtain a number of ordinates as given by the general expression $(3n+4)$, where n is zero, or any positive number; multiply the end ordinates by unity, the 2nd, 3rd, 5th, 6th, 8th, 9th, etc., by 3, and the 4th, 7th, 10th, etc., by 2. Add these results together and multiply the result by $\frac{3}{8}$ the common interval between the ordinates. The quantity thus obtained will be the area of the surface within the given boundaries very nearly for ordinary ship curves.

Practical Example.—Find the area in square feet of a portion of a water-plane whose half ordinates are, 2, 6·5, 9·3, 10·7, 11, 11, 10, 7·4, 3·6, and ·2 feet, respectively, the common interval between them being 14 feet.

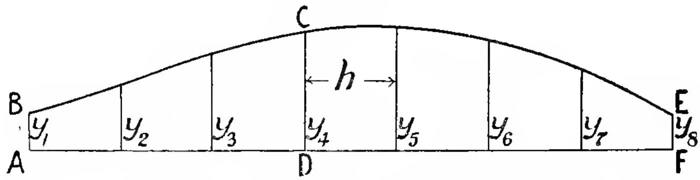
Arranging the figures as in previous examples, the calculation takes the form—

No. of Ordinates.	½ Ordinates.	S.M.	Function of Ordinates.
1	2·0	1	2·0
2	6·5	3	19·5
3	9·3	3	27·9
4	10·7	2	21·4
5	11·0	3	33·0
6	11·0	3	33·0
7	10.0	2	20·0
8	7·4	3	22·2
9	3·6	3	10·8
10	·2	1	·2
			190·0

Area of full-plane between the given end ordinates $= 190 \times 14 \times \frac{3}{8} \times 2 = 1995$ square feet.

Reviewing our rules for areas, we now find that surfaces can be dealt with having 3, 5, 7, 9, 11, 13, etc., ordinates, as required by the First Rule, and 4, 7, 10, 13, 16, etc., ordinates as required by the Second Rule; for surfaces having ordinates whose number is not included in the foregoing, such as those with 6, 8, 12, 14, etc., ordinates, no single rule will apply, and a combination of the rules given must be resorted to. Take, for instance, a surface with eight ordinates as shown in figure 10:—

Fig. 10.



Examining the figure, we note that the portion *A B C D* may be treated by the second rule, and the remainder *D C E F* by the first. Proceeding thus, we get—

$$\text{Area } A\,B\,C\,D = \tfrac{3}{8}h(y_1 + 3y_2 + 3y_3 + y_4)$$

$$\text{and area } D\,C\,E\,F = \frac{h}{3}(y_4 + 4y_5 + 2y_6 + 4y_7 + y_8).$$

Combining these quantities and re-arranging so as to get a common factor outside the bracket, we have—

$$\text{Whole area } A\,B\,E\,F = \frac{h}{3}(1\tfrac{1}{8}y_1 + 3\tfrac{3}{8}y_2 + 3\tfrac{3}{8}y_3 + 2\tfrac{1}{8}y_4 + 4y_5 + 2y_6 + 4y_7 + y_8).$$

Calculate the area of fig. 10, assuming the ordinates to be 12 feet apart, and of the following lengths: 16·7, 24·4, 28·9, 30·3, 29·9, 27·3, 22·3, and 14·9 feet, respectively. The work will be as follows:—

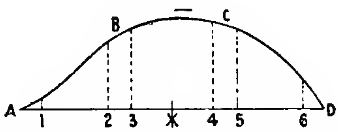
No. of Ordinates.	Ordinates.	S.M.	Function of Ordinates.
1	16·7	$1\frac{1}{8}$	18·8
2	24·4	$3\frac{3}{8}$	82·35
3	28·9	$3\frac{3}{8}$	97·53
4	30·3	$2\frac{1}{8}$	64·39
5	29.9	4	119·6
6	27·3	2	54·6
7	22·3	4	89·2
8	14·9	1	14·9
			541·37

$$\text{Area } A\,B\,E\,F = \frac{541{\cdot}37 \times 12 \times 3}{8} = 2436{\cdot}12 \text{ square feet.}$$

It should be noted that the area of the above surface could have been obtained by combining the First Rule with the Five Eight Rule. The method would be very similar to the above, and we leave the reader to work it out for himself.

TCHEBYCHEFF'S RULE.—This method of finding the areas of plane surfaces having curved boundaries differs in certain important respects from that of Simpson's Rule. The ordinates, for instance, are not equally spaced, as in the latter case, but arbitrarily, according to the number of them employed; nor are they treated by multipliers. All that it is necessary to do to obtain the area in a given case when the ordinates have been placed in position, is to measure the lengths of the latter, add these together, divide the sum by the number of the ordinates, and multiply the result by the length of the base line of the figure.

Fig. 11.



As an example, let us find the area of $ABCD$ (fig. 11). Here we have six ordinates spaced according to rule, and numbered 1, 2, 3, etc., in the sketch; the total length of the base is 20 feet. Applying the rule, we have—

No. of Ordinates.	Ordinates.
1	0·77
2	4·40
3	5·33
4	5·70
5	5·25
6	1·83
	23·28

$$\text{Area of } ABCD = \frac{23{\cdot}28 \times 20}{6} = 77{\cdot}6 \text{ square feet.}$$

So far, the application of the method appears to be simplicity itself; but a little trouble is introduced in the spacing of the ordinates. This is done from the middle of the base line as a starting point, and symmetrically to right and left, the distances being in accordance with the figures obtained when the half length of the base is multiplied by certain fractions given in Table 1.

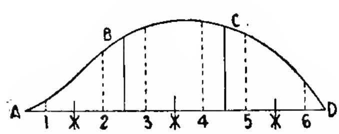
In obtaining the spacing of the ordinates in the practical example above, the half length of the base, or 10 feet, was multiplied by ·2666, ·4222, and ·8662, giving as positions to right and left of the point X, the distances in feet, 2·666, 4·222 and 8·662; and in the same way, by using the corresponding multipliers, the positions of any other number of ordinates could be determined. It should be noted that where there is an odd number of ordinates, one occurs at the origin, that is, the middle point of the base line.

TABLE 1.

Number of Ordinates and Multipliers for Same.						
2	3	4	5	6	7	9
·5773 ·5773	·7071 ·0000 ·7071	·7947 ·1876 ·1876 ·7947	·8325 ·3745 ·0000 ·3745 ·8325	·8662 ·4225 ·2666 ·2666 ·4225 ·8662	·8839 ·5297 ·3239 ·0000 ·3239 ·5297 .8839	.9116 ·6010 ·5288 ·1679 ·0000 ·1679 ·5288 ·6010 ·9116

Of course, where a plane is of extended length, it may be necessary to have more ordinates than is provided by any of the columns of Table 1, in which case the area might be obtained by repeatedly applying any of the rules. For instance, if 18 ordinates were required, the two-ordinate rule might be applied nine times, the three-ordinate rule six times, the six-ordinate three times, and the nine-ordinate rule twice. This adaptability of the system has caused some calculators to use only the two-ordinate or three-ordinate rules, repeatedly applying them as necessary, much in the same way as with Simpson's 1st and 2nd rules, which, of course, apply initially to figures with three and four ordinates, respectively.

Fig. 12.



As an illustration of the foregoing, we have again taken the figure *A B C D*, whose area has been found by directly applying the six-ordinate rule, have divided it into three parts, and have obtained the area by means of the two-ordinate rule. Fig. 12 shows the ordinates in their new spacing, and numbered from left to right. The calculation may be arranged as follows:—

No. of Ordinates.	Ordinates.
1	0·77
2	4·04
3	5·75
4	5·87
5	4·98
6	1·83
	23·24

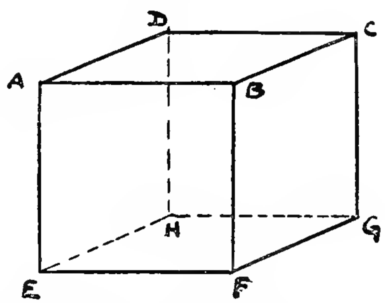
$$\text{Area } ABCD = \frac{23{\cdot}24 \times 20}{6} = 77{\cdot}46 \text{ square feet,}$$

which is seen to be very nearly that by previous rule.

It may be mentioned that the area of this figure by Simpson's First Rule with seven ordinates is 77·03 square feet. Tchebycheff's method is said to give as accurate results as Simpson's, and with a less number of ordinates. It has not been found, however, of such universal usefulness in ship calculations as Simpson's, and, for this reason, the older method is more generally employed.

VOLUME.—We have seen that the common English units of area are:—a square inch, a square foot, and a square yard; the corresponding units of volume are, a cubic inch, a cubic foot, and a cubic yard. While area is a measure of surface, and therefore deals with two dimensions, volume is a measure of space, and has to do with three. In fig. 13, the top surface

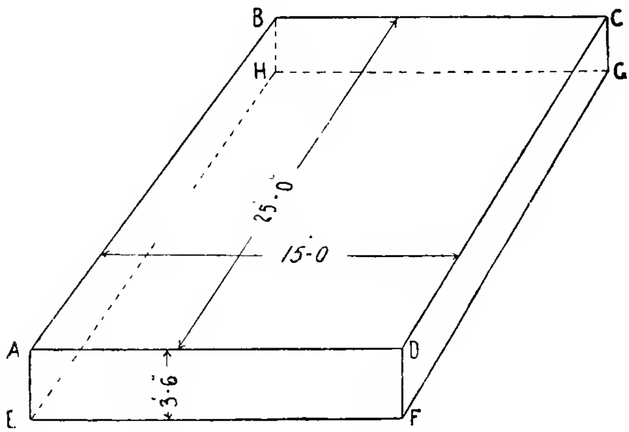
Fig. 13.



A B C D is assumed to represent an area of one square foot; the block is one foot thick, therefore the whole figure *A B C G F E A* represents the new unit, viz.:—a cubic foot. In finding the quantity of space or volume that any object occupies, we merely estimate how many times a standard unit volume, such as a cubic foot, is contained in it. For example, state in cubic feet the volume of a rectangular block 25 feet long, 15 feet broad, and $3\frac{1}{2}$

feet thick. Let fig. 14 represent this block. The area of the upper surface $ABCD = 15 \times 25 = 375$ square feet. If the block were one foot thick, 375

Fig. 14.



would also measure the capacity in cubic feet; the actual volume will obviously be $3\frac{1}{2}$ times this quantity, therefore:—

$$\text{the volume} = 375 \times 3{\cdot}5 = 1312{\cdot}5 \text{ cubic feet.}$$

From this it appears that to obtain the volume of any rectangular solid, such as that in fig. 14, it is merely necessary to find the continued product of the three principal dimensions. There are various rules for obtaining the volumes of regular solids, and we proceed to state a few of them without, however, giving the proofs; these may be obtained by referring to any work on mensuration.

1. Volume of a pyramid with any form of base = area of base $\times \frac{1}{3}$ height (perpendicular).

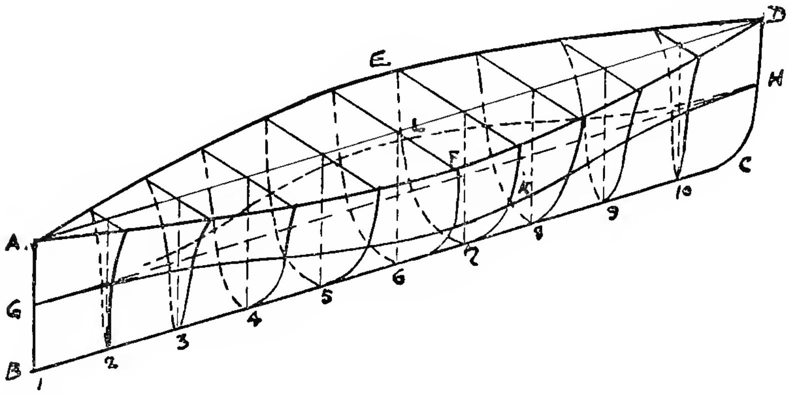
2. Volume of sphere = diameter$^3 \times \frac{3{\cdot}1416}{6}$.

3. Volume of an Ellipsoid = length $\times$ breadth $\times$ depth $\times \frac{3{\cdot}1416}{6}$.

Passing from these, we come next to consider methods of finding the volumes of solids of more or less irregular form, such, for example, as the immersed body of a ship.

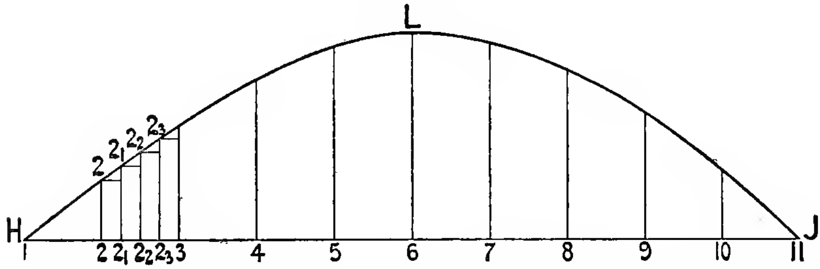
Fig. 15 shows, roughly, a portion of a ship's body—say below the load waterplane—which may be supposed represented by *A E D F A*. Obviously, we have dealt with no rules which admit of direct application here. In finding the volume it is sometimes found convenient to proceed as follows:— First, assume the body to be divided by an odd number of equidistant transverse planes (nine is shown in the figure), and calculate the areas of each of these planes from the keel up to the horizontal waterplane *A F D E A*.

Fig. 15.



Next, take a horizontal line, *HJ* fig. 16, having a length equal, on some scale, to the length of the vessel, and erect equal-spaced ordinates to correspond with the transverse sections of the body previously mentioned. On each of these ordinates, which are numbered 1, 2, 3, 4, 5, etc., in fig. 16, measuring from the base line *H J*, mark off to scale the number of square feet in the corresponding section of the vessel. Draw a fair curve through the points so obtained, and the surface *H L J* will have an area representing the cubic capacity of the body.

Fig. 16.



That the foregoing statement is true may be very simply shown. Let the space between any two sections, such as 2 and 3, be subdivided by ordinates drawn through the points 2_1, 2_2, 2_3, and where they intersect the curve, let lines be drawn parallel to *H J*, as shown. Now, since the ordinate at 2 represents the area of a section of the vessel at that point, the little rectangle $2\ 2_1$ will represent the volume of a vertical layer between sections at the points 2 and 2_1 having a constant section equal to area of vessel at section 2. In the same way the rectangles $2_1\ 2_2$, $2_2\ 2_3$, and $2_3\ 3$, will represent volumes of vertical layers, the areas of whose sections will be those of the vessel at the beginning of each little interval. The sum of the volumes of the vertical layers represented by $2\ 2_1$, $2_1\ 2_2$, etc., between sections at 2 and 3, will be less than the actual volume of the vessel at this part, and the deficiency is obviously represented by the areas of the little triangles between the tops of the rectangles and the curve. But by making the division close enough, the areas of these little triangles can be made as small as we please, so that in the limit the volume of the body, between sections at 2 and 3, will be truly represented by the area of that portion of fig. 14 enclosed by the curve, the bounding ordinates, and the base line. Thus it is clear that, as stated above, the total volume of the body is represented by the area $HLJH$.

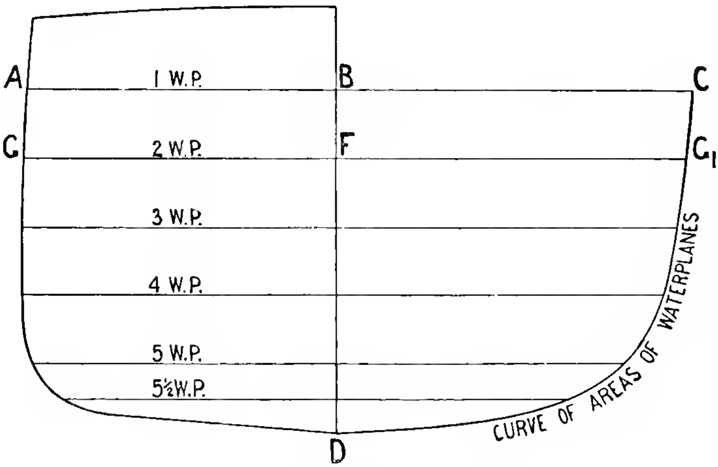
Take a numerical example.—The areas of the vertical transverse sections of a vessel up to the load line, in square feet, are, respectively, 0, 40, 163, 230, 400, 750, 470, 350, 270, 50, and 0, and the common interval between them is 12 feet. Calculate the total immersed volume of the body. It will be seen that this is merely a question of obtaining the area of a figure such as $HLJH$, and the work may therefore be tabulated as follows:—

No. of Section.	Area of Section, in sq. feet.	S.M.	Function of Areas.
1	0	1	—
2	40	4	160
3	163	2	326
4	230	4	920
5	400	2	800
6	750	4	3000
7	470	2	940
8	350	4	1400
9	270	2	540
10	50	4	200
11	0	1	—
			8286

Using the sum of the function of areas: volume of vessel $= 8286 \times \frac{12}{3}$
$= 33144$ cubic feet.

Besides this method of obtaining the volume of a vessel by using the areas of transverse vertical sections, there is another, and for many purposes, a more convenient one, which entails the use of the areas of horizontal sections, or waterplanes. Reverting to fig. 15, the upper plane $AEDFA$ is such a horizontal section; another one is represented by $GKHLG$. To fully take account of the vessel's form, a sufficient number of these horizontal sections are required. In fig. 17, which shows the midship section of a vessel, the traces of these horizontal planes with the plane of the paper are indicated as horizontal lines, numbered 1, 2, 3, etc. Only one half of the body is shown, the vessel being symmetrical about the middle line plane. The area of each of the horizontal planes is first calculated and set off to some scale, to the right of the middle line *B D*, on a horizontal line opposite the water-plane to which it refers. In fig. 17 these areas are represented by *B C* for the first waterplane, $F G_1$ for the second, and so on. A fair curve $C G_1 D$, drawn through these points will obviously, from our previous consideration, enclose an area representing the volume of the vessel from the keel to the first waterplane; and therefore, to obtain the immersed volume of the vessel, it is only necessary to calculate the area of *B C D*.

Fig. 17.



Take one numerical example: If the areas of the waterplanes of the vessel in fig. 17, beginning at the upper one, be 8000, 7600, 7000, 6000, 4500, 2800, and 150 square feet, respectively, and the distance between them be 3 feet, what will be the total volume? The work of finding this area we tabulate as follows:—

No. of W.P.	Area of W.P.	S.M.	Functions of Areas.
1	8000	1	8000
2	7600	4	30400
3	7000	2	14000
4	6000	4	24000
5	4500	$1\frac{1}{2}$	6750
$5\frac{1}{2}$	2800	2	5600
6	150	$\frac{1}{2}$	75
			88825

volume below No. 1 waterplane $= 88825 \times \frac{3}{3} = 88825$ cubic feet.

DISPLACEMENT AND BUOYANCY.—At this point, for a reason which shall appear presently, we must endeavour to explain an important hydrostatic principle known as the Law of Archimedes. This law asserts that if any body be immersed in a fluid it will be pressed upwards by a force equal to the weight of the volume of the fluid which it displaces; and if the body float at the surface of the fluid with only a portion of its bulk immersed, that the volume of fluid displaced will have the same weight as the total weight of the body. Thus, a box-shaped vessel, $100' \times 20' \times 10'$, floating in salt water, with half its depth of 10 feet immersed, will displace—

$$100 \times 20 \times 5 = 10000 \text{ cubic feet of the fluid.}$$

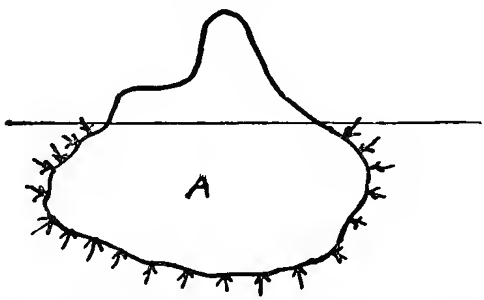
And since we know, or may easily verify by experiment, that a cubic foot of salt water weighs 1025 ozs., or 64 lbs., and therefore that a ton of salt water occupies a space of $\frac{2240}{64} = 35$ cubic feet, we are able to write—

$$\text{Weight of water displaced by vessel} = \frac{10000}{35} = 285 \cdot 7 \text{ tons.}$$

By the Law of Archimedes this weight is equal to that of the vessel and its contents.

The following is a simple proof of this important principle. If the body

Fig. 18



represented by *A* in fig. 18 be placed in a fluid of greater specific gravity than itself, it will float with a part of its bulk above the surface as shown. The immersed portion will be pressed in every direction by the fluid, those pressures which act on a section parallel to the plane of the paper being indicated by arrows. If, now, we imagine the fluid surrounding the body to become solidified, and the body itself to be non-existent, a cavity will remain having the exact shape of the immersed form of the body. If, finally, this cavity be supposed filled to the top with the same fluid and the surrounding solidified fluid be supposed to return to its former state, there will be a free level surface, and consequently the equilibrium will not be disturbed—that is to say, the fluid occupying the cavity will have the same statical effect as the body itself, since the same resultant upward pressure keeps each of them in equilibrium. From this it at once follows that the weight of the floating body is the same as that of a volume of the fluid occupying a space equal to that of the immersed portion of the body. This principle is of enormous value to the naval architect, for by it, when a vessel is floated, he knows that its weight, including contents, is equal to that of the displaced water. He has thus an infallible means of checking his calculations and of forming a basis on which to estimate the amount of cargo the vessel will carry.

We now see the importance of being able to calculate correctly the volume of the immersed body of a ship. We have described two ways in which this work can be done, and pointed out that the method involving the use of horizontal areas is preferable to the other, because of its greater convenience. This is seen, for instance, in the ready means which it affords of obtaining the volume, and therefore the weight of the displaced water at each of the various waterplanes indicated on fig. 17. These intermediate displacements, although not of special value of themselves, when plotted to scale at corresponding draughts, give a curve from which the displacement at any draught up to the load-line may be read off. This curve constitutes what is known as the displacement diagram.

As a practical example let us construct such a diagram in a specific case. Consider the vessel whose waterplane areas were used in the example on page 17. The volume up to the load waterplane was there determined to be 88,825 cubic feet.* Dividing this by 35 we obtain $\frac{88825}{35} = 2538$ tons as the ordinate of the displacement curve at a draught of 15 feet. Referring now to fig. 17, the volume up to the 2nd waterplane, or to a draught of 12 feet, is represented by the area DFG_1. The simplest way of obtaining this volume is to deduct the layer between the 1st and 2nd plane represented by the area BFG_1C from the total volume. The displacement to the 3rd waterplane may be found by direct application of the 1st rule, while for the value to the 4th waterplane, the volume between the 1st and 4th planes should be got by Simpson's 2nd rule, and the result deducted from the total volume. The 1st rule will be suitable for finding the displacement to the 5th plane, the half interval being used in this case. In the following table we show these calculations carried out as suggested; the final results are arranged by themselves for easy reference :—

DISPLACEMENT CALCULATION.

No. of Sect.	Areas of W. Planes	S.M.	Function of Areas.	S.M.	Function of Areas.	S.M.	Function of Areas.	S.M.	Function of Areas.	S.M.	Function of Areas.
1	8000	1	8000					5	40000	1	8000
2	7600	4	30400					8	60800	3	22800
3	7000	2	14000	1	7000			− 1	− 7000	3	21000
4	6000	4	24000	4	24000					1	6000
5	4500	1½	6750	1½	6750	½	2250				
5½	2800	2	5600	2	5600	2	5600				
6	150	½	75	½	75	½	75				
			88825		43425		7925		93800		57800

* There is assumed to be no displacement below the 6th waterplane.

Displacement to load waterplane $= \dfrac{88825 \times 3}{35 \times 3} = 2538$ tons.

Displacement of layer between 1st and 2nd W.P. $= \dfrac{93800 \times 3}{35 \times 12} = 670$ tons.

Displacement to 2nd waterplane $= 1868$ tons.

Displacement to 3rd waterplane $= \dfrac{43425 \times 3}{35 \times 3} = 1240$ tons.

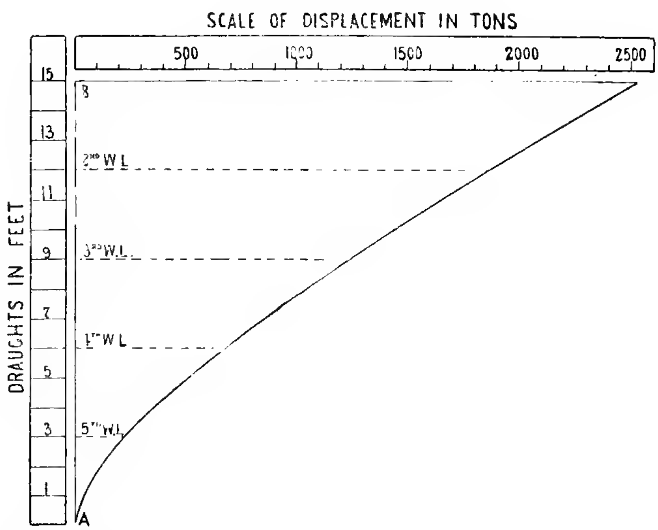
Displacement of layer between 1st and 4th W.P. $= \dfrac{57800 \times 3 \times 3}{35 \times 8} = 1858$ tons.

Displacement to 4th waterplane $= 680$ tons.

Displacement to 5th waterplane $= \dfrac{7925 \times 3}{35 \times 3} = 226$ tons.

The construction of the displacement curve is now an easy matter. Take a vertical scale of draught *A B* (fig. 19), and let the distance from *A* to *B* equal 15 feet; divide it into five equal parts and draw horizontal lines from the points of division. Number these horizontal lines from *B* downwards. Now measure along these lines, to some convenient scale, distances representing the displacements corresponding to these draughts. A fair curve drawn through the points will give the diagram required.

Fig. 19.



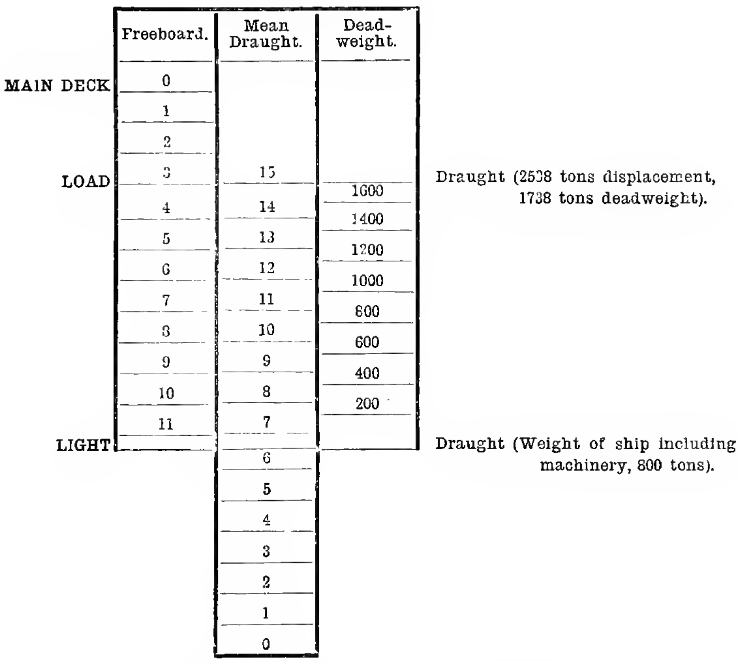
To complete it, a scale of tons should be shown along the top, and the distance between *A* and *B* divided off into feet and inches, so that any draught can be located immediately. The construction lines 1, 2, 3, etc., being no longer required, should be erased. It is necessary to note that in reading from the diagram mean draughts only should be used. For instance, if the displacement of the above vessel were required when floating at 12 feet aft and 9 feet forward, the draught to be taken on the scale should be $\frac{12 + 9}{2} = 10\frac{1}{2}$ feet.* A horizontal line drawn out at this draught would intersect the curve in a point showing on the scale of tons a displacement of 1530 tons. In the same way the displacement at any mean draught, provided it did not exceed 15 feet, could be found.

From the displacement diagram another very useful one may be constructed, called a "scale of deadweight." It is specially constructed for the use of ships' officers and others who may have to do with loading operations. It exhibits in graphic form the weight of cargo put aboard as the vessel sinks in the water, and may be looked upon as a kind of loading meter by which the officer is able to tell, at any moment during loading operations, the amount of cargo he has got aboard, and the amount still to be dealt with to bring the vessel to her assigned load-line.

In fig. 20 we give an illustration of such a diagram deduced from fig. 19. It will be observed to consist of two columns, one of which is a scale of draughts in feet, while the other indicates the amount of immersion caused in the vessel by the addition of each 200† tons in her load. The effect on the draught of quantities less than 200 tons is, of course, found by interpolation.

Fig. 20.

DEADWEIGHT SCALE.



As an example, suppose that the vessel, whose deadweight scale is illustrated by fig. 20, is observed at a certain time during loading operations to be floating

* This is not absolutely correct, see Appendix A.

† In ordinary cases increments of 100 tons deadweight are indicated.

at 7 feet forward and 10 feet 4 inches aft, and that it is required to ascertain how much cargo has been put on board, and how much has still to be shipped to sink her to a mean draught of 15 feet?

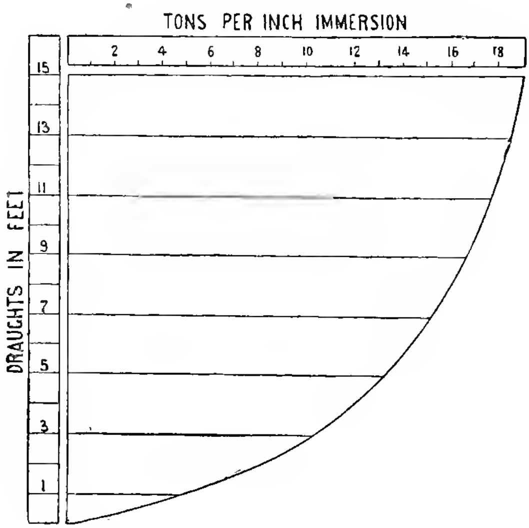
Mean draught at time of observation $= \frac{10 \text{ ft. } 4 \text{ ins.} + 7 \text{ ft.}}{2} = 8$ ft. 8 ins.

At this draught there will be 380 tons aboard.

At 15 feet mean draught the vessel will carry 1738 tons, therefore the amount of cargo still to be shipped $= 1736 - 380 = 1356$ tons.

CURVE OF TONS PER INCH OF IMMERSION.—Sometimes it is desirable to know how much the draught of a vessel would be affected by shipping or discharging a moderate quantity of cargo. If the mean draught were known, this information could be obtained from the displacement diagram or the deadweight scale; it can, however, be more conveniently got by means of a special diagram, called a "Curve of tons per inch of Immersion," which shows graphically the number of tons required to sink or lighten the vessel one inch at any draught. The weight of cargo shipped, divided by a number read from the diagram, will give the number of inches by which the draught has been altered.

Fig. 21.



If A be the area of any waterplane, then the weight of a layer of salt water one inch thick will be $\frac{A \times \frac{1}{12}}{35} = \frac{A}{420}$ tons; which by Archimedes' principle will also equal the number of tons of cargo necessary to sink or lighten the vessel one inch at this draught.

To construct the diagram required take any vertical line representing to scale the full mean draught of the vessel, and at the heights of the waterplanes of which the areas are known, draw horizontal lines. Mark off to scale on each of these lines the corresponding quantities $\frac{A}{420}$, and draw a fair curve through the points so obtained. This will be the curve of tons per inch of immersion. To complete the diagram, as shown in fig. 21, a scale of draughts and of tons must be drawn and the construction lines erased.

Example.—If the vessel, whose diagram is given above, were floating at a mean draught of 9 feet, what would be the increased immersion due to shipping 50 tons of cargo? From the curve at 9 feet draught the tons per inch is found to be 16·6, therefore :—

$$\text{additional immersion} = \frac{50}{16 \cdot 6} = 3 \cdot 01 \text{ inches.}$$

QUESTIONS ON CHAPTER I.

1. State the Trapezoidal Rule for finding areas of plane surfaces having curved boundaries, and point out wherein it is inaccurate. The half ordinates in feet of the load water-plane of a vessel are, commencing from aft, 2, 6·5, 9·3, 10·7, 11, 11, 10, 7·4, 3·6, and ·2, and the common interval between them is 15 feet. Find the area of the plane by using the Trapezoidal Rule.

Ans.—2118 square feet.

2. What are the advantages of Simpson's First Rule for finding plane areas, and for what curve is the Rule accurate? What are the conditions as to the number and spacing of ordinates? The semi-ordinates of the waterplane of a vessel in feet are, respectively, ·1, 5, 11·6, 15·4, 16·8, 17, 16·9, 16·4, 14·5, 9·4, and ·1. The spacing of the ordinates is 11 feet, find the area of plane in square yards.

Ans.—303·6.

3. Given the values of three consecutive and equally spaced ordinates and the common distance between them, what Rule would you employ to find the area between the first and second ordinates? If the ordinates in feet are 5, 11·6 and 15·4, and their spacing 11 feet, find the area between the first two.

Ans.—93·86 square feet.

4. State Simpson's Second Rule. To what class of curve does it apply accurately? Given ·1, 2·6, 5, and 8·3 as the value in feet of the half ordinates of a portion of a ship's waterplane, and 9 feet as the common distance between them, calculate the area including both sides.

Ans.—210·6 square feet.

5. Why are half ordinates sometimes introduced at the ends of plane figures? Deduce the modification in the multipliers of Simpson's First Rule due to the introduction of a half ordinate.

6. What are the main points of difference between Tchebycheff's Rule and Simpson's First Rule for finding plane areas? Compare by an actual practical example the results obtained by applying Tchebycheff's two ordinate Rule and Simpson's First Rule.

7. Given the areas to the L.W.P. of the transverse vertical sections of a vessel, show that the volume of the displacement may be expressed as a plane area. If the tranverse vertical sections in a particular vessel are 4, 100, 180, 240, 260, 242, 190, 120 and 8 square feet, and the common interval is 15 feet, calculate the volume of displacement.

Ans.—20,400 cubic feet.

8. Explain why it is preferable to employ the areas of horizontal sections or waterplanes rather than of transverse vertical sections in calculating the volume of displacement. The areas of the waterplanes of a vessel are 6000, 6000, 4800, 3600, 2400, 1200, and 100 square feet; the common interval between the waterplanes is 2 feet. Calculate the displacement in tons (salt water), neglecting the portion below the lowest plane.

Ans.—1213·3.

9. What is the "Law of Archimedes"? Explain in what way this Law is important to the naval architect.

10. Referring to the latter part of question No. 8, calculate the displacement to the various waterplanes, and plot the diagram of displacement.

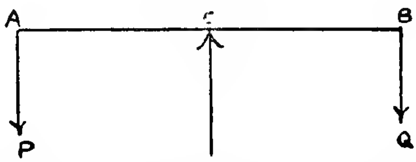
11. How is a curve of "Tons per inch of Immersion" constructed? What use is made of such a curve? The areas of a ship's L.W.P. is 4000 square feet, and the areas of other parallel water sections are, respectively, 3650, 32[illegible]0, 2550, and 24 square feet. The vertical distance between the sections is 2 ft. 9 ins. Construct the curve of "Tons per inch of Immersion?"

CHAPTER II.

Moments, Centre of Gravity, Centre of Buoyancy.

MOMENTS.—If two equal weights be placed one at each end of a weightless lever $A\,B$ (fig. 22), it is obvious that the point at which they may be

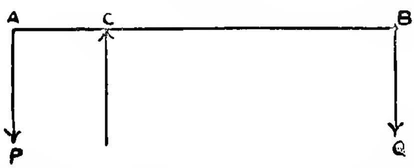
Fig. 22.



supported in equilibrium lies midway between them. If the weights be unequal the balancing point C will not be at the middle but at some other position nearer the larger weight.

In books on elementary mechanics it is shown that in all such cases the exact position of C may be obtained from the relation (fig. 23), $\frac{P}{Q}=\frac{C\,B}{A\,C}$ (1),

Fig. 23.

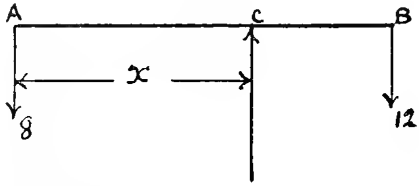


where P and Q are the unequal weights, and $A\,C$ and $C\,B$ the distances of the points of application of the weights from the fulcrum. Cross multiplying, this equation becomes $P \times A\,C = Q,\ C\,B,$ (2).

It appears then, that from a consideration of a balanced system of two parallel forces acting on a rigid bar assumed to be weightless, two items of interest may be deduced: first, that the position of the point of support must be fixed by equation (1); and second, that the moment of the force, or turning effort, about the point of support on one side must be equal and opposite to that on the other, as indicated by equation (2).

Thus, if weights of 8 and 12 lbs. be suspended at A and B (fig. 24), the

Fig. 24.



extreme points of a weightless lever 36 inches long, and if x be the distance of the balancing point C from A, we have, using equation (1), $\frac{8}{12} = \frac{36 - x}{x}$ from which we get $x = 21\frac{3}{5}$ inches.

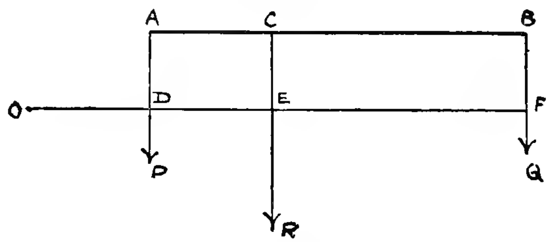
That the point thus determined by x is the one required is proved by the fact that the moments of the weights about this point are equal, *i.e.*,

$$21\tfrac{3}{5} \times 8 = 14\tfrac{2}{5} \times 12.$$

The following is an important theorem:—

The moment of the resultant of a system of parallel forces in one plane acting on a rigid body about any point in the plane is equal to the sum of the moments of the component forces about the same point. Take the simple case of two forces acting in the same direction, as in fig. 25. Let A and B

Fig. 25.



be the points of application of the forces; join $A\,B$ and assume the line to be horizontal. Let O be the point about which the moments are to be taken, and R the resultant of the two forces, which may be called P and Q. Drop a perpendicular from O upon the line of action of the forces, which for simplicity are assumed to be vertical, cutting them in the points D, E, and F, as shown.

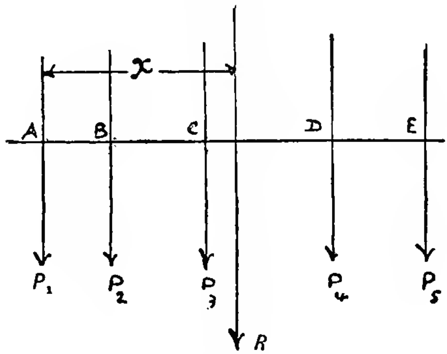
It is clear that the above theorem will hold if $R \times O\,E = P \times O\,D + Q \times O\,F$, that is, if $(P + Q)\,O\,E = P\,(O\,E - D\,E) + Q\,(O\,E + E\,F)$ or, multiplying out and cancelling like terms, if $P \times D\,E = Q \times E\,F$.

Since AB is parallel to DF, this may be written $P \times AC = Q \times CB$. But this relation we know to be true, therefore so must be the above theorem.

The theorem will, of course, hold if there be any number of forces acting, for if the line of action of the resultant be found, the forces acting on either side of this line may be represented by a single force, and this will reduce the case to the one just proved.

We are now able to deal with questions in which it is necessary to find the position of the resultant of parallel forces. Consider the forces P_1, P_2, P_3, etc., shown in fig. 26, which, in the first instance, we shall suppose acting in

Fig. 26.



one plane. In order to determine the position of the resultant we must take moments about some point in the plane. Drop a perpendicular upon the lines of action of the forces, which, as before, we assume to be vertical, cutting them in the points A, B, C, D, and E, and take the moments about a point in this line, say where it intersects the line of action of the force P_1. Assuming the resultant R to act at a distance x from A, we have by the theorem— $R \times x = P_1 \times 0 + P_2 \times AB + P_3 \times AC + P_4 \times AD + P_5 \times AE$, from which, since R equals the sum of the several weights,

$$x = \frac{P_1 \times 0 + P_2 \times AB + P_3 \times AC + P_4 \times AD + P_5 \times AE}{P_1 + P_2 + P_3 + P_4 + P_5}$$

Suppose that the forces P_1, P_2, P_3, P_4, and P_5, are of the following magnitudes, viz.—4, 8, 6, 12 and 10 units respectively, and that the distance AE, which is 12 feet, is divided into equal parts by the lines of action of the forces, the distance of the resultant from A will be—

$$x = \frac{4 \times 0 + 8 \times 3 + 6 \times 6 + 12 \times 9 + 10 \times 12}{40} = 7{\cdot}2 \text{ feet.}$$

If the forces in fig. 26 do not act in one plane, in order to find the centre of the system it will be necessary to determine its perpendicular distances from two vertical planes at right angles. If we suppose one of these planes to be perpendicular to the plane of the paper, and its vertical trace to be represented by the line of the force P_1, by a simple moment calculation about this plane we will determine, not the position of the resultant, but only of the vertical plane containing it parallel to the plane chosen as the axis.

To fully determine the position we must now find another plane also containing the resultant parallel to the plane of the paper. Clearly, since it is in both planes, it must coincide with the line of their intersection. Let the forces in fig. 26 act at the distances shown from the plane containing P_1 normal to the plane of the paper; 7·2 will be the distance from the axis of one of the planes containing the resultant. To find the other one we must know the position of each of the forces from a plane parallel to the plane of the paper. Let these be given by the normal distances $y_1, y_2, -y_3, y_4, -y_5$, each one having the same suffix as the force to which it refers. Calling Y the distance of the plane of the resultant from the axis plane, we have—

$$Y = \frac{P_1 y_1 + P_2 y_2 - P_3 y_3 + P_4 y_4 - P_5 y_5}{P_1 + P_2 + P_3 + P_4 + P_5}.$$

Putting in the numerical values 2, 4, − 7, 9, − 5, for y_1, y_2, etc., this becomes—

$$Y = \frac{4 \times 2 + 8 \times 4 - 6 \times 7 + 12 \times 9 - 5 \times 10}{40} = 1{\cdot}4 \text{ feet.}$$

The two values, $X = 7{\cdot}2$ feet and $Y = 1{\cdot}4$ feet, determine the line of the resultant of the system of the assumed parallel forces.

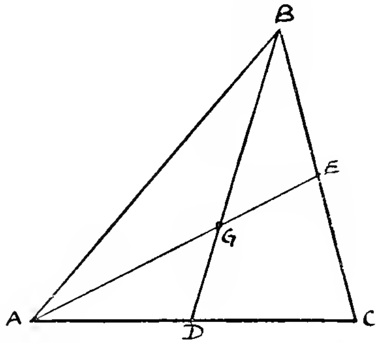
The preceding principle admits of many important applications, not the least of which is that to the finding of centres of gravity, to which we must now turn. We begin with a general definition.

CENTRE OF GRAVITY.—If the mass of any body be supposed divided into an infinite number of parts, the forces or weight due to the attraction of the earth, acting on the various parts, will form a system of parallel forces of which the total weight of the body is the resultant; and the point through which the line of action of this resultant always passes, whatever be the position of the body with reference to the earth, is called the centre of gravity of the body.

The centre of gravity of a body is also sometimes defined briefly as the point at which the weight of the body may be taken to act, no matter what position it may occupy. Thus, in the case of a ship and cargo, the total weight is taken as acting at a fixed point when making stability and other calculations.

It is frequently necessary in dealing with ship calculations to obtain the centre of gravity of an area. In approaching such questions it is usual to keep the idea of weight and to consider the area as consisting of a homogeneous lamina of uniform but infinitely small thickness. Thus, the centre of gravity of a lamina of circular form is at its geometric centre, as evidently the resultant of all the forces due to the weight of the various portions of the lamina must pass through that point. Also the centre of gravity of a lamina of square form is in the point of intersection of the two diagonals. To find the centre of gravity of a triangular lamina such as ABC (fig. 27), we may proceed as follows:— First, bisect AC in D and join BD. This line contains the centre of gravity of all strips of the lamina parallel to AC, consequently the centre of gravity of the triangle must be somewhere in it. Next, bisect one of the other sides, say

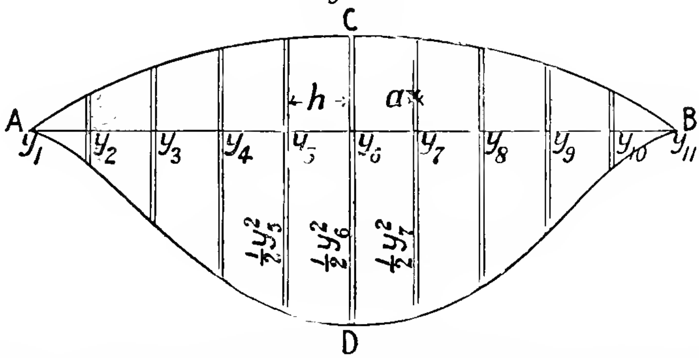
Fig. 27.



BC, in E, and join AE. The centre of gravity of the lamina must obviously also be in AE; therefore, it must be at the point G where the lines AE and BD intersect. G is at a point one-third of BD from D.

In passing to the case of a lamina having a curved boundary, such, for instance, as the half waterplane of a ship, we cannot determine the centre of gravity by such geometrical methods, owing to the irregularity of the form. The usual practice is in effect to divide the lamina into an infinite number of elements, to take the moments of these elements about any two axes chosen at

Fig. 28.



right angles in the plane of the lamina, and to divide the sum of each series of moments by the total weight of the lamina, each quotient being the distance of a line containing the centre of gravity parallel to its corresponding axis, and the centre of gravity itself, the intersection of the two lines. By employing Simpson's

Rules it is only necessary to deal with specimen elements, which greatly simplifies the work. As an illustration consider the half waterplane ABC (fig. 28). Divide AB into a number of parts as shown, and draw ordinates to the curve. Now take a strip of the lamina of very small breadth a at ordinate 5, say; its area will be $y_5 a$, and this may also stand for the weight since the lamina is homogeneous and of uniform thickness.

The moment of this little area about AB as axis will be—

$$y_5 a . \frac{y_5}{2} = \frac{y_5^2}{2} . a$$

With a as base, set down $\frac{y_5^2}{2}$ as an ordinate below AB, and draw in the little rectangle shown in the diagram, which will represent the moment of the strip of area at y_5. In the same way obtain and plot the moments of elementary areas at y_1, y_2, etc. A curve through the extremities of these little rectangles will enclose an area ADB, which will represent the moment of the area ACB about the line AB, and consequently, by the principle of moments, the distance of the centre of gravity of this area from the chosen axis will be given by—

$$\frac{\text{Total Moment Area about } AB}{\text{Total Area}}, \text{ or } \frac{\text{Area } ADB}{\text{Area } ACB}.$$

As a numerical example, let the above half waterplane be 140 feet long, and let 11 ordinates be taken so as to suit the application of Simpson's First Rule to the finding of the areas ADB and ACB.

The figures of this calculation are best arranged in tabular form as shown below. In the first two columns are the numbers of the ordinates and their values in feet, respectively. The third column gives Simpson's multipliers, and the fourth the corresponding functions obtained when the ordinates are treated by these multipliers. It will be seen that so far the work is simply in the direction of finding the area ACB. The next two columns are for obtaining the area enclosed by the moment curve, the fifth giving the squares of the ordinates, and the sixth the functions of the same when affected by the multipliers.

Nos. of Ordinates.	Ordinates (ft.).	S.M.	Function of Ordinates.	Ordinates².	Function of Squares.
1	·3	1	·3	·09	·09
2	2·5	4	10·0	6·25	25·00
3	6·5	2	13·0	42·25	84·50
4	9·3	4	37·2	86·49	345·96
5	10·7	2	21·4	114·49	228·98
6	11·0	4	44·0	121·00	484·00
7	11·0	2	22·0	121·00	242·00
8	10·0	4	40·0	100·00	400·00
9	7·4	2	14·8	54·76	109·52
10	3·6	4	14·4	12·96	51·84
11	·2	1	·2	·04	·04
			217·3		1971·93

Using these figures we obtain at once—

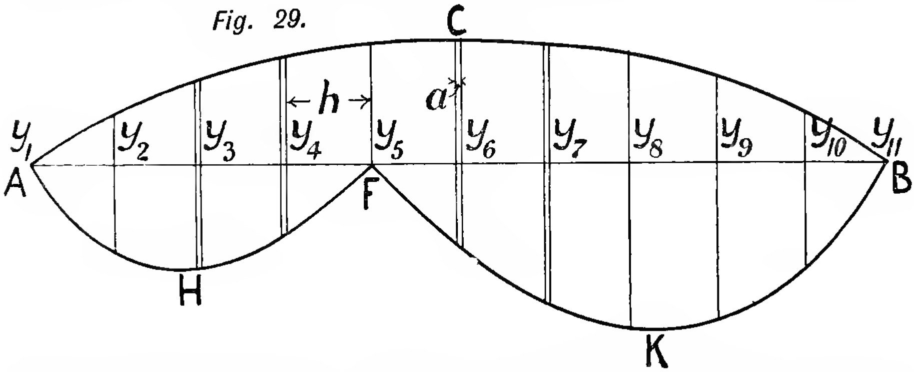
$$\left.\begin{array}{r}\text{Distance of centre of gravity of half}\\ \text{plane from axis } AB\end{array}\right\} = \frac{\text{Area enclosed by moment curve}}{\text{Area of half plane}}$$

$$= \frac{\dfrac{1971{\cdot}93}{2} \times \dfrac{14}{3}}{217{\cdot}3 \times \dfrac{14}{3}} = 4{\cdot}53 \text{ feet.}$$

It should be noticed that in the calculation the whole squares are employed in the table, the division by 2 being done at the end, as shown.

We have, as a result of the preceding calculation, fixed the position of one line containing the centre of gravity. We must now, as already mentioned, determine the position of another line also containing it at right angles to this one. The principle of moments is again employed, and in fixing upon an axis for the purpose, it is usual to choose an ordinate about the middle of the plane, as it will obviously mean a less laborious calculation than if the axis were taken, say, at either end. Care must also be taken to select an

Fig. 29.



ordinate which will allow of Simpson's First Rule being applied in arriving at the areas enclosed by the moment curves. In the present instance either ordinate 5 or 7 might be employed; the middle ordinate No. 6 is unsuitable (*See* fig. 29)

Taking No. 5 as axis, the moment of a small strip at ordinate No. 4 will be $y_4 a h$, h being the common interval between the ordinates and a the breadth of the strip. At ordinate No. 3, the moment of a strip will be $y_3 a \times 2h$, and so on for strips at the other ordinates, the little area in each case being multiplied by the number of times of the common interval it is removed from the chosen axis. The process is repeated for the area on the other side of the axis, the moment of a strip at No. 6 ordinate being $y_6 a h$; that for strip at No. 7, $y_7 a\, 2h$, and so on. To construct the moment diagram, the moments thus found of the small areas are, as in the previous case, set down as little rectangles, each on a base a, on the other side of AB at the points to which they refer, and fair curves drawn as AHF and FKB in the figure. Evidently, the centre of gravity will be on that side of the axis which has the greater moment; and its distance from the axis will be obtained by dividing the difference of the moments by the area of the half plane; thus—

$$\left.\begin{array}{r}\text{Distance of centre of gravity from axis}\\ \text{through ordinate No. 5,}\end{array}\right\} = \frac{\text{Area } FKB - \text{Area } AHF}{\text{Area } ACB}$$

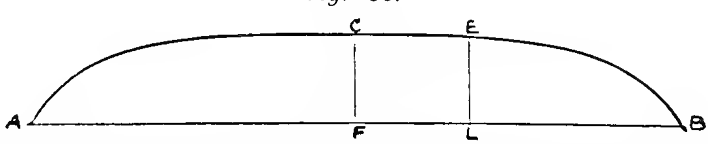
The work of finding the above areas is arranged below. The first four columns are the same as before; in the fifth are the multipliers representing the number of intervals each ordinate is distant from the axis through No. 5; the sixth column gives the functions of the ordinates after treatment by these multipliers as well as those of Simpson's Rule.

No. of Ordinates.	Ordinates.	S.M.	Function of Ordinates.	Mult. for Leverage.	Function of Moments.
1	·3	1	·3	4	1·2
2	2·5	4	10·0	3	30·0
3	6·5	2	13·0	2	26·0
4	9·3	4	37·2	1	37·2
5	10·7	2	21·4	0	94·4
6	11·0	4	44·0	1	44·0
7	11·0	2	22·0	2	44·0
8	10·0	4	40·0	3	120·0
9	7·4	2	14·8	4	59·2
10	3·6	4	14·4	5	72·0
11	·2	1	·2	6	1·2
			217·3		340·4

$$\left.\begin{array}{r}\text{Centre of gravity from axis through}\\ \text{ordinate No. 5,}\end{array}\right\} = \frac{(340{\cdot}4 - 94{\cdot}4)\,\frac{14}{3} \times 14}{217{\cdot}3 \times \frac{14}{3}} = 15{\cdot}85 \text{ feet.}$$

The centre of gravity of the half plane is therefore situated at a point 15·85 feet forward of No. 5 ordinate, and 4·53 feet out from the centre line AB. Obviously, the centre of gravity of the whole plane, since both sides are alike, will be in the middle line and at the same distance forward of the axis, namely, 15·85 feet.

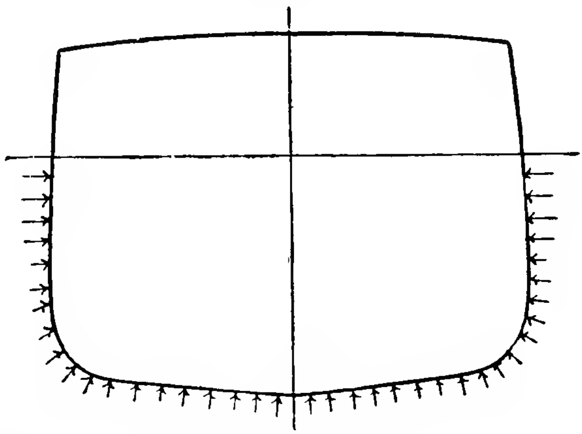
Fig. 30.



The same useful principle of moments is employed if we wish to find the centre of gravity of a portion only of a ship's waterplane, say, of the area $ACEB$ (fig. 30), omitting the space $CFLE$ As before, moments of elementary areas are taken about AB and about an axis at right angles to it, omitting the portion of the area, *C F L E*. The resulting distances obtained by dividing the sum of each of these systems of moments by the reduced area will determine the position of the centre of gravity of the partial plane from the chosen axes.

CENTRE OF BUOYANCY.—It was shown, when treating of displacement and buoyancy, that the weight of any floating body is supported by the upward pressure due to the buoyancy of the water. Fig. 31 represents in section a ship floating freely and at rest in still water, and indicates the water pressures acting on her. It is the resultant of the vertical components of these pressures, which act everywhere normal to the surface in contact, which supports, and is therefore equal to, the total weight of the vessel. It now becomes necessary to state further that the line of action of this resultant, whatever be the position of the vessel, always passes through a certain point, viz., the centre of the immersed bulk, or the centre of gravity of the water that would occupy the same space. This point is called the centre of buoy-

Fig. 31.



ancy. We now proceed to show how this centre may be determined in any given case.

In a vessel of simple box-shape, floating at a level waterplane, the point will obviously be at mid length in the centre line plane, and at a distance below the surface equal to half the draught. In one of constant triangular section floating with a side parallel to the surface it will also be at mid length, but at $\frac{1}{3}$ the depth below the surface. In a cylindrical vessel floating at even keel, it will as before be at mid length and at the same distance below the surface as the centre of gravity of the transverse section. In all these cases, the conditions being given, the point required can be easily determined. In ship-shape bodies, however, owing to the irregularity of form, no such simple methods can be applied. We must, therefore, resort to moment calculations, just as we had to do when finding areas of surfaces enclosed by ship curves.

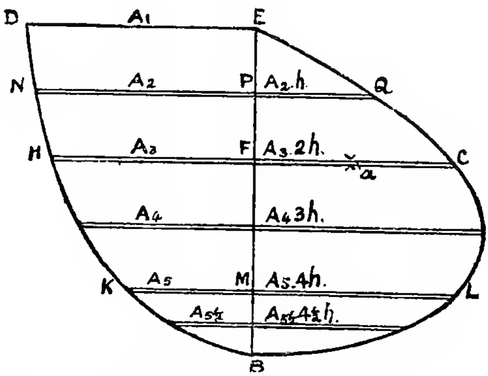
Take, as example, a vessel of ordinary form floating at a draught parallel to the keel-line (*see* fig. 32). In setting out to find the centre of buoyancy, we observe, in the first place, that, since the vessel is symmetrical about the middle line plane, the point required must be somewhere in that plane, and that it will be fully determined if we know its position relative to a vertical and a horizontal line in the plane.

To obtain the vertical position of the point, the immersed body is assumed divided into an infinite number of horizontal layers, and a calculation of moments made with respect to some horizontal plane, such as that of the load-water line.

Fig. 32.

For the horizontal position, the displacement is supposed divided into transverse vertical layers and another calculation of moments made; in this case, with respect to a transverse vertical plane, such as that of the after perpendicular, or of a transverse section in the vicinity of amidships. It is only necessary to correctly plot the results of these calculations in the middle-line plane to obtain the position of the centre of buoyancy. In practice, as in the case of calculations of areas and volumes, by using Simpson's or Tchebycheff's rules, only specimen layers of displacement need be dealt with.

Fig. 33.



Turning to fig. 32 we note that between the upper waterplane W.L. and the keel, four others are introduced at equal distances apart, with, in addition, an intermediate one between the keel and the plane marked No. 5. The plane at half interval is introduced owing to the increased curvature of the ship's form at that part, which makes a closer spacing necessary to obtain accurate results.

Let A_1, A_2, A_3, etc., be the areas of the waterplanes, h the common interval between them, and a a very small thickness of layer taken at each waterplane. The moments of the elementary layers about W.L. will be—0, $A_2 a h$, $A_3 a 2h$, and so on, volumes being treated as weights, the density being constant. Now take any vertical scale of draughts (fig. 33) and mark off horizontally at the planes, 2, 3, 4, etc., the corresponding moments just found, which should be plotted as rectangles of breadths, $A_2 h$, $A_3 2h$, etc., and depths a. A fair curve through the extremities of the little rectangles thus obtained, starting from the point E, will enclose an area representing the total moment of the volume below the upper waterplane. If, on the other side of the axis EB, another diagram be plotted, the ordinates at the various waterplanes being the corresponding waterplane areas, the area of this diagram will represent the total volume of the vessel below No. 1 waterplane.

From our previous considerations it is clear that we may write :—

$$\left.\begin{array}{c}\text{Distance of centre of buoyancy}\\ \text{below No. 1 waterplane,}\end{array}\right\} = \frac{\text{Area } ECB}{\text{Area } EDB}$$

As a numerical example, suppose the areas of the waterplanes in square feet are, beginning from the upper one, 8000, 7600, 7000, 6000, 4500, 1800, and 100, respectively, and that the common distance between them is 3 ft., with a half interval at the lower end.

In obtaining the vertical distance of the centre of buoyancy below the upper waterplane, in this, and all similar examples, it is convenient to arrange the work in tabular form, as shown below.

In the second, third, and fouith columns are the areas, Simpson's multipliers, and functions of areas, respectively. In the fifth column are the multipliers for leverage, and in the sixth, the products of the lever multiples and the area functions. The process is seen to be simply that of obtaining areas such as ECB and EDB by Simpson's Rule.

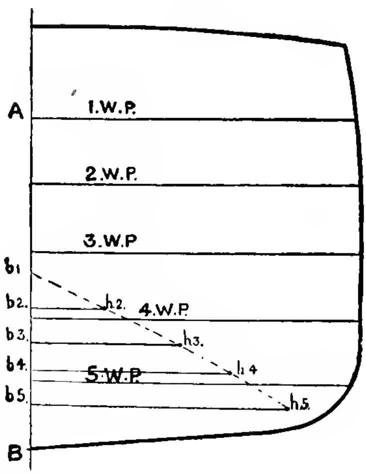
No. of Ordinates.	Areas of Planes.	S.M.	Function of Areas.	Levers.	Function of Moments.
1	8000	1	8000	0	—
2	7600	4	30400	1	30400
3	7000	2	14000	2	28000
4	6000	4	24000	3	72000
5	4500	$1\frac{1}{2}$	6750	4	27000
$5\frac{1}{2}$	1800	2	3600	$4\frac{1}{2}$	16200
6	100	$\frac{1}{2}$	50	5	250
			86800		173850

$$\left.\begin{array}{c}\text{Distance of centre of buoyancy}\\ \text{below No. 1 W.P.,}\end{array}\right\} \frac{173850 \times \frac{3}{3} \times 3}{86800 \times \frac{3}{3}} = 6 \text{ feet.}$$

Volume of displacement below No. 1 W.P. = 86800 cubic feet.

The position of the centre of buoyancy below any of the other waterplanes may now be obtained. Reverting to fig. 33—$\frac{\text{area } EPQ}{\text{area } DEPN}$ gives the distance of the centre of the layer between the 1st and 2nd waterplanes from the 1st waterplane, and by a simple moment calculation, the fall in the centre of buoyancy consequent on the vessel rising to the 2nd waterplane is derived. In the same way, by first finding the centre of the layer between the 1st and 3rd waterplanes, or between the 1st and the 4th waterplanes, the fall in the centre of buoyancy, due to the rising of the vessel to any of these planes, may be determined. We have here a means of constructing a diagram which will show the variation in the height of the centre of buoyancy with change in the displacement, and from which, therefore, the position of the centre of buoyancy for any draught may be read off. Take a vertical scale of draughts AB (fig. 34), and spot off on it

Fig. 34.



the positions of the various centres of buoyancy as calculated for the vessel when immersed to the 1st, 2nd, 3rd, etc., waterplanes.

Through these points, indicated by b_2, b_3, etc., in the figure, set out horizontally distances, $b_2\,h_2$, $b_3\,h_3$, etc., equal to those between the load waterplane and the waterplane to which each centre refers. A fair curve through the points b_1, h_2, h_3, etc., will be the locus of centres of buoyancy required.

If now the height of the centre of buoyancy at any draught be required it is only necessary to draw a line on the diagram parallel to the middle line AB, and at a distance from it equal to that between the load-line and the given draught; the point of intersection of this line with the locus gives the required height of centre of buoyancy.

As showing the work in an actual case, let us construct the diagram for the vessel whose centre of buoyancy at the load draught has already been determined. Reverting to fig. 33, it will be necessary to find the areas PEQ and $DEPN$.

The latter area may be obtained by the Five-Eight Rule already described; area PEQ, however, cannot be correctly found by this Rule. In this case we should proceed as follows:—Multiply the near end ordinate, or A_1, by 3, the middle ordinate, or A_2, by 10, the far end ordinate, or A_3, by -1, and the sum of these products by one twenty-fourth the square of the common interval between the ordinates.

Arranged in tabular form, the figures of the calculation are:—

VOLUME.				MOMENT.		
No. of Ordinates.	Areas of W.Planes.	S.M.	Functions.	Areas of W.Planes.	Multiplier.	Functions.
1	8000	5	40000	8000	3	24000
2	7600	8	60800	7600	10	76000
3	7000	-1	-7000	7000	-1	-7000
			93800			93000

$$\text{Moment of layer } DEPN = \frac{93000 \times 9}{24} = 34875$$

$$\text{Volume of layer } DEPN = \frac{93800 \times 3}{12} = 23450 \text{ cubic feet.}$$

$$\left.\begin{array}{l}\text{Distance of centre of buoyancy}\\ \text{of layer below No. 1 W.P.,}\end{array}\right\} = \frac{34875}{23450} = 1{\cdot}48 \text{ feet.}$$

Having now got the position of the centre of buoyancy of the layer, the distance the centre of buoyancy will fall when the vessel rises to the 2nd waterplane may be easily determined, since the moment of the layer and the moment of the displacement below the 2nd waterplane about the centre of buoyancy of the total displacement are equal. The total volume of displacement we have found to be 86800 cubic feet. The volume of the layer between the 1st and 2nd waterplanes from our calculation is 23450 cubic feet; the volume below the 2nd waterplane will therefore be, $86800 - 23450 = 63350$ cubic feet. Calling d_1 the fall of centre of buoyancy in feet, we may write:—

$$d_1 \times 63350 = 23450\,(6 - 1{\cdot}48)$$

$$\therefore d_1 = \frac{23450 \times 4{\cdot}52}{63350} = 1{\cdot}67$$

For the fall of the centre of buoyancy when the vessel is at the 3rd waterplane, we must find the areas *FEC* and *DEFH*, which may be done thus. using Simpson's First Rule:—

No. of W.P.	S.M.	Areas.	Function of Areas.	Moments from Diagram.*	Function of Moments.
1	1	8000	8000	—	—
2	4	7600	30400	7600	30400
3	1	7000	7000	14000	14000
			45400		44400

$$\left.\begin{array}{r}\text{Distance of centre of buoyancy}\\ \text{of layer below No. 1 W.P.,}\end{array}\right\} = \frac{44400 \times \frac{3}{3} \times 3^{*}}{45400 \times \frac{3}{3}} = 2{\cdot}93 \text{ feet.}$$

The volume of the layer = 45400 cubic feet, so that the volume below the 3rd waterplane will be, 86800 − 45400 = 41400 cubic feet. If d_2 be the fall of the centre of buoyancy in feet, we have—

$$d_2 \times 41400 = 45400\,(6 - 2{\cdot}93)$$

$$\therefore d_2 = \frac{45400 \times 3{\cdot}07}{41400} = 3{\cdot}36.$$

d_3, the fall of the centre of buoyancy when the vessel floats at the 4th waterplane, will be found to be 5·10 feet. To find the requisite areas in the diagram (fig. 33) the Three-Eight Rule should be used; otherwise, the calculation is similar to the preceding ones.

d_4, the fall of the centre of buoyancy to the 5th waterplane, is 6·81 feet We have now sufficient information to construct the locus which, when plotted as described, will be found to give the curve shown in fig. 34.

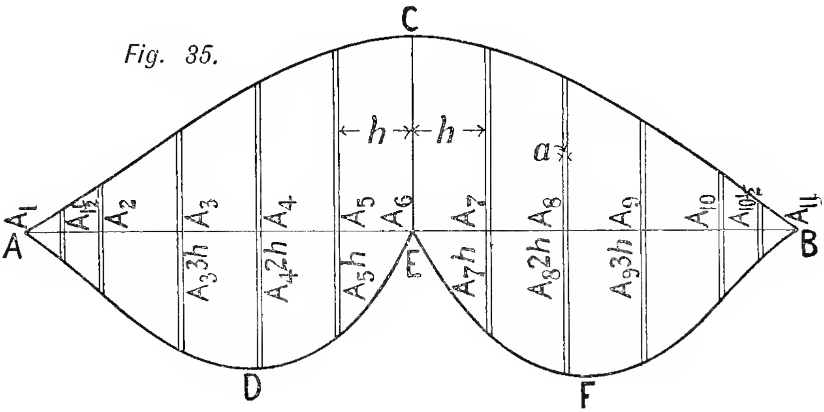
We now proceed to show how the longitudinal position of the centre of buoyancy may be obtained. Reverting to fig. 32, the transverse sections marked 1, $1\frac{1}{2}$, 2, 3, etc., are those at which the elementary layers or slices required for the moment calculation are taken, the number of divisions being arranged to suit the application of Simpson's First Rule. At each end, intermediate sections are introduced, the common distance being there reduced by a half.

Having calculated the various vertical areas, we first take a horizontal line *AB* equal on a convenient scale to the length of the vessel (fig. 35), and draw lines at right angles to it at points in the length marked off to correspond with the positions of the sections. On the upper side of *AB* we then plot the sectional areas just found and draw a curve *ACB* through the extremities of

* Multiplication by the common interval is done at the end as shown.

the ordinates, thus enclosing an area which represents the volume of the vessel below No. 1 waterplane. On the lower side of AB, the moments of the layers are set off. Station 6 is chosen as the axis of moments, as the areas of the moments curves may then be obtained by Simpson's first rule. Calling the vertical areas A_1, $A_{1\frac{1}{2}}$, A_2, A_3, etc., and the distance between them h, the moment of an elementary layer of very small thickness a at section 6 will be zero; at section 5, $A_5 h a$; at section 4, $A_4 2h a$; and so on to the left of the axis. To the right of the axis we have at section 7 a moment $A_7 h a$; at section 8, $A_8 2h a$; section 9, $A_9 3h a$, etc. As in the previous case, the moments are plotted as rectangles, the base a being, in each case, measured along the axis, and the other side of the rectangle, represented by the area and lever multiple appropriate to the section under consideration, erected as an ordinate. Fair curves ADE and EFB are drawn through the extremities of these little rectangles on each side of the axis as shown in fig. 35. The area enclosed by each of these curves and the axis AB represents the longitudinal moment of the volume on the side of the axis to which it refers.

Fig. 35.



By the principle of moments, we are obviously now able to write:—

$$\left.\begin{array}{c}\text{Horizontal distance of centre of buoyancy}\\ \text{from axis through station No. 6}\end{array}\right\} = \frac{\text{area } ADE - \text{area } EFB}{\text{area } ACB}$$

To illustrate the foregoing and show its application, take the following numerical example:—

The areas in square feet of the vertical transverse sections of a vessel up to the load waterplane are respectively, 0, 20, 60, 160, 230, 400, 750, 470, 350, 270, 100, 30, and 0. The sections are 12 feet apart, except at each end, where an intermediate one is introduced. It is required to determine the longitudinal position of the centre of buoyancy.

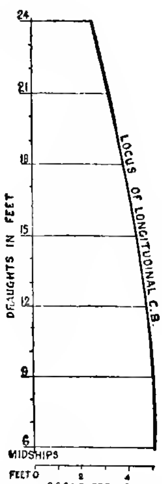
From inspection, we note at once that section No. 6 will form a suitable axis about which to take moments. Keeping this in view, and also remember-ing that, in calculating the moments, multiplication by the common interval is left to the end, we are able to tabulate the figures as follows:—

No. of Section.	Area of Section.	S.M.	Function of Areas.	Multiple for Leverage.	Function of Moments.
1	0	$\frac{1}{2}$	—	5	—
$1\frac{1}{2}$	20	2	40	$4\frac{1}{2}$	180
2	60	$1\frac{1}{2}$	90	4	360
3	160	4	640	3	1920
4	230	2	460	2	920
5	400	4	1600	1	1600
6	750	2	1500	—	4980
7	470	4	1880	1	1880
8	350	2	700	2	1400
9	270	4	1080	3	3240
10	100	$1\frac{1}{2}$	150	4	600
$10\frac{1}{2}$	30	2	60	$4\frac{1}{2}$	270
11	0	$\frac{1}{2}$	—	5	—
			8200		7390

Distance of centre of buoyancy from axis through section No. 6 $= \dfrac{(7390 - 4980)\frac{12}{3} \times 12}{8200 \times \frac{12}{3}}$

$= 3{\cdot}52$ ft. towards No. 7 section

Fig 36.



We saw how to construct a diagram giving the vertical position of the centre of buoyancy at all draughts; a diagram giving similar information can be made for the longitudinal position. The work, however, will be more laborious than in the previous case, as a new set of vertical areas must be found corresponding to each draught, and a complete moment calculation, similar to the one just worked out, made in each case. Assuming the horizontal positions of the centre of buoyancy found up to a series of draughts between the top of keel and the load-line, the diagram may be easily constructed. A vertical scale of draughts is taken, the horizontal distances of the centre of buoyancy, as calculated from some chosen axis, are marked off at the corresponding draughts, and a fair curve drawn through these points. Fig. 36 illustrates this diagram.

QUESTIONS ON CHAPTER II.

1. If two unequal weights be suspended one at either end of a weightless lever, find the point at which the lever must be supported in order to be exactly balanced. If the lever be 48 inches long and the weights 11 lbs. and 5 lbs. respectively, find the balancing point from the end loaded with 5 lbs.

Ans.—33 inches.

2. If weights of 5, 8, 11, 13 and 17 lbs. lie on a table of rectangular outline, their positions taken in the order given and measuring from one end being 3, 4, 5·5, 6, 7·5 feet, and from one side 1, 1·75, 2·5, 3, 2·75 feet, find the position of the point through which the resultant force acts.

Ans.—5·79 feet from end, 2·45 feet from side.

3. Define Centre of Gravity.—The equidistant ½ ordinates of a vessel's waterplane, beginning aft, are ·2, 6·4, 10·2, 11·0, 10·4, 7·0 and ·4 feet, and half ordinates introduced at the extremities in the usual way have values 4 feet and 4·3 feet; find the distance of the centre of gravity from the middle line.

Ans.—4·58 feet.

4. If the longitudinal distance between the ordinates in the preceding question be 14 feet, calculate the position of the centre of gravity with reference to the No. 4 ordinate.

Ans.—·43 feet forward of No. 4 ordinate.

5. Define Centre of Buoyancy.—The area of a ship's load waterplane is 7000 square feet, and the areas of other parallel waterplanes spaced 3 feet apart are respectively, 6500, 5500, 4000, and 2000 square feet (neglecting the volume below the lowest section); obtain the distance of the centre of buoyancy below the load waterplane.

Ans—5·03 feet below load waterplane.

6. Referring to the previous question, calculate the fall in the position of the centre of buoyancy as the vessel rises to each of the given waterplanes, except the last, and plot the locus of centres of buoyancy.

7. Explain how you would proceed to calculate the longitudinal position of the centre of buoyancy? As a practical example, obtain the position of the longitudinal position of the centre of buoyancy of a vessel, the areas of whose transverse vertical sections are, starting from aft, 4, 100, 180, 240, 260 242, 190, 120, and 8 square feet, the sections being spaced 15 feet apart.

Ans.—1·11 feet forward of No. 5 section.

CHAPTER III.

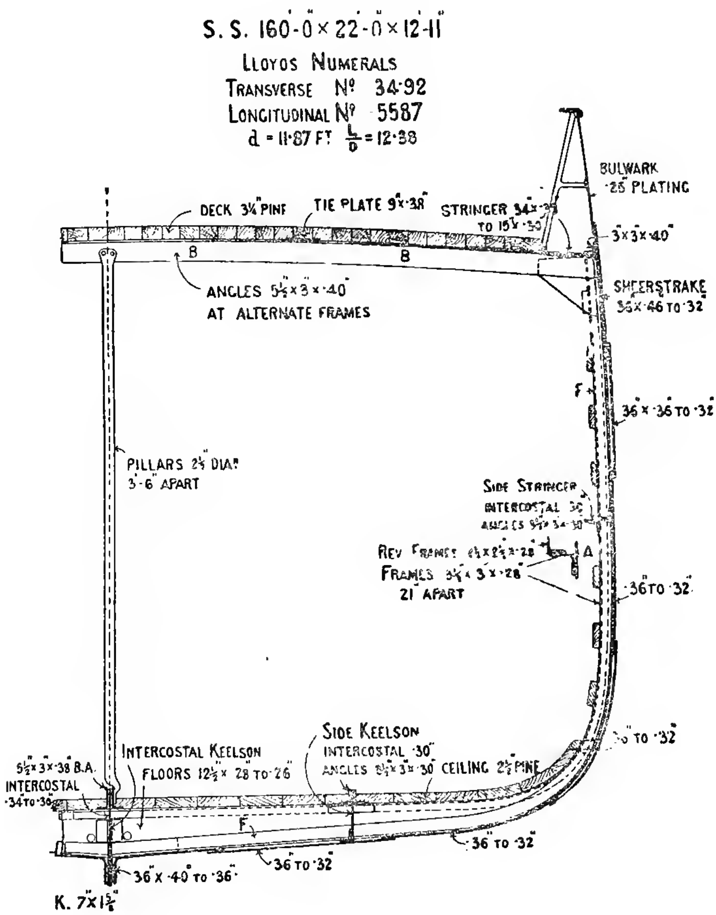
Outlines of Construction.

AT this stage it is desirable to obtain an acquaintance, in a general way at least, with the system of construction of the modern ship, and with the names of the principal parts; in a later chapter we shall take up details.

In the old days, when wood was the medium of construction, vessels were invariably built on what is known as the transverse system; and, as was perhaps natural, the earliest iron ships, when that material began to displace wood, were built on the same plan. There were, of course, important differences in the details of construction due to the great difference in the nature of the materials, but the general principle was in each case the same. As a foundation and sort of backbone to the structure, there was, for instance, the keel running fore-and-aft, and, at equal distances along the keel, transverse vertical frames, or ribs, erected to give the form of the vessel at each point, and to offer a convenient means of fitting the watertight skin or shell. At their upper end the transverse frames were joined by horizontal girders or beams, adapted to keep the frames to their proper shape, and to support a horizontal platform or deck. If the vessel were a large one, usually one or more decks might be fitted below the upper one, this being necessary for strength, and it might be for the convenience of stowing certain cargoes, or of housing passengers.

In fig. 37, which is the midship section of a small steel vessel built on the transverse system, are seen all the characteristics just referred to. *K*, the keel, is a steel or iron bar of considerable depth and thickness. The transverse frames, marked *F*, are angles running from the keel to the gunwale, and associated with bars of similar shape, called reverse frames from the circumstance of their looking in an opposite direction to the frames as shown in horizontal section at *A*. At the bottom of the vessel, deep vertical plates, called floor plates, are riveted to the frames and the reverse frames, the reverse frames being carried along at the upper edge of the floor plates, as shown. *B B* are the beams, which as above mentioned, tie the sides of the ship and resist any tendency to change of transverse form. Vertical change in the transverse form is resisted by means of the pillars, which tie the top and bottom parts of the structure together, and assist the floor plates to carry the cargo. The longitudinal shape is maintained by means of the keelsons and side stringers, which tie the transverse parts together, distribute the stresses, and make the framing into one united structure. The most

Fig. 37.



important part of all is the outer plating or skin, which gives the vessel its floating power. It will be seen to consist of strakes of plating riveted to each other and to the fore-and-aft flanges of the frames. The top deck is also covered in with plating or wood to give strength, and keep the water out. The strakes of plating running fore-and-aft on the outer ends of the beams, and connected to the shell by means of angles, are called stringers; they are valuable elements of strength, as we shall see when we come to consider the stresses to which a ship is liable.

From the diagram it will be seen that the material forming the structure is not evenly distributed throughout. For instance, the shell-plating is thickest at the top and at the bottom. Between the sheer strake, which runs in way of the gunwale or the top deck at side, and the bilge strakes, the plating is reduced in thickness, being a minimum about midway between these points. Also the centre keelson is much heavier than the keelsons and stringers higher up on the sides. The scantlings, too, are not the same right forward and aft. The 'midship thickness and sizes are only maintained for half the vessel's length, or thereabouts, and then a gradually tapering process is begun, minimum sizes being reached at the bow and stern. It should be mentioned that local requirements usually demand heavier materials just at the extreme ends, but in general the principle of reducing the scantlings as above is followed. We shall see presently the reason for all this.

Fig. 37 illustrates only the simplest form of construction of steel vessels. Departures have been made at different times, called forth by the desire of the owners to increase the value of their property as producers of wealth, and these departures have eventually resulted in considerable modifications in the structure of vessels. Thus we have water-ballast tanks. When first introduced these tanks were mere additions to the ship's load, but they have now become incorporated in the structure, and, as we shall see, have added immensely to its strength and safety. In vessels of wood, to build in such tanks was impossible, but in those of steel it is the natural thing to do, as the mild steel used in modern shipbuilding, owing to its nature, can be manipulated in such a manner as to ensure continuity of strength and absolute watertightness with the ballast tank as part of the hull.

Other departures brought about by commercial considerations have resulted in modifications of the framing; these we shall consider in detail when we come to deal more particularly with actual types of the modern cargo steamer. So far we have only attempted to obtain some familiarity with the various parts of a vessel's hull in order to follow intelligently a discussion of the stresses and strains to which ships are liable, which we propose to take up in the next chapter.

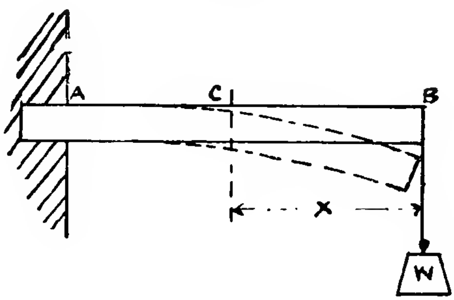
CHAPTER IV.

Bending Moments, Shearing Forces, Stresses, and Strains.

IN this chapter we propose to speak of the stresses and strains to which ships are liable, and as the same principles are involved in calculations of the strength of ships and of simple beams, it will help us to begin with the simplest cases and gradually lead up to those which are more difficult.

Take a beam $A\ B$ (fig. 38), fixed at one end and loaded with a weight W tons at the other, and consider the system of forces in operation at any

Fig. 38.



section. Take one at x feet from the extreme end B. Neglecting the weight of the beam itself, we have here acting :—

(1) A bending moment = $W x$ foot tons, tending to bend the beam as shown dotted.*

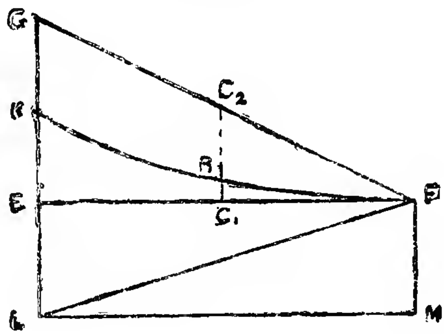
(2) A shearing force = W tons, tending to cause the portion of the beam $C\ B$ to move downwards relatively to the portion $A\ C$.

In this simple case, the bending moment obviously is a minimum at B and a maximum at A, since it varies directly with x. Also the shearing force is

* The deflection is shown much exaggerated for clearness.

the same for all sections of the beam from B to A. To express this in a diagram, take a line EF (fig. 39) to represent the length of the beam. At E set up an ordinate EG, representing on some scale the maximum bending moment, $W \times AB$ foot tons. Join GF; EGF is the diagram of bending moments. From it, by simple measurement, we can obtain the value of the bending moment acting at any point of the length of the beam. For instance, the

Fig. 39.

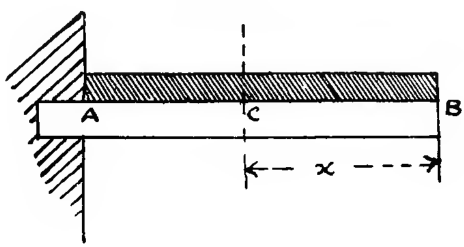


bending moment at a section C of the beam is given by the ordinate $C_1 C_2$ of the diagram, $C_1 F$ being marked off equal to CB.

For the diagram of shearing forces we have merely to construct a rectangle $ELMF$, on EF as base, the side EL representing to scale the force W.

If, instead of being concentrated at the outer end, the load be spread

Fig. 40.



evenly over the surface of the beam (fig. 40), at any section x feet from B, we shall have :—

(1) Bending moment $= wx \times \frac{x}{2} = w\frac{x^2}{2}$ foot tons, w being the load per foot of length. The curve of bending moments will now take the form KRF (fig. 39), and is obviously a parabola having its axis vertical.

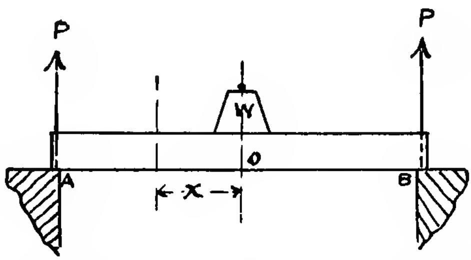
(2) Shearing force $= wx$ tons.

The shearing force will thus vary directly with x, will be zero at B and a maximum at A, where it will equal the total load. The shearing force diagram will be a triangle such as $E\,L\,F$, $E\,L$ giving to scale the shearing force at A.

Consider now a beam supported at each end and loaded in the middle. Fig. 41 illustrates the case. $A\,B$ is the length of the beam, W the load in tons, and P the re-action at each support. At any section x feet from the middle of the beam, the weight of the beam itself being neglected, we have :—

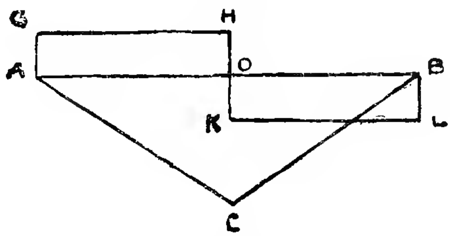
Bending moment $= P\,(A\,o - x)$ foot tons.
Shearing force $= P$ tons

Fig. 41.



The bending moment increases directly as x diminishes. It is therefore a maximum at o, the middle of the beam, and zero at either end $A\,C\,B$, fig. 42 being the diagram. The tendency here is for the beam to become curved convex side downwards, the ends rising relatively to the middle, and it is convenient to describe the bending moment as negative, the diagram $A\,C\,B$ being drawn below the line to indicate this. Where a bending moment gives

Fig. 42.



rise to an opposite tendency, that is for the upper side of the beam to become convex, as in the previous examples, and in the case of beams supported at the middle and loaded at each end or uniformly, it is described as positive, and the diagram is drawn above the line. With regard to the shearing forces, it should be noted that at sections to the left of the middle, the tendency is to cause the left-hand portion of the beam to move upwards relatively to the right, and at sections to the right of the middle the reverse of this, the forces acting being conveniently described as positive and negative respectively. This is expressed in the diagram by plotting the shearing forces for sections to the left of O above the base line, and those for sections to the right of O, below that line. The diagram takes the form $A\,G\,H\,K\,L\,B$, as shown in fig. 42.

As a numerical example, let the length of the beam be 20 feet, and the concentrated load at the middle of it, 12 tons. Neglecting the weight of the beam, the re-actions at the supports will each be 6 tons. At a point, say 2 feet to the left of O, the middle point of the beam, we have:—

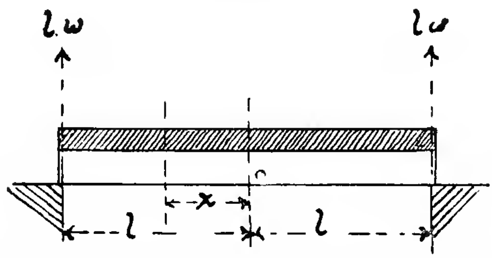
Bending moment $= 6\,(10 - 2) = 48$ foot tons,

and shearing force $= +\,6$ tons.

If the diagram (fig. 42) had been constructed for this beam, the values above of bending moment and shearing force corresponding to a section 2 feet from the middle of the beam, could have been obtained by reading off the ordinates at the corresponding point in the diagram.

If, instead of concentrated at the middle, the load be distributed equally throughout the length of the beam (fig. 43), the diagram of shearing forces and bending moments will be modified somewhat from that given above. Calling the length of the beam $2\,l$, and the load per foot w tons, we have for the re-action

Fig. 43.



at either end of the beam, $l\,w$ tons. At any section of the beam, say x feet to the left of the middle point, there will be acting—

(1) A bending moment $= l\,w\,(l - x) - \frac{w}{2}(l - x)\,(l - x) = \frac{w}{2}(l^2 - x^2)$ foot tons.

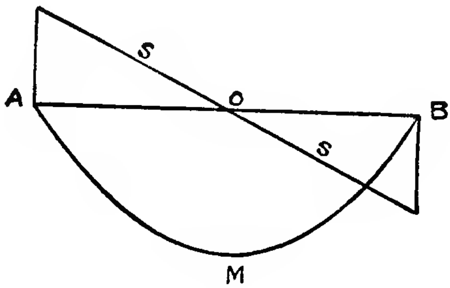
(2) A shearing force $= l\,w - (l - x)\,w = w\,x$ tons.

In plotting the diagrams, we note from the equation above that the curve of bending moments will be a parabola, that it will have zero values at each end of the beam, since x will there be equal to either $+l$ or $-l$, and a maximum value at the middle where $x = o$. The shearing force diagram is obviously a straight line, since the ordinates vary directly as x; it will have a zero value at O, and maximum values at the supports. Calling, as before, the shearing forces to the left positive and those to the right negative, these maximum values are, respectively, $+l\,w$ tons and $-l\,w$ tons. Fig. 44 illustrates the diagram for a distributed load, and it should be compared with fig. 42 for a concentrated load. As an exercise it would be interesting to construct a diagram, assuming a distributed load, in an actual case, say that of the

20-feet beam previously mentioned; but we leave the student to do this for himself.

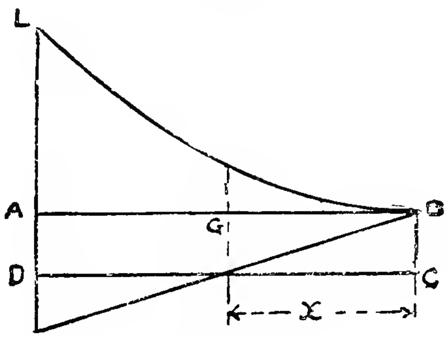
Diagrams of bending moments and shearing forces may be derived by a graphic process, and where the loads are irregularly distributed, as, for instance, in ship problems, this graphic process is the one most convenient to follow. We propose, therefore, to describe it briefly.

Fig. 44.



For this purpose, let us consider again the beam fixed at one end and loaded uniformly. By the proposed method we must start with a curve or diagram of loads. The weight on the beam, including its own weight from A to B, being so much per foot of length, may be represented by the rectangle $A\,B\,C\,D$ (fig. 45). This load is supported by the re-action of the wall

Fig. 45.



on the portion of the beam embedded in it, but we shall only consider the external forces acting on the beam from the wall face outwards.

Now, we know that at any point x feet from the end of the beam (fig. 45) the shearing force $= w\,x$ tons; that is, the ordinate of the shearing force diagram equals the area of the diagram of loads from the end of the beam up to the point under consideration, and, therefore, as already shown, the diagram is a triangle.

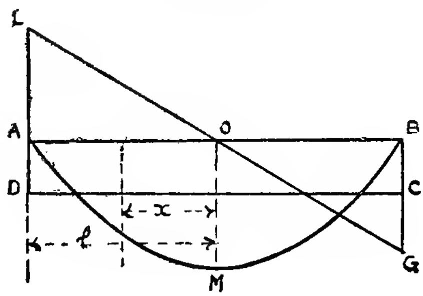
Take now the bending moment. For any section of the beam, say at G,

$$\text{bending moment} = w\,x \times \frac{x}{2} \text{ foot tons, or} = \text{shearing force} \times \frac{x}{2} \text{ foot tons,}$$

that is, equals the area of the shearing force diagram from B to G.

To construct the bending moment diagram it is therefore only necessary to take certain points on the beam, to calculate the area of the shearing force diagram from the end of beam to these points, and to plot the bending moments thus derived on a convenient scale. $B\,L\,A\,B$ (fig. 45), is the form that such a diagram would take in the present case.

Fig. 46.



The construction is quite as simple for a beam loaded uniformly and supported at the ends. Fig. 46 illustrates this case, $A\,B$ being the beam drawn to a convenient scale and $D\,A\,B\,C$ the diagram of loads upon it between the points of support, including its own weight. In plotting the diagram of shearing forces we begin at, say, the left-hand point of support, at which the shearing force is positive and equal to half the load, that is, to half the area of $D\,A\,B\,C$. Its value may be plotted as $A\,L$. From L the diagram of shearing force falls in a straight line, the value of an ordinate at any section, x feet say, from o, the middle point of the beam, being the shearing force at A minus the load represented by the portion of the area of the rectangle $D\,A\,B\,C$ from A to the section. At o the half load and the re-action at A are equal, and there is therefore no shearing force. At sections to the right of o the load exceeds the re-action at A and the shearing force is negative, reaching a maximum value at B, as previously shown.

For the bending moment of a beam loaded as described at a section x feet from middle, we have deduced the equation—

$$\text{Bending moment} = \frac{w}{2}(l^2 - x^2) \text{ foot tons,}$$

l and w having the values previously given.

This may be written—

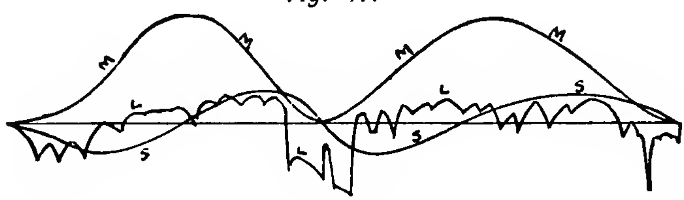
$$\text{Bending moment} = (l - x)\left(\frac{lw + xw}{2}\right),$$

which obviously expresses the area of the shearing force diagram from A to the point considered. Thus, having obtained the shearing force diagram, to get the curve of bending moment, it is only necessary to calculate the area of the former from either end of the beam to various points in its length, to plot these areas as ordinates and to draw a fair curve through their extremities. $AMBA$ is the bending moment diagram for this beam.

As the same principles apply, we are now in a position to consider the case of a floating vessel. In the first place, take a vessel, say a steamer, in the "light" condition, that is to say, completely built, and with all machinery aboard and water in boilers, but without bunker coal, cargo, or consumable stores; and assume her to be floating freely and at rest in still water.

A moment's consideration will make it clear that a tendency to longitudinal straining, with which we are here dealing, must be principally caused by the

*Fig. 47.**



action of the vertical forces made up of the vertical components of the water pressures acting upwards, and of the weight of all the particles in the mass of the vessel acting downwards. In fig. 47, LLL is a diagram of loads for a "light" vessel. We shall show in detail, presently, when we consider the important case of a ship among waves, how such diagrams are constructed; in the meantime it is sufficient to note that fig. 47 shows weight in excess of buoyancy at each end and amidships, and elsewhere, except at one point forward, buoyancy in excess of weight. The excess of weight is obviously due to the small volume and the great weight of the structure at the extremities, and to the concentration of the machinery amidships, and the excess of buoyancy to the empty holds.

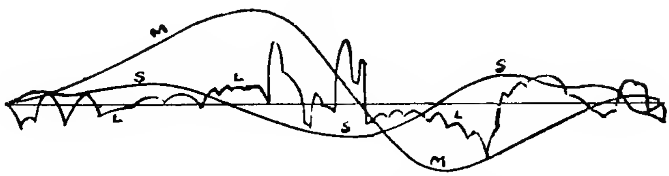
In fig. 47, diagrams of shearing forces and bending moments are also shown. The curve of shearing forces at any point in the length we know to be the area of the curve of loads from either end up to that point, reckoning the portions of area above the axis positive and those below negative. In the present case, the curve takes the form $SSSS$. In the same way, ordinates of the curve of bending moments are given by the area of the diagram of shearing

* The diagrams represented by figures 47 and 48 are taken from a paper read before the North East Coast Institution of Engineers and Shipbuilders, by Mr. Bergström in 1889.

forces from either end up to the points in the length at which they occur. We thus obtain the curve *M M M*, the ordinates of which are seen to be a maximum about the middle of the forward and after holds, and a minimum at about middle length. With a homogeneous cargo filling the holds, the case becomes considerably modified. The curve of loads is now as shown at *L L L* (fig. 48).

It is seen to cross and recross the base line at many points. In way of the machinery the weight is slightly in excess, but is much more so in the mainhold. In the forward and after holds, buoyancy is again predominant,

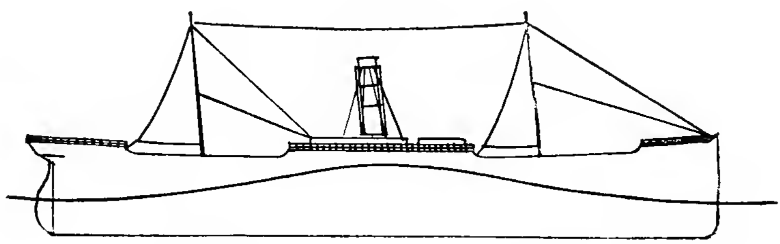
Fig. 48.



while weight is in excess at the extreme ends. The shearing force curve now crosses the axis at three points in the length, while the curve of *B M* has two maximum values, one forward and one aft, tending to strain the vessel in opposite directions.

The bending moment and consequent straining effects on a vessel in still water are, as a rule, inconsiderable compared with those she must withstand when among waves at sea. It is then that the ultimate strength of the structure is called out, in some cases with disastrous results.

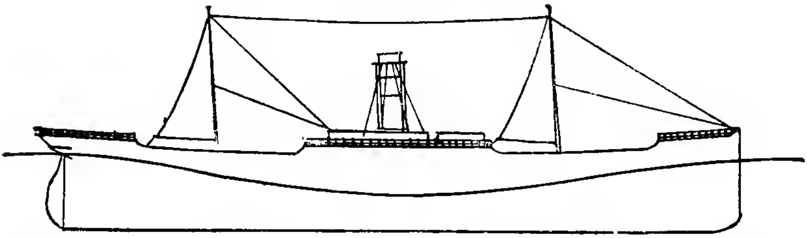
Fig. 49.



Let us try to conceive for a moment the position of a vessel when in a seaway. If the waves be of regular form and speed, the vessel may, at a given instant, be in one of several positions. She may be traversing the waves in a line at right angles to the crests, or be rolling in the trough between the waves: or she may occupy some intermediate position with her length at an oblique angle to the crest lines. The bending moment will be different in every position, and the hull should be designed strong enough for the worst case.

Of the above conditions, the first one, in which the vessel is assumed at right angles to line of wave crests, has been most frequently investigated. It is the condition in which longitudinal straining is greatest, and may, therefore, in this respect be considered to include the other two cases.

Fig. 50.



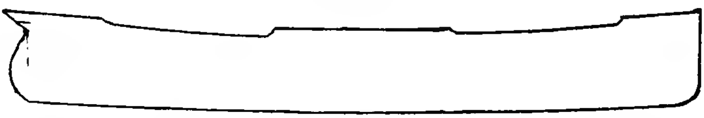
Taking the first condition, we note that it has two critical phases. One of these is indicated in fig. 49, where the vessel, assumed in full sea trim with cargo aboard, but with stores and bunker coal consumed as at the end of a voyage, is shown poised instantaneously in an upright position on the crest of a wave, the latter being at mid-length. The other phase is when the vessel,

Fig. 51.

complete with bunker coal and stores as well as cargo—her worst condition in this case—has a trough amidships and a crest at each end (*see* fig. 50).

As we shall see presently, when we construct the diagrams, the bending moments are reversed in the two cases. The general straining tendency, with the crest amidships, ordinarily is to cause the middle to rise relatively to the

Fig. 52.



