STRUCTURAL DESIGN OF WARSHIPS

HOVGAARD

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BY

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X

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CONTENTS.

CHAPTER I.

GENERAL.

CHAPTER II.

LONGITUDINAL STRENGTH.

CHAPTER III.

TRANSVERSE STRENGTH.

CHAPTER IV.

STRENGTH OF INDIVIDUAL GIRDERS.

CHAPTER V.

STRENGTH OF A RECTANGULAR PLATE UNDER FLUID PRESSURE.

CHAPTER VI.

STRENGTH OF COLUMNS AND PLATING UNDER COMPRESSION.

CHAPTER VII.

RIVETS AND RIVETED CONNECTIONS.

CHAPTER VIII.

CALCULATIONS FOR RIVETED JOINTS.

CHAPTER IX.

WATERTIGHTNESS AND OILTIGHTNESS.

CHAPTER X.

OUTSIDE PLATING.

CHAPTER XI.

WOOD SHEATHING AND THE COMPOSITE SYSTEM OF CONSTRUCTION.

CHAPTER XII.

FRAMES.

CHAPTER XIII.

STEM AND STERNPOST. FRAMING AT THE ENDS OF THE SHIP.

CHAPTER XIV.

INNER BOTTOM PLATING.

CHAPTER XV.

THE DECKS.

CHAPTER XVI.

BULKHEADS.

ABBREVIATIONS.

Am. Soc. Civ. Eng.: American Society of Civil Engineers.
Am. Railw. Eng. Ass.: American Railway Engineering Association.
Br. Inst. Mech. Eng.: British Institution of Mechanical Engineers.
Inst. Nav. Arch.: Institution of Naval Architects, London.
Schiffb. Ges.: Schiffbautechnische Gesellschaft.
Ver. Deutsch. Ing.: Vereines Deutscher Ingenieure.
Am. Soc. Nav. Arch. Mar. Eng.: Society of Naval Architects and Marine Engineers, New York.
Jap. Soc. Nav. Arch.: Japanese Society of Naval Architects.
Jap. Soc. Mech. Eng.: Society of Mechanical Engineers, Japan.

POSITION OF PLATES AND FOLDING TABLES.

PREFACE.

THIS work is based on a series of lectures prepared for a course in Naval Construction which was established by the Massachusetts Institute of Technology in 1901 for officers of the United States Navy detailed to take this course in preparation for the duties as Naval Constructors. The lectures have been developed during the last thirteen years and, having now taken a fairly permanent form, it was thought desirable to print them, partly for reasons of instruction, and partly in the hope that they might be useful to naval constructors and designers of warships in general.

The course in Naval Construction extends over three years and consists of lectures and design work. The general plan of the lectures will be here briefly outlined in order to show the place which the present work occupies in the course.

First year: A historical account of the development of warships, beginning with the introduction of steam-power, iron, and armor.

Preliminary design, comprising a discussion of the different types of warships, determination of the principal elements of design, construction of lines, stability, seaworthiness, general distribution of weights, weight calculations, watertight subdivision, and vibrations of vessels.

Second year: Structural design of warships, comprising materials used in hull construction, strength and strength calculations, riveted joints, and main structural features. Preliminary design and installation of boilers, engines, and propellers, as far as these problems concern the naval architect. Coaling and coal stowage. Liquid fuel. Rudders and steering gear.

Third year: Drainage. Ventilation and heating. Anchors and anchor gear. Boats and boat-handling gear. Disposition and installation of artillery. Ammunition, ammunition transport and stowage, design of ammunition rooms. Torpedo installations. Effects of gunfire. Resistance of armor. Principles of the design of armor protection. Protection against submarine attack. Conning towers.

The present volume covers somewhat more than the first half of the second year's lectures. It presupposes a general knowledge of the theory of Naval Architecture as well as such familiarity with warships as is ordinarily possessed by young naval officers.

The work of collecting the material and preparing it in a form that would be useful to the students presented considerable difficulty. The existing sources of information are very scattered, consisting in text-books, handbooks, essays and articles, specifications, drawings, rules, and regulations. The text-books contain much valuable information, but, excepting some of the French books, they deal with so many other aspects of warship design that the discussion of structural features is necessarily brief and chiefly of a descriptive nature. It is the object here to present a more comprehensive discussion than hitherto given of this special branch of warship design, exhibiting in particular its relation to fundamental principles and conditions. On account of the vastness of the subject it has been found necessary to restrict the scope of the work to the main features of the hull proper. The descriptive part is made subservient to the discussion of principles; but where descriptions and sketches of structural parts are given, they represent, as far as possible, recent practice. Where important points of difference exist in the modes of construction in different navies they are critically compared.

In the study of current practice it was often found difficult to discover the reasons why certain features were adopted and in some cases why they differed in different navies. The explanation is in general that once a certain mode of construction has been introduced in one of the leading navies and found satisfactory it becomes a standard. Gradually the reason for its adoption may be forgotten, and the construction is used as a matter of routine. It may even happen that the conditions which called forth the construction change or cease to exist, and that it survives simply because there is a vague feeling that something will go wrong if it is changed or abolished. A study of such questions cannot fail to be fruitful, because it leads naturally to suggestions for improvements. Moreover, in a work of this nature, it is desirable fully to state and explain the reasons for the adoption of the various structural features, inasmuch as such information will be of value, not only to the inexperienced student of the present time, but also as a matter of record for the future.

Since the lectures, on which the work is based, were written primarily for American naval students, the practice of the United States Navy is given the most prominent place. This is desirable also from another point of view. In the United States Navy the construction of armored ships commenced barely twenty-five years ago, in which short space of time it has grown to become one of the leading navies. The development has been characterised by an intense striving for improvement and perfection. Unhampered by tradition and by too close financial restrictions, the best practice of other navies has been adopted, and many new and original features have been successfully added. The result is that the American warships of to-day are, in general features as well as in details, among the best designed in the world. The descriptive part is, therefore, based chiefly on American practice, but the volume contains also numerous references to the practice in the British, French, and Danish Navies.

The book differs from most other similar works in its more extensive application of mathematics to the problems occurring in warship construction. Mathematical treatment is, in fact, proposed for several problems where, so far, routine has been followed, or where, at the best, experience has been used as a basis for a tentative method. The enormous increase in size and cost of warships and in propulsive and artilleristic power, the improvement in the quality of the materials, and the increasing claims to a saving in weight, and hence to a more perfect adaptation of each structural member to its work, render it necessary to adopt more refined and accurate methods of design than used heretofore. It must, however, be clearly understood that when mathematical treatment is here recommended, it is only as an instrument for a more intelligent and accurate application of experience. Experience, whether gained in actual service or by experiments, must always be the foundation on which our decision rests ; without it the formulas are useless and may be even dangerous to follow. The practical application of the formulas and of the theoretical principles is illustrated by numerous examples.

Although the work is prepared chiefly as a text-book for students of Naval Construction, it goes considerably beyond the limits of ordinary text-books which merely record the established theory and reflect current practice. In the theoretical treatment of the subject much is contained that is believed to be novel, and in the chapters on practical application the author has ventured to advance several suggestions for new structural features. Where methods of design or construction are proposed which are yet untried, this fact is always clearly stated.

Due to the analytical mode of treatment followed throughout this work, where each part of the subject is dealt with independently and in its fundamental aspects, it was found impossible to avoid repetitions. This feature, however, enables the reader to study each chapter separately without too many references to other parts of the work.

I desire to express my indebtedness to Rear-Admiral R. M. Watt, Chief Constructor of the United States Navy, for the great assistance he has rendered in the study of structural details at the time when these

lectures were in the early stages of preparation, and for his permission to publish the drawings and other information concerning the United States Navy contained in this volume. My thanks are due also to Captain J. C. Tuxen, Chief Constructor of the Danish Navy, who has similarly placed at my disposal drawings and other information concerning the Danish Navy.

Professor Henry H. W. Keith, of the Massachusetts Institute of Technology, has rendered most valuable assistance in the preparation of drawings and diagrams as well as in the numerical work ; and Assistant Naval Constructor R. D. Weyerbacher, Assistant Naval Constructor T. B. Richey, and Mr Wai G. Loo, students of Naval Construction in the Massachusetts Institute of Technology, have taken part in the numerical and graphical work connected with some of the problems. Copious references to the various sources of information are made in the text and in the footnotes.

WILLIAM HOVGAARD.

BOSTON, MASS., U.S.A.,
January 1915.

STRUCTURAL DESIGN OF WARSHIPS.

CHAPTER I.

GENERAL.

1. MATERIALS USED IN HULL CONSTRUCTION.

1. **Steel and Iron.**—The material used in the hull structure of a warship is almost exclusively steel, mainly of the quality known as "mild steel" or "medium steel," in Germany called "Flusseisen." Steel of other qualities is used where great strength combined with lightness is specially required and where the material is exposed to great dynamic actions—further, for rivets, bolts, castings, forgings, and for armor.

Iron is used for certain forgings, but extra-soft steel is now often employed as a substitute for wrought iron.

2. **Ultimate Strength and Ductility.**—The quality of the material is ascertained by inspection and by certain prescribed tests, of which the most important are those for ultimate tensile strength and ductility (elongation). Table I. gives the principal requirements for the United States Navy and Tables II. and III. give some figures for the British and the German Navies. The tensile strength of ordinary mild steel (medium steel) we shall in the following take to be 63,000 lb. per sq. in. or 28 ts. per sq. in. as an average value. The shearing strength of rivets,

TABLE I.—PRINCIPAL REQUIREMENTS FOR STEEL AND IRON FOR U.S. NAVAL VESSELS.

Quality of Material.	Use.	Minimum Elastic Limit or Yield Point. lb. per sq. in.	Minimum Tensile Strength. lb. per sq. in.	Elongation.	
				Per cent.	Length inches.
Medium steel. Open-hearth carbon.	Plates and shapes for hulls and hull construction.	...	60,000	25	8*
High-tensile steel. Open - hearth carbon, nickel, or silicon.	Plates and shapes for hulls and hull construction.	...	80,000	20	8*
Special-treatment.	Protective deck plating.	105,000	120,000	17	2†
Medium steel. Open-hearth carbon.	Rods and bars for rivets, bolts, stanchions, davits, etc.	...	≤ 1½-in. diam. 58,000	28	8
			> 1½-in. diam. 60,000	30	2
High-tensile steel. Open-hearth carbon, silicon, or nickel.	Rods and bars for rivets, bolts, stanchions, davits, etc.	...	≤ 1½-in. diam. 75,000	23	8
			> 1½-in. diam. 75,000	25	2
Steel forgings. Class A. Open-hearth nickel or carbon.	Forgings exposed to dynamic actions, as in gun-mounts.	50,000	80,000	25	2
Steel forgings. Class B. Open-hearth carbon.	Stems and sternposts, rudder-stocks, etc.	30,000	60,000	30	2
Steel castings. Class A.	Hawse-pipes, turret-tracks, and other parts subject to crushing and dynamic actions.	35,000	80,000	17	2
Steel castings. Class B.	Stems, sternposts, rudder-frames, struts, and parts subject to tension and bending.	30,000	60,000 to 80,000	22	2
Wrought iron.	Miscellaneous forgings.	...	48,000	26	8
Extra-soft steel used as a substitute for wrought iron.		One-half the ultimate strength	45,000 to 55,000	28	8

* For thicknesses less than ¼ in. and over 1 in. the elongation is measured in smaller lengths.

† Special ballistic test. Attack at an angle of 9° by uncapped 6-in. or 8-in. projectiles. Supporting frames to leave not less than 36 inches between nearest edges.

of whatever grade of iron or steel, is about four-fifths of the tensile strength. For mild steel rivets in plates of mild steel we shall here take the shearing strength 50,000 lb. per sq. in., but when used in plates of high-tensile steel this value is reduced to about 43,000 lb. per sq. in. The tensile strength of rivet iron as used in merchant vessels is about 54,000 lb. per sq. in. and the shearing strength of this material, when used in iron plates, is about 42,500 lb. per sq. in., but when used in steel plates it is reduced to about 37,000 lb. per sq. in.*

TABLE II.—REQUIREMENTS FOR STEEL FOR BRITISH NAVAL VESSELS.

Kind of Steel.	Ultimate Tensile Strength. lb. per sq. in.	Length of Test Piece. Inches.	Elongation. Per cent.
Mild steel	58,250–67,200	8	20
Rivet steel	58,250–67,200	8	25
High-tensile steel for cruisers †	76,150–85,100	8	20
High-tensile steel for destroyers	82,900–96,300	8	15–20

TABLE III.—STEEL FOR SHIPBUILDING USED IN THE GERMAN NAVY.‡

Quality.	Yield Point. lb. per sq. in.			Ultimate Tensile Strength. lb. per sq. in.
	Maximum.	Average.	Minimum.	
Shipbuilding steel I.	49,100	37,000	27,300	48,300–58,300
Shipbuilding steel II.	52,000	39,700	30,900	58,300–66,800
Shipbuilding steel III.	...	...	...	74,500
Rivet steel	...	35,800	...	55,300

Figs. 1 and 2 show curves of stress and strain for steel and iron of various qualities.

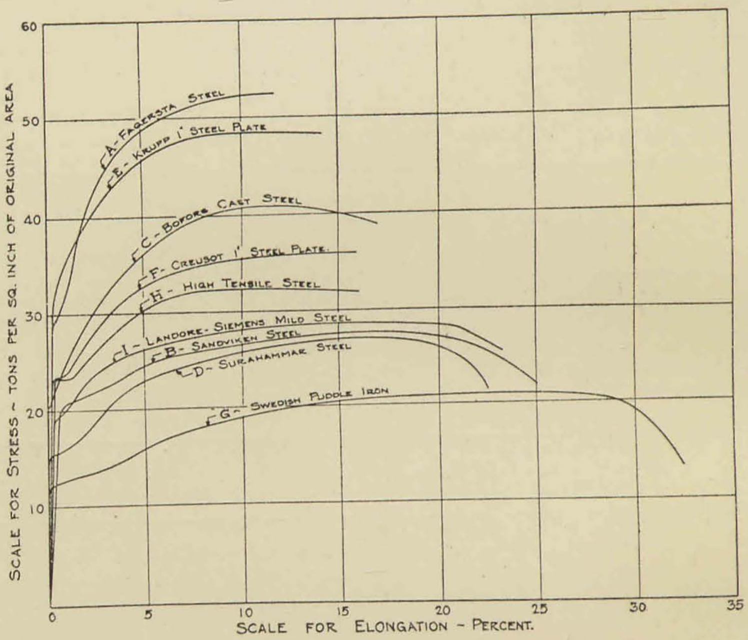
3. **The Elastic Limit.**—The true or primitive elastic limit, *i.e.* the point where the proportionality of stress and strain ceases, is difficult or impossible to determine with certainty in practice. The so-called "breaking-down point" or "yield point," where the material begins suddenly to

* A. C. Holms, *Practical Shipbuilding*, London, 1908, p. 264.

† Elastic limit = 44,800 lb. per sq. in.

‡ Pietzker, *Festigkeit der Schiffe*, Berlin, 1911, pp. 50, 55. The data given for Qualities I. and II. are the result of about 200 tests. The yield point here referred to is the stress at which the permanent set exceeds ·2 per cent. of the length of the test piece; it corresponds to the "practical elastic limit" ("*Elastizitätsgrenze im praktischen Sinne*").

flow (the German *Fliessgrenze*), is better defined, being generally easy to recognise. It lies in shipbuilding steel very near the true elastic limit, and may be regarded in practical work as a limit which should not ordinarily be reached. In the following we shall not, generally, try to distinguish between the yield point and the elastic limit, but shall use



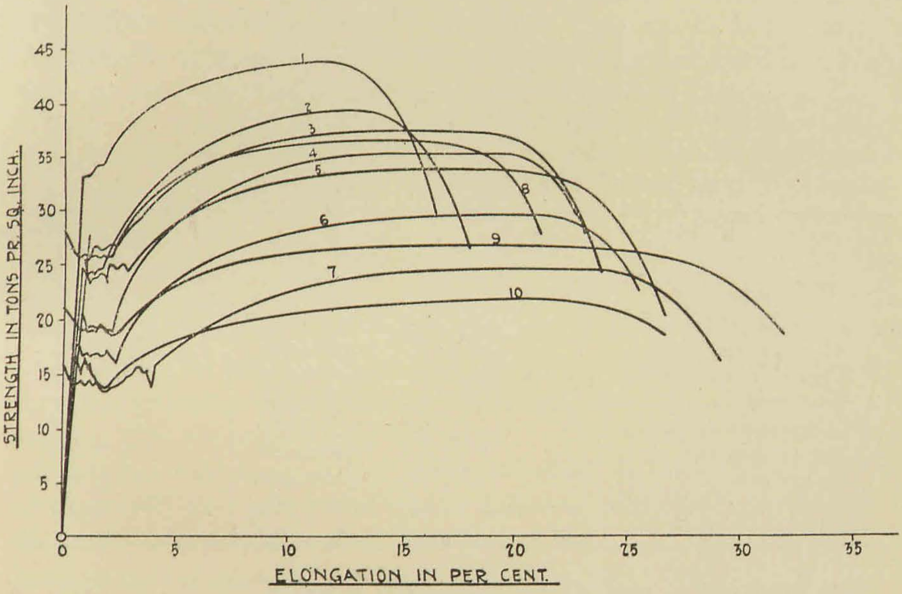
Letter	Name	Length	B'rdth	Thk. or Diam.	Coeff. of Contract	Use
A	FAGERSTA STEEL [SWEDISH]	200 m/m	37·9 m/m	6·5 m/m	·677	BEAK FOR DREDGE SHOVEL
B	SANDVIKEN STEEL [SWEDISH]	200 "	ROUND.	17·4 "	·46	SHAFTING AND STEERING LEAD
C	BOFORS CAST STEEL [SWEDISH]	100 "	"	18·0 "	—	
D	SURAHAMMAR STEEL [SWEDISH]	200 "	SQUARE	—	—	BOILER PLATES
E	KRUPP 1" STEEL PLATE	200 "	ROUND	18·4 "	·541	GUNSHIELD
F	CREUSOT 1" STEEL PLATE	200 "	"	18·1 "	·533	"
G	SWEDISH PUDDLE IRON	200 "	"	—	—	FORGINGS
H	HIGH TENSILE STEEL	20 In.	RECT.	¼ In.		PLATING FOR DESTROYER *
I	LANDORE-SIEMENS MILD STEEL	10 "	—	—	—	

* PROF. BILES – I.N.A. 1907, p 79.

FIG. 1.—Stress-Strain Diagrams for Steel and Iron.

these terms synonymously. For mild steel the yield point is at somewhat less than two-thirds of the breaking stress; for high-tensile steel, as used in shipbuilding, it is from two-thirds to three-quarters of that stress; for "special-treatment" steel for protective deck plating it may be as high as five-sixths. The indefiniteness which may sometimes appear in the position of the yield point is probably due to imperfection in the mode of measurement. Where refined methods of recording the strain are used, the yield point is sharply defined, as seen in fig. 2, which gives the results of some of Professor Dalby's experiments and of certain German tests on structural steel.

4. **The Modulus of Elasticity.**—This modulus, E, is a coefficient or abstract figure which measures the ratio between stress and strain within



ITEM	QUALITY	CHEMICAL COMPOSITION	ELASTIC LIMIT IN. TS. PR. SQ. IN.	ULTM. TENSILE STRENGTH IN. TS. PR. SQ. IN.	ELONGATION IN PER CENT	CONTRACTION IN PER CENT	NOTE
1	CHROME NICKEL STEEL	4% (Cr + Ni)	33·7	44·0	15·5	66	INTERNATIONALE BAUFACHAUSSTELLUNG IN LEIPZIG - 1913 (SHIFFBAU-1913 P. 164)
2	CHROME NICKEL STEEL FOR STRUCTURAL WORK	3% (Cr + Ni)	25·5	39·6	17·0	63	
3	NICKEL STEEL	5% Ni.	27·2	37·5	13·3	69	
4	SPECIAL STEEL		20·4	36·0	22·2	57	
5	NICKEL STEEL	3% Ni.	25·0	33·7	26·5	72	
6	CARBON STEEL	·2% C.	18·0	29·5	24·9	58	
7	MILD STEEL (FLUSSEISEN)		16·4	25·0	29·0	71	
8	STEEL	·4 C	28·0	36·0	21·0		PROF. W. E. DALBY. INST. NAV. ARCH. - 1912
9	STEEL	·12 C.	21·0	26·6	32·0		
10	FARNLEY IRON		16·0	22·0	26·5		

FIG. 2.

the true limit of elasticity. Graphically, it is represented by the steepness of the stress and strain curve within the elastic limit. Its numerical value depends on the units in which the stress is expressed. It is practically the same in all classes of steel, and is usually given as about from 13,000 to 14,000, the units being tons and square inches. In the numerical examples of this work, E will be taken equal to 13,500 (ts. per sq. in.) or 30,000,000 (lb. per sq. in.). Judging from the observed deflections of ships, the modulus of elasticity appears to be smaller in complex riveted structures than in test pieces. Thus, in the experiments with the British destroyer *Wolf*,* the modulus when the boat was in sagging condition was found to vary from about 10,000 at great stresses to 12,000 at small stresses. The cause of these small values will be discussed in a later chapter. Table IV. gives the values of E as employed in the computation of the column curves, figs. 46 and 47.

TABLE IV.—MODULUS OF ELASTICITY USED IN CONSTRUCTION OF THE COLUMN CURVES, FIGS. 46 AND 47.

	E. lb. sq. in.	E. ts. sq. in.
Wrought iron	28,000,000	12,500
All classes of steel	30,000,000	13,400
Compositions (brasses and bronzes)	13,000,000	5,800
White pine	1,400,000	625
Douglas fir	1,700,000	760
Spruce and ash	1,600,000	715
Yellow pine	2,000,000	895
Oak	2,000,000	895

The modulus of elasticity for shearing is called the "coefficient of rigidity." It measures the ratio between the shearing stress and the distortion, and is for steel about 5300 (ts. per sq. in.).

5. **High-Tensile Steel.**—The stress-strain curve for high-tensile steel coincides with the curve for mild steel up to the yield point of this latter. Here the curve for mild steel falls off, while the curve for high-tensile steel continues to a much higher stress before breakdown takes place. This is illustrated by the curves in figs. 1 and 2. High-tensile steel is, therefore, in tension very superior to mild steel and can be used with reduced scantlings. This, however, entails a reduction in stiffness, although high-tensile steel is intrinsically more rigid than mild steel, and the deficiency must be compensated for by providing a more elaborate system of stiffening. High-tensile steel is now used everywhere in the principal strength members of light, high-speed vessels, and of recent years it has been introduced more and more also in the construction of battleships and battle-cruisers.

6. **Wood.**—Wood is employed in the hull construction of warships as follows :—Pine, fir, and teak in decks ; teak in backing behind armor ; teak, elm, and pine in sheathing of the bottom, sometimes

* J. H. Biles, *Inst. Nav. Arch.*, 1905, i. p. 103.

—as in composite vessels—partly or entirely replacing the outer shell plating.

Pine should be well seasoned before it is taken into use, as it is otherwise liable to warp and shrink. When in contact with iron and water, it will absorb rust, and is then liable to decay, wherefore the ends of the planks should not abut on the metal.

Teak possesses almost the same strength as oak ; it is very stiff and does not readily split. It shrinks very little, and does not warp when exposed to alternate moisture and dryness, even in tropical climates. It will stand a great amount of heavy wear and tear, and contains a resinous oil which prevents rusting of the steel or iron with which it is in contact.

Java Teak.—On account of the scarcity and high cost of teak proper, which comes only from Burmah and Siam, a new kind of wood, called "Java Teak," has recently appeared on the market. Java teak, in Java called *Djati*, is botanically closely related to Burmah and Siam teak. Its properties are similar to those of ordinary teak ; its density is more variable but averages about the same. The fibers of Java teak are not so straight as in other kinds of teak, and the average length of the logs is only about half that of ordinary teak logs. Java teak is, therefore, not so well suited for deck planks, but has been used for backing behind armor. It is more rich in chalk and, hence, not so easy to work.

Oak was formerly, in the days of wooden ships, employed extensively in the construction of the hull, both on account of its strength and because it is capable of resisting the decaying action of sea-water. It is, however, unsuitable for use in iron and steel ships, as it contains an acid which attacks the metal.

TABLE V.

	Ultimate Compressive Stress in lb. per sq. in.
White pine *	5400
Douglas fir *	5700
Spruce or ash *	7200
Yellow pine *	8000
Oak *	8500
Java teak †	7800

* Prof. J. B. Johnson, *Materials of Construction*, p. 671. Tests made for U.S. Dept. of Agriculture.

† Danish State Institute for the Testing of Materials.

The tensile strength of wood, in a complex structure, is greatly inferior to that of steel, on account of the impossibility of obtaining an efficient connection between the individual members. Tabling and dowels can only give the scarphs a fraction of the full strength of the timbers which they connect, and metal bolts are too hard to work well with wood. The deeper cause of this imperfection of wood is its fibrous structure, which, on account of the relatively loose connection between the different fibres, entails, first, a small resistance to shearing in direction of the fibres—only about one-twentieth of the tensile strength along the fibres—and, second, a tendency to split, due to a lack of crosswise strength.

The ultimate compressive strength in direction of the fibres of different kinds of wood is given in Table V. These values hold good for wood containing 12 per cent. of moisture. Green timber may be expected to show about half the strength indicated in the table.

7. **Copper and Copper Alloys.**—Pure copper is used for protecting the bottom of sheathed and composite vessels against fouling, but in the hull structure it occurs only as an alloy with other metals.

"Bronze" is an alloy of copper and tin. The percentage of tin varies from about 8 to 10 in "gun-metal" to 25 in the metal used for making bells. There are many varieties of bronze between these extremes, differing also by small additions of other elements.

"Phosphor-Bronze" in its pure form contains 90 per cent. of copper and 10 per cent. of phosphor-tin; but similar alloys, containing also a small percentage of other components, go under the name of phosphor-bronze. Pure phosphor-bronze is very hard and tough, and has a tensile strength of about 35,000 lb. per sq. in.; it is often used for stems, stern-posts, and other outboard castings of sheathed and composite vessels.

"Brass."—The term brass applies to alloys of copper and zinc. The percentage of zinc varies from 10 to 50. The tensile strength of brass in castings is ordinarily from 12 to 14 ts. per sq. in., but when the percentage of zinc exceeds about 45, the alloy changes its character and the strength falls off to about 9 ts. per sq. in. When rolled and annealed, the strength and ductility of brass may be the same as of steel. Brass is used for rivets in structural work required to be non-magnetic.

"Muntz-Metal" is a brass consisting of 60 per cent. of copper and 40 per cent. of zinc. When rolled and annealed it has the properties of steel, being both malleable and strong. It shows in that condition a tensile strength of from 50,000 to 65,000 lb. per sq. in., and an elongation of up to about 30 per cent. Muntz-metal changes its nature in sea-water, but is used for coppering the bottom of sheathed and composite vessels in the British Navy.

"Naval-Brass" consists of 62 per cent. of copper, from 36 to 37 per cent. of zinc, and from 1 to $1\frac{1}{2}$ per cent. of tin. It is, in fact, Muntz-metal with a small addition of tin. The tin enables this alloy to withstand the deteriorating effect of sea-water, and naval-brass is, therefore, suitable for under-water work, such as for fastening the planks of sheathed and composite vessels. When naval-brass is rolled in rods for use in bolts or other important parts subject to stress, it shall, according to the requirements of the United States Navy, show a tensile strength of not less than 60,000 lb. per sq. in., an elastic limit of at least one-half the ultimate strength, and an elongation of not less than 25 per cent. in two inches.

"Manganese-Bronze" is composed of 56 per cent. of copper, about 41 per cent. of zinc, and small quantities of iron, tin, aluminium, and manganese. The term bronze as applied to this alloy is a misnomer, as it is really a kind of brass. Manganese-bronze is used, like phosphor-bronze, for the outboard castings of sheathed and composite vessels. The requirements of the United States Navy for this metal are about the same as for naval-brass.

8. **Zinc.**—This metal is employed in its pure form in a process called "galvanising," which has for object to protect steel or iron against corrosion, and consists in coating these metals with a thin layer of zinc by immersion in a bath of molten zinc. Galvanising should be applied to all plates and shapes of thickness one-eighth of an inch or less, as found in torpedo-vessels and in cabin bulkheads; it may also be applied with advantage to small forgings. Plates and shapes should be galvanised before assembling. The increase in weight by this process is about one-seventh of a pound per square foot.

9. **Aluminium.**—Aluminium has been used, mostly as an alloy with copper, in certain torpedo-boats, yachts, and river-boats, on account of its great lightness. In a torpedo-boat built of an aluminium alloy (6 per cent. copper), by Yarrow, the metal had a tensile strength of from 14 to 16 ts. per sq. in., and an elongation of 3 to 4 per cent. in ten inches.* It appears that the corrosion of this metal was excessive, especially between wind and water and when exposed to galvanic action; but it is possible that other aluminium alloys may give better results. A saving in weight of one-third was effected as compared with steel, but the price of the metal was very high. Recently an alloy "Duralumin" has been produced, which possesses most of the good characteristics of steel, while being only about one-third as heavy as that material, but it is about twelve times dearer than steel.†

* A. F. Yarrow, *Inst. Nav. Arch.*, 1895, p. 274.

† T. G. John, "Shipbuilding Practice of the Present and Future," *Inst. Nav. Arch.*, July 1914.

2. PRINCIPAL ELEMENTS OF HULL STRUCTURE.

Throughout this work the discussion is based chiefly on the structural arrangements of the largest class of sea-going battleships and battle-cruisers. The construction of these ships is considered as the standard, from which that of most other types may be derived. The structural arrangements of smaller warships are, therefore, described and discussed only in so far as they deviate essentially from those of the large vessels.

The hull of a ship is composed of three elements—plates, bars, and rivets. The plates do not require any particular description, being simply sheets of metal of uniform thickness. The bars are used in different

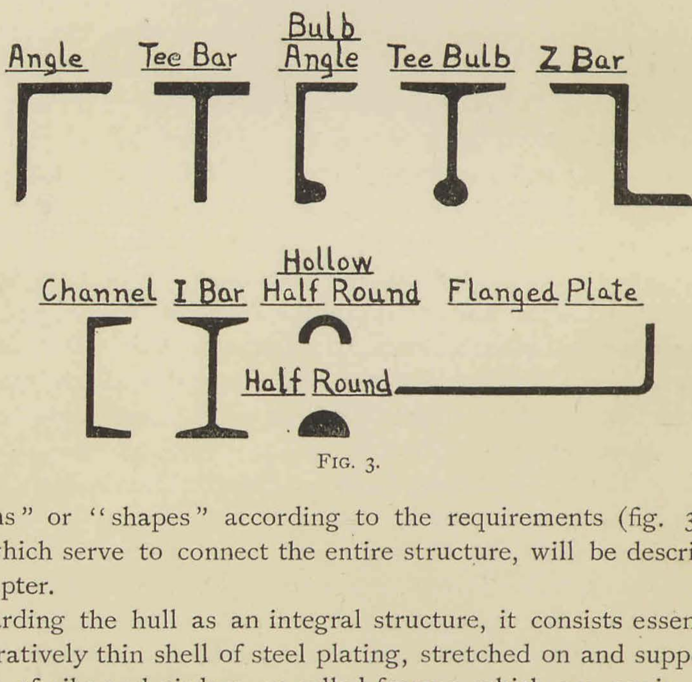


FIG. 3.

"sections" or "shapes" according to the requirements (fig. 3). The rivets, which serve to connect the entire structure, will be described in a later chapter.

Regarding the hull as an integral structure, it consists essentially of a comparatively thin shell of steel plating, stretched on and supported by a system of ribs and girders, so-called frames, which are again stiffened and supported by a network of horizontal decks and vertical bulkheads.

The shell is the primitive and most essential part of the hull, since, by forming a watertight outer envelope, it enables the ship to float on the water. While this is the primary function of the shell, it constitutes, moreover, one of the principal strength members.

The framing consists partly in transverse, partly in longitudinal members. In very small vessels, such as boats, the frames, eventually assisted by beams, so-called thwarts, form the sole support of the outer shell; but in large vessels the frames are inadequate for this purpose, and must be supported by decks and bulkheads in order to secure general strength and rigidity. This latter quality is very important in a ship, since large deflections are injurious both to riveting and calking; it is amply provided for in most warships by the numerous bulkheads. In merchant ships of the cargo-carrying type the number of decks and bulkheads is reduced to a minimum, especially in those that carry cargo in bulk, whence strength and stiffness must here be secured in a greater measure by means of the peripheral structure. There is in this respect a marked difference between warships and merchant ships.

Decks and bulkheads also subdivide the ship, and thus ensure safety and a proper utilisation of the internal space. A discussion of the principles on which the subdivision of warships is based lies outside the scope of these lectures. We shall here merely state that in a large warship there are a number of complete and partial decks, one or two of which may be armored. A limited number of main transverse bulkheads divide the ship into large sections, which are further subdivided by longitudinal bulkheads and by numerous minor transverse bulkheads. At a short distance from the outer shell is fitted an inner bottom, extending practically over the entire immersed portion of the ship. The two shells are held together, supported, and stiffened by the transverse and longitudinal frames, many of which, being watertight, subdivide the double-bottom space into a great number of small watertight compartments. The double bottom is often continued as a cofferdam to a certain height above the water-line. At the extreme ends, the ship is framed by heavy steel castings, the stem and the sternpost, calculated to resist the great forces to which these parts of the ship are exposed.

3. GENERAL REMARKS ON STRUCTURAL STRENGTH AND STRENGTH CALCULATIONS.

The structural design of a warship is based essentially on the requirements to strength, which is in most cases the principal quality to be considered, and which can never be entirely neglected.

1. **Comparison between Merchant Ships and Warships.**—On account of the indefiniteness of the forces to which all ships are subject, calculations for determining the strength of the hull as a whole, as well as the strength of its different parts, must be generally of an empirical nature. Exact calculations are, in fact, only applicable in a few cases.

In the construction of merchant vessels this difficulty has been overcome by the use of rules and tables, by means of which the shipbuilder is able to determine the general construction and the scantlings of the hull as soon as he has settled the type, size, and principal dimensions of the ship. These rules and tables, which are very detailed and complete, are framed by the so-called "Classification Societies" on the basis of the enormous mass of experience and information which these institutions have at their disposal. The object of these societies, *Lloyd*, *Veritas*, *British Corporation*, *Germanischer Lloyd*, and others, is to establish and maintain certified standards of construction. When a ship is built according to their rules, the shipowners, the shippers, and the underwriters have a guarantee that the construction is satisfactory. Since practically all merchant vessels are built to such rules, which leave little room for independent decision, the problem of the design of structural features is in merchant shipbuilding largely taken out of the hands of the shipbuilder and is dealt with chiefly by the experts of the Classification Societies. This does not prevent other naval architects from proposing new departures in ship construction, but, ordinarily, they must be approved by the Classification Societies before they can be adopted. In fact, this procedure has been followed more frequently during recent years than was formerly the case, after the Societies have adopted a more liberal and progressive policy.

In warship design no rules or tables exist to guide the designer, except such as may from time to time be adopted by any navy for certain details. In general, the rapid and great changes that take place in warship design render it unprofitable or useless to frame rules or to compile tables so complete and detailed as those that are used in merchant shipbuilding. The experience which can be obtained with new structural features, even in the largest navies, is—especially with the prevailing secretiveness—far more limited than that which is at the disposal of the Classification Societies, and the structure of warships is much more complex than that of merchant ships. For these reasons the problem before the designer of warships is much more difficult than that which the designer of merchant vessels has to solve.

2. **Design of Warships a Tentative Process.**—The different members of a ship's structure are interdependent, and each one must be designed with due regard to neighbouring members. The general layout of each one of the important parts of the structure must be first determined preliminarily on the basis of the fundamental conditions, and after that the relation between adjoining parts is considered and modifications or adjustments made as necessary to avoid conflicts and to harmonise the construction. Finally, after the study of the general

features is completed, local requirements and their influence on the design are taken into consideration and the details are settled.

3. **Continuity and Uniformity of Strength.**—Since the principal forces acting on a ship vary in a generally continuous and gradual manner, the structure should be designed so as, likewise, to vary continuously and gradually in strength in conformity with the variations of the forces. Any sudden discontinuity in structural strength will cause greatly increased local stresses, and the structure will be liable to show signs of weakness at such points. The strength should, in fact, everywhere be proportioned to the forces. Under statical conditions redundant material gives a useless excess of strength, but under dynamical conditions it is a positive source of weakness. Since, moreover, redundant material implies an excess in weight and cost, it should be carefully avoided.

By observing these rules we attain uniformity of strength, a quality which is of particular importance in light, high-speed vessels. When this quality is present, and as long as the elastic limit is nowhere passed, the structure in its entirety will yield under the action of the external forces as a homogeneous elastic body, each member bearing a due proportion of the strains both in tension and compression. In order to secure this quality, both design and workmanship must be of the highest order.

4. **Significance of the Calculated Stress.**—The object of strength calculations is, in general, to obtain an estimate of the maximum stress, which may be used in one of two ways:

(1) It may be directly compared with the primitive strength of the material, using the elastic limit or, more frequently, the ultimate strength, as found in the testing machine, as a basis of comparison.

(2) It may be compared, often without much regard to its numerical value, with the calculated stress obtained under the same assumptions in another similar structure.

When a structural member is of simple design and the forces acting upon it are fairly accurately known, the first method can be used. A "working stress" is determined upon by simply dividing the ultimate breaking stress with a factor of safety, and the structure is so designed that this working stress is not exceeded. The factor of safety is indeed based upon experience with other similar structures, and this method is therefore not intrinsically different from the second method; but while by this latter the comparison is carried out directly with a concrete case, it is through a factor of safety carried out with an entire class of cases, and its application covers a much wider field both in point of material and construction. Moreover, the calculated stress is not, when a factor of safety is used, supposed to differ materially from the actual stress.

In a warship, however, simple conditions rarely exist. The hull is exposed to many various forces, some of a general nature, others strongly localised and often of extreme intensity. In general, the forces are but imperfectly known. Moreover, the different structural members work together in a rather indefinite manner, causing unknown secondary stresses. Minor cases of overstrain can hardly be entirely avoided ; in fact, it is likely that they occur quite frequently even in the best-constructed vessels. When the elastic limit is passed at a certain point, the material will yield, an adjustment will take place, and additional strains are thrown on neighbouring parts of the structure. Eventually other, more remote parts are overstrained, until finally elastic equilibrium is established or the structure breaks down. If the calculated stress is far in excess of the elastic limit, it is ordinarily a sign that this limit is actually passed ; but since local overstrains are followed by adjustments, the calculated stress will not exist anywhere. In other words, the calculated stress is fictitious and has little absolute significance. Hence, we cannot in the hull of a warship assign any fixed limit to the stress, applicable to all cases. Such a limit can at the best be established only for a particular type of ship or for a particular kind of structural members. In general the second method must be employed, where the calculated stresses are used merely as a means of comparison with concrete cases. The ship chosen as type or model should preferably be of the same class as the design, not too different in size, and should be one with which satisfactory experience has been gained in actual service. It is of advantage if it is built in the same yard as that in which the new ship is to be built, because the quality of the workmanship enters as an important factor in all strength problems, a point which is of particular importance in high-speed vessels.

By basing the design on this method of comparison, features of redundant strength are indeed apt to be repeated, a drawback which can only be obviated by a tentative, cautious reduction in scantlings where redundant strength is suspected to exist. The effect of such reduction should always be carefully watched.

In general, whenever structural failure occurs—for instance, by the rivets working adrift, by buckling or cracking of the plates, or by complete breakdown—it should be studied and analysed, as far as circumstances permit, with the same care as would be a scientific experiment. In this way much valuable information may be obtained as to the relation between actual and calculated stresses. This question should also be studied, whenever opportunities offer themselves, by measurements of strains and deflections of ships in various conditions of service. Full-scale experiments afford, of course, the most perfect means of scientific investigation, but the cost is usually prohibitive.

The formulas and methods of calculation employed in determining the stresses should include all the principal factors that influence the problem, giving to each its proper weight, and should take into account all the important points of difference between the new design and the type ship. All unessential factors should be neglected. As a general rule, complicated formulas and elaborate methods of calculation are of little value. The accuracy attained by refinements in calculation is only apparent, since the probable errors, due to the crudeness of the fundamental assumptions, will ordinarily overshadow the corrections obtained by introducing factors of minor importance. Complexity and bulkiness of calculations tend, moreover, to obscure the important points in the work, and errors are liable to creep in.

5. **Subdivision of the Subject.**—In the early chapters of this work the structural strength of ships will be discussed and the methods of calculations pertaining thereto will be described. We shall deal first with the general strength of the hull, pass on to the strength of individual members, and finally consider local strength. The subject is accordingly grouped as follows :—

(1) Longitudinal strength of the entire ship.
(2) Transverse strength, dealing in particular with transverse bulkheads and frames.
(3) Strength of individual girders such as frames and beams, spanning the distance between rigid walls and diaphragms.
(4) Strength of plating under fluid pressure, a problem which occurs locally in the outer and inner shell, in the bulkheads, and in the platform decks.
(5) Strength of columns and plating under compression.

6. **Notation.**—The following notation is used throughout this work :—

E = Modulus of elasticity.
A = Sectional area of girder or column.
I = Moment of inertia of sectional area of girder.
y = Ordinate above or below the neutral axis of section.
$S = \frac{I}{y}$ = Section modulus.
M = Bending moment.
Q = Shearing force.

p = Normal stress in general, whether tensile or compressive. (In the discussion on riveted joints p is also used to signify the pitch of the rivets.)

q = Shearing stress as distinguished from normal stress.

w = Load per unit length of girder.

s = Spacing of girders (frames, beams, or stiffeners)—in a rectangular plate, the length of the shortest side.

t = Thickness of plating.

$\mu = \frac{s}{t}$ = Ratio between spacing and thickness.

δ = Deflection.

d = Diameter of rivets.

When it is desired to specify the nature of the stresses the following notation is used :—

p_T = Tensile stress.

p_C = Compressive stress.

p_S = Shearing stress.

f_T = Ultimate tensile stress.

f_C = Ultimate compressive stress.

f_S = Ultimate shearing stress.

7. **Definitions.**—The term "amidships" is used to denote that part of the ship which is half-way or about half-way between stem and stern. When referring to parts of the ship near the center-line, such expressions as "near the center-line," "near the keel-line," or "axial" will be employed.

When a beam or girder is fixed in direction at the end, it is said to be "fixed," and the quality is referred to as "fixity." Thus we may speak of a certain "degree of fixity" when a beam is not held quite rigidly at the end. A beam, "fixed" at the ends, and prevented from slipping, so that it will be subject to tension when it deflects, is said to be "absolutely fixed." A beam supported at the ends, but entirely free to turn and to slip at the ends, is said to be "freely supported." If such a beam is free to turn at the ends, but is prevented from slipping, it is said to be "hinged."

The "ton" is always 2240 lb.

CHAPTER II.

LONGITUDINAL STRENGTH.

4. GENERAL CONSIDERATIONS.

1. **Importance of the Subject.**—Many cases are on record where ships have shown signs of weakness due to a lack of longitudinal strength. Tearing or buckling of the plating and shearing of the rivets are symptoms of frequent occurrence, and even complete rupture of the hull has taken place. Strains and breakdowns of this nature have happened both during sea-service and when ships have been aground, and many ships have been severely strained during the launch, where in some cases the alignment of the shafts has been disturbed. The importance of longitudinal strength is enhanced by the modern development in warship construction, which goes in direction of an increase in length and power, a reduction in scantlings, and an increase in size of all classes of ships.

2. **The Ship-Girder.**—A ship may be considered as a hollow girder, the "ship-girder," where the flanges are formed by the decks and the

bottom structure, the web by the sides, often assisted by longitudinal bulkheads. A ship is, like a bridge, subject to the effects of travelling loads; but it differs in that the forces of load and support, and hence the stresses and strains, are more variable in strength and direction and more difficult to estimate. In a seaway, the forces which come into play are but imperfectly known on account of the irregularity of the motion. When a ship is rolling the functions of the web and of the flanges are not sharply defined and the strains are constantly shifting. Each of the four

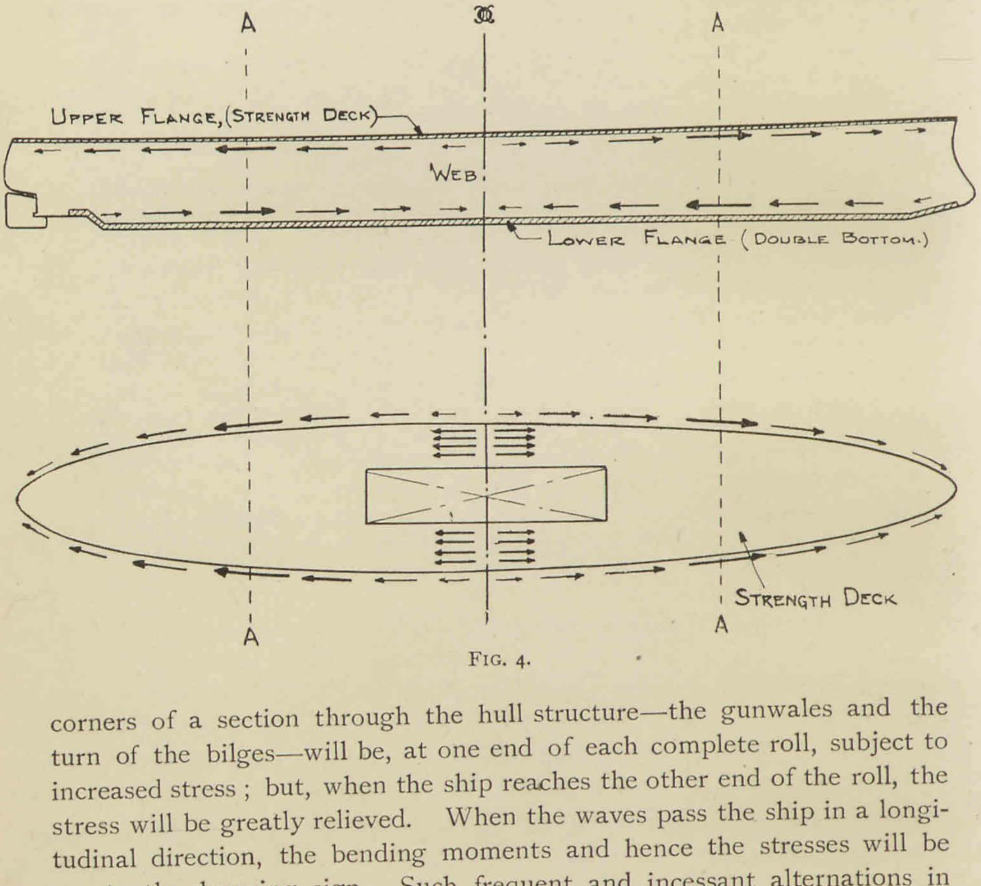


FIG. 4.

corners of a section through the hull structure—the gunwales and the turn of the bilges—will be, at one end of each complete roll, subject to increased stress; but, when the ship reaches the other end of the roll, the stress will be greatly relieved. When the waves pass the ship in a longitudinal direction, the bending moments and hence the stresses will be constantly changing sign. Such frequent and incessant alternations in the intensity and sign of the stresses have no parallel in bridge girders.

3. **Origin and General Distribution of Strains.**—The bending originates in the bottom structure, due to the unequal distribution of weight and buoyancy. The bottom in itself, even if it is double, can offer but small resistance to bending; but, being rigidly connected to the sides (and longitudinal bulkheads), it cannot bend without these members

—that is, the web of the girder—following its motion. To fix our ideas, let us consider a single-deck ship in hogging condition (fig. 4). If, now, the sides were not reinforced by a deck, their upper edge would be liable to tear at the middle. The deck prevents this, but in so doing, causes the sides to distort. Thus, the sides become subject to shearing, while the deck and the bottom will be respectively in tension and compression, induced by the sides and transmitted through the medium of the sheer strakes and the bilge strakes. This transmission, then, takes place at the boundaries between the web and the flanges, and is strongest on about a quarter length of the ship, at AA, where the shearing is a maximum. Fig. 4 shows, in a general way, the distribution of the straining forces along these boundaries. The distribution of the stresses throughout the structure will be discussed more in detail in SECTION 8, *2*. We shall deal separately with each of the two actions—pure bending and pure shearing —and afterwards show how to combine the stresses due to both.

5. BENDING.

1. **Fundamental Formula.**—The tensile and compressive stresses produced in a ship by simple longitudinal bending are determined in accordance with the theory of elastic bending by the formula

$$\frac{p}{y} = \frac{M}{I} \quad . \quad . \quad . \quad . \quad . \quad (1)$$

The validity of this formula as applied to a ship-girder has been proved in a general way by experience, and has been corroborated by the experiments of Sir John Biles on the destroyer *Wolf*.* We shall not enter into a detailed description of the longitudinal strength calculation, which is fully dealt with in text-books on Naval Architecture. It is sufficient here to give a brief review of the method, but it is deemed advisable to state, and in some cases to discuss more fully, the assumptions that are ordinarily made by the designer in applying the formula.

2. **Assumptions.**—It is of importance that the assumptions should conform as nearly as possible to actual conditions, and that they should be so simple as not to necessitate long and complicated computations. Further, it is essential that they should be identical in the cases to be compared, whence they must be precisely defined.

The ship is assumed to be poised on a trochoidal wave of the same length, L, as the ship in the water-line and of a total height, from crest to hollow, equal to $\frac{L}{20}$. The ship is, moreover, assumed to be upright

* *Inst. Nav. Arch.*, 1905, vol. i.

and at rest in a position at right angles to the waves. The longitudinal strength is examined in two standard conditions: one, the "hogging condition," where the ship is placed with its middle on the top of a wave; the other, the "sagging condition," where it is placed with its middle on the wave-hollow, the ends being on the adjacent wave-crests. In order to obtain the stresses under the most unfavorable conditions, the ship, in sagging, should be assumed fully equipped, while, in hogging, it should be assumed to be light, *i.e.* all consumable stores such as fuel and fresh

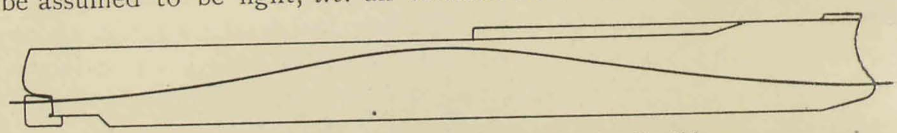


FIG. 5.—Ship on the Crest of a Wave: Hogging Condition.

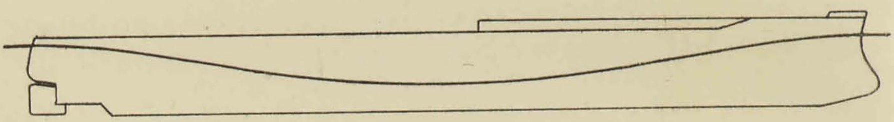


FIG. 6.—Ship in the Hollow of a Wave: Sagging Condition.

water but not the ammunition removed. Frequently the strength is calculated both for sagging and hogging for each of the different conditions of loading—light, normal, and full load. The immersion and trim are generally so adjusted that the ship is in statical equilibrium in the upright position, and the water pressures on the immersed portion of the ship are assumed to be proportional to the depth below the trochoidal wave surface. We shall briefly examine how far we are justified in thus neglecting both the motion of the ship and the waves.

3. **Inclinations of the Ship and Dynamic Actions.**—*Inclination.*—By moderate inclinations the bending moments will not be materially affected, but the section modulus of the ship-girder will be somewhat changed due to the inclined position of the neutral axis. The method of calculating the stresses in such cases, where the plane of the bending moment is not parallel with any of the principal planes of inertia, is described in SECTION 14. We shall here merely state that, according to investigations of Sir John Biles and others, the maximum stresses in inclined positions rarely exceed the maximum stresses in upright position by more than from ten to fifteen per cent.

The Smith Correction.—Due to the orbital motion of the particles of water in the waves, the pressures will not be those corresponding to the depth below the trochoidal surface. In the wave-hollow they will be greater, in the wave-crest they will be smaller. The result is a reduction in the bending moment in both sagging and hogging conditions. Sir W. E. Smith has shown * how this effect can be taken into account, assuming that the presence of the ship in the waves does not influence the wave-pressures. It has been found that the reduction in bending moment calculated on this basis, the so-called "Smith Correction," is, generally, small and will rarely exceed from ten to twelve per cent.

Heaving.—The heaving motion of a ship among waves will cause reductions and increases in the bending moment which can be calculated in each case under given assumptions. We are not at present able to make any definite statement as to the quantitative value of these effects, but it seems safe to say that they will ordinarily be very moderate.

Pitching.—The pitching motion has its greatest effect on from one-quarter to one-third of the length of the ship from the stem and stern. The effect is greatest in the bow, where the bending moment may be con-

CORRIGENDA.

Page 21, line 13.—*Delete* "from the stem to the stern. The effect is greatest in the bow, where the bending moment may be considerably augmented, although it will rarely be as great as under standard conditions amidships."

And substitute

"from the stem to the stern, where the bending moment may be augmented even beyond its value under standard conditions amidships ; probably the effect is greatest in the bow."

Page 21, line 22.—*After* "Alexander in 1911,"

Insert

"and by Mr A. Cannon in 1914."

To face page 21. HOVGAARD, STRUCTURAL DESIGN OF WARSHIPS. (PR. 1286.)

tion gives the curve of bending moments. The integrations are most conveniently carried out by means of the integraph. The bending

* *Inst. Nav. Arch.*, 1883.

and at rest in a position at right angles to the waves. The longitudinal strength is examined in two standard conditions: one, the "hogging condition," where the ship is placed with its middle on the top of a wave; the other, the "sagging condition," where it is placed with its middle on the wave-hollow, the ends being on the adjacent wave-crests. In order to obtain the stresses under the most unfavorable conditions, the ship, in sagging, should be assumed fully equipped, while, in hogging, it should be assumed to be light, *i.e.* all consumable stores such as fuel and fresh

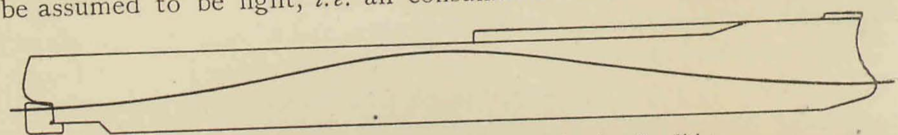
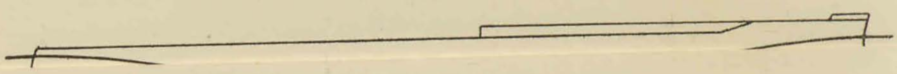


FIG. 5.—Ship on the Crest of a Wave: Hogging Condition.



to the depth below the trochoidal surface. In the wave-hollow they will be greater, in the wave-crest they will be smaller. The result is a reduction in the bending moment in both sagging and hogging conditions. Sir W. E. Smith has shown * how this effect can be taken into account, assuming that the presence of the ship in the waves does not influence the wave-pressures. It has been found that the reduction in bending moment calculated on this basis, the so-called "Smith Correction," is, generally, small and will rarely exceed from ten to twelve per cent.

Heaving.—The heaving motion of a ship among waves will cause reductions and increases in the bending moment which can be calculated in each case under given assumptions. We are not at present able to make any definite statement as to the quantitative value of these effects, but it seems safe to say that they will ordinarily be very moderate.

Pitching.—The pitching motion has its greatest effect on from one-quarter to one-third of the length of the ship from the stem and stern. The effect is greatest in the bow, where the bending moment may be considerably augmented, although it will rarely be as great as under standard conditions amidships. It is advisable not to let the section modulus of the ship-girder fall materially below its midship value till beyond the quarter length from amidships, especially in the fore-body.

For a further study of the influence of heaving and pitching on the stresses in ships the student is referred to papers read before the Institution of Naval Architects by Mr T. C. Read in 1890 and by Mr F. H. Alexander in 1911, as also to the work of Sir John Biles on *Design and Construction of Ships*.

While an investigation of the various dynamical actions is of great interest because it throws light on the relative importance of their effects, it does not seem necessary or profitable to embody these actions in the standard strength calculations. To some extent the resulting corrections neutralise each other, and the error which we are liable to commit by omitting them will be of the second order. In fact, if the ships to be compared are of similar type, the error must be practically insignificant. Bearing in mind, moreover, the arbitrary and uncertain nature of the assumptions on the basis of which the corrections for the dynamical effects are at present determined, the additional complication and work incurred by taking them into account seem unwarranted.

4. **Principal Steps in the Calculation.**—First, curves of buoyancy and weight are constructed and the difference between the ordinates of these curves furnish the curve of loads. By integration of the curve of loads we obtain the curve of shearing forces, which by a second integration gives the curve of bending moments. The integrations are most conveniently carried out by means of the integraph. The bending

* *Inst. Nav. Arch.*, 1883.

moment at any section of the ship can now be obtained, but, generally, only the maximum, usually occurring near the midship section, is considered. In ships where an important structural member, as for instance a longitudinal bulkhead or a deckhouse, is discontinued at a point where the bending moment is still considerable, it may be necessary also to examine the strength at this point.

It remains to determine the moment of inertia and the position of the neutral axis of the section under consideration—generally the weakest section in the region of the maximum bending moment—whereupon the stresses in the most strained fibres can be determined by formula (1) corresponding to each of the standard conditions.

5. **Effectiveness of Longitudinal Members.**—In the calculation of the moment of inertia should be included all effective longitudinal structural members that pass through the section and which are continuous for a considerable part of the length of the ship—generally at least one-half this length, but the determination of what members ought to be considered as "effective" is a point which calls for special mention. It is perhaps the most difficult question in the strength calculation, and that on which opinions are most at variance.

In warship construction the military aspect of the different problems should always be first considered, and in the present case we must, therefore, begin by examining what members are likely to remain effective after an action. This fundamental point being settled, we have to consider that when the ship is in a seaway the structural members are subject alternately to tension and compression, and that the behavior of the material may be entirely different in the two cases. Light plating can generally support great tensile stresses without failure, but is liable to buckle already at moderate compressive stresses. A belt of heavy armor plates, on the other hand, may offer a certain resistance to compression, while it is quite incapable of resisting tension. Likewise, wood planking offers a greater resistance in compression than in tension. While these points are fairly clear, difficulties arise as soon as we try to evaluate the strength quantitatively. Finally, we have to consider the presence of joints and rivet holes, of hatches, and of other openings which in some measure weaken the structure. It is, *a priori*, evident that the moment of inertia and the neutral axis will not be identical in hogging and in sagging.

6. **Armor Protection and Structural Strength.**—In battleships, battle-cruisers, and other armored ships, designed for serious and long-continued fighting, all unprotected parts are liable to be demolished in action. The protected parts are also liable to suffer, but the damage

will be more local and will rarely affect the general strength of the hull. Hence, *only the members that are protected by armor should be included as effective strength members and constructed as such in vessels of the battleship class*, and it follows as a corollary that *the armor should be so distributed as to protect effectively the structural strength of the ship.*

This principle is most important, but appears not, so far, to have been clearly enunciated, and has not been followed everywhere in a complete and logical manner. It applies well to modern battleships where, for other reasons, a complete or nearly complete belt is carried to the height of the second (gun) deck * and where this deck is ordinarily armored. A sufficient depth of protected ship-girder is readily secured, and all that is required, is to make the armor deck continuous from end to end and to construct it as a "strength-deck" by providing efficient butt connections. Even with a very moderate depth of the ship-girder, ample strength can be secured in such a deck without any appreciable expenditure of extra weight. The strength is not so liable to be destroyed by grazing shell as where the strength-deck is unarmored. The armor belt may be of reduced thickness beyond the vitals, but should be capable of protecting the structure against demolition by thin-walled shell for at least two-thirds of the length. At the extreme ends it may be discontinued. With this construction the hull above the belt may be very light, unless requirements other than that to longitudinal strength necessitate heavy scantlings. *It appears, in fact, entirely unnecessary and undesirable to fit heavy sheer strakes and stringer plates in the upper unprotected parts.*

It must not be overlooked, on the other hand, that due to the small depth of the armored ship-girder, the deflections will be relatively great for a given maximum stress, whence the strains on the upper light structure are liable to be excessive. It may, therefore, be necessary here to provide expansion joints.

In many ships an armored casemate for secondary guns is fitted above the belt for a certain length, but the strength-deck should not therefore be raised to the top of the casemate in this region, because it will then be impossible to preserve a proper continuity of strength. By fitting the strength-deck on top of the solid belt throughout the length of the ship there is the further advantage that it will be sheltered in its most important part by the armor of the casemate.

7. **Structural Members in Tension.**—The upper flange of the ship-girder is subject to tension in hogging, the lower flange is subject to tension in sagging. Now, the question has been raised whether the inner parts of these flanges, in particular the plating, are actually capable

* The English "main" deck.

of doing their duty efficiently. It is argued that the web—that is, the sides—can only induce longitudinal forces in those parts of the flanges—that is, the decks and the bottom structure—to which they are directly connected, while the parts that are nearer the center-line will evade this action. This does not, however, appear to be the case, for, as explained in SECTION 8 and as seen from fig. 4 and Pl. I. the tapering form of the ship towards the ends and the horizontal transmission of the forces through shearing will cause the pull on the boundaries to be transmitted to the inner as well as to the outer parts of the flanges. It has been supposed, moreover, that the plating of the flanges, when in tension, might shirk its work by lengthwise bulging, but in a warship the numerous stiffening members will prevent such action. We may, therefore, in tension consider the full intact sectional area of the continuous longitudinal members as effective.

8. **Structural Members in Compression.**—The most important point to consider is here the possibility of failure by buckling of the plating. We must distinguish between general bulging of larger areas of the surface, comprising bending of the stiffening members, and local buckling of the plating between beams or frames. The first form of collapse will be discussed in SECTIONS 7 and 12. We shall here deal with the latter form only.

Where the stress is small, as in the neighbourhood of the neutral axis of the ship, there is no danger of collapse and the full sectional area of the material is effective, but already at a short distance from the neutral axis (*i.e.* the neutral axis corresponding to the upright position) considerable stresses may occur when the ship heels over in a seaway. If the stress reaches a certain point, buckling will take place and the plating will be robbed entirely of its strength. The determinative elements are :—

(1) The ratio between the spacing of transverse stiffeners—frames or beams—and the thickness of the plating $\mu = \frac{s}{t}$.

(2) The presence of longitudinal stiffeners. These act in two distinctly different ways. First, they prevent buckling of a belt of plating of a certain width in their immediate vicinity in virtue of the support which they give to it directly. Second, they act in conjunction with adjacent stiffeners, giving to the intervening plating a certain support, which depends chiefly on the ratio between the spacing of the longitudinal stiffeners and the thickness of the plating.

In order to study the effectiveness of the plating it is necessary to consider each of these factors separately.

9. **Effectiveness of Plating Unsupported by Longitudinal Stiffeners.**—The plating between beams and frames, if sufficiently remote from any longitudinal stiffeners, may be regarded as made up of elemental strips of unit width, each of which acts as an independent column under compression. As explained in SECTION **22**, the strength of such strips is best examined by means of Euler's formula (56), regarding the strips as fixed at the ends. We may form a general idea of the effectiveness of the plating in different parts of the hull by assuming that the working stress in the most strained parts of a ship subject to longitudinal bending does not exceed a certain limit, which we shall take to be $4\frac{1}{2}$ ts. per sq. in. for mild steel and 6 ts. per sq. in. for high-tensile steel. Inserting these figures for p in Euler's formula and solving, we find the value of μ at which collapse is likely to occur, viz. for mild steel

$$\mu = \sqrt{\frac{44400}{4 \cdot 5}} = 99$$

and for high-tensile steel

$$\mu = \sqrt{\frac{44400}{6}} = 86$$

Now, in large vessels the value of μ is about as follows :—

Flat keel plates, where intermediate frames or brackets are fitted (s = 24 in.), μ = 20 or less.
Sheer strakes and deck stringers, μ = 48
Outside plating of standard thickness, μ = 80
Inner bottom plating and light deck plating, μ = 150

It is clear from these figures that the heavy and reinforced strakes such as sheer strakes, deck stringers, and keel strakes can be reckoned to be fully effective, while, on the other hand, the inner bottom and the light deck plating must be considered ineffective where it is not supported by longitudinal stiffeners. For the outside plating μ falls somewhat below the critical value, but there is practically no factor of safety to allow for dynamic effects and it has to be borne in mind that the plating is loaded normally by the pressures of the water. It seems safest, therefore, to consider as ineffective those parts of the outside plating that are remote from the neutral axis and not efficiently supported by longitudinals.

In destroyers μ is rarely less than 80 for the ordinary outside plating and is about 50 for the heavier strakes in the shell and the deck. In the lighter deck strakes it is often 120 or more. Hence, the conclusions are the same as for the large vessels, viz. *that heavy and reinforced strakes* *may be reckoned fully effective, other strakes only where supported by longitudinal stiffeners.* This rule will hold good even if the working stresses are considerably different from those assumed above.

After a preliminary longitudinal strength calculation has been made, each strake can, of course, be examined separately in cases where doubt exists as to its effectiveness.

10. **Effectiveness of Plating near Longitudinal Stiffeners.**—The spacing of the transverse stiffeners is here supposed to be so great that the plating will collapse except where supported by longitudinal stiffeners, *i.e.*, generally, for mild steel $\mu > 100$.

Considering first the case of an isolated longitudinal stiffener, we assume that a band of plating of a certain width on each side is fully effective within the working stress. This width must be a function of μ, but in the absence of sufficient experimental data we shall express it simply as a multiple of the thickness of the plating. The case is distinctly different from that where the plating acts as the flange of an individual girder. The stress is there induced into the plating by the action of the web, and the width of the band which is affected must be smaller—as explained in SECTION 15, it is recommended to reckon it equal to $30t$. In a ship-girder, where the entire plating of the flanges is under a fairly uniform compressive stress and where the stiffeners do not act as girders but simply hold the plating to its duty, the effective width is probably somewhat greater. Judging from Dr J. Bruhn's experiments on the strength of girders * and from an analysis of the *Wolf* experiments, *it is recommended to reckon as effective a band of plating of a width equal to fifty times the thickness at each longitudinal stiffening member*—that is, a strip of width $25t$ on each side. The members need not be continuous, and may not themselves contribute directly to longitudinal strength, but they must be capable of stiffening the plating and must exist for a sufficient length amidships. Thus, the plating along intercostal hold stringers and even along light stiffening angles fitted under a deck between the beams will, according to this rule, be reckoned as effective.

We have, so far, considered the effect of each longitudinal stiffener separately, but it is clear that if we imagine the stiffeners to be approached sufficiently to each other they will combine to support the plating between them, and finally a point will be reached where the plating is fully effective in compression, since buckling will be then prevented by the resistance to

* These experiments were carried out for the Committee of Lloyd's Register. They comprised various forms of girders used in ship construction, and were published in a paper entitled "Some Experiments on Structural Arrangements in Ships," read before the Institution of Naval Architects in 1905. They will be referred to several times in the following.

bending offered by the elemental transverse strips. The spacing of the longitudinal stiffeners for which this condition is fulfilled must be greater than the effective width of the plating for an isolated stiffener—estimated above at $50t$. We have no means of determining this spacing accurately. It will of course depend on the working stress and to some extent on the spacing of the transverse frames. In bridge construction the cover plates on the top flanges of compressive chords are, generally, limited in width

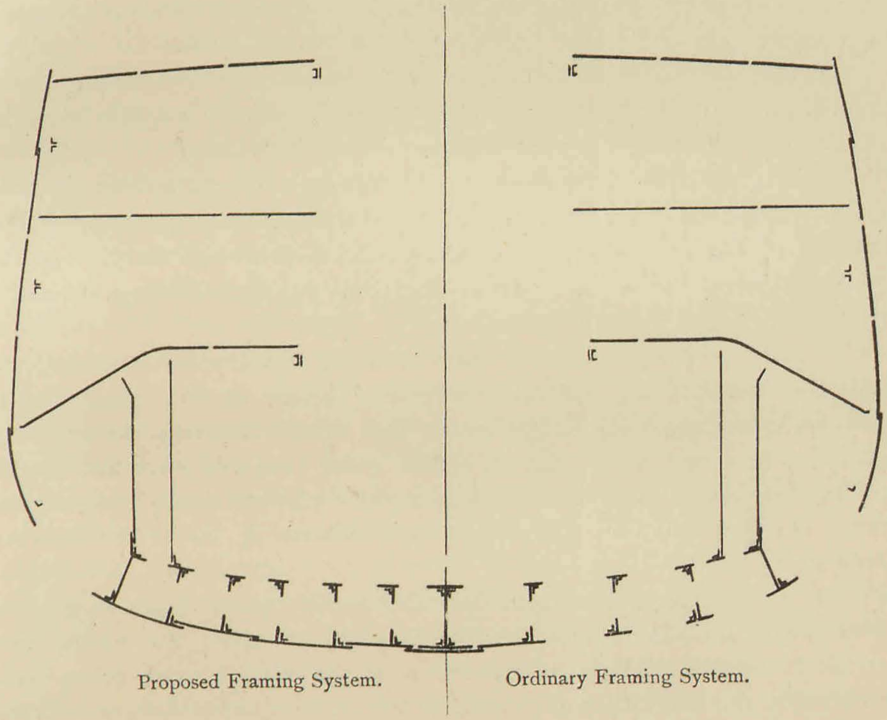


FIG. 7.—Effective Fore-and-aft Members of Light Cruiser (see SECTION 48, fig. 116): Hogging Condition.

to $40t$ between connecting lines of rivets, but the normal working stress is there very high—7 ts. per sq. in. Judging from this and other analogous cases, as well as from Dr Bruhn's experiments, *it is proposed to reckon the limiting spacing of longitudinal stiffeners, for which the plating may be assumed to be fully effective in compression, to be equal to about eighty times the thickness.*

Fig. 7 gives by way of illustration a section of the fore-and-aft members of a cruiser in hogging condition where the ineffective parts of the plating are removed. On the right-hand side the ship is framed in the usual manner with a wide spacing of the longitudinals, on the left-hand side the longitudinals are spaced closer as recommended in SECTION 48, giving a greater effectiveness of the plating.

By following the rules here given, the effective sectional area of the ship-girder will in many vessels be considerably smaller, and hence the calculated stresses and deflections will be greater, than by the usual method of calculation, where the full sectional area of all continuous fore-and-aft members is included in the calculation of the moment of inertia, but it is believed that the result will conform more closely to actual conditions.

11. **Actual Cases of Buckling.**—Numerous cases of buckling are on record, especially in the deck plating of torpedo-vessels, but also buckling of the bottom plating has occurred in vessels of this class.

In a certain ship of the scout type, buckling of the main deck occurred on the first sea trial. The deck was not sheathed with wood. The spacing of the beams was 600 mm. and the thickness of the plating was 7 mm., giving $\mu = \overline{86}$. After the trial the deck plating showed a permanent set in certain places.

It must be borne in mind that buckling is generally arrested by adjacent longitudinal stiffening members before it has reached any considerable magnitude and before the elastic limit is passed. It seems, therefore, likely that it occurs quite frequently in the outside plating of many ships and torpedo-vessels without being noticed, because it is so small in magnitude and because it leaves little or no permanent set.

12. **Armor as a Strength Member.**—Side armor, when in tension, does not contribute to longitudinal strength because the plates have no butt connections. In compression it probably always offers some resistance, the magnitude of which, however, it is impossible to estimate even approximately. It is likely that at small and moderate strains this resistance will be negligible on account of the imperfect contact between the butts of the armor plates. In extreme cases, when the material of the hull structure is strained beyond the yield point, the armor plates may be brought into more intimate contact with one another at the butts and the armor may then become very effective, but in well-designed ships this point should never be reached under ordinary service conditions. It seems advisable, therefore, to disregard the side armor entirely in the strength calculation and to design the hull proper so as to possess sufficient strength without it. The side armor may then be regarded as a reserve, which is called into play in extreme cases, as when the ship is aground or when it is damaged in action.

Deck-armor, on the other hand, should be included in the calculation both in tension and compression, in a measure depending on the efficiency of the butt connections.

13. **Wood Decks, Sheathing, and Backing Behind Armor.**—All members of wood are usually reckoned to be equivalent to steel plating of one-twenty-fifth of their thickness in tension and one-sixteenth of their thickness in compression.*

14. **Effect of Rivet Holes, End-Joints, Hatches, and Other Openings.**—Along every frame, beam, or stiffener there is, when the plating is in tension, an unavoidable line of weakness due to the rivet holes. In a well-designed ship all other lines of weakness are made to offer the same or a greater resistance than this line, which, therefore, is generally chosen as the standard in all strength calculations.

In calculating the neutral axis and the moment of inertia it is customary to deduct the area of these rivet holes from the sectional area of the parts in tension, while no such deduction is made for the parts in compression. This procedure is not, however, rational, and leads to errors in more than one way. The lines or belts of weakness produced along beams and frames by the rivet holes are recurrent but are very narrow. In a large ship, for instance, where the rivets are of $\frac{7}{8}$-in. diameter and the frame space is 48 in., the width of the belt affected by the rivet holes is only about one-fiftieth of the frame space. It is clear that such local weakening cannot greatly affect the position of the neutral axis about which the bending of the ship-girder takes place. The neutral axis cannot have a jog at every frame, but it is likely that in its entirety it will be raised or lowered a little according as the weakening is found in the lower or in the upper part of the ship. Suppose, for instance, that the ship is subject to hogging. The sectional area along the frames and beams above the neutral axis is reduced by about one-eighth, assuming a spacing of the rivets of eight diameters, but if this reduction is distributed over the whole plating it will amount to only about $\frac{1}{8} \times \frac{1}{50} = \frac{1}{400}$, which fraction measures the average additional strain in the upper part of the ship. This will cause the neutral axis to move down about one-thousandth of the distance of the most strained fibre from the neutral axis, a variation which can safely be neglected. Hence, the neutral axis may be calculated as if there were no rivet holes either above or below this axis, and the same reasoning leads us to disregard other small openings, such as side-lights, and the weakening at end-joints. In calculating the moment of inertia the same rule should be followed. The tensile stress determined on this basis is correct for the intact plating between the frames and beams, *i.e.* for the greater

* J. H. Biles, *The Design and Construction of Ships*, i. 263.

part of the structure, but for the section through rivet holes and other points of weakness it must be augmented in proportion to the reduction in sectional area.

By this method, which was proposed by Messrs T. C. Read and G. Stanbury already in 1894,* but which has not received the recognition it deserves, a more correct idea of the distribution of the stresses is obtained than by the method now in general use. We come to realise more clearly the high stresses which may exist locally, for instance at weak end-joints. These stresses are apt to escape attention by the usual method of strength calculation, but may cause serious trouble even although the general strength is satisfactory.

Where large openings are found in the plating, as for instance hatches in the decks, and in particular where such openings are of great longitudinal extent or recur in the same strake, the respective sectional areas of the strakes thus interrupted should be omitted in the calculation. The great stresses found at the hatch corners must be dealt with separately.

15. **Application of Results.**—If the longitudinal strength calculation is carried out with good judgment on sound assumptions consistently applied, it affords a valuable means of comparison between a new design and existing ships of similar type. The calculated stresses depend, however, so much on the underlying assumptions, which are probably not the same in any two navies, that data from different sources can rarely be directly compared. Every navy must rely chiefly on its own experience. We shall not, therefore, give any lengthy compilation of numerical results, or attempt to give any rule for the permissible working stress, but merely state a few facts.

For the British scouts of the *Pathfinder* class the stipulated maximum tensile stress was 6 ts. per sq. in. for mild steel and 8 ts. per sq. in. for high-tensile steel. The maximum compressive stress was $4\frac{1}{2}$ ts. per sq. in. for mild steel and 6 ts. per sq. in. for high-tensile steel. The strength calculations were to be carried out by the usual method.

The British destroyer *Cobra*, which broke and was lost in the open sea, had a calculated tensile stress in the keel of 9·6 ts. per sq. in. when on a wave-hollow.

Certain torpedo-boats showed buckling of the deck with a calculated compressive stress of 7 ts. per sq. in. Boats of the same class from another firm showed no sign of weakness with a calculated stress in the deck of $5\frac{3}{4}$ ts. per sq. in. The assumed height of wave in the strength calculations was as usual one-twentieth of the length.

* *Inst. Nav. Arch.*, 1894, p. 378.

6. SHEARING.

1. **Elementary Considerations.**—The curve of shearing force obtained by integrating the curve of loads gives the total vertical shearing force acting on any transverse section of the ship, but it does not give any information about the distribution of the shearing over the section. By dividing the shearing force with the total area of the section an average stress is found which, however, in a thin-webbed girder like a ship, is much smaller than the maximum. In fact, the shearing stress varies from point to point, but in a ship-girder the variation is far less marked in the web than in the flanges. We cannot determine the vertical shearing stress as such, but at every point an equal horizontal shearing stress is found, and this we can determine, provided the structure is not too complex.

In a solid prismatic beam the problem is simple. Let Q be the total vertical shearing force, I the moment of inertia of the sectional area about its neutral axis, m the moment about the neutral axis of the area above or below the point under consideration, and b the thickness or breadth of the beam at that point. Then the shearing stress at the point is given by the formula

$$q = \frac{Qm}{bI} \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The ship-girder presents a more difficult problem on account of its complex structure, but since the bending under consideration is strictly longitudinal, we need only take into account the longitudinal members.

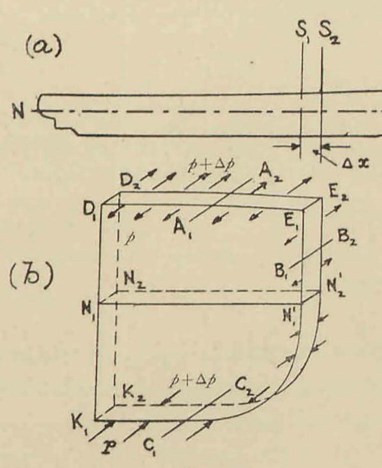


FIG. 8.

2. **The Ideal Single-Deck Vessel.** — We shall commence by studying the shearing stresses in a single-deck vessel without any double bottom, bulkheads, or girders. Since the vessel is symmetrical about the center-line plane, we need only consider one side.

Let S_1 and S_2, (fig. 8, a), be two transverse sections at a distance Δx from each other. NN is the neutral axis of the ship-girder. Fig. 8, b, represents the transverse element enclosed between S_1 and S_2. $N_1N_2N_2'N_1'$ is an element of the neutral surface which we suppose to be plane and horizontal within the length

Δx. A_1A_2, B_1B_2, C_1C_2 are lines of intersection between the outer surface of the vessel and planes normal to this surface and parallel with N_1N_2. These lines will not generally be parallel with N_1N_2 but will form with it an angle θ which represents the obliquity of the surface at the respective points.

Consider now the equilibrium of that part of the deck which is enclosed between section A_1A_2 and the center-line D_1D_2. On A_1D_1 there is acting a total normal force P and, unless the bending moments at S_1 and S_2 are equal, a force $P + \Delta P$ will be acting on A_2D_2 leaving a resultant ΔP which acts parallel with the neutral axis. This resultant must be balanced entirely by longitudinal shearing along A_1A_2 since, by symmetry, there can be no shearing along D_1D_2. The deck being practically parallel with the neutral axis, we have $\theta = 0$

and $$\Delta P = qt\Delta x$$

or $$q = \frac{1}{t}\frac{dP}{dx} \quad . \quad . \quad . \quad . \quad . \quad (3)$$

For sections taken on the side of the ship, such as B_1B_2, ΔP represents again the resultant of the stresses normal to the sections S_1 and S_2 acting on that part of the structure BED which is enclosed between the respective section and the center-line. Suppose now that the side at B_1B_2 is oblique, forming an angle θ with N_1N_2 then the elemental sectional area of the plating is: $t\Delta x \sec\theta$. (See fig. 9.) The shearing stress q must here act parallel with the contour B_1B_2 and its longitudinal component must balance ΔP.

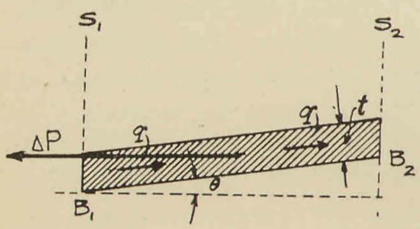


FIG. 9.

Hence $$\Delta P = q \cos\theta \, t\Delta x \sec\theta$$

or again $$q = \frac{1}{t}\frac{dP}{dx}$$

For the section C_1C_2 it is convenient to consider the equilibrium of the piece enclosed between this section and the keel-line K_1K_2 but the formula will be the same.

If p is the normal stress at any point and y the vertical ordinate of this point, referred to the plane of the neutral axis, we have from (1):

$$p = \frac{My}{I_0}$$

where I_0 is the moment of inertia of the entire section (both sides) about the neutral axis. The force P which is the sum of the normal stresses at any section acting between the point under consideration and the center-line, is :

$$P = \int ptds = \int \frac{yM}{I_0} tds$$

the integration to extend along the contour of the section between the point and the center-line of the deck D or the keel K . Either integration will give the same numerical result, but that which involves the smallest amount of numerical work should be chosen. M and I_0 are independent of the integration, and $\int ytds$ is the moment about the neutral axis of the area between the point under consideration and the center-line. This moment we have already denoted by m . Hence we may write

$$P = \frac{Mm}{I_0}$$

and

$$\frac{dP}{dx} = \frac{m}{I_0}Q + M\frac{d}{dx}\left(\frac{m}{I_0}\right)$$

Substituting in (3),

$$q = \frac{1}{t}\left[\frac{m}{I_0}Q + M\frac{d}{dx}\left(\frac{m}{I_0}\right)\right] \quad . \quad . \quad . \quad . \quad (4)$$

If the ship were of the same section throughout the length $\frac{m}{I_0}$ would be constant, whence

$$q = \frac{mQ}{tI_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

which is the same formula as for a prismatic beam.

In actual ships there is considerable obliquity of the sides and change in construction towards the ends, but the value of $\frac{m}{I_0}$ will change but little as long as the depth of the ship remains the same. Towards the ends, the depth generally increases somewhat due to the sheer, $\frac{m}{I_0}$ will be slightly reduced, and the last term inside brackets in (4) will become negative, causing a reduction in q. Generally, however, the value of the last term will be insignificant, and the simple formula (5) may be used as a sufficient approximation.

At the neutral axis this formula becomes :

$$q_0 = \frac{m_0 Q}{t_0 I_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

where m_0 is the moment about the neutral axis of the half section above or below this axis. It must be borne in mind that I_0 is the moment of inertia of the entire section for both sides, Q is the whole vertical shearing force, acting on the entire section, and t_0 is the thickness of the plating in the neutral axis on one side. Unless t_0 is abnormally great, q_0 is generally the maximum value of q.

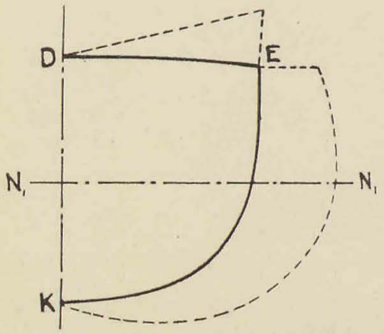


FIG. 10.

Fig. 10 gives a general idea of the distribution of shearing stress in a single-deck ship when the plate thickness is uniform. The stresses are marked off normal to the contour of the section. It is seen that the shearing stress decreases from its maximum at the neutral axis as we go up or down, reaching zero value in the center-line at D and K.

3. **Actual Ships.**—Consider now a large warship, as represented by the diagram fig. 11, provided with double bottom, several decks, longitudinal bulkheads, and constructed of plating of different thicknesses. Here the distribution of the stresses cannot in all cases be determined accurately.

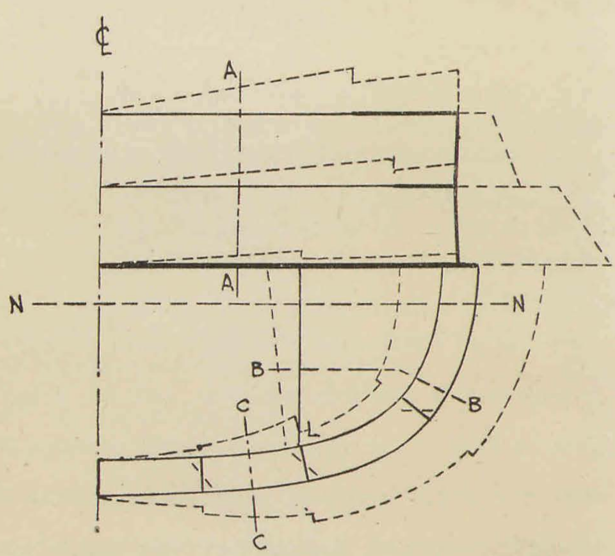


FIG. 11.

For a section like AA taken normal to the decks above the neutral axis, the shearing stress may be found for each deck separately, using formula (5), as in the single-deck ship. The conditions for using this method are that the decks shall be sufficiently stiff and well supported, but longitudinally independent of each other inside (to the left of) the section AA.

For a section like CC taken inside the side bulkhead, normal to the inner and outer shell, the case is more complex, because the longitudinals transmit a certain amount of shearing between the inner and outer shell. A solution may be found, as proposed by Professor Dr K. Suyehiro * of the Engineering College, Tokyo, by equating the moment of the shearing forces about the neutral axis to that of the normal pressures, acting on the elemental section enclosed between CC and the center-line plane.

Let the suffixes 1 and 2 refer to the inner and outer bottom respectively at the intersection with CC. Taking moments about NN we obtain:

$$\Delta[\int yptds] = y_1 q_1 t_1 \Delta x + y_2 q_2 t_2 \Delta x$$

or

$$\frac{Qi}{I_0} = y_1 q_1 t_1 + y_2 q_2 t_2$$

where i is the moment of inertia about the neutral axis of the sectional area between CC and the center-line plane. This equation, together with the equation for equilibrium in a horizontal direction

$$\frac{Qm}{I_0} = q_1 t_1 + q_2 t_2$$

will furnish the value of the shearing stresses q_1 and q_2.

For a section like BB normal to the longitudinal bulkhead and normal to the inner and outer shell, we can determine the average shearing stress from formula (5) using Σt the aggregate thickness of the plating, instead of t while the value of m is, as usual, the moment about the neutral axis of the entire sectional area below BB. The exact determination of the stress on each of the three thicknesses intersected by BB is not possible, since we have only two equations for the three unknown stresses. The distribution of the stresses will depend on the internal elastic properties of the structure. The dotted lines on fig. 11 which indicate the magnitude of the shearing stresses must, therefore, be taken merely as an illustration of the general distribution of the stresses.

4. **Shearing at the Turn of the Bilges.**—Dr Suyehiro, in the paper referred to above, has given a diagram with calculated shearing stresses for the section of a merchant vessel with double bottom extending to the turn of the bilges. He points out that in ships of this construction, where there are no longitudinal bulkheads, the bilge strakes have to resist the entire shearing force induced into the sides (the web) from the heavy and rigid double bottom. He thus explains the signs of weakness which so frequently appear in such vessels in the landings of the bilge strakes and in the tank side angles—a defect which has generally been attributed to other causes.

* "On Shearing Stress in a Ship's Structure," *Jap. Soc. Nav. Arch.*, 1912.

In warships these conditions rarely exist. Wherever a double bottom is fitted, there are longitudinal bulkheads, and in large ships the inner bottom is carried up beyond the turn of the bilges to the armor shelf, whereby the strains on the bilge strakes are relieved. If, however, light cruisers should be built with double bottom to the turn of the bilges but without side bulkheads, this form of weakness ought to be provided against by giving ample strength to the bilge strakes and their seam connections, and by bracketing the frames above the double bottom very efficiently to the margin longitudinals.

7. WRINKLING OF THE WEB.

1. **Origin and Nature of Wrinkling.** — In thin-webbed plate-girders the diagonal compressive stresses, which exist in the web due to shearing, will tend to produce buckling. This action is strongest at the neutral axis, and if the plating is unstiffened, it is liable to take place at very low stresses, resulting in one or more wave-like deformations extending obliquely from flange to flange of the girder. If "web-stiffeners" are fitted, capable of supporting the plating effectively, the length of the "waves" will be much shortened, since they will then extend only from stiffener to stiffener. Buckling of this nature, consisting of a number of wave-like deformations, is called "wrinkling." With closely spaced stiffeners and in unstiffened girders of small depth it will not occur till the stresses are very great, and only if the plating is relatively light.

Wrinkling constitutes a problem which is not yet fully elucidated, but, on the basis of the experimental studies which, so far, have been made and published,* an approximate solution may be given.

Fig. 12 shows the wave-formation in an experiment made by Professor Lilly on a girder with unstiffened web, supported at both ends and loaded at the middle with a concentrated load. The buckling must have commenced directly under the load where the compressive stress was a maximum. The lines of compressive stress were here vertical, whence the wave at this point was horizontal with its crest about at mid-height of the web. Immediately outside this region the lines of principal stress curve over and soon form an angle of 45° with the neutral axis, causing the central wave to bend upwards on both sides of the middle and all the other waves to be inclined at an angle of about 45°, extending from flange to flange. As appears from the diagram, the lower end

* W. E. Lilly, *Engineering*, 1907, vol. lxxxiii. p. 136; J. Stieghorst, *Schiffbau*, 1902–03, vol. iv. p. 263; F. Pietzker, *Festigkeit der Schiffe*, 1911, p. 35.

of each wave was, very approximately, vertically below the upper end of the preceding wave, whence the wave-length was $\frac{h}{\sqrt{2}}$ where h is the free height of the web. In other experiments, where stiffeners were fitted with a spacing equal to h the waves extended diagonally from one corner to another of each section, and such was the case also when the spacing of the stiffeners was not exactly equal to the depth of the web.

In a ship-girder we may look for this action in particular where the outer shell is not assisted in its function as a web by longitudinal bulkheads or by an inner bottom, and where it is not itself properly stiffened. Let us consider a torpedo-vessel without any longitudinal bulkheads.

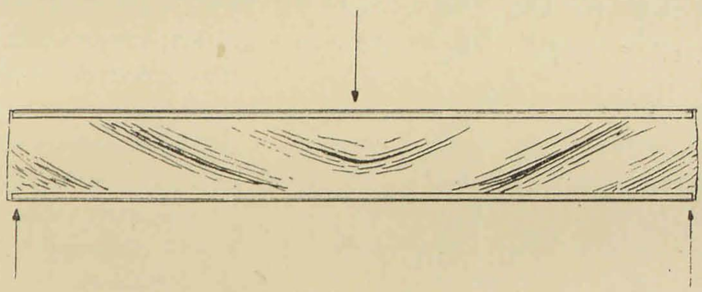


FIG. 12.—From Professor W. E. Lilly's experiments, *Engineering*, 1st Feb. 1907. Courtesy of the Editor of *Engineering*.

The web is here represented by the sides, and the flange strength is concentrated in the deck and the bottom structure. Shearing will exist with small variation in intensity over the entire height of the sides between the lower bilge strakes or the garboard strakes and the sheer strakes. The case differs from that of plate girders used in bridges, in that the spacing of the web stiffeners—here the frames—is much closer relative to the height of the web. Although no failure of this kind is on record in ships, it is of interest to examine what form the phenomenon must take when it occurs.

Suppose the vessel under consideration subject to hogging and consider the shell in a frame space such as FFFF (fig. 13) on about a quarter of the length from the bow. When the shearing stress reaches a certain limit, the plating at the neutral axis will be unable to resist the compressive stress, the effect of which is augmented by the tensile stress acting at right angles to it. A wave-shaped buckling takes place, comprising a wave-ridge A_0A_0 and two wave-hollows, all inclined at $45°$ to the neutral axis and extending from frame to frame. If the stress is increased, the wave-hollows will induce further wave-formation above and below A_0A_0. Judging from experiments, we may expect the wave-length at the neutral axis to be

$$l = \frac{s}{\sqrt{2}}$$

where s is the frame space.

The general distribution of the stresses will be as on Pl. I., where two frame stations FF, FF are marked off on fig. A. Since the waves will place themselves normal to the principal compressive stress, we may expect that below the neutral axis, where the principal compressive stresses become more and more horizontal as the lower flange (the keel) is approached, the waves will place themselves more and more vertically, until in the flange itself they will be in a transverse plane. Above the neutral axis, the waves will place themselves more and more parallel with this axis, as the distance from it is increased. Since the compressive stress here decreases rapidly, the wave-formation will be weaker and weaker, and at the sheer strake, which is chiefly subject to tension, it will have vanished.

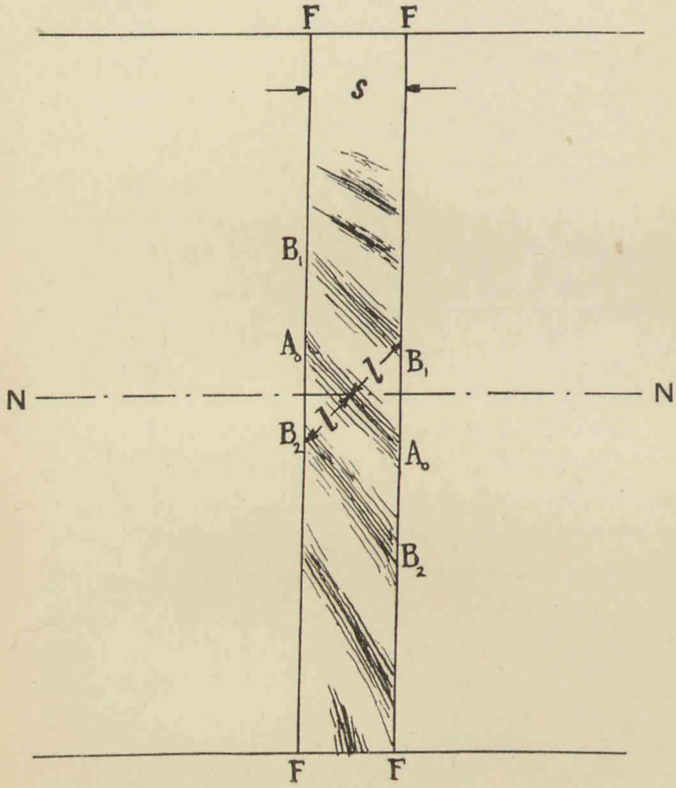


Fig. 13.

Wrinkling in bulkheads, fitted with a pure system of vertical stiffeners, must take the same form as in the sides. In all cases the best remedy is to fit light secondary stiffeners—hold stringers on the sides and horizontal bars on the bulkheads.

2. **Calculation of the Wrinkling Stress.**—In order to determine the minimum shearing stress at which wrinkling is likely to take place,

consider a strip of plating of unit width, normal to the waves B_1 and B_2 (fig. 13) at the neutral axis. The strip is subject to a compressive compound stress, equivalent to a virtual stress of $1 \cdot 3q_0$ as explained in SECTION 8, *1*. The whole strip spans over two waves, but we shall deal with one wave-length only, $l = \frac{s}{\sqrt{2}}$ extending across wave A_0 from wave-hollow to wave-hollow. Such a strip may be regarded as a column fixed at the ends, and hence the critical stress at which wrinkling is likely to take place may be determined from Euler's formula (56) or from the curve in fig. 49. The stress so obtained may be used as a means of comparison between an existing ship and the design. The factor of safety, *i.e.* the ratio between the critical stress and the actual compressive stress, $1 \cdot 3q_0$ is found from the existing ship, and the ratio of frame space to thickness of plating is adjusted in the design so that the same factor of safety is obtained.

In order to obtain an idea, however crude, of the maximum allowable spacing of frames or stiffeners, whether horizontal or vertical, by which wrinkling will be precluded without having recourse to fitting of secondary stiffeners, we shall assume a maximum shearing stress, $q_0 = 4$ ts. per sq. in., and a factor of safety of 3, applied to Euler's value for the critical stress. We have then the virtual compressive stress $1 \cdot 3q_0 = 5 \cdot 2$ ts. per sq. in., and the critical stress $3 \times 5 \cdot 2 = 15 \cdot 6$ ts. per sq. in., corresponding to which Euler's formula (curve, fig. 49) gives $\mu = 53 = \frac{l}{t}$. Hence $s = l\sqrt{2} = 1 \cdot 414 \times 53t = 75t$ but making allowance for the support which the plating receives from the rivets and the flanges of the frames it seems safe to reckon $s = 80t$. We conclude, therefore, that *in vessels such as destroyers, where high shearing stresses may exist in certain parts of the shell plating, the shortest frame space should not, in those parts, exceed about eighty times the thickness of the plating.* The same rule applies to bulkheads and other plate girders in ships, provided the working stress is the same as assumed for the shell plating. It was shown above that a spacing of $80t$ was required also for longitudinals in order to prevent buckling in plates subject to a simple compressive stress. Hence, where buckling is thus prevented, wrinkling is not liable to occur either.

Where shearing stresses are expected to be greater than the limit assumed above, as, for instance, in stiffeners that are to support elastic bulkheads, discussed in SECTION **70**, *2*, the wave-length of the wrinkles and hence the maximum spacing of the web-stiffeners must be calculated independently. Examples on this problem are given in SECTIONS **22** and **70**.

In the *Wolf* experiments the maximum shearing stress was $q = 4{\cdot}23$ ts. per sq. in. The frame spacing was 20 in. and the plating was ·172 in. (7 lb.), whence $s = 116t$.

We have $l = \dfrac{s}{\sqrt{2}} = \dfrac{20}{1{\cdot}414} = 14{\cdot}1$ in. and $\dfrac{l}{t} = \dfrac{14{\cdot}1}{{\cdot}172} = 82$, corresponding to which Euler's formula (56) gives a critical stress $p = 6{\cdot}6$ ts. per sq. in. Since the virtual compressive stress was $1{\cdot}3q_0 = 1{\cdot}3 \times 4{\cdot}23 = 5{\cdot}5$ ts. per sq. in., the factor of safety is only $\dfrac{6{\cdot}6}{5{\cdot}5} = 1{\cdot}2$. This seems to show that the plating of the *Wolf* must have been very near buckling at the point where maximum shearing occurred, viz. at the outer edge of the cradles on which the boat rested, and where the chocks ended abruptly.

3. **Bulging of the Entire Side.**—In torpedo-boats with strongly rounded sides, with light frames, and especially when no longitudinal bulkheads are fitted, the frames may be incapable of localising the wave-formation as supposed in the foregoing. The buckling will then take larger proportions and may span the entire chord between the deck edges and the head of the floors, the frames themselves taking part in the deformation. This action, here referred to as "bulging," is most likely to occur in the bow on about a quarter length. As a result, the depth of the ship-girder will be reduced, the stresses in the flanges increased, and rupture or collapse of the structure may follow. This mode of failure is one to which so far small attention has been given, although it seems likely that it has contributed to certain breakdowns which have occurred in torpedo-boats. The most effective remedy is to fit longitudinal bulkheads, but also transverse bulkheads, deep belt frames, and stanchions will prevent bulging.

4. **Plate Girders in Bridge Design.**—Wrinkling is always considered by structural engineers in the design of plate girders, and is guarded against by fitting stiffeners on the web. According to the rules of the American Railway Engineering Association for the construction of bridges, there shall be web stiffeners, generally in pairs, over the bearings, at points of concentrated loading, and at other points where the thickness of the web is less than one-sixtieth of the unsupported distance between flange angles. The distance between stiffeners shall not exceed that given by the following formula, with a maximum limit of 6 ft. and not greater than the clear depth of the web:

$$s = \frac{t}{40}(12000 - q) \qquad . \quad . \quad . \quad . \quad (7)$$

where s and t are in inches, q in pounds per square inch. Several other formulas are used by structural engineers, and all are based on the assumption that the length of the elemental strip of plating, considered as a column with fixed ends, that is, the wave-length, is equal to $s\sqrt{2}$ and not, as appears from the experiments and as assumed above, $\frac{s}{\sqrt{2}}$. Nevertheless the formulas of structural engineers lead to about the same result, because they include no or a very small factor of safety on Euler's (Rankine's) formula. Substituting, for instance, $q = 4$ ts. per sq. in. $= 8960$ lb. per sq. in. in (7), we find the spacing of the stiffeners, such as the frames in a ship: $s = 76t$ which corresponds well with the figure found above. It will be noticed that the greatest unsupported distance allowed between supporting edges where no secondary stiffeners are fitted is $60t$ as compared with $80t$ recommended above for ships.

8. PRINCIPAL STRESSES AND MAXIMUM SHEARING STRESSES.

1. **Formulas for Calculating the Stresses.**—The case of most practical interest in ship construction is when a simple direct stress p generally due to bending, acts in conjunction with a shearing stress q. The principal stresses are here found from the formula

$$\sigma = \tfrac{1}{2}[p \pm \sqrt{p^2+4q^2}] \qquad (8)$$

The angles which these stresses form with the neutral axis are given by

$$\tan 2\theta = \pm\frac{2q}{p}. \qquad (9)$$

The principal stresses are seen to act at right angles to each other and in opposite directions; *i.e.* one is compressive, the other is tensile. One will be greater than the other except at the neutral axis, where $p = 0$ and where, therefore, $\sigma = \pm q_0$.

The maximum shearing stresses at any point are given by

$$\tau = \tfrac{1}{2}\sqrt{p^2+4q^2} \qquad (10)$$

which determines two equal stresses acting at right angles to each other and always at 45° to the principal stresses. At the neutral axis, where $p = 0$ the maximum shearing stresses τ are equal to q_0 and act horizontally and vertically.

Each of the principal stresses produces a strain e in its own direction, and at the same time, at right angles to itself, a lateral strain of opposite sign equal to $\frac{e}{m}$ where m is a factor depending on the elastic properties of the material. Since the two principal stresses are themselves of opposite sign, the strain which each of them produces will be augmented by the lateral strain produced by the other. The resulting compound strains ϵ are determined by :

$$E\epsilon = \frac{m-1}{2m}p \pm \frac{m+1}{2m}\sqrt{p^2+4q^2} \quad . \quad . \quad . \quad (11)$$

For shipbuilding steel we may reckon $m = \frac{10}{3}$ whence we obtain

$$E\epsilon = \cdot 35p \pm \cdot 65\sqrt{p^2+4q^2} \quad . \quad . \quad . \quad (12)$$

To the strains so produced correspond fictitious simple stresses of magnitude $E\epsilon$, which will be hereafter referred to as the "virtual" principal stresses. At the neutral axis, where $p = 0$ and $q = q_0$ the virtual stresses will be : $\pm 1 \cdot 3q_0$.

2. **General Distribution of the Stresses.***—Figs. A to H on Pl. I. illustrate in a general way the distribution of the stresses in the shell and deck plating of a single-deck ship, and show the effect on the lines of stress of the presence of a deckhouse and of a longitudinal side bulkhead. The ship is supposed to be in the hogging condition. The diagrams are not based on a numerical calculation of a definite case, but conform approximately to the laws which govern stresses in general and to the particular conditions of the assumed state of bending.

The direction of the principal stresses and of the maximum shearing stresses is indicated by the direction of the lines, but it is to be observed that the natural flow of the lines indicated on the diagrams will exist only provided the material is properly distributed. Consider, for instance, the lines of principal tensional stress in the deck (fig. C). If some of the strakes of the deck are unduly heavy, others unduly light, a deflection of the lines towards and through the heavy strakes will take place and the diagram of stresses will not be the simple and ideal one represented by fig. C. An adjustment of the stresses will take place until at every section the longitudinal components of the strains correspond to the distance of the deck from the neutral axis, conforming to the elastic bending of the ship-girder at the section. Due to the deflection of the lines and their crowding together at certain points, greater stresses and strains will there come to exist.

* J. B. Chalmers, *Graphical Determination of Forces in Engineering Structures*, 1881 ; P. Jenkins, *Inst. Nav. Arch.*, 1890 ; J. Bruhn, *Inst. Nav. Arch.*, 1899 ; K. Suyehiro, *Engineering*, Sept. 1, 1911, and *Jap. Soc. Nav. Arch.*, 1912.

3. **Principal Stresses in the Sides.**—Fig. A shows the principal stresses, tensile and compressive, in the sides of the ship and of the deck-house. The lines of tensile and compressive stress run everywhere at right angles to each other and cross the neutral axis under an angle of 45°. On the upper part of the side the lines of tensile stress curve over towards amidships and predominate on half length where they run parallel with the neutral axis and with the deck line. Towards the ends these lines are seen to meet the deck edge at an angle. This is due to the obliquity of the sides relative to the center-line plane and the existence of the deck, which, so to speak, attracts and absorbs the lines of stress in these parts of the ship, causing them to climb over the deck edge, whereafter they continue along the deck in a fore-and-aft direction

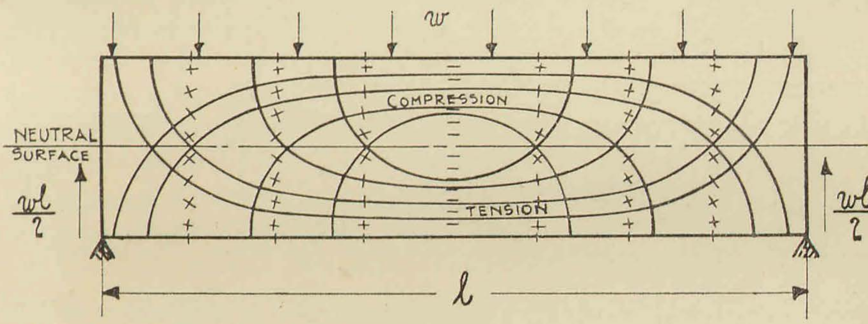


FIG. 14.—Solid Rectangular Beam.

as shown in fig. C. On the lower part of the sides and under the bottom the lines of tensile stress curve down and inwards to meet the keel line at right angles.

The lines of compressive stress are seen to curve upwards from the neutral axis, cross the deck edge at a great angle, which is 90° at amidships, and run across the deck, intersecting the center-line at right angles. In the lower part of the side they curve over towards amidships and arrange themselves similarly to the lines of tensile stress in the upper part, but the majority of the lines of compressive stress dive down round the turn of the bilges and continue under the flat bottom in a fore-and-aft direction. No special diagram is given for the bottom, because it would be quite similar to that of the deck.

For comparison we give in fig. 14 a diagram of the calculated lines of stress in a solid rectangular beam supported at the ends and loaded uniformly. It is reproduced with the kind permission of Professor Arthur Morley from his work on *Strength of Materials.**

* Published by Messrs Longmans, Green & Co., London, 1908.

4. **Principal Stresses in the Deck.**—Fig. C shows the lines of principal stress in the deck in the absence of a deckhouse and of longitudinal bulkheads. In order to explain this diagram, let us consider first some simple cases. In a solid rectangular beam, subject to the same bending action as a ship, all the shearing will take place in vertical planes and the lines of principal stress on the upper face will run parallel with the center-line for the entire length of the beam. In a hollow rectangular box-girder the action of the sides on the flanges will be one of longitudinal shear, which will cause longitudinal strains and stresses in these latter along the line of attachment. These strains will produce shearing in the adjacent inert material of the flanges, causing the lines of principal stress to be deflected inwards at an angle approaching 45°. Gradually, as indicated in fig. 15, these lines curve over to a direction parallel with the center-line and cross the middle section at a distance from the side.

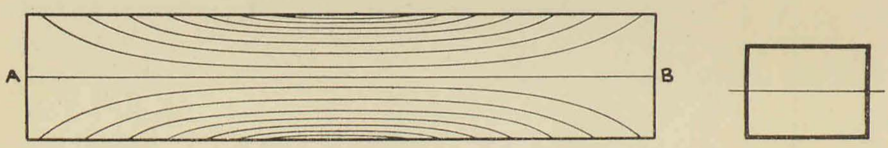


FIG. 15.—Hollow Rectangular Box Girder : Approximate Lines of Stress in Upper Flange.

Thus the shearing action enables the inner portions of the flanges to take part in the work of the girder.

A ship differs from a rectangular box-girder chiefly in that the sides converge towards the center-line at the ends. The action along the deck edges will also here be mainly one of pure shearing, although with deep and stiff framing a certain outward pull may be exerted by the sides on the deck. The resultant pull on the edges of the deck will, therefore, probably act at a small angle with the tangent to the contour of the deck. The longitudinal components of this pull will call forth longitudinal stresses in the deck amidships, creating or tending to create straight lines of longitudinal tensile stress throughout the deck. The transverse components which are of small magnitude will be neutralised by the transverse components on the opposite side of the ship, causing light compressive stresses directed across the deck normal to the lines of tensile stress, intersecting the center-line at right angles. As in the rectangular box-girder, the lines of tensile stress, when starting from the deck edges, will be deflected inwards, but since the sides converge towards the center-line, this deflection will here be far less pronounced. Along the central portion of the deck the lines of stress are deflected by the hatches, at the corners of which they run close together, causing increased local stresses.

5. **Principal Stresses in a Deckhouse and its Influence on the Distribution of the Stresses in the Decks.**—If a deckhouse is of substantial construction, and if it is well connected to the deck, it will form a rigid girder strongly resisting the elongation (and contraction) of the deck. Great shearing will come to exist along the lines of attachment, increasing in intensity towards the ends of the deckhouse (see fig. F). The action will be the reverse of that which took place between the sides of the ship and the deck. The lines of tensile stress will here, as shown in fig. E, be deflected towards and absorbed by the deckhouse, which will thus to some extent relieve the deck of the tensile stresses; but the greater the length and stiffness of the deckhouse, the more violent will be the deflection of the lines of stress, and the more likely it is that excessive local stresses will come to exist at the ends of the deckhouse, in particular at the corners.

The arrangement of the lines of stress on the sides of the deckhouse is shown in fig. A. It is analogous to that in the upper part of the sides, and since the action at the line of attachment is one of pure shearing, the lines of principal stress in the deck as well as in the deckhouse will all meet the line of attachment at an angle approaching 45°. Again, some of the lines will climb over the edge of the deckhouse and cause tension in the deck of this latter. The more yielding the deckhouse, the fewer lines will it absorb; if expansion joints are fitted in sufficient number, it will follow the elongation of the deck practically as an inert structure, and will not influence the distribution of stresses perceptibly.

On the side of the deckhouse are shown a window and a door, illustrating the disturbing effect of these openings on the natural flow of the lines of stress, which are crowded together at the corners of the openings.

6. **Effect of a Longitudinal Bulkhead.**—A longitudinal bulkhead, efficiently connected to the deck and to the bottom and continuous for more than half the length of the ship, will play the same part as the sides, acting as an additional web of the ship-girder. Such a bulkhead will produce in the deck a system of stress lines on each side of its line of attachment similar in nature to those produced by the sides (see fig. G). The transmission of forces between the bulkhead and the deck is, as in the case of the deckhouse, effected entirely by shearing, whence the lines of principal stress will also here meet the line of attachment at an angle of about 45°.

The sides of the ship will be relieved in the same measure as the longitudinal bulkheads are strained. The total tensional pull on the deck amidships will be somewhat smaller than where no bulkhead is fitted and the stresses will probably be better distributed, but at the ends of the bulkhead they are apt to become excessive. It is therefore desirable to continue such bulkheads as far forward and aft as possible. At the ends of very long ships they should taper off to deep girders under the deck, carried so far as to connect with the sheer strakes.

7. **Maximum Shearing Stresses.**—The lines of maximum shearing stress are shown in figs. B, D, F, and H. There are two sets of lines running at right angles to each other and at 45° to the lines of principal stress. In the sides, as shown in fig. B, they converge towards the neutral axis, where the shearing stresses attain their greatest intensity on about one-quarter of the length of the ship from the ends. Amidships the lines of shearing stress are inclined at 45° to the neutral axis, but at this axis the stresses are zero.

On the deck the shearing is greatest near the sides on one-quarter length, as shown in diagram D; here one set of the lines of shearing stress runs tangentially into the deck edge, while the other set stands normal to it. On the half length of the ship the shearing is simply a result of the direct stress and acts at 45° to the center-line. The lines of shearing stress everywhere cross the center-line at an angle of 45°, but the intensity is here zero.

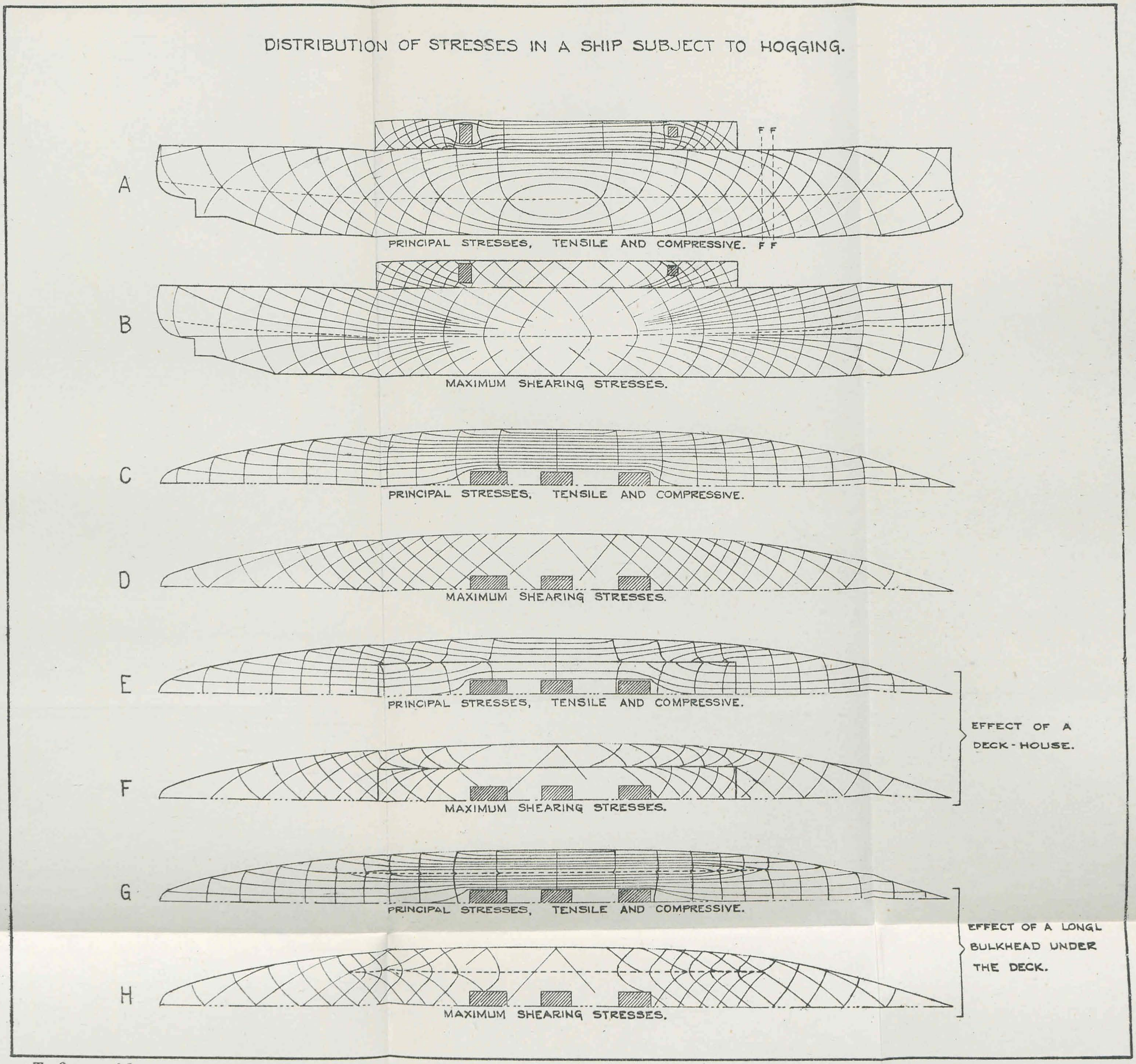
In way of a deckhouse or a longitudinal bulkhead one set of lines runs tangentially into the line of attachments at its ends. (See figs. F and H.)