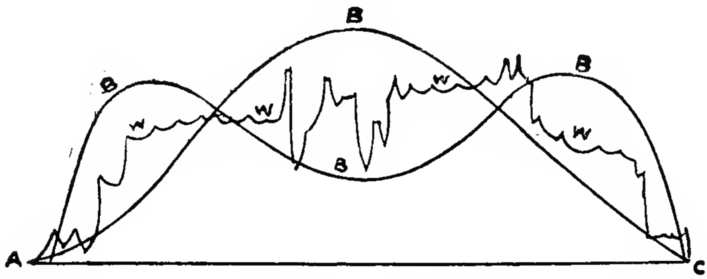
ends, as shown in fig. 51, and with the hollow amidships, to cause the middle to sink relatively to the ends, as in fig. 52. These strains are known as hogging and sagging, respectively.

Diagrams of shearing forces and bending moments for a vessel situated as indicated in figs. 49 and 50, are constructed on the assumption that the waves are stationary, and that the problem may be treated as a purely statical one.

No note is ordinarily taken of the fact that the quick passage of waves past a vessel, particularly one of relatively fine ends, has a tendency to develop an up and down motion in her, altering her virtual weight and buoyancy from moment to moment, and consequently directly affecting the magnitude of the bending moment, and, therefore, the strains brought upon her. It may be mentioned that, where they have been specially allowed for, these vertical oscillations have been found to reduce hogging and increase sagging strains.

In these diagrams, too, it is not usual to allow for the difference in the water pressures in the waves as compared with those in still water, although it is known that they are less in the wave crests and greater in the hollows than at the corresponding depths in still water, due to the effect of the orbital motion of the water particles in the waves, the general effect tending to a reduction of both hogging and sagging bending moments. Other points which are ignored, or are found impracticable so far to deal with, are the effect of longitudinal oscillations, that is, pitching and 'scending, and of rolling motions, although clearly they may have considerable influence on the bending moments.

*Fig. 53.**



It is obvious, then, that diagrams as ordinarily constructed are only approximately true, and should be used merely as a means of comparison between vessels. When so employed, they are most valuable as a guide in new designs for determining the lines to be followed in making departures in construction.

Taking a vessel, then, in the condition exhibited in fig. 49, viz., with a wave† crest amidships, we begin, as in the simpler case of the vessel in still water, by constructing a curve of loads. Such a curve, we know, shows the difference of the forces of weight and buoyancy at all points in the length, and to obtain it we must first find the values of these forces. A curve of forces of buoyancy is easily drawn. It is only necessary to calculate the buoyancy per foot of length, at various cross sections, usually taken at equal distances apart, and then in a diagram, whose base line represents the length of the

---

* The diagrams represented by figures 53, 54, and 55 are from Mr. Bergström's paper referred to on page 51.

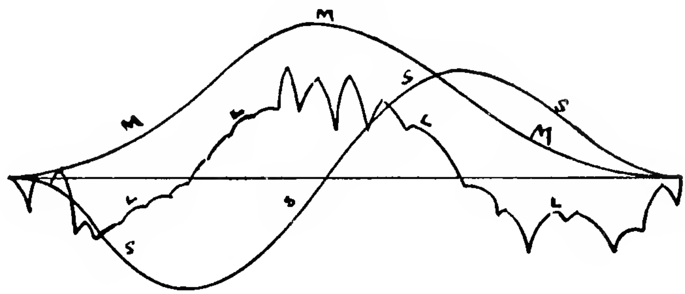
† In these calculations it is usual to assume the wave to have a length equal to the length of the ship, and a height of $\frac{1}{20}$ of its length.

vessel, to mark off the results on some convenient scale as ordinates at corresponding points. Thus we obtain the curve *A B C* (fig. 53).

To draw the curve of weights is more difficult. There are various ways of doing this leading to the same result. One of them is to deal first with the hull material, by calculating the weight per frame space of that which is continuous at chosen points throughout the length, and plotting the results on the same scale as employed for the buoyant forces at corresponding points on the diagram containing the curve of buoyancy; and then on the curve obtained by penning a batten through these points, super-imposing the irregular weights, such as bulkheads, stern-post, propeller and rudder, engines and boilers, tunnel and shafting, coal in bunkers, cargo in holds, etc.

The irregular weights are conveniently plotted as rectangles on bases extending over a portion of the length in the diagram corresponding to that occupied by them on the vessel. In the case of bulkheads, however, the

Fig. 54.



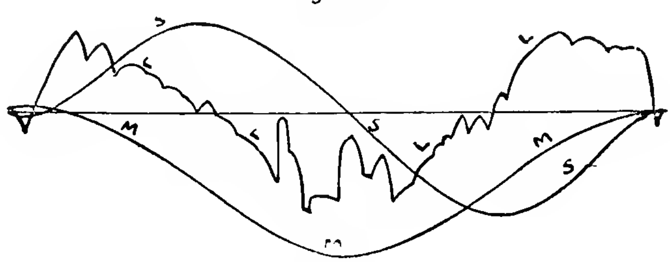
weight is sometimes spread over a frame space, and in the case of coal and of cargo, the weight per foot of hold space is plotted. In the example chosen, a homogeneous cargo of a density to completely fill the holds and bring the vessel to her load-line has been assumed. This is usual in strength calculations, as it would be obviously impracticable to exactly allow for a general cargo, owing to the difficulty of obtaining the positions and weights of the various portions of it. The complete weight curve or diagram is of a very irregular form, as will be seen from the figure. It should be noted that the weight curve must be equal in area to the curve of buoyant forces and have its centre of gravity in the same vertical line, as these are the conditions of equilibrium. If the area of the weight curve be greater or less than that of the curve of buoyancy, the vessel will not float at the assumed water-line but at a deeper or shallower draught, as the case may be; and if their centres of gravity be in different verticals a trimming moment will be in operation, showing the assumed line to be wrong also as to trim.

Having got the curves of weight and buoyancy to correspond as required, we note that the actual load on the vessel at any point is the unbalanced force acting at the point; that is the difference between the two curves. These differences, measured at numerous ordinates and plotted to the same scale as the diagrams of weight and buoyancy, give a curve, or diagram of loads, marked *LLL* in fig. 54. The conditions of equilibrium required in this case are that the area of the portion of the diagram above the base line shall equal the area below that line; also that the centres of gravity of the upper and lower areas shall be in the same vertical.

To construct the curve of shearing forces is a simple matter, since the area of the curve of loads from the end to any point, is the value of the shearing force at that point. In the same way, the curve of bending moment is obtained by integrating the diagram of shearing forces. These two curves in the case assumed—that of an ordinary well-deck cargo steamer—take the forms *SSS* and *MMM* in fig. 54.

In constructing diagrams of loads, shearing forces, and bending moments, for the other extreme condition in which the vessel is astride two consecutive waves, with a hollow amidships (fig. 50), the main point of difference is in the

Fig. 55.



curve of buoyancy, which will now take the form *BBB* (fig. 53). The curve of weights will remain as before,* but the curve of loads will, of course, be of a different form, the weights being in excess amidships and the supporting forces in excess at each end (fig. 55), the tendency being, as already pointed out, to develop sagging strains amidships.

RESISTANCE TO CHANGE OF FORM.—The foregoing shearing forces and bending moments give rise to stresses in the materials, and to consequent tendencies to change of form in the structure. It is, of course, important to prevent the stresses on the materials becoming sufficient to cause rupture, and the tendencies to change of form from becoming actual permanent deformation. We shall show presently that this may be done in three ways—first, by increasing the weight of materials; second, and preferably, by judicious disposition of materials; third, by design of structure.

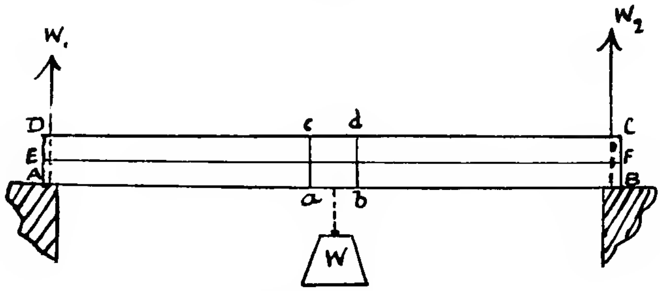
To simplify our explanations, we shall take the case of an ordinary rectangular beam. A ship is really a huge beam or girder, and, consequently,

* Except that the bunker coal and stores, assumed consumed in previous case, must now be allowed for.

what is true for the simple beam when under shearing stresses and bending moments, is also true for the ship.

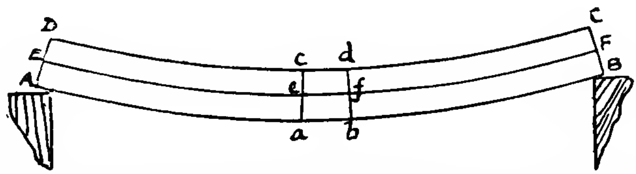
A B C D, fig. 56, is a rectangular beam, which we will assume to be of some elastic material, such as will yield equally under tensional or compressive stresses.

Fig. 56.



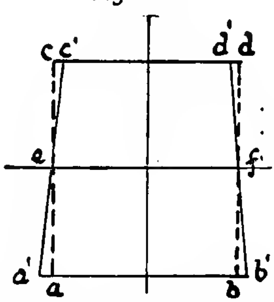
Draw a horizontal line at mid height, and at mid length also draw two vertical lines, *a c*, *b d*, at a little distance apart. Now, place this beam on supports at each end, and load it at the middle and observe what happens (fig. 57).

Fig. 57.



The beam will be seen to sag in the middle; also the top surface *D C* will be found reduced in length, the bottom surface increased, while *E F*, the line at mid height, though taking the curve of the beam, will have its length

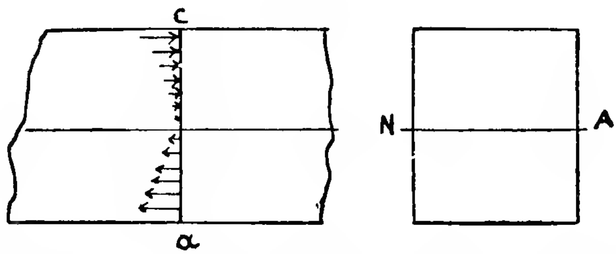
Fig. 58.



unchanged. The lines *a c* and *b d*, which were drawn vertically on the side of the beam, are now inclined to each other, although still straight. Let fig. 58 be an enlarged sketch of this portion of the beam. The original straight beam is shown by dotted, and the beam as bent, by full lines.

Now the stress due to the external bending moment has obviously increased ab to a^1b^1, and reduced dc to d^1c^1. There is thus a compressive stress on the upper part of the beam, and a tensional stress on the lower part. Since the strains are reduced from the outside of the beam to zero at the middle, ef being unchanged in length, the stresses must be correspondingly reduced. Also, it is clear that the strain, and, therefore, the stress at any point in the height of the beam, varies in direct proportion to the distance of that point from the line ef; for example, the strain at b is double that at a point midway between b and f. The surface of which ef is a portion of the section, is known as the neutral surface. Let us now consider the equilibrium of the beam as loaded in fig. 56 at any section such as ac. Taking the portion of beam to the right, there is, as we have seen, an external bending moment due to W_2. No other external forces are acting, if we neglect the weight of the beam itself, so that this moment must be counteracted by the sum of the moments of the molecular forces of the portion of beam to the left acting at the section. Besides these moments there is due to the load

Fig. 59.



a vertical shearing force in operation tending to move the right-hand portion of the beam upwards relatively to the left. This shearing force is counteracted by the resistance of the fibres of material to shearing. We shall return to this point again.

In fig. 59 we show enlarged end and side views of the section ac. NA is a section of the neutral surface and is called the neutral axis at ac. At NA there is no stress due to bending. Above and below this line the molecular stresses push and pull the beam, as shown by the arrows. As the beam does not move in the direction of its length, these horizontal stresses must neutralise each other, that is—

$$\text{Pulling stress} + \text{pushing stresses} = 0 \quad (1).$$

We have seen that the stress at any point of a section varies directly with its distance from the neutral axis: if, therefore, we know the stress at any one point either above or below NA, we are able to write down equation (1), because in materials such as steel or wrought iron the resistance to compression and tension, within the elastic limits, is the same.

When we know the external bending moment and the area and form of the section of the beam, we are able to find the internal stress at any point of the section. Let us find it at unit distance, say one inch from the neutral axis. Calling the stress in tons per square inch at this point s, at two inches from NA it will be $2s$ tons, and, generally, at y inches either above or below NA, it will be—

$$ys \text{ tons.}$$

On a small portion of area a, at this distance from the neutral axis, the stress will be—

$$ysa \text{ tons}$$

and the total stress acting at the section will be the sum of all such elements; we may therefore write:—

Total pushing and pulling stresses at section $ac = s\Sigma ya$ tons,

where the symbol Σ signifies that the sum of the elementary forces is taken. Now there must be no resultant stress acting at the section, so that—

$$s\Sigma ya = 0.$$

But Σya is the moment of the area, and for this to be zero, the neutral axis, about which the moments have been taken, must pass through the centre of gravity of the area of the section. This fixes the neutral axis, and is an important point to remember. To get now the stress at a point one inch from this axis as required, we must equate the sum of the moments of the internal stresses about the neutral axis to M, the external bending moment at the section.

At any distance y inches from NA, either above or below, the moment of the stress acting on a small portion of area a is $sya \times y = sy^2a$ inch tons. And for the whole section we may write:—Sum of moments of internal stresses $= s\Sigma y^2a$ inch tons.

Now, the expression Σy^2a is a well-known quantity in physics: it is called the moment of inertia of the section of the beam. If it be represented by I, and the internal and external moments be equated, we get:—$sI = M$. So that

s, the stress in tons at one inch from $NA, = \frac{M}{I}$.

To find the value of the stress at any point in the section, it is only necessary to multiply s by the distance of the point from the neutral axis. Thus, if y inches be the distance of the upper or lower surface of the beam from NA, and p be the stress there, we shall have:—

Maximum compressive or tensional stress in tons at section $ac = p = y\frac{M}{I}$. (2)

This is the formula which must always be employed when dealing with the strength of beams and girders, such as ships, and is worthy of careful study. It shows that the maximum stress varies directly as M, the external bending moment, and inversely as $\frac{I}{y}$. Consequently, with a given bending moment, the stress is reduced by increasing this quantity and increased by reducing it.

It is easy now to understand why in a ship or other loaded beam a reduction of stress is effected by increasing the sectional area, that is, the weight of materials, by judiciously disposing them, or by changing the design. It merely means that in each case the modulus $\frac{I}{y}$ is increased. Increase of sectional area directly increases the moment of inertia I; the same effect is attained without increase of sectional area by concentrating the latter at points remote from the neutral axis. Deepening the beam or girder will affect the modulus in two ways; I will be increased, but so also will y; as I varies as y^2, however, the effect on the whole will be to increase the modulus and reduce the stress.

It is for these reasons that in the case of a ship it is desirable to have the thickest plates at the upper deck stringers and sheerstrakes, and at the keel and bottom plates, and the thinnest plates midway between deck and bottom, the vicinity of the neutral axis in a ship. The above formula, indeed, tells us that at the neutral axis there is no stress on the materials due to bending moment, and it would thus appear that the scantlings at the neutral axis might be reduced indefinitely. It happens, however, that at this place in the depth a horizontal sliding or shearing action, which is developed by the variation of the bending moment from point to point of the length, and which tends to push the upper and lower portions of the structure in opposite directions, has its maximum value. To counteract this straining tendency, a considerable sectional area of material is needed in the vicinity of the neutral axis. We shall deal more fully with this point presently.

Another reason against thinning down too much the side plating of ships is found in the consideration that when rolling excessively at sea, the sides may frequently become, approximately at least, the top and bottom of the girder, and be called upon to withstand considerable bending stresses.

To apply formula (2) to find the stress at any point of a beam, we must know three things. We must know the external bending moment at the section containing the point, the position of the centre of gravity of the area of the section, and the value of the moment of inertia of the sectional area about a horizontal axis through its centre of gravity. We propose to show in detail how the work is done in the case of a ship, but before dealing with so complex a girder, we shall take one or two practical examples of simple beams. In a previous page we have explained how the external bending moment may always be found; we may therefore assume this item as known. Take then, as a first case, a steel beam of rectangular section 20 feet long, 12 inches deep, and 3 inches thick, under a bending moment of 600 inch-tons at the middle of its length, and let us determine the maximum stress on the material. Begin by writing down the stress formula, viz.—

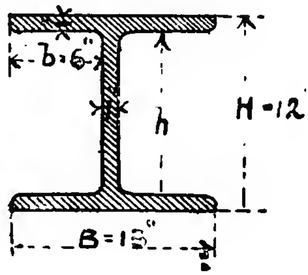
$$p = \frac{M}{I} y.$$

In the above beam the centre of gravity is at mid-depth; therefore $y = 6$ ins. Also the formula for the moment of inertia of the section, in this case a rectangle, about a horizontal axis through its centre of gravity, is $\frac{Ah^2}{12}$, where A is the area of the section and h the full depth. Substituting the given values, we have—

$$I = \frac{36 \times 12 \times 12}{12} = 432 \text{ in.}^4$$

$$\text{and therefore } p = \frac{600 \times 6}{432} = 8{\cdot}3 \text{ tons per sq. inch.}$$

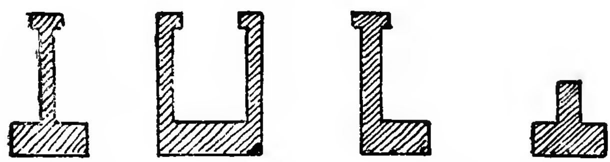
Fig. 60.



Taking the strength of steel at 30 tons per square inch, this stress allows a factor of safety of rather more than $3\frac{1}{2}$, which, in most cases, would be too low, 5 to 6 being common for ship work. The form* of section above given is by no means the most economical for steel beams. This material admits of being rolled into many forms, and to show the great importance of distribution of material as a means of increasing the strength of beams against bending, let us assume the length, depth, and sectional area, and therefore the weight, to remain as before, but the form to be as in fig. 60. The only additional work

* Within elastic limits mild steel and wrought iron are equally strong under compression or tension, but this is by no means true of all materials. Cast iron, for instance, will withstand a six times greater stress under compression than under tension: wood, on the other hand, has its greatest strength under tension. In such cases, for maximum strength on minimum weight,

Fig. 61.



the section must be of special design. The axis of moments must still pass through the centre of gravity, but the stress may be reduced on the weaker side by concentrating the material on that side near the neutral axis. For example, beams loaded at the middle, and supported at the ends, if of cast iron, to be of economical design should have cross sections of such forms as indicated in fig. 61.

to be done here is to find the moment of inertia of the new section about the neutral axis, which, as in the previous case, is at mid-height. The formula for the moment of inertia in this case is—

$$I = \frac{B H^3 - 2 b h^3}{12}$$

where H is the full depth of beam, h the distance between the flanges, B the full breadth, and b the breadth from the outer edge of the flange to the side of the web. Substituting the values given in fig. 60—

$$I = \frac{13 \times 1728 - 2 \times 6 \times 1000}{12} = 872 \text{ in.}^4$$

We therefore have—

Stress at top and bottom of beam $= \dfrac{600 \times 6}{872} = 4 \cdot 1$ tons per square inch,

a maximum stress which is only half of the previous one.

The above is only an illustration; for various reasons, girders of this section are not usually rolled with flanges of greater width than 6 to 7 inches. Taking them at 7 inches, and increasing their thickness to $1\frac{3}{8}$ inches say, with the same weight of material, a girder of 18 inches depth and $1\frac{3}{16}$ inches web could be obtained. The moment of inertia of such a girder would be 1685; and, under the same bending moment of 600 inch-tons, the stress on the upper and lower flanges would be—

$$p = \frac{600 \times 9}{1685} = 3 \cdot 2 \text{ tons per square inch.}$$

Let us turn now to the case of a floating ship. We have seen how to obtain the external bending moment, and to apply the stress formula, it only remains to determine for the material at the transverse section under the maximum bending moment, a method of fixing the position of the neutral axis, and of calculating the moment of inertia about that axis. Now, as we know that the neutral axis passes through the centre of gravity of the sectional area, its position may therefore be easily found. As we shall see presently, the calculation involved is conducted simultaneously with that for the moment of inertia.

In setting out to find the moment of inertia we must bear in mind that we are dealing with a built girder, and that only continuous material lying in a longitudinal direction is to be considered as available for resisting longitudinal strains. In ordinary cases the maximum bending moment occurs at about mid-length; we must therefore choose the weakest section in this vicinity for the moment of inertia calculation, as, of course, if straining were to take place, it would be at this section. Careful note should be made of the fact that material under tension must be calculated minus the area of the holes for the rivets joining the frames to the shell plating, the beams to the deck-plating, etc. This precaution need not be taken with the material in compression, as the rivet, if well fitted, will be as effective to resist this stress as the unpunched plate. Where continuous wood decks are fitted, they are sometimes allowed for, wood being considered equivalent to about $\frac{1}{16}$ of its sectional area in steel. In the case of tension, this must be reduced on account of the butts, which are usually separated by three passing strakes, also on account of the bolt holes. For compression the full area is taken. In modern cargo steamers continuous wood decks are seldom fitted, and there are none in the vessel whose moment of inertia calculation is given below.

As already mentioned, the conditions dealt with in these strength calculations are those depicted in figs. 49 and 50. In the first case, hogging strains usually predominate in ordinary vessels, the upper material being in tension and the lower in compression. In the second case, sagging strains would be expected, causing compressive stresses in the upper works and tensional stresses below. Since the rivet holes require to be deducted from the upper material in the moment of inertia calculations for hogging strains, and from the lower material in that for sagging strains, obviously a separate calculation is needed for each case. Dr. Bruhn* has pointed out that the necessity of two calculations may be obviated by obtaining the moment of inertia without correcting for the rivet holes, the stress so obtained being afterwards increased inversely with the reduced sectional area. The results obtained by this method do not differ much from those by the ordinary one, while the work is less.

In the following example we show in detail how to find the moment of inertia for a cargo steamer of modern type subjected to a hogging bending moment. It will be observed that the full sectional areas are tabulated, the sum of those of the parts in tension being reduced by $\frac{1}{7}$ as an allowance for a line of rivet holes at a frame. It will also be noted that the moment of inertia, in the first instance, is obtained about an assumed axis, the position of the neutral axis being unknown; that the distance between the neutral axis and the assumed one is next determined, and that the value of the moment of inertia about the neutral axis, which is what we require, is obtained from that about the assumed axis, by employing the well-known property of the moment of inertia expressed by the formula: $I = I_1 - A h^2$.

Where I = moment of inertia about neutral axis.
I_1 = moment of inertia about assumed axis.
A = area of material in section.
h = distance between axes.

This principle is also employed in the first instance to obtain the moment of inertia for each item about the assumed axis. For items of small scantlings in the direction of the depth of girder, the moment of inertia is expressed with sufficient accuracy by multiplying the areas by the squares of their dis-

* See his paper on *Stresses at the Discontinuities of a Ship's Structure*, read before the Institution of Naval Architects in 1899.

tances from the axis as in column 7. In the case of the side plating, however, and the vertical parts of the double bottom, such as the centre girder, margin plate, and intercostals, the figures of column 6 have to be increased by the moment of inertia of each item about an axis through its centre of gravity, that is, $\frac{1}{12} A d^2$, where d is the depth of the item, and A the sectional area; these quantities are given in column 8.

The moment of inertia being obtained, the stress in tons per square inch on material at any distance y, either above or below the neutral axis, is quickly found, since $p = \frac{M}{I} y$.

It should be remarked that when applied to a large girder like a ship, special units are employed, M being in foot-tons, A in sq. inches, y in feet, I in feet^2 and inches^2.

Moment of Inertia Calculation.

(Ship under a hogging strain).

S.S. 350′ 0″ × 50′ 74″ × 28′ 0″. Assumed neutral axis above base, 16′ 0″. Depth from base line to bridge deck, 36·10 feet.

Below Assumed Axis.

Items.	Scantlings in Inches.	Sectional Areas $= A$	C.G. from N.A. $= h$	Moments of Areas $A \times h$	h^2	$A \times h^2$	$\frac{1}{12} A d^2$ d = Depth of Items.
½ Centre Girder,	$44 \times \frac{5}{20}$	11·0	14·25	156·7	203	2233	10
Top Angle,	$4 \times 4 \times \frac{10}{20}$	3·8	12·5	47·5	156	593	
Bottom Angle,	$4\frac{1}{2} \times 4\frac{1}{2} \times \frac{12}{20}$	5·0	16·0	80·0	256	1280	
In. Bot. Plating,	$255 \times \frac{8}{20}$	102·0	12·5	1275·0	156	15912	
Margin Plate,	$34 \times \frac{10}{20}$	17·3	14·1	243·9	199	3443	12
Margin Angle,	$4 \times 4 \times \frac{9}{20}$	3·4	15·4	52·3	237	806	
Side Stringer,	$\left\{ \begin{array}{c} 4\frac{1}{2} \times \frac{10}{20} \\ 6\frac{1}{2} \times 4\frac{1}{2} \times \frac{13}{20} \end{array} \right\}$	8·9	3·6	32·0	13	116	
,,	,,	8·9	8·5	75·6	72	641	
½ Keel Plate,	$21 \times \frac{20}{20}$	21·0	16·1	338·1	259	5439	
B Strake,	$54 \times \frac{15}{20}$	40·5	15·95	646·0	254	10287	
C ,,	$58 \times \frac{13}{20}$	37·7	15·85	597·5	251	9463	
D ,,	$56 \times \frac{12}{20}$	33·6	15·7	527·5	246	8266	
E ,,	$56 \times \frac{13}{20}$	36·4	15·6	567·8	243	8845	
F ,,	$58 \times \frac{12}{20}$	34·8	15·5	539·4	240	8352	
G ,,	$57 \times \frac{13}{20}$	37·0	13·3	492·1	177	6549	40
H ,,	$57 \times \frac{12}{20}$	34·2	9·6	328·3	92	3146	63
J ,,	$58 \times \frac{13}{20}$	37·7	5·3	199·8	28	1056	72
K ,, (Part),	$40 \times \frac{12}{20}$	24·0	1·65	39·6	3	72	22
		497·2		6239·1		86499 219	219
		497·2				86718	

Above Assumed Axis.

Items.	Scantlings in Inches.	Sectional Areas $= A$	C.G. from N.A. $= h$	Moments of Areas $A \times h$	h^2	$A h^2$	$\frac{1}{12} A d^2$, $d =$ Depth of Items.
Bridge Deck Stringer,	$42 \times \frac{10}{20}$	21·0	19·2	403·2	369	7749	
Bridge Deck Angle, -	$4\frac{1}{2} \times 4\frac{1}{2} \times \frac{11}{20}$	4·6	19·15	88·1	367	1688	
Bridge Deck Plating, -	$144 \times \frac{6}{16}$	54·0	19·8	1069·2	392	21168	
Upper Deck Stringer, -	$58 \times \frac{10}{20}$	29·0	12·3	356·7	151	4379	
Upper Deck Plating,	$153 \times \frac{8}{20}$	61·3	12·85	787·7	165	10114	
Side Stringer,	$\left\{\begin{array}{c} 6\frac{1}{2} \times 4\frac{1}{2} \times \frac{13}{20} \\ 4\frac{1}{2} \times \frac{10}{20} \end{array}\right\}$	8·9	6·6	58·7	43	383	
"	"	8·9	1·4	12·5	2	18	
P Strake, - -	$40 \times \frac{12}{20}$	24·0	17·9	429·6	320	7680	22
O " -	$49 \times \frac{11}{20}$	27·0	14·7	396·9	216	5832	38
N " -	$44 \times \frac{14}{20}$	30·8	11·2	345·0	125	3850	32
M " - -	$55 \times \frac{12}{20}$	33·0	7·5	247·5	56	1848	58
L "	$57 \times \frac{13}{20}$	37·0	3·3	122·1	11	407	68
K " (Part),	$17 \times \frac{12}{20}$	10·2	·7	7·1	—	—	2
		349·7		4324·3		65116	220
Less $\frac{1}{7}$ for rivet holes,		49·9		617·7		9302	31
		299·8		3706·6		55814 189	189
Above assumed N.A., -		299·8		3706·6		56003	
Below assumed N.A., - -		497·2		6239·1		86718	
		797·0		2532·5		142721 2	
						285442	

N.A. below assumed axis $= \frac{2532 \cdot 5}{797} = 3 \cdot 18$ ft.

N.A. above base $= 16 - 3 \cdot 18 = 12 \cdot 82$ ft.

$y =$ distance of top of vessel from N.A. $= 36 \cdot 0 - 12 \cdot 82 = 23 \cdot 18$ ft.

$$\frac{I}{y} = \frac{285442}{23 \cdot 18} = 12314.$$

The load displacement of the above vessel is 9600 tons, the draught being 23 ft. 9 ins.; if we assume, as is frequently done in approximate calculations, that the maximum bending moment on the wave crest is equal to a thirty-fifth of the displacement multiplied by the length, we shall have in the present instance :—

$$\text{Maximum bending moment} = \frac{9600 \times 350}{35} = 96000 \text{ ft. tons;}$$

and if we use this figure with that just obtained for the value of $\frac{I}{y}$, we shall get for the greatest stress acting on the vessel when under a hogging strain—

$$p = \frac{M}{\frac{I}{y}} = \frac{96000}{12314} = 7·79 \text{ tons per square inch.}$$

To obtain the greatest stress under a sagging strain, as previously pointed out, a new moment of inertia calculation is necessary, otherwise the work is similar to that just explained and need not be here detailed.

With regard to the magnitudes of calculated stresses, it may be said that, generally speaking, these increase with size of vessel. Small vessels have to be built to resist local strains, and are probably too strong, considered as floating girders. At anyrate, their actual calculated stresses, of which records are available, show these to be very small indeed. In 1874, Mr. John investigated the longitudinal strength of iron vessels of from 100 to 3000 gross tonnage, on the basis of Lloyd's scantlings, the following being some of his results :—

Gross tonnage of Ship.	Tensional stress in tons per square inch at Upper Deck.
100	1·67
500	3·95
1000	5·2
2000	5·9
3000	8·09

Later calculations for steel vessels of large size, which have proved satisfactory as to strength, show maximum stresses of between 8 and 9 tons and even higher The *Servia*, a passenger and cargo vessel of 515 ft., had a calculated stress at the upper deck of 10·2 tons per square inch when on the wave crest, and of 8·04 tons when in the wave hollow, while the *Mauretania**, of 760 ft. length, is stated to have a calculated maximum stress of 10·3 tons on the top member. It should be added that a special high tensile steel was largely used in the construction of the upper works of the latter vessel.

With regard to compressive stresses, it is important to note that thin deck plating is liable to buckle when in severe compression, and is therefore not so efficient under a sagging as under a hogging strain; this should be borne in mind, particularly when considering maximum stresses on bridge and awning-decks.

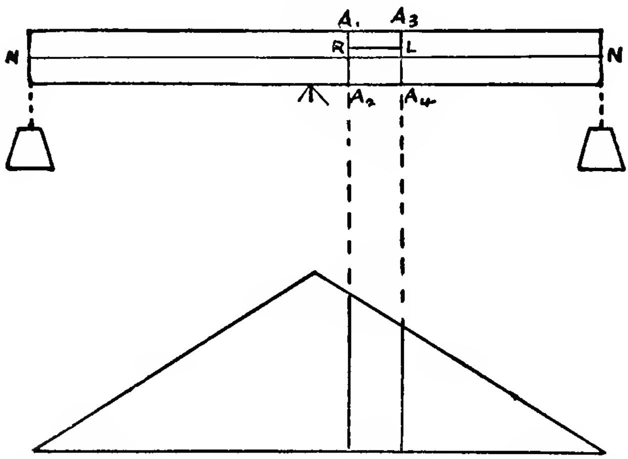
It has already been pointed out that stresses, such as the above, are not the actual stresses experienced by the vessel, since the conditions of figs. 49 and 50 do not fully represent those of a vessel among waves. The results, however, are valuable for comparison. In the case of a proposed vessel, for example, if the calculated stress be not greater than in existing vessels whose

* See the *Shipbuilder* for November, 1907.

records have been satisfactory, the scantling arrangements in the new ship may be considered adequate. If it be much greater, so as to approximate to the calculated stress in vessels which have shown manifest signs of weakness when on service, then additional strength must be added. From our previous considerations, it will be clear that the most economical position for the new material to resist bending, will be either at the top or bottom of the vessel, *i.e.*, as far as possible from the neutral axis, as it will there be of maximum efficiency in reducing the stress.

SHEARING STRESSES.—We come now to consider the effect of shearing forces on a structure. We have already explained how the values of such forces may be obtained at all points in the lengths of beams, including floating vessels, under various systems of loading, and we have now to determine the stresses caused thereby.

Fig. 62.



If the vertical shearing force at any section be taken as F, we may obviously write:—

$$\left.\begin{array}{c}\text{Mean stress per square inch}\\ \text{due to shearing force}\end{array}\right\} = \frac{F}{A},$$

where A is the number of square inches of material in the section. For example, if a rectangular beam of section 8 inches by 4 inches be under a shearing force of 64 tons, then—

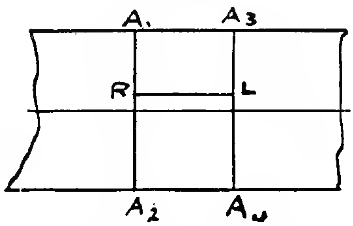
$$\text{Mean stress per square inch} = \frac{64}{8 \times 4} = 2 \text{ tons.}$$

The actual stress at any point of the section may, however, be very different from this mean stress, as we shall now proceed to show.

In fig. 62 we have the diagram of bending moments for a beam supported at the middle and loaded at each end. The bending moment is a maximum at the point of application of the support, and has zero values at each end. For any two vertical sections $A_1 A_2$ and $A_3 A_4$, the bending moments may be read from the diagram. Section $A_1 A_2$ being nearer mid-length has the greater bending moment. NN, the neutral surface, may be considered to divide the beam into two portions, of which the upper one is in tension and the lower in compression.

Consider now the equilibrium of a small portion of the beam $A_1 R L A_3$, shown enlarged in fig. 63. There are pulling forces acting on the end $A_1 R$ and on the end $A_3 L$. As the former section is under a greater bending moment than the latter, the stresses will also be greater. There will thus be a force equal to the difference of the total forces acting on the ends tending to move the portion of the beam $A_1 R L A_3$ towards mid-length. This action is balanced by a shearing force over the bottom surface LR. Clearly, the magnitude of this shearing force will vary with the areas of the ends $A_1 R$ and $A_3 L$. At the top of the beam the shearing force will be zero, and will gradually increase as RL approaches NN, where it will be a

Fig. 63.



maximum. Below the neutral surface, the forces act in opposite directions, and therefore as RL approaches the lower end of the beam, the shearing force will gradually be reduced, becoming again zero at $A_2 A_4$.

If the vertical sections be at unit distance apart, say one inch, the horizontal shearing stress per square inch at any point L of section $A_3 A_4$ will be obtained by dividing the shearing force acting on the surface RL by the breadth in inches of the beam at that point. This is also the value of the vertical shearing stress on the section at the same point, since there cannot be a shearing stress in one plane of a beam without an equal one at the same point in a plane perpendicular to the first. Proceeding in this way, we arrive at the following formula for the shearing stress per square inch at any point of a cross section:—

$$q = \frac{A g}{b I} F$$

where A = Area in square inches of the portion of the cross section above or below the given point.

g = Distance in inches of the centre of gravity of the area from the neutral surface.

F = Vertical shearing force at the cross section in tons.

I = Moment of inertia of the whole section (in inches^4).
q = Stress per square inch at the given point.
b = Breadth of beam in inches at the given point.

Let us apply the formula to the case of the rectangular beam whose mean shear stress was found above to be 2 tons. Substituting values, we get for the stress intensity at the neutral axis:—

$$q = \frac{16 \times 2 \times 12 \times 64}{4 \times 32 \times 64} = 3 \text{ tons per square inch.}$$

Thus the maximum shear stress is, in this instance, 50 per cent. greater than the mean.

The above is for a simple beam of rectangular section, but the same formula may also be applied to the more complex case of a ship. In the latter instance, of course, the beam is of hollow section, and b will be twice the thickness of the shell plating. It is important to note that only continuous longitudinal materials must be used in finding A. Obviously, the value of the shearing stress will vary with P, the vertical shearing force, which is a maximum in ordinary vessels at about one-fourth the vessel's length from each end; so that at the neutral axis at these points of the length the shearing stress may be considerable. We see now why it is inadvisable to unduly reduce the scantlings in the vicinity of the neutral axis.

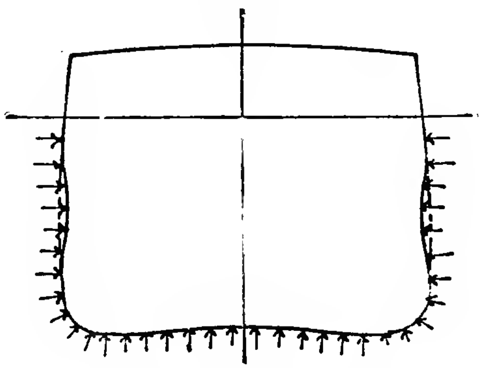
It is also important to give special attention to the rivets in the landings, or longitudinal seams, in this neighbourhood, as the shear stress gives rise to a tendency for the edge of one strake to slide over that of the next. Recent experience with large cargo vessels has shown that the usual plan of double riveting the seams is only sufficient for vessels up to a certain size, say 450 or 480 feet in length. Longer vessels will develop weakness at the longitudinal seams unless precautions be taken to increase the strength of the riveting. Lloyd's Rules now require treble riveted edge seams in the neighbourhood of the neutral axis in the fore and after bodies in vessels of the above length and beyond.

TRANSVERSE STRAINS.—So far, we have dealt exclusively with stresses which tend to strain a vessel longitudinally, and while such stresses are probably of first importance, we must not omit to refer to those which come upon a vessel in other directions.

It has been customary to consider stresses which tend to change the transverse form as next in importance to those affecting a vessel longitudinally. Structural stresses in other directions are, indeed, partly longitudinal and partly transverse, and where the predominant stresses are known for any vessel, the effect of their combination in a diagonal direction may be predicted. Unfortunately, the subject of transverse stresses of ships is a complicated one, and we cannot do more here than indicate generally the external forces which operate on a vessel so as to alter her transverse form, and point out the structural arrangements which best resist this deforming tendency.

Consider in this connection the case of a vessel afloat in still water (fig. 64). The hull surface is pressed everywhere at right angles by the water pressures, indicated in the figure by arrows, and the resulting tendency is towards a general deformation of the vessel's form. Taking the transverse components of the water pressures, these obviously tend to force up the bottom and press in the sides, as shown exaggerated in fig. 64. Such tendencies, however, are prevented from becoming actual strains by the internal framing. The comparatively thin shell plating, which might yield under heavy water pressure, particularly in the way of an empty compartment, is kept in shape by the frames, rigidly connected to the beams and to the floors at their top and bottom ends respectively, and supported between these points by hold stringers and keelsons. In way of the bottom, the deep floors, spaced at comparatively short intervals, and fitted, in the first instance, as supports to the cargo, are splendid preservers of the form. The floors, too, are tied to the beams of the decks by means of

Fig. 64.

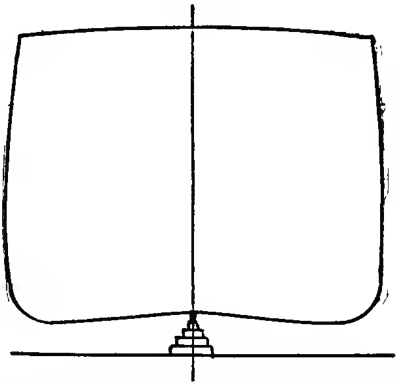


strong pillars, and in this way a stress which comes upon one part of the structure is communicated to it as a whole. Probably the most efficient preservers of transverse form are the athwartship steel bulkheads. Where these occur, the vessel may be considered as absolutely rigid, and care should be taken to spread this excess of strength over the space unsupported by bulkheads by means of keelsons and hold stringers.

Docking Stresses.—A vessel when docked or when aground on the keel, particularly if loaded, has to withstand severe transverse tresses. The reaction of the weight at the middle line will tend to force up her bottom, while the weight of cargo out in the wings will set up a considerable transverse bending moment and cause the bilges to have a drooping tendency. This is shown, much exaggerated of course, in fig. 65. There will be tensile stresses of considerable magnitude acting along the top edges of the floors; and if the vessel be one having ordinary floors, weakness may be developed at the lower turn of the bilge, as the framing has there to withstand a shearing stress due to the weight of the cargo above. The floors should therefore be kept as deep as possible at the bilge, and should be carried well up the sides. In vessels having double bottoms this part of the structure is very strong owing to the deep wing brackets, which bring the resisting powers of the side framing into operation.

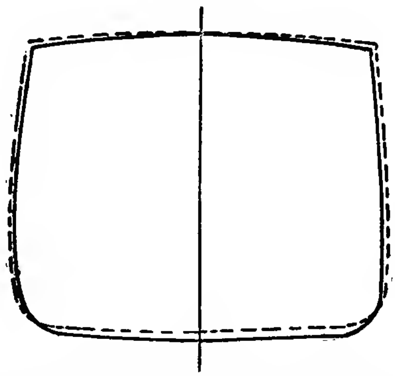
The straining at the middle line will be arrested by the pillars, if efficiently

Fig. 65.



fitted; these will act as struts and communicate the stresses to the deck beams, which will resist a tendency to spring in the middle and to bring the sides together. Thus, as in the case of still water tresses, the straining tendency will be resisted by the structure as a whole. This interdependence of parts, causing equal distribution of stress throughout, is what should be aimed at in design, and special pains should be taken to ensure efficient connections.

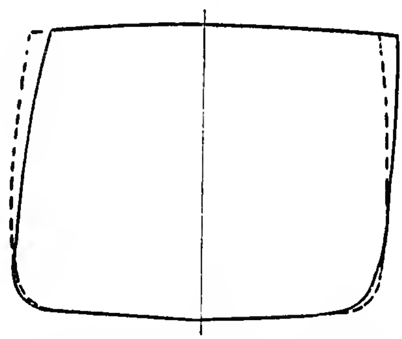
Fig. 66.



TRANSVERSE STRESSES DUE TO INCORRECT LOADING.—A preventable cause of transverse straining is that due to the manner in which heavy deadweight cargoes are sometimes loaded. Frequently, the heaviest items are secured at the middle line of the vessel instead of being spread over the bottom, the wings having therefore comparatively little weight to carry. The straining tendency in such a case is to elongate the transverse form, the water pressures on the sides tending to the same end. This condition is illustrated in fig. 66, the dotted lines representing the normal vessel, and the full lines the vessel as strained. The pillars will be here called upon to tie the top and bottom of the structure, but not unfrequently the rivets connecting the pillars at top and bottom have been sheared in places with consequent dropping of the bottom part of the hull.

TRANSVERSE STRESSES DUE TO ROLLING.—We have pointed out that it is when among waves at sea a vessel meets with the most trying longitudinal stresses, and it may now be added that tendencies to transverse straining are also greatest then. These latter stresses probably reach maximum values when a vessel is rolling in a beam sea, and they are obviously due to the resistance which the mass of the structure offers to change of the direction of motion each time the vessel completes an oscillation in one direction and is about to return. The stress is a racking one, and tends to alter the angle between the

Fig. 67.



deck and the sides, also to close the bilge on one side and to open it on the other. Such a racking strain is exhibited graphically in fig. 67.

The parts of the structure most effective in preventing this change of form are the beam knees, transverse bulkheads, web frames or partial bulkheads, and the ordinary side frames, in which is included the reverse bar, if any. The beam knees should be of good size, efficiently connected to the frames and beams, and fitted well into the corner formed by the side plating and the deck. Change of form at the bulkheads is practically impossible, if they be stiffened sufficiently against collapsing; careful attention should therefore be given to this point. The side frames, owing to their position and close spacing, offer powerful resistance to racking, but in order to attain maximum efficiency they should be securely riveted to the beam knees, and the floors or tank brackets should be carried well up the sides. These brackets virtually reduce the length of the frame, and it is well known that reducing the length in such a case greatly increases the rigidity

Local Stresses.—Besides longitudinal and transverse structural stresses, vessels have to resist other straining tendencies due to local causes. For example, the engines and boilers with their seatings together form a heavy permanent load on a comparatively small fraction of the length, and thus give rise to considerable local stresses. These are provided against in various ways, some details of which are given in a later chapter. It may be said that the general principle is to increase the strength of the structure in way of the loaded zone, and, by means of longitudinal girders and otherwise, distribute the load to the less strained portions of the hull beyond.

Other stresses due to the propelling machinery are those brought on the stern of the vessel by the action of the propeller itself. These are most severe when the vessel is rolling and pitching among the waves, and consist chiefly of vibrations caused by the frequent racing of the propeller and checking of the same, as it rises out of and sinks into the water. The parts that suffer most are the connections of the stern frame to the vessel, and it is highly important, therefore, that these should be made amply strong. We shall see presently, when we come to consider details of construction, what the usual arrangements are in such cases.

Panting Strains.—These strains, which are usually developed in the shell plating forward and aft, where it is comparatively flat, consist of pulsating movements of the plating, as the name indicates. They are partly due to blows from the sea, and partly to the resistance offered by the water to the vessel's progress as she is driven forward by the propeller. An ordinary cargo vessel is not so much troubled by these strains as a fine-lined passenger steamer, for she is slower, and her full ends are better able to resist a tendency to flexibility than the flatter form of the faster boat.

The usual means taken to strengthen the shell against panting, is to fit a hold stringer in the vicinity affected, and connect it well to the shell plating and framing; if the vessel be fairly large, the stringer should be associated with a short tier of beams, which act as struts and prevent movement in the plating. In very fine vessels the floor-plates should be deepened forward and aft. A panting arrangement for a cargo steamer is shown in the chapter on practical details.

A class of strains somewhat akin to those of panting are frequently found developed under the bows of full cargo vessels, in the shape of loose rivets and generally shattered riveted connections. They are now recognised to be due to the pounding which a vessel receives from the waves as she rises and falls among them. As might be expected, they are found much aggravated after a voyage made in ballast trim, for the pitching motions will one instant lift the fore end high out of the water, and the next bring it into it with terrific force. It is scarcely wonderful that this pounding, repeated throughout a long voyage, should produce the results mentioned. Obviously, efficient ballasting is of vital importance, and that this is the opinion of those having shipping interests, was evidenced by the appointment of the Royal Commission, under Lord Muskerry, to consider the desirability of fixing a minimum load-line to sea-going vessels, although, for various reasons, there was no practical result therefrom.

Strains due to Deck Loads, etc.—These loads consist of steam winches, the windlass, donkey boilers, steering gear, etc. The resulting stresses can usually be counteracted by an efficient system of pillaring, with perhaps a few extra beams if the weights be very great.

The Racking Strains brought on the deck of a sailing vessel by stresses from the rigging and masts should be mentioned. In sailing vessels not of sufficient size to require a steel deck, special tie-plates should be arranged in a diagonal direction so as to communicate the stresses from the plating round the mast to the deck beams and side stringers, to all of which the tie-plates should be securely riveted.

QUESTIONS ON CHAPTER IV.

1. Given a beam fixed at one end and loaded with a weight *W* tons at the other, describe the system of forces acting at any section, neglecting the weight of the beam. If the beam be 10 feet long and the load 2 tons, plot the diagrams of shearing forces and bending moments, and give numerical values for a section 4 feet from the free end of the beam.

Ans.—S.F., 2 tons.
B.M., 96 inch tons.

2. Referring to the previous question, if the given load be spread evenly over the beam, indicate the forms which the curves of bending moment and shearing force will then take.

3. A beam 20 feet long supported at each end has a load of 3 tons concentrated at a point 2 feet from the middle of the length. Draw the diagrams of shearing forces and bending moments, and indicate the value of the maximum bending moment.

Ans.—Max. B.M. = 172·8 inch tons.

4. Assuming the load in the previous question to be evenly distributed over the length of the beam, calculate the maximum shearing force and bending moment, and indicate the points in the length at which these are in operation.

Ans.—Max. S.F. = 1·5 tons acting at points of support.
Max. B.M. = 90 inch tons acting at middle.

5. Show that diagrams of S.F. and B.M. may be derived by a graphic process, and employ in your explanation the case of a beam fixed at one end and uniformly loaded.

6. Explain how to construct a curve of loads for a ship floating in still water, and state what tests you would apply to prove the accuracy of your work.

7. What is the connection between curves of loads, shearing forces, and bending moments, and show in one diagram the approximate forms these diagrams would take in the case of a cargo vessel floating "light" in still water.

8. A box-shaped vessel 200 feet long, 30 feet broad, 20 feet deep, floats in still water at a draught of 10 feet. If the weight of the vessel be 1000 tons uniformly distributed, and if there be a cargo of 715 tons uniformly distributed over half the vessel's length amidships, draw the curves of S.F. and B.M. and state the maximum shearing force and bending moment.

Ans.—Max. S.F. = 178·5 tons.
Max. B.M. = 8925 feet tons.

9. If the sides and top and bottom of vessel in previous question are composed of steel plating ½ inch thich, find the greatest stress to which the material is subject under a maximum hogging moment of 8000 feet tons. *Ans.*—1·82 tons per square inch.

10.—Assuming a cargo steamer in loaded condition to be poised on the crest of a wave, sketch roughly the curves of loads, shearing force and bending moment.

11. Referring to the previous question, at what points approximately in the length will the maximum shearing forces act and where will the maximum shearing stress intensity be developed?

12. Taking the box vessel of question 8, and assuming her to be under a maximum shearing force of 400 tons, find the mean shearing stress over the section, and also the maximum shearing stress.

Ans.—Mean shear stress = ·66 tons per square inch.
Max. shear stress = 1·82 " "

13. Enumerate the various local strains to which ships are liable, and the methods adopted to strengthen the vessel against them.

CHAPTER V.

Types of Cargo Steamers.

NOT the least among the many important points to be settled by an owner in deciding upon a new ship, is the question of type. A ship may be suitable as to cost, may be strong enough, have good speed and deadweight capability, and yet may prove herself very unsatisfactory, if not an utter failure, in certain trades. Every owner of experience is aware of this, and is careful to see that he gets a ship suited to his purpose.

Nowadays, an owner who knows his requirements can usually get them carried out in this matter. But this was by no means always the case. At one time it seemed to be thought that ships must be built to certain fixed designs, and cargoes had often to be adapted to suit a vessel's arrangements rather than the latter being made to suit the former, resulting in much annoyance, inconvenience, and expense.

With the expansion in oversea trade, however, but more especially with the changes in materials of construction—from wood to iron, and iron to steel—and the progressive spirit of the age, came a gradual evolution of type, until the cargo steamship of to-day has reached a high standard of excellence, and where applied to special trades, has become almost the last word of efficiency for the purpose intended.

The variations which have marked this evolution and brought cargo vessels to their present stage of development have been, generally speaking, in the following directions, viz.:—(1) in design of structure to provide hulls of degrees of strength suitable for different trades; (2) in form of immersed body and in general outline and appearance; (3) in disposition of materials; (4) in internal construction.

STRENGTH TYPES.—It was long ago recognised that for economical working different cargoes should have different classes of vessels: that cargoes of great density, for instance, which occupy little space in comparison with weight, such as iron ore or machinery, and heavy general cargoes, should be carried in strong vessels having great draught and displacement and limited hold space; and cargoes of less density, such as grain, cotton, wood, and light general cargoes, should be accommodated in ships of relatively greater hold capacity, but less deadweight capability. Thus, until their recent revision, Lloyd's Rules provided special schemes of scantlings for three strength types, viz., three-deck, spar-deck, and awning-deck types. Of these, the first-named was the strongest, and was reckoned to be able to carry any kind of cargo to any part of the world on a greater draught than any other type of vessel of equal dimensions. With regard to the spar and awning-deck types, we have the authority of the late Mr. Martell, a former chief surveyor to Lloyd's Register, for saying that they were not originally intended as exclusively cargo carriers. The upper 'tween decks were really meant to accommodate passengers, and the weather deck and the shell plating and side framing above the second or main deck, were allowed to be of comparatively light construction. But although thus built of smaller scantlings than the corresponding three-deck vessel of same absolute dimensions, these lighter vessels were not of less comparative strength. Their draughts were restricted, their loads reduced, and hence also the leading movements acting upon them; so that their thinner materials were quite sufficient to ensure as low a stress per square inch on the upper and lower works as in the corresponding three-deck type of vessel.

In the development of ship construction, the foregoing types have undergone modification. In Lloyd's latest Rules, only two distinct standard types are mentioned, viz., the full scantling vessel, and an awning or shelter-deck type. The latter has still the characteristics of other vessels of the class, namely, light draught and large capacity, but has otherwise been greatly improved.

FORM TYPES.—With regard to changes of form, it must be admitted that the body of the modern cargo steamer is no thing of beauty. The sentiment which demanded fineness of form and grace of outline has passed away under the pressure of ever-increasing competition. From the fine-lined under water bodies, with displacement co-efficients of from ·6 to ·7, and the nicely rounded top-sides of former days, we have come to sharp bilges, more or less vertical sides and bluff ends, with displacement co-efficients ranging from ·8 upwards. Certainly this side of the development of cargo ships has not proceeded on æsthetic lines.

Appearances apart, however, and considering the case from a purely money-making standpoint, the changes have been in the right direction. The researches of the late Dr. Froude and others, and experience gained from actual vessels, has shown that at moderate speeds like 8 or 10 knots—the speeds of ordinary cargo vessels—the resistance to be overcome in propulsion is largely due to surface friction, the element of wave-making resistance only becoming important at higher speeds. As a considerable increase in displacement and therefore in deadweight capability can be obtained by a moderate increase in surface, the easiest and cheapest way for an owner to increase the earning power of his vessel is obviously to fill her out forward and aft, and this has become the order of the day. Of course, for the best results, the filling out process must be done with judgment. An expert designer can do much even with the fullest co-efficients. In general, vessels of ·8 blocks and upwards should have small rise of floor and relatively sharp bilges amidships, thus allowing most of the fining away to be done towards the extremities. In some cases this method has not been followed. It should be said that in Lloyd's former rules, the half girth appeared as a factor in calculating the numerals, and this induced some builders, for the sake of getting lighter scantlings, to design full cargo vessels with abnormally fine midship sections, thus causing the ends to be very clubby. But such vessels when built invariably proved unsatisfactory. They were found difficult to steer and therefore unmanageable in a seaway, also harder to drive, than vessels of similar block co-efficients designed on normal lines. Under the new rules the girth does not influence the numerals, and there is now no temptation to design freak ships of the kind mentioned; still, owners should not take too much for granted in ordering their cargo "tramps," but should see that they get a maximum of good design with any given conditions.

More striking than the changes of the under-water forms, and those which have caused cargo vessels to be classified into various form types, have been those due to the imposition of deck erections on the fundamental flush-deck steamer.

Very early in the history of the iron merchant ship, the necessity of affording some protection to the vulnerable machinery openings led to the latter being covered by small bridge erections. Then the obvious advantages of having the crew on deck caused the accommodation for the latter to be raised from below and fitted in a forecastle, this erection incidentally forming an admirable shield from the inroads of head seas, and the release of the space under deck making a desirable addition to the carrying capacity. Finally, poops were fitted, experience showing the necessity of raising the steering platform from the level of the upper deck. Thus the three-island type was arrived at, whose outlines are characteristic of many of the cargo steamers of to-day (*see* fig. 68).

The next step in the development of deck erections was in the direction of increased lengths, as, under Government Regulations, which became operative in 1890, considerable reductions in freeboard could thereby be gained, and, particularly as, provided they had openings in their end bulkheads, which, however, might be closed in a temporary manner, the erections were allowed to be exempt from tonnage measurement. Thus long bridges became common, and eventually vessels were built with bridge and poop in one, making, with a disconnected forecastle, one form of the well-deck type (*see* fig. 69).

Fig. 68.

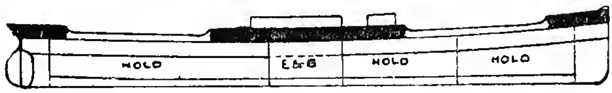


Fig. 69.

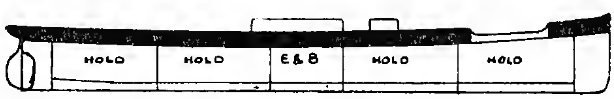


Fig. 70.

Fig. 71.

Fig. 72.

The obvious advantage of having a continuous side and deck, and the admirable shelter which the enclosed space would afford for cattle, etc., very soon produced the suggestion to fill in the former gap between forecastle and bridge; and this was rapidly carried into effect when it was found that by having one or more openings in the deck with no more than temporary means of closing, the space would escape measurement for tonnage. In this way the shelter-deck type (*see* fig. 70) was evolved—a type in recent years much run upon for large cargo vessels, and which, as previously mentioned, is now a standard of Lloyd's rules.

Other modifications have consisted of short bridges on longer ones and on shelter decks, but these can hardly be considered as constituting distinct types.

For the smaller classes of cargo carriers a somewhat special type of steamer has been developed, familiar to all who take an interest in ships, as a quarter-decker, which, in reality, is a one or two-decked vessel with the main deck aft raised (*see* fig. 71). This raising of the after deck was undoubtedly due to considerations of trim. It was found that owing to the finer form aft, and the large amount of space taken up by the shaft tunnel, the tendency with the normal deck line was to trim by the head when loaded, the predominance of cargo at the fore end causing this. To correct this state of things the hold space aft was increased by raising the deck.

While the quarter-deck has certain advantages, such as good trim and general handiness, it has some drawbacks, one of which is the difficulty of making up the strength sufficiently at the break of the main deck. The usual plan is to double the shell plating and overlap the main and quarter deck stringers in way of the break, the hold stringers at this part being also overlapped. In vessels of a size requiring a steel deck or part steel deck, the latter is overlapped where broken to form the quarter deck, and the two portions connected by substantial diaphragm plates. The doubling of shell and overlapping of stringers is also carried out.

The foregoing, or something equivalent, is what must be done to make good the loss of continuity. It is seen to involve a considerable amount of bracketing and troublesome fitting work, which tends to raise the first cost of the vessel. In the vicinity of the break, too, there is much broken stowage space; yet, in spite of all, for some trades this type still remains a strong favourite.

Another modified type, in some respects the opposite of the last in that it leads to an increased hold capacity forward over the normal type, is the partial awning-decker. It is to be supposed that with ordinary cargoes this type would trim badly, but it appears to have been found very suitable for special light bulky cargoes. It was at one time very popular, but of recent years has not been much in evidence. The external appearance of the partial awning-decker is shown in fig. 72. It is seen to be a quarter-decker with the forward well filled in, and the precautions already described for maintaining the strength at the break have also to be taken in this case.

One clear consequence of the long erections now become prevalent is undoubtedly the modern system of distributing the materials of construction.

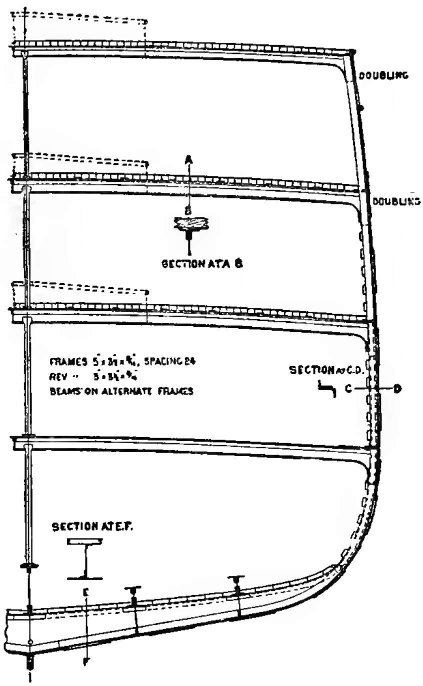
Bridges which are very short have small structural value, as they are not really part of the hull proper, and should not be considered in estimating the longitudinal strength. It is otherwise, however, with bridge erections of substantial lengths, which must withstand the structural bending stresses acting on the vessels of which they form part. Moreover, it follows from the principles expounded in the previous chapter, that the heaviest longitudinal materials should be placed at the deck, stringer, and sheerstrake of an erection, whether it be a long bridge, an awning or a shelter deck, as the moment of inertia of the material at a section is thereby increased, and the stress under a given load, which is of maximum value at these parts, reduced. Modern vessels are now required to be built in this way by the rules of Lloyd's Register and of the other classification bodies, the old practice of making superstructures of light build and massing the strength at the second deck from the top being discontinued. This may be considered to mark an important advance in the scientific construction of ships.

CONSTRUCTION TYPES.—Coming now to the changes that have taken place in the internal construction of vessels, we find these to be of a wide-reaching character. Fig. 73* is the midship section of a large passenger and cargo steamer as built 25 to 30 years ago, and illustrates all the characteristics of the time, viz., thin side framing, numerous tiers of beams, ordinary floors, and deep hold keelsons. The expansion of commerce, however, the opening

* See an interesting paper on "Structural Development in British Merchant Ships," by Mr. J. Foster King, in the *Transactions of the Institution of Naval Architects* for 1907, to which the author is indebted for particulars in preparing some of the sketches in this chapter.

up of new trades, and the specialising of vessels for these trades, led first to increase in the average size of vessels, then to various modifications in their internal economies. The trouble and expense attending the use of dry ballast led to the adoption of water ballast tanks, which, ultimately becoming incorporated in the structure, caused the disappearance from the holds of the huge plate side girders which, as shown in fig. 73, accompanied the fitting of ordinary floors. In a later chapter we shall deal in detail with ballast tanks, but their general design and arrangement may be gathered from figs. 74 to 85. An early modification in the structure was the deepening of the holds by the

Fig. 73.



suppression of the lowest tier of beams, required by the construction rules of the time, and the fitting at every fifth or sixth frame of plate webs having face bars on their inner edges, the hold stringers being deepened to come in line with the inner edge of the plate webs, and the whole forming a strong box-like arrangement which amply made up the deficiency caused by the omission of the hold beams (see fig. 74). This style of construction long remained in favour and is still sometimes preferred, but the loss of stowage capacity, particularly for case cargoes, eventually led to its general abandonment in favour of the deepening of the frame girder itself, the system of framing which in one form or another is found in the cargo steamers building in the yards to-day.

Fig. 75.
330 Feet Steamer

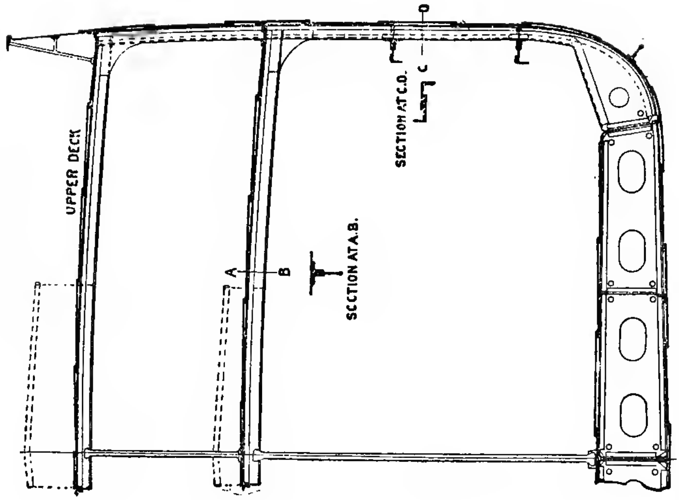
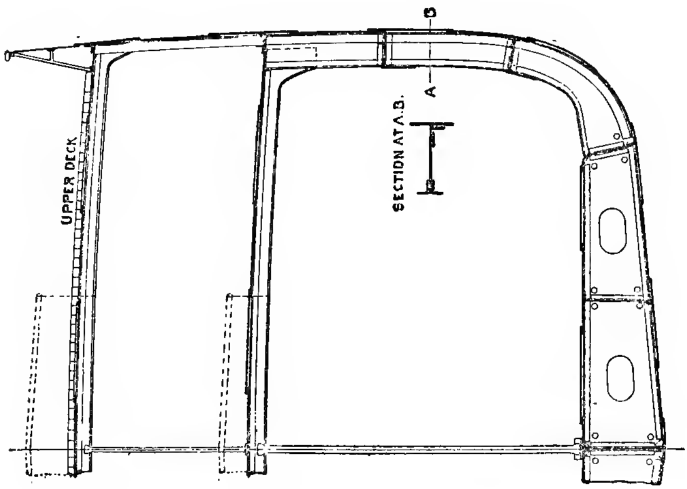


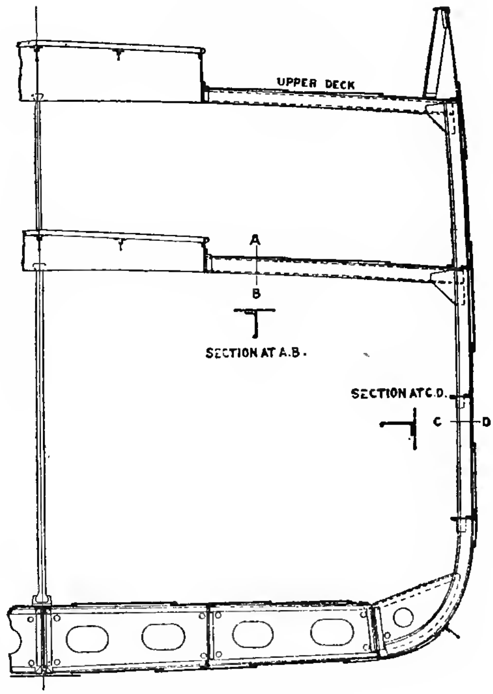
Fig. 74.



A natural development which came, although not quite immediately, was the reduction in size of the hold stringers, which, as stowage breakers, were found not less obnoxious than the webs. Moreover, the deep side framing alone was sufficient for all the demands of local stresses, and owing to their proximity to the neutral axis, the extent to which these hold stringers assisted the ship against bending was comparatively trifling. In fig. 75 is shown the hold stringers of fifteen years ago, and in fig. 76 those of the present day. Their work now is to keep the frames in position, to prevent them side tripping, and to stiffen the shell between the frames.

Fig. 76.

330 Feet Steamer.

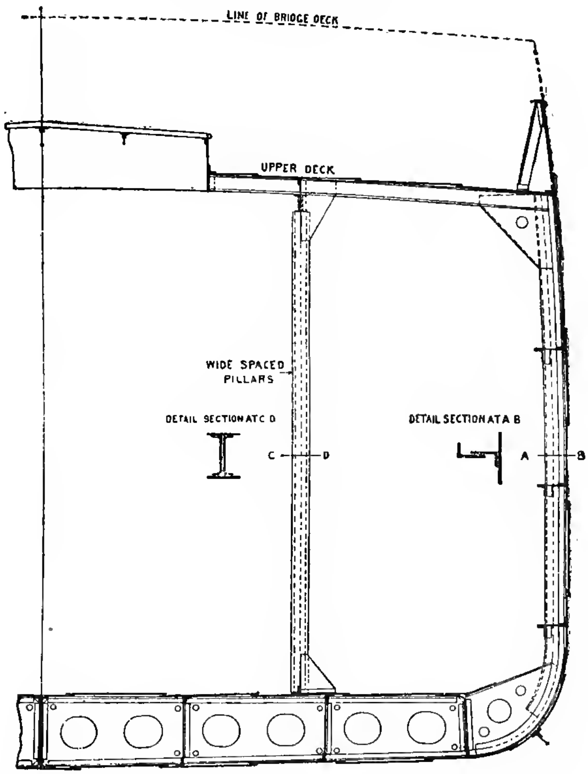


Recent experiments made by Lloyd's Register have gone to show that up to a frame depth of 7 inches (the limit of the experiments) there is no tendency to side tripping, and since then vessels have been built with a reduced number of hold stringers, and in a few recent cases with none at all. Whether the hold stringer will ultimately disappear from the modern ship as an element of construction remains to be seen. This, it may be said, is the view taken by some naval architects, but the general feeling seems to be in favour of its retention in a modified form.

Improvements in the manufacture of steel sections in recent years, and the broad-minded view now taken by the classification societies, have made it possible for builders, following the line of simplification of parts, to still further satisfy the demands of shipowners for large holds clear of beams, stringers, and numerous hold stanchions. Hence has come the well-known single-deck type (*see* fig. 77) Vessels of a size ordinarily requiring, by Lloyd's former rules, three tiers of beams and two steel decks, have been built with a single steel deck and one tier of beams, the structural strength, transverse and longitudinal, being made good by deepening the side frames and increasing the scantlings of the deck, shell plating, and double bottom. Purely

Fig. 77.

SS. 350′ 0″ × 51′ 0″ × 28′ 0″.



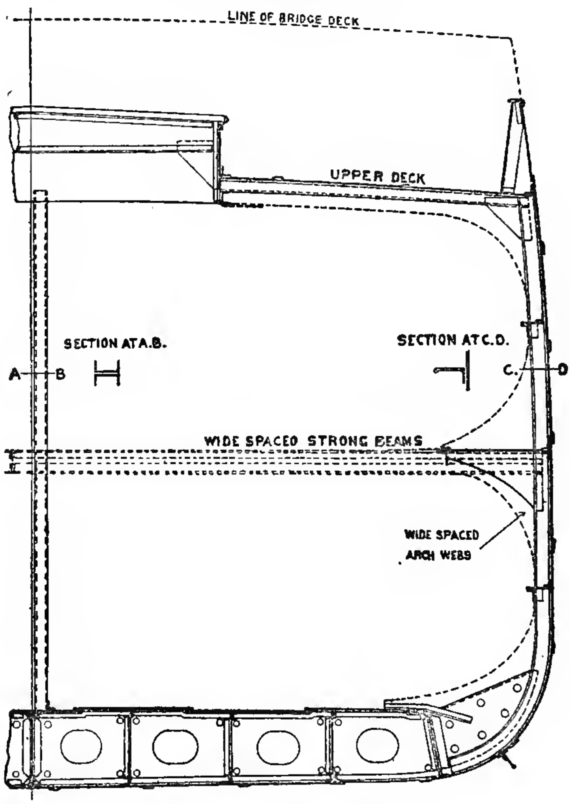
single-deck vessels have gone on increasing in size until they have attained lengths of 350 feet, and depths exceeding 28 feet, and it appears likely the advancement will still proceed so long as the needs of commerce demand it. In Lloyd's latest rules, the construction of single deckers up to a moulded depth of about 31 feet is provided for, but so far as we are aware, no single-deck vessel of ordinary design has been built approaching this depth.

Fig. 78 illustrates a type which may be considered to be in the transition stage towards the pure single decker. It has bulb angle framing and strong hold beams widely spaced in association with arched webs and a broad hold stringer. Many vessels of this type have been built on the N.E. coast and have proved highly satisfactory. The designers and first builders of this type are an important Wearside firm.

With the removal of hold stringers and beams, the presence of numerous hold pillars became specially objectionable. A middle line row for most trades is perhaps no great drawback, but with the increased breadths at-

Fig. 78.

SS. 350′ 0″ × 49′ 0″ × 28′ 0″.

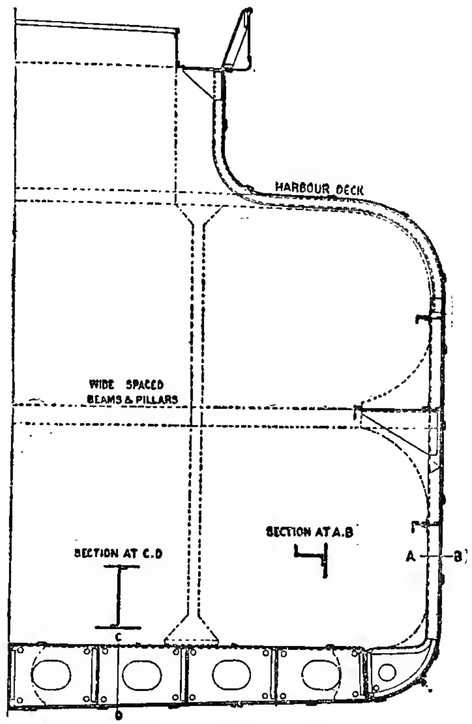


tendant on the steady rise in general dimensions, now the order of the day, additional rows of stanchions between the middle and the side, with the ordinary construction, became imperative. For a time, and up to a certain point, the case of vessels with breadths beyond that at which quarter stanchions are necessary, was met, without resorting to the latter, by increasing the scantlings of the beams and of the middle row of pillars, but a limit was soon reached, and the question of the omission of pillars had to be reviewed from other standpoints. Hence arose the system of fitting wide spaced strong pillars in association with deck girders. Centre line rows of closely spaced pillars with one or two quarter pillars in each hold are now fonnd in vessels of 50 feet breadth and upwards. In some cases the centre row has been omitted, the whole work being done by, say, four specially heavy pillar columns in each hold.

The great convenience of the latter arrangement from a stowage standpoint can readily be conceived, and although it entails a considerable addition in cost over the common arrangement, many shipowners have adopted it.

Fig. 79.

SS. 340′ 0″ × 45′ 6″ × 27′ 3″ and 34′ 3″.



A few vessels have been built so as to be able to dispense with pillars of any kind, and to these we shall refer presently

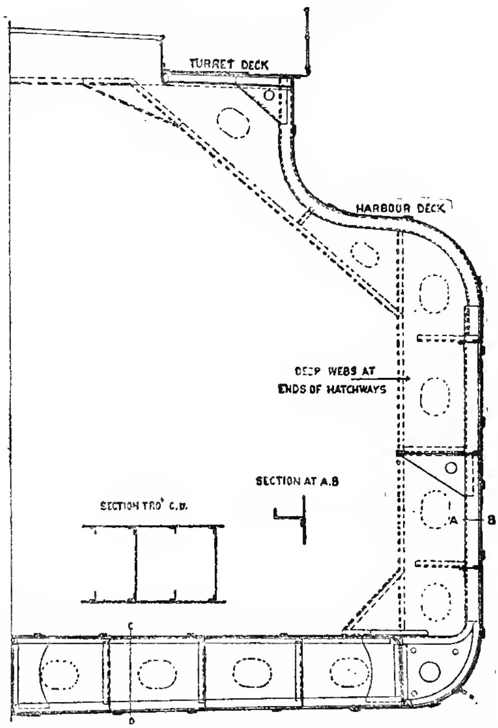
SPECIAL TYPES.—Besides the types of vessels already described, which may be considered the standard ones, there are others of quite distinct character, which the needs of commerce, the enterprise of shipowners, and the genius of shipbuilders, have called into being. Of these, probably the most important is the well-known turret-deck type of Messrs. Doxford. Fig. 79 shows the midship section of one of these vessels, and illustrates the striking differences between them and those of ordinary form.

The principal departure is in the outward form at the topsides, which, instead of being carried up with a moderate tumble home, are curved inwards, forming a central trunk or turret. The working platform is on the top of this turret, which runs forward and aft and contains all hatches, deck machinery, derricks, and everything requisite for efficiently working the vessel.

The internal framing of the majority of these vessels (*see* fig. 79) is on the wide-spaced hold beam and web-frame system; but in recent cases the no hold obstruction principle has been carried out, hold beams and pillars being entirely omitted, and the strength made good by fitting deep web-plates

Fig. 80.

SS. 350′ 0″ × 50′ 0″ × 26′ 3″ and 33′ 6″.



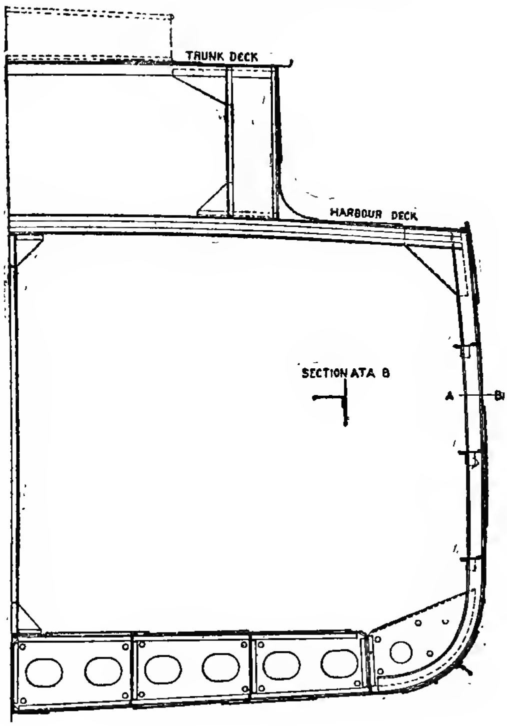
with attachments to the turret deck, ship sides, and tank top, as shown in fig. 80.

Among the advantages claimed for this type over the ordinary ones are its self-trimming qualities, which make it well suited for bulk cargoes; the greater safety which it affords to all vulnerable openings, such as hatches, ventilators, etc., owing to the turret being much higher than the ordinary weather deck; its greater stiffness and longitudinal strength, owing to its shape; increased depth and better distribution of longitudinal materials, the latter circumstance making it possible to reduce the structural weight and thus increase the deadweight. Although it cannot be said that these vessels have a nice appearance, it must be admitted that they have been a long time in service, and seem to be increasing in popularity as purely cargo boats.

Another type, of which there is now a considerable number afloat, is Messrs. Ropner's patent trunk steamer. This class is of normal single deck construction to the main or harbour deck; above this there is a central trunk running fore-and-aft; the top of the latter forms the working deck and is fitted with hatchways, winches, etc. The ship is kept in form by strong beams

Fig. 81.

SS. 350′ 0″ × 50′ 0″ × 25′ 3″ and 33′ 9″.



at the hatchway ends, and the trunk is stiffened by webs and supported by strongly built centre stanchions. This ship, like the turret design, is specially suitable for bulk cargoes like grain, the trunk forming an admirable self-trimmer (*see* fig. 81).

The Dixon & Harroway patent ship is another type whose speciality is its self-trimming arrangements. In this vessel (*see* fig. 82) the upper corner of the hold is plated in, the main frame of the vessel being carried up in the hold space. This corner space is well adapted for ballasting purposes, the high position of the ballast conducing to steadiness in a seaway. This type is of

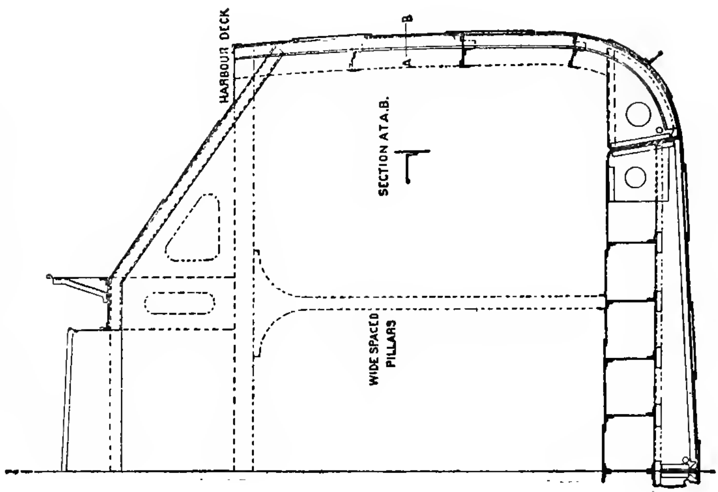


Fig. 83.

SS. 300′ 0″ × 42′ 4″ × 22′ 4″ and 28′ 4″.

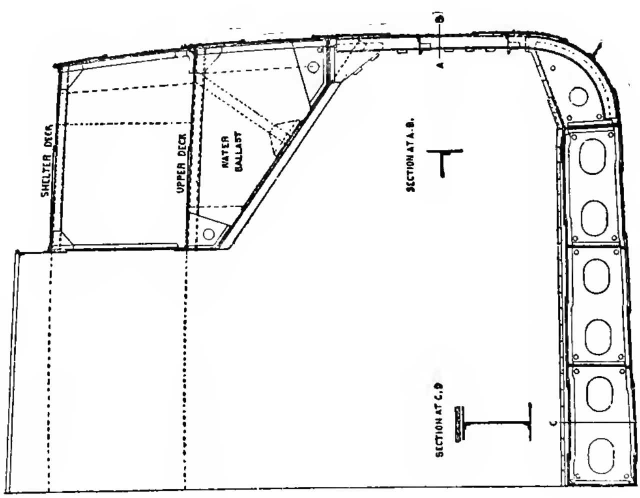


Fig. 82.

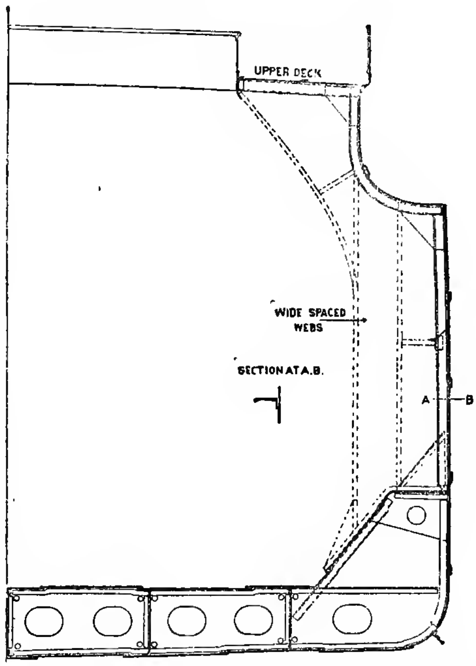
SS. 390′ 0″ × 55′ 9″ × 26′ 6″ and 35′ 0″.

great longitudinal strength and is also well suited to resist the tendency to transverse change of form set up when a vessel is labouring in a seaway. Self-trimming also forms the chief claim to distinction of the vessel shown in fig. 83. It is seen to resemble the last type somewhat with the corner tanks away; and on the latter account is not so efficient from a strength standpoint. As in the Ropner trunk vessel, the ship is worked from a central fore-and-aft platform.

Still another variation of the trunk or turret type is that devised by Mr. Henry Burrell. Like the other vessels just referred to this one is a self-

Fig. 84.

SS. 305′ 0″ × 46′ 9″ × 24′ 0″ and 30′ 3″.

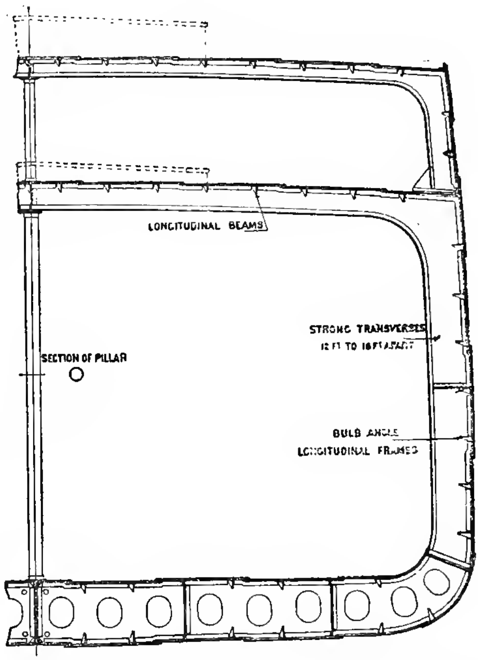


trimmer, and, as well as the upper trunk, has the corners at the bilges filled in (*see* fig. 84) and the inner surface sloped towards the centre, thus obviating the broken stowage space which might otherwise occur at the bilges. Incidentally, the corners thus cut off from the holds form a desirable addition to the ballast capacity. The deck, sides and trunkways are supported by cantilever webs, and there are no hold pillars.

Other special types have been built, or are building, differing more or less from the foregoing, but in general not sufficiently to make it necessary to refer to them. One design, however, that of Mr. Isherwood, is of such distinctive and interesting a character as to warrant its being singled out. This type is framed on the longitudinal system, and in this respect recalls that famous work of Scott Russell and Brunel—the *Great Eastern*. Like the earlier vessel, the new ship has main frames and beams running fore-and-aft, with widely spaced transverse partial bulkheads. There is, however, no double skin on the sides, the inner bottom being of the normal present-day type, except that the main internal framing is longitudinal instead of transverse.

Fig. 85 shows the midship section of a medium-sized cargo steamer framed on this system. The longitudinal beams and frames are seen to consist of bulb angles at wider spacing than on the transverse system. It should be noted, however, that the scantlings of the frames are gradually

Fig. 85.



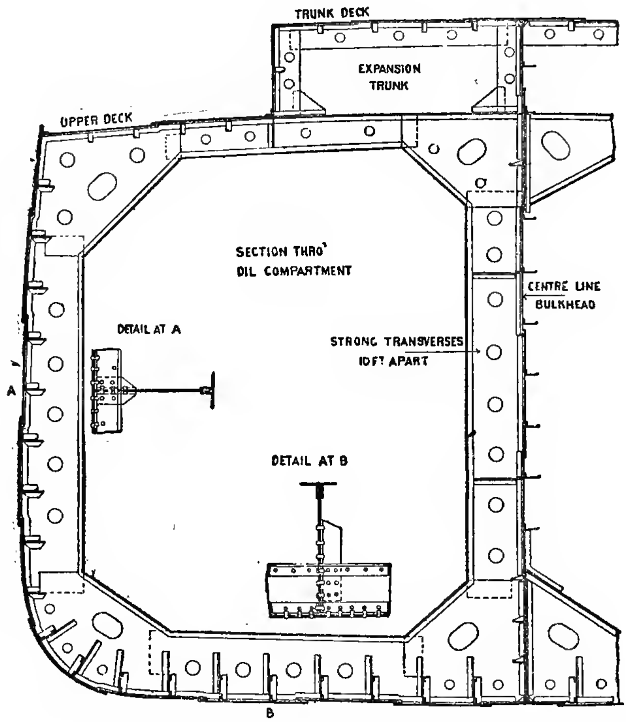
increased towards the bottom of the vessel, where they have to withstand greater loads, the intensity of the water pressure increasing in proportion to the depth below the surface.

The transverse strength is made up by strong transverses or partial bulkheads attached to the shell-plating between the frames, and stiffened on their inner edges by stout angles. The transverses are spaced from about 12 feet to 16 feet apart in ordinary cases, according to size of vessel, the largest vessels having the closest spacing.

The double bottom, as previously mentioned, has fore-and-aft continuous girders, with intercostal transverse floors in line with the tranverses and also midway between them, the latter being required to provide sufficient strength for docking purposes and to resist the excessive stresses which come on the bottom through grounding.

It is claimed for this type of vessel, several samples of which are now afloat and giving good accounts of themselves, that it has greater strength and less relative weight than the normal type. The saving in weight is a point of great importance, as apart from any reduction of first cost which this may represent, it means for the vessel greater deadweight capability and therefore increased earning power.

Fig. 86.



The Isherwood system of construction appears to be specially suitable for oil vessels.* Fig. 86 is a section of an oil steamer framed in this way, the dimensions of which are, viz.:—length, 355 feet; breadth, extreme, 49 feet, 5 inches; depth at centre, 29 feet. The longitudinal frames from the deck to the upper turn of the bilge are bulb angles as shown; on the bottom they are built of plates and bars; the spacing is 29 inches. The beams are bulb angles spaced 27 inches apart. The main oil tanks are 30 feet long, and two strong transverses are fitted in each tank between the boundary bulk-

* See a paper by Mr. Isherwood in the T.I.N.A. for 1908, from which figs. 85 and 86 are taken.

heads. The transverses are fitted to the shell-plating between double angles and have heavy double angles on their inner edges.

The longitudinal frames and beams and longitudinal stiffeners on middle line bulkhead are cut at the transverse bulkheads and efficiently bracketed thereto in order to maintain the continuity of strength. In way of the double bottom, which is fitted for a portion of the vessel's length amidships, alternate transverses are fitted continuously around the bottom to the middle line, and the longitudinal girders are fitted in long lengths between these transverses, and efficiently attached thereto. The remaining transverses are stopped at the deep girder in the double bottom next the margin-plate, and are then fitted intercostally between the longitudinals to the centre line. The margin-plate is fitted intercostally between the transverses, and connected to them by double-riveted watertight collars.

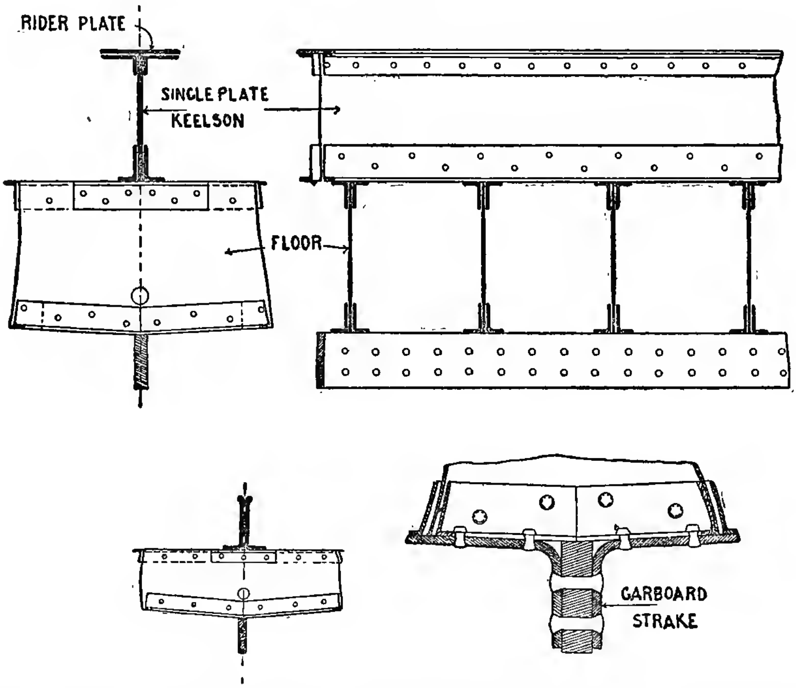
A comparison of the longitudinal stress acting on the bridge gunwale amidships of this vessel with that acting on an oil vessel of the same dimensions built on the ordinary system, showed the former to be 18½ per cent. less than the latter. In spite of this there is stated to be an estimated saving in weight of materials, under the new system, of 275 tons.

CHAPTER V.

Practical Details.

KEELS AND CENTRE KEELSONS.—The keel may be considered the foundation of a ship's structure. The simplest form of keel fitted in iron or steel vessels consists of a forged bar running almost the full length of the vessel. At the ends it is scarphed into the stem and sternpost, the three items together forming a complete longitudinal rib. The bars forming

Fig. 87.



the keel are fitted in lengths averaging about 40 feet, joined together by vertical scarphs nine times the thickness of the keel in length. These scarphs are frequently riveted up previous to the fitting of the shell by means of small tack rivets, so as to allow the keel to be faired. There is an objection to the use of tack rivets, in that, if it be necessary to remove a keel length, say, for repairs after grounding, plates on both sides of the keel have to be removed in order to knock out the tack rivets; for this reason they are sometimes omitted. The main rivets connecting the scarphs together, and also the keel to the shell, are of large diameter, spaced five diameters apart, centre to centre, and arranged in two rows, usually chain style, as in

Fig. 88.

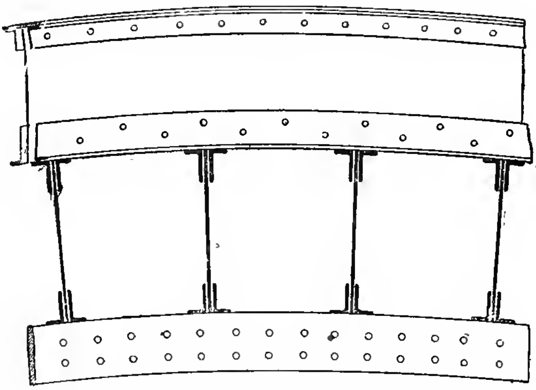
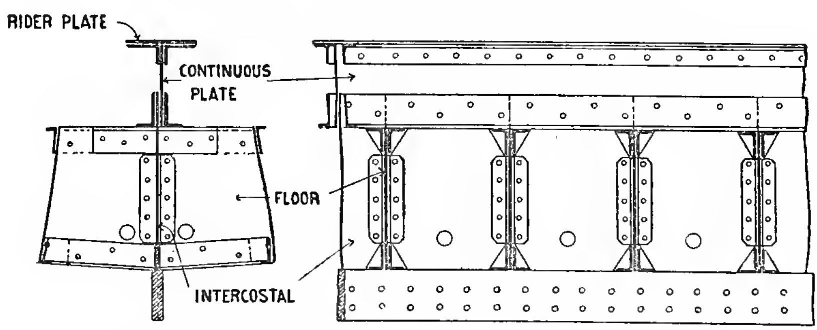


fig. 87, although zig-zag riveting is occasionally adopted; in the latter case, care must be taken to keep the rivets clear of the garboard strake butts.

It will be seen by referring to fig. 87 that the only connection that this keel has to the main structure is through the riveted connection to the garboard strakes. For this reason it is frequently called a hanging keel.

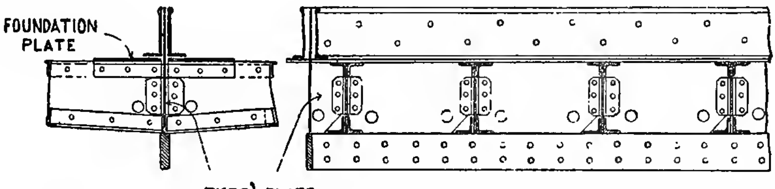
Fig. 89.

BAR KEEL: INTERCOSTAL CENTRE KEELSON



The simple bar keel is sometimes fitted in association with a centre keelson running along the tops of the floors, consisting of double bulb angles in small vessels, and a vertical plate and four angles, two top and bottom, in larger vessels. In the largest vessels, a rider plate is fitted on top of the upper angles and a foundation plate on top of floors below lower angles. This style of keelson, which is depicted in fig. 87, but without a foundation plate, is seen to have no direct connection with the external keel. From what we already know of the bending of beams, we must see that the arrangement is by no means a perfect one. Bending separately, the keel and keelson do not offer the same resistance as if rigidly joined. Moreover, the floors lying at right angles to the line of stress give no support, but develop a tendency to trip, as shown exaggerated in fig. 88. The weaknesses pointed out in the above plan may be largely corrected by fitting plates between the floors, from the

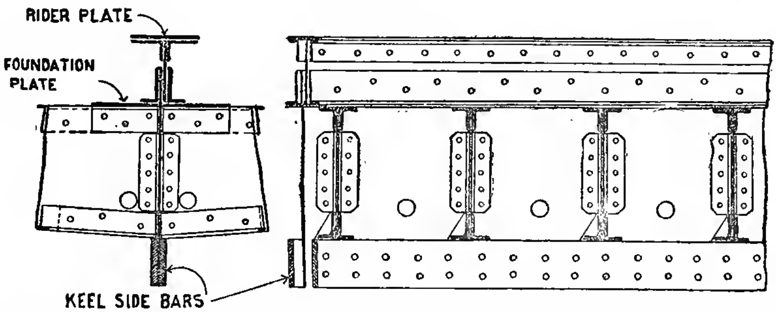
Fig. 90.



keel to the keelson (fig. 89). This transforms the separate beams of small resisting power into one powerful girder. The intercostal plates, too, prevent any possibility of movement of the floors.

A better arrangement than the preceding one consists of a continuous centre through plate, extending from the top of keel to the top of floors, or top of keelson (*see* fig. 90). Sometimes the centre plate is extended to the bottom of the keel, the required thickness of the latter being made up by means of two side bars (*see* fig. 90a). These arrangements, of course, entail

Fig. 90a.



the severing of the floors at the middle line, causing considerable reduction in the transverse strength; the floors are, however, connected to the centre girder by means of double angles, and a flat plate 12 inches wide, and of the same thickness as the centre through plate, is fitted on top of the floors each side of the centre plate, and connected to the latter by angles or bulb angles, thus making good the transverse strength and adding to the longitudinal strength. In the largest vessels the keelson is run up high enough to take four angles with a rider plate on the upper two (*see* fig. 90a).

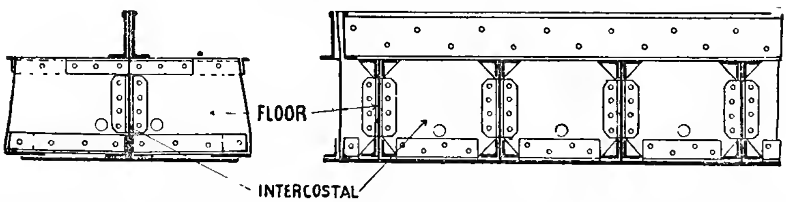
It will be observed that a practical difficulty crops up in the riveting of the keel to the garboard strakes, in the case of a side bar keel, as five thicknesses of plating require to be united by the same rivets. There are two rows of such rivets, of size and spacing similar to the bar keel, and as it is usual to punch these holes before fitting the plates, it can be imagined that very careful workmanship is needed to keep the rivet holes concentric. As a matter of fact, they are frequently more or less obstructed. In such cases, before proceeding with the riveting, the holes should be rimered out. The objectionable plan of drifting partially blind holes—that is to say, of driving a tapered bar of round steel or drift punch into them, so as to clear a passage for the rivet—should not be encouraged. It is known, and has been proved many times in practice, that the bruising which the material round the edges of the holes gets by drifting, renders it brittle and therefore liable to break away, loose rivets resulting in consequence.

An objection common to all projecting keels is the increase of draught which they entail. It is always considered a good feature in a vessel, and

Fig. 91.

FLAT PLATE KEEL

INTERCOSTAL CENTRE KEELSON



particularly in a cargo vessel, to have a moderate draught of water. The reason, of course, is that many ports will be open to a vessel, if of shallow draught, which would otherwise be closed. These considerations have led many owners to adopt what is called a flat-plate keel in preference to the one we have been dealing with. In this case, the ordinary shell-plating is continued under the vessel instead of being stopped on each side of a projecting keel; the middle line strake is increased somewhat in thickness, and is considered to be the keel of the vessel (*see* figs. 91 and 91a). This horizontal plate would of itself be a very inefficient substitute for the rigid vertical bar of the ordinary keel, but it is usually fitted in conjunction with an intercostal or continuous vertical centre plate, the two being connected together by double angle bars. With an intercostal centre plate, the floor plates are continuous; with a continuous centre plate, they are severed at middle line, and abut against the centre plate on each side. In both cases the floor and centre plates are connected by double vertical bars; this prevents any movement of the parts. It should be noticed that with a flat-plate keel, the rolling reducing property of the projecting keel is lost. It is the custom, however, in modern cargo vessels, to make up for this by fitting longitudinal bars or rolling chocks at the bilges (*see* figs. 75, 76, etc.).