9. DEFLECTIONS AND STRAINS.

1. **Calculation of the Vertical Elastic Deflections.**—The elastic deflection of a ship-girder is due primarily to bending, but also partly to shearing. The deflection by bending may be calculated from the formula

$$y = \iint \frac{M}{EI} dx dx \quad . \quad . \quad . \quad . \quad (13)$$

The integration is most conveniently performed graphically. Assuming E to be constant, calculate the value of $\frac{M}{I}$ for a number of points at different sections of the ship. Place the origin O at one end of the ship, and construct a curve for $\frac{M}{I}$ such as ORA in fig. 16, with the length of the ship OA as a base. Integrate this curve twice with an integraph. The resulting curve OP′A′ gives the vertical deflection relative to OA, the tangent to the neutral axis at O. Join the end points of the curve of deflection by a straight line OA′ then the intercepts between the curve



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of deflection and this line, measured vertically, will represent the actual vertical deflections. For instance, at any point P the deflection will be y_B = P′P″

In determining the moment of inertia it is recommended to follow the same rules as in the calculation of longitudinal strength, neglecting the rivet holes but allowing for buckling of light unsupported plating. Evidently the deflection depends on the average stiffness of the hull, which cannot be perceptibly influenced by the single lines of rivet holes which exist along the frames, while buckling of the plating, especially

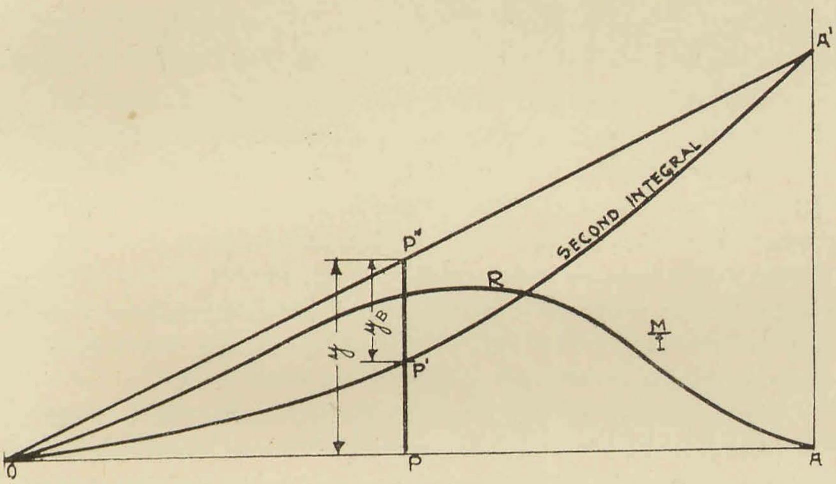


FIG. 16.

when it occurs in several strakes and over a number of frame spaces amidships, may seriously reduce the moment of inertia and thus increase the deflection.

The deflection due to shearing is found from the formula

$$y_s = \int \frac{[\iint q^2 dy dz]}{GQ} dx \quad . \quad . \quad . \quad . \quad (14)$$

where G is the coefficient of rigidity. The double integration inside brackets extends all over the sectional area, OY being the axis of vertical ordinates and OZ the horizontal, transverse axis of abscissæ, which coincides with the neutral axis of the section. The final integration extends longitudinally the entire length of the ship along the axis of OX. It is clear that the calculation will be, generally, very laborious, since the expression inside brackets has to be evaluated for a number of sections, and since the shearing stress q must be determined for every plate and girder on each section. In large ships of complex construction it will not be possible even to find the shearing stress on every member, and the method is therefore applicable only to vessels of simple construction such as torpedo-boats. The final integration is easily performed graphically.

A simple approximate solution can be obtained by assuming the shearing to be uniformly distributed over the web, disregarding the flanges. We have then

$$y_s = \int \frac{Q}{GA_w} dx \quad . \quad . \quad . \quad . \quad . \quad (15)$$

where A_w is the total sectional area of the members which constitute the web, namely the vertical or nearly vertical parts of the outer and inner shell and the continuous longitudinal bulkheads. The deflection will be somewhat overestimated by this method. Ordinarily the deflection due to shearing is much smaller than that due to bending and may be neglected, but where the unsupported length of the ship-girder is short relative to its depth, and where scientific accuracy is required, it should be taken into account. If, for instance, it is desired to study the stiffness of the ends of a ship overhanging the keel blocks when in dock, the deflection due to shearing should be included as part of the total deflection.

2. **Observations of the Vertical Deflections.**—Such observations are of great interest, since, by comparing the measured and the calculated deflections, a check is obtained on the assumptions on which the strength calculations rest. Measurements of the deflections have been made on ships when they were launched, when they were docked, and in a seaway; also the effects of loading and the influence of temperature have been observed. The most accurate and complete experiments of this kind are those on the British destroyer *Wolf*, which were exhaustively analysed and should be carefully studied by all students of this question. Careful measurements of the deflections, accompanied by an analysis, have been made also by Mr T. C. Read and Mr G. Stanbury on two merchant steamers.*

That the deflections, within the elastic limit, may be very considerable was evidenced by measurements made by Naval Constructor S. F. Smith of the United States Navy† on three 500-foot colliers, the *Neptune*, *Orion*, and *Jason*. In the *Neptune*, which was built on the ordinary transverse system of framing, a deflection of the deck amidships of 6½ inches relative to the ends was caused by loading the ship. In the *Orion* and *Jason*, built on the Isherwood system, the deflections were

* *Inst. Nav. Arch.*, 1894.

† *Am. Soc. Nav. Arch. Mar. Eng.*, 1913.

about $3\frac{5}{8}$ inches and 5 inches respectively. It was found, moreover, that a rise of temperature of only 7° caused a deflection of 1 inch.

3. **Comparison between the Observed and Calculated Vertical Deflections.**—In all cases where such comparison has been made, the actual deflections have been found greater than the calculated. This discrepancy may be due to several causes :

(1) Slipping of the riveted joints. A ship is not, as assumed in the calculation of the deflection, a homogeneous elastic structure. Long before the elastic limit is reached, perhaps already at shearing stresses in the rivets of 6–7 ts. per sq. in., a slipping will take place, beginning at the most strained part and gradually spreading as the strains increase.

(2) Buckling of the plating between beams and frames caused by excessive compressive stresses.

(3) Bulging of the sides, deck, or bottom over large areas between the transverse bulkheads due to insufficient stiffness of the structure as a whole.

(4) Elastic deflection due to shearing.

(5) Imperfect workmanship. Sometimes increased deflections may be caused by poor riveting, by strains due to improper fitting of the different parts of the structure, or by initial buckling of the plates.

Which of these causes will predominate in augmenting the deflection beyond the calculated amount will depend on the construction of the ship and on the conditions under which the experiment takes place. In any case it seems rational to make a deduction for the ineffective sectional area of thin and unstiffened plating in compression in calculating the moment of inertia, and in certain cases to take into account the deflection due to shearing. By applying these corrections, the discrepancy, here discussed, will probably in most cases disappear ; but, should the calculated deflections still fall short of the observed deflections, equality may be secured by assuming a reduction in the modulus of elasticity. This reduction must depend chiefly on slipping of the riveted joints and will, therefore, vary with the intensity of the stresses. Hence the apparent modulus of elasticity, so obtained, is not a constant quantity.

4. **Strain Measurements.**—The strains may be measured locally by means of a "strain-indicator," such as that devised by Mr C. E. Stromeyer * and used in the *Wolf* experiments. The strain-indicator gives the linear strain over a certain length, in the *Wolf* 20 inches (one frame space), and if the modulus of elasticity is known, we can thus determine the stress at every point where the strain is measured :

$$p = \mathrm{E}e \quad . \quad . \quad . \quad . \quad . \quad (16)$$

* *Inst. Nav. Arch.*, 1886.

where e is the strain per unit length. Comparing this stress with that found by the usual longitudinal strength calculation, we obtain a check on this latter. If the measured stress, taken over a certain length of solid plate, is greater than the calculated stress, it shows either that other members of the structure have failed, generally by buckling, throwing thus greater strains on the remaining members, or, when the measurement is taken on a thin plate in compression, the discrepancy may be due to buckling of this plate itself between the points on which the indicator is fixed. If a riveted joint is found between these points, the discrepancy may be due to frictional slip or bending of the joint, as evidenced by measurements taken by Mr James E. Howard on s.s. *Ancon*,* where the strains across the butt lap joints of the deck plating were much greater than on the solid plate.

An interesting application of the strain-indicator was made by Sir John Biles in the *Wolf* experiments. Measurements were taken at different points all around a certain section of the boat, and the stresses so obtained were used to determine the elastic moment of resistance of the ship-girder at this section by means of the formula

$$M = \int py\,dA$$

where dA is a horizontal element of the sectional area, and where the integration extends over the whole section. The moment of resistance so obtained was compared with the bending moment, which could be accurately determined from the known conditions of loading.

* *Am. Soc. Nav. Arch. Mar. Eng.*, 1913, Pl. 71 and 72.

CHAPTER III.

TRANSVERSE STRENGTH.

IN most battleships and cruisers the transverse strength is amply provided for by bulkheads, and need not be examined theoretically except for broad vessels in the condition of docking. In torpedo-vessels other than submarine boats, here referred to as "torpedo-vessels," the question of transverse strength requires to be looked into, but in submarine boats it is vital and deserves a full theoretical and experimental study.

10. TRANSVERSE STRENGTH OF A SHIP IN DOCK.

1. **General Transmission of the Load through the Structure.**— The strength of the main transverse bulkheads, considered as girders, is in a large warship enormously great compared with that of the transverse frames, which are almost invariably intercostal in the bottom. We may, therefore, in general disregard the frames and assume as an approximation that only the bulkheads are effective in resisting transverse bending and shearing.

Let us examine how the weights of a ship are transmitted through the structure when in dock. The armor and other weights supported by the side above the armor shelf are transmitted through the side structure behind armor, which constitutes a deep and strong girder resting on the extreme corners of the main transverse bulkheads in the hold. The tops of the contiguous watertight floors, which may be regarded as part of the bulkheads, abut against the armor shelf and form shoulders on which the side structure rests. The weight of the decks, and all that is supported by the decks, is through sides, pillars, and minor bulkheads transmitted to the main transverse and longitudinal bulkheads in the hold. Coal in the lower side bunkers and other weights placed near the sides in the hold are supported by the framing of the bottom and thence transmitted to the bulkheads. The weights that are located near the center-line are either directly resting on the keel structure, and hence on the keel-blocks, or they are transmitted through bulkheads and stanchions without causing any appreciable transverse strains, but they rarely exceed more than about 25 per cent. of the total weight of the ship.

When a ship is docked on both center and side keels, the transverse bulkheads are supported on three points and will not be severely strained. Longitudinal side bulkheads, if placed immediately over the side-keels, as they should be, will be the main transmitters of the load to the keel-blocks. They will probably take the greater part of the load on the main transverse bulkheads, and practically the entire load on the partial transverse wing bulkheads, and transmit them to the keel-blocks. In general it may be said that where side docking keels are fitted and properly located relative to the longitudinal bulkheads, the transverse strains in the ship when in dock will be moderate. A problem of particular importance in modern battleships with central gun-turrets is to transmit to the keel-blocks the enormous and concentrated load of the double turrets with ammunition placed forward and aft. This is best effected by fitting transverse and longitudinal bulkheads directly under the barbettes. There should be one longitudinal bulkhead in the center-line and one on each side, and special docking keels should be worked directly under the latter.

When a ship is docked on the center-keel only, by far the greater part of the weight must be transmitted to the keel through the transverse bulkheads, which, in large and broad vessels, will, therefore, be subject to strong bending and shearing. The longitudinal bulkheads may here be regarded simply as subsidiary girders transmitting the load to the transverse bulkheads. It is, generally, sufficient to examine the strength of the bulkheads amidships, where the load is greatest, and where the bulkheads are more widely spaced and of greater breadth than elsewhere.

2. **Strength Calculation of a Transverse Bulkhead when a Ship is Docked on the Center-Keel only.**—A main transverse bulkhead amidships, generally between two boiler-rooms, is selected, and the load is apportioned to it. Since the distribution of the load must be largely a matter of judgment, it is of importance that analogous assumptions are made in the cases to be compared. Considering the

bulkhead as a plate-girder, supported at the middle and free at the ends, the bending moment can be determined by a graphical process similar to that used for the ship-girder.

The moment of inertia is determined for a section in the center-line plane, where the bending is a maximum. In this calculation should be included the transverse floor under the bulkhead and a strip of the inner and outer shell plating equal to thirty times the thickness (SECTION 15). If a bulkhead, extending from side to side of the ship above the protective deck, is fitted directly above the main bulkhead, it should be taken into account, but if partial it may be neglected. The decks which are directly connected to the bulkhead, *i.e.* generally the protective deck and in some ships the second deck, should like the shell enter into the calculation with a strip of plating equal to thirty times their thickness. The support which the bulkhead may receive through the sides from the upper decks, not connected with the bulkhead, may be neglected, being indeterminate and of small magnitude. When the moment of inertia is found, the stresses can be determined in the usual way. The method is necessarily crude, but may be of guidance in the design of bulkheads and their attachments, as well as in deciding how the ship should be supported in dock.

Example.—A battleship of 16,300 ts. displacement is docked on the center-keel only. Find the maximum stress in a transverse bulkhead extending to the protective deck between two boiler-rooms. An 8-in. wing-turret is placed almost directly over the bulkhead on each side (Pl. II.).

In estimating the load on the bulkhead, all the weights in the ship on one side within the length of one boiler-room were included, but a deduction of 25 per cent. was made in the hull group and in the weight of the protective deck to allow for the load transmitted directly to the keel through the center-line bulkhead. The distribution of the weights was estimated as follows :—

Hull and permanent fittings, equipment, and stores .	189 ts.
Protective deck	51 ,,
Two boilers with uptakes, etc.	108 ,,
One 8-in. wing-turret complete with supports . .	162 ,,
Side armor	113 ,,
Coal	197 ,,
Total load on one side of the bulkhead . . .	820 ts.

The curve of loads was constructed as shown on Pl. II., and the curves of shearing force and bending moment were obtained by integraph.

We find for the section in the center-line :—

Bending moment . . . . . . $M = 19460$ ft.-ts.
Moment of inertia . . . . . . $I_0 = 18960$ sq. in. x (ft.)^2
Distance of the most strained fibre from the neutral axis (in keel plate) . . . $y = 14 \cdot 9$ ft.

Maximum stress (compressive) $p = \frac{19460}{18960} \times 14 \cdot 9 = 15 \cdot 3$ ts. per sq. in.

The moment of the sectional area above or below the neutral axis about this axis is, $m_0 = 790$ sq. in. x ft., whence the maximum shearing stress is

$$q_0 = \frac{m_0 Q}{t_0 I_0} = \frac{790 \times 820}{(\frac{5}{16})18960 \times 12} = 9 \cdot 1 \text{ ts. per sq. in.}$$

and the virtual stress, tensile and compressive, at the neutral axis is $1 \cdot 3 q_0 = 11 \cdot 8$ ts. per sq. in. The critical compressive stress at which wrinkling is likely to occur is found from Euler's formula (56), $p = \frac{44400}{\mu^2}$. The value of μ is here

$$\mu = \frac{s}{t\sqrt{2}} = \frac{48 \times 16}{5 \times 1 \cdot 41} = 109$$

where s is the spacing of the stiffeners, t is the thickness of the bulkhead plating at the neutral axis. Hence $p = 3 \cdot 74$ ts. per sq. in. Since the virtual compressive stress is more than three times as great, it seems certain that wrinkling will occur.

At a point just below the bounding bars under the protective deck, $11 \cdot 4$ ft. above the neutral axis, there is a direct tensile stress

$$p = \frac{19460}{18960} \times 11 \cdot 4 = 11 \cdot 7 \text{ ts. per sq. in.}$$

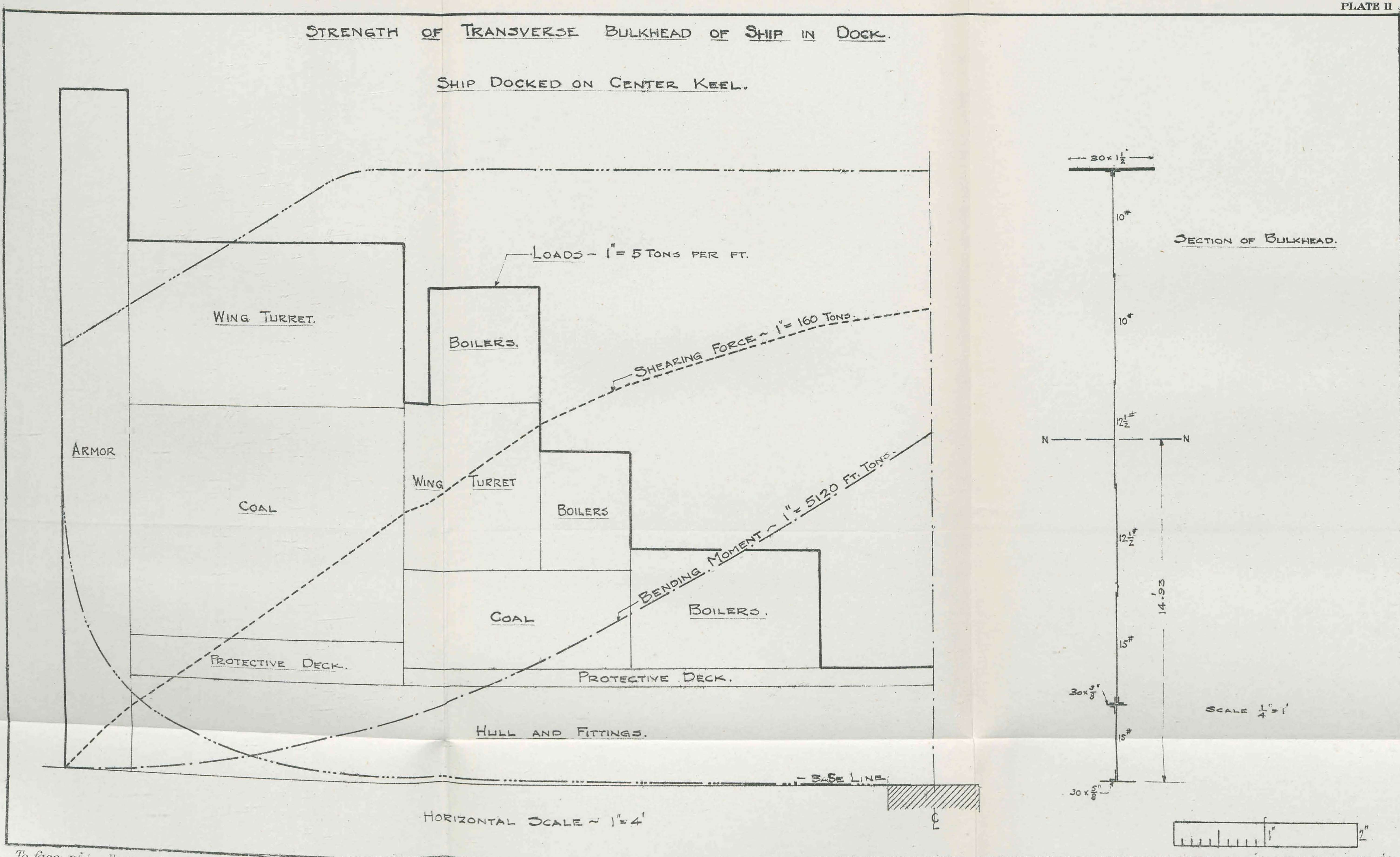
and, since the moment of the sectional area beyond this point about the neutral axis is 589 sq. in. x (ft.), the shearing stress is

$$q = \frac{589}{(\frac{1}{4})} \frac{820}{18960 \times 12} = 8 \cdot 5 \text{ ts. per sq. in.}$$

Hence the virtual tensile stress at this point is

$$E\epsilon = \cdot 35 \times 11 \cdot 7 + \cdot 65 \sqrt{(11 \cdot 7)^2 + 4 \times (8 \cdot 5)^2} = 17 \cdot 5 \text{ ts. per sq. in.}$$

Similarly, the virtual compressive stress in the bulkhead just above the bounding bars connecting it to the inner bottom is found to be $13 \cdot 9$ ts. per sq. in.



 Hovgaard; Structural Design of Warships.

E. & F.N. Spon Ltd London.

Finally, the shearing stress in the rivets in seams and butts near the neutral axis is found from the equation

$$mdtq_0 = n\frac{\pi d^2}{4}\delta^2 q$$

where md is the spacing of the rivets, n is the number of rows, δ is a factor allowing for excess in size of rivet hole above size of rivet, and q is the shearing stress in the rivets. With two lines of $\frac{5}{8}$-in. rivets, spaced 4 diameters apart, and with $\delta = 1 \cdot 1$ we find

$$q = \frac{4 \times (\frac{5}{16}) \times 9 \cdot 1 \times 2}{\pi \times (\frac{5}{8}) \times 1 \cdot 2} = 9 \cdot 7 \text{ ts. per sq. in.}$$

which, with an ultimate shearing strength of 22 ts. per sq. in., gives a factor of safety of little more than two.

It is seen that throughout the bulkhead the stresses are excessive, whence it appears that a ship of this type and size could not be safely docked on the center-keel only, even if we allow for the assistance which the bulkhead receives from the transverse frames. In fact, the ship from which these data are taken is provided with side-keels.

11. STRENGTH OF A CLOSED FRAME-RING. GENERAL MATHEMATICAL TREATMENT.

The frame-ring is assumed to be transverse and its neutral axis in one plane, in which all the straining forces act. In torpedo-vessels the ring consists of a frame on each side and a beam, often stiffened by pillars or longitudinal bulkheads. In submarine boats it is usually formed of one circular or oval frame, but in some boats the contour differs little from that of an ordinary torpedo-vessel. Pillars or longitudinal bulkheads often support the frame-ring. We shall hereafter generally refer to the frame-ring simply as the "frame."

1. **The Principle of Continuity.**—Whatever the form of the frame, the strength may be determined by an approximate method, based on an axiomatic law, which will be here referred to as the "principle of continuity." This law expresses the simple fact that *as long as the frame is not fractured or strained beyond its elastic limit at any point, the continuity of its neutral axis must remain unimpaired.*

Let the closed curve OAO (fig. 17) represent the neutral axis of a continuous frame of any form, loaded by a system of known forces to which it adjusts itself without being strained beyond its elastic limit. Take any point O as origin and the tangent OX at this point as the axis of abscissæ. Let OY normal to the curve at O be the axis of ordinates. Regard the axes of co-ordinates as fixed in space. The frame will deflect relative to the axes, except at the origin where we imagine the frame to be held "fixed," *i.e.* OX remains a tangent at O. Since the deflection is elastic there will be no abrupt change of curvature at any point and since there is no rupture, any two consecutive points will remain consecutive after the deformation. If, then, we integrate the deflections, angular and linear, all round the contour, the sum must be zero for each of them. We shall now show how this process can be used for finding the unknown internal reactions. Consider any point on the frame A the ordinates of which, before the forces were applied, were x_A and y_A while the tangent at this point formed an angle θ_A with OX. Due to the deformation the tangent will turn through an angle $\Delta\theta_A$ relative to OX and the point will be displaced through a distance Δx_A parallel with OX and Δy_A parallel with OY relative to O. The deformation is due partly to bending, partly to compression or tension and shearing, but in the cases here under consideration it is ordinarily sufficient to take only the bending into account.

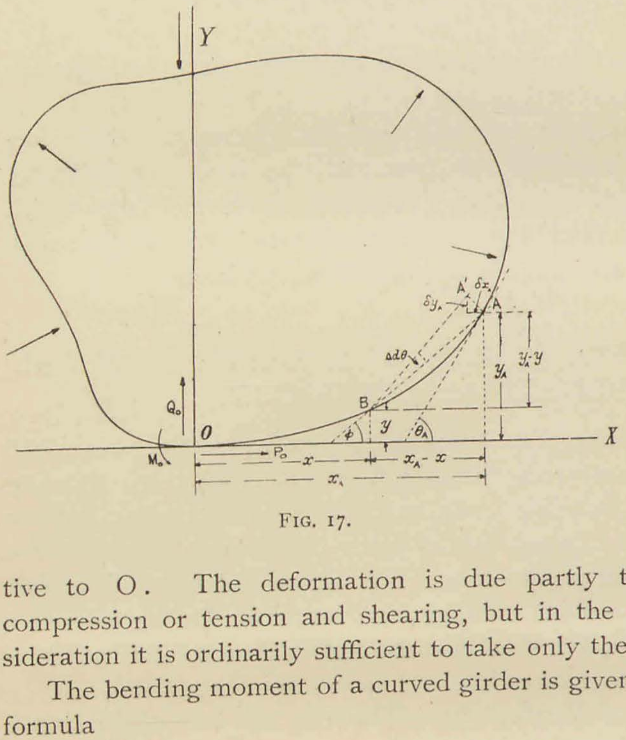


FIG. 17.

The bending moment of a curved girder is given approximately by the formula

$$M = IE\left(\frac{1}{R} - \frac{1}{\rho}\right) \quad . \quad . \quad . \quad . \quad . \quad (17)$$

where ρ and R are the radii of curvature before and after bending respectively.

Let the angular increment at any point, due to the original curvature of the frame, be $d\theta$ corresponding to an element of the girth ds (fig. 18). Then

$$\rho d\theta = ds$$

After bending has taken place, the angle $d\theta$ becomes $d\theta+\Delta d\theta$ and the element ds is changed to $ds+\Delta ds$

whence $$R(d\theta+\Delta d\theta) = ds+\Delta ds$$

Neglecting Δds the elongation of ds we obtain

$$\Delta d\theta = \frac{ds}{R} - d\theta = \left(\frac{1}{R}-\frac{1}{\rho}\right)ds$$

Substituting from (17) we find

$$\Delta d\theta = \frac{M}{IE}ds$$

whence the total angular deflection at A relative to the tangent at O is

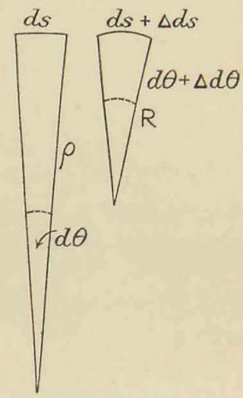


FIG. 18.

$$\Delta\theta_A = \int_0^A \frac{M}{IE}ds \quad . \quad . \quad . \quad . \quad (18)$$

The change of co-ordinates of the point A due to bending of the frame may be found approximately by a method which we shall here briefly explain. On the curve between O and A consider any point B the co-ordinates of which are x and y. Due to the angular deflection of the element of the curve at B the line BA will turn through an angle $\Delta d\theta$ and the point A will move along a circular arc to A′. Let the angle which BA forms with OX be ϕ then the linear displacements of A parallel with the axes of co-ordinates due to turning of the element at B will be

$$\delta x_A = -AA'\sin\phi \quad \text{and} \quad \delta y_A = AA'\cos\phi$$

Now, $AA' = BA\,\Delta d\theta$, $\sin\phi = \frac{y_A - y}{BA}$, and $\cos\phi = \frac{x_A - x}{BA}$

whence we obtain

$$\delta x_A = -\Delta d\theta(y_A - y) \quad \text{and} \quad \delta y_A = \Delta d\theta(x_A - x)$$

Substituting the value of $\Delta d\theta$ found above and integrating from O to A we find the total change of co-ordinates of A *

* We here neglect not only the shearing but also the tension or compression due to the direct force P which acts at any point along the tangent to the neutral axis. This latter force may be taken into account, if desired, by adding a term $\int_0^A \frac{P}{EA}dx$ to the expression for Δx_A in (19) and $\int_0^A \frac{P}{EA}dy$ to the expression for Δy_A in (20).

$$\Delta x_A = -\int_0^A (y_A - y)\frac{M}{IE}ds \quad . \quad . \quad . \quad (19)$$

$$\Delta y_A = \int_0^A (x_A - x)\frac{M}{IE}ds \quad . \quad . \quad . \quad (20)$$

If, now, we extend the integration all round the contour from O to O we shall have $x_A = 0$, $y_A = 0$, $\Delta x_A = 0$, and $\Delta y_A = 0$, and since E is a constant we obtain the following fundamental equations *

$$\int_0^O \frac{M}{I}ds = 0 \quad . \quad . \quad . \quad . \quad . \quad (21)$$

$$\int_0^O y\frac{M}{I}ds = 0 \quad . \quad . \quad . \quad . \quad . \quad (22)$$

$$\int_0^O x\frac{M}{I}ds = 0 \quad . \quad . \quad . \quad . \quad . \quad (23)$$

In frames that are symmetrical about one axis, as is usually the case in ships, it will be sufficient to integrate round half the contour. Other special conditions may exist which will reduce the extent of the integration. When the integration begins and ends at points other than the origin and outside the axes of co-ordinates, say at any points A and B, between which the integrals of the deflections are known to be zero, the equations take the form

* By interchanging the variables in the equations for δx_A and δy_A we obtain $\delta x_A = -\Delta\theta dy$ and $\delta y_A = \Delta\theta dx$

whence
$$\Delta x_A = -\int_0^A \left[\int_0^s \frac{M}{IE}ds\right]dy \quad . \quad . \quad . \quad . \quad . \quad . \quad (19')$$

and
$$\Delta y_A = \int_0^A \left[\int_0^s \frac{M}{IE}ds\right]dx \quad . \quad . \quad . \quad . \quad . \quad . \quad (20')$$

and the corresponding fundamental equations may be written

$$\int_0^O \left[\int_0^s \frac{M}{I}ds\right]dy = 0 \quad . \quad . \quad . \quad . \quad . \quad . \quad (22')$$

and
$$\int_0^O \left[\int_0^s \frac{M}{I}ds\right]dx = 0 \quad . \quad . \quad . \quad . \quad . \quad . \quad (23')$$

which are equivalent to (22) and (23).

These integrals are most conveniently evaluated by the integraph, but in general it involves less labor to use the equations (22) and (23), which, moreover, present the advantage that they can be integrated readily by Simpson's Rule.

$$\int_A^B \frac{M}{I} ds = 0 \quad . \quad . \quad . \quad . \quad . \quad (24)$$

$$\int_A^B (y - y_A) \frac{M}{I} ds = 0 \quad . \quad . \quad . \quad . \quad (25)$$

$$\int_A^B (x - x_A) \frac{M}{I} ds = 0 \quad . \quad . \quad . \quad . \quad (26)$$

Whatever be the limits of the integration, these equations will furnish the values of the unknown internal reactions that enter into the expression for M ordinarily a couple, a direct force, and a shearing force at the origin, and when these are determined, the bending moment, the direct force, and the shearing force at any other point can be found.

The stress can now be calculated from the following approximate formula :

$$p = \frac{P}{A} + \frac{M}{A\rho} + \frac{M}{I} \frac{y}{1 + \frac{y}{\rho}} \quad . \quad . \quad . \quad (27)$$

which for $\rho = \infty$ becomes

$$p = \frac{P}{A} + \frac{M}{I} y \quad . \quad . \quad . \quad . \quad . \quad (28)$$

This is the ordinary formula for a straight girder, and can be used without sensible error in most cases occurring in naval construction.*

2. **The Principle of Least Work.**—The formulas here given for the calculation of the strength of a closed frame-ring may be derived also from the so-called "principle of least work," the enunciation and proof of which, as well as its application to engineering, are due to Alberto Castigliano.† According to this principle, the deformation of a structure,

* The formulas (21) to (28) are all approximative, but will generally be sufficiently accurate for all practical purposes provided the depth of the frame in direction of the radius of curvature is not greater than from about one-fourth to one-sixth of the length of this radius. If this condition is not fulfilled, I must be replaced by the expression $\rho \int \frac{y^2}{\rho + y} dA$ where y is the distance from the neutral axis of an elemental strip, of area dA, of the cross-section of the frame and where the integration extends over the whole section. The error committed in each given case by using I instead of this expression can be easily estimated. Where the frame consists of a simple bar, or a bar with reversed frame, its depth is usually smaller than one-sixth of the radius of curvature, but in some submarine boats, where deep bracket or lattice frames are employed, it may be necessary to use the more exact expression instead of I. Likewise, the additional terms (given in the preceding footnote) which take account of the direct force P are only approximative. If the radius of curvature is very small, a further term $\int_0^A \frac{M}{EA\rho} dx$ must be added to the expression for Δx_A in (19), and $\int_0^A \frac{M}{EA\rho} dy$ to the expression for Δy_A in (20).

† *Théorie d'équilibre des systèmes élastiques*, Turin, 1879.

strained under the action of a given system of forces, will be such that the elastic work done on the structure is a minimum. In other words, *the structure adapts itself with the minimum of effort to the applied forces.*

In order to apply this principle to a given structure, it is necessary first to write down the mathematical expression for the work done in the different elastic straining actions, bending, tension or compression, and shearing (in ship problems generally only bending need be taken into account). The minimum value of this expression is found by differentiating it with respect to each one of the unknown forces or couples that enter into it, and equating the partial differentials so obtained to zero. The resulting equations will be found to be the same as those given above for the deflections, expressing simply the conditions of continuity. This method, which appears to be used in many cases by civil engineers, has been applied by Dr. J. Bruhn to the determination of the transverse strength of ships.* In the problems occurring in ship-construction, it appears simpler, however, to write down at once the expressions for the deflection as explained above instead of going through the lengthy process of applying the principle of least work, which, being less tangible to the mind, is more apt to lead to mistakes.

12. TRANSVERSE STRENGTH OF TORPEDO-VESSELS.

1. **Straining Actions.**—In torpedo-vessels the bulkheads in the engine- and boiler-rooms are spaced far apart, wherefore the frames are here of great importance in point of transverse strength. This fact, as well as the claim to light scantlings, renders it desirable to examine the strength of the frames carefully. The straining effects may be grouped as follows :—

(1) Deformation due to the forces of weight and buoyancy.

(2) Deformation due to dynamic actions. When a vessel is rolling, the inertia forces will tend to distort the frames. When it is forced up against the sea and driven into the waves, the transverse pressures of the water will cause a panting of the frames which is ordinarily most pronounced in the bow, but may be considerable even amidships. In a certain torpedo-boat, where the deck was weakened by a very long hatchway, an actual transverse crushing of the hull took place, resulting in a permanent deformation, whereby the hatchway became narrower at the middle.†

(3) Bulging of the entire side from the gunwale to the head of the floors due to shearing, as explained in SECTION 7, *3*.

* *Inst. Nav. Arch.*, 1901, 1904

† Sir John Thornycroft, *Inst. Nav. Arch.*, 1905, i. p. 111.

(4) Bulging of the sides caused indirectly by longitudinal bending of the vessel.

None of these actions demand explanation except the last, which, to the author's knowledge, has not formerly been discussed in this connection.

2. **Bulging of the Sides Due to Longitudinal Bending.**—Consider a portion of the ship-girder amidships, enclosed between two adjacent transverse bulkheads or other strongly stiffened transverse sections, spaced a distance l from each other. When the vessel is subject to longitudinal bending, these sections come to form an angle ϕ with each other (fig. 19). The resultants P_0 of the stresses which, as explained in SECTION 6, *2*, act in opposite directions above and below the

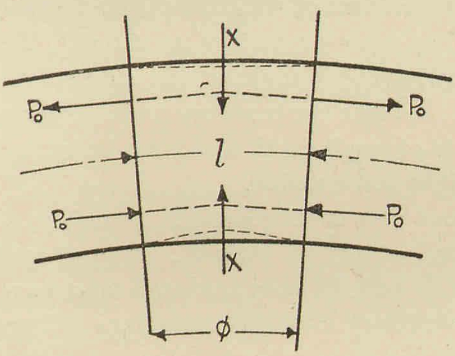


FIG. 19.

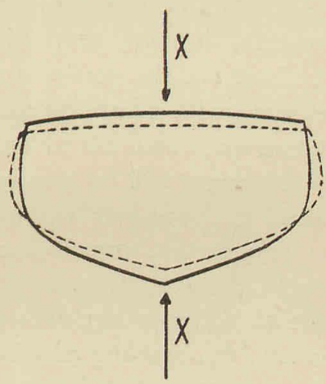


FIG. 20.

neutral axis of the sections will now, due to the deflection of the ship-girder, have vertical components χ which tend to force the deck and the bottom together. Unless the transverse framing possesses sufficient stiffness in itself, or is supported by internal stiffening members, such as longitudinal bulkheads and pillars, these forces will cause outward bending of the sides, eventually accompanied by longitudinal, upwards bulging of that flange which is in compression. The action is the same as may be observed when bending a rubber tube, where ultimately a complete flattening of the tube takes place at the point of maximum bending.

In a ship-girder this action is greatest amidships, where the bending moment and hence the curvature is a maximum. The resultant vertical forces are

$$\chi = P_0\phi = \frac{Mm_0}{I}\phi$$

where m_0 is the moment of the area of one of the sections above or below the neutral axis about this axis. Approximately the angular deflection is given by

$$\phi = \frac{Ml}{IE}$$

if we assume the bending moment and the moment of inertia to be constant between the sections under consideration. Substituting in the expression for χ we obtain

$$\chi = \frac{M^2 m_0 l}{I^2 E} \quad . \quad . \quad . \quad . \quad (29)$$

In order to obtain the force acting on one frame, l is put equal to the frame space s.

This expression shows that for a given bending moment χ will be, roughly, inversely proportional to the product of the cube of the depth of the boat and the sectional area of the deck or bottom structure. The leverage with which the forces χ act, tending to bend the sides of the vessel, must increase with the beam. Hence the tendency to bulging due to this cause will be greatest in boats of small depth and great beam, and where the deck and bottom are of light construction.

In a certain torpedo-boat of about 300 ts. displacement and 200 ft. length, $I = 2746$ sq. in. × (ft.)^2, $m_0 = 347$ sq. in. × ft., $l = 29{\cdot}5$ ft. (the aft boiler-room), $E = 13500$. When the boat is placed on a wave-hollow, the bending moment under ordinary standard conditions is found $M = 3753$ ft.-ts. Hence

$$\chi = \frac{(3753)^2 \times 347 \times 29{\cdot}5}{(2746)^2 \times 13500} = 1{\cdot}42 \text{ ts.}$$

Since there are twenty frame-spaces in the boiler-room, the force per frame-space will be only 159 lb., or, approximately, the weight of one man. The effect will be about the same as if a number of men were lined up on the deck over the boiler-room, one man on each frame.

In this case, then, the force appears insignificant, but it has to be borne in mind: First, that this load is added when the deck is already subject to a great compressive stress and, therefore, apt to sag and buckle, if not properly stiffened by pillars or other means. Second, that the height of the waves may sometimes be much greater than one-twentieth of the length as here assumed. In fact, French observers have recorded waves from 200 to 300 feet long of a height equal to one-tenth of the length.* Third, that due to the vertical oscillations of (the center of gravity of) a boat steaming up against the waves, dynamic actions will

* Sir W. H. White, *Manual of Naval Architecture*, London, 1900, p. 214.

occur which will greatly increase the bending moment, especially when the boat is astride a wave-hollow. Fourth, that when the frames yield by bulging out sideways, the depth of the ship-girder is reduced, the moment of inertia decreases rapidly, χ will increase, and the condition will be still further aggravated.

While, therefore, this action can be ignored under ordinary conditions, it may in extreme cases of straining be serious. In fact, it seems not unlikely that bulging of the sides may have been in some cases the ultimate cause of structural breakdown of torpedo-vessels. This mode of failure should therefore be carefully provided against in such craft by fitting pillars or other means of stiffening between the bulkheads so as to maintain strictly the transverse form of the hull under all conditions.

3. **Application of the Fundamental Equations.** — We shall now show how the formulas, given in SECTION II, may be used in practice for calculating the strength of frames in torpedo-vessels and ships of similar, simple construction.

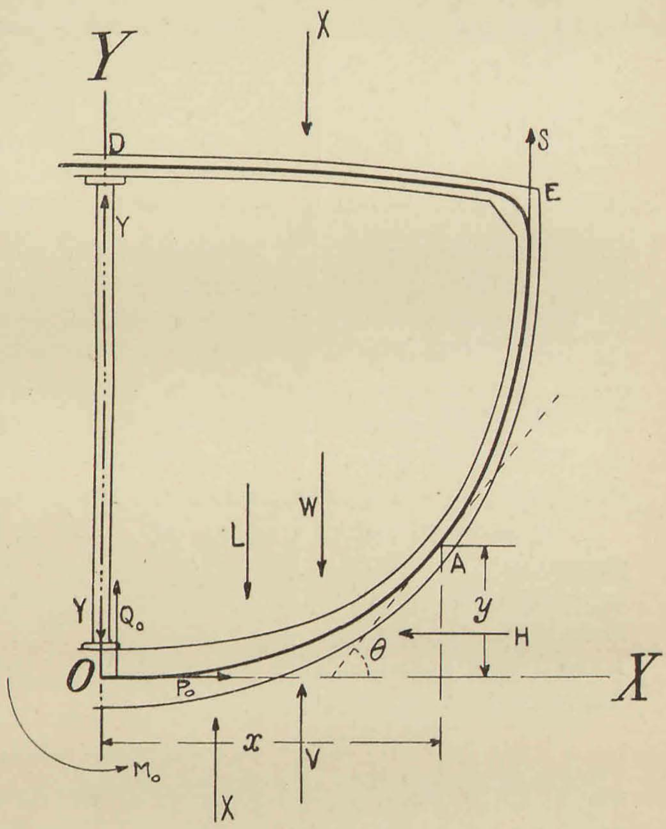


FIG. 21.

Consider the frame-ring of a torpedo-vessel as shown in fig. 21. Since there is symmetry of form and symmetry of forces about the center-line there will be no angular deflection and no linear deflection normal to this line at the points where it intersects the frame. It is of advantage to choose one of these points as origin and the axis of symmetry as one of the axes of co-ordinates, since the expressions for the bending moments are then simplified and the integration needs only to be extended round half the contour of the frame. We shall here place the origin at the keel and the axis of ordinates along the center-line. The tangent to the curve of the neutral axis at O will be the axis of abscissæ. If a pillar is fitted in the center-line, as in fig. 21, the deflection along this line will be zero.

In order to evaluate the integrals of formulas (21), (22), and (23), where the integration in this case extends from O to D it is necessary to resort to some approximate method, as for instance Simpson's Rule, or the integration may be performed graphically. In order to employ Simpson's Rule, the curve of the neutral axis, the heavy line in fig. 21, is divided into a number of equal intervals and the value of $\frac{M}{I}$ is determined for each point of division. Where the frame joins the beam, at E, the curve has a rather sharp bend, but with a proper end connection of the beam there will be no actual discontinuity. There will be probably, however, at E an abrupt change in the value of $\frac{M}{I}$ which renders it necessary, in order to apply Simpson's Rule, to perform the integration for each of the members, the frame and the beam, separately.

The moment of inertia I is calculated from the given design of the frame, a band of shell plating of width $30t$ being included, as usual. The moment of inertia at the beam knee will depend essentially on the riveted connection; ordinarily it may be assumed to be the same as that of the beam. When the frame-ring is of constant section throughout, I will disappear from the equations.

The bending moment M at any point A (fig. 21) is now expressed in terms of the unknown internal elastic reactions at O and the known external forces. The former are completely defined by a direct force P_0 a shearing force Q_0 and a couple M_0. Let the ordinates of A be x and y and consider the equilibrium of the piece OA. Taking moments about A we have

$$M = M_0 + P_0 y - Q_0 x + \Sigma M_1$$

where ΣM_1 comprises the moments of all the external forces acting on OA. Let the inclination of the tangent at A to the axis of abscissæ be θ then the direct force acting on the section at A is

$$P = P_0 \cos\theta + Q_0 \sin\theta + [\text{the sum of the components of all the external forces acting on OA resolved along the tangent at A}].$$

The shearing force at A is determined by a similar expression, obtained by resolving all the forces along the normal to the curve at A.

The external forces acting on the frame may be grouped under the following heads :—

(1) W the weight of the frame with other attached structural parts that are supported by it.

(2) L the weight of the internal loads, cargo, machinery, or other parts carried by the frame.

(3) H and V the horizontal and vertical components of the buoyancy which acts on the frame-space under consideration. If the ship is in dock, the reactions of the blocks take the place of the forces of buoyancy.

(4) S the shearing force resulting from the reaction of adjacent frames due to inequality in the weight and buoyancy forces acting on the frame. S is equal to the difference between the total vertical shearing forces acting at the two sections which enclose the frame, and may be obtained from the curve of shearing forces. It may with sufficient approximation be taken to act at the vertical or nearly vertical parts of the frame, *i.e.* near the side of the boat, since the shearing stresses are practically concentrated in the web of the ship-girder. We here disregard the couple created by S which has its axis in the plane of the frame and tends to bend it out of its plane. If longitudinal bulkheads are found, a certain part of S will be carried by them.

(5) X the vertical forces due to longitudinal bending discussed in *Article 2* of this section.

Using appropriate suffixes we may now write down the expression for the moments of the external forces, referred to above as ΣM_1

$$\Sigma M_1 = M_W + M_L - M_H - M_V \pm M_S - M_X \qquad . \qquad . \qquad (30)$$

The signs of the different terms are chosen so as to conform to fig. 21.

Having determined the complete expressions for M substitute them in the formulas (21), (22), and (23), and integrate. From the equations so obtained M_0, P_0, and Q_0 may be found, and thence M, P, and Q as well as the stresses at the different points of the frame-ring.

4. **Typical Cases.**—We shall now discuss in detail some cases typical for torpedo-vessels and similar light ships.

(1) *The frame-ring is unsupported by pillars.*—Assuming symmetrical loading, there will be no shearing force at O whence $Q_0 = 0$ and the equation for the bending moment becomes

$$M = M_0 + P_0 y + \Sigma M_1 \qquad . \qquad . \qquad . \qquad (31)$$

Since $\Delta\theta_D = 0$ and $\Delta x_D = 0$ the equations to be used are

$$\int_0^D \frac{M}{I} ds = 0 \qquad \text{and} \qquad \int_0^D y \frac{M}{I} ds = 0$$

Having determined the value of $\frac{M}{I}$ and $y\frac{M}{I}$ at each point of division, the integration is performed by Simpson's Rule, whereupon the equations are solved for M_0 and P_0. The bending moment can now be determined at any desired point. The direct force P acting along the tangent at A is given by

$$P = (P_0 - H)\cos\theta + [Q_0 - W - L + V \pm S + X]\sin\theta \quad . \quad (32)$$

(2) *The frame-ring is supported by a pillar in the middle.*—The pillar exerts a simple push or pull, Y, at the points O and D, and may here be considered as incompressible. The bending moment at any point of the frame is

$$M = M_0 + P_0 y - Q_0 x - \Sigma M_1 \quad . \quad . \quad . \quad (33)$$

Since $\Delta\theta_D$, Δx_D, and Δy_D are all equal to zero, the equations to be used are

$$\int_0^D \frac{M}{I} ds = 0 \qquad \int_0^D y\frac{M}{I} ds = 0 \qquad \int_0^D x\frac{M}{I} ds = 0$$

from which the unknown reactions can be determined. P is found from (32), and $Y = 2Q_0$.

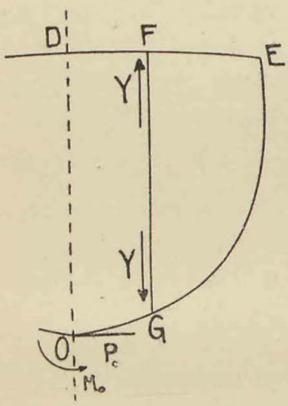


FIG. 22.

(3) *The frame-ring is supported by one pillar on each side* (fig. 22).—Q_0 will be zero, but the pillars will be subject to an unknown force Y which will enter into the expressions for the bending moment and for the direct force at all points on the girth GEF. The pillar is assumed to be hinged at the ends and, therefore, incapable of transmitting any bending moments. While the bending moment for points between O and G and between F and D are given by equation (31), it is for points between G and F of the form

$$M = M_0 + P_0 y + Y(x - c) + \Sigma M_1 \quad . \quad (34)$$

where c is the distance of the pillar from the middle.

We have again

$$\int_0^D \frac{M}{I} ds = 0 \quad \text{and} \quad \int_0^D y\frac{M}{I} ds = 0$$

but since the total vertical deflection is now zero from G to F the ends of the incompressible pillar, the third equation takes the form

$$\int_G^F (x-c)\frac{M}{I}ds = 0 \quad . \quad . \quad . \quad . \quad . \quad (35)$$

The equation for P is the same as in case (1), only between G and F the thrust of the pillar $-Y$ must be added inside the brackets in the last term.

The equilibrium of the frame at such a point as G where it is directly subject to the thrust of the pillar, is maintained by the combined action of the direct forces P_{OG} and P_{GF} and the shearing forces Q_{OG} and Q_{GF} on each side of the pillar, as shown in fig. 23.

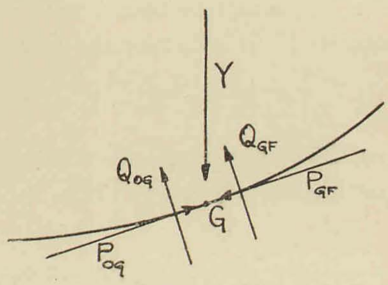


FIG. 23.

If the pillar is normal to the frame, as in the case of a central pillar, we have $P_{OG} = P_{GF}$ and the shearing forces are, each of them, equal to $\frac{1}{2}Y$ as stated above. If the pillar is not normal to the frame, as in fig. 23, the direct forces and the shearing forces are no longer equal, and it is, therefore, in such a case necessary, if we want to find the maximum stresses at G to determine the values of P and Q for each side of the pillar and to use the larger of these forces in the calculation of the stresses. The different forces, shown diagrammatically in fig. 23, are actually the resultants of a complicated system of stresses.

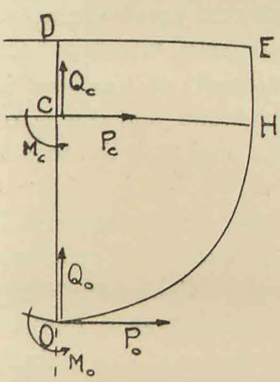


FIG. 24.

(4) *The frame is stiffened by a second deck and by a line of pillars in the middle* (fig. 24).—The conditions to be fulfilled are in this case that at the points O, C, and D, all on the center-line, the tangent to the neutral axis will remain horizontal, and no horizontal or vertical displacement of these points will take place. Hence, we must have:

$$\int_O^D \frac{M}{I}ds = 0 \qquad \int_O^D y\frac{M}{I}ds = 0 \qquad \int_O^D x\frac{M}{I}ds = 0$$

the integration extending from O through H and E to D. Further:

$$\int_C^D \frac{M}{I}ds = 0 \qquad \int_C^D y\frac{M}{I}ds = 0 \qquad \int_C^D x\frac{M}{I}ds = 0$$

the integration extending from C through H and E to D. These six equations will give the six unknown reactions : P_o, Q_o, and M_o at O, and P_c, Q_c, and M_c at C, whereafter M, P, and Q can be found at any point of the structure.

The expression for the bending moment at points on the frame below the lower tier of beams will be as for a single-deck ship, but for points above H it will take the form :

$$M = M_o + P_o y - Q_o x + M_c + P_c(y - y_c) - Q_c x + \Sigma M_1 \quad . \quad (36)$$

In ΣM_1 we must now include the load on the beam CH. The load on the lower pillar is $2Q_o$ and on the upper, $2(Q_o - Q_c)$.

13. TRANSVERSE STRENGTH OF SUBMARINE BOATS.

1. **Submarine Boats of Non-Circular Section.**—The foregoing treatment applies also to submarine boats, but it is necessary to add a few remarks on the special conditions which obtain in these vessels.

Since the weight of a submarine boat is practically always balanced by the buoyancy forces, and since these latter are small compared with the total external water-pressures at the maximum depth of immersion, we commit no great error by neglecting the weight and by assuming that the pressure is uniform all round the contour of the boat and equal to that at its axis. Suppose, for instance, that the maximum head, which a boat is required to withstand, is 150 feet and that the depth of the boat is 12 feet, then the maximum error will be $\frac{6}{150}$ or 4 per cent., but this error is reduced from the fact that the principal weights rest directly on the lower part of the frames.

Submarine boats of oval or elliptic sections are sometimes symmetrical about a horizontal as well as a vertical axis, in which case only a quadrant of the frame needs to be considered in the strength calculation. The pure elliptical section can be dealt with by a more algebraical method than other sections, although it is necessary to evaluate the elliptic integrals graphically or by some other approximate method, but the work is simpler and the solution for a given eccentricity can be readily applied to all elliptical frames of the same eccentricity, whatever their dimensions.*

2. **Examples.**—We shall apply the method here described to the frame of a submarine boat in the three typical cases discussed above—when the frame is stiffened by a pillar on each side, when it is stiffened by a pillar in the middle, and when there is no pillar. Fig. 25 shows

* A solution of this problem is given in a paper by the author : "The Strength of Elliptic Sections under Fluid Pressure," where the equations (22′) and (23′) are used, *Inst. Nav. Arch.*, 1900.

STRENGTH CALCULATIONS FOR SUBMARINE BOATS

$$M = M_0 + P_0 y + Y(x - x_2) - \frac{1}{2} w (x^2 + y^2)$$

STAT.	S.M.	I	$\frac{S.M}{I}$	x	y	x^2	y^2	x^2+y^2	$x-x_2$	$(x-x_2)y$	$(x-x_2)^2$	$(x-x_2)(x^2+y^2)$	xy	$x(x^2+y^2)$
0	1	406·1	·00246	0·0	0·0	00	00	00					00	00
1	4	333·2	·01200	15·1	0·6	228	00	228					9	3443
2	1	132·3	·00756	30·0	3·2	900	10	910					96	27300
														×·738 =
2	½	132·3	·00378	30·0	3·2	900	10	910	0·0	00	00	00	96	27300
2½	2	48·6	·04115	39·6	6·5	1568	42	1610	9·6	62	92	15500	257	63800
3	1½	48·6	·03086	48·6	11·1	2362	123	2485	18·6	206	346	46200	539	120800
4	4	48·6	·08230	64·7	23·6	4186	557	4743	34·7	819	1204	164600	1527	306900
5	2	48·6	·04115	76·4	40·1	5837	1608	7445	46·4	1861	2153	345400	3064	568800
6	4	48·6	·08230	82·0	59·6	6724	3552	10276	52·0	3099	2704	534400	4887	842600
7	2	48·6	·04115	78·4	79·6	6147	6336	12483	48·4	3853	2343	604200	6241	978700
8	4	48·6	·08230	66·6	95·8	4436	9178	13614	36·6	3506	1340	498300	6380	906700
9	1½	48·6	·03086	49·5	107·0	2450	11449	13899	19·5	2087	380	271000	5297	688000
9½	2	48·6	·04115	40·0	110·7	1600	12254	13854	10·0	1107	100	138500	4428	554200
10	½	80·8	·00619	30·0	112·7	900	12701	13601	0·0	00	00	00	3381	408000
10	1	80·8	·01238	30·0	112·7	900	12701	13601					3381	408000
11	4	157·7	·02536	15·1	114·0	228	12996	13224					1721	199700
12	1	183·7	·00544	0·0	114·1	00	13019	13019					00	00
														×·738 =

TWO PILLARS

STAT.	Products for $\int \frac{M}{I} ds$: $\frac{S.M}{I}$	$\frac{S.M}{I}y$	$\frac{S.M}{I}(x-x_2)$	$\frac{S.M}{I}(x^2+y^2)$	Products for $\int \frac{M}{I}(x-x_2) ds$: $\frac{S.M}{I}(x-x_2)$	$\frac{S.M}{I}y(x-x_2)$	$\frac{S.M}{I}(x-x_2)^2$	$\frac{S.M}{I}(x^2+y^2)(x-x_2)$
0	·00246	0.000		0·0				
1	·01200	0·007		2·7				
2	·00756	0·024		6·9				
	·02202	0·031		9·6				
	·01625	0·023		7·1				·738
2	·00378	0·012	0·000	3·4	0·000	0·0	0·0	00
2½	·04115	0·267	0·395	66·3	0·395	2·6	3·8	640
3	·03086	0·343	0·574	76·7	0·574	6·4	10·7	1430
4	·08230	1·942	2·856	390·3	2·856	67·4	99·1	13550
5	·04115	1·650	1·909	306·4	1·909	76·6	88·6	14210
6	·08230	4·905	4·280	845·7	4·280	255·0	222·5	43980
7	·04115	3·276	1·992	513·7	1·992	158·6	96·4	24860
8	·08230	7·884	3·012	1120·4	3·012	288·5	110·3	41010
9	·03086	3·302	0·602	428·9	0·602	64·4	11·7	8360
9½	·04115	4·555	0·412	570·1	0·412	45·6	4·1	5700
10	·00619	0·698	0·000	84·2	0·000	0·0	0·0	00
	·48319	28·834	16·032	4406·1	16·032	965·1	647·2	153740
10	·01238	1·395		168·4				
11	·02536	2·891		335·4				
12	·00544	0·621		70·8				
	·04318	4907		574·6				
	·03187	3·621		424·1				×·738 =
	·53131	32·478	16·032	4837·3	16·032	965·1	647·2	153740

ONE PILLAR

STAT.	Products for $\int \frac{M}{I} ds$: $\frac{S.M}{I}$	$\frac{S.M}{I}y$	$\frac{S.M}{I}x$	$\frac{S.M}{I}(x^2+y^2)$	Products for $\int \frac{M}{I} x\, ds$: $\frac{S.M}{I}x$	$\frac{S.M}{I}yx$	$\frac{S.M}{I}x^2$	$\frac{S.M}{I}(x^2+y^2)x$
0	·00246	0·000	0·000	0·0	0·000	0·0	0·0	00
1	·01200	0·007	0·181	2·7	0·181	0·1	2·7	41
2	·00756	0·024	0·227	6·9	0·227	0·7	6·8	206
	·02202	0·031	0·408	9·6	0·408	0·8	9·5	247
	·01625	0·023	0·301	7·1	0·301	0·6	7·0	180
2	·00378	0·012	0·113	3·4	0·113	0·4	3·4	100
2½	·04115	0·267	1·630	66·3	1·630	10·6	64·5	2630
3	·03086	0·343	1·500	76·7	1·500	16·6	72·9	3730
4	·08230	1·942	5·325	390·3	5·325	125·7	344·5	25260
5	·04115	1·650	3·144	306·4	3·144	126·1	240·2	23410
6	·08230	4·905	6·749	845·7	6·749	402·2	553·4	69350
7	·04115	3·276	3·226	513·7	3·226	256·8	252·9	40270
8	·08230	7·884	5·481	1120·4	5·481	525·1	365·1	74620
9	·03086	3·302	1·528	428·9	1·528	163·5	75·6	21230
9½	·04115	4·555	1·646	570·1	1·646	182·2	65·8	22800
10	·00619	0·698	0·186	84·2	0·186	20·9	5·6	2530
	·48319	28·834	30·528	4406·1	30·528	1830·1	2043·0	285930
10	·01238	1·395	0·371	168·4	0·371	41·9	11·1	5050
11	·02536	2·891	0·383	335·4	0·383	43·6	5·8	5060
12	·00544	0·621	0·000	70·8	0·000	0·0	0·0	00
	·04318	4907	0·754	574·6	0·754	85·5	16·9	10110
	·03187	3·621	0·556	424·1	0·556	63·1	12·5	7460
	·53131	32·478	31·385	4837·3	31·385	1893·8	2063·4	293570

NO PILLAR

STAT.	Products for $\int \frac{M}{I} ds$: $\frac{SM}{I}$	$\frac{SM}{I}y$	$\frac{SM}{I}(x^2+y^2)$
0	·00246	0·000	0·0
1	·01200	0·007	2·7
2	·00756	0·024	6·9
	·02202	0·031	9·6
	·01625	0·023	7·1
2	·00378	0·012	3·4
2½	·04115	0·267	66·3
3	·03086	0·343	76·7
4	·08230	1·942	390·3
5	·04115	1·650	306·4
6	·08230	4·905	845·7
7	·04115	3·276	513·7
8	·08230	7·884	1120·4
9	·03086	3·302	428·9
9½	·04115	4·555	570·1
10	·00619	0·698	84·2
	·48319	28·838	4406·1
10	·01238	1·395	168·4
11	·02536	2·891	335·4
12	·00544	0·621	70·8
	·04318	4·907	574·6
	·03187	3·621	424·1
	·53131	32·478	4837·3

NOTES

w = ·536 TS. PER INCH OF GIRTH

INTERVAL OF GIRTH: 0–2 AND 10–12, 15·00 IN.

" " " 2–10, 20·33 IN.

FACTOR FOR INTERVALS; $\frac{15·00}{20·33}$ = ·738

$P_0 = \frac{1}{2} \times ·536 \times 114·1$ = 30·58 TONS

Two pillars:

$$·5313 M_0 + 32·48 P_0 + 16·03 Y - 4837·3 \cdot \tfrac{1}{2} w = 0$$
$$·5313 M_0 + 993·2 + 16·03 Y - 1296·4 = 0$$
$$M_0 + 30·17 Y - 570·7 = 0 \quad \text{———— (1)}$$

$$16·03 M_0 + 965·1 P_0 + 647·2 Y - 153740 \times \tfrac{1}{2} w = 0$$
$$16·03 M_0 + 29510 + 647·3 Y - 41200 = 0$$
$$M_0 + 40·38 Y - 729·3 = 0 \quad \text{———— (2)}$$

FROM (1) AND (2)

$$Y = \frac{158·6}{10·21} = 15·53 \text{ TONS}$$

SUBSTITUTING IN (1):

$$M_0 + 468·5 - 570·9 = 0$$
$$\therefore M_0 = 102 \text{ INCH TONS}$$

One pillar:

$$·5313 M_0 + 32·48 P_0 + 31·39 \times \tfrac{1}{2} Y - 4837·3 \times \tfrac{1}{2} w = 0$$
$$·5313 M_0 + 993·2 + 15·69 Y - 1296·4 = 0$$
$$M_0 + 29·53 Y - 570·7 = 0 \quad \text{———— (3)}$$

$$31·39 M_0 + 1894 P_0 + 2063 \times \tfrac{1}{2} Y - 293570 \times \tfrac{1}{2} w = 0$$
$$31·39 M_0 + 57920 + 1032 Y - 78680 = 0$$
$$M_0 + 32·88 Y - 661·4 = 0 \quad \text{———— (4)}$$

FROM (3) AND (4):

$$Y = \frac{90·7}{3·35} = 27·07 \text{ TONS}$$

SUBSTITUTING IN (3):

$$M_0 + 799·4 - 570·7 = 0$$
$$\therefore M_0 = -229 \text{ INCH TONS}$$

No pillar:

$$·5313 M_0 + 32·48 P_0 - 4837·3 \times \tfrac{1}{2} w = 0$$
$$·5313 M_0 + 993·2 - 1296·4 = 0 \quad \text{———— (5)}$$
$$M_0 = 570·7 \text{ INCH TONS}$$

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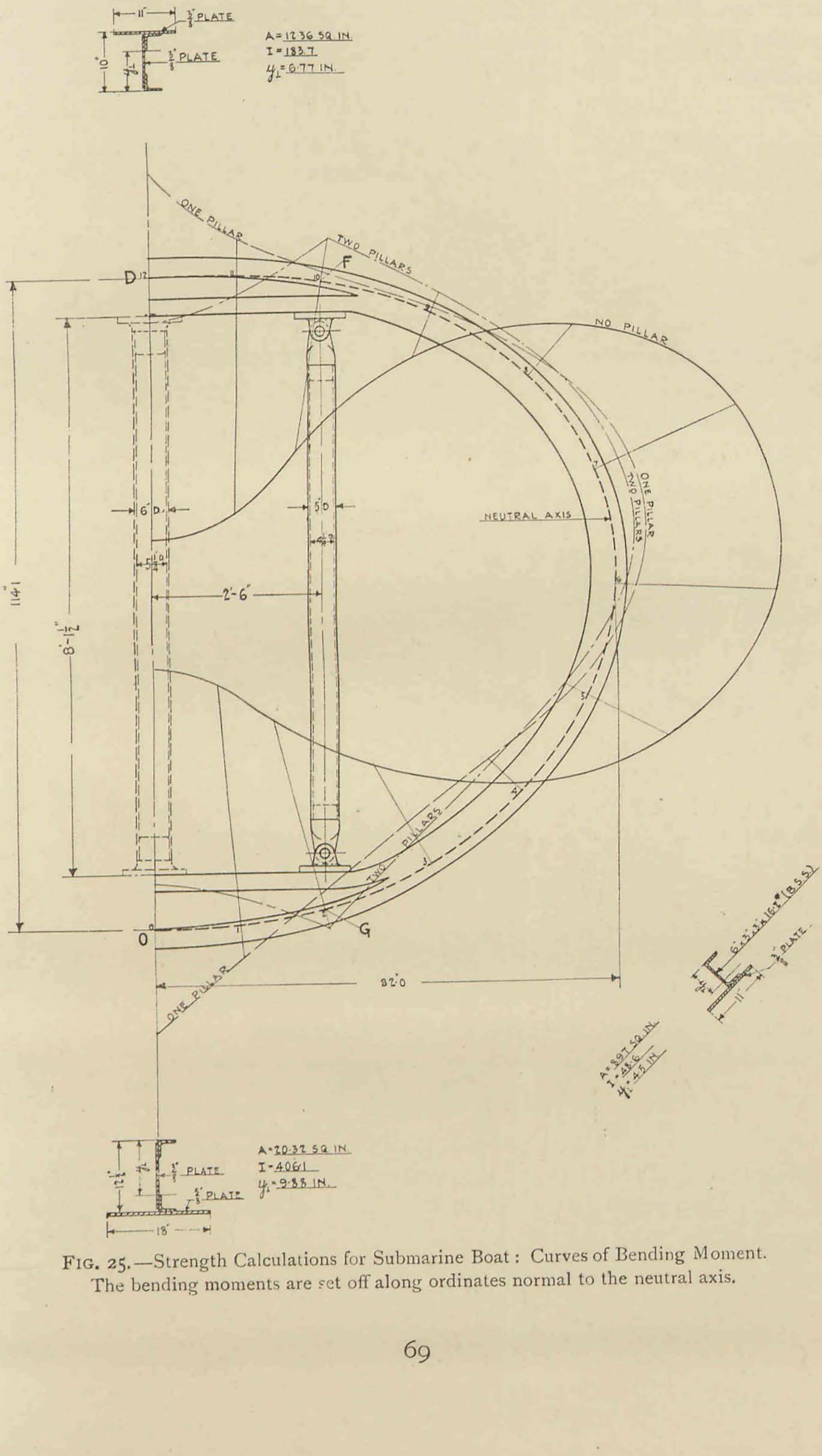

FIG. 25.—Strength Calculations for Submarine Boat: Curves of Bending Moment. The bending moments are set off along ordinates normal to the neutral axis.

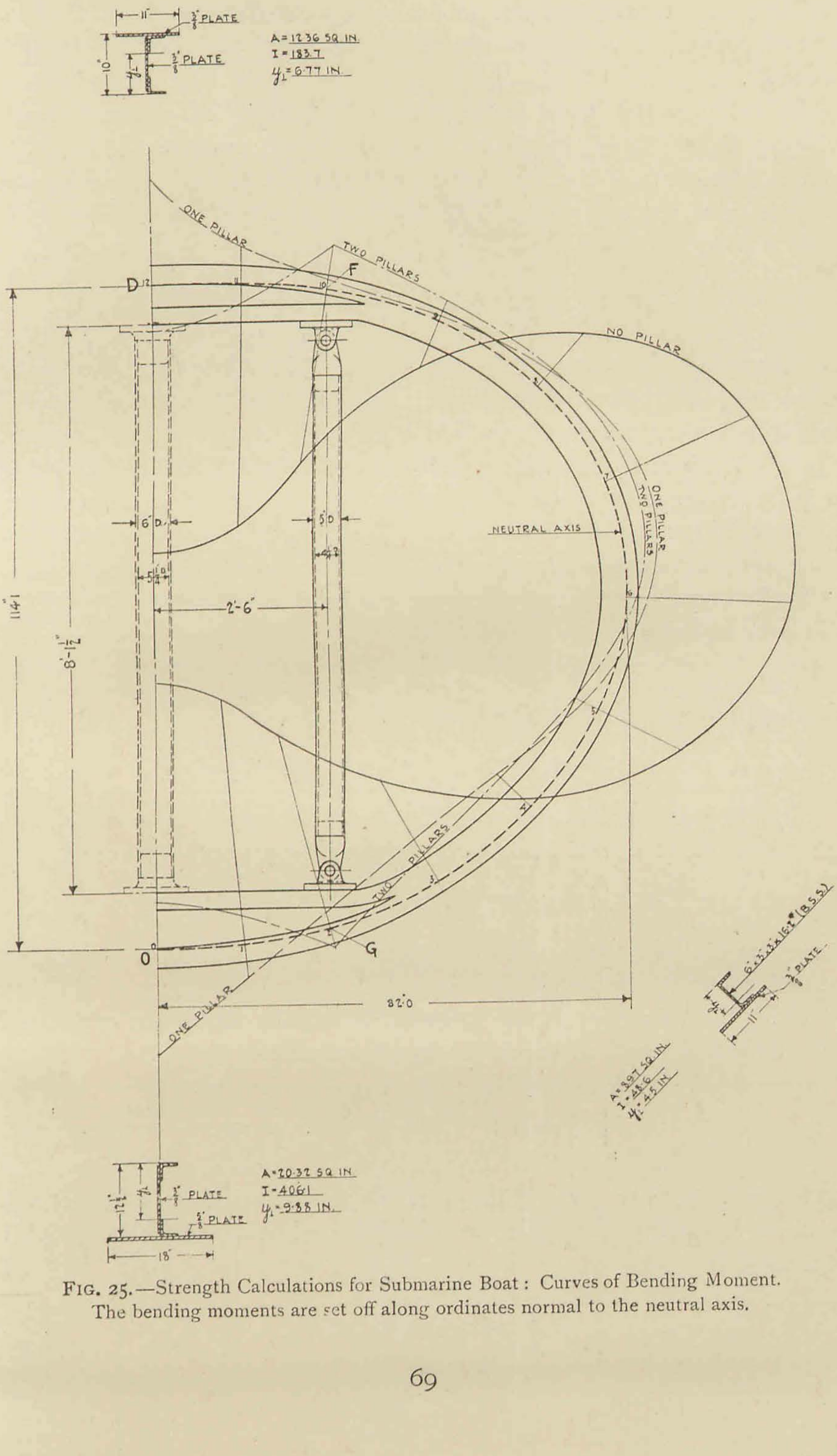

FIG. 25.—Strength Calculations for Submarine Boat: Curves of Bending Moment. The bending moments are set off along ordinates normal to the neutral axis.

the frame, the position of the pillars, and cross-sections of the frame at various points. It gives also the neutral axis and the curves of bending moments.

Referring to the line of the neutral axis, the principal dimensions are: half-breadth = 82 in., depth = 114·1 in., frame space = 18 in. The boat is to withstand the pressure at a depth of 150 ft. Assuming this pressure to be uniform all round the contour, the load everywhere on the frame will be

$$w = \frac{1\cdot5 \times 150}{12 \times 35} = \cdot536 \text{ ts. per in. of girth.}$$

The frames are channels 6 in. × 3 in. × 3 in., strengthened by floor-plates at the top and bottom of the boat. The standard thickness of the shell plating is $\frac{3}{8}$ in.; the flat keel-plate is $\frac{5}{8}$ in.

(1) *One pillar on each side.*—The pillars are placed at a distance of 30 in. from the center-line. The girth from O to D is divided into 12 intervals, numbered from 0 to 12. The length of the intervals from 0 to 2 and from 10 to 12 is 15·00 in., between 2 and 10 it is 20·32 in.

Neglecting the weight of the boat, the bending moment at points on OG and FD is

$$M = M_0 + P_0 y - \tfrac{1}{2}w(x^2 + y^2)$$

At points on GF the bending moment is

$$M = M_0 + P_0 y + Y(x - x_2) - \tfrac{1}{2}w(x^2 + y^2)$$

Since the pressure is uniform, the direct forces at O and D must be equal to half the total horizontal component pressure, whence

$$P_0 = \tfrac{1}{2}w \times y_{12} = \tfrac{1}{2} \times \cdot536 \times 114\cdot1 = 30\cdot58 \text{ ts.}$$

There remain only two unknown quantities, M_0 and Y which are determined by the two conditions:—

First, that the sum of the angular deflections along the girth from 0 to 12 shall be zero, as expressed by the equation

$$\int_0^2 \frac{M}{I} ds + \int_2^{10} \frac{M}{I} ds + \int_{10}^{12} \frac{M}{I} ds = 0$$

Second, that the sum of the vertical deflections from 2 to 10 shall be zero, assuming the pillar to be incompressible. This condition is expressed by

$$\int_2^{10} (x - x_2) \frac{M}{I} ds = 0$$

STRENGTH CALCULATIONS FOR SUBMARINE BOATS

BENDING STRESS

								TWO PILLARS							ONE PILLAR				NO PILLAR		
								$M = P_0 y - \frac{1}{2}w(x^2+y^2) + M_0 + Y(x - x_2)$							$M = P_0 y - \frac{1}{2}w(x^2+y^2) + M_0 + \frac{1}{2}Yx$				$M = P_0 y - \frac{1}{2}w(x^2+y^2) + M_0$		
STAT.	x	y	$x - x_2$	x^2+y^2	I	y_1	y_2	$P_0 y$	$\frac{1}{2}w(x^2+y^2)$	$P_0 y - \frac{1}{2}w(x^2+y^2)$	M_0	$Y(x-x_2)$	M	p_2	M_0	$\frac{1}{2}Yx$	M	p_2	M_0	M	p_2
0	0.0	0.0		00	406.1	9.88	3.25	00	00	00	102		102	.82	-229	00	-229	5.57	571	571	4.57
1	15.1	.6		228	333.2	9.20	3.03	18	61	-43	102		59	.54	-229	204	-68	1.88	571	528	4.80
2	30.0	3.2	0.0	910	132.3	6.50	2.55	98	244	-146	102	0	-44	2.16	-229	406	31	.60	571	425	8.19
3	48.6	11.1	18.6	2485	48.6	4.50	1.88	339	666	-327	102	289	64	2.47	-229	658	102	3.94	571	244	9.42
4	64.7	23.6	34.7	4743	48.6	4.50	1.88	722	1271	-549	102	539	92	3.55	-229	876	98	3.78	571	22	.85
5	76.4	40.1	46.4	7445	48.6	4.50	1.88	1226	1995	-769	102	721	54	2.09	-229	1034	36	1.39	571	-198	18.33
6	82.0	59.6	52.0	10276	48.6	4.50	1.88	1823	2754	-931	102	808	-21	1.94	-229	1110	-50	4.63	571	-360	33.33
7	78.4	79.6	48.4	12483	48.6	4.50	1.88	2434	3345	-911	102	752	-57	5.28	-229	1061	-79	7.31	571	-340	31.48
8	66.6	95.8	36.6	13614	48.6	4.50	1.88	2930	3649	-719	102	568	-49	4.54	-229	901	-47	4.35	571	-148	13.70
9	49.5	107.0	19.5	13899	48.6	4.50	1.88	3272	3725	-453	102	303	-48	4.44	-229	670	-12	1.11	571	118	4.56
10	30.0	112.7	0.0	13601	80.8	5.40	2.60	3447	3645	-198	102	0	-96	6.40	-229	406	-21	1.40	571	373	11.99
11	15.1	114.0		13224	157.7	6.60	3.40	3486	3544	-58	102		44	.95	-229	204	-83	3.47	571	513	11.06
12	0.0	114.1		13019	183.7	6.77	3.61	3489	3489	00	102		102	2.00	-229	00	-229	8.45	571	571	11.22

DIRECT STRESS

									TWO PILLARS						ONE PILLAR				NO PILLAR		
									$P = (P_0 - wy)\cos\theta + (wx - Y)\sin\theta$						$P = (P_0 - wy)\cos\theta + (wx - \frac{1}{2}Y)\sin\theta$				$P = (P_0 - wy)\cos\theta + wx\sin\theta$		
STAT.	A	$\theta°$	$\cos\theta$	$\sin\theta$	y	wy	x	wx	$P_0 - wy$	$wx - Y$	$(P_0 - wy)\cos\theta$	$(wx - Y)\sin\theta$	P	$p_1 = \frac{P}{A}$	$wx - \frac{1}{2}Y$	$(wx - \frac{1}{2}Y)\sin\theta$	P	$p_1 = \frac{P}{A}$	$wx \sin\theta$	P	$p_1 = \frac{P}{A}$
0	20.32	0.0	1.0000	.0000	0.0	.00	0.0	0.00	30.58	0.00	30.58	0.00	30.6	1.51	-13.54	0.00	30.6	1.51	0.00	30.6	1.51
1	19.02	6.5	.9936	.1132	.6	.32	15.1	8.09	30.26	8.09	30.07	.92	31.0	1.63	-5.45	-.62	29.5	1.55	.92	31.0	1.63
2	13.97	16.0	.9613	.2756	3.2	1.72	30.0	16.08	28.86	16.08 .55	27.75	4.43 .15	32.2 27.9	2.30 2.00	2.54	.70	28.5	2.04	4.43	32.2	2.30
3	8.97	31.0	.8572	.5150	11.1	5.95	48.6	26.05	24.63	10.52	21.11	5.42	26.5	2.96	12.51	6.44	27.6	3.08	13.42	34.5	3.85
4	8.97	45.3	.7034	.7108	23.6	12.65	64.7	34.68	17.93	19.15	12.61	13.61	26.2	2.92	21.14	15.03	27.6	3.08	24.65	37.3	4.15
5	8.97	63.2	.4509	.8926	40.1	21.50	76.4	40.95	9.08	25.42	4.09	22.69	26.8	2.99	27.41	24.46	28.6	3.18	36.55	40.6	4.53
6	8.97	86.7	.0576	.9983	59.6	31.95	82.0	43.95	-1.37	28.42	.08	28.37	28.3	3.16	30.41	30.36	30.3	3.38	43.88	43.8	4.88
7	8.97	113.4	-.3971	.9178	79.6	42.67	78.4	42.02	-12.09	26.49	4.80	24.31	29.1	3.25	28.48	26.14	30.9	3.45	38.57	43.4	4.84
8	8.97	136.9	-.7302	.6833	95.8	51.35	66.6	35.70	-20.77	20.17	15.17	13.78	29.0	3.23	22.16	15.14	30.3	3.38	24.39	39.6	4.41
9	8.97	155.6	-.9107	.4131	107.0	57.36	49.5	26.53	-26.78	11.00	24.39	4.54	28.9	3.23	12.99	5.37	29.8	3.32	10.96	35.4	3.94
10	9.40	172.0	-.9903	.1392	112.7	60.41	30.0	16.08	-29.83	.55 16.08	29.54	.08 2.24	29.6 31.8	3.15 3.38	2.54	.35	29.9	3.18	2.24	31.8	3.38
11	11.74	178.0	-.9994	.0349	114.0	61.11	15.1	8.09	-30.53	8.09	30.51	.28	30.8	2.62	-5.45	-.19	30.3	2.58	.28	30.8	2.62
12	12.36	180.0	-1.0000	.0000	114.1	61.16	0.0	0.00	-30.58	0.00	30.58	0.00	30.6	2.47	-13.54	0.00	30.6	2.47	0.00	30.6	2.47

NOTES

P_0 = 30.58 TONS
w = .536 TS. PR. IN. OF GIRTH
Y = 15.53 TS. FOR TWO PILLARS
Y = 27.07 TS. " ONE PILLAR
Y = 0 TS. " NO "

p_1 = DIRECT COMPRESSIVE STRESS Ts. Pr. Sq. in.

p_2 = COMPRESSIVE STRESS DUE TO BENDING. Ts. Pr. Sq. in.

M = BENDING MOMENT IN INCH-TONS

SUMMARY OF RESULTS

	TWO PILLARS					ONE PILLAR					NO PILLAR				
STAT.	P	M	p_1	p_2	$p = p_1 + p_2$	P	M	p_1	p_2	$p = p_1 + p_2$	P	M	p_1	p_2	$p = p_1 + p_2$
0	30.6	102	1.51	.82	2.3	30.6	-229	1.51	5.57	7.1	30.6	571	1.51	4.57	6.1
1	31.0	59	1.63	.54	2.2	29.5	-68	1.55	1.88	3.4	31.0	528	1.63	4.80	6.4
2	32.2 27.9	-44	2.30 2.00	2.16	4.5 4.2	28.5	31	2.04	.60	2.6	32.2	425	2.30	8.19	10.5
3	26.5	64	2.96	2.47	5.4	27.6	102	3.08	3.94	7.0	34.5	244	3.85	9.42	13.3
4	26.2	92	2.92	3.55	6.5	27.6	98	3.08	3.78	6.9	37.3	22	4.15	.85	5.0
5	26.8	54	2.99	2.09	5.1	28.6	36	3.18	1.39	4.6	40.6	-198	4.53	18.33	22.9
6	28.3	-21	3.16	1.94	5.1	30.3	-50	3.38	4.63	8.0	43.8	-360	4.88	33.33	38.2
7	29.1	-57	3.25	5.28	8.5	30.9	-79	3.45	7.31	10.8	43.4	-340	4.84	31.48	36.3
8	29.0	-49	3.23	4.54	7.8	30.3	-47	3.38	4.35	7.7	39.6	-148	4.41	13.70	18.1
9	28.9	-48	3.23	4.44	7.7	29.8	-12	3.32	1.11	4.4	35.4	118	3.94	4.56	8.5
10	29.6 31.8	-96	3.15 3.38	6.40	9.6 9.8	29.9	-21	3.18	1.40	4.6	31.8	373	3.38	11.99	15.4
11	30.8	44	2.62	.95	3.6	30.3	-83	2.58	3.47	6.1	30.8	513	2.62	11.06	13.7
12	30.6	102	2.47	2.00	4.5	30.6	-229	2.47	8.45	10.9	30.6	571	2.47	11.22	13.7

To face p. 71.

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Tables VI. and VII. show how the calculation can be conveniently carried out. The quotient $\frac{\text{S.M.}}{\text{I}}$ Simpson's Multiplier divided by the moment of inertia, which recurs in all the integrations, is first calculated. On account of the difference in intervals, the summation of the columns must be carried out in three sections. The interval in the middle section, 2 to 10, is regarded as the standard, whence the sums for the two other sections, 0 to 2 and 10 to 12, must be multiplied by the factor $\frac{15 \cdot 00}{20 \cdot 32} = \cdot 738$. Due to the rapid change in the value of I in the intervals 2–3 and 9–10, it is necessary here to introduce half ordinates. The measurement of the ordinates as well as the calculations must be carried out with great accuracy, since small errors will produce great differences in the results. The final equations with numerical values inserted are formed and their solution given underneath the respective columns in Table VI.

The direct force at points on OG and FD is found from

$$P = (P_0 - wy)\cos\theta + wx\sin\theta$$

At points on GF

$$P = (P_0 - wy)\cos\theta + (wx - Y)\sin\theta$$

At the points G and F, P will have two values, one to the left of the pillar, determined by the upper equation, and one to the right of the pillar, determined by the lower equation. The angle θ is obtained graphically. The stresses are calculated from the formula (28)

$$p = p_1 + p_2 = \frac{P}{A} + \frac{My}{I}$$

which in this case gives practically the same results as the more complete formula for curved girders (27), since y, the distance of the most strained fiber from the neutral axis, is everywhere small relative to the radius of curvature ρ and since $\frac{M}{A\rho}$ is small. It must be observed that since p_1 is always a compressive stress, p_2 should ordinarily likewise be compressive, as in Table VII. When the bending is excessive, the tensile value of p_2 must also be considered.

The results of the calculation are summarised at the foot of Table VII. It will be seen that the greatest bending moment is found at points 0 and 12, viz. 102 in.-ts., and at point 10, where it is 96 in.-ts. At this latter point the direct force P is 31·8 ts. to the left of the pillar, and the stress is 9·8 ts. per sq. in., which is the absolute maximum stress in the frame. The bending moments at O and D, the lowest and highest points of the frame, are equal, as they must always be with a uniform pressure and a symmetrical arrangement of the pillars. Although the bending moments are great at these points, the stresses are moderate on account of the great strength of the frame. The curve of bending moments given in fig. 25 is plotted on lines normal to the frame at each point. It will be seen that the bending moments are well distributed in this case. The importance of a proper location of the pillars is obvious. On account of the strongly rounded form of the frame the stress due to the direct force forms everywhere a considerable fraction of the total stress.

(2) *One pillar in the middle.*—The same intervals are used as in case (1). The bending moment is

$$M = M_0 + P_0 y + \tfrac{1}{2} Y x - \tfrac{1}{2} w(x^2 + y^2)$$

The conditions to be fulfilled are that the sum of the angular as well as of the vertical deflections shall be zero from 0 to 12, whence

$$\int_0^{12} \frac{M}{I} ds = 0 \quad \text{and} \quad \int_0^{12} x \frac{M}{I} ds = 0$$

The bending moment reaches its absolute maximum at the head and foot of the pillar, 229 in.-ts. The stress is a maximum at the head, where it is 10·9 ts. per sq. in., but another maximum is found at point 7, where it is 10·8 ts. per sq. in.

(3) *No pillar.*—The only unknown quantity is in this case M_0 which is found from

$$\int_0^{12} \frac{M}{I} ds = 0$$

where $M = M_0 + P_0 y - \tfrac{1}{2} w(x^2 + y^2)$. The bending moments and hence the stresses are excessive. The bending moments reach a value of 571 in.-ts. at the points 0 and 12, and the stresses have a maximum of about 38 ts. per sq. in. between points 6 and 7. Collapse will take place by inward bulging at the top and bottom of the frame and breaking at the sides. At some points the tensile stresses will exceed the compressive stresses given in the table.

3. **Accuracy of the Method of Calculation.**—Since small errors will have a great influence on the resulting bending moments, great care must be exercised in measuring the ordinates, in bending the frames, and in the calculations. The form in which the calculations are here presented may be considered as typical, but the integration may of course equally well be performed by the trapezoidal rule and should be checked

by graphical means. In practice it will generally be advisable to use a greater number of ordinates than in the examples here given, where the number of ordinates might with advantage be doubled in order to obtain sufficient accuracy. It must be borne in mind that Simpson's Rule does not give very accurate results where the figures in the columns change value very abruptly, as, for instance, in the intervals 2–3 and 9–10 in the column for $\frac{M}{I}$. Intermediate ordinates must, therefore, be introduced in such cases. Special care should be bestowed on the products $(x-x_2)(x^2+y^2)$ and $x(x^2+y^2)$ and derived columns, which likewise change value very rapidly. It may here be advisable, besides increasing the number of ordinates, to introduce one or two more significant figures, since ultimately the result depends on small differences. The numerical calculations given in Tables VI. and VII. must, in fact, be regarded merely as illustrative of the method.

4. **Submarine Boats of Circular Section.**—Consider first the ideal case of a circular frame of uniform section without any internal stays or other supports and unaffected by the proximity of transverse bulkheads. When such a frame is exposed to a uniform external pressure it will not be subject to bending or shearing, but a uniform compressive stress will exist everywhere, normal to the cross-sections of the frame. This stress is given by the formula

$$p_c = \frac{P}{A} = \frac{dw}{2A} \quad . \quad . \quad . \quad . \quad (37)$$

where d is the diameter of the circle formed by the neutral axis, A is the area of a cross-section including the effective strip of adjacent shell plating, and w is the pressure-load per unit length of girth on an annular belt one frame space wide. Such uniform distribution of the stresses is the one most favorable to strength, wherefore the circular section is the strongest under a uniform pressure. In a submarine boat the pressure is not indeed strictly uniform, but increases somewhat from the top to the bottom, and the most favorable form must, therefore, deviate somewhat from the circle. As explained above, this lack in symmetry of the external forces is, however, largely offset by the action of the internal weights, and the circular form is, therefore, preferred in practice, except where special reasons make it necessary to depart from it.

Breakdown of a circular frame will, in general, occur due to structural instability before the compressive stress reaches the crushing point. The frame first flattens to an oval shape, then bulges inwards over one or two large arcs, and finally collapses in the same way as when a column crushes by bending. The critical pressure-load will probably vary with $\frac{I}{d^2}$* whence we may write

$$w_0 = k\frac{I}{d^2} \quad . \quad . \quad . \quad . \quad . \quad (38)$$

where w_0 is the pressure-load per unit length of girth, I is the moment of inertia of the frame including the effective strip of shell plating, k is a coefficient to be determined from experience with other boats. For a plain cylindrical shell unstiffened by frames the critical load per unit area will accordingly vary as $\frac{t^3}{d^2}$

In the presence of transverse bulkheads, sufficiently close together, collapse will take place in smaller arcs, depending on the distance between the bulkheads. Experiments carried out by Fairbairn and others have shown that for a simple cylindrical shell without frames we may use the approximate formula

$$w_0 = k\frac{t^2}{ld} \quad . \quad . \quad . \quad . \quad . \quad (39)$$

where w_0 is the critical pressure per unit area and l is the distance between the bulkheads or other effective stiffening members.† It appears that no formula has been proposed for determining the critical load for a cylindrical shell supported by bulkheads as in a submarine boat and at the same time stiffened by frames, but it seems reasonable to assume that we may in such a case substitute $I^{\frac{2}{3}}$ for t^2 in (39), and w_0 will then be the load per unit length of girth for one frame space.

The resistance to simple crushing of a submarine boat of circular section should also be examined, for it is conceivable that the crushing pressure may be reached before collapse due to instability occurs. From (37) we obtain the crushing load

$$w_c = \frac{2f_c A}{d} \quad . \quad . \quad . \quad . \quad . \quad (40)$$

where f_c is the ultimate crushing strength of the material. Where no frames are fitted, this formula becomes

$$w_c = \frac{2f_c t}{d} \quad . \quad . \quad . \quad . \quad . \quad (41)$$

* Cotterill, *Applied Mechanics*, London, 1890, p. 320.

† Lloyd's Rules give the following formula for circular furnaces, where the length of the plain cylindrical part exceeds 120 times the thickness of the plating:

$$\text{Working pressure in lb. per sq. in.} = \frac{1{,}075{,}200t^2}{ld}$$

l, t, and d are given in inches, d is the outside diameter of the furnace.

where w_c is the pressure per square inch. It appears that the ratio $\frac{t}{d}$ in submarine boats with unstiffened shell lies, generally, between $\frac{1}{250}$ and $\frac{1}{300}$, depending on the specified maximum depth of immersion. This construction is suitable only for small boats and small depths of immersion and for the ends of larger boats.

A circular or nearly circular frame should not generally be stiffened by pillars or longitudinal bulkheads, because bending moments will then be created which do not exist in an unstiffened frame. Where for some reason such stiffening members must be introduced, the top and bottom of the frame should be reinforced by deep floors as in fig. 25.

CHAPTER IV.

STRENGTH OF INDIVIDUAL GIRDERS.

THE structure of a modern warship consists of a network of rigid surfaces or diaphragms: bulkheads, decks, sides, and the double bottom, which mutually support each other, and it is chiefly through them that the neutralisation of the forces of weight and inertia, of buoyancy and horizontal water-pressures takes place. These forces act both in the plane of and normal to the diaphragms, which must, therefore, be capable of resisting tension and compression, bending and shearing in their own plane, as well as of transmitting the normal load to the adjoining rigid boundaries on which they rest. In order to fulfil these functions, the diaphragms must be provided with a well-developed system of stiffening girders. It is, in particular, such girders, whether simple rolled bars or plate girders, that will be considered in this chapter. Girders formed by the diaphragms themselves, as for instance by transverse bulkheads, have already been discussed in previous chapters, notably in SECTION **10.**

14. SIMPLE ROLLED BARS.

1. **Bars in General.**—In calculating the strength of a bar, it is usually supposed to be loaded in the plane of the web, and the neutral axis NN of the cross-section is assumed to coincide with a line OX_0 drawn normal to the web through the center of gravity O and hence parallel with the axis of the bending moment (fig. 26). The ordinary formula for bending

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

is applied.

This method is, however, far from correct in many cases. Due to unsymmetrical form of cross-section, the neutral axis is not always normal to the web or parallel with the axis of the bending moment; nor is the load always acting in the plane of the web. There results an unsymmetrical distribution of strains and stresses, which acts prejudicially on the strength of the bar, increasing the deflection and stresses and producing several peculiar straining effects.

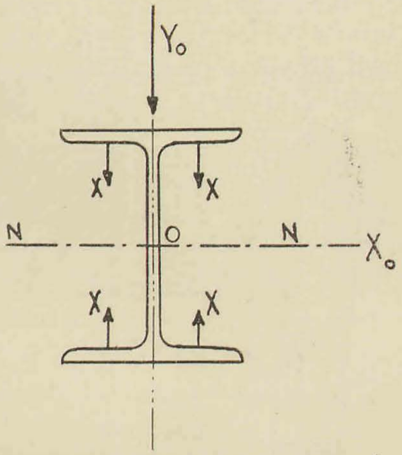


FIG. 26.

2. **The I-Bar** (fig. 26).—The I-bar is symmetrical about both principal axes, and if loaded in the plane of the web, as is generally the case, the ordinary formula for bending will apply, since the axis of the bending moment is parallel with one of the principal axes, which will then coincide with the neutral axis. When the bar deflects, the tension in the lower flange and the compression in the upper flange will call forth resultant forces X X in the flanges as explained in SECTION 12. These forces will tend to bend the flanges towards each other and will cause a slight compression of the web, but the effects will generally be insignificant.

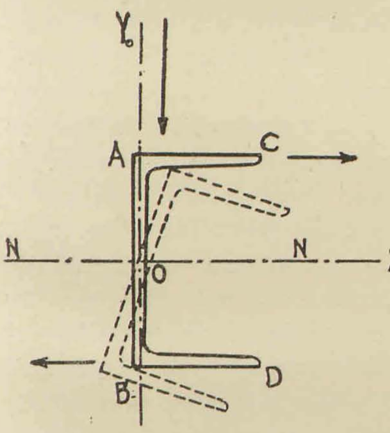


FIG. 27.

3. **The Channel Bar** (fig. 27).—This bar is symmetrical about OX_0 which is, therefore, one of the principal axes of the section. When loaded in the plane of the web, the axis of the bending moment is parallel with OX_0 whence this axis will again, as in the I-bar, coincide with the neutral axis, and the ordinary formula for bending will apply. The channel bar is not, however, symmetrical about OY_0 the line parallel with the web through the center of gravity of the section, and certain unbalanced forces will, therefore, be called into play.

As in case of the I-bar, forces X will act on the flanges, tending to bend them towards each other, but since the flanges are here on one side of the web only, there will result a bending moment on the web (fig. 28). Bending of the web is not indeed likely to take place in shipbuilding channels within the limit of elasticity, but by cold bending, where this limit is passed, it has occurred and has caused considerable difficulty and expense. It has been studied theoretically and experimentally by Mr K. G. Meldahl * by bending channel bars of different section. He found that the resistance of a channel bar to bending of the web is proportional to $\frac{b^2 t_F}{t_W^3}$ where b is the width and t_F the thickness of the flanges, t_W the thickness of the web, showing that notably the width of the flanges and the thickness of the web are determinative of this quality. Meldahl recommends that the value of this expression shall not exceed 50, a condition which is fulfilled for channels of British Standard Section of maximum web thickness, but not for those of minimum web thickness. Thus, for a $10\frac{1}{2}$ in. × $3\frac{1}{2}$ in. × $3\frac{1}{2}$ in. channel bar with a thickness of flanges of ·575 in., the quantity $\frac{b^2 t_F}{t_W^3}$ has the values of 23 and 66 corresponding to the web thicknesses ·675 in. and ·475 in. respectively.

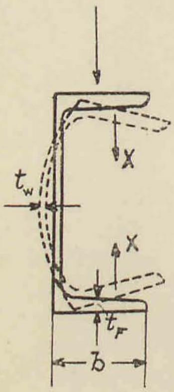


FIG. 28.

Consider now the strains in the flanges (fig. 27). These strains will be greatest near the web, along the edges A and B, and will decrease towards the outstanding edges C and D. The upper flange, being subject to greater contraction along A than along C will tend to warp, *i.e.* bend and deflect, towards the right in its own plane, and when this occurs a reduction in the compression or perhaps even a tension in the outstanding edge C will exist. The lower flange will be subject to elongation, but the edge B will elongate more than D whence this flange will tend to warp to the left. The result will be a twisting of the bar, and when this occurs the principal axis OX_0 will no longer be parallel with the axis of the bending moment, the loading will be unsymmetrical, and sideways bending will take place. This action will now be explained for the case of a zed-bar, where it is more pronounced.

4. **The Zed-Bar.**—The section of the zed-bar has no axis of symmetry. Assume such a bar, as shown in fig. 29, to be loaded in the plane of the web, which is here supposed to be originally vertical. Let the co-ordinate axes normal to and parallel with the web be OX_0 and OY_0. The bending moment then has its axis in OX_0. The principal axes of inertia, OX and OY form an angle θ with OX_0 and OY_0 respectively. OX is the axis about which the moment of inertia is a minimum, OY is the

* *Schiffb. Ges.*, 1903, p. 406.

axis about which the moment of inertia is a maximum. The angle between the axis of the bending moment and the principal axis OX is called β but since in this case the axis of the bending moment coincides with OX_0 we have $\beta = \theta$.

Under these circumstances, where *the axis of the bending moment is not parallel with any of the principal axes*, the girder will not deflect in the plane of the bending moment, nor will it deflect in the plane of any of the principal axes, but in a plane normal to the neutral axis NN which forms an angle ϕ with OX. The distribution of the stresses in the flanges will be as explained for the channel bar, but since the lower flange is on the side of the web opposite to the upper flange, both flanges will tend to deflect in the same direction in their own plane, as indicated by arrows in fig. 29. Hence the whole bar will deflect sideways, in fig. 29 to the right, while, at the same time, it will deflect a certain amount downwards. An adjustment of the stresses will take place until equilibrium is established. All parts to the right of (or below) the neutral axis will be in tension, all parts to the left of (or above) this axis will be in compression. The couple resulting from all the stress forces on the section will have its axis parallel with that of the bending couple and will be of the same magnitude.

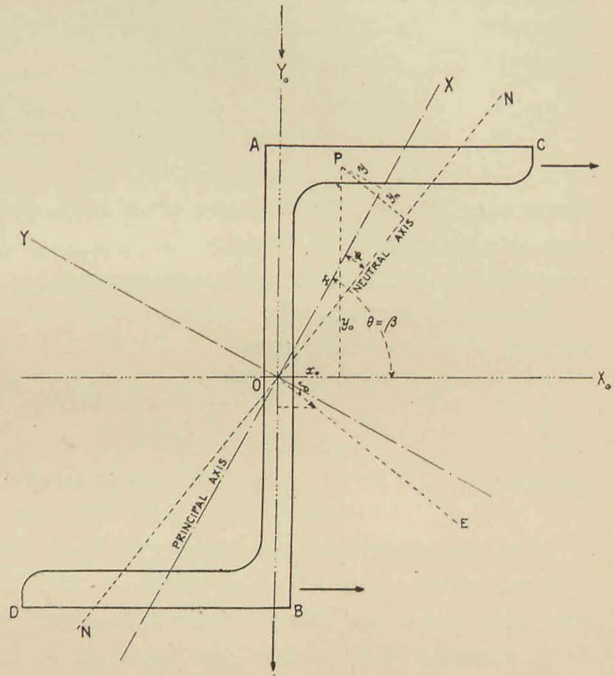


FIG. 29.

The bending takes place about the neutral axis under the action of the component of the bending moment normal to this axis, and the stress at any point P may be determined by the formula

$$\frac{M \cos(\beta - \phi)}{I_N} = \frac{p}{y_N} \quad . \quad . \quad . \quad . \quad . \quad (42)$$

where I_N is the moment of inertia and y_N the ordinate of P both referred to the neutral axis. It is, however, simpler to proceed as follows :—

Resolve the bending couple in two components, $M\cos\beta$ and $M\sin\beta$ having their axes parallel with OX and OY respectively. Then the stress at any point (x, y)—the co-ordinates referred to the principal axes —may be determined as the sum of the stresses due to each of these components, which may be assumed to act independent of each other. Hence, if I_x and I_y are the moments of inertia about the principal axes OX and OY respectively, we have

$$p = M\left[\frac{y\cos\beta}{I_x} + \frac{x\sin\beta}{I_y}\right] \quad . \quad . \quad . \quad . \quad (43)$$

which can be used without any knowledge of the exact position of the neutral axis, provided the most strained points of the section can be located by inspection, as in the present case.

From this formula the equation to the neutral axis can be found by putting $p = 0$ which gives

$$-\frac{y}{x} = \frac{I_x}{I_y}\tan\beta$$

whence

$$\tan\phi = -\frac{I_x}{I_y}\tan\beta \quad . \quad . \quad . \quad . \quad (44)$$

Now the neutral axis can be drawn in on the figure, and the ordinate y_N of the most strained fiber at points A and B can be measured. The maximum stress can then, if desired, be found from equation (42).

It remains to show how the principal axes are determined in cases where, as in the zed-bar, they cannot be located directly from symmetry. Take the center of gravity of the section as origin, and choose the initial axes so as to give a simple calculation of the moments of inertia. In the zed-bar the horizontal and vertical axes OX_0 and OY_0 fulfil this condition. Since the product of inertia about the principal axes OX and OY is zero, we have

$$\int yx\,dA = 0 \quad . \quad . \quad . \quad . \quad (45)$$

but since

$$x = y_0\sin\theta + x_0\cos\theta$$
$$y = y_0\cos\theta - x_0\sin\theta$$

we find by substitution in (45)

$$\tan 2\theta = -\frac{2Z_0}{I_{x_0} - I_{y_0}} \quad . \quad . \quad . \quad . \quad (46)$$

where $I_{y_0} = \int x_0^2 dA \qquad I_{x_0} = \int y_0^2 dA \qquad Z_0 = \int x_0 y_0 dA$

This equation gives θ the angle between the principal axis OX and the initial axis OX_0.

The moments of inertia about the principal axes are found from :

$$\left.\begin{aligned} I_x &= I_{y_0} \sin^2\theta + I_{x_0} \cos^2\theta - Z_0 \sin 2\theta \\ I_y &= I_{y_0} \cos^2\theta + I_{x_0} \sin^2\theta + Z_0 \sin 2\theta \end{aligned}\right\} \quad . \quad . \quad (47)$$

In a zed-bar not stiffened by plating and loaded in the plane of its web, the sideways deflection will often be much greater than the vertical deflection. When the bar is prevented from deflecting sideways, as in the case of a beam attached to deck plating, the deflection in the plane of the web will be much reduced, as shown by Bruhn's experiments.*

Example.—Find the stress and the deflection at the middle of a zed-beam, 6 in. × 3½ in. × 3½ in., as given in fig. 29, of 18 ft. length, fixed at the ends and loaded with a vertical uniformly distributed load of 1½ tons in the plane of the web. Using the units inches and tons, we find :

$$I_{x_0} = 29{\cdot}5 \qquad I_{y_0} = 11{\cdot}5 \qquad Z_0 = 14{\cdot}3$$

The angle between OX_0 and the principal axis OX is given by :

$$\tan 2\theta = -\frac{2Z_0}{I_{x_0} - I_{y_0}} = -1{\cdot}60$$

$$\therefore \quad 2\theta = 122^\circ \quad \text{and} \quad \theta = 61^\circ$$

Since the axis of the couple is horizontal, $\beta = \theta = 61^\circ$.

From (47) : $\qquad I_x = 3{\cdot}6 \quad \text{and} \quad I_y = 37{\cdot}4$

The angle between the neutral axis and the principal axis OX is given by

$$\tan\phi = -\frac{I_x}{I_y}\tan\beta = -{\cdot}172$$

$$\therefore \quad \phi = -9\tfrac{3}{4}^\circ \quad \text{and} \quad \beta - \phi = \theta - \phi = 51\tfrac{1}{4}^\circ$$

The moment of inertia about the neutral axis is found from

$$I_N = I_y \sin^2\phi + I_x \cos^2\phi = 4{\cdot}55$$

The distance of the extreme fiber from the neutral axis is measured from the drawing, $y_N = 2{\cdot}01$ in.

The bending moment at the middle is

$$M_0 = \tfrac{1}{24}Pl = \tfrac{1}{24}\, 1{\cdot}5 \times 12 \times 18 = 13{\cdot}5 \text{ in.-ts.}$$

* *Inst. Nav. Arch.*, 1905, i. 128.

The stress may be found from (42):

$$p = \frac{M_0 y_N}{I_N} \cos(\beta - \phi) = 3{\cdot}7 \text{ ts. per sq. in.}$$

or from (43):

$$p = M_0\left[\frac{y \cos\beta}{I_x} + \frac{x \sin\beta}{I_y}\right] = 3{\cdot}7 \text{ ts. per sq. in.}$$

where $y = 1{\cdot}60$ in. and $x = 2{\cdot}53$ in. (measured from the drawing) are the co-ordinates of the extreme fiber referred to the principal axis. The deflection takes place normal to the neutral axis and is found from:

$$\delta = \frac{M_0 \cos(\beta - \phi) l^2}{16 E I_N} = {\cdot}40 \text{ in.}$$

giving a vertical deflection: $\delta \cos(\theta - \phi) = {\cdot}25$ in., and a horizontal deflection: $\delta \sin(\theta - \phi) = {\cdot}31$ in.

15. ROLLED BARS CONNECTED TO PLATING.

Rolled bars, whatever their section, are in a ship practically always riveted to plating, which they serve to stiffen and support. At the same time the plating assists the bars in two ways: first, by reinforcing the flange to which it is attached; secondly, in case of unsymmetrical shapes, by counteracting their tendency to sideways deflection and twisting. The influence of these actions being determined, the strength is calculated as usual.

1. **Reinforcement of the Flange.**—Through the action of the connecting rivets, stresses will be induced in the plating, which is thus forced to participate in the work of the bar, but this action extends only to within a certain distance from the rivets, beyond which the plating will be practically unaffected or will shirk its duty by buckling or bulging. Hence, only a certain width of the plating on each side of the rivets can be reckoned to be effective and considered as an integral part of the bar. This width will depend primarily on the nature of the riveted connection and on the thickness of the plating, but since the stress lines are curved and since the plating may be supported and stiffened at certain points by intercostal girders or angles crossing the bar in question, the effective width will not be the same everywhere. Bearing in mind the comparative nature of strength calculations in ship construction, it appears, however, allowable to assume that this width is constant and, under ordinary conditions of spacing of the intercostal girders, proportional to the thickness of the plating.

On the basis of an analysis, undertaken by the author, of numerous bulkhead tests,* it is recommended to reckon the effective width of the plating on each side of the bar to which it is connected equal to 15 times its thickness. This estimate appears conservative, judging from Bruhn's experiments † on frames of various sections with plate bands representing the shell plating riveted to them. The total width of the plate bands was in these experiments 40 times the thickness and appeared to be fully effective in all cases except where the riveted connection between the frame and the plate was insufficient and gave way by shearing.

The width of 15 times the thickness on each side, here proposed, is to be reckoned outside the lines of rivets which connect the bar with the plating. If, then, there is only a single row of rivets, the total effective width will be 30 times the thickness, but where there are two rows of rivets, as in case of an I-bar, the width will be increased by the distance

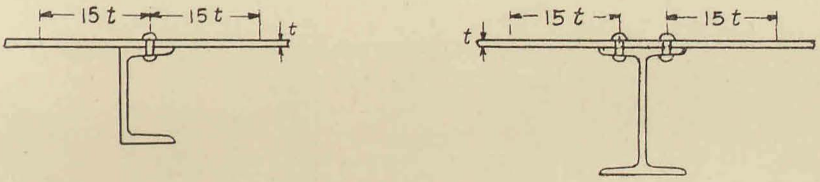


FIG. 30.

between the rows. Where the plating is supported by intercostal girders crossing the bar under consideration, the effective width may be reckoned somewhat greater.‡ Attention must always be paid to the riveted connection between the plate and the bar; if this connection is too weak, the rivets will shear before the full strength of the plate is developed. In Bruhn's experiments the rivets were in single row but were only spaced from five to six diameters apart in most cases.

2. **Tripping and Bending of the Web.**—While the plating will generally prevent sideways deflection and twisting of the bar as a whole, there will remain a tendency to distortion, called "tripping," due to the strains in the free outstanding flange. The tripping takes place away from the heel of the free flange and is probably accompanied by a certain bending of the web and turning of the flange as indicated in an exaggerated measure in fig. 31. The tripping is counteracted by the tendency of the free flange to bend towards the other flange, causing

* *Am. Soc. Nav. Arch. Mar. Eng.*, 1910.

† *Inst. Nav. Arch.*, 1905, i. pp. 140–143, pl. xxxix.

‡ See Bruhn's experiments on intercostal girders fitted under deck beams, where a plate of a width equal to 96*t* proved fully effective. The beams were there spaced 60*t* apart. *Inst. Nav. Arch.*, 1905, i. pp. 133–136.

a bending of the web and a turning of the flange in the opposite direction, as shown in fig. 32. Both actions will be greater the deeper and thinner the web and the heavier and broader the flange; they will increase also with the length of the bar and will be influenced by the bevel, if any. The resulting tendency to deformation is small with the sections ordinarily used in shipbuilding, as shown by Bruhn's experiments, but it appears that tripping is more likely to occur than bending of the web.

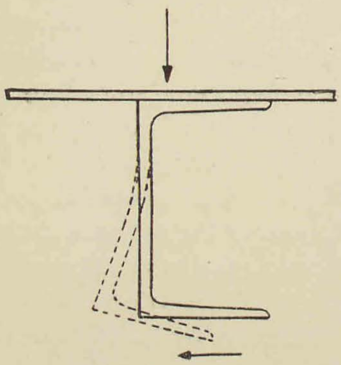


FIG. 31.

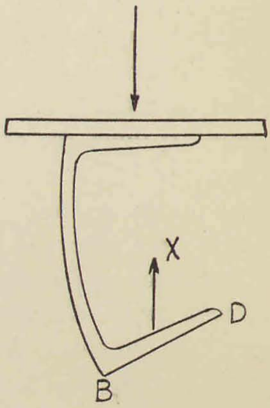


FIG. 32.

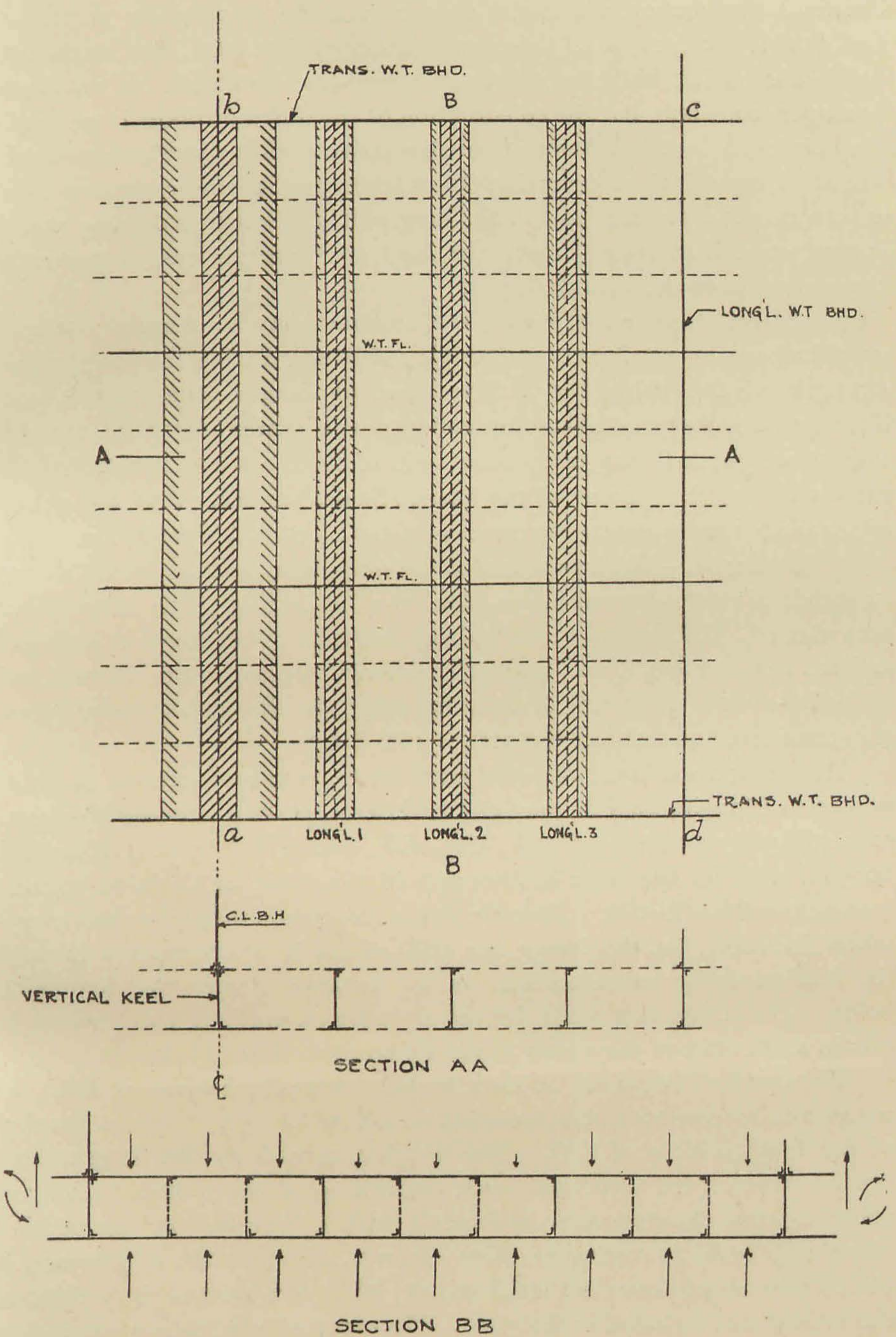
16. CONTINUOUS PLATE GIRDERS.

In a warship continuous girders constructed of plates and angles occur as bulkheads, already dealt with in SECTION 10, as floors, longitudinals, girders under the decks, and as web stiffeners on bulkheads. We shall here as a typical example show how to calculate the strength of a longitudinal.

1. **Strength Calculation for Longitudinals inside the Double Bottom.**—The longitudinals span the distance between the transverse bulkheads and are loaded with weights from the inside and water-pressures from the outside. These opposing forces do not, generally, neutralise each other completely. Free forces, in many cases of great magnitude, remain, and the resultant forces and couples have to be transmitted to the bulkheads, whereby the longitudinals will be subject to shearing and bending.

Fig. 33 gives a diagrammatic sketch of a system of longitudinals in a boiler-room of a battleship. The hatched parts indicate the effective width of the inner and outer bottom plating, determined in accordance with the rule given in last Section.

The longitudinals receive a certain support from the transverse floors, depending on the construction of these latter. With a system of longitudinals and transverse floors of perfectly uniform distribution and of



TRANS. W.T. BHD.
b
B
c
LONG'L. W.T BHD.
W.T. FL.
A
A
W.T. FL.
TRANS. W.T. BHD.
a
LONG'L. 1
LONG'L. 2
LONG'L. 3
d
B
C.L.B.H
VERTICAL KEEL
SECTION AA
SECTION BB

FIG. 33.

identical construction, we might even regard the rectangular portion of the double bottom *abcd* (fig. 33), enclosed between two transverse bulkheads, a side bulkhead, and the center-line bulkhead, as an elastic homogeneous plate, but the construction of the different floors is generally very dissimilar and their strength small relative to that of the longitudinals. We shall, therefore, assume that the sole function of the transverse floors is to transmit a certain load to the longitudinals, but in so doing we, of course, somewhat overestimate the load on these girders, in particular those nearest to the boundaries.

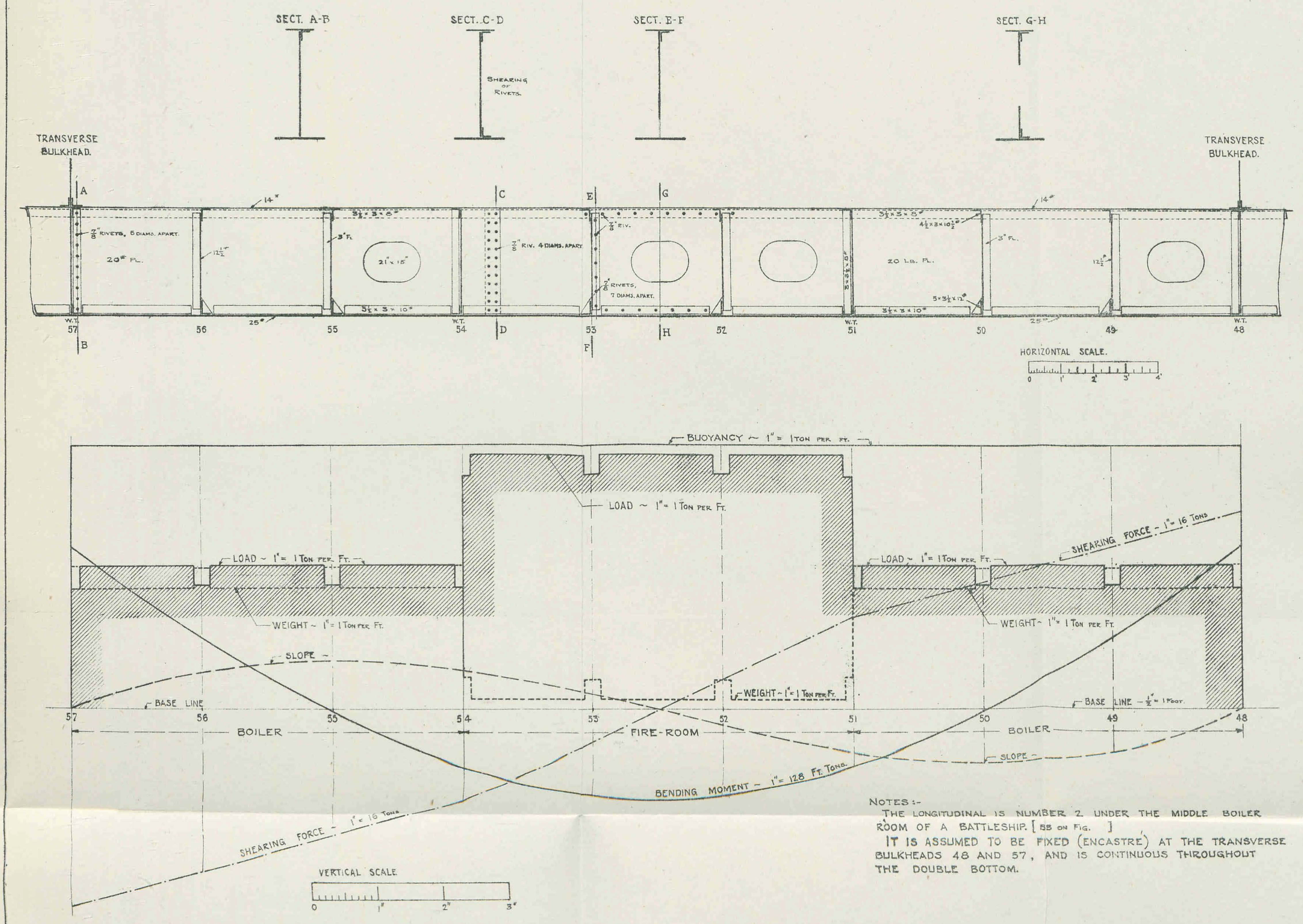
Based on these assumptions, the solution may be worked out by a graphical process similar to that employed in calculating the longitudinal strength of the ship-girder. The curve of loads is constructed, considering each longitudinal as an independent girder carrying the full load of weight and buoyancy on a belt of the bottom structure of the same width as the spacing of the longitudinals, and after that the curves of shearing force and bending moment are obtained by integration.