The flat-plate type of keel is frequently fitted where there is a double bottom (fig. 91b). As the floor plates are then of considerable depth, a much more satisfactory connection with the centre plate is obtainable than with ordinary shallow floors; by Lloyd's Rules double angles are not required in this case, except in the machinery space, where they are always necessary, until the transverse number reaches 66, corresponding to a vessel say, 300′ × 40′ × 26, when double angles are required for half length amidships.

Fig. 91a.

CENTRE THRO' PLATE KEELSON

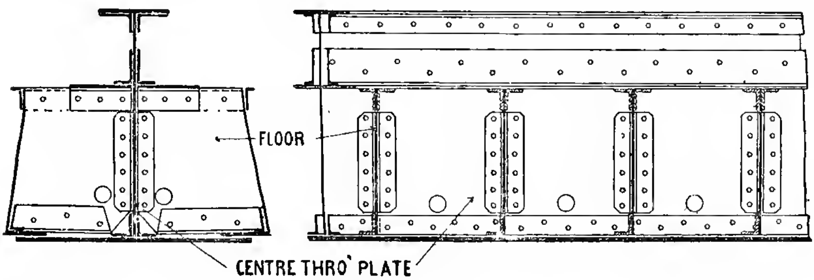
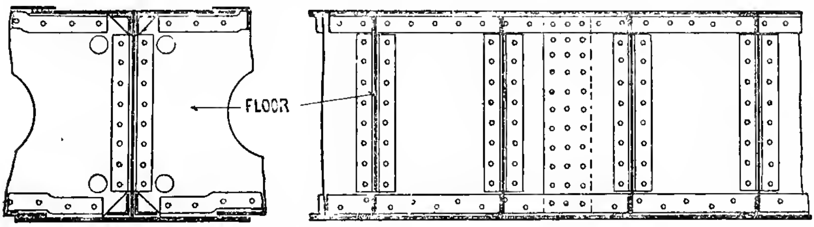


Fig. 91b.

CONTINUOUS CENTRE GIRDER - DOUBLE BOTTOM



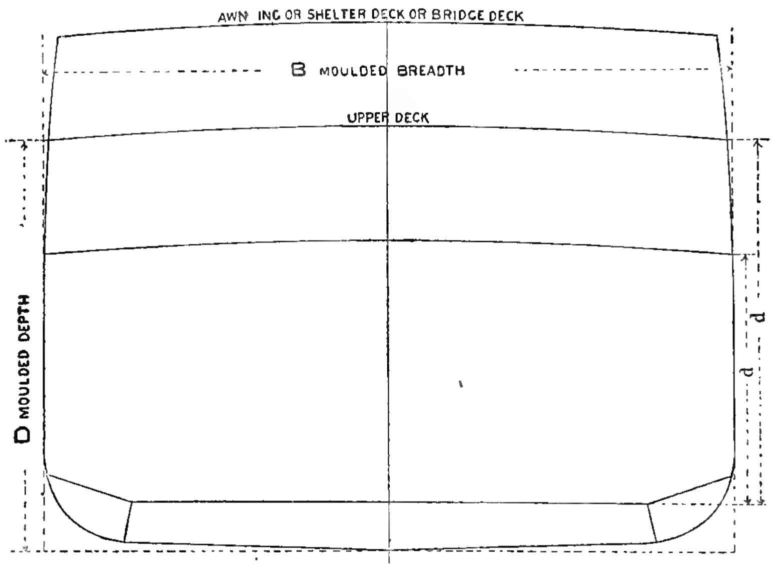
LLOYD'S NUMERALS.—In the previous paragraph reference has been made to Lloyd's Rules, and we now propose briefly to consider the methods adopted in these Regulations for the construction of ships, of assessing the scantlings of the various parts of a vessel. Lloyd's Rules in this particular differ in details from the Rules provided by other classification bodies, but for the purpose of illustration it may be sufficient to refer to them alone, particularly as they represent the common practice of present-day shipbuilding. The numbers under which the Tables of Scantlings are graduated, are derived from the dimensions; it is therefore necessary to define these. The definitions given in Lloyd's Rules are as follows:—

LENGTH.—The length (L) is to be measured from the fore part of the stem to the after part of the sternpost on the range of the upper-deck beams, except in awning or shelter-deck vessels, where it is to be measured on the range of the deck beams next below the awning or shelter deck.

BREADTH.—The breadth (B) is to be the greatest moulded breadth of the vessel.

DEPTH.—The depth (D) is to be measured at mid-length from the top of keel to top of beam at side of uppermost continuous deck, except in awning or shelter-deck vessels, where it may be taken to the deck next below the awning or shelter deck, provided the height of the 'tween decks

Fig. 92.



does not exceed 8 feet; B and D are indicated in fig. 92, which shows an outline midship section of a vessel.

From these dimensions the scantling numbers are obtained thus:—

$$\text{Transverse number} = B + D$$

$$\text{Longitudinal number} = L \times (B + D).$$

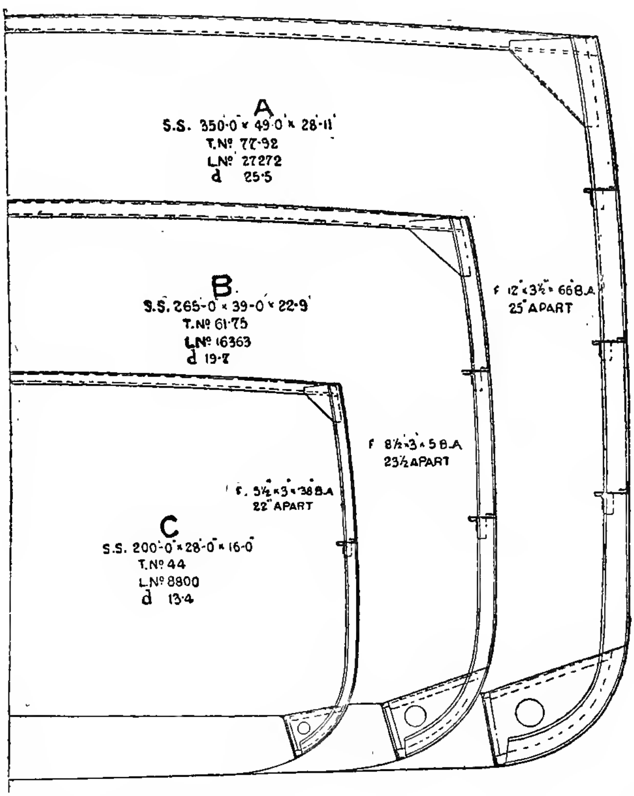
The transverse number regulates the frame spacing and the scantlings of the floors. Thus, taking a vessel of 45 feet breadth and 28 feet depth, we have—

$$\text{Transverse number} = 45 + 28 = 73.$$

And under this number we find in the appropriate Table of the Rules that the frame spacing should be $24\frac{1}{2}$ inches, and the floors 30 inches deep at middle, ·46 of an inch thick for $\frac{3}{5}$ length amidships, tapering to ·38 of an inch at ends.

The scantlings of the frames are governed by the transverse number, *i.e.*, by the size of the vessel, and also by the extent to which the frame is unsupported. The frame is assumed to be supported at the first tier of beams above the base and at the bilge. Reverting to fig. 92, *d* is the unsupported length of frame. Two cases are indicated, one assuming a tier of beams to exist below the upper deck, another assuming the frame to be unsupported from the bilge to the upper deck. It will be observed that at the bilge *d* is measured from a line squared out from the tank at side.

Fig. 93.



In this case there is an inner bottom; when a vessel has ordinary floors, the line is squared out from the height of the floors at middle.

The rules provide scantlings of frames for values of *d* up to 27 feet, this figure apparently marking the limit of a purely single-deck vessel. In fig. 93 the framing of three single-deck vessels of different dimensions is given, and shows how the scantlings increase with increase in size of vessel. The longitudinal number regulates the scantlings of the keel, stem, sternpost, side and bottom plating, double bottom, side stringers, keelsons, lower deck stringer plates, and lower deck plating. It is also employed with a number giving the proportions of length to depth in fixing the scantlings of the upper works.

The importance of distribution of materials and depth of girder, pointed out in Chapter IV., is fully recognised in the Rules. Thus the heaviest materials are concentrated at the side and deck-plating of upper, awning, and shelter decks, and of long bridges. Also the scantlings at these parts are less in a deep vessel than in one that is proportionately shallow.

The depth employed in obtaining the proportions of length to depth for use with the Tables is to be measured at the middle of the length from the top of keel to the top deck at side in all cases, except in way of a short bridge, when the depth is to be taken to the upper deck, which thus becomes the strength deck. The scantlings at the upper deck beyond the ends of a long bridge, are to be determined by taking the depth for proportions to the upper deck.

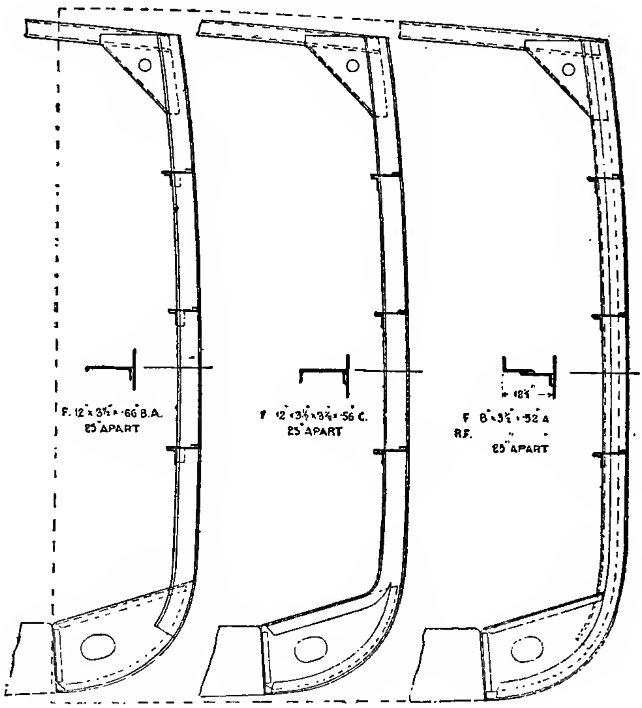
Shallow vessels, which have lengths equal to or exceeding $13\frac{1}{2}$ depths, taken to the upper deck, are required to have a bridge extending over the midship half length, or compensation in lieu. As the bridge deck becomes the strength deck, this means a substantial increase of the depth of the ship girder. In the case of still shallower vessels, namely, those having lengths exceeding 14 depths, the question of the longitudinal strength has to be carefully considered, and Lloyd's Committee require proposals to be laid before them.

FRAMES.—Next to the keel the transverse frame is probably the most fundamental part of a ship's structure, especially in vessels with ordinary floors. As previously explained, it extends from the keel to the top of the vessel in a transverse plane, and gives the form of the ship at the point at which it is fitted (*see* figs. 37 and 74). Frames of vessels built to Lloyd's Rules may be spaced from 20 to 33 inches apart, according to the size of vessel. In special cases, the spacing may exceed 33 inches, if suitable compensation be made. At the fore end, from a fifth the vessel's length from the stem to the collision bulkhead, owing to the pounding stresses to which this part of the vessel is liable at sea, the frame spacing should not exceed 27 inches, unless the frames are doubled to the lowest tier of the beams. In the peaks the frame spacing should not be greater than 24 inches.

Each complete transverse frame may be made up of two angle bars, *i.e.*, a frame and reversed frame, as described in Chapter III.; or it may consist, as in many modern cargo steamers, of a single angle or bulb angle; or it may be of channel section, with the addition, in the case of a large vessel, of a reversed angle. Lloyd's Rules provide tables of scantlings of frames of these various styles. In fig. 94 the side framing required for the vessel marked *A* in fig. 93 is shown, the three equivalent types being indicated.

The fore-and-aft flange of a frame is riveted to the shell-plating, and the transverse flange, in vessels having ordinary floors, is at its lower part attached to a floorplate. When the construction consists of a frame and reversed frame, the latter is riveted to the frame on the sides of the vessel, to the turn of the bilge, whence it sweeps along the top edge of the floor, which being thus stiffened at top and bottom, becomes an efficient transverse girder. Both the frames and reversed frames are usually butted at the centre line, covering angle bar straps being fitted. The frame butt-straps, or heel-pieces, as they are called, are usually about 3 feet long, and are placed back to back with the frame, the floor-plate being between. These heel-pieces should be so fitted as to bear on the top of the keel, when of simple bar type, as in this way stresses due to docking or grounding are communicated

Fig. 94.

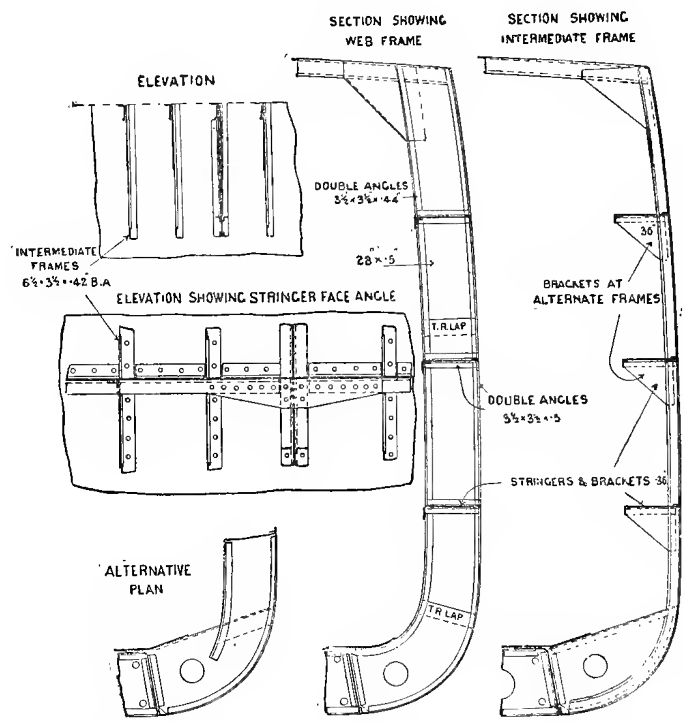


directly to the framing, without unduly straining the rivets connecting the keel to the garboard strakes. Heel-pieces are only fitted for three-quarters the vessel's length amidships, the form at the ends making them unnecessary. Where the middle line keelson is a centre through plate, the heel-pieces are not usually fitted; and where the former is associated with a flat-plate keel it is, of course, impracticable to fit them.

At the decks, the framing on each side of the vessel is connected by cross beams, special attention being given to the beam-knee connections, as the combination of beam, frame, shell-plating and deck-stringer in this neigbourhood is most efficient for resisting the transverse racking stresses to which, as we have seen, a vessel may be subjected when rolling among waves at sea.

WEB FRAMES.—When a transverse rib consists of a deep plate with stiffening angles on its inner edge, it is known as a web frame. Lloyd's Rules permit a system of web frames at six frame spaces apart, with comparatively light intermediate frames, to be substituted for the heavier frames of the ordinary frame table, provided a deck be laid on the tier of beams at the height *d*. In fig. 95, the largest of the three vessels indicated in fig. 93 is shown with web frames. It will be seen that the angles connecting the

Fig. 95.



webs to the shell-plating and to the side stringers are of the same thickness as the webs, and that angles are fitted to the inner edges of the webs and stringers. When webs become of considerable depth, they are really partial bulkheads, and to develop their full efficiency should have a substantial shell connection. Thus webs 24 inches and above, in vessels built to Lloyd's Rules, require double angles to the shell-plating, or equivalent single angles double riveted.

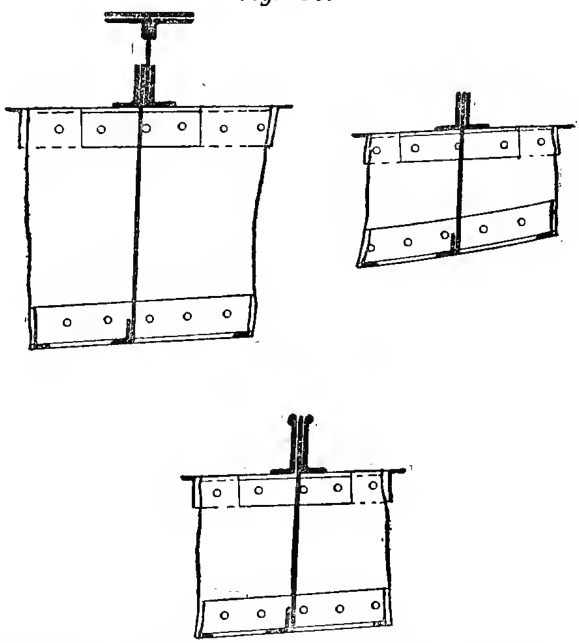
A web frame being held rigidly at the deck by its connection to the beams, and at the bilge by its floor or tank side connection, forms a girder of comparatively short span, at least when compared with the side stringers, which are only properly held at the bulkheads. For this reason it is advisable to make the web frames continuous and the side stringers intercostal, and this is usually done. At the junction of each web and stringer, the discontinuity of the latter is made good by a double angle connection to the webs, and by fitting a stout buttstrap to the stringer face bar (*see* fig. 95). Web frames are attached by bracket knees to beams at their heads, the knees being double riveted in each arm and flanged on their inner edge. At the lower part, when associated with an ordinary floor, the inner edge of the web frame is swept into the top edge of the former, the connection to the floor being an overlapped riveted one. When the connection is to be made to an inner bottom, it should consist of a riveted angle on to the margin plate, with, in addition, a substantial gusset plate or angle bar from the top bar of the web on to the inner bottom plating (*see* fig. 95.)

FLOORS.—These vertical plates will be observed to have a maximum depth, governed by the size of vessel, at the middle line where the transverse bending stresses are greatest. Thence they gradually taper towards the sides, the depth at three-quarters the half breadth, measured out from the middle line on the run of the frame, being half that at centre line. From this point, the upper edge of each floor sweeps into the line of the inside of the frame, terminating at a height from the base line equal to twice its depth at the middle line. There is one such floor at each frame, the floor, frame, and reversed frame forming, indeed, a transverse girder, which is the most characteristic feature of a vessel built without an inner bottom. Except in small vessels, the floor-plates are fitted in two pieces, connected by an overlap or buttstrap at the middle line, or alternately on each side of that line. When a vertical through centre-plate is fitted, the floors are fitted close against it on each side, a riveted connection being made by double vertical angle bars, as shown in fig. 90. The loss of transverse strength due to cutting the floors is also partly made good by a horizontal keelson plate fitted at the centre line on top of floors, referred to when dealing with centre keelsons. When inner bottoms are fitted, this part of the structure undergoes considerable modifications, as we shall see presently.

SIDE KEELSONS.—As well as the centre keelson, vessels with ordinary floors have keelsons midway between the bilge and the middle line. The main function of these keelsons being to keep the frames and floors in their correct relative positions, intercostal plates are fitted between the floors and connected to the shell-plating and to double angles on top of the floors. These intercostal plates need not be connected to the floors, but in order to develop their full efficiency should be fitted close between them. In small vessels, under 27 feet breadth, one side keelson is considered sufficient; in larger vessels, 27 feet and under 50 feet in breadth, two are necessary. In fig. 96, side keelsons for vessels of various sizes are shown.

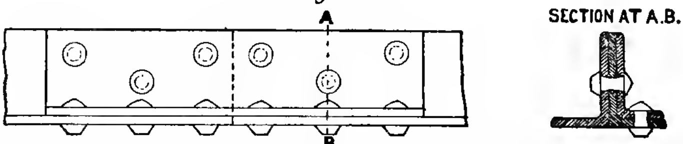
BILGE KEELSON.—In vessels of 50 feet and under 54 feet breadth, in addition to two side keelsons, a bilge keelson is required on each side, and should be carried as far forward and aft as practicable. This keelson, like a side keelson, should have an intercostal plate connected to the shell-plating (*see* fig. 96).

Fig. 96.



SIDE STRINGERS.—Between the bilge and the deck beams the framing is tied together, and stresses to some extent distributed by stringers consisting in small vessels of single angles riveted to reversed frames and lugs, and in larger vessels of similar angles associated with an intercostal plate connected to the shell-plating. Lloyd's Rules require side stringers of this latter type in vessels of all sizes. According to these Regulations, the number

Fig. 97.



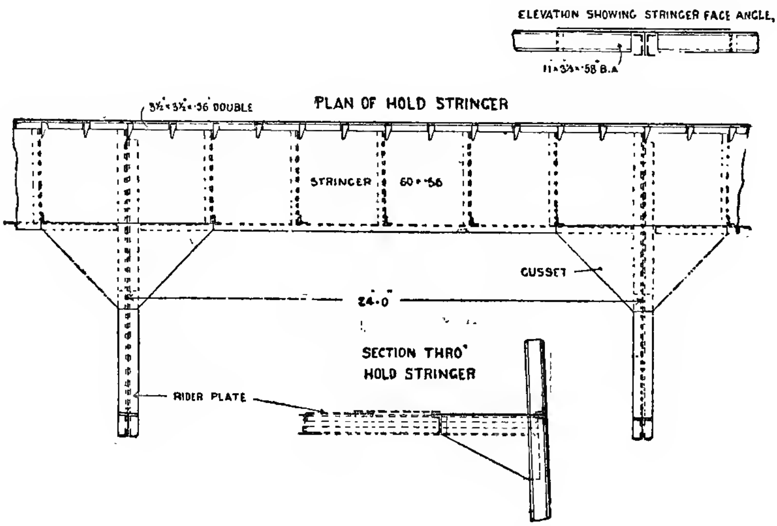
of side stringers depends on the value of *d*. When this is 7 feet and less than 14 feet, that is in very small vessels, one is sufficient; where *d* is 14 feet to 21 feet, two are necessary; and when *d* is 20 feet and under 27 feet, three should be fitted.

All keelson and stringer plates and angle bars, when continuous, should be fitted in long lengths, and to obviate any sudden discontinuity of the strength, adjoining butts should be carefully shifted from each other. Both plates and bars should be strapped at the butts, the angle-bar straps consisting of bosom pieces of the same thickness as the keelson bar and two feet long, having not less than three rivets on each side of a butt (*see* fig. 97).

BEAMS.—A tier of beams is always fitted so as to tie the top of the frames together and support the deck; the functions of beams as elements of strength in the structure of a ship are generally as have been already described (*see* pages 71, 72).

The number of tiers of beams required in any vessel is a question of transverse strength, but it also depends on the trade for which she is intended.

Fig. 98.



Passenger boats usually require one or more decks below the upper one for purposes of accommodation. Many cargo vessels, owing to the nature of their cargo, also need one or more 'tween deck spaces; in most cargo boats, however, as has been said elsewhere, the desire is for deep holds, clear of beams or other obstructions. Lloyd's latest Rules allow considerable scope to the designer in this matter. As was seen in a previous paragraph, they allow him, up to a certain point, to design his vessel entirely clear of beams below the upper deck, if he so wishes. The value of d, in such a case, will, of course, be relatively great, and the scantlings of the side framing relatively heavy (*see* fig. 93). This is the penalty exacted for unobstructed holds, and it is obviously a just one; for each frame is a girder, whose strength is governed by its span.

Spacing of Beams.—According to Lloyd's Rules, a complete tier of beams, such as is required to form the upper point of support of the frames in regulating the scantlings of the latter, may consist of:—

(1) Beams at every frame.
(2) Beams at alternate frames.
(3) Beams widely spaced up to 24 feet apart.

In arrangements (2) and (3) the beams are heavier than in (1); and in (3) a broad stringer must be fitted on the ends of the beams with a heavy face bar, and large horizontal gussets must be fitted between the stringer and the beams (*see* fig. 98). Lloyd's Rules require beams to be fitted at every frame in the following places:—

(1) At all watertight flats.
(2) At upper decks of single-deck vessels above 15 feet in depth.
(3) At unsheathed upper decks, when a complete steel deck is required by the Rules.
(4) At unsheathed bridge, shelter and awning decks.
(5) At upper, shelter, or awning decks in vessels over 450 feet in length, whether the decks be sheathed or not. Under erections, such as poops, bridges, and forecastles in vessels less than 66 feet in breadth, upper deck beams may be on alternate frames, except for one-tenth the vessel's length within each end of a bridge, where they are to be fitted at every frame.
(6) At the sides of hatchways, including those of engine and boiler openings, in all unsheathed steel or iron decks.

Elsewhere, deck beams may be spaced two frame spaces apart, but only if the frame spacing does not exceed 27 inches.

It is easy to grasp the reason for close spacing the beams on unsheathed decks. With thin decks and widely spaced beams, the plating would probably sag between the latter, which would make the decks most unsightly; with beams at every frame the sagging tendency will be very slight. The close beam spacing in way of hatches is necessary in view of the heavy weights which may be brought upon the deck there during loading operations.

When a wood deck is laid, beams may be at alternate frames (except as above stated in vessels over 450 feet). In this case, the steel deck is supported between the beams by the wood deck, the deck fastenings for the wood deck being fitted between the beams. The beams forming the weather decks are usually cambered, so as to throw off water quickly; those of the lower tiers are sometimes cambered and sometimes straight. The usual camber given to weather decks is $\frac{1}{4}$ inch per foot of length of beam. Lloyd's Rules allow the beams of weather decks to be fitted without camber, or with less than the usual amount, in the case of large vessels (30,000 longitudinal number) if at least half the length of the top continuous deck is covered by erections. On the principle of the arch, it might be thought that camber should give additional strength to beams, but, as has been pointed out,* the sides of a ship are not really abutments, so that this can scarcely be the case.

BEAM SECTIONS.—Beams are fitted of different sections according to the strength required; most of these being included in fig. 99. Sections *A* and *B*, and in large vessels, section *E*, are adopted when the beams are at every frame. *C* is not often used for ordinary beams; it is a common section, however, for special beams built to carry the ship's boats. *D* and *H* are used when beams are to be fitted to alternate frames. Sections *F* and *J* are only fitted where extra strength is required. In way of the machinery, the material binding the sides of the vessel together is usually very much cut away owing to the necessity of providing ample space for shipping and unshipping the engines and boilers; it is, therefore, of importance to make any beams that may be got across the vessel in that locality as strong as possible; beams similar to those just mentioned are usually fitted, with satisfactory results. *G* is a form of beam found suitable for the ends of hatchways, the angle bar being, of course, fitted away from the hatch opening. In general, where heavy permanent deck weights are carried, specially strong beams are needed, and the section adopted will be that dictated by the experience of the designer.

Fig. 99.

In order to obtain special strength, and to allow of substantial knee connections between the beam ends and the frames, ship beams are not reduced in depth towards their extremities, as might be done in the case of an ordinary loaded girder supported at the ends. The reasons for having great strength at this part of a ship have already been explained.

BEAM KNEES.—We shall now describe a few methods of forming and fitting beam knees. Several examples taken from Lloyd's Rules, of a common, and very efficient one when the workmanship is good, is shown at fig. 100. It is seen to consist of a triangular plate, fitted into the angle between the top of beam and the ship's side, and well riveted to the beam end and the frame. Sometimes for lightness, and to minimise obstruction to stowage, the inner edge of the knee-plate is hollowed. This knee can be fitted to any of the beam sections given above.

Another way of forming a beam knee is to cut away the lower bulb for a short distance from the end of the beam, and weld in a piece of plate or bulb plate, the knee being afterwards trimmed to the size and shape required. This is called a slabbed knee (*see* fig. 101). Unless great care is exercised, the

* See *Practical Shipbuilding*, by A. Campbell Holms.

welds of these knees will give trouble; for this reason this style is not so popular as the bracket knee, nor as the one we are about to describe. In

Fig. 100.

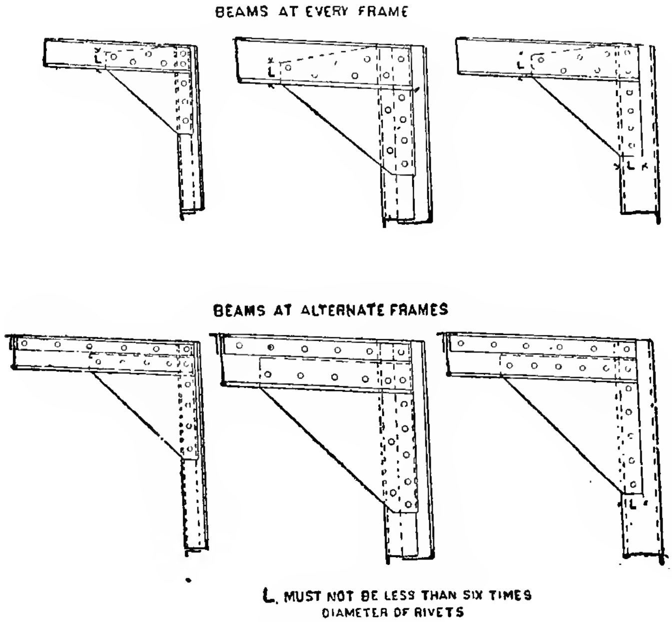


Fig. 101.

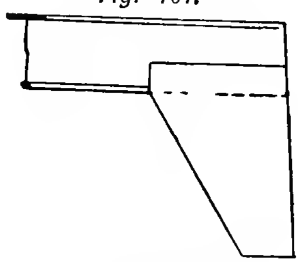
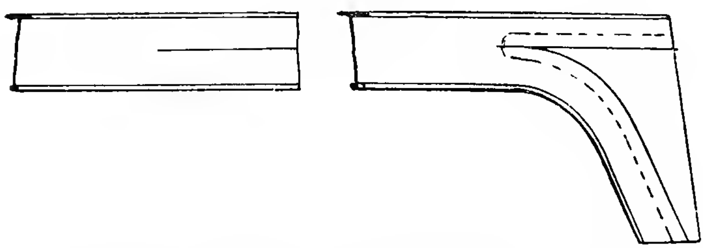


Fig. 102.



this last case, each end of the beam is split horizontally at about the middle of the depth, as indicated in fig. 102, and the lower part is turned down;

a piece of plate is welded into the space so formed, and, finally, the beam is cut to shape and size. This knee also depends on the quality of the welds, but it is stronger than the previous one, and has a fine appearance; it is known as a turned knee.

As the stresses are mainly met by the shearing strength of the rivets, these must be sufficient in number and diameter. Lloyd's Rules require that in knees under 17 inches deep there shall be not less than 4 rivets of $\frac{3}{4}$-inch diameter in each arm, while knees 40 inches deep require nine $\frac{7}{8}$-inch diameter rivets in each arm; the number varies between these limits for knees of intermediate depths.

Obviously, only about half the number of rivets required in bracket knees will be needed for welded knees of the same depth.

The depth and thickness of a beam knee varies with the depth of the beam, and the position of the latter in the ship. Generally, beams which form the top of a hold space are required of maximum depth, the distorting stresses being greatest at these places. Hence, in steamers where there is but a single tier of beams, the beam knees are of greater depth than if there were intermediate decks. The upper deck beam knees in vessels which have a range of wide-spaced beams below the upper deck, are to be of the scantlings of the knees of an upper deck tier where it is the only one. In sailing ships, beam knees at all tiers of beams are to be the same as for upper deck beams of similar scantling in steamers having one tier only. It may be mentioned that Lloyd's Rules require beams in sailing ships to be heavier than those of the same lengths in steamers having one tier only. These requirements are very necessary as sailing vessels have no watertight bulkheads except one fitted at the extreme forward end; they are, therefore, without the rigid transverse stiffening which every steamer possesses in virtue of her bulkheads, and need the bracing given by beams of special strength and depth of knees.

All beam knees should measure across the throats at least $\frac{6}{10}$ of the full depth of the knee.

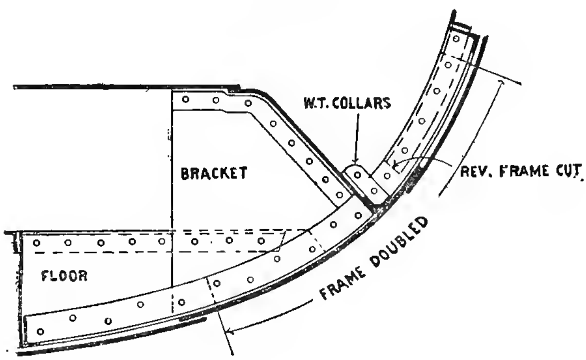
In large single-deck vessels the beams at every frame are to have plate bracket knees varying in size with the moulded depth. Thus, in vessels 23 feet and under 24 feet depth, the knees are to be 33 inches × 33 inches; and in vessels 26 feet and under 27 feet, 42 inches × 42 inches; the knees for vessels of intermediate depths varying between these. Deepening the knees strengthens the frames by shortening the unsupported length; it also stiffens the vessel at the deck corners and arrests any tendency to change of form that might develop when the vessel is labouring among waves at sea.

BALLAST TANKS.—Nearly all modern cargo steamers are constructed to load water-ballast when necessary, the water being carried in double bottoms, in peak tanks, in deep tanks, or in some other space specially devised for the purpose. Frequently, all these methods are employed together in a single steamer, when it is desired to be able to proceed to sea without using supplementary stone or sand ballast.

Ballast tanks are not usually fitted in sailing ships, as, unlike steamers, they have long voyages to perform and load and discharge comparatively seldom. In their case, therefore, rapid means of ballasting are of little use. Another important reason for omitting ballast tanks in sailing ships is the saving in first cost. Still, where there have been special reasons for so doing, double bottoms, and even deep tanks, have been installed in sailing ships.

When water began to be introduced as a means of ballasting, shipbuilders devised many more or less successful plans, to which we need not here refer, for economically carrying it, before they arrived at the efficient system now generally adopted. From the first it was seen that the broken space between the shell-plating and the tops of the floors in the bottom of the ship was admirably suited for this purpose, since use could thus be made of space not otherwise available for profitable employment. In the earliest vessels the tanks were usually only in one or two holds, and to obtain an adequate ballast capacity the tanks

Fig. 103.

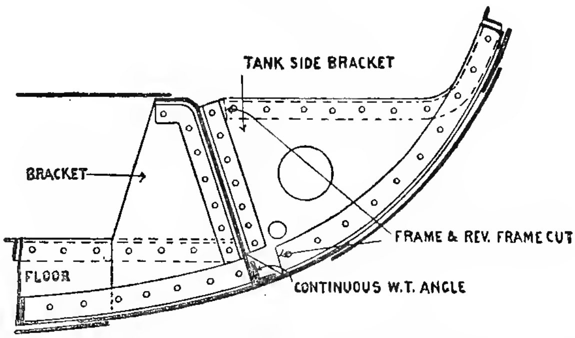


had to be deepened, which had the manifest drawback of encroaching considerably on the cargo space. The advantages, however, of the convenient ballast were considered sufficient to outweigh this loss, and many steamers were thus built. The usual plan followed, was to fit longitudinal girders spaced about 3 feet apart on top of the ordinary floors and to cover them with plating, the tanks being sealed at the sides by carrying the tank top-plating down to the shell and connecting it thereto by means of an angle bar caulked watertight. It was at first found rather difficult to make a satisfactory joint at the ship's side. One method, shown in fig. 103, was to sever the reversed frames in way of the tank, to cut a hole in the margin plate for the frame so that the former could go close against the shell-plating, and then to joggle a bar round the frames and against the shell. The cutting of the reversed frames was compensated for by doubling the frame in the neighbourhood of the tank margin.

Another method, which did not call for the cutting of the reversed frame, consisted in working a smithwrought angle bar round frames and reversed frames, and against the shell-plating, and filling in the little apertures left behind the reversed frames with plugs of wrought iron, the latter being tightly wedged into place and carefully caulked. Neither of these methods was found to be very satisfactory, as the abrupt termination of the tanks gave rise to decided weakness in the structure at the bilges; moreover, even with careful workmanship, watertightness at the margin was difficult to secure.

Both of these objections were eventually overcome by severing the main and reversed frames at the tank margin, and fitting a continuous bar directly on to the shell-plating, the loss of transverse strength being made good by fitting substantial brackets from the frame bar on to the margin plate of the tank top. This arrangement, known as the **M'Intyre System** from the name of its introducer, is, in principle, the one now adopted at the margin in all vessels having a ballast tank extending over the greater part of the length. In the earliest vessels built on this system, the angle connecting the margin-plate to the shell-plating was fitted inside the tank and the margin-plate connected to the tank-top by an angle bar; now, the margin-plate is flanged at the

Fig. 104.



top and the shell bar brought outside the tank, improvements which have led to much better workmanship. Fig. 104 shows the improved M'Intyre System.

In constructing a ballast tank extending over a portion only of the length, a point of importance is the maintenance of the longitudinal strength at the breaks. To stop the tank structure abruptly at any point would accentuate the weakness of sections lying immediately beyond. In such cases the usual plan is to continue the keelsons of the part of the structure clear of the double bottom, so as to scarph the latter for some distance (Lloyd's Rules require a minimum scarph of three frame spaces), and to connect them to the longitudinal girders where practicable.

It very soon came to be recognised that by making a ballast tank continuous, and for the full length of a vessel, other advantages besides the important one of carrying water-ballast could be secured. It was seen, for instance, that the double skin afforded by the tank top-plating would greatly increase a vessel's safety against foundering, in the event of grounding on a rocky bottom; also that the material required for the construction of the tank, being at a considerable distance from the neutral axis, would be very efficient in resisting longitudinal structural stresses. These considerations led to the adoption of a continuous ballast tank in many vessels, and when later, the Board of Trade consented to measure the depth for tonnage in such cases to the inner bottom plating, a fore-and-aft double bottom became the rule in cargo steamers.

The fitting of a full length tank brought immediate changes in the internal framing of this part of the hull. It was now found possible to reduce the depth of the tank as compared with that of one extending over a part only of the length; but this made it impracticable to follow the usual plan in building of fitting longitudinal girders on top of ordinary floors, and a **Cellular System** of construction was introduced, and came to be generally adopted.

There are two principal methods of constructing a double bottom on this system, illustrated in figs. 105* and 106*, respectively. The first consists of longitudinal girders suitably spaced, and floorplates fitted at alternate frames, the girders being connected to the inner and outer bottoms by angle bars. By Lloyd's Rules at least one longitudinal girder is required in vessels under 34 feet breadth, whose breadth of tank amidships is under 28 feet, and two in vessels between 34 feet and 50 feet in breadth, whose breadth of tank amidships is between 28 and 36 feet. Sometimes the parts are flanged in lieu of angles, but although the cost is thus somewhat reduced, there being fewer parts to fit and less riveting, there is a loss in rigidity, for which reason flanged work here is not very common. In way of the engine space, owing to the great vibration there due to the working of the machinery, the floorplates are fitted at every frame and stiffened at their upper edges by double reversed bars; floorplates must also occur at the boiler bearers. As a rule flanged work is not resorted to in this region.† Before and abaft the engine space, at those frames to which no floorplates are attached, brackets, which in medium-sized‡ vessels should be wide enough at the head to take three rivets in the vertical flange of the intermediate reversed angles for $\frac{3}{5}$ the vessel's length amidships, are fitted to the centre girder and margin plate, binding these parts together and strengthening them to resist the stresses set up by the action of the water ballast when the vessel is in motion among waves. The reversed bars in way of these intermediate frames are riveted to the tank top-plating, to which they act as stiffeners; frequently, however, they are dispensed with and the inner bottom slightly increased in thickness in lieu.

The side girders and floors are pierced with manholes to give ready access to all parts of the tank, a considerable saving in weight being also thus effected. The centre girder is more important than the others, forming as it does a kind of internal keel; it has, therefore, heavier scantlings, is not reduced by manholes except perhaps at the extreme ends, and is stiffened top and bottom by heavy double bars (*see* fig. 105). In the earliest vessels built on this plan, the longitudinals were continuous and the floors intercostal, the

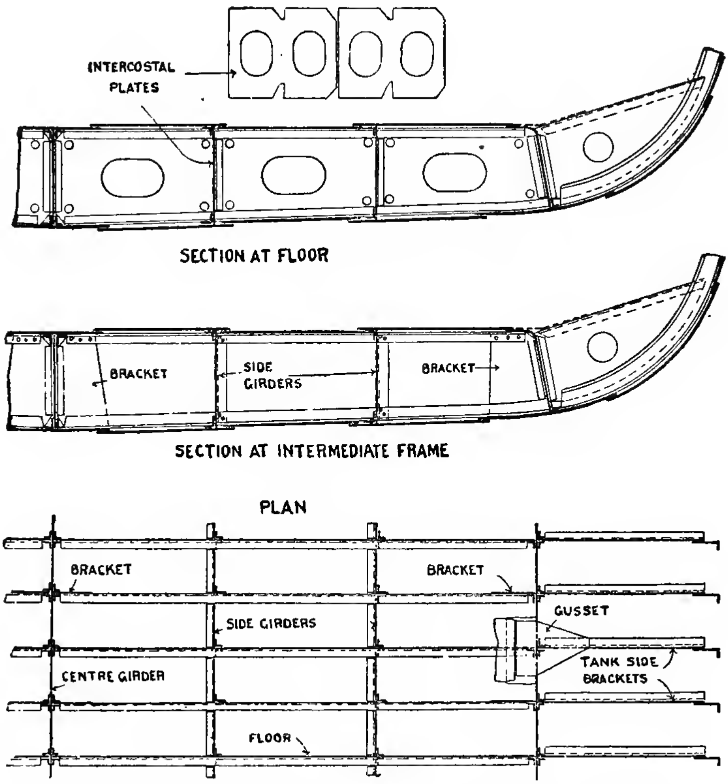
* Figures taken from Lloyd's Rules.

† *See* Remarks on Stiffening of Double Bottom at Fore End, p. 116.

‡ When the longitudinal number is 20,000 and above.

former, owing to their distances from the neutral axis, being considerable elements in the longitudinal strength. Nowadays, it is usual for the floors to be continuous and the girders intercostal, an arrangement leading to greater simplicity of construction—a most important point—and to some reduction in longitudinal strength, which, however, in modern vessels, calculations show to be still ample. An important advantage of the latter arrangement is a great increase in the stiffness of the bottom, the comparatively short floorplates obviously having greater strength and rigidity than long fore-and-aft girders.

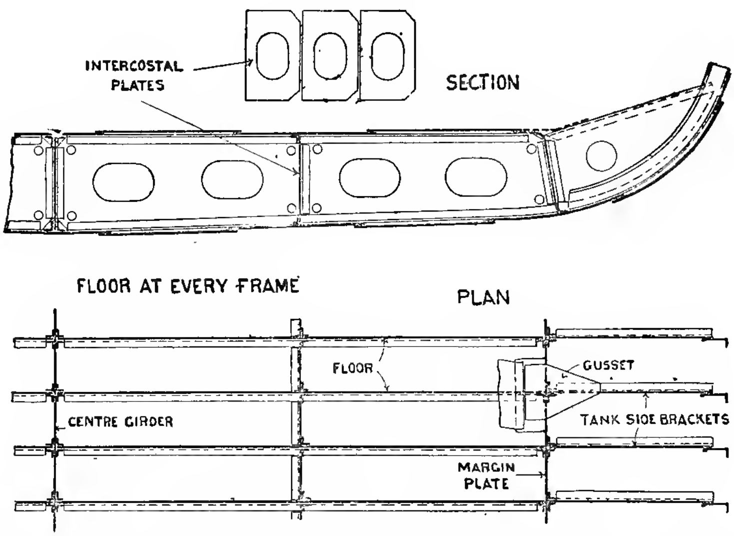
Fig. 105.



When longitudinals are continuous, in order that their efficiency as strength elements may not be impaired, the manholes through them should be as few as possible, and those in different girders shifted well clear of each other transversely.

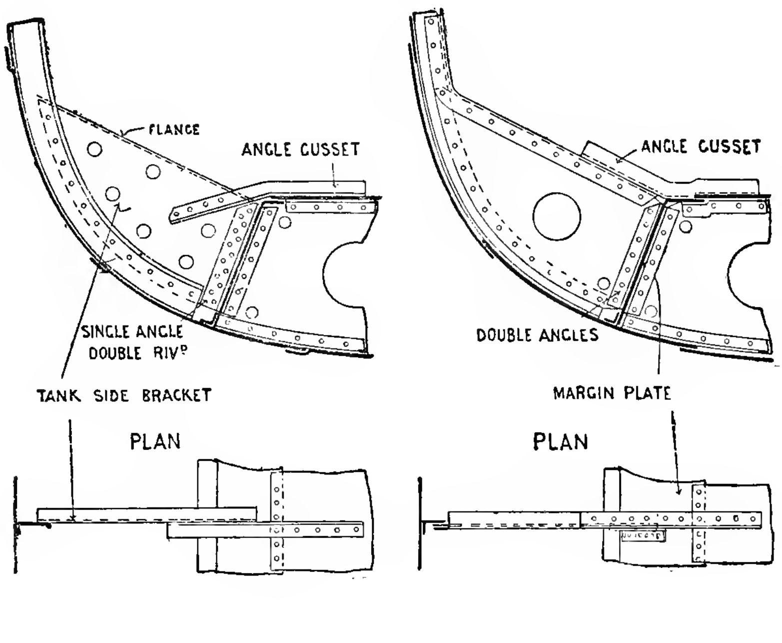
When the rule lengths of vessels exceed 400 feet, and in single-deck vessels which exceed 26 feet moulded depth, the above plan of framing the inner bottom is not considered adequate, and the second method, shown in fig. 106, should be adopted. In this case, the floorplates are at every frame and continuous from centre girder to margin plate on each side, while the longitudinals, except the centre one, are intercostal. The floorplates and side girders have lightening holes, one or two through the floors into each cellular space, and one through every intercostal girder plate. Fewer side girders are required by this plan, only a single one being necessary on each side of the centre if the ballast tank be under 36 feet in width and the breadth of ship under 50 feet, and only two if the tank be under 48 feet and the ship under 62 feet in breadth, the number being proportionately increased in larger vessels; the spacing of the girders with the closer floors gives approximately the same extent of unsupported area of shell and tank top-plating as in the previous case. The intercostals are attached to the floors and to the

Fig. 106.



inner and outer skins by riveted angle bars or flanges; and the floors, as well as being riveted to the frames and to angle bars under the inner bottom plating, have angle connections to the centre girder and the margin-plate; the centre girder attachments, consisting of double angles for half length amidships when the transverse number reaches or exceeds 66. In the latest vessels of this size, single angle attachments between the floors and centre girder have sometimes been adopted, the flanges being double riveted. It is seen that as regards the internal framing of the inner bottom, the longitudinal strength in this last plan is somewhat less than in the previous one, but in view of the tendency towards increase of breadth in modern vessels, demanding considerable transverse strength, and of the greatly enhanced stiffness of the bottom on account of the numerous deep floorplates, it would appear that the continuous floor on every frame method of construction is the better one, particularly as the absence of the bracket work required at the intermediate spaces in the previous plan renders the work of simpler construction. This arrangement is at anyrate a favourite with many builders, who have frequently adopted it in much smaller vessels than those requiring it owing to their size. Lloyd's Rules recognise the greater strength of this plan over the previous one by allowing the shell-plating (except the flat keel and garboard strakes) in way of the tank, when ·52 to ·66 of an inch in thickness, to be slightly reduced; when the plating exceeds ·66 of an inch in thickness, no reduction is allowed.

Fig. 107.



A part of the structure at which weakness has often been found developed in vessels fitted with double bottoms is where the side framing is attached to the margin plates. Experience with actual vessels has shown the need of making this connection amply strong, many of them having exhibited signs of movement in the shape of loose rivets in the angles connecting the side brackets to the margin plates. In Lloyd's Rules a minimum breadth of margin plate is given, with a corresponding minimum number and size of rivets for making the bracket connection. In small vessels, single angles are considered sufficient to join the side brackets to the margin-plate, but with increase in breadth and depth, particularly the latter, double bars or equivalent single bars with double-riveted flanges quickly become necessary for this purpose, over some portion at least of the vessel's length. Experience has shown the fore end to require special attention in this respect, and Lloyd's Rules demand double angles from the collision bulkhead to one-fourth the vessel's length from the stem in vessels of moderate size. Besides the foregoing, with the growth of vessels additional strength becomes necessary at the tank margin, which is provided by fitting gusset-plates to the tops of the wing brackets and to the sides of the tank top-plating. In recent instances, angles have been substituted for the gusset-plates with good results. Detailed sketches of these arrangements are shown in figs. 105, 106, and 107. These gusset-plates or angles are fitted at every fifth, fourth, third, second or single frame, according to the vessel's size, the limits being fixed in each case by the transverse number and the value of d, *i.e.*, the length of unsupported frame.

The fitting of the inner bottom plating calls for little comment. Lloyd's Rules recommend that it be arranged in longitudinal strakes and the butts shifted well clear of each other and of those of the longitndinal girders, when these are continuous, and this is usually done. In some districts, notably the N.E. coast, transverse strips have been fitted under the watertight bulkheads to allow the building of the latter to be proceeded with at an early stage of the work, but the system cannot be otherwise commended. To save the fitting of packing pieces, inner bottom plating is sometimes joggled at the seams, but objection has been raised to the depression thus caused in the surface, as forming lodgments for water, particularly where ceiling is laid, rapid corrosion resulting in consequence. Like other longitudinal material, the thickness of this plating is reduced at the ends; it is increased in way of the machinery space to give additional strength, but more particularly to allow for the corrosion which takes place there.

Structural efficiency in double bottoms would not be obtained were care not bestowed on the riveted connections. It is especially important that the centre girder should not be weakened at the butts; except in the case of small vessels, therefore, these must not be less than treble riveted amidships, and in very large vessels they should be quadruple riveted.* The butts of side girders, where continuous, should be double riveted, and in large vessels treble riveted. The tank top-plating is an important element in both the longitudinal and transverse strength, and the riveting of butts and seams calls for careful consideration. The butts of the middle line strake, and those of the margin plate, must be at least double riveted throughout; in large vessels they should be treble and in the largest vessels quadruple riveted. The remaining butts of inner bottom plating are to be double riveted for half

* In Lloyd's Rules, and also in those of the British Corporation, the requirements as to riveting of butts and edge joints at any part of a ship are fixed by the thickness of the plating and the position of the part.

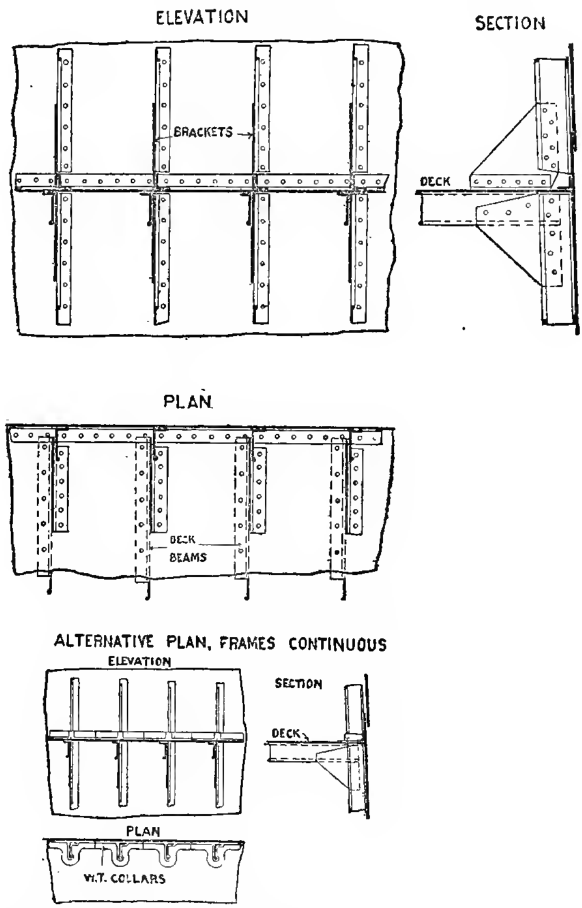
length, and in large vessels treble riveted. The edges of the middle line strake, where the transverse bending stresses are greatest, except in small vessels, should be double riveted, and in the largest vessels this should apply to the remainder of the plating; medium-sized vessels have single riveted edges clear of the middle line strake. All the preceding connections are usually overlapped.

On page 73 reference was made to special stresses which come upon the fore-ends of full vessels when sailing in light trim among waves at sea. To resist these stresses, such vessels are provided with extra strengthening forward in way of the inner bottom. If built on the cellular system with comparatively closely spaced longitudinals, continuous or otherwise, and floors at alternate frames, the floors are to be fitted to every frame, and the main frames doubled from margin plate to margin plate from the collision bulkhead to a fifth of the vessel's length aft, measuring from the stem; also the three strakes of plating next the keel are to maintain their midship thickness forward to the collision bulkhead. If built with continuous floors at every frame, the frames are to be doubled, and the shell-plating increased, as in the last case; but in addition, special intercostal girders must be fitted of a depth equal to half that of the centre girder, and be extended as far forward as practicable. In both cases the rivets through the plating and frames in this region are to be of closer spacing than elsewhere. It should be mentioned that in vessels having ballast tanks constructed with longitudinal girders on top of ordinary floors, and in those without inner bottoms, if of full forms, adequate strengthening of a similar nature to the above is necessary.

PEAK TANKS.—In steamers, after-peaks are now usually adapted for water ballast; in some cases, the fore-peak is also so constructed, but more rarely. The principal value of peak tanks is, of course, their trimming effect; they are like weights situated at the extreme ends of a lever poised at the middle, and have great power in this respect. In strengthening these compartments for their work, we have to bear in mind the special nature of the stresses set up by the load. Unlike ordinary cargo it does not lie on the floors, but presses immediately on to the shell, thus inducing severe stresses on the frame rivets which bind the shell-plating to the structure; for this reason the pitch of such rivets is not to exceed 5 diameters. It is not usual to increase the shell-plating or framing in thickness, as owing to the shape at this part these are amply strong. The boundary formed by the hold bulkhead, however, requires special attention; being flat and of considerable area, care has to be taken to prevent bulging; this is done by thickening the lower plating if the tank is a deep one, and making the stiffeners heavier and closer spaced than at ordinary bulkheads. A centre line bulkhead or washplate is required in all such compartments, to prevent the water from damaging the structure by dashing from side to side in the event of a free surface, and to minimise the effect which such free surface would have on a vessel's stability. Such washplates need not be of strong construction, but should be securely attached to the bulkhead and underside of the deck.

DEEP TANKS.—These usually consist of ordinary hold compartments specially strengthened to carry water ballast; one or two of them are frequently fitted in large modern cargo steamers. Where one only is required, the compartment immediately abaft the engines is usually adapted for the purpose; where there are two, they are generally placed one at either end of the engines and boiler space. When situated thus, the machinery being,

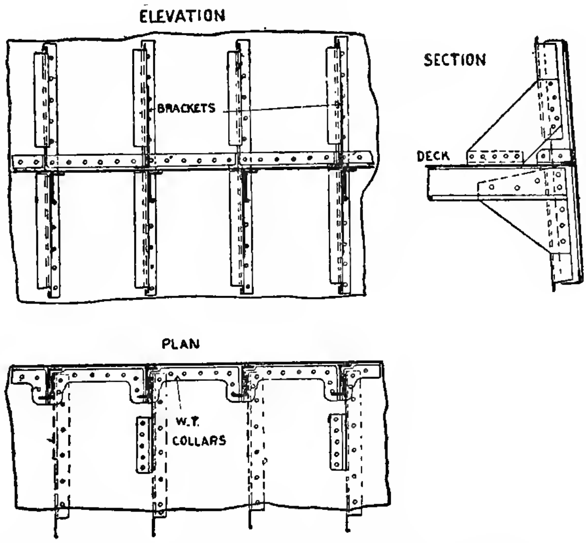
Fig. 108.



of course, assumed amidships, the water ballast has its greatest power in sinking the vessel bodily, without materially changing the trim.

The same system of strengthening is followed here as in the peak tanks, but deep tanks being larger, the stiffening has to be correspondingly increased. The end bulkheads must have heavy vertical and horizontal stiffeners of bulb angle, or other section, fitted on opposite sides and bracketed at their ends, in the one case to the double bottom and deck, and in the other, to the ship's sides. A centre line division is required, and must be of strong construction, as it takes the place of pillars, as well as acting as a wash plate. It should be connected by double angles to the deck and double bottom, and have substantial, close-pitched stiffeners bracketed at top and bottom. Cases have occurred in which the action of the water has swept this bulkhead entirely from its boundary connections, showing the need of having the latter specially strong. The deck forming the top of the tank is required to have beams spaced on every frame, and there should be large beam knee connections to the sides, as severe strains have been found developed at this part due to the action of the water in a partially-filled tank, when the vessel has been in motion at sea. Midway between the centre line and the ship's side, runners are required under the deck beams, and, in order

Fig. 109.



to tie the top and bottom of the tank together against the lifting forces exerted by the contained water when the vessel is in motion among waves, a row of strong pillars are required. The riveting through the frame and shell-plating is of close pitch in way of deep tanks, as the load acts directly on the shell-plating. At the ship's side watertightness is secured in much the same way as at the margin of a double bottom. A continuous bar is fitted and caulked to tank top plating and shell, the side frames, which have, of course, to be severed, being bracketed to the tank top (*see* fig. 108); sometimes, where the side framing consists of frame and reversed frame, only the reversed frames are cut, the frames being doubled in the vicinity and watertight collars fitted; in this last case, the severing of the reversed frames is further compensated for by fitting brackets at alternate frames (*see* fig. 109).

Access to deep tanks for the purpose of shipping cargo is obtained by means of watertight hatchways, one of which, owing to the centre division, is required on each side of the centre line.

OTHER TANKS.—As well as the foregoing, in many steamers special arrangements have been devised for carrying water ballast. Thus, in the Harroway and Dixon type of vessel, corner spaces under the deck are cut off from the holds, and specially strengthened for this purpose (*see* fig. 82). In the Burrell type the bilge corners are built up and utilised for water ballast (*see* fig. 84). In the latest Ropner Trunk type, portions of the trunk space have been specially strengthened to the same end. The M'Glashan system, which consists in continuing the double bottom up the sides of the ship to the height of the deck, is also worthy of mention. The chief point in favour of these arrangements is the high position thus secured for the centre of gravity of the water ballast, leading to steadiness and general good behaviour when in a seaway.

TESTING OF TANKS.—On completion, ballast tanks are tested for watertightness by putting them under water pressure. Each compartment of a double bottom intended for water ballast is pressed by a head of water to the height of the load waterline as being the greatest pressure it need bear in actual service. Peak tanks and deep tanks are tested by a head of water 8 feet above the top of tank, but the head must in no case be less than to the height of the load waterline.

PILLARS.—The importance of pillars in a ship structure has already been pointed out. It was shown that as struts and ties they communicate stresses from one part to another, and thus cause the strength of the various parts of the structure to act together. Short pillars are more effective than long ones, as the latter are liable to collapse by side bending at a much less strain than that represented by the compressive strength of the material. Pillars are, therefore, increased in diameters with their lengths; *e.g.*, in a vessel of 55 feet beam with two rows of pillars, the latter, if just under 8 feet long and supporting a beam of a third deck, should have a diameter of 4 inches, and if just under 22 feet a diameter of $5\frac{3}{4}$ inches; intermediate lengths having diameters between these. The strength of pillars should also advance with the loads they have to bear; for instance, those in the upper erections, since they have only the weight of the deck structure and load to support, may be of comparatively small size; those fitted under second or third decks below the upper deck, which may, therefore, have heavy loads of cargo to support, should be of considerable diameter. In the holds, too, the side pressure of the cargo is liable to bend the pillars unless they be of substantial diameter.

As well as acting as struts and ties, pillars greatly augment the strength of the beams they support. A pillar placed below the middle of a rectangular beam, supported at its ends but not fixed there, will double the strength of the latter and greatly increase its rigidity; beams in ships have fixed ends, but except as modified by this circumstance, the strength value of middle line pillars is equally great. If two pillars be fitted to a beam so as to divide its length equally, the effective span is a third of its original value, and the strength of the beam is correspondingly increased; and so on for any number of pillars. Use of this is made in vessels as they increase in breadth; for instance, beams 43 feet and under in length require only a centre row of pillars, but when they exceed this length, two rows become necessary; when the length of beam exceeds 60 feet, an additional row is required, placed

Fig. 110.

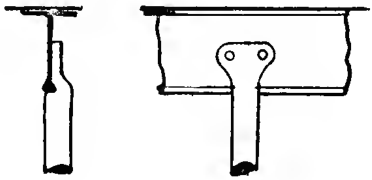


Fig. 111.

Fig. 112.

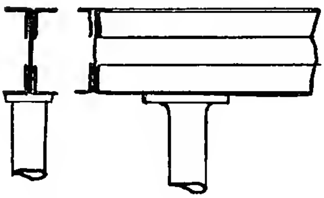


Fig. 113.

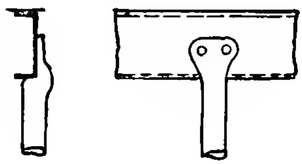


Fig. 114.

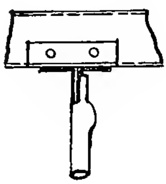


Fig. 115.

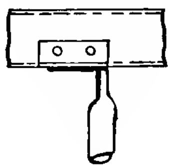
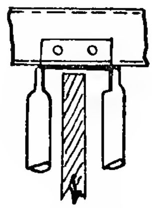


Fig. 116.



one at the centre line, and another at each quarter breadth of the vessel, those in the latter rows being hence called quarter pillars. At the ends of the vessel, as the beam decreases in length, the number of rows of pillars may be correspondingly reduced. Where there are several decks, the various rows of pillars should be arranged as nearly as possible over one another, in order to rigidly join the upper and lower parts of the ship's structure.

HEADS AND HEELS OF PILLARS.—The forms of the heads and heels of pillars are governed by the nature of the part of the vessel to which connection is made. Figs. 110 and 111 illustrate two methods of attachment to bulb-tee beams; the first is the usual one, the second is not so common but has the advantage of gripping the beam round the bulb, and so relieves the rivets when the pillar is under tension. Figs. 112 and 113 show the connection to *H* and channel beams respectively. When beams are fitted on every frame, the pillars being at alternate frames, it is necessary to have runners under them in way of the pillars so as to support the

Fig. 117.

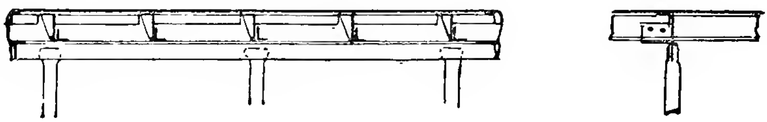


Fig. 118.

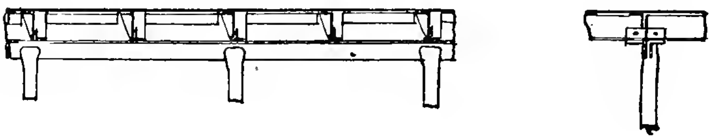


Fig. 119.

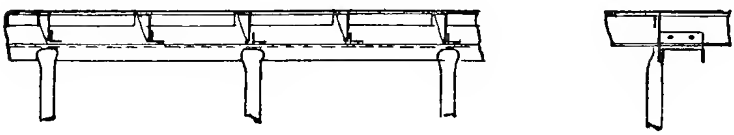


Fig. 120.

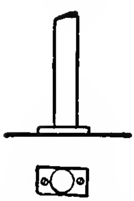


Fig. 121.

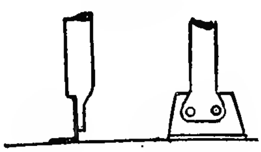


Fig. 122.

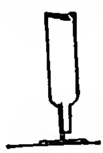
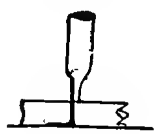


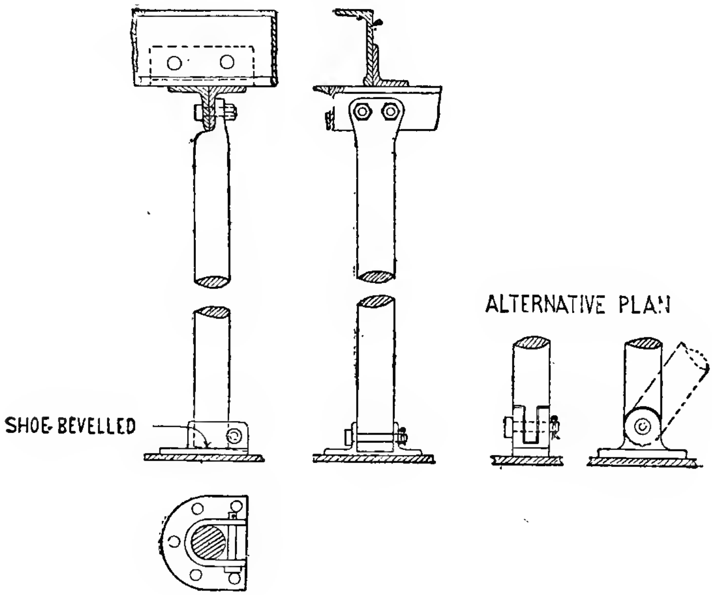
Fig. 123.



intermediate beams. These runners should consist of double angles, but may be of other approved form. Figs. 114 and 115 show two styles of the beam runner; it will be observed that the attachment to the beams is by means of a riveted angle lug; where the beams are of channel section this is unnecessary. Fig. 116 is a suitable form where the pillars have to be reeled for shifting boards, as is frequently the case with those at the middle line in cargo vessels. The plan adopted is to fit consecutive pillars on opposite flanges of the channel runner, thus forming two lines between which the shifting boards may be reeved. Where intercostals to the deck-plating are required, as in the case of quarter pillars to deep tanks, or where pillars are widely spaced, they are fitted in various ways, figs. 117, 118, and 119, also figs. 126 and 127, showing some of these with pillar head attachments. The intercostals transform the beam runners from simple ties into strong girders eminently qualified to stiffen the deck and to distribute the stresses.

The heel attachments of pillars are not so varied in form as those of the heads. A common one is shown in fig. 120, and is seen to consist of a horizontal shoe forged on to the lower end and through riveted to the deck-plating or to the beam, as the case may be. A favourite method when

Fig. 124.



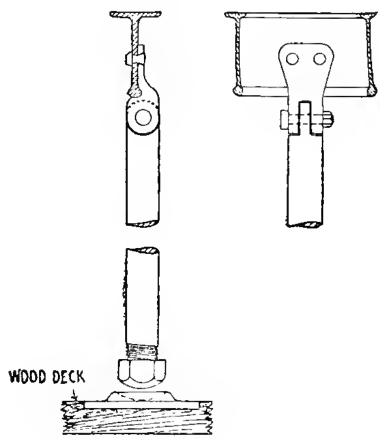
the heel comes on an inner bottom, is to rivet a short bulb angle or tee lug to the plating, and connect the pillar to the vertical flange (*see* figs. 121 and 122). This plan is sometimes followed for making attachments to steel decks when the latter are to be wood sheathed, the vertical flanges of the lugs being made deep enough to allow of the heel of the pillar being above the wood deck (*see* fig. 123) to facilitate the caulking of the latter at this part.

The end attachments of pillars should consist of at least two $\frac{7}{8}$-inch rivets. When they reach a length of 18 feet or a diameter of 4 inches there should be three rivets in each end. Pillars 5 inches or above in diameter require a four-rivet connection at the ends.

Pillars are frequently required to be portable; in way of the hatches, for example, they must be removed when loading many kinds of cargo. Bolt and nut fastenings are often adopted (*see* alternative plan in fig. 124), but occasionally, particularly at the heels, these are found impracticable or undesirable, and arrangements such as those of figs. 124 and 125 are resorted to. When fitted in either of these ways a pillar can, of course, only act as a strut.

WIDE-SPACED PILLARS.—While several rows of close-ranged pillars are valuable to a vessel as regards her strength, from a point of view of stowage they are obviously somewhat of a drawback. With almost all cargoes, the pillars, particularly those in the wings, must give rise to a considerable amount of broken stowage, and although by splitting up the cargo they prevent damage through side pressure when the vessel is rolling, many

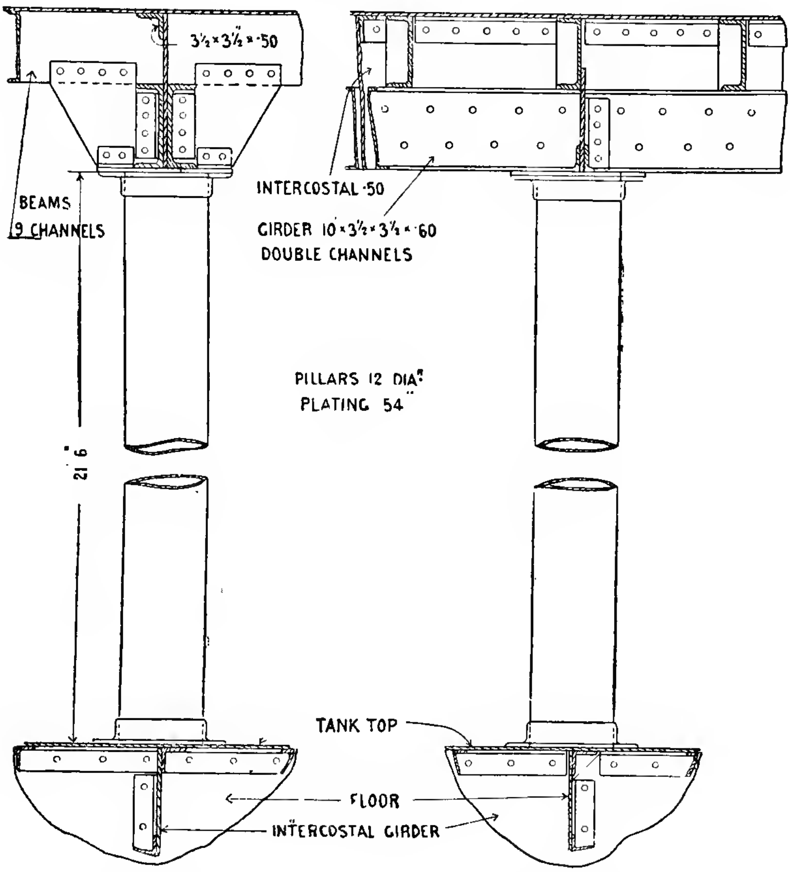
Fig. 125.



owners have sought to dispense with them where possible. For example, in vessels of a breadth requiring say, three complete rows of pillars, it is permitted, and, for the above reasons, usually preferred, to substitute instead one complete middle line row, with two rows in the wings at wider spacing. With this modified arrangement, however, beam runners, having intercostal attachments at each deck, must be fitted in way of each line of quarter pillars, the scantlings of the intercostals and pillars being governed by the spacing of the latter, the breadth of deck to be supported, and the probable load. In Lloyd's Rules, Tables are provided giving the scantlings of wide-spaced pillars and of the girders at their heads.

For many trades, as mentioned in the previous chapter, even when thus spaced, the pillars have been found to be too numerous, and the centre row has been dispensed with, and a very wide spacing adopted for the quarter pillars, in some vessels not more than two aside being fitted even in long holds. In such cases the decks have been supported, and the loads communicated to the pillars, by means of runner girders of enhanced strength, and the greater stresses brought upon the pillars have been met by making the latter of special size and construction. Figs. 126 and 127 illustrate two arrangements to Lloyd's requirements. These pillars, it will be noted, are

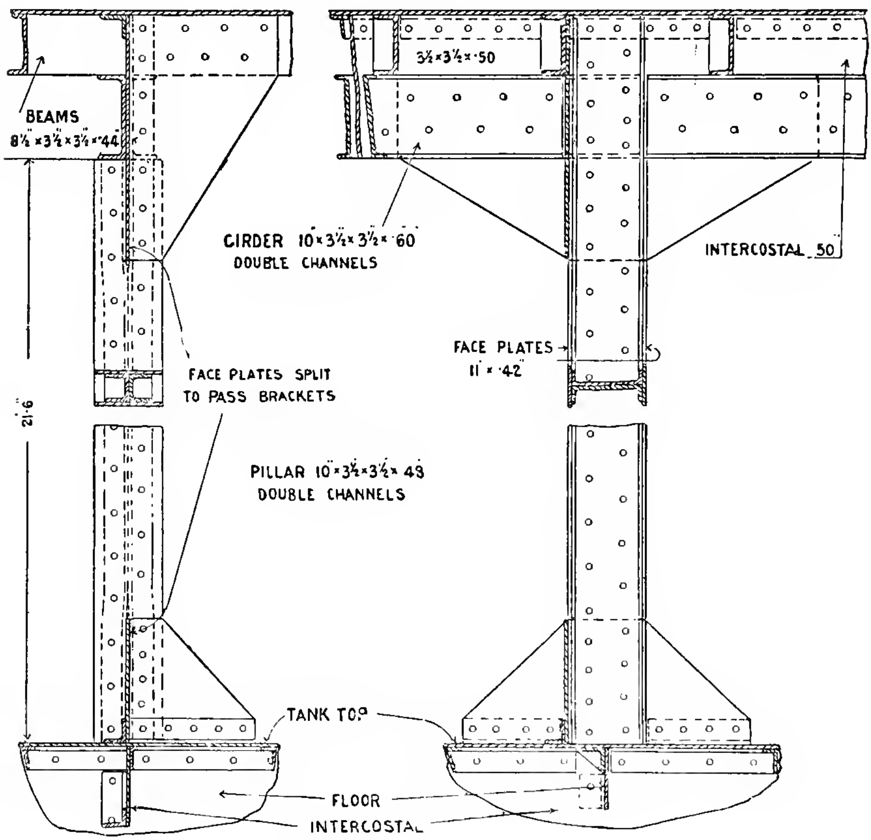
Fig. 126.



stepped on the tank-top at the junction of a floorplate and intercostal girder. This is necessary for rigidity, and when pillars cannot be so placed they must be similarly supported by means of brackets or have seatings built on the tank-top.

OUTER BOTTOM.—The most important part of any ship is the outside shell-plating. Its leading function is to give the structure a capacity to displace water, but, besides this, being spread like a garment over all the inner framing and securely riveted thereto in every direction, it binds the whole together and enables the various parts to efficiently resist the severe stresses brought upon them when the vessel is in lively motion among waves at sea. Every part of the shell-plating is of importance, but owing to their positions some parts must be of greater comparative strength than others. We have seen that the greatest longitudinal bending stresses come upon the upper and lower works, and the least in the vicinity of the neutral axis; so that,

Fig. 127.



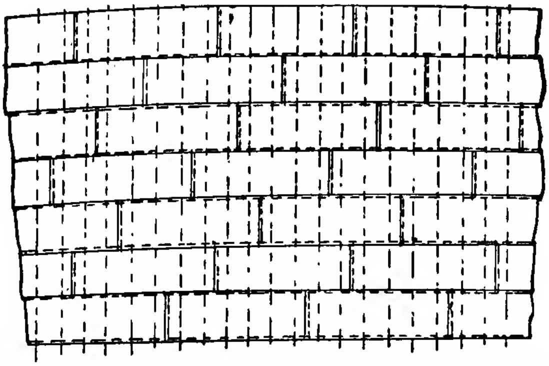
with the vessel upright, the sheerstrake at the top, and the keel, garboard, and adjacent strakes at the bottom, are most severely stressed, while the material at about mid-height—the position of the neutral axis—is stressed least. It has also been pointed out that the above conditions become modified when, through the rolling of the vessel, the side-plating is raised towards the top of the girder away from the neutral axis and has to sustain a much increased stress; this, and the fact that the longitudinal sheering stresses, where they occur in the length, are a maximum at the neutral axis, must be borne in mind when apportioning the scantlings to the various parts of the shell-plating. Towards the ends of the vessel the structural stresses are less than amidships, and the thicknesses are reduced; this applies to all longitudinal materials in a ship.

The Rules of all classification societies require the sheerstrake, the keel-strake, and those adjoining, to be specially heavy, the strakes from above the upper turn of the bilge to the sheerstrake being of smaller scantlings. Of course, in certain places, where severe local stresses may be anticipated, special strength is introduced. Thus, the afterhoods of the strakes which come on the sternposts in steamers are retained of midship thickness to withstand the stresses which the working of the propeller brings upon that part of the structure. For the same reason the plates in the immediate vicinity of the propeller shaft, called boss plates, are increased in thickness beyond that required for the same strakes midships. Usually, the shell-plating is thickened forward where it has to take the chafe of the anchors; and in some special vessels the plating in the vicinity of the stem is thickened to withstand ice pressure.

The actual thickness of the various parts of the shell-plating of a vessel are governed by the size of the latter. For example, in a small vessel, say one 90 feet or 100 feet in length and under 10 depths to length, with a longitudinal number under 3350, the shell scantlings in fractions of an inch would be:—keel-plate, amidships ·44, ends ·36; garboard strakes, where there is a bar keel, amidships ·34, ends ·30; shell-plating, from flat keel-plate or garboard strake, to strake below sheerstrake, amidships ·30, ends ·26; sheerstrake ·32; strake below ·3, ends ·26. In a cargo steamer of average size, say about 360 feet long, with a proportion of length to depth of between 11 and 12, and a longitudinal number of 28,400, the corresponding scantlings would be:—keel-plate, without doubling, ·94 to ·66; garboard strake with a bar keel, ·64 to ·54; from flat keel-plate or garboard strake to upper turn of bilge, ·60 to ·46; from upper turn of bilge to strake below sheerstrake, ·60 to ·44; sheerstrake, ·72 to ·44; strake below sheerstrake, ·62 to ·44. In each case, the second thickness is that at the ends. In each of these sets of scantlings, if the keel strake and sheerstrake be omitted, there is a comparative uniformity throughout; this is what might have been expected from our considerations above. Another point of interest is the small amount of taper towards the ends in the scantlings of the small vessels, compared with those of the other. Structurally, the end thicknesses in the smaller vessel are probably too great, but as even the maximum thicknesses are small, the necessity of allowing for wear and tear prevents the liberal reduction permissible in the heavier material of the larger vessel.

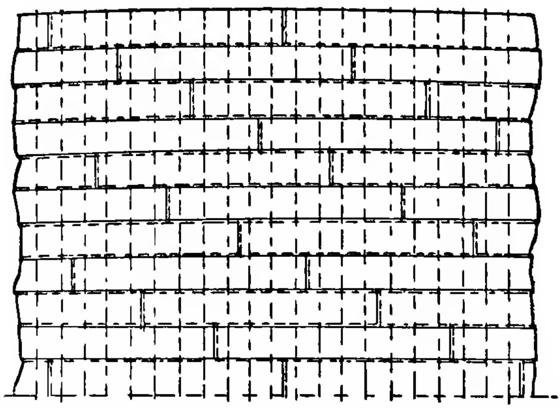
Having decided upon the scantlings, the next point of importance is to arrange the end joints of the plates forming the various strakes. These should be disposed in such a way as to avoid having too many weak points in the same transverse section. Lloyd's Rules stipulate that joints in adjoining strakes must not be nearer to each other than two spaces of frames, and those in alternate strakes at least one space clear. They also demand that the end joints or butts of the sheerstrake be shifted clear of those of the deck stringers by two frame spaces, and the end joints of the garboard

Fig. 128.



strake on one side of the ship clear by a like distance of those of the same strake on the other side. This latter precaution is of course because of the proximity of the garboard strakes, only the keel separating them. Fig. 128 illustrates a shift of joints or butts embodying the minimum requirements of Lloyd's Rules. It will be observed that there are here four passing strakes

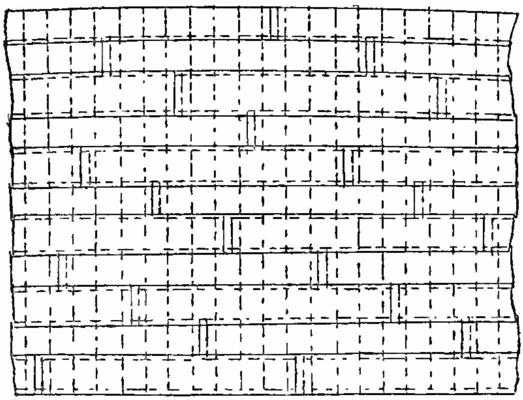
Fig. 129.



between joints which occur in the same frame space. Nowadays, as plates may be rolled to almost any desired length, a better disposition of joints than that of fig. 128 is easily obtainable. It is not, however, necessary to have more than a certain number of passing strakes between consecutive butts in a frame space, no more, indeed, than is required to ensure the same strength at a line of end joints as at a line of frame rivet holes, the latter being taken as the standard because the loss of sectional area is there unavoidable. In strength estimates, of course, allowance must be made for the assistance rendered to the joints by the edge rivets between the joints and the frames. By actual calculation the best arrangement in any given case could be arrived at; such calculations are in practice, however, seldom called for, as a good constructor from his experience is perfectly qualified to devise an altogether satisfactory disposition of joints. Figs. 129 and 130 illustrate different arrangements carried out in cargo vessels recently built.

A point to be noted in arranging shell-strakes is the question of their breadth. As very broad plates lead to a saving in riveting and in the work of erecting—fewer plates being required for the ship than where the strakes are narrow—they are naturally popular with builders. There are objections to their use, however, in that the lines of weakness which occur at

Fig. 130.



the butts are increased thereby, and that the edge laps being fewer than where the strakes are narrow, there is some loss of longitudinal stiffness. For these reasons excessively wide strakes are not adopted by the best shipbuilders. Obviously, plates may be broader in large than in small vessels, as there will still be a sufficient number of strakes in the former case to give a good shift of butts. Lloyd's Rules fix the maximum breadth of strakes at 48 inches in vessels 20 feet in depth, and at 66 inches in vessels 28 feet in depth and above.

The methods of forming the joints of the plates at the edges and ends call for careful attention. In early vessels the edges of the strakes were arranged in clinker fashion (as in fig. 131), but this had several objections, the principal one being the need of tapered slips at the frames; it was, therefore, abandoned in favour of the now universal raised and sunken strake system, shown at fig. 132. An obvious advantage of this style over the preceding one is that only half the number of frame slips are required, which, being parallel, are also less costly and more easily fitted. Other advantages consist in the increased efficiency of construction consequent on having, at least, half the plating directly secured to the framing without packing, and in the possibility

Fig. 131.

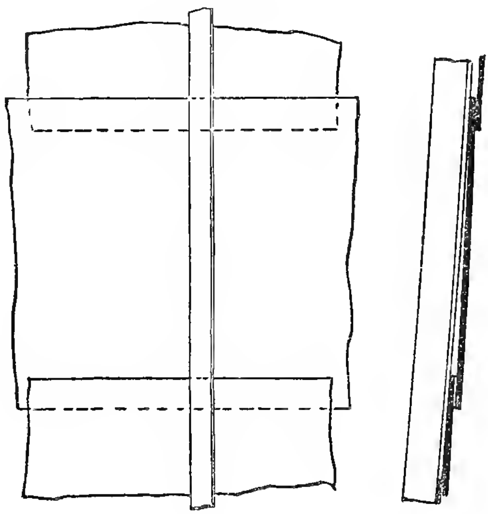
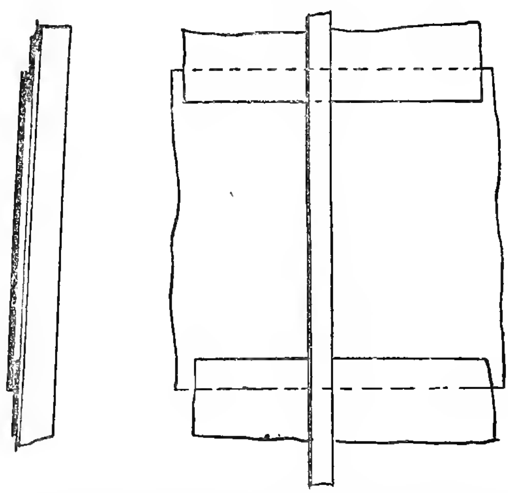


Fig. 132.



of fitting all the inside strakes simultaneously instead of one at a time, the method of plating on the clinker system. In many modern vessels frame slips or packing pieces have been dispensed with altogether, the shell plates being dished or joggled at the edges (as shown in fig. 133), so as to bring their inner surfaces directly on to the flanges of the frames. The advantages claimed are—more efficient riveting, there being two instead of three thicknesses to join, and less weight and cost in the materials of construction. It has also been said that there is a saving in displacement, but there is very little in this, as against the saving in weight of ship at each frame, there is the loss of displacement due to the depression of the plating between the landings.

Fig. 133.

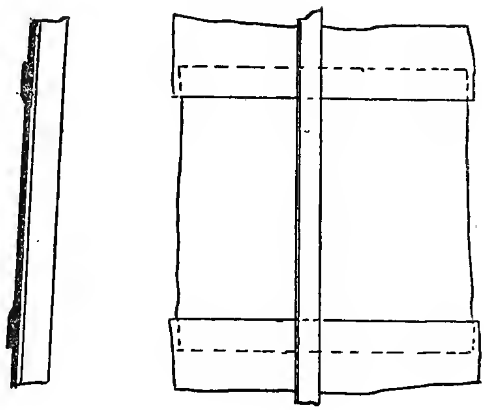
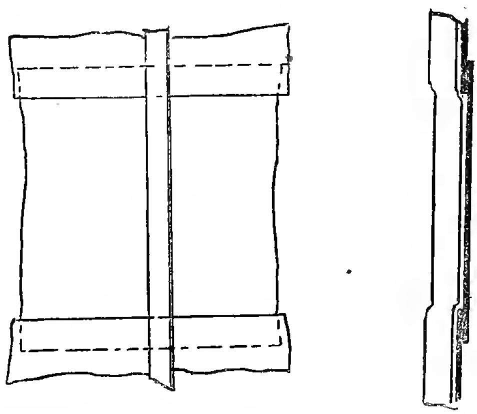


Fig. 134.



This depression, too, it should be noted, causes a reduction in the internal capacity for grain cargoes. Moreover, there is no saving in workmanship, as the joggling of the plates has to be put against the fitting of the packing. On the score of appearance alone, many owners object to the system. Its greatest drawback, however, is probably found in the increased cost and difficulty of carrying out repairs to the shell-plating, when, through the accidents of collision or grounding, these become necessary.

The fitting of packing pieces may also be obviated by joggling the frames (fig. 134). As the plates are not dished, there is a saving in displacement represented by the weight of the packing pieces; also, there is no loss in in-

Fig. 135.

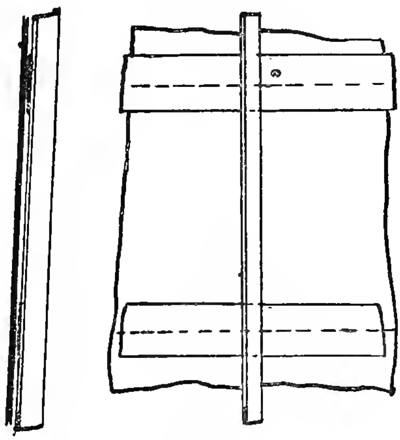
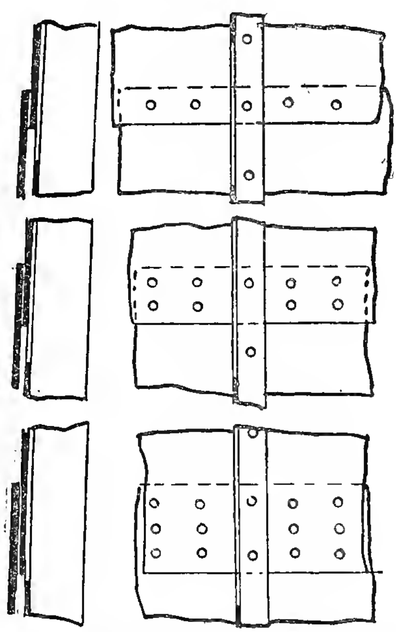


Fig. 136.



ternal capacity. In the case of repairs, if conveniences for joggling be not available, renewed frames may be put in without joggling, packing being used in the ordinary way. For these reasons, this plan has found favour with many owners.

In some yachts and other special vessels, instead of overlapping, the edges are butted, thus necessitating inside strips at the seams (*see* fig. 135). By this method double the number of rivets is required; it also entails a greater weight of material and is considerably more costly than the common method. The flush joint has a decidedly good appearance, but obviously the important considerations of cost and weight are sufficient to debar its use in any but the vessels above referred to.

The number of rows of rivets required in the longitudinal seams is governed by the thickness of the plating, and, therefore, by the size of the vessel. In small craft, in which the shell-plating is less than ·36 of an inch in thickness from the keel-plate to the strake below the sheerstrake, a single row of rivets is sufficient; in larger vessels, in which the plating in the same region is ·46 of an inch, or more, a double row of rivets is required in the seams. The landing edge of the sheerstrake, on account of its importance, should always be at least double riveted.

Until recently, double-riveted seams were considered sufficient even for the largest vessels, but for reasons already given (*see* page 69), in vessels of 480 feet and upwards, built to Lloyd's requirements, or where the thickness of the side-plating is less than ·84 of an inch, it is now necessary to treble rivet the seams in the fore and after bodies for one-fourth the length and one-third the depth in the vicinity of the neutral axis. The seam riveting of vessels of from 450 feet to 480 feet in length, is also to be increased at these parts, proportionately to their length. In very large vessels which have side-plating ·84 inches in thickness or above, the edges must be treble riveted for $\frac{4}{5}$ the length midships. Fig. 136 illustrates single, double and treble riveting at seams.

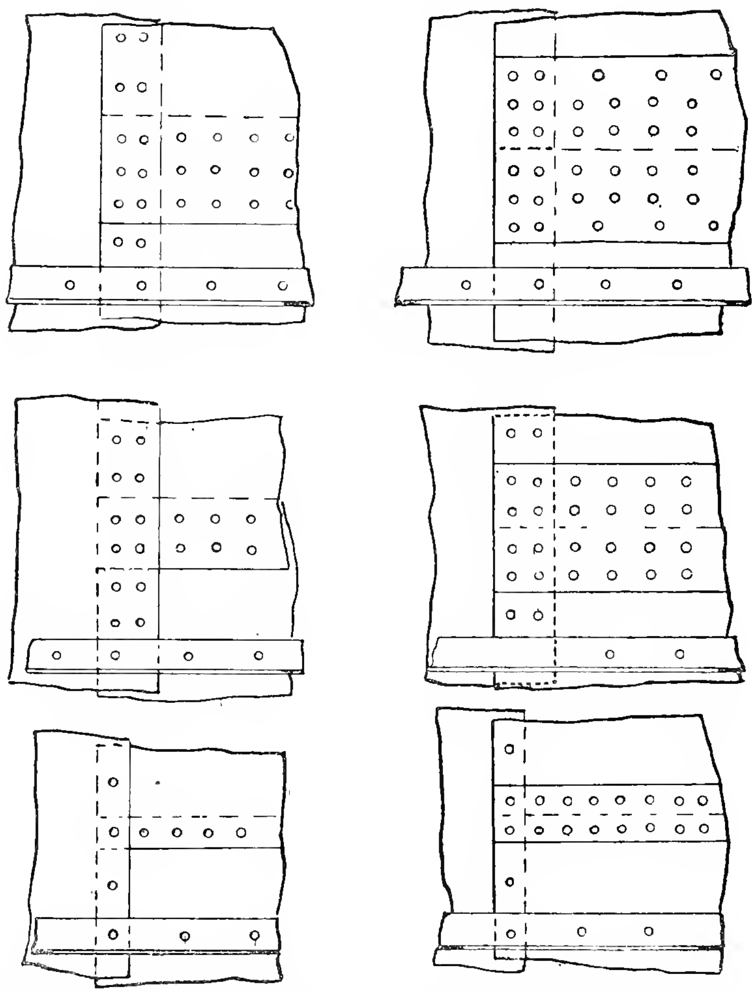
The end joints of shell-plates may be formed either by butting or overlapping; examples of single, double, and treble riveted joints, formed in both these ways, are shown in fig. 137. In making the sketches, overlapped-edge seams on the raised and sunken-plate system have been assumed; with the edge seams formed otherwise, there would be some differences in the details of the end joints.

The question of the number of rivets is decided by the percentage of strength required in the joint compared with the solid plate. In no vessel, however, should the end joints of the shell-plating be less than double riveted. With increase in size of vessels, the need of greater longitudinal strength has made it essential to resort to treble and quadruple riveting at the end joints. In the largest vessels, especially when the proportion of length to depth is excessive, double buttstraps treble riveted are required for the end joints of the sheerstrake and neighbourhood.

In comparing overlapped joints with those having buttstraps, notable points in favour of the former are:—reduction in number of rivets, saving in weight of materials, and reduced cost of construction. It has been objected that the projections due to overlaps cause a drag on a vessel's speed, on account of the dead water which they create; also that the overlapped joint has not the nice appearance of the flush type with the strap inside; but the question of cost has, for cargo vessels at anyrate, quite established the supremacy of the former.

Both lapped joints, and those having single straps, have a tendency to open when under stress, due to the line of the resultant stress not passing through the middle of the joint, thus causing a bending action to be

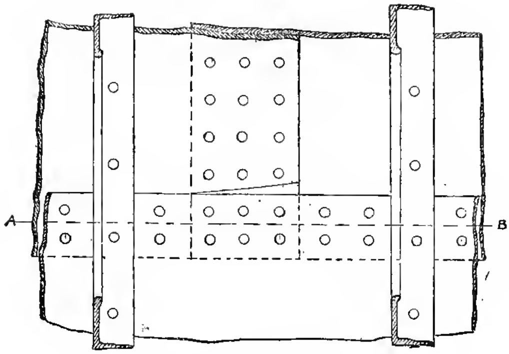
Fig. 137.



developed. Joints having double straps have not this defect, as the resistance to the pull is equal on both sides of the plate.

It may here be mentioned that strained joints situated below the waterline usually leak, this being the unmistakable sign that undue straining has taken place. In such cases, recaulking is resorted to, and, although a cure is

Fig. 138.



SECTION AT A.B.
Cut away where dotted in way of edge lap.

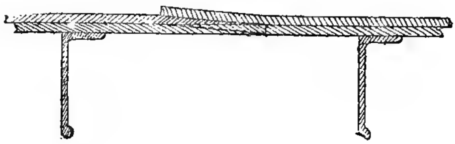
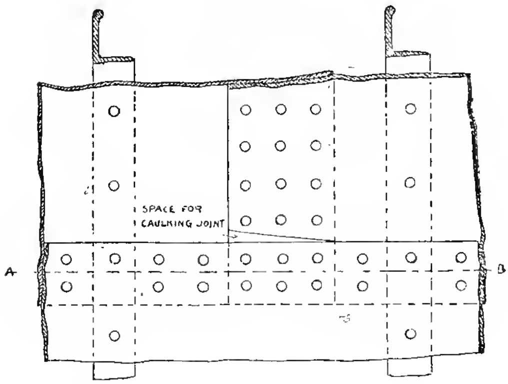
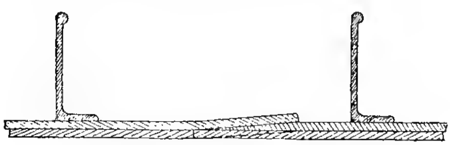


Fig. 139.



SECTION AT A.B.
Dotted part cut away for breadth of edge lap.



generally thus effected with overlapped joints, the same cannot be said for those of the butted type. In the latter case, if the opening at the seam is at all wide, an attempt to recaulk it will but make it more unsightly; moreover, repeated treatment of this sort renders the material brittle and liable to break away. A better way to deal with such a case is to fill the seam with a suitable cement, taking care first to thoroughly clean out the rust; this restores the flush appearance and makes the joint watertight.

With overlapped end joints, some difficulty is experienced in obtaining good work where crossed by the seams of adjoining strakes. Until a few years ago, the usual method of construction was to resort to packing pieces, but this caused unfairness in the landings at each lap joint, and, unless great care were taken in fitting the packing, it could not be satisfactorily caulked. A method now largely adopted is that shown in figs. 138 and 139. In the first figure, which refers to a joint in an outer strake, the end of one plate in way of the joint is seen to be tapered away for the breadth of the edge lap, so as to allow the landing of the outer strake to bear evenly on the inner one without the necessity of packing. In the case of a joint in an inner strake a similar plan is followed (fig. 139). An objection to this scarphing of the seams at the end joints is found in the increased difficulty of executing repairs, where, as may sometimes be the case, the requisite machinery may not be available.

In working shell-plates care must be taken to shear from the faying surfaces—*i.e.*, the surfaces which come together to form a joint—or the rag left by shearing must be chipped off, otherwise it will be difficult to close the work. To facilitate caulking, plates forming outside strakes are usually planed at edges and at one or both ends. In the case of plates forming insides strakes it is necessary to plane one end only, the other end and the edges not requiring to be caulked. When finally shaped and punched shell-plates are secured in place by bolts and nuts which should be sufficient in number to thoroughly close the plates joined, otherwise difficulty will be found in obtaining satisfactory riveting.

RIVETS AND RIVETING.—Nowhere will a lack of efficiency more quickly show itself in the hull of a vessel than at the riveting. The thicknesses of the materials may be well distributed, and the joints carefully shifted from one another, but if the riveting be weak, the straining of the vessel will soon slacken the connections and render her leaky and unseaworthy.

The strength of a riveted joint depends on the aggregate sectional area of the rivets in it, on the spacing of the latter, centre to centre, on the style of the heads and points of the rivets used, and on the material and workmanship.

The number of rivets in a joint varies according as the latter is to be a single, double, treble, or quadruple riveted one—that is, according to the percentage of strength required in the butt as compared with the unpierced plate. We have already indicated generally when and where each class of joint should be employed in a ship, and we now propose to deal with the details of these riveted connections.

The sizes of rivets may be said to be governed by the thickness of the plates they join. If the provision of adequate shearing strength had alone to be considered, wide-spaced rivets of large diameter might be fitted, but for watertight work the distance between the rivets next the caulking edge, especially in thin plates, must not be too great, otherwise the water pressure will cause a tendency to flexibility in the plate edges between consecutive rivets; a comparatively close pitch is also necessary to resist the opening action of the caulking tool. For the same reason, the line of rivets next the caulking edge should not be further from the edge of the strake than about twice the thickness of the plate.