2. **Example.**—Determination of the maximum stresses in the second longitudinal in the central boiler-room of a battleship. The longitudinal here chosen, BB in fig. 33, is midway between the center-line bulkhead and the side bulkhead, and since it is similarly loaded in the forward and aft boiler-rooms, which are of the same length as the central boiler-room, it is considered as fixed at the ends. (Pl. III.)

The section modulus was calculated for the weakest sections in those parts of the longitudinal that are exposed to the greatest bending moments, exemplified by the sections AB, CD, EF, and GH. A strip of plating, 10·3 in. wide for the inner bottom and 18·4 in. wide for the outer bottom, was included as effective. Also the upper, continuous angle bar was everywhere included, but the lower bar only where it is continuous through the section under consideration, as for instance across the lightening holes. Deduction was made for the rivet holes, since it was desired to obtain and compare the stress at the different weakened sections.

The curve of buoyancy is seen to be a straight, horizontal line. It represents the pressure corresponding to a head of 23·5 ft. and a spacing of the longitudinals of 6 ft. The weights carried by the longitudinal consist, besides the structure itself, essentially of the boilers, which are placed against the transverse bulkheads with a fire-place in the middle. Since each boiler is resting on three seatings, of which the middle one is placed very nearly over the longitudinal BB, one-third of the weight of the boilers was included in the load. The curve of weights is symmetrical about the middle of the longitudinal. It will be seen that the forces of buoyancy are far in excess of the weights in way of the fire-place and

STRENGTH DIAGRAM FOR A LONGITUDINAL.



NOTES:-
THE LONGITUDINAL IS NUMBER 2 UNDER THE MIDDLE BOILER ROOM OF A BATTLESHIP. [55 ON FIG.]
IT IS ASSUMED TO BE FIXED (ENCASTRÉ) AT THE TRANSVERSE BULKHEADS 48 AND 57, AND IS CONTINUOUS THROUGHOUT THE DOUBLE BOTTOM.

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that, hence, the longitudinal is subject to hogging strains at the middle. The curve of loads was constructed and was integrated by means of the integraph. The curve of shearing force thus obtained could be at once transferred to its correct position in this case, since, by symmetry, the shearing force at the bulkheads must be of the same magnitude but of opposite sign.

By integrating the curve of shearing forces a curve of bending moments was obtained, but since this curve came to zero at both ends it had to be adjusted, in this case moved upwards parallel with itself, to allow for the unknown bending moments at the ends. In order to find the amount of the correction, the curve of bending moments was integrated, by which a curve of slopes was obtained. The end ordinate of this curve, divided by the length of the base-line, applying the appropriate scale factors, gave the bending moment at the ends and hence the correction needed for the adjustment of the curve of bending moments. On the diagram all the curves are shown in their correct position.

Table VIII. gives the section modulus, bending moment, and maximum stress at each of the sections indicated on Plate III.

TABLE VIII.—BENDING MOMENTS AND STRESSES IN LONGITUDINAL.

Section.	Section Modulus. Sq. in. × in.	Bending Moment. In.-ts.	Maximum Stress. Ts. per sq. in.
AB : Through rivets at bulkhead . . .	333	3830	11·5
CD : Through butt lap (shearing of the rivets)	336	1610	4·8
EF : Through rivet holes at non-watertight floor and notch for transverse frame bar .	340	2090	6·1
GH : Through lightening hole . . .	351	2150	6·0

It will be seen that an excessive or at least a very high stress is found at the bulkheads due to the great bending moments at these points. Elsewhere the stresses are moderate.

17. INTERCOSTAL GIRDERS.

The term "intercostal" refers chiefly to the web of a girder, implying that it consists of insertions between continuous frames or beams. When the web is entirely non-continuous, the girder is considered purely intercostal even if the flanges are continuous. When a considerable part of the web is continuous while the rest is intercostal, we shall refer to the girder as "semi-intercostal"; such girders often possess practically the same ultimate strength as continuous girders. In many cases a girder built as a pure intercostal may obtain an appreciable girder strength by riveting the flanges or angle bars of the intercostal parts to each other through the web of the continuous intersecting girders. Evidently a great variety of intercostal girders exist.

1. **Intercostal Girders inside the Double Bottom.**—Generally the transverse floors are intercostal, but the inner and outer shell plating form continuous flanges and sometimes the outer frame angle is continuous; the inner or reversed frames are usually intercostal and serve merely to connect the web to the inner flange. Several varieties of floors exist, described in SECTION 50. In bracket floors (fig. 129) the web is entirely broken between the brackets, wherefore these frames do not possess any girder strength whatever. In solid plate floors lightened by holes (fig. 128) the web strength depends on the riveted connection between the floors of adjacent panels. When the floor plates are flanged in opposite directions and connected to the longitudinals by openly spaced rivets, the resistance to shearing and hence the strength of the frame as a girder must be small; but when the floors are connected to the longitudinals by angle bars turned the same way and riveted directly to each other through the longitudinals, a greater measure of strength is obtained. Watertight and especially oiltight floors, connected to the longitudinals by double angles with closely spaced rivets, possess a girder strength which probably approaches that of a continuous frame (fig. 130). To the author's knowledge no experiments have been made to determine the relative strength of the different types of floors.

2. **Intercostal Girders outside the Double Bottom.**—Such girders occur as hold stringers and as girders under the decks. They are as a

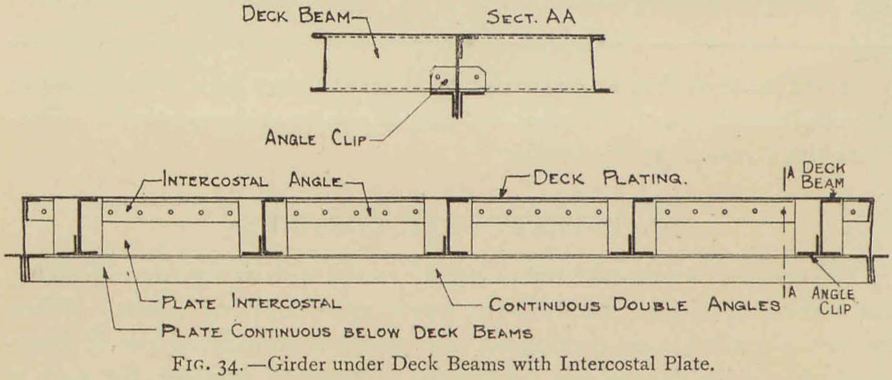


FIG. 34.—Girder under Deck Beams with Intercostal Plate.

rule semi-intercostal. Fig. 34 shows a typical girder of this kind. The web is here continuous below the beams, where two continuous angles, connected by lugs to the beams, form the lower flange of the girder. Experiments * have shown that the strength of girders constructed in this way is about the same as that of continuous girders. In fact, the strength can be estimated with practical certainty by the ordinary formula for bending, including in the moment of inertia the entire sectional area of the intercostal parts and making deduction for the rivet holes only. Failure will not occur till the calculated stress, determined in this way, reaches the ultimate stress of the material.

On the other hand, if the web is entirely intercostal, consisting of separate plates in short lengths inserted between the beams, there is an appreciable falling off in strength, and a much greater loss in strength takes place when the web plates are entirely omitted. Where intercostal girders are required to distribute the load over larger areas, they should, therefore, be provided with a partly continuous web. Purely intercostal girders should be used only where their functions are strictly local, viz. to distribute the load to the nearest continuous girders and to prevent their tripping.

3. **Intersecting Girders.**—Suppose two intersecting girders to be constructed in such a way that both of them preserve a certain continuity at the point of intersection and deflect together at this point. Consider first the simple case of the girders (1) and (2) on fig. 35 intersecting in the middle O each loaded with a uniform but different load and supported only at the ends.

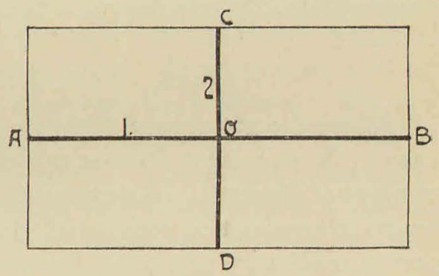


FIG. 35.

The deflection at O is given by the general formula

$$\delta = \frac{kwl^4}{EI}$$

where k is a factor which depends on the way in which the girders are supported at the ends. When a girder is freely resting on the supports $k = \frac{5}{384}$ and when it is fixed at the ends $k = \frac{1}{384}$

In order that the two girders shall deflect as if they were independent of each other, we must have :

$$\frac{k_1 w_1 l_1^4}{I_1} = \frac{k_2 w_2 l_2^4}{I_2} \qquad (48)$$

* J. Bruhn, *Inst. Nav. Arch.*, 1905, pp. 133–136 and pl. xxxvii.

which, for the same mode of support and loading, gives

$$\frac{I_1}{I_2} = \frac{l_1^4}{l_2^4} \quad . \quad . \quad . \quad . \quad . \qquad (49)$$

This shows the rapid rate of increase in moment of inertia with an increase in length of a girder if this condition is to be fulfilled.

If equation (48) is not satisfied, the relatively stronger girder will have to support the weaker by carrying part of its load. Let (1) be the stronger girder, then it has to carry: $W_1 + W_0$ while (2) has to carry: $W_2 - W_0$ where $W_1 = w_1 l_1$, $W_2 = w_2 l_2$, and W_0 is the reaction between the girders acting as a concentrated load at the middle. W_0 can be found by equating the expressions for the deflection.

Where one deep girder is intersected by a number of very light closely spaced girders, as for instance in a bulkhead with one heavy horizontal girder and a number of light vertical stiffeners, an approximate solution can be found, but so many assumptions are involved that the result will hardly be of any practical value.

18. WORKING STRESSES.

Where the girders are of simple construction and the load is definitely known, the calculated stresses can be used directly and independently as a measure of strength. As examples of such cases may be mentioned deck beams in ammunition rooms and girders fitted under the decks for the support of machinery and other known weights. From the experiments on intercostal girders, referred to above, it appears in fact that in all girders of fairly rational construction under known conditions of loading, the calculated stresses can be used directly in connection with a factor of safety.

When all static and dynamic forces are taken into account, a factor of safety of 4 may generally be used. This gives a working stress in bending of about 16,000 lb. per sq. in. or 7 ts. per sq. in. for ordinary mild steel. For the rivets the shearing stress will be about 12,500 lb. per sq. in. or $5\frac{1}{2}$ ts. per sq. in. This is in good accordance with the practice of Structural Engineers in steel railway bridges. Allowing for dead and live load, dynamic actions, and forces incidental to the special conditions, the working stresses in such structures are as follows:*

Axial tension on net section . . .	16,000 lb. per sq. in.
Bending stress	16,000 ,, ,,
Shearing stress on rivets	12,000 ,, ,,
Shearing stress on the webs of plate-girder (gross section)	10,000 ,, ,,

* *Am. Railw. Eng. Ass.*, *General Specifications*, 1910.

CHAPTER V.

STRENGTH OF A RECTANGULAR PLATE UNDER FLUID PRESSURE.

19. DETERMINATION OF THE STRESSES.

1. **Specification of the Problem.**—The plating of a ship is on the whole flat and is usually supported by a network of equidistant frames or stiffeners running at right angles to each other. The pressures vary but little within each of the areas enclosed between the stiffeners. We shall therefore limit the discussion to plane rectangular plates fixed at the edges and subject to a uniform fluid pressure. The complete determination of stresses and deformations even in this simple case is a difficult problem of which, so far, no satisfactory theoretical solution has been found. It is, however, for practical purposes unnecessary to know the stresses and strains at every point of the plate. It is of some interest to know the maximum stress and the point where it occurs, but even these facts are not necessarily required for determining the thickness of the plating, nor are they in all cases sufficient for this purpose. The information which the designer must possess is in fact less specific. In one class of cases, where conditions are such that the plate must not be overstrained at any point, it is merely required to know the head at which the elastic limit is reached. In another class of cases where a moderate permanent set can be accepted, it is required to know the head at which the permanent set has acquired a certain magnitude or at which it begins to increase rapidly. It is also sometimes of interest to know the maximum head which the plate will stand without rupture.

These heads can be determined experimentally for all cases occurring in practice, but at present the experimental material is so limited that it is impossible to give a complete and general solution. The following discussion must be regarded chiefly as an attempt at indicating the direction in which a solution may be found.

The determinative elements of the problem are h the head of water, $\mu = \frac{s}{t}$ the ratio between the length of the short side of the rectangle and the thickness of the plate, and r the ratio between the long and the short side.

2. **Influence on Strength of the Ratio between the Sides.**—Suppose the rectangular plate *abcd* (fig. 36) to be made up of a number of elemental transverse strips of which AA is the middle one. If the rectangle is of very great length relative to its breadth, the strip AA will not receive any support, directly or indirectly, from the end boundaries, but will have to carry the entire load of the pressure on its surface unassisted by adjacent material. Since the plate is fixed at the edges, the strip will behave as a beam fixed at the ends, and since it cannot slip, it must be subject to a certain tension. Hence the maximum stress in the strip will be tensile and must occur at the ends on the pressure side, and here a permanent set must first appear. If we now imagine the short edges to be brought nearer together, a point will be reached where they come to carry an appreciable part of the load, while at the same time the load on the long edges and the stresses at these edges will be to some extent relieved. It appears that the central strip must obtain the least relief in this way and that, therefore, the absolute maximum stress in the plate must be at the points AA, the middle of the long sides. Another maximum will be found on the short edges probably somewhere between the middle point B and the corner, but this maximum must always be smaller than that at the points A until the rectangle becomes a square, when the stress will be the same at A and B.

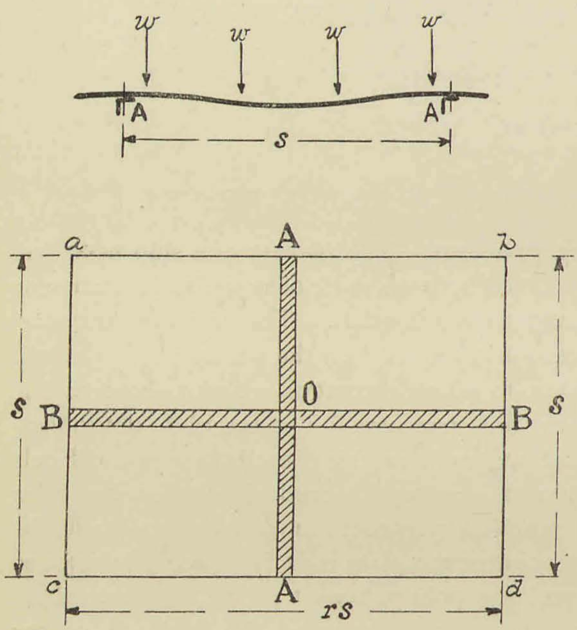


FIG. 36.

The late *Marine-Schiffbaumeister* Pietzker of the German Navy proposed to determine the stress at the points A by means of the ordinary formula for elastic bending, applying a factor to allow for the relief which the strip AA obtains from the short boundaries.* His formula gives

$$p_A = \tfrac{1}{2} K_A w \mu^2 \quad . \quad . \quad . \quad . \quad (50)$$

where w is the load per unit area of the strip and K_A is a factor which depends on r. Pietzker gave a curve for K_A reproduced on Pl. IV., based chiefly on experiments carried out by the German Navy Department and by Professor C. Bach. This curve must, however, be considered only as provisional; in fact, it does not correspond entirely with Bach's experiments, which give somewhat smaller values for K_A. According to the curve, K_A varies from ·64 for $r = 1$, *i.e.* for a square, to unity for $r = 3$. When the ratio is greater than 3 the influence of the short sides on the stress at A is negligible.

20. EXPERIMENTAL SOLUTION OF THE PROBLEM.

It is here proposed to base the determination of the thickness, not on calculated stresses, but on observed deflections, or, in other words, to pass directly from the results of experiments and tests to the design of the scantlings. The only published accurate experiments available for this purpose are those of Bach,† but we are able to supplement them with several practical bulkhead tests carried out with great care in ships of the United States Navy.

Bach's experiments were four in number and comprised two square plates: I., 800 mm. × 800 mm. × 8·4 mm., and II., 800 mm. × 800 mm. × 16·8 mm., and two rectangular plates: III., 800 mm. × 400 mm. × 8·6 mm., and IV., 800 mm. × 400 mm. × 16·5 mm. The head was increased in stages from zero to the maximum, which for one of the plates reached about 1100 feet. The total deflection as well as the permanent set were measured for each stage at a great number of points on the plates, and the stresses were calculated for the head at which the elastic limit was reached, as determined by the deflection at the center.

The distribution of the stresses along the edges of plate IV., shown in fig. 37, is seen to exhibit the characteristics which, according to the foregoing discussion, might be expected. The absolute maximum is

* *Festigkeit der Schiffe*, Berlin, 1911, p. 45.

† "Versuche über die Formänderung und die Wiederstandsfähigkeit ebener Wandungen," *Ver. Deutsch. Ing.*, 1908, pp. 1781, 1876, from which the diagrams figs. 37 and 38 are reproduced with kind permission of Professor Bach.

found at the middle of the long edges and a secondary maximum is found near the corners on the short edges. The stresses in plate III. do not, indeed, conform to this mode of distribution, but probably the behavior of this thin plate was strongly influenced by the edge fastenings. On the whole, the edge fastenings in Bach's experiments did not give conditions exactly similar to those that obtain in the plating of a ship, where complete fixity may be assumed to exist along the lines of support.

Fig. 38 gives the curves of total deflection and permanent set for plate II., measured at the center. The deflection is strictly proportional to the pressure up to a head of 79 feet. At this point the elastic limit

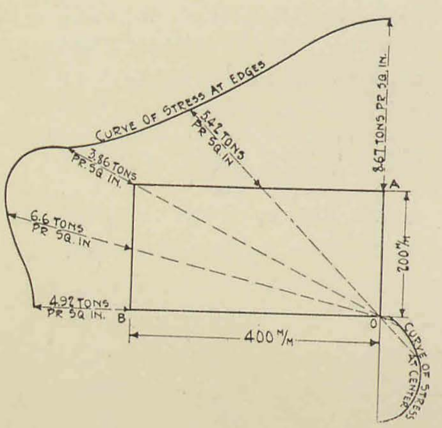


FIG. 37.—Maximum Stresses at the Edges and Center of Rectangular Plate IV. (Bach).

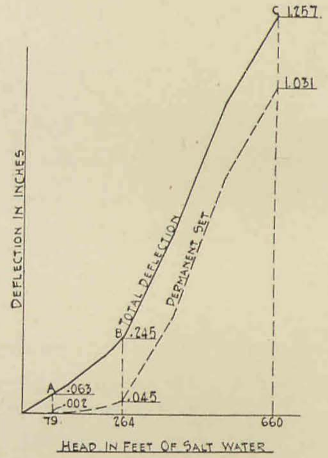


FIG. 38.—Curves of Deflection, Plate II. (Bach).

is reached somewhere in the plate, for then a permanent set begins to appear, increasing slowly until it amounts to about 18 per cent. of the total deflection, at a head of 264 feet. After that the permanent set increases rapidly and both curves rise steeply, approximating to straight lines. The vertical intercepts between the two curves represent the elastic deflection, which is fairly constant, and which, at extreme pressures, forms but a small fraction of the total deflection. The maximum head was 660 feet, at which the plate was still unbroken and did not even show any crack. The other plates exhibited similar characteristics.

There are seen to be three stages in the behavior of the plate. The first stage is one of pure elastic strain. The second is characterised by a local breakdown of the elastic strength, beginning probably at the middle of the long edges and spreading gradually to other parts of the plates. The point of transition between the second and third stages is sharply marked for plate II., as also for plate IV., but for the thinner plates I. and II. it is less pronounced. At this point the permanent set is about one-fifth of the total deflection and the tension is already an appreciable fraction of the total stress. Since the deflection is still small, the tension does not, however, carry much of the load, which is transmitted to the edges essentially by shearing. During the third stage the elastic deflection remains practically constant, whence it appears that also the tension must be constant; but since the deflection, and therefore the inclination at the edges, goes on increasing rapidly, the tension is enabled to carry a greater and greater part of the load. At the maximum head, the plate has a vaulted shape and the total load is carried chiefly in virtue of the tension. The plate is in this condition capable of supporting an enormous pressure-load, many times greater than at the point where the elastic limit is reached, a feature which is most pronounced for thin plates, and when μ is large. This fact is of great interest and may be used to advantage under certain circumstances, as explained in SECTION 70.

The following table gives the principal facts of Bach's experiments :—

TABLE IX.—BACH'S EXPERIMENTS : SUMMARY OF RESULTS.

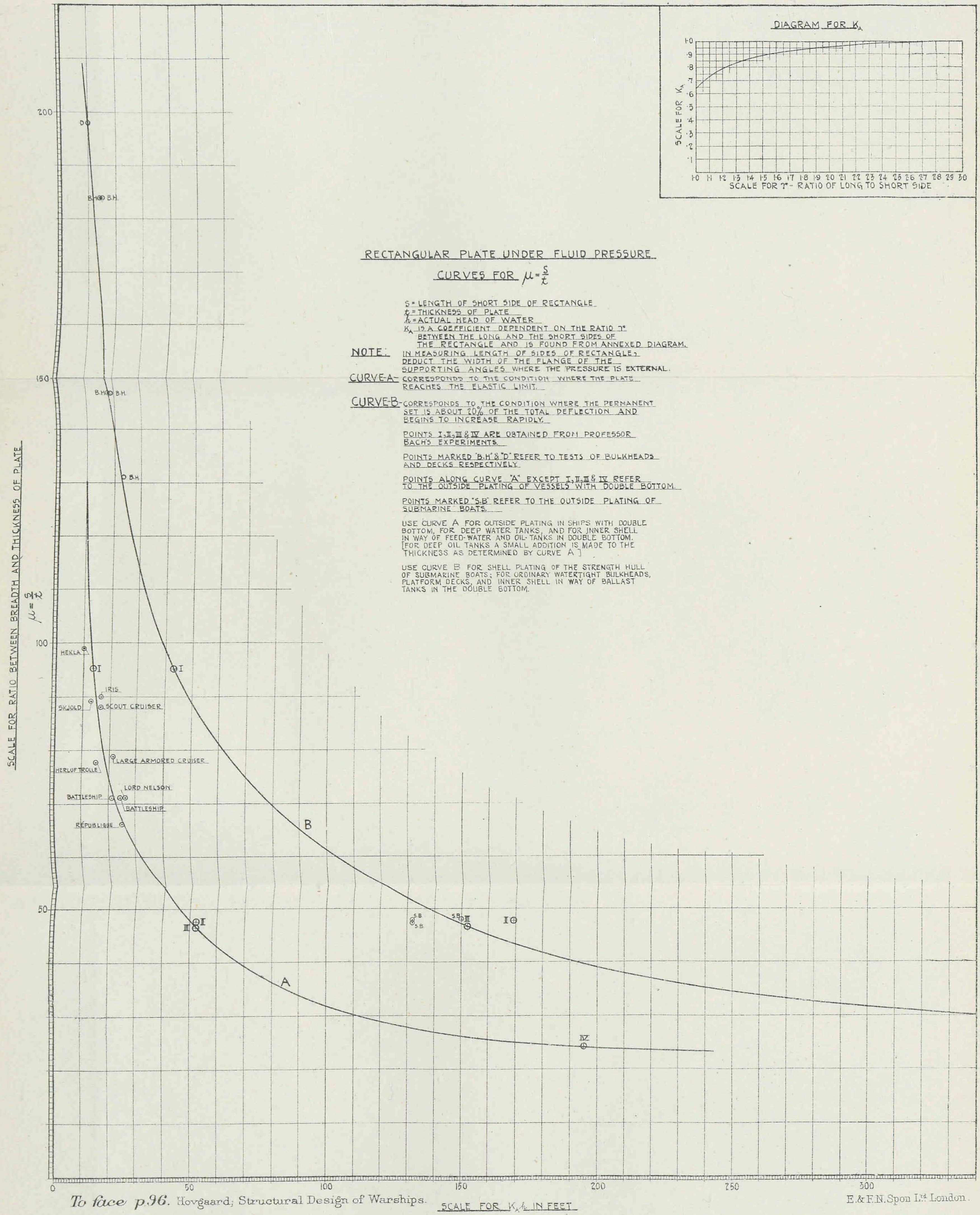
Plate.	Weight of Plating in lb. per sq. ft.	Length of Short Side in inches. s.	$\mu = \frac{s}{t}$	Ratio of Long to Short Side. r.	Head in feet at El. Limit. A.	Head in feet at B.	Head in feet at Max. Pressure. C.	Ratio between Head at C and A.	Ratio between Perm. Set and Total Defl. at B in per cent.	Total Deflection in inches at C.	Total Deflection at C, per cent. of s.
I.	13·5	31·50	95·2	1	20·0	66	792	40·0	20·6	2·2	·71
II.	27·0	31·50	47·6	1	79·0	264	660	8·3	18·3	1·3	·40
III.	13·8	15·75	46·5	2	52·8	158	924	17·5	21·7	1·1	·70
IV.	26·5	15·75	24·2	2	198·0	594	1122	5·7	20·8	0·6	·38

Since the behavior of the plate at different heads depends only on μ and r it is possible to standardise the results by referring them to a certain value of r. We shall here take $r = \infty$ as the value of reference; in other words, we take the rectangular plate in which one side is of infinite length as the standard. In order to use this method we must know the factor for any given value of r with which the actual head must be multiplied to obtain that head which in a rectangle of infinite or very great length would produce essentially the same degree of elastic and plastic deflection. Within and up to the elastic limit this factor must be practically the same as that referred to above as K_A and given on a separate diagram on Plate IV., but we shall here apply K_A not only to the head at the elastic limit, which point will be denoted by A as in fig. 38 but also, in the absence of experiments, to the head at the second point of breakdown B where the permanent set is about one-fifth of the total deflection and where it begins to increase rapidly. We are thus able to draw standardised curves, respectively A and B on Plate IV., connecting $K_A h$ and μ for these characteristic points.

Curve A is drawn through the four points obtained from Bach's experiments, after having multiplied the actual heads by K_A the values of which are ·64 for the quadratic plates and ·96 for the rectangular plates. It will be seen that, practically, the points for plates II. and III. coincide, as they should do, provided the values for K_A are correct, since μ is nearly the same for both plates. The curve for K_A is thus corroborated for that point.

In order to compare the experimental results with practice, a number of spots are plotted representing the outside plating in warships of different types provided with double bottom. In computing the value of μ, t was taken equal to the standard or ordinary thickness of the plating under the bottom amidships between the keel strake and the turn of the bilges. The span s was the frame spacing minus the width of the flange of the frame angle. The head h was the draft of the ship in normal condition, and K_A was found from the diagram on Plate IV. corresponding to the given distribution of floors and longitudinals in the respective ships. It will be seen that most of the spots fall very near the curve A showing that as a rule the outside plating is probably not strained beyond the elastic limit when the ship is at rest in still water. In the fully loaded condition and in a seaway it is likely that the elastic limit will be passed occasionally in most vessels, but probably the strain will be local, occurring only at the middle of the long sides of the rectangular fields of plating, and the permanent set will ordinarily be so small as to be of no practical importance.

Curve B is likewise based primarily on Bach's experiments and is obtained by first plotting the four points corresponding to the condition where the permanent set is about twenty per cent. of the total deflection. The curve is traced through point III. instead of between points II. and III. on account of the uncertainty in the value of K_A and because point III. cannot be moved much to the right without K_A exceeding unity. The direction of the curve is such as to pass through point IV., which falls outside the diagram. A few spots plotted for the shell plating of submarine boats corresponding to the test pressures are seen to fall very close to the curve, but the deflection at these pressures is unknown.



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It would probably not be difficult to obtain reliable spots for submarine boats corresponding to the head where the permanent set is about twenty per cent. of the total deflection. The upper part of the curve is obtained by plotting the results of a number of tests of bulkheads and one test on a platform deck, all carried out in ships of the United States Navy. The spots correspond to the maximum test head, and the permanent set was about fifteen per cent. of the total deflection. The measurements of the deflections were made with great care.

On the whole, spots that are in accordance with good practice correspond fairly well with the respective curves, although the head involves the somewhat uncertain value of K_A. The curves require of course to be further verified by experiments and eventually modified, but such as they are, they may be used with discretion for determining the thickness of the plating or the spacing of frames and stiffeners in various parts of a ship. Curve A may be used for the outside plating of ordinary ships with double bottom and, as explained in later chapters, it may also be used for the bulkheads of deep water tanks and for the inner bottom plating in feed-water and oil tanks. The lower part of curve B may be used for the shell plating of submarine boats, the upper part for the plating of ordinary watertight bulkheads, platform decks, and the inner bottom plating in way of ballast tanks in the double bottom, in fact in all cases where a moderate permanent set can be tolerated.

In ships without a double bottom, and in particular in torpedo-vessels, the thickness of the plating is determined by actions other than the pressures of the water. Spots for such vessels plotted on Pl. IV. will generally fall well below the curve A, showing that the thickness is much greater than required for carrying the water pressures.

We shall illustrate the application of the curves by two examples:

(1) A battleship is to have a draught of 28 ft., a transverse frame spacing of 8 ft. and a spacing of longitudinals of 4 ft. (SECTION 48). The shell flange of the outer frame angles is $3\frac{1}{2}$ in. wide. Determine the thickness of the outside plating.

We have $r = 2$, whence from Plate IV. $K_A = {\cdot}96$ and $K_A h = {\cdot}96 \times 28 = 26{\cdot}9$ ft. Corresponding to this value of $K_A h$ curve A gives $\mu = 65$, and since $s = 48 - 3{\cdot}5 = 44{\cdot}5$ in., we find the thickness of the plating

$$t = \frac{44{\cdot}5}{65} \text{ or about } \frac{11}{16} \text{ in.}$$

(2) In a light cruiser the mean draught is $15{\cdot}5$ ft., the standard thickness of the bottom plating is ·40 in. or about 16 lb. per sq. ft., and

the spacing of the longitudinals within the double bottom is about half as great as the spacing of the transverse frames (SECTION 48). The outer flange of the frame angle is 3 in. Determine the greatest permissible value of the spacing of the longitudinals.

We find K_A = ·96 and $K_A h$ = 14·9 ft., corresponding to which curve A gives μ = 89. Hence s = ·40 × 89 = 35·6 in. and the maximum spacing of the longitudinals is about 39 in. The same numerical result would, of course, be obtained if the frame spacing were reversed.

CHAPTER VI.

STRENGTH OF COLUMNS AND PLATING UNDER COMPRESSION.

21. COLUMNS.

1. **Ship Columns in General.**—In warships columns occur chiefly as hollow or solid "pillars" or "stanchions,"* placed as supports under the beams. The problem of column design is in ship construction less definite than in civil engineering, where the conditions under which compression members work can be more accurately determined. The magnitude of the load, statical and dynamical, which stanchions in a ship have to carry, is generally very difficult to estimate, and the structural members to which the stanchions are attached are subject to angular deflections and unsymmetrical loading which will cause unknown secondary stresses in the stanchions.

Cases are found, however, where a fair estimate can be made of the load, as when stanchions are fitted under barbettes, conning-towers, boats, or other heavy concentrated weights. Also the load on stanchions placed between the keel and the central passage admits of at least a comparative estimate. Secondary stresses can indeed be practically avoided by using pin-ended stanchions, but ship-stanchions are nearly always fixed at the ends, being riveted to the hull structure. This method is preferred, probably because it is simpler and cheaper and because it gives, at least apparently, a greater solidity. It may not, however, always be the best, and there are some cases where, as explained below, pin-ended stanchions should unquestionably be used.

Columns may break down in two distinctly different ways, by instability or by a gradually increasing strain. In both cases the breakdown is ultimately due to bending, but there is the difference that in

* The terms "pillar" and "stanchion" are here used synonymously.

the condition of instability, when the load reaches a certain magnitude, the smallest deviation from ideal central loading will cause instant and complete collapse, even although the stress is at that time everywhere within the elastic limit, whereas by the latter mode the column at moderate, eccentric loads bends over to a certain position of stable equilibrium, and as the load is increased it gradually deflects more and more until the stress at some point passes the elastic limit, when breakdown will take place.

The load at which breakdown by instability occurs is found by Euler's formula, which will be given and discussed in SECTION **22**. It applies with great accuracy to long and slender columns, and will be used here in all cases where the "length-ratio," *i.e.* the ratio between length and radius of gyration of the column $\frac{l}{r}$ is greater than 175. For solid stanchions of circular section this corresponds to a ratio, length to diameter $\frac{l}{d}$ of about 45. For moderate and smaller values of these ratios, such as occur in the stanchions of warships, columns will generally break down by simple bending before the point of instability is reached, depending always on the deviation from the ideal conditions. The formula to be used for stanchions in warships must therefore be one for bending under certain assumed conditions of eccentric loading.

2. **Causes of Secondary Stresses.**—Even with the most careful construction of a column and the most minute precautions in fitting it, the load will never be absolutely central. A small initial curvature will exist in the axis, a small eccentricity is unavoidable in the application of the load, and, even in a pin-ended column, a bending moment may be transmitted to it through the friction of the pins. In other words, conditions are never ideal in practice, and any formula, which is to determine the breaking-down point in bending, must take account of this fact. We shall here consider such small and unavoidable deviations from the ideal conditions as quite distinct from the gross deviations which often occur in practice and which, although not always subject to an exact estimate, are known or suspected to exist, and we shall speak of a column, which is not known to be eccentrically loaded and where everything is done to avoid secondary stresses, as "centrally loaded." The more serious causes of secondary stresses require to be discussed separately, in their pure form (figs. 39, 40, and 41).

(1) A column supports a beam resting freely on its top, loaded by two unequal weights P_1 and P_2 placed at such distances from the column that the beam is in equilibrium. Evidently the shearing forces P_1 and

P_2 acting on the beam on each side at the top of the column, will be unequal, and hence the load $P = P_1 + P_2$ will be eccentrically applied.

(2) A column supports a beam which is rigidly connected to it and

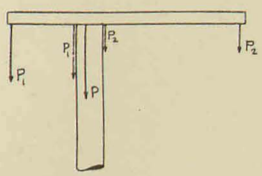
FIG. 39.

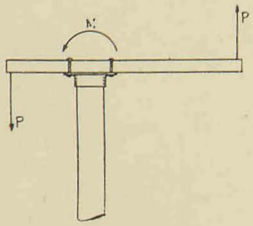
FIG. 40.

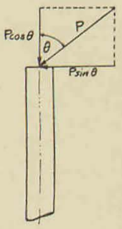
FIG. 41.

acted upon by two equal and opposite forces P. In this case the pillar is not loaded vertically at all, but will be subject to the action of a pure couple M.

(3) A force P directed obliquely at an angle θ to the axis, is applied at the center of the top of a column. Resolving the force into a vertical and a horizontal component, we obtain a central axial force $P\cos\theta$ which produces simple compression in the column, and a transverse force $P\sin\theta$ which produces a bending moment increasing from the top downwards.

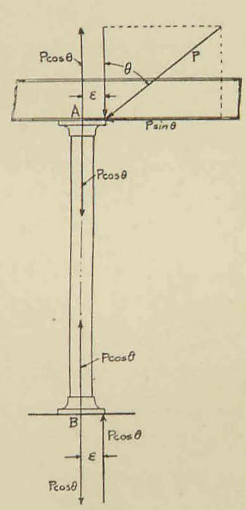
FIG. 42.

All these actions may coexist in a ship-stanchion fixed at the ends. The beam supported by the stanchion may be unsymmetrically loaded, causing an eccentric force parallel with the axis, it may be subject to angular deflection producing a pure bending moment in the stanchion, and when the ship is in an inclined position or when the structure suffers distortion for some reason, the load may be obliquely applied. Whatever the cause of these actions they may be represented by a single resultant force P acting on the top of the stanchion at a certain distance ϵ from its center, and directed at an angle θ with its axis. Resolve P into its vertical and horizontal components, $P\cos\theta$ and $P\sin\theta$ and imagine two equal and opposite forces $P\cos\theta$ to be acting at A the upper end of the axis. It is then seen that the force P is equivalent to a central load $P\cos\theta$ a couple $P\epsilon\cos$ and a transverse force $P\sin\theta$ acting at A.

It is clear that a stanchion is ill adapted to resist a transverse force such as $P\sin\theta$ which will cause it to deflect sideways as indicated in fig. 43, and a highly undesirable state of loading will come to exist, consisting in a combination of a bending moment and eccentric loading. In a warship, however, the stiffness of the hull is ordinarily so great that the distortion of the structure is arrested before the deflection of the stanchion becomes appreciable. In general, therefore, the transverse forces may be neglected, but cases occur where they must be considered and provided against by diagonal bracing or otherwise, as when stanchions are fitted under skid beams for supporting the boats.

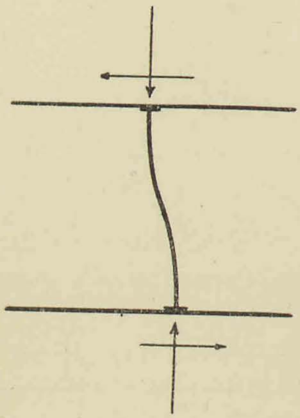

FIG. 43.

Consider next the couple $P\epsilon\cos\theta$ which is transmitted to the stanchion through its head. If the stanchion is rigidly fixed at the lower end B the couple will there be resisted by another couple of exactly equal magnitude and of opposite sign, which may be represented by two forces $P\cos\theta$ one acting downwards at B the other acting upwards at a distance ϵ from B. Since an upwards directed force $P\cos\theta$, the reaction of the central load, already acts at B the downwards force due to the couple will be neutralised. Thus we are finally left with two free forces, of magnitude $P\cos\theta$, one at the top and one at the bottom, acting in opposite directions along the same vertical line at a distance ϵ from the axis of the stanchion, and these two forces, together with the lateral force $P\sin\theta$ are the resultants of all the forces that act on the stanchion. In other words, neglecting the lateral force, we have the pure case of an eccentric load producing throughout the stanchion an axial compression and a uniform bending moment. If the stanchion is hinged at the lower end, the reaction due to the couple will here take the form of a horizontal force F as shown in fig. 44, since no couple can exist at B other than the small one due to pin friction. This force must fulfil the condition: $Fl = P\epsilon\cos\theta$ and the result will be a bending moment increasing from B to A. If the stanchion is "fixed" at the lower end but the structure to which it is attached is of a yielding nature, while still offering some resistance, the result will be intermediate between that for rigid attachment and pin-end. Both a couple and a horizontal reaction will then act at B producing a combined uniform and increasing bending moment in the stanchion.

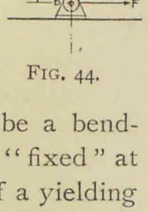

FIG. 44.

In ship construction pillars are either fixed at both ends or hinged at both ends, and we need only consider the horizontal reaction F when the ends are fixed and when the structure is at the same time very yielding, but in warships the stiffness of the structure is generally so great relative to that of the stanchions that F can be neglected. *We are, therefore, justified in disregarding all forces normal to the stanchions and need only consider central or eccentric loading, parallel with the axis.**

Where there is known to be a considerable eccentricity of loading it is best to use pin-ended columns in order to avoid secondary stresses. Such may be the case in submarine boats when stanchions are fitted on the sides as shown in fig. 25, and as further explained in the example at the end of this Section.

3. **Moncrieff's Formula.**—Consider first the "centrally loaded" column. We have seen that a formula for calculating the ultimate load on a centrally loaded column should make allowance for the unavoidable small deviations from ideal conditions which always exist in practice. Such allowance must necessarily be of an empirical nature, but should be based on experiments and experience. The formula should otherwise rest on a rational basis and should conform to all the recorded reliable experiments. The formula proposed by Mr J. M. Moncrieff† fulfils these conditions. It is based on more than two thousand tests made by various experimenters, including practically all reliable tests up to the year 1900, and is corroborated by the results of later tests. Moncrieff adjusted the amount of assumed eccentricity so that the formula came to correspond with the lower limit of the probable ultimate strength of the columns as determined by the experiments. If, for columns of a certain material, a curve is constructed from the formula on the length-ratio as abscissæ and the unit load as ordinates, it will fall just below all the spots which represent the points of failure in the experiments. Hence, irrespective of the way in which the formula is derived, it represents as a matter of fact the results of the experiments. Fig. 45 gives one of Moncrieff's diagrams, illustrating his method.

We shall not here go through the complete development of the formula, but only indicate the principal steps and assumptions. The formula is derived from the fundamental equation

$$f_c = \frac{P}{A} + \frac{P(e+\Delta)y}{I}$$

or

$$f_c = p\left(1 + \frac{(e+\Delta)y}{r^2}\right)$$

* This question has been here dealt with rather fully, because it has been recently the subject of much controversy. See various articles in *International Marine Engineering* during the years 1910, 1911, 1913, 1914, by Mr R. Earle Anderson and Mr A. J. Murray.

† "The Practical Column," *Am. Soc. Civ. Eng.*, 1901, vol. xlv.

where f_c is the ultimate compressive stress of the material, which for steel, wrought iron, and compositions may be taken equal to the ultimate

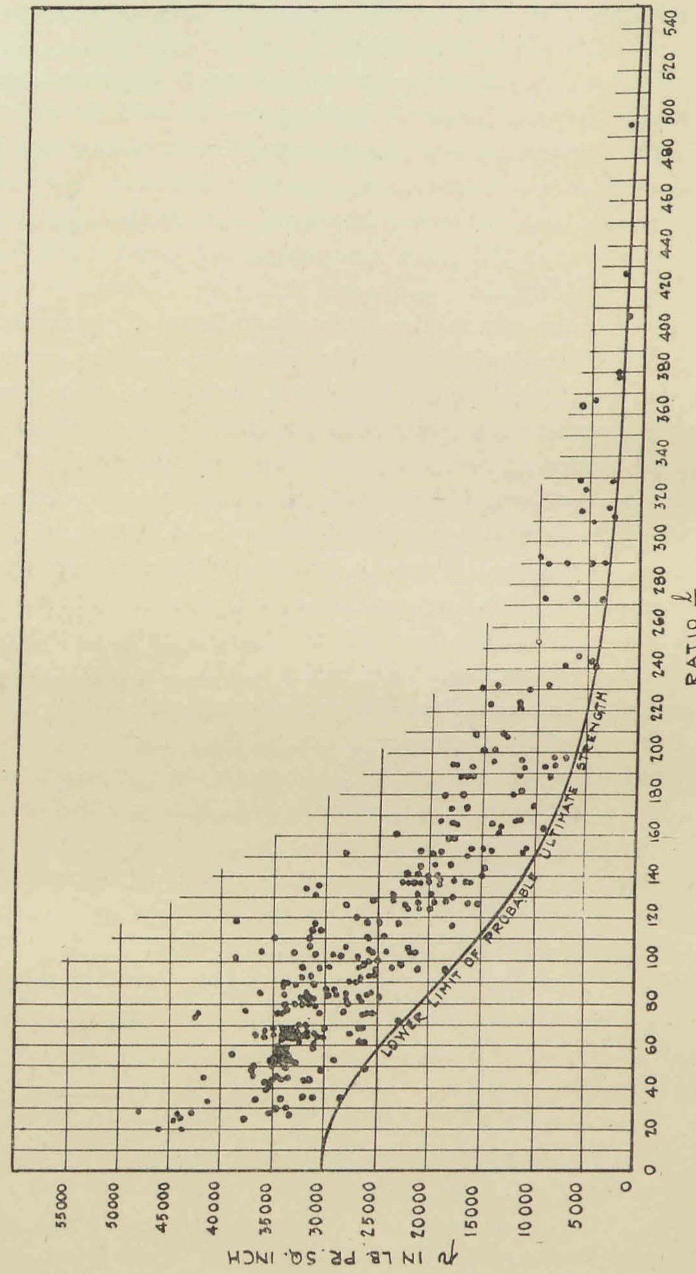

FIG. 45.—Tests of Wrought-Iron Columns with Hinged Ends (*Am. Soc. Civ. Eng.*, 1901, p. 403). (From Moncrieff's Essay on the Practical Column.)

tensile stress. For the various kinds of wood, the ultimate compressive stress is given in Table V. The quantity e is the eccentricity, comprising the actual eccentricity of the load as well as that due to the initial curvature of the column. p is the average or unit load, and with the value assigned to e by Moncrieff it becomes the lowest unit load at which the column according to the experiments is likely to fail. Δ is the deflection at the middle of the column.

Moncrieff now assumed that the curve of deflection is a parabola, whence, for a column with rounded ends,

$$\Delta = \frac{pel^2}{8Er^2 - \frac{5}{6}pl^2}$$

The value of E the modulus of elasticity, to be used in this formula for different materials, is given in Table IV. Substituting this value of Δ in the expression for f_c and solving for $\frac{l}{r}$ we obtain

$$\frac{l}{r} = \sqrt{\frac{48E\left(\frac{f_c}{p} - 1 - \frac{ey}{r^2}\right)}{5f_c + p\left(\frac{ey}{r^2} - 5\right)}}$$

Based on the experimental results, Moncrieff assigned to the term $\frac{ey}{r^2}$ the value ·6 which for a solid cylindrical column corresponds to an eccentricity of $\cdot 3r$ and he thus obtained the formula

$$\frac{l}{r} = \sqrt{\frac{9 \cdot 6E(f_c - 1 \cdot 6p)}{p(f_c - \cdot 88p)}} \quad . \quad . \quad . \quad . \quad . \quad (51)$$

From this formula the Column Tables issued by the Bureau of Construction and Repair of the United States Navy are calculated.* It holds good whenever columns fail by excessive compressive stress, which alone need to be considered in ship-stanchions where the materials are wrought iron or steel, and where flat-ended columns are not used. In cast-iron and flat-ended columns failure by tensile stress may occur, and certain changes of sign must be made in the formula.

When columns are loaded eccentrically and the eccentricity ϵ is known, it may be taken into account by adding ϵ to the unavoidable eccentricity

* These Tables were prepared by Mr R. Earle Anderson, who compared Moncrieff's formula with later tests and proposed a change in his method of applying the factor of safety. A full exposition of Mr Anderson's work in this connection is found in an essay entitled, "Column Tables for Ship Work," which he contributed to *International Marine Engineering*, 1910, pp. 409, 514, and 1911, p. 25.

e in the foregoing formulas. Giving to the term $\frac{ey}{r^2}$ the value ·6 as before, we obtain the equation for eccentric loading

$$\frac{l}{r} = \sqrt{\frac{9{\cdot}6\text{E}\left[f_c - p\left(1{\cdot}6 + \frac{y\epsilon}{r^2}\right)\right]}{p\left[f_c + p\left(\frac{y\epsilon}{5r^2} - {\cdot}88\right)\right]}} \quad . \quad . \quad . \quad (52)$$

4. **Column Curves.**—Based on formula (51) for the centrally loaded column with rounded ends, the curves in figs. 46 and 47 are prepared.* The curves can be used for columns fixed at both ends by simply entering with $\frac{l}{2r}$ as argument instead of $\frac{l}{r}$. Where it is suspected that true fixity of the ends is not attained, a value intermediate between $\frac{l}{r}$ and $\frac{l}{2r}$ should be used.

By means of these curves we avoid altogether the cumbersome formula and place the problem on the same plane as by using the simple so-called "straight-line" formulas adopted by Civil Engineers.

As apparent from the curves, columns with a small length-ratio if of high-tensile steel will carry a much greater load than such columns of mild steel, while long and slender columns will carry about the same load whatever be the quality of the steel. This is because the dominating factor at small values of $\frac{l}{r}$ is f_c—really the limit of elasticity—which is much higher in the harder grades of steel, while the dominating factor at great values of $\frac{l}{r}$ is the modulus of elasticity, which is practically the same in all kinds of steel. This was borne out by some experiments undertaken by Commander Y. Hiraga of the Japanese Navy,† who found that for ratios of $\frac{l}{d}$ from 12 to 26, columns of high-tensile and nickel steel were nearly 40 per cent. stronger than mild steel, but this advantage decreased with the length of the column, and for very long columns all three materials exhibited practically the same resistance to collapse.

* The curves in fig. 46, as well as the following recommendations concerning the construction of columns, are given in Mr Anderson's essay, referred to above. Fig. 46 has been reproduced from *International Marine Engineering*, 1911, p. 27, by kind permission of the Aldrich Publishing Company.

† *Jap. Soc. Nav. Arch.*, 1913.

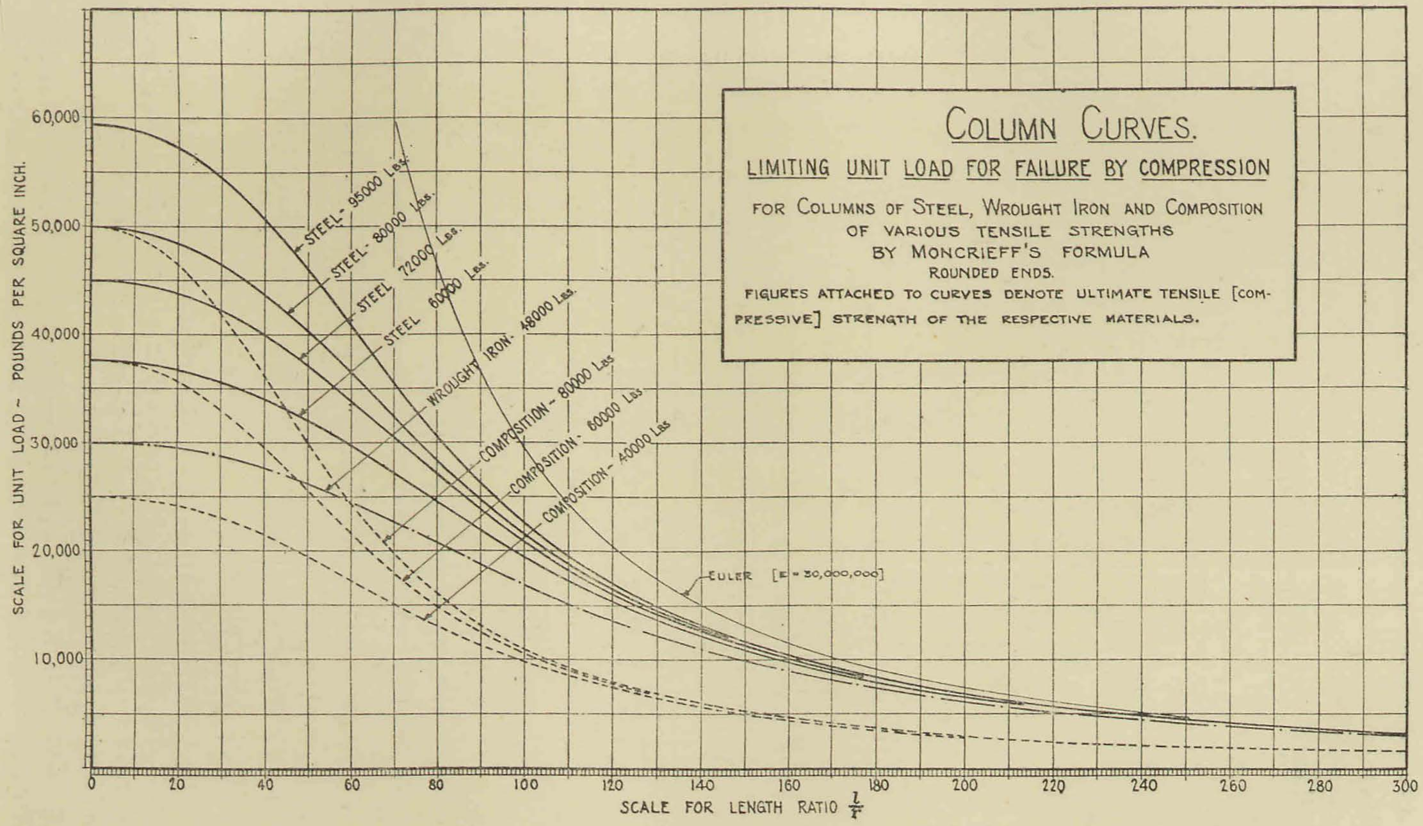
COLUMN CURVES.
LIMITING UNIT LOAD FOR FAILURE BY COMPRESSION
FOR COLUMNS OF STEEL, WROUGHT IRON AND COMPOSITION
OF VARIOUS TENSILE STRENGTHS
BY MONCRIEFF'S FORMULA
ROUNDED ENDS.
FIGURES ATTACHED TO CURVES DENOTE ULTIMATE TENSILE [COMPRESSIVE] STRENGTH OF THE RESPECTIVE MATERIALS.
STEEL - 95000 LBS.
STEEL - 80000 LBS.
STEEL 72000 LBS.
STEEL 60000 LBS.
WROUGHT IRON - 48000 LBS.
COMPOSITION - 80000 LBS.
COMPOSITION - 60000 LBS.
COMPOSITION - 40000 LBS.
EULER [E = 30,000,000]
SCALE FOR UNIT LOAD ~ POUNDS PER SQUARE INCH.
60,000
50,000
40,000
30,000
20,000
10,000
0
20
40
60
80
100
120
140
160
180
200
220
240
260
280
300
SCALE FOR LENGTH RATIO $\frac{l}{r}$

FIG. 46.

For columns of non-uniform cross-section fixed at the ends the area and radius of gyration should be computed from the smallest cross-section. Where pin connection is used and, in general, where the plane of failure can be predicted from the construction of the end attachments,

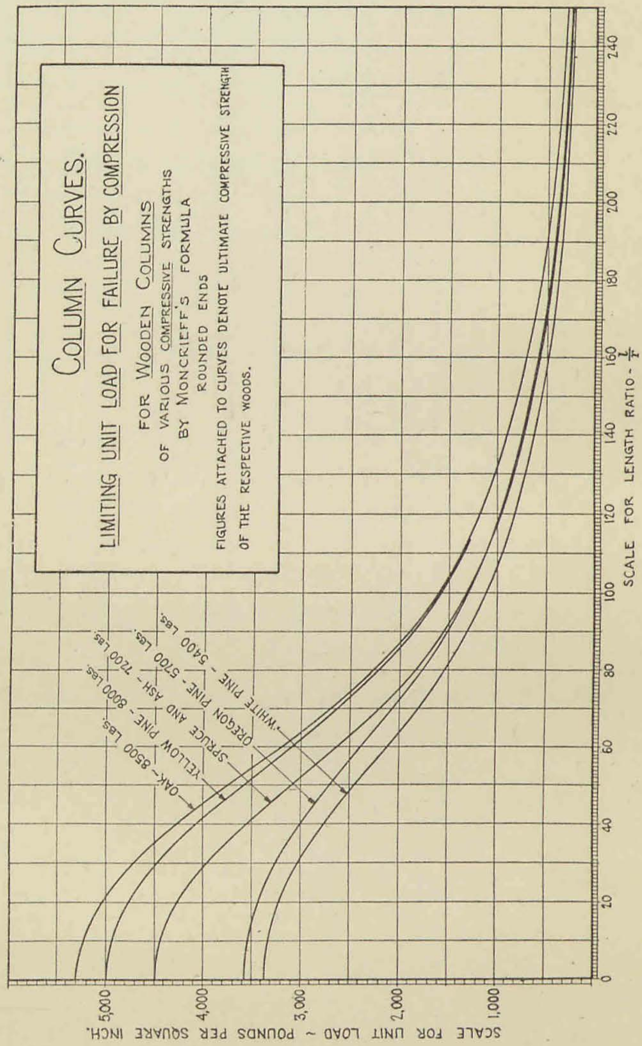

FIG. 47.

the radius of gyration to be used as argument is that corresponding to the probable plane of failure, provided it is not greater than twice the least radius of gyration of the cross-section, in which case the latter should be used.

5. **Construction of Columns.**—Fixed-ended columns should be of uniform cross-section throughout. Pin-ended columns may be tapered

towards the ends. In columns or other compression members, built up of plates and angles, the thickness of each part should be at least one-thirtieth of the distance between its connections to supporting parts. In the case of outstanding parts the thickness should be at least one-tenth of the outstanding breadth. In tubular columns the thickness of the wall should not be less than one-thirtieth of the diameter. These rules for the thickness have for object to prevent wrinkling and buckling ; they are clearly defined by the sketches on fig. 48.

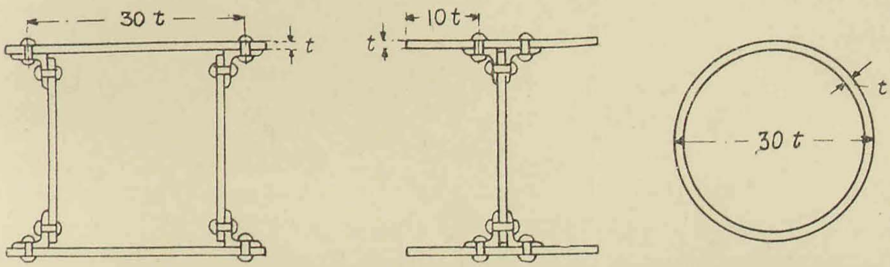

FIG. 48.

6. **Factors of Safety.**—Moncrieff's formula, used with the value of assumed eccentricity proposed above for "central" loading, gives the smallest load for which a column, according to all recorded reliable experiments, is liable to fail. It differs in this respect radically from other formulas which conform to the average and not to the lowest results of the experiments. This must be borne in mind in selecting the factor of safety, for evidently it is permissible here to use a smaller factor than usual. The load determined by Moncrieff's formula is probably very near that at which the elastic limit is reached at some point in a column and it seems therefore reasonable to reduce the factor of safety in about the same proportion as the ratio between the ultimate strength and the elastic limit. Instead of using, for instance, a factor of safety of four, as would ordinarily be employed where the load, including both static and dynamic forces, can be estimated with reasonable certainty, it is here recommended to use a factor of from two to three, varying in accordance with the nature of the material. Where the load cannot be accurately determined, a greater factor of safety should be used. Wherever a direct comparison can be made with another column, placed under similar conditions in a completed vessel, it should be used to determine the factor of safety.

The factor of safety is in practically all problems in engineering applied to the stress, but in most cases this is tantamount to applying it to the bending moment or the load. For instance, in the ordinary formula for elastic bending the result will be the same whether we apply the factor of safety to the bending moment or to the stress. Moncrieff's formula may be regarded as simply expressing the result of a great number of experiments, and any attempt at determining or adjusting the stresses in a column is futile. It seems best and simplest, therefore, to apply the factor of safety directly to p the unit load on the column.*

7. **Compression Members in Railway Bridges.**†—It is of interest to compare the rules here given with the practice of Civil Engineers. In the construction of railway bridges American railway engineers allow an axial compression on the gross section of columns, determined by the formula

$$p_w = \left(16000 - 70\frac{l}{r}\right) \text{ lb. per sq. in.} \qquad (53)$$

This is a "straight-line" formula, and p_w is the working stress or unit load, which is not to exceed 14,000 lb. per sq. in. It will be seen that here also the factor of safety is applied to the unit load and not to a stress involved in an elaborate formula. Comparing the values of p_w obtained from this formula with those determined by the Column Curves, it is found that, for ratios of $\frac{l}{r}$ between 30 and 150, p_w gives a factor of safety of from $2\frac{1}{2}$ to 2, which is in good accordance with the rule given above.

The length of main compression members in railway bridges is not allowed to exceed one hundred times their least radius of gyration $\left(\frac{l}{r} < 100\right)$ and at this limiting value of $\frac{l}{r}$ the compressive working stress is not allowed to exceed 9000 lb. per sq. in. Since the Column Curves give a unit load of about 19,500 lb. per sq. in. for this length-ratio for ordinary structural steel, the factor of safety, according to Moncrieff's formula, would in this case be 2·2, which, again, conforms to the rule here proposed.

8. **Examples.**—*Example* 1.—Determine the factor of safety in an 8-in. pipe stanchion under the central passage in a 22,000-tons battleship.

External diameter of stanchion $8\frac{5}{8}$ in., thickness $\frac{1}{2}$ in. Radius of gyration, $r = 2{\cdot}88$ in. Length 17 ft. 7 in. = 211 in. Regarding the stanchion as fixed at the ends, the length-ratio to be used for the Column

* The mode of selecting and of applying the factor of safety here recommended is in accordance with the practice of the Bureau of Construction and Repair of the United States Navy. See Mr E. Anderson's essay in *Int. Mar. Eng.*, 1910, pp. 516–517.

† *Am. Railw. Eng. Ass.*

Curves is $\frac{l}{2r} = 36{\cdot}6$. From the curves the ultimate unit load with $f_c = 60{,}000$ lb. per sq. in. is $p = 34{,}500$ lb. per sq. in. Sectional area of stanchion, A = 12·76 sq. in. Hence the smallest total load at which the stanchion is likely to break down is

$$P_0 = \frac{12{\cdot}76 \times 34500}{2240} = 196{\cdot}5 \text{ tons}$$

Let us assume that when the ship is in dock, one-quarter of the total weight of the ship is distributed uniformly on the central keel, and that the length of support of the blocks is 416 ft. There is a stanchion on every frame, *i.e.* for every four feet, whence the load on each stanchion will be

$$P = \frac{22000 \times 4}{4 \times 416} = 52{\cdot}9 \text{ ts.}$$

and the factor of safety $\frac{196{\cdot}5}{52{\cdot}9} = 3\frac{3}{4}$

This factor, of course, is of value chiefly as a means of comparison with other similar cases.

Example 2.—To examine the strength of the pillars in a submarine boat.

In the example worked out in SECTION 13 for the strength of a frame section of a submarine boat supported by two pillars, one on each side, the load on each pillar was found, Y = 15·50 ts. Length of pillar, if fixed at the ends, l = 8 ft. $1\frac{1}{2}$ in. = 97·5 in. Extreme diameter 5 in., thickness $\frac{3}{8}$ in. Radius of gyration, r = 1·64 in. Sectional area, A = 5·45 sq. in. $f_c \doteq 60{,}000$ lb. per sq. in. = 26·8 ts. per sq. in.

(1) Suppose first that the pillar is fixed at the ends and loaded centrally, then, corresponding to $\frac{l}{2r} = 29{\cdot}7$, the Column Curves give

$$\begin{aligned} p &= 35500 \text{ lb. per sq. in.} \\ &= 15{\cdot}8 \text{ ts. per sq. in.} \end{aligned}$$

whence

$$\text{Factor of safety} = \frac{5{\cdot}45 \times 15{\cdot}8}{15{\cdot}5} = 5{\cdot}6$$

(2) Actually the load is applied eccentrically, since the vertical shearing force to the left of the pillar is 30 × ·536 = 16·08 ts. acting inwards, while the shearing force immediately to the right of the pillar is 16·08 − 15·50 = ·58 ts. acting outwards. We shall assume that

the resultant of these actions is a compressive force 15·50 ts. acting on the pillar at each end at a distance from the center, $\epsilon = 2·32$ in.

Hence $$\frac{y\epsilon}{r^2} = \frac{2·5 \times 2·32}{(1·64)^2} = 2·16$$

Let E = 13,500 ts. sq. in., and substitute in the formula (52) for eccentric loading,

$$(29·7)^2 = \frac{9·6 \times 13500[26·8 - p(1·6 + 2·16)]}{p\left[26·8 + p\left(\frac{2·16}{5} - ·88\right)\right]}$$

which gives the quadratic equation

$$p^2 - 1293p + 8790 = 0$$

whence $$p = 6·8 \text{ ts. per sq. in.}$$

and $$\text{Factor of safety} = \frac{5·45 \times 6·8}{15·5} = 2·4$$

It is seen that due to the eccentric loading the carrying power of the pillar is reduced by 57 per cent.

(3) Suppose the pillar to be pin-ended and $l = 91$ in. between the axes of the pins, then $\frac{l}{r} = \frac{91}{1·64} = 55·5$, corresponding to which the Column Curves give $p = 30700$ lb. per sq. in. = 13·7 ts. per sq. in., and we get

$$\text{Factor of safety} = \frac{5·45 \times 13·7}{15·50} = 4·8$$

showing that the pin-ended pillar under these circumstances will carry safely twice as great a load as the fixed-ended, eccentrically loaded pillar.

Example 3.—A 5-in. deck stanchion is placed under the beam of a deck, loaded with 300 lb. per sq. ft. The beams are spaced 4 ft. apart and the greatest unsupported length is 18 ft. The ends of the stanchions are supposed to be fixed.

Load on stanchion $$\frac{300 \times 18 \times 4}{2240} = 9·64 \text{ ts.}$$

Length of stanchion, 81 in. External diameter, 5·56 in. Internal diameter, 5·05 in. The Column Curves give an ultimate load of 69·3 ts. for central loading, or a factor of safety of 7·2. If we assume the load to be applied at the periphery of the stanchion, 2·65 in. from the axis, the factor of safety is found to be 3·9, *i.e.* the carrying power is reduced to about one-half of its value under a central load, but there is still a good margin of safety.

22. PLATING UNDER COMPRESSION.

1. **Simple Compression.**—We shall deal first with plating under a simple compressive stress when no shearing force is acting, such as may be found in the deck and bottom plating amidships in a vessel subject to longitudinal bending. The plating is supported by beams or frames running normal to the direction of the stress, but is supposed to be otherwise unstiffened. This condition is probably fulfilled if the part of the plating under consideration is at least forty times the thickness distant from the nearest stiffeners on either side parallel with the direction of the stress (SECTION 5, *10*). We may then regard an elemental strip, spanning the distance between two beams or frames, as an independent column of radius of gyration $r = \frac{t}{3{\cdot}46}$ but the degree of fixity as well as the eccentricity of loading of such a column are difficult to determine. Where the plating is under a considerable normal load, its column strength will be destroyed. For instance, the outside plating under the bottom possesses probably no resistance to compression whatever, except where it is under the direct stiffening influence of longitudinal members. The experimental material concerning these questions is very limited, and a fully satisfactory solution of the problem cannot be given at present. Of the various formulas which have been proposed to deal with it, Euler's formula seems to be the most suitable. It is at once rational and simple and is found to give excellent results for very long columns such as those here under consideration, where $\mu = \frac{l}{t}$ is usually greater than 50 and, hence, $\frac{l}{r}$ greater than 175.

For a column with rounded ends Euler's formula is

$$p = \frac{\pi^2 E}{\left(\frac{l}{r}\right)^2} \quad . \quad . \quad . \quad . \quad . \quad (54)$$

which for plating, with E = 13,500 ts. sq. in., may be written

$$p = \frac{11100}{\mu^2} \text{ ts. per sq. in.} \quad . \quad . \quad . \quad (55)$$

where p is the critical unit load or average stress at which buckling will occur. When the ends are fixed, the formula is

$$p = \frac{4\pi^2 E}{\left(\frac{l}{r}\right)^2} = \frac{44400}{\mu^2} \text{ ts. per sq. in.} \quad . \quad . \quad . \quad (56)$$

The latter formula is represented by a curve on fig. 49, the former on fig. 46. When the plating is continuous over a number of beams or frames, it may probably be considered as being fixed at these, *i.e.* a strip of plating of length equal to the spacing between the stiffening members may be treated as a column fixed at the ends to which (56) can be applied. Generally a factor of safety of 4 may be used, and a factor of 3 should be considered a minimum value.

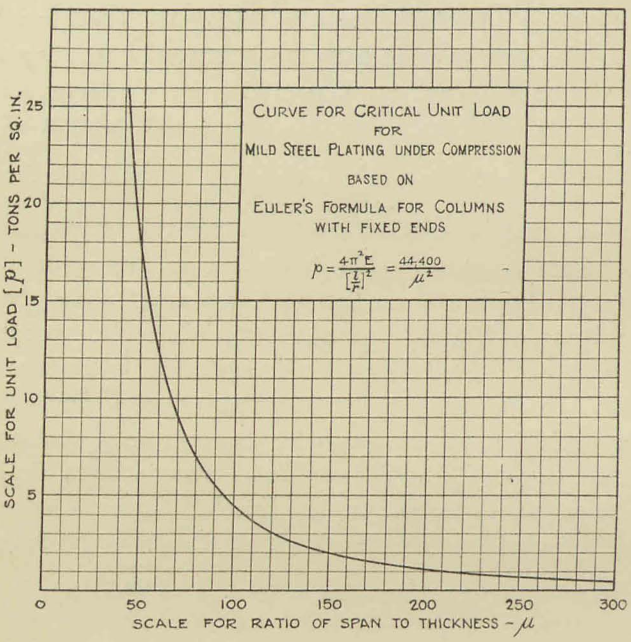

FIG. 49.

When a ship is in dock, the vertical keel and the longitudinals over the side keels are in a state of stress, which, although not free from shearing, will approach the condition of pure compression immediately over the keel-blocks. Where the angles are double, as in the vertical keel, the plating may be regarded as fixed and formula (56) may be used, but where single angles are found the condition will probably be intermediate between the cases of round-ended and fixed-end columns. When μ is smaller than 45, the Column Curves should be used.

2. **Compression Due to Shearing.**—We have seen how shearing in a ship subject to longitudinal bending will cause strong compressive stresses in the outside plating at the neutral axis, and, hence, a tendency to wrinkling. Similar conditions exist in bulkheads subject to shearing, and in longitudinal and transverse continuous frames under the action of concentrated and unevenly distributed loads and reactions. For instance, when a heavily loaded pillar stands on a frame, the web directly below the pillar will be subject to a simple compressive stress and the strength is calculated as explained above, but on both sides of the pillar great shearing stresses will exist in the web, tending to produce wrinkles. As explained in SECTION 7, we here consider an elemental diagonal strip

of plating of a length $l = \frac{h}{\sqrt{2}}$ where h is the depth of the web between flange angles. Regarding this strip as a column under compression, we find the critical load from Euler's formula, and if this does not leave a proper margin of safety as compared with the calculated virtual compressive stress ($1{\cdot}3q_0$) web stiffeners must be fitted. The distance between the stiffeners s must be smaller than the clear depth of the web. We shall illustrate this by an example.

Example.—Let the depth of a frame web between flange angles be 32 in., and the thickness of the web ½ in. Suppose that the maximum shearing stress q_0 is found to be 10 ts. per sq. in., and that the required factor of safety is 3.

Examine first whether there is any necessity for fitting web stiffeners.

Virtual compressive stress on column formed by elemental strip $1{\cdot}3q_0 = 13$ ts. per sq. in.

Length of column $l = \frac{32}{\sqrt{2}} = 22{\cdot}6$ in., which gives $\mu = 2 \times 22{\cdot}6 = 45{\cdot}2$, and the corresponding critical stress according to Euler's formula (56)

$$p = \frac{44400}{(45{\cdot}2)^2} = 21{\cdot}7 \text{ ts. per sq. in.}$$

Hence the factor of safety is only $\frac{21{\cdot}7}{13} = 1{\cdot}67$. It is, therefore, necessary to fit web stiffeners. For a factor of safety of 3, the critical load on the column is $3 \times 13 = 39$ ts. per sq. in., whence

$$\mu = \sqrt{\frac{44400}{39}} = 33{\cdot}7$$

and $$l = \tfrac{1}{2} \times 33{\cdot}7 = 16{\cdot}9 \text{ in.}$$

which gives the required spacing of the web stiffeners $s = 16{\cdot}9\sqrt{2}$ or about 24 in.

In practice and under ordinary conditions of loading stresses will never reach such high figures as here assumed in a girder of this kind, and web stiffeners are ordinarily not required in ship frames, but, as will be shown in the Section on elastic bulkheads, cases may arise in warships where it is necessary to design the structure to stresses approaching the elastic limit.

CHAPTER VII.

RIVETS AND RIVETED CONNECTIONS.

THE strength and watertightness of the hull depend ultimately on the nature and quality of the connection between the individual pieces of which the structure is built up. The connection can be effected in three different ways, by welding, soldering, and riveting.

In *welding*, the union of the metals is effected by hammering or otherwise compressing them while they are in a plastic state, before the melting point is reached. In this way the parts to be united are brought in intimate contact with each other and come to form virtually one piece. In ship construction this process is employed in heavy pieces of wrought iron such as stems and sternposts.

Soldering is employed in the processes known as "autogenous welding." * The union is here effected by fusion either of the metals to be joined, or of a separate piece of metal of similar character used as a solder. In the oxy-acetylene process the heat is produced by a blowpipe, in

* C. Campion and Wm. C. Gray, "On the Strength of Welds produced by the Oxy-Acetylene Process," *Eng. and Shipb. of Scotl.*, July 1914; T. G. John, "Shipbuilding Practice of the Present and Future," and Discussion by Mr P. A. Mudd, *Inst. Nav. Arch.*, July 1914.

"electric welding" by an electric arc, in the "Thermit" process by a chemical reaction in the solder. These processes have been used quite frequently in ship construction in recent years, especially for repair work; for instance, for welding cracks in stems, stern and rudder-posts, for welding landing edges of leaky seams such as box-keel plates to stern-posts, etc. They have been used also in new work to some extent. In certain structural parts of submarine boats, such as coamings, conning-towers, and in the union of complicated castings, the oxy-acetylene process has been successfully applied.

The connections thus effected are not, however, always fully reliable. The zone of metal adjacent to the weld is apt to be injured, and the weakness may appear when the structure is exposed to shock or fatigue. At the present standpoint of the art these processes, therefore, should not be used where stresses are very high or in parts exposed to strong and often-repeated dynamic action; in fact, their application, like that of ordinary welding, is very restricted.

Riveting is, as yet, with the few exceptions referred to above, the only practicable means of uniting the different parts of a ship's structure, and many difficulties will have to be overcome before any process of soldering can take its place.

23. THE RIVETS.

1. **Materials Used for Rivets.**—As a general rule rivets should be of essentially the same material as the parts they connect, but the ductility should be somewhat greater. This rule is, on the whole, followed in warships and rivets are, therefore, generally of steel, mild or high-tensile. At first, when high-tensile steel was introduced, difficulties were experienced. Rivets of this material tended to harden in cooling, they were apt to crack, and the heads were liable to fly off. Mild-steel rivets were reliable, but lost in shearing strength when used in connection with the harder material, and it was necessary to increase the number or size of rivets disproportionately in order to attain a certain strength. The difficulties in the use of rivets of high-tensile steel are now largely overcome, and rivets of a special steel (nickel or crucible), generally driven by hydraulic power, are used for this purpose. Some builders, however, maintain that mild steel is the best material for all rivets that are to be heated, *i.e.* rivets of more than $\frac{3}{8}$-in. or $\frac{7}{16}$-in. diameter.

Rivets of less than $\frac{3}{8}$-in. diameter, whether of iron or steel, are, generally, riveted cold, because, if put in hot, they are liable to waste too much by scaling. Steel rivets of $\frac{3}{8}$-inch diameter or more should always be put in hot whenever watertightness, oiltightness, or strength is required. According to Mr S. W. Barnaby,* steel is not so suitable for small rivets as iron, because, if put in hot, steel rivets are cooled so rapidly on being put into the holes, that they are liable to suffer by the hammering. If riveted cold they are apt to become brittle and treacherous.

In merchant vessels iron rivets are still extensively used in connection with steel plating, the chief reason being that steel rivets require a higher grade of workmanship, which cannot always be secured with certainty. The shearing strength of iron rivets of good quality is more uniform and reliable than that of steel rivets, especially those of the harder grades, but, when used in steel plating, the strength of iron rivets falls off as stated in SECTION I.

2. **General Features.**—Rivets consist in general of a smooth cylindrical "shank," provided with a projection at one end, called the "head," while the other end, after being hammered up by the riveter, forms the "point." Generally the shank in pan and button head rivets is given a slight cone under the head so as to fill the countersink of the plate formed by punching. No cone is used where the hole is drilled or where it is punched small and reamed to size, nor is any cone used in small rivets. In "tap-rivets" and "screw-rivets" the shank is screw-cut.

The length of the rivets should be sufficient to ensure a proper point. A rivet should rather be too long than too short. The allowances for length, over and above the sum of the thicknesses connected, are determined by experience and are usually given in a table.

Pl. V shows the standard forms of rivets used in the United States Navy.

3. **Rivet Heads.**—Ordinarily, the "pan head" rivet is used; it possesses great strength and clamping power, it is well adapted for holding up, and is easily tested. The "countersunk head" is used where necessary to obtain flush work and in staples and bounding bars where watertightness is of great importance or difficult to obtain. Countersunk heads are often used in oiltight work, generally in connection with countersunk points; they are also used in three-ply riveting, and where both sides of a watertight partition are calked. Countersunk heads are easier to calk than other rivet heads; in fact, in many places it may not be possible to calk the whole of a pan head. Countersinking should be carefully done. If the angle of countersink in the plate is smaller than that of the rivet head, watertightness or oiltightness may not be secured.

The "button" or "snap head" is often used where the rivets are closed by power.

* *Inst. Nav. Arch.*, 1902, p. 4.

To face p 118.

Types of Rivets for U. S. Naval Vessels.

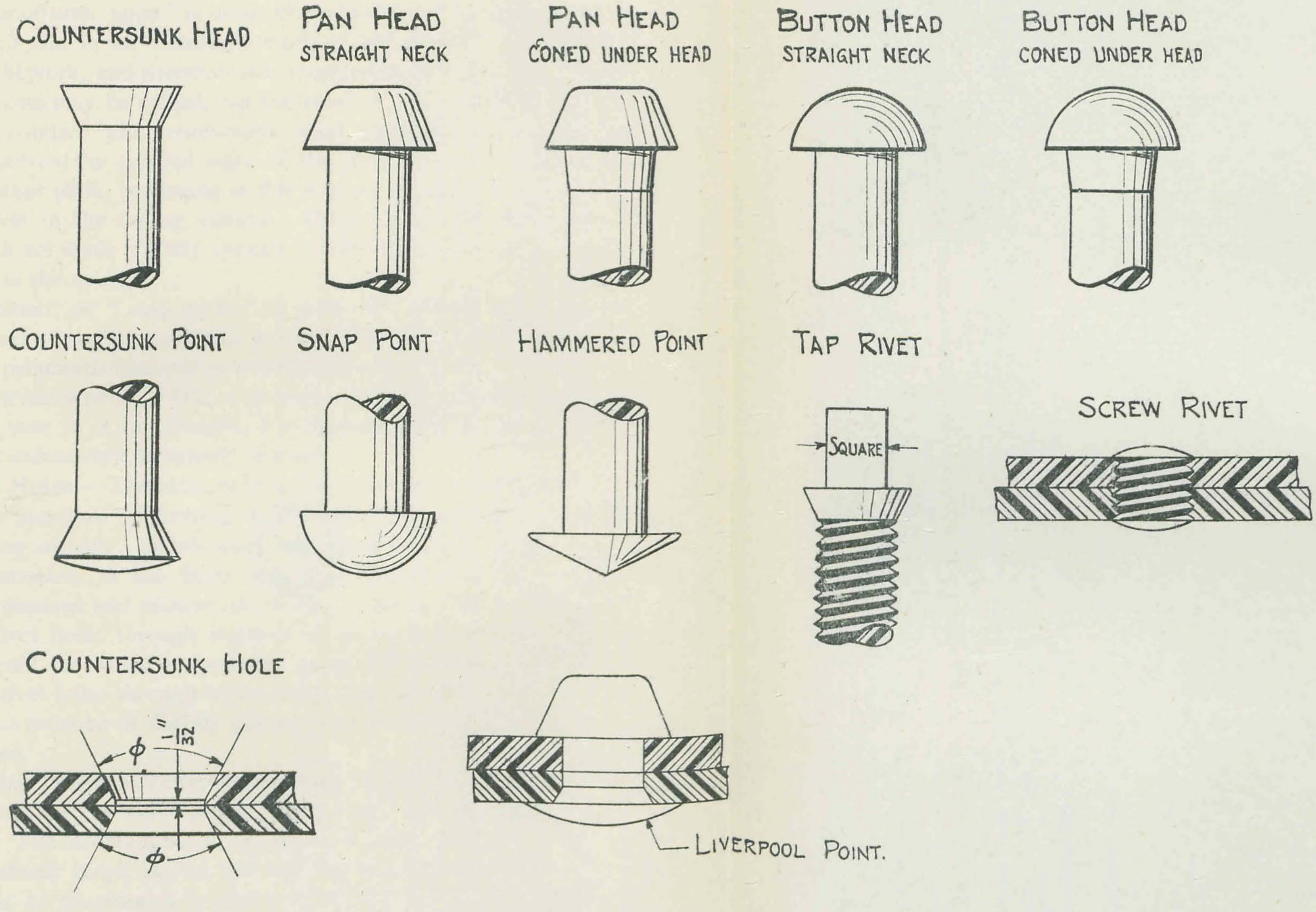

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PLATE V

4. **Rivet Points.**—The "hammered point" is the one most generally used for internal work. It is strong and easy to make, and it requires no chipping.

The "Liverpool point" is a little more full than the hammered point and is countersunk to a depth equal to one-half the thickness of the plate. It has a somewhat greater clamping power than the hammered point. It is well adapted to light work.

The "countersunk point" is used where flush work is required, as in the outer shell and in all watertight work of importance. It is always used in oiltight work, and wherever it is required to calk the rivets. Also hammered points may be calked, but the result is not so reliable as with countersunk points. The countersunk point possesses great clamping power, and presses the tapered edge of the plate round the hole tightly against the other plate, producing in this way an annular watertight area round the rivet in the faying surface. Often the countersunk points in the outer shell are made slightly convex, "full," to provide for corrosion and to increase the strength.

The "button" or "snap point" is used for finished appearance or where rivets are closed by hydraulic power. Sometimes snap points are made with a pneumatic hammer provided with a special die.

The countersunk point, if full, is probably the strongest, the hammered point comes near to it in strength, but the snap point is not so reliable or so efficient unless very accurately centred.

5. **Rivet Holes.**—The holes in plates of less than 1 inch in thickness are ordinarily punched. Punching should, wherever possible, take place from the faying surface. In all work of importance the punched plates should be annealed, or the holes should be punched $\frac{1}{16}$ in. to $\frac{1}{8}$ in. smaller than required and reamed out to size, so as to remove the injured material. Rivet holes through material of more than about 1 inch in thickness should be drilled or punched small and reamed to size afterwards. All rivet holes through high-tensile steel should be drilled.

Rivet holes must be of slightly greater diameter than the rivets before they are closed.

The holding power of countersunk rivets is greater the deeper the countersink is, and, within certain limits, the greater the angle of countersink. The countersink is, therefore, in light plating, where the rivets are relatively large, carried through the full thickness of the plate and the angle of countersink is great, up to 60°. In heavy plating the countersink usually stops a little above the bottom of the hole, leaving a small cylindrical portion, and the angle of countersink is smaller, down to 32° in very heavy plates. The diameter of rivet holes and the angle of countersink, as used for the various thicknesses of plating in the United States Navy, are given in Table X.

6. **Screws Used instead of Rivets.**—"Tap rivets" are screws, generally with a countersunk head provided with a stud which is chipped off after screwing up. The term rivet applied to tap rivets, as also to screw rivets, is really a misnomer, since they are not riveted. Tap rivets are used where ordinary rivets are inapplicable, as, for instance, where the shell plating lands on heavy parts of the stem or sternpost, or where structural parts are connected to armor. Ordinary rivets would there, on account of their great length, be exposed to fracture by contraction on cooling. Generally, rivets should not be longer than about 6 inches, although rivets of much greater length are actually used in many cases, as, for instance, in sternposts of merchant vessels. Where strength is of importance, tap rivets should penetrate to a depth not less than 1 diameter, and should penetrate $1\frac{1}{2}$ diameters where the thickness of the metal will allow.

"Screw rivets" have the advantage that they have no head, which is liable to fly off when the plating is exposed to vibrations by the impact of projectiles. Hence they are used in ceiling or mantlet plates, fitted inside the framing of the armored parts of a ship where men are to be stationed in action, such as casemates or gun-turrets. Screw rivets have no power of drawing the plates together and are, therefore, ill adapted for structural work of any importance.

7. **Screw Bolts with Nuts** are used in wood decks, sheathing, and backing behind armor. They are also employed in the steel structure where the work is required to be removable.

24. GENERAL FEATURES AND CONDITIONS INFLUENCING THE STRENGTH OF RIVETED JOINTS.

1. **General Features.**—Plates may be connected by "lapped" or "butted" joints. Since, in the latter case, straps must be used, which overlap both plates, each butted joint consists really of two lapped joints. Hence the overlap is the fundamental form of joint, which should be first considered in a general discussion of the problem.

Angle bars and shapes are connected by "bosom pieces" or by straps, single or double. The joints so formed are quite similar to those formed by butted plates, and their design is governed by the same principles.

Let us, then, consider the fundamental case of two plates overlapping each other. The connection is effected by rivets, generally arranged in straight lines, "rows," parallel with the edge of the joint. If the rivets in the different rows are placed abreast of each other, *i.e.* in straight lines normal to the rows, the arrangement is called "chain-riveting." If the rivets are displaced relative to each other in the different rows, the arrangement is referred to as "zigzag," "reeled," or "staggered" riveting, the two latter terms being used where the spacing is open and the rows are close together. In fact, reeled or staggered riveting is obtained by slightly displacing the rivets in an ordinary single row alternately to one side and the other, so as to form an extremely flat zigzag line.

The strength of all riveted joints, whatever their nature, is influenced by certain conditions, the effects of which are imperfectly known and difficult to estimate. These effects will be here briefly discussed.

2. **The Grooving-Effect of Rivet Holes.**—Due to the peculiar arrangement of the lines of stress round the rivets, an increase in tensile strength of the metal between the holes is produced, often referred to as "the excess strength due to perforation." According to Professor A. B. W. Kennedy's experiments,* this excess strength, for plates of $\frac{3}{8}$ in. to $\frac{3}{4}$ in. in thickness, appears to be roughly 10 per cent. when the spacing of the rivets is 3 diameters and about 5 per cent. when the spacing is 4 diameters.

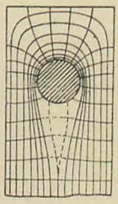

FIG. 50.—Lines of Principal Stress around a Rivet.

Fig. 50 shows the stress lines round a rivet, as determined experimentally by Professor E. G. Coker and Mr W. A. Scoble.†

3. **Weakening of the Material by Punching.**—When holes are punched in a plate, a state of strain is created around the holes, which is equivalent to a local weakening of the material. This weakening depends in some measure on the thickness of the plate, being greater the heavier the plate. The strain may be relieved by annealing, or the injured material may be removed by reaming out the hole. In drilled holes the effect does not occur. The gain in strength by the grooving-effect and the loss by punching counterbalance each other to some extent. Neither of them is accurately known and they will, therefore, be disregarded in the following.

4. **Frictional Resistance.**—The strength of a riveted joint is also influenced by the friction between the plates at the faying surface. The friction will depend on the roughness of the surfaces in contact and on the force with which the rivets press the plates together. It increases somewhat during the first years after the ship is built by the formation of rust. We are unable, at present, to predict what the force required to overcome

* *Br. Inst. Mech. Eng.*, 1885, p. 249. † *Inst. Nav. Arch.*, 1913, i. p. 214 and pl. xxvi.

the friction will be in any given case. Experiments by Wildish,* Bach,† Kennedy,‡ and by the German Navy Department § have given varying results, and further investigation is needed to clear up this difficult question. With power-riveting the resistance is greater than with hand-riveting. As an average of these experiments we may reckon that the resistance offered by a riveted joint at the point where frictional slip occurs is from about 7 to 8 ts. per sq. in., referred to the shearing area of the rivets. Before this point is reached, a small displacement of the plates relative to each other is indeed observed when refined methods of measurements are used, but this slip is largely elastic and practically insignificant. The joint will, therefore, behave essentially as a solid plate as long as the friction is not overcome.

The elastic limit in shearing of the material of the rivets will not be reached until the pull is considerably greater than that at which frictional slip begins; in case of mild-steel rivets this limit occurs probably at a stress of about from 13 to 14 ts. per sq. in. After that the frictional resistance appears to have little influence on the strength of the joint, and it seems certain that it has no influence on the ultimate strength.

Summing up, the frictional resistance, at low and moderate stresses, gives to riveted joints an almost perfect solidity, preventing all straining and looseness of the rivets within the point where frictional slip occurs. This is of particular importance in structures exposed to alternating and dynamic forces and where absolute tightness is required. It follows that in the important strength members of a ship, as also in machinery and gun supports, and in oiltight work, the working stress ought not to reach the point at which frictional slip takes place. In structures where such severe conditions and claims do not obtain, as, for instance, in ordinary watertight bulkheads, the working stress in the rivets may be allowed to approach the yield point of the metal in shearing.

With our present limited knowledge of the subject we cannot make use of the point at which frictional slip occurs in calculating the strength of riveted joints. We are forced, here as elsewhere, to base the calculation on the ultimate strength of the material with due regard to the relative position of the yield point. We should, however, provide an extra great shearing area of the rivets in all joints where frictional slip cannot be tolerated.

5. **Bending of the Joint.**—When an overlapped or single strapped joint is subject to tension or compression, a bending moment will come to act on the joint, because the forces acting on the plates do not lie in the same plane. Let us first consider a joint in tension.

* *Inst. Nav. Arch.*, 1885.
† *Maschinenelemente*, 1891–92, p. 117.
‡ *Br. Inst. Mech. Eng.*, 1885.
§ Pietzker, *Festigkeit der Schiffe*, p. 52.

The plates will bend as indicated in an exaggerated manner in fig. 51. Great tensile stresses will be created in the plates at the faying surface and an uneven distribution of the bearing pressures between the rivet and the plates will take place. The rivets will come under a tensile stress, they will tilt and, in extreme cases, they may pull out or they may shear at a smaller stress than usual. The neutral axis which has a curved jog at the joint will tend to straighten out, whereby the joint will elongate; at the same time the leverage of the bending couple will be reduced. The bending effect is greatest in heavy plates and in single-riveted joints, but since single riveting is never used in ship construction in heavy plates or where great strength is required, and since the butts are always stiffened by adjacent material, this source of weakness may, generally, be disregarded when joints are in tension.

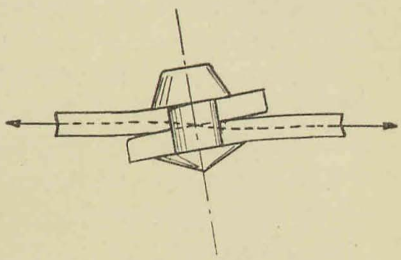

FIG. 51.—Rivet in Lapped Plates under Tension. Dotted line indicates neutral axis.

When a joint is subject to compression the consequences may in certain cases be serious. Consider a butt midway between the frames or the beams. Whether overlapped or single strapped, the bending will in compression increase the jog in the neutral axis and will tend to start buckling. In torpedo-vessels, where the working stresses are high, this action may seriously impair the strength of the plating, whence double butt straps should be used in the important strength members of such vessels, notably in the deck and in the sheer strakes. For the same reason joggling of the plates at the butts should be avoided wherever strength is of importance, and should never be used in strength members of high-speed vessels.

25. MODES OF FRACTURE.

Notation.

t = Thickness of plating.

t' = Thickness of strap.

d = Diameter of rivets.

p = Pitch or spacing of rivets, *i.e.* distance from center to center of rivets in the same row.

$m = \frac{p}{d}$ = Pitch-ratio or spacing-ratio referred to the diameter of the rivets.

n = Number of rows of rivets in a joint.

p_T = Tensile stress.
p_C = Compressive stress.
p_S = Shearing stress.
f_T = Ultimate tensile stress.
f_C = Ultimate compressive stress.
f_S = Ultimate shearing stress.
δd = Diameter of rivet hole at the faying surface of the plates.
adt = Sectional area of rivet hole in a plate, a being a factor that takes account of the increase in area due to countersink and other causes.
e = Efficiency of a riveted joint, *i.e.* ratio between the actual minimum strength of the joint and the strength of the intact plate under a simple tensile stress.
$k = \frac{\pi}{4}\frac{f_S}{f_T}\delta^2$ = A factor often occurring in the strength calculations of riveted joints.

t', δ', and a' have the same meaning as t, δ, and a, but apply to straps.

1. **Elements of a Joint and Modes of Fracture.**—Apart from the general features discussed in last Section, the strength of a joint depends essentially on the following elements :—

The diameter of the rivets relative to the thickness of the plates.
The distance of the rivets in the outer rows from the nearest edge of the plate and the distance between the rows.
The number of rows.
The spacing of the rivets in the rows.

We shall show how these elements may be determined and regulated through a consideration of the different modes of fracture.

Imagine a simple overlapped joint to be exposed to a uniform pull, acting in the plane of the plates and at right angle to the rows. If this pull is sufficiently increased the joint will ultimately be torn asunder, either by the rivets pulling through the plates or by actual fracture of the plates or the rivets. In the majority of cases joints give way by actual fracture, which may occur in the following different ways, here considered analytically, independent of each other :—

a. Breaking of the edge of a plate in front of the rivets in an outer row.
b. Lengthwise shearing of a plate in front of the rivets.
c. Shearing of all the rivets.
d. Crosswise tearing of one of the plates through an outer row of rivets. The term "outer" refers to the joint, not to any of the plates.

These fundamental modes of fracture may be combined. The most important combinations are:

e. Tearing across of one of the plates along an intermediate row and shearing of all the rivets outside this row.

f. Tearing across of both plates along the same or different rows, in the latter case eventually combined with shearing of the intervening rivets.

2. **Minimum or Standard Strength of a Joint.**—The ideal joint is one that offers the same resistance to each of the different modes of fracture. In ship construction it is, however, for reasons that will be explained later, generally impossible to attain such complete uniformity of strength. A certain mode of fracture is selected to which the joint is made to offer a certain minimum or standard resistance, and the strength by all other modes of fracture is made the same or greater. It will be shown in the following that rupture by modes *a* and *b* can be and is, generally, precluded by following certain rules as to diameter and disposition of rivets. In selecting the mode of rupture which is to possess the standard strength we need, therefore, only consider fracture by shearing of all the rivets and crosswise tearing of the plate.

In some navies, as, for instance, the British, the end joints are ordinarily double riveted and the strength of the plates is considerably greater than the strength of the rivets. In merchant vessels, on the other hand, the spacing of the rivets is closer and there are more rows, so that the strength of the rivets is considerably greater than that of the plates, especially where steel rivets are used.

Now, tearing of the plate is more dangerous than shearing of the rivets, because tearing of the plate occurs without warning, and a rent, when once started, is liable to extend further and further. Moreover, the strength of the plate is apt to suffer by corrosion and wear. Hence it would seem rational to make the plate stronger than the rivets. On the other hand, straining of the rivets or, at any rate, visible slip of the joint, is likely to occur very early, probably long before the plate reaches the elastic limit, even although the rivets may ultimately be stronger than the plate. Such slip is highly undesirable and may be dangerous. Owing to the alternating stresses to which a ship is subject, the rivets are liable to work more and more adrift, great strains will be thrown on the adjacent structural members and rupture of these may occur. From the days of iron shipbuilding, when the rivet area in the butt fastenings of the shell in merchant vessels was relatively smaller than it is now, many cases are on record where the butts, notably of the sheer strake, worked adrift. In general, neither the plates of the sheer strake nor the butt-straps were fractured, but often the strake below the sheer strake was torn down through the solid plate immediately under the sheer-strake butt.*

On the whole, it seems best in warships to give a joint practically the same resistance to tearing and shearing, in which case it is immaterial which we call the standard. It is here recommended to use the strength of the rivets as the theoretical standard of reference in the calculations for riveted joints, simply because it is more convenient for this purpose than the strength of the plate. The ultimate resistance offered by the plate is less definite, and therefore less suited as a standard, for while the plate ordinarily tears along the outer row it will in some cases tear along the second row.

Hence, in designing a joint we begin by adopting a rivet area which will give the desired minimum strength, generally about four-fifths of the strength of the solid plate, and after that we give to the plate the same or a slightly greater strength. With the rivet area so determined we shall in general preclude frictional slip, since the ordinary working stresses in the rivets will then rarely exceed the stress at which slip occurs, probably from 7 to 8 ts. per sq. in. In the most strained parts of the structure, however, such as the sheer strakes and the stringer plates, especially in lightly built, fast vessels, stresses may under exceptional circumstances exceed this limit. In such strakes, therefore, it is desirable to augment the ultimate shearing strength of the rivets beyond the standard requirement, by adding one or more rows, while there will be no object in correspondingly increasing the strength of the plate. This point will be again referred to in later chapters. (See Example, SECTION 28, *5*.)

It is of interest to note that structural engineers in the design of girders aim at a similar equality of strength in shearing of the rivets and tearing of the plates as here recommended for warships.

Merchant vessels are not so well stiffened by internal diaphragms as warships, and are subject to more long-continued, hard service, whence the riveted joints will be more severely strained and will require greater solidity. This may explain why the rivet area in the joints of merchant vessels is made so much greater relative to the plate area than in warships. Another reason may be that weakness in the rivets is apt to lead to frequent and expensive repairs of the joints.

3. **Flow of the Material.**—With such ductile materials as are ordinarily used in shipbuilding, rupture will always be preceded by and

* H. H. West, *Inst. Nav. Arch.*, 1884, p. 274.

induced by a certain flow of the metal, whether in the rivets or in the plates or both.

On the sides of the rivet holes the tensile stress in a plate may be perhaps five times as great as the mean stress in the intact plate and will be greater the closer the spacing of the rivets * (fig. 50). When the stress becomes excessive the material will begin to flow in the same way as in the section of greatest contraction in a test piece. The plate between the rivet holes will elongate and decrease in thickness and the holes will stretch oval. Ultimately the plate may tear crosswise between the rivet holes.

At the same time the metal of the plate in front of the rivet, and of the rivet itself, will be subject to compression due to the bearing pressure between the rivet and the plate. The bearing pressure is not uniformly distributed, but has a maximum on the foremost part of the front surface of the rivets, where the compressive stress is probably even greater than the tensile stress on the sides of the hole. If the joint is bent, there will be a further inequality in the distribution of the pressures. Hence the intensity of stress at certain points will be much greater than the calculated mean value of the bearing pressures, and at such points the metal will begin to "flow" or "crush" when the bearing pressure reaches a certain limit. The crushing of the material does not, however, ordinarily exhibit itself so visibly as the stretching just referred to, although it may contribute to the holes stretching oval. Before the crushing takes visible shape it will, generally, due to a local flow of the metal, cause premature shearing of the rivets or tearing and shearing of the plate, whichever is of the softer material. Crushing must, therefore, be considered as the primary or initial cause of several modes of fracture, but not as a mode of fracture in itself.

4. **Effects of Excessive Bearing Pressure on the Plates.**—We shall here, in particular, discuss the effects of the bearing pressures on the plates in cases where the rivets are so hard and strong as not to suffer any great deformation. If one plate is much thicker or of much stronger material than the other, only the thinner or softer plate will suffer. The rivets in such a case keep their position in the stronger plate, while the metal of the weaker plate will flow at the rivets.

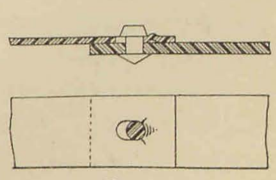

FIG. 52

Usually this action begins by the holes stretching oval. If the stress is increased, the weaker plate will begin to tear or shear, inducing fracture

* See papers by Messrs Coker and Scoble, *Inst. Nav. Arch.*, 1913, i. pp. 214–216; and by Dr. K. Suyehiro, *Jap. Soc. Mech. Eng.*, Spring Meeting, 1914.

by modes a, b, or d: breaking or shearing of the plate in front of the rivets or crosswise tearing between the rivets. In case of very light plating there may be a bulging or curling of the plate in front of the rivets combined with shearing by mode b. Often the rivets will pull through before the joint is completely fractured.

In order to prevent fracture of this particular kind it is necessary and sufficient to avoid sudden changes in strength of the important structural members of a ship. The variation in thickness of the plating should be gradual and proportioned to the variation in the straining forces. Since this rule is observed in all well-designed ships, this kind of fracture will not ordinarily occur in practice.

If the two plates are of the same or nearly the same thickness and of the same material, as is generally the case, they are both liable to suffer by an excessive bearing pressure. The holes will stretch oval and the rivets will tilt. As the pull is increased, shearing and tearing of the plates will begin to take place, and the rivets will take up a more and more inclined position. Finally, if the plates do not fracture beforehand by modes a, b, or d, the rivets will pull out of the plates. This latter form of rupture is most likely to happen if the rivets are countersunk, with rather small heads or points, but sometimes the heads or the points are pulled off.

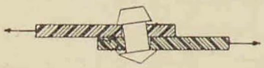

FIG. 53.

5. **Limiting Value of the Diameter Determined by Bearing Pressure.**—By increasing the size of a rivet, its shearing strength is increased as the square of the diameter, while the area of its bearing surface on the plate increases only as the diameter. Hence, the bearing pressure will increase more rapidly than the shearing stress. In order to avoid excessive bearing pressure the diameter must, therefore, not be too large relative to the thickness of the plates, and the material of the rivets, compared with that of the plates, must not be too strong and hard. We may express these conditions mathematically.

Let p_c be the mean bearing pressure and p_s the mean shearing stress on a rivet. Then, for a cylindrical rivet, we have approximately :

$$td\delta p_c = \frac{\pi d^2 \delta^2}{4} p_s$$

or

$$\frac{p_c}{p_s} = \frac{\pi\delta}{4}\frac{d}{t} \quad . \quad . \quad . \quad . \quad . \quad . \quad (57)$$

which shows that the ratio between bearing pressure and shearing stress is approximately proportional to the ratio between the diameter of the rivet and the thickness of the plate. Hence, for a given strength and hardness of the rivet, a point will be reached, when we increase the ratio $\frac{d}{t}$ where the bearing pressure becomes excessive before any shearing of the rivet has taken place. This point may be determined approximately by substituting in (57) for p_c the limiting allowable value of the bearing pressure f_c as found by experiments. For p_s we substitute the ultimate shearing stress f_s of the rivet. Thus we obtain:

$$\frac{d}{t} \leqq \frac{4f_c}{\pi\delta f_s} \quad . \quad . \quad . \quad . \quad . \quad (58)$$

Based on experiments with riveted joints carried out by Professor A. B. W. Kennedy,* we take $f_c = 96{,}000$ lb. per sq. in. as the limiting allowable bearing pressure for mild-steel plating connected by mild-steel rivets of ordinary shearing strength: $f_s = 50{,}000$ lb. per sq. in. For δ we take 1·07 as an ordinary value. Substituting in (58) we find:

$$d \leqq 2\tfrac{1}{4}t$$

which furnishes an upper limit for the diameter in terms of the thickness. If the diameter exceeds this value, the bearing pressure will reach the value f_c before the shearing stress reaches the value f_s and breakdown will take place at a lower figure. If the rivet is countersunk the limiting value will be somewhat higher.

In practice the rule is that *in overlap and single strap joints the diameter of the rivets should not, unless for other reasons necessary or desirable, be more than twice the thickness of the plating.*

This rule leaves a margin of strength, so that even if the rivets are of a material somewhat harder and stronger, relative to the material of the plates, than assumed above, they will shear before the bearing pressure becomes excessive.

With double straps, where the rivets are in double shear, greater bearing pressures may be allowed, and we may, according to Professor Kennedy, reckon $f_c = 110{,}000$ lb. per sq. in. as a permissible limit.† Unless the rivets are of a relatively hard material, the high bearing pressures will, however, cause the shearing strength to be somewhat reduced, and it is, therefore, not safe to reckon the ultimate shearing strength higher than 1·75 times as great as when in single shear. Hence (58) becomes:

$$\frac{d}{t} \leqq \frac{4f_c}{1{\cdot}75\pi\delta f_s} \quad . \quad . \quad . \quad . \quad . \quad (59)$$

Substituting numerical values, we get:

$$d \leqq 1{\cdot}5t$$

* *Br. Inst. Mech. Eng.*, 1885, p. 251.

† *Ibid.*, p. 256.

The practical rule is in this case, that *the diameter shall not exceed one and one-half times the thickness when the rivets are in double shear*, which is again in good accordance with theory.

If, then, the diameter of the rivets is kept within the limits given by these rules, the rivets will ordinarily yield at their normal shearing stress before the material of the plates begins to crush, and the bearing pressure need not be further considered.

6. **The Diameter Used in Practice.**—It is desirable, for reasons to be discussed hereafter, to use as large rivets as possible, and it would, therefore, be of advantage to adhere closely to the upper limit of the diameter determined above. In practice, however, this is not always possible.

In light plating, ¼ in. or less, it is necessary or desirable to use even larger rivets than the rules permit, because there are practical difficulties in using very small rivets. Hence, in very light plating, rupture may take place before the rivets shear, by one of the modes induced by crushing of the plate, as explained above.

In heavy plating, on the other hand, there are difficulties in the way of using rivets as large as the rules permit. In order to obtain at once sound and economical work by hand-riveting, the diameter should not exceed ⅞ in., and 1⅛ in. may be considered an upper limit. With power-riveting larger diameters may be used without difficulty, but 1⅜ in. may be taken as an extreme limit. Hence we find in practice that for plating of 1 in. thickness or more the ratio $\frac{d}{t}$ is about equal to one.

A lower limit to the diameter of rivets, when the holes are punched, is determined by the strength of the punching tool. When the size of the punch is reduced, its sectional area will decrease as the square of the diameter, while the pressure, which the punch must exert in the act of punching a given plate, will decrease only as the circumference of the hole, *i.e.* as the diameter of the punch to the first power. Hence, the compressive stress in the tool will be inversely proportional to the diameter. In practice the lower limit to the diameter of the punch is equal to the thickness of the plate.

It will now be understood why we do not in practice find a constant ratio or any other simple relation between the diameter of the rivets and the thickness of the plating. Table X. gives the diameter of rivets used in the United States Navy. It will be seen that the ratio $\frac{d}{t}$ varies from about 3 in very light plating to 1 in heavy plating, whence, by following this Table, we preclude excessive bearing pressures in all plates of medium and heavy thicknesses, but not in lighter thicknesses.

TABLE X.—RIVETING TABLE FOR U.S. NAVAL VESSELS (1912).

Weight of plating . . . lbs.		-3*	3-7†	7 1/2-8 1/2†	9-12 1/2†	13-19†	20-29†	30-39†	40-50†	51-
Diameter of rivets . . . ins.		1/4	3/8	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4
Diameter of rivet holes . . . ins.		9/32	7/16	9/16	11/16	13/16	15/16	1 1/16	1 7/32	1 11/32
Widths of laps, strips, and straps.	Diams.									
Seam laps. Single riveted	3 1/4	13/16	1 1/4	1 5/8	2	2 7/16	...	...	...	...
Seam laps. Double chain riveted	5 3/4	1 7/16	2 1/8	2 7/8	3 5/8	4 5/16	5	5 3/4	...	...
Seam laps. Double zigzag riveted	5	1 1/4	1 7/8	2 1/2	3 1/8	3 3/4	4 3/8	5	...	...
Seam laps. Treble chain riveted	8 1/4	2 1/16	3 1/8	4 1/8	5 3/16	6 3/16	7 1/4	8 1/4	9 5/16	10 5/16
Butt laps. Double riveted	6 1/4	1 9/16	2 3/8	3 1/8	3 15/16	4 11/16	5 1/2	6 1/4	7 1/16	7 13/16
Butt laps. Treble riveted	9 1/4	2 5/16	3 1/2	4 5/8	5 13/16	6 15/16	8 1/8	9 1/4	10 7/16	11 9/16
Butt laps. Quadruple riveted	12 1/4	3 1/16	4 3/8	6 1/8	7 11/16	9 13/16	10 3/4	12 1/4	13 13/16	15 5/16
Edge strips and single straps. Single riveted	6 1/2	1 5/8	2 7/16	3 1/4	4 1/16	4 7/8	5 11/16	6 1/2	7 5/16	8 1/8
Edge strips and single straps. Double riveted	11 1/2	2 7/8	4 5/16	5 3/4	7 3/16	8 5/8	10 1/16	11 1/2	12 15/16	14 3/8
Edge strips and single straps. Treble riveted	16 1/2	4 1/8	6 3/16	8 1/4	10 5/16	12 3/8	14 7/16	16 1/2	18 9/16	20 5/8
Edge strips and single straps. Quadruple riveted	21 1/2	5 3/8	8 1/16	10 3/4	13 7/16	16 1/8	18 13/16	21 1/2	24 3/16	26 7/8
Double ‡ butt straps. Double riveted	12 1/2	3 1/8	4 11/16	6 1/4	7 13/16	9 3/8	10 15/16	12 1/2	14 1/16	15 5/8
Double ‡ butt straps. Treble riveted	18 1/2	4 5/8	6 15/16	9 1/4	11 9/16	13 7/8	16 3/16	18 1/2	20 13/16	23 1/8
Double ‡ butt straps. Quadruple riveted	24 1/2	6 1/8	9 3/16	12 1/4	15 5/16	18 3/8	21 7/16	24 1/2	27 9/16	30 5/8
Centre of rivet from edge of plate	1 5/8	3/8	5/8	13/16	1	1 1/4	1 7/16	1 5/8	1 13/16	2 1/16
Distance between rows, seam laps, edge strips, and single straps	2 1/2	5/8	15/16	1 1/4	1 9/16	1 7/8	2 3/16	2 1/2	2 13/16	3 1/8
Distance between rows, butt laps, and double straps	3	3/4	1 1/8	1 1/2	1 7/8	2 1/4	2 5/8	3	3 3/8	3 3/4
Width of flange of angle	4	1	1 1/2	2	2 1/2	3	3 1/2	4	4 1/2	5
Centre of rivet from heel of bar	2 1/4	9/16	7/8	1 1/8	1 3/8	1 3/4	2	2 1/4	2 1/2	2 3/4
Area of rivet hole, sq. in.	...	·062	·150	·249	·371	·519	·690	·887	1·167	1·418
Countersink angle, ϕ	...	60°	60°	60°	60°	45°	45°	37°	37°	37°

* Exclusive. † Inclusive. ‡ For plating 25 lbs. and under, use rivets one size smaller.
For countersunk rivets of diameter greater than 1¼ in. : $\phi = 32°$.

The leading Classification Societies specify rivets of a somewhat greater diameter than given by Table X. for some of the thicknesses of plating above 25 lb. Similar differences are found in the riveting table of certain other navies. With the use of mechanical tools for riveting, an increase in the size of rivets, even beyond the requirements of the Rules of the Classification Societies, would probably not present any practical difficulty, and would appear to be advantageous in point of economy and strength.

7. **Mode (a): Breaking of the Edge of the Plate in Front of the Rivets.**—The plate here tears asunder along a line normal to the edge, the band between the rivet hole and the edge behaving much like a loaded beam. This action is most likely to take place when the rivet is relatively large and is placed too near the edge of the plate, and is, in particular, liable to occur when the plate is of a hard and brittle material.

FIG. 54.

8. **Mode (b): Lengthwise Shearing of the Plate in Front of the Rivets.**—We have seen that when the bearing pressure becomes excessive, shearing is liable to be induced in the plating. Generally the plate shears along two somewhat oblique lines, as AB and CD in fig. 55. In very light plates a curling may take place. Probably the action is in most cases not one of pure shearing, but is combined with tearing. Fracture of this nature is much more common with ductile materials than fracture by mode (*a*). It is most likely to occur when the rivet is in double shear and whenever the diameter is very large, when the plate is of soft material, and when the rivet is placed too near the edge.

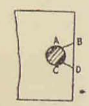

FIG. 55.

In examining the theoretical condition to be fulfilled in order to prevent this mode of rupture, we shall take the length of each of the lines of fracture, as a minimum value, equal to the distance of the center of the rivet from the edge of the plate a minus one-quarter of the diameter. Comparing with shearing of the rivet, we obtain for overlap or single strap :

$$2\left(a-\frac{d}{4}\right)tf_s \geqq \frac{\pi d^2}{4}\delta^2 f_s$$

or

$$a \geqq d\left[\tfrac{1}{4}+\frac{\pi\delta^2}{8}\frac{d}{t}\right] \quad . \quad . \quad . \quad . \quad (60)$$

Taking medium values : $\delta = 1{\cdot}07$ and $\frac{d}{t} = 2$ we get :

$$a \geqq 1{\cdot}2d$$

For very light plating, where $\frac{d}{t} = 3$ and $\delta = 1.17$, we find $a \geqq 1.9d$ while for heavy plating the limiting value of a will be smaller than $1.2d$. For double butt straps, with $\frac{d}{t} = 1\frac{1}{2}$ and multiplying the shearing area by 1.75, we find $a \geqq 1.4d$. When the bearing pressure is excessive, fracture by this mode may take place even if the rivet is placed much farther from the edge than given by these figures, but in such a case the shearing is probably accompanied by tearing and curling of the plate as explained above for the thin plate.

9. **Distance of Rivets from the Edge, and Distance between the Rows.**—Provided the bearing pressure in the rivet holes is not excessive, fracture by the modes of rupture (*a*) and (*b*) can always be prevented by simply making the distance of the rivets in the outer rows from the edges and the distance between adjacent rows sufficiently great. This can be done without in any way affecting the strength against other modes of fracture, the only general restrictions being those imposed by weight and space. Where an edge is to be calked, the distance of the rivets from the edge must not be greater than required for securing sufficient rigidity of the edge.

Through experience and by experiments it has been found that *the center of the rivets should not be nearer to the edge than* $1\frac{1}{2}$ *diameters.* Except for light plating this is well in excess of the theoretical limit determined by (60). In the United States Navy the prescribed distance is $1\frac{5}{8}$ diameters, in the British Navy $1\frac{3}{4}$ diameters. In high-tensile steel plates, as used in large vessels, the distance from the edge should not be less than 2 diameters, because hard steel is more apt than mild steel to give way by breaking of the edge as in mode (*a*). The same margin should be adopted in very light plating of mild steel, but where hard steel in small thicknesses is used, as in torpedo-boats, the rivets should be placed still further from the edge.

The distance between consecutive rows of rivets is somewhat greater than the distance from the edges.

In the United States Navy, in laps, such as the seams, and in single straps, the distance from center to center of the rows must not be less than $2\frac{1}{2}$ diameters; in butt laps and double butt straps it must not be less than 3 diameters; but for zigzag riveting the limiting distance between the rows is reduced to $1\frac{3}{4}$ diameters for rivets spaced 4 diameters apart in the rows. The increased distance between rows specified for butt laps is for the purpose of securing greater resistance to distortion. In double butt straps the increased distance is necessitated by the greater shearing force exerted by the rivets. In zigzag riveting the distance between the rows can be reduced because the rivets are not placed so near each other in successive rows.

In angle bars the rivets should be placed at such a distance from the heel that the head or point will clear the fillet in the bosom of the bar. This is generally attained by making the distance of the center-line of the rivets from the edge of the heel not less than $2\frac{1}{4}$ diameters. The distance of the rivets from the toe of the bar should be at least the ordinary minimum of $1\frac{1}{2}$ diameters plus a small addition for the rounding of the edge. This claim is fulfilled by making the distance of the center of the rivets from the toe not less than $1\frac{3}{4}$ diameters. When a flange is sufficiently wide, the rivets, instead of being placed in a straight line, may with advantage be staggered, forming a flat zigzag line so as to obtain a better distribution of the clamping effect. In such a case the prescribed spacing is measured along the zigzag line.

By a simple summation of the distance of the rivets from the edges and the distance between the rows we are now able to determine the breadth of laps and straps. Using the figures prescribed in the United States Navy we obtain for instance:

Breadth of single riveted overlap: $1\frac{5}{8}d + 1\frac{5}{8}d = 3\frac{1}{4}d$.
Breadth of double riveted seam lap: $1\frac{5}{8}d + 2\frac{1}{2}d + 1\frac{5}{8}d = 5\frac{3}{4}d$.
Breadth of treble riveted butt lap: $1\frac{5}{8}d + 3d + 3d + 1\frac{5}{8}d = 9\frac{1}{4}d$.
Breadth of treble riveted butt strap: $2[1\frac{5}{8}d + 2\frac{1}{2}d + 2\frac{1}{2}d + 1\frac{5}{8}d] = 16\frac{1}{2}d$.

The breadth of overlaps, straps, and edge strips is given in Table X. By adopting the dimensions there prescribed, we preclude fracture by modes (*a*) and (*b*) with the exceptions stated above.

10. **Mode (c): Shearing of all the Rivets.**—In order that shearing of the rivets shall take place at all, we have seen that the diameter must not exceed a certain maximum, beyond which crushing of the plate accompanied by other modes of rupture is liable to occur.

FIG. 56.

The resistance to shearing of individual rivets being known, the resistance of all the rivets may be regulated by simply increasing or reducing the number of rivets and, if necessary, the number of rows. This can be done without affecting the strength of the joint in any other way, provided the spacing of the rivets in the rows is properly adjusted.

Certain theoretical considerations * lead to the conclusion, that on account of the elastic elongation of the material of the plates within a riveted joint, the strength of the joint will not increase in the same propor-

* Milton, *Inst. Nav. Arch.*, 1885, p. 204.

tion as the number of rows when there are more than two rows. This conclusion has been in some measure experimentally verified, although not all tests agree with it.* In the cases where a reduction in strength did exist, it was found much smaller than the theory would lead us to believe.

If desired, this reduction may be taken into account by substituting in the mathematical formulas, given below, a somewhat reduced figure in place of n the number of rows. According to experiments undertaken by Bach † and by the German Navy Department,‡ we should use $n = 2{\cdot}7$ for treble chain riveted joints and $n = 3{\cdot}4$ for quadruple chain riveted joints, instead of $n = 3$ and $n = 4$ respectively. These figures apply to joints exposed to a pull, supposing the plates are of about the same thickness, and provided no rivets are omitted in the rows. The experiments, on which these figures are based, are, however, far from being exhaustive, and the figures can, therefore, only be considered provisional.

11. **Mode (d): Crosswise Tearing of one of the Plates through an Outer Row of Rivets. Spacing of the Rivets.** —Fracture is here probably caused or started by an excessive tensile stress in the plating on the sides of the rivet holes. In soft materials rupture is preceded by a flow at these points, whereby the rivet holes will stretch oval.

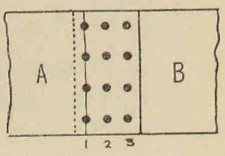

FIG. 57.

The "spacing" p often referred to as the "pitch," is the distance from center to center of the rivets along a row; generally, it is determined by considerations of strength, but it is often modified by the claims of watertightness or oiltightness. The spacing prescribed in practice, in cases where strength is the primary consideration, is based essentially on the condition that the plate shall be as strong as the rivets. This condition determines a lower limit to the spacing, which for single riveting may be as low as 2·9 diameters ($\frac{5}{16}$-in. plating). The practice of the United States Navy is given in Table XI., from which it is seen that the specified spacing for single riveted joints is $3\frac{1}{2}$ diameters and that 3 diameters is to be regarded as an extreme minimum. The spacing is in the practical tables given only to the nearest half diameter and is frequently referred to as the "standard spacing." The determination of the exact spacing for equal strength of plate and rivets will be discussed in the next chapter. We shall now examine how far the standard spacing precludes other modes of fracture.

* *Report: Watertown Arsenal*, 1891, pp. 551, 555.
† Bach, *Ver. Deutsch. Ing.*. 1894, p. 1231. ‡ Pietzker, *Festigkeit der Schiffe*, p. 58.

TABLE XI.—RIVET SPACING.

	Diams
Seams of Plating (Laps and Straps).	
Watertight, single riveted	3½
Watertight, double and treble riveted	4½
Oiltight	3½
Non-watertight	4½
Butts of Plating (Laps and Straps).	
Watertight, single riveted	3½
Watertight, double riveted	4
Watertight, treble riveted	4½
Watertight, treble riveted, alternate rivets in third row omitted	4
Watertight, quadruple riveted	4½
Oiltight	3½
Non-watertight, single riveted	4½
Non-watertight, more than single riveted, same as watertight.	
Staple Bars, and oiltight or watertight bars generally.	
Oiltight	4
Watertight, plating over 15 pounds	5
Watertight, plating 15 pounds and under	4½
Watertight, floor staples, connection to the floor plates	4
Deck and Platform Plating.	
To beams, below but not including the lowest complete watertight deck	5
To beams, above and including the lowest complete watertight deck	8
Bulkheads.	
Bounding bars to watertight and oiltight bulkheads below lowest complete watertight deck; see Staple Bars.	
Bounding bars to non-watertight bulkheads and to bulkheads above lowest complete watertight deck, except where subject to water pressure tests	8
Stiffeners, to plating, watertight bulkheads below lowest complete watertight deck	6
Stiffeners, to plating, where stiffeners are double, that is, on both sides of bulkhead	4
Stiffeners, to plating, turret-support bulkheads	5
Stiffeners, to plating, non-watertight bulkheads and bulkheads above the lowest complete watertight deck	8
Face plates and reinforcing bars to stiffeners	5
Transverse Framing.	
Non-watertight, to outside plating	8
Non-watertight, to inner bottom	5
Watertight and oiltight; see Staple Bars.	
Floor plates, lightened and solid, to clips, frame bars, reversed frame bars, etc., except where watertight	7
Floor plates, flanged bracket type, to frame bars and reversed frame bars, when single riveted	5
Floor plates to frame bars and reversed frame bars when double riveted	7
Floor plates to clips connecting to longitudinals, vertical keel, etc.	5

TABLE XI.—RIVET SPACING—*continued.*

	Diams.
Longitudinals and Keels.	
Watertight and oiltight angles; see Staple Bars.	
Non-watertight longitudinals to outside plating	8
Non-watertight longitudinals to inner bottom	5
Non-watertight longitudinals, plates to inner and outer longitudinal angles	7
Vertical keel angles, where non-watertight, to inner bottom	5
Vertical keel angles, where non-watertight, to flat and vertical keels	7
Vertical keel angles to flat keels on plates beyond inner bottom	7
Foundations.	
Boiler foundations generally	6
Engine foundations generally	6
Turret foundations	5
Gun foundations	5
Miscellaneous.	
Armor shelf angles to plating	5
Turret structures below shelf plate	5
Quilting rivets, generally	14
Where strength and watertightness are not required	8
Where strength is required, but not watertightness	5
In special cases where strength is required in connections of limited extent and in all other exceptional cases, spacing as required by circumstances, but rivets in the same line shall not be spaced closer than	3

12. **Comparison of Crosswise Tearing of the Plate with Crushing in Front of the Rivets.**—By such comparison we obtain a relation between p and d which is independent of the thickness of the plating:

$$nadtf_C \geqq [p - ad]tf_T$$

or:

$$p \leqq ad\left[1 + n\frac{f_C}{f_T}\right] \quad . \quad . \quad . \quad . \quad . \quad (61)$$

If this inequality holds good, the plate will tear before crushing occurs. The ultimate tensile strength f_T we shall here and in the following take to be 63,000 lb. per sq. in. for mild steel. The crushing strength or, rather, the allowable bearing pressure f_C we reckon as before 96,000 lb. per sq. in. for the same material; p is the standard spacing.

Substituting numerical values, we find that (61) is practically always satisfied for double riveted joints and that it is amply satisfied for treble and quadruple riveted joints. In fact, tearing of the plate or shearing of the rivets will by these latter joints take place while the bearing pressures are still moderate. For single riveted joints, where $n = 1$ and where, roughly, a varies from 1·1 for cylindrical holes to 1·4 for countersunk holes, equation (61) requires a spacing smaller than $2\frac{3}{4}d$ and $3\frac{1}{2}d$ respectively. This condition is not ordinarily fulfilled, whence crushing of the plate may here occur before tearing of the plate. Single riveting is used only in light plating, and we have seen already that in very light plating crushing of the plate may occur also before shearing of the rivets. Hence, *very light plating, $\frac{1}{4}$-in. or less in thickness, if single riveted, is likely to give way by crushing of the plate in front of the rivets, with attendant forms of rupture.*

13. **Comparison of Crosswise Tearing of the Plate with Shearing of the Plate as by Mode (b).**—It is here required, for single riveted joints, that the resistance to shearing along the lines AB and CD (fig. 55) shall be at least as great as the resistance to tearing of the metal between the rivet holes. Hence we must have :

$$2\left(a-\frac{d}{4}\right)f_s \geqq (p-ad)f_T \, . \qquad (62)$$

If we take $a = 1\frac{5}{8}d$ and $f_s = 50{,}000$ lb. per sq. in., we find that in single riveted joints the spacing must be not greater than about $3\frac{1}{2}$ diameters—a condition which is usually fulfilled in practice.

Where there are two or more rows of rivets, the inequality (62) is amply satisfied.

If a joint is to be watertight, it is generally calked, and in order that this may be done satisfactorily it is necessary to space the rivets rather closely along the calking edge. Generally a spacing of 4, $4\frac{1}{2}$, or 5 diameters is used, depending somewhat on the thickness of the plating and the intensity of the water pressure. A spacing of 6 diameters, which should be used only in heavy plating, may be considered a maximum along a calked edge. Closer spacing, 3 or $3\frac{1}{2}$ diameters, is necessary in oiltight work, and even a spacing of $2\frac{1}{2}$ diameters has been used in some cases where oiltightness was the primary requirement and where strength was not so important.

14. **Mode (e): Tearing Across of one Plate along an Intermediate Row of Rivets.**—Fracture along an intermediate row, such as 2, fig. 57, can only take place together with shearing of all the rivets between the fractured row and the edge of the intact plate. We may, therefore, without reducing the resistance to this mode of fracture, increase the number of rivets in succeeding rows as we go inwards from the outer row. Joints so designed, illustrated in fig. 72, are generally referred to as "diamond joints." The ideal condition is attained when the resistance to tearing of a plate along any row, together with shearing of the rivets outside that row, is equal to the resistance to tearing along the

outer row, and when, at the same time, it is equal to the resistance to shearing of all the rivets. Such a joint is said to be of "uniform strength." In reality, this term is strictly correct only for fracture by modes (*c*), (*d*), and (*e*), since the resistance to fracture by modes (*a*) and (*b*) cannot be so accurately adjusted.

It will be observed, that the attainable maximum strength of a joint of uniform strength is reached when the outer row contains only one rivet, in which case the strength of the joint approaches that of the intact plate.

15. **Mode (f): Tearing Across of both Plates along the Same or Different Rows of Rivets.**—Fracture by this mode may or may not comprise shearing of rivets. It cannot take place in ordinary chain-riveted joints, except—theoretically—along the outer row on each side and then only if the plates are of the same thickness, but in practice one plate will always give way before the other. Fracture of both plates will, in fact, occur only in diamond joints and joints of similar nature, but, as long as the pitch does not fall below the prescribed limit of $3d$ it will not occur even there if the joints are of uniform strength.

FIG. 58.

16. **Summary.**—Summing up the discussion of this Section, it is concluded that fracture by modes (*a*), (*b*), and (*f*), *i.e.* breaking or shearing of the plate in front of the rivets and tearing across of both plates, are practically prevented by designing the joints in accordance with ordinary good practice. Fracture by pulling the rivets through the plates and by mode (*b*) shearing of the plate in front of the rivets, induced by crushing of the plate, may, however, occur in single riveted joints in light plating and should, therefore, be carefully considered where strength is of importance. It may be prevented by the use of double riveted joints. There remains fracture by modes (*c*), (*d*), and (*e*): shearing of all the rivets, crosswise tearing of the plate, and combinations of these two modes. A mathematical discussion of these modes of fracture forms the subject of the following chapter.

CHAPTER VIII.

*CALCULATIONS FOR RIVETED JOINTS.**

26. INTRODUCTORY REMARKS.

WE shall now show how to design a riveted joint so as to obtain the greatest possible strength consistent with the conditions that obtain in shipbuilding. The strength of a joint is in this chapter considered by itself, *i.e.* independent of the reinforcement which it may receive from adjacent passing strakes or other members of the structure. We begin with the simplest forms and gradually proceed to the more complex forms such as diamond joints and the allied cases of reinforcing and doubling plates.

In shipbuilding it is not, generally, necessary or convenient to use the diamond joint in its complete, ideal form. The great strength attainable by this joint is rarely required, because there is no object in making the joints of the plates stronger than along frames, beams, stiffeners, etc. Moreover, the spacing of frames and beams does not permit very long overlaps at the butts, and the weight of the joints as well as the amount

* This subject was treated first and most fully by French naval constructors. See Clauzel, *Étude sur le Rivetage*, Paris, 1882; Callou, *Mémorial du Génie Maritime*, 1901, and *Construction du Navire*, vol. i., Paris, 1902.

of calking would be unduly increased by long diamond joints. In shipbuilding there are, for these reasons, rarely more than three or four rows of rivets in a lapped joint and, generally, the number of rivets is the same in all the rows.

As explained above, we take the strength by shearing of all the rivets as the standard to which other modes of fracture are compared, and the pitch is chosen so as to make the resistance to tearing of the plate along the outer row equal to or slightly greater than this standard. A joint so designed is often referred to loosely as of "uniform strength," but evidently, if the number of rivets is the same in all the rows, fracture along an intermediate row cannot occur, since it must always offer greater resistance than fracture along an outer row, and hence complete uniformity of strength of such a joint cannot be attained. By omitting alternate rivets in the outer row we may approach a more perfect uniformity.

We shall deal first with chain riveting and in each joint we shall take a typical group of rivets, a "repeating section," to represent the whole joint. This method is permissible in a general discussion of riveted joints; but in concrete cases the exact sectional area of the plate should be measured and the actual number of rivets that are shorn by the different modes of rupture should be counted.

We include in the calculation only that part of a joint, as for instance a butt in the outer shell, which is bounded by the edges of adjacent plates, for in the seams the arrangement of the rivets is independent of the riveting in the joints and must be dealt with separately.

The ratio between the ultimate tensile strength of the plates f_T and the shearing strength of the rivets f_S occurs very frequently in the following. The average value of this ratio is for mild steel plating and mild steel rivets:

$$\frac{f_T}{f_S} = \frac{63000}{50000} = 1\cdot26$$

a figure which will be used hereafter in the examples. For plates of high-tensile steel with rivets of the same material this ratio has about the same value, but where the rivets are of a softer material it may attain much higher figures. In practice it must, of course, be carefully evaluated in each given case.

27. SECTIONAL AREA OF RIVET HOLES.

In order to take account of the weakening which a plate suffers from the presence of rivet holes, it is necessary to determine the exact sectional area of these latter.

It is seen from Table X. that the rivet holes are from $\frac{1}{32}$ in. to $\frac{3}{32}$ in. larger in diameter than the rivets before these are staved up. The result is a weakening of the plate and a strengthening of the rivet. Let d' be the diameter of the rivet hole, and let $\frac{d'}{d} = \delta$.

When the hole is countersunk, the plate is still further weakened. As to the depth of countersinking, practice varies in different navies; frequently the rules of the Classification Societies are adopted. We shall here follow Lloyd's Rules, according to which the countersinking extends through the whole thickness of the plate when the thickness does not exceed ·60 of an inch, but only through nine-tenths the thickness in heavier plates. Let ϕ be the angle of countersink, which in the United States Navy varies from 32° in very large rivets to 60° in small rivets. Let h be the depth of the countersink, then the total sectional area of the rivet hole will be:

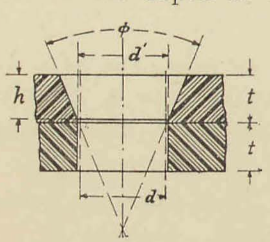

FIG. 59.

$$a' = td' + h^2 \tan\frac{\phi}{2}$$

or, if $a' = atd$

$$a = \delta + \frac{h^2}{td}\tan\frac{\phi}{2} \quad . \quad . \quad . \quad . \quad (63)$$

The factor a is, therefore, the ratio between the sectional area of the plate actually removed by the rivet hole and the net sectional area which would be occupied by the rivet in its original state. If the countersinking extends through the whole thickness:

$$a = \delta + \frac{t}{d}\tan\frac{\phi}{2} \quad . \quad . \quad . \quad . \quad (63')$$

When rivet holes are punched without being countersunk, they are but slightly conical. If the angle of the cone is known, the value of a may be calculated from (63′), but, ordinarily, punched holes may be considered as cylindrical. If the rivet hole is drilled cylindrical, we have $a = \delta$.

Hereafter, when the value of a occurs in the formulas, it must, therefore, be calculated or estimated in each case according as the holes are countersunk, punched, or drilled. Pl. VI. with Table XII. gives the values of a and δ for various thicknesses of plating when the rivets are designed in accordance with Table X.

In the following we have often occasion to use average or ordinary values of these coefficients for the purpose of exemplification and shall then take $\delta = 1{\cdot}07$ and $a = 1{\cdot}33$.

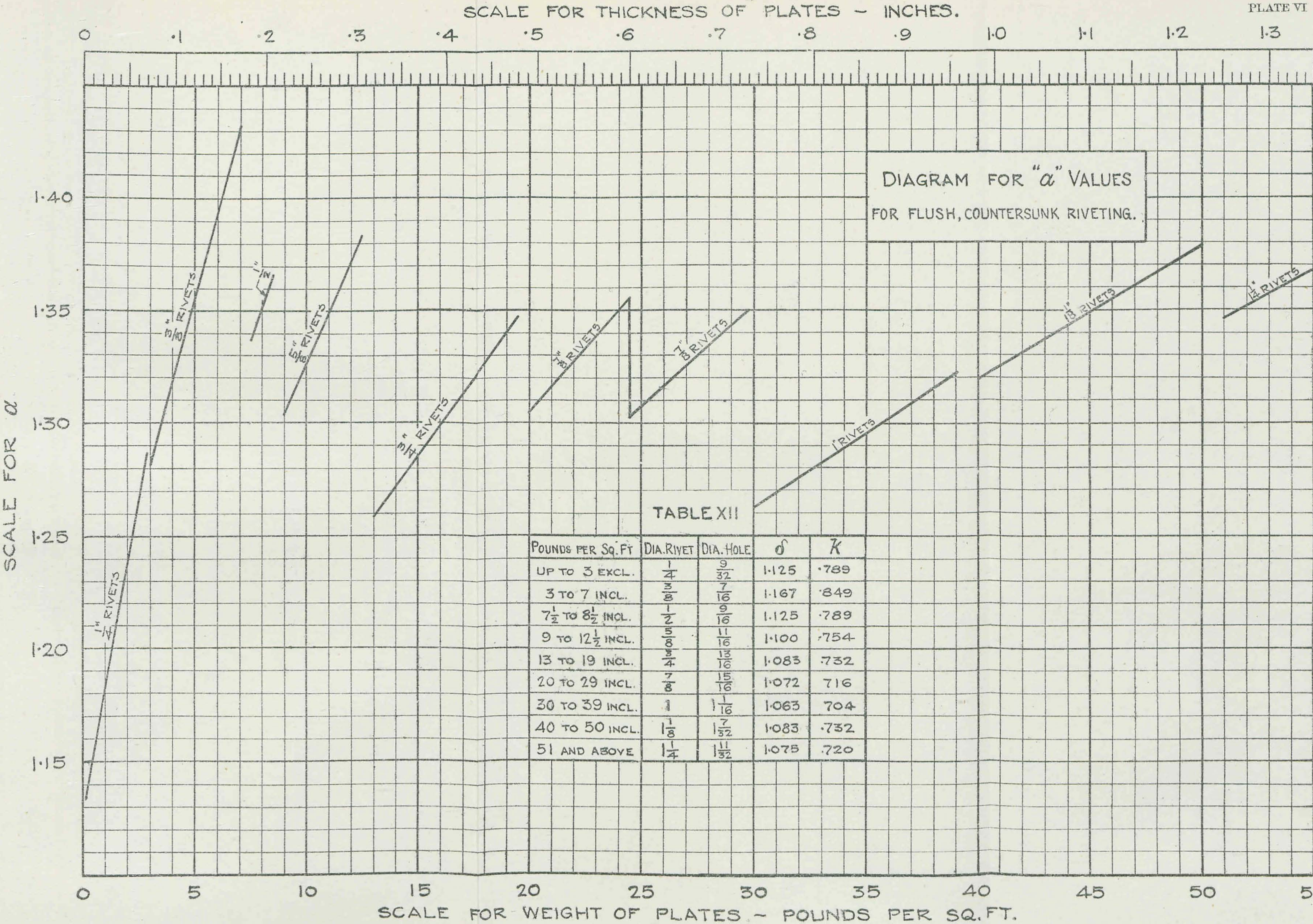

TABLE XII

POUNDS PER SQ. FT	DIA. RIVET	DIA. HOLE	δ	K
UP TO 3 EXCL.	$\frac{1}{4}$	$\frac{9}{32}$	1·125	·789
3 TO 7 INCL.	$\frac{3}{8}$	$\frac{7}{16}$	1·167	·849
$7\frac{1}{2}$ TO $8\frac{1}{2}$ INCL.	$\frac{1}{2}$	$\frac{9}{16}$	1·125	·789
9 TO $12\frac{1}{2}$ INCL.	$\frac{5}{8}$	$\frac{11}{16}$	1·100	·754
13 TO 19 INCL.	$\frac{3}{4}$	$\frac{13}{16}$	1·083	·732
20 TO 29 INCL.	$\frac{7}{8}$	$\frac{15}{16}$	1·072	·716
30 TO 39 INCL.	1	$1\frac{1}{16}$	1·063	·704
40 TO 50 INCL.	$1\frac{1}{8}$	$1\frac{7}{32}$	1·083	·732
51 AND ABOVE	$1\frac{1}{4}$	$1\frac{11}{32}$	1·075	·720

DEPTH OF COUNTERSINKING ACCORDING TO LLOYD'S 1914 RULES.

1" THICKNESS CORRESPONDS TO 40·75 LBS. PER SQ. FT.

To face p142 Hovgaard; Structural Design of Warships.

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28. JOINTS ORDINARILY USED IN SHIPBUILDING.

1. **Overlap and Single Strap. No Rivets Omitted.**—The following treatment applies to overlaps and straps alike.

If the plates are of unequal thickness, we need only deal with that of the smaller thickness. If the plates are of the same thickness, we may consider either plate, unless one of them is weakened more than the other, for instance by countersunk rivet holes, while the holes in the other plate are punched or drilled, in which case, again, the most weakened plate is to be dealt with.

Let n be the number of rows, and choose a repeating section such as that enclosed between the lines FF and FF on figs. 60 and 61, drawn midway between the rivets at a distance p from each other.

The condition to be fulfilled is that the plates, in tearing along an

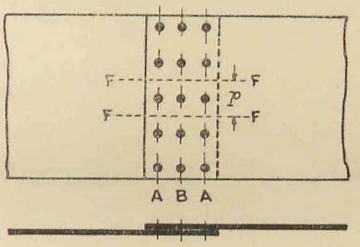


FIG. 60.

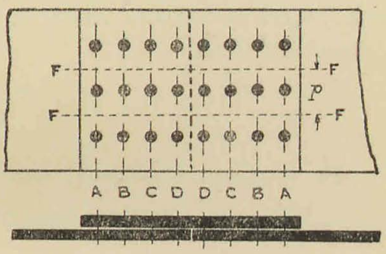


FIG. 61.

outer row, shall offer at least the same resistance as all the rivets in shearing. Hence

$$(p - ad)f_{\text{T}}t \geqq nf_{\text{s}}\frac{\pi d^2}{4}\delta^2$$

or

$$\frac{p}{d} \geqq a + n\frac{\pi}{4}\frac{f_{\text{s}}}{f_{\text{T}}}\delta^2\frac{d}{t}$$

Put $\frac{p}{d} = m$ which is the "spacing ratio" or "pitch ratio" and let

$$\frac{\pi}{4}\frac{f_{\text{s}}}{f_{\text{T}}}\delta^2 = k \quad . \quad . \quad . \quad . \quad . \quad (64)$$

The quantity k is a frequently recurring coefficient, which is independent of the arrangement, number, and pitch of the rivets; it depends on the quality of the materials and varies slightly with the diameter of the rivets. Its value is given in Table XII. on Pl. VI., corresponding to a ratio $\frac{f_{\text{T}}}{f_{\text{s}}} = 1{\cdot}26$. Where it is desired to use an ordinary average value of k we shall in the following reckon $k = {\cdot}72$.

Introducing the symbols m and k in the equation above, we obtain

$$m \geqq a + nk\frac{d}{t} \quad . \quad . \quad . \quad . \quad . \quad (65)$$

which gives the spacing-ratio for any values of n, d, and t.

The strength of a joint may be conveniently measured by the ratio between the minimum resistance offered by any given mode of fracture and the strength of the intact plate. This ratio is called the "efficiency" of the joint and is denoted by e.

Hence, if (65) holds good

$$e = \frac{nf_S\frac{\pi d^2}{4}\delta^2}{f_T pt} = \frac{nkd}{mt} \quad . \quad . \quad . \quad . \quad . \quad (66)$$

When the spacing is taken from Table XI. or given by other practical rules, and is smaller than required by (65), the efficiency must be determined from

$$e = \frac{[p - ad]f_T t}{f_T pt} = 1 - \frac{a}{m} \quad . \quad . \quad . \quad . \quad (67)$$

For a given efficiency, the number of rows n is according to (66) inversely proportional to the ratio $\frac{d}{t}$. Hence, in light plating, where this ratio is relatively great, the number of rows may be smaller than in heavy plating; see Table XIII.

Where strength is required we may say that, as a general rule, single riveting is suitable only for plating under 7½ pounds, double riveting for plating under 20 pounds, and treble riveting for plating under 30 pounds. For still heavier plating, when maximum strength is required, it is necessary to use quadruple riveting or even a still higher number of rows.

For a given number of rows, it is of advantage to use as great a ratio $\frac{d}{t}$ as possible, since in that case the efficiency increases directly as this ratio. This is otherwise obvious from the simple consideration, that when the diameter of a rivet is increased, its shearing strength increases as the square of the diameter, while the reduction in plate area is proportional to the diameter to the first power only. Moreover, for a given strength of joint, an increase in the size of rivets is accompanied by a reduction in the number of rivets. For these reasons *the rivets should be as large as consistent with the various limiting conditions.*

The efficiency is also directly proportional to k which depends chiefly on the ratio $\frac{f_S}{f_T}$. When, for instance, rivets of mild steel are used in plates of high-tensile steel, this ratio is very small, and a relatively great rivet area is required to attain a given efficiency.

The chain-riveted lap with the same number of rivets in all the rows is employed in watertight work as, for instance, in the seams and butt-laps of the outer shell. It is also used in double riveted single straps, where it would be difficult to obtain the requisite rivet area if any rivets were omitted, but in treble and quadruple riveted straps it becomes advantageous to omit some of the rivets in the outer rows, and then the formulas here given do not hold good.

Example.—Double riveted butt-lap. Countersunk rivets. Weight of plating 25 lb. per sq. ft. Determine m and e.

From Pl. VI. : $t = \cdot 61$ in., $d = \frac{7}{8}$ in., $a = 1 \cdot 31$, $k = \cdot 72$, $n = 2$.

According to (65)

$$m \geqq 1 \cdot 31 + 2 \frac{\cdot 72}{\cdot 61} \times \frac{7}{8} = 3 \cdot 38$$

With this spacing we find from (66) the maximum attainable efficiency

$$e = \frac{2 \times \cdot 72 \times \left(\frac{7}{8}\right)}{3 \cdot 38 \times \cdot 61} = \cdot 61$$

If we use the standard spacing $m = 4$, given by Table XI., we obtain from (66)

$$e = \frac{2 \times \cdot 72 \times \left(\frac{7}{8}\right)}{4 \times \cdot 61} = \cdot 52$$

and from (67)

$$e = 1 - \frac{1 \cdot 31}{4} = \cdot 67$$

As might be anticipated, the smallest efficiency is found by shearing of the rivets and falls considerably below the maximum attainable efficiency.

The following table gives the efficiency in some typical cases, first for the spacing determined by formula (65), secondly for the standard spacing. The rivets are assumed to be countersunk and the values of a and k are taken from Pl. VI.

TABLE XIII.—EFFICIENCY OF OVERLAPPED BUTTS.

Type of Riveting.	Single.		Double.					Treble.			Quadruple.		
Weight of plate—lb. sq. ft.	5	10	$7\frac{1}{2}$	10	15	20	25	25	30	35	35	40	45
Diameter of rivet—ins.	$\frac{3}{8}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	1	1	1	$1\frac{1}{8}$	$1\frac{1}{8}$
Standard spacing—diameters	$3\frac{1}{2}$	$3\frac{1}{2}$	4	4	4	4	4	$4\frac{1}{2}$	$4\frac{1}{2}$	$4\frac{1}{2}$	$4\frac{1}{2}$	$4\frac{1}{2}$	$4\frac{1}{2}$
Spacing for uniform strength	4·0	3·3	5·6	5·2	4·3	3·9	3·4	4·4	4·1	3·8	4·6	4·7	4·3
Efficiency of plate (standard spacing)	·61	·62	·67	·67	·68	·67	·67	·71	·72	·71	·71	·71	·70
Efficiency of rivets (standard spacing)	·74	·55	1·07	·96	·75	·64	·51	·68	·64	·55	·73	·75	·66
Efficiency with uniform strength	·66	·59	·76	·74	·70	·66	·61	·70	·69	·66	·72	·72	·69

Countersink depth according to Lloyd's 1914 Rules.

It is seen that by adopting the standard spacing we necessarily lose somewhat in efficiency, except in the rare cases where this spacing happens to be the same as that for uniform strength.

2. **Overlap and Single Strap. Alternate Rivets Omitted in Outer Rows.**—In *overlaps* this type of joint is rarely used in warships except in certain small torpedo-boats, where watertightness is obtained without calking by the use of some obturating material; see fig. 104. When the edge is to be calked, the spacing of the rivets in the row nearest the calking edge must not exceed a certain maximum. Hence, although we can obtain a great shearing strength of the rivets by a closer spacing of the rivets in the inner rows, we cannot increase the efficiency of the joint, which is limited by the spacing in the outer row. This construction, applied to overlaps, is, therefore, well adapted for use in merchant vessels where iron rivets are used and wherever it is desirable to secure an extra-large rivet area. It is adopted by the British Corporation, who for treble and quadruple riveted end joints specify a spacing in the outer rows of $4\frac{1}{2}$ diameters and in the central rows of 3 diameters, giving thus three rivets in the inner rows for every two in the outer rows.

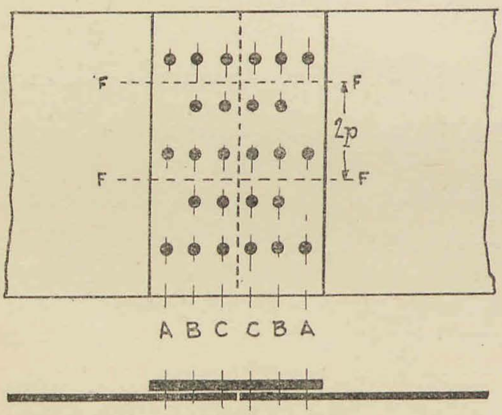


FIG. 62.

In *single straps* this construction is much used in warships for treble and quadruple riveted joints (fig. 62).

Consider a repeating section such as that enclosed between the lines FF and FF distant $2p$ from each other. Suppose first that the plates are of the same thickness, in which case we need only deal with one side of the joint. In addition to tearing of one of the plates along an outer row A we must here examine also the resistance to tearing along the row B next to the outer row, combined with shearing of the rivets in A. Moreover, we have to determine the thickness of the strap so as to prevent rupture along the close-spaced row nearest the butt, C.

Let there be n number of rows on each side of the butt, then there are $2n-1$ rivets in each repeating section. Comparing tearing of the plate along A with shearing of all the rivets, we have

$$[2p - da]f_T t \geqq [2n - 1]f_S \frac{\pi d^2}{4} \delta^2$$

$$\therefore \quad m \geqq \frac{a}{2} + \frac{2n-1}{2} k \frac{d}{t} \quad . \quad . \quad . \quad . \quad (68)$$

If this holds good

$$e = \frac{(2n-1)f_S \frac{\pi d^2}{4} \delta^2}{2f_T pt} = \frac{(2n-1)kd}{2tm} \quad . \quad . \quad . \quad (69)$$

When the standard spacing is used and falls below the value of m determined by (68), the efficiency must be found from the formula

$$e = \frac{[2p - da]f_T t}{2f_T pt} = 1 - \frac{a}{2m} \quad . \quad . \quad . \quad (70)$$

The thickness of the strap t' is best determined by comparing tearing of the strap along C with tearing of the plate along A, whereby we obtain t' expressed in terms of t

$$[2p - 2da']f_T t' \geqq [2p - da]f_T t$$

$$\therefore \quad t' \geqq \frac{m - \frac{a}{2}}{m - a'} t \quad . \quad . \quad . \quad . \quad . \quad (71)$$

The coefficient a' refers to the strap; it is approximately equal to δ and is always greater than $\frac{a}{2}$. Hence t' should theoretically be greater than t but the difference is ordinarily small. In fact, for lighter plating, the strap is in practice quite frequently made of the same thickness as the plates. Finally, compare tearing of the plate along B combined with shearing of the rivets along A with tearing of the plate along A

$$[2p-2da]f_T t+f_S\frac{\pi d^2}{4}\delta^2 \geqq [2p-da]f_T t$$

$$\therefore \quad d > \frac{a}{k}t \quad . \quad . \quad . \quad . \quad . \quad (72)$$

For riveting in accordance with Table X. this condition is fulfilled for all lighter plates up to and inclusive of 17 lb. In other words, for such plating tearing along B will not occur and formulas (68), (69), and (71) may be used.

For plates heavier than 17 lb. per sq. ft. tearing along B will occur earlier than tearing along A and we must, therefore, in such a case compare resistance to tearing along B to shearing of all the rivets

$$[2p-2da]f_T t+f_S\frac{\pi d^2}{4}\delta^2 \geqq [2n-1]f_S\frac{\pi d^2}{4}\delta^2$$

$$\therefore \quad m \geqq a+(n-1)k\frac{d}{t} \quad . \quad . \quad . \quad . \quad . \quad (73)$$

If this holds good, find e as above from equation (69), but if not, we must find e from the equation

$$e = \frac{[2p-2da]f_T t+f_S\frac{\pi d^2}{4}\delta^2}{2f_T pt} = 1-\frac{a}{m}+\frac{kd}{2tm} \quad . \quad . \quad (74)$$

In order to prevent tearing of the strap, we must have

$$[2p-2da']f_T t' \geqq [2p-2da]f_T t+f_S\frac{\pi d^2}{4}\delta^2$$

$$\therefore \quad t' \geqq \frac{m-a}{m-a'}t+\frac{kd}{2(m-a')}$$

If (73) holds good as an equality this becomes :

$$t' \geqq \frac{2n-1}{2(n-1)}\frac{m-a}{m-a'}t \quad . \quad . \quad . \quad . \quad . \quad (75)$$

If the two plates in a strapped joint are of unequal thickness, design first the joint for the thinner plate so as to obtain the desired strength or efficiency. This being secured, there can be no object in giving greater strength, *i.e.* greater absolute resistance, to the joint on the side of the heavier plate. Hence, make the shearing area of the rivets in the heavy plate the same as in the thinner plate. This might be done by using a smaller number of larger rivets, whereby an excess of strength would be attained in the strap and in the heavy plate, but this excess is of no use since the strength is already determined by the resistance to shearing of the rivets. Hence it is preferable to simplify the design by using the same size of rivets, viz. that corresponding to the heavier plate, and the same arrangement of the rivets in both plates.

Example 1.—Single strap, treble riveted.

The rivets are countersunk in the plates but punched or drilled cylindrical in the strap. Thickness of plates $\frac{5}{8}$ in.

Find spacing-ratio, efficiency, and thickness of strap for uniform strength. $t = \frac{5}{8}$ in., $d = \frac{3}{4}$ in., $k = \cdot 73$, $a = 1\cdot 29$, $a' = \delta = 1\cdot 08$, $n = 3$.

Since $d > \frac{a}{k}t$ use formulas (68), (69), and (71).

$$m = \frac{1\cdot 29}{2} + \frac{5}{2} \times \cdot 73 \times 2 = 4\cdot 30$$

$$e = \frac{5 \times \cdot 73 \times \frac{3}{4}}{2 \times \frac{5}{8} \times 4\cdot 30} = \cdot 85$$

$$t' \geqq \frac{4\cdot 30 - \cdot 65}{4\cdot 30 - 1\cdot 08} \times \frac{5}{8} = \cdot 43 \text{ or about } \tfrac{7}{16} \text{ in.}$$

Example 2.—Single strap, quadruple riveted.

The rivets are countersunk in the plates, but punched or drilled cylindrical in the strap. Thickness of plates 1 in.

Find spacing-ratio, efficiency, and thickness of strap for uniform strength. $t = 1$ in., $d = 1\frac{1}{8}$ in., $k = \cdot 73$, $a = 1\cdot 32$, $a' = \delta = 1\cdot 08$, $n = 4$.

Since $d < \frac{a}{k}t$ use formulas (73), (69), and (75).

$$m = 1\cdot 32 + 3 \times \cdot 73 \times \frac{9}{8} = 3\cdot 78$$

$$e = \frac{7 \times \cdot 73 \times \left(\frac{9}{8}\right)}{2 \times 3\cdot 78} = \cdot 76$$

$$t' \geqq \frac{7}{6} \times \frac{3\cdot 78 - 1\cdot 32}{3\cdot 78 - 1\cdot 08} = 1\cdot 06 \text{ or about } 1\tfrac{1}{16} \text{ in.}$$

3. **Double Butt Straps. No Rivets Omitted.**—In double butt straps the action of the straining forces is symmetrical, wherefore the bending, which exists in single butt straps, is avoided. Moreover, the rivets are in double shear and may, therefore, be of smaller diameter

than in other joints. A reduction in diameter is in lighter plates in any case necessary in order to avoid an excessive bearing pressure, as explained in SECTION **25**, *5*, where it was shown that the diameter ought not to exceed one and one-half times the thickness of the plate. According to Table X., a reduction in diameter of $\frac{1}{8}$ in. is prescribed for plating up to 25 lb., but for heavier plating, where the standard diameter is already relatively small, no reduction is required.

Double butt straps of this type are used to best advantage in non-watertight work where great strength is required. Where great strength is to be associated with watertightness, they are not so well adapted, since the rivets in the outer rows must be closely spaced for satisfactory calking and thus the special advantage of this joint, its high efficiency, cannot be realised. Calking, moreover, is rendered difficult because, relative to the thin straps, the rivets are far from the edges and very widely spaced. For instance, in $\frac{1}{2}$-in. plating the straps are about $\frac{1}{4}$ in. thick; the rivets, of $\frac{3}{4}$-in. diameter, are placed a distance of $\frac{5}{4}$ in. or five times the thickness of the straps from the edges, while the spacing may be 5 diameters or fifteen times the thickness of the strap. Under these circumstances, the edges of the strap are not well supported by the rivets, they will tend to bend under the calking tool and lack rigidity when the ship is working.

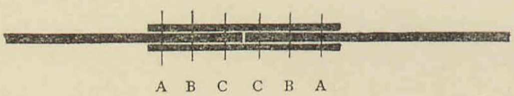


FIG. 63.

Assume the shearing strength of the rivets in double shear to be 1·75 times as great as in single shear, and compare tearing of the plate along an outer row A with shearing of all the rivets

$$[p - d\delta] f_T t \geqq 1 \cdot 75 n f_S \frac{\pi d^2}{4} \delta^2$$

$$\therefore \quad m \geqq \delta + 1 \cdot 75 n k \frac{d}{t} \quad . \quad . \quad . \quad (76)$$

and

$$e = \frac{1 \cdot 75 n k d}{m t} \quad . \quad . \quad . \quad . \quad . \quad (77)$$

If the spacing is smaller than determined by (76)

$$e = 1 - \frac{\delta}{m} \quad . \quad . \quad . \quad . \quad . \quad (78)$$

The factor δ is here used instead of a since the holes in the plate will generally be drilled cylindrical.

In order to prevent the straps from tearing along the row C nearest the butt, we must have :—

$$2[p - da']f_T t' \geqq [p - d\delta]f_T t$$

where a' is the factor for rivet area in the straps, assuming the holes to be of the same type in both straps.

$$\therefore \quad t' \geqq \frac{m-\delta}{m-a'}\frac{t}{2}. \qquad \qquad (79)$$

If the holes in the straps are drilled, $a' = \delta$ and the thickness of the straps need not be greater than half the thickness of the plate. If the holes are punched or countersunk, $a' > \delta$ and the straps should be thicker than $\frac{t}{2}$.

On account of the high efficiency of double straps they permit the use of double riveted joints in many cases where, with overlap or single strap, treble riveting would be necessary.

Example.—Double butt strap. Double riveted.

$t = \frac{5}{8}$ in., $d = \frac{7}{8}$ in., $k = \cdot72$, $\delta = 1\cdot07$, $n = 2$. Let $a' = \delta$

$$\therefore \quad m \geqq 1\cdot07 + 1\cdot75 \times 2 \times \cdot72 \times \frac{7}{5} = 4\cdot60$$

$$e = \frac{1\cdot75 \times 2 \times \cdot72 \times 7}{4\cdot60 \times 5} = \cdot77$$

$$t' = \frac{t}{2} = \tfrac{5}{16} \text{ in.}$$

The efficiency of a treble riveted overlap or single strap would be only ·70.

4. **Double Butt Straps. Alternate Rivets Omitted in Outer Rows.**—This joint is frequently used in shipbuilding in non-watertight work, for instance in the vertical keel, generally treble riveted.

The repeating section on each side of the butt comprises $(2n-1)$ rivets. Compare tearing of the plate along A with shearing of all the rivets :

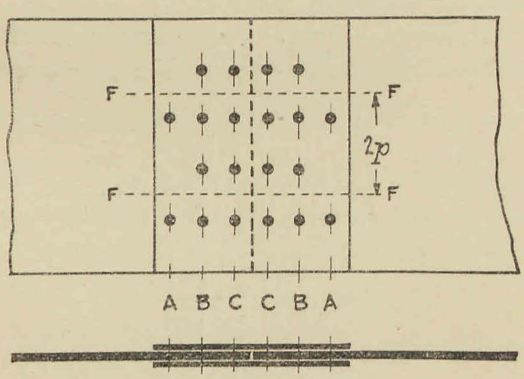


FIG. 64.

$$[2p - d\delta]f_T t \geqq 1 \cdot 75(2n - 1)f_S \frac{\pi d^2}{4} \delta^2$$

$$\therefore \quad m \geqq \frac{\delta}{2} + \cdot 875(2n - 1)k\frac{d}{t} \quad . \quad . \quad . \quad (80)$$

$$e = \frac{\cdot 875(2n - 1)kd}{mt} \quad . \quad . \quad . \quad . \quad (81)$$

or, if (80) does not hold good :

$$e = 1 - \frac{\delta}{2m} \quad . \quad . \quad . \quad . \quad . \quad (82)$$

In order that the straps shall not tear along C they must have a thickness t' determined by

$$2[2p - 2da']f_T t' \geqq [2p - d\delta]f_T t$$

or

$$t' \geqq \frac{2m - \delta}{2(m - a')} \frac{t}{2} \quad . \quad . \quad . \quad (83)$$

which is always greater than $\frac{t}{2}$.

In order to prevent tearing of the plate along B we must have

$$[2p - 2d\delta]f_T t + 1 \cdot 75 f_S \frac{\pi d^2}{4} \delta^2 \geqq [2p - d\delta]f_T t$$

or

$$d \geqq \frac{\delta t}{1 \cdot 75k} \quad . \quad . \quad . \quad (84)$$

which is fulfilled for all thicknesses.

This joint is rarely used quadruple riveted, since its efficiency as treble riveted is sufficiently high in most cases. It may be used as a substitute for the quadruple riveted overlap or single strap, as will appear from the following example.

Example.—Double butt strap. Treble riveted. Alternate rivets omitted in the outer rows. Thickness of plates 1 in.

$t = 1$ in., $d = \frac{9}{8}$ in., $k = \cdot 73$, $\delta = 1 \cdot 08$, $n = 3$. Assume $a' = \delta$.

$$m = \frac{1 \cdot 08}{2} + \cdot 875 \times 5 \times \cdot 73 \times \frac{9}{8} = 4 \cdot 13$$

$$e = \frac{\cdot 875 \times 5 \times \cdot 73 \times \frac{9}{8}}{4 \cdot 13} = \cdot 87$$

$$t' \geqq \frac{2 \times 4 \cdot 13 - 1 \cdot 08}{2[4 \cdot 13 - 1 \cdot 08]} \times \frac{1}{2} = \cdot 59 \text{ in. or about } \tfrac{5}{8} \text{ in.}$$

For a quadruple riveted single strap the efficiency was found to be only ·76.*

From equation (59) we find the bearing pressure

$$f_c = \frac{1{\cdot}75\pi f_s \delta d}{4t} = \frac{1{\cdot}75\pi \times 50000 \times 1{\cdot}08 \times \frac{9}{8}}{4} = 83{,}500 \text{ lb. per sq. in.}$$

which is well within the permissible limit, 110,000 lb. per sq. in. for rivets in double shear.

5. **Special Type of Double Butt Strap for Watertight Work.**— By making one strap wider than the other so as to take one more row of rivets on each side of the butt, we may combine watertightness with a high efficiency of the joint. The narrow strap is placed on the calking side with close-spaced rivets, while the rivets in the outer row of the wider strap are openly spaced, generally by omitting alternate rivets.

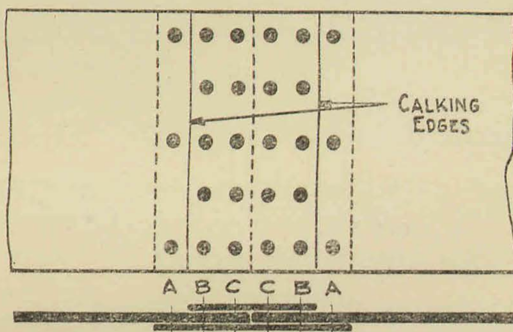


FIG. 65.

This joint is used in the outside plating of many destroyers and also frequently in merchant vessels (the Cunard liner the *Lusitania*) and in the shell of boilers.

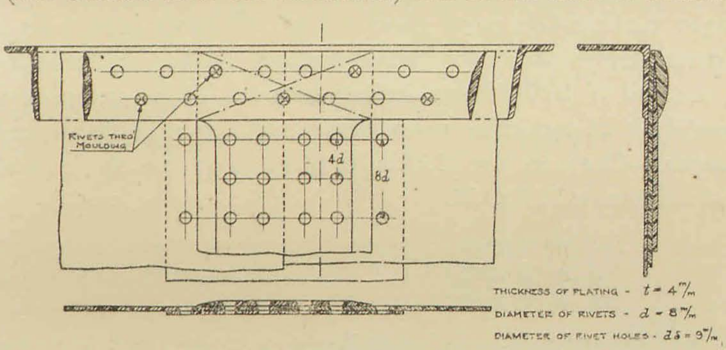


FIG. 66.—Double Butt Strap in Sheer Strake of Torpedo-Boat.

Let us assume that the rivets are countersunk in the outer strap and in the plates in row A but cylindrical in the inner strap and in the plates in rows B and C. Proceeding as in case of ordinary double butt straps, we find the efficiency

$$e = \frac{[3{\cdot}5(n-1)+1]kd}{2mt} \quad . \quad . \quad . \quad . \quad (85)$$

* See Example 2, Article 2 of this Section.

where m is the spacing-ratio in the inner rows and n is the total number of rows on one side of the butt.

As in other cases where rivets are omitted in the outer row, there is a limit to the thickness of the plates beyond which fracture along row B is liable to occur before fracture along row A. This limit is determined by

$$t = \frac{kd}{(2\delta - a)} \quad . \quad . \quad . \quad . \quad . \quad (86)$$

and lies at about $t = 1$ in.

For plates of less than 1-in. thickness the spacing-ratio is

$$m \geqq \frac{a}{2} + [1 \cdot 75(n-1) + \tfrac{1}{2}]\frac{kd}{t} \quad . \quad . \quad . \quad (87)$$

and the thickness of the straps

$$t' \geqq \frac{2m - a}{2m - \delta - a'} \frac{t}{2} \quad . \quad . \quad . \quad . \quad . \quad (88)$$

For plates thicker than 1 in.

$$m \geqq \delta + 1 \cdot 75(n-1)\frac{kd}{t} \quad . \quad . \quad . \quad . \quad . \quad (89)$$

and
$$t' \geqq \frac{(m-\delta)(3 \cdot 50(n-1)+1)}{1 \cdot 75(2m - \delta - a')(n-1)} \frac{t}{2} \quad . \quad . \quad . \quad (90)$$

In light plating this joint gives a high efficiency even with $n = 2$ and is particularly suited for the important strakes in torpedo-vessels both on account of its lightness and on account of its symmetry, which latter quality is of particular value where buckling is liable to occur.

Example.—Consider the double butt strap in the sheer strake of a torpedo-boat shown in fig. 66.

$t = 4$ mm.	$a = 1 \cdot 40$
$d = 8$ mm.	$k = \cdot 79$
$d\delta = 9$ mm.	$n = 3$

Find first the spacing-ratio for uniformity of strength and the corresponding efficiency.

From (87): $$m = \cdot 70 + \frac{4 \times \cdot 79 \times 8}{4} = 7$$

From (85): $$e = \frac{[3 \cdot 5 \times 2 + 1] \cdot 79 \times 8}{2 \times 7 \times 4} = \cdot 90$$

A spacing of 7 diameters for the inner rows would, however, be too wide for watertight work. In fig. 66, which represents an actual case, the spacing is 4 diameters, which gives the following efficiencies:

From (85): $e = \dfrac{8 \times \cdot79 \times 8}{2 \times 4 \times 4} = 1\cdot58$ for the rivets

while for the plate $e = 1 - \dfrac{a}{2m} = 1 - \dfrac{1\cdot40}{2 \times 4} = \cdot83$

The great excess of rivet area found in this joint is a feature which, as explained in SECTION 25, 2, is highly desirable in the most strained parts of torpedo-vessels in order to secure great rigidity and to prevent frictional slip of the rivets under all circumstances. It is of interest to examine what will be the stress in the plate, p_T, when slip of the rivets takes place, which we shall assume to occur when the shearing stress is 7 ts. per sq. in. Comparing the tensile strength of the plate between the rivet holes in the outer row with the shearing strength of the rivets, we have

$$(2m - a)dtp_T = [3\cdot5(n-1)+1]\frac{\pi d^2}{4}\delta^2 \times 7$$

$$\therefore \; p_T = \frac{8 \times \pi \times 9 \times 9 \times 7}{(8 - 1\cdot4)4 \times 8 \times 4} = 16\cdot9 \text{ ts. per sq. in.}$$

whence it appears that the plate will reach the elastic limit at about the same time as the rivets begin to slip. The general stress in the solid plate outside the joint will be $\cdot83 \times 16\cdot9 = 14\cdot0$ ts. per sq. in.

6. **The Backstrap Joint.**—Imagine an ordinary overlapped joint to be reinforced by an internal strap taking one row of rivets inside the calked edge and one row outside this edge. In both of the outer rows every second rivet is omitted, while the rivets along the calked edge are spaced for watertight work. Then we have the strapped overlap shown in fig. 67. The strap will be here referred to as the "backstrap," and the joint as the "backstrap joint."

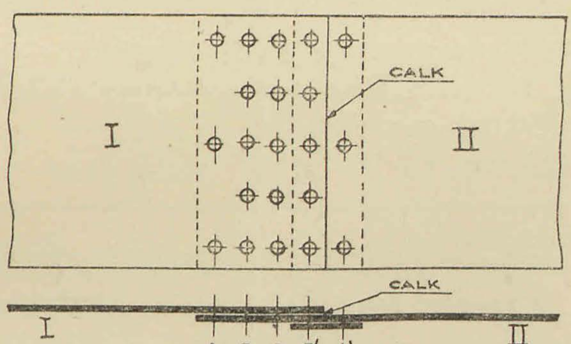


FIG. 67.

Proceeding as usual, we find the efficiency:

$$e = \frac{(2n-1)kd}{2mt} \quad . \quad . \quad . \quad . \quad . \quad (91)$$

where n is the number of rows excluding the outer row through the backstrap. Again, a limit to the thickness of the plates is found beyond which fracture along row B or B′ is liable to occur before fracture along A or A′. Assuming that the rivets are countersunk on the calked side, this limit is determined by

$$t \leqq \frac{kd}{a} \quad . \quad . \quad . \quad . \quad . \quad . \quad (92)$$

which gives t equal to about one-half inch as the limiting value.

For plates of less than $\frac{1}{2}$-in. thickness the spacing-ratio is

$$m \geqq \frac{a}{2}+\frac{2n-1}{2}\frac{kd}{t} \quad . \quad . \quad . \quad . \quad . \quad (93)$$

The thickness of the backstrap is found by comparing simultaneous tearing of the strap and the plate along row B′ with tearing of the plate along an outer row

$$t' \geqq \frac{2\delta - a}{m-\delta}\frac{t}{2} \quad . \quad . \quad . \quad . \quad . \quad (94)$$

For plates thicker than one-half inch

$$m \geqq a+(n-1)\frac{kd}{t} \quad . \quad . \quad . \quad . \quad . \quad . \quad (95)$$

and

$$t' \geqq \left[\frac{m-a}{2(n-1)} - a + \delta\right]\frac{t}{m-\delta} \quad . \quad . \quad . \quad . \quad (96)$$

provided (95) holds good as an equality.

The following table gives a comparison between the backstrap joint and certain other joints.

TABLE XIV.

Thickness of Plate and Diameter of Rivets.	Ordinary Chain-riveted Overlap.	Single Strap. Alternate Rivets Omitted in Outer Rows.	Backstrap. Alternate Rivets Omitted in Outer Rows.
$t = \frac{1}{4}''$ $d = \frac{5}{8}''$	$n = 3$ $m = 6{\cdot}9$ $e = {\cdot}81$ ...	$n = 3$ on each side of butt $m = 5{\cdot}4$ $e = {\cdot}88$ $t' = \frac{9}{32}''$	$n = 3$ $m = 5{\cdot}4$ $e = {\cdot}88$ $t' = \frac{1}{32}''$
$t = 1''$ $d = \frac{9}{8}''$	$n = 4$ $m = 4{\cdot}6$ $e = {\cdot}71$ ...	$n = 4$ on each side of butt $m = 3{\cdot}8$ $e = {\cdot}76$ $t' = \frac{17}{16}''$	$n = 4$ $m = 3{\cdot}8$ $e = {\cdot}76$ $t' = \frac{5}{32}''$

It is seen that the weight of the backstrap itself is insignificant, although it is, of course, in practice made heavier than required by the formula. Comparing the backstrap joint with the ordinary overlap, the former gives a superior efficiency with a much closer spacing of the rivets along the calked edge. In fact, an efficiency of ·81 is obtained in the ordinary overlap only by adopting a spacing of nearly 7 diameters, which is much too wide for watertight work. Comparing with the single strap, the backstrap joint gives the same efficiency with a great saving in weight and riveting, and with all the advantages that follow from overlapped as compared with butted joints.

Backstrap may, therefore, be used with advantage wherever a high efficiency is to be attained in watertight work, and where a saving in weight is of special importance, but where buckling is liable to occur, as in the sheer strakes and deck strakes of torpedo vessels, the double butt strap described in the last article is to be preferred.

29. OBLIQUE FRACTURE.

We have so far dealt only with lines of fracture normal to the direction of the stress, but in many cases fracture may occur along lines oblique to this direction, as, for instance in zigzag riveting and in diamond joints, where a line through the boundary rivets may show less resistance than a transverse line through the outer row. Oblique fracture differs essentially from simple tearing of the plate along a line normal to the direction of the stress in that the tearing action is combined with shearing. It is necessary to study this question in its fundamental aspects.

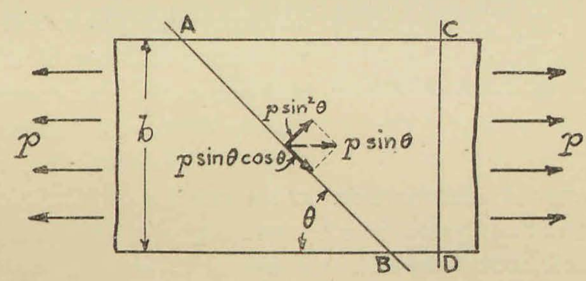


FIG. 68.

Consider a steel plate, assumed to be perfectly isotropic, subject to a uniform tensional stress p and let us examine the strength of the plate along a line AB forming an angle θ with the direction of the stress.

The stress per unit area along such a line will be $p\sin\theta$ and its components will be a normal stress :

$$p_n = p\sin^2\theta$$

and a tangential stress :

$$p_s = p\sin\theta\cos\theta$$

At large values of θ where AB is nearly normal to the direction of the stress, fracture along AB will take place essentially by tearing, but at smaller values of θ the shearing action will predominate. We may therefore expect that fracture will occur when $p\sin\theta$ reaches some value between the ultimate breaking stress of the material f_T and the ultimate shearing stress f_S which for mild steel is equal to about $\frac{4}{5}f_T$. Let us call this limiting stress kf_T where k is a function of θ and lies between 1 and $\frac{4}{5}$. The theory of elasticity does not furnish any reliable method of determining k and no exhaustive experiments are known to have taken place by means of which its value can be estimated in an accurate manner. It seems certain, however, that unless the line AB is weakened in some way, $p\sin\theta$ cannot reach the value kf_T before fracture takes place along a line normal to the stress, such as CD, *i.e.* before $p = f_T$. In other words, fracture will always, in an isotropic material exposed to a simple tensile stress, take place along a line normal to the stress unless an oblique line of weakness exists.

Let AB in fig. 69 be such a line, weakened by rivet holes, so that the sectional area of metal along this line is reduced in the ratio e', which is, therefore, the efficiency under a pull normal to AB.

Since the lines of stress will probably, as indicated in fig. 69, cross the line AB more nearly at right angles than if no rivet holes were found, the action, when fracture takes place, will have more the character of tearing and less of shearing than if the holes did not exist. Moreover, the grooving effect of the rivet holes will increase the strength along AB. For these reasons we conclude that k will be increased, and the more so the smaller θ is. Hence we shall probably not make any great error in assuming k to be equal to unity at all values of θ. It is, however, well to consider $k = 1$ as an upper limiting value.

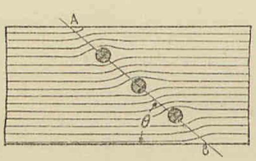


Fig. 69.

Fracture along AB will accordingly occur when

$$p\sin\theta = e'f_T$$

or

$$p = e'f_T\operatorname{cosec}\theta$$

The efficiency of the plate under oblique fracture is therefore:

$$e = \frac{bte'f_T\operatorname{cosec}\theta}{btf_T} = e'\operatorname{cosec}\theta \quad . \quad . \quad . \quad (97)$$

Now suppose e to be the standard efficiency at which we aim, being, for instance, the strength of the shell along a transverse frame or of the deck along a beam. We must then, according to (97), make:

$$bte'\operatorname{cosec}\theta \geqq bte$$

which can be expressed by the following empirical rule: *In order to prevent fracture along a line oblique to the direction of the stress, and weakened by rivet holes, the net sectional area of metal along this line—or, generally, the net length* $be' \operatorname{cosec}\theta$ *—should be at least as great as along a line normal to the direction of the stress, possessing standard strength.*

The spacing-ratio required to prevent fracture along an oblique line is $m' \geqq \frac{p'}{d}$ where p' is the pitch along this line, but since $e' = 1 - \frac{ad}{p'} = e\sin\theta$ we obtain

$$m' \geqq \frac{a}{1 - e\sin\theta} \quad . \quad . \quad . \quad . \quad (98)$$

30. ZIGZAG RIVETING.

One of the most important cases where oblique fracture has to be considered is zigzag riveting.

In the United States Navy and in the British Navy this mode of riveting is in large vessels only used in special cases, while in the French Navy it is used almost exclusively. In all navies zigzag riveting is employed extensively in torpedo-vessels.

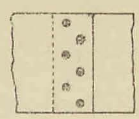


FIG. 70.

The advantages of zigzag riveting are:

(1) The rows may be placed somewhat closer together, whereby the width of the overlap may be reduced and weight may be saved. Hence its use in the landing of plates on stem and sternpost and in double riveted angle bars. We have seen that in the United States Navy the minimum distance between rows, which for chain riveting is $2\frac{1}{2}$ to 3 diameters, is for zigzag riveting $1\frac{3}{4}$ diameters for rivets spaced 4 diameters apart in the rows. In the French Navy the distance between the rows for zigzag riveting is usually equal to about 2 diameters.

(2) The strength of the plates is probably better developed by zigzag riveting than by chain riveting, although this point has not been proved conclusively.

(3) A more perfect watertightness may be obtained, in particular in case of light plates, on account of the greater closeness of the rows and the better distribution of the clamping effect.

On the other hand, zigzag riveting requires more care in the design of the joints and in marking off the rivet holes on the plates. Some rivets are apt to be lost where a joint is crossed by a structural member as, for instance, where a seam is crossed by a frame. In a butt joint with several rows of rivets there will, with a given spacing, be one rivet less in every second row than in the case of chain riveting.

The formulas for chain riveting apply also to zigzag riveting, but in addition it has to be ascertained that the distance between the rows is such that fracture along a zigzag line cannot occur. This may be done by applying the rule for oblique fracture, established above, according to which the net sectional area of metal of a plate along a zigzag line through the first and second rows of rivets shall be at least equal to that along the first or outer row.

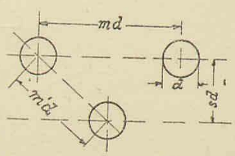


FIG. 71.

Let us examine what the minimum permissible distance between the rows must be in the simple case of two rows of rivets of the same spacing, $p = md$ arranged symmetrically zigzag as in fig. 71.

Let the distance between the rows be sd and the pitch along the zigzag line $m'd$. Then we must have:

$$m'd - ad \geqq \frac{md}{2} - \frac{ad}{2}$$

but

$$m'^2 d^2 = s^2 d^2 + \frac{m^2 d^2}{4} .$$

whence

$$s \geqq \sqrt{\frac{a^2 + 2ma}{4}} \quad . \quad . \quad . \quad (99)$$

For an average value of $a = 1{\cdot}33$, and for $m = 4$, we find $s \geqq 1\frac{3}{4}$, as prescribed in the United States Navy; for $m = 5$ we find s equal to about 2.

Professor Kennedy * gives the rule, chiefly, it appears, for boiler work, that the net area of plate along a zigzag line of fracture ought to be not less than about one-third in excess of the net area straight across the joint. This gives:

$$m'd \geqq \tfrac{2}{3} md + \tfrac{1}{3} da$$

and

$$s \geqq \tfrac{1}{6} \sqrt{7m^2 + 16ma + 4a^2}$$

which is somewhat greater than ordinarily used in shipbuilding. For instance, with $m = 4$ and $a = 1{\cdot}33$ this formula gives: $s \geqq 2{\cdot}4$.

31. DIAMOND JOINTS.

Although it is possible in diamond joints to obtain a very high efficiency, they are rarely, as explained above, used in their pure form in ship construction. The principle, however, finds application in bulkhead liners and other compensating plates, as well as in certain butt connections.

* *Br. Inst. Mech. Eng.*, 1885, p. 217.

We will deal first with overlapped joints.

1. **Mathematical Treatment.**—Two plates A and B are connected by an overlapped joint. The breadth of the plates is b and their thickness t_A and t_B. Let $t_B > t_A$. The rows of the rivets are numbered from 1 to n from A towards B. The number of rivets in any row p is denoted by N_p and the number of rivets between and including any two rows, such as p and r where $r > p$ is denoted by $\sum_p^r N$. The values of α for the two plates are distinguished by the respective suffixes. The total number of rivets in the joint is: $N_t = \sum_1^n N$.

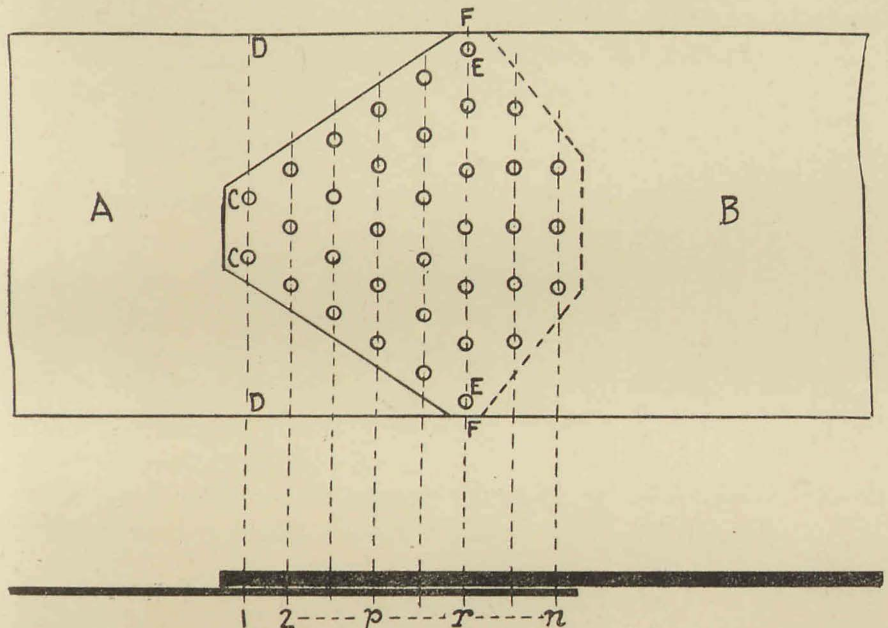


FIG. 72.

The attainable strength is determined by the number of rivets in row 1, and reaches its maximum when this row contains only one rivet. We shall, however, here as in previous cases take shearing of all the rivets as the standard mode of fracture of minimum resistance. Generally the efficiency e to be attained is given, and we have :—

$$e \leqq \frac{f_S \frac{\pi d^2}{4} \delta^2 N_t}{f_T b t_A} = \frac{kd^2}{bt_A} N_t$$

$$\therefore \quad N_t \geqq \frac{bt_A e}{kd^2} \quad . \quad . \quad . \quad . \quad . \quad (100)$$

N_t may be made equal to the lowest integer satisfying this inequality.

Apart from shearing of all the rivets, fracture may occur in one of three ways:

(1) Tearing of plate A along any row p together with shearing of the rivets between and inclusive row 1 to $p-1$.

(2) Tearing of plate B along any row p with shearing of the rivets from row $p+1$ to row n both inclusive.

(3) Tearing of both plates, along the same or different rows, eventually with shearing of intervening rivets.

Comparing fracture by modes (1) and (2) with shearing of all the rivets, we obtain :—

$$[b - N_p da_A] f_T t_A + f_S \frac{\pi d^2}{4} \delta^2 \sum_1^{p-1} N \geqq f_S \frac{\pi d^2}{4} \delta^2 \sum_1^n N$$

$$[b - N_p da_B] f_T t_B + f_S \frac{\pi d^2}{4} \delta^2 \sum_{p+1}^{n} N \geqq f_S \frac{\pi d^2}{4} \delta^2 \sum_1^n N$$

from which :

$$b - N_p da_A \geqq \frac{kd^2}{t_A} \sum_p^n N$$

$$b - N_p da_B \geqq \frac{kd^2}{t_B} \sum_1^p N$$

$$\therefore \quad N_p \leqq \frac{b}{da_A} - \frac{kd}{a_A t_A} \sum_p^n N \quad . \quad . \quad . \quad . \quad (101)$$

$$N_p \leqq \frac{b}{da_B} - \frac{kd}{a_B t_B} \sum_1^p N \quad . \quad . \quad . \quad . \quad (102)$$

When $p = 1$ equation (101) gives its minimum value for N_p

$$N_1 \leqq \frac{b}{da_A} - \frac{kd}{a_A t_A} N_t \quad . \quad . \quad . \quad . \quad (103)$$

When p increases, N will according to (101) increase steadily up to the point where $p = n$.

Conversely, equation (102) gives the maximum value of N when $p = 1$, decreasing as p increases, until it reaches a minimum when $p = n$:

$$N_n \leqq \frac{b}{da_B} - \frac{kd}{a_B t_B} N_t \quad . \quad . \quad . \quad . \quad (104)$$

Since, in any case, we must adopt the smaller of the two values for N, determined by (101) and (102), the two equations mutually limit each other. For smaller values of p we are obliged to follow (101), but a point will be reached where both equations give the same value of N and from thence on, as p increases, N will be controlled by (102).

We may give to (101) a more convenient form:

$$N_p \leqq \frac{b}{da_A} - \frac{kd}{a_A t_A}\sum_p^n N = \frac{b}{da_A} - \frac{kd}{a_A t_A}[N_t - \sum_1^{p-1} N]$$

$$\therefore \quad N_p \leqq N_1 + \frac{kd}{a_A t_A}\sum_1^{p-1} N \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (105)$$

which enables us to determine N for successive rows from row 1 up to the point where the value for N given by (102) is exceeded. This point may be most readily determined tentatively by inserting successive values of N_p in (102).

After that equation (102) is used, which, since $\sum_1^p N = N_p + \sum_1^{p-1} N$, may be given the form:

$$N_p \leqq \frac{bt_B - kd^2 \sum_1^{p-1} N}{a_B dt_B + kd^2} \quad . \quad . \quad . \quad . \quad (106)$$

Generally there will be a greater number of rivets on the side of the light plate than on the side of the heavy plate. Moreover, the number of rivets in the last row N_n will be, ordinarily, greater than N_1 the number of rivets in the first row.

Having adopted a certain minimum pitch md the number of rivets in a row must not exceed the value:

$$N_{max.} \leqq \frac{b}{md} \quad . \quad . \quad . \quad . \quad (107)$$

reckoning that the distance from the edges of the end rivets is about $\frac{md}{2}$.

If the values of N_p found by (105) and (106) for the middle rows are greater than $N_{max.}$ one or more extra rows with $N_{max.}$ rivets are introduced until (106) steps in and cuts down the number. Hereafter, the indications of (106) are followed until the total number of rivets, N_t is obtained.

It may occur that N_n as found from (104) is greater than necessary for obtaining the total number of rivets, in which case row n may be simply given the remaining number of rivets. When in such a case only one or a few rivets are left for the last row, it may often be advisable, in order to obtain a more uniform distribution of the rivets, to omit some of the rivets in the inner rows and place them in the last row. This arrangement is, in particular, desirable where the butt edge of plate A has to be calked, but, at the same time, fracture along the boundary must be guarded against. The total number of rivets required by (100) will always be obtained, since N_n as determined by (102) is positive when $\sum_1^p N$ reaches the value $\frac{bt_A e}{kd^2}$ required for N_t according to (100).

It remains to consider fracture by mode 3, where tearing of both plates occurs at the same or at different rows, but it will be found that such fracture cannot occur as long as the conditions given by (105) and (106) are fulfilled, and as long as the spacing of the rivets in the row or rows where N is a maximum is not less than about $2\frac{1}{2}$ diameters.

Since the number of rivets in the rows is reduced towards the end of the plates below the values allowed by (105) and (106), the breadth of the plates may here be reduced, but must always fulfil the condition:

$$b_p \geqq N_p md \quad . \quad . \quad . \quad . \quad . \quad (108)$$

Moreover, the breadth must not fall below the limit determined by equations (101) and (102). In these equations N_p, $\sum_p^n N$ and $\sum_1^p N$ are now given quantities, while the breadth, which we shall here refer to respectively as b_{Ap} and b_{Bp} may be considered as unknown. Solving for these unknown quantities, we obtain:

$$b_{Ap} \geqq N_p da_A + \frac{kd^2}{t_A}\sum_p^n N \quad . \quad . \quad . \quad (109)$$

$$b_{Bp} \geqq N_p da_B + \frac{kd^2}{t_B}\sum_1^p N . \quad . \quad . \quad . \quad (110)$$

It will be observed that (109) applies to that end of the joint where N_p was determined by (102), while (110) applies to that end where (101) was used to determine N_p. Having calculated the limiting values of N and b for the different rows, we proceed to determine the detail arrangement of the rivets, but on account of the complex and somewhat arbitrary nature of this process, it is best to solve the problem tentatively, essentially by graphical means.

2. **Graphical Design.**—First the lines for the rows are laid off with the prescribed spacing between the rows, $2\frac{1}{2}$ diameters for chain riveting and about $1\frac{3}{4}$ diameters for zigzag riveting.

The breadth of the plates is marked off along these lines, choosing the greater of the two limiting values obtained from (108) and from (109), (110). The boundary determined by these points will usually be irregular and broken or curved and will diverge rapidly from the center-line of the joint as we pass from the ends towards the middle. Usually it is preferred to design the contour with straight sloping lines, but in any case the arrangement of the rivets along the boundaries should satisfy two conditions, first, that the spacing of the rivets along the boundaries must not exceed about 5 diameters if the edges are to be calked, second, that fracture along the boundaries must be precluded. It is necessary to study this latter claim somewhat in detail.

3. **Fracture along the Boundaries.**—Such fracture depends, first, on the angle θ which the bounding lines, and hence also the lines through the bounding rivets, form with the direction of the pull and, second, on the spacing-ratio m' of the rivets along the boundaries. If θ is too large or m' too small, fracture may take place by tearing through the rivets in line 1 and the oblique boundary rivets, following the broken line FECCEF (fig. 72), instead of fracturing transversely through line 1 across the intact plate, following the line DCCD.

We have already seen, SECTION 29, that for a certain efficiency e of a plate subject to oblique rupture the spacing-ratio along the line of rupture is given by (98)

$$m' = \frac{\alpha}{1 - e\sin\theta}$$

If then we make $e = 1$ in this equation we obtain the spacing-ratio along EC which will give to the line of rupture FEC the same resistance as the line CD across the intact plate. In other words, if we make

$$m' \geqq \frac{\alpha}{1 - \sin\theta} \quad . \quad . \quad . \quad . \quad (111)$$

there will be no tendency to rupture along the oblique boundary lines.

With $m' = 5$ as a proper spacing-ratio for calking, and $\alpha = 1{\cdot}33$ as a maximum value of this coefficient, we find θ equal to 47°.

Since the spacing of the rows is given by

$$a = m'd\cos\theta$$

we find $a = 3{\cdot}4d$ when $\theta = 47°$.

It is, therefore, recommended, for watertight work and where the taper of the plates begins at the edges—*i.e.* where there are no rivets between E and F (fig. 72)—to *make* θ *equal to about* 45° *to* 50°, *the spacing of the rivets along the boundaries* 4½ *to* 5 *diameters, and the spacing of the rows* 3 *to* 3½ *diameters.* If the spacing $m'd$ does not fit in with the length of the oblique line, the taper may start at some distance from the edges as in the following example (fig. 73).

If the edges are not calked, as in very light plating where packing material is used or in cases where watertightness is not required, values of m' greater than five and, hence, values of θ greater than 45° may be accepted. For $\theta = 60$ we have $m' = 10$ and $a = 5d$ and for greater values of θ we get a still wider spacing of the rows. When $\theta \geqq 60^\circ$ it will, therefore, be advantageous to insert intermediate rows, in which, however, the boundary rivets must be omitted and care must be exerted to prevent a zigzag fracture along the boundary.

The value of θ being decided upon, the designer has considerable freedom in the remaining part of the adjustment. As long as the various limiting conditions, discussed above, are observed, there will be no necessity for adhering strictly to the regular arrangement of the rivets, assumed in the formulas. The rows need not be absolutely straight. The rivets may be arranged chain fashion, zigzag, or staggered as found expedient. The spacing of the rows and of the rivets in the rows need not be uniform, and may be increased beyond the minimum if desired. The number of rivets in any given row may be reduced below the value determined by the formula, provided an equivalent addition is made in other rows. Eventually an extra row may be inserted. The bounding rivets are usually placed the prescribed distance, $1\frac{5}{8}$ diameters, from the edge, but in non-watertight work this distance may, if desired, be somewhat increased. If the plates are of equal thickness, the arrangement of the rivets will be symmetrical and the calculation need only be performed for one side of the joint. In such a case it will generally be necessary to omit one or more rivets in the middle rows to avoid redundant strength.

4. **Example. Design of an Overlapped Diamond Joint.**— Given: $t_A = \frac{1}{2}''$, $t_B = \frac{5}{8}''$, $b = 24''$, $d = \frac{7}{8}''$, $m \geqq 3$

Required efficiency $e = 95$. The edge of plate B to be calked. From Pl. VI., $a_B = 1{\cdot}31$, $a_A = \delta = 1{\cdot}07$, $k = {\cdot}72$

$$\text{From (100), } N_t \geqq \frac{24 \times {\cdot}50 \times {\cdot}95 \times 64}{{\cdot}72 \times 49} = 20{\cdot}7\text{; make } N_t = 21$$

$$\text{From (103), } N_1 \leqq \frac{24 \times 8}{7 \times 1{\cdot}07} - \frac{{\cdot}72 \times 7}{1{\cdot}07 \times {\cdot}50 \times 8} \times 21 = {\cdot}90\text{; make } N_1 = 1$$

N_1 cannot be smaller than unity, and the required efficiency will be

$$\text{secured since } 1 - \frac{da_A}{b} = 1 - \frac{7 \times 1{\cdot}07}{8 \times 24} = {\cdot}96$$

From (105), $N_p \leqq 1+\frac{\cdot 72 \times 7}{1 \cdot 07 \times \cdot 50 \times 8}\sum_1^{p-1} N = 1 + 1 \cdot 18 \sum_1^{p-1} N$

For N_2, $\sum_1^{p-1} N = 1 \quad \therefore \quad N_2 = 2 \cdot 2$; make $N_2 = 2$

For N_3, $\sum_1^{p-1} N = 3 \quad \therefore \quad N_3 = 4 \cdot 5$; make $N_3 = 4$

For N_4, $\sum_1^{p-1} N = 7 \quad \therefore \quad N_4 = 9 \cdot 3$

From (107), $\qquad N_{max.} \leqq \frac{24 \times 8}{3 \times 7} = 9 \cdot 1$

Equation (106), which controls the number of rivets in the right-hand part of the joint, gives

$$N_p \leqq \frac{24 \times (\frac{5}{8}) - \cdot 72 (\frac{7}{8})^2 \sum_1^{p-1} N}{1 \cdot 31 \times (\frac{7}{8})(\frac{5}{8}) + \cdot 72 \times (\frac{7}{8})^2} = 11 \cdot 8 - \cdot 435 \sum_1^{p-1} N$$

For N_4, $\sum_1^{p-1} N = 7 \quad \therefore \quad N_4 = 8 \cdot 8$, hence make $N_4 = 8$, which satisfies at once (105), (106), and (107).

For N_5, $\sum_1^{p-1} N = 15 \quad \therefore \quad N_5 = 5 \cdot 3$, make $N_5 = 5$ provisionally

For N_6, $\sum_1^{p-1} N = 20 \quad \therefore \quad N_6 = 3 \cdot 1$

Since there are only 6 rivets needed in rows 5 and 6 to make up the total number of 21, we make $N_5 = 4$ and $N_6 = 2$. The maximum value of N for the last row is found from (104)

$$N_n \leqq \frac{24 \times 8}{7 \times 1 \cdot 31} - \frac{\cdot 72 \times 7}{1 \cdot 31 \times 5} \times 21 = 4 \cdot 7$$

This condition is complied with since $N_6 = 2$.

The breadth of the plates at any row is limited by the minimum spacing of the rivets $3d$ and formula (108)

$$b_p \geqq 3 \times \tfrac{7}{8} \times N_p$$

The breadth of plate A is limited by

$$b_{Ap} \geqq \frac{7 \times 1 \cdot 07}{8} N_p + \frac{\cdot 72 \times 49}{\cdot 50 \times 64} \sum_p^n N \quad . \quad . \quad . \quad (109)$$

and the breadth of plate B by

$$b_{Bp} \geqq \frac{7 \times 1 \cdot 31}{8} N_p + \frac{\cdot 72 \times 49}{5 \times 8} \sum_1^p N \quad . \quad . \quad . \quad (110)$$

from which we find

Values of b in ins.

Row No.	Number of Rivets.	Formulas.		
		(108).	(109).	(110).
B 1	1	2·6	...	2·1
B 2	2	5·3	...	4·9
B 3	4	10·5	...	10·8
B 4	8	21·0	...	22·4
A 5	4	10·5	10 3	...
A 6	2	5·3	4·1	...

The result of plotting the greatest values of b for each of the rows is shown in light dotted lines in fig. 73. For row 4, b is equal to the full width of the plate, 24 in. The spacing of the rows is $2\frac{1}{2}d$.

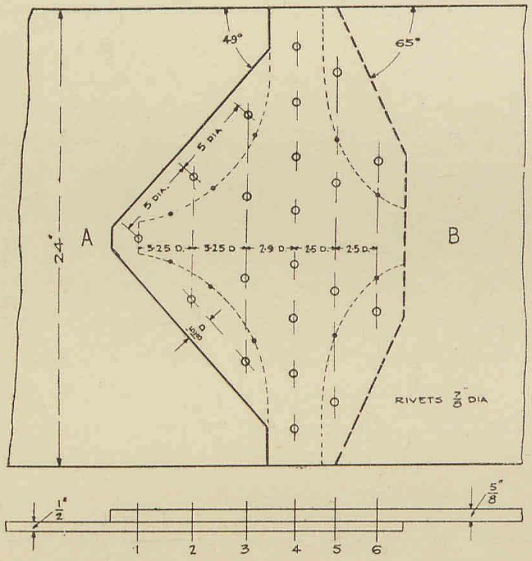


Fig. 73.

Since, by this solution, rupture along the boundary will occur before rupture across line 1, and since the curved contour is undesirable, another solution is worked out by a tentative process, the result of which is shown in fig. 73 in heavy lines. The bounding lines are here straight. The oblique lines on the A-side form an angle of 49° with the center-line, this great angle being admissible because the value of a_A is so small. The spacing along the oblique boundary is $5d$, which is the minimum if fracture is to be prevented along this boundary and which is at the same time satisfactory for calking. The spacing of the rows is $3\frac{1}{4}d$. On the B-side the bounding lines taper at 65° with the center-line and the rows are $2\frac{1}{2}d$ apart.

5. **Strapped Diamond Joints.**—The strapped joints are treated as a combination of two overlapped joints. If the plates and the strap are all of the same thickness, the rivets will be arranged in two symmetrical groups, each of them tapered the same amount both ways, and placed one on each side of the butt. If the strap is heavier than the plates, the arrangement of the rivets will be of the type shown in fig. 74.

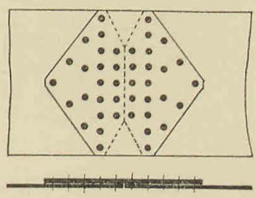


FIG. 74.

In ship construction, where the required efficiency is moderate, an arrangement as in fig. 75 may be employed with single or double strap. This joint gives a fair strength with a small number of rivets, and may be used in non-watertight work. Zigzag rupture must be guarded against by a proper spacing of the rows.

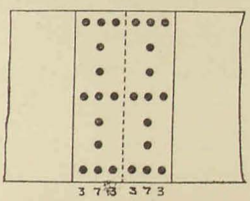


FIG. 75.

Diamond joints with double straps are dealt with theoretically as those with single straps, but $2t'$ must be substituted for t' in the formulas, and the shearing action of the rivets must be multiplied by 1·75, *i.e.* $1{\cdot}75k$ is to be used instead of k.

32. BULKHEAD LINERS AND DOUBLING PLATES.

Wherever the strength of the shell plating and the decks falls below the standard, reinforcements should be fitted.

Lines of weakness are found where the rivets are closely spaced, as for watertight or oiltight work. Local points of weakness exist at doors and hatches, port-holes, deck lights, side lights, sea valves, etc.

1. **Bulkhead Liners.**—We shall open the discussion with a study of the reinforcements, fitted on the shell plating, and often also on the inner bottom plating, to compensate for the loss in strength along the close-spaced lines of rivets at watertight and oiltight bulkheads and frames. In the raised and sunken system these reinforcements take the form of enlarged liners, generally referred to as "bulkhead liners," fitted between the bounding angles and the raised strakes. In the flush system they consist of continuous parallel strips which should be placed on the outside of the plating where possible, as for instance behind armor backing.

Consider the raised and sunken system. The bulkhead liners, being only fitted under the raised strakes, must be so designed as to bring the total longitudinal strength at the weakened section up to that at a non-watertight frame. Suppose that the strakes are of the same width b between the edges of adjacent strakes. Disregard the seam laps which are of equal strength at all the frames. Let

e = Required efficiency of a raised reinforced strake along the watertight or oiltight frame.

e' = Efficiency of a sunken strake along the watertight or oiltight frame.

e_0 = Standard efficiency along the non-watertight frames.

It is then required that

$$e = 2e_0 - e' \quad . \quad . \quad . \quad . \quad (112)$$

Suppose, for instance, that the standard efficiency is ·83, and the efficiency of the sunken strakes along a watertight frame is ·71, then the required efficiency of the raised reinforced strake is $e = 2 \times \cdot 83 - \cdot 71 = \cdot 95$.

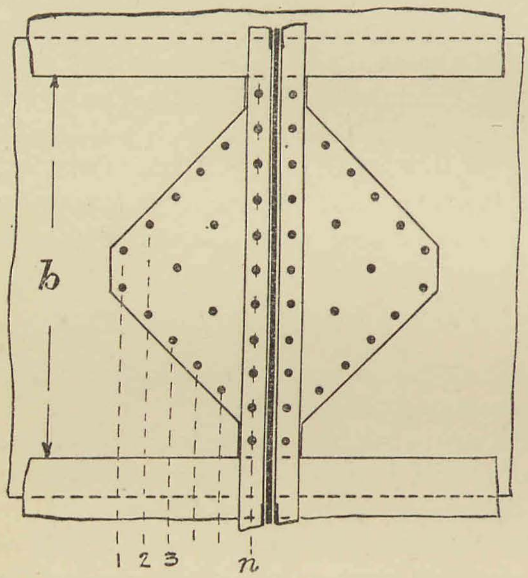


FIG. 76.

Bulkhead liners are generally lozenge- or diamond-shaped and the riveting is arranged essentially on the same principles as in diamond joints, the chief difference being, that since there is in this case no joint or butt in the plating, shearing of all the rivets cannot take place. We shall, therefore, make tearing of the plate along the outer row of rivets the local standard to which other modes of fracture are compared. Referring to fig. 76, the strength along line 1 must not fall below the required efficiency e

$$\frac{b - N_1 da}{b} \geqq e$$

or

$$N_1 \leqq \frac{1 - e}{da} b \quad . \quad . \quad . \quad . \quad (113)$$

We need only consider one side of the liner, since a line of symmetry exists either along the rivets of the bounding angle or—in cases where these are double—along the bulkhead plating. The riveting of the bounding angle being given by the tables, N_n is known. The lines between 1 and n must fulfil the condition that tearing of the plate along any line p together with shearing of all the rivets from line 1 to $p-1$ both inclusive, shall offer greater resistance than tearing of the plate along line 1. Hence

$$(b - N_p da)f_T t + f_S \frac{\pi d^2}{4} \delta^2 \sum_1^{p-1} N \geq (b - N_1 da)f_T t$$

from which we obtain :

$$N_p \leq N_1 + \frac{kd}{at} \sum_1^{p-1} N \quad . \quad . \quad . \quad . \quad (114)$$

which is the same as (105). The number of rivets on one side of the central row or rows may be found by making $p = n$ in this equation

$$\sum_1^{n-1} N \geq \frac{at}{kd}[N_n - N_1] \quad . \quad . \quad . \quad . \quad (115)$$

The total number of rivets in the liner will thus be $2\sum_1^{n-1} N$ plus N_n or plus $2N_n$ according as the bounding angle is single or double.

It remains to consider the possibility of tearing of the liner, but this can only occur together with simultaneous tearing of the plate. Such combined fracture may take place in three different ways, according as fracture of the liner occurs inside or outside the line of fracture of the plate or along the same line. In either case it is prevented if the resistance to combined fracture is greater than that to fracture of the plate alone.

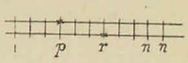


FIG. 77.

Let t stand for tearing of the plate or the liner along a certain row, and let s stand for shearing of the rivets. Top suffixes of t refer to the liner (L) or the plate (P). Bottom suffixes of t give the number of the row. The three conditions may then be expressed as follows :

$$t_p^L + t_r^P + \sum_{p+1}^{r-1} s \geq t_r^P + \sum_1^{r-1} s$$

$$t_r^L + t_p^P + \sum_{p+1}^{r-1} s \geq t_p^P + \sum_1^{p-1} s$$

$$t_p^L + t_p^P \geq t_p^P + \sum_1^{p-1} s$$

The first of these equations may be written :

$$t_p^L \geq \sum_1^{p} s \quad . \quad . \quad . \quad . \quad . \quad (116)$$

If this holds good, $t_r^{L} \geqq \sum_1^r s$ and hence the second of the equations may be written :

$$\sum_1^r s + \sum_{p+1}^{r-1} s \geqq \sum_1^{p-1} s$$

or

$$\sum_p^r s + \sum_{p+1}^{r-1} s \geqq 0$$

and this condition is always fulfilled. The third of the above equations may be written :

$$t_p^{L} \geqq \sum_1^{p-1} s$$

which is fulfilled if (116) is satisfied. In other words, all three conditions are fulfilled if (116) is satisfied. This equation may be written :

$$[b_p' - N_p d a'] f_T t' \geqq f_S \frac{\pi d^2}{4} \delta^2 \sum_1^p N \quad . \quad . \quad . \quad (117)$$

and is seen to involve two unknown quantities, b' the breadth of the liner, and t' its thickness ; but since $b' = b$ on line n we may find t' by making $p = n$ and $b'_p = b$.* The thickness of the liner so determined will always be smaller than the thickness of the plating and we may, therefore, in accordance with usual practice make $t' = t$ or, to be exact, equal to the thickness of adjacent sunken strakes. There will then, ordinarily, be no danger of fracture of the liner. Substituting the value of t' in (117), we may determine b'_p the breadth of the strap at the different rows. We now proceed as explained for the diamond joint. First the breadth of the tapering part is checked and adjusted, corresponding to the minimum spacing of the rivets. Next the angle of the oblique bounding line is determined so as to preclude rupture along the bounding rivets while at the same time a spacing along the boundary suitable for calking is secured. The edge is drawn the prescribed distance $1\frac{3}{8}d$ from the bounding rivets. Where the taper does not span over the whole breadth of the plate, the formula (111) does not apply. It is simplest, in such case, to examine the possibility

* Since in this case both p and r are equal to n, we may, if we so desire, use the third of the fundamental equations $t_n^{L} \geqq \sum_1^{n-1} s$ or

$$[b - N_n d a'] f_T t' \geqq f_S \frac{\pi d^2}{4} \delta^2 \sum_1^{n-1} N \quad . \quad . \quad . \quad . \quad . \quad . \quad (117')$$

which gives the allowable minimum value of t'.

of oblique fracture by measuring on a large-scale diagram the sum of the net distances between the rivet holes all along the boundary and to compare it with the net distance across the plate at line 1. The diameter of the rivets as drawn on the diagram must be αd.

2. **Example. Design of a Bulkhead Liner** (fig. 76).—Given: $t = \frac{5}{8}$ in., $b = 48$ in., $d = \frac{7}{8}$ in. Spacing-ratio to be nowhere less than three; in the bulkhead angles it is to be five, and along the boundaries of the liner not greater than five. Required efficiency, $e = \cdot 95$.

The rivets are countersunk in the plate and $\alpha = 1 \cdot 31$, but since they are not countersunk in the liner we may reckon $\alpha' = \delta = 1 \cdot 07$; $k = \cdot 72$.

From (113)

$$N_1 = \frac{8(1 - \cdot 95)}{7 \times 1 \cdot 31} \times 48 = 2 \cdot 1 ; \text{ make } N_1 = 2$$

$$N_n \geqq \frac{b}{5d} = \frac{48}{4 \cdot 38} = 10 \cdot 96 ; \text{ make } N_n = 11$$

This leaves $2\frac{1}{8}$ in. in row n between the center of the end rivets and the edge of the adjacent plates.

From (115)

$$\sum_1^{n-1} N \geqq \frac{1 \cdot 31 \times 5}{\cdot 72 \times 7} \times 9 = 11 \cdot 7 ; \text{ make } \sum_1^{n-1} N = 12$$

From (114)

$$N_p \leqq 2 + \frac{\cdot 72 \times 7}{1 \cdot 31 \times 5} \sum_1^{p-1} N = 2 + \cdot 77 \sum_1^{p-1} N$$

whence for

$$p = 2, \quad \sum_1^{p-1} N = 2, \quad N_2 \leqq 3 \cdot 5 ; \quad \text{make } N_2 = 3$$

$$p = 3, \quad \sum_1^{p-1} N = 5, \quad N_3 \leqq 5 \cdot 9 ; \quad \text{make } N_3 = 3$$

$$p = 4, \quad \sum_1^{p-1} N = 8, \quad N_4 \leqq 8 \cdot 2 ; \quad \text{make } N_4 = 4$$

This small value of N_4 is chosen so as to make $\sum_1^{4} N = \sum_1^{n-1} N = 12$.

For $p = 5, \quad \sum_1^{p-1} N = 12, \quad N_5 \leqq 11 \cdot 2 ; \quad \text{make } N_5 = N_n = 11$

The minimum thickness of the liner is found from (117) by making $b'_p = b = 48$ in. and $p = n$

$$t' = \frac{\cdot 72 \times 23 \times \left(\frac{7}{8}\right)^2}{48 - 11 \times 1\cdot 07 \times \left(\frac{7}{8}\right)} = \cdot 34 \text{ in.}$$

In practice $t' = t = \frac{5}{8}$ in., corresponding to which the breadth of the liner in the different rows is found from (117)

$$b'_p \geqq \frac{\cdot 72 \times 49}{5 \times 8} \sum_1^p N + \frac{7 \times 1\cdot 07}{8} N_p = \cdot 88 \sum_1^p N + \cdot 94 N_p$$

The breadth b'_p is also determined corresponding to the minimum spacing $3d = 2\cdot 63$ in., and the larger of the values for each row is chosen. In this way a very compact arrangement of the rivets and a very small liner may be obtained, but rupture along the boundaries would not be prevented since the rivets would be here too close together. In order to preclude such rupture and at the same time secure a spacing suitable for calking of the edges, the boundaries must be drawn at an angle $\theta = 45^\circ$ and the bounding rivets must be spaced about five diameters apart, as recommended in SECTION 31, *2*. This solution, with a consequent rearrangement of the rivets, is shown in fig. 76. An extra rivet is added on each side in order to make the strap work better with the plate.

Bulkhead liners of much smaller width than that here designed are frequently used, but they do not possess so high an efficiency, and rupture along the boundaries is not always prevented. Even if a liner of small width is given a great length, it will not be so efficient as a wide liner, because the lines of stress that pass through the liner, being strongly diverted from their natural course, will crowd together at certain places and a very uneven distribution of the stresses will come to exist.

3. **Doubling Plates.**—When a strength member such as, for instance, a strake in the outer shell or in the strength deck, is weakened by an opening being cut in it, the lines of stress will be deflected somewhat in the same way—although not exactly in the same way—as the stream lines of a current flowing round an obstacle. This deflection is of practical importance only within a certain distance from the opening, and it is within this region, at the places where the stress lines are crowded together, that the plating should be reinforced. Fig. 78 gives a crude picture of the lines of stress round a rectangular hatch in a deck subject to a uniform pull.

The distribution of the stresses is very uneven. The plating along the sides parallel with the pull and especially at the corners is subject to great tensile stress, while at the sides normal to the pull the plating is subject to compression. Evidently, tearing at the corners is the most likely form of fracture, but also buckling at the middle of the end edges must be guarded against.

The reinforcement should, therefore, in way of large openings, such as doors and hatches, consist of a doubling plate on each side, parallel with the pull, and should extend beyond the end edges for a distance not

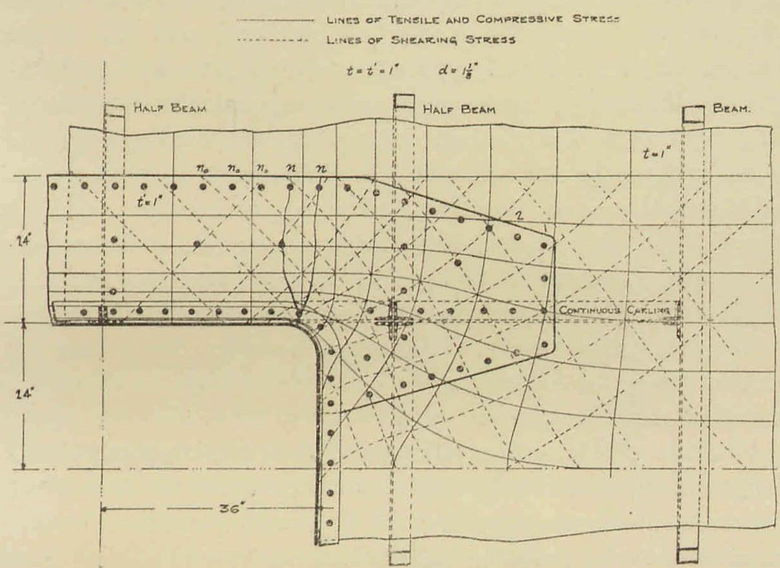


FIG. 78.—Doubling Plate in Way of Deck Hatch.

less than about two-thirds the width of the opening. The doubling plates should enclose the corners and extend along the end edges for not less than about one-quarter the width of the opening. Usually the edges of the openings are reinforced and stiffened by a coaming plate or by beams, frames, carlings, or stiffeners, but if such reinforcements are not found, a plate band or other means of stiffening should be fitted all around the contour. The corners should be well rounded, as otherwise an excessive crowding of the stress lines will here take place. Smaller openings may be entirely encompassed by the doubling plates. Sometimes it may be necessary to place the doubling entirely on one side of the opening, but this is not in general desirable as it involves an additional deflection of the stress lines.

The minimum requirement to the width and thickness of the doubling plates is evidently that the strength of any section across the opening, after deduction of rivet holes, shall be at least that of the normal standard. Ordinarily, this requirement is supposed to be fulfilled by simply replacing the material removed by the opening, but when we deal with important strength members, especially in high-speed vessels, the reinforcement so obtained may not always be adequate. Let us suppose, for instance, that the removed material is replaced by fitting a doubling plate of the same thickness as the original plate and of such a width that the total net sectional area is equal to that along a non-watertight beam or frame. Suppose also that the doubling extends to a proper distance beyond the opening at both ends and that it encloses the corners. As seen from fig. 78, the stresses, even if fairly distributed over the two thicknesses, must necessarily be greater at the corners than normally existing in the plating. Should the elastic limit be passed at any point and cracks started, there is a danger that such cracks may gradually spread. It is, therefore, advisable, at least in vessels of the high-speed type, to use doubling plates heavier than the plate to be reinforced or else to provide additional local reinforcements at the corners of the opening. The latter course is generally to be preferred, because it involves less extra weight. Before we consider the nature of such local reinforcement it is necessary to study the ways in which fracture at the corners can take place :

(1) Fracture of the ordinary (lower or inner) plating combined with shearing of all the rivets in the doubling plate outside the line of fracture. Such fracture can always be prevented by simply providing an adequate rivet area outside the probable line of fracture.

(2) Fracture of both plates. The presence of a coaming plate with its angle of attachment cannot offer any appreciable resistance to this mode of fracture. It is clear that the reinforcement which is to fulfil this purpose must not only cover the region of maximum stress but must extend well beyond this region and must be efficiently connected to both plates. One form of reinforcement consists in patches of plating enveloping the corners. Another, structurally more perfect form, is obtained by making the carlings or stiffeners on the sides of the opening continuous for at least one frame space beyond the ends of the opening. This mode of strengthening is shown on fig. 78, where the carling is so well connected to both plates that fracture at the corner cannot take place without tearing of the carling.

As in case of bulkhead liners, fracture across the outer row of rivets is taken as the local standard. In fact, the procedure of design for the doubling plate outside the opening is precisely as for the bulkhead liners.

First the rivets are arranged around the contour with a spacing suitable for watertight work and after that they are placed on the beams, the carlings, and the coaming angle. If the number of rivets so obtained, outside the probable line of fracture at the corners, is smaller than that required to prevent fracture of the plate, additional rivets are distributed as tack rivets where most needed. No tack rivets should be placed in the region near the corners of the hatch. Abreast of the opening, tack rivets are placed at suitable intervals, from 12 to 16 diameters apart.

Since doubling plates may take very different shapes and since the arrangement of the rivets must depend on local conditions and on the judgment of the designer, we shall not here attempt to give standard formulas, but merely illustrate the mode of procedure by an example.

4. **Example. Design of Doubling Plates Fitted in Way of a Deck Hatch** (fig. 78).—Required efficiency $e_0 = \cdot 84$. Rivets along all boundaries to be spaced for watertight work. Hatch opening: 48 in. × 72 in. $t = 1$ in.; $d = \frac{9}{8}$ in. The rivets are cylindrical in the deck plating, but countersunk in the doubling plates; hence $\alpha = \delta = 1\cdot 08$, $\alpha' = 1\cdot 32$, $k = \cdot 73$. Denote breadth of hatch by $h = 48$ in. Make the breadth of each of the doubling plates equal to: $\frac{h}{2} = 24$ in. and find the required thickness: t'.

We need deal only with one side of the hatch. Abreast of the hatch, at the rows n_0, there are nowhere more than three rivets. In row n, at the corners, there are two or at most three rivets, according as we take one or the other line of fracture, shown on the figure, as the most probable. Fracture of both thicknesses, starting at the corner or at any point on the long sides and passing through three rivets, must not occur before tearing of the plate outside the doubling, whence

$$\left(\frac{h}{2} - 3\delta d\right)t + \left(\frac{h}{2} - 3\alpha' d\right)t' \geqq \cdot 84ht$$

$$\therefore \quad t' \geqq \frac{\cdot 34h + 3\delta d}{\left(\frac{h}{2}\right) - 3\alpha' d}t = 1\cdot 02 \text{ in.}$$

Since the doubling plate receives considerable assistance from the coaming structure in this case, we are justified in making $t' = t = 1$ in. For the outer row of rivets we have

$$N_1 \leqq \frac{\cdot 16h}{\delta d} = 6\cdot 3$$

Comparing fracture of the deck plate along row n, *i.e.* from the corner outwards through three rivet holes, combined with shearing of all rivets outside row n with fracture along line 1 we obtain an equation for $\sum_1^{n-1} N$.

If we make $N_1 = 4$ as in fig. 78 we have:

$$\sum_1^{n-1} N \geqq \frac{8}{9 \times \cdot 73}\left[1 \cdot 08[3-4]+\frac{24 \times 8}{9}\right] = 24 \cdot 6$$

We have made $\sum_1^{n-1} N = 30$ including in this number two of the rivets in the coaming angle.

With $N_1 = 4$ we are allowed to make $N_2 = 7$ and the following numbers still greater and we might thus secure the required minimum of twenty-five rivets on a small number of rows, but, as explained above, it is desirable to extend the doubling plate well beyond the opening. Hence the arrangement shown in fig. 78 is recommended with practically seven rows of rivets outside that in the coaming angle. The carling is extended continuously for 5 feet beyond the hatch, and takes here six rivets through both thicknesses at each end.

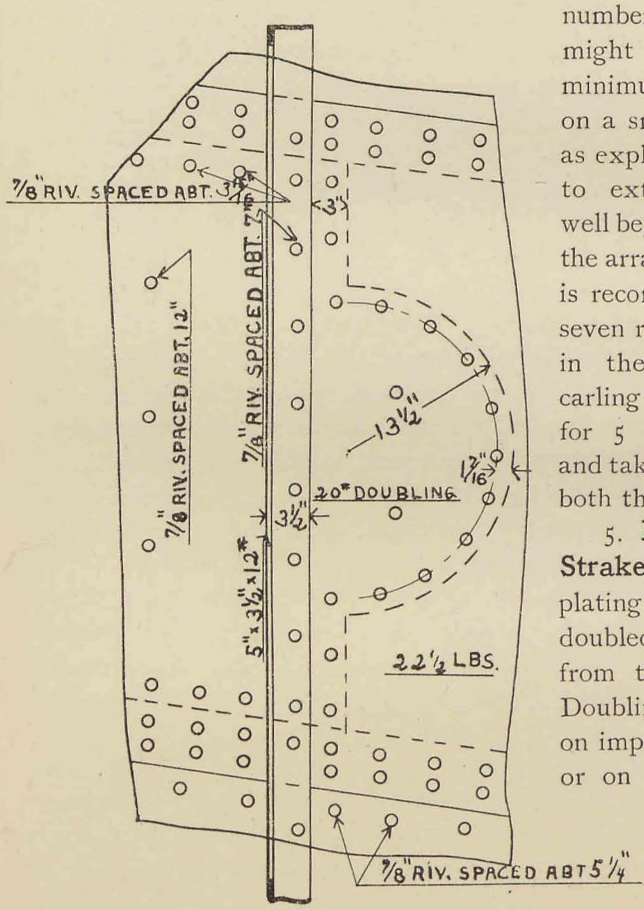


FIG. 79.—End of Doubling Plate.

5. **Ending of Doubling Strakes.**—In large ships the shell plating at the ends is generally doubled for a certain distance from the stem and sternpost. Doubling is frequently used also on important strakes on the sides or on the strength deck for a certain length amidships. The ends of doubling strakes, when fitted on the outside, must have close-spaced riveting in order to permit calking, and if they are straight constitute lines of weakness similar to those along the watertight frames or bulk-

heads. In order to avoid this weakening, the ends of the doubling strakes are shaped like a bulkhead liner of a polygonal or semicircular contour around which the rivets are arranged. A few tack rivets are added.

In this way a line of increased resistance to rupture is formed, the strength of which may be estimated approximately by measuring the net sectional area of metal between the rivet holes following all along the contour. In the accompanying sketch the net area is nearly the same as that across the intact plate ; in other words, the efficiency is almost one. We must also examine cross lines of fracture as, for instance, through the extreme three or five rivets, in order to ascertain that the strength does nowhere fall below the standard.

This rough but simple mode of design may of course also be used in case of bulkhead liners, but since these occur in such great numbers it is preferable to design them with greater care, as explained above.

33. RELATION BETWEEN THEORY AND PRACTICE.

1. **Spacing of the Rivets.**—If we compare the spacing of rivets given in Table XI. with that determined by the formulas for maximum strength, it is found, as pointed out already, in general to differ widely. Correspondence between these spacings exists, in fact, for each kind of joint only for one particular combination of rivet diameter and plate thickness. For smaller thicknesses the standard spacing falls below, for greater thicknesses it rises above the theoretical spacing. On closer examination, however, it is found that this discrepancy is more apparent than real. In general, an efficiency of two-thirds to three-quarters is desired, and, as seen from Table XIII., this efficiency may be attained for light plating with single riveting, but the heavier the plating is, the more rows of rivets are required. Now, the spacings used in practice are chosen so that within the range of each particular joint (single, double, treble, or quadruple riveted) they give approximately the maximum strength, and do not, therefore, deviate more than about one-half diameter from the theoretically best spacing. In fact, the correspondence between theory and practice is fairly good except where the claim to watertightness or oiltightness renders it necessary to adopt a closer spacing than determined by the formulas. For instance, when double riveting is used in plating of less than 10 lb., the ideal spacing would be rather wide for efficient calking, wherefore in practice a spacing of 4 diameters is used (see Table XIII.).

A strict adherence to the ideal spacing is neither possible nor desirable. The claim to simplicity requires that the prescribed standard spacing shall be a simple multiple of the diameter, but even the standard spacing can rarely be followed exactly, because it does not, in general, fit the given length of the row. On the whole, the theoretical spacings can only be regarded as a guide, the more so as the actual conditions often deviate from the assumptions on which the calculations are based. Where strength is of paramount importance, it is, however, advisable to approach as closely as practicable to the ideal spacing.

In merchant shipbuilding the rivets are, on the whole, spaced much more closely than is customary in warships. For instance, according to Lloyd's Rules the spacing-ratio for treble riveted butts of the outer shell plating is 3½ as against from 4 to 4½ used in warships. The rivet area in the joints of merchant vessels is, therefore, as already stated, very great. Where steel rivets are used, the strength of the rivets in treble riveted butt laps may be more than 50 per cent. in excess of that of the plates.

2. **Working Stresses.**—There is in shipbuilding no established practice as regards the working stresses allowed on the rivets in the ship-girder; the complexity of the problem and the empirical nature of the strength calculations have so far prevented the adoption of any definite limits. In calculating the strength of beams, longitudinals, and other girders fitted to support forces of known magnitude, it is safe to use a shearing stress on the rivets of 5 ts. per sq. in., but in the substructure of guns and wherever great dynamical forces are liable to act the shearing stress should not exceed 4 ts. per sq in.

In the construction of railway bridges * a shearing stress of 12,000 lb. per sq. in. (5·4 ts. per sq. in.) is allowed for shop-driven rivets and a bearing pressure of 24,000 lb. per sq. in. (10·7 ts. per sq. in.).

* *Am. Railw. Eng. Ass.*

CHAPTER IX.

WATERTIGHTNESS AND OILTIGHTNESS.

34. CALKING.

THE primary function of riveting is to unite the different members of the hull and to secure the requisite strength, but in many parts of a ship's structure watertightness is of equal importance. A certain degree of watertightness is always obtained by riveting. In fact, if the rivets are spaced sufficiently close together and if the workmanship is good, perfect watertightness can be obtained without the use of other, special means and in course of time the watertightness will become still more absolute and reliable by the formation of rust between the faying surfaces, whereby all the interstices will be filled up. Since it is not, however, in all cases, desired to adopt a very close spacing of the rivets, and since the workmanship cannot be relied upon to be perfect everywhere, all watertight work is calked wherever possible.

Calking consists in forcing the metal along the edge of one or both structural members, which are joined together, in between the faying surfaces, so as to prevent the passage of water or oil. Efficient calking demands that the distance between the rivets along the calked edge shall not be too great relative to the thickness of the plate, for otherwise the plate will yield to the calking tool, and the calking cannot be relied upon when the joint is severely strained. Hence, as we have seen, a maximum spacing is given for watertight work, varying from $3\frac{1}{2}$ to 5 diameters. If the spacing is expressed in terms of the diameter of the rivet, as is usually the case, it is rational to make it smaller for light plating than for heavy plating, since the rivets are relatively larger for the lighter plating. This is actually done, as seen from Table XI., where the spacing for double and treble riveted watertight work (heavy plates) is greater than for single riveted watertight work (light plates). See also the discussion of double butt straps.

In overlapped joints the edge to be calked should first be planed. With a "splitting tool" the plate is "nicked" or "set up," as shown in fig. 80, and after that the metal at the edge is driven against the adjacent plate with a "setting-in" or "edge-calking" tool. In this way an almost rectangular groove is formed on the face of the edge, and the metal so displaced forms a shoulder which causes the calked plate to be forced a little away from the other plate. Watertightness is produced and maintained primarily by the elasticity of the calked plate, which holds the shoulder in close contact with the other plate.

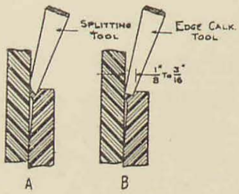


Fig. 80.

In butt calking the edges of both plates are planed, and after being fitted closely together they are nicked on each side of the butt by the splitting tool. After that the two edges are forced together with a flat or hollow tool, a so-called "butt tool," which forms a rectangular groove with flat or convex bottom. The face of the butt tool is often serrated.

In cases where the faying surfaces do not make a good contact, a tool with rounded, blunt edge, a so-called "fuller," is used after the splitting tool and before the butt tool is applied.

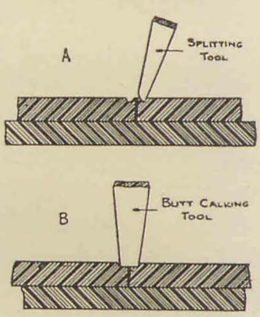


Fig. 81.

Butt calking is not so efficient as lap calking because any straining and, in particular, bending of the joint is more apt to open the calking in a butt than in an overlap. Moreover, unless the fit of the plates is very perfect, contact by butt calking takes place only along a narrow surface between the projecting edges of the metal, and is easily destroyed by corrosion. In such a case the butt must be recalked, but if this process is repeated, it will be increasingly difficult to obtain a satisfactory result, and at last calking becomes impossible. The only remedy then left is to weld (solder) the butt or else to fit an external (double) butt strap.

All calking should be metal to metal. The interposition of packing materials or filling pieces should be avoided as far as possible. Calking should always take place on one side of the watertight surface and not partly on one side, partly on the other, since all lines of calking should be closed loops. In order to facilitate calking, all watertight surfaces are, therefore, on the calking side, as far as possible, kept free from stiffeners, frames, and other fixtures. In some cases both sides of a watertight division are calked. In testing a watertight division, the water pressure is generally applied to the non-calked side, whereby any fault in the calking can be easily detected and made good. If the water pressure is applied to the calked side, it is often difficult to locate a leak, since the water may ooze between the faying surfaces of the plates and angles and may appear at a great distance from the defective point. In order to repair the fault it will be necessary first to empty the compartment. On the other hand, structural weakness, if it exists, is more readily revealed by applying the pressure to the calked side, because then the plating, if insufficiently stiffened, will be deflected away from the bounding angle, the calking will be broken, and a leak will appear. Partitions with bounding angles on one side only are sometimes calked on both sides, in which case the heel of the angle is calked as well as the toes (fig. 82).

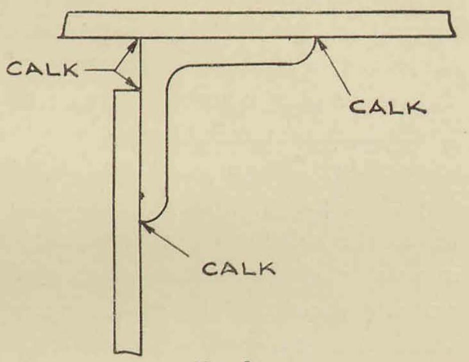


FIG. 82.

Rivets are calked only where watertightness is of particular importance and should in such case be countersunk on the point. Where an otherwise sound rivet shows a small leak, it may be calked whatever the type of point.

Riveting through three thicknesses, so-called "three-ply" riveting, should be avoided as far as possible, because it is difficult to obtain perfectly fair holes and the rivets do not draw the plates so well together as with two thicknesses. Especially where the central thickness forms a watertight division, the water may pass from one side of the division to the other along the shanks of the rivets if these do not fill the holes completely. This difficulty may be met by calking round the respective fixture, as in case of a stanchion riveted through the inner bottom to the reversed bars of floors or longitudinals. Generally, in three-ply riveting, stopwaters are fitted on one side of the central thickness, and rivets are used larger than given by the tables, in order to draw the different parts better together.

Light plating of a thickness smaller than about $\frac{3}{16}$ in. (5 mm.) cannot be efficiently calked. Especially at overlaps the calking tool will tend to lift the calked plate and separate it from the other plate. With strapped butts this difficulty does not exist, wherefore this joint is better suited for calking than overlaps in case of thin plating, but the calking cannot be relied upon where the structure is exposed to great strains.

35. PACKING MATERIALS.

In general, where the plating is very thin, where watertightness is of importance, and especially where the joints are liable to great strains, as, for instance, in the outer shell of torpedo-boats, watertightness is obtained by the use of packing materials, consisting of oily or tarry matter. Packing material is used also between doubling plates, since it is difficult, especially on the curved surface of the outer shell, to obtain a perfect fit between the plates ; it not only prevents the water from entering, it also supports the plates and prevents their working.

Wherever an uncalked edge cuts through a watertight surface, it is necessary to use so-called "stopwaters" at the point of intersection. Stopwaters consist of packing materials applied locally. For instance, between the faying surfaces of the seams in the outside plating water may enter everywhere at the uncalked inboard edge, and if no special precautions are taken, it may flow along the seams from one compartment to another. Stopwaters must, therefore, be interposed in the seams in way of watertight floors and bulkheads. Stopwaters are used, moreover, where the calking edge is inaccessible for some reason as, for instance, in a watertight bulkhead, when stiffeners are placed on the calked side, crossing the seams of the plating. They should here be applied to the seams in way of each of the stiffeners.

Packing material may consist of canvas, burlap, flannel, felt, or paper soaked in red lead, in tar, or in a mixture of red and white lead. In some cases a stiff mixture of red and white lead putty is the most suitable. Where doubling plates fit imperfectly together, it may be necessary to pump red lead in between them. One of the best packing materials is, however, hemp felt sheeting soaked in tar, often called tar-felt. Red lead is apt to get dry in course of time and then takes the form of cakes or powder which is incapable of excluding the water. Tar-felt preserves its obturating qualities for any length of time. Under deck armor, under the flanges of piping, and under many various fittings, grommets are used, made of lamp-wick soaked in red and white lead.

All packing materials, being oily or tarry, have the drawback that they tend to prevent the formation of rust, which is so valuable in maintaining the watertightness after the calking has deteriorated.

36. SPECIAL PRECAUTIONS FOR OILTIGHTNESS.

Oiltightness is more difficult to attain than watertightness, because oil is more searching than water, it prevents the formation of rust, and greasy matter, such as red lead, cannot be used as stopwater, because it is dissolved and washed away by the oil. Hence it is necessary in oiltight work to space the rivets more closely than in watertight work —generally from 3½ to 3 diameters. The rivets must be sound, the holes fair, and the workmanship of the highest class throughout. As far as practicable, the rivets should be driven by power. Ordinarily, pan-head rivets with a full point are employed, but often countersunk rivets are preferred. When rivets with countersunk head are used, exceptionally good workmanship is required; the countersunk head must fit exactly into the countersink in the plate, the heads of the rivets must be well heated, and must be of ample depth.