If rivets were of large diameter compared with the plates joined, their shearing strength would greatly exceed the strength of the material between them and the edge of the plate, and the connection would consequently fail, when under stress, by the rivets tearing through the plate edges. To prevent this happening, a maximum diameter of rivet is fixed at about twice the thickness of either of the plates joined. With the rivet at one diameter from the edge of the plate, it can be shown by a simple calculation that the shearing strength of the rivet is approximately equalled by the resistance of the material to tearing, and thus the joint is not more likely to fail in one direction than another. As plates increase in thickness with increase in size of ships, the diameter of rivets become considerably reduced from the maximum given above. Obviously, there is a limit to the size of rivet which can be efficiency worked by hand, and when great strength is required in a riveted joint and machine riveting is not available, this is obtained by increasing the rivets in number rather than in diameter. A lower limit to the size of rivets which may be used in any case is fixed by the punching machine, which cannot punch holes of a diameter much less than the thickness of the plates, as the punch is liable to crush up under the load. In the subjoined table we give Lloyd's Rules for the diameter of rivets in steel ships. It will be seen that rivets $1\frac{1}{8}$ inch diameter are required for plates 1 inch in thickness or thereabouts. These heavy plates are usually restricted to the keel or the sheerstrake, where ordinary machine riveting may be employed, but in the big Cunarders *Lusitania* and *Mauretania*, the riveting of a large part of the shell was done by special hydraulic machines with gaps sufficient to take the full width of a strake, the strakes being fitted and riveted up complete, one at a time, and consecutively.

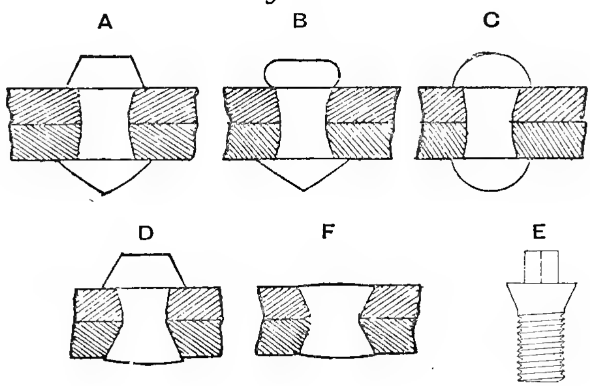
Thickness of Plates in Inches.	·22 and under ·34	·34 and under ·48	·48 and under ·66	·66 and under ·88	·88 and under 1·14	1·14 and under 1·2
Diameter of rivet in inches -	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$

Where joints are to be watertight, to ensure efficient caulking the rivet spacing should not in general exceed 4 to $4\frac{1}{2}$ diameter, centre to centre. The latter spacing, for example, is permitted in the joints of bulkhead plates, in gunwale and margin plate angles, and elsewhere; the former in the fore-and-aft seams of shell-plating, the edge and end joints of deck-plating, in end joints of shell-plating where these are quadruple riveted overlaps, in double buttstraps, butts of margin plates, floor-plates and tie-plates, in the athwartships and fore-and-aft joints of inner bottom plating, and elsewhere. In some positions, the need of providing sufficient strength entails the adoption of a closer spacing than required for watertightness. Thus the rivets in the end joints of shell-plating, where buttstraps are fitted, have a spacing of $3\frac{1}{2}$ diameters in the rows parallel to the joints, and of 3 diameters in the rows at right angles to the joints; with overlapped end joints, except where they are quadruple riveted overlaps, the rivets are $3\frac{1}{2}$ diameters spacing in both directions. Although the plate is thus pretty severely riddled with holes, there is still a sufficiency of strength left in the material between them to prevent the failure of the connection in that direction.

In a few places, where special reasons demand it, the spacing for water-tight work is extended to five diameters centre to centre. The rivets in bar keels, those in the angles of keels of flat plate type, also the rivets connecting water-tight bulkhead frames to the shell plating, are of this spacing. In the case of keels, as the rivets are large and the riveting is usually done by hydraulic machines, which effectually close the surfaces, the spacing is found close enough to obtain good caulking. In the case of bulkhead frames, although to obtain watertightness a closer rivet pitch might be desirable, to resort to it would accentuate the line of weakness through the frame rivet holes. The standard weakest section is that through the ordinary frame rivet holes; the section in way of the closer pitched bulkhead frame rivet holes is made up to this by doubling the shell in way of the outside strakes in the vicinity of the bulkhead, or other compensation is made. When unhampered by the necessity for caulking, we find that the spacing is widened. In the frames, beams, keelson angles, bulkhead stiffeners, etc., the distance between rivets may be seven diameters. In the case of frames, this spacing becomes modified in certain circumstances. When the frames are widely spaced, viz., 26 inches and upwards, to make up the number of rivets their spacing must be reduced to 6 diameters. A similar reduction is required when the framing is deep, viz., 11 inches and above, to develop its full efficiency. When each frame consists of a channel bar and inner reversed bar, the spacing must be reduced to 5 diameters, because the rivets connecting the frame to the shell plating are liable to a relatively high stress, due to the neutral axis of the frame girder being drawn towards the inner edge of the latter by the reversed bar. Experiments carried out by Lloyd's Register have borne this out. In way of deep ballast tanks, and peak tanks, the spacing of frame rivets is to be 5 diameters, and in way of the oil compartments of bulk oil vessels, 6 diameters apart, owing to the circumstance that the weight acts directly on the shell plating, and the strength of the framing is brought into play entirely through the rivet connections.

RIVET HEADS AND POINTS.—There is much variety in the methods of forming the heads and points of rivets; a few of the commoner styles are shown in fig. 140. The rivet marked *A* is known as the pan type, and rivets employed in the shell, decks, tank top, and in handwork generally where strength is the first consideration, are usually made thus:—The pan-head is an efficient form, the shoulder under the head giving it good binding power when riveted up. The rivet-head marked *B* is in favour with some shipbuilders. It is considered that the plug-shape when hammered fits tightly into the hole, and secures watertightness independently of the laying up process; with pan-heads, such hammering frequently brings a strain on the head without affecting the shank of the rivet. The plug-head rivet is sometimes used for decks, tank tops, etc., but owing to its lack of strength it has not found favour for shell work. Clearly it has not the binding power of the pan-head, and having little or no shoulder, under severe stress it is liable to be drawn through the rivet hole. At *C* a snap form of head is

Fig. 140.



indicated. It is occasionally employed in handwork at places in sight, where a nice appearance is desired, such as in casings, bulkheads, etc. The flush head (see *F*) is only used in special cases where a surface clear of projections is required. It is a somewhat expensive form to work, as, of course, the plating must be countersunk to receive it, but it has fair holding power and makes efficient work.

Of rivet points the commonest, and most efficient, is the flush one (*see D*). In general, it is associated with a pan-head, as, for example, in shell work. It is usually finished slightly convex, as shown, in order to maintain the strength. Being flush, the holding power of the rivet has to be obtained by giving the point the shape of an inverted cone. The widening of the hole in the plate for this purpose is called countersinking, which entails the drilling of each hole after punching. The flush point is sometimes adopted where plug, snap, and flush heads are employed. Usually the snap head goes with a snap point (*see C*). It cannot be said that this point, while it looks well, is always reliable. The manner of using the snap tool, with which the point is finished off, is the cause of much of this; frequently the snap is applied before the rivet is thoroughly beaten into the hole, with the result that many such rivets afterwards work loose. The snap style of head and point is used in machine riveting, but the results are then invariably satisfactory. This is, of course, due to the great pressure available, which squeezes the rivet thoroughly into the hole and at the same time closes the joint. The hammered point indicated at *A* and *B* is very efficient, as in order to obtain the conical shape, it is necessary to subject the material to a severe beating-up process, which causes the rivet to thoroughly fill the hole, thus obviating the chief defect of the snap point.

At *E* a tap rivet is illustrated. This type is really a bolt, and is used where the point is inaccessible for holding up. It is frequently employed in connecting shell plates to stern frames at the boss and at the keel.

In fitting tap rivets, the holes are first threaded, the rivet being then inserted and screwed up with a key fitted to the square head on the rivet. When the rivet is sufficiently tight the head is chipped off and the rivet caulked.

In shipwork generally, the holes for rivets are made in the plates and bars by punching. The positions are spaced off or marked from templates, and are punched from the faying surfaces, *i.e.*, the surfaces which come together when fitted in place. One reason for taking this precaution is to ensure that the rag, which is frequently left round the hole on the underside of the plate or bar after punching, shall be clear of the jointed surfaces; another is to take advantage of the shape of the punched hole to increase the efficiency of the riveting. As is well known, a punch in penetrating a plate makes a cone-shaped hole, which has its smallest diameter at the point at which the punch enters, and plates which are to be joined together are so fitted that corresponding holes form two inverted cone frustrums, the finished rivet having thus much greater holding power than if it were merely cylindrical. Rivets are usually manufactured with cone-shaped necks to readily fill up the space under the head (*see* fig. 140).

A disadvantage of punching holes in steel plates is found in the deterioration of the material in the vicinity of the hole which is thereby caused. This deterioration takes the form of brittleness, the steel having thus a liability to break away when through stress the rivet bears upon it. When rivet holes are countersunk this unsatisfactory material is largely removed. The strength may also be restored by annealing the plates after punching, *i.e.*, heating them to a cherry red and then allowing them to cool slowly.

Drilled holes are not largely employed in shipwork because of the greater cost. Drilling, unlike punching, does not impair the quality of the material, but the cone shape which is got by punching could only be obtained in drilled holes by specially countersinking them. Sometimes, when considerations of cost are not allowed to intervene, as at the sheerstrakes and upper deck stringers of some large vessels, the holes are drilled in place by portable electric tools. By this means, perfectly concentric holes are obtained, and a good quality of riveting thus assured.

STRENGTH OF RIVETED CONNECTIONS.—Riveted connections have been frequently experimented upon with a view to obtaining the conditions of maximum efficiency. Iron rivets are found to have maximum shearing strength when in iron plates; in steel plates their shearing strength is less—a $\frac{7}{8}$-inch rivet, for instance, in iron plates has a shearing strength of 13·6 tons, which falls to $11\frac{1}{2}$ tons in steel plates. This appears to be due to the increased shearing effect of the harder plates upon them. Iron rivets are, however, much used in steel ships, as they are more easily worked than steel rivets, and their deficiency in strength is readily made up by increasing them in number. Steel rivets in steel plates give excellent results when carefully worked. Indeed, the quality of workmanship in riveted connections is of first importance.

A point worthy of note is the friction which exists between parts riveted together. This is due to the contraction which takes place in the rivets while cooling, causing the surfaces in contact to press on one another. Careful experiments* have shown that the frictional resistance caused by 1-inch rivets, when the points and heads are countersunk, is 9·04 tons per rivet, and by $\frac{3}{4}$-inch rivets, 4·95 tons; with snap heads and points, the results were—1-inch rivets, 6·4 tons, $\frac{3}{4}$-inch, 4·72 tons. In hydraulic work the frictional strength is greater. It is probable that connections are seldom stressed beyond what can be resisted by the friction between the surfaces; in this case the rivets will not be under stress at all, and there will therefore be no movement in the joints to disturb the caulking and cause leakage.

Of course, frictional resistance has its highest efficiency only when care is taken in fitting the plates and in riveting them. When this is not done the efficiency of a joint may be low indeed.

One fruitful cause of unsatisfactory riveting are blind or partially blind holes. As mentioned above, rivet holes are usually marked from templates, and the plates and bars are punched before being erected into place. Obviously, to obtain an exact correspondence of holes with so many separate processes is a most difficult matter, so that even in work of fair quality a moderate number of holes are found out of line; in careless work, the percentage may be very large. When only slightly unfair, the holes may be corrected by using a steel drift punch. This tool should not, however, be driven into holes which overlap to any great extent, as the tearing of the steel by the punch has a very pernicious effect upon it, much the same, indeed, as that caused when punching the plates in the first instance, *i.e.*, the material becomes brittle and liable to fracture under stress. The best way to cure partially blind holes is to rimer them out to a larger size and use rivets of increased diameter.

DECKS.—Next to the shell-plating, the decks are perhaps the most im-

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* See an interesting paper by Mr. Wildish, in the *Transactions of the Institution of Naval Architects* for 1885.

portant features of a ship's structure. The top deck serves the purpose of making the holds watertight and suitable for the carriage of perishable cargoes. It is also available as a flat to walk upon, from which the crew may perform the various operations required in working the ship. But, besides this, it occupies a commanding position as a feature in the design. We saw, when dealing with strains, that the top and bottom members of a beam are of great value in resisting longitudinal bending tendencies, as they occupy positions most remote from the neutral axis. In a beam or girder, such as the hull of a ship, the top member is formed by the upper deck and the topmost parts of the shell-plating; the importance of this deck as an item in the strength is therefore obvious. 'Tween decks, although they may not be disposed so suitably as the upper one to resist longitudinal bending, are yet splendid stiffeners of the hull, tying the sides together and offering powerful resistance to racking tendencies. These intermediate decks are a necessity in large passenger vessels, the sleeping and other accommodation being provided in the space thus cut off below the upper deck. In cargo vessels they are also necessary for some kinds of freight, although for general trading purposes large holds without obstructions, such as intermediate decks, are now much favoured (*see* Chapter V.)

The minimum number of plated decks required by an ocean-going steamship, structurally speaking, varies according to her size, *i.e.*, taking Lloyd's Rules, according to her longitudinal number.

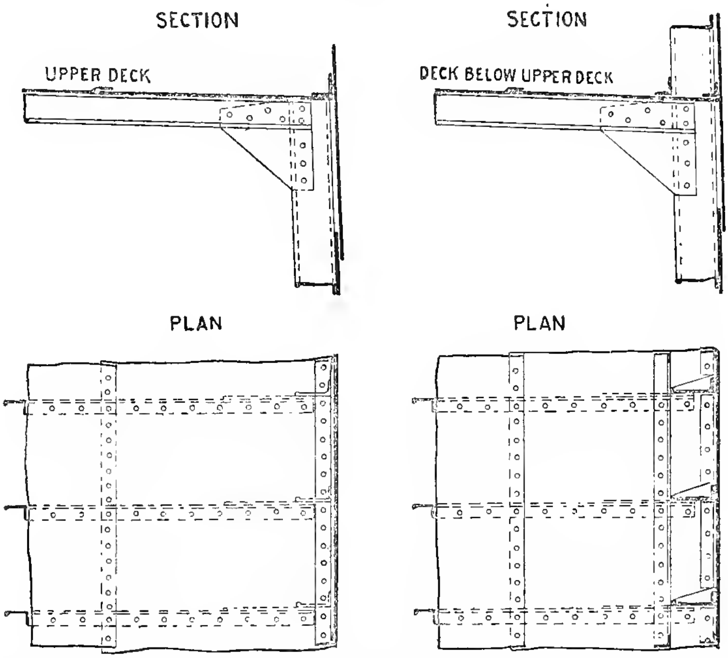
Of course, in small steamers and sailing vessels, no steel deck may be structurally necessary, the strength being sufficient without it. In such a case the necessary watertightness of the holds may be secured by covering the beams with a wood flat or deck, caulking the seams with oakum and paying them with pitch.

A wood deck has some advantages over one of steel. It has, for instance, a finer appearance, and is pleasanter to walk upon, for which reason it is always fitted in passenger vessels, even when a steel deck is required structurally. For vessels trading in hot climates wood weather decks are desirable, as the effect of the sun's rays on unsheathed steel or iron decks is such as to make it almost impossible to move about on them. The decks of ordinary tramp steamers, however, are seldom wood-sheathed on account of the cost, and because a steel or iron deck is found to stand much more knocking about than one of wood, for which reason in many small cargo vessels the upper deck is fitted of steel or iron, although uncalled for by considerations of strength.

DECK DETAILS.—The most important part of the deck is the stringer; indeed, all tiers of beams must have stringer plates riveted to their upper surfaces, whether a complete deck be fitted or not. These plates form a margin strake to the deck, by means of which, through the medium of an angle bar, it is connected to the shell plating. At weather decks this bar is continuous; at intermediate decks the stringers are slotted out to allow the frames to pass through, and the attachment to the shell is obtained by means of short, intercostal lugs between the frames, a continuous angle bar, however, being fitted by way of compensation, along the stringer just inside the frames.

In fig. 141 the usual methods of fitting stringer plates at an upper and at an intermediate deck are illustrated. As will be seen, in conjunction with the shell plating, it forms a powerful T-shaped girder eminently adapted to resist tendencies to deformation of transverse form. The upper-deck stringer plate is specially important as affording considerable resistance to longitudinal bending. The end joints of this strake must be at least double riveted, even in small vessels; in larger ones, treble and quadruple riveting is essential; while in the largest vessels, treble-riveted double straps are required. Both the latter methods of forming a stringer end joint are shown in fig. 142.

Fig. 141.



The first plates to be fitted on a tier of beams are those of the stringer, as they bind the structure together and keep it in proper shape. When a wood deck only is to be fitted, the beams are further held to their work by having two fore-and-aft lines of tieplates fitted, one on each side of the centre abreast of the hatch openings, or in some other convenient position. In sailing vessels severe racking stresses are communicated to the deck through the masts, and to counteract these, a system of diagonal tieplates is fitted in conjunction with those running fore-and-aft (*see* fig. 143). Where the two lines of tieplates cross, to ensure that only a single thickness shall project above the surface of the beams into the wood deck, one is joggled under

Fig. 142.

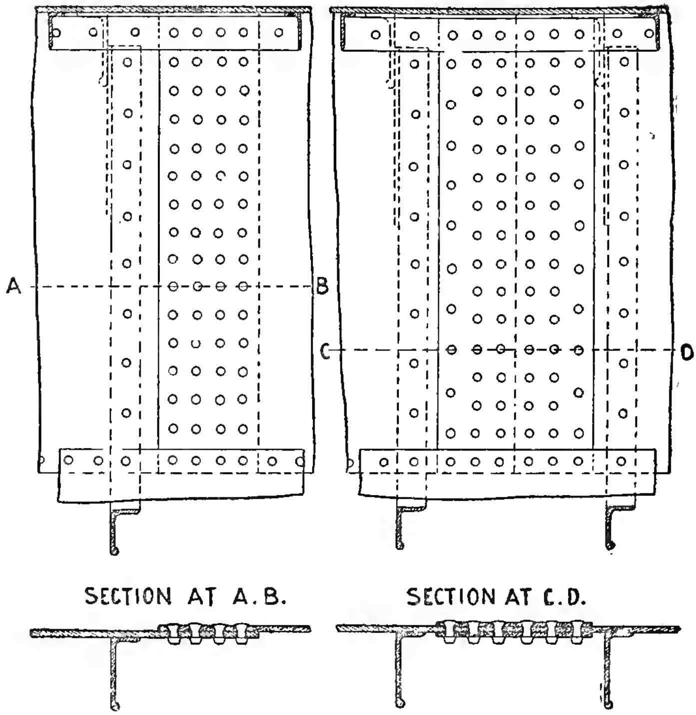
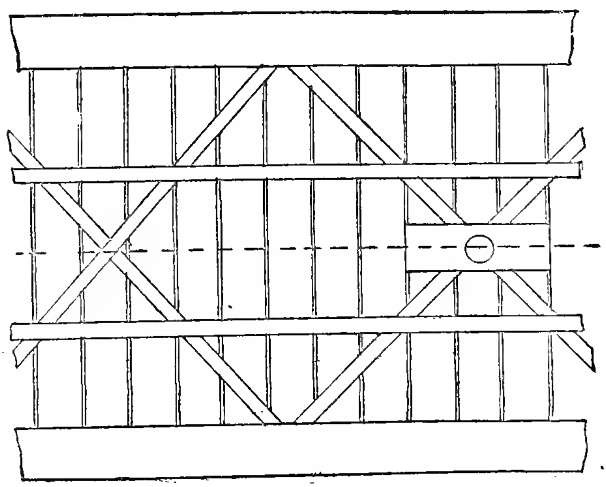


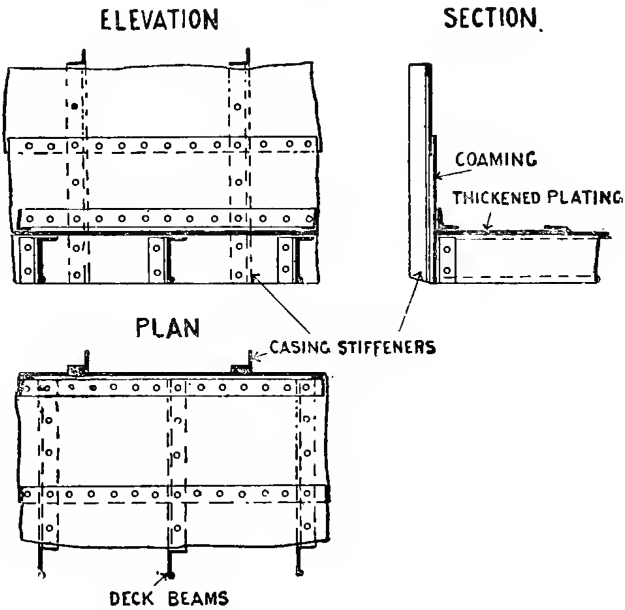
Fig. 143.



the other, or is cut and a double-riveted lap or butt joint made. Tieplates are of the same thickness as the stringer plates of the deck on which they are fitted.

If a steel or iron deck is to be fitted, the tieplates are, of course, dispensed with, and the deck plating, which is usually considerably less in thickness than the stringer plates, is arranged in fore-and-aft strakes of considerable breadth so as to minimise the number of rivet seams. The end joints of deck plates in ordinary merchant vessels are invariably overlapped, and should be double riveted for half length amidships, single riveting being sufficient at the ends. The seams are usually single-riveted overlaps. When decks are unsheathed, the end overlaps should be arranged looking toward midships, as

Fig. 144.



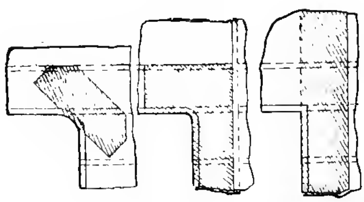
this allows of better drainage. For the same reason the fore-and-aft strakes should be fitted clinker fashion, and the seams so placed as to impede drainage to the scuppers as little as possible. When decks are to be covered with wood, the clinker arrangement makes the fitting of the deck-planking more difficult; in such a case, the raised and sunken system of deck-plating allows of better work. In many recent cargo vessels the edges of the unsheathed steel decks are joggled, which obviates the fitting of slips at the beams, but it has been objected that the depressions thus caused in the surface of the deck form lodgments for drainage water.

At all deck openings compensation has to be made for the cutting away of the material, the extent of this compensation depending on the strength required in the deck, and whether the latter has been fitted for strength purposes, and not merely as a flat to walk on. In some cases increase in thickness of the strake of plating alongside the openings is found sufficient; in others, where the openings are very large, doublings are fitted. The strake of plating alongside the machinery openings on the upper deck of large vessels is an important one. Frequently, it is strengthened sufficiently so as to make, when combined with a strong vertical coaming plate, a rigid girder well adapted to resist longitudinal strains (*see* fig. 144).

The corners of large deck openings are particular points of weakness on account of the sudden discontinuity of the deck-plating, etc., and unless precautions are taken there is a probability of fracture occurring at these points when the vessel is under severe stress.

The usual precaution taken is to fit corner doublings as shown in fig. 145, but as well as this, at the upper deck within the half length, and also within the same range at shelter, awning and bridge decks, if there be such, the stresses being greatest at such places, girders should be fitted under the decks in line with the hatch coamings, to which they should be efficiently

Fig. 145.

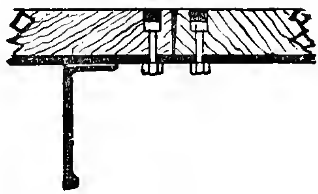


joined, or abreast of them, if not in the same line, in the vicinity of the corners of the openings, so as to bridge over the weak points. Such arrangements have been found to effectively strengthen vessels which had shown signs of straining at the hatch corners; they are now required by Lloyd's Rules.

Gutterways are usually fitted round the margin of weather decks where these are to be laid with wood. They are formed simply by running an angle bar fore-and-aft at a fixed distance from the ship's side, and riveting it to the stringer plate. Steel gutterways are frequently coated with cement as a preventative against undue corrosion.

WOOD DECKS.—In laying a wood deck, whether it be on top of one of steel or iron, or merely on a tier of beams, it is important that the planking should be fitted close down on the metal work. In way of the tieplates and stringers in a non-plated deck, and of the edge seams and end laps where a deck is plated throughout, the underside of the planking should be scored out so as to obtain a solid bearing and an even upper surface. With a plated deck on the raised and sunken system, the planking fitted over the sunken strakes is thicker than that over the raised strakes by the thickness of the plating. Sometimes the difference in thickness is made up by slips of wood, but this is most objectionable, as spaces are thus left between the wood and steel decks which a slight defect in the caulking of the wood deck causes to become receptacles for the lodgment of water; when this happens pitting and general decay of the steel deck quickly follow. Before laying planks the steel work should be coated with a suitable preservative composition, such as Stockholm tar powdered with Portland cement, and each plank should be separately coated with tar before being bedded down. This prevents the likelihood of lodgment spaces for water existing between the metal and wood to cause decay. The best wood for weather decks is undoubtedly teak, as it is of an oily nature and is well suited to stand changes of temperature, but it is somewhat expensive. Pitch pine is frequently used for weather decks of cargo vessels, where these are sheathed. It is less costly than teak, but more of it is required, as a pine deck must be thicker than one of teak. Pitch pine does not wear so uniformly, but it is of a hard grain and fairly durable. Yellow pine makes a handsome deck, and is therefore much used in passenger steamers. It is very soft and

Fig. 146.



therefore requires frequent renewals. On this account and because of its extra cost, it is seldom used in cargo vessels.

Care should be taken when laying wood decks to have the hard side of planks uppermost; this reduces the likelihood of the deck wearing into holes in places. Three intermediate planks should separate butts in the same frame space. The plank butts should be of vertical type (*see* fig. 146) and arranged to come between beams where a steel deck is fitted, and on the beams where there is no steel deck. They should be fastened with bolts at each beam, or between the beams, where there is a steel deck; and to ensure that the planking shall lie perfectly flat, when it exceeds 6 inches in breadth it should have double fastenings. Between 6 inches and 8 inches broad, a bolt and nut and a screw-bolt is considered sufficient; when the planks are over 8 inches broad, two bolts and nuts are required. Deck bolts should be galvanised, and should have their heads well bedded in white lead, with grommets of oakum. When screwed up from below, the heads should be sufficiently sunk in the deck to allow of a dowel being fitted over the top.

Pine decks should not be laid until the wood is thoroughly seasoned. If this precaution be not taken, the deck is likely to open at the seams and become leaky as well as unsightly. Lloyd's Rules require a period of four to six months to elapse, according to thickness, from the time of cutting to the time of using. Pitch pine planks for weather decks should be seasoned for six months. The above periods of seasoning are not required where a satisfactory artificial method of seasoning is adopted.

It should be mentioned that wood decks are now being superseded in passenger and crew spaces by compositions like litosilo, corticene, etc. Decks thus covered are comfortable to walk on, have a good appearance, and, when carefully laid, have generally been found to wear well.

CARGO HATCHWAYS.—There must be at least one deck opening into each main compartment of a vessel to allow of cargo being shipped into, and discharged from it. In the latest cargo steamers these hatchways are of considerable size, so as to be suitable for special cargoes of large measurement, such as pieces of machinery. Lengths of 24 to 28 feet, and breadths of 16 feet, are common, while these dimensions have frequently been exceeded.

We have already indicated some of the means adopted to prevent these large gaps in the deck from becoming dangerous points of weakness, and it now remains to show how the hatch openings are framed.

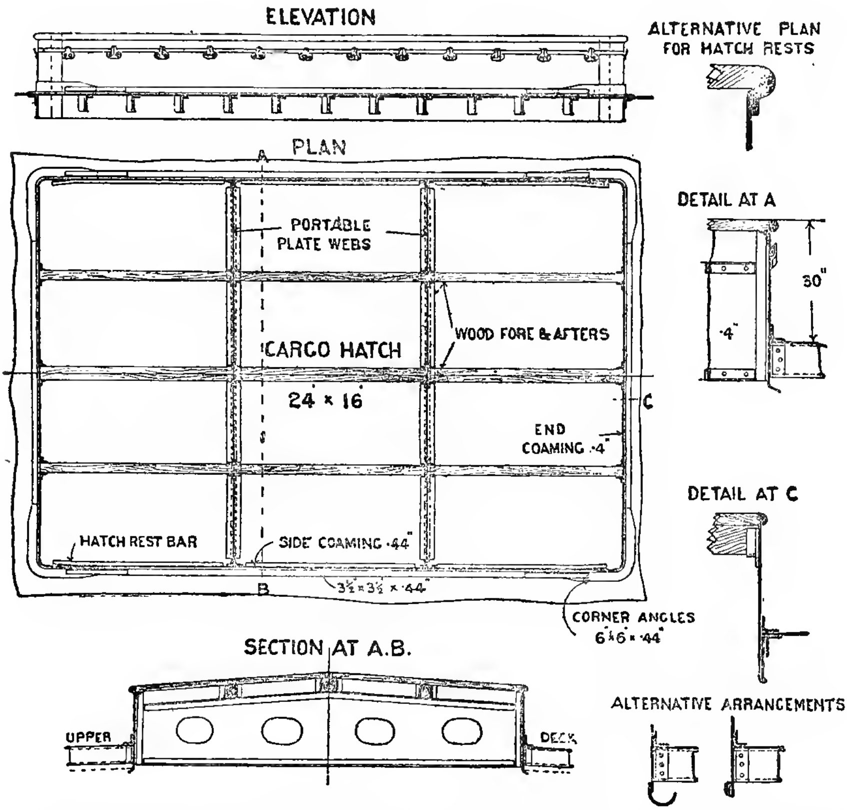
The main portion of this framing consists of vertical coaming plates fitted fore-and-aft and athwartships and carried down to the lower edge of the deck beams, those in a fore-and-aft direction forming an abutment for the beams that have been cut, and those fitted athwartships stiffeners to the continuous hatch-end beams, to which they are securely riveted. The connection between the hatch coamings and the severed beams is effected by means of angle lugs, fitted single where beams are at every frame, and double where they have a two frame spacing (*see* fig. 147). Lloyd's Rules require that there shall be three rivets in each flange of these lugs when attached to beams $7\frac{1}{2}$ to $9\frac{1}{2}$ inches deep, the number being increased to four where the depth of beam is 10 to 12 inches.

The deck-plating is fitted so as to abut against the coamings, a riveted attachment being secured by means of a strong angle bar. In non-plated decks broad tieplates are fitted on the beam ends and against the coamings, and in this way a strong T-shaped girder is obtained round the edge of the opening. It should be mentioned that when decks are laid with wood, the vertical flange of the hatch-coaming bar is fitted of sufficient depth to project $\frac{1}{2}$ inch above the wood, so as to facilitate the caulking of the latter.

Weather deck hatch coamings (*see* fig. 147) should be of considerable height above the deck so as to protect the comparatively weak covers which seal the openings from receiving the full force of the heavy seas which in rough weather frequently fall upon the deck. On upper decks, coamings should have a minimum height of 2 feet except under awning or shelter decks. In certain classes of vessels, which have deep wells between the front of the bridge deck and the aft end of the forecastle on the upper deck, the coaming height should not be less than 2 feet 6 inches, as these spaces are specially liable to flooding. Obviously, on bridge, awning and shelter decks, which are situated high above the waterline, hatch coamings may be of reduced height; they need not, in fact, exceed 18 inches.

Weather deck coaming plates, in order to be efficient as girders and as protecting walls to the hatchways against inroads from the sea, should be of a substantial character. For instance, the coamings of hatches under 12 feet in length should be ·36 of an inch thick, while those having lengths of 16 to 24 feet should have side coamings ·44 of an inch thick; end coamings

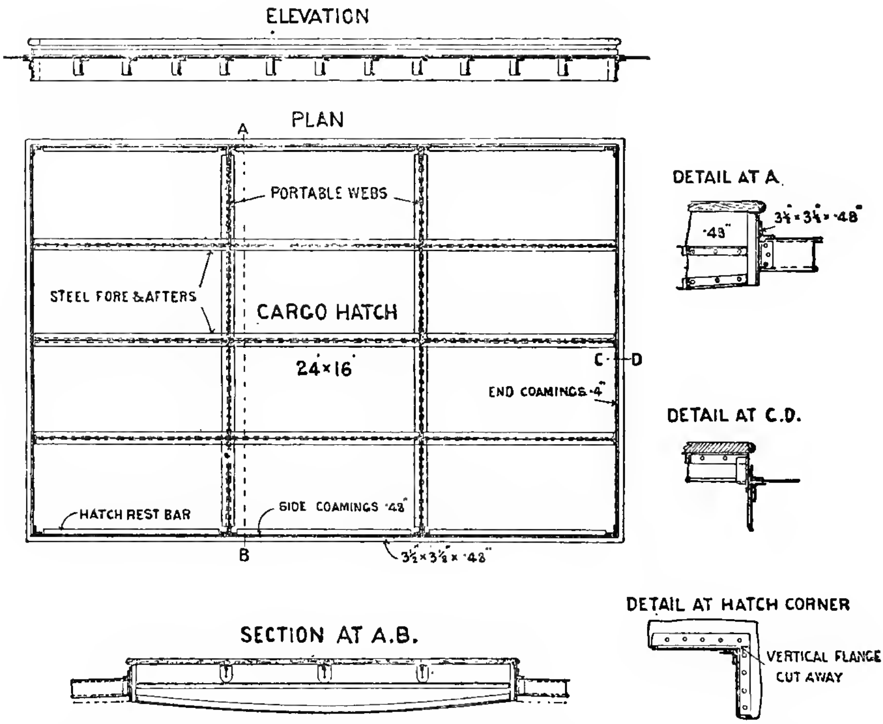
Fig. 147.



in the larger hatchways are allowed to be ·04 of an inch less in thickness than the side coamings, owing to their shorter length, and to the fact that they have no beam ends to support like the side coamings.

Below the weather deck hatch coamings need not be so deep, as, of course, the openings are not exposed to the sea, and high coamings would impede the efficient stowage of cargo in the 'tween decks (*see* fig. 148). In 'tween deck hatches 10 feet and under 14 feet long, the total depth of side coamings from underside of hatch end beams may be 16 inches, and in those from 18 feet to 24 feet, a depth of 20 inches is considered sufficient. The loss of strength due to reducing the depth of the hatch coamings at lower decks is made up to some extent by increasing the thicknesses by ·04 of an inch, as compared with upper deck hatches of the same length. Round corners are usually preferred for hatches on weather decks. This style makes the fitting of the wood covers at the corners somewhat difficult; it is also less convenient for fastening the tarpaulins, but otherwise it has obvious advantages. To begin with, the tendency for the deck to strain at the hatch corners is less where these are round than where square. Round

Fig. 148.

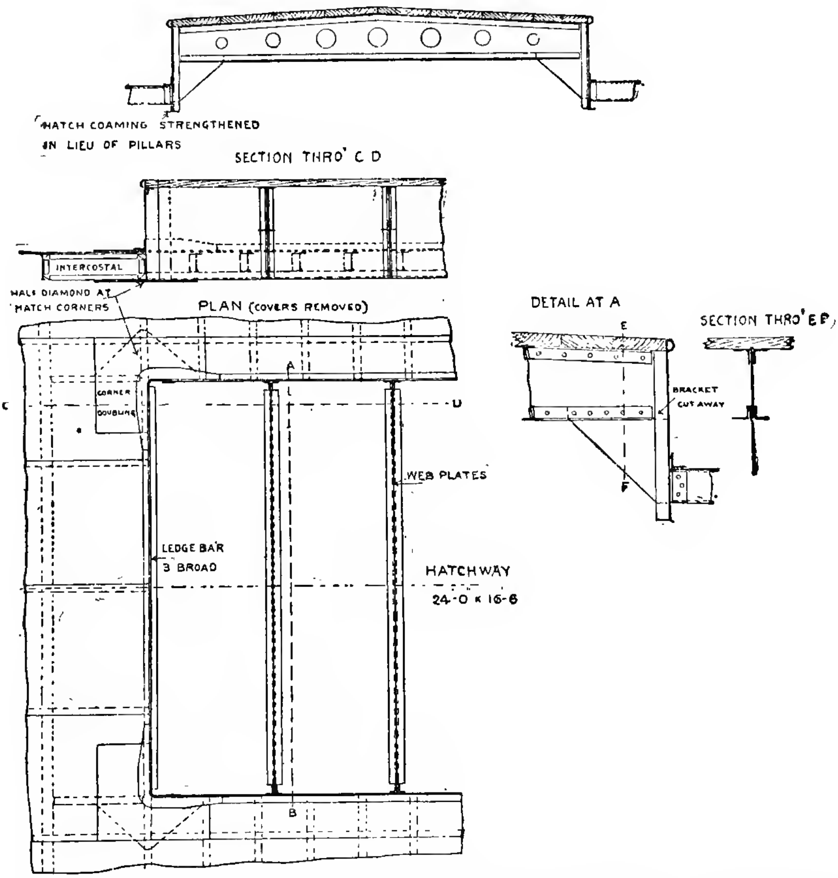


corners are less likely to damage cargo which may collide with them; they have also a nice appearance. The same advantages of having round corners, obviously, do not extend to 'tween deck hatches and, consequently, they are usually of square type (fig. 148).

In order to strengthen hatch coamings against inroads from the sea and to provide adequate support to the wood covers, portable athwartship beams are fitted. In hatches 10 and under 16 feet long, one such beam, formed of a plate with double angles at top and bottom, or other equivalent section, is required; in those of 16 to 20 feet in length, the portable beam becomes a web-plate extending to the bottom of the coamings, fitted with double angles top and bottom. Two web-plate girders of this description are required in hatches from 20 to 24 feet in length. These portable beams are frequently bolted between double angles riveted to the coamings, when they act as ties as well as struts, and to some extent compensate for the gaps in the deck made by the hatch openings; occasionally, they are arranged to ship into special shoes.

Fig. 149.

SECTION AT A B



Web-plate beams in hatchways below the upper deck should be equal in thickness to the coamings to which they are attached, and should extend to the lower edge of the coamings. Where the latter are shallow, as in the case of 'tween deck hatches, the web-plates are to be a quarter deeper in the middle than at the ends, and stiffened top and bottom by double angles.

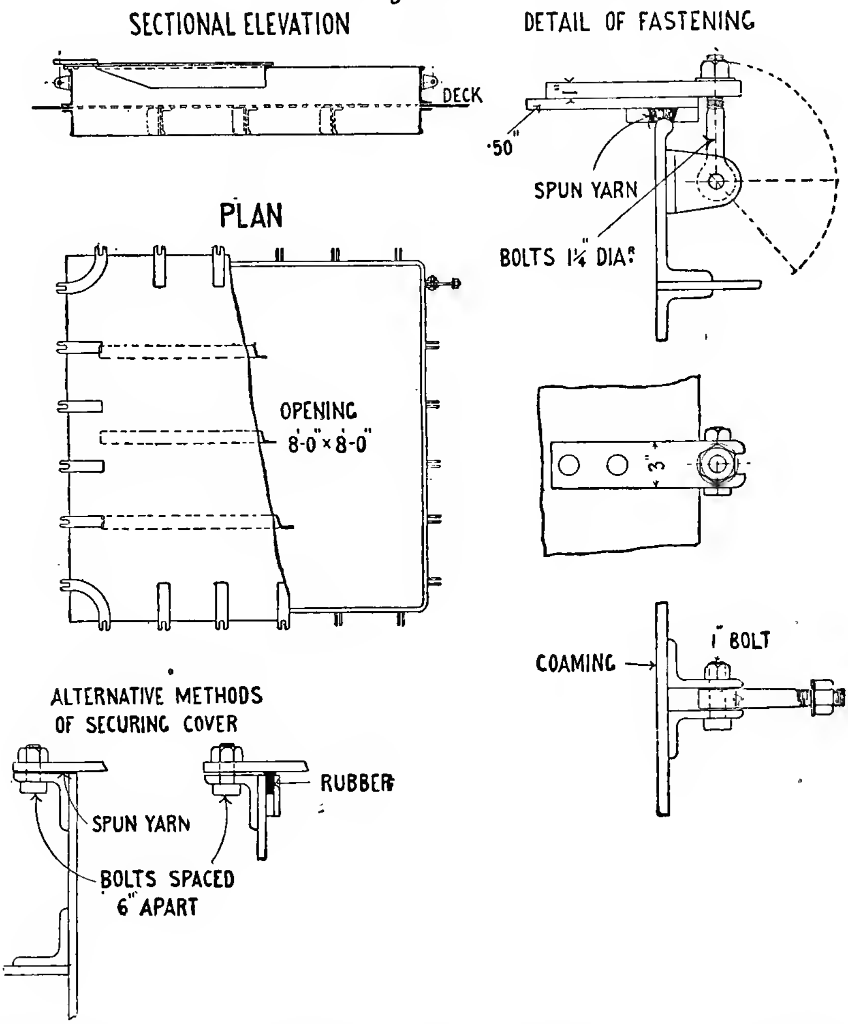
The top angles of the portable webs, as already hinted, form lodgments for the wood covers, but, as previously mentioned, heavy seas frequently fall upon the deck, and the covers have to sustain a substantial share of the weight; they therefore need additional support. This is provided by fitting strong steel or wood fore-and-aft bearers. In small hatches from 6 to 10 feet broad, a single bearer fitted at the centre is sufficient; in larger hatches three or more in the breadth are required. The fore-and-afters should fit into iron or steel shoes securely riveted to the end coamings and to the intermediate webs, if any, and the shoes should afford a bearing surface not less than 2 inches broad. To support the wood covers at the hatch sides, ledge or rest bars are fitted, giving a bearing surface at least $1\frac{3}{4}$ inches broad. This ledge iron is riveted to the hatch coamings between the webs, except where the side moulding and ledge rest consist of a single special section, when it is, of course, continuous for the length of the hatch (*see* fig. 147). As well as at the top of coamings, mouldings are frequently fitted at the bottom on the inside to take the chafe of cargo. The latter requirement is sometimes met by flanging the lower edge of the side coamings instead of fitting mouldings. At weather deck hatches, to ensure watertightness, strong tarpaulins are fitted over the wood covers, usually two or three to a hatch, one placed above another. The tarpaulins are secured in position by means of flat iron bars wedged into cleats riveted to the hatch coamings, round which they are spaced about 24 inches apart.

In many recently built cargo vessels hatch beams have been fitted all in one direction, *i.e.*, either all athwartships or all fore-and-aft, the direction being that of the shorter dimension of the opening, which, in ordinary cases, is athwartships. Lloyd's rules now provide for arrangements of wholly transverse webs for hatches ranging in breadth from 12 to 20 feet. The supports at the coamings for the wood covers in this case should have a bearing surface 3 inches broad. Fig. 149 shows a hatch framed in this way.

HATCHWAYS INTO DEEP TANKS.—These should be strongly framed and have means of closing in a watertight manner as they must withstand the testing pressure on the tank, viz., an 8-feet head of water above the crown of the tank, without straining or showing a leak. To simplify construction, watertight hatchways are made no larger than absolutely necessary. Usually they are about 6 feet to 8 feet square and, as already pointed out, owing to the presence of the middle-line bulkhead with which deep tanks are provided, are fitted two abreast. The coamings of these hatchways frequently consist of deep bulb angles (fig. 150), but sometimes they are built of plates and angles. The cover is a plate of substantial thickness, with angle or bulb angle stiffeners at about [illegible] feet spacing. It is secured in position by nuts and fall-back bolts, or nuts and through bolts, and watertightness at the joint is effected by packing it with spun yarn or rubber. To gain admission to the tank without removing the hatch cover, a watertight manhole door is usually fitted in the latter.

CARGO PORTS AND DOORS.—Many vessels are fitted with small side ports to give access to the 'tween decks. These are found useful in loading certain classes of bale goods, and allow the 'tween decks to be stowed while the holds are being filled through the main hatchways. When side ports do not exceed, say, 3 feet square, sufficient compensation for cutting the opening in the shell-plating is provided, by doubling the strake above it for a short distance and fitting a stout angle around the edge of the

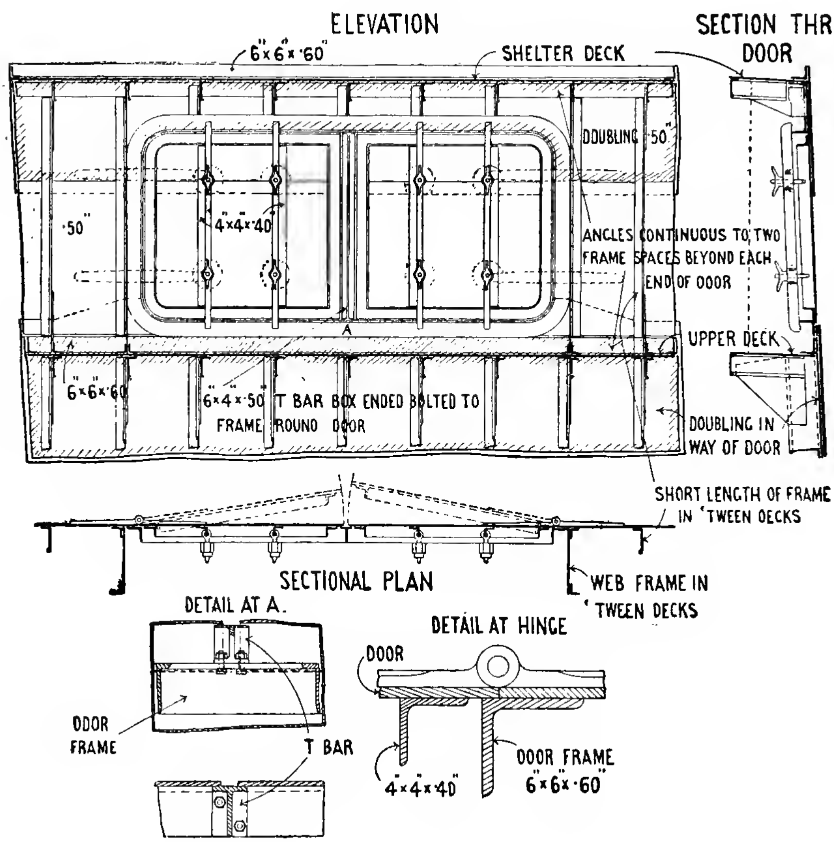
Fig. 150.



opening. The door is sometimes secured by bolts and nuts at sufficiently close spacing to ensure a watertight joint with canvas and red lead between the surfaces; usually, however, strong backs are fitted inside, one or two being used according to the size of the door; and to obtain watertightness at the joint spun yarn or rubber packing is used.

In certain vessels—in those engaged in the cattle trade, for instance—very large doors are fitted in the ship's side in way of the bridge or shelter 'tween decks. These doors make big gaps in the side-plating and have to be carefully compensated for. Usually the shell-plating is doubled above and below the opening and for some distance, say two frame spaces, beyond each end of it; web frames are also fitted in the 'tween decks at each end of the doorway. The shell doublings make good the longitudinal

Fig. 151.

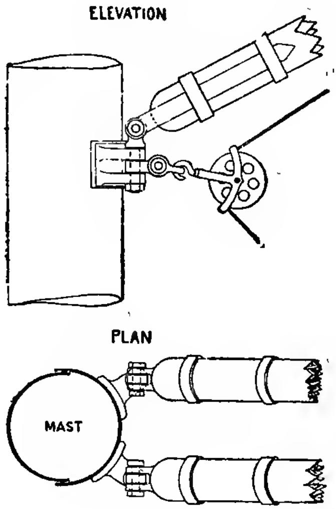


strength, and the web frames restore the loss entailed in the cutting of the side frames in way of the opening. Fig. 151 gives details of a cattle door 12 feet long and 5 feet 6 inches deep, as fitted in a large modern cargo and passenger steamer.

DERRICKS AND DERRICK POSTS.—Large and numerous hatchways are of little value unless an efficient installation of appliances for working the cargo in and out of them be also provided. This is specially the case with steamers whose economical working demands the utmost despatch in the loading and unloading of cargo. Sailing ships usually make long voyages and are seldom in port; they can, therefore, afford to spend a longer time there than the more ubiquitous steamer. Moreover, their working expenses are much less than those of the latter. For these reasons an expensive system of cargo gear is seldom fitted in sailing vessels. Hand-power winches are considered sufficient, and the cargo gear is usually suspended from the lower yards or from convenient wire spans.

The cargo gear of modern steamers may consist of (1) ordinary derricks with steam winches, (2) hydraulics derricks, (3) steam cranes, (4) electric cranes. Electrical appliances, although frequently proposed, have not yet come much into use. Steam cranes are frequently fitted in coasting vessels, as they hoist

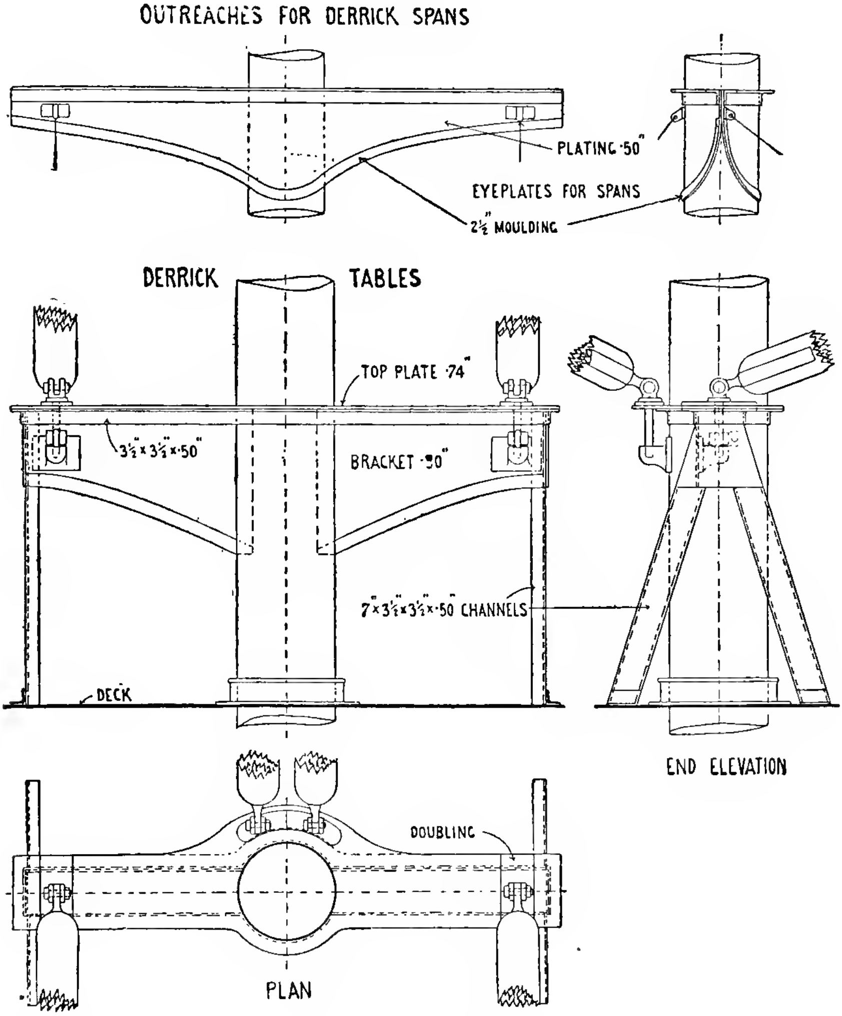
Fig. 152.



and slew quickly, and thus minimise the time a vessel need remain in port —an important consideration where a vessel has to be loaded and discharged every day or two, or even more frequently. Moreover, steam cranes may be placed anywhere about the deck. They take up a great deal of room, however, and are more expensive than steam winches and derricks, for which reasons they are seldom fitted in ordinary ocean-going cargo vessels. Hydraulic derricks are sometimes fitted on first-class passenger steamers, as they work smoothly and without noise. They are costly to install, as, of course, a powerful pumping engine is required in order to maintain an artificial head of water.

The system of working cargo almost universally adopted in ordinary cargo vessels is that comprising steam winches and ordinary derricks. The latter may be constructed of wood or steel; if for small lifts, say, from three to five tons, pitch-pine derricks are commonly fitted. They are hinged, if practicable, on the masts, which, in cargo steamers, are now little else than derrick posts. One derrick and winch per hatch is sufficient where the holds are of moderate size; where they are large, however, and where loading or discharging can be carried out on both sides of the vessel at once, two derricks and

Fig. 153.



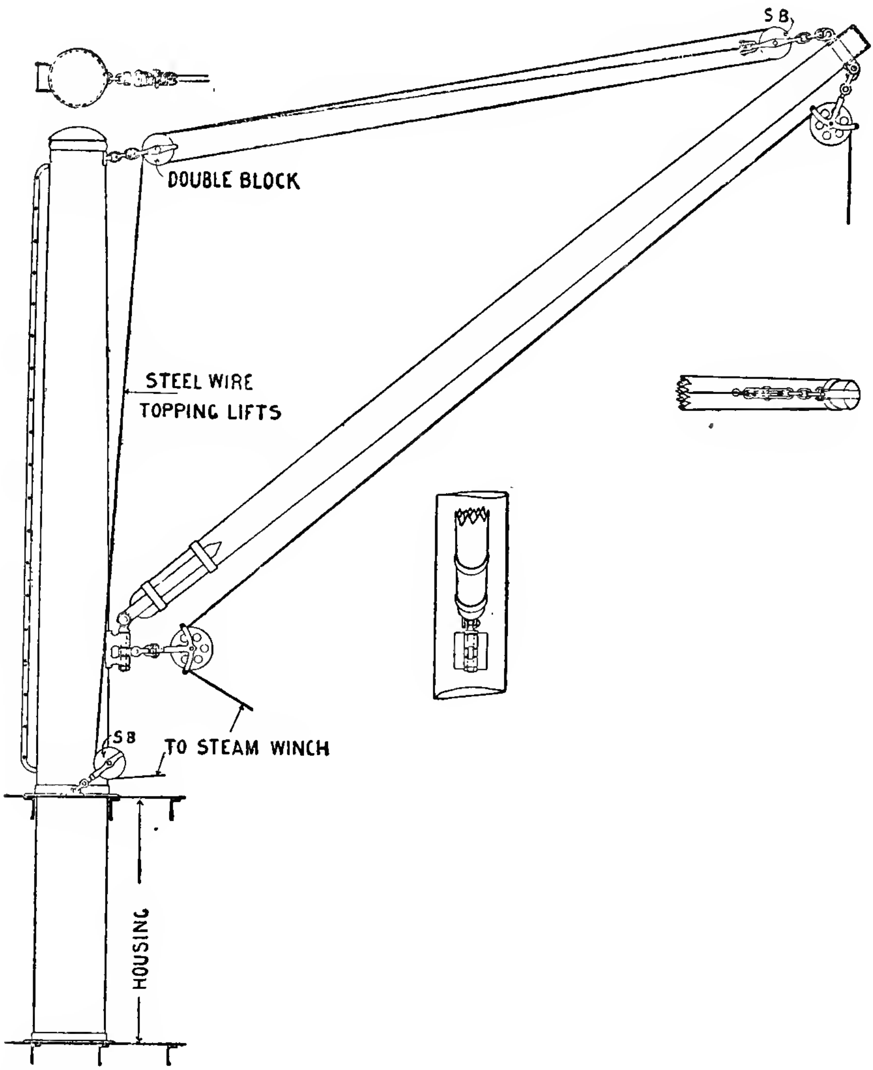
winches are necessary. We thus see that in the usual arrangement, with a mast between two hatches, the former may have to support four large derricks with their respective loads. In special cases, where separate derricks are employed for hoisting and for slewing, the number of derricks per mast may exceed four. In fig. 152 is shown the usual way of hinging derricks on masts.

Steam winches must be placed with careful regard to the derricks. Single winches are, of course, situated at the middle line with the middle of the winding drums in line with the derricks. Double winches should be placed on each side of the centre of the ship, with a sufficient distance between them to allow a man to pass. To obtain compactness, the inner drums are frequently dispensed with, and to ensure direct leads from the derricks to the winding barrels, the axis of the winches are inclined to the middle line. Frequently, it is preferred to have the winches square to the middle line, as the winchmen are then better able to observe operations; in these cases, direct leads to the barrels are obtained by means of snatch blocks on the decks, or, better, by extending the derricks out transversely on tables (fig. 153), so as to come in line with the middle of the winches. With this arrangement it is desirable to have the point of suspension in each case immediately over the heel of the derrick, otherwise difficulty will be experienced in slewing the latter. This drawback is found, for instance, where a single derrick is worked from a mast having considerable rake, and no arrangement is made to bring the point of suspension over the heel of the derrick. If the derrick is fitted forward there is a strong tendency for it to lie overboard, and if aft on the mast, to lie over the middle of the hatch; considerable power being required to slew the derrick, particularly if loaded, against either of these biassed directions. The advantage of plumb derricks is therefore obvious, and some vessels are built with vertical masts to this end.

When a mast is situated too near a hatch to allow of a derrick hinged on it being sufficiently long to plumb the centre, and swing clear of the ship's side, it becomes necessary to resort to the use of derrick posts. These may be placed between the hatch and the ship's side, and with a comparatively short derrick a sufficient outreach may be easily obtained. Derrick posts were at first objected to as being unsightly, but their great utility has outweighed considerations of this sort, and they are now to be found in many up-to-date cargo steamers. Where the lifts are at all great,. derrick posts should be made of considerable height, otherwise excessive stresses will be brought upon them, as well as on the derricks and the spans. This can be readily demonstrated by drawing a diagram of forces. Derrick posts to carry a 10-inch to 12-inch derrick, and lift ordinary cargo, should be 20 inches to 24 inches diameter at the deck, and have a height of 24 to 28 feet. Large derrick posts are constructed of $\frac{1}{2}$-inch steel plates, a little taper being allowed in the thickness towards the top. To give rigidity, a housing equal to the height of one 'tween decks is desirable; where this cannot be obtained, deep brackets must be fitted to the deck. Fig. 154 shows a derrick-post and appurtenances as ordinarily fitted.

SEATS FOR STEAM WINCHES, Etc.—As before mentioned, decks should be stiffened locally in way of steam winches, by fitting plating on the beams, and by supporting the latter by special pillars. If winches are to be placed on a wood deck, the part under each winch should be of hardwood, or the wood deck increased in thickness locally, as the wear and tear is great at these places. Sometimes a steel angle bar is fitted round the winches and riveted to the deck-plating, the wood deck being butted against this bar, and the spaces enclosed coated with cement or left bare. In such a case, the

Fig. 154.



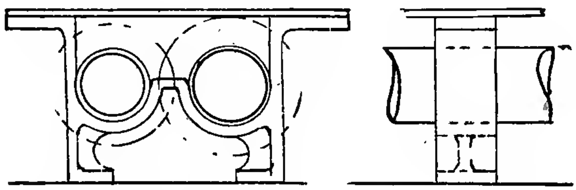
winch sole plates are either bolted directly to the steel or iron deck, or raised on Z or channel bar stools. The latter plan is the better one, as the stools stiffen the deck and take most of the vibration caused by the working of the winches. In ordinary cargo steamers, which have, as a rule, unsheathed decks, winch stools like the above are commonly fitted.

The steam supply and exhaust pipes to the winches (where the exhaust steam is returned to a tank in the machinery space) are usually led from the machinery-casing along the deck just outside the line of the hatchways, and are supported on stools of cast or wrought iron (fig. 155), except where they can be conveniently held by clips riveted to the casings or to the hatch coamings. Sometimes separate exhaust pipes are led from each winch across the deck to the ship's side. This latter plan has only cheapness to commend it, as the cloud of escaping steam always present about the deck during loading or discharging operations, is most objectionable.

To protect the winch pipes from damage, solid plate or sparred iron covers are fitted over them.

MASTS.—In a sailing-ship the masts are probably as important as the hull itself, since her power of locomotion depends on them; in a steamer they have a much reduced value, and are even not indispensable,

Fig. 155.



some classes of steamers having none at all. As seems fitting, therefore, we shall first consider the masts of a sailing-ship, and afterwards indicate the modifications usual in the case of a steamer.

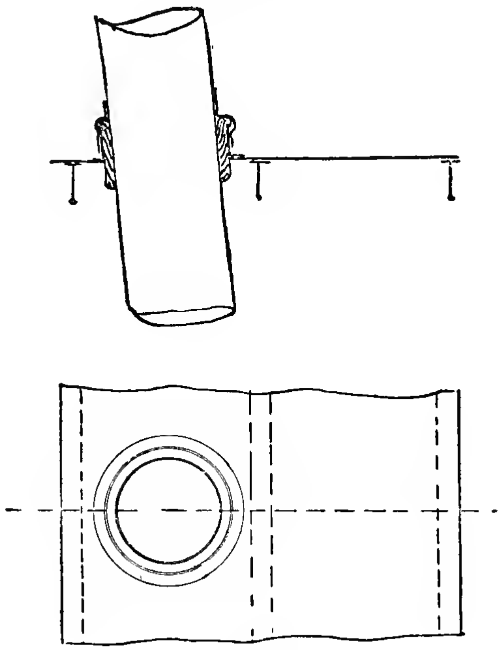
In a modern sailing-ship of average size, the masts, like the hull, are constructed mainly of steel, their diameters and scantlings being graduated in accordance with the strains they may be called upon to bear through the action of the wind-pressures on the sails. As the mast bending moments vary with the lengths of the masts, length is the natural basis on which to fix scantlings, and, in compiling tables of the latter, this method is usually followed. Taking Lloyd's Rules, a lower mast 60 feet long has a maximum diameter of 20 inches, with plating $\frac{7}{20}$ inch thick, and one 96 feet long a diameter of 32 inches, and a plating thickness of $\frac{10}{20}$ inch, the scantlings of masts of intermediate lengths lying between these.

Mizen-masts of barques carry no cross-yards, and support a less sail area than mainmasts; reduced diameters and scantlings are therefore allowed in their case.

The maximum diameter of a mast and the greatest thickness of plating are at the deck, as the bending moment is obviously greatest there; towards either end, the diameter and the thickness of the plating are somewhat reduced. The number of plates in the circumference of a mast is governed by practical considerations. Lower masts, with a rule length of 72 feet and under, are built with two plates in the round; those above this length should not have less than three plates. These plates are usually overlapped at edge and end joints; sometimes the latter are butted, in which case the straps should be fitted outside, as opening at the joints due to bending of the masts is thus prevented, and better work can be made in the fitting of internal angles where these are necessary.

Fig. 156.

ELEVATION.



PLAN.

As the principal stresses on masts are cross-breaking ones, the end joints are very important, and in all cases should be treble-riveted above the deck; in way of the housing—by which is meant the part of the masts below the deck—double-riveting is permitted. The seams should be double-riveted; but in masts under 84 feet long, single-edge riveting is considered sufficient, provided the loss of stiffening effect due to reducing the lap is made good by fitting internal angle bars. Above 84 feet length, double-riveted seams are required as well as internal stiffening angles, as the bending moments on masts of this length may be very great.

Rigidity is given to masts by securely fixing them into the hull, and staying them by means of steel wire ropes. Fig. 156 shows the modern method of framing a mast-hole and wedging the mast at the upper deck—the deck at which this is usually done. As will be observed, a stout plate is fitted on the beams, which, in non-plated decks, must have a breadth equal to three diameters of the mast. This plate is riveted to the beams and (in non-plated decks) to diagonal tieplates (fig. 143), which unite it to the side stringers and distribute the stresses communicated from the mast through the wedging. A bulb angle ring about 4 inches greater in diameter than the mast is riveted to the deck-plate, and when the mast is shipped the space between this ring and the mast, the plating of which should be doubled in this neighbourhood, is tightly wedged with hard wood. Above the deck the wedging is neatly rounded, and a canvas cover or coat, usually double, is bound to the mast and over the ring to prevent leakage of water into the hold space. The doubling of the mast at the deck is to give strength, but particularly to compensate for corrosion and pitting which may take place in way of the wedging, the material being there inaccessible except when the vessel is under special survey. When the mast-plating is doubled, the wedges need not be removed before the third special survey, *i.e.* about every 12 years.

At the heel the mast should be supported on a strong stool, as very great downward stresses are communicated to the mast through the rigging, and if due provision be not made to resist these the mast may be forced downwards, the plating at the heel crushing up or the stool collapsing if not efficient. Such movement of the mast would cause the rigging to become slack and valueless as a support against bending.

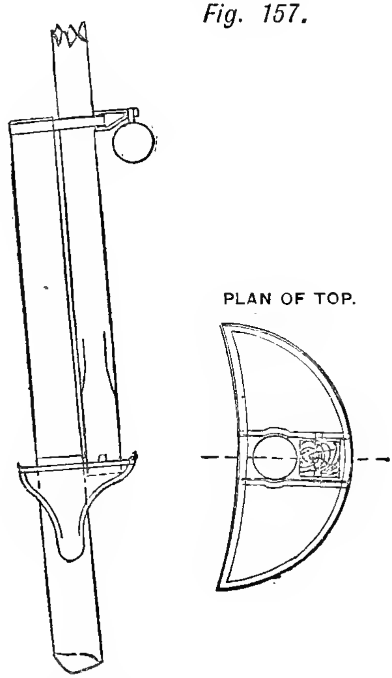
Where a mast is stepped on a centre keelson, a good stool may be contrived by fitting a strong plate immediately under the mast, and supporting it by brackets connected to the keelson and floorplate on each side of the middle line. For wedging purposes a ring is fitted on the plate round the mast-heel, and to keep the mast from turning, an angle or tee lug, riveted to the plate, is fitted through the bottom of the mast. The mast-heel plating is usually doubled for about 2 feet up from the bottom.

The main portion of a mast, and that upon which the principal diameters and scantlings are fixed, is known as a lower mast, but above this there are a topmast, a topgallant mast, and a royal mast. These upper spars are sometimes constructed of wood, but in modern sailing vessels of fair size steel topmasts are common. Lower masts and topmasts are occasionally built as single tubes, but usually they are separate, the union between them being effected by overlapping in the manner indicated in fig. 157. In this case the topmast, which is of wood, passes through a cap hoop at the lower masthead, and is supported by a rectangular bar of iron, or fid, which passes through the heel of the topmast and rests on strong cheek-plates riveted to the lower mast. This overlapping method is sometimes adopted for uniting the topmast with the upper portions, the topgallant and royal masts usually consisting of a single wood spar; but where the topmast is of steel the upper spar is frequently housed into its upper end.

The scantlings of steel topmasts, like those of lower masts, vary with the length. Topmasts 38 feet long and above should have internal stiffening bars. The edge seams of topmast plating may be single-riveted, but the end joints, like those of lower masts, and for the same reason, should be treble-riveted.

Obviously, so tall and comparatively slim a structure as a mast such as we have described, must be strongly stayed in order to hold it to its work of supporting the cross-yards and sails and resisting the wind pressure. We have mentioned that steel wire ropes are used for this purpose. Lower masts are stayed laterally by shrouds, which loop round the mast at the

Fig. 157.



hounds and extend down to the gunwale, where they are attached to chain plates riveted to the sheerstrake. Shrouds of smaller size are also fitted to the topmast and top-gallant mast. These are not carried down to the ship's sides, but are fastened to the mast just below the lower mast-top and the topmast trestle-trees respectively, the necessary spread being obtained by means of the mast-top and topmast cross-trees. As well as shrouds, the upper spaces are further held by backstays fastened to the gunwale and to the mast. In a fore-and-aft direction the masts are stayed to one another by powerful wire ropes at various heights. The stays of the foremast are run down to the forecastle deck, the upper ones being attached at their lower ends to a bowsprit; this arrangement, which is to allow of sufficient spread in the stays, also permits of large-sized staysails.

Obviously, all this rigging will have little staying value if it be slack, as in that case the mast which first receives the stresses due to the wind pressure might break before the strength of the wire could be called upon. Cases are on record of masts collapsing through lack of efficiency in the stays in this respect. To obviate such disasters, the standing side rigging of all vessels should be provided with rigging screws of simple design, by means of which the shrouds and backstays may be readily tightened up at any time.

BOWSPRIT.—In modern sailing-vessels of fair size this spar is constructed of steel, and as it has to withstand considerable bending stresses, due to the pull of the maststays attached to it, it is built of substantial diameter and thickness of plating; the latter, indeed, is about the same as for a lower mast of equal diameter.

Usually the bowsprit is housed in the forecastle, passing through an aperture in a transverse bulkhead, or knighthead plate, fitted at the fore-end of the forecastle, and abutting against a vertical plate extending between the upper and forecastle decks, and strongly bracketed to the main deck-plating. To secure it in position, the bowsprit is wedged in way of the knighthead plate, angle rings being fitted to the latter around the aperture to take the wedging. Internal stiffening angles are fitted in the middle of each plate in the round, and, in addition, when the spar exceeds 28 inches in diameter, a vertical diaphragm plate is fitted in way of the wedging and extended some distance either way beyond. The end joints of the bowsprit plating outside the wedging should be treble-riveted; inside the forecastle, they may be double-riveted.

Sometimes, instead of being housed in the forecastle, the bowsprit is sloped away on the lower side at its after-end and bedded on the forecastle deck-plating, to which it is securely connected by strong angle bars; the forecastle deck being stiffened in this neighbourhood by fitting the beams on every frame. The outer part of a bowsprit when fitted as a separate spar is called a jibboom. The latter is usually built of wood and is fitted through the cap-band of the bowsprit. Frequently, in large modern sailing-ships the bowsprit and jibboom are made of steel in one length, when it is known as a "spiked bowsprit."

The bowsprit is stayed laterally by means of wire shrouds, and strong bobstay bars are fitted to eye attachments on the stem and the underside of the cap-bands at the fore-end of the bowsprit and jibboom.

YARDS.—The cross-yards of small sailing-vessels are constructed of wood, usually pitch pine. In large ships having steel masts, the lower yard and the one above it are frequently built of the same material as the mast. The greatest diameter of a yard is, of course, in the middle, and is taken at $\frac{1}{48}$ of its length; at the ends it is tapered to half of this. When built of steel, yards have single-riveted seams and treble-riveted end joints. The lower topsail and lower topgallant yards on each mast are usually fixed to the latter, but with attachments designed to allow of free movement to any angle; the upper topsail and upper topgallant yards are attached to parrel hoops which fit loosely on the mast, thus admitting of each yard being hoisted into position by means of appropriate running gear.

A special feature in the masts and yards of sailing ships are the mountings. These are very elaborate and are made of great strength, as the safety of a vessel might be seriously threatened if even one stay attachment were to give way, on account of the increased stress which would thus be brought on the others.

MASTS OF STEAMSHIPS.—The masts of a steamer do not call for much comment. As has been said, in most modern cargo steamers they are mainly fitted for ornament; incidentally, they can be usefully employed as derrick-posts and standards for signal lamps, etc. The main function of masts in sailing-ships, which is to carry sails, has been almost done away with in steamers. No yards or square-sails are now carried; two small fore-and-aft sails on each mast are all that are usually arranged for, and these are fitted not for propulsion but to give steadiness in rough weather. In many modern cargo vessels even these are omitted. This omission of sails has been deplored in some quarters, and there certainly seems a lack of economy in neglecting to use the power of the wind for propulsion when it is available. The steadying effect of sails when a vessel is in a seaway is well known.

Owing to their auxiliary character, a steamer's masts are of smaller diameters and scantlings than those of same length in a sailing-ship. Where fore-and-aft sails only are carried, Lloyd's Rules permit the diameters to be a fifth less, and the plating of a thickness to correspond with this reduction.

It has been pointed* out that in the case of a steamer's masts, whose main duty is to withstand the strains due to the working of derricks, the breadth of the ship should be considered in fixing upon the diameters and scantlings. The broader a vessel, the greater will be the outreach of the derricks, and, therefore, the greater the bending stresses on the masts. At present no notice is taken of this in Rules for masts, and in the case of a very broad vessel, where the mast has, say, to support four derricks, they are frequently none too strong for their work.

It is customary to make a steamer's masts of pole type, *i.e.*, in one piece from heel to topmast head. As the sail spread is unimportant, no greater height than this is necessary, so that topgallant and royal masts are dispensed with, the finishing pole being fitted into the topmast. Frequently, the topmasts are of wood, and made to ship telescope fashion into the upper ends of the lower masts, appropriate gear being provided for the purpose. Such an arrangement is demanded to allow of the vessel passing under bridges in reaching ports like Manchester.

The edge seams of the mast-plating may be single-riveted, but the end joints, like those of a sailing-ship's masts, must be treble-riveted above the

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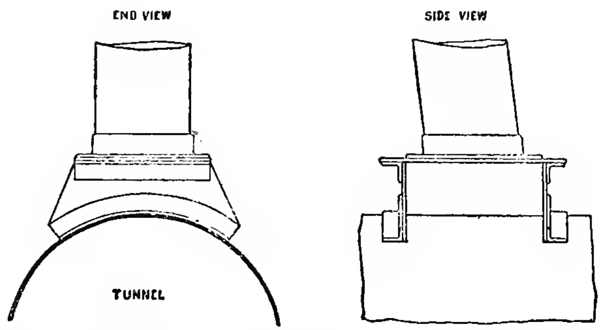
* See a paper by Mr. W. Veysey Lang, read before the Institute of Marine Engineers in February, 1909.

deck or partners, and double-riveted in the housing. When the masts are of considerable length, the strength should be augmented by fitting and securely riveting internal angle bars up the middle of each plate in the round. The masts must be supported athwartships and fore-and-aft by strong steel wire standing rigging.

The usual arrangement is, say, three or four shrouds, immediately abreast the masts on each side, with two fore-and-aft stays. This is not the best arrangement for the purpose intended. Instead of the close-spaced shrouds, it is better to fit two with as great a spacing as possible, as much, in fact, as will still permit the derricks to swing clear of the side. For access to the masthead an iron ladder may be riveted to the mast, or two additional shrouds may be fitted at close-spacing with ratlines to the masthead.

The need of strong work at the mast-heels has been pointed out in the case of sailing ships, and similar remarks apply to steamers; for as well

Fig. 158.



as the bending stresses already referred to, the working of the derricks give rise to considerable downward thrusts, steamers should therefore be strengthened in way of mast steps. If these come on an inner bottom, brackets should be fitted to the centre girder, unless the mast-heel happens to be immediately over the junction of a floor-plate with the centre girder. Fig. 158 shows the arrangement when a mast is stepped on a tunnel. The stiffening of the latter in way of the step, which usually consists of stout angle-bars riveted to the plating, is not shown in the sketch. A steamer's masts are usually wedged at the upper-deck, and the arrangement is very similar to that described for a sailing-vessel.

BULKHEADS.—This name is given to all vertical partitions, whether fore-and-aft or athwartships, which, in a ship, separate compartments from one another. Many of these partitions are of little value structurally, as those of wood between cabins, or those which, though built of iron or steel, are only intended to act as screens and are therefore of the lightest description. Bulkheads, however, which divide a steel vessel into watertight compartments, are of immense importance structurally and otherwise, and it will, therefore, be of interest to consider their principal functions and the leading features in their construction.

These main partitions, which are usually placed transversely, are strongly built and made watertight, so that in the event of a compartment being filled with water the containing bulkheads shall be strong enough to support the pressure and have joints tight enough to prevent the fluid escaping into adjoining compartments. Generally, main watertight transverse bulkheads are of value for the following reasons:—

1. As Elements of Strength.—Where they occur the hull is practically rigid transversely, so that they effectually prevent any tendency to deformation in that direction; they also afford support to the longitudinal framing, *i.e.*, to the side stringers and keelsons, when the latter are efficiently bracketed to them. The keelsons, indeed, of themselves have but little rigidity, but when braced to the bulkheads they become efficient as girders, and transmit the stresses brought upon the hull to the massive bulkheads, thus spreading the strength of the latter over the region of the structure lying between them.

2. As Safeguards Against Spread of Fire.—Their importance in this respect can hardly be over-estimated, isolating as they do the various holds with their contents from each other. Many instances are on record of vessels having been saved from total destruction by fire through the medium of their bulkheads.

3. As Preventatives Against Foundering Consequent on the Piercing of the Hull by Striking a Rock or Otherwise.—We are already familiar with the effect on the flotation of bilging a compartment, and have seen that if the latter be large the loss of buoyancy may be sufficient to sink the vessel. The importance of restricting the lengths of compartments by a sufficient number of watertight bulkheads is therefore obvious.

In the case of a steam-vessel, there are certain conditions which fix the lower limit of the number of bulkheads required. With the machinery amidships, for instance, there should be at least four: one at a short distance abaft the stem, one at each end of the machinery compartment, and one placed at a reasonable distance from the sternpost. With the machinery aft, a minimum of three watertight bulkheads might be allowed, the after-bulkhead forming one end of the machinery compartment.

Of the above divisions the forward one, which is fitted as a safeguard in the event of collision, is probably of chief importance. It should not be placed too far aft, or the loss of buoyancy due to bilging the peak compartment may be sufficient to cause the vessel to go down by the head. Lloyd's Rules require it to be fitted at a twentieth of the length from the stem, measuring at the height of the lower deck. The collision bulkhead, as it is called, has proved of immense service in saving vessels, and has often enabled them, though seriously damaged by collision, to make a port in safety.

It may here be said that the collision bulkhead is the only one usually fitted in sailing ships. In this case the transverse strength is made up otherwise, and the question of cost, to mention no other, has put to one side any idea of fitting numerous bulkheads.

The importance of the bulkheads which isolate the engines and boilers from the cargo spaces scarcely needs emphasis. The chance of fire and other damage to cargo, if there were no efficient tight divisions, is clearly apparent. It is also necessary that the machinery compartment should be quite self-contained, so that the bilging of neighbouring compartments would not mean the extinguishing of the boiler fires.

The after-bulkhead is required so as to isolate leakage which may be caused by the breaking of the stern tube, or by general vibration due to the action of the propeller. Usually, it is placed near enough the stern to prevent any loss of buoyancy consequent on the bilging of the after compartment being sufficient to endanger the vessel. Incidentally, it forms a splendid stiffener at this part, an important consideration when the machinery is placed aft.

Although no surveyor to the Board of Trade, under the existing regulations, could refuse to grant a declaration of survey that the hull, even of a passenger steam-vessel, whatever her length, was sufficient for her work, if fitted with bulkheads equivalent to the foregoing, such an arrangement can clearly be considered satisfactory only in small steamers. With increase in size, additional bulkheads very soon become desirable, partly because of the need of providing greater transverse strength, but as this may be met otherwise, mainly because safety in the event of fire or bilging demands an adequate sub-division of the holds.

Thus we find, taking Lloyd's latest Rules for example, that when steam-vessels reach 285 feet in length, five bulkheads are necessary, the distance between the collision and boiler room bulkheads being sub-divided. In vessels of 335 feet, the after hold is in turn sub-divided, making six bulkheads, the number of watertight bulkheads becoming seven, eight, nine, and ten, when vessels reach lengths of 405, 470, 540, and 610 feet respectively.

Obviously, the question of sub-division is of first importance in purely passenger vessels, no form of life-saving appliance being so efficient as a good system of watertight bulkheads. Few first-class passenger steamers are therefore now built but can float safely with, say, any two* compartments in open communication with the sea, while some have even a better sub-division, and partly on this account, a few of these have been subsidised by the Government to act as auxiliary cruisers in time of war.

A point of great importance in the fitting of bulkheads is that they should extend well above the loadwater line; otherwise the bilging of one compartment may cause sufficient sinkage to submerge the tops of the bulkheads, in which case the water would find its way into all the compartments

* In the report of the Bulkhead Committee of 1890, the highest class of sub-division is given as that which would enable a vessel to float safely, in moderate weather, with any two compartments in open communication with the sea.

and thus sink the vessel. In general, bulkheads should extend to the top deck of the main structure. In vessels with continuous superstructures, such as an awning or shelter deck, the bulkheads (with the exception of the collision bulkhead, which should extend to the awning or shelter deck) are usually stopped at the deck below, *i.e.*, the upper deck, in virtue of the greater freeboard and reserve buoyancy of this class. In the 'tween decks of these vessels, a deep web frame or partial bulkhead is to be fitted on each side immediately over the watertight bulkheads, or other efficient strengthening must be provided.

CONSTRUCTION OF BULKHEADS.—Although an ordinary watertight bulkhead may never be called upon to sustain the pressure due to a compartment on either side becoming filled, it must be constructed strong enough to meet such an eventuality. It should, therefore, be built of plates of substantial thickness, and be strongly stiffened. Lloyd's Rules require a thickness of ·26 of an inch in bulkheads having a depth from upper deck to floors of from 8 to 12 feet, *i.e.*, in the smallest vessels, and of ·46 of an inch in those in which the same depth is from 44 to 50 feet, *i.e.*, in very large vessels. The plates are fitted vertically or horizontally, are usually lapped at edges and at end joints, and single-riveted, the rivets being spaced for watertight work, *i.e.*, $4\frac{1}{2}$ diameters apart. At the points where the end joints come on the edges, the two plates of the former are thinned down for the breadth of the edges laps so as to obviate the fitting of slips.

In the stiffening of watertight bulkheads, the plan recommended by the Bulkhead Committee of 1890 is now usually followed, the stiffeners being arranged generally in a vertical direction (*see* also stiffening of collision bulkheads). In the former Rules of Lloyd's Register, bulkhead stiffeners were required to be arranged horizontally as well as vertically—a cross-bracing arrangement which assured the strength of the bulkhead in a transverse as well as a vertical direction, making it efficient to resist pressures tending to force in the ship's sides, which a purely vertical arrangement of stiffeners is not adapted to do. By the cross arrangement, too, the unsupported area is less than by the vertical. Still, the advantages of an entirely vertical arrangement of stiffeners are considerable. To begin with, as the depth of a bulkhead is less than the width, stiffeners are shorter and therefore more efficient arranged vertically than when arranged horizontally—the rigidity of girders varying inversely as the cubes of their lengths. Again, the pressure which a bulkhead may be called upon to withstand is greatest at the bottom, and a range of closely-pitched vertical stiffeners bracketed to the tank top are effectively placed to resist this.

The spacing of stiffeners in ordinary watertight bulkheads should not exceed 30 inches. In the case of a collision bulkhead, as a vessel's safety may depend on this bulkhead's ability to withstand the dashing about of masses of water admitted to the fore peak through collision, the spacing of vertical stiffeners should not exceed 24 inches, and in this case there should also be horizontal stiffeners consisting of bulb angles on the opposite side, spaced 4 feet apart, bracketed to the ship's sides. As the horizontal stiffeners are short, the vessel being narrow at this part, they add immensely to the rigidity of the bulkhead.

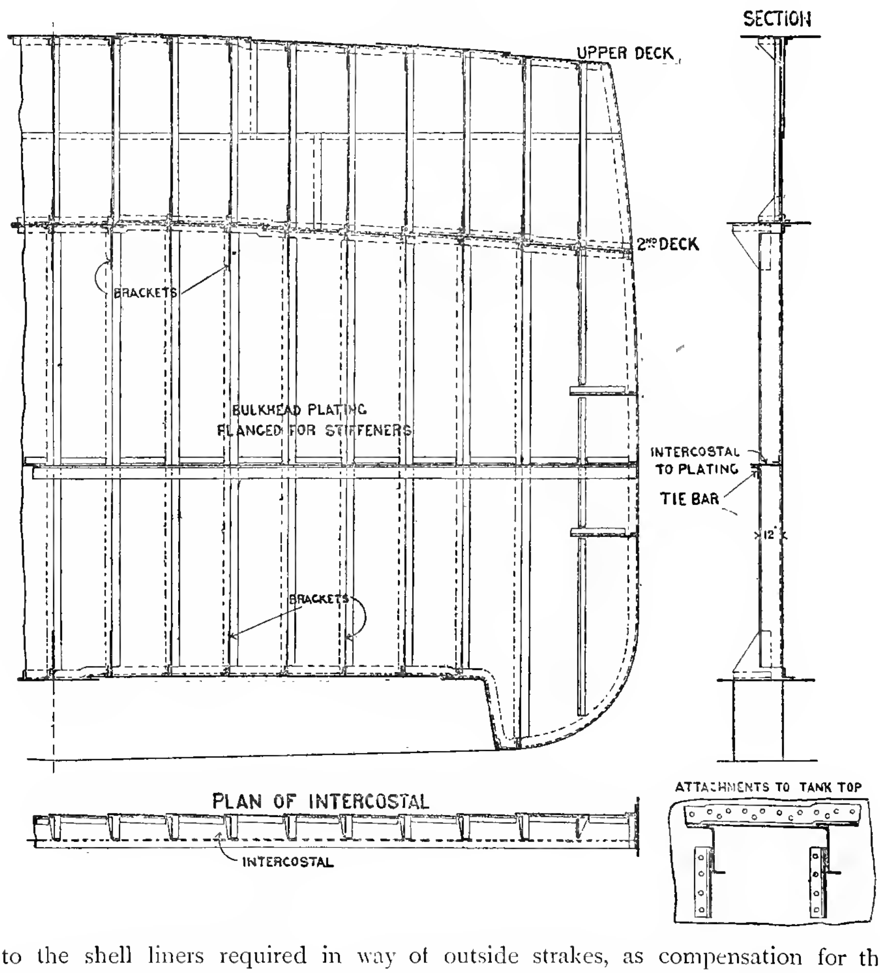
Bulkheads which form the ends of oil compartments in vessels designed to carry oil in bulk, or which form the ends of deep-water ballast tanks, should be of extra strength, because, as well as fulfilling the main function of ordinary bulkheads in affording sufficient structural strength, they must be able to resist the pressure of the mass of fluid which the compartment contains, the speed of the vessel being communicated to the fluid in a compartment through the bulkhead at its after-end; also, as any compartment on occasion may not be quite full, its bulkheads should be strong enough to meet the very severe stresses which the dashing about of large masses of fluid in a partly-filled tank would give rise to. Lloyd's Rules provide scantlings for the bulkheads of oil vessels.

Frequently, the edges of plates forming bulkheads are flanged to act as stiffeners. This entails a vertical arrangement of somewhat narrow plates, since the distance between stiffeners must not be more than 30 inches. There is here a saving in riveting, and fewer parts require to be put together; this system is therefore rather popular, especially as experiments have shown the arrangement to be as strong as the ordinary one, and as mild steel may be readily flanged cold. Lloyd's Rules require that when flanged stiffeners are 12 inches or more in depth, intercostals are to be fitted between the stiffeners and connected to the bulkhead plating and to a bar on the face of the stiffeners (*see* fig. 159). These intercostals, which are to be spaced not more than 10 feet apart, should greatly stiffen the bulkhead by preventing any tendency to trip on the part of the stiffeners.

In Lloyd's Tables the scantlings of the stiffeners are shown to vary with the full depth of the bulkhead as governing the maximum pressure that could come upon it. When the bulkhead is divided into zones by the abutment of steel decks, the scantlings of the lower stiffeners, that is, those between the tank top, or in single bottom vessels, the top of floors and lowest laid deck, are governed by the full depth as fixing the intensity of the pressure, and the length of the stiffener as fixing the load. 'Tween deck stiffeners are, in the same way, governed by the distance from the top of the bulkhead to the lower part of the 'tween decks, and by the length of the stiffener. Stiffeners in way of holds and 'tween decks, except the upper 'tween decks, should be bracketed top and bottom. This follows from the consideration that a uniformly loaded girder fixed at the ends is 50 per cent. stronger and five times more rigid than one with free ends. Lloyd's Rules permit bulkhead stiffeners, in small vessels, to be fitted without end brackets, provided their scantlings be increased beyond the tabular requirements. In oil vessels, which are generally built without inner bottoms in way of the oil holds, the knee brackets at the lower ends of the bulkhead stiffeners should be fitted between the floors to the shell.

At the edge every watertight bulkhead should have a strong connection to the shell-plating, inner bottom (where one is fitted), and deck-plating. Double angles are frequently fitted to the shell-plating and inner bottom and make a good job, but it is becoming increasingly common, particularly in cargo vessels, to have instead large single bars double-riveted in both flanges; the latter arrangement is cheaper and is probably not less strong. Lloyd's Rules make provision for both methods. Reference has already been made

Fig. 159.



to the shell liners required in way of outside strakes, as compensation for the closer spaced rivets—necessary for caulking—through the shell angles of water-tight bulkheads. When the shell liners are not fitted, as with joggled plating or framing, bracket knees between the shell-plating and the bulkhead in way of outside strakes are necessary, except where the hold stringers are 5 feet or less apart. The joints on one side only of a bulkhead require to be caulked. Where hold stringers and keelsons pass through bulkheads, caulked angle collars should be fitted on the watertight side, and to give a finished appearance, plate collars, uncaulked, on the other side. Frequently, hold stringers are stopped at the bulkheads, and the longitudinal strength is made good by fitting substantial bracket plates connected to the bulkhead by angles, and to the stringers by a riveted lap (*see* figs. 159 and 160).

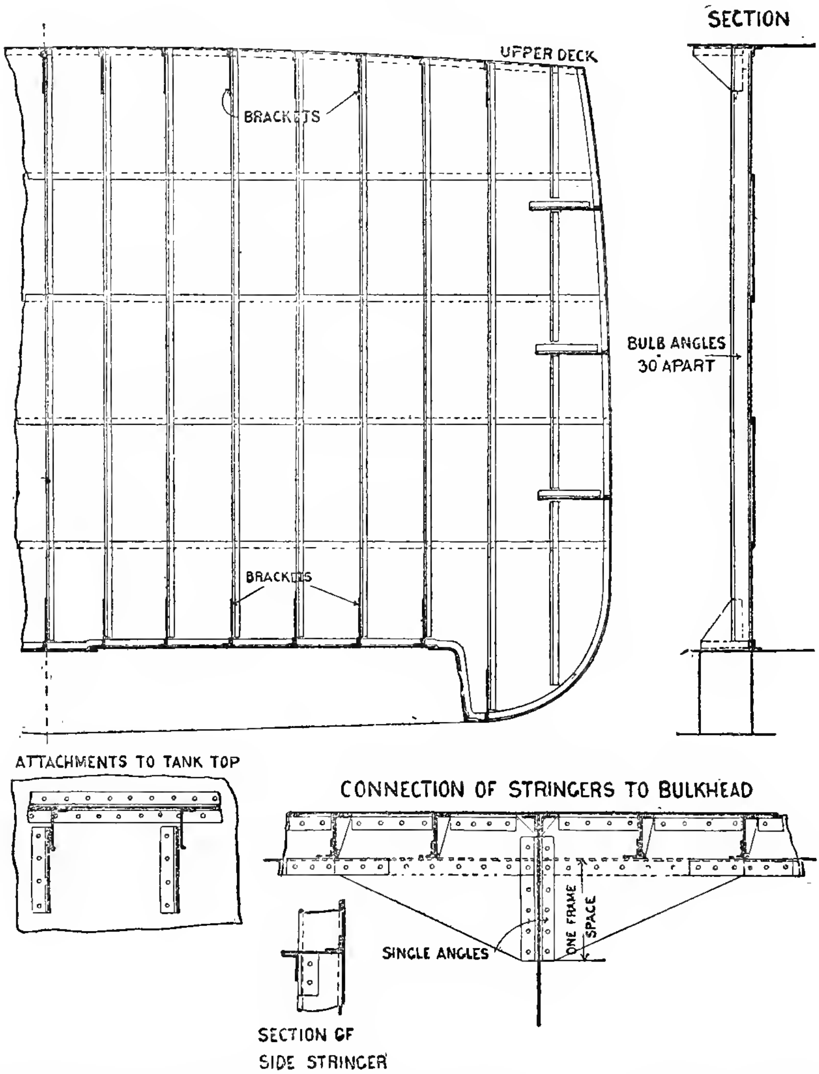
The subordinate bulkheads of a ship, such as screens and casings, do not call for a lengthened description. They are constructed of light plates and bars, the former having single-riveted joints. They are not usually watertight, and where perforated by beams, dust tightness is secured by fitting plate collars. Where screens take the place of pillars, as in the case of side bunker casings and centre-line bulkheads, additional rigidity is called for, and is obtained by increasing the thickness of plating and making the stiffeners of substantial size, the latter being fitted two frame spaces apart in line with the beams and attached thereto. Where vessels have open floors, the centre-line bulkhead is attached to the vertical plate of the centre keelson. A centre-line bulkhead is usually stopped in way of the hatches, so as not to interfere unduly with stowage, and, when required for grain cargoes, the continuity of the division is made good by wood shifting-boards. Although interrupted in this way, when properly built, a centre-line bulkhead is a splendid vertical web, excellently adapted to resist longitudinal deflecting stresses.

When machinery casings in 'tween decks have to take the place of quarter pillars, they must be strongly built, and the stiffeners should be riveted to the beams.

DOORS IN WATERTIGHT BULKHEADS.—It is, of course, desirable that watertight bulkheads should be intact, as their efficiency as subdivisions of a hold is then at its highest. In some cases, however, doorways must be cut in them. For instance, the need of a direct means of access from the engine-room to the shaft tunnel, calls for a door in the watertight bulkhead at the after-end of the engine-room, where the tunnel abuts upon it. Again, in most cargo steamers, a reserve coal bunker lies immediately before the forward boiler-room bulkhead, in which doors must be fitted so that the coal may reach the stokehold floor. In special cases doors have been fitted at the ceiling level in all the watertight bulkheads of a vessel, when it has been desired to pass from hold to hold without going on deck. Besides the foregoing, particularly in passenger vessels, doors are frequently made in watertight bulkheads at the height of the 'tween decks, so that passengers may readily get from place to place in the region devoted to their accommodation.

In designing doors for watertight bulkheads it is necessary to remember that one placed near the foot of a bulkhead would have to withstand considerable pressure, if from any cause a compartment on either side of it became flooded. The framing of the doorway and the door itself are therefore made specially strong. Usually these parts are of cast iron of substantial thickness. The door is made of wedge shape, as also the groove in which it works, any degree of tightness of the joint, which is a metal to metal one, being thus obtainable. As in the case of bilging a door would quickly become inaccessible, arrangements must be provided for working them from a high level. Doors in engine and boiler-room bulk-

Fig. 160.

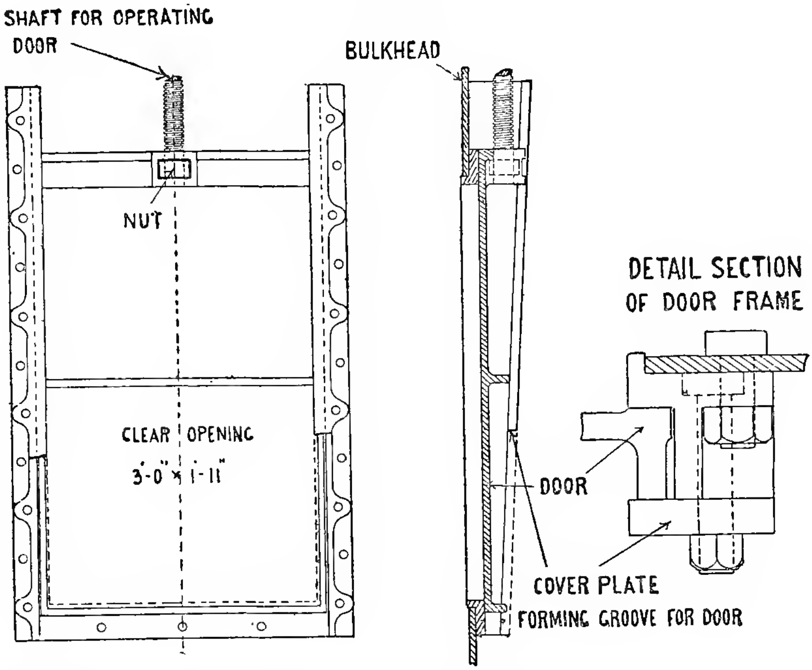


heads are usually wrought from an upper platform; doors in other bulkheads are worked from the deck. Where doors open in a vertical direction (fig. 161) the apparatus for working them commonly consists of a vertical shaft with a screw at one end working in a fixed nut in the door. Where they open in a horizontal direction, the vertical shaft is fitted at its lower end with a small pinion wheel which works a fixed rack on the door.

Doors in bulkheads which give access into 'tween decks need not be designed to withstand great water pressure. Usually, they consist of plates hinged to the bulkhead and secured by snibs so fitted as to be readily operated from either side of the bulkhead. The joint between the door and the door frame on the bulkhead is made watertight by means of spun yarn or rubber packing (fig. 162).

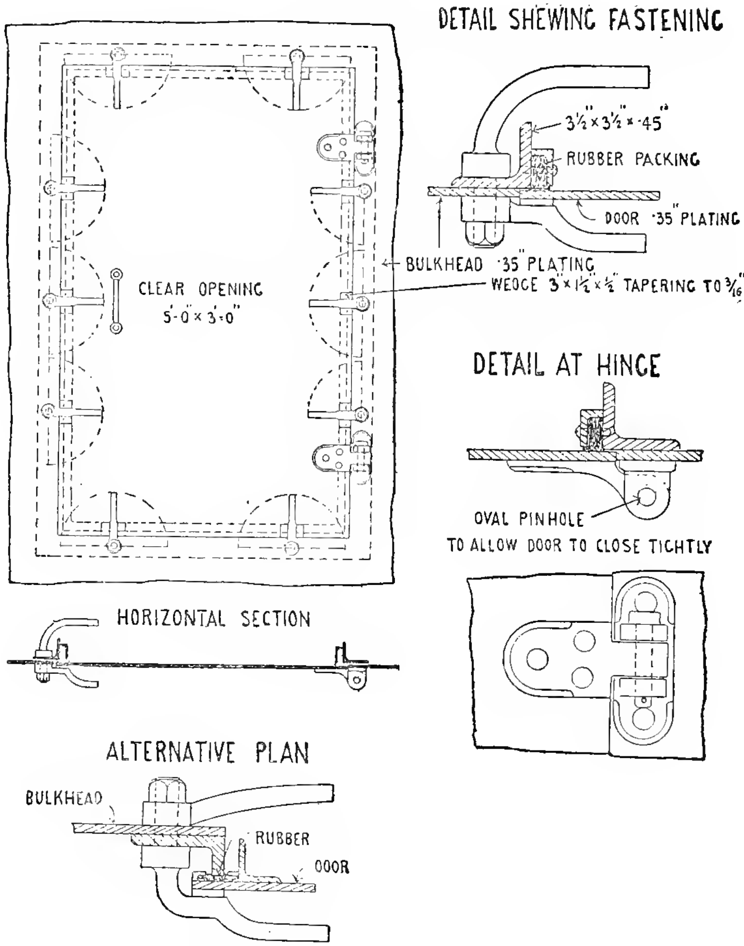
STEMS, STERNPOSTS, AND RUDDERS.—In merchant vessels the

Fig. 161.