Rivet holes should either be drilled or punched small and afterwards reamed. The increase in diameter of a hole by reaming should be equal to at least one-eighth of the thickness of the material. In no case should "drifting" of the holes be allowed. All plates, angles, and clips should have perfectly flat and smooth faying surfaces and special care should be exercised to avoid burrs and chips. This is of particular importance in three-ply riveting. The calking should be performed very thoroughly. Heads and points of rivets should be calked where necessary. On oiltight floors and longitudinals the angle bars connecting them to the inner and outer shell should be double and calked on both sides.

Angle clips connecting a non-oiltight member to an oiltight member should be calked all around its edge, as, for instance, the clips connecting ordinary floor-plates to an oiltight longitudinal. Care should be taken to leave all calking edges in way of oiltight work clear and accessible, and clips should be cut short for this purpose if necessary. At least one inch should be left between the calking edge and adjacent clips.

Bosom straps of oiltight angles should take four rivets on each side of the butt.

"Oil-stops" are used on seams, laps, staples, etc., where the plating is 7½ pounds or less and so cannot be efficiently calked; in plating of greater thickness they should be used only where absolutely necessary. Oil-stops are made from lamp-wick, canvas, or felting soaked in a mixture of shellac and white or red lead. Often canvas soaked in a mixture of pine tar and shellac is used.

CHAPTER X.

OUTSIDE PLATING.

37. DIFFERENT SYSTEMS.

1. **Connection of Butts and Edges.**—The plates of the outer shell are arranged in longitudinal "strakes." In the same strake the ends of the plates, the so-called "butts," are placed between the frames, generally parallel with them, and are connected either by straps or overlaps.

Butt straps are ordinarily single and placed on the inside of the plating, but in some cases, where special strength is desired, double butt straps are used. Lapped butts were introduced at the end of the eighties and were first used in tank steamers. The term "lapped butt" is a misnomer, since the term butt implies that the plates do not overlap. Formerly the ends of the plates were always connected by butt straps, while the edges were usually overlapped, and the term butt was rightly applied to the end joints. Gradually the conception of a butt became associated with the end joints in contradistinction to the edge joints, and when, later, the nature of the end joints was changed from butts to overlaps, the term "lapped butt" came into use. The term "lapped end joint" is more rational, but longer.

The strapped butts can be made stronger than the overlapped by omitting some rivets in the last rows, but otherwise the overlapped butts are superior in every respect. They permit a saving in weight and riveting, the workmanship is simpler, since the fitting of an overlap requires less accuracy than that of a butted joint, and, finally, the calking is more durable and is more easily repaired. Lapped butts are indeed well suited, not only for shell plating, but for all ordinary assemblages of plating, and should, therefore, be employed, except where some special reason renders the use of butt straps more desirable. Where an overlapped butt is required to be of greater strength than ordinarily attainable in watertight work, the backstrap, described in SECTION 28, *6*, should be used. The cases where butted joints are preferable to overlapped joints will be discussed in the following. Lapped butts should be so arranged that the outside edge of the overlap faces aft, for if it faces forward, it will be exposed to rapid deterioration due to the action of the water streaming against it.

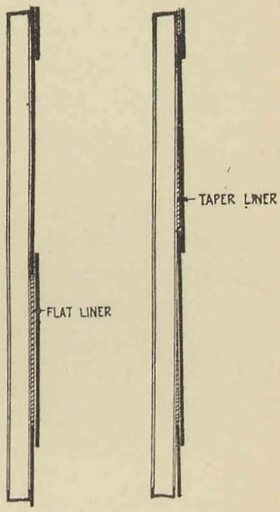


FIG. 83.

Adjacent strakes are connected along their edges by joints which are referred to as "seams" or "landings." These joints may be flush, in which case "edge-strips" are used, but generally they are overlapped. In the former case the plates are worked flush, not only at the seams, but also at the butts. In the latter case they are arranged either on the "raised and sunken system," where only alternate strakes rest directly on the frames, or on the "clinker system," where all the strakes rest with one edge on the frames, the other on an adjacent strake. The clinker system is employed chiefly as a supplement to the raised and sunken system; it is somewhat more expensive, because it necessitates the use of tapered liners unless the plates are joggled. It has the advantage that plates can be removed without disturbing the adjacent strakes.

2. **The Raised and Sunken System.**—The space between the frames and the outer strakes is here filled in with "liners" of the same thickness as the adjacent plates and, excepting the bulkhead liners, they are of the same breadth as the flange of the frames. In order to save weight, the liners may be lightened by punching oval holes in them between the rivets. When butt straps are used with the raised and sunken system, the straps on the inner strakes extend the whole breadth of the plates; while the straps on the outer strakes extend only between the edges of adjacent inner plates. Where a butt of an outer strake falls in a frame space adjacent to a watertight frame or bulkhead, the strap and the bulkhead liner are often merged into one on account of the restricted space. This construction has the drawback that the strap cannot be made thicker than the adjacent plates. Generally, an independent bulkhead liner and a lapped butt will be both lighter and stronger, and the combined strap and liner should, therefore, not be used except where unavoidable. This occurs in ships with a small frame space, as in merchant vessels, and in cases where the butts, for some reason, are not placed in the middle of the frame spaces.

3. **Joggling of Frames and Plates.**—Two methods are used to save the weight of the liners with the raised and sunken system, and may equally well be used in the clinker system.

FIG. 84.—Joggled Frame.

FIG. 85.—Joggled Plating.

The first of these methods is "joggling of the frames" at the edges of the sunken strakes so as to bring them in close contact with the raised strakes. It is employed, for instance, in the *Lusitania* in connection with the clinker system in the lower part of the bottom. It is best suited to light framing, such as the frame bars inside the double bottom. The second method consists in "joggling of the plates," by bending the overlapping edges of the plates in the raised strakes sharply so as to permit the plates to fay closely against the frames.

Both systems combine the advantages of overlap with the absence of liners, but both require the use of special tools and extra handling. The saving in weight by the omission of liners is, in joggling of the plates, more than counterbalanced by the loss in buoyancy. Probably the plates suffer some loss in strength by the process, and the transverse strength and rigidity of the ship is reduced. This, apparently, is the chief reason for not using it in warships for the outer shell, where the wide frame spacing and the small plate thickness do not warrant any further weakening of the shell. On the whole, joggling of the frames seems preferable to joggling of the outside plating. For internal work, where the plates are lighter and where the gain in weight is not destroyed by a loss in buoyancy, joggling of the plates appears advantageous, and is frequently so used in warships. Both methods are employed extensively in merchant ships.

4. **Brackets instead of Bulkhead Liners.**—In merchant ships, brackets, as shown in fig. 86, are commonly used instead of bulkhead liners. The brackets are extra deep and generally attached to the hold and side stringers. They serve at once to reinforce the shell plating, to stiffen the bulkhead, and to preserve the continuity of the stringers. They are particularly suitable where the plating or the frames are joggled

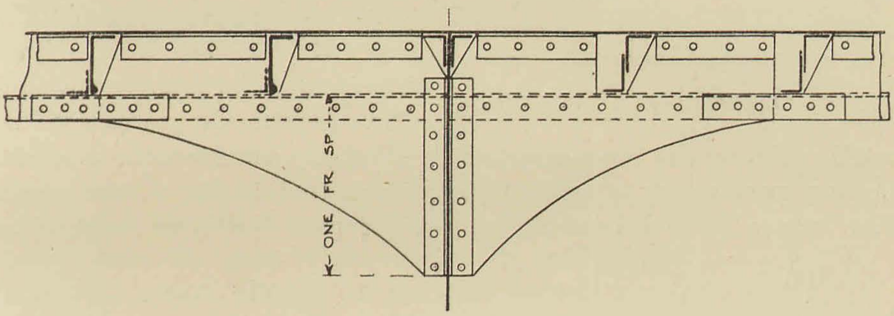


FIG. 86.—Side Stringer Brackets at Watertight Bulkheads.*

and where, therefore, no liners can be fitted. It appears that this feature could be adopted with advantage in destroyers where the shell is not stiffened by a double bottom or by intermediate decks.

5. **Doubling Plates.**—In many ships one or more strakes are doubled in order to obtain increased longitudinal strength and, frequently, certain areas of the shell are doubled to secure local strength. With the raised and sunken system, doubling is effected by fitting the additional plates between the edges of the inner and outer strakes. The doubling plate of an outer strake is given the same thickness as the adjacent inner strakes. The butts are always worked flush. Where strength is not of importance, the plates of one layer may serve as butt straps for the other, but in strength members straps are fitted to one or even both courses. The straps should be fitted directly on the plates they connect. If, for instance, both courses are strapped, and all the straps are fitted on the inside, as is sometimes the case, the inner plates will be weakened at the butts of the outer plates.

* From *Lloyd's Rules and Regulations*, 1912–13, p. 130.

6. **Chamfer or Tapered Liners.**—With the raised and sunken system, as also with the clinker system, a problem arises in cases where overlapped butts are used together with overlapped seams, viz. how to obtain watertightness where the butts meet the seams. At such points there are three thicknesses of plating overlapping each other. The middle thickness B is inserted with its corner between an inner plate A and an outer plate C. Unless special precautions are taken, calking is impossible at this place, since a wedge-shaped vacant space will be left between plates A and C, as at *ab* (fig. 89, XX). Two methods are used to meet this difficulty. Either the corner of the middle thickness is chamfered, which is usually performed in a special slotting machine, or a tapered liner is inserted in the vacant space. The former method involves less weight than the latter, and when

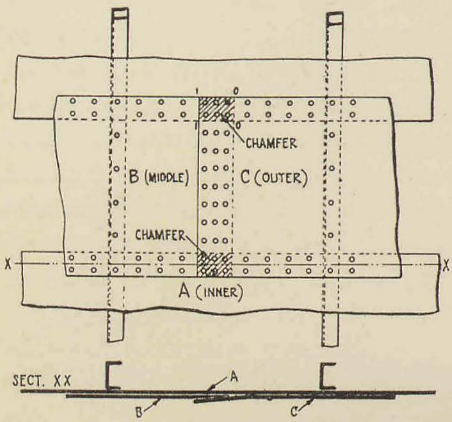


FIG. 87.—Chamfered Butt Lap in Outer Strake.

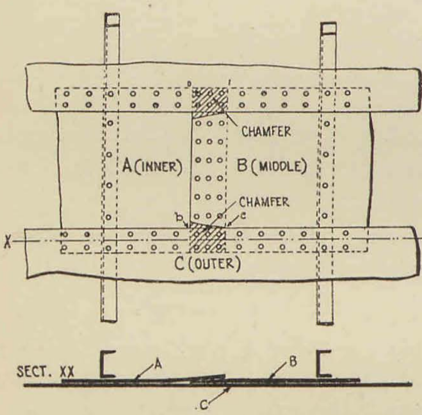


FIG. 88.—Chamfered Butt Lap in Inner Strake.

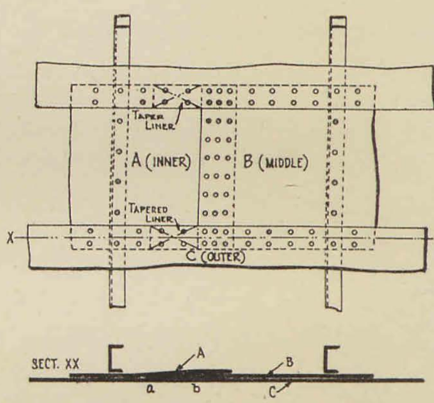


FIG. 89.—Butt Lap in Inner Strake with Tapered Liner.

applied to outer strakes it is simple and permits easy and satisfactory calking of the seam, as evident from fig. 87, XX. When, on the other hand, chamfering is used on plates of the inner strakes, it is desirable to chamfer in two different planes as shown in fig. 88, but then the plate B will be very thin at point *b* and is liable to break under the calking tool, causing unsatisfactory calking; moreover, there is apt to be undue corrosion at such a point. If the chamfer is only carried out in one plane, as in the outer strakes, it will be difficult to calk along the edge *bc* in a satisfactory manner, unless the workmanship is so accurate that a perfect fit is obtained between the two plates along the edge. For these reasons chamfering of the corners is in the United States Navy used only in the outer strakes, while tapered liners are considered preferable for the inner strakes. Fig. 89 shows the application of tapered liners. Section XX gives an edge view, from which it is seen that a clear and accessible calking edge is obtained. Where the frame space is small, the presence of tapered liners may render it necessary to place the butt laps out of the middle of the frame space. In many merchant vessels chamfer is used both on the inner and the outer strakes.

7. **The Flush System.**—This system is, generally, employed only above the water-line, either in order to produce a smooth appearance or in order to facilitate fitting of the armor. In the French Navy the flush system has formerly been used extensively also below the water-line, partly, it appears, on the theory that the resistance to propulsion is smaller by this system than by others. Experiments, however, seem to show that the gain in this respect is insignificant. Another advantage, claimed for the flush system, is that it lends greater rigidity to the shell than the raised and sunken system.

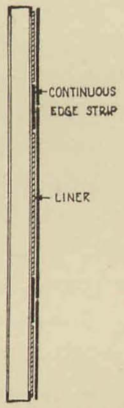


FIG. 90.—French System of Flush Plating.

When the flush system is used for appearance, both edge strips and butt straps are worked on the inside. The strips are usually continuous for the sake of longitudinal strength and because it is desired to cover the longitudinal seams as completely as possible, but in such a case liners must be fitted on the frames. The strips should be as long as procurable, in particular on the sheer strake. The liners may be lightened by holes or washer-liners, *i.e.* circular disks or washers, may be fitted round the rivets. In this way much weight is saved, but the plates are not properly supported by the frames, and looseness of the rivets is apt to occur from the waves pounding on the sides. It is better to joggle the frames round the edge strips and thus avoid the use of liners altogether.

On flush plating behind armor backing, the edge strips may without inconvenience be worked on the outside, whereby, again, the use of liners is avoided. Sometimes also the butt straps are then fitted on the outside. The straps are always fitted between the strips.

8. **Comparison of the Raised and Sunken System with the Flush System.**—By the raised and sunken system with lapped butts only the calking edges, *i.e.* the edges of the outer strakes and the aft butt edges of all the plates, need to be planed; all other edges may be sheared. No great accuracy is required in dimensions and form of the plates. By the flush system both edges and butts must be planed and great nicety of workmanship is demanded in preparing the plates so as to obtain the perfect fit necessary for efficient calking and, at the same time, avoid compressive strains. The flush system requires nearly twice as much riveting as the raised and sunken system.

With the raised and sunken system the weight of overlaps at edges and butts plus the weight of liners is in large warships about 17½ per cent. of the net shell weight, and where the butts are strapped it is about 20 per cent., including the straps, but with the flush system the weight of edge strips, butt straps, and liners reaches about 30 per cent. of the shell weight. It is clear, also, that the flush system must be much more costly than the raised and sunken system. The only advantages that can be claimed for the flush system are the more finished appearance, a greater stiffness of the shell, and a certain reduction in the shearing stresses on the rivets. Appearance should not, however, be given any weight in warships when in the way of technical advantages. The requisite stiffness can be attained more simply and cheaply by internal girders and frames, and a greater resistance to shearing of the rivets, where this may be locally required, can be attained more cheaply by the addition of a line of rivets. The raised and sunken system is, therefore, superior to the flush system from almost every point of view, and is in fact the one most generally adopted.

38. GENERAL ARRANGEMENT OF THE STRAKES.

1. **Breadth of the Plates.**—The work of planning the general lay-out of the strakes should begin with a study of the location of the edges on the midship section, but this, at once, leads to a discussion of the breadth of the plates.

Great breadth is advantageous, inasmuch as the number of seams, the amount of riveting, and hence the weight and cost are reduced, when the breadth is increased. On the other hand, apart from limitations in manufacture and in facilities of handling and riveting (by hydraulic power), an increase in breadth of the plates in a given ship will cause increased difficulties in shaping and fitting the plates on the strongly curved parts of the hull, and broader plates give wider butts, which are less favourable to longitudinal strength. With a given form of hull, these objections are not of so much weight in a large ship as in a small one. In fact, as far as they are concerned, *the breadth of the plates in similar ships may be, roughly, proportional to the linear dimensions of the ships.*

It has been claimed for narrow strakes, that they give increased stiffness to the shell on account of the great number of seams, but this mode of obtaining stiffness is irrational and expensive. Much greater stiffness, involving less riveting and weight, can be obtained by a greater development of the longitudinal frames.

Based on these considerations and on experience, a certain practice has been established by which the breadth of the strakes is settled within rather narrow limits, being, in fact, as great as the aforesaid restrictions permit. At the present time the ordinary or "standard breadth" of the plates amidships is in large battleships from 55 to 65 inches, in light cruisers from 50 to 55 inches, and in large destroyers from 40 to 45 inches. In merchant ships the standard breadth is limited by the Classification Societies to 72 inches as a maximum in very large vessels, and 54 inches in small vessels. If these breadths are exceeded, the riveting of the butts must be specially strengthened.

Sometimes the raised strakes have been made narrower than the sunken strakes by twice the overlap, so as to make the edges of the raised strakes, the so-called "sight-edges," equidistant for the sake of appearance. This, however, seems irrational, since the strength of the shell will then be unevenly distributed and the raised strakes cannot be given the full maximum breadth.

2. **The Flat Keel.**—The keel strake is here called the "flat keel" in contradistinction to the vertical keel. In merchant vessels it is often called the "flat-plate keel" to distinguish it from the vertical bar-keel carried in early iron ships and still used in many smaller vessels. The keel strake is always of extra strength and in all but the smallest vessels it is doubled. The functions of the flat keel, besides forming an integral part of the shell, are to act as the lower flange of the keel-girder, and to resist and distribute the great local forces to which it may become subject when the ship is in dock or when it is aground.

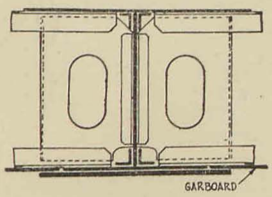


FIG. 91.

The keel strake is, usually, of less than standard breadth, since there is no necessity for extending this heavy strake much beyond the width of the keel-blocks. When the flat keel is doubled, the lower strake should be wider than the upper, since thereby the work of construction, erection, and repair is facilitated. The adjacent strake, the "garboard strake," will then be an inner strake, as on fig. 91.

3. **The First Strake below the Armor Shelf and the Sheer Strake.**—The first strake below the armor shelf forms the boundary of the bottom plating and is generally worked as an outer strake, flush with the side armor, to which it is often connected by a cover plate. The frame bar may then, without being joggled, run up into the bosom of the continuous angle under the armor shelf, and the harpin, which is used during construction to hold the upper end of the frames in position, may be applied in the most favorable way, that is, at the top of the frames and in way of an outer strake.

The strake at or immediately below the strength deck, the "sheer strake," is, like the strake below the armor shelf, worked as an outer strake and for the same reasons. It is frequently of more than standard breadth and should extend above the strength deck, where it should be connected by a substantial angle to the deck stringer.

4. **Other Strakes.**—With the exceptions here mentioned, the strakes on the midship section are made of standard breadth as far as local conditions permit, and are arranged raised and sunken or flush in accordance with the system adopted. Sometimes it is necessary to work one of the strakes clinker fashion. In determining the exact location of the seams on the midship section, the position of the longitudinals and bilge keels must be studied at the same time, so as to ensure that they are all clear of each other.

5. **The Seams in the Body Plan and on the Half-Block Model.**—The location of the plate edges being determined amidships, their further extension may be obtained preliminarily on the body plan of the line drawing. In the upper part of the ship, above the protective deck, the edges are sheered to the decks. In the lower body they are drawn about normal to the frame stations, somewhat curved near the midship section, but otherwise nearly straight and converging towards the stem and stern as viewed on the body plan.

The further study of the arrangement of the seams is best carried out on a half-block model, where the edges can be obtained in their true shape. The model is usually prepared to one-quarter inch scale, but for small vessels one-half inch scale is used. First, the frames, the bulkheads, the decks, the armor shelf, the load water-line, and other lines, which are already given and which are not subject to change, are marked off on the model. After that the plate edges are transferred to the model from the body plan, and are here determined more accurately and faired. This process should be accompanied by a study of the arrangement of longitudinals, bilge-keels, docking-keels, and the location of propeller-struts, bosses, sea-valves, air-ports, and other appendages or fittings on the outer shell, all of which are laid off on the model. Attention to be given to the accessibility of the calking edges.

6. **Intersection of the Seams with Longitudinals, Longitudinal Bulkheads, and Decks.**—Such intersections will take place under very small angles and will, therefore, be of great extension. They necessitate riveting through three thicknesses, and are liable to cause conflicts in the distribution of the rivets. They are, therefore, undesirable, in particular where the intersecting members are watertight. Longitudinals may, generally, be so placed as to clear the seams, but crossing of the decks and the seams is often unavoidable. Where intersection takes place, the seam may be jogged so as to cut the member in question at right angles. Fig. 92, A, shows this solution in its general form, involving a jog in the edge of both of the adjoining plates. By making a butt coincide with the jog as on fig. 92, B, it is only necessary to cut a jog in one of the plates. The best solution is obtained by the arrangement shown in fig. 92, C, where a so-called double stealer is introduced, a feature that will be explained later.

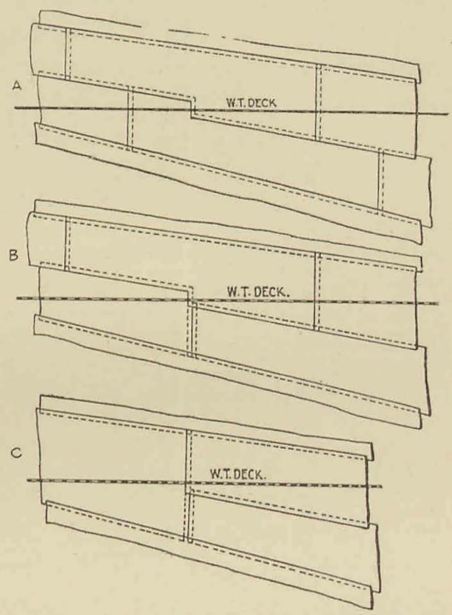


FIG. 92.—Intersection of Seam with Watertight Deck. Starboard side, aft, looking inboard.

7. **General Trend of the Seams.**—In a vessel of the modern battleship type with rather flat bottom and flat sides along the middle portion and a sharp turn of the bilges amidships, the general trend of the seams under the bottom, and especially near the center-line, should be the same as that of the lines *a*, *a*, *a*, fig. 93, A, parallel with the center-line plane and horizontal amidships, but bending upwards at the ends of the ship. Above the turn of the bilges, and especially on the upper part of the sides, above the armor shelf, the seams should lie in nearly horizontal planes, sheered to the deck edges, and should be of the same general character as the lines *b*, *b*, *b*, . . . in fig. 93, B. Towards

the ends, these two systems of lines blend and merge into each other at the turn of the bilges, as in fig. 93, C.

In ships of fine form with a good rise of floor, the general trend of the seams should be of the same character, but the two systems will merge earlier, *i.e.* further from the ends, and since the turn of the bilges is here more gentle, the seams in this region should be placed in nearly diagonal planes.

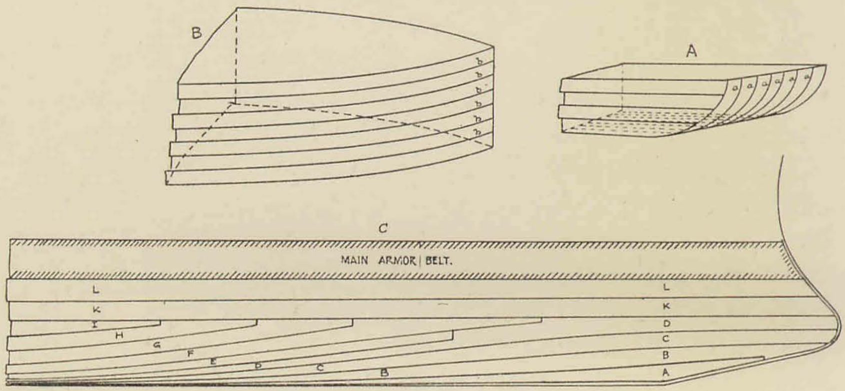


FIG. 93.—Distribution of Plate Edges.

8. **Developable Plates and Straight Edges.**—The general trend of the seams being determined, several points have to be considered in settling their exact form and location. When plates are far from being developable, they have to be heated and forged into shape, involving much extra cost. When the edges are strongly curved, much material is wasted. As far as possible, the plates should, therefore, be developable and the edges should be nearly straight in the developed condition. These claims are satisfied on the simple surfaces represented in fig. 93, A and B, but cannot be wholly fulfilled in a ship on account of the undevelopable nature of a great part of the bottom.

A surface may be developable and still have strongly curved edges in the developed state, such as, for instance, the surface of a truncated cone, while a non-developable surface may have straight edges, like, for instance, a screw surface bounded by two generators. Hence the arrangement of seams which gives most nearly straight edges is not, everywhere, that which gives the most developable plates. The designer must often, and perhaps generally, adopt a solution intermediate between the two.

The best way to arrive at such a compromise is to use a rather flat, broad spline as a guide in drawing the seams on the model. If a spline is constrained to follow the surface of the model without being bent sideways, it determines a developable strake with straight edges ; if it is bent sideways, it determines a developable strake with slightly curved edges. In studying the curvature of individual plates and their edges, it is often advantageous to use strips of stiff paper, which, when pressed against the surface of the model, will indicate to what extent the plate is developable and, if cut to conform to the edges drawn on the model and straightened out, will show the curvature of the edges.

9. **Stealers.**—Another point that influences the design of the seams is the reduction in girth forward and aft, which renders it necessary to taper most of the strakes in breadth towards the ends. In order, now, to avoid making the strakes too narrow, some of them must be dropped before they reach the stem or the stern. Such strakes are called "lost strakes," "drop strakes," or "stealers." They, naturally, occur in the region where the seams from the bottom meet the seams from the sides. In some ships, however, the bottom strakes are simply made to land on a horizontal strake, that runs parallel with and below the armor shelf (fig. 93, C). A similar arrangement is used frequently in merchant ships with a long, parallel, box-shaped middle body, for which form it is particularly adapted (*Lusitania*), but in a warship of fine form it is liable to involve strongly curved edges. Preferably, the seams should be laid out according to the principles stated above, where the two systems, that from the bottom and that from the sides, blend with each other in the region of the turn of the bilges forward and aft. When a strake becomes too narrow, it should be made a stealer. Its place may be occupied by widening one of the adjacent plates, as shown on fig. 94, but by this construction much material is lost in the widened strake and considerable labor is involved. Of recent years a simpler solution has been largely adopted, by which two strakes are stopped at the same place, and a single strake takes their place. The two lost strakes will be here referred to as a "double stealer" (see fig. 95). The only objection to the double stealer seems to be that two butts of adjacent strakes are placed in the same frame space, but by tapering the plates so much that their total breadth at the final butt does not materially exceed the standard breadth amidships, this objection has no weight, since stealers always occur near the ends of the ship, where the longitudinal strength is of reduced importance.

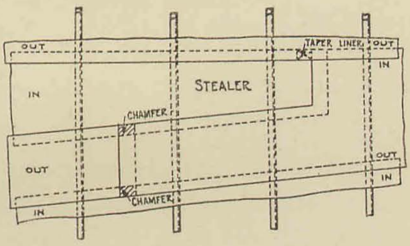


FIG. 94.—Ordinary Stealer. Starboard side, forward, looking inboard.

With the raised and sunken system, when a stealer is replaced by widening the adjacent strake, this latter becomes raised along one edge and sunken along the other. If the butts are of the overlapped type, the widened strake, which is always the middle thickness, is given a chamfer or a tapered liner at the corner, where the butt meets the seam.

In the British Navy the widened strake is chamfered along the entire length of the butt of the stealer and a reinforcing strap is fitted on the inside to make up for the reduction of strength along the overlap.*

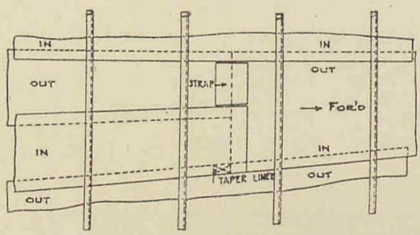


FIG. 95A.—Double Stealer. Port side, forward, looking outboard.

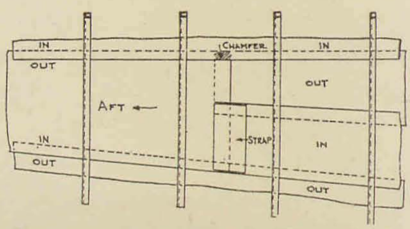


FIG. 95B.—Double Stealer. Port side, aft, looking outboard.

When a double stealer is used in the raised and sunken system, the butt of one of the strakes is provided with a strap while the other is overlapped. In order to make the outer edge of the overlapped butt face aft, the overlap should in the fore part of the ship be applied to the sunken stealer as in fig. 95A, in the aft part to the raised stealer as in fig. 95B. In the British Navy chamfering and wedges at the corners are avoided; the plate which replaces the two stealers is worked flush along one of its edges and is there connected by an edge strip.

10. **Boss Plates.**—At the bosses, which enclose the stern tubes, the seams should not be placed in the concave parts at C C, fig. 96, where the plates are liable to work a great deal, but rather along the middle of the convex part at A or, still better, entirely clear of the boss, as at B B. The latter arrangement is suitable for small ships, where the boss may be made of one plate. The boss plate should preferably be in an outer strake.

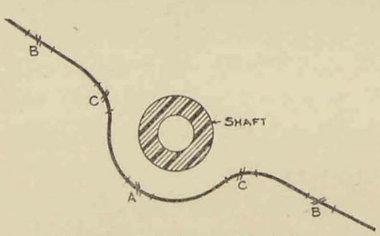


FIG. 96.—Plating at Shaft Boss.

11. **Fairing of the Seams.**—When the seams are arranged on the model, they are transferred to the body plan on the "loft floor," where the fairing should be finally checked in the sheer and half-breadth. This fairing process may also be carried out on a 1-in. or ¾-in line drawing,

* *Shipyard Practice*, McDermaid, 1911, p. 32.

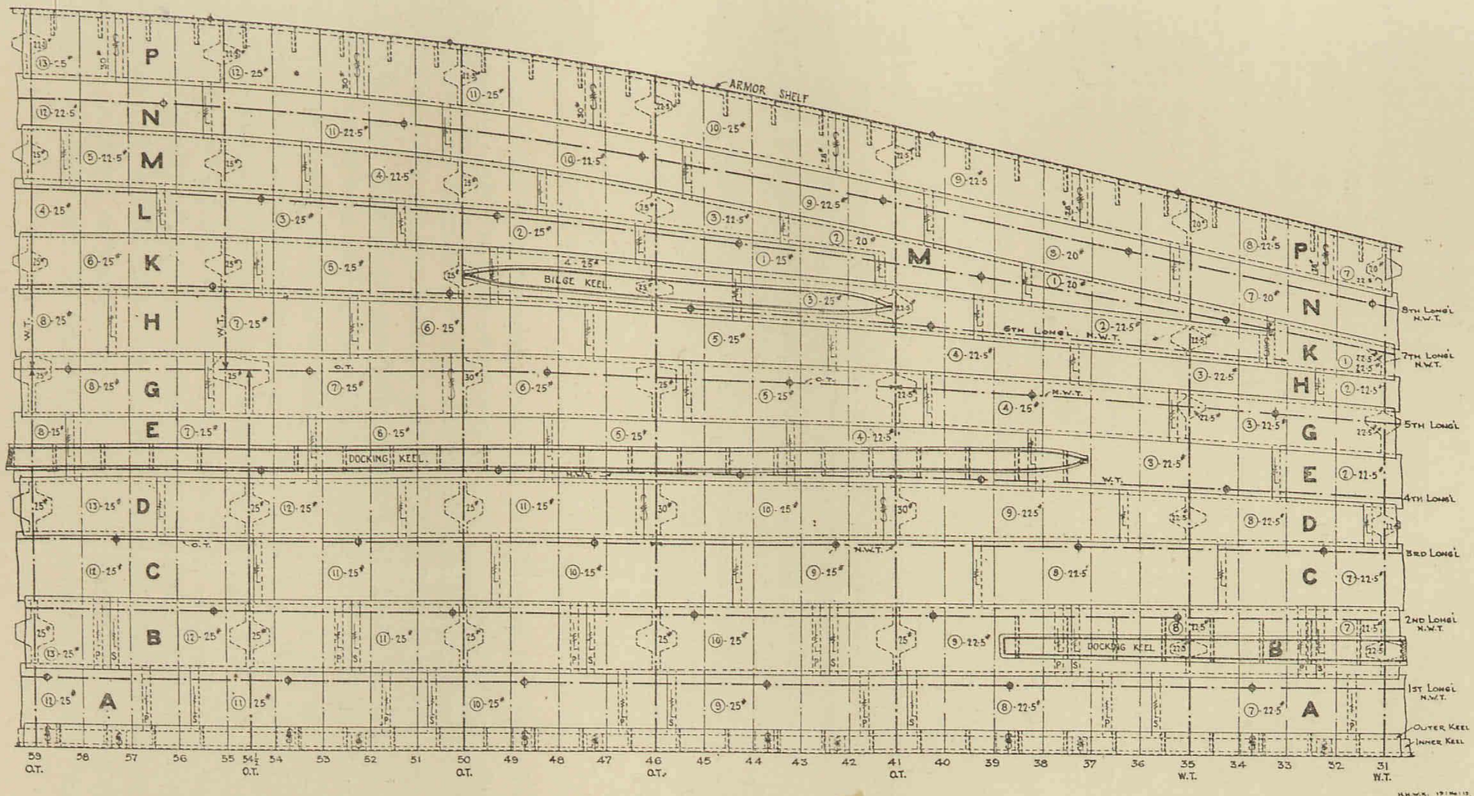


FIG. 97.—Portion of Lower Shell Expansion of a Large Battleship.

often laid off on a marble slab. When the seams are completely faired, they are transferred to the "scrieve board."

For ordering the plates, the breadths are measured from the body plan or the scrieve board, while the lengths are measured from the half-block model.

12. **Shell Expansion.**—For use in the yard an expansion of the shell plating is prepared, ordinarily to ¼-in. scale. This plan is constructed by first marking off the frame stations in their proper longitudinal position on the keel-line as a base. The girth measurements, reckoned from the center-line at the keel to the various edges, are then transferred from the model to the corresponding stations on the expansion plan, and by drawing in the plate edges through the points so obtained, a distorted picture of the shell is produced. On this plan are shown butts, longitudinals, and, in fact, all the lines on the model, besides the dimensions and thickness of each plate. The plates are identified by designating each strake by a letter. Generally the garboard strake is marked A and the following strakes B, C, D, etc. The plates in each strake are numbered 1, 2, 3, etc., beginning usually from the bow. The plates are lettered and numbered in the same manner on the two sides, but are distinguished from each other where necessary by the addition of the letter P (Port) or S (Starboard).

The shell expansion is very useful as a means of guidance for the workmen, showing at a glance the general lay-out of the plating and giving the principal data for each individual plate.

39. ARRANGEMENT OF BUTTS.

1. **Length of the Plates.**—The arrangement of the butts depends primarily on the length of the plates relative to the frame spacing. Great length is of advantage in so far as it entails a reduction in the number of butts, but although the steel-works can roll the plates to practically any length that could be desired, a limit is imposed by the capacity of the tools available in the shipyard for the preparation and transport of the material. In the strongly curved parts of the shell, the length must be kept within certain limits in order to avoid excessive curvature of the plates and of their edges, a limitation which, as explained in case of the breadth of the plates, is felt more in small ships than in large ships. It would appear rational to use longer plates amidships, where the bottom is nearly plane or cylindrical, than at the ends, where the curvature is usually greater and where the plates are tapering in width. Moreover, in point of longitudinal strength, a reduction in the number of butts is

of greater importance amidships than at the ends. Generally, however, a uniform standard length is adopted throughout the bottom, being the largest multiple of the frame space which the tools of the yard can handle conveniently, and this length is departed from only where necessary to meet local requirements. By giving the same length to the plates of the shell and of all structural members adjacent to the shell, a systematic and uniform shift of butts throughout the hull structure is facilitated.

The ordinary length of shell plates is from 20 to 28 feet, which corresponds to from five to seven frame spaces in large ships and from ten to twelve frame spaces in smaller ships, but in large merchant vessels much larger plates are sometimes used. In the *Lusitania*, for instance, the standard length of the plates is 33 ft. or twelve frame spaces.

2. **Shift of Butts.**—The butts generally form points of weakness and should, therefore, be carefully arranged so as to cause a minimum reduction in longitudinal strength. The shell presents a number of necessarily weakened transverse sections along the frames, where the plates are pierced by an unbroken row of rivets running from the keel to the upper deck. Along the non-watertight frames the rivets are, generally, eight diameters apart, and hence the strength of the shell at a frame section is reduced in the ratio 1 to $1-\frac{a}{8}$. Along watertight frames and bulkheads, where the rivets are more closely spaced, the shell is or should be strengthened, either by bulkhead liners or by brackets, so as to bring the strength up to that along the non-watertight frames, which may then be taken as a standard.

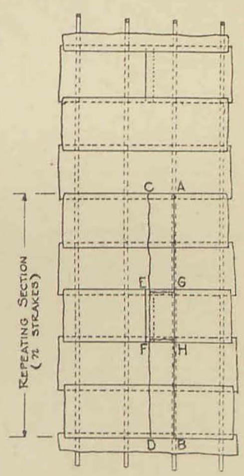


FIG. 98.—Rupture of Plating.

In the discussion of riveted joints we considered only the individual plates connected by the joint under consideration. It is clear, however, that complete rupture of a butt can only take place in conjunction with rupture of adjacent plates and other longitudinal members with which it may be connected, since all contiguous members that are carried continuously past the butt will act as reinforcement of it.

We shall deal first with the shell plating by itself, apart from adjacent members such as longitudinals and decks. We shall, moreover, assume that the longitudinal stresses are fairly uniformly distributed over the region considered, and that the breadth and thickness of the different strakes is about the same. Under these suppositions we may take, as representing the entire shell, a repeating section comprising one butted strake and as many intact strakes as are passing between consecutive butts in the same frame space. Such a repeating section, comprising altogether n strakes, as shown in fig. 98 between the lines CA and DB may rupture in different ways.

3. **Simultaneous Rupture of a Butt and Adjacent Intact Strakes.**—It is required that such a line of rupture, represented by CD (fig. 98), shall offer at least the same resistance as a line along a frame, AB. Take the resistance to tearing across of an intact plate as unity and let the resistance to tearing of a plate along a non-watertight frame be z. Denote the resistance of a butt by x.

We have then :

$$x + n - 1 \geqq zn \qquad \text{or} \qquad n \geqq \frac{1-x}{1-z} \quad . \quad . \quad . \quad (118)$$

Let $z = \cdot 83$ as an ordinary average value, and take $x = \frac{2}{3}$, which is the ordinary efficiency of butt connections in the outside plating, corresponding, for instance, to treble riveted laps in 25-lb. plating. We must then, according to (118), have $n \geqq 2$, whence one passing strake between consecutive butts in the same frame space will be ordinarily sufficient to prevent rupture across the intact strakes. The lowest efficiency likely to be found in butt connections is $x = \frac{1}{2}$ which, for instance, would correspond to double riveted butts in 25-lb. plating. With this value of x we should have $n \geqq 3$ showing that in this case, which may be considered extreme, two passing strakes will be required. Hence, in general, there will be no necessity for increasing the number of passing strakes beyond two, a conclusion which agrees well with the rule adopted in practice, both for warships and merchant ships, that *there shall be not less than two passing strakes between consecutive butts in the same frame space.*

4. **Rupture of a Butt and Shearing of the Rivets in the Seams between the Butt and the Nearest Frame.**—If the lines of rupture GE—EF—FH combined offer less resistance than the line GH then rupture of the repeating section along the broken line AGEFHB will be more likely to occur than along the frame from A to B. Let y represent the resistance to shearing along one of the lines EG and FH, being actually the ratio between this resistance and that to tearing of an intact plate. The requirement to be fulfilled is

$$x + 2y \geqq z$$

or

$$y \geqq \tfrac{1}{2}(z - x) \quad . \quad . \quad . \quad (119)$$

Ordinarily x does not fall below ·66 and hence, with $z = \cdot 83$, we get $y \geqq \cdot 09$. This value of y can easily be secured in most cases with the open frame spacing used in warships provided the butt is midway between the frames, which, for this reason, should preferably be the case. Where the butt is not so placed, where the frame space is small, and where the butts are very weak, it is advisable to examine whether (119) is satisfied. If this condition is fulfilled and if there is at least one frame space between the butts of adjacent strakes, there will obviously be no danger of rupture along a broken line such as ABCDEFG in fig. 99A, drawn stepwise through the butts in different frame spaces. In practice, however, *it is considered desirable to have not less than two frame spaces between the butts of adjacent strakes.*

5. **Example.**—In a battleship with treble riveted butt laps and double riveted seams, the number of passing strakes between consecutive butts in the same frame space is two as a minimum, whence $n = 3$. Spacing of rivets along the frames, eight diameters. Breadth of plates 60 in. There are at least eight rivets to shear in the seam between the butt and the frame. Compare the different modes of rupture of a repeating section.

$$t = \tfrac{5}{8} \text{ in.}, \quad d = \tfrac{7}{8} \text{ in.}, \quad m = 4\tfrac{1}{2} \text{ in butts and seams.}$$

We find: $x = \cdot 67, \quad z = \cdot 84$

Condition (118) is satisfied since:

$$\cdot 67 + 3 - 1 > \cdot 84 \times 3 \qquad \text{or} \qquad 2\cdot 67 > 2\cdot 52$$

Deduct 8 in. from the breadth of the plate to allow for the overlaps at the seams, whence $b = 60 - 8 = 52$ in.

$$y = \frac{8 \times \frac{\pi d^2}{4} \delta^2 f_s}{52 t f_T} = \frac{8kd^2}{52t} = \frac{\cdot 72 \times 49}{52 \times 5} = \cdot 14$$

and condition (119) is satisfied, since:

$$\cdot 14 > \tfrac{1}{2}(\cdot 84 - \cdot 67) = \cdot 085$$

Hence, rupture is most likely to take place along the frame.

6. **Different Shifts of Butts.**—With a length of plating equal to four frame spaces, such as used in the early days of iron shipbuilding, the so-called "brick arrangement" is probably the best. The butts fall on the middle of the plates of adjacent strakes, leaving two frame spaces between adjacent butts as prescribed above, but only one passing strake between consecutive butts in the same frame space.

With a length of plating equal to five or more frame spaces, such as now generally used, it is easy to fulfil the requirements. The accompanying diagrams (fig. 99) show suitable arrangements with plate lengths of 5, 6, and 7 frame spaces.

Fig. 99A represents the so-called "diagonal arrangement," which is the best with a plate length of five frame spaces. It gives four passing strakes between consecutive butts in the same frame space, and has been extensively used. Counting the number of frame spaces in successive steps between the butts of adjacent strakes, we obtain by going to one side the series 2, 2, 2, . . . , by going to the other side the series 3, 3, 3, . . .

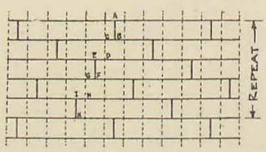


FIG. 99A.—Shift of Butts in Shell Plating. 5 frame spaces. Period 2 2 2 or 3 3 3. (4 passing strakes.)

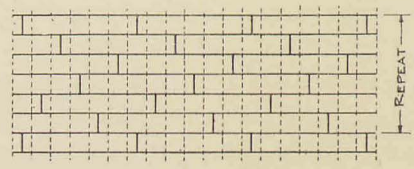


FIG. 99B.—Shift of Butts in Shell Plating. 6 frame spaces. Period 2 3 4 4 3 2 either way. (5 passing strakes.)

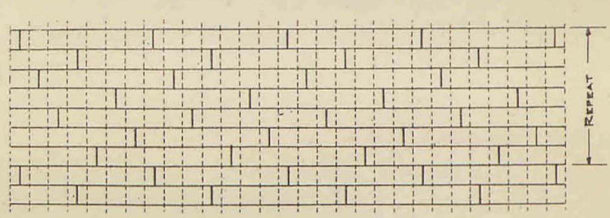


FIG. 99C.—Shift of Butts in Shell Plating. 7 frame spaces. Period 3 5 4 4 4 5 3 or 4 2 3 3 3 2 4. (6 passing strakes.)

With a plate length of six frame spaces, which is quite generally used in large ships, the butts may be arranged as in fig. 99B, where there are everywhere five passing strakes. The shift of butts follows a series of period 2 3 4 4 3 2 whether going to the right or left. This arrangement is used in the British Navy.

With a plate length of seven frame spaces and with a shift of butts as in fig. 99C, we obtain uniformly six passing strakes. The period is 3 5 4 4 4 5 3 going one way and 4 2 3 3 3 2 4 going the other way. With a plate length of eight frame spaces we may use the periods 3 3 3 . . . or 5 5 5 . . ., which give seven passing strakes. With a plate length of ten frame spaces, periods of 3 3 3 . . . and 7 7 7 . . . give a fair distribution with nine passing strakes, while periods of 4 4 4 . . . and 6 6 6 . . . give four passing strakes.

All these combinations give a greater number of passing strakes than required, and, generally, there are also more than two frame spaces between butts in adjacent strakes. It is desirable to adopt as a basis of the design some such arrangement, because thereby a systematic distribution of the butts and hence a certain uniformity of strength is secured over the entire shell. Where local requirements make it desirable or necessary, departures can of course be made from the systematic arrangement.

7. **Gradual or Progressive Rupture.**—We might be led to believe from the foregoing that rupture through the butts may be entirely prevented by simply arranging them as prescribed above. It must, however, be borne in mind that the foregoing treatment is based upon an ideal uniform distribution of material and stresses, which is approached only in certain regions of the shell, as under the flat bottom amidships in large battleships on both sides of the keel strakes. In other parts of the shell the distribution of the stresses is far from uniform. On the sides of the ship the longitudinal stresses vary from a maximum in the sheer strake to zero at the neutral axis and increase again downwards towards the turn of the bilges. Generally, the sheer strake is thicker or of stronger material than the other strakes, and often an abrupt change in strength takes place between the sheer strake and the strake next below. Within a given strake, the stresses at different sections vary inversely as the efficiencies, and wherever a weakness exists, the strains will be increased and additional local stresses will be called into play in the nearest parts of the adjacent strakes. A uniform distribution of the stresses throughout the repeating section, such as assumed above, will hardly ever exist in practice.

Fracture is most likely to start in the sheer strakes, which are more strained than other parts of the shell. Let us consider a butt in a sheer strake and suppose, as very frequently is the case, that the butt connection is weaker than the plate along the frames, but that the shearing strength of the edge rivets between the butt and the frame is ample to prevent fracture of the butt combined with shearing of the edges. There would then, theoretically, if the shell is designed in accordance with the foregoing rules, be no danger of rupture through the butt with simultaneous rupture of adjacent strakes of a repeating section. When, however, the sheer strake is in tension, the butt joint, being weaker than the plate along the frame, and therefore subject to greater stresses, will sooner reach the elastic limit. It will begin to yield, and through the medium of the edge rivets, which are by supposition amply strong, a great strain will be thrown on the edge of the plate next below the sheer strake. In other words, the lines of stress will curve around the strained spot in the sheer strake and pass through the nearest part of the adjacent strake. If, now, this strake is relatively light, it is liable to tear at the edge. Once fracture is produced, it will easily extend further downwards gradually as the butt yields more and more. It is clear that under these circumstances the intact plates of the repeating section will have no opportunity of acting in unison as assumed above. Irrespective of the number of intact passing strakes between the butts, *the plates will tear, one after the other, in the same way as a sheet of paper or cloth is rent by forces acting along one of the edges.* It is probable that the line of rupture, thus started at the butt, will soon strike over to one of the adjacent frames and thence follow the frame rivets.*

Hence, in order to avoid progressive rupture, *the sheer strakes must not present any special points of weakness and the transition in strength between the sheer strakes and the strakes next below must not be too abrupt.*

8. **Shift of Butts of Sheer Strakes, Keel Strakes, and Garboard Strakes.**—It is of particular importance that the butts of the sheer strakes make a good shift with the butts of the deck stringers, which will thus form an effective reinforcement of the sheer strakes and prevent incipient fracture in these from extending to the deck plating. Where the sheer strakes are doubled, the butts in the two courses should not be nearer to each other than two frame spaces. The same rules are followed for the keel strakes. When the arrangement of the butts on the two sides of the ship is symmetrical about the keel, the butts in the garboard strakes will fall in the same frame spaces, but where the flat keel is doubled they are actually separated by two passing strakes. With a plate length of six frame spaces this arrangement must, in fact, be accepted if the rule is to be observed that there shall be at least two frame spaces between butts in adjacent strakes. When the length of the plates is greater than six frame spaces, the butts in the garboard strakes may with advantage be shifted clear of each other on opposite sides of the keel.

Where brackets are fitted to the vertical keel and where intermediate floors are worked midway between the frames, the butts must be placed on the quarter frame spaces, in which case the butts in the garboard strakes should be shifted at least one half frame space clear of each other (fig. 97).

40. THICKNESS OF OUTSIDE PLATING.

1. **Enumeration of Straining and Wasting Actions.**—The outside plating must be capable, simultaneously, of resisting several actions,

* An example of the breaking of a ship by progressive rupture is described in Reed's *Shipbuilding in Iron and Steel*, London, 1869, p. 15.

which in many parts of the shell produce cumulative and compound stresses. These actions will be first considered separately and may be grouped as follows :

(1) The external water pressures, statical and dynamical.

(2) Bending of individual frames, transverse or longitudinal, whereby stresses are induced in those parts of the outside plating which co-operate with them in their function as girders.

(3) General straining of the ship-girder.

(4) Local actions.

(5) Corrosion.

2. **Water Pressures.**—The claim that the plating shall withstand the external water pressures is fundamental ; for a given spacing and arrangement of the frames, it determines an absolute minimum value of the thickness at each point of the bottom. For ships with a double bottom this thickness may be found from Curve A , Pl. IV. ; but where no double bottom is fitted, the outside plating is ordinarily for other reasons heavier than required to resist the water pressures.

In submarine boats the water pressure is the dominating factor and the problem is very definite, since the maximum depth of immersion to which the boat is to be tested is exactly known. The test head varies from 120 ft. in some navies to 200 ft. or more in others, depending on the type of the boat and on the service for which it is designed. Corresponding to a given head and frame spacing the thickness of the shell plating in submarine boats may be determined from Curve B , Pl. IV.

3. **Co-operation of the Plating with the Frames.**—The stresses in the outside plating due to its co-operation with the continuous frames to which it is attached and of which in a measure it forms part, will exist only in the immediate vicinity of the frames and will have their maximum values at the points of support and midway between these. Generally, in warships, the continuous girders are longitudinal, and the stresses which they induce in the plating are, therefore, likewise longitudinal. The method of determining the stresses is explained in SECTIONS 15 and 16.

4. **General Straining of the Ship-Girder.**—The requirements to general strength are satisfied with a fairly uniform thickness of the shell plating, only certain strakes must be reinforced, notably the sheer strakes and the keel strakes, partly to secure general strength in the most efficient manner, partly to prevent progressive rupture. These requirements demand distinctly different means of strengthening.

(1) *Increase in thickness.*—It is evidently good economy to make the most strained strakes heavier than the others, since the material is

thus used to best advantage as far as general strength is concerned. The keel and the sheer strakes should, therefore, be of increased thickness. It must not be overlooked, however, that the stress in a given strake is not materially reduced even by a considerable increase in thickness. Suppose, for instance, that the thickness of the keel strakes and the sheer strakes are increased, then the neutral axis will remain practically in the same position, the moment of inertia will be greater and the stresses will be somewhat reduced throughout the structure, but there will be no local and marked decrease in stress in the strengthened plates.

(2) *The use of high-tensile steel.*—Where progressive rupture is apprehended in a certain strake, an increase in thickness will not, accordingly, be of much avail, since the stress is not thereby sensibly reduced. Evidently the remedy is to construct the strake of a material which will stand the required stress without reaching the yield point, whence a high-tensile steel, possessing a higher elastic limit than the material in adjacent strakes, should be employed. This mode of reinforcement should, in particular, be applied to the sheer strakes, since progressive rupture is most likely to start there, but in high-speed ships the keel strakes also should be similarly strengthened.

The strake next below the sheer strake should be intermediate in thickness between that and the ordinary plating.

The keel strake is usually doubled, primarily in order to secure local strength when the ship is in dock or when it is aground. At the same time the flat keel, besides adding strength to the lower flange of the ship-girder, forms together with the vertical keel and the central strake of the inner bottom plating a strong independent girder which acts as a backbone to the bottom structure. Where a complete center-line bulkhead is fitted, the keel structure acts as its lower flange. The upper keel strake and the garboard strakes should be of the same thickness, somewhat heavier than the ordinary plating. The lower keel strake is usually much heavier in order to form a sort of "rubbing plate."

Where continuous longitudinal side bulkheads are fitted for more than half the length amidships, the contiguous strakes of the shell plating should be given a somewhat increased thickness.

In vessels where the double bottom extends only to the turn of the bilges, the bilge strakes should be strengthened so as to enable them to resist the shearing forces as explained in SECTION 6.

5. **Reduction in Thickness towards the Ends of the Ship.**—The maximum thicknesses are, ordinarily, maintained only for about

half the length of the vessel amidships. Toward the ends, where the longitudinal stresses are smaller, the thicknesses are reduced, and since, with a rational construction of the hull, the decrease in stresses is gradual, the reduction in thickness should likewise be gradual. The reduction is limited by the claims to strength under water pressures, especially at the bow, where the dynamic action of the waves may be very considerable. The amount of the reduction depends on the type and size of the ship and is best determined by practice and experience. It varies from ten to fifteen per cent. in battleships to from twenty to twenty-five per cent. in torpedo-boats, but in the sheer strakes of some ships it is more than fifty per cent. According to Lloyd's Rules, the reduction in the ordinary strakes is from about ten per cent. in small thicknesses to about twenty-five in great thicknesses.

6. **Comparison with other Ships.**—Ultimately the determination of the thickness must rest on a comparison with existing ships. Thus, as far as water pressures are concerned, the curves A and B, Pl. IV., although primarily based on experiments, depend for their practical usefulness on a verification by comparing them with the best practice. In the case of the two other principal actions, bending of individual girders and of the ship-girder, the calculated stresses can only be used for a direct comparison with ships already built.

7. **Compounding of the Stresses.**—The stresses due to the different straining actions will combine and form compound stresses of greater or smaller magnitude, but when the type ship is fairly similar to the design we may assume that the compounding will take place approximately in the same manner in both. Hence, if each of the individual stresses in the design is equal to or less than those in the type ship, we conclude that the same will be the case with the compound stresses, and it is, therefore, unnecessary to calculate these latter.

When the type ship is of dissimilar construction or of very different size the case is not so simple. The continuous frames may be transverse in one ship and longitudinal in another, so that even if the general straining actions are the same in the design and the type ship, the compound stresses will be entirely different. The same would be the case if the frame spacings in the ships to be compared were very different. If then a type ship that deviates materially from the design is to be used, it will be necessary to determine the compound maximum stresses by a combination of the individual stresses according to sign and direction and to use the result so obtained as a basis of comparison.

In such a case, the fact would have to be taken into account, that when the sagging strains are a maximum the ship rides across a wave-hollow, whence the water pressures amidships are much smaller than when the ship is floating in still water. In the hogging condition, on the other hand, where the ship rides with its middle on a wave-crest, the water pressures are much increased amidships. The variation in pressure due to this cause may amount to nearly fifty per cent. and affects the stresses in the outside plating directly due to the water pressures as well as those induced into the plating by the bending of continuous frames. The correctness of the compound stresses, even if calculated with the greatest care, will, however, always be open to doubt on account of the crudeness of the underlying assumptions and because all the concurrent causes of stress cannot be included in the calculation. Thus, the wave-height is assumed to be the same fraction of the length of the ship in small and in large vessels, which is manifestly not the case, the dynamic forces as well as the inclination of the ship are neglected, and the stresses due to water pressures can only be calculated very crudely. It appears, therefore, on the whole that *compounding of the stresses is of no practical value.* It is, moreover, unnecessary since ships of a type similar to the design are generally available for comparison.

8. **Local Actions.**—It remains to discuss reinforcements applied to the outside plating to provide for local straining or wearing actions of various nature.

The keel strake has been already discussed. In the bow, the plating is increased in thickness or doubled from the stem to a certain distance aft with the primary object of providing against wear and tear due to anchors and cables. For this purpose doubling is best suited, because it prevents chafing at the edges. The reinforcement of the plating in the bow serves also to support the stem and enables the shell better to resist the tearing action in case of ramming or collision.

In the stern, from the rudder-post forward, a similar increase in thickness, or a doubling of the shell, is used in order to resist the vibratory forces due to the propellers and the lateral bending action of the rudder. On account of the narrowness of the ship in this region, the flange strength under such bending must be derived entirely from the shell plating, while the floors and possibly a platform deck or breasthook perform the function of the web. According to Lloyd's Rules, the plates connected to the stern frame in steamers must be of the thickness required for the same strake amidships.

In unarmored, lightly built vessels the water-line strake should be given some extra thickness in order to allow for the active corrosion which is liable to take place between wind and water. Strengthening of this strake, including doubling at the bow, is particularly desirable in ships that have to navigate in waters where ice may be encountered.

Bulkhead liners are fitted for about two-thirds the length of the vessel amidships in way of watertight and oiltight floors and bulkheads. Around gunports, sidelights, sea-valves, and other openings in the outside plating, reinforcements should be applied wherever the strength falls below the standard. Finally, strengthenings are used where the side of the ship is exposed to the blast of guns fired in close proximity to the plating.

Additional thickness, $\frac{1}{16}$ in. to $\frac{1}{8}$ in., is given to plates of strongly curved form, which are to be furnaced, such as dished keel plates and boss plates. This is to allow for the deterioration which such plates suffer by treatment in the oven and by hammering. By repeated heating the thickness will be materially reduced throughout, and by hammering a local stretching and weakening of the material is apt to take place at the strongly curved parts.

9. **Corrosion.**—If the thickness of the shell is determined, as recommended above, by a comparison with ships that have stood the test of long service, we may consider the waste by corrosion to be sufficiently provided for.

10. **Standard Thickness.**—Strakes, other than those specially reinforced for general or local strength, are given a uniform or fairly uniform thickness, determined with due regard to all the various straining actions.

This thickness, which will be here referred to as the "standard thickness," varies from about $\frac{5}{8}$ in. in large battleships to $\frac{3}{8}$ in. in light cruisers and $\frac{1}{4}$ in. in large destroyers.

In large merchant vessels much greater thicknesses are used than is customary in warships. Lloyd's Rules give thicknesses up to nearly one inch for the outside plating under the flat part of the bottom. In the *Lusitania* the plating is in this region $\frac{21}{20}$ in. amidships.

In submarine boats the thickness varies from $\frac{5}{16}$ in. in small boats to $\frac{5}{8}$ in. in large boats.

41. PLATING BEHIND ARMOR.

The plating behind armor was formerly in most navies worked double. It was desired to give a very substantial support to the armor bolts and to offer a strong resistance to the forces created by impact of projectiles on the armor; in fact, the plating behind armor often formed an appreciable addition to the total protection. Since it was unnecessary with double plating to fit straps at the butts or along the edges, a perfectly flush surface was obtained, facilitating the fitting of armor bolts and wood backing. The French still use double plating behind armor on account of its greater yielding capacity (*déformabilité*).

Difficulties are, however, liable to occur with double plating in point of watertightness where the armor is below or near the water-line. Generally a wood backing is interposed between the armor and the

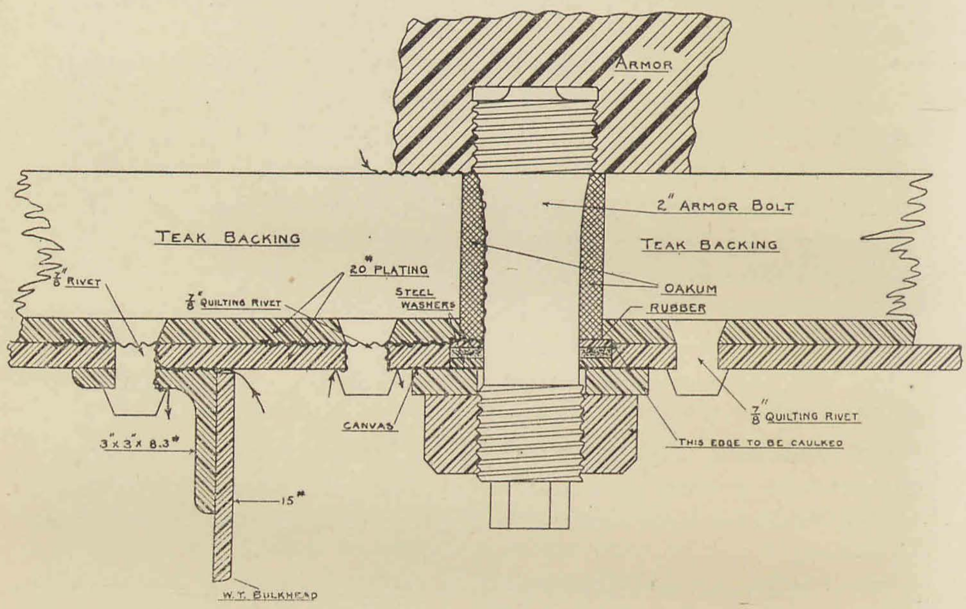


FIG. 100.—Watertightness of Double Plating behind Armor.

plating, but as neither the armor nor the backing can be reckoned upon to be watertight for a great length of time, the plating behind armor must be relied upon ultimately. Now, even if the calking of this plating is in perfect condition, water may in course of time find its way along the shanks of the armor bolts to the space between the two layers of plating, and may thence enter the ship along the shanks of defective rivets. Quilting rivets are, therefore, often placed round the armor bolts as near to the edge of the holes as practicable, whereafter the seam formed by the two plates in the bolt hole is calked. Another difficulty, liable to occur in this as in all other cases of double plating, is that if a compartment adjacent to the armored side is flooded, water may enter on one side of a bulkhead along the shank of defective rivets, perhaps by way of the heel of a bounding angle, and, percolating between the plates, it may reach the compartment on the other side of the bulkhead, as indicated in fig. 100. The use of double plating behind armor is, for these reasons, undesirable on the immersed part of the sides, and appears to be now abandoned for this region in most navies.

In modern ships a single layer of heavy plating is generally used, varying in thickness from ⅝ in. in smaller ships to 1 in. in larger ships. The plating is always worked flush in order to facilitate the adjustment of the length of the armor bolts, and in order to obtain a uniform, and hence minimum, thickness of the backing throughout. Ordinarily the seam straps are worked on the outside, but in case of light armor, where no wood backing is fitted, the seam straps must be worked on the inside and the frames should then be joggled over the straps.

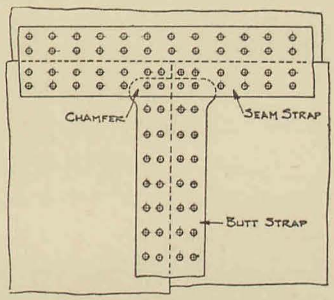


FIG. 101.—Butt Strap inserted under Seam.

The butt straps are often worked on the inside, but in some ships, as for instance in English battleships, they are worked on the outside. In order to obtain better conditions for calking, the ends of the butt straps may be chamfered and inserted under the seam straps. This method is used in the Danish armorclad *Herluf Trolle* (fig. 101).

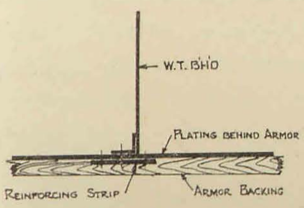


FIG. 102.—Reinforcing Strip instead of Bulkhead Liner.

Along the watertight bulkheads, vertical strips may be fitted externally on the plating as reinforcement instead of bulkhead liners.

In French ships of the *République* class (fig. 133) the plating behind armor is directly connected to the outside plating about two feet below the armor and takes the place of the armor shelf, being bent slightly outward at the foot. The sloping protective deck butts up against the plating behind armor and gives thus a very efficient support to the lower edge of the belt. By this construction it is intended to minimise the danger of leakage at the foot of the armor, a point of great importance, as evidenced in the battle of Tsushima. According to the report of Admiral Rodjestvensky, serious leakages of this nature occurred in several of the ships of the *Borodino* class, and it appears from press reports that special measures are taken in later Russian ships to prevent this form of damage.

42. PLATING OF DECKHOUSES AND OTHER SUPERSTRUCTURES.

The construction of deckhouses and similar structures depends on whether they are intended to form part of the ship-girder or not, and on the local actions to which they may be subject. A study of the stress diagram Pl. I. shows that a deckhouse, in order to be an effective strength member, must extend over at least one-half of the length of the vessel amidships. If it is much shorter it will cause great local stresses at the corners without adding materially to the longitudinal strength. A long forecastle, extending from the bow to the aft gun-turrets or some equivalent fraction of the length, is in this respect the most satisfactory type.

If, then, a superstructure is of sufficient length and if it is desired to utilise it in this way, it should be constructed on the same principles as the main hull. The thickness of the plating should be determined on the basis of the longitudinal strength calculation and the upper bounding strake should be reinforced like a sheer strake. The superstructure should extend the full breadth of the ship so as to make its outer walls flush with the sides, since otherwise severe stresses will come to exist in the deck plating. No doors or other large openings should be cut in the sides and compensation should be given for all openings which reduce the strength below the standard.

Theoretically it is very advantageous to use a superstructure as a strength member, since the stiffness of the ship-girder is in this way greatly increased and even a small weight of material properly applied in a superstructure gives a great moment of inertia. In armored vessels such as battleships and battle-cruisers, destined for hard fighting, it is, however, as explained in SECTION 5, *6*, altogether irrational to depend on the strength of such structures. If they are unarmored, they are liable to be demolished in action, and if protected by armor, they are usually pierced by gunports which destroy their value as strength members. In unarmored vessels conditions are more favorable, since the superstructure is there no more liable to destruction than the main hull. Yet it is not, generally, of advantage to construct the superstructure as a strength member, because it will always be difficult to avoid excessive stresses at the connections with the main hull. Ordinarily, therefore, superstructures should be lightly built and provided with expansion joints. Plating of from 5 lb. to $7\frac{1}{2}$ lb. with margin strakes of 10 lb. will be generally sufficient. Only the forecastle should be constructed of somewhat heavier plating, where it is exposed to the action of the sea. In ships of extremely small depth, on the other hand, as for instance certain types of river gunboats, it may be absolutely necessary to utilise the superstructure as a strength member.

43. RIVETING OF OUTSIDE PLATING.

Shell rivets are hammered up from the outside and are generally countersunk on the point. Only in certain torpedo-boats the rivets are hammered up on the inside.

1. **The Butts.**—In battleships the butts are generally treble chain riveted, either overlapped or strapped with alternate rivets omitted in the back rows. In the former case an efficiency of about $\frac{2}{3}$ is attained, in the latter case the efficiency is about $\frac{5}{6}$. In English battleships (1906) the butts are connected by double riveted straps. In the plating behind armor, quadruple riveted straps are often used with alternate rivets omitted in the back rows, whereby an efficiency of from $\frac{3}{4}$ to $\frac{4}{5}$ is attained. In ships of the scout class, the butts, whether lapped or strapped, are double chain-riveted, which, with the small thicknesses of plating used in these ships, gives the same efficiency as treble riveting in the large ships, about $\frac{2}{3}$.

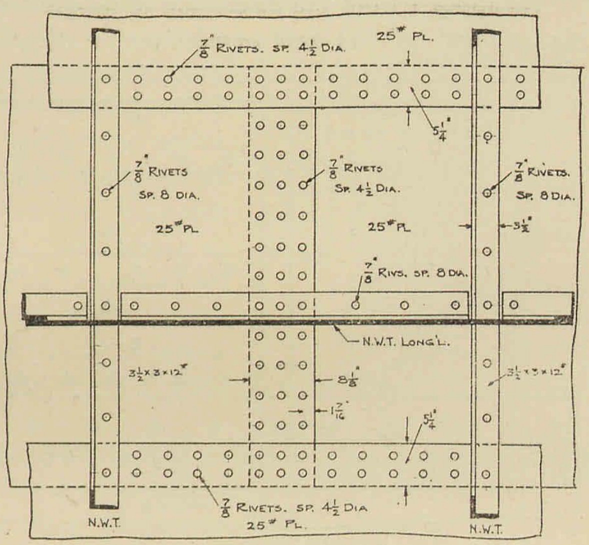


FIG. 103.—Riveting in Shell Plating below Protective Deck.

In torpedo-vessels of all classes the butts are likewise generally double riveted. Single riveting gives here a too low efficiency and involves the danger of excessive bearing pressures in the rivets, leading to crushing of the material and breakdown of the joints in some form or other. In small torpedo-boats, where the plating is too thin to permit calking, and where watertightness is obtained by some form of stop-water, it is possible to attain a very high efficiency with overlapped joints, since the rivets in the back rows may here be widely spaced. This fact is taken advantage of in certain boats of French design, as shown in fig. 104. We find there two closely spaced rows of rivets arranged zigzag along the middle of the joint and two openly spaced rows on each side of the middle rows. The backstrap-joint (SECTION **28**, *6*) seems, however, preferable and may be used with advantage in all the ordinary strakes of torpedo-vessels whether the butts are calked or not.

In merchant ships double riveted butts are, according to Lloyd's Rules, used only in small vessels up to about 1000 register tons ; in larger ships the butts are treble or quadruple riveted.

As appears from the discussion of progressive rupture in SECTION **39**, *the strength of the butt connections in the sheer strakes should in all*

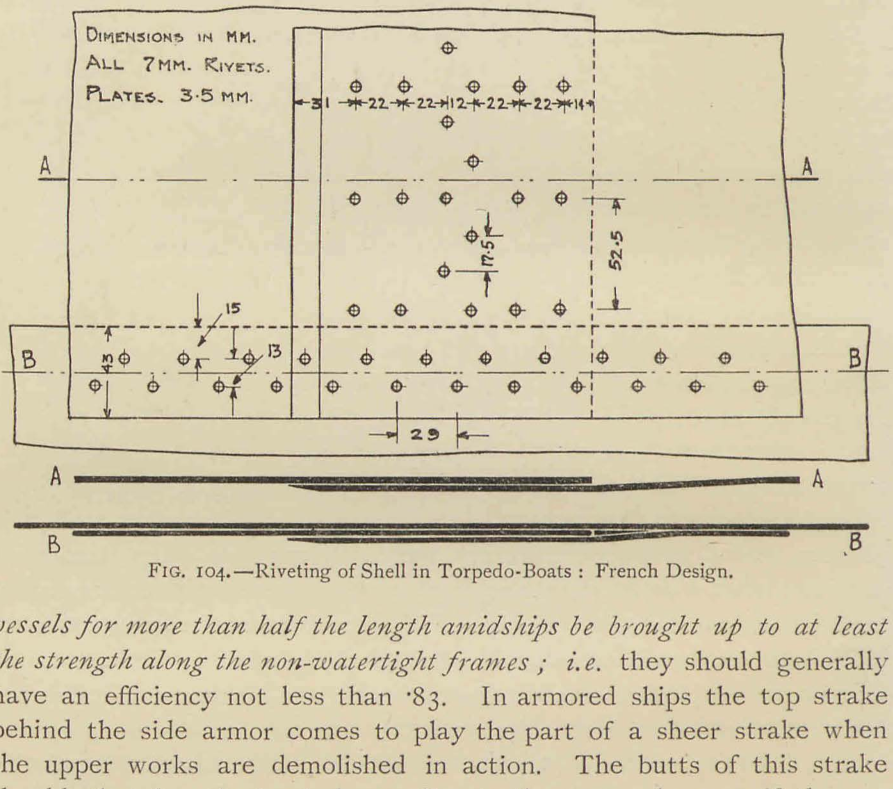


FIG. 104.—Riveting of Shell in Torpedo-Boats : French Design.

vessels for more than half the length amidships be brought up to at least the strength along the non-watertight frames ; i.e. they should generally have an efficiency not less than ·83. In armored ships the top strake behind the side armor comes to play the part of a sheer strake when the upper works are demolished in action. The butts of this strake should, therefore, be treated according to the same rule, even if the use of double butt straps is thereby rendered necessary.

In the sheer strakes of torpedo-vessels and light cruisers the double butt strap discussed in SECTION **28**, *5*, where the inner strap is of greater width than the outer, may be used with advantage. Whether the rivets in the sheer strakes of such vessels are countersunk or not, the points should always be riveted full so as to secure a good holding power. The best result is obtained by first drilling the rivet holes cylindrical and thereafter giving them a slight countersink so as just to remove the sharp edge. In this way the reduction in the strength of the plate is a minimum and an efficiency of about ·85 may be attained along the non-watertight frames. In such a case snap points or full-mashed points should be used.

The butts of *the flat keel plates*, and generally also of the garboard strakes, are connected by straps. The straps of both keel strakes are placed on the inside, extending ordinarily from the toe of the lower keel angle to the edge of the plates which they connect. When the vertical keel is watertight or oiltight, it is profitable, in order to facilitate calking, to carry the straps of the flat keel plates across the center-line, and to joggle the lower keel angles over the straps. Since great strength is essential in the keel construction, and since the outer keel strake is usually very thick, the straps should in all but the smallest ships be treble or quadruple riveted according to the thickness of the plates, and the alternate rivets in back rows should be omitted.

In doubling plates the rivets along the butts are generally arranged in a single row and spaced for watertight work. It appears, however, that along the butts of inside doubling plates the rivets might with advantage be more widely spaced, 6–7 diameters, since these butts are not calked. To make up for the loss in rivet area so incurred, the number of quilting rivets may be increased. Fig. 105 shows the riveting of the butts of a double stealer in the shell of a battleship.

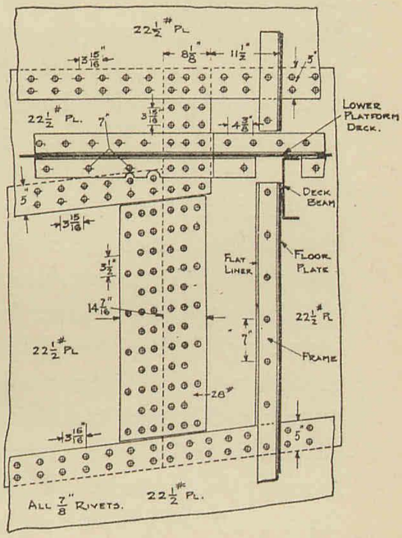


FIG. 105.—Riveting of Shell Plating at a Double Stealer.

2. **The Seams.**—Ordinarily the seams are double riveted ; only small vessels of moderate length, such as gunboats of less than about 1500 ts. displacement, have single riveted seams. Destroyers and torpedo-boats, even those of very small size, have double riveted seams, usually zigzag. Such vessels are exposed to great strains, and it would be difficult, with single riveting, to obtain watertightness when the plating is so light that it cannot be calked.

According to Lloyd's Rules, double riveted seams are used generally in vessels of moderate size, single riveted seams are permitted where the plating is less than about ·4 in. in thickness, and treble riveting is required on the sides of very large and long vessels according to certain rules. Treble riveting is prescribed also in ships of moderate size in the region where shearing is a maximum, *i.e.* near the neutral axis on about one-quarter of the length from the ends. Excessive shearing in this region has frequently occurred in ships of great fullness and large size, due to the great forces which come to exist when such vessels pitch in a seaway. In battleships the shearing action is not likely to be so great as in large merchant vessels of full form, and since, moreover, the hull is better stiffened, there seems no necessity for introducing such extra riveting in the seams. Scouts, and torpedo-vessels, that are liable to be forced up against the sea at high speed, may possibly become subject to excessive shearing forces in the bow. The side plating of such vessels should, therefore, be carefully watched for the appearance of strains in this region. Some recent light cruisers have treble riveted seams near the neutral axis in the bow.

In double riveted seams one rivet is omitted on each frame, viz. that farthest from the calking edge (fig. 105). Only in the sheer strakes the rivets of both rows are retained on the frames.

The edge of the inner keel strake is connected to the outer keel strake by a single row of rivets, spaced for watertight work. The same practice is followed in other doubling plates, but in addition a number of "tack-rivets" or "quilting-rivets" are distributed regularly over the plates in order to make them work together. Tack-rivets should be from 12 to 16 diameters apart, depending on the curvature of the plates. In railway bridges, when two or more plates are used in contact, tack-rivets are to be not more than 12 in., or 14 to 16 diameters, apart.*

* Am. Railway Eng. Ass.

CHAPTER XI.

WOOD SHEATHING AND THE COMPOSITE SYSTEM OF CONSTRUCTION.

44. WOOD SHEATHING.

1. **Advantages and Disadvantages.**—The primary purpose of sheathing the bottom is to permit coppering as a means of antifouling. Sheathing, if properly fitted, will not only prevent galvanic action between the copper and the steel plating, but will protect the shell against corrosion. It also adds considerably to the strength and, in particular, to the stiffness of the bottom. In small ships its presence will justify the omission of a double bottom. The cost of maintenance of sheathed vessels is considerably less than that of steel ships, since they do not require so frequent docking. Sheathing, on the other hand, adds to the displacement and the first cost of the ship and, if badly fitted, may cause serious corrosion of the shell and expensive repairs.

Sheathing is used to best advantage on ships destined for service in tropical seas where fouling is rapid, and especially where docking accommodation is limited and expensive or under foreign control. Its application is a question that has been the subject of much discussion. At present it is rarely applied and only to gunboats and smaller cruisers, but formerly it was fitted also to many larger cruisers.

2. **Watertightness of Sheathing. Single *versus* Double Sheathing.**—It is important that the sheathing should be perfectly watertight, and that all metallic connection between the copper and the hull should be avoided, but such complete isolation of the steel hull from the sea and from the coppering cannot be obtained without minute precautions.

The stem and the stern-posts and other external castings which are not covered by wood are made of bronze or brass in order to prevent waste by galvanic action. Formerly two layers of sheathing were used, the inner secured to the hull by iron bolts, the outer secured to the inner by brass screws. It was found, however, in many cases, that after a short period of service, the water would find its way behind the sheathing and spread over large areas between the wood and the hull plating. The result was an excessive galvanic action upon the iron bolts which secured the inner thickness, while no deterioration of the iron or steel hull took place. This action on the iron bolts was so serious that double sheathing was ultimately abandoned altogether. At the time when two thicknesses of sheathing were used with coppering, a single thickness was used in many ships in connection with zinc plating. No attempt was made to isolate the zinc from the iron, since the zinc, being electro-positive to iron, acted as a protector to the hull. Hence, the sheathing was not calked, and a free connection was allowed between the two metals and the sea-water. It was found, however, that in some cases the zinc would waste away very rapidly and irregularly, in others it would last a very long time, but then its antifouling qualities were poor. Zinc, moreover, presented a rougher surface than copper. About the end of the eighties of the last century single sheathing with coppering was definitely introduced. The planks were hereafter secured by bolts of composition, and the coppering consisted of either pure copper or of muntz-metal. The system of single sheathing has given satisfaction when carefully fitted and will be here described.

3. **Single Sheathing.**—The sheathing should cover the entire immersed surface of the hull with the exception of the bronze or brass castings and should extend to some feet above the water-line. Below the flat keel, a deep wood keel of elm or teak is, generally, fitted in order to take rabbets for the garboard strakes. A false keel of oak is nailed on to the main keel by means of spikes. The planks are of pine or teak; they are laid in strakes from 30 to 40 feet long, from 8 to 9 inches wide, and from $3\frac{1}{2}$ to 4 inches thick, and should be given a good shift of butts. They are fastened to the shell with $\frac{3}{4}$-in. or $\frac{7}{8}$-in. screw-bolts of composition or naval-brass, and are rabbeted into the stem and the sternpost. The bolts are inserted through holes in the planks and screwed into the bottom plating. On the inside a nut is hove up on a countersunk washer with hemp grommet set in red lead. The head of the bolt is hove up to a depth of about one inch beneath the outer surface of the plank on a hemp grommet steeped in red lead. The hole outside the head is filled up with cement or with white lead filling.

In order to secure an efficient bolting of the planks, the shell plating should be not less than three-eighths of an inch in thickness. In ships of small size this implies a rather heavy shell, and sheathing is, therefore, not suited to small, lightly built vessels, such as torpedo-boats. In larger vessels, on the other hand, where plating thicker than $\frac{3}{8}$-in. would ordinarily be used, it may be permissible to reduce the thickness, generally to $\frac{3}{8}$ in., on account of the reinforcement which the plating receives from the sheathing. In most vessels the thickness is maintained uniform to the ends, but otherwise the bottom is plated in the usual manner.

Generally there are two or three bolts in each plank in every frame space. At the butts there are two bolts in each plank. In marking the shell plating for the bolt holes, care must be taken that the holes do not come on the rivets. The butts of the planks should be placed between the frames.

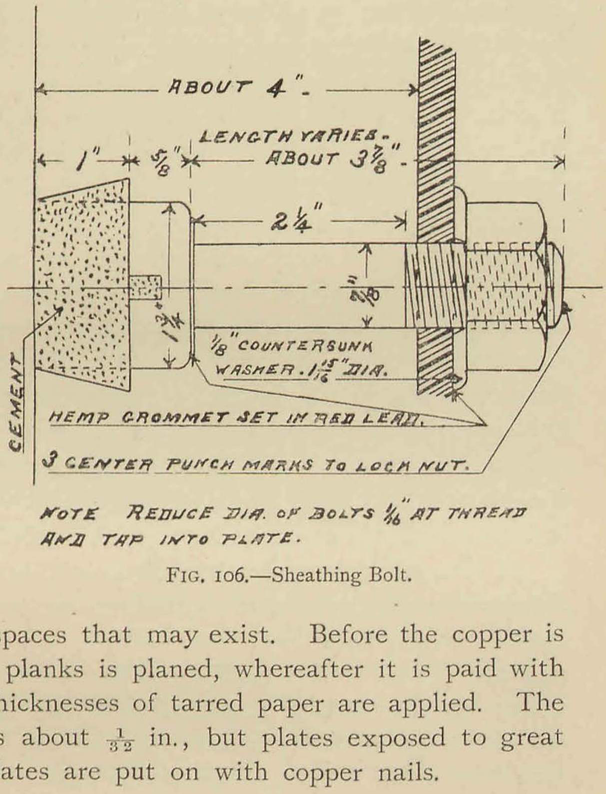


FIG. 106.—Sheathing Bolt.

Before laying the planks, the bottom is covered with a mixture of white and red lead, applied with a brush. After the planks are fitted, the edges and butts are carefully calked, whereupon a mixture of white and red lead and linseed oil is injected behind the planks in order to fill any small spaces that may exist. Before the copper is put on, the surface of the planks is planed, whereafter it is paid with coal tar, and one or two thicknesses of tarred paper are applied. The thickness of the copper is about $\frac{1}{32}$ in., but plates exposed to great wear are heavier. The plates are put on with copper nails.

45. THE COMPOSITE SYSTEM OF CONSTRUCTION.

1. **Advantages and Disadvantages. History.**—By this system the keel, stem, sternpost, and almost the entire outer shell are constructed of wood. Steel is employed in the outer shell only in the most important strakes, but the entire internal structure, including transverse and longitudinal frames, bulkheads and deck beams, is of steel, and does not differ in construction from a steel ship of the same class and size.

The system presents the advantages that it permits coppering of the bottom, and that the wooden sides make the ship more habitable in hot climates. It gives greater structural strength than can be obtained by the exclusive use of wood, and the bottom is locally stronger than that of a steel ship. The system, in fact, combines several of the advantages of metal construction with those of wood construction, and was, therefore, naturally adopted in the period of transition, from about 1860 to 1870, when iron commenced to be introduced on a large scale instead of wood in shipbuilding. At that time it looked, in fact, as if the composite system would supersede iron for sailing vessels of the merchant marine, especially in such trades as the tea-trade, where freights were high and where a quick return voyage was of the utmost importance. The clean bottom of composite vessels was here a great advantage. The system was, likewise, extensively introduced in the navies in fast cruisers provided with both steam and sail power. In the merchant marine, however, it was completely abandoned already about 1870. At that time the Suez Canal was opened, and steamers largely superseded sailing vessels in those trades where composite ships were used; but several drawbacks, inherent in the composite system, contributed to its abandonment. The system is expensive, and involves not only great first cost, but also high cost of maintenance due to deterioration by galvanic action. It does not give so great a longitudinal strength as attainable with an all-steel hull. It involves great shell weight, because the planking must either be of two thicknesses or, if of one thickness, it must be very heavy in order to permit efficient calking.

In the military navies the composite system, due to its structural deficiencies, was not found suitable to the large and long, high-powered cruisers which came into use towards the end of the century. On account of the inflammability of the wooden sides it was, moreover, unsuitable for all ships intended for severe fighting. The system was, therefore, abandoned in cruisers and its use was restricted to gunboats and other small vessels of moderate power built for police or surveying service in the tropics. It is particularly suitable for gunboats designed for protracted service on rivers and on coasts where docking facilities are scarce and where fighting is not likely to occur.

2. **Mode of Construction.**—Usually the frames are transverse, held together externally by the keel strake, the sheer strakes, and the bilge strakes, all of steel. Where only one thickness of planking is used, it is necessary to fit diagonal rider plates or ties in order to prevent distortion of the hull by longitudinal shearing. These rider plates should be securely connected to the sheer strakes and to the bilge

strakes and should be riveted to the frames which they cross; they should be worked like lattice, two systems of plates, inclined at an angle of 45° to the frames, crossing each other at right angles. If arranged in this manner, the rider plates will only have to resist tension and may, therefore, be made rather narrow and slender. The keel, stem, and sternpost are of wood, but should be stiffened and held together by continuing the flat and vertical keel plates up along the stem and sternpost to the height of the upper deck. Where two thicknesses of planking are fitted, the planks of one layer act as seam straps for the other. By this construction a good longitudinal connection is secured and it is unnecessary to fit diagonal rider plates.

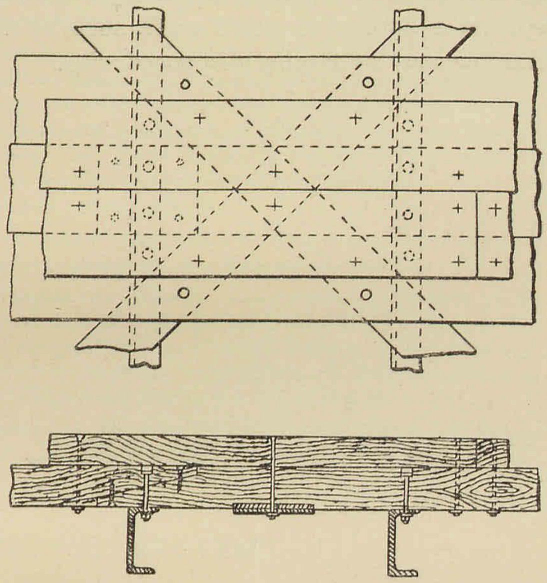


Fig. 107.—Composite Construction: French System.*

The principal difficulty by this system is to obtain a good connection between the steel frames and the planks, for the planks do not get as good a support from the narrow steel frames as from the broad and closely spaced frames in a wooden ship. The planks are generally teak or pine, in certain parts elm. They are from about 9 to 12 inches wide, and when only one thickness is used, about 4 inches thick, but if two thicknesses are used, the inner is 3½ inches, the outer 2½ inches. The inner planks, or, if only one thickness is fitted, all the planks, are secured to each frame by two screw-bolts of galvanised steel or yellow metal. In way of plates or diagonals, the bolts are placed on them. The bolts are tapped into the frames or plates and secured with a nut on the inside as in sheathed vessels. The outer planks are fastened to the inner by copper bolts, driven and clenched upon metal rings on the inside. The butts of the inner layer are scarphed and are secured to the frames on which they are placed, obtaining thus a good support. The butts of the outer layer are placed between the frames and receive two copper bolts on each side of the butt.

* J. Rougé, *Construction du Navire*, Paris, 1904–05, i. 298.

The construction of the decks in composite ships is quite analogous to that of the sides, and will be described in a later chapter.

Between the pure composite system here described and that of sheathing a variety of constructions are found. The steel shell may be more fully developed than in the pure composite type, without being complete as in the sheathed vessel. The wood planking may stop at a height of a few feet above the water-line, and the stem and sternpost may be of bronze or brass. A very strong construction may be obtained by arranging the planks diagonally in two thicknesses inclined in opposite directions at an angle of about 45 degrees to the vertical. When this system is applied to small craft, such as steam or sailing launches, the framing may be largely dispensed with.

CHAPTER XII.

FRAMES.

46. PRINCIPAL SYSTEMS.

ACCORDING as the frames are disposed transversely, longitudinally, or both transversely and longitudinally, the system of arrangement is called the "transverse," the "longitudinal," or the "mixed system."

1. **The Transverse System.**—This system, with closely spaced frames, was adopted in the early iron vessels in analogy with the framing of wooden ships, and is still used in its pure form in small vessels. The longitudinal strength is supplied by the shell plating and the decks, assisted by the keel structure. Generally, one or more side keelsons or stringers are worked intercostally with continuous plates or angles inside the frames. As in wooden ships, the beams which support the decks are transverse. For small vessels of a moderate ratio of length to depth, this system is rational, and during the early days of iron shipbuilding it was found perfectly satisfactory. The longitudinal strains were small and the shell plating, which was of relatively great thickness, provided ample longitudinal strength. As ships increased in size, and as the

speed increased, involving a higher ratio of length to depth, greater longitudinal strength and stiffness were required than obtainable by the transverse system. In fact, if these qualities were to be obtained without the use of longitudinal frames, the shell plating would need to be excessively heavy—a feature which could not be accepted in view of the increasing claims to economy in hull weight.

2. **The Longitudinal System.**—Already in the forties and fifties of the last century a number of steamers were built, possessing the feature

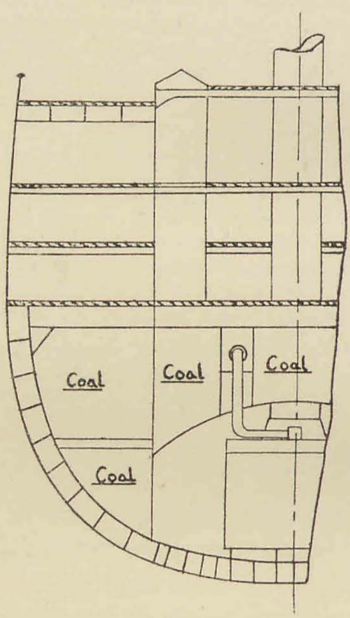


FIG. 108.—Midship Section, *Great Eastern.**

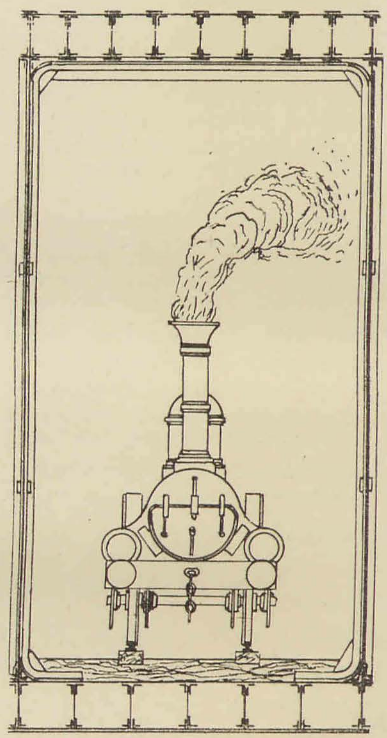


FIG. 109.—Section of Britannia Bridge.*

of longitudinal framing in a more or less marked degree. The most notable application of the longitudinal system was found in the *Great Eastern*, built in 1859. The structural arrangements of this ship were due to the joint labors of the French Civil Engineer Mons. I. K. Brunel and the English Naval Architect Mr Scott Russell.† We find here the clear conception of the ship's structure as a hollow girder. The *Great Eastern* was, in fact, built essentially on the box-girder principle applied

* Reproduced by kind permission of Mons. A. Croneau from *Construction Pratique des Navires de Guerre*, Paris, 1894, i. pp. 133, 134.

† Sir W. H. White, *Manual of Naval Architecture*, 1900, p. 368.

to the Britannia Bridge, which connects the Isle of Anglesey with the coast of Wales. The framing of the *Great Eastern* was entirely longitudinal, not only in the top and bottom flanges, but also on the sides up to the lower deck. Both the bottom and the upper deck were double, and the longitudinal framing was fitted between the two layers of plating, which it served to support and connect. The longitudinal strength of the upper and lower flanges of the girder, formed by the ship's structure, was thus directly and specially provided for. The transverse strength and stiffness were maintained by bulkheads, complete or partial.

3. **The Cellular System.**—The longitudinal system was never adopted in the pure form in which it was applied in the *Great Eastern*, perhaps chiefly because it involved considerable difficulties in erection, but the principle of longitudinal framing soon found extensive application in the so-called "cellular system," which was introduced universally in the double bottom of both merchant ships and warships. It may be described as a mixed system, since it consists of an intersecting network of transverse and longitudinal frames. The predominance of the longitudinal element varies considerably in the different varieties of the cellular system. In most merchant ships only the vertical keel and a "margin plate" on each side, forming the boundaries of the double bottom, are continuous, the other longitudinals are intercostal. In warships the longitudinal frames are more strongly developed.

4. **The Isherwood System.**—Of recent years the so-called "Isherwood system" has been introduced quite extensively in the merchant marine. It is regarded by many and is often referred to as a longitudinal system, but it is not difficult to show that this view is erroneous.

The closely spaced transverse frames and beams are here omitted, but the transverse strength is obtained by a number of widely spaced deep transverse girders, called "transverses," which extend all around the bottom, the sides, and under the decks. These girders are spaced from 10 to 20 feet apart and are of sufficient strength to perform the functions ordinarily belonging to the transverse frames and to the beams. Between the transverses, the shell plating as well as the decks are supported by a system of continuous longitudinal stiffeners of small depth, piercing the transverses, with which they are connected by clips, and which are slotted to permit their passage. The spacing of these so-called longitudinals is about one-half the breadth of the plating, two longitudinals being placed on each strake. The system is seen to consist essentially of a number of widely spaced transverse girders, while relatively light, longitudinal stiffening ribs enable the shell to span the distance between these girders. The main part of the framing is, therefore, transverse, and the longitudinal ribs virtually form part of the shell, which they serve to stiffen. Only the fact that these ribs are continuous through the transverse girders gives to the system a longitudinal character. *The Isherwood system may, therefore, properly be characterised as a mixed, essentially transverse system.*

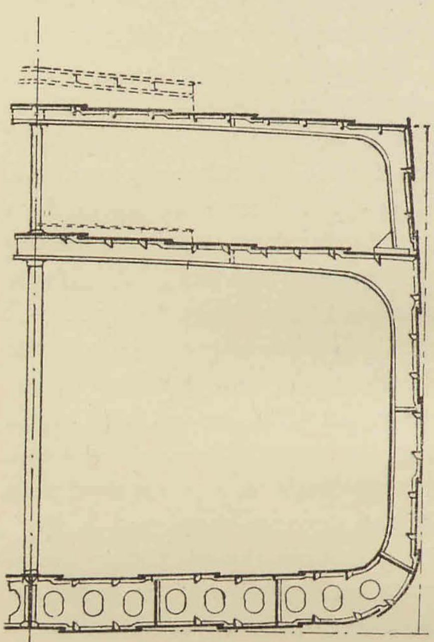


FIG. 110.—"Isherwood" System of Framing.*

Structurally, the chief advantages of this system are, first, that a greater moment of inertia can be obtained on a given weight by the deep transverses than by the ordinary transverse frames, and second, that the inner and outer shell, being well stiffened longitudinally, are more effective as strength members than by the ordinary construction, where they are liable to buckle when in compression.

The Isherwood system has been approved by the Classification Societies, and has found favor with many shipbuilders and shipowners. There is claimed for it a considerable saving in weight. It lends itself better than the pure longitudinal system to erection and permits an extensive use of riveting by power. The numerous curved and bevelled frames of the ordinary system of construction are here replaced by nearly straight longitudinals, which require little work in shaping. Repairs can be carried out with great ease.

47. GENERAL ARRANGEMENT OF THE FRAMES.

1. **Comparison between Warships and Merchant Ships.**—All ships, whether warships or merchant ships, when they are afloat, are subject to the pressures of the water and have to support the weight of the hull with all that it contains and carries ; when moving in a seaway, they are all subject to dynamic actions, and when they are aground or in dock, the pressures of the water will be, in all ships, partly or wholly

* From J. W. Isherwood, "A New System of Ship Construction," *Inst. Nav. Arch.*, 1908, pl. x. fig. 2.

replaced by local reactions. There is, however, a marked difference between warships and merchant ships, especially those of the cargo-carrying type, in the way in which the forces of weight and buoyancy neutralise each other through the structure.

In a warship the dead load, consisting of armor and ordnance, is largely placed on the decks and on the sides, and is transmitted through the sides and the bulkheads to the framing system of the bottom, before it can be supported by the forces of buoyancy. In a cargo vessel the load consists of the cargo, which entirely or largely rests directly on the inner bottom or on the frames, or is transmitted to the frames through pillars. Hence the function of the bulkheads and the sides as transmitters of weight will be, relatively, unimportant. The forces of buoyancy and weight will for the greater part neutralise each other directly through the frames of the bottom. These frames will be, therefore, far less exposed to bending and shearing than in warships, where the excess of buoyancy is often considerable between the bulkheads. (Example, SECTION 16.)

Warships are, moreover, exposed to several actions which need not be considered in merchant ships, notably attack by artillery, torpedoes, and mines. By the impact of projectiles the structure is exposed to violent impulsive forces, strictly local where the side is unarmored and where the projectile is not explosive, but more widespread where the side is armored and where shell effect is added to impact and penetration. By the explosion of mines or torpedoes, the action is extremely violent and impulsive over a rather sharply defined area of the bottom near the center of explosion, where the plating is directly subject to the gas pressures. Outside this area more moderate pressures, transmitted through the water, will exist over a large region of the immersed part of the bottom.

2. **Principal Functions of the Frames.**—The complex system of forces which strains the structure under the various actions is always in equilibrium, statically or dynamically, but this equilibrium or neutralisation of the forces cannot be established without stresses being created in some or all of the strength members of the structure. The stresses may be due to simple tensile or compressive forces, but, generally, they are due also to bending and shearing. The frames perform their functions as strength members in several distinctly different ways, which may be grouped as follows:

(1) They act as integral parts of the ship-girder when this is exposed to longitudinal or transverse bending, and in this capacity co-operate with the inner and outer shell plating, taking their share of the direct tensile

or compressive forces. At the same time they perform the important function of stiffening the plating, preventing it from shirking its work by bulging and buckling.

(2) They act as girders between the rigid diaphragms of the structure : bulkheads, decks, double bottom, etc., transmitting to them the weights, water pressures, and other forces with which they may be loaded.

(3) They support the inner and outer shell locally against water pressures and other normal forces.

Besides these three principal functions, there are others, not so fundamental or of such a general nature, but which are nevertheless very important.

(4) Acting as stiffeners of the bottom and the sides the frames have to support forces, usually compressive, in the direction of their length. Most important under this head is the load which the transverse frames on the sides have to support, viz. side armor and other weights resting on the sides.

(5) Inside the double bottom, some of the frames act as watertight or oiltight partitions.

3. **Grouping of the Frames.**—Since the functions, which the frames are called upon to perform in a warship, differ radically in nature according to location, it is necessary to consider the frames separately in each of the different parts of the ship, which we shall divide into the following groups :

The lower, nearly horizontal part of the double bottom.
The double bottom on the sides.
The region outside the double bottom below the armor shelf.
The sides behind armor.
The unprotected sides above the level of the armor shelf.

4. **Framing of the Lower Part of the Double Bottom.**—*General longitudinal and transverse strength.*—The claim to general longitudinal strength is evidently best fulfilled by a system of continuous longitudinals. Transverse strength is already provided for by the bulkheads, but in order to stiffen the longitudinals and the bottom structure in general a system of intercostal transverse frames should be worked between the longitudinals.

Transmission of forces to the rigid surfaces of the structure.—The flat part of the double bottom constitutes a practically horizontal platform which is subject to the forces of buoyancy, applied as a tolerably uniform pressure on its lower surface. Distributed irregularly over its upper surface we find the vitals of the ship and other weights resting directly on it, but the main support of the bottom against the upward forces of buoyancy is derived from the numerous bulkheads and from the sides, which transmit to it the weights from the upper parts of the ship. In some cases lines of stanchions perform the same function. We have thus to deal with a loaded platform, subdivided in rectangular areas and bounded by rigid supporting diaphragms.

The vertical keel is generally supported by stanchions or center-line bulkheads and thus forms a rigid line of support. Hence, transverse frames will obtain a good support at the center-line and since the span across to the side bulkheads is shorter than the distance between the transverse bulkheads in engine and boiler rooms, it would appear of advantage to base the support of the double-bottom platform on a system of transverse continuous frames. Since, however, the vertical keel must be continuous, the transverse frames would here be severed and would not have the high degree of fixity obtainable with longitudinals carried continuously under the transverse bulkheads. Moreover, we know from experience that sufficient girder strength can be obtained with longitudinal frames even in engine and boiler rooms, and outside these compartments the transverse bulkheads are more closely spaced. Hence, a system of longitudinal frames is at least acceptable, even if not everywhere absolutely the best in point of girder strength. The intercostal transverse frames, advocated above, are well adapted to assist the longitudinals in their capacity as individual girders, since they help to transmit the load on the inner and outer shell to the longitudinals and support them laterally at the same time. The intersections between the frames form points of great strength and stiffness, of particular importance when the ship is in dock or aground.

Support of the inner and outer shell plating against water pressures.—The principal claim to the framing in this regard is that it should leave but narrow or small areas of the plating unsupported. This might be attained either by a purely transverse or by a purely longitudinal system, provided the frames were spaced close together, but a close spacing of the frames in the double bottom is undesirable because it renders access to and work in the compartments difficult. It is, therefore, preferable to use a network of more widely spaced transverse and longitudinal frames; as already recommended for other reasons.

Watertight and oiltight subdivision.—It is necessary to subdivide the double bottom minutely, and hence a number of frames, transverse and longitudinal, must be made watertight or oiltight. The rectangular network of framing, at which, so far, we have arrived through our study, lends itself well to such subdivision, but the frames should be so designed that this end is attained with a minimum of work and with due regard to the claims to continuity of the frames and to facility of erection. As a general principle, it is advantageous to make the watertight frames continuous, since in that case the amount of calking is reduced to a minimum and the simplest and most efficient construction is obtained, although this consideration ought not to take precedence over the requirements as to strength. Where two watertight frames cross each other, the longitudinal frame should be made continuous because it is the most important in point of strength.

Hence, from every point of view, we arrive at the conclusion that the lower part of the double bottom should be framed on the mixed system, and that the longitudinals should be given precedence both as to strength and continuity. The longitudinals should, as far as space permits, be carried to the ends of the double bottom, even if this extends the full length of the ship.

5. **Framing of the Double Bottom on the Sides of the Ship.**—*General longitudinal and transverse strength.*—In the lower part of the side structure, including the turn of the bilges, the greatest principal stresses will be longitudinal, especially under the circumstances of most severe strain when the ship is heeling over in a roll. The framing here, therefore, should be of the same nature as in the flat part of the bottom. Higher up on the sides, where the shearing forces predominate, the principal stresses are oblique, as illustrated on Pl. I. It has been suggested on this ground to use diagonal framing on the sides, but oblique intersections between the frames and other structural members and the seams of the plating are objectionable because there would be difficulties in laying off, in construction, and in erection. Diagonal framing is, therefore, ill suited to ship construction. Moreover, the web strength of the ship girder is already amply secured, and the stresses are on the whole moderate in this region of battleships, whence the arrangement of the frames should here be determined on the basis of other considerations.

Transmission of forces to the rigid surfaces of the structure.—In order to perform this function, the frames should, again, be placed parallel with the shorter sides of the rectangles bounded by the rigid diaphragms and should be continuous through the boundaries. In most battleships the distance from bulkhead to bulkhead does not differ greatly from the height between the protective deck and the flat part of the bottom. By making the transverse frames continuous, erection is facilitated, a certain arch effect is obtained, and the frames will be well adapted for carrying the weights on the sides. The end attachments, however, are not so efficient as with continuous longitudinals. Generally, therefore, the two systems are equivalent in this case.

Support of the inner and outer shell plating against water pressures and pressures due to the explosion of mines and torpedoes.—The water pressures on the sides are, on the whole, more moderate than under the bottom, although they are rarely counteracted by internal forces, except as may be the case where coal bunkers are adjacent to the sides. Far greater are the requirements to strength and stiffness of the sides when mines or torpedoes explode in contact with the outside plating, a danger to which the sides are much more exposed than the bottom.

Whether the pressures are due to one cause or the other, the best support of the plating is again obtained by a mixed system of framing. The points of intersection between the frames are capable, to a remarkable degree, of resisting the pressures due to under-water explosions. While the shell plating between the frames is blown inwards and the frames themselves collapse between the points of intersection, the structure yields but little at these points, except in the immediate vicinity of the centre of explosion.*

Support of the side armor and other weights resting on or transmitted to the sides of the ship.—The main part of this weight is probably taken by the transverse bulkheads, and the sides below the armor shelf can easily carry the remaining load provided the inner and outer bottom plating are properly stiffened by a mixed system of frames as just recommended.

The requirements as to *subdivision* lead to the same conclusions as for the lower part of the double bottom.

Summing up, we conclude that the framing on the sides within the double bottom should be on the mixed system as in the lower part of the bottom, but, unless there is an appreciable difference in the unsupported length of the transverse and longitudinal frames, it is indifferent which of the two sets is made continuous, except at the turn of the bilges, where the longitudinals should be continuous.

6. **Framing Systems within the Double Bottom in Existing Ships.**—We shall now describe and discuss the practice of some of the leading navies.

Fig. 111 gives in diagrammatic form the framing system of the French battleship *République*.† The vertical keel and the first longitudinal are continuous throughout, and the latter is watertight. The

* *Schiffb. Ges.*, 1913, p. 639.

† J. Rougé, *Construction du Navire*, Paris, 1904–05, i. 342; and *Constructions Navales, Coque*, Paris, 1912, p. 131.

third and fourth longitudinals are continuous and watertight where they are under a longitudinal bulkhead, respectively in boiler and engine rooms ; elsewhere they are non-watertight and intercostal. The seventh longitudinal is watertight and is continuous between the main transverse bulkheads ; the object of making this longitudinal watertight being

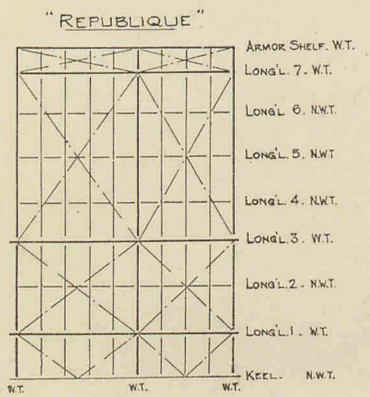


Fig. 111.

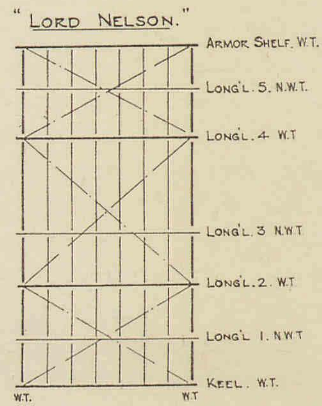


Fig. 112.

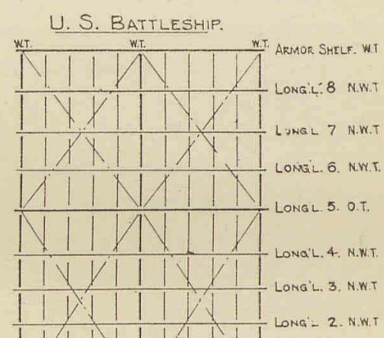


Fig. 113.

General Arrangement of Framing below Armor Shelf.

probably to limit the inflow of water into the double bottom in case of damage to the armor shelf. The other longitudinals are intercostal and non-watertight. The transverse frames are continuous between the first and the third or fourth longitudinals and thence to the seventh longitudinals. The continuity of the transverse frames is preserved in French ships to such great extent chiefly in order to facilitate erection and because the transverse system is considered simpler and cheaper than the longitudinal.

Fig. 112 shows the framing system of the English battleship *Lord Nelson.** All the longitudinals, including those on the sides of the ship, are here continuous, and the transverse frames are purely intercostal.

Fig. 113 shows the framing of a battleship of the United States Navy. All the longitudinals are continuous as in the English ships. The only notable difference, in point of continuity, is that the outer angle of the transverse frames is continuous in the American ships from the keel to the protective deck on each side, the object being probably to secure fairness of form of the ship during erection and, at the same time, to obtain some measure of transverse continuity. The vertical keel is non-watertight, but in recent ships the claims to subdivision of the double bottom in feed-water and oil tanks have rendered it necessary to make the vertical keel watertight or oiltight under all engine and boiler rooms. Apart from such claims, the only watertight longitudinal is the fifth on each side, placed at the turn of the bilges.

The American system of framing seems, on the whole, the most satisfactory. While it gives precedence to longitudinal strength and continuity, a certain weight is also given to transverse strength and to facility of erection. The French method of working a watertight longitudinal closely below the armor shelf seems very advantageous.

7. **Framing outside the Double Bottom below the Armor Shelf.**—In many vessels the double bottom stops before it reaches the trimming tank bulkheads forward and aft. Assuming this to be the case, we shall deal here with the framing between the ends of the double bottom and the trimming tanks. The framing inside the trimming tanks will be discussed in a later chapter. The great length and high speed of modern ships render it necessary, now more than formerly, to provide a considerable measure of longitudinal strength at the ends. When a ship is forced up against the sea, the bending forces may be very great at the bow, and with the overhanging stern of modern ships great longitudinal bending exists in the stern portion when the ship is in dock. The vertical keel and at least one of the first longitudinals on each side should, therefore, be carried continuously beyond the double bottom to the ends of the ship, and should be efficiently connected to the stem and the sternpost whether directly or by breast-hooks.

Above these longitudinals the frames should be on the transverse system, because the platform decks and the protective deck give good support for transverse frames. Intercostal floor-plates should be fitted between the continuous longitudinals in the bottom, while semi-intercostal hold stringers should be fitted on the sides in continuation of the

* J. J. Welch, *Naval Architecture*, London, 1907, chaps. vi. and vii.

longitudinals in the double bottom in order to support the transverse frames, to stiffen the plating, and to tie the shell to the frames.

Where the double bottom extends only to the turn of the bilges, the frames between the double bottom and the armor shelf (protective deck) should be on the transverse system, well bracketed to the margin longitudinals.

8. **Frames behind Armor.**—Support of the armor is here the all-important consideration. The energy of the projectile is consumed in three ways: (*a*) breaking up of the projectile, (*b*) perforation and breaking up of the armor, (*c*) elastic work and damage to the supporting structure. The more unyielding the supporting structure is, the less will be the liability of the armor to crack and break, and the less will be the damage done to the structure itself. Fig. 114 suggests the general features of a design which is believed to possess the required rigidity.

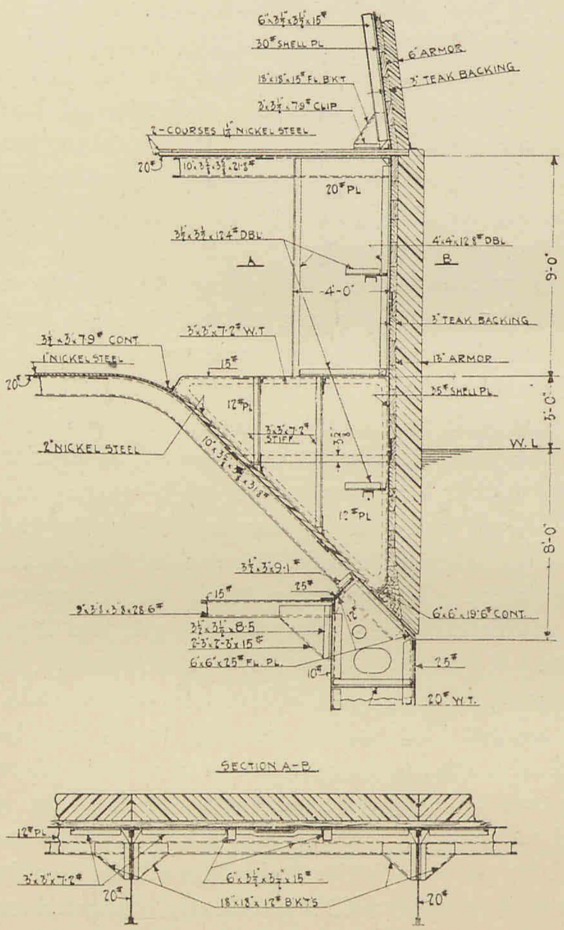


FIG. 114.—Proposed Framing behind Armor in a Battleship.

Behind the armored side are found two or more decks of great strength and rigidity, which should be used as main lines of support, whence the stiffening girders, whether bulkheads or frames, should be normal to these. Where the armor deck is sloping, a tier of substantial beams covered by deck plating should be worked in continuation of the flat part of the deck, providing an intermediate line of support between the lower edge of the sloping deck and the deck next above. The seams of the armor plates should fall on the deck edges and the butts should be supported by partial bulkheads or by very deep web frames as on fig. 114. Between these rigid lines of support a system of vertical frames of lighter construction should be worked, supported by deep semi-intercostal horizontal girders midway between the decks. While these are the general features recommended, the details of the supporting structure must be determined experimentally on the proving ground. Absolute rigidity should be secured, *i.e.* there should be no serious straining of the structure or of the riveting when the armor is perforated.

The construction will be simplified by arranging the armor in one strake extending from the lower to the upper armor deck as on fig. 114. The plates will in such a case be rather narrow but of great height.

Vertical frames are not only well adapted to stiffen the plating behind armor, but will also assist it in its task of carrying the weight of the armor and transmitting it to the structure below the armor shelf. As explained in SECTION **10**, *r*, the plating together with its framing forms a deep girder which rests essentially on the rigid points of support formed by the transverse bulkheads in the hold.

9. **Framing behind the Unprotected Side above the Level of the Armor Shelf.**—The sides are here everywhere supported by decks, whence a pure transverse system of framing is the most efficient.

10. **Local Reinforcements.**—Extra framing should be worked, as needed, in places where great weights are concentrated, as for instance in the double bottom under heavy gun turrets and under the main engines. Such extra framing, when transverse, is best worked as intermediate frames on the half frame spaces. Extra longitudinal frames, worked under the main engines, should be located with due regard to the engine bed-plates so as to support the engines as directly as possible. In the bow, especially in light and fast vessels, so-called "panting stringers" are worked in order to support the plating when the ship is steaming against a head sea.

11. **Framing Systems of Smaller Ships.**—*Light cruisers.*—In light cruisers the double bottom rarely extends beyond the end of the ammunition rooms forward and aft, and transversely it generally stops at the turn of the bilges. Within the double bottom, the mixed longitudinal system should prevail as in battleships and battle-cruisers. Beyond the double bottom, the vertical keel should be carried continuously to the stem and sternpost, and the other longitudinals should be continued in the form of semi-intercostals as near to the ends as they can be conveniently worked.

On the sides below the protective deck the transverse system is the most efficient on account of the strongly curved form of the bottom.

Transverse frames will here act as arches, spanning the short distance between the margin of the double bottom and the protective deck. Longitudinal girders should be fitted in sufficient measure to stiffen the plating between the transverse frames (fig. 116).

Above the protective deck the framing should likewise be essentially transverse. In fact, the transverse system should prevail everywhere except in the double bottom.

Torpedo-vessels.—Torpedo-vessels have no double bottom, the bulkheads are spaced widely apart, and the shell plating is very light, whence the framing acquires greater importance than in other vessels.

The longitudinal strains are liable to be very great not only amidships but also in the fore body. Thus the bottom plating is known to have buckled in some boats under the conning-tower as a result of driving up against the sea. Transverse strains also are liable to be considerable, due to the scarcity of bulkheads and the violent motion of these vessels in a seaway.

It follows from these conditions that the longitudinal frames should be well developed in the bottom, comprising always a continuous vertical keel extending from end to end, and one or more continuous longitudinals on each side according to the size and proportions of the vessel. The transverse frames should possess great stiffness and be continuous all round the contour of the section (fig. 117).

The stiffening of the shell plating to prevent buckling and wrinkling is of more importance here than in other ships, whence the shell should everywhere be well stiffened by longitudinal girders.

Submarine boats.—The chief function of the frames is here to support the shell under the great external water pressures, and since the transverse bulkheads are spaced far apart, this end is best attained by a purely transverse system of frames. A semi-intercostal vertical keel should, however, be fitted and also, where necessary, other longitudinal girders, depending on the size and proportions of the boat. Locally, longitudinal girders should be worked on the sides of hatches, domes, conning tower, and in other places where frames are cut. Such girders must either be supported by pillars or possess sufficient strength to span the distance between the bulkheads.

48. SPACING OF THE FRAMES.

1. **Fundamental Conditions.**—(1) *Water pressures.*—In order that the plating shall be able to support the pressures of the water without undue overstraining, μ the ratio between the shorter frame space

and the thickness of the plating must not exceed a certain value, depending somewhat on r the ratio between the longer and the shorter frame space. With the present frame spacings, μ varies from about 70 in large ships to 80 in cruisers and destroyers, which values may be considered satisfactory as long as the draught is not very different from that of existing ships. As far as this action is concerned, it is indifferent which set of frames, the transverse or the longitudinal, has the shortest spacing. Also in the choice of r the designer has considerable freedom, since within ordinary limits its influence on the strength of the plating is only of secondary importance.

(2) *Simple tension and compression.*—The fore-and-aft members of the bottom structure must possess a certain sectional area of metal, sufficient to resist the tensile and compressive stresses when the ship is subject to longitudinal bending. This may be attained with practically any spacing of the longitudinals by simply adjusting the thickness of the plating in the inner and outer shell, but the best result is probably attained by spacing the longitudinals at a distance from one another of about eighty times the thickness of the outside plating. With this spacing buckling of the outer shell will be ordinarily prevented (SECTION 5, *9* and *10*). A still smaller spacing will rarely be desirable for other reasons.

(3) *Wrinkling.*—As explained in SECTION 7, wrinkling is precluded when the longitudinals are spaced so as to prevent buckling.

2. **Present and Past Practice.**—In order to fix our ideas, we shall confine the discussion to three types of warships, representing recent practice, viz. a battleship of 30,000 ts., a light cruiser of 5000 ts., and a destroyer of 1000 ts. displacement. The principal dimensions and certain other characteristics of these type ships as now generally constructed are found in Table XV., p. 240, where also the same data for ships from the early nineties are given for comparison. Submarine boats will be dealt with separately afterwards.

An inspection of this table brings out in a quantitative manner the well-known fact that a marked advance in size, length, speed, and ratio length to depth has taken place during the last twenty years in all classes of warships. Hence the claims to longitudinal strength are much greater now than formerly. The mere increase in size, not being accompanied by a corresponding increase in thickness of the bottom plating, necessitates a more strongly developed framing system or else a more rational distribution of the material. Excepting the destroyers, the requirements of transverse strength have not advanced at the same rate as those of longitudinal strength, since the beam has not increased in the same proportion as the length and the spacing of the transverse bulkheads has remained unaltered or has been reduced. In spite of the relatively much greater claims to longitudinal strength, the frame spacing which was in use about twenty years ago has remained practically unchanged, and it appears, therefore, that the time has come for a careful scrutiny and revision of this feature. Since conditions differ greatly in the various types of ships, we shall discuss each one separately.

TABLE XV.

	Battleship.		Light Cruiser.		Destroyer.	
	1890–95.	Recent.	1890–95.	Recent.	1890–95.	Recent.
Displacement . . . . . tons	14,000	30,000	3500	5000	300	1000
Length . . . . . . feet	380	650	300	450	200	280
Breadth . . . . . . „	75	95	43	48	20	27
Draught . . . . . . „	27	28	16·5	15·5	7	9
Ratio: length to depth . . .	8·4	14·4	12·0	14·3	14·8	17·5
Speed . . . . . knots	18	23	20	26	30	34
Spacing of transverse frames . feet	4	4	3	3	1·75	1·75
Spacing of longitudinal frames . „	6–8	6–8	6–7	6–7	...	3–4
Spacing of transverse bulkheads in boiler rooms . . . „	35–40	30–35	30–35	30–35	20–35	35–40

3. **Battleships.**—Dealing first with the lower part of the bottom, we find in existing ships a spacing of 4 ft. for the transverse frames and from 6 to 8 ft. for the longitudinals. Since incipient buckling of the outer shell is probably not thus prevented, it is proposed, in accordance with the recommendations in *Article 1* (2), to reduce the spacing of the longitudinals to about eighty times the thickness of the outside plating, that is, about 4 ft. in vessels of the battleship class. The effectiveness of the inner bottom plating will be hereby much increased, although buckling may still occur. The spacing of the transverse frames may be now safely increased in the same ratio as that in which the spacing of the longitudinals is reduced, since the plating will then receive the same support against water pressures as at present. The weight of the framing system will by this reversion of the existing frame spacing remain practically unaltered, but the strength of the fore-and-aft members of the bottom will be increased, and the effectiveness of the outer and inner shell plating as longitudinal strength members will be much improved. It might be objected that the transverse strength will be inadequate because the number of transverse frames is reduced. General

transverse strength is, however, provided for by the bulkheads, and local transverse strength may be secured, if necessary, by abandoning altogether the bracket floors now extensively used, employing instead solid plate floors, which possess greater girder strength. It must be borne in mind also that the number of watertight floors, which are the strongest transverse frames, will not be reduced.

Light transverse stiffening bars will probably have to be fitted to the inner bottom plating and on the longitudinals intermediate between

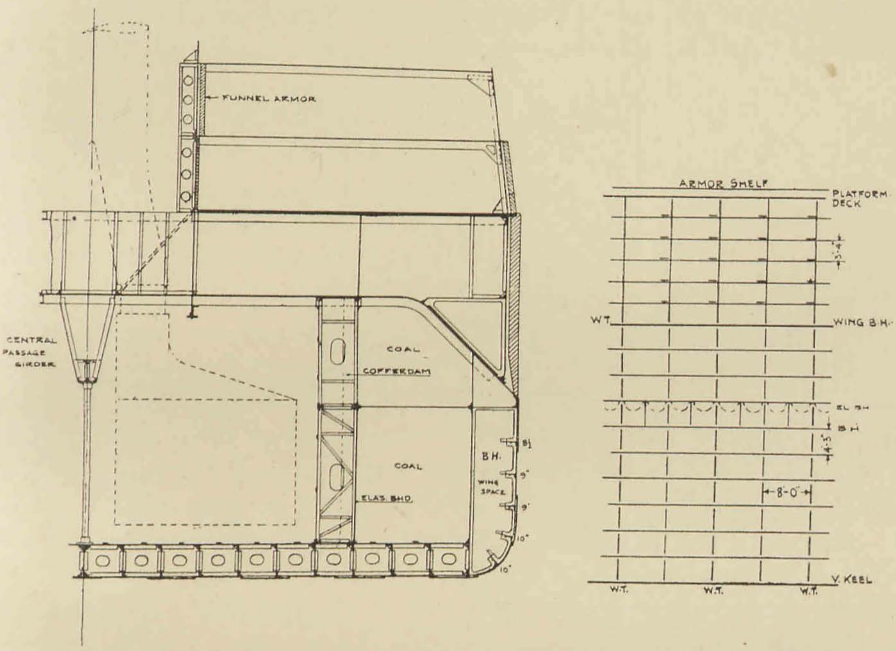


FIG. 115.—Midship Section of Battleship, showing proposed Framing System, Elastic Bulkhead, and Support for Funnel Armor and Side Armor.

the floors, but a corresponding saving in weight can be effected by a slight reduction in the thickness of the inner and outer shell, leaving the total weight of the bottom structure unaltered.

The change here suggested is illustrated in fig. 115, where the longitudinals in the bottom are spaced about 4 ft., the transverse frames 8 ft. apart.

On the sides of the ship below the armor deck we have to consider primarily the function of the frames as individual girders, longitudinal strength being here of smaller importance. Where an inner bottom is fitted on the sides, the present arrangement of the frames is satisfactory, but it seems preferable in point of safety against submarine attack to substitute a wing bulkhead for the inner bottom, and to subdivide the wing space by watertight bulkheads fitted 8 ft. apart in continuation of the transverse frames in the double bottom. These bulkheads will provide ample transverse strength of the sides, and it remains only to support and stiffen the intervening plating of the outer shell.* This seems to be best attained by fitting a system of longitudinal girders consisting of deep channels or bulbs, spaced from 3 to 4 ft. apart, and carried continuously through the transverse wing bulkheads, to which they should be efficiently connected by brackets (fig. 115 and *Example 2*, SECTION 70, *2*).

Above the protective deck there appears no reason to deviate from the present practice of frame spacing.

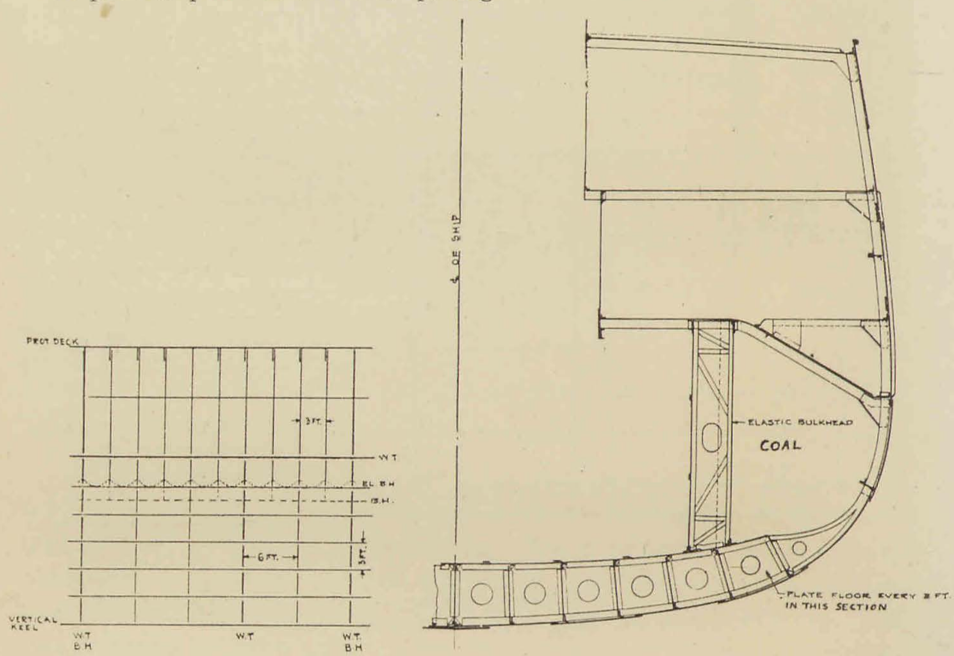


FIG. 116.—Proposed Framing System of Light Cruiser.

4. **Light Cruisers** (fig. 116).—In existing ships of this class and of the size here under consideration the transverse frame space is usually about 3 ft. The longitudinal frame space is more varying but is commonly from 6 to 7 ft. The arguments which led to a reversal of the transverse and longitudinal frame space in the bottom of battleships will again in light cruisers lead to the same result. It is, therefore, proposed within the double bottom to adopt a transverse frame space of from 5 to 7 ft. and a longitudinal frame space of 3 ft. In the

* The longitudinal wing bulkhead should be stiffened as usual by vertical stiffeners between the transverse bulkheads. These stiffeners, which should be fitted on the inside, are not shown in fig. 115.

arrangement suggested on fig. 116, the transverse frame space is 6 ft. A spacing of 3 ft. for the longitudinals will again correspond to about eighty times the thickness of the outside plating, which should be sufficient to support the water pressures and to prevent buckling. On the sides, outside the double bottom, the frame spacing should remain as at present, the transverse frames being placed 3 ft. apart and the longitudinal semi-intercostal girders about half a deck height apart or midway between the decks.

5. **Destroyers.**—In destroyers, conditions are somewhat different. The beam has increased at about the same rate as the length, the transverse bulkheads are generally farther apart than formerly, and in many boats no longitudinal bulkheads are fitted. Hence, as stated above, the framing system should provide not only great longitudinal but also great transverse strength. Moreover, the shell must be very carefully stiffened against buckling and wrinkling, necessitating a close spacing of the longitudinal stiffeners both in the bottom and on the sides.

In most of existing destroyers the fulfilment of these requirements does not appear to be attained in a rational way (fig. 137). The transverse frames are closely spaced, about 21 in. in boats of 1000 ts. displacement, and consist essentially of light bars, but every fifth or sixth frame is of greater depth and strength. The floors are very deep. Evidently a smaller number of deep frames could be substituted with advantage for the light bars, since the same moment of inertia could be thus obtained on a smaller weight, the only condition being that the shell must be stiffened sufficiently to enable it to transmit the load to the frames. The spacing of the longitudinals is very erratic in existing boats, although there is of late a tendency towards a more systematic distribution of these girders. The longitudinal strength as well as the longitudinal stiffening of the shell is everywhere attained only with a great expenditure of weight. Excepting the vertical keel, all other longitudinals are intercostal. Those in the bottom have to bridge over the deep floors and must, therefore, in order to get connection with the bottom, be excessively deep and heavy, but also the longitudinals on the sides are relatively heavy because they have to pass inside the frames. It must be borne in mind that the longitudinals on the sides, unless they are excessively deep, are quite incapable of transmitting the load of the water pressures from bulkhead to bulkhead. Their functions are, therefore, first, to transmit the load of the water pressures to the transverse frames, and second, to stiffen the shell against buckling and wrinkling. It is clear that even with a fairly wide frame spacing, longitudinals of very light scantlings will be sufficient for these purposes, and, as far as buckling is concerned, a greater number of light, closely spaced stiffeners are preferable to a smaller number of deep stiffeners.

Based on these considerations the framing system illustrated in fig. 117 is proposed, where, again, the frame spacing now commonly used is practically reversed.

The longitudinals are everywhere spaced about 21 in. apart, which is sufficiently close for the thicknesses of plating suitable for boats of this

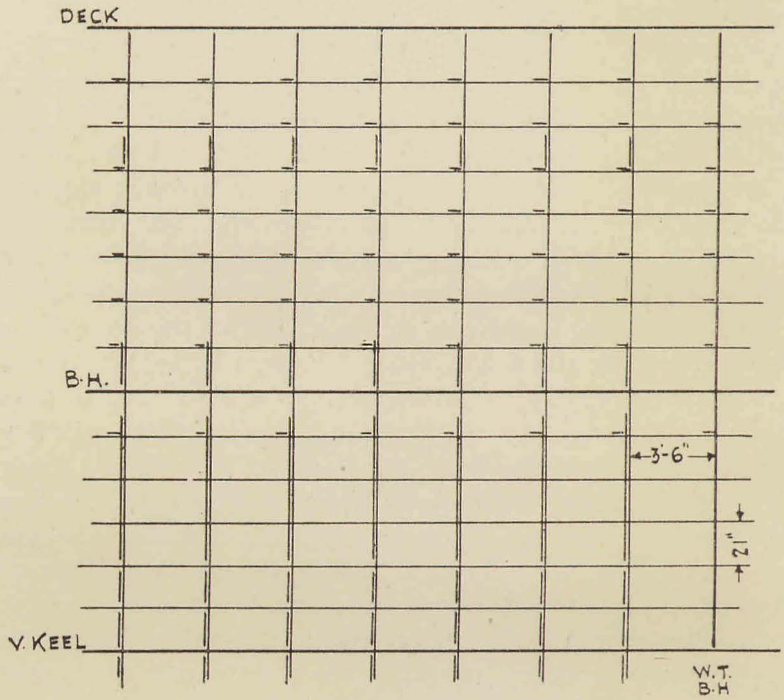


FIG. 117A.—Proposed Framing System of Destroyer.

size. Where the plating is heavier than ¼-in., as in the keel and sheer strakes, the spacing is somewhat greater. The longitudinals in the bottom below the turn of the bilges are continuous throughout or provided with deep brackets at the bulkheads and are constructed of plates and angles. All other longitudinals are light bars, continuous between the transverse bulkheads, to which as well as to the transverse frames they are connected by brackets. The transverse frames are spaced 3 ft. 6 in. apart and are continuous all around the contour of the vessel. They run unbroken across the top of the vertical keel and the longitudinals in the bottom, but on the sides and under the deck they are slotted to allow the light continuous longitudinals to pass through. Their strength is calculated so as to replace the transverse frames now ordinarily employed, providing the same aggregate moment of inertia. The details of the construction will be further discussed in the next Section.

Recapitulating: the proposed changes in the framing of destroyers are based on the following arguments:—(1) Greater claims to longitudinal

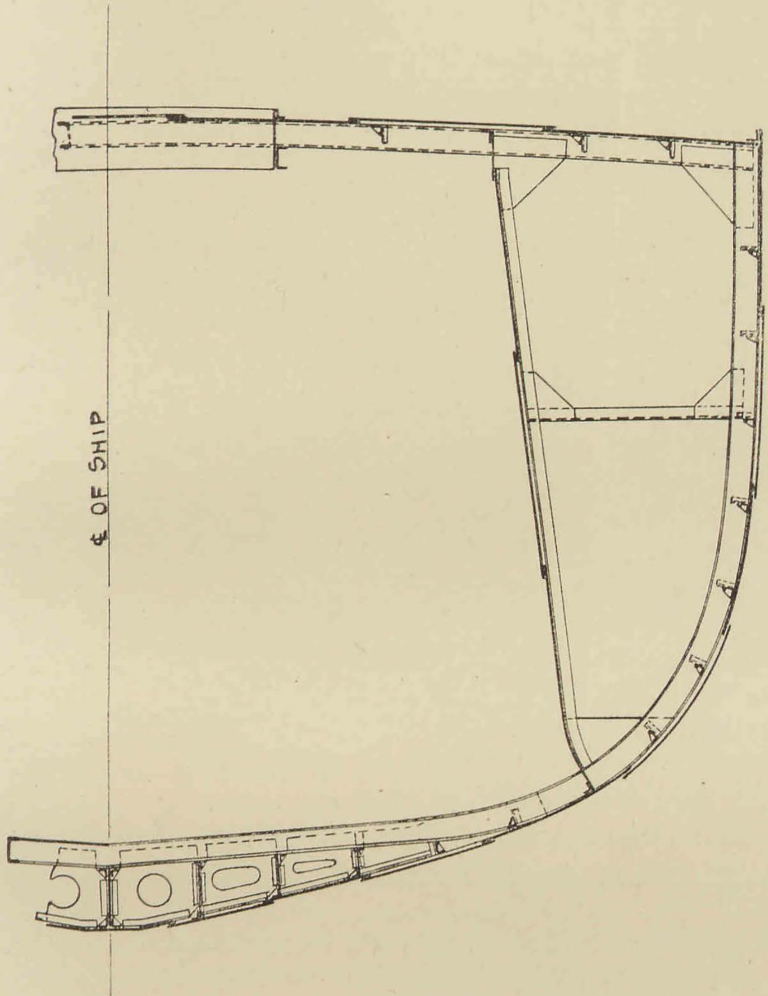


FIG. 117B.—Proposed Framing System of Destroyer.

strength necessitate a stronger development of the longitudinal frames in the bottom. (2) Transverse strength may be obtained more efficiently by a smaller number of deep frames than by a greater number of light frames. (3) Buckling of the thin shell plating is best prevented by closely spaced light stiffeners, rather than by a few girders of great weight.

6. **Submarine Boats.**—The spacing of the frames in submarine boats depends exclusively on the requirements as to depth of immersion.

It may be determined by means of Curve B, Pl. IV., in conjunction with the thickness of the plating and the stipulated pressure head.

49. CONSTRUCTION OF LONGITUDINAL FRAMES.

We shall here describe and discuss the longitudinals as they are usually constructed in modern large warships, continuous within the double bottom and partly or entirely intercostal without. Where longitudinals within the double bottom are intercostal, they are constructed like the floors, described in the next Section. The central longitudinal is generally referred to as the "vertical keel." The other longitudinal frames are referred to simply as "longitudinals" and are numbered, beginning with number one for the longitudinals nearest the keel on either side. The longitudinal on which the side armor rests is called the "armor-shelf."

1. **Strength Calculations.**—In deciding upon the construction of continuous longitudinals, they must be considered first, as members of the general hull structure, and second, as individual girders. Finally, the local strength under concentrated loads must be examined.

(1) As members of the general structure, the longitudinals will be subject to simple tension and compression in direction of their length, and the stresses are determined by the longitudinal strength calculation. These stresses, however, hold good only for the gross sectional area of the longitudinals; at the intersections with the transverse frames, at the butts, and where holes are cut in the plates, the stresses will be increased. The following table gives, by way of illustration, the efficiency in tension at various sections of the longitudinal, the girder strength of which was calculated in the *Example*, SECTION 16 :—

TABLE XVI.—EFFICIENCIES AT VARIOUS SECTIONS OF A LONGITUDINAL.
(See Pl. III.)

Section.	Mode of Fracture.	Efficiency.
AB	Through rivet holes at bulkheads or watertight frames, including tearing of upper angle.	·76
CD	Through butt lap. Shearing of the rivets, tearing of upper and lower angle.	·81
EF	Through notch for transverse frame bar and rivet holes at non-watertight frame. Tearing of upper angle, shearing of one rivet in way of notch.	·76
GH	Through lightening hole, including tearing of both angles.	·72

The efficiencies are obtained by dividing the net sectional area of the longitudinal, including the plate and the angles that are continuous through the section, with the gross sectional area of the plate and the upper angle. If the lower angle were continuous through the watertight frames, the efficiency would at these points be ·84 instead of ·76.

In compression as in tension it is probably safe to include also the intercostal angle as effective in the sectional area through a lightening hole (fig. 7). Buckling is not likely to occur in longitudinals.

(2) The strength of a longitudinal, considered as an individual girder, is calculated as described in SECTION **16**, where it is illustrated by the example just referred to. It is seen that the strength in that particular case is fairly uniform, and that the stress does not generally exceed from 5 to 6 ts. per sq. in. except at the bulkheads, where it reaches 11·5 ts. per sq. in. on account of the great bending moment at these points. Similar conditions exist quite frequently in boiler-rooms, whence it is desirable in such compartments to reinforce the longitudinals where they cross the bulkheads. This may be accomplished by placing a bracketed bulkhead stiffener on each of the longitudinals, or by fitting a bracket against each side of the bulkhead similar to those on the hold stringers of merchant vessels. Such brackets will serve, at the same time, to distribute the reactions between the bulkheads and the bottom structure over larger areas.

The strength of semi-intercostal hold stringers, constructed as indicated on figs. 34 and 127, may be calculated as if they were continuous. When continuous girders are worked on the sides of the ship, as in the framing system recommended in last Section for a battleship, their strength may be determined from the Bulkhead Tables, given and explained in Chapter XVI., the scantlings being calculated and selected by a process similar to that described for stiffeners of deep water tanks. (See *Example* in SECTION **69**.)

(3) Local, concentrated loads were dealt with in SECTION **22**, where it was explained how to calculate the strength of the longitudinals by considering the direct compressive stress at the point of application of the force, and the resistance to wrinkling elsewhere in the girder. The most important case of concentrated loading of longitudinals arises when a ship is in dock, when also the girder strength of the longitudinals over the docking keels may be called into play. We shall deal separately with this problem (see also SECTION **10**).

2. **Strength of the Longitudinals on which a Ship is Resting when in Dock. The Central Passage.**—Warships are docked either

on the center keel only or on the center keel together with one or two side keels on each side.

When a ship is docked on the center keel only, we may reckon the pressure on the keel blocks to be fairly uniformly distributed along the length of the line of support. Only at the middle portion the pressures will probably exceed the mean value by from ten to twenty per cent. A rough calculation will show that the keel, where no center-line bulkhead is fitted, is quite incapable, by itself, of transmitting the reactions to the transverse bulkheads. In order to reinforce or assist the keel and transmit to it the weights from the upper part of the ship, a deep box girder, usually a "central passage," is in large warships fitted under the armor deck, connected with the keel by a row of pillars. A central passage girder is shown in section in fig. 115 and is given in elevation in fig. 118. The flaring section is proposed in order to give the girder lateral stiffness, avoiding thus internal diagonal bracing. The keel and the central passage will have the same deflections, but will otherwise act as two independent girders, each carrying a share of the load proportional to its moment of inertia. The strength of these girders is calculated as explained for the longitudinals, only the reaction of the keel blocks takes the place of the forces of buoyancy.

Particular attention should be given to the rivets connecting the vertical keel plates to the lower keel angles in large vessels. Unless special precautions are taken or special reinforcements are fitted, the rivets immediately over the keel blocks are likely to be overstrained. This action should be relieved by faying the vertical keel plates as closely as possible upon the flat keel and by fitting intermediate brackets half-way between the floors. Such brackets will serve also to stiffen the vertical keel under the strong compressive stresses which exist when a ship is so placed in the dock that the keel blocks fall between the frames. Ordinarily the blocks are 16 in. wide and are spaced 4 ft. from center to center, but for very large ships it appears desirable to make the blocks wider, preferably about 24 in., and the ship should be placed in such a position in the dock that either the frames or the brackets fall on the blocks.

When a ship is docked on side keels as well as on center keel the pressure on the center keel is much reduced. It is, however, advisable to apply the same means of reinforcement to the vertical keel as in ships docked on the center keel only, for in such large ships to which side keels are fitted, the load on the center keel is still much greater than it can safely carry without special reinforcements. Each of the longitudinals immediately over the side keels will probably have to support about three-eighths of the total weight of the ship and should, like the vertical keel, be provided with double angles, intermediate brackets, and be made to fay closely against the outside plating. They should be somewhat heavier than the other longitudinals.

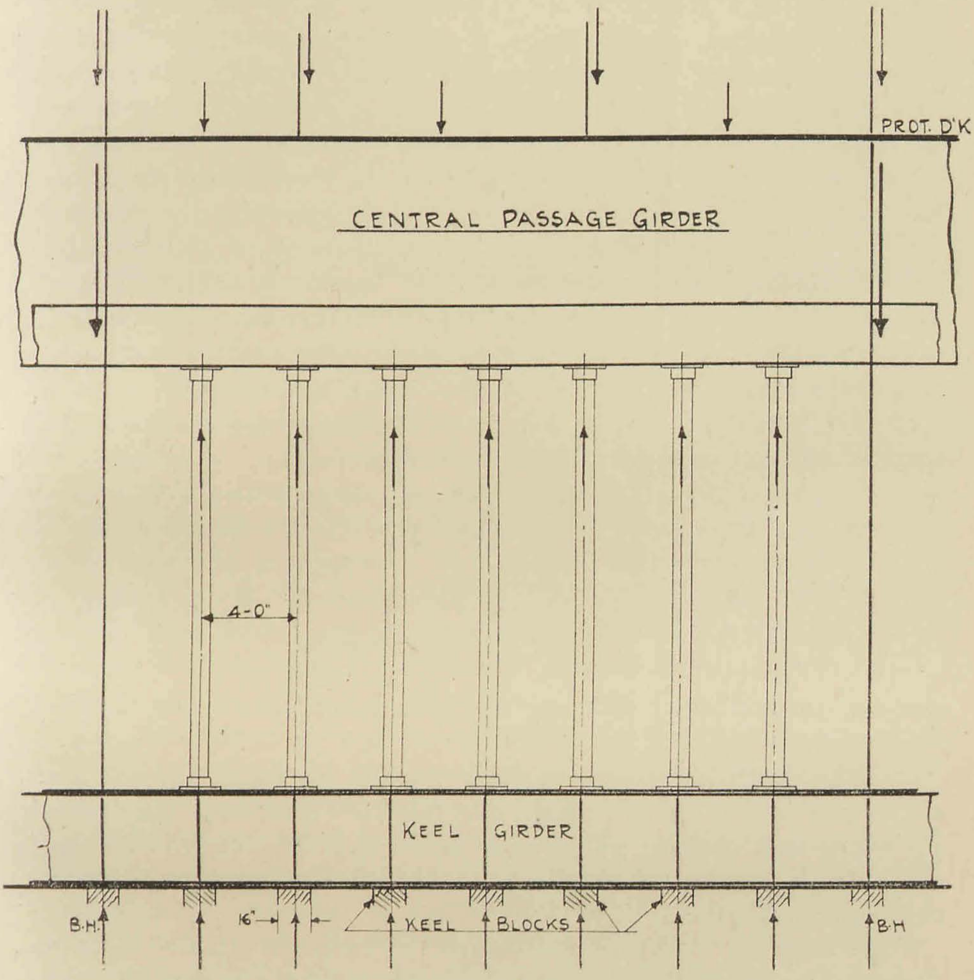


FIG. 118.

3. **Example.**—A ship of 12,000 ts. displacement is docked on the center keel only. The mean pressure on the keel is 3·5 ts. per inch run. The keel blocks are spaced 4 ft. apart from center to center and are 16 in. wide in longitudinal direction. The vertical keel is constructed as shown on the accompanying sketch. The height of the vertical keel plate between the angles is 32 in., its thickness is $\frac{5}{8}$ in. The keel angles are connected to the vertical keel plate by $\frac{7}{8}$-in. rivets, spaced

5 diameters apart. The vertical keel is stiffened by transverse floors, which are spaced 4 ft. apart, and by intermediate brackets. Each of the floor plates and brackets is connected to the vertical keel by six $\frac{7}{8}$-in. rivets. Suppose first that the ship is placed with the frames or brackets directly on the blocks, and let us determine the shearing stress on the rivets in the vertical flanges of the lower keel angles.

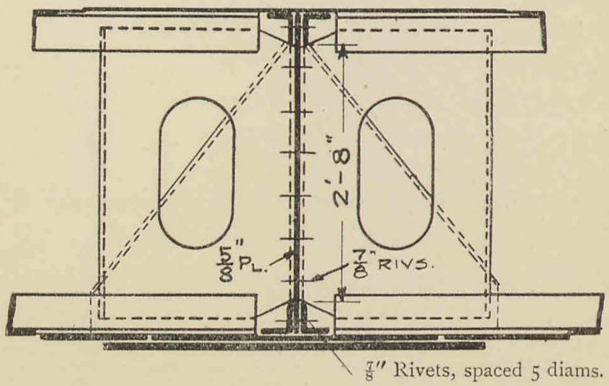


FIG. 119.

The total reaction of each keel block is $48 \times 3 \cdot 5 = 168$ ts., which we shall assume to be taken by the rivets on 24-in. length of keel angles together with the rivets in the floors or brackets. The total rivet area is:

$$1 \cdot 75 \times \frac{\pi}{4}\left(\frac{7}{8}\right)^2 (1 \cdot 07)^2 \left[\frac{24}{5 \times \frac{7}{8}} + 6\right] = 13 \cdot 8 \text{ sq. in.}$$

whence the stress on the rivets is:

$$p_s = \frac{168}{13 \cdot 8} = 12 \cdot 2 \text{ ts. per sq. in.}$$

showing the desirability of fitting the vertical keel plate closely against the flat keel.

Suppose next that no intermediate brackets are fitted, and that the ship is so placed on the blocks that they fall midway between the frames. The vertical keel plate will then be under a great compressive stress and liable to buckle. Let us examine the relation between the actual and the critical load.

An elemental vertical strip midway between the frames is at a distance from these of about 40 times the thickness of the plate and may be regarded as an independent column. The sectional area of this column is $\frac{5}{8}$ sq. in. and the load on it is roughly $\frac{168}{16} = 10 \cdot 5$ ts. or $16 \cdot 8$ ts. per sq. in. We have $\mu = \frac{32 \times 8}{5} = 51 \cdot 2$ whence from the curve for Euler's formula (fig. 49) we find the critical load $p = 16 \cdot 5$ ts. per sq. in. It appears then that the critical load will be reached or passed, and that buckling will take place.

4. **Direction of Plane of Longitudinals.**—Whether we regard the longitudinals as supporting or as stiffening members, they will be most effective when placed normal to the surface of the plating, since they will otherwise be subject to a tripping tendency. Hence *the longitudinals should be placed as nearly as possible normal both to the inner and outer shell*, except where special reasons make deviation from this rule desirable. Such is the case where longitudinals serve as a shovelling flat in coal bunkers, or where they are made to merge into breasthooks, bulkheads, flats, or decks. Where a local twist of a longitudinal occurs, a reinforcement should be fitted, which may consist of an auxiliary longitudinal girder worked between floors normal to the longitudinal, and connected to it by angles at about mid-depth.

5. **Principal Features of Construction.** — Within the double bottom the longitudinals are constructed of plates and angles. Generally, the plates are of the same length as in the outer shell, being limited by the same conditions as to handling and tools. With this length it is easy to secure a good shift of butts with adjacent strakes of the inner and outer bottom. The depth of the longitudinals is the same as that of the double bottom, generally 3 ft. to 3 ft. 6 in., but in some ships it is as much as 5 ft. In the smallest vessels in which a double bottom is fitted, the depth is reduced to the minimum consistent with the claims to working inside the double bottom. A depth of 2 ft. may be considered as the limit below which it is not advisable to go, although a depth of 18 in. is found at the margin of the double bottom in certain gunboats.

The thickness of the vertical keel and of the longitudinals in way of docking keels should be not less than the standard thickness of the outside plating; it is in some ships slightly greater. Other longitudinals in the bottom, below and including the turn of the bilges, may be somewhat lighter than the shell plating with which they are connected. In deciding their thickness, the general longitudinal strength of the ship should be taken into consideration. The longitudinals on the sides are in many ships still lighter than those in the bottom. In determining the thickness of side longitudinals, their function as individual girders should be primarily considered. The armor shelf should be at least of the standard thickness of the outside plating.

6. **The Angles of the Longitudinals.**—The angles serve to connect the plates of the longitudinals to the inner and outer shell and contribute at the same time to the flange strength. Ordinarily the angles are single, but double angles, both inner and outer, should be fitted on the vertical keel and on the longitudinals over docking keels, as

explained above. Double angles should be fitted also on oiltight longitudinals in order to secure greater rigidity and in order to permit calking on both sides.

The angles, or at least the upper one, are generally continuous and should be as long as can be worked conveniently. Their butts should make a good shift with the other elements of the longitudinals. Where the outer angles of the transverse frames are continuous, it is necessary to slot the longitudinals over them and to make the outer longitudinal angles intercostal. At watertight and oiltight transverse frames both angles of the longitudinals should be continuous, providing thus a much-needed reinforcement at these weakened sections (fig. 123B).

Where the longitudinals do not stand normal to the inner or outer shell, the angle bars, as a general rule, should be so placed as to have an "open bevel," *i.e.* so that the angle between their flanges is obtuse. By a "close bevel" the work of bevelling, punching of rivet holes, and riveting is rendered difficult, and the bar is more weakened than by an open bevel. In watertight longitudinals both the inner and outer angles should be placed on the calking side.

7. **Holes in Longitudinals.**—Holes must be cut in the non-watertight longitudinals in such locations and of such size as to permit access

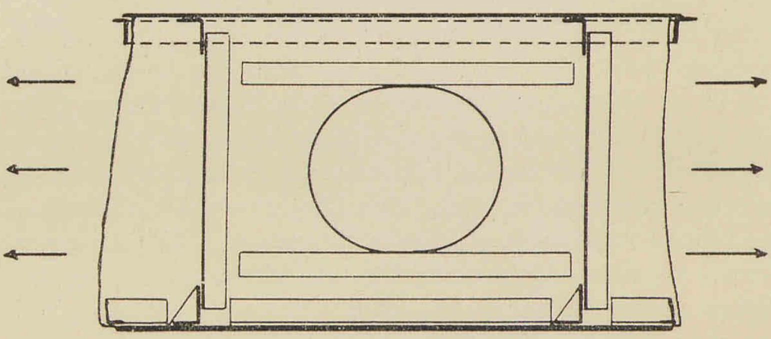


FIG. 120.

to all parts of the double bottom. By cutting holes weight may be saved and the resilience of the longitudinals as members of the ship-girder is increased. In general, holes are cut in every frame space except where butts occur in the same frame space in adjacent inner or outer shell plates or in the longitudinal itself.

Great caution should be exerted in locating and cutting these holes, which should not, with the present depth and spacing of the frames, be larger than strictly necessary for access. In many cases it is desirable to compensate for the reduction in strength, a question which must be decided through experience guided by a calculation of the maximum stresses. The weakening caused by the holes may be classed under three heads :

(1) The holes reduce the cross-sectional area of metal of the girder and, therefore, reduce the strength in simple lengthwise tension or compression. If the holes are too large in a direction normal to the longitudinal, the strength will fall below that at the crossings of the floors, and reinforcements should be introduced. The reinforcements may consist of bars or plate bands fitted above and below the hole close to the edge, as indicated in fig. 120.

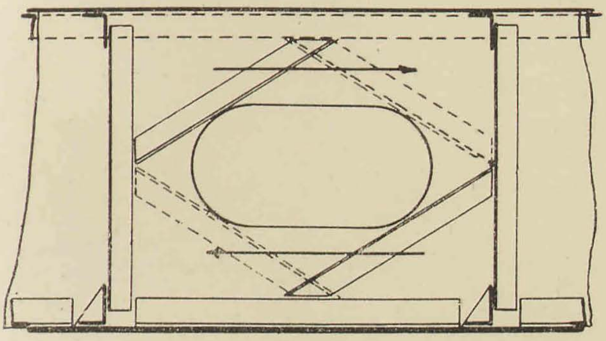


FIG. 121.

In the example given in SECTION 16, the lightening holes are somewhat too large, the efficiency of the girder in tension at sections across the holes being only ·72 as compared with ·76 at the non-watertight floors (Table XVI.).

(2) The holes reduce the web strength of the longitudinal considered as an independent girder. If the holes are very long, even if of small height, diagonal bars should be fitted as shown in fig. 121, reinforcing the edges of the holes at the corners and running approximately in direction of the principal stresses at these points.

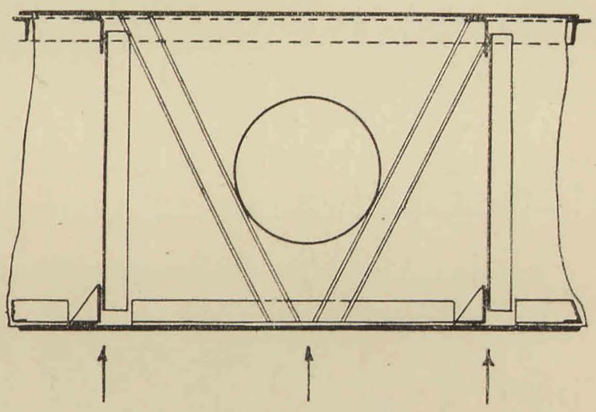


FIG. 122.

(3) The holes reduce the crosswise stiffness of the plate and hence its power to carry concentrated vertical loads, as for instance the reactions of the keel blocks when the ship is in dock. In longitudinals fitted over docking keels the holes, if any, should, therefore, be as small as possible. If necessary, diagonal reinforcing bars should be fitted, applied as in fig. 122.

Drain holes—so-called "limber holes"—and "air holes" should be cut as shown in figs. 123A and 124A.

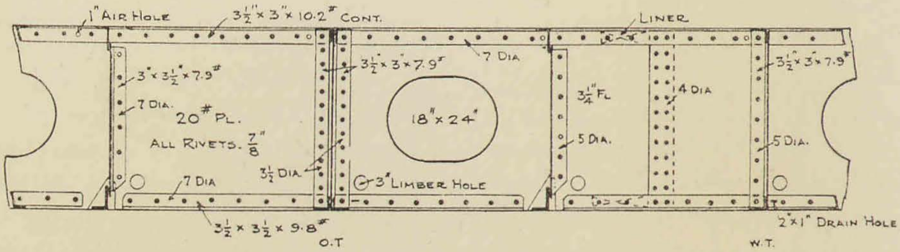


FIG. 123A.—Non-Watertight Longitudinal (U.S. Navy).

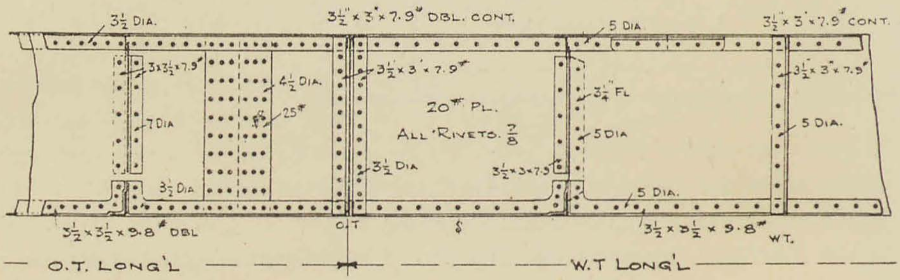


FIG. 123B.—Oiltight and Watertight Longitudinal (U.S. Navy).

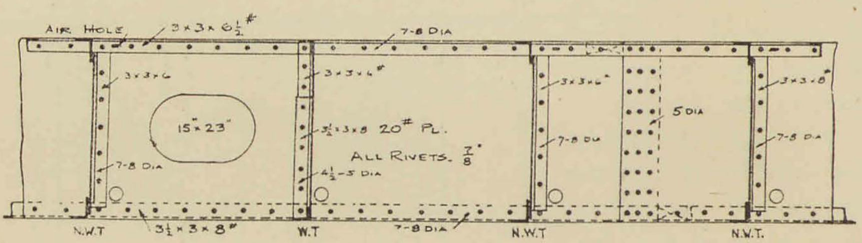


FIG. 124A.—Non-Watertight Longitudinal (British Navy).

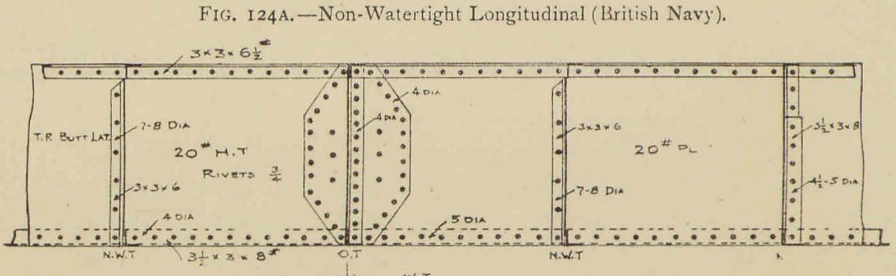
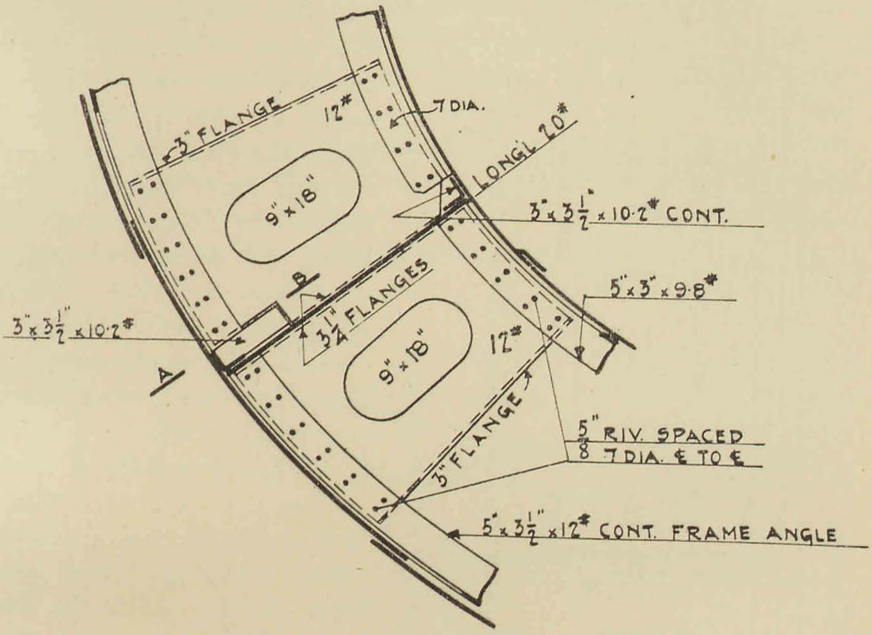


FIG. 124B.—Oiltight and Watertight Longitudinal (British Navy).

8. **Butt Connections.**—The strength of the butt connections—*i.e.* their efficiency—should not fall below that at the lines of intersection with the transverse floors. Generally, double-riveted lapped butts will

be sufficient, as appears from Table XVI., but in some navies treble riveted laps are used. The vertical keel, where it is non-watertight, is



Section A–B.

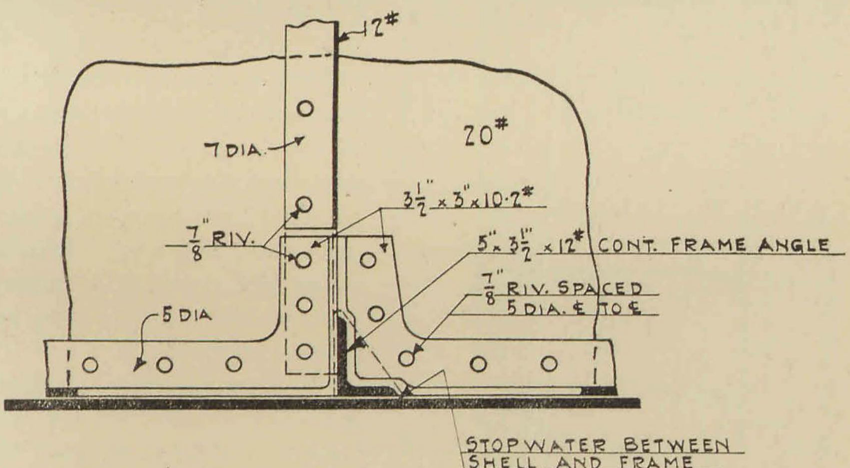


FIG. 125.—Intersection between Watertight Longitudinal and Transverse Floor.

often given double straps, treble riveted, but where it is watertight or oiltight, single straps or overlaps should be used. Other watertight or oiltight longitudinals should likewise have either single butt straps or overlaps—treble riveted in order to secure stiffness. Single straps should be fitted on the non-calked side, and where both sides are calked, as in oiltight work, the straps should either extend the full width of the plates with the bounding angle stapled over them, or the ends should be chamfered and inserted under the angles in order to cover the point where the end of the strap crosses the butt line. This difficulty does not exist in overlapped butts, which may as well be calked on one side as the other. Overlaps, which are simpler, lighter, and cheaper than single straps, seem indeed preferable for watertight and oiltight work.

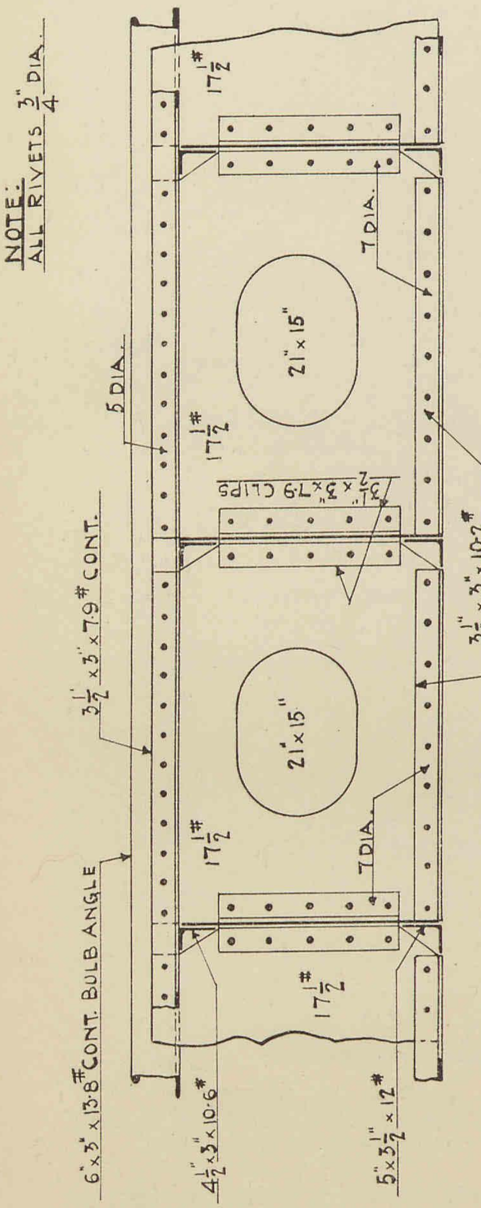


Fig. 126.—Intercostal Longitudinal in Way of Deep Floors.

9. **Construction of Details.**—Figs. 123A and 123B show the typical construction of non-watertight, watertight, and oiltight longitudinals within the double bottom in ships of the United States Navy. It will be noticed, that where a watertight longitudinal meets a non-watertight transverse frame, the outer angle of the longitudinal is intercostal and stapled in order not to break the continuity of the frame bar (fig. 125), but where it meets a watertight or oiltight frame, the outer angle is continuous, as explained in *Article 6.*

Figs. 124A and 124B show the construction used in the British Navy.* Where oiltight frames meet continuous longitudinals, enlarged liners are fitted as reinforcement.

10. **Longitudinals Outside the Double Bottom.**—When the longi-

* Prepared on the basis of various sketches and information given in E. L. Attwood, *War-Ships*, London, 1904; and N. J. McDermaid, *Shipyard Practice*, London, 1911.

tudinals emerge from the double bottom, their construction is generally modified. In the lower part of the bottom where the transverse floors are deep, the longitudinals are likewise of great depth, so as to stiffen the floors effectively. Ordinarily the longitudinals are here intercostal, although, as explained above, the first longitudinal may with advantage be continuous in long and fast vessels. In any case a continuous member—a bulb angle, channel, or simple angle bars—should be worked on the top of the floors so as to tie the different members of the framing structure together (fig. 126). The intercostal plates should be connected by clips to the outer shell and to the floors.

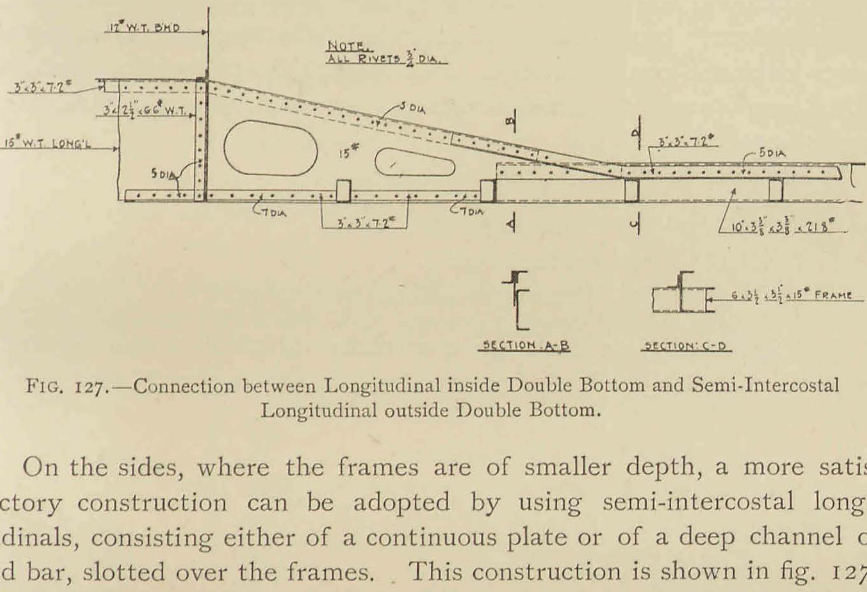


FIG. 127.—Connection between Longitudinal inside Double Bottom and Semi-Intercostal Longitudinal outside Double Bottom.

On the sides, where the frames are of smaller depth, a more satisfactory construction can be adopted by using semi-intercostal longitudinals, consisting either of a continuous plate or of a deep channel or zed bar, slotted over the frames. This construction is shown in fig. 127, which illustrates also the transition from the deep longitudinal inside the double bottom to the shallow girder outside.

Panting stringers should be of essentially the same semi-intercostal type as other hold stringers.

50. CONSTRUCTION OF TRANSVERSE FRAMES.

The transverse frames are generally placed with their web in transverse planes normal to the load water-line, and are numbered, beginning from the bow or the stern. Intermediate frames are given fractional numbers, as, for instance, $64\frac{1}{2}$, being the frame half-way between 64 and 65. The angles connected to the outer shell—the "frame-bars"—should face towards amidships in order to avoid a close bevel. The inner or "reversed bars" are given the same bevel as the outer so as to form a fair surface on which an inner bottom or ceiling can be fitted and to which longitudinal hold stringers can be attached. The same rule is followed for the flanges of frames consisting of simple channel or zed bars.

1. **Frames Within the Double Bottom.**—These frames are generally referred to as "floors," a term which has been handed down from the time of wooden ships, where it was applied to the frame timbers lying across the keel. In early iron ships it was applied to the lower deep part of the transverse frames, which were often likewise continuous across the keel, and the term is still so used in steel ships outside the double bottom and everywhere in ships where there is no double bottom. It has now been extended to include all the transverse frames inside the double bottom. Such floors are usually intercostal and will be here described. Continuous floors are constructed like continuous longitudinals.

Floors occur in three different varieties—

(1) Lightened plate floors.
(2) Bracket floors.
(3) Watertight and oiltight floors.

(1) *Lightened plate floors* are constructed of plates extending from longitudinal to longitudinal, lightened by holes which should be designed

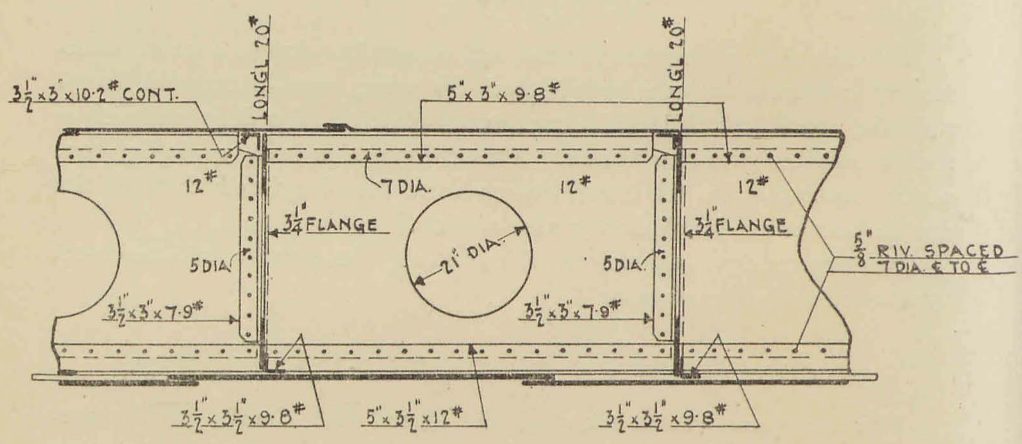


FIG. 128.—Lightened Plate Floor.

with due regard to stiffness and strength as explained for the longitudinals. This type of floor is used where heavy weights are resting on the inner bottom and where the structure is exposed to great dynamic forces as, for instance, under engines, gun-turrets, in way of shaft-struts and shaft-tubes, etc. They should be used also where a certain girder strength is required, and are for this reason recommended where the spacing of the transverse frames is very great, as proposed in SECTION 48.

The floor plates are connected to the longitudinals by flanges or by angle bars. Flanging is cheaper and lighter than angle bars, but does not give so great rigidity to the connection. Angle bars should therefore be used on the sides where the transverse frames help to carry the weight of the side armor and the structure is exposed to underwater explosions. In order to obtain the greatest possible girder strength, flanges as well as angle bars should face the same way in the different floor plates of each frame. This construction should be used, in particular, where the transverse frame space is very great.

The connection of the floors to the inner and outer shell should be effected by angle bars sufficiently deep to obtain an efficient riveted connection. In ships of the United States Navy the outer angle is always continuous while the inner is intercostal. In the British Navy both angles are intercostal.

(2) *Bracket floors* consist of brackets of light plating, two in each section, lightened by holes, and leaving a space between them sufficient

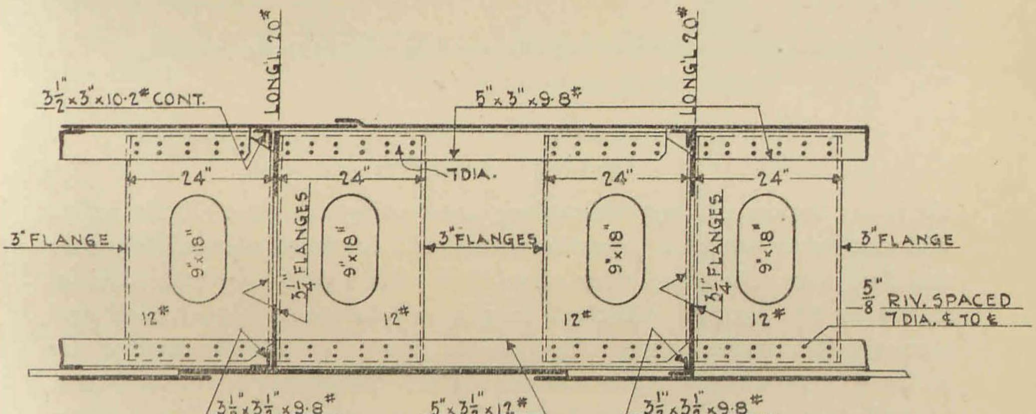


FIG. 129.—Bracket Floor.

for access—at least fifteen inches wide. They are connected to the outer and inner shell plating by angle bars—the main and reversed frames—which should be of sufficient depth to support the plating. The connection of the brackets to the longitudinals may be by flanges except where the longitudinals are oiltight, in which case angle clips should be employed to permit calking round the connection. The brackets should be flanged on the free edge so as to increase their stiffness. Bracket floors should be used where there is no special reason for employing the stronger plate floors and where watertightness or oiltightness is not required.

(3) *Watertight and oiltight floors* are built of solid plate and, if intercostal, staple angles are worked all around the contour as shown in fig. 130. The angles are always double in oiltight work and sometimes also in watertight work. In all such frames and, in particular, in oil-tight frames, it is of importance to secure great rigidity in order to preserve the efficiency of the calking under all conditions. When subject to great fluid pressures the floors will tend to bulge, pulling down

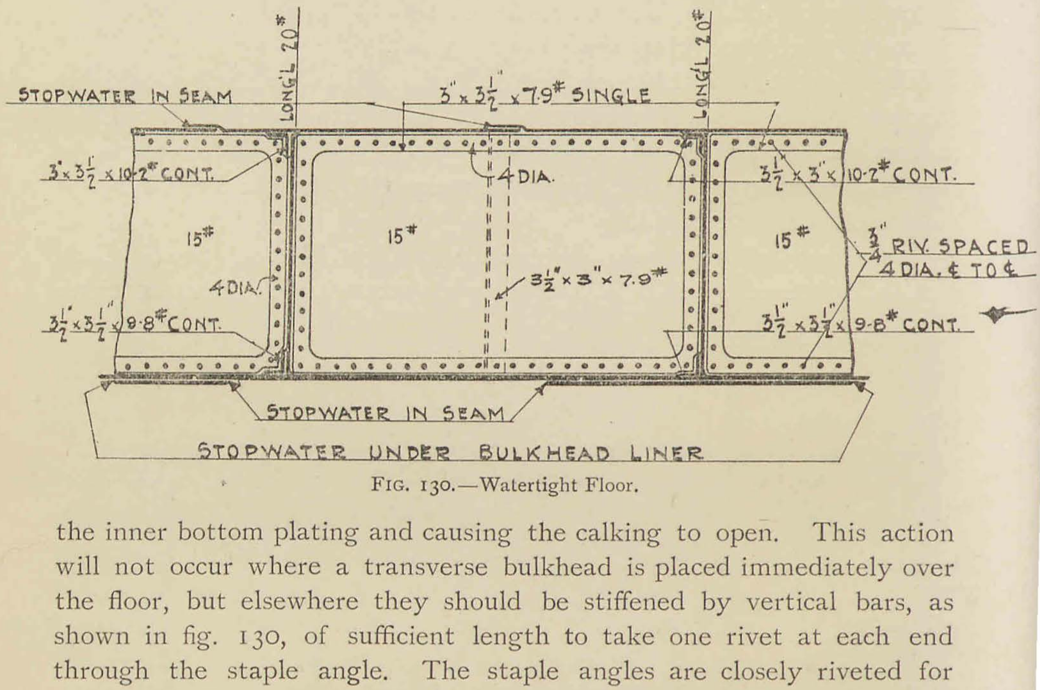


FIG. 130.—Watertight Floor.

the inner bottom plating and causing the calking to open. This action will not occur where a transverse bulkhead is placed immediately over the floor, but elsewhere they should be stiffened by vertical bars, as shown in fig. 130, of sufficient length to take one rivet at each end through the staple angle. The staple angles are closely riveted for watertight or oiltight work and are calked completely round the toes. Oiltight floors are calked on both sides. Stopwaters must be used in several places as indicated in fig. 130.

Figs. 131, 132 show typical floors as used in the British Navy,* reproduced by kind permission of Professor J. J. Welch from his work on Naval Architecture. The diagonal bars in fig. 132 serve to distribute the pressures when the ship is in dock.

Intermediate frames.—Under the engine foundations, and in other places where the bottom is subject to great strains over a large area,

* J. J. Welch, *Naval Architecture* (Potter), London, 1907, figs. 80 and 89, and pp. 128, 129.

complete intermediate frames constructed of lightened floor plates are inserted. In reserve feed-water tanks and in oil tanks additional stiffness should be given to the inner bottom by working intermediate reversed bars on the half frame spaces, stapled down on the longitudinals, where they take a few rivets.

If, as proposed in SECTION 48, the now commonly used frame spacing is reversed, and the transverse frames spaced 8 ft. apart, stiffening bars should be fitted under the inner bottom on the half, or perhaps on the one-third and two-thirds frame spaces, stapled down the full depth of the longitudinal on one side.

2. **Frames Outside the Double Bottom.**—Below the protective deck in large ships these frames should generally be of the belt type, consisting of lightened plates with frame and reversed frame angles, but in localities where the frames are not subject to great forces and where they are well supported by decks, they may be simple channel or zed bars. The foot of such bars should be connected to the vertical keel by floor plates or, as the case may be, to the margin plate of the double bottom and to the protective deck, by deep brackets.

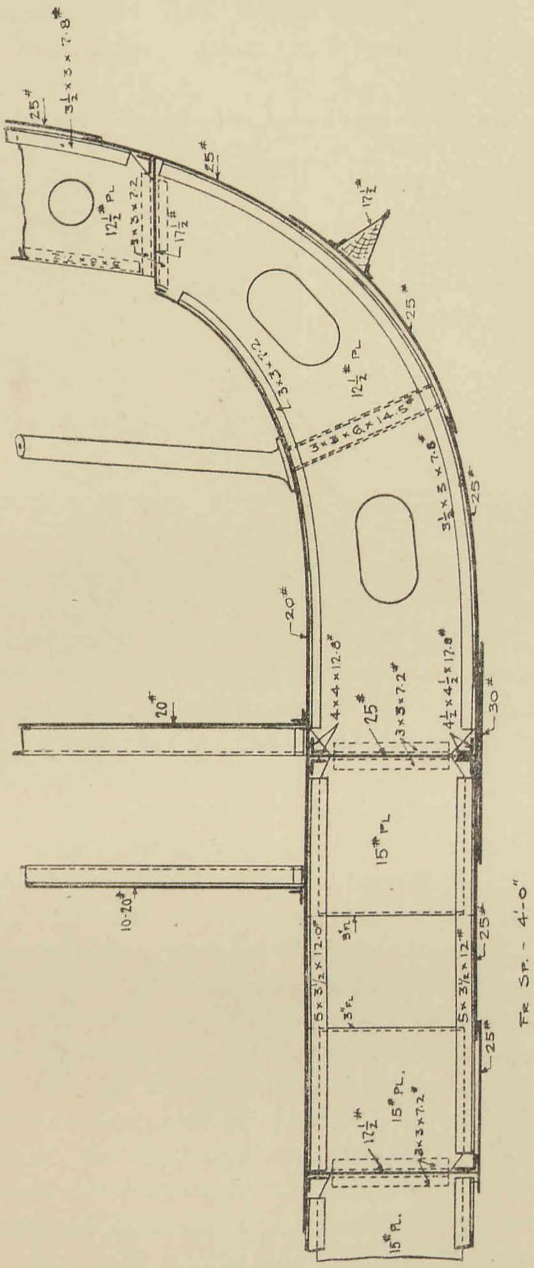


FIG. 131.—Bracket Frame and Plate Frame, *Lord Nelson.*

The strength of frames of this class may be conveniently calculated by means of the Bulkhead Tables, the scantlings being determined as for stiffeners of deep water tanks.

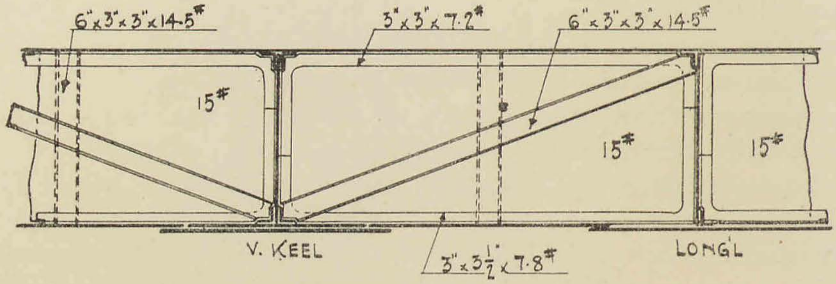


FIG. 132.—Watertight Frame adjacent to Vertical Keel (English System).

3. **Frames Behind Armor.**—The frames may here be of two kinds, web frames and simple channel or zed bars, as explained in SECTION 47, *8*, and as shown in fig. 114. Sometimes all the frames are of the same type, either web frames or deep bars. Web frames are constructed of plates, usually lightened by holes, but since such frames may be subject to extremely high shearing stresses, it seems better not to cut any holes in them. They should be connected by double angles to the plating behind armor and to the deck on which they stand, and be directly connected to the beams at the top. They are often deeper at top and bottom than at the middle, so as to secure a great shearing area of the rivets in the end connections. The inner edge should be reinforced by double angles. Bars, worked intermediately between the web frames, should be provided with efficient brackets at the ends.

In the French battleship *République* the framing consists of a number of watertight web plates—28 in. deep and 24 in. apart—which, together with an inner skin, form a very rigid, minutely subdivided cofferdam. Transverse bulkheads are fitted at intervals of about twelve feet and a wing bulkhead is worked about three feet inside the cofferdam. This construction is shown in fig. 133, which, as well as several other sketches in this work, is reproduced with the kind permission of M. J. Rougé, Ingénieur Principal de la Marine, from his work, *Construction du Navire*, École d'Application du Génie Maritime, Paris, 1904–5.

In the battleships of some navies the frames behind armor cut through the armor shelf and are attached to the floors below the shelf, as in the Danish armorclad *Peder Skram* (fig. 134). This arrangement may be adopted with advantage in all cases where the armor shelf is not continued in a deck and where, therefore, the foot of the frames behind armor cannot obtain an efficient support by means of brackets.

4. **Frames of Smaller Vessels.** — In *light cruisers* the frames inside the double bottom are constructed as in the battleships, but the

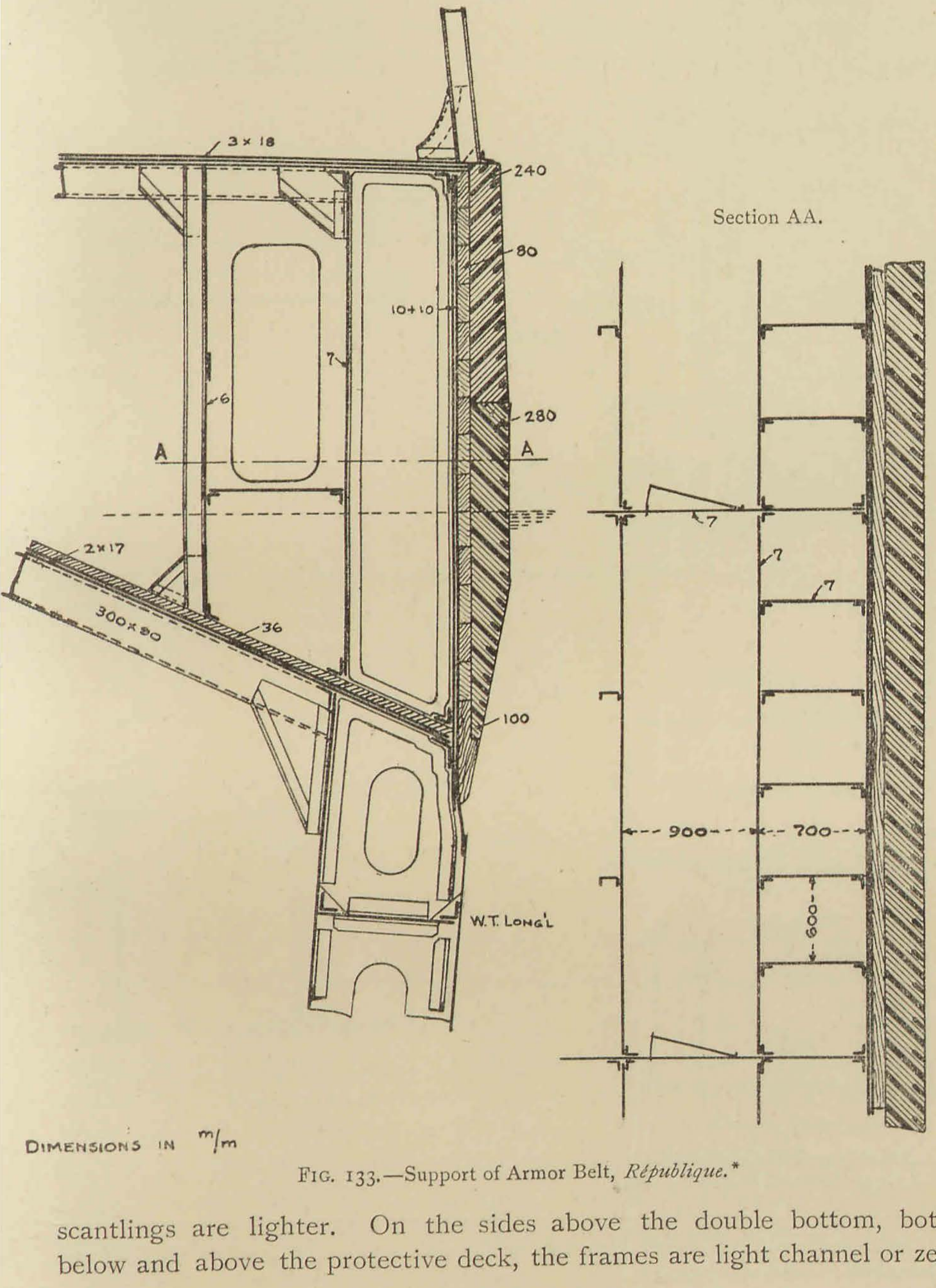


FIG. 133.—Support of Armor Belt, *République*.*

scantlings are lighter. On the sides above the double bottom, both below and above the protective deck, the frames are light channel or zed

* J. Rougé, *Construction du Navire*, i. p. 365.

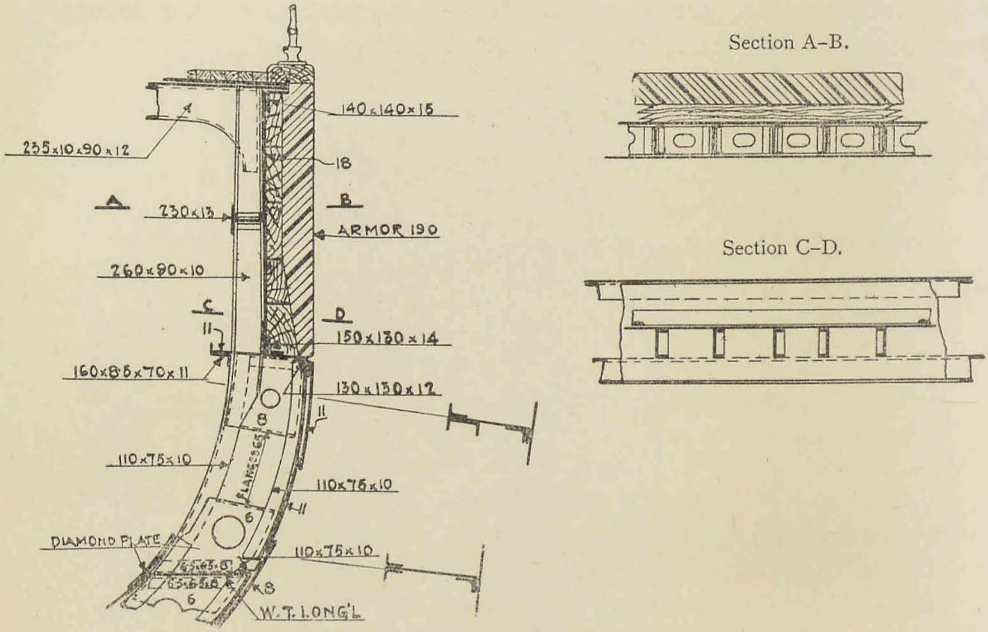


FIG. 134.—Framing behind Armor, Danish Armorclad *Peder Skram*.

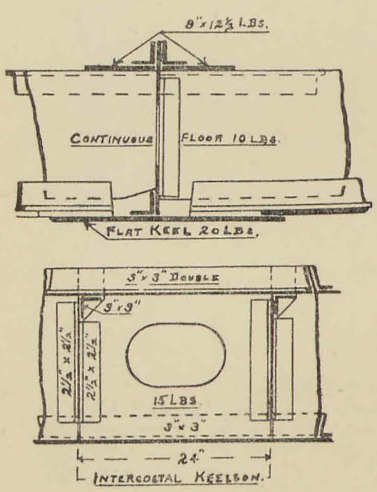


FIG. 135.—Keel Construction, Third-class Cruiser, English System.*

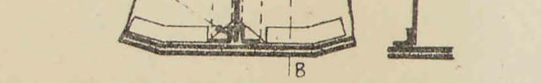


FIG. 136.—Keel Construction of Small Sheathed French Cruiser.†

* By courtesy of Mr Edward L. Attwood, from his work on *War-Ships*, published by Longmans, Green & Co., London, 1904, p. 34.

† A. Croneau, *Construction Pratique des Navires de Guerre*, Paris, 1894, i. p. 215.

bars, connected by deep brackets to the margin longitudinal of the double bottom, as shown in fig. 116. Where a side armor is fitted, the frames should consist of deeper, more substantial bars.

In gunboats and small cruisers of moderate speed transverse strength is of primary importance. Where no double bottom is fitted, the floor

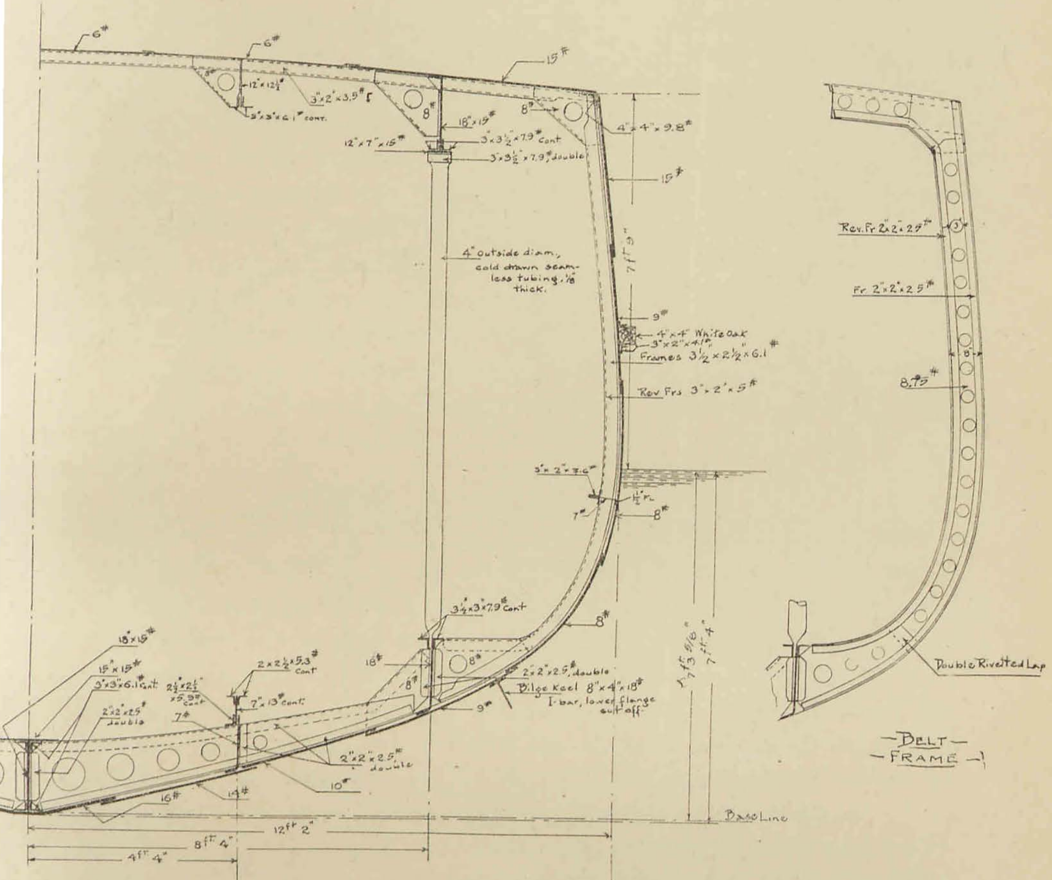


FIG. 137.—Midship Section of 600-ton Oil-burning Destroyer of Usual Construction.*

plates together with the reversed bars may with advantage be continuous across the keel, while the longitudinal strength is maintained to some extent by making both the outer and inner keel angles continuous, the latter running over the top of the floors. This construction, shown in fig. 135, has been used in the British Navy in certain third-class cruisers.

In smaller French ships exactly the reverse system has been used, the vertical keel being here continuous, while the transverse continuity is

* Prepared in the course of Naval Construction at the Massachusetts Institute of Technology by Assistant Naval Constructor A. W. Carmichael, U.S.N.

preserved by carrying the main frame, which is a channel, as a reversed bar along the inner edge of the floor plates, across and above the vertical keel.

In *destroyers* the frames usually consist of light bulb angles or angles with reversed bars connected to floor plates at the bottom. Belt frames, consisting of plates lightened by holes and strengthened by angles on

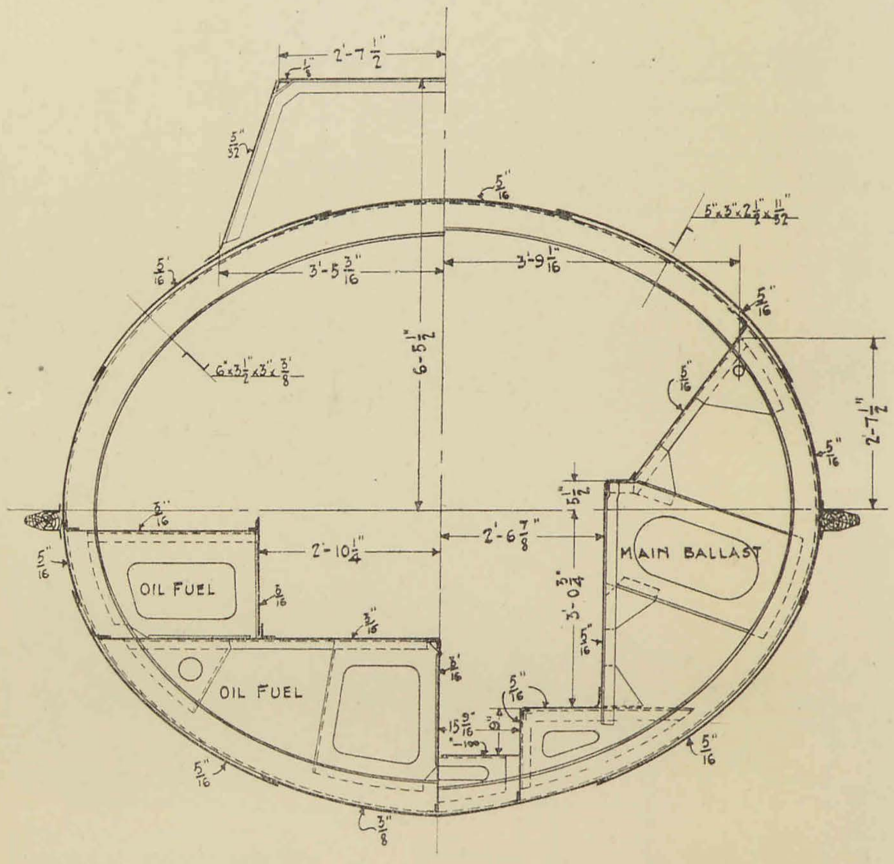


FIG. 138.—Midship Section of Danish Submarine *Havmanden*, Whitehead type.

their inner and outer edges, are fitted at intervals (fig. 137). By *the system of framing proposed in* SECTION 48 and illustrated in fig. 117, the transverse frames are relatively deep channels, slotted over the longitudinal bars on the sides and carried continuously across the keel. The floor plates are intercostal in order to obtain an efficient connection to the deep longitudinals. The longitudinal bars on the sides are continuous between the bulkheads to which, as well as to the transverse frames, they are connected by brackets.

In torpedo-boats the frames are simple angle bars with floor plates at the foot. A reversed bar is worked on top of the floor and carried up along the frame for a short distance.

In *submarines* of the Whitehead type (fig. 138) the frames are internal and consist of channels or Z-bars or simple angles running all round the contour of the section, which is often of oval form. Floors are worked under engines and storage battery, *i.e.* practically throughout the length of the boat. Where the section is strongly oval it appears advantageous to fit floors both at the top and bottom of the frames. The ballast- and fuel-tanks are placed inside the strength hull.

In boats of the *submersible* type the tanks are placed chiefly outside a circular strength hull between it and an outer somewhat lighter ship-shaped hull. Between the two shells are worked, in boats of the Laurenti-Fiat type, a framing consisting of brackets and lattice work, connecting light angles on the strength hull to similar angles on the outer shell. Generally no frames are fitted inside the strength hull. In the Germania type of submersible the strength hull is made of circular-welded sections riveted or bolted together, apparently without any framing, internal or external.

51. RIVETING OF THE FRAMES.

The spacing of the rivets varies a great deal according to the actions to which the riveted connections are exposed. Where the function of the rivets is simply to hold a plate and an angle together with little or no shearing or tension, the spacing may be very open and is usually 7 to 8 diameters. Such is, generally, the case in the connection between the frames and the outer shell, but where the frames are of small depth and of relatively great unsupported length, and especially where subject to panting, the rivets should be spaced from 5½ to 6 diameters apart.

Where rivets are subject to tension, as in case of the rivets which connect the frames to the inner bottom plating, the spacing should be fairly close, not more than 5 diameters. The rivets in the vertical flange of the lower keel angles in ships docked on center keel only should be spaced from 4 to 5 diameters apart. The floor brackets in large ships should be double riveted to the frame angles and to the reversed bars in order to secure a good rigidity, but the spacing in the rows may be 6 or 7 diameters.

Where watertightness is required, the ordinary spacing of from 4½ to 5 diameters should generally be used, but the rivets in staple angles of watertight floors should be spaced 4 diameters apart, so as to support the calking better when the pressure is applied on the calked side.

CHAPTER XIII.

STEM AND STERNPOST. FRAMING AT THE ENDS OF THE SHIP.

52. STEM AND BOW-FRAMING.

1. **Conditions and Claims.**—The stem forms the bounding frame in the bow. Regarding the stem as the continuation of the vertical keel and as the limiting transverse frame, we may say that in the stem the longitudinal and the transverse framing systems merge into each other.

Formerly the stem was designed for the special purpose of ramming, but this mode of attack is not now considered of importance. Accidental ramming, *i.e.* collision, is, however, a contingency to be considered and the stem is exposed, moreover, to violent strains by grounding and by impact on quay-walls or other obstructions. To meet these actions the stem should be capable of resisting and of distributing great pressures. It must therefore possess great strength and stiffness and its connection to the structure of the ship must be proportionately strong. By collision or ramming the action is at first a great longitudinal thrust, but when the ships swing together, as they will usually do, a lateral bending effort will come to exist, tending to break the stem off sideways. Locally, there may be a strong tendency to break the stem crosswise and to rip up the plating as, for instance, when the stem meets an armor deck in the rammed ship. When a ship runs aground on a rocky bottom there may be a violent local pressure under the forefoot, likewise tending to break the stem crosswise.

2. **Material.**—In the early days of iron shipbuilding the stem was made of wrought iron and this material is still used extensively in

merchant vessels, but in modern warships the stem is usually of forged or cast steel. Forged steel combines great toughness with strength and stiffness, but can only be used for stems of simple form. Cast steel has the advantage that it can be given almost any form; it possesses great stiffness and a fair amount of strength, but it is deficient in toughness. Cast steel is, therefore, employed where the connection to the structure of the ship is of a complex nature, necessitating the presence of flanges, rabbets, webs, and other projections. It has been objected to cast steel that it is unreliable and, especially, that it is apt to crack and break when exposed to shock such as may occur by ramming and collision. The French were for a long time reluctant to use cast steel for this reason. Steel castings can, however, now be produced in a perfectly homogeneous and reliable quality, and, moreover, the action by ramming and collision cannot properly be described as one of shock or sudden impact. The velocity of striking will rarely exceed from ten to twenty feet per second, which is small relative to the great mass of a large ship. The ram will generally strike the soft part of the bottom of the other ship below the armor, and no serious obstacle is encountered till the stem brings up against the armor shelf. Hence the action is to be likened to a slow but irresistible punching, where a hard rigid and massive body, placed end on to a softer and thin-skinned, weaker body, penetrates forcibly into this latter. The impact is further reduced in violence by the yielding of the rammed ship, which will move bodily and generally also by swinging round and heeling over. The relatively gradual nature of the impact by collision is evidenced by the observation so frequently heard after even serious collision cases in large vessels, that the shock was not felt at all, and surprise is generally expressed at the magnitude of the damage (*Titanic*). The objections to cast steel as material for the stem cannot, therefore, be considered valid, and it is now, generally, used for this purpose in all except the smallest warships. In sheathed and composite vessels the stem is of bronze, generally either manganese bronze (U. S. N.) or phosphor bronze (Br. N.). This metal, which is electro-negative to copper, is here employed to avoid the deterioration to which a steel casting would be exposed when in contact with the copper sheathing. Since bronze is inferior to steel in strength and ductility, it is necessary to make bronze castings more massive than steel castings. The bronze stem should extend to the upper edge of the copper sheathing, above which iron or steel may be employed. In composite vessels of pure type the stem is of wood.

3. **Form and Extension.**—When the stem is designed for ramming, it should project under water from eight to ten feet forward of the end of

the load water-line, and the extreme point should be at a depth of about ten feet below this line. Although ramming is now contemplated only as an accident, the stem is still given some projection below water. In this way weight is saved in the upper works and it has been found by model experiments that this form, especially if combined with bulbous transverse sections, is favorable in point of resistance at high speeds.* The projecting form is, however, liable to cause difficulties in handling of the anchors, since the chain may come to ride across the ram and the anchor may chafe on the broad, bulbous part of the hull. Usually the contour of the stem rises nearly vertically from the water-line to the main deck. In earlier ships the casting extended to the weather deck, whether main or forecastle deck, but in many recent large vessels it stops at the second deck. The foot of the stem should be carried down to the straight part of the keel line, so as to permit a simple connection to the flat keel plates. It should not be carried further aft than necessary for this purpose, since the difficulties in casting, transportation, and handling increase greatly when a certain size is exceeded, depending upon the appliances and the machinery available. Sometimes limitations in these respects make it necessary to cast the stem in two pieces, which are then united with a scarph.

4. **General Principles of Construction.**—The stem should nowhere be made heavier than required for strength, even when it is designed for ramming. Concentration of a great mass of material does not in itself increase the effect of ramming. The mass and hence the inertia of the stem will always be small compared to that of the ship, and the main point is, therefore, to make the connection between the casting and the ship so strong that the enormous mass of this latter can be made effective in ramming without any breakdown of the structure. The ram is simply a tool, the driving force is supplied by the inertia of the ship. The forces of impact created by ramming and collision should be distributed as widely and uniformly as possible over the structure of the striking ship, whence the material must be so arranged that the strength will vary very gradually, passing from the heavy and rigid casting in the stem to the extensive and ramified structure which is found at only a short distance aft.

5. **Connection to the Hull Structure.**—In accordance with these principles the connection of the stem casting to the structure in a large ship should be effected in the following ways (fig. 139) :—

(1) The outer bottom plating, which is generally doubled in the bow, should be rabbeted into the stem and connected to it by double riveting.

* D. W. Taylor, "Some Model Basin Investigations, etc.," *Soc. Nav. Arch. Mar. Eng.*, 1911.

(2) The end of the flat keel plates should be rabbeted into the foot of the stem, which they should envelop with a lengthwise overlap sufficient for an efficient riveted connection. The foot of the stem must therefore be provided with a double notch ("stops").

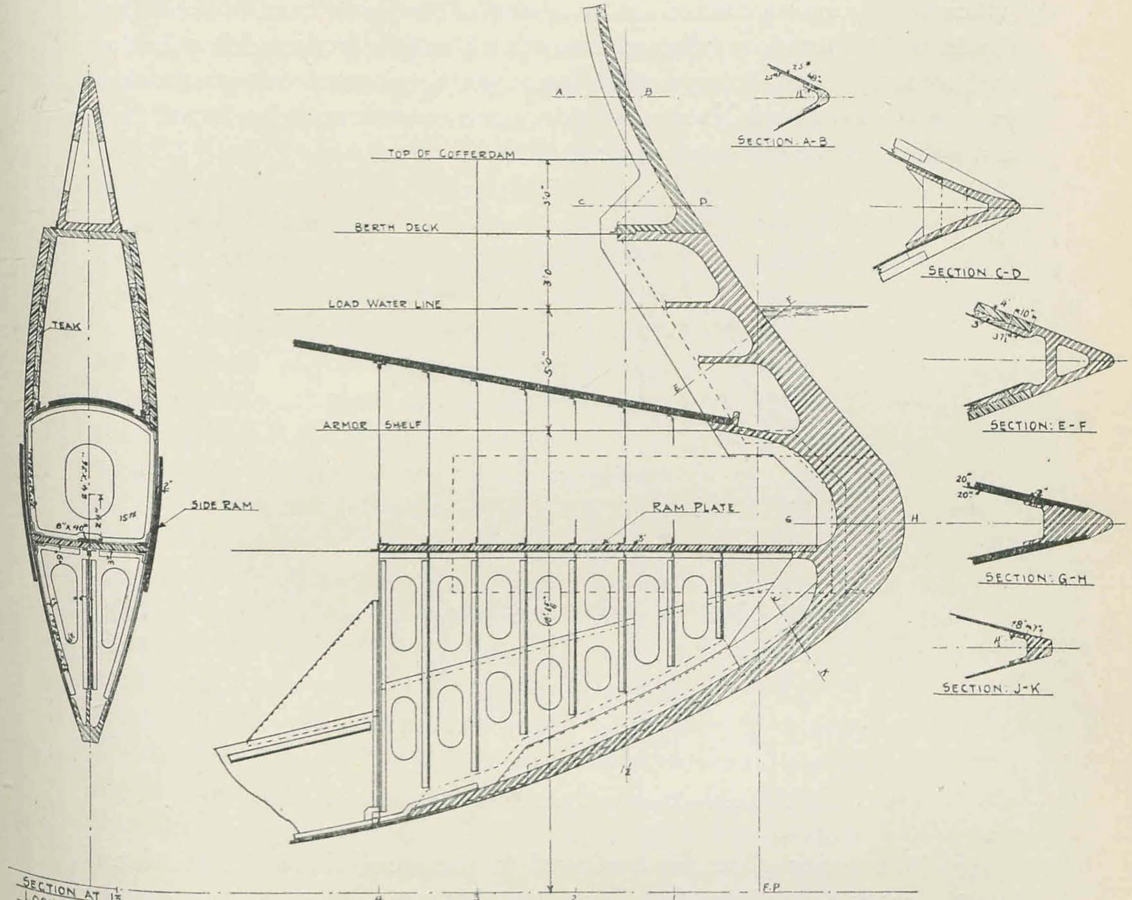


FIG. 139.—The Stem of a Battleship designed for Ramming.

(3) All the decks should be connected to "webs" ("lugs") on the casting.

(4) Several of the longitudinals should be continued to the bow and should merge into horizontal breasthooks constructed of plates lightened by holes and connected to webs on the stem in the same way as the decks (fig. 140). Breasthooks should be placed at the level of each of the platform decks as well as midway between these. Longitudinals should connect the platform decks with the respective breasthooks.

(5) The vertical keel should be efficiently connected to the foot of the

stem either by riveting it to a central vertical web on the casting or where space permits by means of the lower keel angles.

(6) The transverse floors that stand on the foot of the stem should be connected to webs on the casting unless the breadth of the stem is sufficient to permit a connection by means of the frame bars. The floors should be lightened by holes and should fill out the entire section below the protective deck, being worked intercostally between the breasthooks. In large ships intermediate floors of the same construction should be fitted in addition on the half frame spaces.

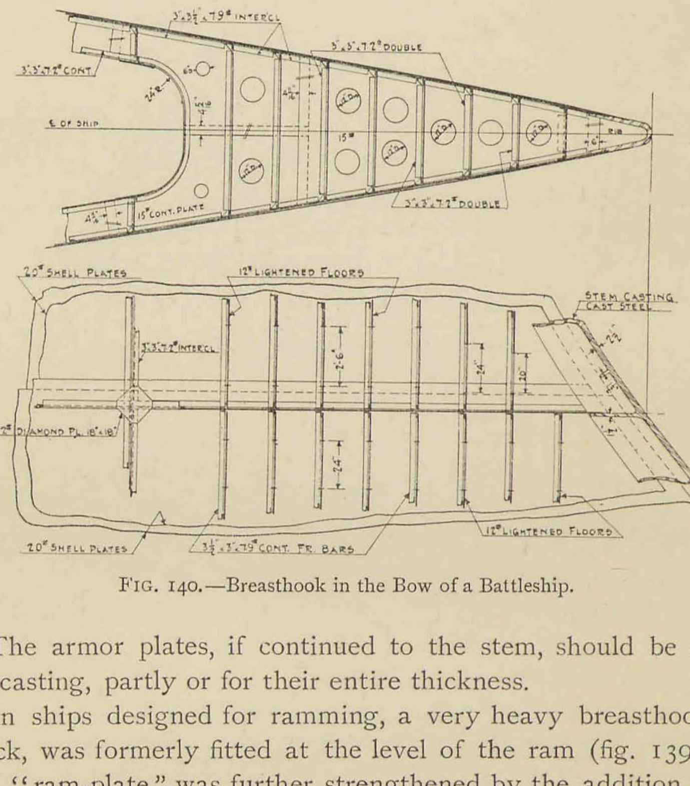


FIG. 140.—Breasthook in the Bow of a Battleship.

(7) The armor plates, if continued to the stem, should be rabbeted into the casting, partly or for their entire thickness.

(8) In ships designed for ramming, a very heavy breasthook, often 2 in. thick, was formerly fitted at the level of the ram (fig. 139). This so-called "ram-plate" was further strengthened by the addition of heavy plates, "side-rams," fitted outside the shell plating in way of the ram-plate, but these features are not found in recent ships.

With the construction here described, the longitudinal thrust will be well provided for, and the shell will possess great resistance to local tearing. The forefoot will be well supported and stiffened, capable of resisting great local efforts when grounding.

Sideways bending of the stem will be resisted by the entire structure acting as a girder of which the shell plating on each side forms the flanges, while the breasthooks and the decks act as webs. For this reason care should be taken not to weaken the breasthooks too much by lightening holes or by a single riveted seam in the center-line, since it is here that the maximum shearing will occur and the breasthooks would be thus robbed of their web strength.

6. **Casting and Details of Construction.** — The difficulties in obtaining a sound casting increase with weight and size. In very large and heavy castings the shrinkage is apt to be excessive during the cooling process, causing great internal stresses and hence a tendency to contraction flaws. The stem should be as light as consistent with strength and soundness of casting, and should, therefore, be well cored out, the stiffness being secured by webs placed half a deck-height apart or closer where required for local strength. The average thickness in the main body of the casting for a large ship should be from 2 to $2\frac{1}{2}$ in., increasing to 3 in. or more in parts exposed to great local strains. The thickness of the flanges tapers to about $1\frac{1}{2}$ in. or 1 in. at the edges in large ships and to $\frac{3}{4}$ in. or even $\frac{5}{8}$ in. in small ships. The width of the flanges should not be greater than necessary to take the rabbets. The webs should be just large enough for their purpose, *i.e.* they should be of the same depth as the flanges which they serve to stiffen, except where the connection to a deck or breasthook calls for greater depth. The thickness of the webs in large castings should be from 1 in. to $1\frac{1}{2}$ in., except in case of the web that takes the protective deck, which may need to be somewhat heavier.

As a general rule, great and sudden differences in thickness should be avoided, because, in the process of casting, the slender parts cool more rapidly than the heavier, and tend to suck the material from them, whereby cavities may be produced. This can to some extent be avoided by fitting risers at the points where this condition exists. Where light projections are found on the massive parts, the same action will take place and is liable to cause flaws, rupture, or distortion and strains. It follows that in small castings the projecting parts may be made of lighter scantlings than in large castings.

The front of the stem should not be so blunt as to offer excessive resistance to driving through the water. On the other hand, it should not be so sharply curved as to injure the chain cables when these are riding over the stem. A rounded section, struck with a radius of not less than about $1\frac{1}{4}$ times the size of the chain, will usually fulfil these requirements.

7. **Riveting.**—The riveting of the stem is necessarily heavy and the rivets are often of great length. Through rivets should be used where possible, *i.e.* where not longer than from six to nine diameters ; in large

vessels they are from 1 in. to $1\frac{1}{2}$ in. in diameter. Where through riveting is impracticable, tap rivets are used. The riveting in the rabbets should be zigzag in order to reduce the width of the rabbets as much as possible. The rivets connecting the shell plating to the stem casting should be $\frac{1}{4}$ in. larger than otherwise specified for the plating.

8. **Scarphing.**—When the stem is scarphed, the joint should not be placed too near the ram point. The scarph may be a "table-scarph" with rabbet and tongue, placed crosswise, as in fig. 139, or it may be a "side-scarph," placed lengthwise as shown in fig. 141. A side-scarph should be provided with a tapered key for drawing the two parts together; heavy screw-bolts must be used on account of the great thickness of the material. According to Lloyd's Rules, the length of a side-scarph should be nine times the thickness of the stem.

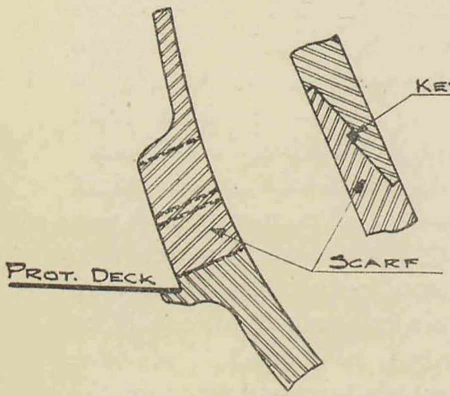


FIG. 141.—Scarph in Stem Casting.

9. **Upper Part of Stem.**—In recent ships of high freeboard in the bow, the stem casting usually stops at the upper edge of the armor or at the second deck, above which it is replaced by a heavy curved "contour-plate," the lower end of which is wrapped round the top of the casting and thoroughly riveted to it.

In many French battleships, where the armor belt is continued to the stem, the casting stops at the lower edge of the belt. The armor plates are carried out to butt against each other as shown in fig. 142, and extend so far downwards as to form the ram. The weight of this construction seems out of proportion to its strength, and the great mass of armor thus concentrated at the extreme end of the ship is of little value in point of protection.

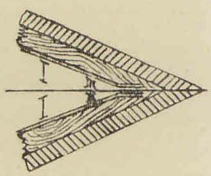


FIG. 142.

10. **Smaller Vessels.**—The stem of cruisers does not differ essentially from that of battleships, but the construction is simpler and lighter.

In the stem of sheathed vessels rabbets are cast both for the shell plating and for the planks. The rabbet for the sheathing is given an under bevel to hold the hood end of the planks.

The stem in most torpedo-vessels is simply a forged bar of iron or steel, in larger boats provided with a rabbet on each side to take the plating. In some boats the lower part is cast and scarphed to an upper part of forged iron or steel.

53. STERNPOST AND STERN-FRAMING.

1. **Functions of the Sternpost.**—The sternpost completes the framing system in the aft end of the ship, and may be required, moreover, to perform the following functions :—

(1) To carry the weight of the rudder and to guide and limit it in its motion.

(2) To support the propeller shaft or shafts. This may be done either directly, the sternpost holding the aft end of a central stern-tube, or indirectly through shaft struts, the lower arms of which are often connected to the sternpost.

(3) To take the upward reaction when the ship is in dock or aground.

(4) To protect the rudder and the propeller from damage by grounding. In many modern ships, where the deadwood is cut away and where partly or entirely underhung rudders are used, neither the rudder nor the central propeller obtains such protection in a direct manner, but they are kept clear above the lowest point of the keel.

2. **Material.**—The general remarks made on the material for the stem apply also to the sternpost. On account of the more complex form of the latter, cast steel is here employed more generally than for the stem, in particular in large ships. In small vessels, especially gunboats and torpedo-boats, the form of the sternpost may be so simple that wrought iron or wrought steel may be used. Such forgings are generally made from scrap iron or scrap steel. Scrap steel gives a stronger and more ductile forging, but at the welds the strength is hardly equal to that of scrap iron, which is, therefore, mostly preferred in this case.

3. **The Rudder-Post.**—That part of the sternpost which carries the rudder is called the "rudder-post"; it is vertical and is provided with projections, called "gudgeons," for taking the pintles. It is subject to great alternate transverse forces due to the pressures on the rudder.

In fig. 143 is shown the rudder-post of a *twin-screw battleship*. Since the deadwood is cut away and since the rudder is partly underhung, the post is here of reduced height. Fig. 144 shows the sternpost of a small coast-defence armorclad. In many English battleships the rudder-post extends down to the keel-line and the deadwood is not cut away till about seven feet forward of the post, in order to provide a support for the stern when the ship is in dock.

Fig. 145 shows the rudder-post of a *triple-screw ship*, as used in the French and German navies. The rudder-post here projects downwards as a "skeg" and carries only one pintle.*

* J. Rougé, *Construction du Navire*, i. p. 457.

In *single-screw ships* of the gunboat type the stern is usually shaped as in merchant vessels (fig. 146). The "stern frame" consists of the

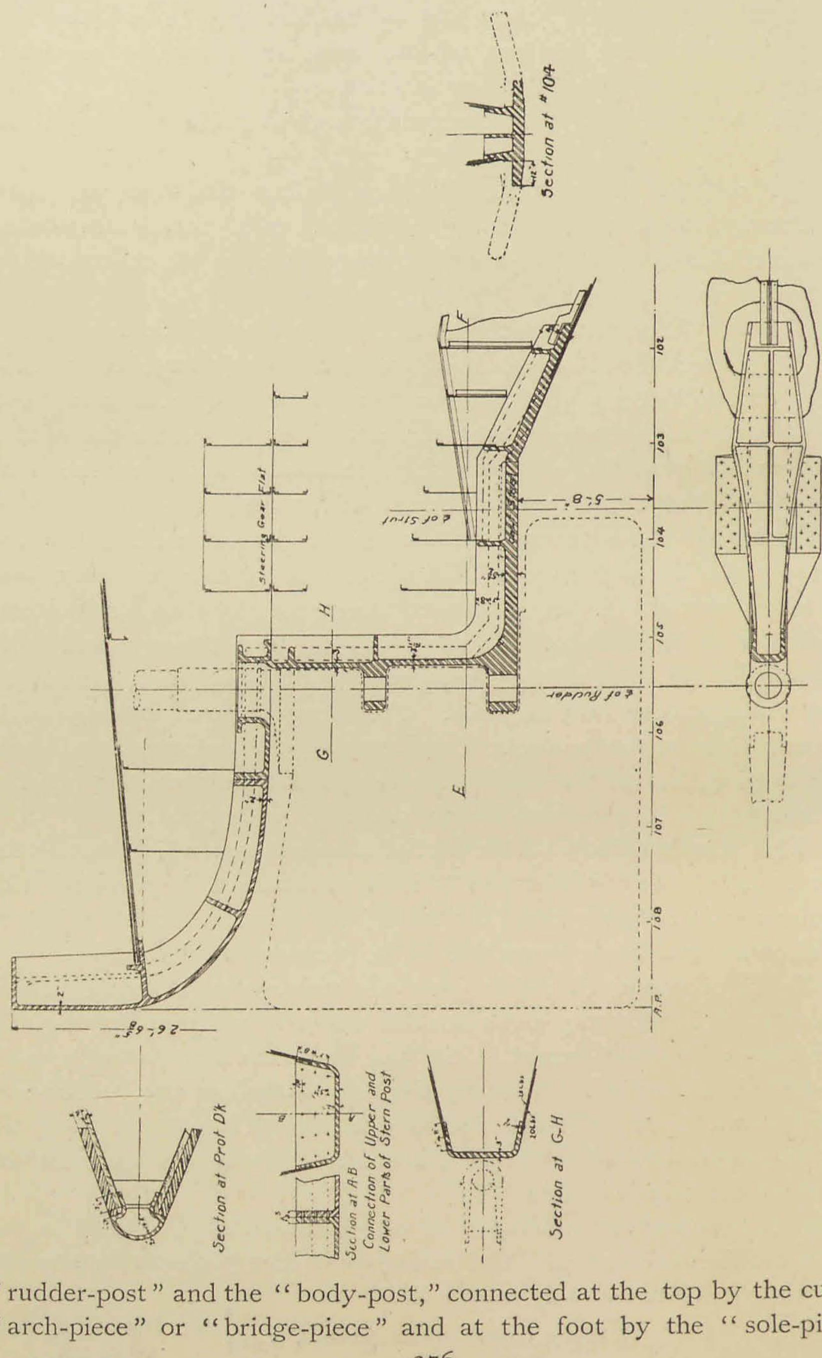


FIG. 143.—Sternpost of Sea-going Battleship.

"rudder-post" and the "body-post," connected at the top by the curved "arch-piece" or "bridge-piece" and at the foot by the "sole-piece."

The stern frame envelops the propeller, which works in the aperture between the two posts. By this construction there is, ordinarily, no eye or boss for holding the rudder-head. The rudder-post projects vertically into the ship above the counter and serves as a means of connection

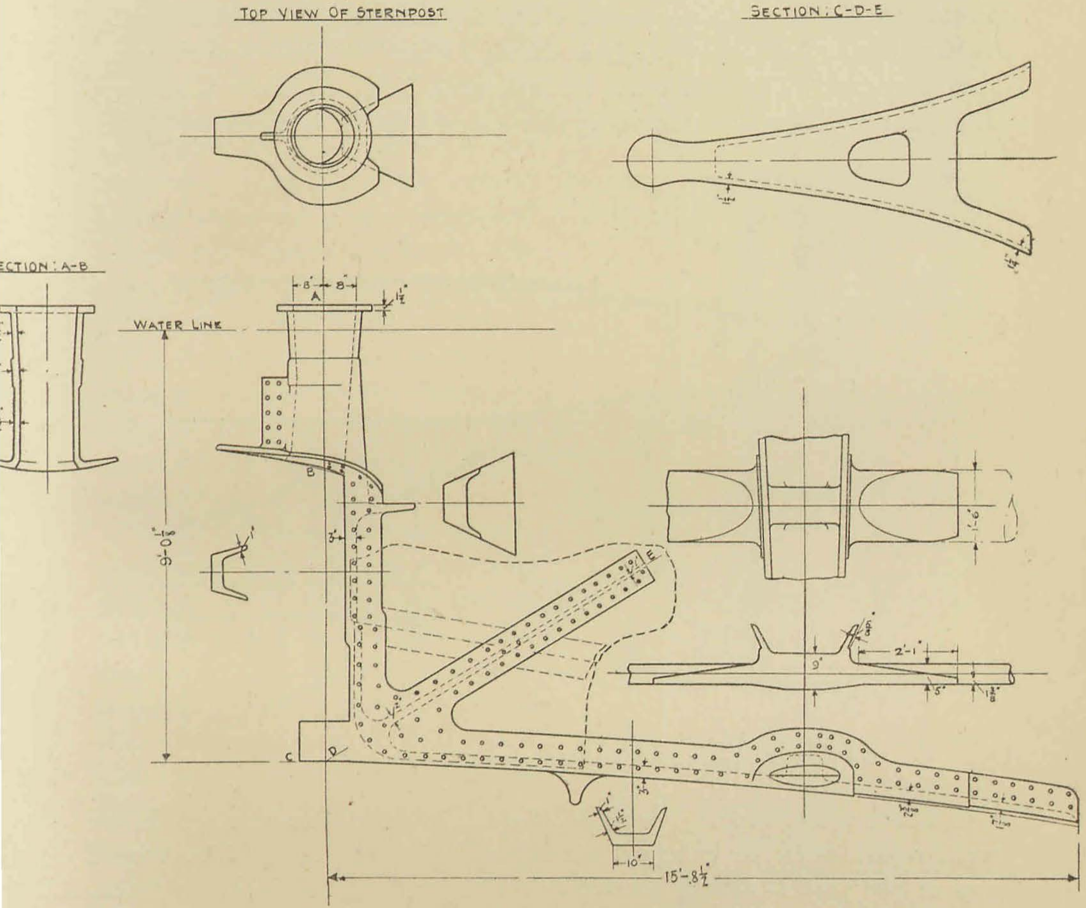


FIG. 144.—Sternpost of Danish Armorclad *Herluf Trolle.*

to the hull. This construction is, however, heavy and should be used only when the sternpost is forged and, therefore, is necessarily of simple form. When the sternpost is cast, a lighter and more efficient connection can be obtained by means of flanges and webs at the top of the rudder-post.

Returning to the larger ships where the sternpost is cast, the aft face of the rudder-post is usually broad and flat, increasing in width from the lower gudgeon to the top, so as to provide strength against the lateral bending forces due to the action of the rudder. In order that this feature shall not act prejudicially on the resistance of the ship, the rudder should be given the same width as the post at every point of the front edge, and should taper aftwards so as to give a "fairwater" finish. Projections, "hardover-stops," are cast on the rudder-post near the top of the rudder in order to limit the rudder angle. The top of the rudder-post is usually swelled out to form an eye or boss through which the rudder-stock enters into the ship. The hole must be sufficiently large to house a bearing, and generally also a stuffing-box. In many ships it is necessary to tilt

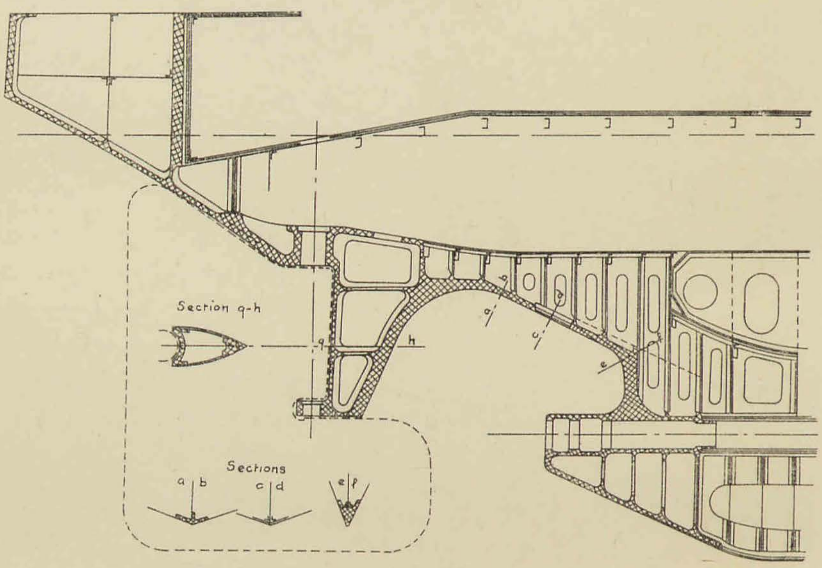


FIG. 145.—Sternpost and Stern Framing of French Battleship *République.*

the rudder when shipping or unshipping it, in which case the hole in the rudder-post must be enlarged correspondingly (fig. 144). The bossed top of the post is usually formed as a circular flange, stiffened by webs, for attachment of the stuffing-box and for taking the bearing which supports the weight of the rudder.

4. **The Body-Post and the Bottom-Piece.**—The body-post exists only in single- and triple-screw ships. It takes the end of the stern-tube. The aft edge of the body-post must not be too blunt in its vertical and nearly vertical part. If necessary, a fairwater is worked, consisting of plates made watertight and filled with white pine. The sole-piece of single-screw ships should possess considerable lateral strength so as to resist the transverse resultant of the rudder pressure. It is, therefore, often of flat form or provided with broad horizontal flanges. There

is a great variety in the form of the contour of the stern profile and hence in the form of the stern casting. As a general rule the casting is carried sufficiently far forward from the foot of the rudder-post or body-post to take the lower arms of the shaft-struts and to obtain a good connection with the keel structure. The connection of this "bottom-piece" to the strut arms may be effected by means of projections to which palms on the lower strut arms are riveted (see figs. 143, 144, 147). In English ships the palms of the lower strut arms pierce the outside plating above the stern casting and are riveted together inside the ship resting upon a heavy horizontal plate. In ships with four propellers the outboard struts are too far forward to connect to the sternpost, and the same holds good for the struts of the side shafts in triple-screw ships.

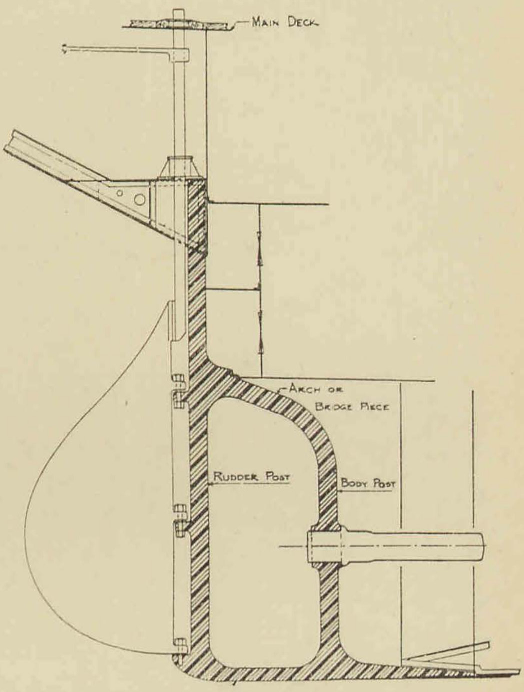


Fig. 146.—Stern Frame of Single Screw Ship.

5. **Connection between Sternpost and Hull Structure.**—The construction is similar to that already described for the stem. The rudder-post or, in ships with central propeller, the body-post is provided with rabbeted flanges for connection with the outside plating. Webs are fitted to take breasthooks or flats; usually the steering-room flat is connected to the top of the rudder-post. Where the breadth of the casting is too small to allow connection by angle bars, the bottom-piece should have transverse vertical webs for the floor plates and a longitudinal web for the vertical keel. From the bossed top of the rudder-post a short arm provided with flanges projects aft and connects with a curved contour plate, which in modern ships takes the place of the upper part of the sternpost. In many earlier ships the casting was continued to the protective deck or even to the top of the armor as in fig. 143.

The framing of the stern portion of the ship consists, as in the bow, chiefly of transverse floor plates lightened by holes. In large sections the floor plates are replaced by belt frames running all round the sides and under the breasthooks or decks. This construction, shown in fig. 147, is very suitable in way of the shaft struts, where exceptional strength and stiffness are required.

Large sternposts are often cast in two pieces connected by a scarph, usually placed on the rudder-post. With the increasing size of ships it has become more and more difficult to cast the sternpost in one piece,

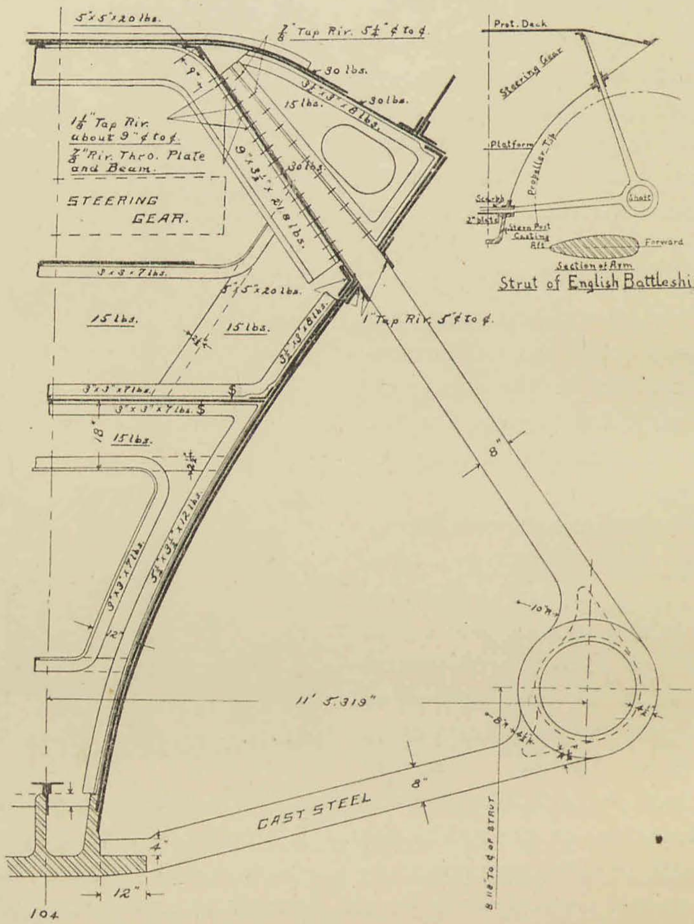


FIG. 147.—Section at Shaft Strut.

while at the same time the larger dimensions render it easier and lighter to employ structural work. In many modern ships, therefore, the sternpost is replaced by several entirely separate castings connected by plates and frames which form part of the ordinary hull structure, the use of castings being restricted to the most complicated parts.