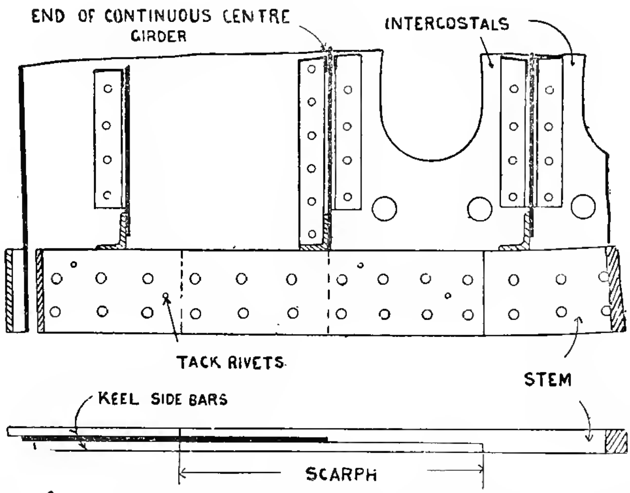
stem consists of a solid forged bar of iron or steel, or of rolled steel, of suitable breadth and thickness, and forms the fore-end of the hull. Nowadays, stems are usually straight above the load-waterline with a slight rake—say, two feet—forward, to minimise the effect of a collision, should this happen, and to overcome the impression of falling aft at the head which a quite vertical stem gives. The clipper stem, so common at one time, is now seldom built on steamers; in sailing ships, as it is a suitable construction with a bowsprit, and also has a fine appearance, it is always found. When associated with a hanging or bar keel, the stem becomes a continuation of the same, being connected to it by a vertical scarph similar to that employed for uniting the lengths of the keel bar. When the keel is of centre through plate or side bar type, a modification of the ordinary vertical scarph is adopted. By referring to fig. 163, it will be seen that the after-end of the stem is slotted out to receive the ends of the centre girder and the two side bars, which together make up the thickness of the keel. The total length of this scarph should be about eighteen times the keel thickness, or double the

Fig 162.



length of scarph required for an ordinary bar keel, to allow a reasonable distance between the terminating points of the flat side bars. There are several ways of making a connection between a stem bar and a flat plate keel. Fig. 164 shows one adopted by many builders. The lower part of the stem is carried three or four feet on to the fore length of the keel, and is securely riveted to intercostal plates, which in turn are riveted to the floors. The lower ends of the frames in this vicinity extend below the top of the stem bar to the line shown dotted in the figure, and the fore end of the keel-plate is dished so as to come under the stem and yet fay against the ship's side. The keel-plate may be said to end where it rises on to the side of the stem (fig. 164), as in front of that point it becomes an ordinary strake of shell-plating. The preceding is an efficient plan, and obviates the necessity of tapering down and spreading out fanlike the after-end of the stem—a more costly arrangement, but one which gives good work and formerly frequently adopted. It will be observed from fig. 165, which illustrates this method, that the keel-plate is dished round the after part of the stem, and continued for the distance of a frame space or two under it. Thence, as in the previous case, the keel-plate is lifted on to the side of

Fig. 163.

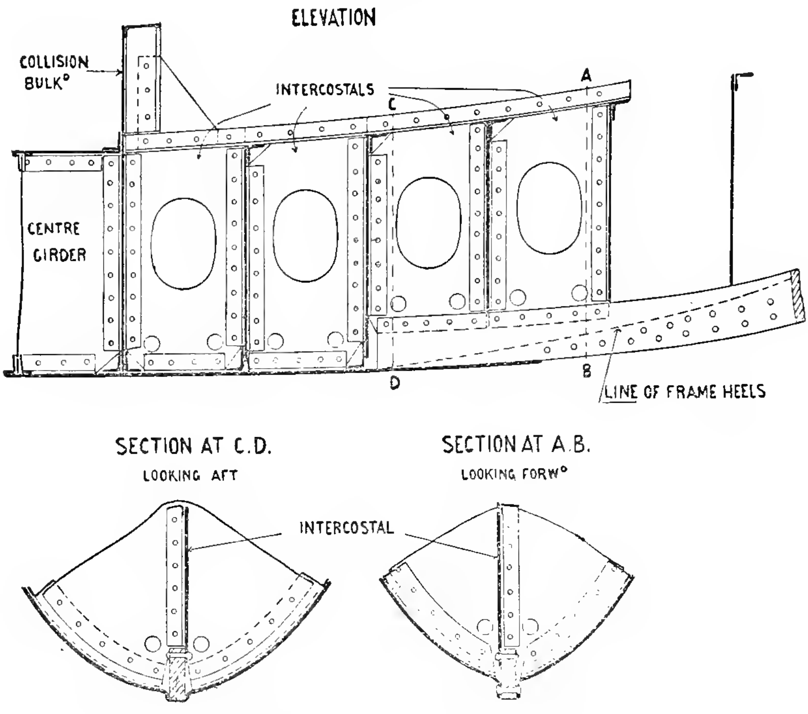


the stem and through riveted to it. Through riveting is also adopted at the after-end of the stem, if practicable, otherwise tap riveting is resorted to. The centre keelson-plate is carried intercostally for a few frame spaces forward of the after-end of the stem and attached to a tongue formed on the stem, as in the sketch, or in lieu of a tongue to bottom bars tap-riveted to the stem.

As well as by means of the keel connection, the stem is thoroughly bound with the structure by the main shell-plating. The strakes at their forward ends are arranged to lap on each side of the stem, and rivets sufficient in length to pass through all three thicknesses are employed (*see* fig. 166). It will be noticed that the shell-plating is kept back $\frac{1}{2}$-inch from the front of the stem; this is to protect the caulking. At least two rows of rivets are required to connect the shell-plating to the stem, and these should have the same spacing as the keel rivets, viz., 5 diameters, centre to centre. Below the load waterline the stem should be maintained at full thickness, as it is there liable to severe strains by grounding or collision; above that point it is usually reduced somewhat. In practice it is tapered to the top, where the sectional area has three-quarters its maximum value.

To facilitate construction and reduce the cost of repairs, in the event of damage to the stem, the latter is usually made in two parts with a scarph at about the light waterline. Mention may here be made of the practice of

Fig. 164.



using tack rivets in stem scarphs. They are fitted to join the parts together for the purpose of erection and fairing, but they are a drawback when a portion of the stem has to be removed, as plates on both sides of the stem have to be taken off in order to punch them out, for which reason they are frequently omitted.

STERNPOSTS.—The sternpost forms the after-end of the hull structure. In sailing-ships and paddle steamers, and also in some twin-screw steamers, it consists, like the stem, of a simple bar, with the addition of forged gudgeons for hinging the rudder. In single-screw steamers, however, this part of the ship becomes more complicated, for in addition to providing facilities for carrying the rudder, the propeller shaft, which leaves the hull at this point, must be supported by it. Moreover, the strain caused by the continual working of the shaft has to be counteracted, and this can only be done by making the post and its connections to the hull of ample strength. Fig. 167 shows the stern frame of an ordinary cargo steamer. The stem of this vessel is 11 inches by 3 inches, and the increase in strength of sternpost necessary, for the reasons given, is represented by the increase of the thickness

Fig. 165.

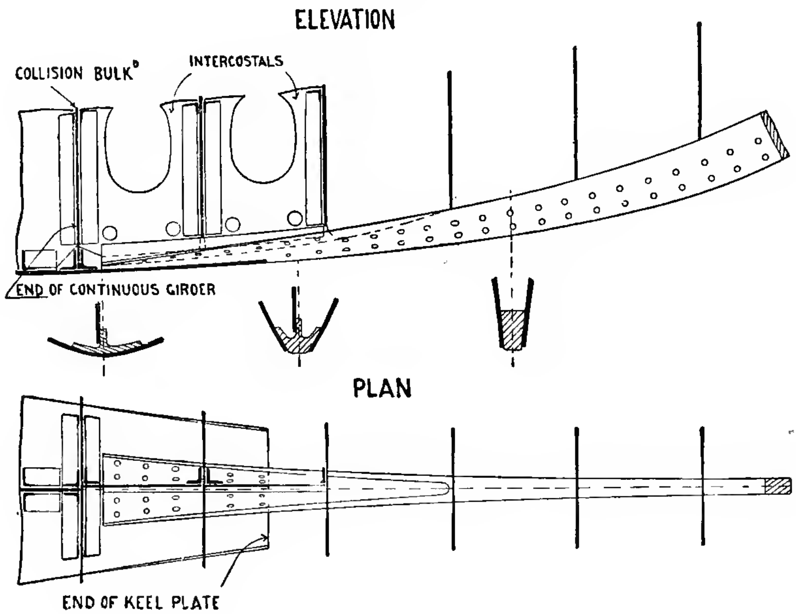
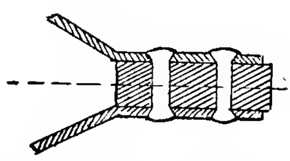


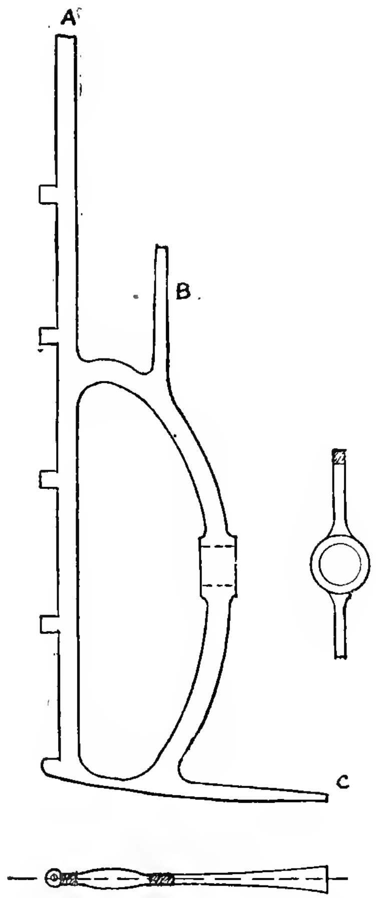
Fig. 166.



of the propeller post, which is joined to the shell-plating, to 9 inches, the breadth remaining 11 inches, while the rudder post, which is not called upon to withstand such severe stresses, may be $9\frac{1}{2}$ inches × 9 inches. Besides this, as previously mentioned, the after lengths of the shell-plating, which come upon the propeller post, are augmented in thickness above adjoining plates, being usually of midship thickness, while the plates in way of the bossing round the shaft are still further thickened. The shell-plating is attached to the stern frame by two rows of rivets of large diameter, increased below the boss in vessels over 350 feet in length to three rows. The hull attachment is further improved by securely connecting the upper arms—marked *A* and *B** in fig. 167—to floorplates, also by extending the arm *C* well forward, and connecting it to the keel-plate and middle line keelson.

The size of the aperture is fixed by the diameter of the propeller, for the efficient working of which ample clearance must be allowed. It is of

Fig. 167.



importance to keep the propeller as low down as possible so as to ensure its always being under water, as when partly immersed, the efficiency is much reduced. For this purpose the lower part in way of the aperture is reduced in depth and increased in width, the sectional area being increased 15 per cent. over that of the propeller post, as this part has frequently to with-

* Arm *B* is required by Lloyd's Rules in vessels whose longitudinal number is 16,000 and above.

stand severe grounding stresses. The main purpose of the after-post is for hanging the rudder, for which the necessary braces or gudgeons are provided, as shown. These should be spaced sufficiently close to properly support the rudder. In Lloyd's Rules tabulated distances are provided on the basis of the diameter of the rudder stock; in the Rules of the British Corporation, gudgeons are required to be spaced 4 feet apart in vessels of 10 feet depth, and 5 feet 6 inches apart in vessels of 40 feet depth and upwards, the spacing in vessels between 10 feet and 40 feet depth being found by interpolation. Gudgeons should have a depth equal to $\frac{7}{10}$ of the diameter of the rudder stock. These details of the sternpost are best considered in association with the rudder. When vessels are of large size it becomes impracticable to make the stern frame in one piece. Moreover, as the part outside the hull proper is most liable to damage, it facilitates repairs and makes them less costly, if this portion can be easily disconnected from the remainder. We, therefore, usually find that stern frames in large modern single-screw steamers are built up, as shown in fig. 168, with scarphs as shown. The upper joint can be disconnected without disturbing the main structure, while the lower one only interferes with the after-length of the lowermost strake of shell-plating. These scarphs should have a length equal to three times, and a breadth equal to $1\frac{1}{2}$ times, the width of the frames, and be secured by four rows of rivets.

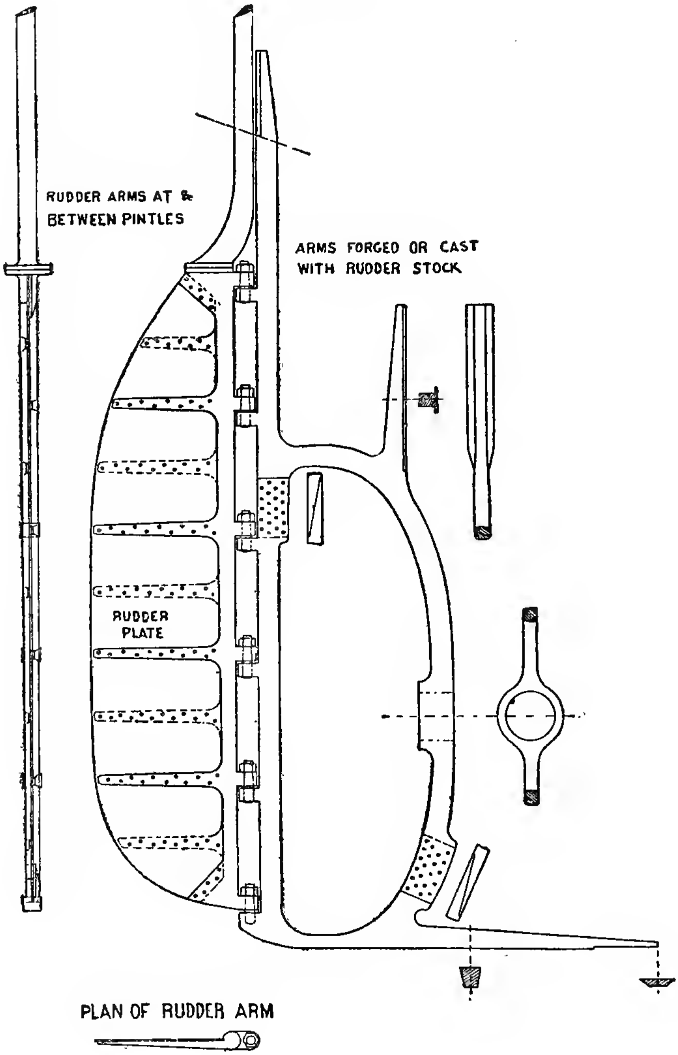
It should be said that stern frames are built as just described, *i.e.*, in several pieces, only when they are made of cast steel; but there seems no good reason, except the extra expense and difficulty of forging the scarphs, why, in ordinary simple cases, forged stern frames should not be so made, considering the advantages accruing thereto.

A word may here be said in regard to the relative merits of cast steel and forgings for stern frames, rudders, etc. The rules of the classification bodies permit the use of cast steel for such items, subject to their withstanding certain tests, and as castings are cheaper than forgings they are popular with some builders. But, from an owner's standpoint there are objections to castings. They are, for instance, not as reliable as forgings, for while flaws in the latter are rare, inherent weaknesses, acquired in the processes of manufacture, frequently exist in stern frames and rudder castings, and these, though not disclosed by the usual tests, are sure to manifest themselves subsequently when the parts are in place and doing their work. Again, a defect in a forged frame may frequently be effectively, quickly, and cheaply repaired, but a serious flaw in a steel casting simply means its renewal, which, in addition to considerable expense, may cause loss to the owners in delaying the ship. It is only fair to state, however, that in recent years there has been improvement in the manufacture of large steel castings.

Of course, where the forms of stern frames and rudder are complicated, as in the case of some war vessels and large passenger liners, steel castings are resorted to because forgings are quite impracticable.

So far, reference has been made exclusively to the stern frames of single-screw steamers, but those of modern twin-screw vessels call for special mention. In the simplest form, as in the case of small vessels, the stern

Fig. 168.



frame proper is of the familiar **L**-shape fitted to sailing-ships, the projecting propeller shafts being supported by means of a ʌ bracket on each side. This form is illustrated in figs. 169, 170. In the first figure it will be observed that the upper palm of the bracket is fitted directly on to the shell, which is doubled in the vicinity, and the lower one is through-riveted

Fig. 169.

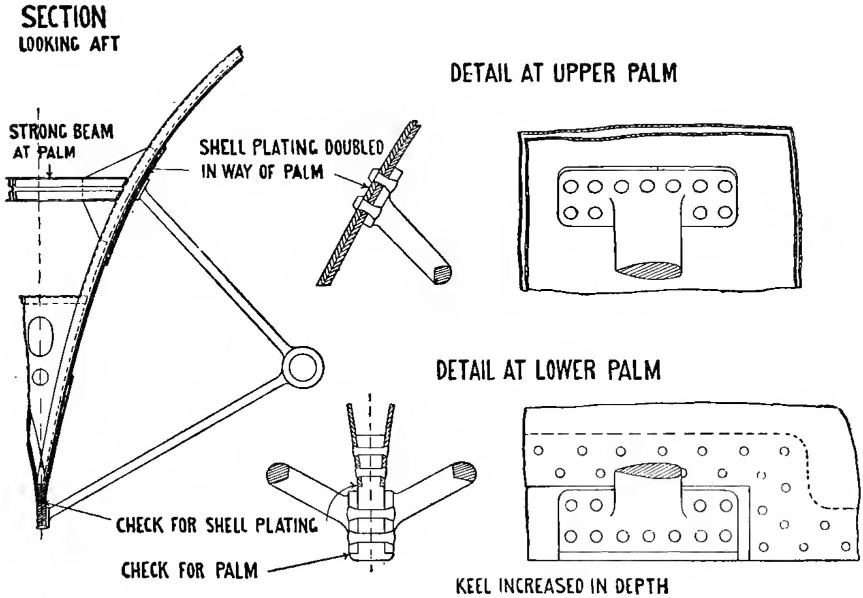
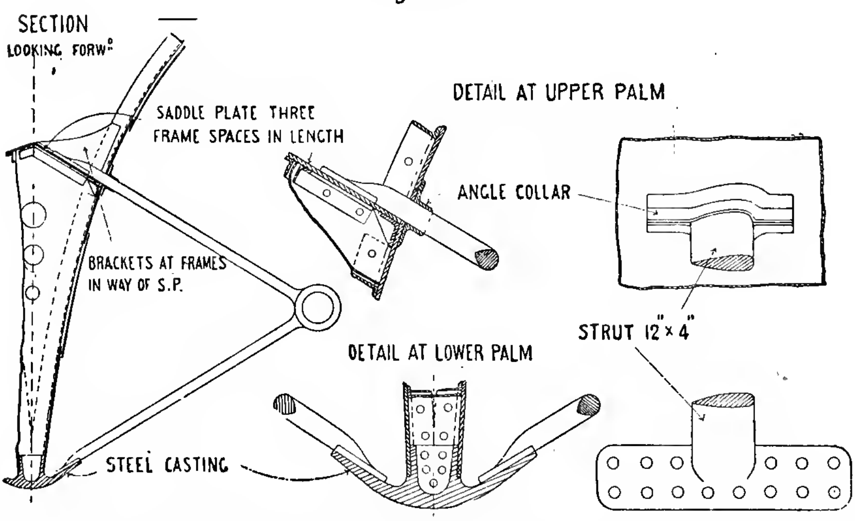


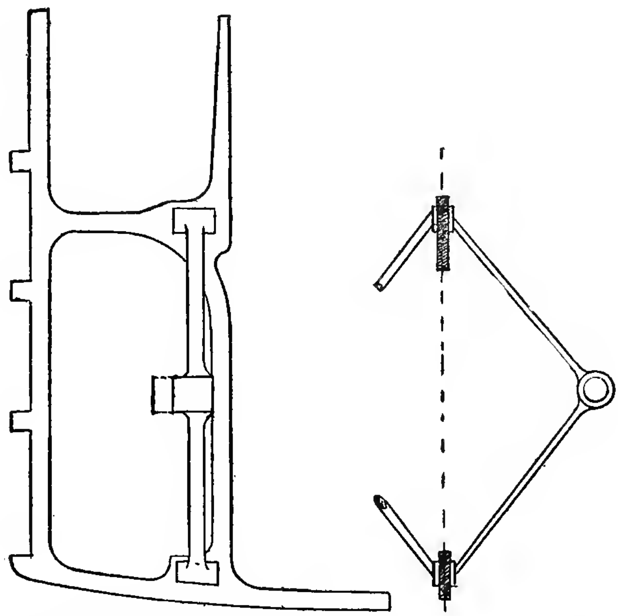
Fig. 170.



to the keel, the latter being made deeper for the purpose at the place required, a strong plate beam, connected to the sides by deep bracket plates, being fitted across the ship in way of the palms, to give the necessary rigidity at this part. In the second case, the upper palm is riveted to a plate inside the ship, an angle collar being fitted round the strut where it passes through the shell-plating, and the lower palm is riveted to a projection forged or cast on the lower part of the stern frame.

Very often, in order to keep the lines of shafts near the middle line, and thus minimise vibration as well as protect the propellers, the latter are overlapped, a screw aperture thus becoming necessary. The usual arrangement in such cases, when associated with Λ brackets, is as shown in fig. 171. The aperture must be of sufficient width in a fore-and-aft direction

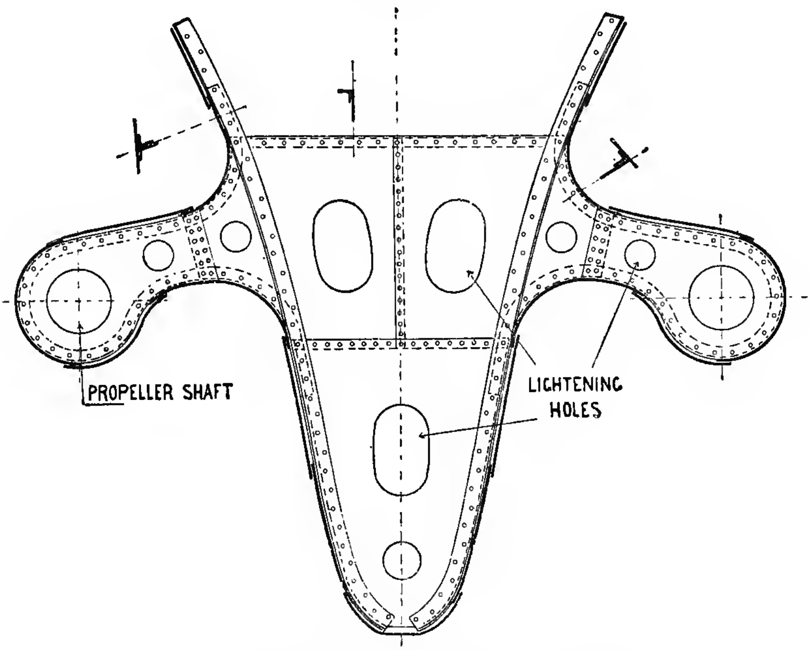
Fig. 171.



to take both propellers; it need not, however, be so high as for a single screw, the upper point of the propeller path being clear of the middle line. In the special instance before us, the stern frame is designed in such a way as to take the palms of the Λ brackets, the whole being riveted together. Other arrangements might easily be devised, although that shown is very neat. The different fore-and-aft positions of each propeller is arrived at by making the shaft bossing longer on one side than on the other.

The Λ bracket system of supporting the propeller shafts, though simple, is not suitable where high speeds have to be attained. Experiments have shown that in such cases the projecting brackets cause a serious augmentation of resistance. It was found, for instance, in one case, that of a twin-screw vessel of fine form, the propeller shafts of which were encased in tubes supported by two sets of struts, that the resistance caused by the tubes amounted to $4\frac{1}{2}$ per cent., and by each set of struts to about $10\frac{1}{2}$ per cent. of the total hull resistance. Various attempts have been made to overcome this objection by giving a suitable shape to the arms, which from a more or less circular section, in early vessels, became of a flattened oval shape in those more recently built. The results obtained in this way were better, but the resistance was still serious. The strut resistance being mainly due to the disturbance of the stream lines, an attempt was latterly made to eliminate this by bossing the form of the vessel round the shafts, right up

Fig. 172.



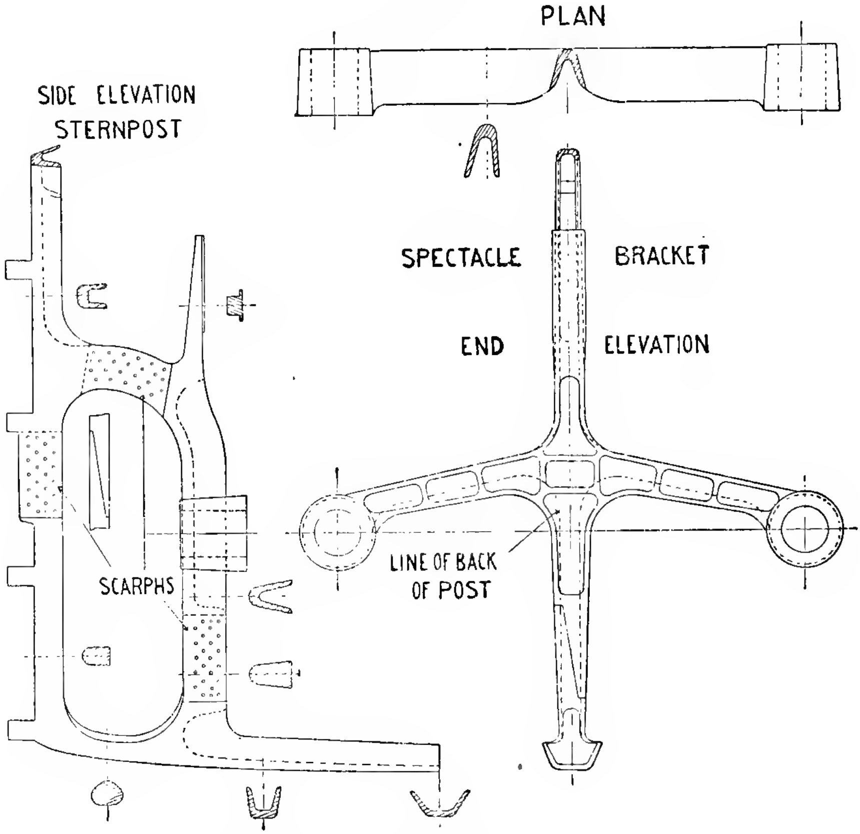
to the stern frame, thus allowing the streams an unbroken run aft.* This plan, although somewhat costly, has otherwise proved most satisfactory, and is now frequently adopted, particularly in fast vessels of large size.

As well as reducing resistance to speed, bossing the hull round twin-screw shafts has an obvious advantage in that it adds much to the strength at the after end. It also obviates the possibility of any lateral strain being brought upon the shafts, as might happen where they are exposed.

* A model of the liner *Kaiser Wilhelm der Grosse*, when tried in the experimental tank at Bremerhaven, was found to have 12 per cent. more resistance with propeller brackets than when fitted with shaft bossing.—*Engineering*, 9th October, 1908.

When properly constructed the bossing becomes an integral part of the hull structure. Fig. 172 shows in section the method of constructing the fin—as it is sometimes called—where the distance from the ship's side to the line of shafting is considerable. As will be seen, each main frame is carried down in the ordinary way, and on the reverse side a bar suitably shaped is fitted and made to overlap the main frame for some distance above and below the points at which it leaves the normal frame line. The

Fig. 173.



frame-work is strengthened by web-plates, as shown. All the frames in the bossing need not be built in this way. For a considerable distance the eccentricity in form may be met by bossing out the main frame, the object being to obtain the required shape and strength as economically as possible.

For the purpose of forming bearings for the shafts and a termination to the bossing, a special steel casting, sometimes described as a spectacle frame, is fitted Fig. 173 shows this frame associated with an ordinary propeller frame having an aperture—a fairly common arrangement. In the figure the spectacle frame forms part of the propeller post; frequently it is a distinct casting bolted to the propeller post, which is complete without it.

RUDDERS.—The rudder is that part of a vessel which controls the direction of her movements when afloat and in motion. As the axis about which a vessel turns is in the vicinity of amidships, and as the rudder takes the deflecting force, obviously the best position for it is at either end of the vessel. The after-end is most convenient for the purpose, and, with a few exceptions, it is always placed there.

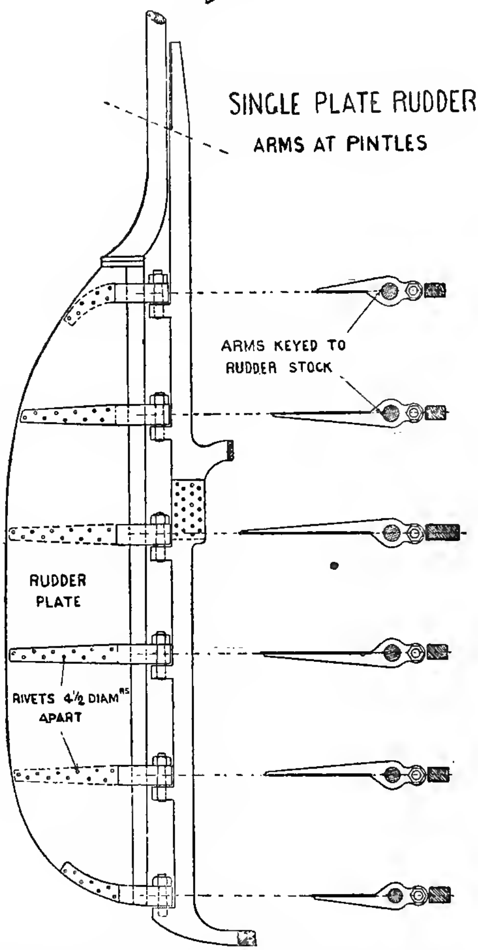
In most mercantile vessels the rudder is hinged about an axis at its forward end (figs. 168, 174); in war vessels, and in some few merchant ships, what is termed a balanced rudder is fitted, having the axis so placed that about a third of the area lies before it. The obvious advantage of the latter type consists in the ease with which it can be put over to port or starboard. It has a disadvantage, however, in being somewhat costly, a sufficient reason to debar its adoption in ordinary cargo vessels, especially, as owing to their low speed, a rudder of common type can be operated by a steering gear of moderate power.

Fig. 168 shows the rudder frame of a modern cargo steamer of large size. It is seen to consist of a vertical main frame or stock with arms at right angles to it, the latter being spaced close enough to afford sufficient support to a heavy plate which gives the contour of the rudder. The arms, which are forged or cast with the rudder frame, are arranged on alternate sides of the plate, as shown in the sketch. The rivets attaching the arms to the rudder-plate should be of large size, and the arms kept back a little from the outside edge of the plate to protect them from being torn off. The rudder is attached to the stern frame by means of bolts or pintles, which ship into gudgeons on the after-part of the stern frame. These gudgeons are forged or cast solid with the stern frame, and are afterwards bored out at the ship as required, care being taken to keep their centres in line so that the rudder may have a true axis. Formerly, the rudder pintles were also forged on the rudder frame, but are now usually portable bolts, as in the illustration given.

Fig. 174 shows a style of rudder frequently fitted in modern vessels. It has a circular stock and arms that are separate forgings fitted one at each pintle. This is about double the spacing of the previous case, to allow for which the wider spaced arms are made relatively heavier. In fitting the parts together the post is turned in way of the arms, which are shrunk on, a key being fitted to prevent the arms turning. Usually a groove is cut in the back of the stock for the rudder-plate to fit into, the stresses on it being thus communicated directly to the stock and the rivets in the arms to some extent relieved.

The weight of the rudder, in most cases, is taken by the bottom gudgeon of the sternpost. Fig. 175 shows this arrangement in detail. The socket for the bottom pintle is not continued through the gudgeon as with the others, but sufficient housing is allowed to prevent any danger of accidentally unshipping. In the present instance, the depth of the socket is 4 inches. To minimise friction, the bottom of the pintle is rounded, and a suitable bearing provided by fitting a hemispherical steel disc into the gudgeon socket. Experience with this style of bearing has not shown it to be completely

Fig. 174.



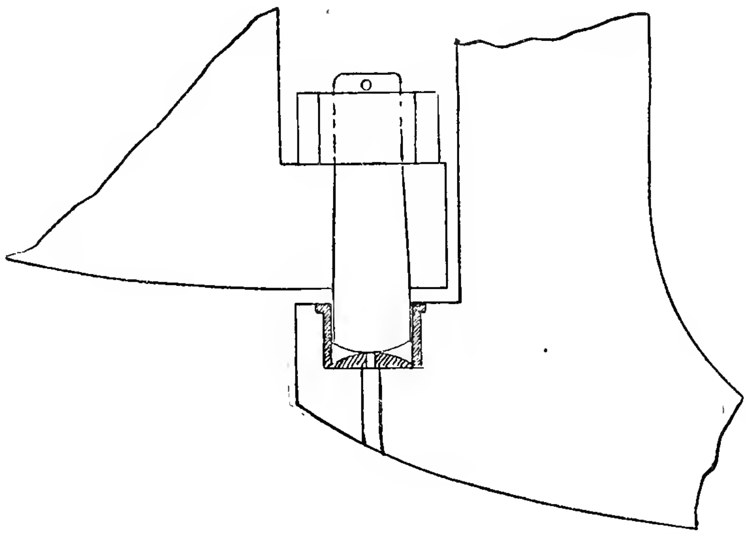
satisfactory. The weight of the rudder soon produces wearing, which is usually uneven, the friction then becoming greater than if no disc were used. The hole from the bottom of the socket to the heel of the post is to enable the disc to be easily removed.

Rudder pintles are all alike except the bottom one, which is somewhat shorter than the others, and the "lock" pintle, to which we shall refer presently. The part of each pintle which fits into the rudder frame is tapered from bottom to top, to prevent its being knocked out. On the head of the pintle a large nut is fitted, which secures it in position, any slackening tendency being guarded against by a steel pin which is driven through the pintle immediately over the nut, as shown.

To prevent accidental unshipment of the rudder, a locking arrangement must be devised. A simple plan is to make one of the pintles—preferably the top one—with a bottom collar (fig. 176).

Another point of importance for the satisfactory working of the rudder is to provide a means of limiting the turning angle, which, in ordinary cases, should not exceed 35 to 40 degrees. Referring to fig. 177, which shows a common design of stopper, it will be seen that the movement of the rudder

Fig. 175.



beyond a certain inclination is checked by widening out one of the gudgeons on the sternpost and altering the shape of the rudder stock in the vicinity, so that each surface may bear solidly on the other at the required angle. In a very large vessel two such stoppers would be needed, and they should be fitted so as to distribute the pressure equally over the sternpost. Stoppers must also be fitted on deck, the rudder movement being here controlled by stopping the quadrant or tiller arm. Where a good brake is fitted to the tiller, or the quadrant is geared on to the steam steering engine, no deck stops are necessary, the control being sufficient without them.

The foregoing is a description of a rudder such as is fitted in an ordinary cargo vessel, and it will be observed that only bare essentials are provided for. Where an owner does not object to extra expense in order to obtain greater efficiency, refinements are introduced. For instance, it is advantageous to bush the gudgeons with brass or lignum-vitæ (fig. 178), and more so to also line the pintles with brass or gun-metal (fig. 179). By these means the rudder is made to work more smoothly, and as the parts, when worn, can be renewed with little trouble or expense, a high standard of efficiency is easily maintained. The objection to carrying the weight of a rudder on the bottom gudgeon has already been referred to. This has sometimes been overcome by causing the rudder to bear on several or on all the gudgeons, circular discs or washers of white metal being inserted between the rudder lugs and the gudgeons for this purpose. Another plan is to fit solid washers, cone-shaped at bottom, into each gudgeon, with pintles having tapered points to suit. The weight of the rudder is thus distributed over all the gudgeons, and there can be little or no side movement of the rudder. By both these arrangements, of course, more power will be required to turn the rudder than when it is supported on a footstep bearing only, but this is

Fig. 176.

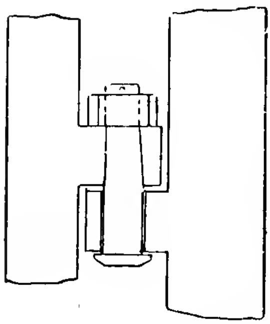
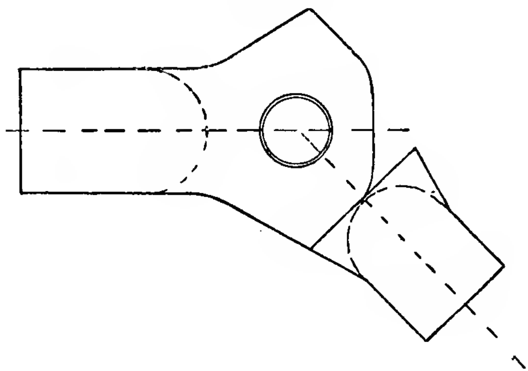


Fig. 177.



no drawback where there is an efficient steam steering gear. Occasionally rudders are fitted which do not bear on the gudgeons, the weight being taken by a thrust block inside the vessel, usually fitted at the level of the transom floor. With balanced rudders this is the invariable plan, the bottom pintle, where there is one, serving merely as a guide. The fitting of an internal bearing to a rudder of ordinary type adds to the cost, but it has the advantage of accessibility, an important consideration when dealing with working parts.

When rudders increase greatly in size and weight, it becomes necessary to devise a simple means of shipping and unshipping them without disturbing the steering gear and inboard stuffing boxes. It is customary, in such cases, to fit a coupling just under the counter, and this is found to answer the purpose admirably. Horizontal couplings, as illustrated by figs. 168, 174 and 180, are common, although others of a vertical type are sometimes fitted (figs. 181 and 181a). With such an arrangement, to unship a rudder it is only necessary to unscrew the pintle nuts, thus allowing the pintles to drop out, and to disconnect the rudder coupling. By means of block and tackle, the rudder may then be easily moved out of its usual position.

Fig. 178.

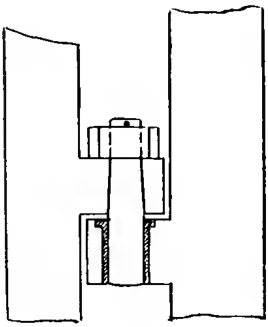


Fig. 179.

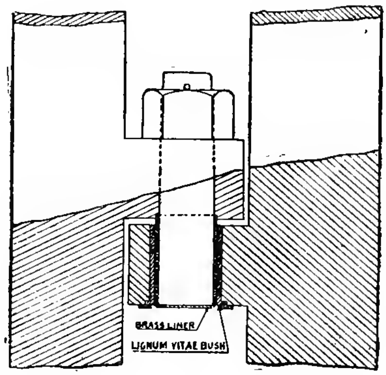
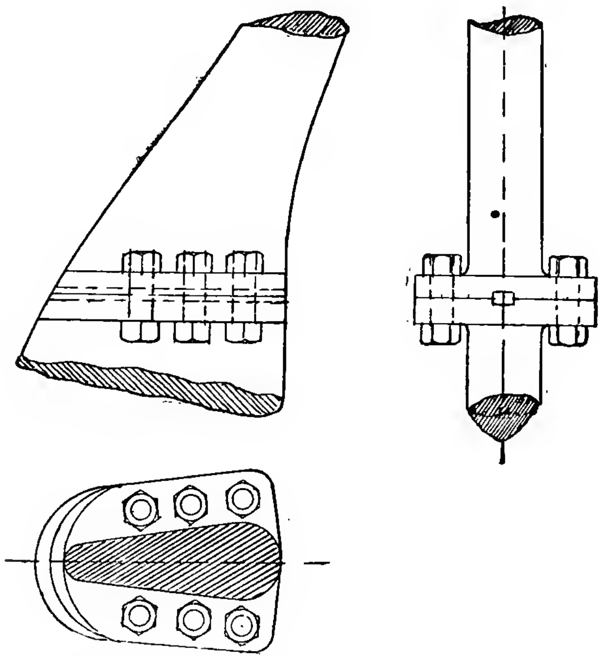


Fig. 180.



Nowadays, the rudder proper is usually formed by a single heavy plate, as previously described; another plan, once universal, and still sometimes followed, is to design the frame to the desired contour, as illustrated in fig.

182. Each side of this frame is covered by thin plating, through-riveted, the space thus enclosed being filled in solid with wood or cement. This style is

Fig. 181.

MᶜLACHLAN'S VERTICAL COUPLING

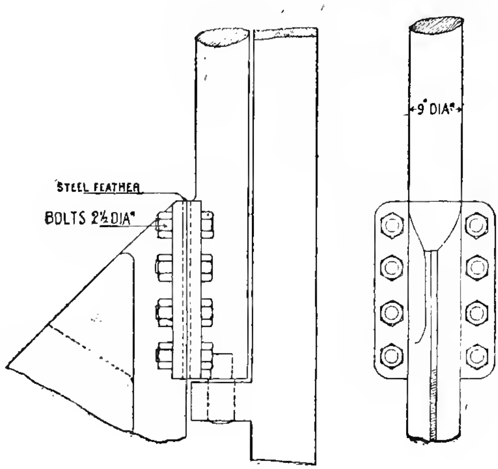
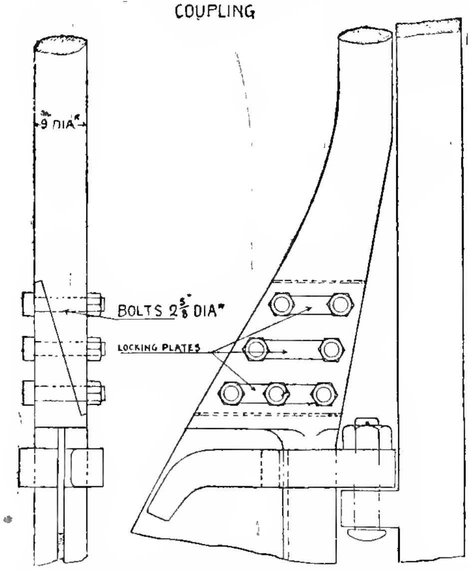


Fig. 181a.

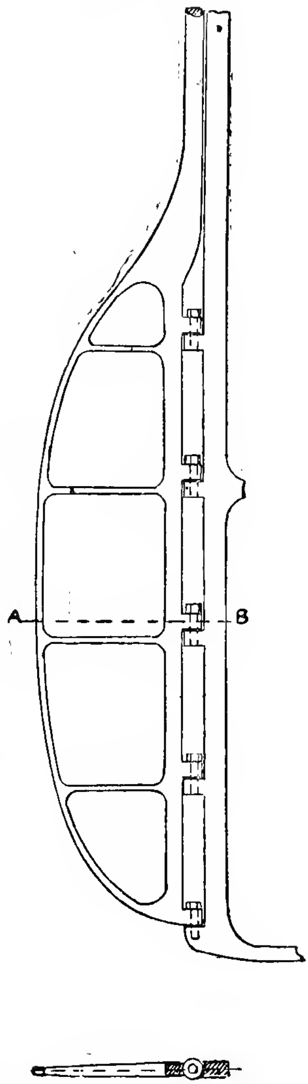
WEDGEWOOD'S SCARPHED COUPLING



not so strong as that of the single plate; it is also more liable to decay through corrosion, as the inside surfaces of the rudder-plating are obviously inaccessible for cleaning. These were the chief reasons of its abandonment in ordinary vessels in favour of the single-plate type. In special cases, such as yachts, it is still retained for its finer appearance.

The sizes of the various parts of a rudder are governed by the area and shape of the latter, and the speed of the vessel. Knowing these particulars, the twisting moment can be determined and the requisite diameter for the head of the rudder stock calculated. The aggregate sectional area of the arms supporting the single plate depends to some extent on the bending moment to be sustained, but it should be increased beyond this requirement to allow for shocks from the sea, to which the rudder in stormy weather may be subjected. The sizes of the pintles should also be sufficient to withstand these shocks and provide for wear and tear, considerable at these parts. The strength of the coupling joint must be equal to that of the stock. This entails flanges of considerable thickness and a sufficient number of coupling bolts, the moment of whose aggregate strength about the rudder axis should be equal to the twisting moment, and, therefore, to the torsional strength of the rudder stock.

Fig. 182.

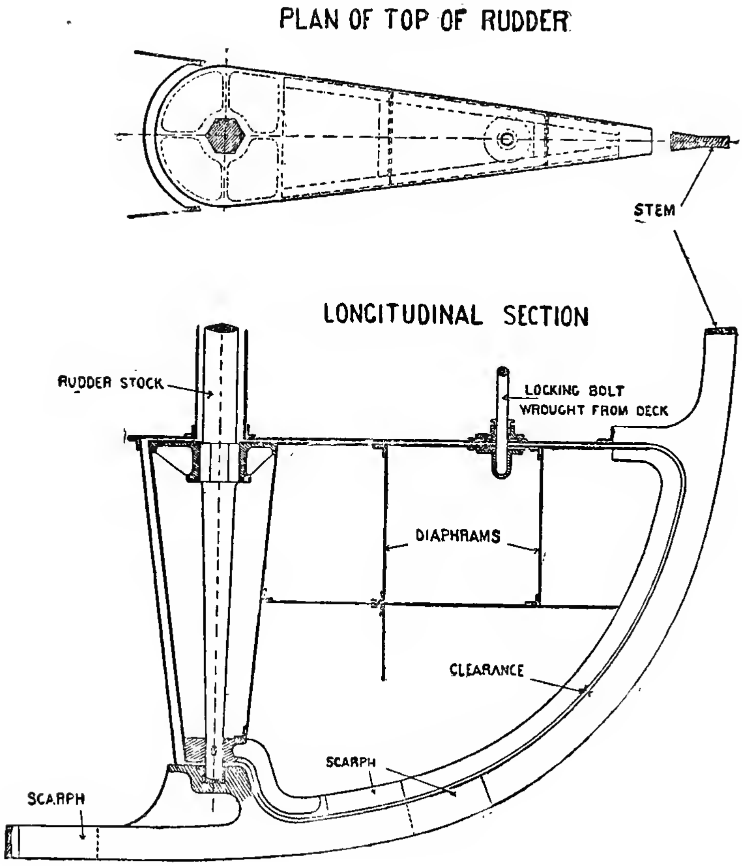


SECTION THROUGH A B.

These are the principles which must be followed in making detailed calculations. Of course, if a ship is to be built to Lloyd's Rules, such calculations, on the part of the builder at all events, are unnecessary, as detailed dimensions of rudders are provided in carefully compiled tables. In these the diameter of rudder stock is given for various speeds under numbers which represent the product of the total area of the rudder in square feet abaft the centre line of the pintles, and the distance in feet of the centre of gravity of this area abaft the same line.

As previously remarked, rudders are sometimes fitted at the fore ends of vessels, such, for instance, as have to navigate channels too confined to turn in. These rudders are usually designed to come inside the line of the stem, and to follow the shape of the vessel, being thus more or less buoyant (fig. 183). The rudder stock is carried to the weather-deck and worked by a simple hand-gear. As a bow rudder is mainly for emergency purposes, when not in use it is locked in a fore-and-aft position by means of a strong bolt.

Fig. 183.



CHAPTER VII.

Equilibrium of Floating Bodies: Metacentric Stability.

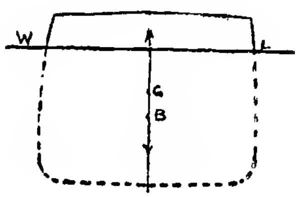
FROM our considerations in Chapters I. and II., we know something of the forces in operation when, as depicted in fig. 184, a vessel is floating freely and at rest in still water. We know, for instance—

1. That the total upward forces, or buoyancy, must equal the total downward forces or weight;

2. That the resultant of the downward forces acts through *G*, the centre of gravity of the weights, and the resultant of the upward forces through *B*, the centre of gravity of the displaced fluid, already defined as the centre of buoyancy.

It is now necessary to note that these two equal and opposite resultant

Fig. 184.



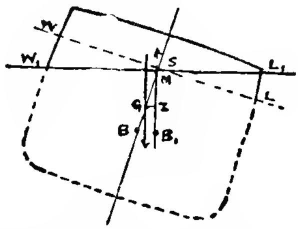
forces must act in the same vertical line, for, if the lines of action did not coincide, a turning moment would be in operation to disturb the equilibrium.

Now, suppose an external force to act upon the vessel and cause her to heel over, as shown in fig. 185. No weights have been added, therefore the displacement is unchanged, and the volume lifted out of the water on one side must be counterbalanced by the volume immersed on the other; that is, the wedges W_1SW and L_1SL are equal.

As the immersed body is now altered in form, the centre of buoyancy is no longer at *B* but takes up some new position B_1; and as there has been no change in the disposition of the weights, the centre of gravity *G* is not altered in position. The two equal resultant forces act down through

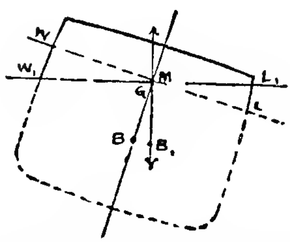
G, and up through B_1 respectively, their lines of action having a perpendicular distance GZ between them, as drawn in the figure. The turning moment acting on the vessel obviously tends to restore her to the original

Fig. 185.



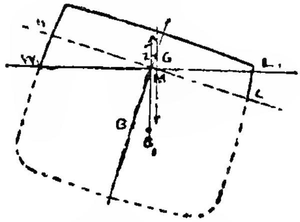
position, and she is therefore said to be in stable equilibrium. Next, suppose the centre of gravity to be raised from the position in fig. 184 say, by pumping out a ballast tank, and by putting a quantity of cargo

Fig. 186.



into the 'tween decks in order to keep the displacement the same, or by some other means. First, let G become exactly coincident with M (*see* fig. 186). As before, the weight will act downwards through G, and the

Fig. 187.



buoyancy upwards in the line B_1M. The forces will therefore act in opposite directions in the same vertical line, and being equal in magnitude will neutralise each other. In this case there will be no lever tending to heel the vessel, which will not tend to depart from its inclined position. The condition is said to be one of neutral equilibrium. Now, let G be raised above M. A glance at fig. 187 will show what will happen if the vessel be inclined as before. The two resultant forces will act in different lines, causing a heeling moment to be in operation on the vessel. There is, however, a very important difference between this heeling moment and that existing when G was below M, the tendency being now, not to right the vessel, but to incline her further from the initial position. With G above M, therefore, the vessel, when in the upright position, is in unstable equilibrium.

We thus see that the relation between the points G and M in a floating vessel entirely determines the nature of her equilibrium. M is called the metacentre from its being the *meta* or limit beyond which the centre of gravity G must not rise, if a condition of stability is to be maintained. It may be defined as follows :—

DEFINITION OF TRANSVERSE METACENTRE.—*If a vessel be floating upright at rest and in equilibrium, at a certain draught, and be then inclined through a very small angle, the point in which the vertical line through the new centre of buoyancy intersects the middle line of the ship, is called the transverse metacentre at that draught.* For every draught there is, in ordinary vessels, a different position of metacentre. The point also changes with every inclination from the upright. It is usual, however, and sufficiently correct for practical purposes, to assume it as fixed for inclinations up to 10 or 11 degrees. This is important, as within these limits, if we know the distance GM, we can determine the vessel's righting power, since—

$$\left.\begin{array}{r}\text{Moment of Statical Stability in foot-}\\ \text{tons at any angle } \theta\end{array}\right\} = W \times GZ = W \times GM \times \text{Sin } \theta,$$

W, the displacement, being given in tons, and GZ or GM in feet.

This is known as metacentric stability, GM being called the metacentric height. It must be borne in mind that this method applies only up to the angles above given; beyond these it is unreliable, as M changes rapidly in position, and GM has no longer its initial value, which is the only one that is used by the metacentric method. Further on we shall see, when considering actual curves of stability, that in many cases considerable error would be involved, even at moderate angles, by using the above formula for calculating the moment of stability.

A knowledge of a vessel's metacentric height is, however, useful for many purposes. It is an excellent guide, for instance, for determining whether or not a vessel may be safely shifted in harbour, or whether ballast tanks may be run up, or, in the case of a vessel carrying oil in bulk, how the loading of cargo should be proceeded with. In conducting the first of these operations, there need be no inclination from the upright exceeding that for which the moment of stability may be written—

$$W \times GM \times \text{Sin } \theta,$$

so that in order to shift the vessel with confidence, it is only necessary to make sure that the value of GM is sufficient. In the two last operations GM should be great enough to allow for the reduction in its value due to the presence of free liquid in the vessel. We shall return to this point again. Besides the foregoing, if the vessel be of known type, the metacentric height will furnish a good basis from which to predict the probable nature of her stability at large angles of inclination.

The great importance of the points G and M will now be manifest, and a shipmaster *ought to know* for every condition of lading of his vessel in which she may have to put to sea, what GM or metacentric height he has available.

In considering these two centres, the influences controlling the position of each should be carefully noted. Obviously, G is fixed by the distribution of the weights, and we shall show presently how it may be determined in any given case. The point M, however, is not affected by the weight distribution, but only by the underwater volume of the vessel, and by the shape of the waterplane. This appears from the formula that gives the height of the point relatively to the centre of buoyancy, which may be written—

$$\left.\begin{array}{c}\text{Height of transverse metacentre}\\ \text{above centre of buoyancy}\end{array}\right\} = BM = \frac{I}{V},$$

where I is the moment of inertia of the waterplane about its middle line as axis, and V the volume of displacement.

The numerator of the right-hand member of this equation may be explained in a popular way, as follows:—Imagine the area of the whole waterplane to be divided into an infinite number of parts, and the distances of the centres of these elements from the middle line ascertained; then, if each of these small areas be multiplied by the square of its distance from the axis, and the sum of the products be taken, the result will be the moment of inertia required.

Although it involves some calculation to obtain the above moment of inertia in the case of an ordinary-shaped vessel, owing to the varying nature of the boundary line of the waterplane, it may be quickly obtained for any figure of simple form such as a square, a circle, or a triangle, as established formulæ are then available. We have a case in point in a floating box-shaped vessel. Here the outline of the waterplane is a rectangle, and the moment of inertia of this figure about the major axis is $\frac{LB^3}{12}$, where L is the length of the vessel, and B the breadth.

Applying the formula for the height of the transverse metacentre above the centre of buoyancy we have—

$$BM = \frac{I}{V} = \frac{\frac{L \times B^3}{12}}{L \times B \times D} = \frac{B^2}{12\,D},$$

D being the mean draught. If the actual dimensions of the vessel be—length, 150 feet; breadth, 30 feet; draught, 15 feet; then—

$$BM = \frac{30 \times 30}{12 \times 15} = 5 \text{ ft.}$$

Almost as simple a case occurs when the vessel is of constant triangular section with the apex down. The waterplane is, as before, a rectangle, so that the expression for I is unchanged, but the displacement is obviously only half the previous value, and we now have—

$$BM = \frac{\frac{L \times B^3}{12}}{\frac{L \times B \times D}{2}} = \frac{B^2}{6D}.$$

We therefore note that a floating vessel of this form has its transverse metacentre at twice the height above the centre of buoyancy of another having a rectangular section, the extreme dimensions in each case being the same. Moreover, in a vessel of triangular section, the centre of buoyancy is at a greater height above the base line than in the other case, so that the absolute height of the metacentre is, on this account, still further increased. Now, the forms of the 'midship sections of ordinary ship-shaped bodies lie between the two extreme cases just considered, and, neglecting for the moment the influence of tapered lines, the general effect of change of design upon the position of the transverse metacentre may be grasped. It is important to note in the above formula that BM is independent of the length, while the breadth appears in the second power. This shows the influence of breadth on stability, and explains why broad shallow vessels have always high metacentres.

A unique case occurs where the vessel is a floating cylinder with its axis horizontal. In ordinary vessels the metacentre, as we have seen, may be considered as a fixed point only for one draught. In this case, however, the vertical through the centre of buoyancy will intersect the middle line at the same point at all draughts. This will be apparent, if we consider that, since the immersed section is part of a circle, a normal to any waterline through its middle point will pass through the centre of buoyancy, and intersect the middle line of the vessel at the centre of section. Thus, for a vessel of cylindrical section, there is only one position for the transverse metacentre.

In applying the formula $BM = \frac{I}{V}$ to ship-shaped bodies, the work, as already stated, mainly consists in obtaining the value of I. In actual calculation it is usual to divide the waterplane into an even number of equal parts, suitable for the application of Simpson's First Rule, to treat the cubes of the ordinates, measured at the points of division, as ordinates of a new curve, and find the area of the latter in square feet, two-thirds of the quantity so obtained being the moment of inertia of the waterplane about the middle line. To obtain the value of the height of the transverse metacentre above the centre of buoyancy, this moment of inertia, as we have seen, must be divided by the volume of the ship's displacement in cubic feet up to the waterplane or draught considered. As practical examples of the foregoing, and in order to impress the method upon us, we shall calculate the value of *BM* in two particular cargo vessels, both of modern type. The first is a small deadweight carrier of full co-efficient, having the following dimensions:—length, 275 feet; extreme breadth, 39 feet, 6 inches; moulded depth, 20 feet, 3 inches; the load draught is 18 feet, 9 inches, and the displacement 4535 tons. We shall deal with the vessel when in this condition. The work is given in the table below. In the first and second columns we have the numbers of the half ordinates of the load-waterplane, reckoning from the after end, and their breadths as measured at the points of division; in the third column are tabulated the cubes of these half ordinates, and in the fourth and fifth, Simpson's Multipliers and the products of these multipliers with the cubes, respectively.

No. of ½ Ordinates.	½ Ordinates.	(½ Ordinates).^{3}	Simpson's Multipliers.	Functions of (½ Ordinates).^{3}
1	—	—	½	—
1½	10·2	1061	2	2122
2	16·3	4331	1½	6496
3	19·0	6859	4	27436
4	19·5	7415	2	14830
5	19·5	7415	4	29660
6	19·5	7415	2	14830
7	19·5	7415	4	29660
8	19·5	7415	2	14830
9	19·3	7189	4	28756
10	14·9	3308	1½	4962
10½	8·5	614	2	1228
11	—	—	½	—
				174810

$$\left.\begin{array}{r}\text{Height of transverse metacentre}\\ \text{above centre of buoyancy}\end{array}\right\} = BM = \frac{174810 \times 27{\cdot}5 \times 2}{3 \times 3 \times 4535 \times 35} = 6{\cdot}75 \text{ feet.}$$

It will be observed that the figure 3 appears twice in the denominator of the expression for the value of *BM*, once as required by Simpson's Rule, and once for the moment of inertia calculation.

The other vessel chosen for illustration is of somewhat finer form, and much larger. Her dimensions are—length, 469 feet, 4 inches; breadth, extreme, 56 feet; depth, moulded, 34 feet, 10 inches. This vessel at a draught of 27 feet, 6 inches, has a displacement of 15,814 tons. We shall find the value of *BM* at this draught, arranging the work in tabular form, as in the previous case—

No. of ½ Ordinates.	½ Ordinates.	(½ Ordinates).³	Simpson's Multipliers.	Functions of (½ Ordinates).³
1	—	—	½	—
1½	13·0	2197	2	4394
2	21·5	9938	1½	14907
3	27·5	20797	4	83188
4	27·9	21717	2	43434
5	27·9	21717	4	86868
6	27·9	21717	2	43434
7	27·9	21717	4	86868
8	27·9	21717	2	43434
9	27·0	19683	4	78732
10	18·8	6644	1½	9966
10½	10·2	1061	2	2122
11	—	—	½	—
				497347

As before—

$$B\,M = \frac{497347 \times 46{\cdot}93 \times 2}{3 \times 3 \times 15814 \times 35} = 9{\cdot}36 \text{ feet.}$$

APPROXIMATE METHODS FOR FINDING $B\,M$.—If in the two preceding examples the vessels were treated as of rectangular form of the same extreme dimensions, and the rule applied, we should get for the small vessel—

$$B\,M = \frac{39{\cdot}5 \times 39{\cdot}5}{12 \times 18{\cdot}58} = 7 \text{ feet;}$$

and for the large one—

$$B\,M = \frac{56 \times 56}{12 \times 27{\cdot}25} = 9{\cdot}6 \text{ feet.}$$

The draught in each case is reduced by the depth of a flat keel. These results are sufficiently near the actual values to suggest the possibility of framing an approximate rule for readily obtaining the height of the metacentre for ordinary vessels, but employing the formula as for box-shaped vessels, with factors or co-efficients introduced to make up the differences between the types. Now, we know that the moment of inertia of a waterplane of rectangular shape about the middle line is $\frac{B^3\,L}{12}$, where B is the full breadth and L the length; this may also be written—

$$I = C_1 \times B^3 \times L, \text{ where } C_1 = \frac{1}{12}, \text{ or } {\cdot}08\dot{3}.$$

For ship-shaped load waterplanes of moderately fine form, C_1* will vary from ·05 to ·055; and where they are of full form, from ·06 to ·065. Thus, we are able to arrive at a ready expression giving the value of the numerator in the formula for $B\,M$. The denominator may be similarly

* These co-efficients are from Sir Wm. Whyte's *Manual of Naval Architecture*, to which the reader is referred for figures applying specially to war vessels.

treated, as the volume of a ship's displacement may always be written—$V = L \times B \times D \times C_2$, where C_2 is a co-efficient varying with the form. Using these approximations, we get—

$$BM = \frac{I}{V} = \frac{C_1 B^3 L}{C_2 L B D} = \frac{C_1}{C_2} \times \frac{B^2}{D} = k\frac{B^2}{D}.$$

For many classes of merchant vessels $k = \cdot 09$, while in vessels of full form it may become as low as ·08, or even less; in fine ships, such as yachts, k may rise to ·15.

As a test, let us apply the approximate formula to the two examples for which we have made detailed calculations. For the small vessel, which has full ends, $k = \frac{C_1}{C_2} = \frac{\cdot 0635}{\cdot 787} = \cdot 0806$; and therefore—

$$BM = \cdot 0806 \times \frac{39\cdot 5 \times 39\cdot 5}{18\cdot 58} = 6\cdot 77 \text{ feet.}$$

In the larger vessel, the ends ot the load waterplane and the underwater body are both finer, and making due allowance—

$$k = \frac{C_1}{C_2} = \frac{\cdot 063}{\cdot 770} = \cdot 0818,$$

$$\text{and } BM = \cdot 0818 \times \frac{56 \times 56}{27\cdot 25} = 9\cdot 40 \text{ feet.}$$

Values of height of metacentre above centre of buoyancy thus obtained are therefore seen to approach the actual figures very closely.

In order to fix the position of the metacentre in the vessel, it is necessary to know the height of the centre of buoyancy. In approximate calculations, for vessels of ordinary form, this value may be taken as varying between $\frac{8}{20}$ and $\frac{9}{20}$ of the mean moulded draught, measured downwards from the waterline, the latter figure being used for full vessels. For detailed calculations, the position of the centre of buoyancy, as obtained from correct drawings of the vessel, must, of course, be employed.

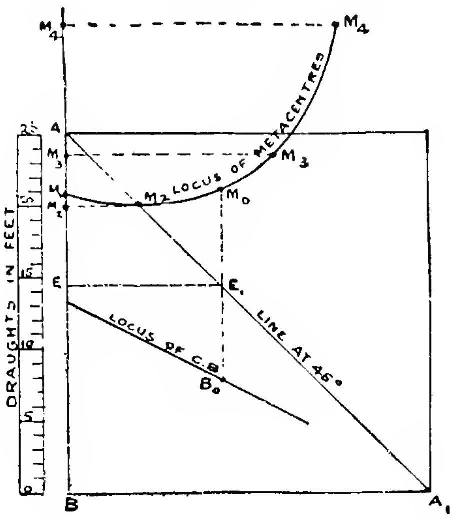
While it is most important to know the position of a vessel's transverse metacentre when floating at her load draught, it is frequently necessary to know it for other draughts. For some classes of vessels the launching condition is a critical one, and the amount of GM available then should be known. Cases are on record of vessels capsizing through deficient stability, while being launched.

Another important condition for which the metacentric height should be known is that called "light-ship," which means that the vessel is complete, including machinery, but is without cargo or bunker coal. This condition forms an excellent basis from which to calculate the value of GM for the vessel when laden with any kind of cargo.

Still another condition calling for special consideration is that when in ballast. Modern cargo vessels frequently perform voyages in ballast trim, and calculations should be made to find the disposition of ballast which will give a value of GM, ensuring the good behaviour of the vessel at sea.

Thus we have, including the loaded one, four conditions for which it is essential to know the positions of the transverse metacentre and the centre of gravity. To enable us to find the former quickly at any draught, a diagram is constructed showing the change in the position of M with change in draught. (The centre of gravity must be dealt with specially, as we shall show afterwards). The curve of metacentres is usually plotted on a diagram, such as fig. 188, on which the curve of centres of buoyancy is also drawn—the distance between the two curves at any point being the value of BM at the corresponding draught. In plotting the curve of metacentres, the procedure is the same as for the curve of centres of buoyancy, which we may assume to be already

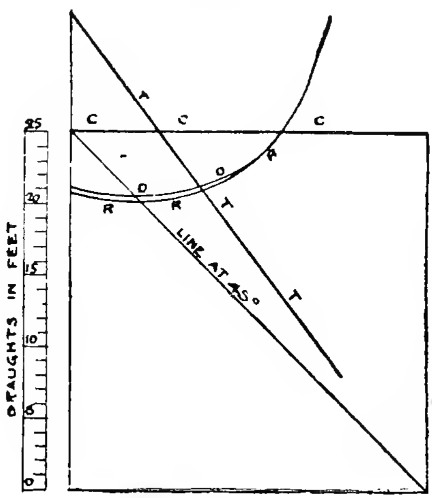
Fig. 188.



plotted. That is to say, referring to fig. 188, the height of M for various draughts is calculated and spotted off on AB. Then each of these points M_1, M_2, M_3, etc., is translated out horizontally, a distance equal to that between the load draught and the draught to which it refers, and a curve is drawn through them. To complete the diagram, a line AA_1 at 45° to the vertical AB is drawn from the point A, where the load waterplane intersects AB. To obtain, now, from such a diagram, the value of BM at any draught, BE say, it is only necessary to draw a horizontal line at that draught to intersect the line AA_1 at some point E_1, and to draw through the latter point a vertical line to the curves of buoyancy and metacentres at B_0 and M_0. It will be clear, after a little consideration, that B_0M_0 is the height of metacentre above the centre of buoyancy corresponding to the draught BE.

In fig. 189 are shown, in one diagram, curves of metacentres constructed in this way for prismatic floating bodies having cross sections of rectangular, triangular, and circular form, marked respectively *R R*, *T T*, *C C*. The curve marked *O O* applys to a cargo vessel of ordinary form with full lines. This diagram is very instructive. It will be noticed that the locus of *M* for a vessel of triangular section is a straight line which falls as the draught diminishes—a characteristic to be found in the diagrams of fine vessels as they approach very light draughts, the immersed volume being then more or less triangular in form. The locus for a circular section, as might be expected, is a horizontal straight line, *M* coinciding with the centre of the section for all draughts. The curve for the

Fig. 189.



vessel of box form resembles that for the ordinary ship in being convex to the base; that is to say, the position of *M* at first falls, as the vessel lightens. For a box-shaped vessel, $BM = \frac{B^2}{12D}$, so that, since *B* is constant, the value of *BM* continually increases as *D* diminishes. The convex shape of the curve of metacentres is, therefore, entirely due to the fact that, at first, the centre of buoyancy falls more quickly than the value of *BM* increases. This peculiarity in the curves of metacentres of vessels of full form should be carefully noted, as we see that if the position of the centre of gravity be assumed unchanged, while a vessel rises from the load draught to another somewhat less, the initial stability will be reduced, although the "freeboard," or height of the deck above the waterplane, will be increased.

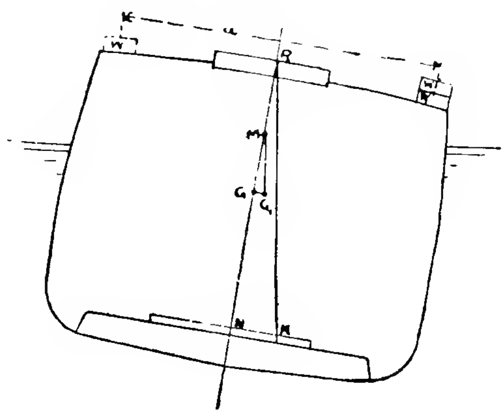
METHODS OF FINDING THE POSITION OF THE CENTRE OF GRAVITY.—A knowledge of the position of the transverse metacentre at any draught, as provided by such diagrams as the above, is of itself of no value whatever in predicting a vessel's initial stability. For example, we may have two similar vessels with identical curves of metacentres, and yet at the load draught one may have excessive initial stability, and the other be unstable. As stated already, it is the relation between the positions of *M* and *G* which is of paramount importance. In the similar vessels just referred to, the condition as to stability has been entirely influenced by the position of the latter point. In the stable vessel the heavy items have been placed low down; in the other, the opposite has been the case. This shows how much the behaviour of a vessel at sea depends on those who have charge of her stowage.

Fortunately, *G* may be determined very easily by means of an experiment, and being thus known for a given condition, the effect of a new disposition of cargo on the initial stability may be closely estimated. As well as by experiment, *G* may be found directly by calculation. This is the method employed by the naval architect in the preliminary stages of a ship's design in arranging the positions of the fixed weights. In warships, yachts, and other vessels, which sail at practically constant displacements, the estimate for the centre of gravity must be very carefully made, since any defect in the stability on completion cannot, without great expense, be corrected. In freight-carrying vessels the stowage of cargo, as we have seen, greatly influences the final position of *G*, and its "light-ship" position is therefore not of the same importance.

We cannot, in this work, elaborate in all its details the calculation method of finding *G*. It is, however, perfectly simple in principle, consisting, in fact, of a huge moment calculation, in which every item of a ship's weight, including her cargo, is multiplied by its distance from two datum lines at right angles in the middle-line plane, the centre of gravity being fixed by the values obtained when the sum of each of these systems of moments is divided by the total weight of the vessel. The datum lines are taken in the middle-line plane, as obviously, since both sides of the ship are alike, the centre of gravity must lie in that plane. The experimental method, which we now proceed to explain, briefly consists in heeling the vessel by moving a weight across the deck, observing the consequent effect upon the ship's centre of gravity, and thence deducing the value of *GM*. From this, since the position of *M* is known, the height of *G* may then be determined.

In carrying out an experiment it is important to see, in the first place, that the vessel is floating quite freely, *i.e.*, not aground at any point, or unduly hampered by neighbouring vessels, or too tightly moored to the quay. Indeed, it would be better if moorings could be unloosed altogether. The condition of the vessel should next be noted. If she be "light," the holds, ballast tanks, and bunkers, should be empty. If some coal still remain in the bunkers, it should be trimmed level so that its weight and position of centre of gravity may be determined. The weights and positions of all items which may still require to go on board to complete the vessel, such as deck machinery, small boats, etc., must also be noted. A correction is made afterwards to allow for the effect on the final position of centre of gravity by the removal or addition of these weights. If the vessel should be in loaded trim, the ballast tanks will probably be empty, but they should be carefully sounded, and any loose water pumped out. Loose water in ballast tanks or holds is particularly detrimental to such an experiment. The apparatus may now be got ready. This consists of the heeling weight, a plumb line and bob, and a straight edge. For a vessel of fair size, the weight should not be less than about ten tons, so as to ensure a definite inclination when the weight is moved across the deck. Successful experiments have been carried out with a lighter heeling weight, when the dis-

Fig. 190.



tance moved has been considerable, as, of course, the heeling moment, which consists of the product of the weight into the distance moved, may be made up of a heavy weight into a short distance, or *vice versa.*

The plumb line is usually hung in the middle-line plane of the vessel. A convenient place, when the holds are empty, is at a hatchway, the line being suspended at the upper-deck coaming, and the movements of the bob weight marked on a straight edge arranged for the purpose on top of the ceiling. With a loaded vessel this will, of course, not be possible, but a mast-stay, if screwed up tightly, will do quite well to suspend the bob weight from, and a straight edge on which to record the movements of the latter could be fitted in a suitable position near the deck. The method of conducting the actual experiment may now be described. In the first place, half the heeling weight is arranged on each side of the upper deck, at a place allowing an unobstructed passage across the ship. The vessel is in the upright position, and the point N where the plumb bob crosses the straight edge is carefully marked (fig. 190).

The weight on one side is then moved through a distance a feet across the ship, as shown, causing the plumb line to move out of the centre and take up a position RK. Now, moving the heeling weight from one side of the ship to the other causes the centre of gravity of the whole structure to move in the same direction through a distance given by the equation—
$GG_1 = \frac{w \times a}{W}$, where w is the heeling weight in tons, a the distance it is moved in feet, and W the total displacement in tons. The point G_1 is clearly the centre of gravity of the vessel in the inclined condition. Since there is equilibrium, the upward line of action of the resultant buoyant force must be in the vertical G_1M; and M being the intersection of this vertical line with the middle line, it is, by definition, the metacentre. From inspection, the triangles NRK and GMG_1 are similar, and therefore,

$$\frac{GM}{GG_1} = \frac{NR}{NK}.$$

The inclination being very small, the distance NR may be taken as the length of the plumb line. We therefore get—

$$GM = \frac{\text{length of plumb line}}{NK} \times GG_1.$$

This is the metacentric height (uncorrected), and all that remains to be done to obtain the position of the centre of gravity above the base line, is to deduct this distance from the height of the transverse metacentre above the same line at this draft, as measured from the diagram of metacentres. Corrections are afterwards made to allow for the removal of the inclining weights, and for the addition and deduction of other weights, if such be necessary to bring the vessel into the desired condition.

It is the custom with certain shipbuilders to heel their vessels for the position of the centre of gravity when in the first two of the four conditions previously mentioned, viz., the "launching" and "light" conditions. With other shipbuilders the "light-ship" condition is the only one dealt with. The information obtained for the light condition is frequently supplied to the shipmaster, to be used as a basis for making estimates of the position of the centre of gravity when in any actual service condition, such as when in ballast, or when fully loaded. In these estimates it is, of course, necessary to have a strict account of the weights of the various items of cargo or ballast put on board, with the positions of their centres above the base, or any other datum line, and to combine the whole in a moment calculation. For ballast conditions, particularly where the ballast consists of water in fixed compartments, and for loaded conditions with homogeneous cargoes, this method is quite reliable; for loaded conditions with miscellaneous cargoes, however, it is not so satisfactory, as it is to be feared that the care required to ascertain the weight and the centre of gravity of every individual item of the cargo, would not always be exercised; and without such care, the calculated position of the ship's centre of gravity might be very wide of the mark. To make sure of the condition of his vessel with regard to initial stability, when fully loaded with a mixed cargo, a shipmaster can always resort to a special heeling experiment, which we have seen to be simple in character and absolutely reliable. As a practical example of such an experiment, let us take the case of the smaller of the two vessels (*see* page 182), whose metacentre at the load draught we have found to be 6·75 feet above the centre of buoyancy, as recorded particulars of a heeling experiment carried out on her when fully loaded are available. At the time of the experiment the vessel, including the heeling weights, had a displacement of 4535 tons. The plumb line was hung from a stay and was 23 feet, 6 inches long. The inclining weights, arranged in two lots of five tons, were placed one lot on each side of the deck, at equal distances from the centre line, the distance between their centres being 33 feet. The deflection NK of the plumb line caused by moving one portion of the weight across the deck from port to starboard, was found to be $6\frac{1}{4}$ inches. As a check, the weight transferred across the deck was replaced in its old position, and an observation taken of the plumb line. It should, of course, have returned to the middle line, but scarcely did so. The other portion of the inclining weight was next shifted from starboard to port, and the resulting deflection of the plumb line noted; it was $5\frac{3}{4}$ inches. In the calculation, the mean of the observations was taken, viz., 6 inches. Now—

$$GG_1 = \frac{33 \times 5}{4535} = \cdot 0363,$$

and, therefore,

$$GM = \frac{23\cdot 5}{\cdot 5} \times \cdot 0363 = 1\cdot 71 \text{ feet}.$$

Assuming the positions of the metacentre and the centre of buoyancy to be given, we have the following:—

Height of metacentre above centre of buoyancy	= 6·75	feet.
Height of centre of buoyancy above base line	= 8·75	feet.
Height of metacentre above base line	= 15·5	feet.
Distance of centre of gravity below metacentre	= 1·71	feet.
Height of centre of gravity above base line	= 13·79	feet.

The inclining weights were removed from the ship, and this was the only correction necessary. These weights being situated above the ship's centre of gravity, the effect of their removal was to lower the latter point. Calling the displacement of the vessel including the inclining weights W, the inclining weights w, the uncorrected height of the centre of gravity above the base h, the height of the centre of gravity of the inclining weights above the base l; and taking moments about the base (the weights being in tons, and heights in feet), we have—

$$\left.\begin{array}{r}\text{Corrected height of centre}\\ \text{of gravity above base}\end{array}\right\} = \frac{W \times h - w \times l}{W - w} \text{ feet,}$$

$$= \frac{4535 \times 13{\cdot}79 - 23 \times 10}{4535 - 10} = 13{\cdot}77 \text{ feet.}$$

There was a slight fall in the position of the centre of buoyancy due to the reduction in draught, but also a slight increase in the value of BM, the height of M above the base remaining as before. The corrected metacentric height thus became $15{\cdot}5 \quad 13{\cdot}77 = 1{\cdot}73$ feet.

To estimate the change in the position of the centre of gravity due to raising or lowering weights already on board, or removing them from the vessel altogether, is now a very simple matter. Let us take a specific case:—Assuming 250 tons of cargo are to be discharged from the bridge 'tween decks of the above vessel, at a certain port, find to what extent the centre of gravity and metacentric height will be affected. Taking moments about the base line, we have—

$$\left.\begin{array}{r}\text{New height of centre of gravity}\\ \text{above base line}\end{array}\right\} = \frac{4525 \times 13{\cdot}77 - 250 \times 26{\cdot}5}{4525 - 250} = 13{\cdot}02 \text{ feet.}$$

To get the corrected GM, we must allow for the fact that the position of M is altered by the change in displacement and in the form of the water-plane. Assuming no change of trim to take place—

$$\text{Change in draught} = \frac{\text{Weight of cargo removed}}{\text{Tons per inch of immersion}}$$

$$= \frac{250}{22} = 11{\cdot}36 \text{ inches, or } {\cdot}95 \text{ feet.}$$

The new draught is, therefore, $18{\cdot}75 - 0{\cdot}95 = 17{\cdot}8$ feet. From the curve of metacentres the corresponding height of transverse metacentre above the base line is $15{\cdot}8$ feet, and thus we have—

$$GM = 15{\cdot}8 - 13{\cdot}02 = 2{\cdot}78 \text{ feet.}$$

As a further example, take the case of the large cargo steamer previously dealt with (*see* page 182). This vessel's weight, when in the "light" condition, *i.e.*, ready for sea, but with no coal or cargo aboard, is 5134 tons, the centre of gravity being 20 feet, 6 inches above the top of keel. Assuming the displacement scale and diagram of metacentres to be available, let us find the GM when laden with 10,680 tons of cargo and bunker coal distributed as follows:—6420 tons of cargo in lower holds, 3320 tons in shelter 'tween decks, and 940 tons of coal in bunkers. Tabulating our data, we have—

Light weight of ship	=	5134 tons.
Deadweight	=	10680 tons.
Total displacement	=	15814 tons.

Turning to the displacement scale we find the draught corresponding to this displacement to be 27 feet $6\frac{1}{4}$ inches to bottom of keel. The transverse metacentre above the base line at this draught (from diagram) is 23·5 feet.

To obtain the position of the centre of gravity, we must make a moment calculation, as follows:—

	Weight in Tons.	C. of G. above base.	Moments.
Light Ship	5134	20·5	105247
Cargo in holds	6420	15·5	99510
Cargo in shelter 'tween decks	3320	41·0	136120
Coal in bunkers	940	24·0	22560
	15814		363437

$$\left.\begin{array}{r}\text{Height of centre of gravity}\\ \text{above base line}\end{array}\right\} = \frac{363437}{15814} = 22{\cdot}98 \text{ feet,}$$

$$\text{so that} \qquad GM = 23{\cdot}5 - 22{\cdot}98 = {\cdot}52 \text{ feet.}$$

This value is less than would be considered safe, unless the corresponding curve of stability were particularly favourable. If we haven't got this information, it will probably be considered desirable to remove some of the cargo from the shelter 'tween decks into the main 'tween decks, so as to lower the centre of gravity 4 or 5 inches. If 10 feet be the distance through which such cargo may be lowered, the quantity affected is given by the equation—

Weight of cargo to be lowered × 10 = Displacement × fall in centre of gravity.

Substituting values we obtain—

$$\text{Weight to be lowered} = \frac{15814 \times {\cdot}33}{10} = 522 \text{ tons.}$$

The new position of centre of gravity will be $22{\cdot}98 - {\cdot}33 = 22{\cdot}65$ feet above the base line, and the value of GM ${\cdot}52 + {\cdot}33 = {\cdot}85$ feet.

The burning out of the bunker coal has often an important effect on the stability. Let us find what it would be in the present case. Assuming the weight of coal to be 940 tons, and its centre of gravity 24 feet above the base, the effect of burning out the coal in the present instance would be to lower the centre of gravity. Taking moments about the base line—

$$\left.\begin{array}{r}\text{New height of}\\ \text{centre of gravity}\end{array}\right\} = \frac{15814 \times 22{\cdot}65 - 940 \times 24}{15814 - 940} = 22{\cdot}56 \text{ feet.}$$

The fall in draught of water would be—

$$\frac{940}{53} = 17\tfrac{3}{4} \text{ inches,}$$

53 being the tons per inch of immersion at the load draught. The new draught would therefore be $27'\ 6\frac{1}{4}''\quad 1'\ 5\frac{3}{4}'' = 26'\ 0\frac{1}{2}''$. From the diagram,

the height of metacentre above base at this draught is 23·25 feet; so that under the assumed conditions—

$$GM = 23{\cdot}25 - 22{\cdot}56 = {\cdot}69 \text{ feet.}$$

The burning out of the coal would thus reduce the initial stability.

As there would be draught to spare, it might be considered desirable to run up some water ballast in order to bring GM to about its previous value. Let sufficient water be supposed admitted to lower the vessel's centre of gravity 3 inches. Then, assuming the centre of gravity of the ballast to be 2 feet above the base, and taking moments about that line, we have, representing the weight of the ballast by B—

$$(14874 + B)\ 22{\cdot}31 - B \times 2 = 14874 \times 22{\cdot}56$$

from which $$B = \frac{14874 \times 22{\cdot}56 - 14874 \times 22{\cdot}31}{20{\cdot}31} = 183 \text{ tons.}$$

The added ballast would increase the draught $\dfrac{183}{53} = 3\frac{1}{2}$ inches, and from the diagram we find that the metacentre would rise half-an-inch, therefore—

$$\text{New value of } GM = {\cdot}69 + {\cdot}25 + {\cdot}04 = {\cdot}98 \text{ feet.}$$

APPROXIMATE METHOD OF CALCULATING THE EFFECT IN THE INITIAL STABILITY DUE TO ADDING OR REMOVING WEIGHTS OF MODERATE AMOUNT.—In order to make estimates of a vessel's metacentric height, or initial stability, like the foregoing, considerable data must be available. In many instances a shipmaster may not have this information. By an approximate rule* he may, however, still find the effect on the initial stability of raising or lowering, adding or removing, a moderate weight, provided he knows its amount and the distance of its centre of gravity from the load-waterplane. If w be the weight in tons, and h its distance in feet from the load-waterplane, by this rule, the stability at an inclination of θ degrees will be affected to the extent $w \times h \times \text{Sin } \theta$ foot tons, the correction being a decrease if w is added at some point above the waterplane, or removed from some point below it, and an increase if the conditions be the opposite of these. Thus, the effect of running in 183 tons of ballast on the initial stability of the vessel referred to above, will, by this approximate method, be to increase it by the amount—

$$(26{\cdot}04 - 2)\ 183 \times \text{Sin } \theta \text{ foot tons} = 4399 \text{ Sin } \theta \text{ foot tons.}$$

By the exact method, viz., Righting Moment $= W \times GM \times \text{Sin } \theta$, the increase is the difference between the stability after adding the ballast and that existing before, that is—

$$(15057 \times {\cdot}98 - 14874 \times {\cdot}69) \text{ Sin } \theta \text{ foot tons} = 4492 \text{ Sin } \theta \text{ foot tons,}$$

so that the approximation is a good one.

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* See a paper by Sir Wm. Whyte in volume XIX. of the Transactions of the Institution of Naval Architects.

As a further example, suppose 200 tons of deck cargo to be taken on board at 11 feet above the load-waterline. The effect will be to reduce the initial stability by the amount (200×11) Sin θ foot tons, which at 10 degrees is $2200 \times \cdot 1736 = 381 \cdot 9$ foot tons. If the cargo were removed instead, the initial stability would be increased by the same amount. This method only gives reliable results when the addition or removal of the weights causes no appreciable change in the form of the load-line.

SAFE MINIMUM GM.—The question of a minimum value of GM has been the subject of much debate and difference of opinion. Those who have favoured a large value have been confronted with the fact that great stiffness conduces to bad behaviour at sea, as will appear when we come to the subject of rolling. On the other hand, a very small value indicates a crank vessel, and may mean, although not necessarily so, an altogether unsafe one. The only secure manner of dealing with vessels in this respect is to compare them with others whose performances at sea are known, and to adopt values of metacentric heights thus suggested.

In designing the various classes of warships, the value of the metacentric height must be carefully considered from the point of view of what may be expected from each vessel on active service. This is specially necessary, since GM cannot be readily altered after the completion of the vessel, the weights being then fixed and the displacement more or less constant. If the only consideration is to obtain a steady gun platform, GM should be small, as a minimum of rolling motion at sea is thereby assured. If, however, other considerations intervene, such as a liability to lose a portion of the inertia of the load-waterplane when in action, to which some war vessels, owing to their design, are subject, the steady gun platform must be sacrificed and a sufficiently large initial value of GM provided to meet all eventualities.

It is beyond our province to fully discuss the subject as affecting warships, but coming to the case of trading merchant vessels, there seems to be a consensus of opinion in favour of limiting the minimum value of GM in steamers of about medium size to one foot when filled with a homogeneous cargo, which just brings them to the load-waterline. Cases are on record of vessels which have given a good account of themselves with smaller metacentric heights, when loaded as above. In one oft-quoted instance, the GM was as low as ·6 feet, yet the vessel proved herself in every way a good sea-boat. The natural feeling, however, is to have a margin on the side of safety, and this is considered to be provided in ocean-going steamers when the metacentric height has the minimum value given above.

For sailing-ships, of course, a much higher value of GM is requisite, in order that they may not be unduly heeled when under canvas. The best authorities give 3 to $3\frac{1}{2}$ feet as a minimum value, and where a homogeneous cargo will not admit of this, ballast should be carried.

QUESTIONS ON CHAPTER VII.

1. A vessel is floating at rest in still water; discuss the changes in the character of the equilibrium as the centre of gravity is raised.

2. Define the transverse metacentre; write down the formula for the height of the transverse metacentre above the centre of buoyancy, and state its numerical value in the case of a rectangular vessel 38 feet broad, floating on even keel at a draught of 20 feet.
Ans.—6·01 feet.

3. What is metacentric stability? A vessel of 4000 tons displacement has a metacentric height of 18 inches; calculate the stability in foot tons at an inclination of 10 degrees.
Ans.—1041·6 foot tons.

4. The equidistant ordinates of a vessel's load-waterplane, measured on one side of the middle-line at intervals of 16 feet, are—·2, 6·8, 9·6, 10·0, 10·2 10·0, 9·9, 8·8, and 1·8 feet, and half ordinates introduced at the ends are 3·8, and 7·3 feet, respectively. Find the height of the transverse metacentre above the centre of buoyancy, the displacement being 530 tons.
Ans.—3·46 feet.

5. A prism of circular section floats in still water with its axis horizontal. Show that the metacentre is at the centre of the section for all draughts.

6. Two prisms of rectangular and triangular section, respectively, float on even keel at the same draught; they are also the same breadth at the waterline. Show that the height of metacentre above the centre of buoyancy in the first case is half that in the second. If the breadth be 35 feet and the draught 18 feet, what are the values in the two cases?
Ans.—5·67 feet; 11·34 feet.

7. A raft is supported by, and rigidly attached to, two rectangular pontoons placed parallel to each other and 6 feet apart from centre to centre. Each pontoon is 4 feet wide and 2 feet deep, and floats half immersed when the raft is laden. Calculate the metacentric height, assuming the centre of gravity of raft and lading to be 3 feet above the waterline.
Ans.—6·83 feet.

8. Explain an approximate method of calculating the height of transverse metacentre above the centre of buoyancy, and work out the numerical value in the case of a full-formed cargo steamer of 48 feet beam and 25 feet draught.
Ans.—7·36 feet.

9. How is a metacentric diagram constructed, and what are its uses? Construct such a diagram for a homogeneous rectangular prism afloat in still water, assuming it to be 30 feet broad and 20 feet deep.

10. The displacement of a vessel is 400 tons; the transverse metacentre is $5\frac{3}{4}$ feet above the centre of buoyancy, and the centre of gravity 3 feet above the centre of buoyancy. If 12 tons be moved 8 feet across the deck, find the inclination of the vessel.
Ans.—5 degrees.

11. Explain how you would find the position of the centre of gravity of a ship—
(1). By calculation.
(2). By experiment.

12. The centre of gravity of a certain cargo vessel of 4500 tons displacement is found to be 16 feet above the base; the following weights are then added, viz., 600 tons at 10 feet, 400 tons at 15 feet, and 500 tons at 11 feet above the base; the following weights are removed, viz., 600 tons from 2 feet, and 100 tons from 14 feet above the base. Calculate the new height of centre of gravity.
Ans.—16·39 feet above the base.

13. How would you estimate the change in initial stability of a vessel due to raising or lowering, adding or removing, a weight of moderate amount?

(1). Accurately.

(2). Approximately.

14. Estimate approximately the change in stability at an inclination of 10 degrees, due to running in 150 tons of ballast at 2 feet above the base, and discharging 100 tons of cargo from 30 feet above the base; the load draught to begin with being 22 feet.

Ans.—Stability increased 660 foot tons approximately.

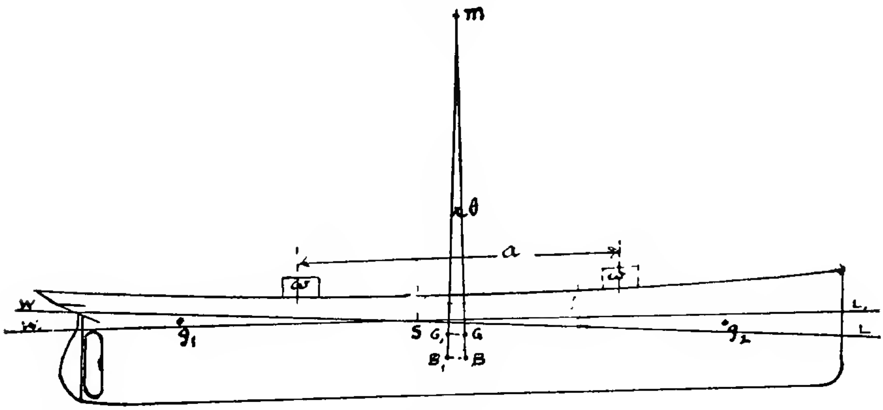
CHAPTER VIII.

Trim.

IN the previous chapter we examined the condition of vessels when heeled through small angles in a transverse direction; in the present one we propose to deal with longitudinal inclinations—that is, with the subject of trim.

The vessel in fig. 191 is supposed to be heeled to a very small angle θ in a fore-and-aft direction, by transferring a weight w tons from forward to aft through a distance a feet. W_1L_1 and WL represent, respectively, the lines of

Fig. 191.



flotation before and after the transference. The draught is increased at the stern and decreased at the stem, as shown. The sum of the distances LL_1, and WW_1, is called the change of trim, and the finding of this for any proposed condition of lading constitutes the trim problem. We shall endeavour to explain two methods of solution—one that ordinarily given in text books; the other not so well known.

In fig. 191, the point of intersection S of the new and the old waterlines is at about the middle of the length. It actually coincides with the centre of gravity of the waterplane W_1L_1, which, in most cases, is a little aft of the middle of the length; however, it is sufficiently correct to assume

SL_1 and SW_1 to be equal, and this simplifies the work somewhat. Referring to the figure, obviously—

$$\text{Change of trim} = WW_1 + LL_1 = (W_1S + SL_1) \tan \theta.$$
$$= \text{length of load line} \times \tan \theta \quad (1).$$

Now the movement of the weight w causes the centre of gravity of the vessel to be drawn aft through a distance GG_1 given by the equation $GG_1 = \frac{w \times a}{W}$ (W being the displacement in tons), and the resultants of the vertical forces of weight and buoyancy, when the ship is once more in equilibrium, act through the point G_1. This is indicated in fig. 191, as also the previous line of vertical forces through G, corresponding to the condition before the shifting of w. Their point of intersection m is called the longitudinal metacentre, and the distance Gm the longitudinal metacentric height, the latter being of considerable importance in trim problems, as we shall see presently. The angle between the two lines is clearly θ, the inclination between the water lines, and therefore—

$$GG_1 = Gm \tan \theta,$$

or $$Gm \tan \theta = \frac{wa}{W}$$

so that $$\tan \theta = \frac{wa}{W \times Gm}.$$

From (1) $$\tan \theta = \frac{\text{Change of trim}}{\text{Length of load-line}}.$$

Calling the length of load-line L, and equating these two values of tan θ, we obtain the relation—

$$\frac{\text{Change of trim}}{L} = \frac{wa}{W \times Gm} \quad (2),$$

which is the ordinary formula for change of trim due to any given shift of weights already on board.

One or two simple examples will illustrate the application of this formula. Take a vessel 200 feet long of 2000 tons displacement, in which the quantity Gm is 190 feet, and find the effect on the flotation due to shifting 50 tons aft through 80 feet. Transposing a little, and introducing the quantities given, we have—

$$\text{Change of trim} = \frac{50 \times 80 \times 200}{2000 \times 190} = 2{\cdot}1 \text{ feet, say 25 inches.}$$

The draught forward will therefore be reduced, approximately, $12\frac{1}{2}$ inches, and the draught aft increased by the same amount.

Again, if it were found that the propeller tips in this vessel were showing 9 inches above the water when in the original condition, the minimum weight w required to be shifted, say, 120 feet aft, so as just to immerse the screw, would be—

$$w = \frac{1{\cdot}5 \times 2000 \times 190}{200 \times 120} = 23{\cdot}75 \text{ tons.}$$

In dealing with questions like the foregoing, and indeed, in working out any trim problem, it frequently saves time to find at the outset the *moment to alter trim one inch.* To do this it is only necessary to substitute in (2) 1 inch (or $\frac{1}{12}$ foot, since the units are in feet) for "change of trim," and after transposing, write down the equation giving the moment required. Thus,

$$\text{Moment to alter trim 1 inch} = w \times a = \frac{W \times Gm}{L \times 12} \text{ foot tons.}$$

In the preceding example—

$$w \times a = \frac{2000 \times 190}{200 \times 12} = 158{\cdot}3 \text{ foot tons,}$$

and using this figure we get the same results as before more simply. Thus, for the effect of shifting 50 tons aft through 80 feet we have—

$$\text{Change of trim in inches} = \frac{\text{Trimming moment}}{\text{Moment to alter trim 1 inch}} = \frac{50 \times 80}{158{\cdot}3} = 25.$$

In the second question, as the change of trim necessary to submerge the propeller tips is 18 inches, there must be a total trimming moment of $18 \times 158{\cdot}3 = 2849{\cdot}4$ foot tons, and as we know that w, the weight, may be moved 120 feet aft, obviously—

$$w = \frac{2849{\cdot}4}{120} = 23{\cdot}75 \text{ tons.}$$

The *moment to alter trim one inch* is thus seen to be very useful, and in order to have it ready to hand for all possible conditions as to draught, it is frequently found convenient to plot a curve of its values with variation in draught. In the calculations it is usual to employ Bm instead of Gm (*see* fig. 191) as, of course, the exact position of the centre of gravity at the various draughts is unknown, and Bm, as we shall see presently, may be readily found. The difference between Gm and Bm, however, is usually small, and for practical purposes the trim moments thus found are sufficiently accurate. The plotting of the diagram is a simple matter, and as an exercise, the reader should draw one for any case for which he may have the data.

LONGITUDINAL METACENTRE.—The only term in equation (2) calling for special explanation is Gm, already described as the longitudinal metacentric height—m being the longitudinal metacentre. The point m is obviously analogous to M the transverse metacentre, with which we are already familiar. In fact, the definition of M given on page 179 will also apply to m, if for transverse the word longitudinal be substituted. The points M and m have similar functions in respect to stability; but vessels have enormous righting power in a longitudinal direction, and detailed calculations of longitudinal stability are therefore unnecessary. The principal use of m is found, as illustrated above, in dealing with questions of trim.

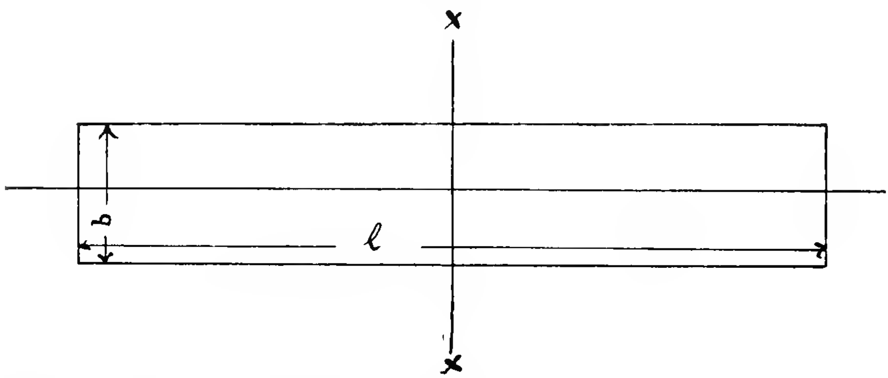
CALCULATION OF Bm (fig. 191).—As in the case of the transverse metacentre, the height of the longitudinal metacentre is first found above the centre of buoyancy. The same formula is used, viz.:—

$$Bm = \frac{I}{V},$$

but here I is the moment of inertia of the waterplane with respect to a *transverse* axis through its centre of gravity. We shall see presently, that for ordinary vessels the calculation of I is rather more laborious than in the previous case; for box-shaped or other vessels of simple form, this is, however, not so. Take, for example, a box-shaped vessel of length l, breadth b, and draught d. Fig. 192 shows the waterplane, which is, of course, a rectangle; $x\,x$, drawn at right angles to the middle line, is the axis of the moment of inertia—

$$I \text{ about } x\,x = \frac{l^3 b}{12}.$$

Fig. 192.