In some ships a so-called "heel-piece," as shown in fig. 148, is fitted at the knuckle where the straight keel ends and the cut-up of the deadwood begins. In this way much expensive and difficult plate work is avoided.

In the English battleship *Dreadnought*, and in many following ships of the British and other navies, two entirely or almost entirely underhung rudders and four propellers are used. The deadwood is entirely

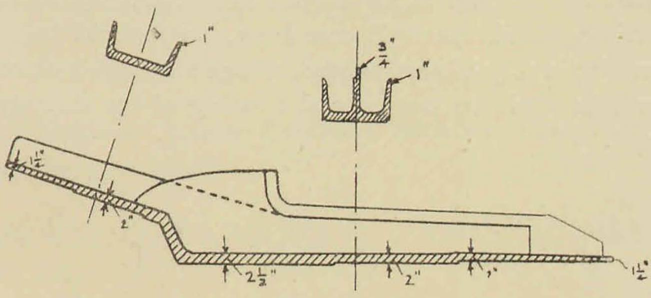


FIG. 148.—Heel-piece, Danish Armorclad *Peder Skram*.

cut away and there is no sternpost. Separate "rudder-head castings" are in such vessels fitted at each of the rudders to form bearings for the rudder-stocks and to hold the stuffing-boxes.

The stern construction used in the *Dreadnought* was first adopted by Sir John I. Thornycroft in his early torpedo-boats. These boats had

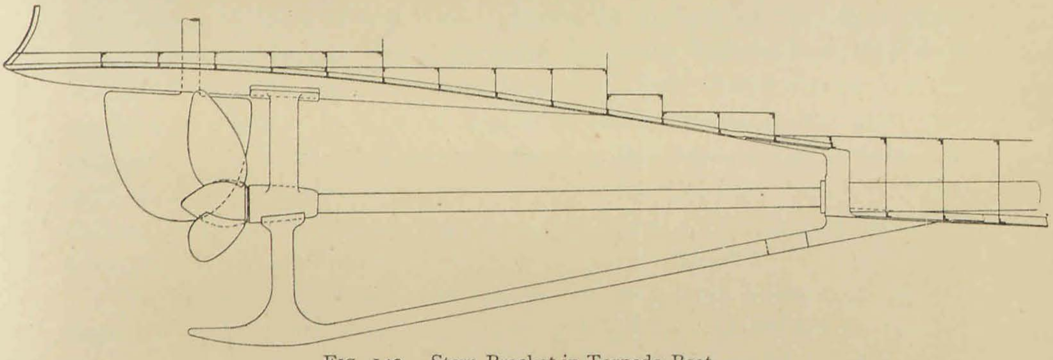


FIG. 149.—Stern Bracket in Torpedo-Boat.

underhung rudders and a single propeller projecting below the keel. In order to protect the propeller, a bracket of forged or cast steel was fitted as shown in fig. 149, connected with a scarph to the shaft bracket. A similar "stern bracket" is often fitted also in twin-screw torpedo-boats for the protection of the propellers, in some cases supporting the pintle of a central rudder.

6. **Details of Construction.**—In designing the rudder-post and its attachments to the hull, the action of the rudder must be carefully con-

sidered. In particular the unsupported skeg used in triple-screw ships is liable to be severely strained ; in fact, one or more cases of fracture have occurred. Strength calculations, based on the ordinary theory of bending, can be made in this case.

The "deadwood" of twin-screw ships may with advantage be stiffened by a diagonal web as used in the Danish armorclads (fig. 144).

All castings should be well cored out, but should be stiffened by webs, especially in way of rudder-head, gudgeons, and shaft struts. The general remarks made on casting and riveting of the stem apply also to the sternpost.

CHAPTER XIV.

INNER BOTTOM PLATING.

54. GENERAL ARRANGEMENT. LENGTH, BREADTH AND THICKNESS. LOCATION OF MANHOLES.

THE inner bottom should be carried continuously under all bulkheads, transverse and longitudinal, so as to preserve its longitudinal strength. It rests on the transverse and longitudinal frames, to the inner angles of which it is riveted. Ordinarily the plating is laid on the clinker system with lapped butts, in some navies (U.S.N.) joggled at the seams. With the raised and sunken system water will lodge at the upper (outer) seam of each strake, but with the clinker system this may be avoided, if there is any rise of floors, by making all the sight edges face toward amidships. In some ships the inner bottom is worked flush in way of oil tanks and feed-water tanks with butt straps and edge strips on the upper side. It may be argued in favor of this system that butt calking as well as the difficulties caused by the intersection of butt laps and seam laps are avoided, and a somewhat greater stiffness of the plating than by overlapped work is obtained. Overlapped work is, however, most commonly used.

The length and breadth of the plates may with advantage be the same as in the outside plating. The general lay-out should be studied on a model and an expansion prepared, showing edges and butts, manholes, boiler and engine seatings, and all other fixtures and connections of importance. The arrangement of the plates should be similar to that in the outer shell. The strakes which run along the top of the longitudinals should be continued beyond the limits of the double bottom for two or more frame spaces, tapering in breadth so as to avoid an abrupt change in longitudinal strength.

The inner bottom plating is not ordinarily subject to great water pressures or to the dynamic effects of waves and is less likely than the outside plating to be damaged by grounding. It is, therefore, not of so vital importance and is, further, more easily inspected. For these reasons it is always considerably lighter than the outside plating, but must, of course, possess sufficient strength to resist the pressure of the water when the outer shell is broken. Where the double bottom compartments are used merely as *empty cells*, considerable deflection of the inner bottom is permissible, since flooding will only occur when the outer shell is damaged and minor leakages will not be of serious consequence. As far as general structural strength is concerned there is rarely a necessity for employing anything but very light plating and the ultimate strength is always ample as long as the riveting is sufficiently strong. The plating in such compartments may, therefore, be given the minimum thickness required for efficient calking, *i.e.* from $7\frac{1}{2}$ to 10 lb. In some cases the plating of cellular compartments, which stood the test satisfactorily, showed a permanent set of about thirty per cent. of the total deflection.

Ballast tanks for regulating the trim and stability and for righting the ship when it takes a heel are ordinarily empty, but may be occasionally flooded for a considerable length of time. The head will usually be very small, but the claim to watertightness and, hence, to stiffness of the plating is rather severe. This claim will generally be satisfied when the thickness of the plating does not fall below that determined from Curve B, Pl. IV., assuming the same pressure head as for a watertight bulkhead placed in the same position relative to the water-line (SECTION 68). The strength will then be sufficient under extreme conditions of flooding, and when the compartments are used as ballast tanks the deflection will be small. The plating should nowhere be less than 10 lb.

In *feed-water* and *oil tanks* a more absolute tightness and hence a greater rigidity of the inner bottom is required. Again, the head is usually small, but, as in ballast tanks, the plating may be subject to a much greater head when the ship is in damaged condition, and it should not then be strained much beyond the elastic limit. Hence the thickness should be determined from Curve A, Pl. IV., using again the same head as for a watertight bulkhead in the same position but not less than 12 ft. above the top of the tank. In order to avoid excessive thickness, extra means of stiffening must be introduced so as to reduce the width of the unsupported areas of plating. Where intermediate floors are not already fitted for other reasons, substantial angle bars should be worked on the underside of the plating midway between the frames, stapled at the ends for connection by a few rivets to the longitudinals. Again, the plating should nowhere be less than 10 lb.

Example.—A double bottom compartment to be used for storage of oil is required to stand a head of 31 ft. when the ship is in damaged condition. The main frames are 48 in. apart and intermediate stiffeners are fitted on the half frame spaces, making $s = 24$ in. Since $r > 3$ we have $K_A = 1$ and $K_A h = 31$ ft. Corresponding to this head, Curve A gives $\mu = 61$, whence the required thickness is $\frac{24}{61}$ in. or about 16 lb. per sq. ft.

The inner bottom plating usually varies from 10 to 15 lb. in large vessels, being 10 lb. on the sides and at the ends where the compartments are merely cellular spaces, and from 12·5 to 15 lb. in the lower part of the double bottom amidships where the feed-water and oil compartments are located. These thicknesses, so far, have been found satisfactory in practice, and are on the whole in fair agreement with the rules given above. In light cruisers and gunboats, where the test pressures are smaller and the frame spacing closer than in large vessels, the inner bottom plating is usually 10 lb. throughout, apart from local reinforcements.

The central strake, often called the "center keelson" or "flat plate keelson," forms the upper flange of the important keel-girder and should be given about the same thickness as the standard of the outer shell. The thickness of the strakes adjacent to the center keelson should be intermediate between those of the center keelson and the next strakes. Extra thickness should be given also to the plating in way of engine beds, thrust-blocks, and wherever the structure is subject to concentrated loads, rough wear, or great dynamic forces. Doubling plates should be fitted under important stanchions, in way of manholes, and wherever the plating is reduced in strength below the standard. Each of the larger compartments of the double bottom should preferably have two manholes placed in diagonally opposite corners so as to facilitate a complete ventilation of the compartment. The manholes should be located so as to be always easily accessible, and should make a good shift with each other in different strakes. Manholes should not be placed in the same frame space with a butt of an adjacent strake or longitudinal. These claims cannot always be completely satisfied. Local conditions, such as the position of engines or boilers, will have to be considered. Oil tanks, which are generally of small size, can rarely be given more than one manhole. Hence, the location of manholes often involves a compromise.

55. RIVETING OF THE INNER BOTTOM.

In large vessels the riveting is generally double both in butts and seams, only the center keelson having treble riveted butts. With plating from 10 lb. to 15 lb., this will give practically the same efficiency of the butts as attained in the outside plating with treble riveting, viz. about ·7. The efficiency along the oiltight frames, where $m = 3\frac{1}{2}$ or even less, will fall below this standard, wherefore compensating liners should be fitted on such frames. The seams are double riveted on account of the claims to watertightness.

In small vessels the butts should be double riveted, but the seams may be single riveted except in oiltight work, where the riveting should be double. As stated in Table XI., the spacing of rivets connecting the inner bottom plating to frames and longitudinals should be 5 diameters. This close spacing is adopted because when the double bottom is under pressure the rivets will be in tension. The inner bottom plating is in this respect not so favourably placed as the outside plating, which is held against the frames by the pressure of the water.

Where angle bars, clips, or other parts are riveted to the upper side of the inner bottom plating, there is always a danger of leakage along the shank of the rivets and thence along the faying surface. Such leakage may be prevented by calking around the fixture, as in case of a stanchion, or else stopwaters must be used. The difficulty is most serious with three-ply riveting. The points of the rivets should everywhere be left convex especially in way of water and oil tanks.

CHAPTER XV.

THE DECKS.

56. INTRODUCTION.

1. **Nomenclature.**—The nomenclature used in different navies is given in fig. 150, which is self-explanatory.

The following additional remarks refer to the nomenclature used in the United States Navy.

A partial deck above the lowest complete deck and below the main deck is called the "half deck." If two armored decks are fitted, the heavier is called the "protective" deck and the lighter the "splinter" deck, irrespective of which is the upper of the two. These names are used in addition to the regular names: "second," "third," "fourth" deck, etc. Where a portion of the protective or splinter deck is sloped, the sloping portion is defined as the "inclined protective deck" or "inclined splinter deck." Where a protective deck is stepped a complete deck height, the respective portions are distinguished by means of the terms: "middle protective section" and "forward (or after) protective section" in addition to their regular names. The nomenclature used in the United States Navy is followed throughout this work.

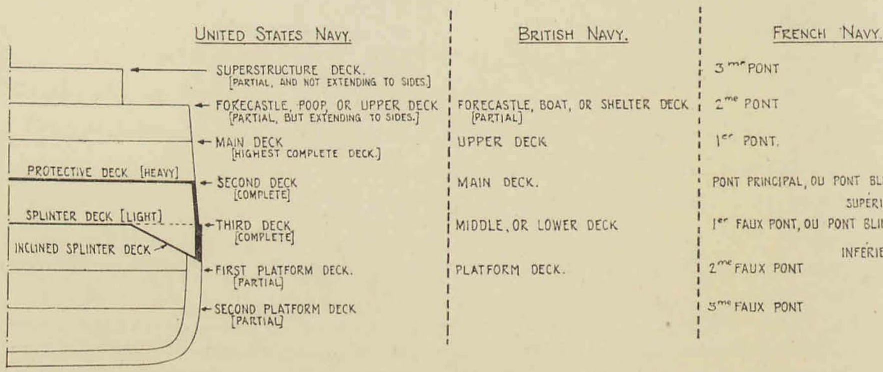


FIG. 150.—Nomenclature of Decks.

2. **Principal Functions and Claims.**—Let us consider first the principal functions of the decks and the features of construction which they necessitate.

(1) *The decks form rigid diaphragms, serving to stiffen the structure, transversely and longitudinally.* They should, therefore, be so constructed as to possess great rigidity in their own plane, and should be efficiently connected to the sides and to the bulkheads. Rigidity is best obtained by transverse stiffeners, so-called "beams," but in many cases also longitudinal girders are required. The function of the decks in supporting the sides of the ship against water pressures and general deformation is far less important than in merchant vessels, because the bulkheads are more numerous, but, on the other hand, the decks of a warship, where they butt up against the armored parts of the sides, fulfil the important function of forming a direct and rigid support for the armor. Torpedo-vessels are in this respect an exception. The bulkheads are here widely spaced and only one deck is fitted, which is of the same importance as in single-deck merchant vessels.

(2) *The decks in the upper part of the ship form the principal elements in the upper flange of the ship-girder.* The decks that are to perform this function and in particular the strength deck must be longitudinally continuous from end to end and should be, each of them, of uniform strength throughout, *i.e.* the strength should be apportioned to the straining forces. The connection to the sides should be efficient, for through it the sides transmit the longitudinal shearing which induces the tensile and compressive stresses in the decks (fig. 4). In fact, if this connection were incapable of transmitting the shearing forces, the decks would bend

without being stretched or compressed and would thus be valueless as flanges of the ship girder.

(3) *The decks serve the same purpose as the floors of a building*, constituting platforms which permit a full utilisation of the internal space. They should therefore be capable of carrying the load that is placed on them, for which purpose they must be stiffened by beams, and where heavy loads are concentrated, longitudinal girders must be fitted locally. The beams and girders transmit the load to the structures which support the deck, viz. the sides of the ship, the bulkheads, and the pillars. The lines of support should, as far as possible, be vertical, direct, and unbroken, being carried continuously down to the framing system of the ship. Unless absolutely unavoidable, no accumulation of load should take place by transmission from one deck to the decks below. This rule being observed, the decks may be constructed much lighter than the bottom to which the accumulated load of the entire weight of the ship is transmitted.

The protective decks present a special problem because they have to support the forces created by impact of projectiles.

(4) *The decks contribute to the watertight subdivision of the ship*, and should, therefore, be made watertight.

(5) *The weather deck serves the same purpose as the roof of a building*, affording shelter to the space below, thus making it habitable. In fact, this was one of the primary objects of introducing a deck in the early days of shipbuilding. The weather deck should not only be watertight, but should also be so constructed as to be a bad conductor of heat and sound. These latter claims are best fulfilled by fitting a sheathing of wood planks or a covering of linoleum on a steel deck, although sometimes, where a wood deck is fitted, the plating is partly or entirely omitted.

3. **Distance between the Decks.**—Where living rooms, working rooms, or passages are found, the decks should be not less than 6½ ft. apart, reckoned from the top of one deck to the under edge of the beams of the deck next above. A height of about 7 ft. is, in fact, desirable. This will permit the suspension of ventilating ducts, pipes, and leads under the beams, still leaving sufficient headroom. The total height between decks in the above-water part of the ship should, therefore, be from 7½ to 8 ft. Between the platform decks and in hold spaces, smaller heights are often found, when the total height available below the armor deck is not a multiple of the ordinary deck heights.

Where the sheer of the main deck is considerable, it is best to give the decks next below the main deck the same or nearly the same sheer, so as not to leave an excessive headroom at the ends in the upper part of the ship. Usually the extra space can be utilised to better advantage in the hold.

57. THE DECK PLATING.

1. **Material for Decks. Steel *v.* Wood.**—In most vessels, except those of composite construction, complete steel decks are laid, even on light superstructures and bridges. The first cost of a steel deck is smaller than that of a wood deck, it is easier to maintain watertight, and therefore less expensive to keep in repair. Steel decks, on the other hand, become slippery when wet, especially when covered with snow, they conduct heat better than wood decks and, therefore, transmit changes of temperature more quickly, and are more apt to cause condensation of water vapour. They also transmit sound better than wood decks. For these reasons steel decks are rarely used uncovered in warships. Decks exposed to the weather are on the unsheltered parts generally sheathed with wood and are elsewhere covered with linoleum, except where ammunition or stores are permanently stowed on the deck. The great objection to wood decks, viz. their inflammability, is to some extent removed when the wood is laid on a complete steel deck, because incipient fires will be then in most cases extinguished by the cooling effect of the plating and by the exclusion of air from the underside of the planks. In case of serious fires, however, when the plating is heated to a high temperature over large areas, and especially if the deck is demolished by shell explosions, the wood sheathing will burn completely.

In merchant vessels of the cargo-carrying type, decks are laid of steel plating without any kind of covering, except above the living spaces of officers and crew, where a wood sheathing is generally fitted.

2. **Watertightness.**—All decks should be worked watertight, whether to keep out rain and sea-water, or to prevent the water from leaking through when cleaning deck, or for the purpose of watertight subdivision. In some ships this rule is not observed in case of the second and third deck, where only a quasi-watertightness is attained along the sides by means of a mixture of cement and coke, but this does not appear satisfactory, especially at the ends of the ship, where these decks may become immersed in serious cases of bilging.

3. **Arrangement of Strakes and Plates.**—The deck plating is in warships generally worked flush, because this system facilitates fitting wood or linoleum on the deck. Where longitudinal strength is of importance, as in the main deck, the seam-strips should be worked continuous and should, therefore, preferably be placed on the upper side of

the deck. Where a wood sheathing is fitted, this will not cause any great difficulty, but where the deck is to be covered with linoleum, as may be the case inside a deckhouse, the strips should be placed on the underside of the deck. In order to avoid the use of liners, the strips are in such case worked intercostally between the beams. All decks below the main deck, that are covered with linoleum, have the strips so fitted. As explained above, the strips may then with advantage be made to serve as stiffeners for the lighter strakes of deck plating by giving them a tee-section. The butt straps are always placed on the underside, midway between the beams.

In decks used for stowage of coal, ammunition, or stores, where no covering is fitted, the raised and sunken system with overlapped butts is the most suitable. This system is generally used in the decks of merchant vessels of the cargo-carrying type, and may also be used with advantage in torpedo-vessels when no deck covering is fitted, but in important strakes the butts should be double strapped.

The deck plating is usually laid in longitudinal strakes, with the edges parallel with the center-line, excepting the outer strake on each side, the so-called "stringers," which follow the contour of the sides, forming a margin on which the inner strakes land. A glance at the stress diagrams of the deck on Pl. I. shows how well suited this arrangement is to take the forces to which a deck is subject in its capacity as the upper flange of the ship-girder. The stringers, together with the adjoining sheer strakes, form continuous girders from end to end of the ship, well adapted to transmit the strong shearing forces at quarter length and to resist the direct longitudinal forces amidships. The inner strakes, being parallel with the center-line, follow very nearly the lines of stress throughout the length of the ship, and the arrangement is simple and cheap. In large ships there are often two stringer strakes on each side.

In ships where the steel deck is not complete, stringer plates should always be laid and generally also so-called "funnel stringers" along the large hatch openings on each side. The main stringers and the funnel stringers may be connected by diagonal tie-plates, but in some ships the plating is worked complete from side to side amidships for one-third to one-half the length of the ship, tapering off into the main stringers at the ends.

The breadth of the plates is generally the same as that of the outer shell plating, in large ships about five feet. Often the stringer plates are wider, up to six feet or more, but they should taper towards the ends.

The length of the plates is likewise the same as for the outer shell. The butts are shifted on the same principles as in the outside plating, and a good shift should, in particular, be secured between the butts of the stringers and the sheer strakes.

The thickness of the plating on the different decks and the distribution of thickness on individual decks depend on various considerations which require to be discussed in detail.

4. **Choice of Strength Deck.**—It is desirable to construct the uppermost complete deck—the main deck—as the strength deck and to concentrate in it as much fore-and-aft material as practicable, since in this way the greatest strength of the ship-girder is obtained on a given weight of metal.

In vessels of the battleship class, however, it is best, as explained in SECTION 5, *6*, to choose as strength deck that which is located on top of the belt, usually an armor deck, and to make all decks above the belt very light. The French battleship *République* is constructed in this way (fig. 133). The second deck, which is here the upper armor deck, fitted on top of a complete armor belt about 7½ ft. above the water-line, is constructed of three thicknesses of 18 mm. (28 lb.) plating and is the strength deck of the ship. It is consequently by the French called the main deck (*le pont principal*). The plating on the main and upper (partial) deck is 6 mm. (9½ lb.) throughout, only the upper deck has a stringer strake 6 mm. + 8 mm. This construction seems rational and is even better suited to more recent types of battleships where the armor belt is higher than in the *République* and where, therefore, a greater protected depth of ship-girder can be obtained.

In light cruisers where there is no side armor or only a light partial belt the main deck should be the strength deck.

We shall begin by discussing the case of an unarmored strength deck whether fitted on top of an armor belt or entirely unprotected as in a light cruiser or a torpedo-vessel.

5. **Thickness of an Unarmored Strength Deck.**—A certain aggregate sectional area of the continuous strakes is required to secure sufficient longitudinal strength of the ship-girder, and is readily determined when the allowable working stress is given. In deciding upon the distribution of this area we must bear in mind that when the ship is upright there will tend to be a fairly uniform distribution of the lines of stress in all the strakes between the large hatches and the sides, as apparent from the stress diagrams Pl. I., but when the ship is rolling, the stringers will become the most strained parts. *It appears rational, therefore, to make all the strakes between the hatches and the stringers of a fairly uniform standard thickness and to make the stringers somewhat heavier*, but here as in case of the sheer strakes *the stringers, forming the*

margin strakes of the deck, should be of high-tensile steel in order to prevent progressive rupture. Judging from the strength calculations of existing ships, the standard thickness of the deck plating should probably be about the same as that of the outside plating and the thickness of the stringers the same as that of the sheer strakes. At present, however, this rule is not followed in practice. In most of existing warships there is a very marked difference in thickness between the stringers and the adjacent inner strakes, whence a great discontinuity in strength exists in the seam between these strakes. The stringer plates vary from 12½ lb. in destroyers to 50 lb. in battleships, while the adjacent inner strakes vary from 5 lb. in destroyers to 12½ lb. in battleships, although in some vessels the transition is less abrupt. When a deck so constructed is subject to tension or compression the stresses will be greatly increased in certain regions as explained in SECTION 8, *2*. Unless the light inner strakes are strengthened by longitudinal stiffeners or by a wood deck they can, in fact, hardly be relied upon in severe cases of straining, especially when in compression. These strakes are, moreover, in themselves, incapable of supporting even the ordinary normal load.

By giving to all the strakes outside the hatches a substantial thickness, as here recommended, these deficiencies are avoided and the plates will be able without special reinforcement to co-operate effectively with the longitudinal strength members to which, in certain cases, they are attached. Most important among such members are longitudinal bulkheads, if continuous and of great length and if, as in many torpedo-vessels, they extend from the bottom to the deck.

The non-continuous strakes between the large hatches, being of no importance in point of longitudinal strength, should be made as light as consistent with other claims. Generally, 5-lb. plating in destroyers and 10-lb. plating in battleships will be sufficient, but in order to support the ordinary load normal to the deck, stiffening angles must be fitted between the beams.

The thicknesses adopted for the plating of the strength deck amidships should be maintained for about three-fifths of the length, but towards the ends they may be reduced. The reduction is for the stringer plates generally very considerable, approaching fifty per cent., while the thickness of the inner lighter strakes remains unaltered. If the plating were of more uniform thickness, as here recommended, the reduction in thickness towards the ends might be applied to all the continuous strakes, and would not amount to more than from twenty to twenty-five per cent.

Locally, the strength deck should be reinforced by increasing the thickness or by doubling plates, where openings are cut in the continuous strakes. Heavier plates or doubling plates should also be worked round barbettes, in way of windlass, gipsy heads, winches, conning-tower supports, and other structures or fittings subject to heavy strains.

6. **Thickness of Plating in Decks below Strength Deck.**—The deck next below the strength deck, being nearer the neutral axis, is of minor importance in point of longitudinal strength, especially in ships where the strength deck is armored and protected by side armor. It may therefore be constructed of light plating, from 10 to $12\frac{1}{2}$ lb., but the margin strakes should be heavier so as to form girders for the support of the sides.

Platform decks may in general be constructed of 10-lb. plating, but in ammunition rooms it may be necessary to use greater thicknesses, depending on the weight of the ammunition and the spacing of the beams. The lower platform deck is liable to become subject to great water pressures, and forms virtually a third bottom to the ship. Its minimum thickness may be found from Curve B, Pl. IV., but it should not in any case have less than 12-lb. plating. Suppose, for instance, that a platform deck is to withstand a test head of 25 ft., and that the beams are spaced 4 ft. apart; we find then, from Curve B, assuming $K_A = 1$, $\mu = 125$, whence $t = \frac{48}{125} = \frac{3}{8}$ in.

According to Lloyd's Rules, the thickness of plating in decks on which cargo is stowed and where the beams are spaced 4 ft. apart, shall be not less than $\frac{1}{120}$th of the spacing, *i.e.* about 16 lb. If we allow $3\frac{1}{2}$ in. for the flange of the beams, we have $\mu = \frac{48}{44 \cdot 5 \times 120} = 90$. Since the plating ought not under ordinary stowage conditions to pass the elastic limit, we shall use Curve A in this case, which gives $K_A h = 14$ or, with $K_A = 1$, $h = 14$. Hence, it appears that the plating will be able to support a uniformly distributed load of 14×64 or about 900 lb. per sq. ft., without passing the elastic limit.

7. **Decks of Superstructures.**—At the ends of all superstructure decks, there will be a discontinuity in strength which it is difficult to bridge over without a great expenditure of material. In some cases longitudinal partition bulkheads exist inside the superstructure at the ends and may be utilised for this purpose. Ordinarily it is best, as explained in SECTION 42, not to include superstructures as members of the ship-girder. Their decks should, therefore, be lightly constructed and where necessary provided with expansion joints. Generally the plating is from 7 lb. to 10 lb. with margin strakes of from 10 lb. to 15 lb.

8. **Boundary Connections.**—The primary object of boundary connections is to connect the decks to the sides, casings, barbettes, and other structures, but at the same time they generally serve to secure watertightness. On the weather deck, where the frames terminate below the deck, as also around plated structures and coamings which pierce the deck, there is no difficulty in attaining these objects by simply working angle bars along the boundaries. Where the frames cut through the deck plating, watertightness can only be attained by smithing the angles so as to fit closely round the frames, thus obtaining a continuous calking edge. Angles so prepared are called "staple angles" or "collars"; their form depends primarily on the profile of the frames. Fig. 151 shows various types. It will be noticed that the stringer plates are cut out for the frames so that they can be fitted home in direct contact with the outside plating. Since points of weakness are thus produced, at which tearing of the plates is liable to start, reinforcement must be given to the edge of such stringer plates on strength decks. This may be done, as shown in fig. 151, by fitting a continuous "stringer angle" inside the frames and giving the staples a dished form.

In order to prevent water from passing between the frames and the shell plating from one side of the deck to the other, it is necessary to fit stopwaters behind the frames in way of the decks.

Platform decks should have double bounding angles, the upper stapled where necessary and calked, the lower intercostal and uncalked.

9. **Riveting.**—The rivets should have countersunk points on the upper side of the deck so as to leave a smooth surface. The efficiency of the joints in the strength deck should be such as to secure relatively the same longitudinal strength as in the outside plating, *i.e.* an efficiency of not less than about two-thirds. In the stringer plates the efficiency of the butts should be the same as along the beams, about ·83, but in light, fast vessels the strength of the rivets, moreover, should be well in excess of the strength of the plates, as explained for the sheer strakes, so as to secure absolute solidity of the joints.

The seams are usually single riveted throughout the decks, even on strength decks, but if the inner strakes are of substantial thickness, as here recommended, it is advisable to use double riveting in the seams between the stringer plates of the strength deck and adjacent strakes. The stringer angle, which connects the stringer strake to the sheer strake and transmits the shearing between these members, should likewise be double riveted, as is usually the case.

In platform decks both seams and butts should be double riveted.

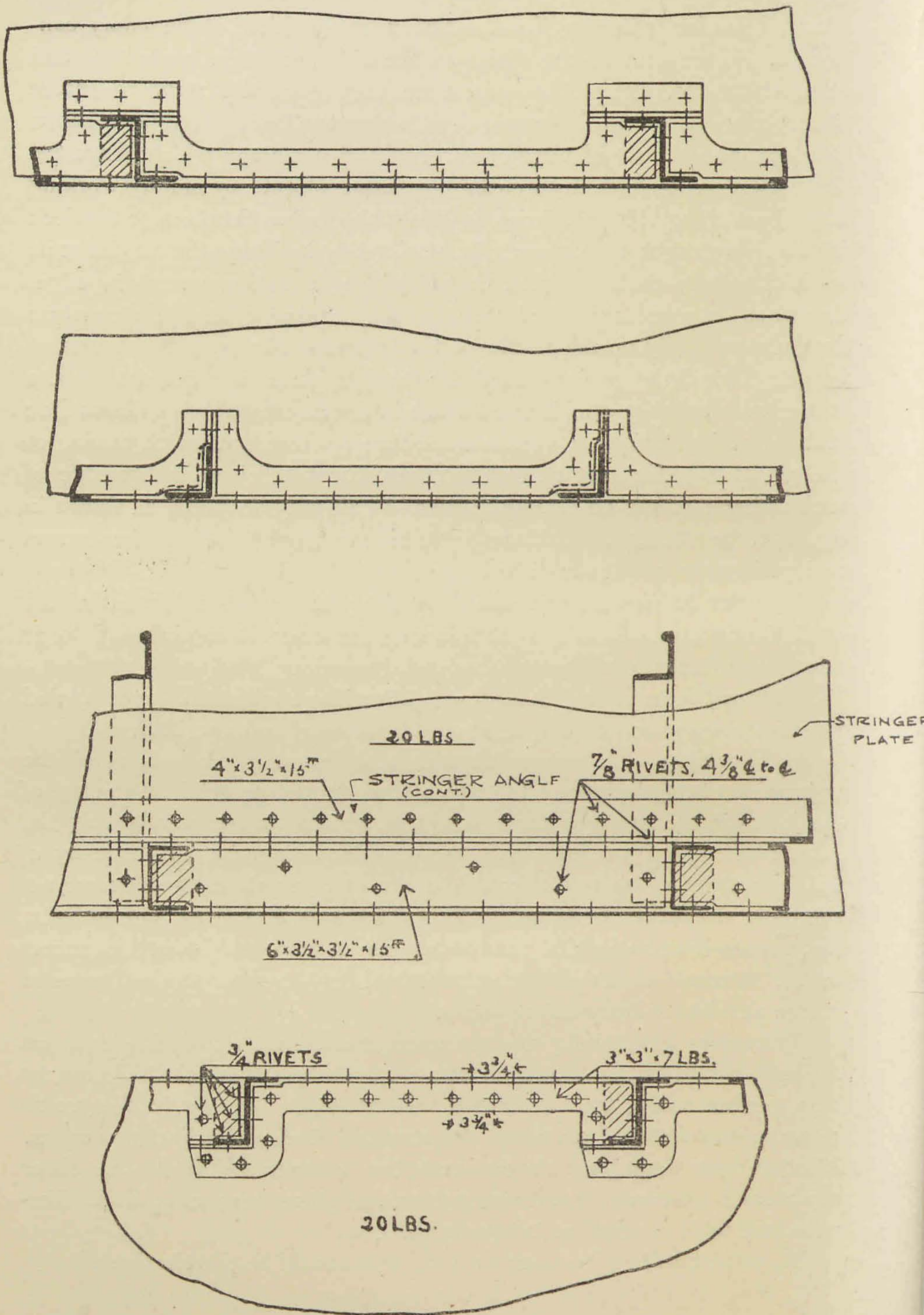


FIG. 151.—Staple Angles on Watertight Decks.

58. PROTECTIVE DECK PLATING.

1. **General Requirements.**—In most armored and protected warships a protective deck is fitted in the region of the water-line, generally sloping down at the sides. In battleships this deck is usually so near the neutral axis that it is of little or no importance as a member of the ship-girder. No special efforts, therefore, need to be made to obtain great longitudinal strength of this deck. In modern large armored vessels also the second deck—the English "main deck"—located on top of a more or less complete armor belt, is frequently armored. As explained above, this deck should then be made the strength deck and special efforts should be made to develop its longitudinal strength. An armored strength deck is also found in monitors and small monitor-like coast-defence vessels.

In all protective decks the butts and seams form lines of weakness which must be reinforced by riveted connections or by means of a rigid support, so as to enable such decks to fulfil their main purpose, to prevent projectiles or fragments and structural debris from breaking through and penetrating into the hold.

Besides longitudinal strength and the resistance to impact of projectiles and other attack, we have also to consider the question of watertightness, which in protective decks presents considerable difficulty.

2. **Protective Decks of the Sloping Type Fitted near the Neutral Axis.**—Such decks should be constructed of one or two layers of shipbuilding steel of small or moderate thickness, on top of which a layer of armor plates of greater thickness and of a harder material is fitted. In many ships armor plates are only placed on the slopes. The thicknesses of these various layers depend entirely on the type and size of the ship and on the system of protection. Generally the mild-steel plates are from 20 lb. to 30 lb. and the armor plates from 40 lb. to 100 lb. Protective decks are always worked flush.

If only one thickness of mild steel is used, the plates are connected by butt straps and seam straps on the underside. While ample structural strength may be secured in this way, the joints will not be well adapted to resist impact of projectiles unless the rivet area is made unduly large. Moreover, when the deck is struck by projectiles, the straps and strips are liable to be torn from the deck and projected into the hold. For these reasons it is preferable to use two thicknesses, one serving as edge connection for the other, whereby, further, a more uniform strength and a greater yielding capacity of the deck are attained.

On the slopes the seams should follow the deck-at-side line, to which also the knuckle line between the flat and the sloping part should be parallel. On the flat part of the deck the seams should run parallel with the center-line. The length and breadth of the mild-steel plates and the arrangement of the butts are as described for other decks. Frequently a model is constructed in order to study the lay-out of the plating, as in case of the outer shell.

The armor plates need not be given any connection at the butts or seams, but should be strongly and rigidly supported along these lines. Hence the butts should fall on the beams and preferably on bulkheads. The armor plates are, speaking generally, made of as large an area as practicable, but the dimensions should be chosen with due regard to the available lines of rigid support.

3. **Armored Strength Decks.**—Such decks are usually constructed of two rather heavy courses of nickel steel of equal thickness, from 30 lb. to 50 lb. In the French Navy three thicknesses are often used (*République*, 3 × 18 mm.). Where still greater thicknesses are used and, especially, where a harder material is employed, which is difficult or impossible to calk, a watertight deck, consisting of a single course of mild-steel plating, should be fitted under the armor plates.

The armor plates should always be so placed that one course may serve to connect the butts and seams of the other, but, if sufficient strength is not attained in this way, butt straps must be fitted in addition or the butts must be scarphed as shown in fig. 157. The strength required for the butt connections must be determined by the longitudinal strength calculation. In some cases butt straps are only fitted to the lower course, in others it may be necessary to fit butt straps to both courses, the straps of the lower being fitted under the deck, those of the upper on top of the deck. Where adequate strength can be attained by scarphing the plates at the butts this seems to be the best solution, since much weight is saved and the heavy butt straps are avoided.

4. **Connection of Protective Decks to the Sides.**—In battleships protective decks of the sloping type are generally continued to the side so as to form a shelf for the armor, or they are directly connected with the armor shelf. The frames as well as the outside plating are in such a case ordinarily discontinued below the protective deck and the outside plating is by a substantial angle bar connected to the lower course of the sloping deck (fig. 114).

In protected cruisers the outside plating is carried continuously past the edge of the sloping armor deck. If the frames are discontinued at this deck, the connection to the shell plating takes place by an angle bar fitted above or below the deck ; sometimes double angles are used. If the frames are continued through the deck, it is necessary to employ staples, but in order to avoid bevelled angles, the protective deck should

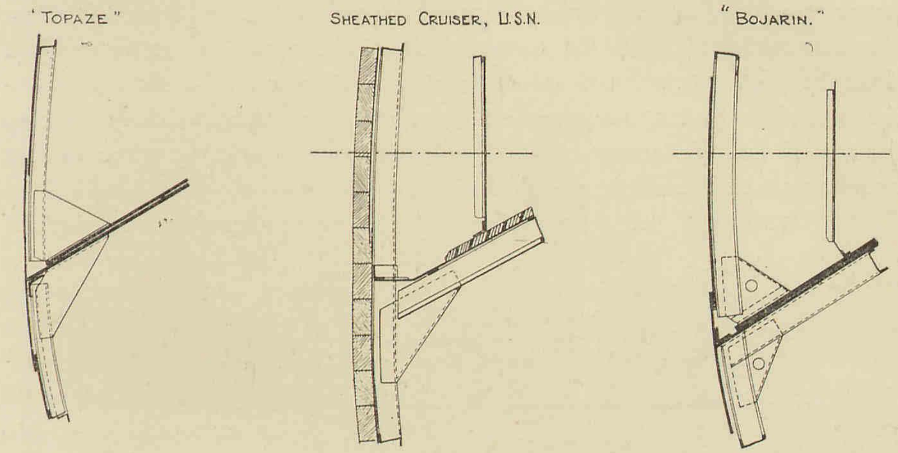


FIG. 152.

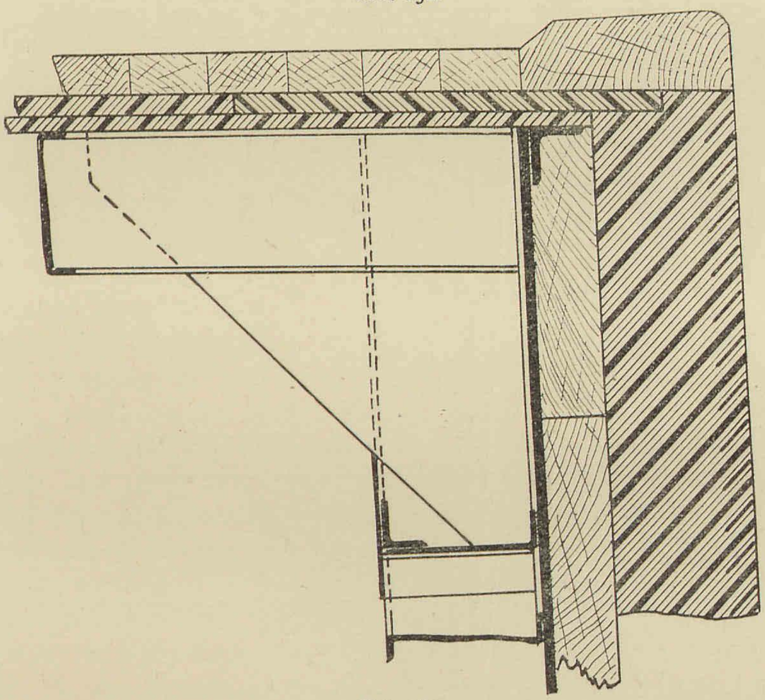


FIG. 153.

be flanged at the lower edge so as to meet the ship's side at right angles (fig. 152). The upper course of the sloping deck, which is usually a strake of armor plating, should stop at a distance from the side sufficient to allow calking of the angle bar.

When the armor deck is flat and placed on a level with the top of the side armor, the deck plates should be housed in a rabbet in the armor so as to give an efficient support to this latter. If a deck plate is not so rabbeted it should butt up directly against the armor with a close fit (fig. 153). A less satisfactory construction, which is sometimes used, is to let the upper course of the deck armor simply overlap the top of the side armor, while the lower course butts up more or less loosely against the plating behind armor.

5. **Riveting.**—Let us consider first the case where two courses of

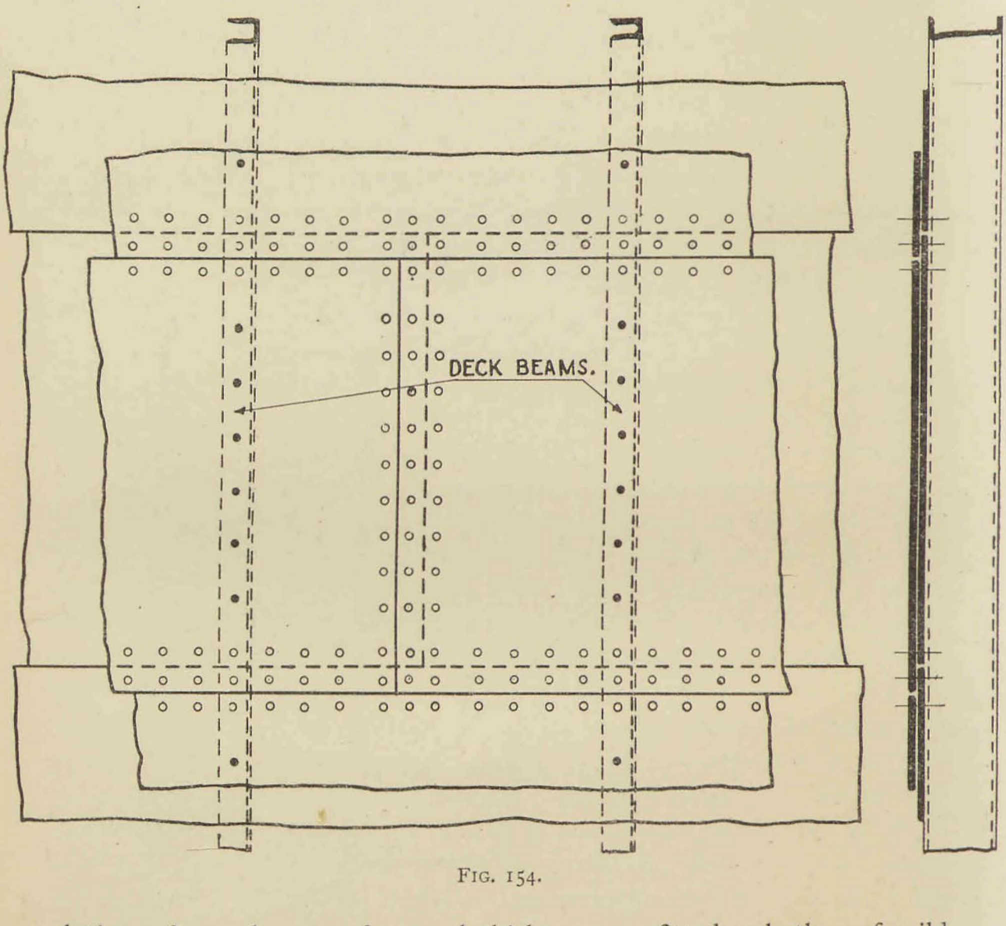


FIG. 154.

plating of equal or nearly equal thickness are fitted, whether of mild steel or some harder quality of metal.

The riveting of the seams is usually as in figs. 154 and 155, where three single lines of rivets serve to connect four plates. It will be noticed, that shearing of the middle row of rivets is sufficient to produce complete rupture, and that, hence, such seams form marked lines of weakness, which from the point of view of protection are very objectionable. In some ships also the butts are connected on the same principle, but this plan, although involving the minimum amount of riveting, is clearly unsatisfactory whether we regard the deck as designed for protection or for longitudinal strength. This arrangement is illustrated in fig. 154.

Ordinarily the ends of the plates are connected as in fig. 155, where the butts of the two courses are placed in different, generally adjacent beam spaces. In this case, in order that complete rupture shall take place within a certain double strake by shearing of the rivets, it will be necessary to shear two rows of butt rivets plus three rows of edge rivets for the length of one beam space. In general this mode of rupture will offer greater resistance than tearing of one of the plates along a line of rivets at the butts. Hence, shearing of the rivets is precluded and it is sufficient, as far as structural strength is concerned, to place the butts of the two courses one beam space apart as in fig. 155. The best result is obtained by placing both seams and butts of the two courses clear of each other as in fig. 156.

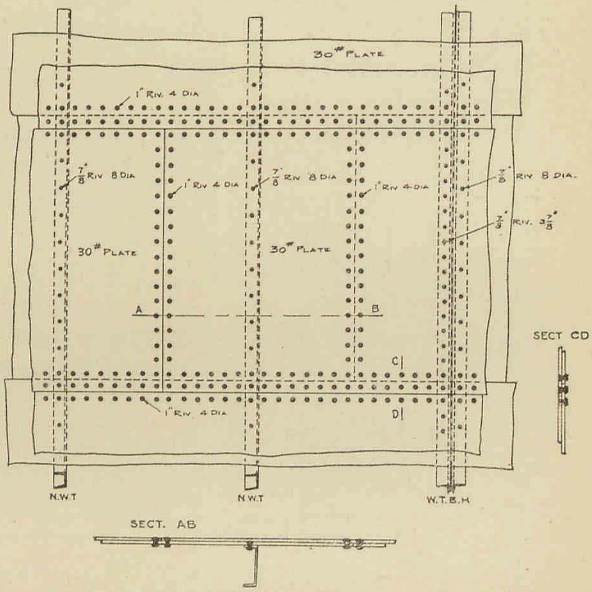


FIG. 155.

The spacing of the rivets along edges and butts is from four to five diameters, as required for watertight work, but depending somewhat on the thickness of the plating. Along the beams the spacing is in some navies from seven to eight diameters. Only the lower course is riveted to the beams. In the *République* the two lower thicknesses are riveted directly to the beams with a spacing of six diameters, giving a rivet area of about 10 sq. in. per ton weight of plates. The upper thickness is connected to the two lower by tack rivets.

Where the butts of the armor plates are scarphed, and where in addition a deck of ordinary mild steel is fitted under the two layers of heavy plating, as in fig. 157, the rivets of the scarphs always pass through the lower mild steel deck, which comes to form thus an additional reinforcement of the butt connections. The rivets in the butts of the upper course pass through all three thicknesses.

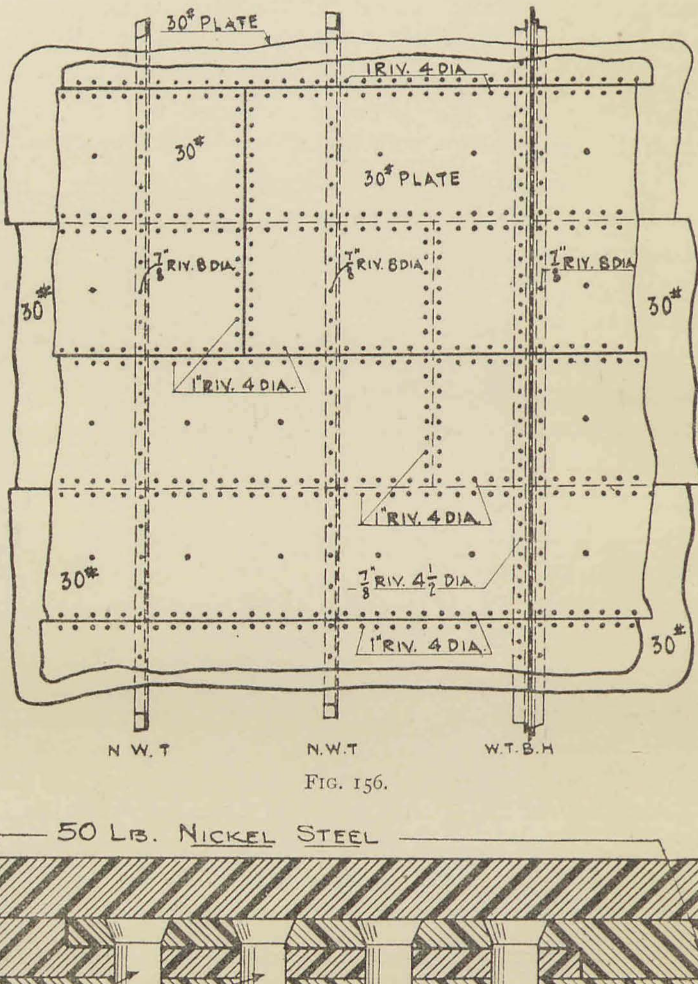


FIG. 156.

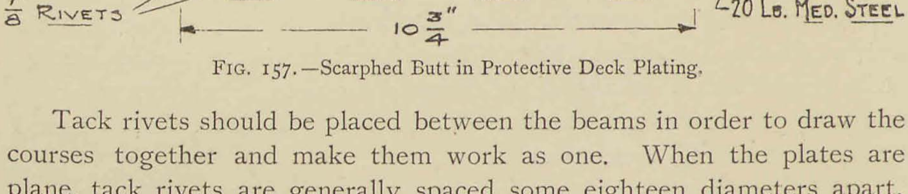


FIG. 157.—Scarphed Butt in Protective Deck Plating.

Tack rivets should be placed between the beams in order to draw the courses together and make them work as one. When the plates are plane, tack rivets are generally spaced some eighteen diameters apart, but where the plates are curved, they should be spaced closer together.

Where a single thickness of heavy deck armor is fitted on a watertight deck located near the neutral axis, as, for instance, the armor plates placed on the slopes of a splinter deck, there need be no butt or seam connections. The armor plates are secured to the watertight deck by heavy rivets, up to $1\frac{1}{4}$-in. diameter, spaced some eight diameters apart round edges and butts. When the armor is very heavy—more than 100 lb. per sq. ft.—it should be secured by tap bolts from underneath, similar to those used for light side armor.

The angle bar which connects the lower course of the sloping deck—the armor shelf—to the shell plating should be double riveted. The rivets should be not more than five diameters apart in the rows and arranged zigzag.

6. **Calculation of Longitudinal Strength for a Repeating Section of a Protective Deck.**—We shall here examine, as a typical

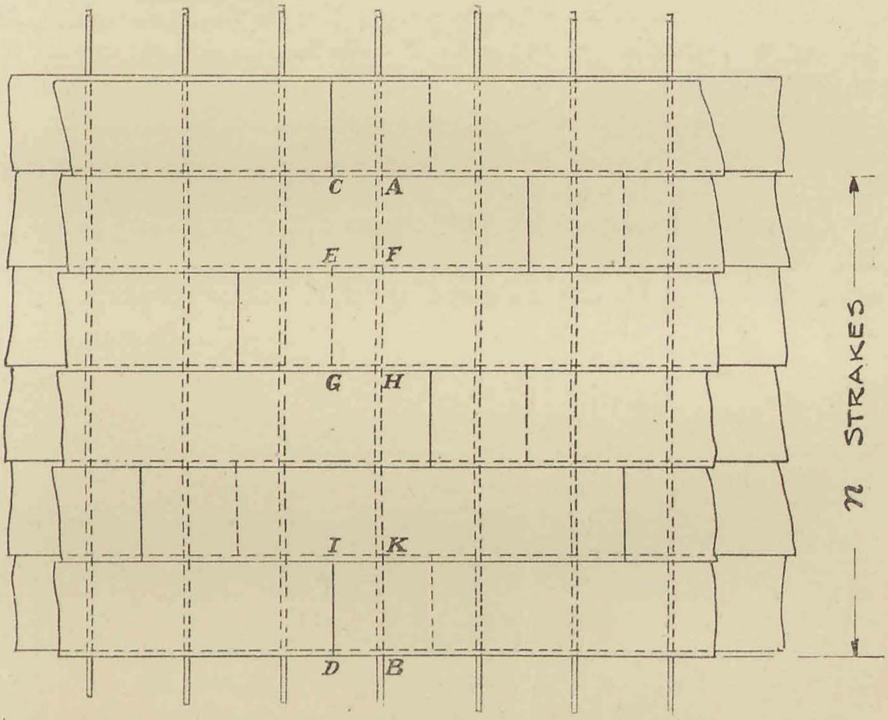


FIG. 158.—Rupture of Double Thickness Deck Plating.

case, the strength of a deck, consisting of two courses of plating of equal thickness.

Assume the riveting to be as in fig. 155 and the arrangement of the butts to be as in fig. 158. Each repeating section comprises n strakes and contains within any beam space one butt of the upper and one of the lower course. At every butt one strake is completely broken, while the other, acting as a strap, is weakened by the rivet holes along the butt. The efficiency of a plate of the upper course thus weakened will be denoted by v_U that of the lower course by v_L. Since the holes in the upper course are countersunk, while those in the lower are cylindrical, v_U is somewhat smaller than v_L. Following the same mode of procedure and using the same notation as for the outer shell (SECTION **39**), let us compare the combined strength of both courses of plating along a line through the butts CD with that of both courses along a beam AB

$$2(n-2)+v_U+v_L \geqq n(1+z)$$

$$\therefore \quad n \geqq \frac{4-(v_U+v_L)}{1-z} \quad . \quad . \quad . \quad . \quad (120)$$

Substitute numerical values corresponding to two courses of 30-lb. plating, worked as in figs. 155 and 158. We have $z = \cdot 83$, $v_U = \cdot 68$, $v_L = \cdot 73$, whence $n \geqq 15$. This value of n far exceeds the value in the present case, where $n = 5$, and is not obtainable in practice.

Rupture through the butts is, therefore, always more likely to occur than rupture along the beams.

Compare next rupture through the butts with rupture along the broken line AFEGHKIDB. Bearing in mind that y stands for the relative strength of the rivets to be sheared along the seams for pieces such as IK, DB, etc., we have

$$(n-2)(1+z)+4y+v_U+v_L \geqq 2(n-2)+v_U+v_L$$

whence

$$y \geqq \frac{(n-2)(1-z)}{4} \quad . \quad . \quad . \quad . \quad (121)$$

Substituting numerical values for 30-lb. plating as above, we find

$$y \geqq \frac{3 \times \cdot 17}{4} = \cdot 13$$

In fig. 155 we have the plate breadth $b = 65$ in., $d = 1$ in. and there are 6 rivets to shear along each of the seams IK, DB, etc. Thus

$$y = \frac{6\frac{\pi d^2}{4}\delta^2 f_S}{65 \times \frac{3}{4} \times f_T} = \frac{8d^2k}{65} = \frac{8 \times \cdot 704}{65} = \cdot 087$$

This is considerably smaller than the required value, and in general, even with a spacing of 6 diameters along the beams, it will be found that the inequality demanded by (121) cannot be attained. Hence the broken line represents the weakest section and the efficiency will be

$$e = \frac{(n-2)(1+z)+4y+v_U+v_L}{2n} = \cdot 73$$

Bearing in mind that the total thickness of the plating is $1\frac{1}{2}$ in., this efficiency actually means a very great strength of the protective deck, far greater than that attainable with ordinary decks. Where decks are constructed in the way here assumed, it will be, therefore, in general, unnecessary to fit butt straps, but since cases occur where they must be used it is of interest to examine their effect.

Suppose then that butt straps are fitted to both courses, applied directly to the respective plates above and below the deck, and riveted in such a way that the plates of the upper course are not weakened at the butts of the lower. Let the efficiency of the strapped butts be x and compare again the lines CD and AB

$$2(n-2)+2x+v_L+1 \geqq n(1+z)$$

$$\therefore \quad n \geqq \frac{3-2x-v_L}{1-z} \quad . \quad . \quad . \quad . \quad (122)$$

Assuming the butt straps to be treble riveted, we may reckon $x = \cdot 69$ whence $n \geqq 5\cdot 2$. This condition is here practically fulfilled, and rupture along the beams and through the butts will offer about the same resistance. Theoretically, rupture through the butts is likely to occur first and will, therefore, here be compared with rupture along the broken line AFEGHKIDB. We have

$$(n-2)(1+z)+4y+2x+2 \geqq 2(n-2)+2x+v_L+1$$

$$\therefore \quad y \geqq \tfrac{1}{4}[(n-2)(1-z)+v_L-1] \quad . \quad . \quad . \quad (123)$$

Substituting numerical values, we find $y \geqq \cdot 04$, which can be easily attained. Hence rupture through the butts is most likely to occur and the efficiency is

$$e = \frac{2(n-2)+2x+v_L+1}{2n} = \cdot 91$$

showing a gain of about 25 per cent. in strength as compared with the double deck without straps. The inconvenience, weight, and cost by fitting straps both above and below the deck would, however, be considerable.

The case here discussed represents ordinary practice; decks of different construction may be dealt with on the same principles.

7. **Watertightness.**—Whatever the construction of the protective decks, there are generally two and in some cases three thicknesses of plating and it is, therefore, necessary to take special precautions to secure watertightness. Nickel steel plates of the softer grades may be calked, but accurate workmanship is required in fitting the plates, since the edges must be very close together. Where a deck of shipbuilding steel is fitted under the armor plates, it should be always carefully calked.

On the sloping deck the armor plates are generally not calked, but stopwaters are used in order to prevent water finding its way between the armor plates and the lower courses and thence down along the shanks of the rivets.

On armored strength decks, constructed of two thicknesses of armor plates without any deck of shipbuilding steel being fitted underneath, practice differs. In some navies both courses are calked, the lower one with particular care ; in other navies only the upper course is calked. It is argued, that if both courses are calked, it is difficult to detect a leak ; water may percolate between the two thicknesses and leak through the lower course far from the leak in the upper course. In the British Navy, for instance, the lower thickness is left uncalked, but the edges of the upper thickness are calked against the lower ; stopwaters are very sparingly used, and when the upper thickness is complete, its edges and butts are all calked. In the Danish Navy tar-felt, interposed all over the faying surface between the two thicknesses, has given excellent results in connection with calking of the upper course. Red lead proved ineffective after a few years.

In all cases where a watertight deck of shipbuilding steel is fitted under the armor plates, only the upper course of these latter should be calked.

59. DECK BEAMS AND OTHER DECK FRAMING.

1. **Spacing.**—The spacing of the beams is generally the same as that of the frames, one beam being fitted on every frame, forming with it a continuous closed frame-ring in a transverse plane. In decks carrying ammunition or other heavy weights, it may be necessary to fit beams on every half frame space. In certain smaller vessels, especially where the frame space is small and where the decks are covered with wood, it may be sufficient to fit beams on every second frame.

2. **Different Types of Beams.**—Beams are usually made of one bar and consist in the important decks of bulb angles or channels, the latter finding most favor in warships of recent years. In some special cases

tee-bulbs and I-beams are employed or beams are built up of plates and angles. In lighter decks simple angle bars are used.

Channel bars have the advantage, as compared with bulb angles, that they lend themselves better to connection with the frames and to the fitting of girders and ceiling. They also possess a somewhat greater moment of inertia for the same weight, but, on the other hand, they suffer more than the bulbs from the defect of unsymmetrical form. Where certain channel beams are required to be of extra strength, reversed bars or face plates should be added to the lower flange. This mode of reinforcement is preferable to an increase in depth because it is desirable to maintain a constant depth of the beams throughout the deck. Moreover, the section modulus will be thus greatly increased since the deck plating that works with the beam will be better balanced and, finally, the symmetry of the beam section will be in some measure established. Sometimes two channels are riveted to each other back to back.

Tee-bars and I-bars do not lend themselves so well to connection with the frames. Tee-bulbs are, however, often used under wood decks where no steel deck is found, in order to facilitate fastening of the planks. I-beams are used in special cases where great strength is required. Where tee- or I-beams meet the frames, the flange on one side of the beams must be removed in way of the frames.

Beams built up of plates and angles are seldom used in warships. In merchant vessels they are often fitted as "strong" beams, placed at wide intervals in the holds or machinery spaces, where it is not desirable or possible to have closely spaced beams. No decks are fitted on such beams, but broad stringer plates are riveted to the beam ends in the hold in order to distribute the thrust or pull of the beams over the sides. Since the main object of "strong" beams is to provide struts or ties between the sides of the ship, and thus increase the stiffness of the structure, it is not, generally, necessary to fit them in warships where the transverse bulkheads are more closely spaced. Sometimes, however, it may be desirable to fit beams of extra strength across large hatches. More often built-up beams are fitted in warships as girders for the purpose of supporting gun-turrets, uptakes and funnels, and other heavy concentrated weights.

3. **Strength.**—In determining the strength and hence the scantlings of the beams, we must rely largely on practice and experience, since the forces acting on the beams are in most cases difficult to estimate. Theoretical considerations and empirical calculations are not, however, without interest.

The two primary functions of the beams are to stiffen the decks so as to enable them to resist forces in their own plane, and to strengthen them so that they can support the normal load to which they are subject. The transverse forces acting in the plane of the decks are difficult or impossible to estimate, but experience seems to show that if the decks are constructed strongly enough to carry the normal load, they will also possess ample transverse stiffness. The longitudinal stiffness of the decks will be considered separately hereafter. We are, therefore, for the present concerned only with the normal load to be carried by the decks.

4. **Normal Load.**—In estimating the magnitude of this load we may disregard permanent heavy and concentrated weights, which should always be supported independently and transmitted directly to the framing system of the ship. There remains the load due to the crew, occasional transport or stowage of weights, the blast of guns, and the action of the sea.

The greatest load due to men gathering on the deck may be estimated, as in buildings for public assembly, at 150 lb. per sq. ft., which should be provided for as a minimum on all decks where men may gather. This load may be considered sufficient for bridges and superstructure decks and decks in officers' and crew's quarters, in so far as they are not exposed to being charged more heavily in other ways.

The load on decks subject to temporary stowage of such weights as ammunition, coal, and provisions or other heavy stores, may be estimated to reach 250 lb. per sq. ft., which is considered as a proper requirement for a warehouse floor. In order to allow for the inertia forces when the ship is moving in a seaway, an addition of about 15 per cent. should be made, bringing the load on such decks up to about 300 lb. per sq. ft.

The effects of gun blast need only be considered on weather decks. When a gun is fired over a deck, great pressures and sub-pressures will come to exist in way of the blast cone. We know little about the magnitude of these pressures, but considerable downward deflections of the deck beams have been observed by the author in some cases, while in other cases the rivets connecting the deck stanchions to the beams have sheared under the effort of the deck to lift. Considering that a load of 300 lb. per sq. ft. corresponds to only one-seventh of an atmosphere, it will be seen that the load due to this cause, whether acting as a pressure or a sub-pressure, may easily exceed that which we have assumed for temporary stowage of weights. On the other hand, this action is local and rarely extends over more than a fraction of the length of each beam.

By the shipping of seas there will be a temporary accumulation of water on the weather deck, and the static load will in many cases be accompanied by a considerable dynamic pressure depending on the normal velocity with which the water strikes the deck. A layer of water about five feet deep produces a static load of 300 lb. per sq. ft. Here, again, we are unable to make any exact estimate of the pressures, but it is clear that the chances of shipping large quantities of water are much greater in ships of low freeboard than in ships of high freeboard. A recent case of a cargo steamer, s.s. *Gorm*, which foundered in the Bay of Biscay on her first voyage, the fore-deck being crushed in by the sea, shows how serious this question is for low-freeboard vessels.

Only weather decks are exposed to the effects of gun blast and shipping of seas, but are in warships usually provided with a wood sheathing, which adds considerably to the strength and serves to distribute the load over greater areas. The reinforcement which the decks receive in this way is difficult to estimate but, judging from experience, it may be assumed to provide a sufficient margin of strength for the indeterminate effects here considered. For sheathed decks exposed to these actions we shall, therefore, reckon the same load as for other decks —ordinarily 300 lb. per sq. ft.—disregarding in the strength calculation the effect of the wood deck. Where guns of large caliber are to be fired over a deck, local reinforcements should be applied, however, as dictated by experience.

On decks where ammunition, coal, or stores are permanently stowed, the load may reach much higher figures than here estimated for other decks, but can generally be calculated with a fair degree of accuracy. An average load of 1400 lb. per sq. ft. is known to be reached in certain ammunition rooms.

5. **Strength Calculation of Beams.**—In calculating the moment of inertia, a strip of deck plating of a width equal to $30t$ should be included as usual. As stated above, when a steel deck is sheathed with wood, its strength and stiffness will be greatly increased. The planks act partly by the support which they give to the steel plating, preventing it from shirking its duty by bulging, partly in a direct manner by the resistance which the wood offers to compression. Evidently the latter resistance is very difficult to estimate since it must depend on the condition of the calking of the seams. To the author's knowledge no experiments bearing on this problem have been undertaken, and we shall, therefore, not attempt to include the wood deck in the calculation. The local strength of the end connections of the beams will be discussed separately hereafter; for the present it suffices to state that we may regard the beams as fixed at the ends. Assuming the intermediate points of support, whether pillars or longitudinal bulkheads, to divide the beams in a number of

equal parts, and the load to be uniformly distributed, we may regard the beams as fixed also at these points.

Let l be the length of an unsupported section of the beam, w the load per unit length, S the section modulus, then the maximum stress will occur at the points of support and will be given by

$$p = \frac{wl^2}{12S} \quad . \quad . \quad . \quad . \quad . \quad (124)$$

from which it appears that for a given unit load and a given allowable stress, the section modulus should be proportional to the square of the unsupported length of beam. Or, with a given beam and an assumed standard load, there must be a certain maximum spacing of pillars or other points of support which should not be exceeded. This is the rule adopted both in warships and merchant vessels.

Example 1.—A deck in a battleship is constructed of beams: 10 in. × $3\frac{3}{8}$ in. × $3\frac{3}{8}$ in. × 21·8 lb. channels, spaced 4 ft. apart, and covered with $\frac{1}{2}$-in. plating. It is subject to a uniformly distributed load of 300 lb. per sq. ft. The allowable stress is 7 ts. per sq. in. Find the maximum spacing of the points of support.

We have $$w = \frac{4 \times 300}{2240 \times 12} = \frac{1}{22 \cdot 4} \text{ ts. per in.}$$

Including in the effective section of the beam a strip of plating $30 \times \frac{1}{2} = 15$ in. wide we find S = 23·8 sq. in. × (in.), whence

$$l^2 = \frac{12Sp}{w} = \frac{12 \times 23 \cdot 8 \times 7 \times 22 \cdot 4}{144} = 311$$

$$\therefore \ l = 17 \cdot 6 \text{ ft.}$$

If the effective strip of plating is reckoned to be $50t$ = 25 in. wide instead of 15 in. in order to allow for the stiffening effect of a wood deck, we find S = 24·5 and l = 17·9 ft. The width of the strip has therefore but small influence on the result in this case; whether we reckon it $30t$ or $50t$ the maximum length of unsupported beam is found about 18 ft., which corresponds well with ordinary practice.

Example 2.—The deck of a shell room for 14-in. ammunition is constructed of 10 in. × $3\frac{3}{8}$ in. × $3\frac{3}{8}$ in. × 21·8 lb. channels, spaced 2 ft. apart and covered with $\frac{3}{8}$-in. plating. The beams may be considered as fixed at the longitudinal bulkheads, which are 12 ft. apart. The shells weigh 1400 lb. each and are stowed uniformly, 24 projectiles on each beam space, but a passage, 2 ft. 6 in. wide, is left in the middle. Find the maximum stress in the beams.

Assume that 24 projectiles are distributed uniformly along each beam space, which appears to be the most severe probable condition. We have then

$$w = \frac{24 \times 1400}{144 \times 2240} = \cdot 104 \text{ ts. per in.}$$

Including a strip of plating $30 \times \frac{3}{8} = 11\frac{1}{4}$ in., we find S = 24·2 sq. in. × (in.), whence

$$p = \frac{\cdot 104 \times 144 \times 144}{12 \times 24\cdot 2} = 7\cdot 4 \text{ ts. per sq. in.}$$

which is about at the allowable limit.

In Lloyd's Tables for determining the scantlings of beams, the length is used as argument. The spacing of the frames is therein virtually implied and separate tables are given for beams fitted to every frame, for beams fitted to alternate frames, and for widely spaced "strong" hold beams. The scantlings may be selected from three different columns, corresponding to the cases where one, two, or three rows of pillars are fitted. The unsupported length varies from 8 to 20 feet.

Consider, for example, a beam 76 ft. long supported by three pillars. From Lloyd's Tables we find the scantlings : 10 in. × $3\frac{1}{2}$ in. × $3\frac{1}{2}$ in. for a channel bar with ·50-in web and ·575-in. flanges. Suppose the spacing of the beams to be 2 ft. 6 in., and the deck plating to be $\frac{1}{2}$ in. thick. A cargo of density 40 cb. ft. per ton is stowed 7 ft. 6 in. high on the deck, corresponding to a load of 420 lb. per sq. ft. Find the maximum stress in the beam.

$$l = 19 \text{ ft.} \qquad w = \frac{2\cdot 5 \times 7\cdot 5}{40 \times 12} = \cdot 039 \text{ ts. per in.}$$

S = 30·5 sq. in. × (in.), including a strip of plating 15 in. wide, whence the maximum stress will be :

$$p = \frac{\cdot 039 \times \overline{19}^2 \times \overline{12}^2}{12 \times 30\cdot 5} = 5\cdot 5 \text{ ts. per sq. in.}$$

Bearing in mind that a somewhat heavier load than here assumed may be stowed on the beams, this result shows that the method of calculation and the assumptions which it is here proposed to apply to the beams of warships agree well with the practice in merchant ships.

6. **Uniform Scantlings Adopted in Practice.**—The assumptions that the load is uniformly distributed and that the points of support are equidistant are not, of course, exactly fulfilled in practice, wherefore the maximum bending moment may exist between the supports instead of at the supports. As long as the distance between the supports does nowhere

exceed the maximum determined above and as long as the load is not too strongly concentrated, these varying conditions are probably met satisfactorily by *making the beams of uniform section throughout their length*, the strength being apportioned to the maximum bending moment as determined above. The beams of a deck should, moreover, as a general rule, be made of *the same depth throughout the length of the ship*, because then the hanging of pipes and ducts is simplified and can be carried out with the minimum loss in headroom. Only at the ends of the ship, where the breadth falls appreciably below the allowable unsupported length of beam, it may sometimes be of advantage to use beams of smaller depth and lighter scantlings than elsewhere. Such reduction in strength of the beams should not, however, be made where the deck is exposed to the blast of heavy guns.

7. **Beams under Protective Decks.**—Protective decks are heavy in themselves and the lower protective deck, in particular, is loaded with great weights, transmitted to it from the decks above through pillars, bulkheads, and casings, while other weights, such as uptakes and funnels, coal, etc., are placed directly upon it. It is not everywhere possible to carry the lines of support directly down to the framing system, and where this condition exists the beams must be assisted by longitudinal girders or extra-strong beams must be fitted.

The support to be given to protective decks as such, depends on the quality and thickness of the armor and on the way in which it is expected to resist attack. In any case, the decks should be so supported as to be capable without structural breakdown to withstand the forces due to the impact of projectiles when ricochet on or perforation of the deck occurs. With the soft and rather tough quality of protective deck armor used at present, it is generally considered desirable not to make the support very rigid. Hence we find that the beams of the protective deck are usually of the same scantlings as those of the main deck. Whether the support, thus given to the deck armor, is in all cases sufficient appears doubtful, but this question can only be settled by experiments.

8. **Crown of Beams.**—The beams of decks that are entirely or partly uncovered, and thus exposed to the action of the weather and the sea, should be given a slightly vaulted form in order to facilitate the water running off the deck into the side gutters. Other decks should be flat except perhaps the second deck in large ships, where great quantities of water are often used for cleaning purposes. The influence of this feature on the strength of a beam is probably insignificant because of the flatness of the arc and because the sides of the ship are not sufficiently rigid to act as abutments. The amount which the arc of the beam rises above

its chord is called the "crown" or "round-up." It varies from $\frac{1}{50}$ of the length of the beam in small warships to $\frac{1}{100}$ in large warships. The crown of the second deck is generally smaller, about one-half of that of the main deck. Lloyd's Rules prescribe a crown of $\frac{1}{48}$ of the beam. The curve to which the beams are bent is, ordinarily, a circle, and in order that the deck may form a fair surface, the radius of this circle should be the same for all the beams. Hence it is sufficient to construct the curve for the maximum beam; the arc of other beams will be portions of this curve.

The construction may be performed as indicated in fig. 159. AB represents the half beam, AC the crown. ABDC is a rectangle. AB and DB are divided into the same number of equal parts.

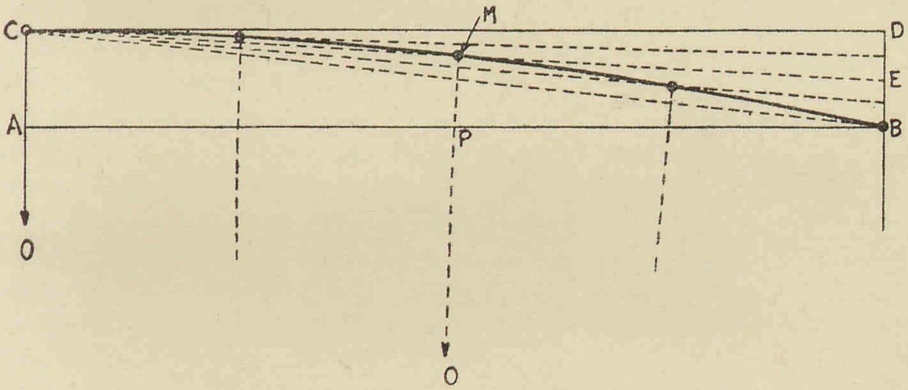


FIG. 159.—Construction for Camber of Deck Beams.

Straight lines are drawn from C to the points of division on DB, and perpendiculars are dropped on these lines from the points of division of AB. The foot of these perpendiculars will be points on the desired circular arc. In order to prove this proposition, consider any one of these perpendiculars, such as PM, dropped normally to CE, and produce MP to cut CA produced in a point O, not shown on the diagram. If, now, we can prove that OA is a constant, then M lies on the circle constructed on CO as diameter since CMO is a right angle.

Δ OAP is proportional to Δ CDE, whence: $\frac{OA}{CD} = \frac{AP}{DE}$ but by the construction: $\frac{AP}{DE} = \frac{AB}{DB}$ $\therefore$ $OA = CD\frac{AB}{DB}$ which is a constant.

9. **End Connections.**—The connection between beams and frames has to resist the following actions:

(1) Racking or distorting strains when the ship is moving in a seaway, producing alternating couples in the beam ends. In warships the

beams under all decks up to and including the second deck are generally relieved of these strains by the bulkheads, but the main and upper decks are not always so well stiffened and often support heavy weights which cause great lateral inertia forces when the ship is rolling. Hence, the racking strains at the beam ends may here be considerable and will necessitate great solidity in the connections.

(2) Bending and shearing due to normal load on the deck.

(3) Direct thrust, end on to the beams, behind side armor, when this is struck by projectiles. The beams here act as struts supporting the

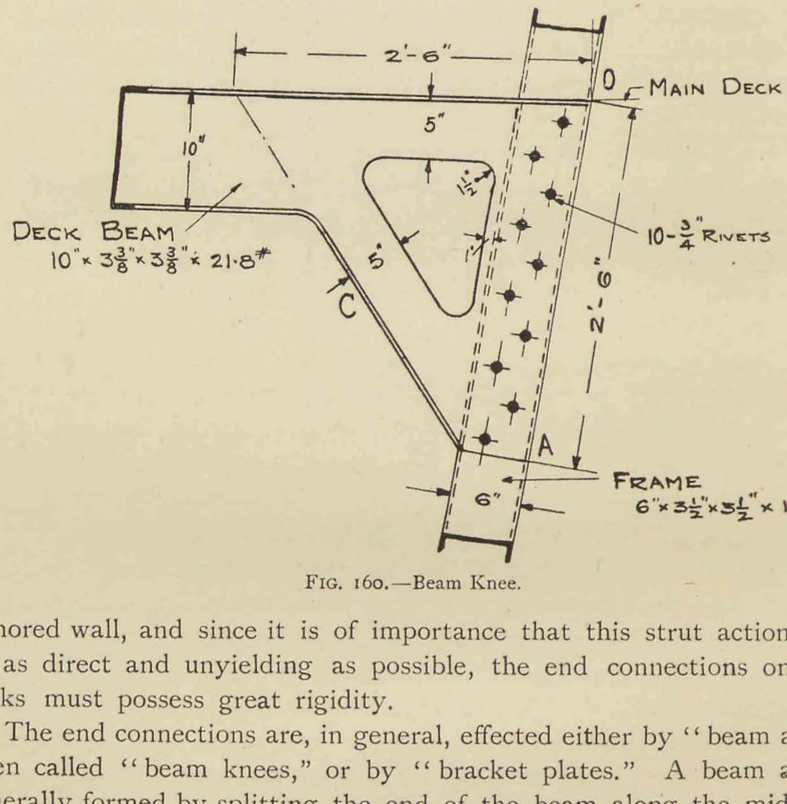


FIG. 160.—Beam Knee.

armored wall, and since it is of importance that this strut action shall be as direct and unyielding as possible, the end connections on such decks must possess great rigidity.

The end connections are, in general, effected either by "beam arms," often called "beam knees," or by "bracket plates." A beam arm is generally formed by splitting the end of the beam along the middle of the web, turning down the lower part, and welding a piece of plate in between. Often a hole is left at the center of the triangle so formed in order to save weight. Brackets are triangular plates, riveted to the beams and generally flanged on the outstanding edge; their thickness is the same as that of the web of the beam. Brackets are usually inserted between the beam and the frame or they may be joggled. The principal dimensions of beam arms and brackets are the depth OA, and the width across the throat, OC in figs. 160 and 161. The depth determines the number of rivets connecting the arm or bracket to the frame and is generally from $2\frac{1}{2}$ to 3 times the depth of the beam itself. The width across the throat should be sufficient to prevent collapse at this point; it must be greater in a bracket, when the edge is not flanged, than in a beam arm where the beam is provided with a bulb or a lower flange. According to Lloyd's Rules this width should be not less than six-tenths of the depth of the beam, *i.e.* not less than 1·5 to 1·8 times the depth of the arm. When, as is ordinarily the case, $\frac{3}{4}$-inch rivets are used, it may be considered good practice, with bulbs and channels of usual profile, to place one rivet for every inch of depth of the beam in the connection to the frame. With zigzag riveting and a depth of beam arm equal to three times the depth of the beam, this gives a distance of eight diameters between the rivets in the rows.

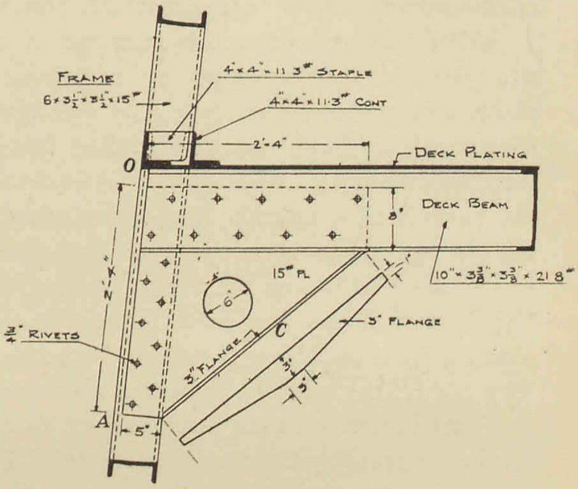


FIG. 161.—Beam Bracket.

Wherever possible, the beams should be connected directly to the frames by one or two rivets taken through beam, bracket and frame. Deep plate frames and floors may take the place of brackets, provided the overlap of the beam is sufficient to take the required number of rivets. Thus, protective deck beams generally cut through the inner bottom and connect directly with the floors. Platform deck beams should not pierce the inner bottom, but should be connected to it by brackets, whence the respective strake of the inner bottom should be of a somewhat increased thickness.

10. **Strength of End Connections.**—Comparing the strength of beam arms and brackets, we have to consider first the riveted connections. Since a bracket can always be given, and should always be given, the same number of rivets and the same length along the beam as along the frame, it would appear that the bracket connection could always be given the same strength as the arm connection. The arm connection is, however, always superior in that it is not liable to the slipping which may occur in the rivets that connect the brackets to the beams. In case of decks placed behind side armor, an almost perfect strut action can be

secured with beams provided with arms by bringing the arm in close and intimate contact with the side plating. Beam arms should, therefore, be used under all decks which serve to support side armor and wherever great stiffness of the connection is required, as under the main deck. Beam arms are, moreover, often used where a finished appearance is considered of importance. They are somewhat more expensive than brackets.

With the end connection constructed as described above, the beam will yield by bending long before simple shearing of the rivets takes place, and we need therefore only consider the former of these actions. The strength of the beams in bending has already been discussed for the intermediate points of support. At these points the strength resides in the beam itself assisted by the deck plating, but at a beam arm or bracket the strength depends on the rivets that connect the arm or bracket to the frame, assisted by the rivets that connect the stringer plate to the bounding stringer angle. An investigation of some typical cases seems to show that the ultimate strength of this riveted connection is, ordinarily, about fifty per cent. greater than that of the beam.

This excess of strength is justified by the requirement of solidity and by the alternating nature of the actions to which the connection is exposed. Slipping of the rivets is in this case highly objectionable. It is difficult to prove that the construction here recommended for the end connection of the beams is always the one best proportioned to meet the requirements, but it is in accordance with practice, and experience seems to show that it is at least sufficiently strong. In view of the rigidity of the ship's side and the great strength of the end connections, it appears that we are justified in regarding the beams as fixed at the ends, as assumed in the foregoing.

11. **Continuity of Beams.**—Ordinarily beams are worked in one length from side to side if not interrupted by openings in the deck, but where, for some reason, the beams are built up of two lengths, the connection is usually effected by double butt straps on the web and a single butt strap on the lower flange, if any. On the upper flange the deck serves as a strap. All straps should be treble riveted, and the scarphs should be shifted well clear of each other. Scarphs are sometimes used on the protective deck beams in ships with sloping armor deck, in which case there is a sharp knuckle line between the flat and the sloping part. In most navies, however, the beams are bent to shape, forming a gently rounded knuckle. The beams should always, where possible, be carried continuously across the ship from side to side, but in way of hatches and barbettes they must be severed. Such partial beams are called "half-beams."

12. **Carlings and Coaming Plates.**—The sides of smaller hatches are framed by longitudinal bars of the same section as the beams, so-

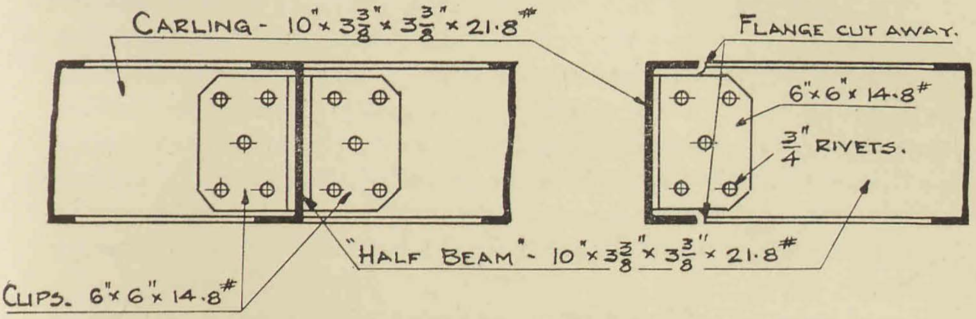


FIG. 162.—Connection of Deck Beams to Carlings.

called "carlings," against which the half-beams butt up, and to which they are connected by double angle clips. The carlings, which transmit the load on the half-beams to the nearest continuous beams, must be efficiently connected to these latter, usually by double clips. Where several beams are cut, stanchions should be fitted at the corners of the framework round the hatch. Ordinarily, the length of the hatches in a fore-and-aft direction is made a multiple of the beam space, so that the beams may form part of the bounding framework. It may often be necessary to reinforce such bounding beams by reversed angle bars.

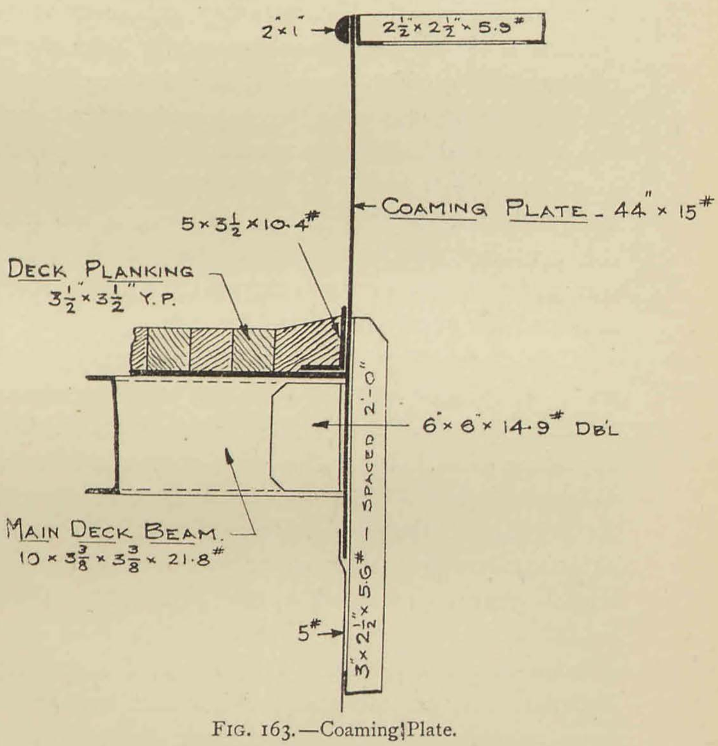


FIG. 163.—Coaming Plate.

In very large hatches, such as boiler- and engine-room hatches, a carling of the same section as the beams would not be capable of transmitting the load of the half-beams to the points of support, unless it could be supported by several pillars. Generally, the presence of a casing prevents the use of pillars and makes it profitable to introduce, instead of the carling, a deep and rather heavy "coaming plate," which forms part of the casing. The coaming should be connected to the deck plating by a substantial angle, and the half-beams should be connected to the coaming by double clips. The stiffeners of the casing should be vertical and connected directly to the coaming plate so as to give an effective support to this latter. The coamings on the protective deck, to which the load on the upper coamings is transmitted, should be heavier and supported by hold stanchions where possible ; they should be connected to the deck by double angles.

13. **Longitudinal Framing of the Decks.**—The need for longitudinal girders under the decks arises chiefly from the claim to longitudinal stiffness. Strength can be secured by simply providing a sufficient sectional area of metal in the plating, but stiffness can only be obtained by the addition of girders. Several other reasons exist for the use of longitudinal girders. Where beams are severed by large hatches, it may be necessary to introduce girders, spanning the distance from bulkhead to bulkhead, and similar girders may be required under heavy concentrated weights when pillars or bulkheads cannot be fitted. Under secondary guns, windlasses, and other weights of moderate magnitude, especially where dynamic actions occur, shorter girders are used, of sufficient length to distribute the load over several beams. When the deck plating is so light as to be incapable of supporting the ordinary normal load, it must be locally stiffened between the beams by light girders.

The claim to general longitudinal strength and stiffness is of course the one to be first and foremost considered, and by providing these qualities in a sufficient measure other claims are frequently satisfied at the same time. Again, in this case, conditions differ so much in the different classes of ships that each must be discussed separately.

(1) *Battleships.*—The strength deck of battleships usually possesses sufficient strength without the addition of longitudinal girders, and stiffness is generally secured by the deck being either sheathed with wood or armored.

Where no wood sheathing is fitted and where the plating is very light, as may be the case for instance on the main deck inside superstructures and on the second deck, light stiffeners are fitted under the deck, usually consisting of longitudinal angle bars worked intercostally between the beams. In French vessels these angles are stapled down on the web of the beams as shown in fig. 164, but in certain other navies tee-bars are used which at the same time serve as edge strips and which are not connected to the beams. The object of these stiffeners is primarily to enable the plating between the beams to support the normal load without too great deflection, but they at the same time render the plating more effective under compressive forces.

The lower protective deck is always heavily loaded, as explained above, and cannot everywhere be adequately supported by pillars or bulkheads, especially in engine- and boiler-rooms. Generally a rigid line of support is provided along the center-line, either by a bulkhead or by a central passage, but between the center-line and the side bulkheads it is often difficult to provide direct supports in the desired locations. Where this condition exists it is advisable to work a substantial longitudinal girder under the deck on each side midway between the center-line and the side bulkheads, supported by stanchions as far as possible, but otherwise capable of transmitting the load to the adjacent transverse bulkheads. In many ships a light bulkhead, pierced by doors, is found on each side between the protective and second deck, enclosing the uptakes and engine hatches and forming the inner wall of long passages, and in this case the girders should be worked directly under them, for although such bulkheads do not possess much girder strength they are capable of transmitting a considerable load from above.

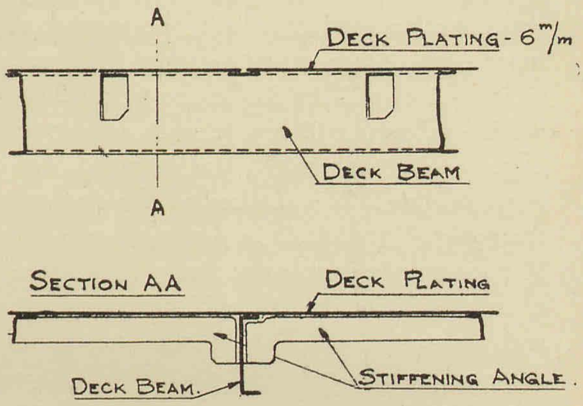


Fig. 164.—Stiffening of Light Deck Plating.

(2) *Light Cruisers.*—In light cruisers the longitudinal strains are greater than in battleships and a more complete utilisation of the available fore-and-aft material is required. Hence, along funnel and engine hatches longitudinal girders should be fitted under the strength deck on each side connecting the different side coaming plates, which become thus integral parts of the fore-and-aft structure. The side coamings should for this purpose be arranged as far as possible in straight lines and the girders should extend some distance beyond the hatches at each end. Such girders, which will be here referred to as "coaming girders," will add to the sectional area of the deck structure and thus increase the longitudinal strength directly, they will reinforce and stiffen the strakes adjacent to the hatches (the funnel stringers) and relieve the stresses at

the hatch corners, and they will help to support the half-beams and transmit the load on these to the points of support. Coaming girders are not, to the author's knowledge, fitted in existing vessels of this class, but appear very advantageous, especially where they can be worked in conjunction with a longitudinal casing bulkhead between the main and second deck. This feature should therefore be kept in view in the early stages of the design when the location and width of the hatches in the strength deck are being settled.

In many light cruisers the strength deck is sheathed with wood, but when this is not the case the danger of buckling must be carefully considered. The stringer plates are so heavy and so well supported by the sheer strakes that they will not require any further stiffening, but the inner strakes are often so light that unless stiffened by longitudinal girders they will be liable to buckle when the ship is in the sagging condition, and will not be able to support the ordinary normal load. Light angle bars, intercostal between the beams, are sufficient for this purpose, but if it is desired at the same time to add to the longitudinal strength of the deck in tension, semi-intercostal girders with a continuous member under the beams should be fitted. As explained above, a better solution is obtained by making the strakes between the stringers and the funnel hatches of more substantial thickness so as to secure a value of μ not greater than 80. If, in such a case, intercostal tee-bars are fitted under the seams, buckling will probably be entirely precluded.

(3) *Destroyers.*—In vessels of this class particular attention must be given to the construction of the deck, which is here unsheathed and which is in a less favorable position than the bottom, for while the deck is weakened by funnels, ventilators, and hatches the bottom is intact and for various reasons of heavier construction. On account of the concentration of weights amidships, the sagging moments are apt to be great and the compressive stresses in the deck very high, whence the material of the deck should be in a great measure worked into longitudinal girders so as to secure adequate stiffness. This point is of increased importance after the introduction of high-tensile steel, which has led to smaller thicknesses of the deck plating.

The distribution of the material in the deck should be such that a proper relation is established between the thickness of the plating and the spacing, depth, and scantlings of the longitudinal stiffeners. It is irrational, as is often done, to fit a few deep and very strong girders under a light deck, because their great strength as individual girders is yet inadequate, and the sectional area of metal which represents their value as members of the ship-girder may be distributed to much greater advantage over a larger number of light girders which can more effectually support and stiffen the deck plating. The beams are now usually of the same type as the transverse frames on the sides, consisting of closely spaced light bars with deeper beams interspersed in continuation of and of the same construction as the belt frames (fig. 137). This system necessitates a great depth and hence a limitation in number of the longitudinal girders, ill adapted to work with the extremely light plating usually employed in the inner strakes of the deck. The girders are of the semi-intercostal variety and are fitted at rather irregular intervals, often more than one and one-half frame spaces apart, incapable of preventing incipient buckling of the plates between the girders. Their fore-and-aft extension is in many boats somewhat erratic and is made dependent on local conditions, foreign to the question of longitudinal strength. Frequently the girders follow a broken line whereby their capacity for resisting longitudinal forces is much reduced and great local strains are created in the deck plating. Sometimes a still more unsatisfactory construction is adopted by "scarphing" the girders, two parts of a girder, instead of being actually connected, running parallel with and overlapping each other a certain distance apart. These undesirable features can generally be avoided when planning the preliminary design.

Although there has been recently a marked advance in the design of torpedo-vessels and notably a tendency towards a more systematic and rational distribution of the longitudinal members, as in the Chilian destroyer *Almirante Lynch*, there appears to be yet much room for improvement. As far as the deck is concerned, it is believed that greater strength and stiffness than by the present system can be attained on a smaller weight by *designing the framing of the deck on the same principles as proposed for the sides in* SECTION 48, *5*. This leads to the following construction, illustrated in fig. 117.

The deck plating between the stringers and the funnel openings is of about the same thickness as the outer shell, in large destroyers about ¼ in. The stringers are somewhat heavier. The framing is identical with that on the sides except where special conditions render deviations necessary. The beams are rather deep channels, worked in continuation of the frames and, like these, slotted over the longitudinals. They are spaced about 3 ft. 6 in. apart in a destroyer of 1000 ts. and should possess the same moment of inertia as the aggregate of the beams which they replace. The longitudinals consist of light bars spaced about 21 in. apart, and are continuous between the bulkheads to which as well as to the beams they are connected by brackets. A somewhat heavier coaming girder is worked on each side of the funnels as explained for the light cruisers. In coal-burning destroyers the bunker bulkheads support and stiffen the frame-rings, but in oil-burning destroyers, where no side bulkheads are found, a line of pillars should be fitted instead. Where a pillar can be placed on every frame there will then, with the framing system here proposed, be no necessity for fitting deep longitudinal girders under the deck in way of the pillars, since the object of the pillars is simply to stiffen the frame-rings, but where pillars cannot be fitted except at wide intervals, as may be the case in the engine-rooms, semi-intercostal girders of great depth must be worked both under the deck and in the bottom.

60. WOOD DECKS.

As explained in the foregoing, wood decks are in warships generally fitted as a sheathing on complete steel decks, while the use of wood decks proper is confined to ships designed for peace duties.

1. **Materials.**—The materials ordinarily used for wood decks are pine and teak. In the United States Navy pine is preferred, viz. yellow pine from the Gulf ports. Pine should be well seasoned before it is used, and should not be laid within a period of six months after being cut. Teak is the material by far best suited for this purpose and is used extensively in the British Navy, although Dantzig crown deals are often employed for the mess decks. Teak is particularly suited for weather decks, ammunition rooms, and passages, and in all places where the deck is exposed to heavy wear and tear. It is also the best material for margin planks. It is somewhat heavier than pine, but may be used in smaller thicknesses without detriment to the calking. Lloyd's Rules permit a reduction in teak of one-sixth as compared with pine. The chief reason why teak is not universally used for decks is its high cost.

Wood decks are laid in planks which should be free from "sap" and objectionable "knots." Especially the upper side of the planks should be free from knots. Sap-wood is nearest the bark of the tree ; it is soft and liable to rapid decay. Knots may become loose, and, if found on the upper surface of the planks, form hard points which do not wear at the same rate as the neighbouring parts.

2. **The Planks.**—The planks are of uniform width and thickness and the seams are worked parallel with the center-line. In order to house the "nibs" of the planking and in order to avoid the ends abutting on the metal, a margin strake, often referred to as the "waterway," is fitted round the boundary of the deck inside the inner gutter angle. The margin strake is somewhat thicker and wider than the other strakes, and is cut out as shown in fig. 165 in order to avoid the acute angle at

the end of the planks. When a plank runs off to a fine edge, the fastening cannot be placed so near the end as required for efficient calking. Margin planks are also fitted round barbettes, deckhouses, casings, hatches, etc. The bounding angle bar, against which the margin strake is fitted round such fixtures, should project a little above the wood so as to prevent the water lodging between the plank and the wall or coaming plate. At the waterways the angle bar should have the same height as the edge of the plank.

Teak planks are generally about 9 in. wide and 3 in. thick on the decks, but smaller thicknesses, down to 2 in. and 1½ in., are used in case-

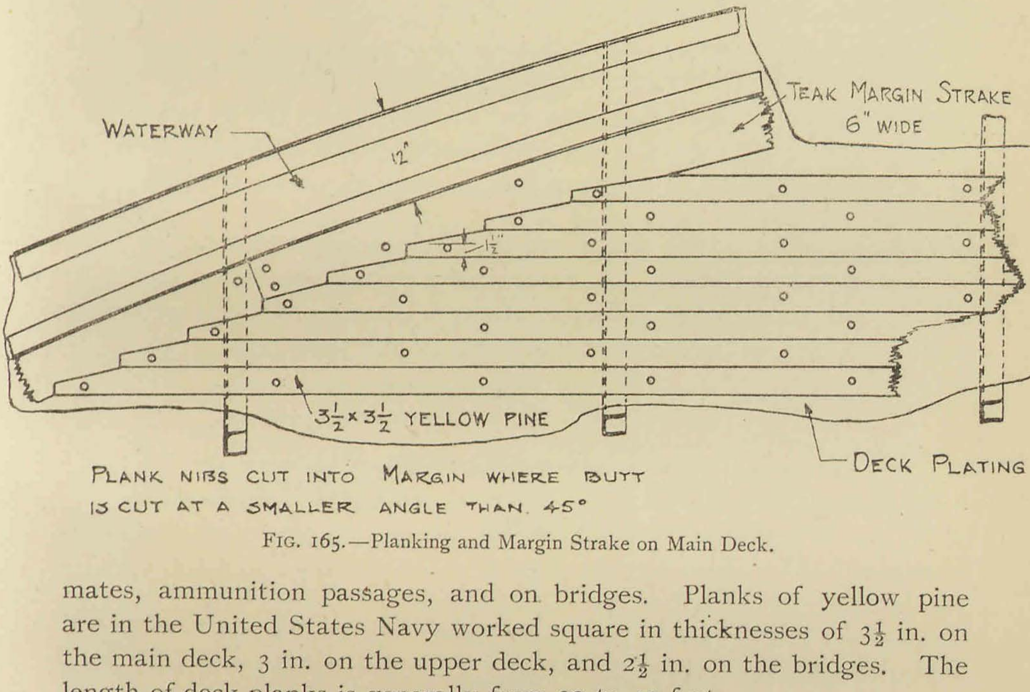


FIG. 165.—Planking and Margin Strake on Main Deck.

mates, ammunition passages, and on bridges. Planks of yellow pine are in the United States Navy worked square in thicknesses of 3½ in. on the main deck, 3 in. on the upper deck, and 2½ in. on the bridges. The length of deck planks is generally from 20 to 40 feet.

When a wood deck is laid on a steel deck, the planks are fastened to the plating between the beams. There should be not less than two bolts in each frame space. Where the width of the planks permits, the bolts should be staggered so that adjacent bolts shall not be in the same fore-and-aft line. The butts are generally placed on the beams, and the ends of the planks are fastened by one or two bolts on each side of the beam.

Where no steel deck is fitted, the planks are laid directly on the beams and fastened to them. Beams of tee-section are most suitable in this case, since they give a better support to the planks and permit a better fastening than profiles with a single upper flange. The butts of the planks must necessarily be placed on the beams, but in order that the bolts shall not be too near the end of the planks, a piece of plating

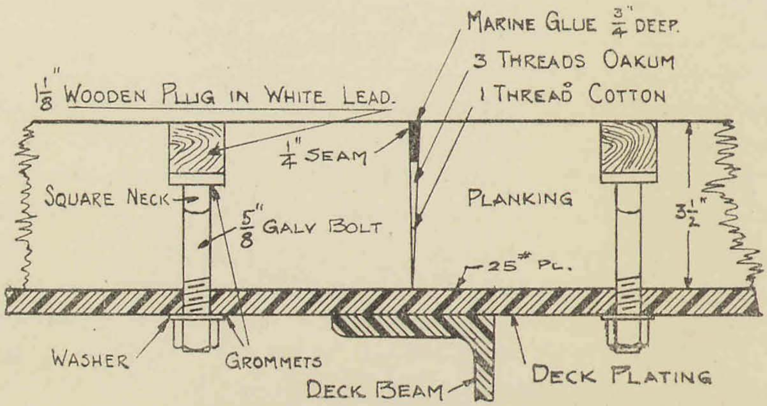


FIG. 166.—Butt in Main Deck Planking.

of the same width as the planks and of sufficient length to take the bolts is riveted to the beams as shown in fig. 167. The butts should be shifted well clear of one another so as not to leave less than three or four passing strakes between consecutive butts on the same beam.

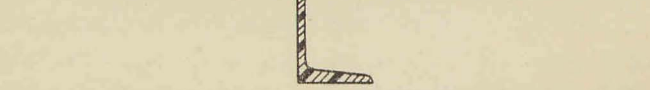


FIG. 167.—Butt in Deck Planking where no Deck Plating is Fitted.

3. **The Fastenings.**—The planks are fastened by galvanised iron bolts with nuts, varying in diameter from ½ in. in light decks to ⅝ in. in heavy decks. The heads are round, but the necks are square, to prevent the bolts turning in the wood when the nuts are hove up. In special cases the bolts are tapped in the plating. A grommet is placed under the head, which is sunk about one inch below the upper surface of the planking. "Wood plugs," in England called "dowels," are driven into the cavities so formed. The grain of the plugs should run in the same direction as that of the planking. The dowels are steeped in white lead

and should fit snugly. The nuts are set up with washers and grommets soaked in white lead.

In decks with very light steel plating the bolts are placed close to or on the beams and in addition "lag-bolts" are sometimes fitted from underneath between the beams. Lag-bolts are wood screws; they have not the holding power of through-bolts but leave the upper surface of the deck intact. Usually each plank receives one through-bolt and one lag-bolt per frame space.

4. **Watertightness.**—Wood decks are made watertight by calking edges and butts, which are therefore left with a slight opening at the surface into which threads of oakum are laid down. After calking, the seams and butts are left about $\frac{3}{4}$ in. deep for paying with marine glue or hot pitch. Seams of bridges may be puttied. Putty gives a finer appearance to the deck, but is not so reliable in point of watertightness.

With sheathing of steel decks there is the inconvenience that if there is a leakage in the wood deck and if the steel deck is calked it may be very difficult to locate the leak. The best result is probably obtained by not calking the steel deck at all, but if it is calked, it should be done very carefully. Before the planking is fitted, all metal surfaces to be against the wood should be given a thick coat of red lead, applied immediately before the planking is laid.*

61. LINOLEUM AND OTHER DECK COVERINGS.

1. **Linoleum.**—Linoleum is the deck covering mostly used as a substitute for wood; it is in the British Navy called "corticine." It is a preparation of linseed oil mixed with ground cork. The mixture is pressed upon canvas and hardened by oxidation of the linseed oil. If not thoroughly seasoned before being laid down, it will stretch and bulge and is then very liable to crack. Linoleum should, therefore, before it is taken in use, be stored a couple of years in such a way that air can get free access to it and the oxidation of the linseed oil can be completed.

Linoleum is generally used in the thickness known as "medium weight" (4·85 mm.), but where it is exposed to excessive wear a "heavy-weight" quality, $\frac{1}{4}$ in. thick, is used, which weighs about 1 lb. per sq. ft.

Linoleum is employed mostly in living spaces, passages, state rooms, etc., but sometimes it is laid on open decks, as on superstructures, bridges, etc., and on the decks of torpedo-boats.

* For a more complete discussion of wood decks the reader is referred to A. C. Holms' work on *Practical Shipbuilding*.

Before laying the linoleum, the surface of the deck must be perfectly dry, clean, and smooth ; all recesses and rough places must be filled with cement, and suitable strips fitted in way of all projections. When secured to the deck the linoleum should be at a temperature of about from 60° to 70° F. and the compartment in which it is laid should be maintained at about the same temperature. In places where linoleum is exposed to moisture, the canvas on the underside should be thoroughly permeated with linseed oil or paint.

Linoleum is secured to the deck by a cement or by a waterproof glue. A solution of orange shellac and methylated spirit is much used. The edges are stopped with putty. Round openings, and often also along boundaries and along seams and butts, strips of brass or galvanised iron are fitted for the purpose of protecting the edges and for holding the linoleum in place. The strips are generally secured by wood screws, screwed into the steel plating, but where the deck is very thin, bolts are used.

While linoleum fulfils the functions of a wood deck in some measure, being a bad conductor of heat and sound and non-slippery, it is, like wood, inflammable, and is far less durable than this material. If, by neglect, it becomes worn and cracked it is a source of uncleanliness and disease-producing conditions, since it will then permit an unseen accumulation of filth and moisture.

2. **Other Substitutes for Wood.**—Great efforts have been made to produce a fireproof material which possesses the good qualities as a deck covering peculiar to wood. Various substances, such as asphalt-like cement, have been proposed, and on torpedo-boats a bitumastic cement mixed with sand has been tried with some measure of success. Also matting and painted canvas have been used. So far, however, no fully satisfactory substitute for wood has been found.

Locally, ceramic tilings are used in pantries, wash-rooms, water-closets, bathrooms, and many other places. In some cases rubber tiling is employed.

62. DECK STANCHIONS.

Stanchions are in England called "pillars"; the two terms will here be regarded as synonymous.

1. **General Requirements.**—Based on considerations of strength, a maximum length of unsupported beam is determined, as explained above, and stanchions are then located in such a way that this maximum shall not be exceeded, while at the same time care is taken not to

obstruct passages, the working of guns, or other important service. Wherever possible, bulkheads are utilised as support for the beams. The stanchions under the different decks should be worked vertically over each other where practicable, so as to form continuous lines of support, carried directly or through bulkheads down to the framing system of the ship. By breaking the lines of support, as, for instance, by entirely omitting one or more stanchions on one of the decks, bending moments are introduced in the structure, which, in order to be properly resisted, may require an expenditure of extra material. A similar result, although in a smaller degree, will follow if a stanchion is placed out of line. Locally, stanchions are fitted under guns, windlasses, and heavy weights not otherwise supported.

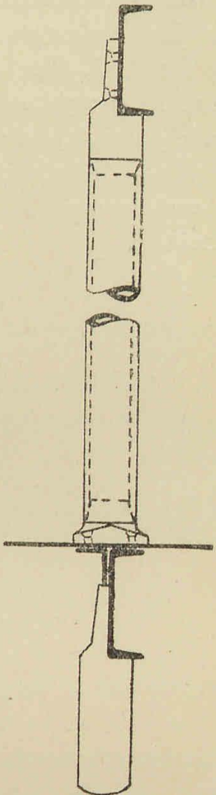


FIG. 168.—Deck Stanchion.

Stanchions, although primarily designed for compression, may in many cases be subject to tension, as, for instance, when a ship is working in a seaway, when a lower deck is more heavily loaded than an upper, and when guns are fired over a deck. Stanchions should, therefore, be efficiently fastened at both ends, so as to be capable of acting, not only as struts, but also as ties. This requirement is in warships of particular importance under decks exposed to the blast of guns.

2. **Construction and Fitting of Stanchions.**—Stanchions are generally made of wrought iron or steel pipes with heads and heels welded in solid. In some cases, however, heads and heels are of cast steel with the pipe expanded into the casting. A usual size between decks above the protective deck in large ships is 5-in. external diameter and $\frac{1}{4}$-in. thickness, but in boiler-rooms, under barbettes, and elsewhere in the hold of large vessels where the stanchions are much longer and the superincumbent weights are greater, stanchions of 8-in. to 10-in. diameter are used. The thickness of the pipe varies from one-tenth to one-twentieth of the diameter.

Heads of stanchions should abut against the lower flange of the beam and lap on the web, extending well beyond half depth of the web to prevent it from buckling and tripping. In way of the heel reversed angle clips should be riveted to the beams for taking the rivet connections.

Heads and heels are usually fastened by rivets, although in some cases it may be necessary to use taps. The heel should stand directly on the steel deck and should be riveted directly to the beam ; it should not be bolted to a wood deck, except where, as in case of skid beam supports, the stanchions do not form an integral part of the ship's structure. Doubling plates should be fitted under heavy stanchions. Rivet holes should be absolutely fair when the stanchion is fitted snugly against the deck below and the beam above. In some cases where stanchions are required, their presence may be at times very inconvenient, as when fitted near capstans, torpedo-tubes, etc. Such stanchions are, therefore, made portable, being, generally, hinged at the head.

Pin-ended stanchions should be used, as explained in SECTION **21**, *2*, in certain cases where the load is great and known to be strongly eccentric.

CHAPTER XVI.

BULKHEADS.

BULKHEADS are watertight, oiltight, or non-watertight. Watertight bulkheads may be grouped in two classes, ordinary watertight bulkheads, which are only accidentally and rarely subject to water pressures, and water-tank bulkheads, which are intended to be exposed to water pressures for long periods. We shall deal first with ordinary watertight bulkheads.

63. PRINCIPAL FUNCTIONS AND CLAIMS.

1. **Watertight Subdivision.**—Watertight bulkheads subdivide the internal space in a number of compartments which contribute to the safety of the ship and, at the same time, permit a proper utilisation of the space. They should be sufficiently strong to withstand the pressures of the water in the most severe cases of flooding likely to occur, and should under these conditions be watertight, but the watertightness need not be absolute. Small leakages due to weeping of rivets or joints are of

no consequence provided they are in the aggregate so small as to be within the capacity of the daily (secondary) drainage system of the ship.

2. **Bulkheads Considered as Members of the Ship-Girder.**—In this capacity bulkheads act essentially as diaphragms and webs, resisting compressive and distorting forces in their own plane, and imparting thus stiffness to the hull.

Longitudinal bulkheads in the hold, extending for more than half the length of the vessel amidships, should be so constructed as to be capable of assisting the outside plating in its function as the web of the ship-girder. The continuity of such bulkheads should, therefore, be preserved as far as possible by carrying them continuously fore and aft through the transverse bulkheads and vertically through the platform decks. Above the protective deck, the longitudinal bulkheads should likewise be continuous fore and aft, but should not cut through the decks. Longitudinal bulkheads which are thus enabled to work as members of the ship-girder should be connected efficiently at their upper and lower edges to the decks and to the bottom. They should not terminate abruptly at the ends, and should possess sufficient strength and stiffness to avoid excessive stresses and deformations. These claims are of particular importance in long, lightly constructed ships of high speed.

Transverse bulkheads should be capable of performing the same functions transversely as required of the longitudinal bulkheads longitudinally. They have in particular to resist the racking strains when the ship is working in a seaway and the transverse shearing and bending when in dock. Particular attention should be given to the transverse bulkheads which are placed on the end blocks when the ship is in dock, because these bulkheads—perhaps assisted by some of the floors—have to transmit the weight of the overhanging portions of the ship to the blocks. In all bulkheads that are liable to be severely strained when the ship is in dock, the shearing strength of the boundary connections should be carefully examined.

3. **Support of Weights.**—All bulkheads should be vertically continuous between the inner bottom and the protective deck, so as to be capable of directly supporting and transmitting the superincumbent weights. Bulkheads are peculiarly adapted for this function, inasmuch as they distribute the load over large areas of the bottom structure and support it equally well in different directions in their own plane. This latter quality is of particular importance when a ship is in a seaway, where the load is constantly changing direction. Bulkheads are in this respect greatly superior to pillars, which can only support forces normal to the decks.

64. ARRANGEMENT OF PLATING AND STIFFENERS.

1. **General Construction.**—If bulkheads were required to withstand water pressures only, they might with advantage be constructed as elastic membranes of very small stiffness. The best result would then be obtained with very light plating, efficiently riveted at the seams and butts and supported by a close network of light stiffeners, well connected at the boundaries. A good watertightness might be secured with such a bulkhead on a small weight, but the deflection would be great, since the bulkhead would have to resist the pressures entirely in virtue of tension. Great deflections are, however, undesirable because, where pipes and leads pass through a bulkhead and where doors are found, excessive strains are apt to be produced causing local leakage and other difficulties. Moreover, the claims on bulkheads as strength members and diaphragms of the hull render it necessary to endow them with considerable stiffness. It is for these reasons preferable to base the strength of bulkheads primarily on the stiffeners, which by themselves should be capable of resisting the water pressures. The deflection and hence the tension in a bulkhead as a whole and in individual stiffeners should be practically insignificant. The deflection of the plating between the stiffeners may be relatively greater. In certain submarine boats important bulkheads are designed to resist entirely by tension, having a strongly vaulted form and no stiffeners.

2. **The Plates Run Parallel with the Stiffeners.**—This system, of which fig. 169 shows different varieties, has been much used in the French Navy in connection with vertical stiffeners and flush plating. Usually tee-bars, acting both as stiffeners and edge strips, are placed on the seams. If there is only one row of rivets in each flange, a strip of plating is fitted on the opposite side, but often the tee-bars are sufficiently wide to take two rows in each flange, in which case the seam strips may be omitted.

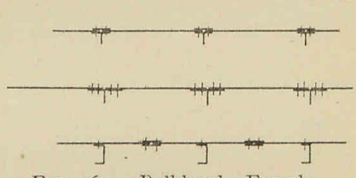


FIG. 169.—Bulkheads, French Warships.

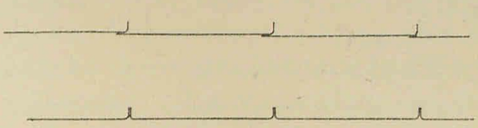


FIG. 170.—Bulkheads, Flanged Plating.

The plates are generally in one length, and hence of one thickness for the entire height of the bulkhead. In bulkheads of larger size, channels or I-bars are used, and the edge joints are placed midway between the stiffeners, connected by double strips. Frequently the stiffeners are formed by flanging the plates as shown in fig. 170, a construction much employed in torpedo-vessels in various navies, sometimes with vertical and sometimes with horizontal seams.

3. **The Plates Run Normal to the Stiffeners.**—This system is used in the British and in the United States Navy and, to some extent, in the French Navy, generally with horizontal plates and vertical

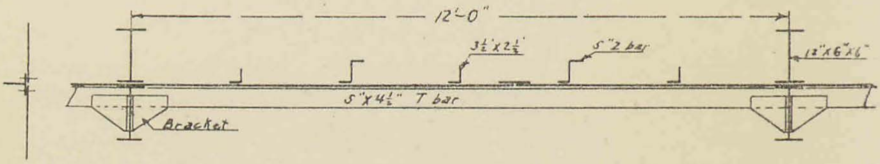
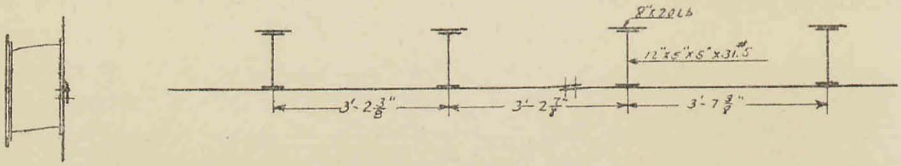
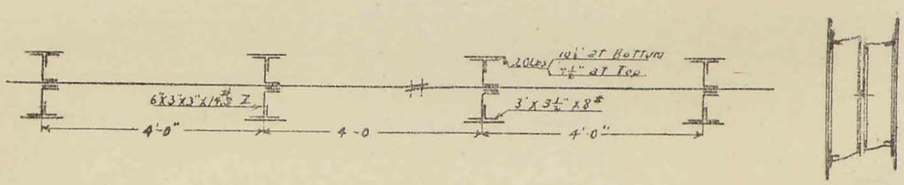


FIG. 171.—Bulkheads, Large English Warships.

stiffeners. In the British Navy the construction shown on fig. 171 has been used extensively.* The plates are here worked flush and the edges are connected by horizontal tee-bars which at the same time serve to stiffen the plating along the seams. The main stiffeners are vertical. At intervals of about 8 or 12 feet, double stiffeners, each consisting of



Transverse Bulkhead, Recent Battleship, U.S.N.



Center-line Bulkhead, Earlier Battleship, U.S.N.