Since the volume of displacement is $l\,b\,d$—

$$Bm = \frac{l^3 b}{12\,l\,b\,d} = \frac{l^2}{12 d}.$$

If it be given that $l = 150$ feet, and $d = 15$ feet,

then $$Bm = \frac{150 \times 150}{12 \times 15} = 125 \text{ feet.}$$

If this vessel were of constant circular section, with its longitudinal axis in the waterplane, the numerator of the expression giving Bm would be unchanged, but the volume of displacement would be less. In such a case, we should have—

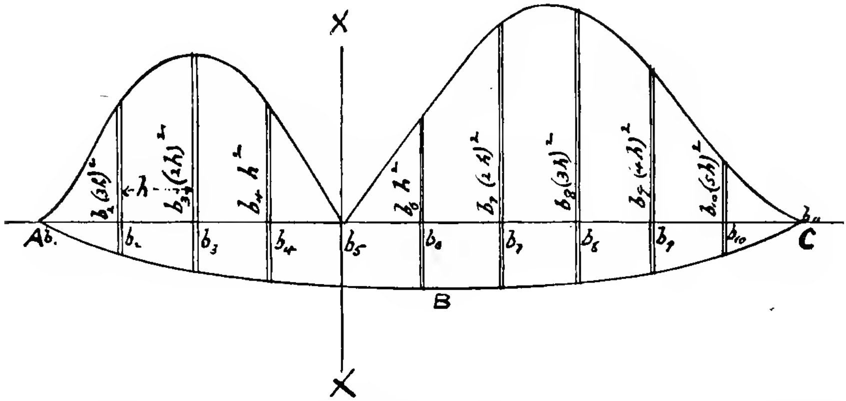
$$Bm = \frac{l^3 b}{\frac{12\,l\,b^2 \times \cdot 7854}{2}}$$

Simplifying and substituting values—

$$Bm = \frac{150 \times 150}{6 \times 30 \times \cdot 7854} = 159 \cdot 1 \text{ feet.}$$

Coming to ship forms, we find that the varying nature of the outline of the waterplane and of the form of the underwater body introduces complexity into the calculation. No simple formula is available for the moment of inertia of the waterplane. To find this quantity it is usual to divide the waterplane by ordinates in number suitable for the application of Simpson's Rule, to calculate the moment of inertia of elementary strips of area at each of these ordinates about some chosen axis, to treat these moments as ordinates of a new curve, and calculate the area of the latter. This area, subject to a further correction to be explained presently, gives the moment of inertia required. The foregoing may, perhaps, be better understood by a simple graphic explanation. In fig. 193, ABC represents a half of a load waterplane of a vessel. XX is the axis about which the moment of inertia is to be calculated, and is usually taken in the vicinity of the mid-length.

Fig. 193.



The ordinates b_1, b_2, b_3, etc., are numbered from aft, and the common interval between them is h. Calling the breadth of each little strip a, we may write—

Moment of inertia of elementary strip at b_1 about $XX = o \times (4h)^2 a = o.$

" " b_2 " $= b_2 \times (3h)^2 a = 9 b_2 h^2 a.$

" " b_3 " $= b_3 \times (2h)^2 a = 4 b_3 h^2 a.$

" " b_4 " $= b_4 \times (h)^2 a = b_4 h^2 a.$

" " b_5 " $= b_5 \times o. \ a = o.$

In the same way, moments of inertia of strips of area on the other side of XX may be found. At the points of division on the middle line AC, the the moments of inertia of the strips are erected as rectangles, the base in each case being the small breadth a, and the ordinates, the quantities o, $9 b_2 h^2$, $4 b_3 h^2$, etc. Fair curves drawn through the tops of these rectangles enclose areas as shown, which, added together, represent the moment of inertia of the plane about the chosen axis.

Now, the axis of the moment of inertia must contain the centre of gravity of the waterplane area. If $x\,x$ is not so placed, which generally would be the case, a correction must be made. This is done by means of the formula explained on page 63, viz.,

$$I_1 = I + A\,h^2$$

where I is the moment of inertia of the plane about an axis through the centre of gravity, I_1 its moment of inertia about any parallel axis $x\,x$, h the distance between these axes, and A the area of the waterplane. Applying this formula to the above case, it is necessary to find the distance h, which is obtained if the position of the centre ot gravity of the plane is known. We have already seen how to find this latter point, and, having obtained I_1, the value of the required quantity may be at once written down.

As a numerical example, take the 470 feet vessel for which we have calculated the transverse $B\,M$. The table below exhibits the full work, and scarcely calls for explanation. It may be mentioned, however, that columns 4 and 6 are introduced to determine the area of the waterplane, and the position of its centre of gravity from the assumed axis at ordinate No. 5; also that, as each ordinate of the moment of inertia diagrams (fig. 193) involves the square of the interval between the ordinates, the finding of the areas introduces the cube of the interval in the expression for the moment of inertia, as shown below.

No. of Ordinates.	Ordinate.	S.M.	Functions of Ordinates.	Levers for Moments.	Functions for Moments.	Levers for M.I	Functions of Moment of Inertia.
0	—	$\frac{1}{2}$	—	5	—	5	—
$\frac{1}{2}$	13·0	2	26·00	$4\frac{1}{2}$	117·00	$4\frac{1}{2}$	526·50
1	21·5	$1\frac{1}{2}$	32·25	4	129·00	4	516·00
2	27·5	4	110·00	3	330·00	3	990·00
3	27·9	2	55·80	2	111·60	2	223·20
4	27·9	4	111·60	1	111·60	1	111·60
5	27·9	2	55·80	0	799·20	0	
6	27·9	4	111·60	1	111·60	1	111·60
7	27·9	2	55·80	2	111·60	2	223·20
8	27·0	4	108·00	3	324·00	3	972·00
9	18·9	$1\frac{1}{2}$	28·35	4	113·40	4	453·60
$9\frac{1}{2}$	10·2	2	20·40	$4\frac{1}{2}$	91·80	$4\frac{1}{2}$	413·10
10	—	$\frac{1}{2}$	—	5	—	5	—
			715·60		752·40		4540.8
					799·20		

excess forward = 46·80

Centre of gravity of L.W.P. forward of No. 5 ordinate $= \dfrac{46{\cdot}8 \times 46{\cdot}93}{715{\cdot}60} = 3{\cdot}06$ feet.

Area of L.W.P. $= \dfrac{715{\cdot}6 \times 46{\cdot}93 \times 2}{3} = 22388$ square feet.

M.I. about axis through No. 5 ordinate $= \frac{4540 \cdot 8 \times (46 \cdot 93)^3 \times 2}{3} = 312890090.$

M.I. about axis through C.G. of L.W.P. $= 312890090 - (22388 \times (3 \cdot 06)^2)$

$= 312680458.$

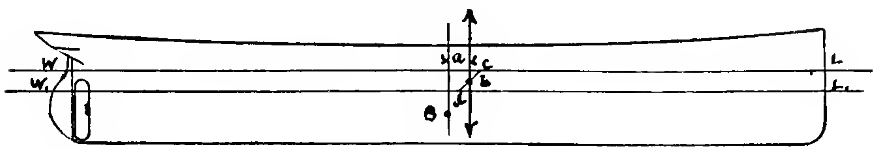
The displacement of the vessel is 15814 tons;

so that, $$Bm = \frac{I}{V} = \frac{312680458}{15814 \times 35} = 565 \text{ feet.}$$

For easy reference, values of Bm, such as the above, are usually calculated for various draughts and plotted in a diagram from which, given a draught, the corresponding Bm may be read off. Of course, for accurate trim calculations, Gm and not Bm is required; the distance between B and G should therefore be deducted from the calculated distance Bm, G being usually above B. In the above case, for instance, the centre of gravity is 6 feet above the centre of buoyancy, and, consequently—

$$Gm = 565 - 6 = 559 \text{ feet.}$$

Fig. 194.



The foregoing principles will be more clearly understood by the following trim calculations for a few actual cases:—

Example 1.—Suppose 150 tons of cargo, having its centre 80 feet before the centre of gravity of the load-waterplane, is to be discharged from the above vessel; what will be the new draughts forward and aft, it being given that the vessel to begin with is on even keel at 27 feet 6 inches?

In questions such as this, and in those involving additions of cargo, it is usual to assume, in the first place, that the removed or added weight has its centre in the vertical plane, containing the centre of gravity of the layer of volume which rises out of or sinks into the water. Under this assumed condition, a vessel will obviously rise or sink through a parallel distance. For, in fig. 194, if b be the centre of gravity of the layer through which a vessel rises by the removal of a weight w, and a be the distance of b from the vertical through B, the original centre of buoyancy of the vessel, then—

$$w \times a = \text{Moment tending to depress bow,}$$

the effect of the removal of the layer of displacement being to cause the centre of buoyancy of the vessel to move aft: and if, as assumed, the weight before removal had its centre in the vertical containing b, another moment due to removing w will also be in operation, in this case tending to raise the bow. These two moments are equal and neutralise each other, and thus the vessel rises to the waterplane $W_1 L_1$ without changing trim. On the other hand, if the vessel be floating at the waterplane $W_1 L_1$, and the weight w be added with its centre of gravity in the same vertical as b, the moment due to the increased buoyancy will be $w \times a$, and will tend to raise the bow, while that due to the added weight, being of equal amount but of opposite sense, will tend to depress the bow; so that the final effect will be to sink the vessel to the line $W L$ without change of trim.

It should be noted that, where the weights to be added or removed are moderate, b may be taken as in the vertical containing the centre of gravity of the waterplane.

Applying these principles to the working out of our question, it is assumed that the 150 tons of cargo is immediately over the centre of gravity of the waterplane, the effect of its removal therefore being to cause the vessel to rise through a parallel distance—

$$\frac{150}{\text{Tons per inch of immersion}} = \frac{150}{53 \cdot 3} = 2\tfrac{3}{4} \text{ inches, nearly.}$$

The reader already knows how to find the tons per inch at a given waterplane.

We next take account of the fact that the cargo is 80 feet forward of the assumed position. A little consideration will show that the removal of 150 tons from a point 80 feet before the centre of gravity of the waterplane, has the same trimming effect as the addition of the same weight at 80 feet abaft that point. By removing the weight from its true position, we therefore get—

Moment trimming vessel by stern = 150 × 80 = 12,000 foot tons.

Now,

$$\text{Moment to alter trim 1 inch} = \frac{W \times Gm}{L \times 12} \text{ foot tons,}$$

$$= \frac{15814 \times 559}{470 \times 12} = 1570 \text{ foot tons,}$$

so that,

$$\text{Trim by stern} = \frac{12000}{1570} = 7\tfrac{3}{4} \text{ inches, nearly.}$$

The final draughts will be—

Forward, $27'\ 6'' - 2\tfrac{3}{4}'' - 3\tfrac{7}{8}'' = 26'\ 11\tfrac{3}{8}''$;
Aft, $27'\ 6'' - 2\tfrac{3}{4}'' + 3\tfrac{7}{8}'' = 27'\ 7\tfrac{1}{8}''$.

We have assumed the change of trim to be divided equally at stem and stern. This is not quite correct. Allowance should be made for the fact that S (*see* fig. 191), the point in which the water-lines intersect, which coincides with the centre of gravity of the load-waterplane, is not usually at the mid-length, and the change of trim, forward and aft, should be allotted according to the proportion—

$$\frac{L L_1}{W W_1} = \frac{S L}{S W}.$$

In small changes of trim, however, it is not usual to proceed to this refinement, as the difference is inappreciable; in the present case, it is less than a quarter-of-an-inch.

Example 2.—A vessel 360 feet long, 48 feet broad, and drawing 26 feet aft and 20 feet forward, has to cross a bar which will only admit of a draught of 25 feet. She has a fore-peak tank of 160 tons capacity. Show by calculation whether the filling of this tank will trim the vessel sufficiently to enable her to pass over the bar. The centre of gravity of the water ballast is 166 feet forward of the centre of gravity of the load-waterplane, the tons per inch of immersion is 33, and the moment to alter trim one inch, 700 foot tons.

From the information given, we may write:—

$$\left.\begin{array}{r}\text{Sinkage, assuming ballast immediately over}\\ \text{the centre of gravity of load-waterplane}\end{array}\right\}=\frac{160}{33}=4\cdot 84,\ \text{say } 5 \text{ inches.}$$

$$\left.\begin{array}{r}\text{Trim by head due to actual position of}\\ \text{ballast}\end{array}\right\}=\frac{166\times 160}{700}=38 \text{ inches.}$$

Assuming waterplanes to cross at mid-length—

$$\begin{aligned}&\text{New draught forward} = 20'\ 0'' + 5'' + 1'\ 7'' = 22'\ \ 0'';\\ &\text{New draught aft} \quad\ \ = 26'\ 0'' + 5'' - 1'\ 7'' = 24'\ 10'';\end{aligned}$$

we thus see the vessel may, with care, be safely navigated across the bar.

In the two previous cases, the weights causing the change of trim are small in comparison with the total displacements; had they been large, it would have been incorrect to assume b of the parallel layer to be in the vertical containing the centre of gravity of the original waterplane. Its true position is obviously somewhere in the line joining the centres of gravity of the two planes enclosing the layer; dc is this line in fig. 194. If the areas A, A_1, of the planes WL and W_1L_1, be known, the position of b may be determined from the relation—

$$\frac{db}{bc}=\frac{A}{A_1}.$$

Very little error is involved if b be taken as the mid-point of dc, and in most cases this is done.

Another point to be borne in mind is, that in dealing with large changes of trim, the plane of flotation, after movement of the weights, may have become so altered in shape as to materially affect the value of the moment of inertia, and, therefore, of the metacentric height. In actual calculations, it is customary to first approximate the trim by using the Gm given by the original waterplane; and, for a final result, to employ a mean Gm between those corresponding to the approximate and the original waterplanes. Taking an actual case, let it be required to find the draught and trim in the vessel of Example 2, after discharging 1000 tons of coal and cargo and loading 300 tons of water ballast. The reduction in displacement is 700 tons, and, assuming the weights to have their centres in the vertical plane containing *b*, the centre of buoyancy of the layer—

$$\text{Thickness of parallel layer} = \frac{700}{33} = 21 \text{ inches.}$$

b is found to be one foot forward of the centre of gravity of the original waterplane, and the leverages of the weights are measured from *b*. The work of calculating the trimming moment may be tabulated as follows:—

	Item.	Distance from *b*.	Forward Moment.	Aft Moment.
			foot tons	foot tons
Weights taken off	400 tons bunker coal	14 ft. forward		5600
	350 tons cargo from main hold	46 ft. forward		16100
	250 tons cargo aft hold	121 feet aft	30250	
Weights put on board	200 tons W.B. in aft main tank	71 feet aft		14200
	100 tons W.B. in aft tank	131 feet aft		13100
			30250	49000
				30250
				18750

There is thus an aft-trimming moment in operation of 18,750 foot tons.

The moment to alter trim 1 inch in the initial condition is 700 foot tons; as a first approximation, we therefore get—

$$\text{Change of trim} = \frac{18750}{700} = 27 \text{ inches, nearly.}$$

The approximate draughts are—

$$\text{Forward,} \quad 20'\ 0'' - (1'\ 9'') - (1'\ 1\tfrac{1}{2}'') = 17'\ 1\tfrac{1}{2}'';$$
$$\text{Aft,} \quad 26'\ 0'' - (1'\ 9'') + (1'\ 1\tfrac{1}{2}'') = 25'\ 4\tfrac{1}{2}''.$$

Allowing for the difference in the displacement and the metacentric height, the moment to alter trim 1 inch in the above approximate condition is found to be 650 foot tons; so that a mean moment is 675 foot tons. Employing this figure we get with more exactness—

$$\text{Change of trim} = \frac{18750}{675} = 28 \text{ inches, nearly.}$$

The final draughts will therefore be—

$$\text{Forward,} \quad 17'\ 1\tfrac{1}{2}'' - \tfrac{1}{2}'' = 17'\ 1'';$$
$$\text{Aft,} \quad 25'\ 4\tfrac{1}{2}'' + \tfrac{1}{2}'' = 25'\ 5''.$$

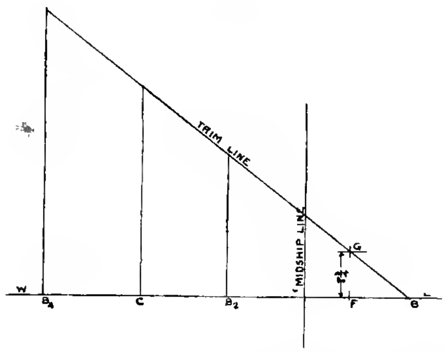
MR. LONG'S METHOD.*—A method of dealing with questions of trim, which differs somewhat from the preceding one, and which has several

* While Mr. Long makes no claim to the invention of this elegant method of solving trim problems, his paper contains what appears to be the first published description of it. As seems fitting, therefore, we have called the method by his name.

points of advantage, is described by Mr. Long in a paper read recently before the North-East Coast Institution of Engineers and Shipbuilders.

In this system, use is made of what are called trim lines or curves to find the trim corresponding to any mean draught and longitudinal position of the centre of gravity. A trim line is obtained as follows:—First, a level line is drawn, as WL in fig. 195, to represent the mean draught for which the trim line is required. On this a point B is taken as the position of the longitudinal centre of buoyancy at level keel, and a datum line drawn showing its relation, say, to amidships. The horizontal distance from B of the centre of buoyancy, with the vessel trimming 2 feet by the stern, is then calculated and marked off at B_2; also the distance abaft B of the centre of buoyancy, with the vessel trimming 4 feet aft, is plotted as at B_4. At B_2 and B_4 verticals are erected, and the corresponding trims marked off, the same scale being used throughout. Through the points thus found and the point B a line is drawn; this is the trim line required.

Fig. 195.



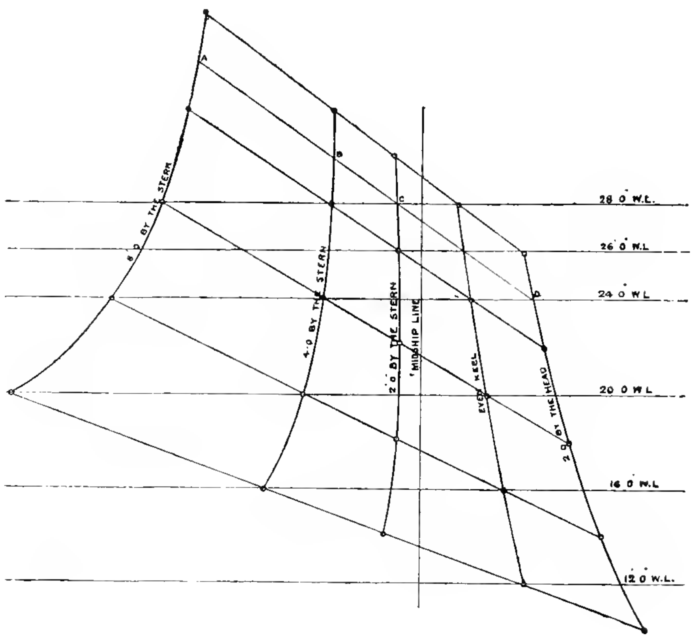
Obviously, we have here all the information necessary to determine any trim up to 4 feet, due to the movement of weights on board; for, if the distance the centre of gravity travels aft on account of the movement of the weights be ascertained and plotted from B along the level line to C, say, and a vertical be raised to intercept the trim line at D, CD must be the trim by the stern, as the centre of buoyancy and centre of gravity are again in the same vertical line.

For forward trims the trim line should be continued below its level line to indicate the movement of the centre of buoyancy in that direction. It should be noted that the centre of gravity and centre of buoyancy are here assumed to travel the same distance when a change of trim takes place. This is not quite true, as a glance at fig. 191 will show; B being below G, and therefore more remote from M, moves a greater distance. For quite accurate work, therefore, the distance plotted from B towards W (fig. 195) should be the calculated travel of the centre of gravity plus $BG \tan \theta$

(*see* fig. 191). It is not necessary to proceed to this refinement in ordinary cases, as the error thus involved is inappreciable.

One advantage of the trim line system is the absence of formulæ. No calculations are required except a simple one for the travel of the centre of gravity consequent on the movement of the weights. This will be seen by an example. It will be interesting to work out Example 1 (page 203) by this method; we are able to do this, as fig. 195 is the trim line at the load draught of the vessel referred to. Employing the figures given, we get—

Fig. 196.



$$\left.\begin{array}{r}\text{Travel of centre of gravity}\\ \text{on removal of weight}\end{array}\right\} = \frac{80 \times 150}{15814 - 150} = \cdot 766 \text{ feet.}$$

Plotting this distance from *B* to *F*, and erecting a perpendicular to meet the trim line at *G*, we obtain *FG*, or $7\frac{3}{4}$ inches, as the trim by stern required. This is the same result as before.

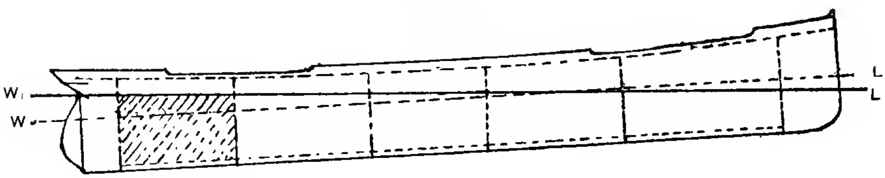
The trimming weight in the foregoing example is small in comparison with the displacement, and for such cases we know the ordinary metacentric method is as accurate as any. Where the weights and moments are great, however, only approximate results are obtainable by the ordinary method due to the fluctuating nature of the metacentric height. In this respect the trim line method excels the other, as it is practically accurate for all changes of trim and draught, however large.

It is perhaps scarcely necessary to point out that a trim line is only reliable at its own draught, and that where the change of displacement is considerable, a new curve is required. Experience goes to show that in ordinary cases the tendency of trim lines is to become less steep with reduction in draught. For this reason they should be drawn for useful draughts, as those of the load, ballast, and light conditions, and this would probably be sufficient in most cases. By constructing cross curves of trim, however, as is done for stability, a trim line for any draught within the limits of the cross curves can at once be obtained. Such a diagram obviously provides full trim information for a vessel. Fig. 196 represents the case of the steamer of Example 1. The horizontal lines are the waterplanes for which trim lines have been drawn. The points in which the latter intersect their corresponding level waterplanes, and where they indicate trims of 2 feet, 4 feet, and 8 feet by the stern, and 2 feet by the head, are enclosed by small circles. Curves through these points give the cross curves required. If, now, a trim line at an intermediate draught, say of 26 feet, be desired, it is only necessary to draw a level line at this point, and at heights of 2 feet, 4 feet, etc., parallel lines to cut the corresponding cross curves at A, B, C and D, a line through these points being the trim line required.

BILGING.—Given a diagram like that just described, any trim question relating to the ship for which the diagram is drawn can be readily and quickly dealt with. Consider, for instance, the important trim problem of finding the floating condition of a vessel consequent on one or more of her compartments being bilged and in free communication with the sea. Such a case is depicted in fig. 197, in which a vessel is shown bilged in the after compartment of the hold. $W_1 L_1$ is the line of flotation after the accident, with the ship once more at rest; $W L$ the original waterline. The problem is to determine the line $W_1 L_1$.

Now the change of trim is here caused, not by an added or deducted weight, but by a loss of buoyancy, and it is usual to treat the problem as one of loss of buoyancy. By an exercise of imagination, however, the question may be more easily dealt with; for, if the hole into the bilged compartment be assumed closed—the vessel being once more at rest—the trim will not be affected, but an important change will have taken place in her floating condition, as the lost buoyancy will have been restored and the water in the compartment become, for practical purposes, a weight carried. Viewed thus, it is only necessary to obtain the weight and the position of the centre of gravity of this water for a complete solution of the problem. The process of calculation is tentative in character and may be as follows:—First, the weight of water in the compartment up to the original waterplane $W L$ should be found, and the parallel sinkage determined, assuming compartment open to the sea and the admitted water placed with its centre of gravity in the vertical plane containing the centre of gravity of the added layer of displacement. This distance, measured in the trim diagram above the height of the original waterplane, will give the point from which the level line and corresponding trim line should be drawn. The trim can then be obtained, as already described, by finding the travel aft of the centre of gravity, assuming the weight to be translated to its true position. This is, of course, a first approximation. It will next be necessary to calculate the weight of water in the compartment, assuming the surface to rise to the level of the new draughts, and to use it in the same way in another trim estimate. If the second approximation should differ much

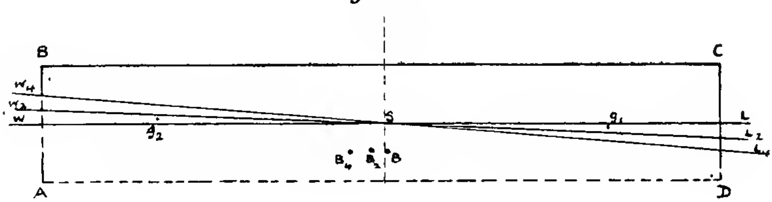
Fig 197.



from the first, it may be necessary to proceed to a third. But the experience of the calculator must guide him here.

As a numerical example, take a box-shaped vessel, 210 feet long, 30 feet broad, and 20 feet deep, drawing 10 feet forward and aft; and suppose an empty watertight compartment at the extreme after-end, 10 feet long, to be in free communication with the sea. It will be necessary first to draw out the trim diagram. This is a simple matter owing to the regular nature of the vessel's shape. We begin by obtaining the trim line at 10 feet

Fig. 198.



draught. $ABCD$, fig. 198, shows the vessel in side elevation, WL is the level keel water-line, W_2L_2 and W_4L_4 those when 2 feet and 4 feet by the stern, respectively. Now, assuming the vessel to be floating in salt water, her displacement is—

$$\frac{210 \times 30 \times 10}{35} = 1800 \text{ tons,}$$

and in passing from the water-line WL to water-line W_2L_2, the wedge of displacement LSL_2 moves to the position WSW_2. As SL is half the vessel's length, and LL_2 1 foot, the volume of the wedge is—

$$\frac{105 \times 1 \times 30}{2} = 1575 \text{ cubic feet,}$$

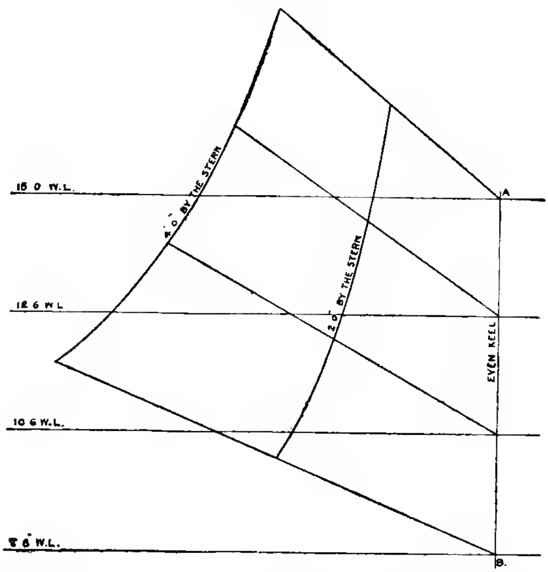
and in moving aft, its centre of gravity travels a horizontal distance $g_1 g_2$,

$$\text{or,} \quad \frac{210 \times 2}{3} = 140 \text{ feet.}$$

The corresponding movement of the vessel's centre of buoyancy is from B to B_2, and from what we know of moments, obviously—

$$B B_2 = \frac{1575 \times 140}{1800 \times 35} = 3{\cdot}5 \text{ feet.}$$

Fig. 199.



The horizontal travel of the centre of buoyancy, with the vessel 4 feet by the stern, is clearly just double this amount, or 7 feet. This is all the information needed to construct the trim line at the initial draught. The trim lines corresponding to other displacements would be obtained in the same manner. Fig. 199 is the complete diagram for this vessel, and shows cross curves with a range from 7 feet 6 inches to 15 feet draught. We are now in a position to deal with our bilging question.

Beginning with the initial condition, we have—

$$\text{Weight of water in bilged compartment} = \frac{10 \times 10 \times 30}{35} = 85{\cdot}71 \text{ tons,}$$

and,

Parallel sinkage assuming water situated amidships and compartment open to the sea $\left.\right\} = \frac{85 \cdot 71 \times 35 \times 12}{200 \times 30} = 6$ inches,

also,

Horizontal travel* aft of vessel's centre of gravity, assuming the water at the increased draught to move into its true position and the ship's bottom to be intact $\left.\right\} = \frac{90 \times 100}{1890} = 4 \cdot 76$ feet.

Referring to fig. 199, we can draw at once the trim line corresponding to a level line at 10 feet 6 inches, and by measuring 4·76 feet along this level line from the vertical *AB*, and erecting a perpendicular, we get 2 feet $10\frac{1}{4}$ inches as the trim by the stern. The draughts of the vessel will thus be—

$$\text{Forward,} \quad 10' \ 0'' + 6'' - 1' \ 5\tfrac{1}{8}'' = 9' \ 0\tfrac{7}{8}'';$$
$$\text{Aft,} \quad 10' \ 0'' + 6'' + 1' \ 5\tfrac{1}{8}'' = 11' \ 11\tfrac{1}{8}''.$$

In the second approximation, we start with the vessel in this trim. The weight of water in the bilged compartment will now be—

$$\frac{11 \cdot 86 \times 10 \times 30}{35} = 101 \cdot 66 \text{ tons.}$$

The

$$\text{Parallel sinkage} = \frac{101 \cdot 66 \times 35 \times 12}{210 \times 30} = 6\tfrac{3}{4} \text{ inches nearly,}$$

and taking the centre of gravity of the water at the middle of the length of the compartment, we get as before—

Travel of vessel's centre of gravity due to admission of water $\left.\right\} = \frac{101 \cdot 66 \times 100}{1901 \cdot 66} = 5 \cdot 35$ feet aft.

From the trim diagram we find the corresponding trim by the stern to be 3 feet $2\frac{3}{4}$ inches.

Dividing this equally forward and aft, and adding $6\frac{3}{4}$ inches as the parallel sinkage, the draughts become—

$$\text{Forward,} \quad 10' \ 0'' + 6\tfrac{3}{4} - (1' \ 7\tfrac{3}{8}'') = 8' \ 11\tfrac{3}{8}'';$$
$$\text{Aft,} \quad 10' \ 0'' + 6\tfrac{3}{4} + (1' \ 7\tfrac{3}{8}'') = 12' \ 2\tfrac{1}{8}''.$$

By a third approximation we obtain the draughts—

$$\text{Forward,} \quad 8' \ 11''$$
$$\text{Aft,} \quad 12' \ 2\tfrac{3}{4}'',$$

in which condition the vessel will float in equilibrium whether the after compartment be now open to the sea or not. Of course, the same result could be obtained by the ordinary method, *i.e.*, by calculating the height of the longitudinal metacentre, the moment to alter trim, and the heeling moment due to

---

* The trimming of the vessel causes the water in the compartment to change level, and a small quantity of the water to move aft; this affects the position of the ship's centre of gravity, and therefore the trim, but to no appreciable extent, except in the case of large compartments.

the admission of the water, and finally dividing the latter by the moment to alter trim. We do not propose to deal with the problem in this way, as the principles involved have already been fully explained. The student, however, should work it out himself as an exercise.

APPROXIMATE CALCULATIONS.—Although a commanding officer may know nothing regarding his vessel beyond her dimensions and displacement, he is still able to estimate, roughly, at least, the trimming effect due to the addition, removal, or movement of weights. In the formula—

$$\text{Moment to alter trim 1 inch} = \frac{W \times Gm}{12 \times L} \text{ foot tons,}$$

if we assume Gm to be equal to L, which is roughly true in the case of ordinary cargo vessels at their load displacements, the trimming moment per inch becomes $\frac{W}{12}$ foot tons.

Applying this simple formula to the example on page 198, we get—

$$\text{Moment to alter trim 1 inch} = \frac{2000}{12} = 166{\cdot}6 \text{ foot tons,}$$

and

$$\text{Change of trim} = \frac{50 \times 80}{166{\cdot}6} = 24 \text{ inches,}$$

which compares with 25 inches obtained by the exact method.

In the case of Example 1, page 203,

$$\text{Moment to alter trim 1 inch} = \frac{15814}{12} = 1318 \text{ foot tons.}$$

For the effect of discharging 150 tons from a position 80 feet before the centre of gravity of the load waterplane, we thus have—

$$\text{Trim by stern} = \frac{150 \times 80}{1318} = 9 \text{ inches, nearly.}$$

The ordinary formula gives $7\frac{3}{4}$ inches; the approximation is thus near enough for practical purposes, an inch or two either way, in ordinary cases, not being of great importance. As the value of Bm rises rapidly with reduction in draught, the formula is inapplicable for draughts other than the load draught. Also, it is unsuitable in the case of vessels which are of abnormally shallow draught in relation to length, as Gm and L are then far from being even approximately equal.

An approximate formula, giving closer results than the foregoing, has been devised by M. Normand.* By this rule, for the height of the longitudinal metacentre above the centre of buoyancy in ordinary cargo steamers, we have—

$$Bm = {\cdot}0735 \frac{A^2 \times L}{b \times V} \text{ feet.}$$

---

* See a paper in the Transactions of the Institution of Naval Architects for 1882.

where A = area of load waterplane in square feet.
L = length on the load waterline in feet.
b = breadth of ship amidships in feet.
V = volume of displacement in cubic feet.

Assuming $Bm = Gm$,

$$\text{Moment to alter trim 1 inch} = \frac{\frac{V}{35} \times \cdot 0735 \frac{A^2 \times L}{b \times V}}{L \times 12}$$

$$= \cdot 000175 \frac{A^2}{b} \text{ foot tons.}$$

This is Normand's formula.

Now, if T be the tons per inch of immersion,

$$T = \frac{A}{420}$$

$$A = 420\,T$$

$$A^2 = 176400\,T^2.$$

Substituting this value of A^2, the formula takes the convenient shape—

$$\text{Moment to alter trim 1 inch} = \frac{30\cdot 9 \times T^2}{b} \text{ foot tons.}$$

Applying the Rule to Example 1, page 203, we get—

$$\text{Moment to alter trim 1 inch} = \frac{30\cdot 9 \times 53\cdot 3 \times 53\cdot 3}{56}$$

$$= 1567 \text{ foot tons;}$$

the previous value being 1570 foot tons.

In the case of Example 2—

$$\text{Moment to alter trim 1 inch} = \frac{30\cdot 9 \times 33 \times 33}{48} = 701 \text{ foot tons,}$$

which compares with 700 tons, the exact value.

Besides the foregoing, we have in the trim-line method a ready means of making approximate estimates of trim. It happens that the trim lines in ordinary cases are practically straight, and make certain definite angles with the corresponding level lines, also that these angles, in different vessels of similar type, are about the same at corresponding draughts. It has been suggested, therefore, that in type vessels a note should be made of the trim angles at useful draughts, such as those of the load, ballast, and light conditions. The trim line in the case of any new design could then be plotted at once with sufficient accuracy for preliminary calculations. If only one trim should be required and the angle of the corresponding trim line is known, it can be found quickly by means of the formula—

$$\text{Change of trim in inches} = \frac{w \times d}{W} \times C \times 12,$$

where w = weight shifted,
d = distance shifted,
W = whole displacement,
C = tangent of the trim line angle.

C varies considerably with type of vessel. Ships of very light draught relatively to their length, have flatter trim lines than those of ordinary proportions, but an average value at the load draught of cargo steamers 300 to 500 feet in length, and of the usual fullness, is ·9163, corresponding to a trim line angle of $42\frac{1}{2}°$. Assuming this trim line angle in the case of Example 1, page 203;

$$\text{Change of trim} = \frac{150 \times 80}{15814 - 150} \times \cdot 9163 \times 12 = 8 \cdot 4 \text{ inches},$$

which is a good approximation. The student should apply the rule in other cases; an officer might try it on his own ship.

TRIM INFORMATION FOR COMMANDING OFFICERS.—An important use to which the trim line method may be applied, is the supplying of information to masters and others who have to deal with loading and ballasting operations. With a good-sized diagram, showing the trim curves of his vessel, and a scale, a master should be able to decide in a few minutes any question of trim, provided the weights to be shipped and unshipped, and their movements, be known. We have already fully explained the procedure. If builders would supply such trim diagrams to new vessels, with instructions as to their use, we are confident they would come to be highly appreciated.

QUESTIONS ON CHAPTER VIII.

1. Define the longitudinal metacentre. A prism of rectangular section 200 feet long, and 33 feet broad, floats at a draught of 11 feet forward and aft, calculate the height of the longitudinal metacentre above the centre of buoyancy.

Ans.—303 feet.

2. The equidistant ordinates of a vessel's waterplane measured on one side of the middle line are—·2, 7·2, 10·6, 12·0, 12·0, 12·0, 10·9, 9·6, and 1·9 feet, and half-ordinates at the ends have values 3·8 and 7·7 feet, respectively; find the height of the longitudinal metacentre above the centre of buoyancy, the volume of displacement being 20,000 cubic feet, and longitudinal interval between the ordinates 15 feet.

Ans.—102·26 feet.

3. Obtain the expression giving the change of trim consequent on moving a small weight w tons longitudinally through a distance a feet. A vessel 300 feet in length floats at a level draught of 17 feet; she has a longitudinal metacentric height of 400 feet, and a displacement of 4500 tons; a weight of 50 tons is moved aft through 100 feet; find the new draught forward and aft.

Ans.—Draught { Forward, 16 feet, 7 inches.
Aft, 17 feet, 5 inches.

4. Deduce the moment to change trim one inch in the case of the vessel of the last example.

Given that the tons per inch of immersion is 30, calculate the new draught forward and aft when the following weights have been placed on board in the positions named.

Weights (Tons).	Distance from Centre of Gravity of Waterplane.	
20	100	before.
45	80	
15	40	
60	50	abaft.
40	80	
30	110	

Ans.—Moment to change trim one inch, 500 foot tons.

Draught { Forward, 17 feet, $3\frac{3}{4}$ inches.
Aft, 17 feet, $10\frac{1}{4}$ inches.

5. Suppose a weight of moderate amount to be put on board a vessel, where must it be placed so that the ship shall be bodily deeper in the water without change of trim?

Describe, clearly, why it is that vessels in passing from salt water to fresh water usually change trim slightly as well as change their draught of water.

6. It is desired that the draught of water aft in a steamship shall be constant, whether the coals are in or out of the ship. Show how the approximate position of the centre of gravity of the coals may be found, in order that the desired condition may be fulfilled.

7. What is a trim line? Describe how such a line is obtained, and explain its uses.

8. A box-shaped vessel, 260 feet long, 40 feet broad, and 25 feet deep, floats at an even draught of 20 feet, construct the trim line for this draught. If 100 tons be shipped aft, with its centre 100 feet from amidships, find the new draught forward and aft, using the trim line.

Ans.—Draught { Forward, 19 feet, $6\frac{3}{4}$ inches.
Aft, 21 feet, $1\frac{1}{4}$ inches.

9. Referring to the vessel of the previous question—if a watertight compartment situated at the extreme after-end be bilged and in free communication with the sea, what will be the new floating condition, the bilged compartment being 10 feet long, the full breadth of the vessel, and with half its space occupied by cargo?

Ans.—Draught { Forward, 19 feet, $2\frac{3}{4}$ inches.
Aft, 21 feet, $7\frac{1}{4}$ inches.

10. It is desired that a certain vessel shall float with any two compartments in open communication with the sea. Describe in detail the calculations involved.

11. A steamer 330 feet long, 48 feet broad, drawing 24 feet aft, and 20 feet forward, has to cross a bar over which there is a depth of water of 23 feet, 6 inches. The vessel has a fore-peak tank with a capacity of 100 tons. Given that the centre of gravity of this tank space is 150 feet forward of the centre of gravity of the load-water plane, find, by an approximate method, if filling the tank will modify the draught sufficiently to admit of the vessel crossing the bar. The displacement in tons to begin with is 7800.

12. Referring to the previous question, if it be given that the longitudinal metacentric height is 345 feet, and the tons per inch of immersion 33, estimate correctly the effect on the draught of filling the fore-peak tank.

Ans.—Draught { Forward, 21 feet, 2 inches.
Aft, 23 feet, 4 inches.

CHAPTER IX.

Stability of Ships at Large Angles of Inclination.

IN Chapter VII. we saw how to obtain the righting or upsetting moment for a vessel when inclined through initial angles about the upright position. We learned that up to angles of 10 or 11 degrees, the metacentre may be considered as fixed in position, and that inside these limits the

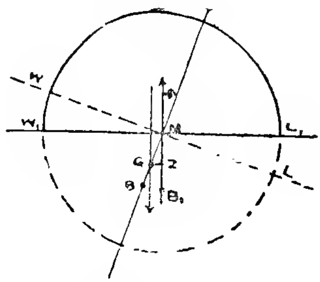
$$\text{Heeling Moment} = W \times GM \times \text{Sin. } \theta.$$

Thus, taking the two cargo vessels for which the values of GM were obtained, we have—

Righting moment at 10 degrees (small vessel) $= 4525 \times 1{\cdot}73 \times {\cdot}1736$
$= 1359$ foot tons.

Do. do. (large vessel) $= 15814 \times {\cdot}85 \times {\cdot}1736$
$= 2333$ foot tons.

Fig. 200.



For inclinations much exceeding 10 to 12 degrees, however, except in the instance of a single type of vessel, the righting moment cannot be obtained by this method. The exception referred to, is where a vessel is so designed that all the immersed cross sections are circular segments with a common centre in the middle line. We have already shown that for floating bodies of this form the line of upward pressure passes through the same point M for all transverse inclinations, so that, if there is no disturbance in the weights, the distance GM will remain unchanged as the vessel heels from angle to angle. Fig. 200 represents a vessel of constant circular section inclined to some angle θ, GM is the metacentric height, and if W be the displacement, we have, by applying the metacentric method—

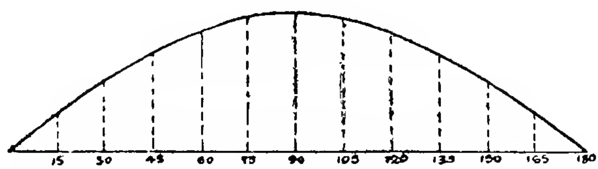
$$\text{Righting moment in foot tons} = W \times GM \times \text{Sin. } \theta$$
$$= W \times GZ,$$

GZ being the arm of the righting couple.

The only variable in this expression is the sine of the angle; therefore, to construct a stability curve for this simple case, it is only necessary to draw a horizontal line, set off on it, to a convenient scale, the various angles at 15, 30, 45, etc., degrees, erect perpendiculars at the points of division, on these perpendiculars scale off the various values of righting moment as obtained above, and draw a fair curve through the points so found.

As a specific case take a cylindrical vessel, 20 feet in diameter, floating with its axis in the waterplane. We shall deal only with the levers or righting arms, so that the length of the vessel is immaterial. If we assume the centre of gravity to be 2 feet below the centre of the figure, we may, using a table of sines, at once write down the value of the righting arm for any inclination. Obviously, the righting arm increases from zero at 0 degrees to a maximum value at 90 degrees, and thence gradually decreases

Fig. 201.



to zero again at 180 degrees; obviously, too, by drawing a diagram, the righting arm or lever at an angle θ, say, is the same as at the angle $180° - \theta$. It is, therefore, only necessary to calculate values from 0 degree to 90 degrees; and at intervals of 15 degrees, which is close enough for the purpose of constructing a curve, these are as follows:—

Inclination, in Degrees.	Sine of Angle.	Righting Arms, in Feet.
15	·2588	·5176
30	·5	1·00
45	·7071	1·414
60	·866	1·732
75	·9659	1·932
90	1·0	2·00

Fig. 201 is the curve constructed from this data. On a little consideration it will be seen to represent, for every draught, the curve of righting arms of all vessels of circular section, whatever their length or diameter, in which $GM = 2$ feet; and since the righting moment at any angle is equal to GZ, or the ordinate of the curve at that angle, multiplied by the displacement, if the scales be always altered to suit, this curve will also represent the *righting moments* of all vessels of all circular section having a metacentric height of 2 feet.

SUBMARINE VESSELS.—The cylindrical vessel just referred to is assumed to float with part of its bulk above the surface—the case of

Fig. 202.

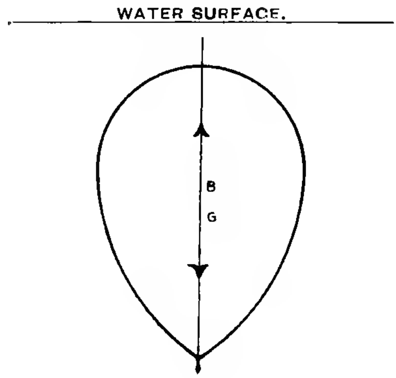
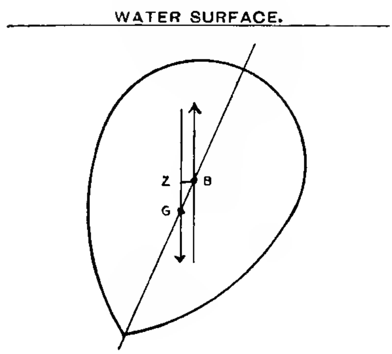


Fig. 203.



ordinary vessels; but when properly designed, a vessel may have stability even when totally submerged. The submarines, now so much in evidence, are examples in point. A stability curve of a submerged vessel may be easily obtained by a method analogous to that employed in the previous case. Figs. 202 and 203 show a submarine floating upright and heeled, respectively. *B*, the centre of buoyancy, is also the centre of bulk; *G* is the centre of gravity. Here, *G* being below *B*, when the vessel is heeled as in fig. 203, the tendency of the resultant forces of weight and buoyancy is to restore the vessel to the position in fig. 202. If G were above B, and the vessel then inclined, the tendency would be to heel still further until G became vertically below B—the position of stable equilibrium.

Applying the formula $BM = \frac{I}{V}$, since $I = 0$, BM is zero, and therefore, B and M are coincident. Thus, in this special case, B and BG have much the same functions as M and GM in the preceding one, for at any angle θ the righting or upsetting moment $= BG \sin \theta$; so that, as in the case of the cylinder floating at the surface, the lever varies directly as the sines of the angles of inclination, has zero values at 0 degrees and 180 degrees, and a maximum value at 90 degrees. Clearly, if BG is 2 feet, fig. 201, the stability curve for a cylindrical vessel floating on the surface may also be taken to represent the curve of righting arms for the submerged vessel.

Fig. 204.

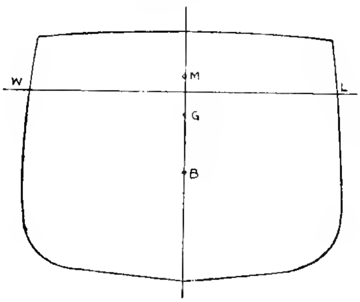
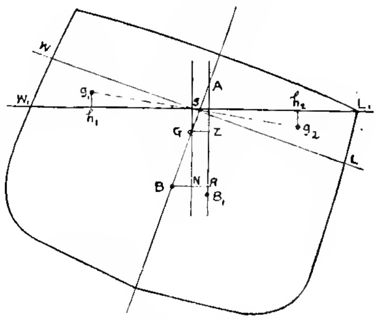


Fig. 205.



A noteworthy point in curves of righting arms of totally submerged vessels is that they represent the stability when inclined in any direction, either transverse or longitudinal, which follows from the fact that the line of buoyancy must always pass through the same point, viz., the centre of bulk. This is, of course, by no means the case in vessels floating at the surface, which have enormous righting power when inclined longitudinally.

VESSELS OF NORMAL FORMS.—The case of an ordinary vessel, it need hardly be said, admits of no such simple treatment as those just dealt with, owing to the increased difficulty of obtaining values of GZ. Fig. 204 shows, in cross section, an ordinary vessel floating upright at a waterplane WL. M is above G, therefore the condition is one of stable equilibrium. Fig. 205 shows the same vessel heeled to a large inclination. The movement has caused the centre of buoyancy B to travel out to B_1, through which the resultant buoyant pressure now passes. No weights are supposed to have been shifted during the heeling, so that the centre of gravity G is unchanged in position. Unlike the case of the cylinder, the line of upward force does not intersect the middle line at the metacentre,

GM in fig. 204 and GA in fig. 205 having different values. Obviously, then, the equation—

$$\text{Righting Moment} = W \times GM \times \text{Sin } \theta$$

is no longer applicable, and in order to obtain the values of righting arms or righting moments at large inclinations we must resort to another method.

In this case, to find the lever GZ between the verticals through G and B when the vessel is inclined to any angle, we must proceed as follows:—Referring to fig. 205, as previously pointed out, the transference of the wedge of displacement from WSW_1 to LSL_1, compels the centre of buoyancy to travel from B to B_1. A line joining these points is parallel to the line joining g_1g_2, the centres of gravity of the wedges, and

$$BB_1 = \frac{\text{Volume of wedge} \times g_1g_2}{\text{Volume of displacement}}.$$

Now, through B draw a horizontal line to cut the verticals through G and B_1, in N and R; and from g_1 and g_2 drop perpendiculars g_1h_1, g_2h_2 on W_1L_1; then clearly,

$$BR = \frac{\text{Volume of wedge} \times h_1h_2}{\text{Volume of displacement}}.$$

Also, $\quad BR - BN = NR = GZ;$

and $\quad BN = BG \sin \theta;$

so that, $\quad GZ = \dfrac{\text{Volume of wedge} \times h_1h_2}{\text{Volume of displacement}} - BG \text{ sin. } \theta;$

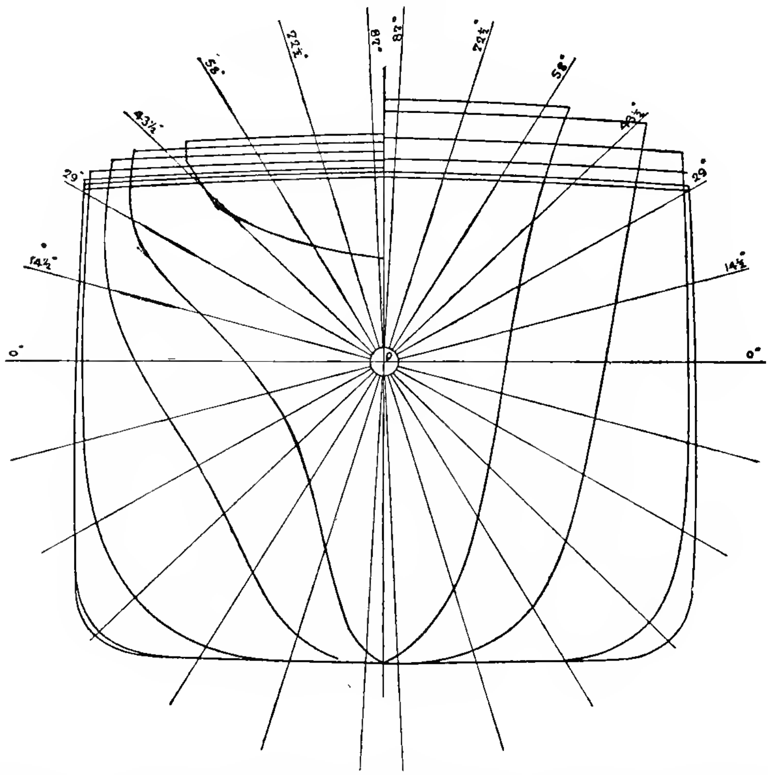
this is the value of the stability lever required, and the equation is known as Atwood's formula. The only portion of this expression which cannot be quite easily obtained, is $v \times h_1h_2$, the horizontal moment due to the transverse movement of the wedge of displacement, and we now propose to show how this is calculated, and the stability lever or moment arrived at.

VOLUMES AND MOMENTS OF WEDGES.—A body plan of the ship is prepared with transverse sections, spaced as for a displacement calculation, and with radial planes drawn to represent the floating condition of the vessel when upright, and when inclined at all the inclinations required to give data for the construction of the stability curve. The sections should represent the full volume available for buoyancy, and be drawn to the top watertight deck and to the outside of the shell-plating. With regard to the radial planes, it is found convenient to draw them so as to intersect in the middle-line plane (*see* O, fig. 206), although this does not usually ensure that the in and out wedges shall be of equal volumes, as, of course, they must be; but it allows all the inclined planes up to any maximum inclination to be drawn at once, while the correction due to the inequality of the wedges can easily be made afterwards.

To furnish spots close enough to obtain the correct form of the stability curve, the radial planes should be drawn at intervals of about 10 to 15 degrees. Discontinuities in the vessel should be carefully dealt with. The entrance into the water of the deck edge, for instance, causes a sudden change in the form of the immersed wedge, and to ensure accuracy in finding the volumes and moments of wedges by Simpson's Rules, a radial plane should occur at this point, with a suitable number of radial planes on each side of it.

These particulars attended to, the measurement and tabulation of the ordinates, or breadths of sections at the various radial planes on each side of the point *O*, is proceeded with, those of the immersed side being kept separate from the emerged, and those of the various planes separate from

Fig. 206.



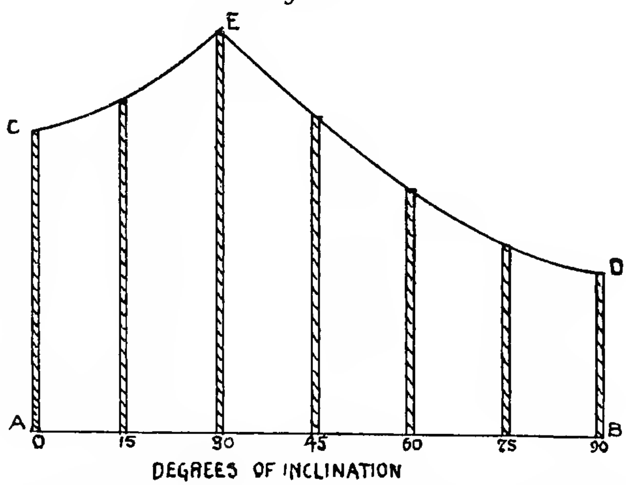
each other. At each radial plane, the volume of an elementary wedge and its moment about a fore-and-aft horizontal axis through *O* is next calculated.

To show how this is done, let b be the length in feet of an ordinate of a radial plane, say, on the immersed side; then the sectional area at this ordinate of a very small wedge of the immersed volume, treating it as a segment of a circle, will be $\frac{b^2}{2}\theta$ square feet, θ being the circular measure of the wedge angle; and if t be the thickness in feet of a thin transverse slice, its volume in cubic feet will be $\frac{\theta}{2}b^2 t$.

Having obtained such values for slices at various sections in the length of the wedge, to find the volume of the latter becomes simply a matter of finding the area of a plane surface, for, if a base line representing to scale the length of the vessel be taken, and at points corresponding to the positions of the various sections the quantities $\frac{\theta}{2} b^2 t$ be set off as rectangles, each on the little quantity t as base, and a curve be drawn through the tops of the rectangles, an area will be enclosed representing the sum of the volumes of all the slices into which the wedge may be supposed divided, and, therefore, the volume of the whole elementary wedge.

The moment of an elementary wedge may be similarly dealt with. For instance, taking the same radial plane and ordinate, the distance from O, of

Fig. 207.



the centre of gravity of a thin transverse slice of the elementary wedge is $\frac{2}{3} b$ feet, and the geometrical moment of the slice in foot units about a fore-and-aft axis through O,

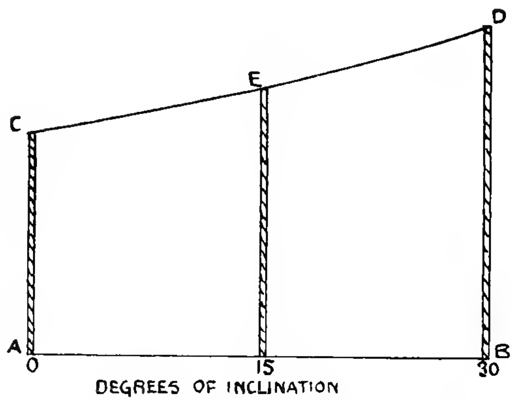
$$\frac{2}{3} b \times \frac{\theta}{2} b^2 t, \text{ or } \frac{\theta}{3} b^3 t.$$

To express the whole moment as an area, it is only necessary to plot, at the same points in the length as in the case of the volume, rectangles, each on a base t, giving the various values of $\frac{\theta}{3} b^3 t$, and draw a curve. The volumes and moments of the elementary wedges may now be found by calculating the above areas.

It is next necessary to combine the figures of the elementary wedges to obtain those of the full wedges of immersion and emersion. We shall show presently how this is done in an actual case, but it may also be explained graphically. Take a base line $A B$ (fig. 207), and let it represent on some scale the circular measure of the maximum wedge angle. Mark off points at the various angles at which the volumes of the elementary wedges have been calculated, and plot rectangles, each on a base θ (θ = the circular measure of the elementary wedge angle), representing the volumes of the corresponding elementary wedges. A curve through the tops of these rectangles will enclose an area *A C D B*, representing the sum of the volumes of all the elementary wedges, that is to say, the volume of the whole wedge.

To find the volume of any wedge within the limits of *A C D B*, it is simply necessary to plot an ordinate at the correct wedge angle, and by Simpson's Rules calculate the area thus cut off. Separate diagrams are necessary for the immersed and emerged wedges. Coming to the moments of the full wedges, it must be noted that while the moments of the elementary wedges are, in the first instance, calculated about a longitudinal

Fig. 208.

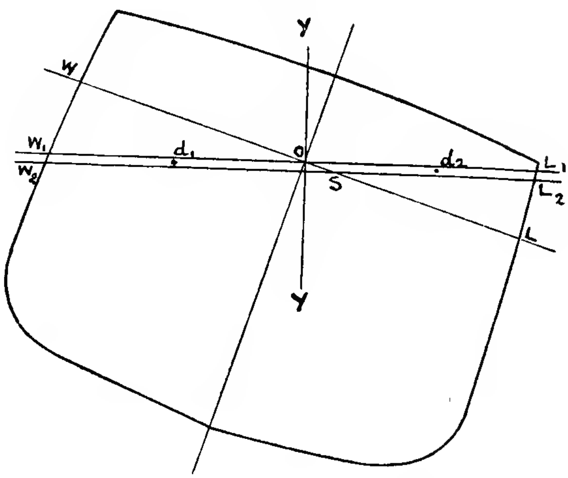


axis through *O* (the point of intersection of the radial planes) the moments required for statical stability are taken about a longitudinal vertical plane through *O* (*see* *y y*, fig. 209), and, therefore, in combining the elementary moments to obtain those of the full wedges of immersion and emersion, each of the former has to be multiplied by the cosine of the angle which the particular elementary wedge makes with the horizontal. For instance, in calculating the moment of wedges of 30 degrees, the moment of the elementary wedges at 0 degrees about a longitudinal axis through *O* has to be multiplied by the cosine of 30 degrees, and those at, say, 15 and 30 degrees, by the cosine of 15 and 0 degrees, respectively. Fig. 208 is the complete diagram of moments, the abcissæ being in circular measure, *A B* representing 30 degrees. Since the sum of the moments is required, the diagram takes account of the wedges on both sides of the axis *y y*. Thus the little rectangle at *A C* is the sum of the moments of the in and out elementary wedges at 0 degrees multiplied by cosine 30 degrees; the rectangle at *E F* the sum of the moments of the in and out elementary wedges at 15 degrees multiplied by cosine 15 degrees; and the rectangle at *D B* the corresponding quantity at 30 degrees multiplied by cosine 0 degrees. Clearly, from our preceding remarks, the whole area *A C D B* represents the sum of the moments of the immersed and emerged wedges at 30 degrees about the axis *y y*.

It will be seen that, in the case of the moments, a new diagram is required for each inclination at which the righting arm or moment is calculated, as the elementary wedge at the limiting inclination must always be multiplied by cosine 0 degrees, and the others by the cosine of the angle which each of them makes with the limiting radial plane.

Such are the principles to be followed in finding the volumes and moments of the various in and out wedges, and they are seen to present

Fig. 209.



no greater difficulty than is involved in the application of Simpson's Rules to the calculation of plane areas.

CORRECTION OF WEDGES.—It must not be forgotten that the moments of the various wedges, found as above, have to be corrected on account of the immersed and emerged wedges, as drawn in the body plan, being unequal in volume. This may be done as follows:—Suppose the immersed wedge is in excess, then the vessel is shown deeper in the water than she should be, and the vertical distance between the true and the assumed waterplanes, or—

$$\text{Thickness of layer} = \frac{V_1 - V_2}{\text{Area of inclined waterplane}}$$

where V_1 and V_2 are the volumes of the in and out wedges, as drawn.

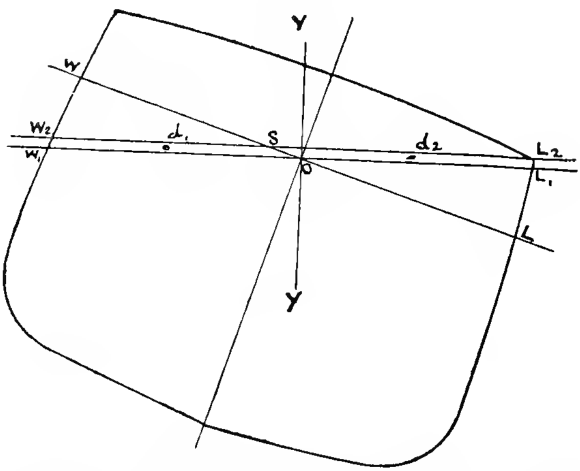
Let fig. 209 represent the case dealt with; let WL be the upright water-plane, $W_1 L_1$, the uncorrected inclined plane, and $W_2 L_2$ the corrected inclined plane. Since the correct wedges are $W S W_2$ and $L_2 S L$, the moment of the volume $W_1 O S W_2$ is to be added, and that of $L_1 O S L_2$ deducted, from the moments of wedges as calculated. Call volume $W_1 O S W_2$ v_1, and volume $L_1 O S L_2$ v_2, and let $o d_1$ and $o d_2$ be the distances of their centres of gravity from axis $y y$, then the correction is—

$$v_1 \times o d_1 - v_2 \times o d_2 \quad (1).$$

If the centre of gravity of the whole layer $W_1 L_1 L_2 W_2$ be at a distance x on the immersed side of the axis—

$$v_1 \times o d_1 - v_2 \times o d_2 = (v_1 + v_2) x,$$

Fig. 210.



and equation (1) will be negative, and the correction a deduction. If x be on the emerged side, (1) will be positive, and the correction an addition.

Now, suppose the emerged wedge to be in excess (*see* fig. 210). In this case, the moment of the volume $W_1 O S W_2$ will be deducted from, and that of volume $L_1 O S L_2$ added to, the calculated moment of the wedges. Using the same symbols, we have—

$$\text{Correction} = v_2 \times o d_2 - v_1 \times o d_1 \quad (2)$$

and this is also equal to $(v_1 + v_2) x$, where x is the distance of centre of gravity of the whole layer from the axis $y y$. If x be on the emerged side, equation (2) will be negative, and the correction a deduction; if on the immersed side, positive, and the correction an addition. A little consideration will make this quite clear. Rules for the correction of the moments of wedges may now be stated as follows:—

1. If the immersed wedge be in excess, and the centre of gravity of

the layer on the immersed side of the axis of moments, the correction will be a deduction; but if it be on the emerged side, an addition.

2. If the emerged wedge be in excess, and the centre of gravity of the layer on the emerged side, the correction will be a deduction, but if it be on the immersed side, an addition.

In most cases the layer is small, and the centre of gravity of the inclined plane may be used for that of the layer. This simplifies the work, but if the layer be large, its centre of gravity must be calculated, and its correct distance from the axis employed. Thus we arrive at the value of the quantity $v \times h_1 h_2$ at any inclination, and by Atwood's formula the length of the stability lever may be written down.

As a practical example, let us obtain the stability curve for an actual vessel, such, for instance, as the large cargo steamer whose dimensions and other particulars are given on page 182.

This vessel, laden to her full draught, has, with a certain distribution of cargo, a metacentric height of ·85 feet. The centre of gravity is 22·65 feet above the base line, and the centre of buoyancy 14·4 feet above the same line, so that BG, the distance between the centre of buoyancy and centre of gravity, is $22·65 - 14·4$, or 8·25 feet. Let this be the basis of our calculation.

Fig. 206 shows the body plan drawn out as already directed, with transverse sections and radial planes, the former showing the vessel's shape at each tenth part of the length, and also at intermediate positions towards the ends, and the latter being drawn at intervals of $14\frac{1}{2}$ degrees so as to ensure a radial plane striking the deck edge, which becomes immersed at 29 degrees.

Before starting to measure the ordinates, sheets must be prepared (*see* Table I.) with a suitable number of columns to take the calculations for the areas of the radial planes and the functions for the volumes and moments of the elementary wedges. Two such sheets are required for each radial plane, the immersed and emerged sides, as previously mentioned, being kept separate. In Table I. we give the calculations for the elementary wedges on each side of the axis at 29 degrees inclination. As the work is the same for each elementary wedge, the method followed is amply illustrated in this table, which is drawn up in accordance with explanations given for the general case.

Having obtained the requisite information for the various elementary wedges, it is utilised to determine the values of the righting arms when inclined to angles increasing by increments of $14\frac{1}{2}$ degrees.

Let us deal with the vessel when inclined, say, to 29 degrees. The elementary wedges required are those at 0 degrees, $14\frac{1}{2}$ degrees and 29 degrees, respectively. The information is combined, as in Table II., which is seen to consist of the numerical work entailed in deducing the areas of the volume and moment diagrams previously described.

TABLE I.

ELEMENTARY WEDGE 29 DEGREES, EMERGED SIDE.

No. of Ord.	Ords.	S.M.	Products for Area.	Ords.^{2}	Products for Volume of El. Wedge.	Ords.^{3}	Products for Moments of El. Wedge.
0	·3	½	·1	—	—	—	—
½	10·5	2	21·0	110	220	1158	2316
1	19·5	1½	29·2	380	570	7415	11122
2	29·8	4	119·2	888	3552	26463	105852
3	31·9	2	63·8	1018	2036	32462	64924
4	31·9	4	127·6	1018	4072	32462	129848
5	31·9	2	63·8	1018	2036	32462	64924
6	31·9	4	127·6	1018	4072	32462	129848
7	31·9	2	63·8	1018	2036	32462	64924
8	30·8	4	123·2	949	3796	29218	116872
9	20·5	1½	30·7	420	630	8615	12922
9½	11·2	2	22·4	125	250	1405	2810
10	—	½	—	—	—	—	—

3) 792·4 = 264·1 3) 23270 = 7757 3) 706362 = 235454

ELEMENTARY WEDGE 29 DEGREES, IMMERSED SIDE.

No. of Ord.	Ords.	S.M.	Products for Area.	Ords.^{2}	Products for Volume of El. Wedge.	Ords.^{3}	Products for Moments of El. Wedge.
0	·3	½	·1	—	—	—	—
½	26·1	2	52·2	681	1362	17780	35560
1	28·8	1½	43·2	829	1243	23888	35832
2	30·6	4	122·4	936	3744	28652	114608
3	31·0	2	62·0	961	1922	29791	59582
4	31·1	4	124·4	967	3868	30080	120320
5	31·1	2	62·2	967	1934	30080	60160
6	31·1	4	124·4	967	3868	30080	120320
7	31·1	2	62·2	967	1934	30080	60160
8	30·7	4	122·8	942	3768	28934	115736
9	24·8	1½	37·2	615	922	15253	22879
9½	13·4	2	26·8	180	360	2406	4812
10	—	½	—	—	—	—	—

3) 839·9 = 279·9 3) 24925 = 8308 3) 749969 = 249989

Emerged wedge 235454

Both wedges 485443

TABLE II.

Calculation for Stability (Statical) at 29 Degrees Inclination.

Immersed Wedge. Inclin.	Functions of Squares.	S.M.	Functions for Volumes.	Functions of Cubes (both sides).	S.M.	Products of Functions of Cubes (both sides).	Cosines of Angles of Inclination.	Functions for Moments of Wedge.
0	6225	1	6225	332636	1	332636	·8746	290923
$14\frac{1}{2}$	6746	4	26984	364564	4	1458256	·9681	1411737
29	8308	1	8308	485443	1	485443	1·000	485443
			41517					3)2188103

Emerged wedge 39994

2) 1523

761

$\frac{1}{3}$ Ang. interval ·084

63·92

Long. interval 46·93

Im. wedge in excess 3000 cubic ft.

Stability corr. 3000 × ·506 = 1518

3)2188103

729367

$\frac{1}{3}$ Ang. interval ·084

61266·8

Longitudinal interval 46·93

Moment (uncorrected) 2875251

Correction (deduct) 1518

Vol. of displacement 553490)2873733

BN = 5·19

BG Sin. $\theta = 8{\cdot}25 \times {\cdot}4848$ = 4·0

GZ = 1·19 feet

Emerged Wedge.

Inclin. of Radial Planes.	Functions of Squares.	S.M.	Functions for Volumes.
0	6225	1	6225
$14\frac{1}{2}$	6503	4	26012
29	7757	1	7757
			39994

Function of area of W.P. at 29° (Immersed side) } 279·9 (Table I).

Function of area of W.P. at 29° (Emerged side) } 264·1 (Table I).

544·0

Longitudinal interval 46·93

Total Area of W.P. 25530 square ft.

Functions of Ords.^{2} of W.P. at 29° (Im. side) 8308 (Table I.)

Functions of Ords.^{2} of W.P. at 29° (Em. side) 7757 (Table I.)

2) 551

275·5

Longitudinal interval 46·93

25530)12929

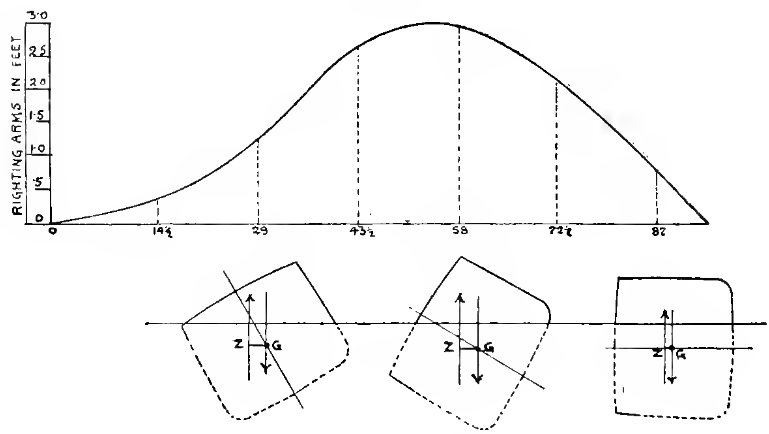
C.G. of W.P. towards immersed side ·506 feet.

Righting arms estimated in this way for the the three inclinations above mentioned are given in Table III., and the corresponding curve of stability in fig. 211. Of course, for the righting moment at any inclination, the ordinate given by fig. 211 must be multiplied by the displacement in tons, and this information is also included in Table III.

TABLE III.

Righting Arms and Righting Moments. S.S. 469′ 4″ × 56′ 0″ × 34′ 10″ Displacement, 15814 tons.		
Magnitude of Wedges in Degrees.	Righting Arms in Feet.	Righting Moments in Foot Tons.
$14\frac{1}{2}$	·34	5376
29	1·20	18977
$43\frac{1}{2}$	2·59	40958
58	2·94	46493
$72\frac{1}{2}$	2·13	33683
87	·81	12809

Fig. 211



In plotting a stability curve the correct contour near the origin is readily obtained if the tangent to the curve at that point is drawn. This may be done as follows:—At a point on the base line indicated by 57·3 degrees, erect a perpendicular and mark off on it, to the same scale as the righting levers, a distance equal to *GM*, the metacentric height. Join the point thus obtained with the origin of the stability curve. This is the tangent required, and it will be found that the curve will tend to conform with this line as it nears the origin. As an example, the stability curve of fig. 211 is reproduced in fig. 212, and the tangent to the curve at the origin is drawn as described, the metacentric height in this being ·85 feet. The explanation of the foregoing is as follows:—

By the metacentric method—

$$GZ = GM\,\theta$$

and

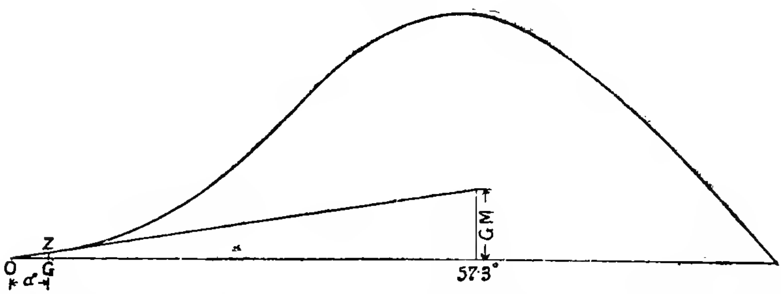
$$\frac{GZ}{\theta} = \frac{GM}{1}.$$

Now the denominators of these fractions are in circular measure; let them be expressed in degrees, θ becoming a°, and 57·3 degrees being substituted for 1, there being that number of degrees in the angle whose circular measure is 1; the equation may now be written—

$$\frac{GZ}{a^\circ} = \frac{GM}{57{\cdot}3^\circ}.$$

Referring to fig. 212, if GZ be the stability lever at a point close to the origin of the curve, and a° the distance in degrees measured along

Fig. 212.



the base line up to this lever, the small portion OZ of the curve will lie in a straight line, tangent to the curve at this place.

$$\left.\begin{array}{l}\text{The tangent of the angle which this line,}\\ \quad\text{and therefore the stability curve near}\\ \quad\text{the origin, makes with the base}\end{array}\right\} = \frac{GZ}{a^\circ}$$

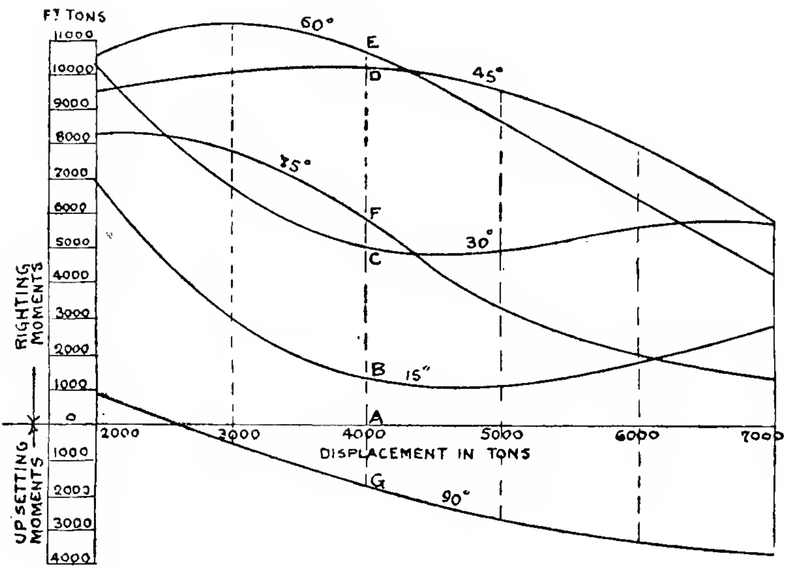
$$\text{or} \quad \frac{GM}{57{\cdot}3^\circ}$$

which is the value employed above in setting off the tangent to the curve at the origin.

CROSS CURVES.—The foregoing method of obtaining curves of stability is seen to involve considerable calculation. It has also another drawback. For most vessels several stability diagrams are required. Such curves should, indeed, be available for at least the four conditions referred to when dealing with the metacentric height, viz., the launching, light-ship, ballast, and fully-loaded conditions. By the above method we should thus have four troublesome calculations. And, moreover, if the vessel happened to be loaded or ballasted to draughts other than those originally allowed for, it would not be possible to ascertain her condition as regards stability without fresh calculations.

This defect in Barnes' method, as that by means of the wedges is called, was quickly seen when, a good many years ago, scientific attention was turned in earnest to this important subject. Many ingenious schemes were propounded for arriving quickly at the knowledge of a vessel's stability under all conditions of draught and lading, of which the best and simplest, and most generally employed, is that known as the cross-curve method. Here the abscissæ of the stability diagram is in terms of displacement, instead of

Fig. 213.

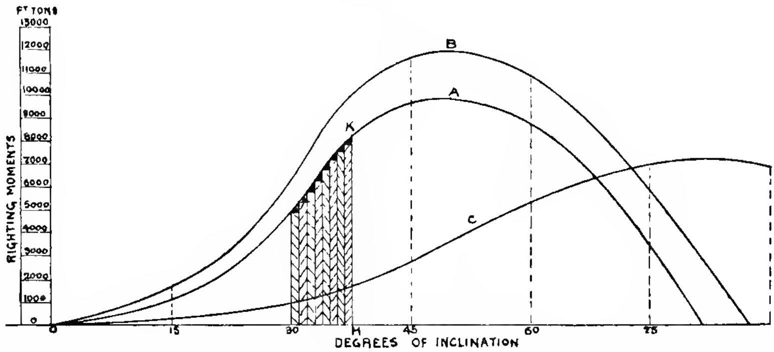


egrees of inclination, as in the ordinary case. In the complete diagram there is a series of curves, each exhibiting for one inclination variations in the righting arms or righting moments, as the case may be, with change in the displacement.

In fig. 213 a cross-curve stability diagram is depicted, with curves at 15, 30, 45, 60 degrees, and so on. The great value of this diagram is that it supplies us at once with the stability at every displacement or draught, and every inclination from the upright within the limits of the calculation. For example, suppose we require to know the vessel's stability when floating at a draught corresponding to a displacement of, say, 4000 tons; it is only necessary to draw a line at this point in the scale of tons perpendicular to the base of the diagram, to measure the distances *A B*, *A C*, *A D*, etc., cut off by the curves at 15, 30, 45 degrees, etc., to set off these distances as ordinates in a diagram having degrees of inclination as abscissæ, and draw a curve through the points so obtained. The result, subject to a correction, is an ordinary curve of stability such as might be obtained by Barnes' method (*see* curve *A*, fig. 214). We thus see that the cross curves lie in planes perpendicular to those of the ordinary curves, and it is from this circumstance the former derive their name.

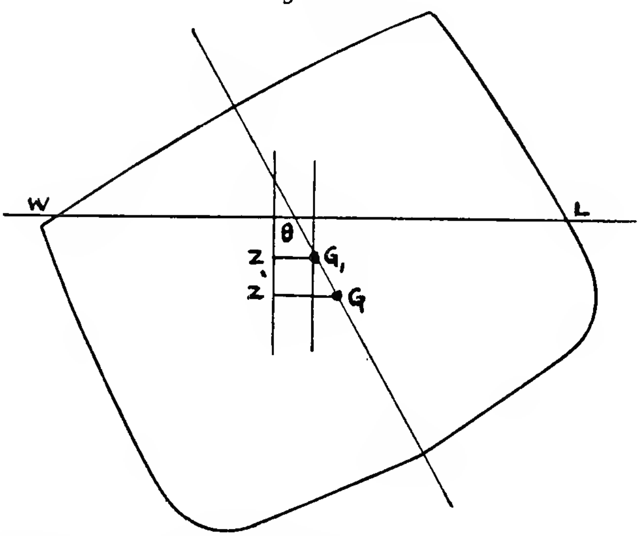
The relation between these two sets of curves may be simply illustrated

Fig. 214.



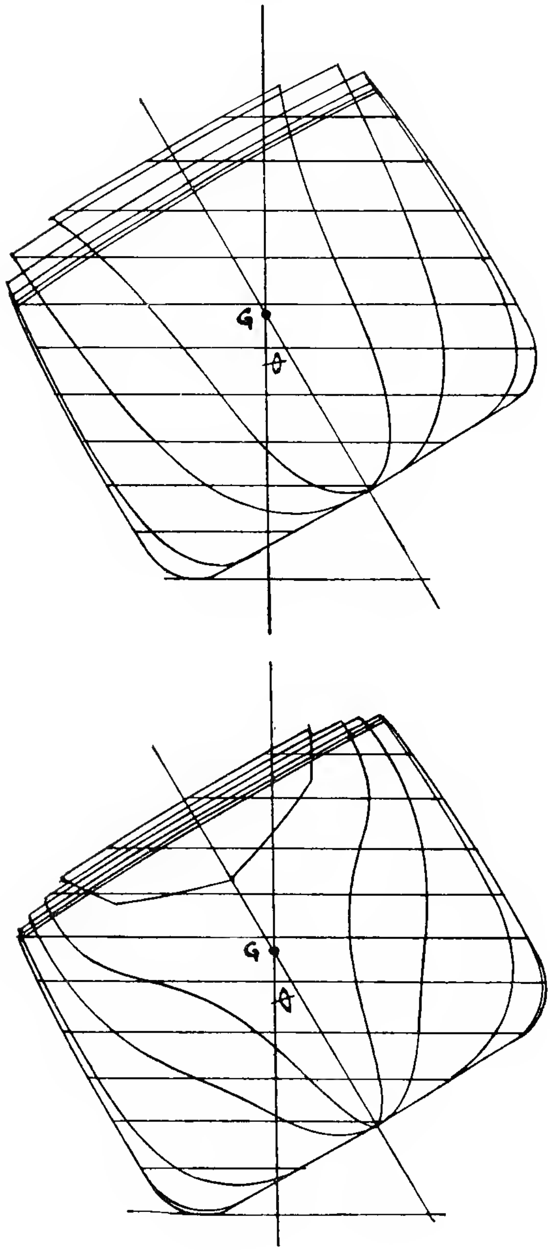
as follows:—Take a model representing half a solid cylinder, and assume it to be cut by planes perpendicular to the base and parallel to the longitudinal axis; these will intersect the curved surface of the model in straight lines. Next, suppose it to be cut by planes perpendicular to the base and

Fig. 215.



to the longitudinal axis; the lines of intersection with the model surface will obviously be half circles. If these half circles are considered to be ordinary curves of stability, the straight line intersections will be the corresponding cross curves.

Fig. 216.



CORRECTION FOR POSITION OF CENTRE OF GRAVITY.—It is to be noted that an ordinary curve of stability obtained by simply transferring distances from the cross-curve diagram, as described above, will truly represent the stability condition only if the centre of gravity, at the particular displacement, is coincident with that used in constructing the cross-curve diagram. This, of course, would not usually be the case, and, as already indicated, a correction must be made.

If the true position of the centre of gravity G be below the assumed one G_1 (*see* fig. 215), the ordinary curve of stability, as transferred, must be increased throughout by the amount $G G_1$ sin. θ, if the curve shows righting arms, and by $W \times G G_1$ sin. θ foot tons, if righting moments. Should the true position of the centre of gravity be above the assumed one, the process of correction would be the same, except that it would now, in each case, be a deduction. Assuming, for illustration, G_1 to be 6 inches above G, in the case represented by fig. 215, the stability curve would take the amended form B (fig. 214).

CALCULATIONS FOR CROSS CURVES OF STABILITY.—These are very simple and may be briefly described:—First, the body plan is prepared with transverse sections showing the form of the vessel at regular intervals, as in ordinary displacement calculations. It is an advantage to have the fore-and-after bodies drawn on each side, and this is sometimes done, although a separate drawing is frequently used for each body (*see* fig. 216). Next, tangent to the midship section, a base line is drawn inclined as required to the middle line of the vessel, and above the base, waterplanes are introduced at sufficiently close intervals to take account of the vessel's form, and to suit Simpson's Rule, these waterplanes intersecting the middle-line plane in parallel lines; usually an intermediate waterplane is introduced between the base line and the first waterplane, but this is not shown in fig. 216. As the deck edge is a point of discontinuity, a waterplane should occur there. Lastly, a position of the centre of gravity is assumed, and a vertical line drawn through it to form the axis of the stability moments. This position of the centre of gravity remains constant throughout all the cross-curve stability calculations.

The next step is to find the area of each waterplane and the transverse position of its centre of gravity from the chosen axis. It should be noted that a plane which does not cut the middle line has to be specially laid off and its area and centre of gravity determined.

To calculate the transverse position of the centre of gravity of a waterplane, which is not symmetrical about the middle line as in the present case, it is convenient to follow the method described for finding the transverse position of the centre of gravity of a half waterplane (*see* page 29), that is, taking one side of the plane at a time, put the ordinates and the ordinates squared separately through Simpson's Rule, and add the products in each case. Deal in the same way with the ordinates on the other side. Subtract one total of functions of squares from the other; divide by 2, and again by the sum of the totals of the products of the ordinates by their respective multipliers. The result is the distance of the centre of gravity from the line of intersection of the middle-line plane with the waterplane in question. All the planes are treated in this way, and the areas and transverse positions of the centres of gravity thus obtained, are combined as shown in the subjoined table to obtain :—

(1) The displacement below each of the waterplanes.

(2) The horizontal distance of the vertical through the centre of buoyancy corresponding to the displacement below each waterplane from the vertical through the centre of gravity.

Referring to the table, if H is the sum of the products of the waterplane areas with their respective multipliers up to any waterplane, and M the sum of these products multiplied in each case by the distance of the centre

Areas of Waterplanes.	Simpson's Multipliers.	Functions for Volumes.	Distances of C.G. of Planes from Axis.	Products for Moments about Axis.
A_1	$\frac{1}{2}$	$\frac{1}{2} A_1$	d_1	$\frac{1}{2} A_1 d_1$
$A_{1\frac{1}{2}}$	2	$2 A_{1\frac{1}{2}}$	$d_{1\frac{1}{2}}$	$2 A_{1\frac{1}{2}} d_{1\frac{1}{2}}$
A_2	$1\frac{1}{2}$	$1\frac{1}{2} A_2$	d_2	$1\frac{1}{2} A_2 d_2$
etc.	etc.	etc.	etc.	etc.
		H		M

of gravity of the plane from the chosen axis, then, h being the common interval between the planes, we have, up to the chosen waterplane—

$$\text{Displacement in tons} = H \times \frac{h}{3} \times \frac{1}{35}$$

$$\left.\begin{array}{c}\text{Distance in feet of centre of gravity}\\ \text{from axis}\end{array}\right\} = \frac{M}{H}.$$

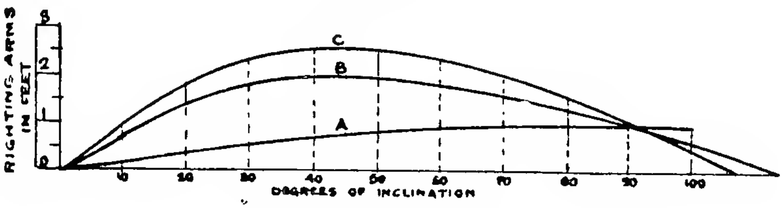
This work is performed for a sufficient number of displacements, and the results are plotted to obtain the cross-curve of stability at the given inclination, the displacements forming the abscissæ, and the horizonral distances of the centre of buoyancy from the axis, the ordinates. For other inclinations the process is the same. There are other methods of obtaining the data required for constructing cross-curves of stability. One of these consists in the use of a modification of Barnes' tables. For a description of this method the student is referred to Attwood's *Theoretical Naval Architecture.*

THE MECHANICAL INTEGRATOR.—Although cross-curves of stability can be correctly derived by arithmetical processes as above described, it is now customary to employ a Mechanical Integrator for this purpose, Amsler's being the one in common use. This instrument, which has several little wheels which run on the paper, also a long arm with a pointer, is placed in reference to the drawing in such a way that the movement of the pointer round the various sections of the body plan causes readings to be indicated on dials associated with the little wheels, from which, when affected by certain multipliers, the displacement and the moment of the displacement about a chosen axis may be derived. To find the distance of the centre of buoyancy from the axis it is only necessary to divide the moment by the displacement. Full descriptions of the work of calculating the stability in this way are provided in Reed's *Stability of Ships*, and Attwood's *Theoretical Naval Architecture*, and to these the student is referred for further information.

CAUSES WHICH INFLUENCE THE FORMS OF STABILITY CURVES—BEAM.—We saw in a previous chapter that beam is the element in design most powerfully affecting the height of the transverse metacentre above the centre of buoyancy; that, in fact, the height in similar vessels varies as the square of the beams. It is, therefore, clear that beam will also intimately affect the forms of stability curves, particularly at initial angles. This may be shown very simply, as follows. By the metacentric method we have—

$$\text{Righting arm at inclination } \theta = GM \times \text{Sin. } \theta,$$

Fig. 217.

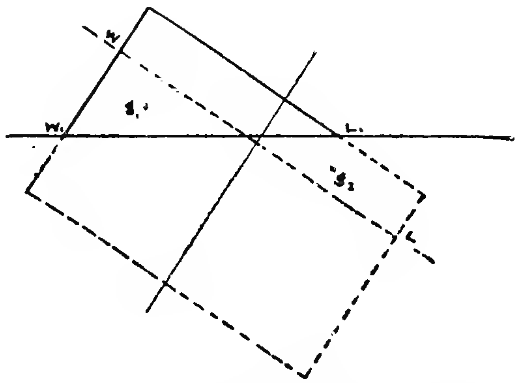


from which it is seen that the lever of stability varies directly as the metacentric height, and that, therefore, a broad vessel with a high metacentre and a large value of GM will have a stability curve initially steeper than a narrow one, and *vice versa*. Using the box form for purposes of illustration, we show in fig. 217 the actual effect on the whole stability curve of adding to the beam. Curve A is for a vessel 100′ × 20′ × 20′, floating at 15 feet draught, with the centre of gravity at 9 feet above the base. Curve B is for a vessel 100′ × 30′ × 20′, the particulars as to draught and position of centre of gravity remaining as before. As expected, the latter curve is seen to be steeper and to have enhanced values of righting arms, although these advantages are associated with a shortening of range. This shortening of range becomes more pronounced as the vessel increases in beam; curve C, for instance, corresponds to a vessel of 35 feet beam, with other particulars the same as the preceding cases.

INFLUENCE OF FREEBOARD.—Another important element affecting stability is freeboard, *i.e.*, clear height between load-waterplane and top-deck.

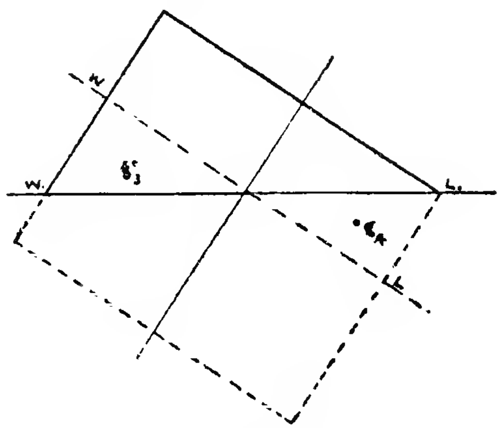
The box-shaped vessels depicted in figs. 218 and 219 are of the same breadth and draughts, but fig. 219 has greater freeboard. The stability curves for these vessels are depicted at *C* and *D* in fig. 220, and it is seen that, up to the angle at which the deck edge in the one with low freeboard becomes immersed,

Fig. 218.



they are identical. At this point the curve for the shallow vessel receives a check, and as the deck enters the water, quickly reaches a maximum and begins to fall. The curve of the other vessel, however, continues to rise rapidly with further inclination. The explanation is simple, if indeed not

Fig. 219.

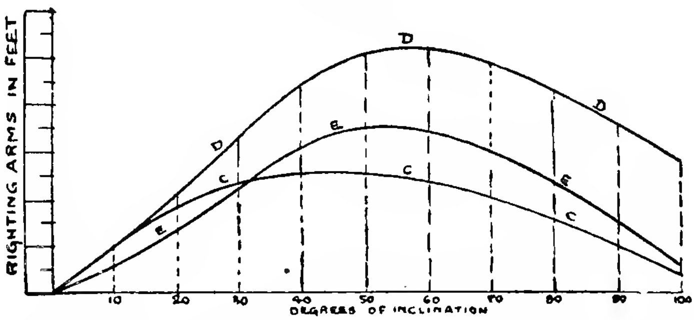


quite obvious. In the figures the vessels are shown inclined to the angle which just brings the high freeboard vessel's deck awash. The deck edge of the other one is, of course, considerably under water. The points $g_1 g_2$ mark the centres of buoyancy of the wedges of immersion and emersion in the shallow vessel, and the points $g_3 g_4$ the corresponding centres in the other. Clearly, the distance $g_1 g_2$ is less than $g_3 g_4$, and, as the volumes of the wedges are greater in fig. 219 than in fig. 218,

$$v_1 \times g_1 g_2 < v_2 \times g_3 g_4$$

where v_1 and v_2 are the volumes of the wedges in each case. As the quantities in this equation are measures of the stability, the relation between the curves at this inclination is apparent. Beyond this angle both deck edges are under the surface and the centres $g_3 g_4$ begin to approach each other, but much more slowly than the centres $g_1 g_2$; the actual differences in the values of the righting arms of the two vessels are shown in fig. 220. It should be stated that the centre of gravity is assumed at the same height in each case; in each case also the breadth is 35 feet, and draught 15 feet, while one has 5 feet freeboard (curve C), and the other 10 feet (curve D)

Fig. 220.



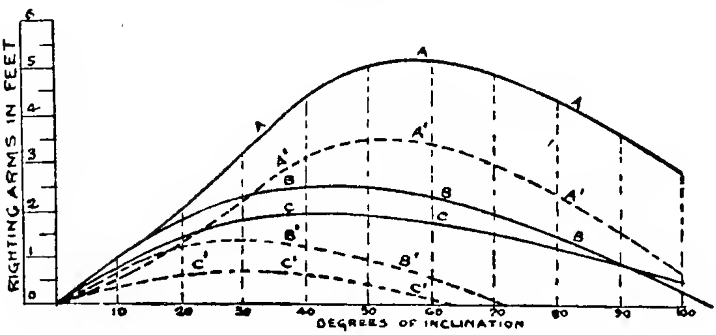
In this comparison we see at a glance the enormous influence of increased freeboard in augmenting the righting levers and extending the range of stability curves. In the low freeboard vessel the lever is a maximum at an inclination of 45 degrees; in the other, this is not attained until an inclination of 60 degrees is reached, the value of the righting lever at this angle being more than double the maximum in the other case.

Comparing figs. 217 and 220, we note that increased freeboard has a more beneficial effect on the forms of stability curves than increased beam. While both elements increase the righting arms, the former also extends the range which the latter rather tends to curtail.

In the foregoing comparisons we have assumed the centre of gravity to be at the same height throughout. Where the breadth alone is affected this assumption is fair enough, but where the depth is increased, as in the high freeboard vessel, a change in the position of the centre of gravity must be assumed to take place, in the direction of raising it. For a useful comparison, therefore, the stability curve should be modified to some extent. If we assume the centre of gravity to be raised 2 feet by the 5 feet addition to the depth, the corresponding stability curve will be that marked *E* (fig. 220). A considerable reduction is seen to have taken place in the lengths of the righting levers. These are, however, except at initial angles, still considerably greater than those of the low freeboard vessel with the lower centre of gravity. The range is also greater.

CHANGE IN HEIGHT OF THE CENTRE OF GRAVITY.—The powerful influence of raising the centre of gravity is manifest from the last illustration. To a great extent the position of this point is dependent upon the nature of the stowage. A shipmaster may therefore often make the stability of his vessel what he pleases. If he finds that she is deficient in stability, he cannot correct the defect by increasing the beam or the freeboard, but he can, it may be, stow the heavy weights low down in the hold, and the light ones higher up, and, by thus lowering the centre of gravity, attain the same end.

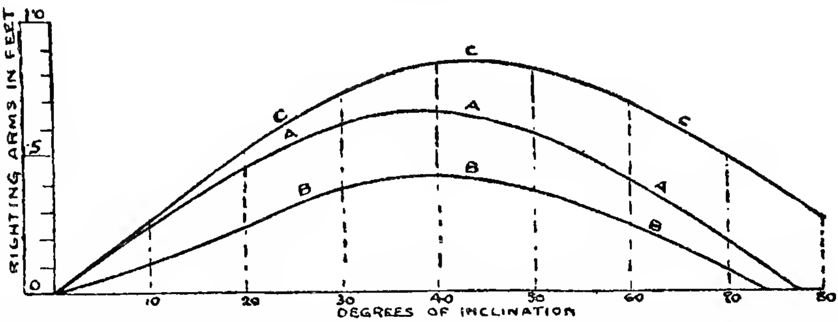
Fig. 221.



It may happen that the stowing of a vessel needs to be conducted with a view to attaining a high position of centre of gravity. Such would be called for in a broad, shallow vessel with a consequent high metacentre. A low centre of gravity in this case would mean a very large metacentric height, and this, as we shall see in the chapter on rolling, is by no means desirable. To get a high centre of gravity, the heavy portions of cargo should, of course, be stowed high up in the holds or in the 'tween decks, and the light cargo low down. To illustrate the effect on the forms of stability curves of raising the centre of gravity, we have performed this operation on three different box-shaped vessels, and the resulting curves, with those from which they have been deduced, are exhibited in fig. 221. *A*, *B*, *C*, are the original curves of stability; $A,^1B,^1C^1$, the curves as affected by a rise of 2 feet in the position of the centre of gravity in each case. Comment on the diagram is unnecessary.

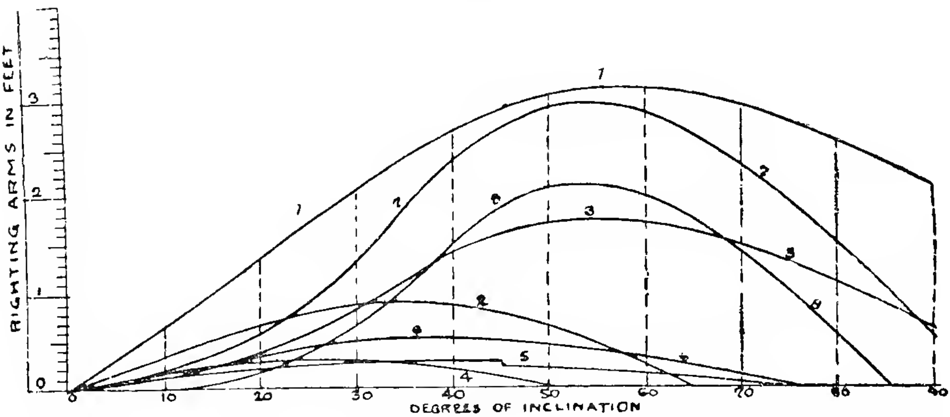
CURVES FOR VESSELS OF NORMAL FORM.—In showing the modifications in curves of stability due to variations in beam, freeboard, and position of centre of gravity, we have cited cases of box-shaped vessels only; vessels of ordinary form are, however, affected similarly. In fig. 222, for example, the effect due to beam and freeboard is shown.* *A* represents the stability curve of a cargo steamer whose dimensions are—length, 289·5 feet; breadth, 32·1 feet; depth, moulded, 23·1 feet. This vessel was as-

Fig. 222.



sumed to have 300 tons of coal in bunkers, and to be laden with a homogeneous cargo which completely filled the holds and 'tween decks and brought her down to her load-waterline. She had a regulation freeboard of 4 feet 7 inches. Curve *B* shows the effect of reducing the breadth by 2 feet, the conditions of lading being as before. The righting power, as

Fig. 223.



in the case of the box vessel, is seen to be much reduced by this alteration.

Curve *C* illustrates the stability curve of the same vessel after an increase of 6 inches in the freeboard, the density of the cargo being kept

* See Reed's "Stability of Ships," p. 104.

as before, and the surplus space assumed to be at the ends of the 'tween decks.

Further illustrations of stability curves of actual ships are depicted in fig. 223, the particulars of the vessels being given in the table below. Considerable variation in stability is here shown. Compare, for example, curves 1 and 4. The first may be considered an example of excessive stability; the other of deficiency in this respect. This was borne out when the vessels were on service. During the voyages made by each when in the conditions stated in the table,* No. 1 vessel met severe weather and

	Description.	Length.	Breadth.	Depth.	Meta-centric Height	Free-board.	Displacement.	Cargo.
		feet.	ft. ins.	feet.		ft. ins.	tons.	
1	Three-deck vessel -	285	35 0	24.5 REG.D	3·5	6 6	3800	Pig iron.
2	Raised quarter-deck vessel -	245	32 0	17·6 ,,	1·5	3 0	2350	Coal or grain.
3	Spar-deck vessel -	245	32 0	24·0 ,,	·8	8 0	2730	Coal or grain.
4	Three-deck vessel, flush, except for small forecastle	245	33 0	23·0 ,,	·7	4 0	3500	Coal or grain.
5	Quarter-deck vessel	135	21 0	ft. ins. 11 10 M.D.	·75	1 1½	612	Coal.
6	Quarter-deck type, erections ⅔ length	220	32 0	17 0 ,,	·92	2 1½	2300	Coal.
7	Shelter-deck vessel modern type -	470	56 0	34 10 ,,	·85	7 9	15800	Coal or grain.
8	Shelter, deck vessel, modern type -	470	56 0	34 10 ,,	·2	7 9	15800	General.

rolled so much that the hull was found to be considerably strained. Had her initial stability been more moderate, her behaviour would have been better, and there would probably have been little or no straining. No. 4 vessel, on the other hand, is a type of many that were lost when making voyages laden with coal or grain. The elements conducing to deficient stability in this case are the small metacentric height, low freeboard and almost flush deck, the only erection being a short forecastle. Vessels Nos. 2 and 3 are of about the same dimensions as No. 4, but they have certain important differences which account for their improved stability. No. 2, for instance, although she has less freeboard than No. 4 to the main deck, has a long raised quarter-deck erection, which increases the reserve buoyancy; her metacentric height is also greater. No. 3 vessel has double the freeboard of No. 4, and we are already familiar with the effect of this element on the stability.

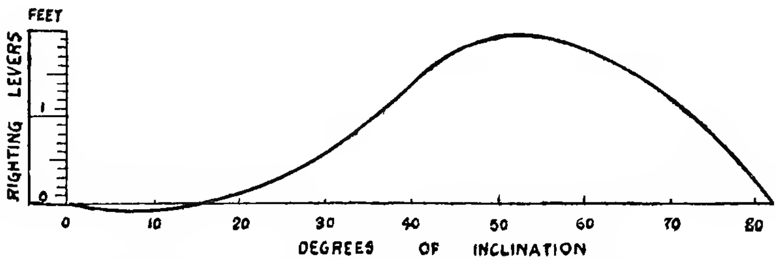
Vessels Nos. 5 and 6 are additional examples of weakness in stability. No. 5, indeed, was lost on her maiden voyage when in the condition given above. Her behaviour after leaving port showed her to be very tender.

*Examples 1, 2, 3 and 4 are from a paper by the late Mr. Martell in the T.I.N.A. for 1880.

She first listed to one side, remained in that position for some time, then returned to the upright, but almost immediately lolled over to the other side. She continued for some time to heel from one side to the other in this manner, until eventually she was struck by a sea which caused her to heel further over, from which position she could not right herself. The water then poured into the engine-room through the casing door, and the vessel went down by the stern.* The curve of No. 6 is not so bad as that of No. 5. The maximum lever is ·51 feet, and the range is seen to be 78 degrees. But the vessel was always considered very tender and had to be handled with care.

Curves Nos. 7 and 8 exhibit two stability conditions of a large modern cargo vessel of good freeboard. No. 8 is particularly interesting, as it shows that, with so small a metacentric height as ·2 feet, there may be associated a stability curve quite satisfactory as to range and lengths of righting arms. In some instances, indeed, vessels having curves of stability of large area and range have had negative metacentric heights. Such vessels are unstable

Fig. 224.



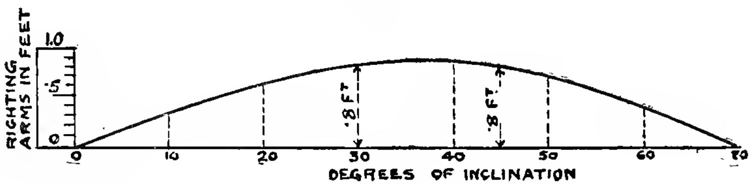
in the upright position and loll over to one side or the other. In the case of vessel No. 8, if the centre of gravity were raised 6 inches, the metacentric height would be $-·3$ feet, and the vessel would be unstable from the upright to an angle of 13 degrees, her stability curve lying below the base line. She would thus loll over to this angle for her position of equilibrium. Beyond 13 degrees the curve would rise above the base line, the maximum lever reaching 1·95 feet at an angle of 53 degrees, and the range 81 degrees. The curve is depicted in fig. 224, which shows the vessel to have considerable reserve of stability. In such a case as this, a vessel's ultimate safety obviously does not depend on there being a positive metacentric height (*see* p. 194). The latter, however, is necessary in order that she shall float upright and not be too easily inclined by the action of external forces. In the present instance, to gain a positive *GM*, if the vessel were in quiet water, the centre of gravity might be lowered by filling a compartment of the double bottom, but it may be pointed out that it would not be proper to do such a thing in all cases of instability in the upright condition. It

* See an interesting paper by Mr. Pescod on "Stability of Small Steamers," read before the North-East Coast Institution of Engineers and Shipbuilders, in 1903.

would have been improper in the case of vessel No. 5 for instance, when in the condition described above, and would probably have hastened the disaster which eventually came upon her (see also chapter on Loading and Ballasting).

SAFE CURVE OF STABILITY.—The curves of vessels Nos. 4, 5 and 6, which show unsafe conditions of stability, also cause the question naturally to arise, "What *does* constitute a safe minimum curve of stability?" In attempting an answer for any given case, two things are to be chiefly borne in mind—the size of the vessel and the nature of her cargo.

Fig. 225.



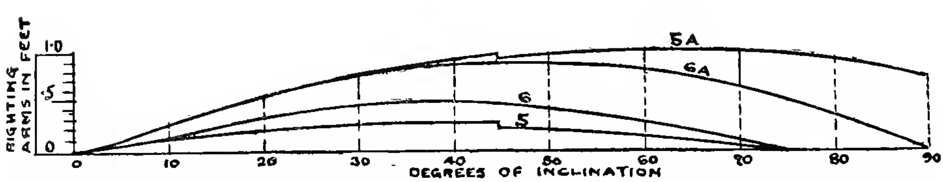
From the relation:—

$$\text{Righting moment} = \text{displacement} \times GZ$$

we make the deduction that vessels of small displacement will be more affected by movements of weights on deck or water in holds, by shifting of cargo or by the action of wind or waves, than those of large displacement with the same curve of righting arms, and that, therefore, a curve of righting arms suitable enough for a very large vessel, may be quite inadequate for a very small one.

Again, vessels intended chiefly to load bulk cargoes, such as grain, which are liable to shift in heavy weather, should have a margin of righting power in excess of the safe minimum limit. Many cargo vessels have to take all

Fig. 226.



kinds of cargoes, and for them it is not easy to decide upon a minimum curve of stability. A well-known firm of shipbuilders, however, have fixed upon a curve for ordinary medium-sized cargo steamers, of which fig. 225 is a copy, and experience has shown that vessels so provided are safe and comfortable sea boats. Referring to this figure, it will be observed that the righting arm at 30 degrees and 45 degrees is, in each case, ·8 feet, and that there is a range of 70 degrees.

In fig. 226 we have reproduced the stability curves Nos. 5 and 6 (fig. 223), and have shown in 5 *A* and 6 *A* the corresponding safe minimum curves, using fig. 225 as a basis. To obtain a minimum righting arm of ·8 feet at 30 degrees in these cases, the centre of gravity in No. 5 would require to be lowered 1·04 feet, and in No. 6, ·5 foot, making the metacentric heights 1·79 feet and 1·42 feet, respectively. It will be noticed that the righting arm at 45 degrees, and the range in each case, exceed those of the standard curve (*see* fig. 225). These curves are for small vessels, and, for reasons already given, we do not say that even the modified curves leave nothing to be desired; the stability conditions, however, exhibited by them are a great improvement on those of the original curves.

DYNAMICAL STABILITY.—The dynamical stability of a vessel at any angle is the work done in inclining her from the upright to that angle. It should be carefully distinguished from statical stability, which is the moment supporting the vessel at the given inclination and tending to return her to the original position. In heeling a vessel, work is done as follows:—

(1) In raising the centre of gravity.
(2) In depressing the centre of buoyancy.
(3) In creating waves and eddies.
(4) In surface friction.

Fig. 227.

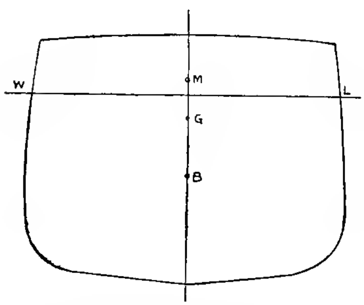
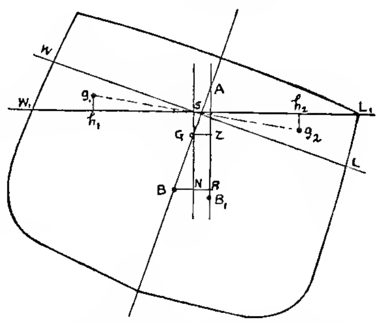


Fig. 228.



Comparing figs. 227 and 228, which show a vessel upright and inclined, respectively, we note the movement of the centres of gravity and buoyancy, the former point being obviously nearer the load-waterplane, and the latter further from it, in fig. 228 than in fig. 227. Items (1) and (2) constitute the dynamical stability as usually calculated; items (3) and (4) cannot, from the nature of the case, be correctly estimated, and in practice are therefore ignored. The result, however, is on the safe side.

The quickest way of obtaining the dynamical stability is by means of a curve of moments of statical stability; for, as shall be shown presently, the area of such a diagram from the origin to any angle truly represents the work done on the vessel in inclining her to that angle, omitting, of course, the effect of surface friction, wave and eddy-making resistances.

As a preliminary, consider the following:—If a force F lbs. acting on a body causes it to move in any direction through a distance h feet, then, works on mechanics tell us that the—

$$\text{Work done on the body} = F \times h \text{ foot lbs.}$$

For example, if F be 10 lbs. and h 5 feet—

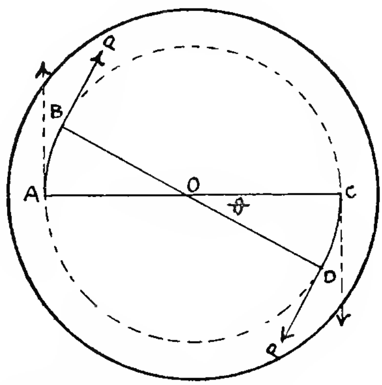
$$\text{Work done} = 10 \times 5 = 50 \text{ foot lbs.}$$

If the force moves along a curved path, the length of the path traversed is employed in calculating the work done. Consider now a case of two equal parallel forces acting on a body free to turn. The body will revolve about an axis passing through its centre of gravity. If the points of application of the forces be fixed, the latter will move with the body, and in turning it through any angle θ (*see* fig. 229) if the forces be in lbs. and the distances in feet,

$$\begin{aligned}\text{Work done} &= (P \times A\,B) + (P \times C\,D) \\ &= P(A\,B + C\,D) \text{ foot lbs.}\end{aligned}$$

Since $A\,B = A\,O \times \theta$, and $C\,D = O\,C \times \theta$, θ being the circular measure of the angle, we may write—

Fig. 229.



$$\begin{aligned}\text{Work done} &= P(A\,O + O\,C)\ \theta \\ &= P \times A\,C \times \theta \text{ foot lbs.,}\end{aligned}$$

that is, the work done up to any angle is given by the product of the turning couple and the circular measure of the angle.

Applying these principles to the case of a ship, let a vessel be assumed floating at rest in stable equilibrium, then let an external heeling couple be supposed to act upon her. If, starting at zero with the vessel upright, this heeling couple be assumed to grow so as always to be equal to the righting couple, the righting moment diagram at any point will represent the value of the heeling or upsetting couple at the same point, and, generally, the curve of righting moments will also represent the curve of upsetting moments.

Reverting to fig. 214, curve A shows an ordinary curve of righting moments. Consider an ordinate at 30 degrees, say. On the above assumption, it gives the value of the upsetting couple at that angle. Let now the vessel be further inclined through one degree. If the upsetting couple be assumed constant through this small inclination, the work done by it will be equal to the ordinate at 30 degrees multiplied by the circular measure of one degree. If the base line of the righting moment diagram be in terms of circular measure, and a line parallel to the base be drawn through the top of the ordinate at 30 degrees, the work done by the upsetting couple will be represented by the area of the little rectangle thus enclosed. At 31 degrees, if the upsetting moment be assumed augmented so as to equal the righting moment at that angle, and to remain constant while the vessel is heeled through one degree, the work done will be represented by the little rectangle between 31 and 32 degrees, the base line, and a line parallel to the base through the ordinate at 31 degrees.

Proceeding thus by intervals of one degree, the work done by the upsetting moment from the origin to any angle OH will be represented by the area OHK less the sum of the little triangles between the curve and the tops of the little rectangles (those indicated in fig. 214 are shown in black). But by making the intervals infinitely small, the difference between the area OHK, and the work done by the upsetting couple, is made infinitely small. We may, therefore, ultimately say, that on inclining the vessel through the angle OH, the work done, or dynamical stability, is truly represented by the area of the curve of statical stability from the origin up to that angle. An ordinary curve of stability thus assumes a new importance, since, besides showing the variation in the statical righting moment from point to point, as the vessel is inclined from the upright, it also measures the amount of energy that must be expended to incline her.

Practical Example.—Assuming the values of the righting moments in curve A, fig. 214, at intervals of 15 degrees, to be 0, 1200, 4900, 9500, and 8000 foot tons, respectively, what is the dynamical stability at 60 degrees? This is merely a case of obtaining the area of curve A from the origin to an ordinate at 60 degrees by means of Simpson's Rules. It is convenient to arrange the figures as follows :—

Degrees of Inclination.	Righting Moments in foot tons.	S.M.	Functions of Moments.
0	0	1	0
15	1200	4	4800
30	4900	2	9800
45	9500	4	38000
60	8000	1	8000
			60600

$$\left.\begin{array}{r}\text{Dynamical stability at 60}\\ \text{degrees inclination}\end{array}\right\} = 60600 \times \frac{\cdot 2618}{3} = 5288 \text{ foot tons,}$$

·2618 being the circular measure at 15°.

Curve C, fig. 214, shows the complete curve of dynamical stability for this vessel.

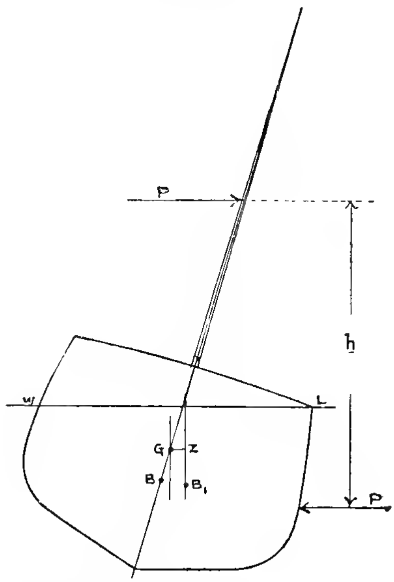
A knowledge of the dynamical stability* is particularly useful in the case of sailing-ships as a guide in fixing the area and distribution of the sails. In such estimates, the sails are assumed braced to a fore-and-aft plane and the pressure to act dead upon them.

Let fig. 229 represent a sailing-vessel heeled to, and held steadily at, an angle θ. Here the upsetting and righting moments are obviously equal. That is,

$$P \times h = W \times GM \times \text{Sin } \theta,$$

P being the total wind pressure in tons at the angle θ, h the vertical distance

Fig. 229.



*The dynamical stability may also be obtained by an equation known as Moseley's formula. Simply stated, this consists in multiplying the vertical movement between the centre of buoyancy and centre of gravity during the inclination by the weight of the vessel. Reverting to figs. 227 and 228, BG is the distance between the centres when the vessel is upright; B_1Z, the distance when she is inclined. Therefore—

$$\left.\begin{array}{r}\text{Work done in inclining}\\ \text{vessel to given angle}\end{array}\right\} = (B_1 Z - BG)\, W \text{ foot tons.}$$

Now, $B_1 Z = B_1 R + RZ$, and $RZ = BG \cos \theta$. To find $B_1 R$ we must multiply the volume (v) of the wedge of displacement transferred across the ship by the vertical travel of its centre of gravity *i.e.*, $g_1 h_1 + g_2 h_2$, and divide by the volume of displacement (V). Thus,

$$B_1 R = \frac{v}{V}(g_1 h_1 + g_2 h_2).$$

Substituting in first equation, we get—

$$\text{Work done in foot tons} = W\left(\frac{v}{V}(g_1 h_1 + g_2 h_2) + BG \cos \theta - BG\right)$$

$$= W\left(\frac{v}{V}(g_1 h_1 + g_2 h_2) - BG(1 - \cos. \theta)\right)$$

in feet at the same inclination between the centre of effort, or centre of gravity of the sail area, and the centre of lateral resistance, the point through which the resultant fluid pressure is taken to act, W the displacement in tons, and GM the metacentric height in feet. It may be mentioned that the centre of lateral resistance is usually assumed to be at mid draught.

From the above formula, the magnitude of the angle to which a sailing-ship will heel under a given sail area and wind pressure is seen to vary inversely as GM, and as it is desirable to have the deck as level as possible, the importance of a large GM in this class of vessel is apparent. In medium-sized sailing-ships it should not be less than 3 to $3\frac{1}{2}$ feet. In any special case, the best way of showing the effect of the wind pressure on the stability is to draw a curve of upsetting moments due to the wind pressure on the same diagram as the corresponding curve of stability. This may be done as follows: First, the upsetting moment in the upright position is calculated, this being equal to the total wind pressure on the sails in tons multiplied by the vertical distance in feet between the centre of effort and centre of lateral resistance. Then the upsetting moment acting when the vessel is inclined to the upright is obtained. In this case, the effective sail area, and therefore, the effective total wind pressure, are reduced, being equal to the values corresponding to the upright position multiplied by the cosine of the angle of inclination; also the effective leverage is equal to the leverage in the previous case multiplied by the cosine of the angle of inclination. Thus, at any angle θ—

$$\text{The upsetting moment} = \left\{\begin{matrix}\text{Upsetting moment in} \\ \text{upright position}\end{matrix}\right. \times \cos.^2 \theta \text{ foot tons.}$$

Employing symbols, let—

A = Total sail area in square feet.

H = Distance in feet between centre of effort and centre of lateral resistance (vessel upright).

p = Wind pressure per square foot in lbs.

Then, with vessel inclined at some angle θ,

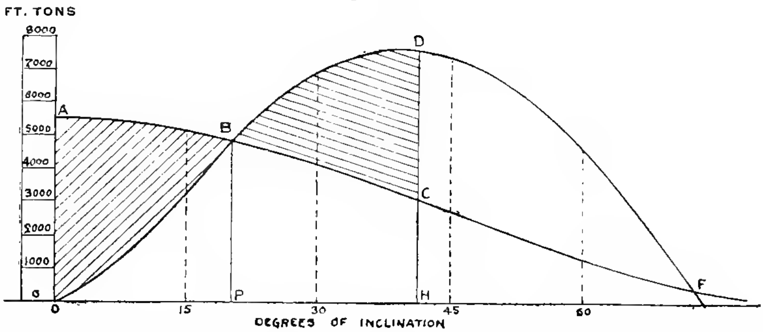
$$\begin{aligned} \text{Effective sail area} &= A \cos. \theta. \\ \text{Effective lever} &= H \cos. \theta. \\ \text{Upsetting Moment} &= A \cos. \theta \times H \cos. \theta \times p \\ &= A H p \cos.^2 \theta. \end{aligned}$$

In calculating the steady angle of heel for a vessel under full sail, it is usual to assume a wind-pressure of 1 lb. per square foot, which is taken to be the force of the wind in a fresh breeze. In estimating the effect of a squall, however, a much larger wind-pressure is assumed.

Values of upsetting moments obtained in this way are set off at corresponding points on the base line of the stability diagram, and a "wind curve" drawn.

EFFECT OF A SQUALL.—In fig. 230, OBD is an ordinary stability curve, and $OABCF$ a curve of upsetting moments due to the wind pressure or wind curve constructed in the way described. Referring to this diagram, it will be observed that at an inclination *O P* the upsetting and righting moments are equal. This tells us that but for the energy which the vessel has stored up in virtue of the unresisted moment area *O B A*, she would be held at the inclination *O P*. As it is, she passes beyond *P* to some angle *O H*, when the energy of motion is overcome by the stability reserve *B C D*. *O H* is approximately twice *O P*, thus a sudden squall striking the vessel

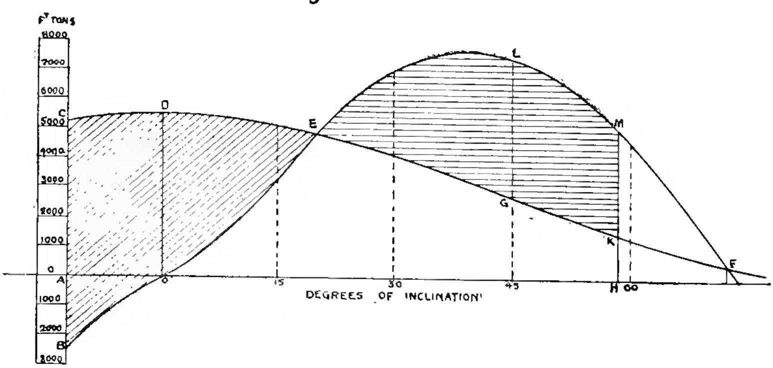
Fig. 230.



when upright causes her to heel to about twice the angle that a steady force gradually applied would heel her to.

The wind curve frequently crosses the stability curve at two points, such as *B* and *F*. The portion of the stability curve below the line *A F* is absorbed by the upsetting force, while the portion above the line, namely, *B D F*, is called the reserve of dynamical stability, and as we have seen, is available to overcome any energy of motion which the vessel may have

Fig. 231.



when she reaches the inclination at which the wind and stability curves have equal ordinates.

In the case of a sailing-ship rolling freely at sea under full sail, probably the greatest demand will be made on the reserve of dynamical stability when a squall strikes her as she is about to return to the upright after completing a roll to windward.

Fig. 231 illustrates this condition. The stability curve above the line refers to inclinations on one side of the upright; that below the line to those on the other side. The inclination to windward is represented by *O A*. At this point there is a righting lever *A B* tending to return the vessel to the upright position; and as the energy which heeled her thus has been expended, the influence of the righting moment is about to be felt. At this instant the squall is supposed to strike the vessel. *A C D E G F* is the wind curve, and the returning moment becomes suddenly increased from *A B* to *B C*, causing the vessel to return rapidly to the upright, her angular velocity continuing to increase until the angle corresponding to the point *E* on the other side of the upright is reached, when it is a maximum. Beyond this, the righting moment is in excess of the upsetting moment, and as the vessel becomes further inclined to leeward, her kinetic energy and angular velocity gradually decrease, the vessel coming finally to rest at some angle *O H*, when the excess of upsetting moment, represented by the area *B C E*, is absorbed by the excess of dynamical stability *E K M E*. Her energy of motion being now expended, the vessel begins to return by virtue of her excess of righting moment and, if the wind curve be assumed to remain as before, she will oscillate for a little about the angle corresponding to the point *E* and finally come to rest at that angle.

We have neglected the retarding effect of the friction of the water and the hull surface, and of such head resistances as bilge keels. These considerably reduce the inclinations to which the wind heels the vessel, and if the wind were suddenly to fall, would, with the assistance of the air resistance on the sails, gradually bring her to rest.

QUESTIONS ON CHAPTER IX.

1. What is a curve of statical stability? How is such a curve usually drawn?

2. Distinguish between the terms "righting arm" and "righting moment." A vessel has a displacement of 4000 tons and a metacentric height of 2 feet; what are the values of the righting arm and righting moment when the vessel is heeled to an inclination of 10 degrees?
Ans.—R.A. = ·348 foot; R.M. = 1392 foot tons.

3. In the case of vessels of ordinary form, correct values of righting arm and righting moment cannot be obtained at large inclinations by using the metacentric height; explain why. For what special form of vessel is the metacentric height method correct?

4. Construct a curve of righting arms for a vessel of cylindrical section, 15 feet in diameter, floating with its axis in the waterplane, the centre of gravity being 18 inches below the middle of the section.

5. A vessel's metacentric height is 2 feet 6 inches; show how you would construct the tangent to the curve of righting arms at the origin, and prove that your method is correct in principle.

6. Show that in a submarine vessel floating below the surface, the centre of buoyancy and metacentre coincide with the centre of bulk, and explain what in this special case are the conditions of equilibrium.

7. If, in the previous case, the centre of gravity be below the centre of bulk, show that if the vessel be turned about a horizontal axis passing through the centre of bulk, the curve of stability will be the same whatever be the direction of the axis.

8. Prove Atwood's formula for the statical stability of a vessel at any angle of heel. Show how a statical stability diagram is constructed, and explain its uses.

9. Explain the terms "angle of maximum stability," and "range," as applied to curves of stability. A box-shaped vessel 35 feet broad, and 35 feet deep, floats at a level draught of 17 feet 6 inches. The metacentric height of the vessel being 2 feet, construct the curve of righting arms, indicating the "angle of maximum stability," the value of the "maximum righting lever," and the "range."

10. Describe in detail how you would proceed to obtain the statical stability at large angles of inclination of a vessel of known form.

11. A vessel having a deep forward well ships a heavy sea. Assuming the water ports to be set up with rust and inoperative, discuss the effect of the filling of the well on the vessel's stability, and state whether her safety is likely to be thus endangered.

12. In Barnes' Method of calculating the statical stability of a vessel, show clearly how the "wedge correction" is made. The uncorrected sum of the moments of the wedges of immersion and of emersion for an inclination of 30 degrees is 263,000, and the vessel's displacement is 2000 tons. The volume of the layer is 600 cubic feet, and the horizontal distance of the centre of gravity of the radial plane at 30 degrees from the intersection with the middle-line plane is 1·5 feet. Find the value of the righting arm (1) assuming the immersed wedge in excess and the centre of gravity of the radial plane on the immersed side; (2) assuming the immersed wedge in excess and the centre of gravity of the radial plane on the emerged side; (3) assuming the emerged wedge in excess and the centre of gravity of the radial plane on the immersed side; (4) assuming the emerged wedge in excess and the centre of gravity of the radial plane on the emerged side.

NOTE.—The centre of gravity of the layer may be assumed to be in the same vertical with the centre of gravity of the radial plane, and $BG = 5$ feet.

Ans. $\begin{cases} \text{(1) and (4), Righting Arms} = 1·24 \text{ feet.} \\ \text{(2) and (3),} \quad \text{,,} \quad = 1·27 \text{ ,,} \end{cases}$

13. A vessel whose displacement is 3000 tons, has a righting-arm of 1 foot at an inclination of 30°. Cargo weighing 300 tons, whose centre of gravity is in the middle line at a depth of 1 foot below the common centre of gravity, is discharged; and the vertical through the centre of buoyancy of the layer through which the vessel rises when at an inclination of 30° is 6 inches on the immersed side of the vertical through the centre of buoyancy of the vessel corresponding with the load draught at that inclination. Find the length of the righting arm after the removal of the cargo.

Ans.—$\frac{8}{9}$ths of a foot.

14. What are cross curves of stability? How are these related to the ordinary stability curves?

15. A vessel of constant circular section, 120 feet long and 14 feet in diameter, has its centre of gravity 1·5 feet below the axis; construct to scale cross curves of stability for transverse inclinations of 30°, 60°, and 90°.

16. Referring to the previous question, deduce an ordinary curve of stability for the vessel when floating with its axis in the waterline. If the centre of gravity be 9 inches below the position assumed in making the cross curves, show how the necessary correction would be made at the various inclinations, and plot the new curve.

17. The plans of a vessel being given, state fully how you would prepare the body plan for the calculation of a cross curve of stability.

18. A vessel inclined transversely is cut by a series of horizontal equidistant planes at intervals of 3 feet, the intersection of the middle-line plane being parallel to the top of keel. The first plane touches the vessel's bottom tangentially. The areas of the successive planes are 0, 300, 590, 880, and 1150 square feet, and the horizontal distances in feet of their respective centres of gravity from the vertical through the vessel's centre of gravity, omitting the tangent plane, are 1·4, ·9, ·4, and ·1, on the immersed side. Construct the cross curve of stability.

19. What are the causes which influence the forms of curves of stability? Give an example of such curves for

(1) a flush-deck vessel of low freeboard;
(2) the same vessel fitted with a continuous watertight shelter deck.

20. Discuss the comparative effect on curves of stability of increase of breadth and increase of freeboard. Taking a rectangular vessel 100 ft. long, 20 ft. broad, 15 ft. deep, floating at a level draught of 12 ft., with the centre of gravity at 7 ft. above the base; show the effect on the stability curve of

(1) an increase of 4 ft. in beam,
(2) an increase of 4 ft. in freeboard,

the draught and position of centre of gravity remaining the same in each case.

21. What is meant by the dynamical stability of a vessel? In inclining a vessel from the upright position explain the several ways in which work is done.

22. A rectangular pontoon 100 feet long, 25 feet broad and 25 feet deep, floats in salt-water at half depth with one of its sides horizontal; the metacentric height is 1 foot. Calculate the dynamical stability at an angle of 45°.

Ans.—487 foot tons.

23. Prove that the work done in inclining a vessel from the upright to any angle, is equal to the area of the curve of righting moments up to that angle.

24. The ordinates of a curve of righting arms measured at equal angular intervals of 10°, starting from the upright, are—0, ·4, ·7, ·9, and 1·0 feet. Find the dynamical stability at 40° inclination, the displacement being 2500 tons.

Ans.—1102 foot tons.

CHAPTER X.

Rolling.

THE time that a vessel, rolling freely in undisturbed water, takes to complete an oscillation from port to starboard, or *vice versa*, is called her period of a single roll. Theoretical investigations in this subject are based on the assumption that the rolling medium is a perfect or frictionless fluid, so that in calculating the period of roll, the fact that water offers substantial resistance to the movement of the vessel is ignored. The result thus obtained is found to be nearly true, since, from actual rolling experiments, fluid resistance, while greatly limiting the arc of oscillation, appears to have little influence on the period.

Early investigators were wont to think that if a vessel had great initial stability, and was, therefore, difficult to move, she would also be steady in a seaway. They were struck with the apparent analogy between a rolling ship and an oscillating pendulum, and thought that a ship might be looked upon as a simple pendulum suspended at the metacentre of length equal to GM, the distance between the metacentre and the centre of gravity.

Now, the period in seconds of a single swing of a simple pendulum, from left to right, or *vice versa*, is

$$T = 3{\cdot}1416\sqrt{\frac{l}{g}},$$

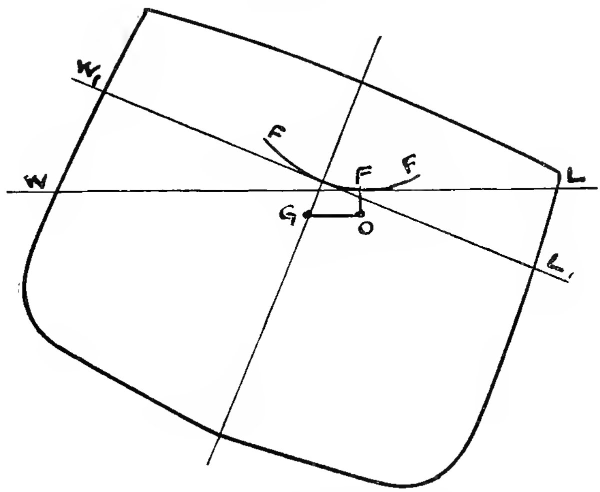
where l is the length of the pendulum, and g the acceleration due to gravity.

If the above analogy between the pendulum and the ship were correct, GM might be substituted for l in this formula, and, consequently, the ship's rolling period would lengthen with increase in GM. We find, however, such to be by no means the case, all experience going to show that vessels of small metacentric heights are of longer periods, that is, make fewer rolls per minute, than those having large metacentric heights. Thus, the assumption that a ship is a simple pendulum, with its whole weight concentrated at the centre of gravity, and with a fixed axis of oscillation at the point M, is clearly an erroneous one.

As a matter of fact, a ship has no fixed axis of oscillation. The instantaneous axis is for most vessels not at M, but in the vicinity of the centre of gravity, and it is usual to assume it as passing through that point, While accepting this as a fair approximation, it must not at the same time be forgotten that the centre of gravity itself, though fixed relatively to the ship, really describes a path in space as the vessel rolls. To obtain the instantaneous axis we may proceed as follows:—

Referring to fig. 232, let *W L* be the waterplane, *F F* the curve of flotation, *i.e.*, a section of the surface tangent to the various waterplanes which cut off a constant displacement as the vessel rolls, and *G* the centre of gravity. Now, neglecting the presence of the ship, assume the surface of flotation and the level water surface to become solid, and the former to roll or slip without friction along the latter as the vessel oscillates. *F*, the point of contact of the surfaces *F F* and *W L*, is a point in the oscillating vessel, and will move, at any instant, about a centre somewhere in the line *F O*. Another determinable point in the vessel is the centre of gravity. It has vertical motion only, since the forces acting when the vessel is rolling freely

Fig. 232.



are purely vertical, therefore, the centre about which *G* turns is in the line *G O*. The axis of the vessel at the instant considered is obviously at *O*, the point of intersection of the lines *F O* and *G O*. The point *O* in most cases is near *G*, so that very little error is introduced by the assumption that the axis passes through *G*.

Let us consider what actually takes place when a ship is rolling unresistedly in still water. This case is a purely hypothetical one, but it offers a convenient starting-point.

Suppose a vessel floating freely and at rest is acted upon by an external force causing her to roll through some angle, say, to port. The work done is represented by the dynamical stability of the vessel at the angle at which she comes to rest. She has then energy due to position, which, on removal of the external force, takes effect in returning her towards the upright.

When the vertical is reached the energy of position becomes transformed into energy of motion, the vessel attaining a maximum angular velocity. The energy of motion now carries the vessel to starboard, to the same angle as that reached on the port side, where she once more regains energy of position, which, in turn, sends her back to the upright. And so the rolling goes on, since, by our assumption, there is no external resistance.

The formula for a single roll in the above hypothetical circumstances is—

$$T = 3{\cdot}1416 \frac{k}{\sqrt{g\,m}},$$

where T is the time in seconds of a single roll, m the metacentric height in feet, and g the acceleration due to gravity ($= 32{\cdot}2$ feet per second per second).

The symbol k is known as the transverse radius of gyration. What this quantity is may be explained by stating that if the whole weight of the oscillating vessel could be concentrated at a point distant k from the axis of oscillation, the effect would be the same as with the vessel as she is, that is, the period of oscillation would be the same. To find the value of k in any case, it is necessary to assume the vessel's weight divided into very small elements w, say, and to obtain the distance between the centre of each of these elements and the rolling axis; then, if r be taken to represent any of these distances, and W be the total weight of the vessel—

$$k^2 = \frac{\text{Sum of all the products } w \times r^2}{W}.$$

The numerator of this expression for k^2 is the moment of inertia of the vessel about the chosen axis; calling this I

$$k = \sqrt{\frac{I}{W}}.$$

Using the value given for g, the formula for the period may be written—

$$T = {\cdot}554 \frac{k}{\sqrt{m}}.$$

We now see why vessels having large metacentric heights are of quicker motion, *i.e.*, shorter period, than those with smaller values, for evidently T becomes reduced with increase of m, and *vice versa*. Also, the period is increased or decreased with corresponding changes in the value of the radius of gyration, which varies according to the distribution of the weights on board the ship, being increased by spreading them out from the centre of gravity, and decreased by crowding them about that point.

As a practical example, let us obtain the rolling period for a vessel of 3000 tons displacement, whose metacentric height is 2·5 feet, and radius of gyration 17·16 feet. By substitution we get—

$$T = {\cdot}554 \frac{17{\cdot}16}{\sqrt{2{\cdot}5}} = 6 \text{ seconds}.$$

Vertical movements of weights have greater influence on the period than horizontal movements. For instance, in the above vessel, if 120 tons were raised 14 feet, *i.e.*, from a position 7 feet below the centre of gravity to one 7 feet above it, the period would be increased to 6·8 seconds, while winging out this weight 14 feet from the centre would only lengthen the period to 6·1 seconds. As well as by calculation, the still-water rolling period may be found experimentally. To do this it is only necessary to set the vessel rolling by some artificial means, and to note the number of complete rolls she makes in a certain time. The period of a single roll may then be got by dividing the time by the number of rolls. This follows from the fact that the time taken per roll, for all inclinations up to which the curves of statical stability are straight lines—that is, about 12 to 15 degrees—is the same, a characteristic known as Isochronism.

It may be well to state here that it is important to know the value of a vessel's still-water rolling period in order to predict her probable behaviour at sea. Vessels seldom roll to dangerous angles in still water, but, as we shall see presently, may do so among waves, unless precautions have been taken to provide them with suitable still-water periods.

SEA WAVES.—Before dealing with the rolling of a vessel among waves, it will be necessary to give some attention to the structure of the latter in the light of modern theory, in order to obtain a clear idea of the action of water on a vessel when the surface of the former is undulated into wave shape.

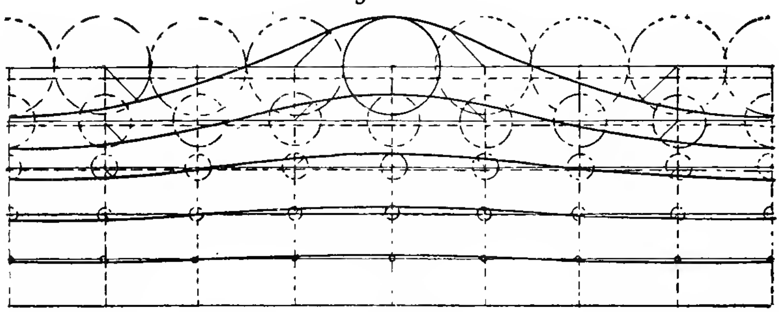
Waves are generated by the action of wind on the sea, and are the principal agents causing ships to roll. There have been various theories as to the action of wave water, the most satisfactory of which, and the one now generally accepted as representing the case best, being that known as the trochoidal theory.

The groundwork of this theory is, that only the form of the wave travels, and that the particles of water affected move in small circular orbits about horizontal axes. That some such action does take place will be obvious to anyone who observes the movements of a piece of driftwood afloat among waves. It will be noted that the wood does not travel with the wave, but merely moves backwards and forwards, showing clearly that the water particles supporting it move only a short distance as the wave passes.

According to this theory, a section of a wave in the plane of the line of advance, has for its outline a trochoid, *i.e.*, a line described by a point having uniform circular and linear motion. A rough, but simple way of drawing a trochoid, is as follows:—Take any point between the centre and the circumference in a circular paper disc, and let the latter be rolled without slipping along a horizontal line; the path described by the point is a trochoid. The reader should try this for himself. The theory also states that originally horizontal layers below the surface become, when under wave motion, distorted into trochoidal forms of the same general character as that of the upper surface, and that columns of water originally vertical curve towards the wave crest. The hollows and crests of the various trochoids are immediately under each other, and, therefore, the trochoids are all of the same length. But they become flatter as the depth below the upper surface increases, the particles moving in smaller and smaller orbits, until finally the wave, form disappears. Fig. 233, which exhibits in section part of a trochoidal wave, illustrates some of the points referred to. In this figure the original surfaces and subsurfaces in still water are shown by dotted horizontal lines, the same surfaces when in wave form by full curved lines. The orbits of surface and subsurface particles are also indicated. The lines containing the centres of these orbits (shown full) are seen to be at higher levels than the corresponding still-water lines, showing that in the wave, as well as kinetic energy, or energy of motion, the particles have also potential energy or energy of position.

Another fundamental point in this theory is, that the pressure of a particle in the wave is affected by the centrifugal force generated by its orbital motion, and acts normally to the particular surface in which the particle lies; so that, as the slope varies at each subsurface, the directions of

Fig. 233.



the normals also continually vary. All this has to be remembered when considering the resultant of the wave forces which act on a vessel afloat among waves.

The length of a wave is the horizontal distance from crest to crest, or hollow to hollow; the height is the vertical distance from hollow to crest; the period of a wave is the time it takes to move a distance equal to its own length. From calculations based on the trochoidal theory, we have the following:—

$$\text{Period of wave in seconds} = \sqrt{\frac{2 \times 3{\cdot}1416 \times \text{length}}{g}} = \sqrt{\frac{\text{length}}{5{\cdot}124}}.$$

$$\left.\begin{array}{l}\text{Speed of wave in feet per}\\ \quad\text{second}\end{array}\right\} = \sqrt{\frac{\text{length} \times g}{2 \times 3{\cdot}1416}} = \sqrt{5{\cdot}124 \times \text{length}}.$$

Results obtained from these formulæ are found to agree closely with those of observations of actual waves. Atlantic storm waves 600 feet in length, for instance, have observed periods of 11 seconds, and from the formulæ, using this length of wave, we get—

$$\text{Period} = \sqrt{\frac{600}{5 \cdot 124}} = 10 \cdot 8 \text{ seconds.}$$

$$\text{Speed} = \sqrt{5 \cdot 124 \times 600} = 55 \cdot 4 \text{ feet per second.}$$

The heights of waves have an important influence on rolling. The magnitude of the maximum angle of slope of a wave depends upon the ratio of the height to the length, and it will be found that the extent of the arc through which a vessel oscillates, is largely governed by the value of this slope angle. The heights of well-defined ocean waves are usually found to vary from $\frac{1}{20}$ to $\frac{1}{10}$ of the lengths, in long and in short waves, respectively, the steepness of waves decreasing with increase in length.

ROLLING AMONG WAVES.—We have seen that the effect of the internal wave forces is to cause the resultant buoyant pressure on a surface water particle to act normally to the wave slope, and it must now be added that the same forces act upon the weight of any small floating particle or body, and cause the resultant force also to act normally to the wave slope, but in a line opposite to that of buoyancy. The truth of this was proved experimentally by Dr. Froude in the following manner:—Taking a small float of cork he fitted it with an inclined mast, from the top of which he suspended a simple pendulum. He then placed the float on waves, when it was observed that the pendulum did not hang vertically but took up a position perpendicular to the wave slope.

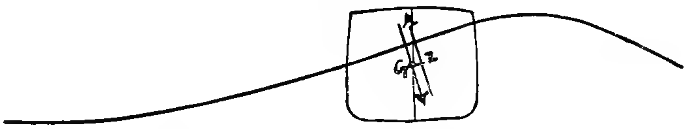
Now, a ship displaces a considerable amount of wave water, and cannot, properly speaking, be looked upon as a surface particle. It intersects many subsurfaces, the pressures on the particles of which act normally to the curves of these subsurfaces, and the resultant pressure, on the whole body, is normal to a mean subsurface; but in actual calculations it is usual to consider the vessel as very small relatively to the wave, and to treat it for all practical purposes as a surface particle, the resultant lines of pressure and weight being assumed to act normally to the slope of the upper surface of the wave.

On this assumption, a vessel among waves will tend to place her masts parallel to the normal to the wave slope, which virtually becomes her upright position of equilibrium. This is illustrated in fig. 234, where the centre line, that is, the line of the masts, is shown inclined to the wave normal, with a moment $W \times GZ$ in operation tending to bring them into parallel lines and thus place the deck parallel with the wave slope. In calculating the angle of inclination of the vessel to the vertical at any instant when among waves, this modified righting moment, which is assumed to be proportional to the angle between the line of the ship's masts and the wave normal at that instant, is employed.

It is not our intention to attempt a description of these calculations, as they are difficult and would be quite out of place in a work of this kind. It is easy, however, in general terms, to predict the behaviour of a vessel among waves when the periods of ship and waves are known.

Where the still water period of a vessel is very short in comparison with the period of the waves she is among, caused by her being specially broad and shallow, or having a cargo of great density placed low down in her holds, she will tend to keep her masts close to the wave normal, as depicted in fig. 235. Her motions will be quick and jerky, and although she will generally keep her decks clear of water, she cannot otherwise be

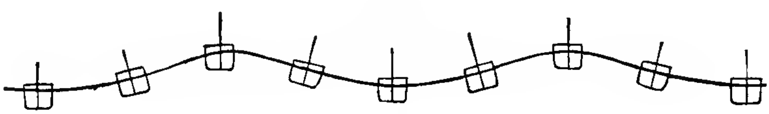
Fig. 234.



considered satisfactory. Her rapid motions are likely to strain the hull, especially during rough weather, and she will obviously be an uncomfortable boat in which to travel.

Different results are obtained when the ratio of the periods is reversed, *i.e.*, when the still water period is very long compared with the wave period. A vessel so circumstanced will be an easy roller, as may be readily explained. Assume such a vessel, for example, to be upright when a wave approaches her. Under the influence of the wave forces, she will immediately begin to heel, but her period being long compared with that of the wave, she will not have gone far when the wave normal, having passed

Fig. 235.



through its maximum angle to the vertical, at about the mid-height of the wave, will have returned to the upright, bringing a crest under the vessel. As the crest passes, the tendency of the wave pressures in the back slope will be to arrest the inclination of the vessel, and return her to the upright. And so the next hollow will find her a little inclined to the other side. This inclination will, in turn, be arrested by the following wave, and thus the departure from the upright of such a vessel will be small, and she will maintain a comparatively level deck. Such a state of things is highly desirable in warships to ensure a steady gun platform, and for obvious reasons it is also sought after in merchant vessels.

From the formula—

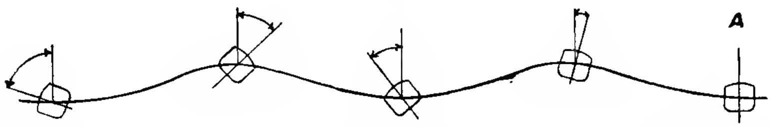
$$T = \cdot 554 \sqrt{\frac{k^2}{m}}$$

it is seen that in order to obtain a long rolling period, k, the radius of gyration, must be increased, and m, the metacentric height, must be reduced, as much as possible. This would mean concentrating the weights away from the middle line, narrowing the beam, or raising the position of the centre of gravity. More important considerations than those of rolling limit the extent to which the foregoing modifications may be carried out. It is impracticable, for instance, in merchant vessels to bank the weights against the sides, although with general cargoes something may be done by judicious stowage, while to bring down the value of m by reducing the beam or stowing the weights high in the vessel might seriously affect the stability, and no careful designer would recklessly do that. Experience must be the guide here, it being remembered that, generally speaking, vessels of great displacements may have smaller values of m than those of small displacement.

A critical case arises when the half period of the waves is the same as the ship's still-water period, and she is rolling broadside on to the former, a state of things known as synchronism.

In fig. 236 the effect of this coincidence of the effective time of the

Fig. 236.



two periods on a vessel's behaviour when rolling in a frictionless fluid, *i.e.*, without resistance, is depicted. Referring to this figure, at A the vessel is in the hollow, and is supposed to be upright when the wave reaches her. As she rises on the latter, the internal wave forces cause her to heel from the upright, and her period agreeing with the half period of the wave, she reaches the end of a roll at the first wave crest. On the back slope the wave forces will assist the ordinary statical moment to return her to the upright, and to a maximum inclination on the other side of the vertical, which she will reach at the hollow, and which will be greater than if she had oscillated under her statical moment alone. The wave forces in concert with her statical moment will again change the direction of motion, and at the next crest, where she will complete another roll, her maximum inclination will be further increased. Thus she will continue to roll, reaching a greater maximum inclination at each crest and hollow, until she finally upsets.

Theoretical investigations show that the increment of roll due to the wave impulse is equal to $\frac{\pi}{2}$, or about $1\frac{1}{2}$ times the maximum wave slope. Thus, if this were 6 degrees, the maximum inclination would be increased each time by 9 degrees, and her arc of oscillation by 18 degrees; so that the effect of a few such waves would be to put the vessel on her beam ends.

Dr. Froude proved the truth of the foregoing theory by experimenting with little models in a tank. Waves were generated having a period double that of the models in still water, and the latter when placed in the tank were upset after the passage of a few waves.

We thus see that a vessel is most seriously situated when broadside on to waves whose period of advance is double that of her own still-water period. In some respects the ship, in receiving the wave impulse as above described, resembles an oscillating pendulum which has additional force applied to it periodically at the end of an oscillation in one direction, and just when it is about to return, the effect of which is to increase the angle of swing each time. Another illustration is given by a body of soldiers crossing a bridge, when the period of march keeps time with the period of vibration of the bridge, a state of things which, continued long enough, would greatly increase the amplitude of the vibrations, and might eventually bring the structure down.

Summarising the foregoing remarks and deducing obvious inferences there from, we note:—

(1) That vessels whose periods are very short in comparison with the waves, will tend to place their masts parallel to the wave normals; that the angular velocity of such vessels may, in stormy weather, become very great and the rolling heavy; that excessive transverse straining may thus be developed, with a particular tendency to throw out the masts.

(2) That vessels of long periods (single roll), if among waves with half periods considerably less, are likely to be slow rollers, and to incline through moderate angles from the upright; that this is a most desirable state of things in both mercantile and war vessels, and after sufficiently allowing for stability, should be aimed at in new designs.

(3) That vessels having periods which keep time with those of the waves are badly, if not dangerously, situated; that such synchronism, as has been borne out in actual cases of which records are available, is likely to conduce to heavy rolling and severe transverse straining.

A practical example of the effect of synchronism is afforded in the case of H.M.S. *Royal Sovereign*, a large warship which from her design was expected to be very steady among waves at sea, and in general proved herself to be so; but on one occasion, when sailing in company of other vessels, there being a slight swell on the sea at the time, she rolled considerably, her maximum arc of oscillation reaching to 32 degrees, while the other vessels, which were of quicker periods, were but slightly affected. Observation showed the waves to have a period which synchronised approximately with the *Royal Sovereign's* single roll period of 8 seconds. On another occasion, when broadside on to a series of synchronising waves, she rolled through maximum arcs of 50 to 60 degrees. Excessive rolling is also reported of another vessel of this class, H.M.S. *Resolution*, the circumstances pointing to sychronism between the ship and apparent wave periods.

These examples show how difficult it is to altogether avoid the effects of synchronism. Actual observation has shown ordinary storm waves to average 500 to 600 feet in length, with periods of 10 to 11 seconds, only in exceptional cases longer waves being met with. Consequently, vessels having still-water periods of 8 seconds like the *Royal Sovereign* class should be expected to practically escape synchronism. Experience, however, has shown that circumstances may arise which shall cause the unexpected to happen. But even where there is synchronism, a vessel of long period is better situated than one of short period. In the former case, the waves keeping time are longer and less steep and have smaller maximum slope-angles than in the latter. Thus, the increment added to the angle of roll at each hollow and crest is less in the vessel of long period than in the other.

Of course, a master with his vessel well in hand is usually able to help matters considerably when his vessel is rolling excessively on account of synchronism. Should the rolling become heaviest when she is lying broadside on to the waves, he may disturb the coincidence of the periods by changing to an oblique course, which would lengthen the effective time of the wave. Should, however, the synchronism be developed when sailing on an oblique course, he may affect a cure in various ways. He may turn his vessel into the wave trough, or direct her head to the line of crests, or if he wishes to keep the original course, he may change the effective time of the waves by increasing or reducing the speed of the vessel. Thus, by skilful navigation, much can be done even with a vessel of bad design.

RESISTANCE TO ROLLING.—Although synchronism will always tend to make rolling heavy, as in the case of the *Royal Sovereign*, the resistance due to the friction of the water with the surface of an oscillating vessel, with that spent in the creation of waves, etc., will minimise the rolling at all times. Suppose, for instance, a vessel is set rolling in still water, and that, at a given instant, the external force causing her motion is removed, thus allowing her to roll freely. Her maximum range of oscillations will immediately begin to diminish. In any single oscillation, the difference between the maximum angle, say to starboard, from the following one to port, will be a measure of the resistance overcome. But when among waves whose period keeps time with that of her own motion, the periodical impulse given by the wave will cause the maximum inclination to be increased with each oscillation, so long as the resistance of the water is less than the increment of force due to the wave impulse. With increase of angle, however, the speed of oscillation will increase, since vessels describe the largest arcs in nearly the same time as the smallest. And, since the resistance of the water increases rapidly with the speed, a range of oscillations is soon reached, to sustain which the repeated impulse due to synchronism is necessary. Moreover, although the period when the arcs of oscillation are large are only slightly greater than when they are small, the difference is sufficient to disturb the synchronism. The wave impulse does not occur at the same critical moment each time, and a fraction of the resistance being thus unbalanced, it takes effect in reducing the angular velocity, the rolling becoming less heavy. This reduction may increase the period, and again cause synchronism, with consequent increase in the rolling, which, as before, will in turn be arrested. We thus see that oscillations sufficient to place a vessel on her beam ends, or to overturn her, are unlikely to occur in a resisting medium such as water.

ANALYSIS OF RESISTANCE.—Many years ago, Dr. Froude in carrying out experiments on the resistance of vessels to rolling, divided it into three parts, viz., that due (1) to the hull surface; (2) to keel, bilge-keels, deadwood, and the flat parts of the vessel at either end; (3) to surface disturbance. He obtained quantitative results by calculating items (1) and (2) from the plans of the vessels, and placing the difference between the sum of these items and the actual resistance obtained from experiments to the credit of item (3). The results showed the hull surface resistance to be less than 2 per cent., and the keel, bilge-keel, and flat surface resistance from 18 to 20 per cent. of the total, leaving about 80 per cent. as due to surface disturbance and the creation of waves.

While it is known that the creation of a very small wave would be sufficient to account for this residual resistance, subsequent experiments and investigations have shown that item (2) has probably been under-estimated. In making his calculations for the resistance of flat surfaces, Dr. Froude took 1·6 lbs. as a co-efficient of resistance per square foot at a velocity of one foot per second, and assumed the resistance to vary as the square of the velocity. This co-efficient he had obtained previously by oscillating a flat board in deep water. In the case of bilge-keels, it is now pretty well established that this figure, taken with the surface area of the bilge keels, does not represent the extinctive value of these appendages. On the assumption that the whole work of extinction, due to the fitting of bilge keels, might be credited to a virtual increase in the co-efficient of resistance per square foot of bilge area, and that the resistance varied as the square of the velocity, Sir Philip Watts pointed out that, in the case of the warships *Volage* and *Inconstant*, instead of 1·6 lbs., the co-efficients at a mean velocity of one foot per second should be 8·7 and 7·2 lbs., respectively.

In rolling experiments carried out in 1895 on H.M.S. *Revenge*, a warship of large displacement, similar results were obtained, the corresponding co-efficient being 11 lbs. for a swing of 10 degrees, rising to about 16 lbs. for a swing of 4 degrees.

Commenting on these results, Sir Wm. Whyte* pointed out that, as well

* See a paper by Sir Wm. Whyte in the Transactions of the Institution of Naval Architects for 1895.

as offering direct resistance, bilge keels create further resistance by indirectly influencing the stream-line motions that exist about an oscillating ship.

Investigation* has fully borne this out. Professor Bryan has shown that the motion of a rolling vessel, particularly if she be of sharp form at the bilge, gives rise to counter currents in the water which strike the surface of the bilge-keels and increase their extinctive value; also, that the presence of the bilge-keels cause discontinuous motions in the surrounding stream lines, the result of which is a gradual reduction in the speed of the streams as they approach the keels and an increase of pressure on the hull surface, giving rise to turning moments tending to arrest the angular motion of the ship. Crediting these resistances to the bilge-keel area, Professor Bryan estimates the effect of the counter currents as equivalent to doubling Dr. Froude's co-efficient, and the effect of the discontinuous motion to quadrupling it.

It should be stated that the foregoing analysis is based on the assumption that the vessel has no forward motion, but rolling motion only. From the results of rolling experiments† with a destroyer, it is shown that the effect of discontinuous motion is much reduced, when a ship has motion ahead, the diminution increasing with the speed ahead for the same angle of roll; the apparent reason being that the keel surfaces strike the water at an oblique angle, and thus do not create such masses of dead water as when rolling without forward motion.

But however the work done by bilge-keels be analysed, the point of most importance concerning them is that they are invaluable as a means of reducing rolling. Experience has amply shown this in a general way, but figures deduced from actual experiments are perhaps more convincing. H.M.S. *Repulse*, a large battleship, was, as an experiment, fitted with bilge-keels 200 feet long and 3 feet deep, and when amongst synchronous waves at sea, was found to reach only half the maximum angles of oscillation attained by her accompanying sister vessels, which were without bilge-keels. In the years 1894-5, rolling experiments, with and without bilge-keels, were conducted on the *Revenge*, a vessel of the same class as the *Repulse*. In still water it was found that, starting with an inclination of 6 degrees, without bilge-keels it took 45 to 50 swings to reduce the angle to 2 degrees, and with bilge-keels, similar to those on the *Repulse*, only 8 swings. Again, it was noticed that, starting at 6 degrees inclination, after 18 swings the vessel without bilge-keels reached an angle of $3\frac{3}{4}$ degrees, and with bilge-keels, an angle of 1 degree. In the case of the destroyer above referred to, the decrement of roll for 4 degrees mean angle of roll was, without bilge keels, ·24 degrees, and with bilge-keels, ·5 degrees. Most important of all is perhaps the effect of bilge-keels on rolling when vessels have motion

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* See a paper in the Transactions of the Institution of Naval Architects for 1900.

† See an interesting paper by Mr. A. W. Johns in the Transactions of the Institution of Naval Architects for 1905.

ahead. In the case of the *Revenge*, starting at an angle of 5 degrees from the vertical in each case, after 4 swings the inclination with no motion ahead was 2·95 degrees, and at a speed of 12 knots, 2·2 degrees; after 16 swings the corresponding inclinations were 1·15 degrees and ·25 degrees. In the case of the destroyer the resistance to rolling for 4 degrees mean angle of roll, at 17 knots, was $3\frac{1}{2}$ times greater than when not under weigh.

The reason of the greater extinctive value of bilge-keels in vessels under weigh is due to their having at each instant new masses of water to set in motion, and the energy so imparted being continually left behind and thus lost to the ship.

The experiments with the destroyer brought out another point, viz., that forward motion tends to reduce the rolling period both with and without bilge keels. The double period in seconds with no motion ahead was found to be—with keels, 5·59; without keels, 5·61. At about 17 knots speed the corresponding figures were 5·4 and 5·46 respectively. At higher speeds the reduction was still more marked. Another point to be noted in respect to bilge-keels is that they are more effective in small quick-rolling vessels than in large vessels of slow angular motions. This follows because the resistance bilge-keels meet with from the water increases with their speed of motion through it, and because the power of this resistance in arresting angular motion is greatest when the oscillating body to which the bilge-keels are attached is of relatively small weight and inertia. The importance of having these appendages on small vessels of quick-rolling period is thus apparent. The advantages of bilge-keels are now generally recognised, and they are regularly fitted to both war and merchant vessels. In warships they are frequently of considerable depth; in merchant vessels they are seldom more than 12 to 15 inches deep, and are often less; but even when so limited in size, their effect on the behaviour of vessels at sea has been most beneficial.

WATER CHAMBERS.—Besides bilge-keels, various other more or less successful methods have been advanced for minimising the rolling of ships. The best known of these consists in having a chamber partially filled with water fitted across the ship, so that when the vessel rolls, say, from port to starboard, the water, having a free surface, rushes in the same direction and acts against the righting moment operating to return her to the upright position. The efficiency of the method has been found to depend on the depth of water in the chamber. This was borne out by observations of the behaviour at sea of H.M.S. *Inflexible*, which had such a chamber, her mean angle of roll being reduced 20 to 25 per cent. with the chamber about half full, this being the best result obtained. The value of a water chamber was further tested in a series of still water rolling experiments with the *Edinburgh*, a warship of the same class as the *Inflexible*. The chamber in this case was 16 feet long, 7 feet deep, and had a full width of 67 feet; by means of bulkheads the chamber could also be tried at breadths of 43 feet and $51\frac{1}{2}$ feet, respectively. Increasing the breadth was found to have a powerful effect, the extinctive value at 67 feet being three times that at 43 feet. It was also found that the most effective depth of water was that which made the natural period of the wave traversing the chamber, the same as the natural rolling period of the ship.

Experiments with a model of the water chamber, fitted on a frame designed to oscillate at the same period as the ship, showed the efficiency of the system to be greatest at small angles of inclination.

For various reasons the water chamber method has not become popular. In the case of warships, changes in design have led to longer natural rolling periods, and a reduced necessity for special means of extinguishing rolling. The *Inflexible*, for instance, had a GM of 8 feet, while modern battleships have seldom greater metacentric heights than about 3 feet.

In the case of merchant vessels, the expense of fitting up a water chamber, and the loss of valuable space which would be entailed by its presence in the hold, has stood in the way of its general adoption, particularly as the inexpensive method of fitting bilge-keels has produced satisfactory results.

THE GYROSCOPE.—A proposal for extinguishing rolling motion, which some authorities think is likely to be widely adopted in the future, has been brought forward in recent years by Dr. Otto Schlick. In Dr. Schlick's words, "the method depends in principle on the gyroscopic action of a flywheel, which is set up in a particular manner on board a steamer, and made to rotate rapidly." The principle of the apparatus, and the method of application, is fully explained by Dr. Schlick in an interesting paper read before the Institution of Naval Architects in 1904, and to this the reader is referred for details.

So far the apparatus has, we believe, only been practically applied in the case of two vessels, but from the reported results of these trials, the system appears to be a highly efficient one. The first of these vessels is a German torpedo-boat, 116 feet long, and of about 56 tons displacement. In this case* the gyroscope, which was fitted for purely experimental purposes, has a flywheel one metre in diameter, with a proportionate weight to weight of ship of 1 to 114. It is steam driven, the periphery of the wheel being provided with rings of blades, and the wheel enclosed in a steam-tight casing, and worked as a turbine.

The casing containing the wheel is carried on two horizontal trunnions having their axis athwartships, the steam supply and exhaust passing through the trunnions as in an oscillating engine. When the vessel is upright and at rest the spindle of the flywheel is vertical, and the latter when in motion thus rotates in a horizontal plane; also the apparatus is free to become inclined in a fore-and-aft direction. With the gyroscope in action, the effect of the transverse heeling of the vessel is to cause the apparatus to become inclined, and moments to be produced reducing the velocity of the vessel's oscillations and also their magnitude. To control the fore-and-aft move-

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* See a paper by Sir Wm. Whyte in the T.I.N.A. for 1907.

ments of the gyroscope and the rotary movement of the flywheel, an arrangement of brakes is provided.

At the commencement of the experiments the torpedo boat was rolled in still water. With the gyroscope at rest, a double-roll period of 4·136 seconds was obtained; with the apparatus in action, and the flywheel running at 1600 revolutions a minute, the period was found to be 6 seconds, an increase of 45 per cent.

The roll extinguishing effect was found to be enormous. Starting from an angle of 10 degrees, with the gyroscope at rest, it took twenty single oscillations to reduce the inclination to half a degree; with the gyroscope in action, the same angle was reached in little more than two single oscillations.

The sea trials were quite as remarkable; when through the state of the sea, the vessel was caused to roll considerably, the effect of the action of the apparatus, when brought into play, was practically to extinguish the rolling motion. On two occasions, for instance, the vessel, with the gyroscope fixed, reached arcs of rolling of 30 degrees, which, on the apparatus being allowed to act, became immediately reduced to 1 degree or $1\frac{1}{2}$ degrees. The results of other observations were equally convincing.

The other vessel referred to as being fitted with Dr. Schlick's apparatus is the coasting passenger steamer *Lochiel.* Very few details of the gyroscope in this case are available, but it is stated that the flywheel is driven electrically. From the reports, the roll-extinguishing effect appears to be quite as great as in the torpedo-boat. On occasions the vessel was found to be rolling through arcs of 32 degrees, the gyroscope being at rest, and the effect of bringing it into action was to reduce the arc to from 2 to 4 degrees, oscillations scarcely perceptible to those on board.

It remains to be seen how far this unique system of extinguishing rolling motions at sea will be adopted in the future, but it is not unlikely that the success of the *Lochiel* trial may lead to the installation of gyroscopes in other vessels of the same class, and also in steamers engaged in cross-channel passenger traffic.

PITCHING AND HEAVING.—A vessel among waves will have motions in many directions, depending on her position with regard to the crest lines. If broadside on, the principal motions will be those of transverse rolling, but there will be also more or less vertical dipping oscillations due to the wedges of immersion and emersion being instantaneously dissimilar in volume. If head on to the waves, while there will be some transverse rolling, the chief motions will consist of pitching, *i.e.*, longitudinal rolling about a transverse axis through the centre of gravity, and heaving, due in part to the dipping motions above mentioned, and in part to the excesses of weight and buoyancy which occur as the vessel rides over the waves. If, however, the vessel lie in an oblique direction relatively to the wave crests, simultaneous skew rolling and pitching motions will be set up, as well as heaving.

The conditions in each of these cases will be modified to a greater or less extent by the forward motion due to the propeller. We have seen how transverse oscillations are thus affected, and we shall consider presently the influence of speed on pitching and heaving.

The period of unresisted pitching in still water may be determined by the formula which gives the period of similar transverse oscillations, provided K be the radius of gyration about a *transverse* axis through the centre of gravity, and M the *longitudinal* metacentric height.

T, as before, being the period in seconds, the formula is—

$$T = 3{\cdot}1416 \sqrt{\frac{K^2}{g\ M}}.$$

As an example, take a vessel of 4000 tons displacement, with a radius of gyration of 110 feet, and a longitudinal metacentric height of 285 feet. In this case—

$$T = 3{\cdot}1416 \sqrt{\frac{110 \times 110}{32{\cdot}2 \times 285}} = 3{\cdot}61 \text{ seconds.}$$

The corresponding transverse rolling period for this vessel is about 8 seconds, *i.e.*, more than double the other; and this, in most cases, is the proportion between the two periods.

In pitching, a vessel always tends to place her masts normal to the effective wave slope. When a vessel is long in comparison with the waves, the effective wave slope will depart very little from the horizontal, and the pitching will be slight; when the opposite is the case, *i.e.*, when the vessel is short relatively to the waves, the extent of the pitching will be governed by the natural pitching period, the period of the waves, and the course and speed of the vessel relatively to the waves. If the wave period be long, and that of the vessel very short, she will follow the slope of the wave; but if the wave period be naturally short, or, if it be made so by the speed of the vessel, pitching is likely to become excessive, as the vessel will fail to rise on each successive wave crest; her ends will thus become buried, and the periodical impulses received from the waves will conduce to larger longitudinal oscillations. Pitching, then, which is in the first instance caused by the passage of waves, will, like rolling, become excessive if the wave period keeps time with her natural pitching period.

Every seaman likes his vessel to be lively fore and aft, *i.e.*, to have a short pitching period. In such a case a vessel will follow approximately the wave slope, especially if she be short relatively to the wave length, and will rise on the waves instead of burying her ends into them. This vessel will be drier than a slower moving boat, and will not be subjected to the same hammering stress which continual plunging into waves is bound to set up.

In order to obtain a short pitching period, thus seen to be desirable, the radius of gyration must be reduced or the longitudinal metacentric height increased. This follows from the formula for the period given above. The metacentric height cannot be affected to any appreciable extent, as the length and displacement, on which its value depends, are fixed by more important considerations than those of pitching; the radius of gyration, however, may obviously be reduced by concentrating the heavy weights amidships, and in a merchant ship this should be done as far as possible in stowing the cargo.

Of course, just as in the case of transverse rolling, a master may frequently help matters. Should pitching become excessive when he is sailing head to sea, due to synchronism, he may change to an oblique course and thus lengthen the apparent wave period, and give his vessel time to rise on the waves; or, without changing his course, he may reduce speed and attain the same end.

On the subject of vertical heaving and dipping it is unnecessary to say much. From what we know of synchronism, it is clear that motions of this kind are likely to be excessive, if the period of dipping is in approximate unison with that of the waves. Pronounced heaving will not endanger a vessel's safety, although, as was noticed in a previous chapter, the longitudinal bending moments, and therefore the stresses brought upon the hull, may be considerably affected thereby. It may be said that, as usually constructed, vessels are provided with a sufficient margin of strength to meet all such demands.

QUESTIONS ON CHAPTER X.

1. How would you set about obtaining the instantaneous axis of an oscillating ship?

2. Write down the formula for the period in seconds of a single oscillation of a ship on the supposition that there is no resistance. What use is made of this formula by the naval architect?

3. Given that the metacentric height in a certain vessel is 2 feet and the radius of gyration 18, calculate the period of a single roll. *Ans.*—7 seconds.

4. The radius of gyration of a vessel is 16, and the single roll period 5 seconds; find the metacentric height. If the metacentric height be reduced one foot, what would be the periodic time? *Ans.*—3·14 feet; 6·06 seconds.

5. What is Isochronism? To what inclinations are vessels of ordinary form isochronous? In rolling through large angles how will the period be affected?

6. Explain briefly the modern theory concerning the form and action of sea waves?

7. Calculate the period in seconds and the speed in feet per second of a wave 500 feet long. *Ans.* 9·88; 50·6.

8. What is meant by the term "effective wave slope"? Explain why a vessel among waves tends to place her masts parallel to the normal to the wave slope as virtually her upright position of equilibrium.

9. Show that the behaviour of a vessel at sea depends largely on the relation between her still-water period and the period of the waves she may encounter.

10. What difference in the behaviour of a vessel would you expect if her single roll period were—

(1) longer than the half period of the waves;
(2) much shorter than the half period of the waves?

11. Explain the terms "steady," "crank," and "stiff," as applied to a vessel's condition when among waves at sea.

12. Under what circumstances is the rolling of a ship likely to be most severe?

The single roll period of a vessel is 6 seconds; how would you expect her to behave if broadside on to a regular series of waves of 12 seconds period?

13. Referring to the previous question, if the maximum slope angle of the waves be 5 degrees and the vessel upright when a wave reaches her, in what position would you expect her to be after the passage of six waves, assuming the water to offer no resistance?

14. If a shipmaster finds the rolling of his vessel to become exceptionally severe, to what cause may he justly attribute it? What steps should he take with a view to reducing the rolling motion?

15. Give an analysis of the resistance encountered by a vessel when rolling freely.

16. What are bilge-keels? In what way do they affect the rolling motions of vessels? Discuss the effect of motion ahead on the action of bilge-keels.

17. Bilge-keels are more effective in reducing rolling in small vessels of short period than in large slow-rolling vessels. Explain why.

18. What are water chambers? Show how they tend to diminish the rolling of ships in which they are installed.

19. Explain the terms "pitching" and "heaving." Give the formula for the pitching period of a vessel in seconds. The pitching period of a vessel being 4 seconds, and her longitudinal metacentric height 400 feet, calculate the radius of gyration.

Ans.—144·4 feet.

20. Is a short or long pitching period preferable? Give a reason for your answer?

21. Under what circumstances is a vessel likely to pitch excessively? Explain how a master, who has his vessel well under control, might help matters in such a case and obtain easier fore-and-aft motions.

CHAPTER XI.

Loading and Ballasting.

IT should now be clear that to efficiently load a vessel does not mean simply to fill her with cargo in the shortest possible time. In the previous chapters we have endeavoured to show that the nature of a vessel's sea qualities depends upon the manner in which the weights, including the cargo, are distributed, so that skilful stevedoring is almost as important as efficient designing.

We have already seen that the characteristics controlling a vessel's sea qualities have a conflicting interdependence, which makes it difficult in any given case to arrange for the values necessary to the best all-round results; that with great stability heavy rolling is frequently associated, and with great steadiness a dangerously small margin of stability. It is thus clear that considerable care and experience is necessary in order to put cargo properly into a vessel. The superintendence of this work should, therefore, be entrusted only to thoroughly-experienced persons, and owners who take no precautions of this sort may find the subsequent behaviour of their vessels to be scarcely all that might be desired.

An intelligent and experienced officer can, with care, usually do much to bring about a satisfactory condition of his vessel. Even if he does not gain all he may strive for, his vessel should still be safer and more comfortable than if loaded in any haphazard way.

GENERAL CARGOES.—In loading general cargoes, an officer who knows his business will be guided by the characteristics of his vessel. If she be narrow and deep, he will place the heavy weights low in the holds and the lighter weights higher up, thus ensuring a comparatively low position of the centre of gravity, necessary on account of the metacentre being low in position in vessels of this type. If the vessel be broad and shallow, the metacentre will be relatively high, and to obviate a too great value of GM he will aim at a higher position of centre of gravity, placing the heavy weights higher in the vessel.

Besides this, following the principles of Chapter VIII., he will see that the weights are distributed longitudinally in such a way as to secure a suitable trim. Thus, with sufficient stability, steadiness among waves and a satisfactory fore-and-aft flotation may be secured.

The vessel's steadiness may be further improved, without affecting the stability, if, without raising them, the heavy items of cargo can be banked against the ship's sides, as the radius of gyration is thus increased and the roll period lengthened. Actual experience appears to indicate that, in ordinary cases, very little can thus be done to improve a vessel's condition, but the effect of "winging" the weights should not be lost sight of.

The nature of a cargo, it is hardly necessary to point out, is always a determining factor of the style of loading. It is also admitted that circumstances may not always be favourable to good stowage. Suitable cargo may not be available for shipment at the correct time, and, in consequence, the heavy items may occupy positions either too high or too low, and at the centre of the vessel rather than at the sides; but such a state of things may be considered exceptional. When the weights and other particulars of the various items for shipment are available, a good plan is for the officer in charge to make a rough estimate of the position of the centre of gravity. In this way the best places for individual items of cargo may be determined before commencing operations, and, although in the process of loading departures may require to be made, these may readily be allowed for. On completion of the stowage, the metacentric height may, as previously suggested, be checked by means of an inclining experiment, and, if necessary, corrected by transposing some of the weights. Also, the roll period may be ascertained by forcibly heeling the vessel and counting the number of rolls as already described. It is to be feared the value of such experiments is not fully appreciated. Owners make much of the trouble and loss of time involved, and do not give the encouragement they might to their commanding officers, and hence we find well-proportioned and designed vessels developing tendencies to excessive rolling, which the exercise of a little care at the time of loading would have done much to obviate.

It cannot be doubted that the carrying out of the experiments above described would afford invaluable experience to a commanding officer as to how particular kinds of cargo should be stowed in his ship to obtain the best results at sea. Such an officer might be said to "know his own ship." It sometimes happens, however, that a man is called upon to take charge of the loading of a ship of whose qualities he is in total ignorance.

In such a case an officer should be quick to notice changes in the vessel's condition during the process of loading. If he should observe her to suddenly list to port or starboard, he may take it her stability, in the upright position at least, is dangerously small, the sudden movement being caused by the raising of the centre of gravity above the metacentre, and the vessel being put into a state of unstable equilibrium. Her stability curve will resemble fig. 224, that is, she will be unstable from the upright to the angle at which she has come to rest. The officer must on no account attempt to cure such a list by moving weights to the high side, as he might quite correctly do if the list had been a gradual one due to uneven loading. In the present case the raising of the weights would make matters worse, and, if the reserve of stability were small, might culminate in actual disaster. The only cure is to bring down the centre of gravity by lowering the position of weights already on board, or shipping additional weights low down in the holds. A good way is to run up a compartment of a ballast tank, but, as will be seen later on, this might be dangerous if the stability reserve were small.

HOMOGENEOUS CARGOES.—In the foregoing remarks we have assumed a more or less general cargo. The case, however, is different with certain homogeneous cargoes, as we shall now proceed to show.

Suppose, for instance, a vessel has her whole cargo space filled with a homogeneous cargo, of such density as to just bring her to the load water-line. This is a trying condition of loading, as an unfavourable position of the centre of gravity cannot now be corrected by shifting about the cargo. The only plan open is to discharge part of it, and this few owners would contemplate with any satisfaction. Such a resort, however, unpleasant though it be, would, under such circumstances of loading and position of centre of gravity, be unavoidable if the safety of the ship at sea were to be considered at all.

Of course, vessels intended frequently to load homogeneous cargoes of this critical density can always be designed to carry a full cargo with perfect safety. The naval architect would, in such a case, make this the one condition in which the vessel should have sufficient stability and trim properly, since it is the only one over which stowage has no control.

The importance of good design has been demonstrated by the results of actual experience. The late Dr. Elgar, in a paper on "Losses at Sea," read before the Institution of Naval Architects in 1886, made an analysis of British shipwrecks over a certain period, and showed conclusively that many of the disasters were due to bad design. Few of the vessels lost were, indeed, of such proportions as to admit of sufficient stability when fully laden with a homogeneous cargo such as above described, and many of them *were* so laden.

It should be mentioned that the proportionately narrow and deep class of vessels, to which these mainly belonged, is no longer popular; the modern tendency is towards greater breadth, and this is in the right direction.

With homogeneous cargoes of other densities, as with general cargoes, something may be done to correct a high position of the centre of gravity due to faulty design. With those of lighter density, for instance, the whole cargo space may be filled as before, and the margin of draught taken up by running in water ballast; if the vessel has no tanks, heavy dry ballast may be put in the bottom of the holds before the cargo is loaded. With cargoes of greater density, the whole internal space will not be required, and so the position of the centre of gravity can be affected by leaving an empty space in the holds, or in the 'tween decks, according as it is desired to diminish or increase the value of GM.

SPECIAL HOMOGENEOUS CARGOES—OIL.—Bulk oil, as a freight, is becoming increasingly important, and special care is necessary in dealing with it. In explanation of this, suppose an oil-carrying vessel, in the process of loading, to be slightly heeled by some external means. Fig. 232 illustrates the case and is a section through a partially filled compartment. It will be noted that the act of heeling has transferred the small wedge of oil $S_1 O S_3$ across the ship into the position $S_2 O S_4$, causing G, the centre of gravity, to be drawn out in the same direction to G_1. The forces of weight and buoyancy act through G_1 and the deflected centre of buoyancy, and, as drawn, form a couple tending to right the vessel, the arm of the couple being $m Z_1$.

It is thus seen that the effective centre of gravity, so far as the initial stability is concerned, is raised to m, and the metacentric height reduced from $G M$ to $m M$. If we assume that the liquid in the hold is of the same density as the water in which the vessel is floating, the reduction $G m$ in feet may be obtained in any actual case from the formula—

$$G m = \frac{i^*}{V},$$

where i is the moment of inertia of the free surface of the fluid in foot units, and V the volume of displacement of the ship in cubic feet. This formula is

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* This formula is obtained as follows:—Referring to fig. 232,

Let b = half breadth of free surface in feet.
l = length of free surface in feet.
w_1 = weight per cubic foot of liquid cargo in lbs.
w_2 = weight per cubic foot of water in which vessel is floating in lbs.
V = volume of displacement in cubic feet.

Assuming the vessel to be heeled as in fig. 232, and compartment to be rectangular at the level of the free liquid surface,

Volume of wedge $S_1 O S_3$ or $S_2 O S_4 = \frac{1}{2} b^2 l \theta$ cubic foot.

Weight of wedge $S_1 O S_3$ or $S_2 O S_4 = \frac{1}{2} b^2 l \theta w_1$ lbs.

Moment of wedge $S_1 O S_3$ or $S_2 O S_4$ about fore-and-aft axis through O $\Big\} = \frac{1}{2} b^2 l \theta w_1 . \frac{4}{3} b$ foot lbs.

And, since weight of vessel $= V \times w_2$ lbs.

$$G G_1 = \frac{2}{3} \frac{b^3 l \theta}{V} \times \frac{w_1}{w_2}.$$

Let a vertical line be drawn through G_1, and call the point in which it intersects the middle line, m. Then, the inclination being small,

$$G G_1 = G m \theta,$$

therefore,

$$G m = \frac{G G_1}{\theta} = \frac{\frac{2}{3} b^3 l}{V} \times \frac{w_1}{w_2}$$

But $\frac{2}{3} b^3 l$ is the moment of inertia of the liquid surface about the axis through o, *i.e.*, the middle line. Calling this i, we get by substitution,

$$G m = \frac{i}{V} \times \frac{w_1}{w_2},$$

which becomes—

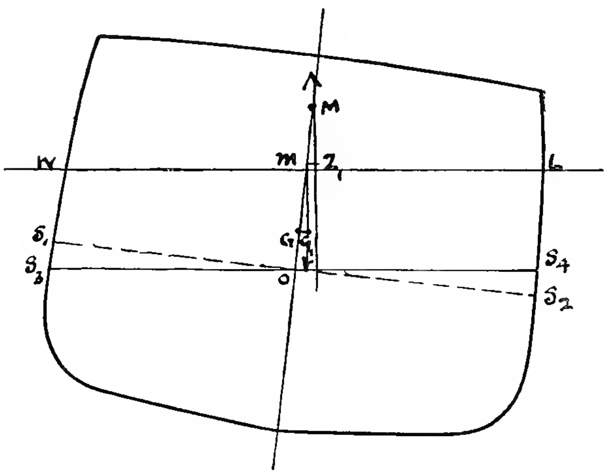
$$G m = \frac{i}{V},$$

when, as assumed above, liquid cargo and water in which the vessel is floating are of same density.

seen to be similar to that for the height of the transverse metacentre above the centre of buoyancy, except that G takes the place of B as the point from which the resulting distance must be measured. Clearly, the greater the value of i, that is, the larger the free surface of oil, the greater will be the reduction in the metacentric height. It should be specially noted that the reduction does not depend on the *quantity* of oil in the compartment, as a small quantity having a large free surface will have more effect than a large quantity with a small free surface.

Practical Example.—A midship compartment of a vessel of 4,500 tons displacement is partly filled with liquid of the same density as the water in which the vessel is floating. It being given that the free surface is 30

Fig. 232.



feet long, 38 feet broad, and rectangular in shape, estimate the reduction in metacentric height. Applying the formula, we get—

$$G\,m = \frac{30 \times 38 \times 38 \times 38}{12 \times 4500 \times 35} = \cdot 87 \text{ feet.}$$

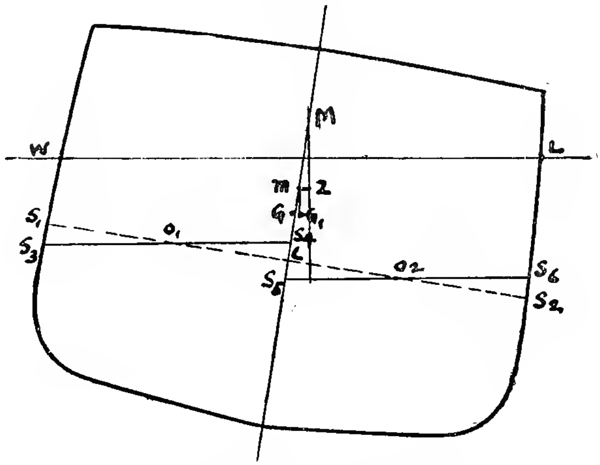
If the liquid in the compartment were oil different in density from the water supporting the vessel, the above value would require to be multiplied by the ratio of the density of the oil to that of the water. Thus, if the cargo were petroleum, and the vessel afloat in salt water, the reduction in metacentric height would be—

$$G\,m = \cdot 87 \times \cdot 8 = \cdot 69 \text{ feet,}$$

the ratio of the density of petroleum to that of salt water being $\cdot 8$.

In the above case there is assumed to be no middle line bulkhead. Such a bulkhead, however, is never omitted in modern oil-carrying vessels, as it is of great value in minimising the detrimental effect of a free surface. This is shown in fig. 233. The continuous line, $S_1 S_2$, indicates the oil surface with the vessel upright, and the two lines $S_3 S_4$, $S_5 S_6$, the surface when the vessel is heeled, the presence of the bulkhead restricting the movement as shown. The wedge transferred here from one side to the other of each portion of the divided compartment is half the breadth and one-fourth the volume of that of the previous case; also the travel of the centre of gravity of the wedge of fluid is a half, and the moment an eighth. But two wedges of fluid move instead of one, so that the total moment is one-fourth of what it was in the previous case. The reduction in metacentric height due to the restricted oil surface, since it varies directly as the moment, is thus also a fourth.

Fig. 233.



From the foregoing considerations, there follow two results of importance. The first is the advisability of restricting the lengths of oil compartments; the second, the necessity of exercising great care in loading fluid cargoes. In conducting the latter operation, it is highly important to keep the vessel upright, as a slight inclination caused by the movement of a weight of moderate amount on board is considerably accentuated by the action of the liquid cargo, which rushes in the direction of the inclination.

It is customary to draw out a diagram showing the angle to which the vessel may heel as the liquid rises in the hold. If the vessel has sufficient stability, it may be possible to fill two holds simultaneously. When the liquid is first poured into the vessel its free surface is small, and the reduction in metacentric height, due to loss of moment of inertia of surface, may be less than the increase due to the fall in the centre of gravity consequent on the admitted liquid being low in the vessel; but as the liquid rises, its upper surface broadens rapidly, and m quickly overtakes and passes M, the vessel becoming unstable.

In a case* investigated by the late Professor Jenkins, m coincided with M when the liquid reached a depth of 15 inches. Heeling then began, and rapidly increased as the oil rose in the hold, the vessel reaching a maximum inclination of $19\frac{1}{2}°$. After that she began slowly to right herself, finally returning to the upright when m had passed below M, which took place when the liquid came within a few inches of the top of the tank.

A point of special importance in loading oil vessels, which, perhaps, need scarcely be pointed out, is that adjacent compartments should be filled or emptied simultaneously; for if one side only were dealt with the inclining effect would naturally be great. Vessels have frequently been inclined to dangerous angles when this precaution as to loading has been neglected; and there are cases on record even of actual capsizing from this cause. Of course, when an oil compartment is quite full, no movement is possible, and the oil becomes virtually a solid homogeneous cargo.

Expansion Trunkways.—A point which must not be overlooked in connection with bulk oil cargoes, is the loss due to evaporation, and unless specially provided against, the reduction in bulk may lead to free surfaces in the holds. Accordingly, every oil compartment has one or more open trunkways rising above it, and sufficient oil is pumped into the vessel to fill the holds and partially fill these passages. The horizontal areas of the trunkways are kept as small as possible, consistent with the volume of oil in them above the level of the tops of the compartments being fully sufficient to allow for loss due to evaporation without bringing the oil level below the trunkways. These trunkways, too, being open to the holds, also serve the purpose of allowing the oil to freely expand and contract in volume with change of temperature.

GRAIN CARGOES.—Dr. Elgar, in the paper previously referred to, pointed out that between the years 1881 and 1883, the period covered by his analysis, vessels carrying grain had a greater number of losses than all other cargoes except coal. This is striking, as the number of vessels carrying grain is a small proportion of those engaged in the coal trade, and points to the existence of special characteristics in the nature of grain cargoes and their stowage. Investigation has proved these surmises to be correct.

It is found that bulk cargoes, such as grain, even when loaded with care, have a tendency to settle down during a voyage and to leave empty spaces immediately under each deck. These spaces have been estimated at 5 to 8 per cent. of the depth of hold, and in fairly large vessels may, therefore, be of considerable magnitude. After such settlement, the grain has a free surface, and it is here that the danger lies, for when the vessel is

* See a paper in the Transactions of the Institution of Shipbuilders and Engineers in Scotland for 1889.

rolling at sea, the grain tends to put its surface parallel with the wave slope, and, if the rolling is heavy, shifting is the inevitable result.

The angle to which the free surface of grain must be inclined before sliding motion will ensue, may be easily obtained. If wheat, for example, be poured on to a floor until there is a heap, it will be found to take the form of a cone-shaped pyramid. When sliding has stopped, the angle which the side of this pyramid makes with the floor, is called the angle of friction or repose, of this kind of grain; for, if more wheat be poured on to the heap, the angle of the cone will be increased, and the particles will run down the side of the cone until the same angle as before is attained. This is one of the principal differences between a liquid and a grain cargo. On the slightest inclination of the vessel, liquid puts itself parallel with the water surface; with grain the tendency is the same, but friction between the particles prevents any movement until a certain inclination is reached; this inclination, in fact, if the vessel be heeled in quiet water, being the angle of repose of the grain. The value of this angle has been obtained for various kinds of grain; for wheat it is 23¼ degrees, for two kinds of Indian corn, 26½ degrees and 28¼ degrees respectively, for mixed peas and beans, 27½ degrees.

The late Professor Jenkins, who investigated this subject,* drew attention to some points of importance with regard to the sliding angle. He showed that the accelerative forces, which act on a vessel and her cargo when rolling at sea, cause shifting to take place at a much smaller angle than the still-water angle of repose. In the case of grain with an angle of repose of 25 degrees, he found it to be, in a certain vessel of which the particulars as to stability and radius of gyration were assumed, as low as 16½ degrees. He also found that heaving motions, when accompanied by rolling, will, at a certain point during each oscillation, cause still further diminution of this angle. In the example above, it proved to be rather less than 14½ degrees. As this angle is frequently exceeded by vessels rolling among waves, the probability of shifting, where there is a free surface, becomes manifest. It should be mentioned that shifting would take place in the above vessel at 14½ degrees at one point only during the oscillation, namely, when she had arrived at the end of a roll and was about to return; also, that the whole surface would not slide at this angle, but only a portion of it at the upper part of the side about to descend. At any other part, the angle of shifting would be greater, reaching a value in excess of the still-water angle of repose at the other extreme of the free surface, *i.e.*, on the side about to ascend. Of course, when the vessel became inclined to the other side of the vertical, this state of things would be reversed. On the whole, the effect of rolling appears to increase considerably the tendency to shifting.

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* See a paper on the Shifting of Cargoes in the Transactions of the Institution of Naval Architects for 1887.

Professor Jenkins showed further that the decrease of angle at which sliding begins is greater, the greater the stability, but at the same time pointed out that the effect of a shift of cargo is more serious in the case of a vessel of small stability than in that of one of great stability. He also showed that the part of the cargo most subject to movement is that above the centre of gravity, which, in double-decked vessels, would apply to the 'tween decks. Government Regulations prohibit the carriage of grain in bulk in 'tween decks except such as may be necessary for feeding the cargo in the holds and is carried in properly constructed feeders; generally, it is largely carried in bags; dangerous shifting of the cargo at this part is thus obviated. The stowage of grain in the holds of vessels having a 'tween decks requires special care.* In the case of single-decked vessels, when shifting of cargo has taken place the effect may be rectified by opening the hatches when the weather permits, and filling up the empty spaces with bags of grain carried for the purpose. Where there is a 'tween decks this cannot be done, as the holds are inaccessible, and shifting once begun cannot be corrected. It is usual to fit trimming hatches through the lower deck, and these to some extent allow the settlement in the holds to be made up from the cargo in the 'tween decks, the grain in way of the trimming hatches being in bulk; but empty spaces under the beams are still likely to exist between these hatches.

Allowing for certain exceptions, the Government Regulations require one-fourth the grain to be carried in bags in all spaces which have no efficient feeding arrangements. In such cases, before stowing the bags, the grain must be trimmed level and covered with boards. For the reasons given above, this rule, which is calculated to prevent serious shifting of cargo, should obviously apply specially to compartments constituting the lower holds of vessels having one or more 'tween decks.

As a safeguard against the effects of possible shifting of cargo, grain-carrying vessels, whether the grain be in bags or in bulk, are required to have a centre division in the holds and in the 'tween decks, which restricts the extent of the movement of a grain cargo much as it restricts the movement of a cargo of liquid. Generally, the centre division consists of portable wood boards fitted edge on edge and reeved between the centre line of pillars, which are reeled for the purpose; but in some modern vessels it consists of a permanent steel bulkhead (*see* page 161) except in way of the main hatchways, the exigencies of stowage demanding portable boards at these places.

COAL CARGOES.—Owing to its density, coal can, in general, be stowed at such a rate as to ensure a certain amount of empty space in the 'tween decks, the vessel at the same time being down to her load mark. There is, therefore, no apparent reason why coal-laden vessels should not

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* See a paper by the late Mr. Martell in the Transactions of the Institution of Naval Architects for 1880.

have sufficient stability. From the great number of such vessels which have been lost, some of them known to possess a fair amount of stability at the start of their fatal voyages, it has been conjectured that shifting of the cargo may have been the cause of not a few of the disasters. Colliers are not usually fitted with shifting boards, and there is no restriction placed on the stowage of coal in the 'tween decks, as with grain; and, although the angle of repose of coal is considerably greater than that of grain, the diminishing process it undergoes during rolling motions at sea may doubtless often bring it within the range of a vessel's oscillations in stormy weather. From which considerations it would appear that the suggested reason of shifting for the loss of many coal-laden vessels may be quite near the mark.

The lessons to be deduced here by the ship's officer are, first, to aim at so loading his vessel as to ensure easy motion when among waves at sea; and second, to see that no vacant spaces are left under the decks, as these inevitably lead to shifting of the cargo.

TIMBER CARGOES.—In the case of cargoes of the heaviest woods, the full hold capacity is not required, and loading should simply follow the lines already indicated for ordinary heavy deadweight cargoes. With cargoes of mixed timbers, satisfactory conditions of stability and trim can always be attained by a proper distribution of the light and heavy woods. In the case of cargoes of the lightest timbers, however, the problem of stowage becomes more difficult, because, as well as full holds, there is usually a considerable deck load carried.

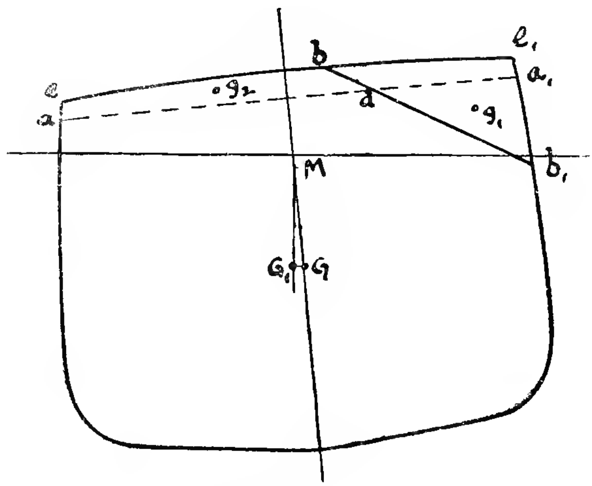
Many vessels that are good deadweight carriers, and quite suitable for general trades, could not, without a considerable amount of ballast, be safely employed to carry a cargo of timber of the last description, the high position of the centre of gravity, due to the presence of the deck cargo, making the stability quite inadequate. In some cases, indeed, there might be actual instability in the upright position, and no seaman would care to face a sea voyage in a ship having a pronounced list to port or starboard. In this trade, vessels should be specially broad in relation to draught, as this ensures a relatively high position of metacentre, and a sufficient margin of stability, without having to resort to ballast. It should, of course, be noted that a deck cargo of wood, when well packed and securely lashed in place, affords valuable surplus buoyancy, which has a marked effect on the form of the stability curve, giving it increased area and range, although at initial angles, owing to a high position of the centre of gravity, the righting arms may be small.

We have already referred to curves of this type (*see* curve No. 8, fig. 223), and have pointed out that in such cases the value of GM may, with perfect safety, be quite small; and as this ensures a long rolling period, a vessel so circumstanced should prove an easy roller, and thus a comfortable boat in a seaway

The value of GM in a case like this, should be limited by the consideration of the vessel being stable, and not too tender, in the upright position throughout the voyage. That is, over and above a sufficient margin to cover diminutions from causes that may be anticipated, such as the burning out of the bunker coal and the increase in weight of the deck cargo through becoming saturated with sea water, a certain minimum value of GM, such as previous experience with the vessel may suggest, should be provided. A metacentric height of greater value is unnecessary, and, to the extent of the excess, might be considered as actually detrimental to the vessel.

EFFECT ON STABILITY OF A SHIFT OF CARGO.—To calculate the quantity involved in any particular shift of cargo is not always an easy matter. In the case of oil cargoes, where there is a free surface, the

Fig. 234.



quantity shifted through the heeling of the vessel may be accurately determined, since the oil cargo is always horizontal, but, as we have seen, owing to friction between the parts, dry homogeneous cargoes do not move so quickly nor so definitely as oil, and the same rules cannot be applied. Still, with grain, an approximation may be made to the heeling effect of the worst shift of cargo that is likely to take place in a given case, when the plans of the vessel are available.

Fig. 234 is the midship section of a grain-laden vessel. The horizontal dotted line $a\,a_1$ shows the grain level, assuming the settlement to have taken place evenly, and is drawn at a distance below the deck, corresponding to the anticipated maximum settlement of the grain, *i.e.*, about 8 per cent. of the depth of the hold. $b\,b_1$ is the ultimate line of the grain surface when the cargo has shifted, and may be taken to lie to the horizontal at the angle of repose of the grain. The wedge of grain $a_1 d\, b_1$

now occupies the position $a\,c\,b\,d$, its centre of gravity moving from g_1 to g_2, and the common centre of gravity of vessel and cargo moving from G to G_1.

If w = weight of cargo shifted in tons,
W = displacement of vessel in tons,
$g_1 g_2$ = travel of centre of gravity of shifted cargo in feet,

$$G\,G_1 = \frac{w \times g_1 g_2}{W} \text{ feet;}$$

$G\,G_1$ will be parallel to the line joining g_1 and g_2; the point G will thus be raised relatively to the keel as well as moved laterally. For small angles, however,

$$G\,G_1 = G\,M \times \theta \text{ (nearly).}$$

From which equation θ, the angle at which the vessel comes to rest, may be obtained.

This lateral movement of the centre of gravity obviously means a reduction of the righting arms, and in a vessel originally tender might endanger her safety. In the case assumed there is no middle line bulkhead; if such were fitted, the angle of heel, as in the case of a liquid cargo, would be reduced.

Practical Example.—In a grain-laden vessel of 48 feet beam, 9000 tons displacement, and 18 inches $G\,M$, 50 tons of cargo is shifted transversely through a distance of 27 feet. Calculate the angle of heel.

In this case—

$$G\,G_1 = \frac{50 \times 27}{9000} = G\,M \times \theta$$

$$\therefore \quad \theta = \frac{50 \times 27}{9000 \times 1{\cdot}5} = {\cdot}10$$

$$= 6^\circ \text{ nearly.}$$

If the shift of cargo were considerable, it might be necessary to allow for the vertical as well as the transverse movement in determining the angle of heel.

Let h = the vertical distance between g_1 and g_2,
d = the horizontal distance between g_1 and g_2;

then, Vertical movement of centre of gravity $= \dfrac{w \times h}{W}$ feet,

Horizontal movement of centre of gravity $= \dfrac{w \times a}{W}$ feet.

This determines the position of the centre of gravity after shifting has taken place. Join this point with M, the metacentre, then the angle between this line and the middle line is the angle of heel required, provided it does not exceed 10 to 15 degrees.

In the previous example, if the cargo shifted had been 150 tons, the vertical movement 5 feet, and the transverse movement 20 feet—

$$\text{Tan } \theta = \frac{\frac{w\,d}{W}}{GM - \frac{w\,h}{W}}$$

$$= \frac{w\,d}{GM \times W - w\,h}$$

Substituting values—

$$\text{Tan } \theta = \frac{150 \times 20}{1{\cdot}5 \times 9000 - 150 \times 5}$$

$$= {\cdot}235$$

$$= 13^\circ \text{ nearly.}$$

Since shifting of cargo may be more or less expected in grain-laden ships, the stability of such vessels should be carefully considered before setting out on a voyage, the effect of the shifting of the greatest quantity of cargo that may be anticipated, estimated, and a safe margin of stability provided.

EFFECT OF BURNING OUT BUNKER COAL.—This is a point of importance in fixing upon a value of GM with which to start a voyage. While it is desirable, in the interests of steadiness at sea, to avoid excessive initial stability, if it is known that consumption of the coal will entail a diminution of the metacentric height, the latter to begin with should be sufficiently great to allow for such loss. In many cargo vessels the centre of gravity of the bunkers is higher in position than the common centre of gravity of ship and load, and the removal of the coal thus leads to an increase of the metacentric height and righting levers; but in some steamers there are large reserve bunkers extending no higher than the lower deck, and the rise in the vessel's centre of gravity, due to the burning of the coal, causes considerable reduction in the metacentric height, reductions amounting to as much as $1\frac{1}{2}$ feet being not unknown.* The need of investigating this question of the bunkers is thus manifest.

Curves showing the condition with bunker coal in and out are now supplied by many shipbuilding firms with new vessels for the guidance of the officers.

BALLASTING is the name given to the loading of deadweight other than cargo, to enable a vessel to make a voyage in safety.

Unfortunately, vessels do not always find an available freight at the port of discharge, and have consequently often to make the return journey, or proceed, at least, to another port, without a remunerative cargo. Every seaman knows that it would be imprudent to attempt such a voyage, if it meant crossing the sea with a chance of rough weather, with an absolutely "light" ship. He knows, in the case of a sailing-ship, that the stability would probably be insufficient to withstand the heeling effect of a spread of

* See a paper by the late Dr. Elgar in the T.I.N.A. for 1884.

sails; in the case of many steamers, that the stability would be dangerously small, owing to a relatively high position of the centre of gravity; and, in all such cases, even assuming sufficient stability, that their slight grip of the water, and their greatly exposed surfaces, would cause them to be the sport of wind and waves, with a probability of serious damage before the end of the voyage. Accordingly, if he cannot get cargo, he takes ballast aboard, *i.e.*, sand, gravel, rubbish, or water. Frequently, a combination of water and one of the other forms of ballast is used. In steamers, bunker coal is useful in this way.

As to the total amount of ballast required, it should be sufficient to secure the following:—

(1) An adequate immersion of the propeller in steamers to prevent racing of the engines and breaking of the tail shaft, and undue strains being brought upon the stern frame.

(2) A stability curve of suitable area and range, with considerable righting moments at large angles of inclination in all vessels.

(3) A good floating depth to give grip of the water, and to reduce, as far as possible, when among waves, the effective wave-slope angle, which, as we have seen, directly affects the magnitude of the arcs through which a vessel will roll.

It would appear that a great variety of opinion exists as to the ratio which should hold between the amount of ballast and the full deadweight. In a paper by Mr. Thearle on "Ballasting of Steamers for Atlantic Voyages" in the Transactions of the Institution of Naval Architects for 1903, the actual figures in 41 cases are given, and show the ballast draught to vary from ·5, to ·72 of the load draught, and the amount of ballast from ·24 to ·51 of the full deadweight, shelter-deck vessels with high sides having the greatest proportion of ballast and draught, and small vessels with flush decks or with very short erections the smallest. Mr. Thearle points out, that for good results, the amount of ballast in ordinary tramp steamers when making voyages across the Atlantic in winter should not be less than one-third of the full deadweight, with the vessels 4 to 5 feet by the stern, and the propellers about two-thirds immersed, experience having shown that damage in the form of loose rivets at the ends is likely to result where there is a less proportion of ballast. A point in ballasting not less important than having a sufficiency of deadweight, is that the latter should be carefully stowed. The same principles should, in fact, be followed here as in ordinary loading operations; that is, the ballast should not be placed too high, or the stability may be endangered, nor too low, or there may be an undue depression of the ship's centre of gravity, and a consequent abnormal increase in the metacentric height, a state of things, as we have seen, inevitably leading to excessive rolling and great straining of the hull.

In vessels having 'tween decks, part of the ballast, if of sand or rubbish, should be placed there; in single-decked vessels some special arrangement should be made to raise the centre of gravity, either by fitting temporary bulkheads to bank up the ballast in the holds, or by carrying part of it properly secured on deck, or by any plan which experience and the special circumstances may suggest. Unfortunately, such precautions are not always taken, and thus many vessels in ballast are unduly stiff. Nowadays, particularly in steamers, water ballast is largely used. It has the obvious advantage over sand or rubbish of being more easily, quickly, and cheaply loaded and discharged. Most modern cargo steamers have double bottoms and peak tanks; some large vessels have also one or more deep tanks extending from the bottom of the vessel to the first or second deck; while in a few instances ballast tanks have been built into the corners under the deck, also on top of the deck between the hatches, and in other places. For details of the construction of ballast tanks, the reader is referred to the chapter on practical details.

A double bottom, of course, except in the special case in which it extends up the ship's side, is not the best place for ballast. The amount carried in a double bottom, however, is not of itself sufficient for a sea voyage, and if the remainder is loaded in deep tanks, or in corner tanks under the deck, of which the capacity has been carefully considered, a satisfactory immersion and a metacentric height such as to ensure good behaviour at sea may be attained.

Where the ballast supplementary to that in the double bottom consists of stones or rubbish, it should be disposed so as to obtain a suitable position of the centre of gravity.

Scarcely less important than the vertical distribution of ballast, is the placing of it longitudinally. In steamers, as already noted, there should be a preponderance aft to properly immerse the propeller. But this allowed for, the remainder should be disposed so as to obtain a suitable pitching period. This period is, we know, lengthened by winging out the weights towards the extremities and shortened by concentrating them amidships. To obtain, therefore, a satisfactory quick fore-and-aft motion, and avoid the constant tendency which a slow-moving vessel has to bury her extremities in the waves, supplementary ballast, whether water in deep tanks, or stones and rubbish, should be placed towards amidships, while peak tanks, where such are required, should be kept within moderate limits.

Unfortunately, concentrating the ballast amidships in this way is likely to lead to the development of considerable bending moments, but these cannot well be avoided, and the strength of vessels should be made sufficient to meet all such demands.

DANGER OF FILLING BALLAST TANKS AT SEA.—Mention has already been made of this point, which should be abundantly clear from the remarks on the loading of liquid cargoes. It is to be feared that many officers do not fully appreciate the danger of this practice. The ballast is loaded in order to increase the metacentric height and, therefore, the stability, but, as we have seen, the presence of the free surface during the process may deprive a vessel of her effective metacentric height and cause her to heel to a dangerous angle, if not to capsize. It is important to remember that the formula—

$$G\,m = \frac{i}{V},$$

as previously remarked, shows that it is the extent of the area of the free surface, not the magnitude of the quantity of liquid in the tank, which influences the metacentric height. And commanding officers should see that when the ballast is out, the tanks are quite empty, particularly in the case of midship compartments which are of considerable breadth.

As a concrete example, take the following:—In a certain vessel of 7000 tons displacement, it is intended to run up a 'midship compartment of the double bottom, which is 80 feet long, 35 feet broad, 4 feet deep, approximately rectangular in shape, and has a capacity for 320 tons of salt water.

Given that the distance between the ship's centre of gravity and the top of the tank is 12 feet, calculate the reduction in metacentric height when the water in the tank is 1 foot deep, the metacentre being assumed to remain at the same height above the base throughout.

We have here to consider two things, viz., the fall in the centre of gravity due to the admission of the water, and the virtual rise in the centre of gravity due to the free liquid surface. Taking the centre of gravity of the admitted water to be at half its depth, we have—

$$\text{Fall in centre of gravity of vessel} = \frac{80 \times 15{\cdot}5}{7080} = {\cdot}17 \text{ feet.}$$

In finding the effect of the free surface, if we suppose the fore-and-aft girders, including the middle one, to be pierced with holes, the whole breadth of the tank will be available in estimating the moment due to the shifting wedges of water, and, therefore—

$$i = \frac{80 \times 35 \times 35 \times 35}{12} = 285833 \text{ (foot units),}$$

and

$$\text{Virtual rise in centre of gravity} = \frac{285833}{7080 \times 35} = 1{\cdot}15 \text{ feet.}$$

We thus get,

$$\text{Reduction of metacentric height} = 1{\cdot}15 - {\cdot}17.$$
$$= {\cdot}98 \text{ feet.}$$

This reduction is serious, and in the case of many vessels would cause instability in the upright position. It is the general practice, however, to fit the centre line division without perforations. In that case, the virtual rise in the centre of gravity would be a fourth of the above amount, or ·29 feet, and the reduction of the metacentric height would only be—

$${\cdot}29 - {\cdot}17 = {\cdot}12 \text{ feet,}$$

showing the powerful effect of a watertight centre division in a double bottom.

STABILITY INFORMATION FOR COMMANDING OFFICERS.—A common plan with many shipbuilders in supplying stability information to new vessels for the guidance of the officers, is to provide diagrams of curves depicting the nature of the stability under certain anticipated conditions of loading and ballasting, along with such remarks as may be necessary for the proper interpretation of the curves. In closing the present chapter, we shall give two examples of such stability diagrams, and shall discuss briefly how they may be employed in the actual working of vessels. Fig. 235 is a diagram for a modern cargo steamer of the following dimensions:—Length 395 feet 6 inches, breadth 51 feet 6 inches, moulded depth 29 feet 3 inches, mean load-draught 23 feet $11\frac{1}{2}$ inches. The vessel has a short poop, bridge, and forecastle, disconnected, and a main 'tween decks; she is adapted to carry water ballast in a double bottom and in both peaks.

Fig. 236 is a similar diagram for a smaller cargo steamer also of modern design. The dimensions are:—Length 351 feet 0 inches, breadth 49 feet 3 inches, moulded depth 28 feet 5 inches, mean load-draught 23 feet $6\frac{3}{4}$ inches. The erections consist of poop, bridge, and forecastle; there is also a main 'tween decks, and accommodation for water ballast in a double bottom and in the after peak.

Curves *A* to *H* in each diagram refer to the following conditions:—

1st condition (curve *A*).—Light ship, *i.e.*, vessel complete, water in boilers, but no cargo, bunker coal, stores or fresh water aboard, and all ballast tanks empty.

2nd condition (curve *B*).—Same as 1st, but with bunker coal, stores, and fresh water aboard.

3rd condition (curve *C*).—Vessel ready for sea, water in boilers, bunker coal, stores, and fresh water aboard, and the holds and 'tween decks filled with a homogeneous cargo of such density as just to bring the vessel to her legal summer load-line.

4th condition (curve *D*).—Same as 3rd, but with bunker coal, stores and fresh water consumed, approximating to the condition at the end of a voyage.

5th condition (curve *E*).—Vessel ready for sea, water in boilers, bunker coal, stores and fresh water aboard, and all ballast tanks filled.

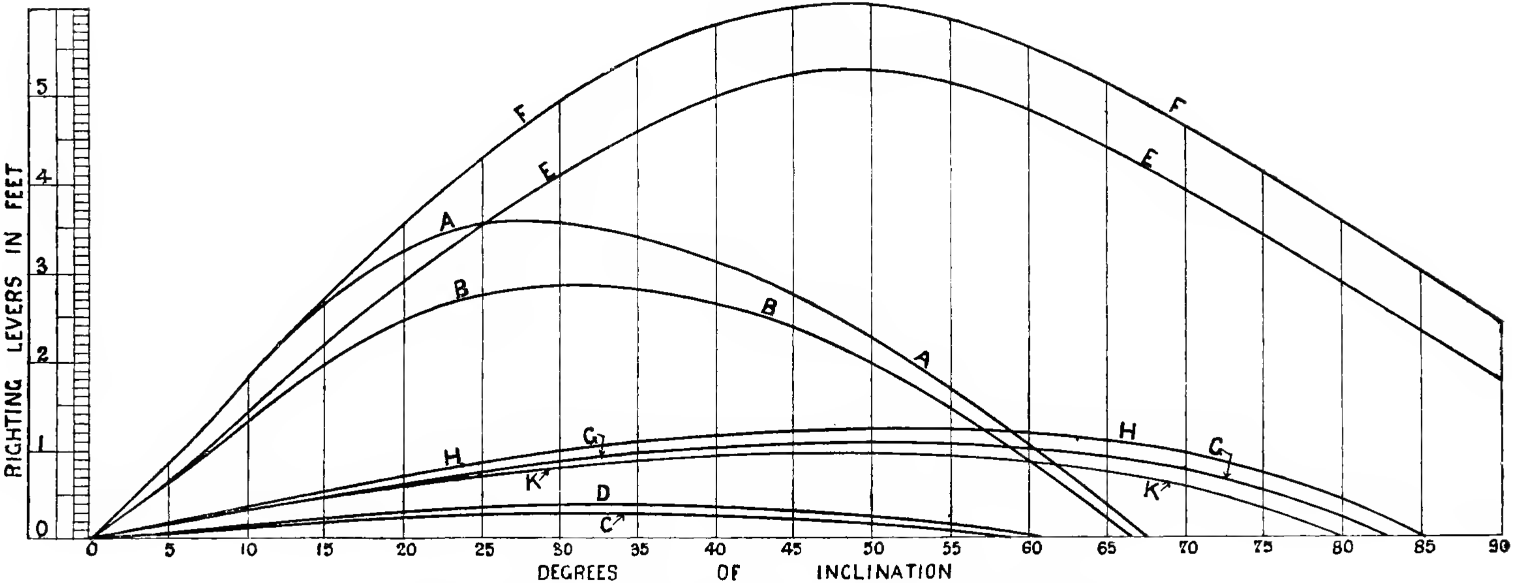
6th condition (curve *F*).—Same as 5th, but with bunker coal, stores and fresh water consumed.

7th condition (curve *G*).—Same as 3rd, but laden with a coal cargo, part of the bridge 'tween decks being empty.

8th condition (curve *H*).—Same as 7th, but with bunker coal, stores and fresh water consumed.

In the case of the larger vessel, it will be observed that the 3rd condition (curve *C*, fig. 235) is a critical one, the stability reserve being very small. When loading a cargo of the given density, it would probably be considered desirable, in the interests of the vessel's safety, to remove some of the cargo from the bridge 'tween decks, and run up a compartment of

Fig. 235.

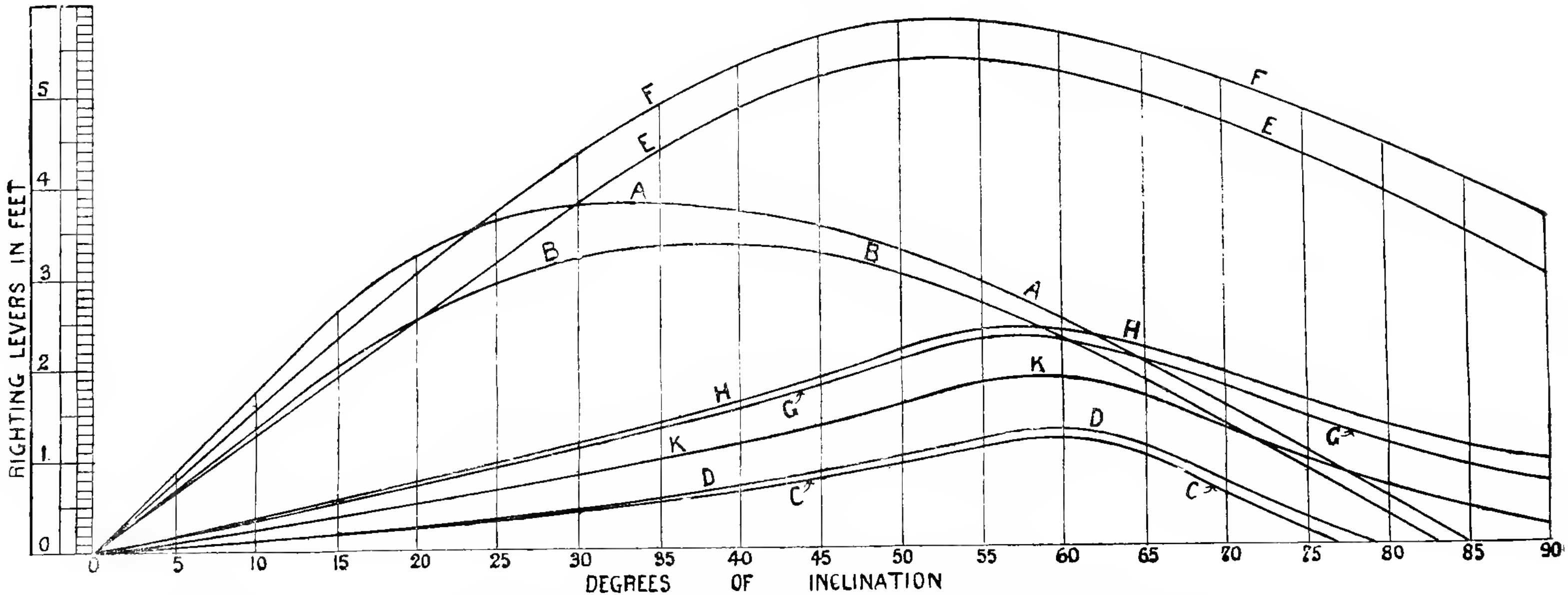


S.S. 395′ 6″ × 51′ 6″ × 29′ 3″. TABLE OF CONDITIONS.

Index to Curves.	Water Ballast.	Cargo.	Bunker Coal Stores and F.W.	Displacement.	Mean Draught.		Metacentric Height.	Maximum Righting Lever.	Angle of Maximum Righting Lever.	Height of C.G. above Base.
	Tons.	Tons.	Tons.	Tons.	Feet.	ins.	Feet.	Feet.	Degrees.	Feet.
A	—	—	—	3150	7	$10\frac{3}{4}$	10·59	3·57	$29\frac{1}{4}$	21·11
B	—	—	592	3742	9	2	7·22	2·85	$32\frac{1}{4}$	21·34
C	—	7085	592	10827	23	$11\frac{1}{2}$	0·66	0·26	31	20·25
D	—	7085	—	10235	22	9	0·75	0·34	37	20·12
E	1362	—	592	5104	12	$0\frac{3}{4}$	7·57	5·32	$49\frac{1}{4}$	16·60
F	1362	—	—	4512	10	$9\frac{3}{4}$	9·93	6·10	$49\frac{1}{4}$	15·82
G	—	7085	592	10827	23	$11\frac{1}{2}$	1·90	1·10	52	19·01
H	—	7085	—	10235	22	9	2·07	1·22	$49\frac{1}{4}$	18·8

T

Fig 236.



S.S. 351′ 0″ × 49′ 3″ × 28′ 5″ TABLE OF CONDITIONS.

Index to Curves.	Water Ballast.	Cargo.	Bunker Coal Stores and F.W.	Displacement.	Mean Draught.		Metacentric Height.	Maximum Righting Lever.	Angle of Maximum Righting Lever.	Height of C.G. above Base.
	Tons.	Tons.	Tons.	Tons.	Feet.	ins.	Feet.	Feet.	Degrees.	Feet.
A	—	—	—	2730	7	11	10·10	3·85	34	18·60
B	—	—	338	3068	8	$9\frac{1}{4}$	7·77	3·40	39	19·00
C	—	6000	338	9068	23	$6\frac{3}{4}$	0·70	1·20	58	19·32
D	—	6000	—	8730	22	9	0·69	1·26	58	19·21
E	1016	—	338	4084	11	$4\frac{1}{4}$	7·33	5·40	54	15·57
F	1016	—	—	3746	10	6	8·93	5·84	56	14·97
G	—	6000	338	9068	23	$6\frac{3}{4}$	2·01	2·30	$60\frac{1}{2}$	18·01
H	—	6000	—	8730	22	9	2·06	2·41	$61\frac{1}{2}$	17·84

the double bottom so as to bring down the centre of gravity and improve the stability.

In vessels of this size and description, safety demands that the righting arms at inclinations of 30 degrees and 45 degrees should not be less than about ·8 of a foot. Let us find what fall in the centre of gravity would be necessary to secure this in the present case. At 30 degrees the righting lever is ·26 feet; it has thus to be increased by (·8 − ·26) = ·54 feet.

Assuming the draught to remain unchanged—

$$\left.\begin{array}{l}\text{Increase of righting arm at}\\ \text{inclination of 30 degrees}\end{array}\right\} = \text{Fall in centre of gravity} \times \sin. 30 \text{ degrees.}$$

$$\cdot 54 = \text{Fall in centre of gravity} \times \cdot 5,$$

$$\therefore \text{ Fall in centre of gravity} = \frac{\cdot 54}{\cdot 5} = 1 \cdot 08 \text{ feet.}$$

The curve of stability under the new conditions may now be obtained by increasing the ordinates of curve *C* throughout by the amount—

1·08 × sine of angle of inclination.

In this way curve *K*, fig. 236, has been derived.

With a stability diagram like fig. 235 or fig. 236 ready to hand, an officer should be able in most cases to satisfy himself as to the state of his vessel. In making deductions, however, he must be careful to note that he can only deal with draughts for which he has curves; also, that differences in stowage may quite alter the nature of the stability.

This would appear to limit the utility of the curves, but it may be pointed out, with regard to draughts, that a ship is usually either light, fully loaded, or in ballast, so that in this respect the curves should be found generally applicable. Differences in stowage give rise to more trouble. The stowage of a general cargo, or of homogeneous cargoes other than those for which special curves are provided, lead to variations in the positions of the centre of gravity from those of the standard conditions at the same draught. To obtain a stability curve for any such new condition, the amount of the rise or fall of the centre of gravity from the position of a standard case must be known, and, with the information usually given, the only way of obtaining this would be by means of a special heeling experiment. If, however, the position of the centre of gravity above the base corresponding to the various conditions were stated, as in the examples given, a change in the position of the centre of gravity might be approximated to by a simple moment calculation. It would only remain, then, to deduce the righting levers of the new curve from those of the appropriate standard curve, by deducting or adding at each inclination the amount—

Rise or fall of centre of gravity × sine of angle of inclination.

The tables of conditions supplied with the diagrams of stability curves are very useful in working out problems like the foregoing. Suppose, for instance, it were intended to load the smaller of the two given vessels with a full general cargo. It would be first necessary, by means of the capacity plan, which is part of the equipment of all modern vessels, to approximate to the positions of the various weights forming the cargo; then, by a moment calculation, to combine these weights and heights with those of the light ship (taken from the table) to obtain the height of the common centre of gravity of vessel and cargo. Such a case is worked out in detail on page 192.

Suppose this done in the present case,* and the centre of gravity found to be ·5 feet above the position corresponding to a full coal cargo (curve *G*, fig. 236). The levers of the required curve are equal to those of curve *G*, decreased throughout by the amount—

$$\cdot 5 \times \text{sine of angle of inclination},$$

as already described. In the subjoined table the righting levers at 15 degrees, 30 degrees, etc., are tabulated, and the stability curve, marked *K*, is plotted in fig 236.

Angles of Inclination.	Ordinates of Standard Curve (Coal Cargo).	Rise of C.G. × Sine of Angle of Inclination.	Ordinates of required Stability Curve.
Degrees.	Feet.	Feet.	Feet.
15	·52	·13	·39
30	1·10	·25	·85
45	1·78	·35	1·43
60	2·27	·43	1·84
75	1·40	·48	·92
90	·70	·50	·2

As another example, and in this case referring to the larger vessel, suppose 200 tons of coal to be put into the bridge 'tween decks in excess of the amount allowed for in curve *G* (fig. 235), the same quantity being omitted from the lower hold to keep the mean draught as before. Let the height through which the centre of gravity of the coal has been raised be 20 feet, then—

$$\left.\begin{array}{r}\text{Rise of centre of gravity}\\ \text{of vessel}\end{array}\right\} = \frac{200 \times 20}{10827} = \cdot 37 \text{ feet.}$$

The levers of curve *G* (fig. 235), reduced by ·37 × sine of angle of inclination, furnish the data for constructing the stability curve of the new condition (this curve is not shown in the diagram).

When a diagram of metacentres is available, along with a table of conditions, the *GM* for any condition of loading and ballasting can be quickly determined. Thus, if it were decided to load supplementary ballast in the larger vessel when in the 5th condition, with a view to increasing her grip of the water and making her more navigable, the new *GM* might be arrived at in the following manner:—

* No reference has been made above to the question of trim, but, of course, in all loading problems this must be kept in view and the cargo distributed to obtain the best results, approximate calculations being made where necessary.

Referring to the table of conditions, the metacentric height in the given standard case is found to be 7·57 feet. As this is excessive, the supplementary ballast should be stowed high, particularly as the stability curve (*see* *E*, fig. 235) is of great area and range.

With regard to the amount of the additional ballast, let it be sufficient to make the total equal to about a third of the full deadweight, this having been shown to be a good average. Including bunker coal, stores and fresh water, the deadweight is 7677 tons, the total ballast should therefore be—

$$\frac{7677}{3} = 2559 \text{ tons.}$$

Including bunker coal and water ballast, 1954 tons is already loaded (*see* Table), therefore—

$$\begin{aligned}\text{Supplementary ballast} &= 2559 - 1954 \\ &= 605 \text{ tons.}\end{aligned}$$

or, in round figures, say 600 tons.

In the present case, if such would also suit the trim, it would be an advantage to put the whole amount into the bridge 'tween decks. If only half of it can be so placed, and the remainder is passed into the main 'tween decks, the height of the centre of gravity will be as obtained below, the centres of the supplementary ballast being assumed taken from the capacity plan as before.

Items.	Weights.	Heights of C.G. above base.	Moments about base.
Ballast Condition (from table), -	5104	16·6	84726
Extra ballast in main 'tween decks,	300	26·0	7800
Extra ballast in bridge 'tween decks,	300	33·5	10050
	5704		102576

$$\text{Height of centre of gravity above base} = \frac{102576}{5704} = 18 \text{ feet.}$$

From the deadweight scale, the increased load is found to sink the vessel $14\frac{1}{4}$ inches, which, added to 12 feet $0\frac{3}{4}$ inches, the mean ballast draught in the table, gives 13 feet 3 inches for the new condition. At this draught the metacentre is 23·3 feet above the base, so that—

$$\begin{aligned}\text{New } GM &= 23{\cdot}3 - 18{\cdot}0 \\ &= 5{\cdot}3 \text{ feet.}\end{aligned}$$

There is thus a reduction of over 2 feet in the metacentric height, which, with the increased immersion, should ensure an improvement in the vessel's behaviour at sea.

In the same way the effect of loading supplementary ballast in the smaller vessel may be determined.

As a final example—suppose the smaller vessel, laden with coal, has to discharge part of her cargo at a certain port, and afterwards proceed to sea under the reduced load. The question is, how should the unloading be done so as to leave her in a favourable condition for prosecuting the voyage?

Let the amount to be discharged be 2000 tons. From the deadweight scale, assuming the vessel to rise evenly. the reduction in draught due to unloading the coal is found to be 4 feet 9 inches. In order to keep the draught aft right, the coal should be taken out forward, say 1000 tons from the fore main-hold, and the remainder from the main and bridge 'tween decks.

Taking the centres from the capacity plan, and the figures for the loaded condition from the table, the calculation for the height of the centre of gravity is as follows:—

Items.	Weights.	Heights of C.G. above base.	Moments about base.
Displacement	9068	18·01	163315
Cargo removed from lower hold	– 1000	13·00	– 13000
Cargo removed from main 'tween decks	– 600	25·00	– 15000
Cargo removed from bridge 'tween decks	– 400	32·00	– 12800
	7068		122515

$$\text{Height of centre of gravity above base} = \frac{122515}{7068} = 17{\cdot}33 \text{ feet.}$$

The new mean draught is thus $23'\ 6\frac{3}{4}'' - 4'\ 9''$, or $18'\ 9\frac{3}{4}''$, and (from the metacentre diagram) the corresponding height of metacentre above the base is 19·9 feet.

$$\therefore \quad \text{New } GM = 19{\cdot}9 - 17{\cdot}33$$
$$= 2{\cdot}57 \text{ feet.}$$

This is a larger metacentric height than in the fully loaded condition, but as the displacement is less, it may be considered satisfactory.

If considered necessary, the trim might also be approximated in the above case by measuring from the capacity plan the distances between the centre of gravity of the load waterplane and the centres of the loads to be discharged, calculating the trimming moment, and dividing it by the moment to alter trim estimated in any of the ways described in Chapter VIII.

QUESTIONS ON CHAPTER XI.

1. In superintending the loading of his vessel, enumerate the points which should be kept in view by the commanding officer.

Explain why great initial stability in a vessel is conducive to bad behaviour at sea.

2. How should the items of a general cargo be stowed to obtain the best results at sea?
   (1) In the case of a vessel proportionately broad and shallow.
   (2) In the case of a vessel proportionately narrow and deep.

3. If, in the process of loading, a vessel is observed to suddenly list to port or starboard, what may be inferred as the probable cause, and how should the subsequent loading be conducted so as to bring the vessel back to the upright?

4. Whether does a general or a homogeneous cargo afford greater facilities for loading so as to produce a comfortable vessel at sea? Give reasons for your answer.

5. A vessel has been loaded to her maximum draught, with a homogeneous cargo which entirely fills her, and the master desires to ascertain the metacentric height before sailing. Explain how he may readily obtain this knowledge.

If the metacentric height were found to be deficient, what steps should the master take to correct it?

6. Write down the formula for the reduction in the metacentric height due to the presence of a free liquid surface in the hold.

One of the compartments of an oil steamer is partially filled with petroleum. Calculate the reduction in the metacentric height due to the free surface, given that the compartment is situated amidships, is 30 feet long, 42 feet broad, and approximately rectangular in shape at the level of the oil, and that the total displacement is 6500 tons. *Ans.*—·65 feet.

7. Show that the presence of a middle-line bulkhead greatly modifies the effect of a free liquid surface in the holds. Assuming a middle-line bulkhead in the vessel of the previous question, what would be the reduction in metacentric height?

8. Enumerate the precautions which should be taken in loading a vessel with oil in bulk.

Why are trunkways fitted in oil vessels?

9. Explain why it is that grain cargoes loaded in bulk are frequently found to shift during a voyage when bad weather has been encountered.

A grain-laden vessel of 7000 tons displacement has a metacentric height of 2 feet 6 inches. If 100 tons of cargo shift transversely through a distance of 18 feet, what will be the angle of heel, assuming the vessel to have been upright before the shifting took place?

Ans.—6°, nearly.

10. Show that the burning out of bunker coal may have an important influence on a vessel's condition.

If the coal in a particular vessel, whose margin of stability is small, is contained in 'tween decks as well as lower bunkers, how should it be worked out in the interest of the safety of the vessel?

11. A steamer 470 feet in length, 15,600 tons displacement, drawing 27 feet 6 inches forward and aft, has a reserve bunker containing 500 tons of coal. The centre of gravity of the latter is 10 feet below that of the vessel, and 35 feet before the centre of gravity of the load-waterplane. The tons per inch is 60, the longitudinal metacentric height is equal to the length of the vessel, and the transverse metacentric height at the start of the voyage is 1·5 feet. Assuming the transverse metacentre to remain at the same point while the vessel rises to the lighter draught, estimate approximately the draught and transverse metacentric height when the coal in the reserve bunker is consumed.

Ans.— Draughts { Forward, 26 feet, $2\frac{3}{4}$ inches. Aft, 27 feet, $4\frac{3}{4}$ inches. }
Metacentric height, 1·17 feet.

12. What is meant by ballasting? What should be aimed at in ballasting a steamer for a sea voyage? How would you expect a vessel to behave if laden with heavy ballast placed low down in the holds?

13. A cargo steamer of 7000 tons deadweight is to be ballasted for an Atlantic voyage. With bunker coal, stores and fresh water aboard, and water in boilers, the displacement is 3350 tons, the draught forward 8 feet and aft 10 feet 9 inches, and the centre of gravity 19·5 feet above the base. Water ballast is then loaded as follows:—1000 tons in double

bottom, centre 2 feet above base and 2 feet before 'midships; 650 tons in a deep tank abaft engine-room, centre 14 feet above base, and 54 feet abaft 'midships; 140 tons in the fore-peak, centre 16 feet above base, and 163 feet before 'midships. Assuming the centre of gravity of the parallel layer to be 3 feet forward of 'midships, the average tons per inch in way of increased immersion to be 34, the average moment to alter trim 1 inch 670 foot tons, and the height of the transverse metacentre above the base with the ballast aboard 19 feet; estimate approximately the draught and metacentric height when in ballast trim.

Ans.—Draughts: Forward, 11 feet, $4\frac{3}{4}$ inches. Aft, 16 feet, $1\frac{1}{4}$ inches. Metacentric height, 3·7 feet.

14. If, while on her voyage, a vessel should exhibit signs of "tenderness," show that it would be unsafe to attempt to remedy matters by running up a compartment of the double bottom.

Referring to example 11, it is proposed to fill a compartment of the double bottom, with a view to making good the reduction in metacentric height due to the consumption of the coal. The tank chosen for the purpose is approximately rectangular in shape, and has a perforated centre division; it is 60 feet long, 44 feet broad, and 4 feet deep, and has capacity for 300 tons of salt water. Assuming the top of the tank to be 18 feet below the centre of gravity of the vessel, estimate approximately the metacentric height when the tank is half full, and also when it is full.

Ans.—·58 feet. 1·56 feet.

15. A vessel whose load displacement is 7500 tons is being loaded in dock for a summer voyage. If the water in the dock weighs 63 lbs. per cubic foot, to what extent may the centre of the disc—that is, the legal summer load-line in salt water—be immersed? The area of the load waterplane is 12,000 square feet.

Ans.—4 inches.

16. In loading a general cargo, how should the heavy items be disposed longitudinally to ensure a vessel behaving well at sea? Give reasons.

17. What stability information should be supplied with a new vessel for the guidance of the officers?

APPENDIX A.

CHANGE OF DRAUGHT IN PASSING FROM FRESH TO SALT WATER.—Taking salt water to weigh 64 lbs. to a cubic foot, and fresh water 62·5 lbs., the number of cubic feet to a ton in each case is—

$$\text{Salt water, } \frac{2240}{64} = 35,$$

$$\text{Fresh water, } \frac{2240}{62 \cdot 5} = 35 \cdot 84.$$

Let now W = a vessel's displacement in tons,
then immersed volume in salt water = $35 \times W$ cubic feet,
and immersed volume in fresh water = $35 \cdot 84 \times W$ cubic feet.

In passing from fresh to salt water, the vessel thus rises out of the water to the extent—

$$35 \cdot 84\, W - 35\, W = \cdot 84\, W \text{ cubic feet.}$$

If A be the area of the waterplane in square feet, and d the distance through which the vessel rises in inches—

$$d = \frac{\cdot 84\, W \times 12}{A} \quad (1).$$

Let T be the tons per inch of immersion in salt water;

$$\text{then,} \quad T = \frac{A \times \frac{1}{12}}{35} = \frac{A}{420},$$

$$\therefore \quad A = 420\, T.$$

Substitute in (1), and we get—

$$d = \frac{\cdot 84\, W \times 12}{420\, T} = \frac{W}{41 \cdot 7\, T}.$$

If instead of fresh water, river water weighing 63 lbs. per cubic foot be assumed, the immersed volume will be—

$$W \times \frac{2240}{63} = 35\frac{5}{9}\, W \text{ cubic feet,}$$

and the difference in volume in river and in sea water,

$$\frac{5}{9}\, W \text{ cubic feet.}$$

Substituting this in the expression for d, we get—

$$d = \frac{W}{63\, T}.$$

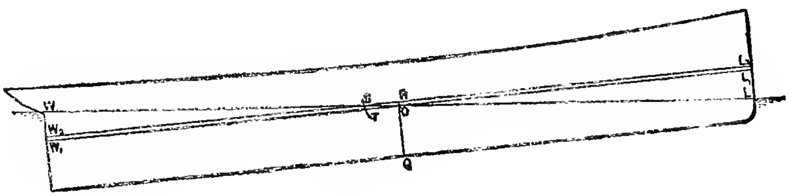
Thus, if a vessel of 9500 tons displacement, whose tons per inch at the load draught is 35, were to pass from river water at 63 lbs. per cubic foot into sea water, she would rise—

$$\frac{9500}{63 \times 35} = 4{\cdot}3 \text{ inches.}$$

MEAN DRAUGHT.—In reading a displacement from a displacement scale, it is the usual custom to employ the vessel's mean draught, *i.e.*, the sum of the actual draughts read off at the stem and stern divided by 2. This assumes the waterplane drawn at this mean draught to cut off wedges of equal volumes forward and aft, which in ordinary cases would not happen. To cut off a displacement closely approximating to that of the vessel when out of the normal trim, a line should be drawn parallel to the base through the point in which the actual waterline intersects the locus of the centres of gravity of waterplanes.

Fig. 237 shows a vessel trimming by the stern; $W L$ is the line of flotation at which it is required to know the displacement. Let $S T$ be the locus of the centres of gravity of the waterplanes. Through S, the point of intersection of this locus and the line $W L$, let $W_2 L_2$ be drawn parallel to the

Fig. 237.



base. The displacement to the waterplane $W_2 L_2$ will be very nearly equal to that to the original waterplane $W L$. At amidships draw $Q R$ normal to the keel line. $O Q$ is the mean draught corresponding to $W L$, but this mean draught, marked off on the displacement scale, would obviously give a reading less than the actual displacement by the amount of the layer between $W_1 L_1$ and $W_2 L_2$. The draught $Q R$ should therefore be employed.

Since the triangles $W W_1 O$ and $S R O$ are similar—

$$\frac{O R}{R S} = \frac{W W_1}{W_1 O},$$

$$\therefore \; O R = \frac{W W_1}{W_1 O} \times R S.$$

But $W W_1$ is half the trim, $W_1 O$ half the vessel's length, and $R S$ the distance abaft amidships of the centre of gravity of waterplane $W_2 L_2$. Thus $O R$, the amount to be added to the mean draught to get the draught to use with the displacement scale, is readily obtained.

In an actual case, if $R S$ were 4 feet, the length of vessel 360 feet, and the trim 6 feet by the stern, we should get—

$$O R = \frac{3}{180} \times 4 \times 12$$

$$= \frac{4}{5} \text{ of an inch,}$$

corresponding to an increase of displacement over that given by the mean draught of about 30 tons.

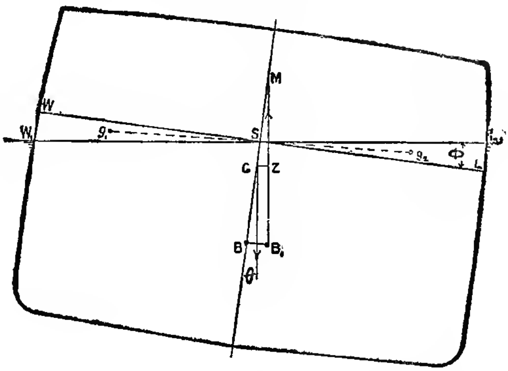
PROOF OF FORMULA $BM = \frac{I}{V}$.—In this formula, which expresses the height of the metacentre above the centre of buoyancy—

BM = Height of metacentre (transverse or longitudinal) above centre of buoyancy.

I = Moment of inertia of the waterplane about the middle line as axis in the case of the transverse metacentre, and about a transverse axis through the centre of gravity of the waterplane in the case of the longitudinal metacentre.

V = Volume of displacement.

Fig. 238.



Consider first the transverse metacentre. Fig. 238, which illustrates the case, is a transverse view of a vessel inclined through a small angle θ from the upright. Before the inclination took place, the centre of buoyancy was at B; it is now at B_1, and has thus travelled the distance BB_1. The line of the resultant upward pressure passes through B_1 and intersects the middle line in M, which by definition is the transverse metacentre.

In the act of heeling the wedge of displacement WSW_1 passes across the ship into the position LSL_1, its centre of gravity moving from g_1 to g_2 in a line parallel to BB_1. If V be the volume of the ship's displacement, and v the volume of either wedge—

$$V \times BB_1 = v \times g_1 g_2,$$

or $$V \times BM \times \theta = v \times g_1 g_2. \quad (1)$$

Where θ is the circular measure of the angle of inclination, which is assumed to be very small.

V is assumed to be known, so that to find BM it is only necessary to calculate the value of the quantity $v \times g_1 g_2$, *i.e.*, the moment due to the movement of the wedge of displacement across the ship. θ being small, S, the point of intersection of the water lines WL and $W_1 L_1$, is in the middle line.

Calling the half breadth of the waterplane amidships, b,

$$\text{Sectional area of wedge } WSW_1 \text{ or } LSL_1 = \frac{b^2}{2}.\theta,$$

and $$\text{Volume of a thin slice of either wedge } = \frac{b^2}{2}.\theta.\delta x,$$

δx being the thickness of the slice.

Also $$\left.\begin{array}{r}\text{Moment due to movement of this}\\ \text{volume across the ship}\end{array}\right\} = \frac{b^2}{2}.\theta.\delta x \times \frac{4}{3}b$$
$$= \frac{2}{3}b^3.\theta.\delta x.$$

Now the moment of the whole wedge is equal to the sum of the moments of all the slices into which it may be supposed divided. That is—

$$\left.\begin{array}{r}\text{Moment due to movement of}\\ \text{whole wedge}\end{array}\right\} = \Sigma\frac{2}{3}b^3.\theta.\delta x.$$

But $\Sigma\frac{2}{3}b^3\,\delta x$ is the expression for the moment of inertia of the waterplane about the middle line as axis. Calling this I, we get—

$$\left.\begin{array}{r}\text{Moment due to movement}\\ \text{of wedge}\end{array}\right\} = I \times \theta.$$

Substituting this for $v \times g_1 g_2$ in (1)—

$$I.\theta = V.BM.\theta$$

or, $$BM = \frac{I}{V}.$$

Take now the longitudinal metacentre. The inclination here is a fore-and-aft one, but except as modified by this circumstance, the proof is the same.

Fig. 239 shows a fore-and-aft view of the vessel with a slight inclination aft. B is the centre of buoyancy when floating at the waterplane W_1L_1, B_1 its position when at the line WL, m,* the intersection of the verticals through B and B_1, being the longitudinal metacentre; O, the projection in the plane of the paper of the line of intersection of the waterplanes WL and $W_1 L_1$, is called the centre of flotation and occurs at the same point in the length as the centre of gravity of the waterplane WL. h_1, h_2 are the centres of the immersed and emerged wedges. As in the previous case, we have—

$$v \times h_1 h_2 = V \times BB_1$$

v being the volume of either wedge, and V that of the displacement.

The inclination being very small, $BB_1 = Bm \times \theta$. So that—

$$v \times h_1 h_2 = V \times Bm \times \theta \quad (2).$$

* m is used here instead of M to distinguish the longitudinal metacentre from the transverse.

To calculate the quantity $v \times h_1 h_2$, consider a small element of the volume of the emerged wedge distant x from o. The thickness of this element is $x \times \theta$, and if y be the breadth of the vessel at the place, and δx the dimension of the element in the direction of the vessel's length, its volume will be—

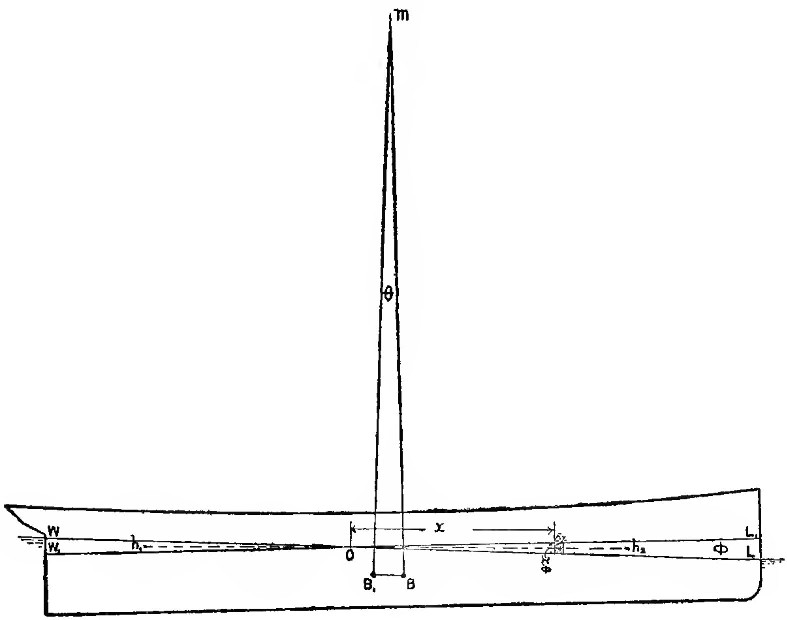
$$x.y.\theta.\delta x$$

and its moment about a transverse axis through o

$$x^2.y.\theta.\delta x.$$

The moment of the whole wedge is the sum of the moments of the

Fig. 239.



elements, or

$$\Sigma x^2.y.\theta.\delta x,$$

and the moment of both wedges, double this quantity, or

$$2\,\Sigma x^2.y.\theta.\delta x.$$

But $2\,\Sigma x^2.y.\delta x$ is the moment of inertia of the waterplane about a transverse axis through its centre of gravity o. Calling this I, and substituting the value for the moment of the wedges thus obtained in (2), we get—

$$I.\theta = V.B\,m.\theta$$

or,

$$B\,m = \frac{I}{V}.$$

CO-EFFICIENTS OF FORM.—These are useful in comparing one vessel with another. The following are the usual co-efficients employed :—

1. Co-efficient of area of load-waterplane.
2. Co-efficient of area of immersed midship section.
3. Block co-efficient.
4. Prismatic co-efficient.

1. Co-efficient of Area of Load-Waterplane.—This is the ratio of the area of the load-waterplane to that of a rectangle enclosing it. In a new design it is important to know how this ratio compares with the corresponding ratio of a vessel of known performance.

Example.—A vessel 360 feet long, 48 feet broad, is used as a basis in designing another vessel 340 feet long and 46 feet broad. The load-waterplane area in the standard case is 13,000 square feet, and it is intended to give the new vessel the same co-efficient of load-waterplane. Calculate the area of load-waterplane in the latter case.

The standard co-efficient is $\frac{13000}{360 \times 48} = \cdot 752$; the area of the load-waterplane in the new vessel will thus be :—

$$340 \times 45 \times \cdot 752 = 11505 \text{ square feet.}$$

The load-waterplane co-efficient is also useful in approximate calculations like the following:—A vessel of 330 feet length and 45 feet breadth floats at her load draught. If 150 tons of cargo be discharged from a compartment amidships, calculate the decrease in draught, assuming the co-efficient of the plane of flotation to be ·83, and the vessel to rise to a parallel waterplane.

$$\text{Area of L.W.P.} = 330 \times 45 \times \cdot 83 = 12325 \cdot 5 \text{ square feet.}$$

$$\text{Tons per inch immersion} = \frac{12325 \cdot 5}{420} = 29 \cdot 3.$$

$$\therefore \text{ Decrease in draught} = \frac{150}{29 \cdot 3} = 5 \cdot 12 \text{ inches.}$$

2. Co-efficient of Immersed Area of Midship Section.—This is the ratio of the area of immersed midship section to that of a rectangle having a depth equal to the vessel's moulded depth, and a breadth equal to the breadth of the vessel. Thus, in the case of a vessel of 28 feet breadth and 8 feet mean draught, which has an area of immersed midship section of 210 square feet, this co-efficient is—

$$\frac{210}{28 \times 8} = \cdot 937.$$

The immersed midship section co-efficient, like the previous one, is useful in designing, and should be carefully considered where speed is an important condition.

3. Block Co-efficient.—This is a volume ratio and expresses the relation between the immersed volume of a vessel's body and that of a rectangular figure surrounding it.

If

$$L = \text{length of vessel},$$
$$B = \text{breadth of vessel},$$
$$D = \text{draught of vessel},$$

Then, $$\text{Block co-efficient} = \frac{\text{volume of displacement}}{L \times B \times D}.$$

Example.—A vessel 500 feet long, 57 feet broad, has a moulded draught of 28 feet. Calculate the displacement, assuming a block co-efficient of ·76.

$$\text{Displacement} = 500 \times 57 \times 28 \times \cdot 76$$
$$= 606480 \text{ cubic feet.}$$

Again, find the block co-efficient of a vessel, given the following particulars:—Length 185 feet, breadth 26 feet, mean draught in salt water 10 feet, displacement, 1000 tons.

$$\text{Block co-efficient} = \frac{1000 \times 35}{185 \times 26 \times 10} = \cdot 727.$$

This co-efficient is of great value in comparing the forms of vessels, but it must be used with care. It is easy to show that two vessels of the same dimensions, block co-efficient, and displacement, may be very different in shape. In one case the midship section may be full and the ends fine; in the other, the midship section may be fine and the ends full. Generally speaking, in cargo boats having large block co-efficients, any fining of the body that is done should be reserved for the ends, the midship section being kept full. Where this has not been done, vessels hard to drive and difficult to steer have resulted.

4. Prismatic Co-efficient.—This expresses the ratio of the volume of displacement to the volume of a prism, whose section is the vessel's immersed midship section, and length the length of the vessel. Thus, in the case of a vessel 140 feet in length, which has an immersed midship section area of 210 square feet, and a displacement of 640 tons, the prismatic co-efficient is—

$$\frac{640 \times 35}{140 \times 210} = \cdot 76.$$

It will be readily seen that this co-efficient affords a closer means of comparing immersed forms than the block co-efficient, and in the case of the two vessels above referred to, would show the one of fine midship section and full ends to be of poor design, the prismatic co-efficient being relatively higher than in the other vessel. It should be observed that a relation exists between co-efficients 2, 3, and 4.

If V be the volume of displacement, A the area of immersed midship section, C_2, C_3, C_4, the co-efficients 2, 3, and 4 above described, then—

$$C_2 = \frac{A}{B \times D}$$

$$C_3 = \frac{V}{L \times B \times D}$$

$$C_4 = \frac{V}{L \times A}$$

Now, $$A = C_2 \times B \times D$$

$$\therefore \quad C_4 = \frac{V}{L \times B \times D \times C_2}$$

$$= \frac{C_3}{C_2}$$

So that if any two of the foregoing co-efficients be known, the third can be obtained.

It is usual to plot curves in the displacement scale diagram, showing how the above co-efficients vary with change of draught. From these curves the co-efficients at any draught may be obtained by simple measurement.

APPENDIX B.

Table of Natural Tangents, Sines, and Cosines.

Angle in Degs.	Tangent.	Sine.	Cosine.	Angle in Degs.	Tangent.	Sine.	Cosine.	Angle in Degs.	Tangent.	Sine.	Cosine.
0	—	—	1·0000	10	·1763	·1736	·9848	20	·3640	·3420	·9396
¼	·0043	·0043	·9999	10¼	·1808	·1779	·9840	20¼	·3689	·3461	·9381
½	·0087	·0087	·9999	10½	·1853	·1822	·9832	20½	·3738	·3502	·9366
¾	·0130	·0130	·9999	10¾	·1898	·1865	·9824	20¾	·3788	·3542	·9351
1	·0174	·0174	·9998	11	·1943	·1908	·9816	21	·3838	·3583	·9335
1¼	·0218	·0218	·9997	11¼	·1989	·1950	·9807	21¼	·3888	·3624	·9320
1½	·0261	·0261	·9996	11½	·2034	·1993	·9799	21½	·3939	·3665	·9304
1¾	0305	·0305	·9995	11¾	·2080	·2036	·9790	21¾	·3989	·3705	·9288
2	·0349	·0349	·9993	12	·2125	·2079	·9781	22	·4040	·3746	·9271
2¼	·0392	·0392	·9992	12¼	·2171	·2121	·9772	22¼	·4091	·3786	·9255
2½	·0436	·0436	·9990	12½	·2216	·2164	·9762	22½	·4142	·3826	·9238
2¾	·0480	·0479	·9988	12¾	·2262	·2206	·9753	22¾	·4193	·3867	·9222
3	·0524	·0523	·9986	13	·2308	·2249	·9743	23	·4244	·3907	·9205
3¼	·0567	·0566	·9983	13¼	·2354	·2292	·9733	23¼	·4296	·3947	·9187
3½	·0611	·0610	·9981	13½	·2400	·2334	·9723	23½	·4348	·3987	·9170
3¾	·0655	·0654	·9978	13¾	·2446	·2376	·9713	23¾	·4400	·4027	·9153
4	·0699	·0697	·9975	14	·2493	·2419	·9702	24	·4452	·4067	·9135
4¼	·0743	·0741	·9972	14¼	·2539	·2461	·9692	24¼	·4504	·4107	·9117
4½	·0787	·0784	·9969	14½	·2586	·2503	·9681	24½	·4557	·4146	·9099
4¾	·0830	·0828	·9965	14¾	·2632	·2546	·9670	24¾	·4610	·4186	·9081
5	·0874	·0871	·9961	15	·2679	·2588	·9659	25	·4663	·4226	·9063
5¼	·0918	·0915	·9958	15¼	·2726	·2630	·9647	25¼	·4716	·4265	·9044
5½	·0962	·0958	·9953	15½	·2773	·2672	·9636	25½	·4769	·4305	·9025
5¾	·1006	·1001	·9949	15¾	·2820	·2714	·9624	25¾	·4823	·4344	·9006
6	·1051	·1045	·9945	16	·2867	·2756	·9612	26	·4877	·4383	·8987
6¼	·1095	·1088	·9940	16¼	·2914	·2798	·9600	26¼	·4931	·4422	·8968
6½	·1139	·1132	·9935	16½	.2962	·2840	·9588	26½	·4985	·4461	·8949
6¾	·1183	·1175	·9930	16¾	·3009	·2881	·9575	26¾	·5040	·4500	·8929
7	·1227	·1218	·9925	17	·3057	·2923	·9563	27	·5095	·4539	·8910
7¼	·1272	·1261	·9920	17¼	·3105	·2965	·9550	27¼	·5150	·4578	·8890
7½	·1316	·1305	·9914	17½	·3152	·3007	·9537	27½	·5205	·4617	·8870
7¾	·1361	·1348	·9908	17¾	·3201	·3048	·9523	27¾	·5261	·4656	·8849
8	·1405	·1391	·9902	18	·3249	·3090	·9510	28	·5317	·4694	·8829
8¼	·1449	·1434	·9896	18¼	·3297	·3131	·9496	28¼	·5373	·4733	·8808
8½	·1494	·1478	·9890	18½	·3345	·3173	·9483	28½	·5429	·4771	·8788
8¾	·1539	·1521	·9883	18¾	·3394	·3214	·9469	28¾	·5486	·4809	·8767
9	·1583	·1564	·9876	19	·3443	·3255	·9455	29	·5543	·4848	·8746
9¼	·1628	·1607	·9869	19¼	·3492	·3296	·9440	29¼	·5600	·4886	·8724
9½	·1673	·1650	·9862	19½	·3541	·3338	·9426	29½	·5657	·4924	·8703
9¾	·1718	·1693	·9855	19¾	·3590	·3379	·9411	29¾	·5715	·4962	·8681

Angle in Degs.	Tangent.	Sine.	Cosine.	Angle in Degs.	Tangent.	Sine.	Cosine.	Angle in Degs.	Tangent.	Sine.	Cosine.
30	·5773	·5000	·8660	42½	·9163	·6755	·7372	55	1·4281	·8191	·5735
30¼	·5831	·5037	·8638	42¾	·9243	·6788	·7343	55¼	1·4414	·8216	·5699
30½	·5890	·5075	·8616	43	·9325	·6819	·7313	55½	1·4550	·8241	·5664
30¾	·5949	·5112	·8594	43¼	·9407	·6851	·7283	55¾	1·4686	·8265	·5628
31	·6008	·5150	·8571	43½	·9489	·6883	·7253	56	1·4825	·8290	·5591
31¼	·6068	·5187	·8549	43¾	·9572	·6915	·7223	56¼	1·4966	·8314	·5555
31½	·6128	·5224	·8526	44	·9656	·6946	·7193	56½	1·5108	·8338	·5519
31¾	·6188	·5262	·8503	44¼	·9741	·6977	·7163	56¾	1·5252	·8362	·5482
32	·6248	·5299	·8480	44½	·9826	·7009	·7132	57	1·5398	·8386	·5446
32¼	·6309	·5336	·8457	44¾	·9913	·7040	·7101	57¼	1·5546	·8410	·5409
32½	·6370	·5373	·8433	45	1·0000	·7071	·7071	57½	1·5696	·8433	·5373
32¾	·6432	·5409	·8410	45¼	1·0087	·7101	·7040	57¾	1·5849	·8457	·5336
33	·6494	·5446	·8386	45½	1·0176	·7132	·7009	58	1·6003	·8480	·5299
33¼	·6556	·5482	·8362	45¾	1·0265	·7163	·6977	58¼	1·6159	·8503	·5262
33½	·6618	·5519	·8338	46	1·0355	·7193	·6946	58½	1·6318	·8526	·5224
33¾	·6681	·5555	·8314	46¼	1·0446	·7223	·6915	58¾	1·6479	·8549	·5187
34	·6745	·5591	·8290	46½	1·0537	·7253	·6883	59	1·6642	·8571	·5150
34¼	·6808	·5628	·8265	46¾	1·0630	·7283	·6851	59¼	1·6808	·8594	·5112
34½	·6872	·5664	·8241	47	1·0723	·7313	·6819	59½	1·6976	·8616	·5075
34¾	·6937	·5699	·8216	47¼	1·0817	·7343	·6788	59¾	1·7147	·8638	·5037
35	·7002	·5735	8191	47½	1·0913	·7372	·6755	60	1·7320	·8660	·5000
35¼	·7067	·5771	·8166	47¾	1·1009	·7402	·6723	60¼	1·7496	·8681	·4962
35½	·7132	·5807	·8141	48	1·1106	·7431	·6691	60½	1·7674	·8703	·4924
35¾	·7198	·5842	·8115	48¼	1·1204	·7460	·6658	60¾	1·7856	·8724	·4886
36	·7265	·5877	·8090	48½	1·1302	·7489	·6626	61	1·8040	·8746	·4848
36¼	·7332	·5913	·8064	48¾	1·1402	·7518	·6593	61¼	1·8227	·8767	·4809
36½	·7399	·5948	·8038	49	1·1503	·7547	·6560	61½	1·8417	·8788	·4771
36¾	·7467	·5983	·8012	49¼	1·1605	·7575	·6527	61¾	1·8610	·8808	·4733
37	·7535	·6018	·7986	49½	1·1708	·7604	·6494	62	1·8807	·8829	·4694
37¼	·7604	·6052	·7960	49¾	1·1812	·7632	·6461	62¼	1·9006	·8849	·4656
37½	·7673	·6087	·7933	50	1·1917	·7660	·6427	62½	1·9209	·8870	·4617
37¾	·7742	·6122	·7906	50¼	1·2023	·7688	·6394	62¾	1·9416	·8890	·4578
38	·7812	·6156	·7880	50½	1·2130	·7716	·6360	63	1·9626	·8910	·4540
38¼	·7883	·6190	·7853	50¾	1·2239	·7743	·6327	63¼	1·9839	·8929	·4500
38½	·7954	·6225	·7826	51	1·2348	·7771	·6293	63½	2·0056	·8949	·4461
38¾	·8025	·6259	·7798	51¼	1·2459	·7798	·6259	63¾	2·0277	·8968	·4422
39	·8097	·6293	·7771	51½	1·2571	·7826	·6225	64	2·0503	·8987	·4383
39¼	·8170	·6327	·7743	51¾	1·2684	·7853	·6190	64¼	2·0732	·9006	·4344
39½	·8243	·6360	·7716	52	1·2799	·7880	·6156	64½	2·0965	·9025	·4305
39¾	·8316	·6394	·7688	52¼	1·2915	·7906	·6122	64¾	2·1203	·9044	·4265
40	·8391	·6427	·7660	52½	1·3032	·7933	·6087	65	2·1445	·9063	·4226
40¼	·8465	·6461	·7632	52¾	1·3150	·7960	·6052	65¼	2·1691	·9081	·4186
40½	·8540	·6494	·7604	53	1·3270	·7986	·6018	65½	2·1942	·9099	·4146
40¾	·8616	·6527	·7575	53¼	1·3391	·8012	·5983	65¾	2·2199	·9117	·4107
41	·8692	·6560	·7547	53½	1·3514	·8038	·5948	66	2·2460	·9135	·4067
41¼	·8769	·6593	·7518	53¾	1·3638	·8064	·5913	66¼	2·2726	·9153	·4027
41½	·8847	·6626	·7489	54	1·3763	·8090	·5877	66½	2·2998	·9170	·3987
41¾	·8925	·6658	·7460	54¼	1·3890	·8115	·5842	66¾	2·3275	·9187	·3947
42	·9004	·6691	·7431	54½	1·4019	·8141	·5807	67	2·3558	·9205	·3907
42¼	·9083	·6723	·7402	54¾	1·4149	·8166	·5771	67¼	2·3847	·9222	·3867

Angle in Degs.	Tangent.	Sine.	Cosine.	Angle in Degs.	Tangent.	Sine.	Cosine.	Angle in Degs.	Tangent.	Sine.	Cosine.
67½	2·4142	·9238	·3826	75	3·7320	·9659	·2588	82½	7·5957	·9914	·1305
67¾	2·4443	·9255	·3786	75¼	3·7982	·9670	·2546	82¾	7·8606	·9920	·1261
68	2·4750	·9271	·3746	75½	3·8667	·9681	·2503	83	8·1443	·9925	·1218
68¼	2·5065	·9288	·3705	75¾	3·9375	·9692	·2461	83¼	8·4489	·9930	·1175
68½	2·5386	·9304	·3665	76	4·0107	·9702	·2419	83½	8·7768	·9935	·1132
68¾	2·5714	·9320	·3624	76¼	4·0866	·9713	·2376	83¾	9·1309	·9940	·1088
69	2·6050	·9335	·3583	76½	4·1652	·9723	·2334	84	9·5143	·9945	·1045
69¼	2·6394	·9351	·3542	76¾	4·2468	·9733	·2292	84¼	9·9310	·9949	·1001
69½	2·6746	·9366	·3502	77	4·3314	·9743	·2249	84½	10·3853	·9953	·0958
69¾	2·7106	·9381	·3461	77¼	4·4193	·9753	·2206	84¾	10·8829	·9958	·0915
70	2·7474	·9396	·3420	77½	4·5107	·9762	·2164	85	11·4300	·9961	·0871
70¼	2·7852	·9411	·3379	77¾	4·6057	·9772	·2121	85¼	12·0346	·9965	·0828
70½	2·8239	·9426	·3338	78	4·7046	·9781	·2079	85½	12·7062	·9969	·0784
70¾	2·8635	·9440	·3296	78¼	4·8076	·9790	·2036	85¾	13·4566	·9972	·0741
71	2·9042	·9455	·3255	78½	4·9151	·9799	·1993	86	14·3006	·9975	·0697
71¼	2·9459	·9469	·3214	78¾	5·0273	·9807	·1950	86¼	15·2570	·9978	·0654
71½	2·9886	·9483	·3173	79	5·1445	·9816	·1908	86½	16·3498	·9981	·0610
71¾	3·0325	·9496	·3131	79¼	5·2671	·9824	·1865	86¾	17·6105	·9983	·0566
72	3·0776	·9510	·3090	79½	5·3955	·9832	·1822	87	19·0811	·9986	·0523
72¼	3·1239	·9523	·3048	79¾	5·5300	·9840	·1779	87¼	20·8188	·9988	·0479
72½	3·1715	·9537	·3007	80	5·6712	·9848	·1736	87½	22·9037	·9990	·0436
72¾	3·2205	·9550	·2965	80¼	5·8196	·9855	·1693	87¾	25·4517	·9992	·0392
73	3·2708	·9563	·2923	80½	5·9757	·9862	·1650	88	28 6362	·9993	·0349
73¼	3·3226	·9575	·2881	80¾	6·1402	·9869	·1607	88¼	32·7302	·9995	·0305
73½	3·3759	·9588	·2840	81	6·3137	·9876	·1564	88½	38·1884	·9996	·0261
73¾	3·4308	·9600	·2798	81¼	6·4971	·9883	·1521	88¾	45·8293	·9997	·0218
74	3·4874	·9612	·2756	81½	6·6911	·9890	·1478	89	57·2899	·9998	·0174
74¼	3·5457	·9624	·2714	81¾	6·8968	·9896	·1434	89¼	76·3900	·9999	·0130
74½	3·6058	·9636	·2672	82	7·1153	·9902	·1391	89½	114·5886	·9999	·0087
74¾	3·6679	·9647	·2630	82¼	7·3478	·9908	·1348	89¾	229·1816	·9999	·0043
								90	Infinite	1·0000	·0000

Weights of Materials used in Shipbuilding.

Material.	Per Cubic Foot.	Material.	Per Cubic Foot.
Steel	490 lbs.	Oak, Danzic	50 lbs.
Wrought Iron	480 ,,	Elm, English	35 ,,
Cast Iron	450 ,,	Elm, American	44 ,,
Gun Metal	534 ,,	Mahogany	53 ,,
Brass, Cast	518 ,,	Greenheart	64 ,,
Lead	712 ,,	Ash	46 ,,
Tin	462 ,,	Teak	52 ,,
Zinc, Sheet	449 ,,	Pine, White	35 ,,
Copper	549 ,,	,, Red	36 ,,
Aluminium, Cast	160 ,,	,, Yellow	30 ,,
Oak, English	58 ,,	,, Pitch	45 ,,

Rates of Stowage.*

Cargoes.	No. of Cubic Feet to 1 Ton.	Remarks.
Coal, Scotch	44	
Coal, Welsh	40	
Coal, Newcastle	44	
Manchester Bales	50	The figure may reach 160.
Pig Iron	9	Stowed with as little wood packing as possible.
Alkali in Casks	47	
Wheat	46	Varies from 40 to 52.
Flour	45	
Maize	46	
Barley	58	
Oats	72	
Cargo, rice in bags		Varies from different causes, weight placed in each bag, amount of paddy, etc. Cargo rice generally contains 20 per cent. paddy.
Tea	83-120	
Raw Sugar in Baskets	50	
Cotton, American	130	
Cotton, Indian	60	Machine pressed.
Cotton, Egyptian	70-220	
Jute	49-77	The closer being very much pressed by hydraulic power.
Wool, undumped	235	
Wool, washed and dumped	100	
Wool, greasy and dumped	84	
Potatoes	50	
Bacon and Hams in cases	64	
Peas and Beans	43-53	
Beef, frozen and packed	90-95	
Beef, chilled and hung in quarters	120	
Mutton, New Zealand	105-110	
Mutton, River Plate	115	

In the above table of Stowage Rates no attempt is made to allow for broken stowage, the figures being obtained from measurements of parcels where the lost space was little or nothing.

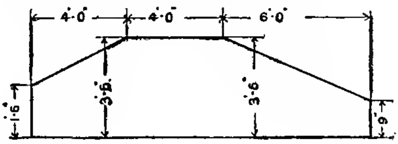
* From a paper by Professor Purvis in T.I.N.A. Vol. 26.

APPENDIX C.

Additional Questions.*

I.

1. Define the term area as applied to a plane surface. Calculate the area of the plate shown in the following sketch:—



Ans.—36·75 square feet.

2. Calculate the area in square feet of the following:—(1) A square of 11·5 feet side. (2) A rectangle of 15 feet length, $3\frac{3}{4}$ feet breadth. (3) Triangle 6·5 feet base, 8·25 feet height. (4) A circle of 12·25 feet diameter.

Ans.—(1) 132·25. (2) 56·25. (3) 26·81. (4) 117·85.

3. The ordinates in feet of a plane curve are 3, 5·5, 7·5, 8, and 9 respectively, the common interval being 8 feet. Between the first and second ordinates, a half ordinate 4·6 feet is introduced, and another of 8·6 feet between the fourth and fifth ordinates. Calculate the area in square feet.

Ans.—220·4.

4. State the Five-Eight Rule; upon what assumption is it based? Show how the Five-Eight Rule and Simpson's First Rule may be combined.

The half ordinates in feet of a portion of the load waterplane of a vessel are 3, 7, 8, 8·5, 6·5, and 5 respectively, and the common distance between them, 12 feet. Calculate the area in square feet, employing a combination of the Five-Eight Rule and Simpson's First Rule.

Ans.—822.

5. Referring to question No. 1, if the plate be steel $\frac{5}{8}$ of an inch thick, what is its weight in lbs.?

Ans.—937 lbs.

6. A solid wrought iron pillar, 18 feet in length, is $4\frac{1}{4}$ inches in diameter. Find its weight.

Ans.—851 lbs.

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* Many of these examples are based on questions set at the Board of Education Examinations in Naval Architecture.

7. A portion of a cyclindrical steel shaft tube, $1\frac{1}{2}$ inches thick, is 20 feet long, and its external diameter is 16 inches. Calculate its weight.

Ans.—4646 lbs.

8. A deck 9000 square feet in area is to be laid with pitch-pine planks 4 inches thick and 5 inches wide. There are two openings 16 feet × 12 feet and one opening 24 feet × 12 feet, which are not to be covered. Calculate

(1) The number of running feet of deck planking;
(2) The weight of the wood deck, excluding fastenings.

Ans.—(1) 19,987. (2) 124,920 lbs.

9. A derrick post 18 inches external diameter is built of $\frac{1}{2}$-inch steel plates. Estimate the weight of a length of 15 feet, neglecting straps and rivets.

Ans.—1402 lbs.

10. Define displacement. The areas of the vertical transverse sections of a ship up to the load waterplane in square feet are respectively 25, 105, 180, 250, 295, 290, 235, 145, and 30, and the common interval between them is 20 feet. The displacement in tons before the foremost section is 5, and abaft the aftermost section is 6. Find the load displacement in cubic feet and in tons (salt water).

Ans.—30,900; 882·8.

11.—Deduce a formula by which the tons per inch immersion at any draught may be ascertained.

Given that the half ordinates in feet of the load waterplane of a vessel are respectively ·2, 4, 8·3, 11·3, 13·4, 13·4, 10·4, 7·2, and 2·2, and the length of the plane 130 feet, calculate the tons per inch immersion in salt water.

Ans.—5·42.

12. A prism of rectangular section 120 feet long and 30 feet broad, floats at a draught of 15 feet. Calculate the displacement in tons in salt water; also construct the curve of displacement and the curve of tons per inch of immersion for this vessel.

Ans.—1543.

13. The tons per inch at the successive waterplanes of a vessel, which are $1\frac{1}{2}$ feet apart, are respectively 6·5, 6·2, 5·6, 4·5, and 0. Construct the curve of tons per inch on a scale of 1 inch to 1 foot of draught, and 1 inch to 1 ton.

14. How is a "deadweight scale" constructed? Of what use is it to the commanding officer?

15. What is meant by "mean draught"? Show that in the case of a vessel floating considerably out of her normal trim, it is incorrect to use the mean draught in reading the displacement from the displacement scale.

A vessel 300 feet in length floats in salt water and trims 8 feet by the stern. If the waterplane intersects the locus of the centres of gravity of waterplanes at a point 3 feet abaft amidships, measured parallel to top of keel, and the tons per inch immersion be 23, estimate the difference between the actual displacement of the vessel and that obtained from the displacement scale, using the mean draught.

Ans.—22 tons.

16. Obtain an expression giving the extent to which a vessel rises in passing from fresh to salt water.

A vessel whose displacement is 4000 tons, leaves a harbour in which the water is partly salt, and proceeds to sea. If the water in harbour weighs 1015 ozs. per cubic foot, calculate the number of inches through which the vessel will rise on reaching salt water, given that the tons per inch is 30.

Ans.—1·18.

17. A vessel of box form is 210 feet long, 30 feet broad, and has an even draught of water of 10 feet when floating in sea water. If a sheathing of teak 3 inches thick were worked over the bottom, and also over the ends and sides to a height of 12 feet above the bottom, what would be the additional weight, taking teak at 50 lbs. per cubic foot, and what would then be the draught of water.

Ans.—68 tons, 10 feet $2\frac{1}{4}$ inches.

18. A vessel carries in her hold a cube, each side of which is 10 feet. If the cube be put overboard and attached to the ship by means of a chain, what will be the effect upon the vessel's draught, the cube being supposed of greater density than salt water. The area of the vessel's waterplane is 4000 square feet.

Ans.—Vessel rises 3 inches.

19. A rectangular pontoon 100 feet long, 50 feet broad, 20 feet deep, is empty, and floating in sea water at a draught of 10 feet. What alteration will take place in the floating condition of the pontoon if the centre compartment is breached and in free communication with the sea, if—

(*a*) The pontoons were divided into five equal watertight compartments by transverse bulkheads, extending the full depth of the pontoon?

(*b*) The watertight bulkheads stopped at a deck which is not watertight, 12 feet from the bottom of the pontoon?

Ans.—{(*a*) Vessel will sink bodily 2 feet 6 inches.
(*b*) Vessel will founder.

II.

1. Show how the principle of moments is applied in obtaining the centre of gravity of a plane area such as a vessel's waterplane.

2. The half-ordinates of a load-waterplane of a vessel in feet, commencing from aft, are, respectively—·1, 5, 11·6, 15·4, 16·8, 17, 16·9, 16·4, 14·5, 9·4, and ·1, and the common interval is 11 feet. Find—

(1) The area of the plane in square feet;

(2) The distance of its centre of gravity from the 17-feet ordinate, stating whether the centre of gravity is before or abaft that ordinate.

Ans.—(1) 2732·4. (2) 2·83 feet forward.

3. A vessel is 180 feet long, and the transverse sections from the load-waterline to the keel are semicircles. Find the longitudinal position of the centre of buoyancy, the half-ordinates of the load-waterplane being 1, 5, 13, 15, 14, 12, and 10 feet, respectively.

Ans.—73·76 feet from 10-feet ordinate.

4. Given a diagram showing the locus of the centre of buoyancy, constructed as described in Chapter II., explain how the height of the centre of buoyancy corresponding with any waterplane may be ascertained.

5. Construct the locus of the centre of buoyancy for an upright prism of rectangular section, and also for a prism whose section is an equilateral triangle, and which floats with one of its faces horizontal.

6. Illustrate by a simple example the arrangement of the numerical work usually followed in an ordinary displacement paper for obtaining the displacement and position of the centre of buoyancy of a ship.

7. The load displacement of a ship is 5000 tons, and the centre of buoyancy is 10 feet below the load-waterline. In the light condition the displacement of the ship is 2000 tons, and the centre of gravity of the layer between the load and the light lines, is 6 feet below the loadline. Find the vertical position of the centre of buoyancy below the loadline in the light condition.

Ans.—16 feet.

IV.

1. Distinguish clearly between hogging and sagging strains. What causes these strains, and at what parts of a loaded cargo steamer are they likely to be a maximum?

2. What is a "curve of weight," and a "curve of buoyancy"? Describe how these curves are constructed for a vessel afloat in still water. What conditions must these curves comply with in relation to each other?

What are the usual assumptions made in constructing curves of weight and buoyancy for a vessel afloat among waves?

3. A vessel of box form 240 feet long, 40 feet broad, 20 feet deep, floats in salt water at a level draught of 8 feet. If the vessel's weight is 1000 tons evenly distributed, and she is loaded at each end for length of 70 feet with 600 tons, also evenly distributed, draw the curves of weight and buoyancy.

4. Write down the formula employed in calculating the longitudinal stress on the material at any point of a section of a beam under a longitudinal bending moment.

5. What assumptions are made in applying the formula referred to in the previous question to the case of a ship?

6. A steel beam of I section is 12 inches deep, $\frac{1}{2}$ inch thick, and has 6-inch flanges top and bottom. Calculate the moment of inertia of the section.

Ans.—254 inch units.

7. Referring to the previous question, if the beam is 20 feet long and supported at each end, and loaded in the middle with a weight of 6 tons, calculate the maximum tensile and compressive stresses in tons per square inch. The weight of the beam itself may be neglected in working out the problem.

Ans.—8·5, 8·5.

8. State the maximum longitudinal stress, as ordinarily calculated, in a large and in a small steel cargo steamer, when poised on waves of their own length. Give reason for any difference in the values.

9. In the case of some large steel passenger vessels having long superstructures of light build, the latter are cut about mid length, and a sliding joint made. What is the reason for this?

10. Enumerate the stresses to which ships are subjected which tend to produce changes in their transverse forms. State what parts assist the structure to resist change of form.

V.

1. Sketch and describe the three-deck, spar-deck, and awning-deck type of vessels.

2. In designing a cargo vessel of full form, state generally how you would proceed to shape the body with a view to securing the best results.

3. Sketch in profile a well-deck and a quarter-deck type of vessel. What are the essential features of each type?

4. Describe briefly the trend of development in the construction of cargo steamers. Sketch in section a vessel on the web frame system, also one with deep frames.

5. Sketch in outline midship section of a turret-deck steamer. What are the advantages claimed for this type over cargo vessels of ordinary form?

6. Describe with the sketches the Ropner Trunk Steamer, and the Isherwood Patent Ship. What are the chief features of these types?

VI.

1. Sketch and describe an ordinary bar keel.

2. How is the scarph of a bar keel formed? What is the length of a scarph in terms of the thickness of the keel? How are the rivets arranged, and what is their spacing?

3. Before proceeding with the framing it is necessary to set the keel

straight on the blocks. How are the keel lengths temporarily joined together so that this may be correctly done?

4. What is a side-bar keel? Is it a better or worse form than that referred to in the previous questions? Give reasons.

5. Mention any practical difficulty attendant on the constructing of a keel on the side-bar system, and state what means are taken to overcome it.

6. What are the advantages and disadvantages of projecting keels? Sketch and describe a form of keel which entails no outside projection, and show that the arrangement is satisfactory from a point of view of strength.

7. Show a good shift of butts of the flat keel with reference to those of the vertical keel and angles connecting them; also with reference to the garboard strakes of plating.

8. Describe, and show by sketches in section and side elevation, how an intercostal plate keelson (or vertical keel) is worked and secured in an ordinarily transversely framed vessel with a flat plate keel.

9. What are bilge-keels? Why are they fitted? Sketch an efficient form of bilge-keel, indicating the connections to the hull.

10. Why are hold keelsons fitted? Sketch a side and a bilge-keelson. What advantage is gained by fitting intercostal plates to the shell between the keelson bars?

11. How is the strength maintained at the joints of hold keelson bars? Make a sketch showing details of riveting, etc.

12. What is the usual spacing of transverse frames? Show by a sketch how a frame, reverse frame, and floorplate are connected.

13. What are frame heel pieces? Where are they fitted? They are not usually fitted at the ends of a vessel. Why?

14. Sketch an ordinary floor. How far does it extend up the ship's side? Where are floorplates usually joined? Make a sketch at a joint, showing the rivets.

15. Show in section the common forms of ship beams, and state where each section should be employed.

16. Why are beams cambered? Is their strength increased thereby? What is the usual camber of upper-deck beams?

17. Deck beams are sometimes fitted at every frame, and sometimes at alternate frames. State the circumstances in which each arrangement may be employed to most advantage.

18. Why are deck beams not reduced towards their ends, as on the principle of the girder they might be.

19. Describe the usual methods of forming "bracket," "slabbed," and "turned" beam knees, and state which, in your opinion, is most efficient.

20. Sketch a bracket knee showing in detail the connections to the frame and beam. What are Lloyd's requirements as to the number of rivets in beam knees?

21. In the design of a certain vessel, requiring by rule a tier of lower-deck beams at the usual spacing, it is proposed to modify or dispense with the latter in order to improve the facilities for stowage. Show how this might be done without reducing the strength.

22. What are web frames? Why are they fitted? Sketch a web frame showing all connections in a vessel having ordinary floors.

23. Discuss the relative merits of making the web frames continuous, and hold stringers intercostal, and *vice versa*. Show in detail the connection of a web frame to the margin-plate of an inner bottom.

24. What is meant by "deep framing"? What are the advantages of this system of construction over that consisting of combined ordinary framing and web frames.

25. Sketch and describe a M'Intyre ballast tank. What are its essential features?

26. Describe the cellular system of constructing double bottoms; compare it in details with the system referred to in the previous question.

27. Assuming continuous longitudinals and intercostal floors, show by sketches the construction of a cellular double bottom for a length of one compartment, indicating the man-holes through the longitudinals and tank top, and showing details of the connections of the longitudinals to the floors, tank top and shell.

28. Show by a sketch how the plating of the tank top or inner bottom is usually arranged, giving details of the butt and edge connections. At certain parts the plating is increased in thickness. Name these parts, and state why the increase is made.

29. Signs of straining have frequently been observed in the riveting connecting the tank knees to the margin-plate of the double bottom, particularly at the upper part of the knees. Show by a sketch the means usually taken in modern vessels to prevent such straining.

30. What are the advantages and disadvantages of flanging the edges of plates in lieu of fitting angles?

31. Why are wash-plates fitted in deep ballast tanks and in peak tanks?

32. Show by a sketch how a deep tank is made watertight at the deck.

33. What considerations determine the diameters of pillars in a ship? In fitting pillars to beams, where should they be placed in order to develop their greatest efficiency?

34. What is the limit of breadth of ship allowed by Lloyd's Rules for one and for two rows of pillars, respectively?

35. Show by sketches the usual methods of attaching pillars at their heads and heels.

36. In the case of a deck having beams at every frame, show how it may be efficiently supported by a tier of pillars at alternate frames.

37. Sketch an arrangement of wide-spaced pillars, showing how the deck between the pillars is supported. Give details of the attachments to tank top and deck.

38. Certain parts of the shell-plating of a ship are thicker than others. Name these parts, and give reasons for the increased thickness.

39. What are Lloyd's requirements regarding the length of shell-plates and the position of end joints? Sketch a good arrangement of shell butts or joints.

40. Sketch and describe the various plans adopted of fitting shell-plating, indicating specially a system by which the fitting of frame packing pieces is obviated.

41. What are Lloyd's requirements as to the number of rows of rivets in shell landings? Show by rough sketches a single, a double, and a treble-riveted edge lap, indicating the thickness of the plates, width of laps, and diameter and spacing of rivets.

42. What are the advantages and disadvantages of overlapped end joints as compared with butted joints?

43. Lapped joints and butted joints having single straps, show a tendency to open when under stress. What is the reason of this?

44. How would you proceed to stop a leaky end joint of butted type in a strake of bottom plating?

45. It is the practice in many shipyards to scarph overlapped joints where they are crossed by the landings so as to avoid the use of packing pieces. Show by sketches how this is done (*a*) in the case of a joint in an outside strake; (*b*) in the case of a joint in an inside strake.

46. In a riveted joint, discuss the general considerations which govern the diameter and pitch of rivets, and their distance from the edge of the joint.

47. Explain why, as a rule, the ratio of the size of rivets to the thickness of the plates they connect becomes reduced as plates increase in thickness. Show that a limit to this ratio is fixed by practical considerations.

48. Sketch and describe the various heads and points common in ship work, and state where each is used.

49. State the diameters of rivets required by Lloyd's Rules for plates of the following thicknesses—$\frac{1}{4}''$, $\frac{1}{2}''$, $\frac{3}{4}''$, and $1''$, respectively. What should be the pitch of rivets in watertight work?

50. What is the spacing of rivets in frames and beams? Why is the rivet spacing closer in bulkhead frames than in frames elsewhere, and how is the loss of strength thus caused made good?

51. Why are the rivets connecting the framing to the shell-plating of closer pitch in way of deep water ballast tanks and peak ballast tanks than elsewhere.

52. Rivets are usually manufactured of cone shape under the heads. Why?

53. Two plates have to be joined by rivets. Discuss the advantages and disadvantages of—

(*a*) Punching the rivet holes;
(*b*) Drilling the rivet holes.

Describe how the punching and fitting of the plates should be conducted to secure efficient work.

54. Iron rivets are found to have a higher strength efficiency in iron plates than in steel plates. Give a reason for this. Why are iron rivets employed in steel shipbuilding in preference to steel rivets?

55. What is a drift punch? Explain its uses. Show that in certain circumstances the use of a drift punch might lead to bad workmanship.

56. What are the principal functions of a deck stringer? Show by sketches how you would connect and fasten a stringer to the beams, framing, and plating of a ship.

57. How would you proceed in arranging the fastenings in a stringer plate at the butts? A stringer plate is 50 inches wide and $\frac{1}{2}$-inch thick; sketch the riveting in a beam and at a butt, and show that the arrangement is a good one.

58. A steel ship is found on her first voyage at sea to be structurally weak longitudinally. How would you attempt to effectually strengthen the ship with the least additional weight of material?

59. What are deck tieplates? Sketch an arrangement of tieplates on the main deck of a sailing-ship, showing how they are fitted. Explain why they are arranged diagonally as well as fore-and-aft.

60. Decks require to be strengthened in way of large openings. Show by a sketch the usual compensation at the sides and corners of a large upper-deck cargo hatch.

61. Discuss the relative values of teak, pitch pine, and yellow pine, as materials for deck planking.

62. Describe in detail how you would proceed to lay a wood deck

(*a*) Where no steel deck is fitted;

(*b*) Where there is a steel deck.

Show by sketches the connections at a butt joint of the deck planking in each case.

63. What is the Rule height for hatch coamings at upper and at bridge decks. Show by detail sketches how the end and side coamings of an upper-deck hatchway are bound to the deck structure.

64. How are hatch openings protected against inroads from the sea? Sketch an arrangement of beams for supporting the covers of a main cargo hatch in a modern vessel.

65. Describe the mechanical appliances usually installed in cargo steamers for loading and discharging cargo.

66. Sketch a derrick, showing how it is supported at the heel, and detail the arrangements for topping and slewing it.

67. Assuming two winches to be fitted to one hatch, sketch roughly two arrangements by which direct leads to the winch barrels may be obtained.

68. In what circumstances may it be desirable to hinge the derricks on special posts instead of on the masts? Sketch a derrick-post and derrick, and show how the former is connected to the deck.

69. How are steam winches supported (*a*) on an unsheathed steel deck? (*b*) Where a wood deck is laid? What arrangement is made to minimise vibration of the deck due to the working of the winches?

70. Show in section the construction of a lower mast in a large sailing-ship. At what parts is the mast-plating doubled? Why are the doublings fitted?

71. Show by rough sketches how a mast is wedged at a deck, and how it is supported at the heel.

72. Sketch an appliance fitted in modern vessels for tightening up the standing rigging. Show how it is connected to the ship.

73. What is a "spiked bowsprit"? Show how a bowsprit is supported and stayed.

74. What reduction in diameter is allowed in a steamer's masts as compared with those of a sailing-ship? How is a steamer's mast supported at the heel where it is stopped at a lower deck?

75. State the advantages of having a good system of watertight bulkheads in a steamer. What are Lloyd's requirements in respect to watertight bulkheads for a steamer of 300 feet and one of 400 feet length, respectively?

76. Explain why, in ordinary cases, only one transverse watertight bulkhead is fitted in a sailing-ship.

77. How are watertight bulkheads usually built and stiffened? Show by a sketch the arrangement of the plating, spacing of stiffeners, and details of the attachments of the latter in a main transverse watertight bulkhead of a large cargo steamer.

78. In many modern cargo steamers the stiffeners below the deck are fitted vertically only. What are the advantages of the arrangement? Are there any disadvantages?

79. Taking the case of a fore-peak bulkhead, which is deep and narrow, how would you arrange the stiffeners so as to get the greatest efficiency with the least weight of material?

80. A hold stringer consisting of a bulb plate and double angles passes through a watertight bulkhead. Show how you would make the bulkhead watertight around the girder.

81. Referring to the previous question, if the stringer were stopped on each side of the bulkhead, show by a sketch how you would endeavour to maintain the strength at the junction.

82. Sketch and describe a common method of fitting a stem bar in a modern cargo vessel, where there is a flat plate keel.

83. Sketch roughly an iron or a steel sternpost of a cargo steamer, showing how it is connected with and fastened to the keel. Why is the sole piece of the sternpost of a single-screw ship frequently made broad and shallow in way of the aperture?

84. Sketch a Λ bracket arrangement as fitted for supporting the after-end of each propeller shaft in a small twin screw steamer.

What is the principal objection to Λ brackets? Describe a plan by which this is overcome in many modern high speed vessels.

85. Sketch and describe a modern single plate rudder, showing the spacing of the arms and details of the pintles.

86. Commonly, a rudder is supported by the bottom gudgeon of the sternpost. Sketch the arrangement and indicate the means taken to ensure that the rudder shall work without undue friction.

87. Show by rough sketches the usual method of preventing the accidental unshipping of a rudder and of limiting the angle through which the rudder turns.

88. Sketch a rudder coupling, the diameter of rudder stock being 9 inches; indicate the number, position, and diameter of the bolts.

VII.

1. Define stable, unstable, and neutral equilibrium as applied to the case of a vessel floating freely in still water. Illustrate your definitions by suitable sketches.

2. Explain briefly what are the elements in the design of a vessel which control the position of the transverse metacentre. Show that the position of the metacentre is only of relative importance.

3. Describe in detail an "inclining experiment." State what precautions should be taken in order to ensure a reliable result.

An inclining experiment is to be conducted on a certain vessel, her displacement at the time being 2600 tons, and mean draught 8 feet 6 inches. The inclining weight is 6 tons, arranged in two lots of 3 tons, one on each side of the upper deck. The pendulum is 29·5 feet in length. The following is done:—First, one lot of the inclining weights is moved from port to starboard through 40 feet. The deflection of the pendulum is observed and the weight returned to its original position. Then the second lot is moved from starboard to port through the same distance, an observation taken, and the weight, as before, returned. The mean deflection of the pendulum is found to be 1·9 inches. Estimate from the information given the metacentric height of the vessel when in the above condition.

Ans.—8·6 feet.

4. Obtain and prove the expression for the height of the transverse metacentre above the centre of buoyancy.

5. A vessel is 30 feet wide, 15 feet deep, and the centre of gravity of the vessel and its lading is at the middle of the depth of the vessel for all variations in the draught of water. Construct to scale the metacentric diagram.

6. Sketch the metacentric diagrams of any two vessels of different types with which you are acquainted. Give reasons for any differences in the form of the curves.

7. A vessel 140 feet long, and whose body plan half sections are squares, floats with its sides upright, and the centres of all the sections lie in the plane of flotation. The lengths of the sides of the sections, including the end ordinates, are ·8, 3·6, 7·0, 8·0, 6·4, 3·0, and ·7 feet, respectively, the sections being equally spaced. Calculate the distance between the centre of buoyancy and the metacentre.

Ans.—4·51 feet.

8. In the case of what class of vessel must the centre of gravity be below the centre of buoyancy, for equilibrium?

9. A vessel of constant rectangular section, 200 feet long, 40 feet broad, draws 20 feet of water when intact. Two rectangular watertight compartments, 10 feet in width, measuring in from the ship's side, and 10 feet in depth, the bottom of each being 6 feet below the original waterplane, extend each side of amidships for a length of 60 feet.

If the centre of gravity of the vessel is 15 feet above the keel, find the metacentric height—(*a*) When the vessel is intact. (*b*) When the side compartments (assumed empty) are in open communication with the sea.

Ans.—$\begin{cases}(a) & 1·66 \text{ feet.} \\ (b) & ·07 \text{ feet.}\end{cases}$

VIII.

1. Obtain the expression which gives the height of the longitudinal metacentre above the centre of buoyancy.

2. Calculate the longitudinal metacentric height for a log of wood 20 feet long and of square section, the side being 2 feet 6 inches, when floating freely and at rest at a draught of 1 foot 6 inches.

Ans.—21·72 feet.

3. A raft 15 feet long is constructed of two logs of timber 18 inches in diameter and 4 feet between centres, and is planked over with wood 3 inches thick, forming a platform 12 feet by 5 feet. All the wood is of the same density, and the raft floats in sea water with the logs half immersed. Find the longitudinal metacentric height and the moment to alter trim 1 inch. (*See* note to question No. 6 on opposite page).

Ans.—31·33 feet, 295 foot lbs.

4. A small weight is placed on board a vessel in any longitudinal position. Explain how you would proceed to find the changes in the draughts forward and aft.

5. A cargo vessel is 48 feet broad on the load waterline. Given that the tons per inch of immersion is 35, calculate approximately the moment to alter trim 1 inch.

Ans.—788 foot tons.

6. A vessel of circular section, 80 feet long and 20 feet diameter,

floats with the axis in the waterplane. Calculate the trimming effect of shifting a weight of 15 tons from mid length to a point 10 feet from the after end. The centre of gravity is 2 feet below the waterplane.

NOTE.—The centre of buoyancy may be fixed in relation to the transverse metacentre.

Ans.—$18\frac{1}{4}$ inches by the stern.

7. Describe any simple method of providing commanding officers with such information concerning their own vessels as will enable them to deal quickly and correctly with trim problems.

8. The trim line of a certain vessel corresponding to the load draught makes an angle of 42 degrees with the horizontal.

Plot the trim line, and from it obtain the change of trim due to shifting 50 tons through 100 feet aft. The displacement is 8000 tons.

Ans.—$6\frac{3}{4}$ inches by stern.

IX.

1. Given that the righting levers of a vessel at angle of 15°, 30°, 45°, 60°, 75°, and 90° respectively, are ·74, 1·53, 2·1, 2·18, 1·65, ·9 feet, construct the curve of stability, and indicate the maximum righting lever and the angle at which it occurs. The metacentric height is 2·62 feet.

Ans.—2·22 feet, 55°.

2. What are the features in a vessel affecting the range of the curve of stability? Show that a great metacentric height may be associated with a short range.

3. Draw in one figure the curves of stability of two dissimilar types of vessels with which you are acquainted, and give the reasons for any differences which exist in the nature of the curves you show.

4. Some merchant vessels will not remain in an upright position when unloaded. Explain the reason of this. Draw the curve of stability of a vessel when in the condition named.

5. A sailing-ship is heeled by the pressure of the wind on the sails. Assuming her to be at a steady angle of heel, show in a sketch the forces acting, and state the relation of the moments of these forces to each other.

6. A vessel of box form 200 feet long, 40 feet broad, 20 feet deep, floats in sea water at a level draught of 15 feet. Assuming a metacentric height of 2 feet, construct the curve of statical stability.

7. Draw cross curves of stability for a vessel of square section at angles of 45° and 90° respectively, assuming the centre of gravity to be 1 foot below the centre of the section.

8. Having given the value of the righting arm of a vessel at a certain inclination when at her load displacement, the position of the centre of

gravity being known, show how you would find it at the same inclination when at a reduced displacement, due to the consumption of the bunker coal.

X.

1. What is meant by the phrase "Period of a single roll"? It is desired to obtain the period of roll of a cargo vessel when in a given condition. How could this be ascertained experimentally?

2. What is the transverse radius of gyration? How is it obtained?

3. What effect has the variation of the metacentric height upon the value of the rolling period?

In a given vessel what is the difference between the rolling periods corresponding to a metacentric height of 2 feet and 4 feet, respectively, assuming the transverse radius of gyration to be 18 feet and the same in both cases?

4. Explain why waves that are relatively high in relation to their lengths are more powerful in causing vessels to roll heavily than waves that are relatively low.

5. Describe a simple experimental method of proving that a vessel when broadside on to a series of regular waves always tends to place her masts parallel to the normal to the wave slope.

6. State the length and period as actually observed of large Atlantic storm waves ordinarily met with.

What, by inference, should the natural roll period approximate to in the case of a vessel intended to trade in the Atlantic, in order to obtain the best results.

7. Mention an appliance that has been recently employed to minimise the rolling motions of vessels at sea. Show by quoting the results in any actual case, what success has attended the new system.

XI.

1. A vessel is to be loaded with a general cargo of which the weights and other particulars are known. How would you proceed to find the vertical position of the centre of gravity? Assuming the displacement scale and the diagram of metacentres to be available, how would you determine the metacentric height with the proposed system of loading?

2. The stability curve at the load draught in a certain vessel is of considerable area and range, but shows upsetting levers at angles near the origin. How do you account for this? In the case of such a vessel, what considerations would influence you in fixing upon a value of metacentric height with which to start a voyage.

3. What is the chief objection to deck cargoes? Show that a deck cargo of timber, if well stowed and secured, may improve a vessel's sea qualities.

4. What is the angle of repose for wheat?

In certain circumstances, grain carried in the hold of a steamer is found to slide at a much smaller inclination than its normal angle of repose. Describe these circumstances, and explain the causes to which they give rise which lead to the reduction in the sliding angle.

5. What proportion of the full deadweight should a modern steamer carry in making an Atlantic voyage in ballast? To what extent should the propeller be immersed? What has frequently happened when a voyage has been made in too light a trim, and rough weather has been encountered?

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