FIG. 172.

two deep I-bars placed on opposite sides of the bulkhead, are fitted, and between these are placed zed-bars of moderate weight, alternating with light angle bars. The spacing of the vertical stiffeners is about two feet. The double I-stiffeners virtually divide the bulkhead in several independent smaller bulkheads. The butt straps are on the same side of the bulkhead as the vertical stiffeners, while the horizontal tee-bars are on the opposite side—the calking side. By this arrangement there is no interference between the stiffeners, and the calking edges are clear.

* Attwood, *War-Ships*, published by Longmans, Green & Co., London, 1904, p. 61.

Recently it appears * that overlapped work has been introduced in the British Navy and a pure system of vertical stiffeners adopted by omitting the tee-bars, but the principle of subdividing the bulkhead in sections by double I-stiffeners is retained.

In the United States Navy (fig. 172) the strakes run horizontally and both seams and butts are overlapped. Formerly the stiffeners were double, and the plating was arranged on the raised and sunken system (fig. 176). This necessitated the use of liners under the stiffeners on all the strakes, and the double stiffeners had drawbacks which will be discussed below. In recent ships the stiffeners are, with few exceptions, single, *i.e.* they are all placed on one side of the bulkhead, the non-calked side (fig. 177). Generally, they are all of the same uniform section throughout the bulkhead, spaced four feet apart in large vessels. The strakes are worked clinker fashion and the seams are joggled, requiring only short tapered liners on the stiffeners, of a length sufficient to take two rivets.

4. **Two Sets of Stiffeners at Right Angles to Each Other.**—A mixed system, partly horizontal, partly vertical, was formerly very generally used, especially in the merchant marine. The vertical stiffeners were fitted on one side of the bulkhead, the horizontal stiffeners on the other. In merchant vessels the vertical stiffeners were rather light and closely spaced, while the horizontal stiffeners were deep heavy girders, placed at a greater distance, generally one deck height, from each other. Sometimes horizontal stiffeners of smaller depth were fitted in addition between the deep girders.

A mixed system, consisting of closely spaced light vertical stiffeners and one deep horizontal double stiffener, has been proposed for bulkheads of moderate width.† Fixity of the horizontal stiffener to be attained by connecting it at the corners by deep knees with similar stiffeners worked round the other walls of the compartment, the ends of one stiffener serving to fix the ends of adjacent stiffeners when the compartment is flooded. The vertical stiffeners to be unbracketed and intercostal. This system, however, can only be used with advantage where all the four walls of several adjacent compartments are of relatively short horizontal extension. Moreover, the horizontal deep stiffeners take up much space and, being double, they are necessarily inefficient, as explained below.

5. **Comparison between the Different Systems.**—Let us compare first the arrangement of the stiffeners. The load per foot run, and hence the water pressure, which a stiffener of a given section can support, is inversely proportional to the square of its length. Hence, from the

* *Manual of Seamanship*, London, 1909, v. ii. pl. xviii.

† Pietzker, *Festigkeit der Schiffe*, fig. 108, p. 141.

sole point of view of strength under water pressure, it is most advantageous to place all the stiffeners across the shortest dimension of a bulkhead—if a bulkhead is of small height and great width, the stiffeners should be vertical and *vice versa*. In order to obtain the maximum strength with a given stiffener, it should be "fixed" at the ends, but this is possible only if the structures, to which the ends are connected, possess sufficient rigidity. In the main bulkheads of the machinery spaces the breadth is ordinarily greater than the height and the horizontal boundaries are formed by the rigid bottom structure and by the protective deck, both of which afford good attachment for stiffeners. Vertical stiffeners are, therefore, here the most efficient. Beyond the machinery spaces, the bulkheads are subdivided by decks into sections of small height and great

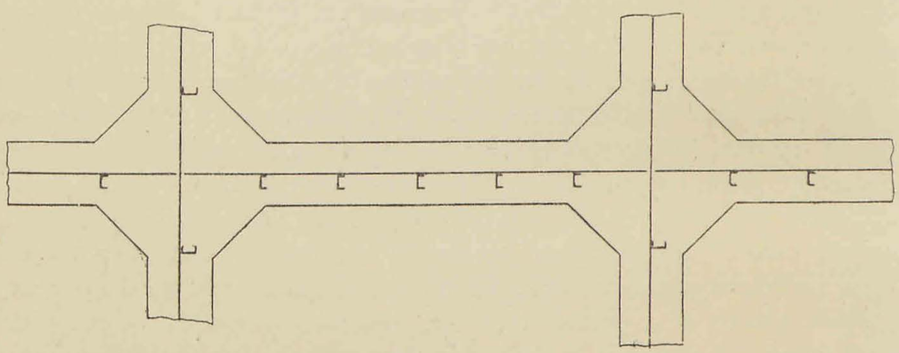


FIG. 173.

width, whence, again, vertical stiffeners are the most suitable. Cases occur, frequently indeed, in side bunkers, where the breadth of the bulkheads is much smaller than the height, but even there vertical stiffeners are generally preferred, because it is difficult to provide a good end attachment for horizontal stiffeners and because such stiffeners are unfavorable to coal stowage and are apt to be in the way of the bunker doors.

At the extreme ends of the ship, and sometimes in the wing spaces, very narrow bulkheads of great height may be found, while at the same time the side boundaries are sufficiently rigid for the bracketing of horizontal stiffeners. In such cases horizontal stiffeners should be used. An important example is the collision bulkhead, where the horizontal stiffeners can be bracketed to longitudinals or to the outside plating, which is here usually doubled or at least very heavy.

As far as the resistance to water pressures is concerned, we conclude that a simple system of parallel, generally vertical stiffeners is the best. Vertical stiffeners are also those best adapted to support superincumbent weights and they lend themselves well to an efficient connection with the framing system of the ship. When we consider bulkheads in their capacity of diaphragms, a mixed system is probably the most efficient, but in warships the horizontal compressive forces as well as the racking strains to which individual bulkheads are exposed, are usually of small magnitude, and their strength when the ship is in dock is generally ample in ships provided with side keels. In ships of large size, docked on center keel, the transverse bulkheads are more severely strained, and it is desirable to subdivide the plating in small fields in order to prevent wrinkling. This is best attained by fitting light horizontal bars on the side opposite to the main stiffeners, a method which may be used with advantage also in tank-bulkheads where great stiffness of the plating is required, but such bars must be regarded chiefly as a local reinforcement of the plating and cannot properly be classed as bulkhead stiffeners. It might be thought that when the ratio width to height of the bulkhead is smaller than two, the horizontal bars so fitted would exert an influence of the same order as that considered in the theoretical treatment of the rectangular plate, there taken account of by multiplying the pressure with a factor of reduction K_A. When, however, the horizontal bars are light relative to the vertical and when they are fairly long, they will act only in virtue of their tension and will not be called into play at ordinary test heads. They can, therefore, be disregarded in the strength calculation of the bulkhead as a whole.

There appears then no necessity for adopting the mixed system in ordinary watertight bulkheads, and *a pure system of vertical stiffeners may be considered the best in most cases.* This system is now used, not only in many warships, but also almost universally in the merchant marine, after being adopted by Lloyd's and Veritas a few years ago. Only where vertical stiffeners are formed by flanging of the plates in large bulkheads, Lloyd's Rules require horizontal tie bars with intercostal plates to be fitted at large intervals.

The spacing of the stiffeners is usually the same as that of the frames, except where heavy weights are to be supported by the bulkheads, in which case it should be smaller, ordinarily one half frame space. The stiffeners of longitudinal bulkheads should be placed directly under the deck beams, and on transverse bulkheads as far as possible on longitudinals. With a close spacing of the longitudinals as recommended in SECTION 48, all the stiffeners on transverse bulkheads may be so placed.

Turning now to the plating, the horizontal arrangement of the strakes has the advantage that the thickness can be gradually increased from the top of the bulkhead to the bottom, consistent with the increase in pressure. Apart from the question of strength it is advisable to give the lowest part of the bulkhead some extra thickness, because, more than the rest of the bulkhead, it is in many cases exposed to rapid corrosion by moisture, as, for instance, in coal bunkers and machinery spaces. Further, since the maximum tension in the plating, when under water pressure, will exist across vertical lines, horizontal seams and their calking will be less severely strained than vertical seams. Finally, horizontal seams form lines of stiffness, which together with the stiffeners subdivide the plating into small rectangles.

On the whole, then, it appears that *the horizontal arrangement of the strakes is the best.*

65. CONSTRUCTION OF STIFFENERS.

Stiffeners generally consist of simple bars of various sections, often strengthened by angles and face plates. The sections mostly used at present are angle bars, channels, bulb-angles, tee- and I-bars. Where special strength is required, stiffeners are built up of deep plates reinforced by angles, so-called "web stiffeners"; where a light construction is desired, they may be formed by flanging the edges of the bulkhead plating.

1. **Double Stiffeners compared with Single Stiffeners.** — The term "double stiffener" is here applied to a combination of two single stiffeners, one on each side of the bulkhead, riveted to each other through the bulkhead plating. It is clear that the depth on one side of the bulkhead of such a stiffener, really its half depth, can be made smaller than the depth of the equivalent single stiffener. This is, under certain conditions, an advantage as, for instance, on a center-line bulkhead between two engine-rooms, when it is desired to obtain a perfectly symmetrical arrangement; in fact, space may be virtually gained in this way. Double stiffeners, moreover, offer the same support against pressure from either side, which might appear at first sight an important advantage in case of transverse and center-line bulkheads. Where single stiffeners are fitted, and when the pressure is applied on the same side as the stiffeners, the plating hangs on the rivets, which are ill adapted to this kind of strain. In ordinary tests, however, bulkheads with single stiffeners are found to show the same behavior whether the pressure is applied to one side or the other, but, probably, the difference would appear if the tests were carried nearer the point of destruction. It is believed, therefore, that if a bulkhead is constructed in accordance with the rules and tables given below, by which, in particular, a good connection between the stiffeners and the plating is secured, this point can be dis-

regarded within the limits of ordinary test requirements. On side bulkheads, exposed to the effects of torpedo and mine explosions, violent pressures, far exceeding the test pressures, are liable to exist. The stiffeners of such bulkheads should, therefore, always as far as possible be placed on the inside, since otherwise the plating is liable to be torn away from the stiffeners by the rivets pulling through.

Structurally, a double stiffener has the inherent defect as compared with a single stiffener that a riveted connection and hence a line of weakness exists in the web along the neutral axis where the shearing is a maximum. The shearing stresses will be greatest between the points of inflection and the apices of the brackets, at EC and FD in fig. 175. There will here, as indicated in fig. 174, be a tendency of the stiffener to move upwards on one side and downwards on the other, resisted only by the rivets which connect the flanges of the stiffeners through the bulkhead plating. Now, if there is only one line of rivets, it will be exposed to excessive shearing even at moderate loads on the bulkhead, a slipping of the stiffeners relative to each other will take place, and the points of inflection will move nearer to the apex of the brackets. When the load is sufficiently increased, the points of inflection will move right up to the brackets, and if these possess sufficient strength, the double stiffener will henceforth behave as if hinged at the apices C and D.

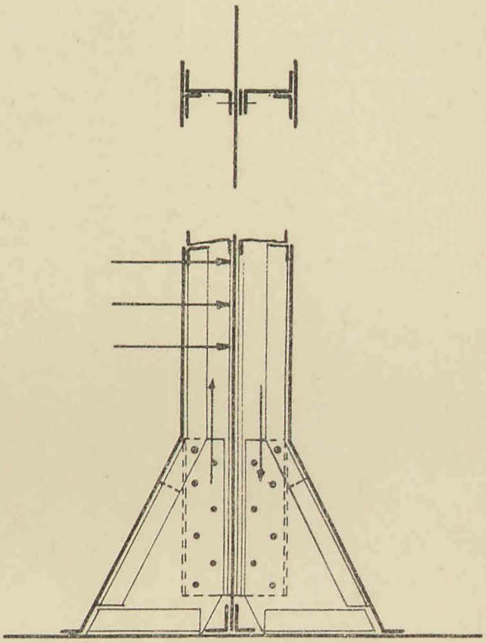


FIG. 174.

Numerous carefully conducted tests of bulkheads, accompanied by accurate measurements of the deflections, have been made in the United States Navy. These tests, which comprise bulkheads with double as well as with single stiffeners, have been analysed by the author and the results recorded in a paper read before the Society of Naval Architects and Marine Engineers in New York, 1910.* The tests on bulkheads

* *An Analysis of the Strength of Watertight Bulkheads.* See also *Am. Soc. Nav. Arch. Mar. Eng.*, 1905, p. 229.

with double stiffeners fully bear out the explanation of the cause of their weakness as explained above. The observed deflection of such stiffeners, bracketed at the ends, was found to be about four times as great as its

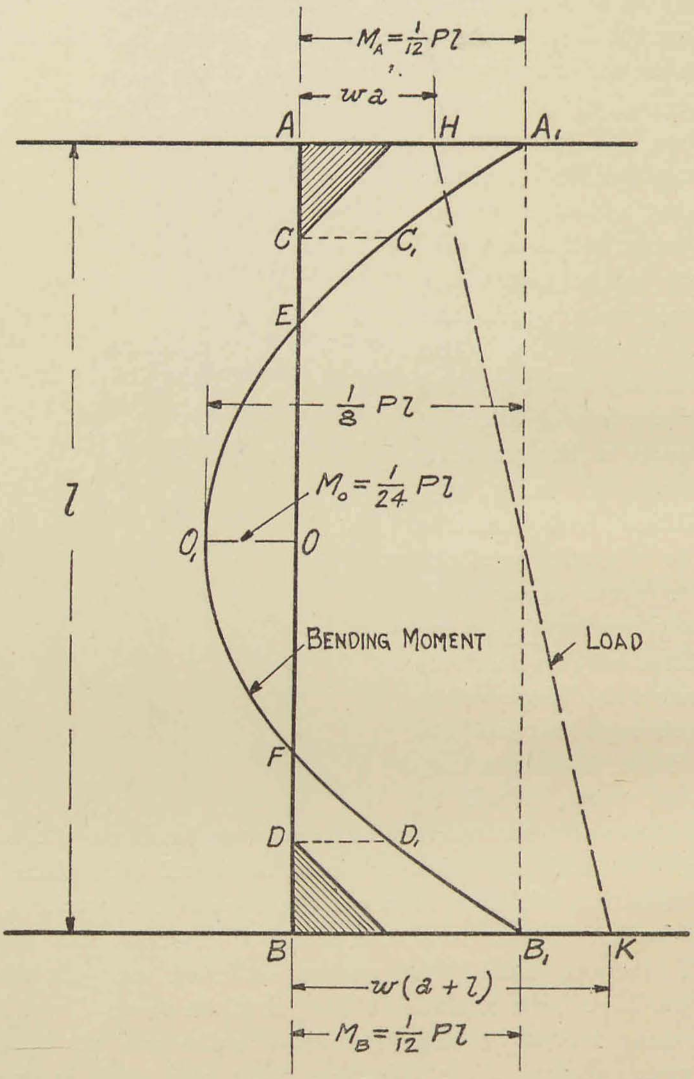


FIG. 175.

theoretical value calculated for fixed ends and for an effective length equal to the total actual length of the stiffeners. If, on the other hand, it was assumed that the stiffeners were hinged at the apex of the brackets, a good correspondence between the observed and the calculated deflections was obtained. See fig. 176, which gives the results of one of the tests. The great deflection could not be attributed to the brackets, which showed no sign of weakness whatever, and which from their construction might well be expected to be able to "fix" the stiffeners. It is clear, therefore, that the deficiency must be sought in the stiffeners themselves as explained above. Such weakness of the double stiffeners could, of

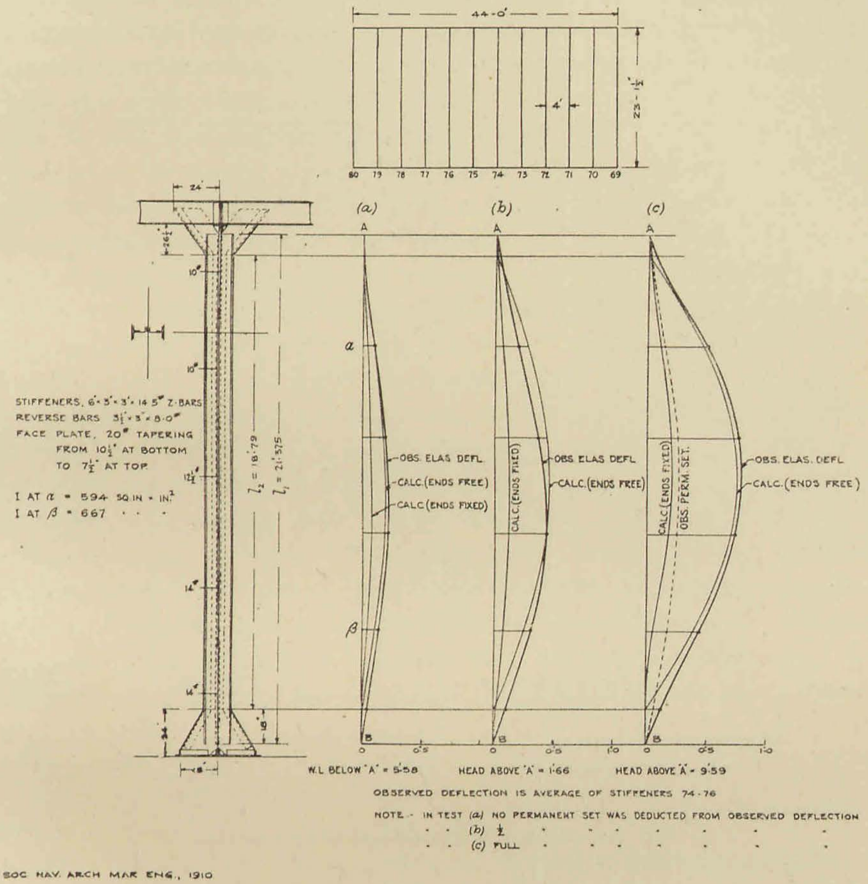
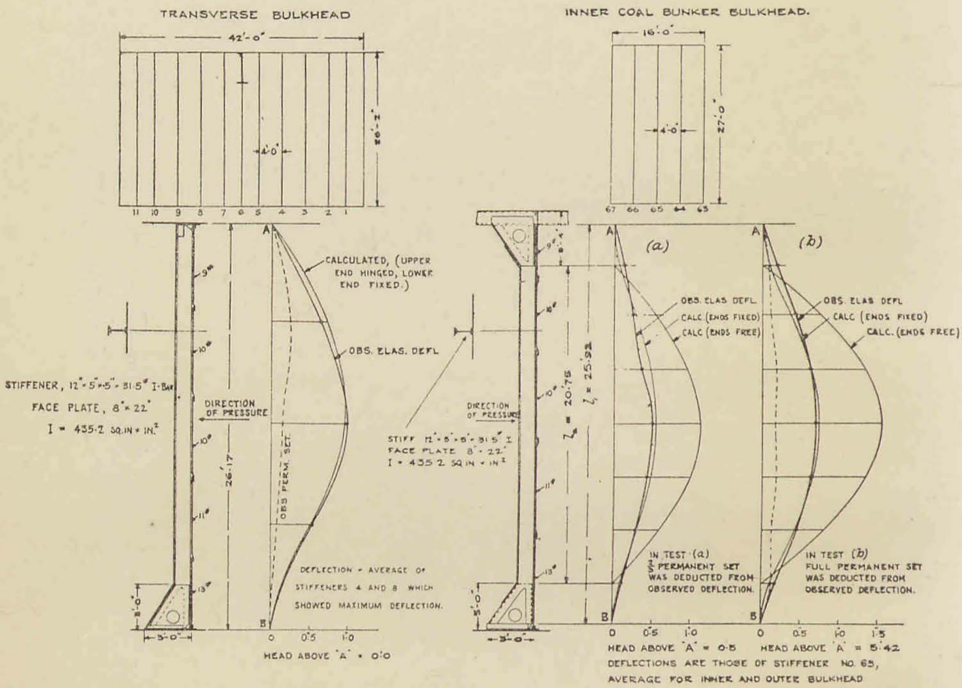


FIG. 176.—Test of Centre-line Bulkhead between Engine-Rooms. Double Stiffeners.

course, be avoided by a more efficient connection between the flanges that meet in the neutral axis, using two or more lines of rivets, but then the cost and weight would be much increased. Another serious objection to double stiffeners is the unavoidable accumulation of material at the neutral axis, viz. the bulkhead plating, the flanges of the two bars, and the liners, all of which do not contribute appreciably to the moment of inertia. Moreover, the rivets, passing through four thicknesses, are ill adapted to resist shearing.

Single stiffeners may be constructed of solid bars, as, for instance, channels or I-bars having no line of weakness or undue accumulation of material in the neutral axis. By riveting a face plate or a reversed bar of appropriate scantlings to the outstanding flange, the bulkhead plating will come to work to best advantage with the stiffener, and the section modulus will be the highest attainable with the given section. The width of the effective strip of bulkhead plating may be assumed, as usual, equal to about thirty times the thickness plus the distance between



SOC. NAV. ARCH. MAR. ENG., 1910

FIG. 177.—Tests of Bulkheads with Single Stiffeners.

the lines of rivets that connect the stiffener to the plating. Stiffeners of important bulkheads, even in small vessels, should preferably be I-bars, because they allow two lines of rivets to be placed in the bulkhead plating as well as in the face plate. The rivets in the two lines should be zigzagged relative to each other and should not be spaced more than five diameters apart in the lines. In the vicinity of the apex of the brackets and near the points of inflection the spacing should be somewhat closer. In the tests referred to above, single stiffeners, constructed as here indicated and efficiently bracketed, gave exceedingly favorable results as compared with similar tests carried out on bulkheads with double stiffeners. The cases compared were large bulkheads in battleships and armored cruisers. The height of the bulkheads, the head of water, and other conditions were practically the same for the two types of bulkheads. The single stiffeners were about forty per cent. lighter per foot run than the double stiffeners, and their calculated moment of inertia was only about two-thirds of that of the double stiffeners. Nevertheless, the single stiffeners, wherever brackets were fitted, behaved as if they were fixed at the ends, *i.e.* at points A and B (fig. 175), showing thus deflections much smaller than those of the double stiffeners under the same circumstances. Fig. 177 gives the results of two tests on bulkheads with single stiffeners. In one of these the stiffeners were attached at the top by lugs to the protective deck and were consequently assumed to be hinged at this point. It is seen that also in this case the correspondence between the observed and the calculated deflections is very good, testifying to the correctness of the assumptions and to the general reliability and efficiency of single stiffeners. It is, therefore, concluded that, *except where local conditions necessitate the application of double stiffeners, single stiffeners should be used.*

2. **End Connections.**—The load on a vertical stiffener of a bulkhead may be represented by a straight sloping line such as HK (fig. 175), drawn on the vertical line AB as a base. At the top of the stiffener the head is a at the foot it is $a+l$ where l is the length of the stiffener, ordinarily equal to the height of the bulkhead. Let us assume that each stiffener carries the entire pressure load on a rectangle of plating of height l and of width equal to s the spacing of the stiffeners. The total load P which is the sum of a uniform, "rectangular," load and an increasing, "triangular," load may then be expressed as follows :

$$P = \frac{sl}{35}\left(a+\frac{l}{2}\right) \text{ tons} \qquad (125)$$

where a and l are given in feet.

The form of the curve of bending moments depends only on the nature of the load, but is found to differ little between the two extremes of pure uniform load and pure triangular load, as appears from fig. 178. For practical purposes we may assume the curve of bending moments to be that for a uniformly distributed load of magnitude P. While the form of the curve of bending moment is thus given, depending only on the value of a and l, the relative position of the base line, from which readings are taken, depends on the way in which the ends of the stiffener are held. If the ends are free to incline, the base line is A_1B_1 (fig. 175), which gives a maximum bending moment at the middle of the stiffener

$M_0 = \frac{1}{8}Pl$. If the ends are fixed vertically, the base line will be AB and the bending moments at the ends will be $M_A = M_B = \frac{1}{12}Pl$ *i.e.* two-thirds of the maximum when the ends were free. At the middle, the bending moment will now be $M_0 = \frac{1}{24}Pl$ only half of the value at the ends. If the ends are not held strictly vertical, but are allowed to incline to some extent, the position of the base line will be intermediate between the positions AB and A_1B_1 and may tilt one way or the other. Hence, by fitting brackets, even if they are not entirely rigid, there will be an equalisation of the bending moments and

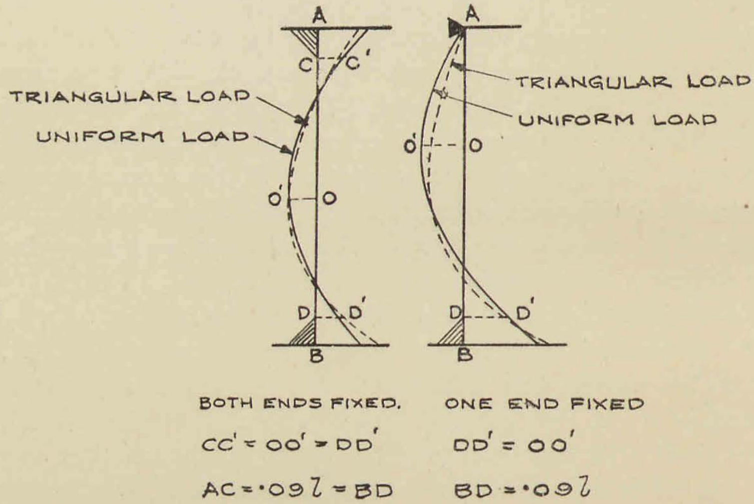


FIG. 178.—Curves of Bending Moments of Bulkhead Stiffeners.

a reduction in their maximum value with a corresponding equalisation and reduction in the stresses.

The brackets not only reduce the bending moments on the stiffeners, they also provide efficient boundary attachments, which relieve the stress on the rivets in the bounding angles of the bulkhead; they enable the bulkhead better to carry the superincumbent weights, and by distributing the forces, transmitted through the bulkhead, over larger areas of the adjacent structure, they assist it in its function as a diaphragm.

The stiffeners of all important bulkheads should, therefore, be provided with brackets at both ends.

Tests as well as experience have clearly shown the beneficial effect of brackets, but have at the same time brought out the fact that they, in order to be efficient, must not only themselves possess considerable strength and stiffness, but must also be efficiently connected to a rigid part of the ship's structure. The simplest way of fixing the end of a stiffener is by connecting it by an overlap to a member of the hull framing, as for instance a floor plate or longitudinal (fig. 179, *a*). In large ships, where an inner bottom is fitted, this method is not applicable, and brackets must be used. We shall here consider the two principal types of brackets, illustrated by fig. 179, *b* and *c*. In type *b* the bracket consists of a triangular plate attached by a clip or by flanging to the hull structure. The stiffener simply overlaps the bracket and is riveted to it in the same way as it is riveted to the floor plate in type *a*. Here, as in type *a*, the upper rivets will be subject to great shearing stresses, and, moreover, the outstanding edge of the bracket, unless reinforced in some way, will be liable to collapse when under compression. It is, in fact, likely that long before the elastic limit is reached in any part of the stiffener proper, breakdown of the bracket connection will occur, either by shearing of the rivets or by buckling of the edge of the bracket. By the tests, undertaken in the U.S. battleship *Illinois* by Naval Constructor Woodward,* the brackets were of this type and gave way by buckling of the unstiffened outstanding edge. Reversed bars were then fitted at both ends of the stiffeners, reinforcing the edge of the brackets and extending for a certain distance along the stiffeners themselves, and this remedy proved satisfactory. Brackets should, in fact, always be stiffened on the free edge, either by an angle bar or by flanging. A still better solution may be obtained by splitting the stiffener at the ends and bending one leg sufficiently outwards to connect with and reinforce the edge of the bracket, as shown in fig. 179, *c*. When a stiffener is provided with a reversed bar, a face plate, or both, these members should likewise be continued along the edge of the brackets, at least for a certain distance, so as to strengthen the stiffener at the apex of the brackets. The curve formed by the outstanding flange at these points must not be sharp. When constructed in this way, the brackets may be regarded as forming integral parts of the stiffener and their section modulus may without difficulty be made to conform to the great bending moments which exist at the ends when the stiffener is fixed. It has been objected that when a pressure is applied to the bulkhead on the side opposite to the stiffeners, this form of bracket is not efficient because the outstanding flange is not, on account of its curvature, well adapted to resist compression and will be liable to collapse. If, however, the brackets are given a good height and if the curvature at the apices is easy, this objection has no weight in practice, as evidenced by numerous tests. This case is not analogous to that of a column exposed to buckling, because the outstanding flange can neither deflect laterally nor in the plane of the bracket. In fact, the

* *Am. Soc. Nav. Arch. Mar. Eng.*, 1898, p. 172.

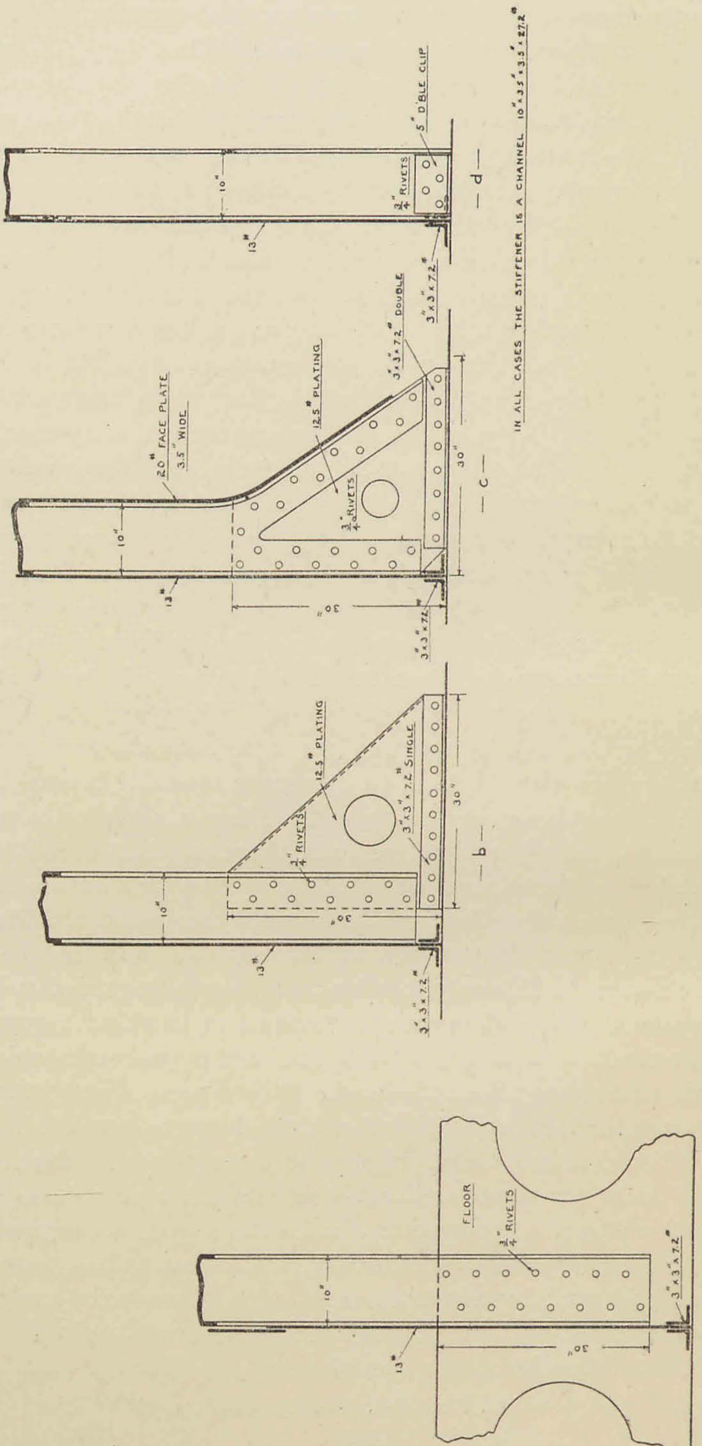

IN ALL CASES THE STIFFENER IS A CHANNEL 10"×3.5"×3.5"×27.2#
— d —
5" D'BLE CLIP
3/4" RIVETS
3"×3"×7.2#
10"
13#
— c —
20# FACE PLATE 3.5" WIDE
12.5# PLATING
3"×3"×7.2# DOUBLE
3/4" RIVETS
30"
— b —
12.5# PLATING
3"×3"×7.2# SINGLE
3/4" RIVETS
— a —
FLOOR
3/4" RIVETS
3"×3"×7.2#

FIG. 179.

flange can only yield by direct crushing after the elastic limit is passed, and this point will not be reached in a well-designed bulkhead.

The connection between the stiffener and the hull structure should be calculated to resist the full bending moments at the ends of the stiffener, a condition which is easy to fulfil when the brackets and the bulkhead plating are connected to the hull structure by double angles, as here recommended. Greater difficulty is sometimes found in providing sufficient stiffness in the hull structure itself. Where possible, the stiffeners should be so placed that the brackets will stand on or directly connect with floors, longitudinals, and beams, or else special reinforcements must be fitted. The type of bracket here recommended (fig. 179, *c*), or its equivalent, should be applied to the stiffeners of all important bulkheads.

In order to equalise, as far as possible, the stresses in the stiffeners, the height of the brackets should be such that the bending moment at the apex is about the same as at the middle of the stiffener, *i.e.* $\frac{Pl}{24}$ · Referring to fig. 175, we should have $CC_1 = DD_1 = OO_1$. Moreover, the section modulus of the brackets should increase from the apex to the base in such a way that the stress shall nowhere exceed that at the points C, O, and D. In other words, the curve of stress should be of the general character indicated in fig. 180, where the stresses at C, O, and D are approximately equal and all slightly greater than at A and B. It will be found that under severe conditions of loading the required approximate equalisation of the bending moments is attained by *making the height of the brackets about equal to one-tenth of the length of the stiffener, and this height should, therefore, be considered as a minimum.*

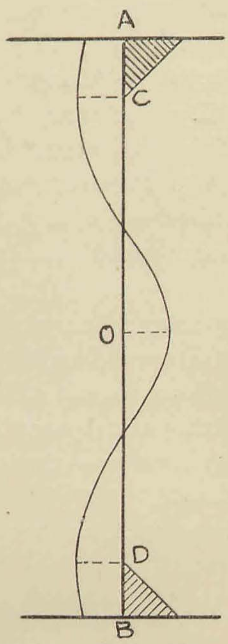


FIG. 180.—Curve of Stress.

It is clear, however, that the height of brackets must be regulated, not only by the length, but also by the strength of the stiffener, *i.e.*, practically, its depth. In case of deck beams, the depth of the knees is about three times the depth of the beam. Adopting this rule also for bulkhead stiffeners, we obtain thus a second minimum value for the height of the brackets. The horizontal side of the brackets should likewise be of a length equal to about three times the depth of the stiffener. In bulkheads of great height, the upper brackets may be constructed somewhat lighter than the lower. Where it is impossible to obtain a rigid attachment at the top of the stiffener, the bracket should be replaced by lugs. In such a case the stiffener will behave as if freely supported at the top and, if fixed at the lower end by an efficient bracket, the maximum bending moment in the free part of the stiffener will not exceed $\frac{1}{14}Pl$ and will occur a little above mid height. At the apex of the bracket the bending moment will be slightly less. In bulkheads of small height and exposed only to small pressures, as between decks at or above the water-line, the stiffeners need not be bracketed, since relatively great deflections can here be accepted. The upper end of such stiffeners may be riveted directly to the beams or may be connected to the deck plating by double clips. The lower end should be connected to the deck by double clips. The stiffeners should take a rivet through the bounding angle of the bulkhead at each end. As shown by experiments, the stiffeners will under these circumstances behave as if they were hinged at the ends, whence the maximum bending moment will occur near the middle and will be approximately equal to $\frac{1}{8}Pl$. This form of attachment is shown in fig. 179, *d.*

3. **Reinforcements.**—Where a door opening is cut between two stiffeners, a bar should be fitted horizontally like a carling above the door. If the door is not placed near the foot of the bulkhead, a similar bar should be fitted below the door. When a door or some other opening severs a stiffener, cross girders of the same section as the main stiffeners should be similarly fitted and efficiently connected to the adjacent uncut stiffeners. In important bulkheads these latter should be buttressed by extending the height of the lower brackets to the cross girder above the opening, or other equivalent means of reinforcement should be fitted. Such compensation for openings in bulkheads is very important, as otherwise marked points of weakness will come to exist.

66. THE BULKHEAD PLATING.

1. **Principal Dimensions.**—The length and breadth of the plates should be as great as practicable, generally the same as for the outside plating.

The thickness must be sufficient to resist the water pressures but, since flooding is contemplated only as an accident, a considerable deflection and even a small permanent set can be accepted at test heads provided the watertightness is not impaired. Judging from experiments and tests, this condition is fulfilled if the thickness is determined in accordance with Curve B, Pl. IV., but while the scantlings so obtained may be satisfactory from all points of view as far as the lower strakes are concerned, they may not always be sufficient to meet other claims in case of the upper strakes. We have to consider the functions of the bulkheads also as stiffening diaphragms and plate girders, in which capacities the top strakes may be as severely strained as the bottom strakes. Provided the forces are known, we are able to calculate the maximum principal and shearing stresses from which the thickness of plating required to resist these stresses can be determined. It may thus be found necessary to augment the thickness of some of the strakes beyond the indications of Curve B.

Sometimes additions must be made to the thickness determined from Curve B in order to secure efficient calking, to provide local strength, and to allow for corrosion.

It will ordinarily be found that after all the different actions are taken into account, the resulting variation in thickness from the top to the bottom is very moderate, even in large bulkheads. In practice the plating is from 14 to 15 lb. in the bottom strake and from 9 to 10 lb. in the top strake of the main bulkheads of large warships. In the British Navy it appears that the thickness is uniform throughout in such bulkheads.*

2. **Edge and Butt Connections.**—On account of the great deflections allowed in bulkhead plating, the edge and butt connections must be very efficient. Single riveting gives an efficiency of only ·55 for 10-lb. plating and ·43 for 15-lb. plating, and under strong deflection single riveted joints are liable to suffer deformation and leakages will occur. This was evidenced in the White Star liner s.s. *Republic*, the loss of which appears to have been caused ultimately by a gradual leakage of the edges and butts in the aft engine-room bulkhead, which were single riveted. *It is, therefore, recommended to use double riveting in all important bulkheads*, including in this term practically all bulkheads below the water-line. An efficiency of about ⅝ will be thus attained as in the outer shell.

3. **Boundary Connections.**—The connection at the boundaries is effected directly by angle bars, single or double, which go all around the contour of the bulkhead, but these bars are assisted by the brackets of the stiffeners, which probably transmit to the structure the greater part of the forces acting on the bulkhead. Hence, the chief duties of the bounding bars are to secure watertightness and to resist the tension which may exist in the plating. The rivets in the vertical bars, being unassisted by stiffeners, are liable to be strained more than the rivets in the top and bottom bars. Under certain circumstances the main bulk-

* *Manual of Seamanship*, 1909, vol. ii. pl. xviii.

heads may be exposed to great shearing forces which cause straining of the rivets in the boundary connections.

Single bounding bars are always fitted on the calked side, whence, if the pressure is applied to this side, the plating will tend to deflect away from the toe of the angle, and the calking is liable to break. This tendency is, of course, greatest along the lower boundary and along the sides of large bulkheads below the water-line, especially where the bulkhead is not stiffened by platform decks. The remedy is to fit "doubling bars," often called "backing bars," which lend support to the main bounding angles. This feature will also enable the bulkhead better to perform its duties as a strength member and diaphragm of the hull structure. In large ships, doubling bars should, speaking broadly, be fitted along the bottom and side boundaries of all bulkheads up to the level of the water-line, except where the bulkheads are subdivided into smaller areas by platforms or by other bulkheads, in which case the doubling need not extend higher than the level of the upper platform deck. In all main bulkheads, however, transverse and longitudinal, the doubling should extend all round the contour. Lloyd's Rules require double angles or else double riveted single angles to the outside plating and double bottom, single angles to the decks.

Bounding bars are usually 3 in. × 3 in., providing for one line of rivets. The pitch is ordinarily four and a half or five diameters, except in the shell flange of the backing bar where it is seven or eight diameters. With single riveted single bounding angles, the strength of the connection is very small. Comparing shearing of the rivets with tearing of the plate, the average efficiency is only one-third. Single bounding bars will, therefore, give decided lines of weakness where they are not reinforced by the brackets of bulkhead stiffeners or longitudinals. With double bounding angles the rivets are in double shear and the efficiency will approach two-thirds.

The bounding bars are stapled on the calked side round all girders and beams that cut through the bulkhead. Fig. 181 shows different types of staples. The general principle underlying their design is that the lines of calking should everywhere form closed circuits. Several circuits may combine and run into each other, but there should be no cusps or discontinuities in the calking lines. The bounding bars are generally joggled over the laps or straps of the inner bottom, bulkheads, and other surfaces on which they are worked. Short liners, taking two rivets each, should be fitted at the jogglings where necessary. Stop-waters are used as indicated in fig. 181. Backing bars are not stapled or calked in ordinary watertight bulkheads.

Where a transverse bulkhead extends to the outer shell the bounding bars take the place of the transverse frame.

Brackets fitted instead of bulkhead liners (SECTION 37, *4*) afford a very

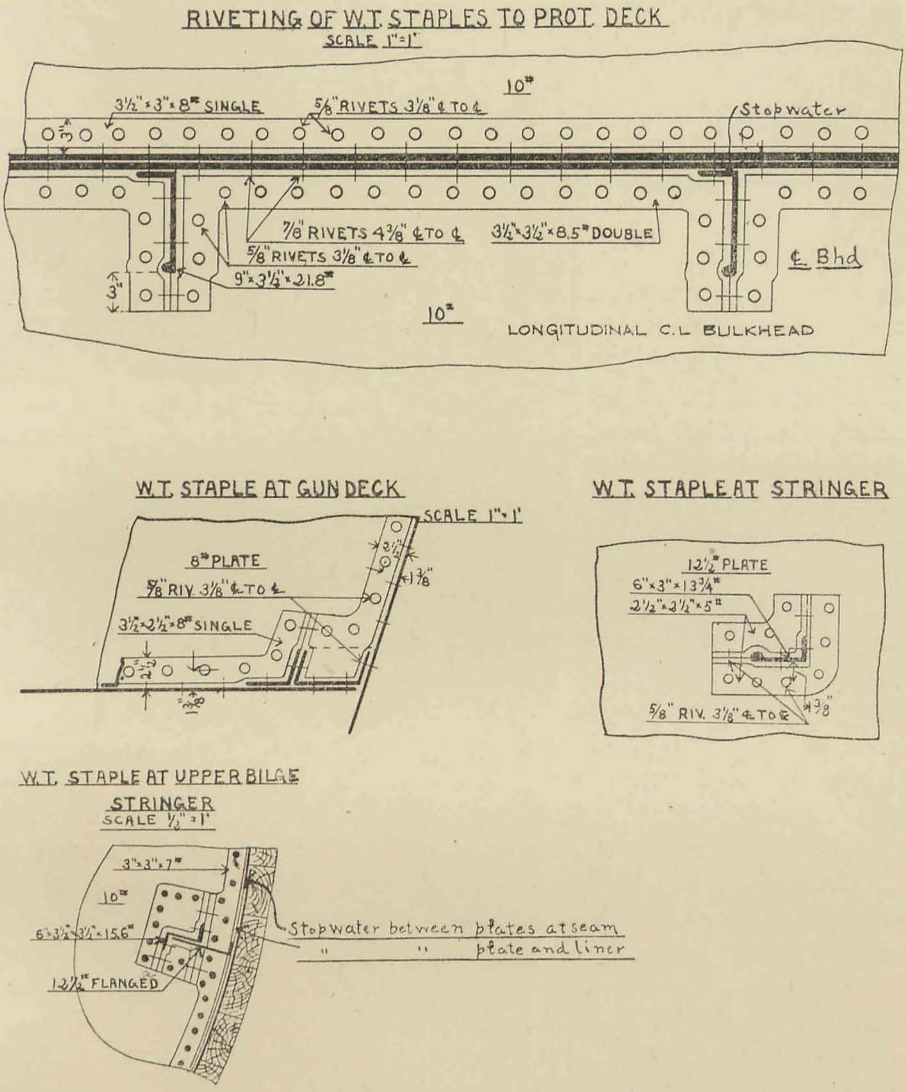


FIG. 181.—Staples on Bulkheads.

valuable boundary connection to a bulkhead. Especially where the stiffeners are vertical, such brackets, fitted on the sides, will relieve the bounding angles and serve to transmit all reactions between the shell and the bulkhead, enabling this latter to perform its functions as a diaphragm in a more efficient manner.

67. TANK BULKHEADS.

1. **Water-Tank Bulkheads.**—Bulkheads of "deep tanks," "trimming tanks," and other compartments, destined to contain water for long periods, should possess a degree of watertightness distinctly superior to that of ordinary bulkheads, but this can only be attained by giving them a much greater stiffness. The stiffeners must, therefore, be so constructed and spaced that the deflection of the bulkhead as a whole as well as the deflection of the plating between the stiffeners will be considerably smaller than in ordinary bulkheads. These claims appear to be best fulfilled by fitting a set of vertical main stiffeners on the uncalked side, constructed as usual but of greater depth and closer spacing, and a horizontal set of light stiffening bars on the calked side. The object of the former set is to support the bulkhead as a whole, while the latter set is to stiffen the plating, subdividing it into rectangles of small area. According to experiments and experience it appears that bulkheads of this class should be so designed that the maximum deflection of the stiffeners will be not greater than one-half of what would be accepted in an ordinary bulkhead. Hence the aggregate moment of inertia of the main stiffeners, including as usual the reinforcement which they receive from the bulkhead plating, should be at least about twice as great as in an ordinary bulkhead designed to work under the same pressures. The spacing of the stiffeners should be so chosen that, without going to excessive thicknesses in the lower strakes, the plating is nowhere strained beyond the elastic limit. This is practically the same requirement as for the outer shell, whence the value of μ may be found from Curve A, Pl. IV. Generally a spacing of 2 ft. or 2 ft. 6 in. will be found suitable, but with this spacing the thickness of the upper strakes as determined from Curve A may in some cases be rather light for efficient calking and will have to be slightly augmented. Here, as elsewhere, the indications of the curves must be regarded as minima values.

Since the main stiffeners run at right angles to the seams, they should preferably be placed on the uncalked side, that is, on the pressure side of the bulkhead. The riveted connection between the plating and the stiffeners should, therefore, be of ample sectional area, best obtained by using two lines of rivets on each stiffener, and the rivets should have a full point. As explained in SECTION 65, *1*, the stiffeners of side bulkheads should always be placed on the side nearest the center-line.

The horizontal stiffeners, being light bars, of relatively great length, should not be taken into account in the strength calculation of the bulkhead as a whole, but their spacing will influence the coefficient K_A

used in finding the thickness of the plating. There should be two such stiffeners on every strake, placed on a quarter width of the plates, symmetrically with respect to the seams. When so placed, the spacing will be about the same as that of the vertical stiffeners, K_A will be equal to ·64, and the maximum deflection relative to the stiffeners of the plating in the unsupported areas will be reduced to about two-thirds of what it would be in the absence of horizontal stiffeners. These latter should be provided with suitable brackets at their ends in order to relieve the strains on the boundary connections. The bounding angles should either be double or single with double zigzag riveting and should extend all around the boundaries.

Collision-bulkheads belong to the same class as deep tank bulkheads and should be similarly constructed but, as already explained, the main stiffeners should be horizontal and the light stiffening bars consequently vertical. The main stiffeners should be placed on the aft side of the bulkhead—the calking side—and should be well bracketed to the longitudinals or to the outside plating.

2. **Oiltight Bulkheads.**—The claim to tightness is here still more severe than for bulkheads of water-tanks. Sufficient stiffness of the bulkhead as a whole will be secured by adopting the same construction and disposition of the stiffeners, but in order to obtain perfectly reliable calking it is advisable to increase the thickness of the plating by from one-sixteenth to one-eighth of an inch as compared with that of water-tank bulkheads. As already explained in previous chapters, the spacing of the rivets should be closer than in ordinary bulkheads and the utmost care should be exercised in the work of riveting and calking. In fact, it is in these details, rather than in general design, that oiltight bulkheads differ from the bulkheads of water-tanks.

It will be noticed that while it is recommended to use Curve A in determining the thickness of the plating in deep water-tanks, it was proposed to use Curve B for the inner bottom plating in way of ballast-tanks in the double bottom. Likewise for deep oil-tanks a relatively greater thickness is recommended than for similar tanks in the double bottom. The reason for these differences is that in deep tanks the pressure under normal conditions is much greater than in the compartments in the lower part of the double bottom and is liable to be increased by the motion of the water or oil. Moreover, the test head, being on the whole less severe for deep tanks, will give a relatively smaller thickness of the plating when used as an argument on Pl. IV.

Where oil-tanks are adjacent to compartments not containing oil or water and not left as empty cells, "cofferdam-bulkheads" should be fitted. A cofferdam-bulkhead consists of two single bulkheads placed a few feet apart. The intervening space, the "cofferdam," may be filled with water, either permanently or in case of emergency, whence one of the bulkheads should be oiltight, the other watertight. Any oil which may find its way into the cofferdam when it is filled with water will float harmlessly to the surface. Generally, each of the two bulkheads is supported by a system of vertical stiffeners of moderate depth, tied together by flanged bracket plates. These plates act as swash-plates but, at the same time, they enable the bulkheads mutually to support and stiffen each other. In fact, by a proper arrangement of the swash-plates, supplemented by some lattice-work, it is easy to make the strength and rigidity of cofferdam-bulkheads exceed that of deep tank bulkheads as ordinarily constructed. When oil-tanks extend across the ship, as is frequently the case in torpedo-vessels, they must be subdivided by longitudinal bulkheads into two or more compartments, each of which should be again subdivided by "swash-bulkheads," so constructed that they permit a certain circulation of the oil, usually at their foot. Swash-bulkheads prevent violent motion of the oil and, if fitted sufficiently close together, they will not be subject to great forces. They at the same time give excellent support to the transverse bulkheads at the ends of the tank. If only one longitudinal bulkhead is fitted, it must be very strongly supported and constructed. In a certain tank steamer, where only a center-line bulkhead was fitted, it was completely torn from its fastenings and at the end of the voyage it was found at the bottom of the tank.

The bounding bars of oiltight bulkheads should be doubled all round the contour, and in large bulkheads they should be stapled and calked on both sides, but only one of the bars need have oiltight spacing of the rivets.

68. PRACTICAL DESIGN OF BULKHEADS.

1. **Rules of Construction.**—The conclusions and recommendations of the foregoing discussion form the basis for the determination of the scantlings and for the Bulkhead Tables given below. They will be here formulated as rules of construction.

(1) The stiffeners should ordinarily be vertical, uniformly spaced, and of the same section throughout. Horizontal stiffeners should be used only where the height of the bulkhead is considerably greater than its breadth and where they, at the same time, can obtain a rigid end attachment. They may also be used in the form of light bars on tank-bulkheads and in some other special cases when the main stiffeners are vertical in order to support the plating locally between these latter.

(2) The stiffeners should generally be spaced the same distance apart as the frames of the ship and should, whenever possible, be placed on the frames, whether these are transverse or longitudinal. In tank bulkheads and in other cases where extra strength or stiffness is required the stiffeners should be spaced half a frame space apart.

(3) The stiffeners should be single and placed on the uncalked side of the bulkhead. On important bulkheads they should be provided on the free flange with a face-plate or reversed bar, to which, as well as to the bulkhead plating, they should be connected by a double line of rivets. The rivets should be not more than five diameters apart, and near the apex of the brackets the spacing should be somewhat closer. Face-plates should not much exceed the width of the flange to which they are attached.

(4) The stiffeners should be provided with efficient brackets both at top and bottom, constructed as indicated in fig. 179, *c*, by splitting the main bar of the stiffener. The height and width of the lower bracket should be not less than three times the depth of the stiffener, nor less than one-tenth of the length. The sides of the upper brackets in bulkheads of great height may be somewhat smaller. The brackets should be attached to rigid members of the ship's framework or else equivalent reinforcements should be introduced. In unimportant bulkheads and where it is impossible or impracticable to provide a rigid end attachment the stiffeners should be held at the ends by angle clips as shown in fig. 179, *d*.

(5) The plating should be arranged in horizontal strakes, decreasing slightly in thickness from the bottom upwards. The seams and butts should be double riveted everywhere below the deck nearest the water-line, usually the protective or splinter deck.

(6) The bounding angles of the main bulkheads and of all tank bulkheads should be doubled all around the boundaries. In less important bulkheads the doubling may stop at the level of the water-line or at the deck nearest below the water-line.

2. **Determination of Scantlings.**—Supposing these rules to be complied with, we shall now proceed to determine the scantlings of a bulkhead of given height, subject to a certain head of water.

Suppose first that the stiffeners are bracketed at both ends and that they have no intermediate points of support. With the construction of brackets here proposed, the effective length of the stiffeners l may be taken equal to the entire height of the bulkhead except where, as may occur in a longitudinal bulkhead, the stiffeners are bracketed to the deck beams, in which case l should be measured from the underside of the beams. If a is the head of water above the top of the stiffeners, $l+a$

is always the total head of water at the foot. The value of a which determines the test head should be chosen so as to correspond to the deepest draught to which the ship may reasonably be assumed to be immersed when in damaged condition. Due regard should be had to the trim, to the type and size of the ship, and to the location of the bulkhead. Bulkheads placed near the ends or near the sides are liable to be exposed to greater heads of water than when placed near the half length or at the center-line of the ship. Allowance should be made also for the increased pressures which may come to exist when the water in a flooded compartment is put in motion by pitching and rolling of the ship. In the s.s. *Republic* the final breakdown of the engine-room bulkhead occurred when the ship was taken in tow and brought head on to the sea. The determination of the test head must always be largely a matter of judgment, but much guidance may be obtained by basing the judgment on a simple formula which in a rational manner takes into account the main elements that influence the problem. We shall here attempt to frame such a formula expressing the height of the deepest draught water-line above the normal load water-line at any point distant x forward or aft of the midship section. Let this height be H_x. At amidships H_x here denoted by H_0 must depend on the difference in draught between the normal and full load conditions, on the sinkage due to flooding, and on the heel. We shall assume that these factors are taken into account by making H_0 a certain fraction of the draught and propose $H_0 = \frac{D}{4}$. This gives a test head amidships which is a mean between that adopted in the British and the United States Navies. It is assumed further that the severest case of flooding that needs to be considered is when the main deck * is immersed at the bow or stern, this deck being virtually the "bulkhead deck" in warships, since above it watertight bulkheads are rarely, if ever, fitted. Let F be the freeboard of the main deck at the bow or the stern, according as the bulkhead under consideration is located in the fore-body or the after-body. We have then

$$H_x = \frac{D}{4} + \frac{2\left(F - \frac{D}{4}\right)}{L} x \quad . \quad . \quad . \quad . \quad (126)$$

where L is the length of the ship. For the collision bulkhead we have approximately $H_x = F$, *i.e.* this bulkhead is to be tested to a head of water corresponding to the height of the main deck.

Let us apply this formula to a battleship where L = 650 ft.,

* The English "upper deck."

D = 28 ft., and F = 20 ft. in the bow. Then we have at amidships H_0 = 7 ft., in the bow $H_{\frac{L}{2}}$ = 20 ft., and for a bulkhead for instance 130 ft. forward of the midship section we find

$$H = 7 + \frac{2(20-7)}{650} \times 130 = 12 \text{ ft.}$$

At this point, then, a bulkhead, the top of which is, say, 3 ft. above the normal load water-line, would have to be tested to a head of 9 ft. above the top, whence a = 9 ft.

As explained above (SECTION 65, *2*), the bending moment for bracketed stiffeners is reckoned equal to $\frac{Pl}{24}$ where P is the total load on the stiffener and is given by (125)

$$P = \frac{sl}{35}\left[a + \frac{l}{2}\right] \text{ tons}$$

whence

$$M = \frac{sl^2}{70}\left[a + \frac{l}{2}\right] \text{ inch-tons} \quad . \quad . \quad . \quad (127)$$

From the analysis of bulkhead tests referred to above, the allowable or working stress in the stiffeners, calculated as here described, was found to be 10 ts. per sq. in.* With this stress, bulkheads, constructed essentially as recommended above, were able satisfactorily to stand the severe tests prescribed in the United States Navy. The deflection was moderate and the watertightness satisfactory. For instance, bulkheads with bracketed stiffeners, of 25 ft. height, exposed to a head of 7 ft. above the top, showed a maximum elastic deflection of the stiffeners of less than two-thirds of an inch and a permanent set of about one-sixth of an inch. This corresponds to $\frac{\delta}{l} = \frac{1}{450}$ and $\frac{1}{1800}$ respectively. In architectural work the allowable deflection of beams is generally reckoned $\frac{1}{360}$ of the span.

With such small deflections the tension in the stiffeners is negligible and we need only consider their strength in bending.

The required section modulus, $S = \frac{I}{y}$ may now be determined since its value must not fall below that of $\frac{M}{p}$. From (127) and with p = 10 ts. per sq. in. we find the minimum value

$$S = \frac{sl^2}{700}\left[a + \frac{l}{2}\right] \quad . \quad . \quad . \quad . \quad (128)$$

* *Am. Soc. Nav. Arch. Mar. Eng.*, 1910, p. 95.

If the water in the flooded compartment does not reach the top of the bulkhead, the maximum bending moment may still be reckoned approximately equal to $\frac{Pl}{24}$ but then

$$P = \frac{s}{70}[l-a]^2 \text{ tons} \quad . \quad . \quad . \quad . \quad (129)$$

where a is now the height of the top of the bulkhead above the water-line of deepest immersion. Hence

$$S = \frac{sl}{1400}[l-a]^2 \quad . \quad . \quad . \quad . \quad (130)$$

The section modulus was calculated from (128) and (130) for the range of values of l and a likely to be useful in practice and the results are given in Table XVII. This table holds good for stiffeners bracketed at both ends and spaced 4 ft. apart, but the section modulus for any other spacing can readily be found, since the values of S are directly proportional to s.

It remains to select a stiffener which possesses the required section modulus. This work is facilitated by the Tables XVIII.–XXIII., which give the section moduli for different types of stiffeners both with and without face plates or reversed bars. Since the stress p which enters in the calculation of the section modulus in Table XVII., was based on an analysis of experiments where the bulkhead plating was assumed to contribute to the strength of the stiffeners according to a certain rule, the same assumption and the same rule were adopted in computing the S-values for the Tables XVIII.–XXIII. In this latter computation, however, the thickness of the bulkhead plating was not exactly known, whence it was necessary for each stiffener to estimate approximately the sectional area of the strip of plating which could be expected to work with it. The most satisfactory approximation was obtained by making the sectional area of the strip equal to that of the flange attached to the bulkhead, since in that case the width of the strip would be roughly equal to thirty times its thickness with appropriate addition where there are two lines of rivets, as assumed in the analysis of the experiments. Tables XVIII.–XXIII. are thus in good accordance with the experimental results. Where a face-plate is added to the free flange or flanges of the stiffener it was likewise given the same, or about the same, sectional area as the flange or flanges attached to the bulkhead.

The use of the tables is simple. Corresponding to the given values of l, a, and s the required section modulus is found from Table XVII. and the most appropriate stiffener can then be readily selected from

Tables XVIII.-XXIII. For water-tank or oil-tank bulkheads, when no cofferdam is fitted, the section modulus is first found as for an ordinary bulkhead, usually corresponding to a reduced spacing of the stiffeners. A stiffener is chosen which possesses this section modulus, its moment of inertia is calculated, and finally a stiffener of twice this moment of inertia is selected. Another and simpler method may be used which, however, in most cases will give a somewhat heavier stiffener. Let s be the spacing of the stiffeners in the tank bulkhead. Find the stiffener appropriate for an ordinary bulkhead under the same head but with a spacing of the stiffeners of $2s$. The stiffener so obtained will then evidently with a spacing equal to s fulfil the required condition as to moment of inertia.

When the stiffeners are not bracketed at both ends the section modulus cannot be found from Table XVII. but must be calculated independently by determining the maximum bending moment in the free part of the stiffener and dividing it with 10 ts. per sq. in., the allowable working stress. The stiffener can then be selected from the Tables as usual. It has already been shown that when no brackets are fitted, the stiffener may be considered as freely supported at both ends and the maximum bending moment will be $\frac{1}{8}Pl$. When a stiffener is bracketed at the foot only, while the top, being connected by lugs, may be considered as freely supported, the maximum bending moment in the free part of the stiffener is found a little above mid-height and may be reckoned equal to $\frac{1}{14}Pl$. When a stiffener, bracketed at both ends, is supported at the middle, as for instance by a platform deck, the maximum bending moment between the brackets is found at the intermediate point of support and will be $\frac{1}{48}Pl$. In this case the sides of the brackets can be reduced, those at the top to one-half, those at the bottom to two-thirds of the ordinary dimensions. For other special cases the bending moment must be determined independently by the usual methods.

The thicknesses of the plating recommended in Table XVII. apply only to ordinary watertight bulkheads and are based on experience, guided by the results of tests and by theoretical considerations. The table is constructed in accordance with the simple rule that the lowest strake is made $\left(6+\frac{l+a}{4}\right)$ lb. and the other strakes are reduced, going upwards, by about one pound for every six feet. The thicknesses so determined agree well with the best practice, provided the assumed deepest draught water-line is chosen in accordance with formula (126). They do not deviate much from the values given by Curve B, Pl. IV., for a spacing of the stiffeners of about 3 ft. 3 in. When the spacing is smaller than about three feet, the thicknesses might indeed be reduced throughout if only water pressures had to be considered, but other claims to the bulkheads, notably in their capacity as diaphragms and girders, make it inadvisable to go appreciably below the thicknesses given in the table. For a spacing of four feet the figures in the table must be regarded as minimum values. The weight per square foot of plating is given for the top and bottom strakes, the weight for intermediate strakes may be found by interpolation in accordance with the rule given above.

For water-tank bulkheads the thickness of the lower strake may be determined by Curve A, Pl. IV., and the other strakes are reduced in thickness at the same rate as in ordinary bulkheads. For oiltight bulkheads, find first the thicknesses as for a water-tank bulkhead and make an addition of from 2 to 5 lb. according to the size of the tank.

3. **Examples.**—*Example* 1.—An ordinary watertight bulkhead is 24 ft. high and is to be tested to a head of 12 ft. above the top. The spacing of the stiffeners is 4 ft. Find the scantlings of the stiffeners and the thickness of the plating. We have

$$l = 24\,, \quad a = 12\,, \quad s = 4\,.$$

From Table XVII. we find S = 79 which according to Table XVIII. is amply obtained with an I-bar 12 in. × 5·09 in. × 5·09 in. × 35 lb. provided with face-plate 8 in. × $\frac{1}{2}$ in.

The plating will be 11 lb. at the top and 15 lb. at the foot.

Suppose a bulkhead of the same dimensions and the same test head is to serve for an oil-tank, then a satisfactory construction would be obtained by simply fitting the same 12-in. I-bars 2 ft. apart, in which case the deflection would be reduced to one-half of its amount in an ordinary bulkhead; but we may also proceed as follows:

The section modulus for an ordinary watertight bulkhead where the stiffeners are spaced 2 ft. apart is $\frac{1}{2} \times 79 = 39{\cdot}5$, which could be provided by a 10-in. channel bar with face-plate, where $y = 5{\cdot}5$ in. Hence $I = 5{\cdot}5 \times 39{\cdot}5 = 217$. The required moment of inertia per stiffener is therefore $2I = 434$, which may be obtained with a 12 in. × 5 in. × 5 in. × 31·5 lb. I-bar with face-plate 8 in. × $\frac{3}{8}$ in. This stiffener, which is somewhat lighter than that found by the first method, has a section modulus $S = 70$, giving $I = 6{\cdot}38 \times 70 = 447$.

The horizontal bars may be 3 in. × 3 in. spaced about 2 ft. 3 in. apart.

The thickness of the plating determined as for a deep water-tank bulkhead is found from Curve A, Pl. IV. For the lowest strake $h = 36$ ft., and for $r = 1{\cdot}125$, $K_A = {\cdot}75$, $K_A h = 27$ ft., whence $\mu = 65$

and $t = \frac{24}{65} = 15$ lb. The upper strake will then, with the same rate of reduction in thickness as in ordinary bulkheads, be 11 lb. Since the bulkhead is to be oiltight these figures should be augmented to 18 lb. for the bottom strake and 14 lb. for the top strake.

Example 2.—In the battleship, the framing system of which is shown in fig. 115, the lowest longitudinal girder in the wing space is to support the shell plating at a head of about 35 ft. when the ship is in a seaway or when in damaged condition. The girder is bracketed at the transverse wing bulkheads, which are spaced 8 ft. apart. The spacing of the girders is about 3 ft. 4 in. The total load is therefore $P = \frac{8 \times 3{\cdot}33 \times 35}{35} = 26{\cdot}6$ ts.

Assuming perfect fixity at the points of intersection with the bulkheads, and that the maximum stress occurs in the free part of the girder as for bulkhead stiffeners, we have

$$M = \tfrac{1}{24}Pl$$

but in the case of a water-tank bulkhead we want to obtain twice the moment of inertia of that required for an ordinary bulkhead and therefore select a stiffener suitable for a spacing of twice 3 ft. 4 in. Hence

$$M = \tfrac{1}{24}2Pl = \tfrac{1}{24}2 \times 26{\cdot}6 \times 12 \times 8 = 213 \text{ in.-ts.}$$

and with $p = 10$ ts. per sq. in. we have $S = 21{\cdot}3$, which can be obtained with a bulb angle 10 in. × 3·5 in., British Standard Section (Table XXII.).

For the upper girder, placed 13 ft. higher, we find that a bulb angle 8·5 in. × 3·5 in. will be sufficient.

69. TESTS OF BULKHEADS.

It would seem rational to test each bulkhead to a pressure corresponding to a water-line determined by some formula such as (126), but, so far, more crude rules are adopted in practice by which the location of the bulkhead in the ship is disregarded. In the British Navy the assumed deepest draught water-line is 5 feet above the normal load water-line. In the United States Navy it is, in large ships, 35 feet above the bottom of the keel, or from 8 to 10 feet above the normal load water-line; in small ships it is 5 feet above this line. Lloyd's Rules require that deep water-ballast tanks shall be tested by a head of water 8 feet above the top of the tank, but not less in any case than to the height of the load water-line.* Oil compartments are to be tested by a

* This rule is adopted in the *International Convention on Safety of Life at Sea*, 1914.

head of water 8 feet above the highest point of the expansion trunk. Germanischer Lloyd requires oil-tanks to be tested by a head of water at least 11·5 feet (3·5 meters) above the highest point of the tank deck or, when the tank deck is below the load water-line, 11·5 feet above this line.

Before a compartment is tested, it should be ascertained that all rivets are driven, all holes plugged and all calking completed. A filling-pipe, through which the water is pumped into the compartment, an air-pipe, and a stand-pipe are fitted. The stand-pipe is open at the top and should rise to such a height that the desired head of water is obtained. It will thus act at once as an overflow pipe and as an indicator. All air should be carefully emptied from the compartment before the air escape is plugged, otherwise the indications of the stand-pipe will be uncertain due to cushioning of the air and the pulsations of the filling pump.

All compartments adjacent to the outer shell should be tested. That part of the outer shell which will be submerged after launching, should be tested while the ship is on the ways. All compartments forward and abaft the machinery spaces should be tested, and at least one of the large main bulkheads in an engine- or boiler-room. This latter test is often carried out by constructing a cofferdam at the bulkhead. Where several ships of identical design are built, the main bulkheads, tested in the different ships, should not be the same, and an effort should be made to test each type of large bulkhead. If all six bounding walls of a compartment have satisfactorily stood the test of adjacent compartments it is unnecessary to fill such a compartment.

The bulkheads that are not tested under pressure may be examined for watertightness by means of a hose. By the hose a great head of water can be readily produced, but the pressure is local and can only serve to discover such leakages as exist in the bulkhead initially or under a very small pressure. This method of testing does not, therefore, reveal what will take place when the bulkhead as a whole is strained under great pressures. Moreover, the efficiency of the hose depends much on how it is used. The edges and rivets may also be examined by means of the testing knife and hammer.

It is clear from the foregoing that we must distinguish between two kinds of leaks :

(1) Initial leaks, that exist with any head of water and which are due to imperfect workmanship.

(2) Leaks due to straining of the bulkhead, which do not, in general, occur till the head of water is near the maximum, when excessive deflection may cause breakage of the calking and leakage of the rivets.

In other words, there are two distinct elements to be tested, workmanship and design. It appears, therefore, that if one of the main bulkheads has been tested, and its general strength found satisfactory, the other main bulkheads, if similarly constructed, need only be tested and examined for workmanship by the hose and by the testing knife and hammer. In testing, it is advantageous to apply the pressure so that the calked side is open to inspection.

70. PROTECTIVE BULKHEADS.

The term "protective bulkhead" is here applied only to bulkheads designed as protection against underwater explosions. The problem of providing such protection is at the present time one of prime importance in the design of warships; in fact, it may be said that the future of the large seagoing battleship depends essentially on its solution. Apart from a subdivision of the hold in watertight compartments, two methods are used to attain this object: (1) by plane armor bulkheads, (2) by so-called "elastic" bulkheads, *i.e.* bulkheads in which the plating is dished or curved and resists the pressures in virtue of tension.

1. **Plane Armor Bulkheads.**—In recent vessels of the battleship class it is customary to fit an armored side bulkhead in way of the vitals below the protective deck on each side. In some ships this bulkhead is fitted outside the bunkers, only from six to eight feet inside the outer shell, in others it is fitted inside the bunkers much farther from the side. As in protective decks two modes of construction are used:

(1) Heavy armor plates of from 1-in. to 2-in. thickness of nickel steel are fitted on a layer of mild steel plating of about ½-in. thickness. Usually the armor plates have no seam or butt connections, but the mild steel plates are provided with straps.

(2) Two and sometimes three layers of plating of mild or high-tensile steel of moderate thickness, aggregating from 1¼ in. to 2½ in., are worked as explained for protective decks. Usually the joints are unstrapped, but by a good shift of the seams and butts a fair efficiency is attained.

It is clear that in bulkheads of type (1), as constructed at present, the armor will open along the joints at relatively moderate gas pressures and, probably, the mild steel plating will tear. If, then, this type is adopted, the armor plates should be scarphed at the seams and butts as explained for protective decks or else provided with strapped connections. The plates should be as large as practicable. The efficiency of the joints will, however, always be low, and great bending stresses will come to exist locally at the stiffeners and perhaps also midway between these.

Bulkheads of type (2) are better able to withstand gaseous pressures than those of type (1) on account of their greater yielding capacity, although again the flat form renders them unfit to resist pressures of extreme magnitude. They might be given a high efficiency at the joints by means of straps, but this would add considerably to the weight and cost.

In both types of bulkheads the stiffeners are now usually of about the same scantlings as in ordinary watertight bulkheads, capable of resisting only low pressures. It is likely, therefore, that such bulkheads will be, partly or entirely, blown bodily inwards under the enormous pressures created by underwater explosions.

That existing protective bulkheads possess serious points of weakness, as here explained, seems to be confirmed by the disappointing results of experiments recently carried out with armored bulkheads in various countries. It appears that if the strength of the armor is to be developed even in a very moderate measure, it is necessary to provide better edge connections at seams, butts, and boundaries and far more substantial stiffeners than now ordinarily fitted. *It would seem rational to adjust the strength of the stiffeners to match the strength of the plating, so that the elastic limit will be reached in both of these parts of the structure at approximately the same pressure load.*

Let us now try to determine the load at which the armor plating reaches the elastic limit, assuming first that the edge connections as well as the stiffeners possess adequate strength. Consider, for instance, a bulkhead of type (1) constructed of $1\frac{1}{2}$-in. nickel-steel plating with an elastic limit of 23·7 ts. per sq. in. The support which the light mild-steel plating gives to the armor will be here disregarded. The stiffeners are supposed to be vertical, spaced 48 in. apart, and connected to the plating by double 4-in. angles. An elemental horizontal strip of plating between two stiffeners is regarded as "fixed" at the ends. Its unsupported length between the edges of the angles, where the greatest bending stresses occur, is $48-8 = 40$ in. Its width is 1 in. Let P be the total load on the strip, then the maximum bending moment is $M = \frac{1}{12} \times 40 \times P$; but $M = \frac{p_T I}{y}$ where $p_T = 23{\cdot}7$ ts. per sq. in., $I = \frac{1}{12}t^3 = \frac{3{\cdot}375}{12}$, and $y = \frac{3}{4}$ in. Hence $P = 2{\cdot}67$ ts. and the load per inch run of the stiffeners is $\frac{48}{40} \times 2{\cdot}67 = 3{\cdot}21$ ts., which corresponds to a head of about 340 ft. of water. This, then, is the point at which the elastic limit will be reached, and the metal will begin to crack or flow locally at the lines of support along the stiffeners. In reality, the bulkhead will give way long before this by failure of the edge connections even with the best practicable construction. Also the stiffeners will break down unless they are made much stronger than in ordinary bulkheads, but the requirements in this respect as well as the detail design of the stiffeners will be fully discussed in the next Article.

2. **Elastic Bulkheads.**—Since the primary object of protective bulkheads is to resist very high gaseous pressures, it appears rational to design them on the same principle as the shell of a cylindrical boiler, which, as known from experience, can be easily made to resist pressure heads of several hundred feet without going to excessive thicknesses of plating. It seems, in fact, as irrational to construct a protective bulkhead with a plane surface as it would be to construct a boiler in the form of a rectangular box. We have seen from Bach's experiments, described in SECTION **20**, that when a plate is even slightly dished, it is capable of carrying enormous pressures without rupture. This is due to the fact that the dished plate acts essentially as an elastic membrane, every part of which is in tension, the even distribution of stresses being favorable to a full development of the strength of the material. The greater the initial curvature, the greater will be within certain limits the carrying power of the plate. The best form for the plating between two bulkhead stiffeners is evidently that of a circular cylindrical surface, and the theoretical maximum strength is reached when the cylinder has its center in the plane of the bulkhead, *i.e.* when its diameter is equal to the free distance between the stiffeners. This can easily be shown.

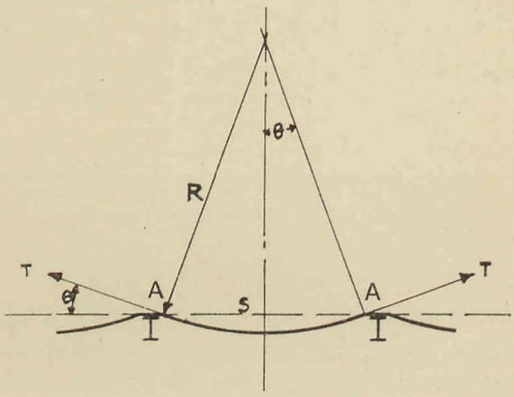


FIG. 182.

Let the circular arc AA in fig. 182 represent a section normal to the plating between two stiffeners. Let R be the radius of the arc, 2θ the total angle subtended by the arc, s the unsupported span, and p the gaseous pressure. T is the tension in an elemental strip of the plating. We have then

$$2T \sin\theta = ps$$

but

$$2R \sin\theta = s$$

whence

$$T = pR$$

The tension in the plating for a given pressure is therefore proportional to the radius of the cylinder and since this is a minimum when the center is on the line AA the carrying capacity of the strip is a maximum when $2R = s$. Let p_T be the tensile stress in the plating and t the thickness, we have then for any value of R

$$p = \frac{2p_T t \sin}{s} \quad . \quad . \quad . \quad . \quad . \quad (131)$$

It is essential that local strains shall not exist anywhere in the plating, wherefore the supporting surfaces on the stiffeners should be given an easy curve.

Elastic bulkheads were first introduced in the French Navy, although not in the form here advocated. In the battleships of the *Bretagne* class the bulkheads are, according to press reports,* constructed, as shown in fig. 183, of three thicknesses of 10-mm. plating, two of which are plane, while the third is dished. The two plane thicknesses are evidently

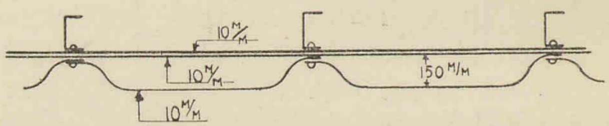


FIG. 183.

intended to provide watertightness as in an ordinary bulkhead, and to arrest flying fragments, while the dished plates are to resist the gaseous pressures after the other plates are broken through. By this construction space is saved as compared with the ideal cylindrical form, but the resistance offered by the plates to gas pressures is considerably smaller. The mode of attachment of the dished plates is not efficient, since the plates will be torn away from the stiffeners long before their full strength is developed. Finally, the strength of the bulkhead as a whole is limited by the strength of the stiffeners, which are of the same type and scantlings as in an ordinary bulkhead.

As far as resistance to gas pressures is concerned, a much stronger bulkhead can be obtained by curving the plates to a form approaching that of semi-cylinders and by giving to the stiffeners a strength corresponding to the enormous resistance which the plating will then offer. The general arrangement of such a bulkhead is shown on fig. 115 and details of construction are given on fig. 184. The bulkhead is of the double or cofferdam type. The outer wall is built of curved plates, the inner—the "cofferdam bulkhead"—is of ordinary construction. Deep web stiffeners separate the two bulkheads and are firmly attached to the

* *Le Yacht*, October 11, 1913, p. 654.

hull structure at top and bottom. The curved plating is arranged in vertical strakes connected midway between the stiffeners by double butt

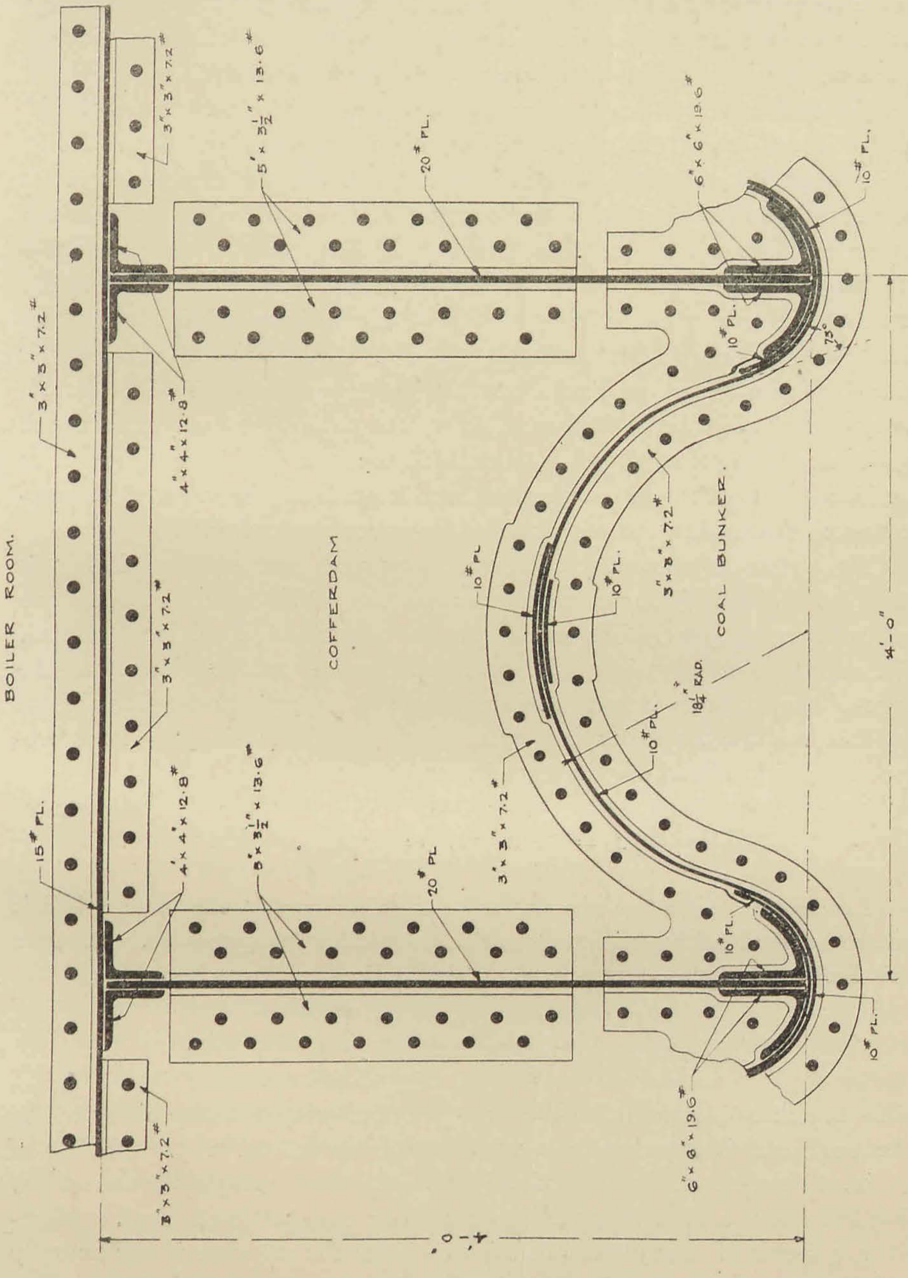


FIG. 184.—Light Elastic Bulkhead, Corrugated Type.

straps of the same type as used in the shell of boilers and described in SECTION **28**, *5*. The flanges of the outer angle bars of the stiffeners are bent rather sharply and curved face-plates are riveted to them, serving to strengthen the stiffeners and affording at the same time a curved and elastic support for the plating. The elastic bulkhead has double bounding bars round the entire contour, it is calked as far as possible or stopwaters are used, but absolute watertightness is evidently unessential.

We shall now try to determine the scantlings of such a "corrugated" bulkhead and investigate, in particular, where its weak points are likely to be found. Let us suppose, for instance, that the bulkhead is to resist a head of 340 ft. which was the outside limit to the pressure that the armor bulkhead, considered in the last Article, was capable of withstanding. Referring again to figs. 115 and 184, which show the solution at which we arrive, the spacing of the stiffeners as well as their depth and hence the depth of the cofferdam is 4 ft. This spacing fits in well with the framing system, and a depth of 4 ft. permits convenient access everywhere inside the cofferdam. As will be shown presently, this depth is sufficient for attaining the required strength and the space thus sacrificed does not seem excessive in a large ship, in view of the importance of providing an adequate under-water protection.*

The radius of curvature of the unsupported part of the plating is about 18 in. and of the bearing surface on the stiffeners $6\frac{5}{8}$ in., the two curves merging gently into each other so as to avoid all local strains. The unsupported span is 35 in. and the tangent to the plating at the edge of the supports is inclined at an angle $\theta = 73^\circ$ to the plane of the bulkhead. The load per inch run of the stiffeners is 3·21 ts. corresponding to the head of 340 ft., whence

$$ps = \frac{35}{48} \times 3{\cdot}21 = 2{\cdot}34 \text{ ts.}$$

Let us now examine what must be the thickness of the plating if it is not to be strained beyond the elastic limit under this pressure. Assume this limit to be $p_T = 17$ ts. per sq. in. We have then from (131)

$$t = \frac{ps}{2p_T \sin\theta} = \frac{2{\cdot}34}{2 \times 17 \times {\cdot}956} \text{ or about } \tfrac{1}{16} \text{ in.}$$

This thickness is much too small for practical purposes, especially where the bulkhead is adjacent to a coal bunker as in fig. 115. Plating of $\frac{1}{16}$ in. thickness cannot be calked and leaves no margin for corrosion. Bunker doors and other openings cannot be satisfactorily fitted. Moreover, a factor of safety must be allowed since the pressure is not actually

* On the midship section of a light cruiser shown in fig. 116, the depth of the cofferdam is only 2 ft. 3 in., but the plating of the elastic bulkhead is not there so strongly curved.

uniform; in fact, pressures, far greater than the average, are liable to exist locally where the bulkhead is struck directly by the gas jet. It is, therefore, proposed to make the curved plating of $\frac{1}{4}$ in. thickness, which must indeed be considered a minimum for such a bulkhead. The efficiency of the seam straps as designed on fig. 184 is ·87.

The stiffeners are divided by a platform deck into two parts, of which we need here only consider the lower, which is considerably longer than the upper and more directly exposed to the pressures. The length of the lower part is 15 ft. and will be considered as "fixed" at both ends. It is clear that such a short and deep girder is more liable to fail by shearing than by bending. The weak points will be the web, which will be liable to buckle, and the riveted attachments at the foot. We shall, therefore, first consider these parts of the structure. The total load on one stiffener is

$$P = \frac{15 \times 340 \times 4}{35} = 583 \text{ ts.}$$

whence the shearing force at the foot is, $Q = 292$ ts.

With the construction adopted on fig. 184, where the web is of $\frac{1}{2}$ in. thickness, we find the moment of inertia of the section, $I_0 = 21{,}500$ sq. in. × (in.)^2, and the moment about the neutral axis of the sectional area on one side of this axis, $m_0 = 368$ sq. in. × (in.). The maximum shearing stress at the foot of the stiffener will therefore be

$$q_0 = \frac{Qm_0}{tI_0} = \frac{292 \times 368}{\frac{1}{2} \times 21500} = 10 \text{ ts. per sq. in.}$$

and the virtual principal stress at the neutral axis 13 ts. per sq. in. In the example SECTION **22**, *2* it was found that with this stress and with $\frac{1}{2}$-in. plating web stiffeners should be fitted 24 in. apart in order to obtain a proper safety against wrinkling. Similarly we find that on one-quarter of the length from the end, where the shearing stress under a uniform load is half as great as at the end, the spacing of the web stiffeners should be about 35 in. These conditions are amply fulfilled by the web stiffeners shown in fig. 115. At the middle of the stiffeners, where there is no shearing under a uniform load, a hole is cut to permit passage from one section to another.

The rivet area required for each stiffener at the foot of the bulkhead, reckoning the elastic limit in shearing to be 13 ts. per sq. in., is $\frac{292}{13} = 22{\cdot}5$ sq. in., which will be supplied by about fifty $\frac{3}{4}$-in. rivets. This area is present provided we include the rivets in the bounding bars in the vicinity of the stiffeners.

Consider finally the resistance of the stiffeners to bending. The section modulus is about 800 sq. in. × (in.), which, with an elastic limit of 17 ts. per sq. in., will enable the stiffeners to support a bending moment of 13,600 in.-ts. This corresponds to a head of about 530 ft., considerably greater than required. Even assuming imperfect fixity at the ends the stiffeners will be amply strong in bending.

It appears then, as stated above, that the most probable mode of failure is by straining of the rivets at the foot, possibly accompanied by buckling of the web. Even considerable straining of the rivets with consequent leakage will not, however, be a serious matter as long as the inner bulkhead is fairly tight.

A somewhat simpler design, which will be here referred to as the "cusp" type, is shown in fig. 185, where the bevelled angles on the

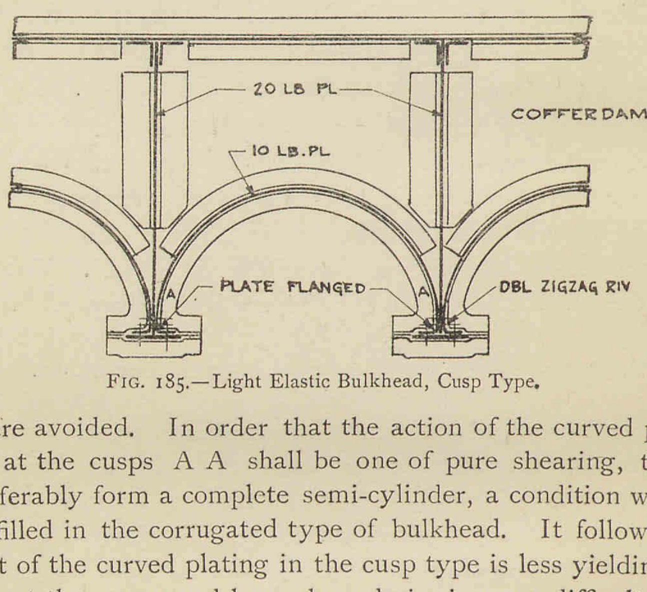


FIG. 185.—Light Elastic Bulkhead, Cusp Type.

stiffeners are avoided. In order that the action of the curved plating on the rivets at the cusps A A shall be one of pure shearing, the plating should preferably form a complete semi-cylinder, a condition which need not be fulfilled in the corrugated type of bulkhead. It follows that the attachment of the curved plating in the cusp type is less yielding, a good connection at the upper and lower boundaries is more difficult to obtain, and less space is left inside the cofferdam.

The principle underlying the design of elastic bulkheads finds extensive application in submarine boats, where, as stated in SECTION 64, *r*, certain transverse bulkheads are often of dished form, and also the inner longitudinal walls of internal tanks are in many boats strongly curved, concave on the pressure side. Probably this principle will find an even wider application in the larger submarine vessels of the future.

3. **Comparison between Plane Armor Bulkheads and Elastic Bulkheads.**—Assuming the same construction of the stiffeners as well as the existence of a cofferdam bulkhead in either case, it is evident that

there must be a considerable saving in weight by the elastic bulkhead. Moreover, the strength of the curved plating will be more uniform and reliable and can easily be made far greater than that of the armor bulkhead whatever the strength of its edge connections. It may be objected to elastic bulkheads that they possess no longitudinal strength, but this quality may be secured by a proper construction of the cofferdam bulkhead.

The only serious disadvantage of the light elastic bulkhead as compared with the plane armor bulkhead is its smaller resistance to flying fragments and to attack by projectiles such as those from the Davis torpedo-gun. These dangers can be much reduced by stowing coal outside of the elastic bulkhead,* but coal protection may not always be available and in ships burning oil-fuel exclusively it cannot be obtained at all. Hence, while the light elastic bulkhead appears to offer the best solution of the problem of underwater protection in light cruisers where no weight can be spared for armor, it is not fully satisfactory in large battleships and battle-cruisers. In such vessels it is preferable to combine the features of armored and elastic bulkheads and thus obtain at once a great resistance against gas pressures and against flying fragments. We shall examine this combination.

4. **Elastic Armor Bulkheads.**—When the plating of an elastic bulkhead is of great thickness, its curvature need only be very small and ample strength can be secured even with a low efficiency of the joints. We shall investigate and illustrate the possibilities of this mode of construction by a concrete example.

Assume the plating to consist of two thicknesses of $\frac{3}{4}$-in. nickel steel with an elastic limit of 23·7 ts. per sq. in., the conditions being otherwise the same as in the example of Article 2. Pressure head 340 ft. or 3·21 ts. per in. run of the stiffeners. Unsupported span of elemental strip between two stiffeners is here 36 in., whence

$$ps = \frac{36}{48} \times 3{\cdot}21 = 2{\cdot}41 \text{ ts.}$$

Allowing for impulsive action and for uneven distribution of the pressures, let us reckon that the pressure load is liable to be locally about four times as great, or that $ps = 10$ ts.

Suppose that the joints of the two layers of plating are not strapped, but that they are merely shifted well clear of each other, then the strength at the vertical joints will be that of one $\frac{3}{4}$-in. plate weakened by rivet holes. Assuming the joints so designed that the efficiency of

* *Oberon* experiments. Brassey, *The British Navy*, ii. p. 21, London, 1882.

the unbutted plate is $\frac{3}{4}$, then there remains $\frac{3}{4} \times \frac{3}{4} = \frac{9}{16}$ in. as effective thickness. The angle of inclination θ at the end of the arc between the stiffeners is found from (131):

$$\sin\theta = \frac{ps}{2p_{T}t} = \frac{10}{2 \times 23{\cdot}7 \times \frac{9}{16}} = {\cdot}375$$

$$\therefore \quad \theta = 22^\circ$$

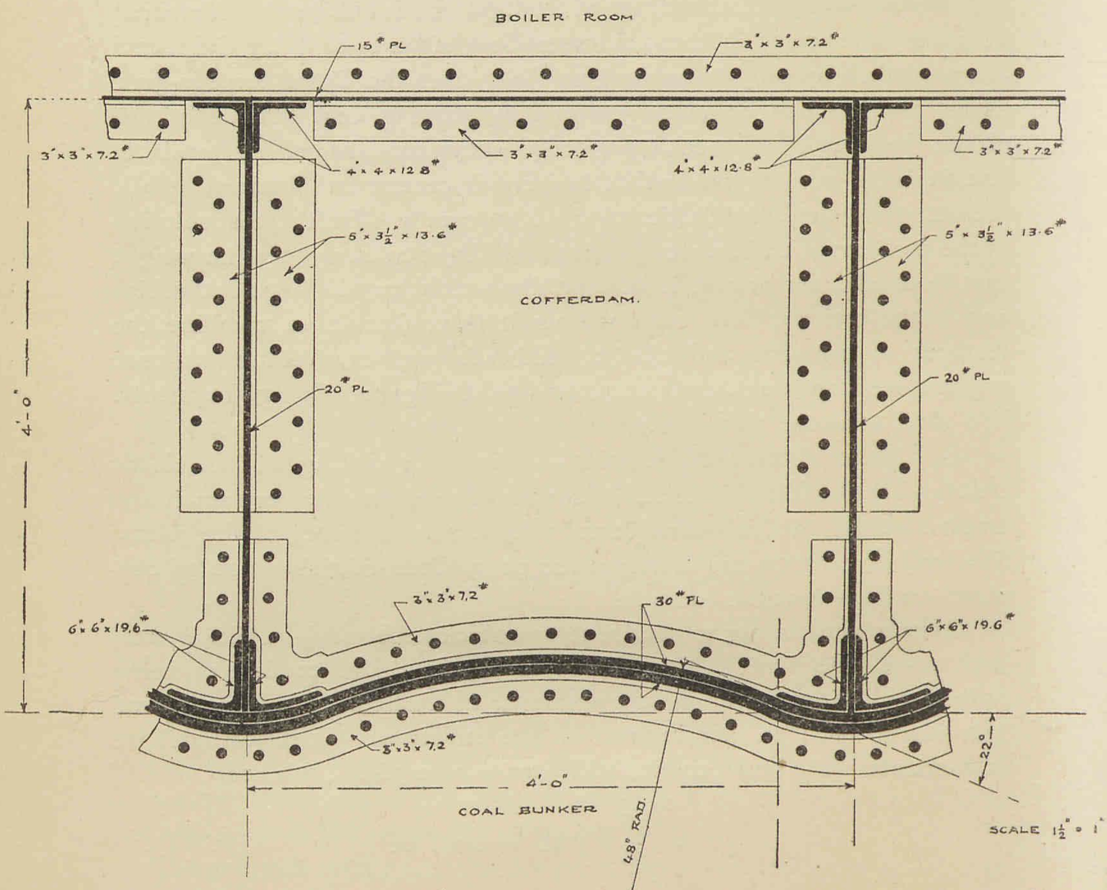


FIG. 186.—Elastic Armor Bulkhead, corrugated type.

The radius of the arc is

$$R = \frac{s}{2\sin\theta} = \frac{36}{2 \times {\cdot}375} = 48 \text{ in.}$$

and the depth of the arc inside the chord will be $48(1 - {\cdot}927) = 3{\cdot}5$ in. The bulkhead so designed is shown in fig. 186; it possesses at least the same resistance to the impact of flying fragments as a plane bulkhead of the same thickness and a vastly greater resistance to the explosive pressures. There will be no difficulty in making it perfectly watertight or even oiltight, and the calking is less likely to be broken than in a light elastic bulkhead. It is conceivable that an elastic armor bulkhead would possess sufficient strength even when fitted at a distance of some eight or ten feet from the outer shell, but as a precaution it would be well if possible to have coal protection inside. In such a case the space outside the bulkhead might serve as an "explosion chamber," and the inner bunker bulkhead would perhaps render the cofferdam bulkhead superfluous.

Commander G. E. Elia of the Italian Navy has proposed an elastic bulkhead without any stiffeners, curved in the vertical plane to one large arc extending from the armor shelf to the turn of the bilges. As designed and presented by Mr. T. G. Owens,* the bulkhead is a combination of an outer light elastic and an inner armored elastic bulkhead. The principle is the same as in the bulkheads proposed above, but the span of the arc being here much greater there will be an enormous tangential pull at the upper and lower boundaries normal to the outer shell. A similar bulkhead but of polygonal shape and of three thicknesses is fitted in the *Danton*.†

5. **Necessity of Experiments.**—The most important novel features of the bulkheads recommended above are :

(1) A plating of curved form.

(2) A system of stiffeners, the strength of which is commensurate with the strength of the plating.

While the advantage of adopting these features seems obvious *a priori*, the type and location of the bulkhead, the thickness and curvature of the plating, the value of coal protection, and many questions of detail can only be settled experimentally. Experiments are, in fact, urgently needed ; they should be full scale and carried out analytically, each one of the various features being tested separately. The great expenditure entailed by this procedure seems fully warranted by the importance of the subject.

71. NON-WATERTIGHT BULKHEADS.

Practically all bulkheads below the water-line or below the deck next above the water-line should be watertight, but in the upper part of the ship non-watertight bulkheads may be employed to a considerable extent, especially above the second deck. Four different types of bulkheads are

* *Inst. Nav. Arch.*, 1914, p. 11 and Pl. III. † *Le Yacht*, Oct. 11, 1913, p. 653.

used for non-watertight subdivision: (1) Structural bulkheads, (2) Pilaster bulkheads, (3) Corrugated metal bulkheads, (4) Wire-mesh and expanded-metal bulkheads. Only the first of these types will be here discussed, as the others are of no structural importance.

Structural bulkheads are usually of from 5 to 7½ lb. plating stiffened by light angles. They are used where a certain measure of local strength is required as in store-rooms and wherever it is desired to stiffen the structure or to support weights. Heavier scantlings are used under the conning-tower and under other very great weights.

Structural bulkheads should generally extend from the plating of one deck to that of another, but partial transverse bulkheads are connected at the top by an overlap to the beam. In longitudinal bulkheads the plating must be scored out for the beams and plate collars fitted so as to secure fairly tight work. Airtightness is of particular importance in bulkheads surrounding water-closet spaces, sick-bay, and compartments belonging to the sick-bay. Coaming plates of somewhat heavier plating, 10 lb., should be fitted at the bottom of all light bulkheads of this class in order to provide for strength, watertightness, and corrosion. Usually the bottom angle of non-watertight bulkheads is calked. Transverse bulkheads, when complete from side to side, and all longitudinal bulkheads should have a fairly substantial bounding plate also at the top for the sake of strength. The stiffeners should take rivets through the bounding angles both at top and bottom, and where they fall on beams they should be directly riveted to these at the top. The seams of the plates are usually single riveted, while the butts are often connected by double riveted single butt straps. The plating should be worked in the greatest possible width, in general there will be only one strake besides the bounding plates.

The uptake and engine-hatch enclosure bulkheads belong to this class and are commonly built of 5-lb. plating in large ships. As explained above, the coaming plates should here be very substantial and efficiently connected to the decks.

Table XVII.—Thickness of Bulkhead Plating

l = length of stiffener. a = head of water above top of stiffener. s = span of stiffeners. $S = \frac{I}{y}$ = section modulus

l, Feet.	a, Feet.	Weight of Plating. Upper Strake, lb.	Weight of Plating. Lower Strake, lb.	S. s = 4 ft.
8	−4	6	7	·37
8	−2	7	8	·82
8	0	7	8	1·5
8	2	8	9	2·2
8	4	8	9	2·9
8	6	9	10	3·7
8	8	9	10	4·4
8	10	10	11	5·1
8	12	10	11	5·8
8	14	11	12	6·6
8	16	11	12	7·3
8	18	12	13	8·0
8	20	12	13	8·8
8	22	13	14	9·5
8	24	13	14	10·2
8	26	14	15	11·0
8	28	14	15	11·7
8	30	15	16	12·4
8	32	15	16	13·2
8	34	16	17	13·9
8	36	16	17	14·6
8	38	17	18	15·4

l, Feet.	a, Feet.	Weight of Plating. Upper Strake, lb.	Weight of Plating. Lower Strake, lb.	S. s = 4 ft.
10	−4	7	8	1·0
10	−2	7	8	1·8
10	0	8	9	2·9
10	2	8	9	4·0
10	4	9	10	5·1
10	6	9	10	6·3
10	8	10	11	7·4
10	10	10	11	8·6
10	12	11	12	9·7
10	14	11	12	10·9
10	16	12	13	12·0
10	18	12	13	13·1
10	20	13	14	14·3
10	22	13	14	15·4
10	24	14	15	16·6
10	26	14	15	17·7
10	28	15	16	18·9
10	30	15	16	20·0
10	32	16	17	21·1
10	34	16	17	22·3
10	36	17	18	23·4

l, Feet.	a, Feet.	Weight of Plating. Upper Strake, lb.	Weight of Plating. Lower Strake, lb.	S. s = 4 ft.
12	−4	6	8	2·2
12	−2	7	9	3·4
12	0	7	9	4·9
12	2	8	10	6·6
12	4	8	10	8·2
12	6	9	11	9·9
12	8	9	11	11·5
12	10	10	12	13·2
12	12	10	12	14·8
12	14	11	13	16·5
12	16	11	13	18·1
12	18	12	14	19·8
12	20	12	14	21·4
12	22	13	15	23·0
12	24	13	15	24·7
12	26	14	16	26·3
12	28	14	16	28·0
12	30	15	17	29·6
12	32	15	17	31·3
12	34	16	18	32·9

l, Feet.	a, Feet.	Weight of Plating. Upper Strake, lb.	Weight of Plating. Lower Strake, lb.	S. s = 4 ft.
14	−4	7	9	4·0
14	−2	7	9	5·8
14	0	8	10	7·8
14	2	8	10	10·1
14	4	9	11	12·3
14	6	9	11	14·6
14	8	10	12	16·8
14	10	10	12	19·0
14	12	11	13	21·3
14	14	11	13	23·5
14	16	12	14	25·8
14	18	12	14	28·0
14	20	13	15	30·2
14	22	13	15	32·5
14	24	14	16	34·7
14	26	14	16	37·0
14	28	15	17	39·2
14	30	15	17	41·4
14	32	16	18	43·7

l, Feet.	a, Feet.	Weight of Plating. Upper Strake, lb.	Weight of Plating. Lower Strake, lb.	S. s = 4 ft.
16	−6	6	9	4·6
16	−4	6	9	6·6
16	−2	7	10	9·0
16	0	7	10	11·7
16	2	8	11	14·6
16	4	8	11	17·6
16	6	9	12	20·5
16	8	9	12	23·4
16	10	10	13	26·3
16	12	10	13	29·3
16	14	11	14	32·2
16	16	11	14	35·1
16	18	12	15	38·0
16	20	12	15	41·0
16	22	13	16	43·9
16	24	13	16	46·8
16	26	14	17	49·7
16	28	14	17	53·0
16	30	15	18	56·0

l, Feet.	a, Feet.	Weight of Plating. Upper Strake, lb.	Weight of Plating. Lower Strake, lb.	S. s = 4 ft.
9	−4	6	7	·64
9	−2	7	8	1·3
9	0	7	8	2·1
9	2	8	9	3·0
9	4	8	9	3·9
9	6	9	10	4·9
9	8	9	10	5·8
9	10	10	11	6·7
9	12	10	11	7·6
9	14	11	12	8·6
9	16	11	12	9·5
9	18	12	13	10·4
9	20	12	13	11·3
9	22	13	14	12·3
9	24	13	14	13·2
9	26	14	15	14·1
9	28	14	15	15·0
9	30	15	16	16·0
9	32	15	16	16·9
9	34	16	17	17·8
9	36	16	17	18·7

l, Feet.	a, Feet.	Weight of Plating. Upper Strake, lb.	Weight of Plating. Lower Strake, lb.	S. s = 4 ft.
11	−4	6	8	1·5
11	−2	7	8	2·5
11	0	7	9	3·8
11	2	8	9	5·2
11	4	8	10	6·6
11	6	9	10	8·0
11	8	9	11	9·3
11	10	10	11	10·7
11	12	10	12	12·1
11	14	11	12	13·5
11	16	11	13	14·9
11	18	12	13	16·3
11	20	12	14	17·6
11	22	13	14	19·0
11	24	13	15	20·4
11	26	14	15	21·8
11	28	14	16	23·2
11	30	15	16	24·5
11	32	15	17	25·9
11	34	16	17	27·3

l, Feet.	a, Feet.	Weight of Plating. Upper Strake, lb.	Weight of Plating. Lower Strake, lb.	S. s = 4 ft.
13	−4	6	8	3·0
13	−2	7	9	4·5
13	0	7	9	6·3
13	2	8	10	8·2
13	4	8	10	10·1
13	6	9	11	12·1
13	8	9	11	14·0
13	10	10	12	15·9
13	12	10	12	17·9
13	14	11	13	19·8
13	16	11	13	21·7
13	18	12	14	23·7
13	20	12	14	25·6
13	22	13	15	27·5
13	24	13	15	29·5
13	26	14	16	31·4
13	28	14	16	33·3
13	30	15	17	35·2
13	32	15	17	37·2

l, Feet.	a, Feet.	Weight of Plating. Upper Strake, lb.	Weight of Plating. Lower Strake, lb.	S. s = 4 ft.
15	−6	6	8	3·5
15	−4	7	9	5·2
15	−2	7	9	7·2
15	0	8	10	9·6
15	2	8	10	12·2
15	4	9	11	14·8
15	6	9	11	17·4
15	8	10	12	19·9
15	10	10	12	22·5
15	12	11	13	25·1
15	14	11	13	27·6
15	16	12	14	30·2
15	18	12	14	32·8
15	20	13	15	35·4
15	22	13	15	37·9
15	24	14	16	40·5
15	26	14	16	43·1
15	28	15	17	45·6
15	30	15	17	48·2

l, Feet.	a, Feet.	Weight of Plating. Upper Strake, lb.	Weight of Plating. Lower Strake, lb.	S. s = 4 ft.
17	−6	6	9	5·9
17	−4	6	9	8·2
17	−2	7	10	10·9
17	0	7	10	14·0
17	2	8	11	17·3
17	4	8	11	20·6
17	6	9	12	23·9
17	8	9	12	27·3
17	10	10	13	30·6
17	12	10	13	33·9
17	14	11	14	37·2
17	16	11	14	40·5
17	18	12	15	43·8
17	20	12	15	47·1
17	22	13	16	50·0
17	24	13	16	54·0
17	26	14	17	57·0
17	28	14	17	60·0

AND SECTION MODULUS OF STIFFENERS.

where I is the moment of inertia about the neutral axis, y is the distance of the most strained fiber from the neutral axis.

l, Feet.	a, Feet.	Weight of Plating.		S.
		Upper Strake, lb.	Lower Strake, lb.	s = 4 ft.
18	− 6	6	9	7·4
18	− 4	7	10	10·1
18	− 2	7	10	13·2
18	0	8	11	16·7
18	2	8	11	20·4
18	4	9	12	24·1
18	6	9	12	27·8
18	8	10	13	31·5
18	10	10	13	35·2
18	12	11	14	38·9
18	14	11	14	42·6
18	16	12	15	46·3
18	18	12	15	50·0
18	20	13	16	54·0
18	22	13	16	57·0
18	24	14	17	61·0
18	26	14	17	65·0
18	28	15	18	69·0

l, Feet.	a, Feet.	Weight of Plating.		S.
		Upper Strake, lb.	Lower Strake, lb.	s = 4 ft.
20	− 6	7	10	11·2
20	− 4	7	10	14·6
20	− 2	8	11	18·5
20	0	8	11	22·9
20	2	9	12	27·4
20	4	9	12	32·0
20	6	10	13	36·6
20	8	10	13	41·1
20	10	11	14	45·7
20	12	11	14	50·0
20	14	12	15	55·0
20	16	12	15	59·0
20	18	13	16	64·0
20	20	13	16	69·0
20	22	14	17	73·0
20	24	14	17	78·0
20	26	15	18	82·0

l, Feet.	a, Feet.	Weight of Plating.		S.
		Upper Strake, lb.	Lower Strake, lb.	s = 4 ft.
22	− 6	6	10	16·1
22	− 4	7	11	20·4
22	− 2	7	11	25·1
22	0	8	12	30·4
22	2	8	12	36·0
22	4	9	13	41·5
22	6	9	13	47·0
22	8	10	14	53·0
22	10	10	14	58·0
22	12	11	15	64·0
22	14	11	15	69·0
22	16	12	16	75·0
22	18	12	16	80·0
22	20	13	17	86·0
22	22	13	17	91·0
22	24	14	18	97·0

l, Feet.	a, Feet.	Weight of Plating.		S.
		Upper Strake, lb.	Lower Strake, lb.	s = 4 ft.
24	− 6	7	11	22·2
24	− 4	7	11	27·4
24	− 2	8	12	33·2
24	0	8	12	39·5
24	2	9	13	46·1
24	4	9	13	53·0
24	6	10	14	59·0
24	8	10	14	66·0
24	10	11	15	72·0
24	12	11	15	79·0
24	14	12	16	86·0
24	16	12	16	92·0
24	18	13	17	99·0
24	20	13	17	105·0
24	22	14	18	112·0

l, Feet.	a, Feet.	Weight of Plating.		S.
		Upper Strake, lb.	Lower Strake, lb.	s = 4 ft.
26	− 6	7	11	29·7
26	− 4	8	12	36·0
26	− 2	8	12	42·8
26	0	9	13	50·0
26	2	9	13	58·0
26	4	10	14	66·0
26	6	10	14	73·0
26	8	11	15	81·0
26	10	11	15	89·0
26	12	12	16	97·0
26	14	12	16	104·0
26	16	13	17	112·0
26	18	13	17	120·0
26	20	14	18	127·0

l, Feet.	a, Feet.	Weight of Plating.		S.
		Upper Strake, lb.	Lower Strake, lb.	s = 4 ft.
19	− 6	6	9	9·2
19	− 4	7	10	12·2
19	− 2	7	10	15·7
19	0	8	11	19·6
19	2	8	11	23·7
19	4	9	12	27·9
19	6	9	12	32·0
19	8	10	13	36·1
19	10	10	13	40·2
19	12	11	14	44·4
19	14	11	14	48·5
19	16	12	15	53·0
19	18	12	15	57·0
19	20	13	16	61·0
19	22	13	16	65·0
19	24	14	17	69·0
19	26	14	17	73·0

l, Feet.	a, Feet.	Weight of Plating.		S.
		Upper Strake, lb.	Lower Strake, lb.	s = 4 ft.
21	− 6	6	10	13·5
21	− 4	6	10	17·3
21	− 2	7	11	21·7
21	0	7	11	26·5
21	2	8	12	31·5
21	4	8	12	36·5
21	6	9	13	41·6
21	8	9	13	46·6
21	10	10	14	52·0
21	12	10	14	57·0
21	14	11	15	62·0
21	16	11	15	67·0
21	18	12	16	72·0
21	20	12	16	77·0
21	22	13	17	82·0
21	24	13	17	87·0

l, Feet.	a, Feet.	Weight of Plating.		S.
		Upper Strake, lb.	Lower Strake, lb.	s = 4 ft.
23	− 6	6	10	9·0
23	− 4	7	11	23·7
23	− 2	7	11	29·0
23	0	8	12	34·8
23	2	8	12	40·8
23	4	9	13	46·9
23	6	9	13	53·0
23	8	10	14	59·0
23	10	10	14	65·0
23	12	11	15	71·0
23	14	11	15	77·0
23	16	12	16	83·0
23	18	12	16	89·0
23	20	13	17	95·0
23	22	13	17	101·0

l, Feet.	a, Feet.	Weight of Plating.		S.
		Upper Strake, lb.	Lower Strake, lb.	s = 4 ft.
25	− 6	7	11	25·8
25	− 4	7	11	31·5
25	− 2	8	12	37·8
25	0	8	12	44·6
25	2	9	13	52·0
25	4	9	13	59·0
25	6	10	14	66·0
25	8	10	14	73·0
25	10	11	15	80·0
25	12	11	15	88·0
25	14	12	16	95·0
25	16	12	16	102·0
25	18	13	17	109·0
25	20	13	17	116·0

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TABLE XVIII.—CARNEGIE STEEL COMPANY'S SHAPES.

I-Bars.

Size.	Thickness of Web.	Weight per Foot Run.	Dimensions of Face-Plate.	Section Modulus.	
				Without Face-Plate.	With Face-Plate.
in.	in.	lb.	in.	sq. in. × in.	sq. in. × in.
5 × 3·00 × 3·00	·210	9·75	$4 \times \frac{5}{16}$	5·5	10·5
5 × 3·15 × 3·15	·357	12·25	$4 \times \frac{3}{8}$	6·5	12·3
5 × 3·29 × 3·29	·504	14·75	$4 \times \frac{7}{16}$	7·4	14·0
6 × 3·33 × 3·33	·230	12·25	$5 \times \frac{3}{8}$	8·4	17·8
6 × 3·45 × 3·45	·352	14·75	$5 \times \frac{7}{16}$	9·6	20·2
6 × 3·58 × 3·58	·475	17·25	$5 \times \frac{1}{2}$	10·8	22·6
7 × 3·66 × 3·66	·250	15·00	$5 \times \frac{3}{8}$	11·8	22·5
7 × 3·76 × 3·76	·353	17·50	$5 \times \frac{7}{16}$	13·1	25·3
7 × 3·87 × 3·87	·458	20·00	$5 \times \frac{1}{2}$	14·5	28·1
8 × 4·00 × 4·00	·270	18·00	$6 \times \frac{3}{8}$	16·1	31·0
8 × 4·09 × 4·09	·357	20·50	$6 \times \frac{7}{16}$	17·7	34·8
8 × 4·18 × 4·18	·449	23·00	$6 \times \frac{1}{2}$	19·2	38·4
9 × 4·33 × 4·33	·290	21·00	$6 \times \frac{3}{8}$	21·2	37·7
9 × 4·45 × 4·45	·406	25·00	$6 \times \frac{7}{16}$	23·5	42·3
9 × 4·61 × 4·61	·569	30·00	$6 \times \frac{1}{2}$	26·7	47·5
10 × 4·66 × 4·66	·310	25·00	$7 \times \frac{3}{8}$	27·4	49·0
10 × 4·81 × 4·81	·455	30·00	$7 \times \frac{7}{16}$	30·9	55·0
10 × 4·95 × 4·95	·602	35·00	$7 \times \frac{1}{2}$	34·5	62·0
10 × 5·10 × 5·10	·749	40·00	$7 \times \frac{9}{16}$	38·1	68·0
12 × 5·00 × 5·00	·350	31·50	$8 \times \frac{3}{8}$	40·3	70·0
12 × 5·00 × 5·00	·350	31·50	$8 \times \frac{7}{16}$	40·8	76·0
12 × 5·09 × 5·09	·436	35·00	$8 + \frac{7}{16}$	43·5	77·0
12 × 5·09 × 5·09	·436	35·00	$8 \times \frac{1}{2}$	44·0	83·0
12 × 5·25 × 5·25	·460	40·00	$8 \times \frac{1}{2}$	51·0	89·0
12 × 5·25 × 5·25	·460	40·00	$8 \times \frac{9}{16}$	51·0	95·0
12 × 5·37 × 5·37	·576	45·00	$8 \times \frac{9}{16}$	55·0	98·0
12 × 5·37 × 5·37	·576	45·00	$8 \times \frac{5}{8}$	56·0	103·0
12 × 5·49 × 5·49	·699	50·00	$8 \times \frac{5}{8}$	59·0	106·0
12 × 5·49 × 5·49	·699	50·00	$8 \times \frac{11}{16}$	60·0	111·0
15 × 5·50 × 5·50	·410	42·00	$8 \times \frac{7}{16}$	66·0	108·0
15 × 5·55 × 5·55	·460	45·00	$8 \times \frac{1}{2}$	69·0	117·0
15 × 5·65 × 5·65	·558	50·00	$8 \times \frac{9}{16}$	74·0	128·0
15 × 5·75 × 5·75	·656	55·00	$8 \times \frac{5}{8}$	80·0	138·0
15 × 6·00 × 6·00	·590	60·00	$8 \times \frac{9}{16}$	91·0	143·0
15 × 6·10 × 6·10	·686	65·00	$8 \times \frac{5}{8}$	96·0	153·0
15 × 6·19 × 6·19	·784	70·00	$8 \times \frac{11}{16}$	102·0	164·0

Note.—A strip of bulkhead plating of the same sectional area as the face-plate is assumed to work with the stiffeners.

TABLE XIX.—CARNEGIE STEEL COMPANY'S SHAPES.

Z-Bars.

Without Reversed Bar.				With Reversed Bar.		
Size.	Thickness of Metal.	Weight per Foot Run.	Section Modulus.	Size of Reversed Bar.	Weight per Foot Run.	Section Modulus.
in.	in.	lb.	sq. in. × in.	in.	lb.	sq. in. × in.
$4 \times 3\frac{1}{16} \times 3\frac{1}{16}$	$\frac{1}{4}$	8·2	4·2	...	...	...
$4 \times 3\frac{1}{16} \times 3\frac{1}{16}$	$\frac{7}{16}$	13·8	6·3	...	...	...
$4 \times 3\frac{1}{16} \times 3\frac{1}{16}$	$\frac{5}{8}$	18·9	7·8	...	...	...
$5 \times 3\frac{1}{4} \times 3\frac{1}{4}$	$\frac{5}{16}$	11·6	6·1	...	...	...
$5 \times 3\frac{1}{4} \times 3\frac{1}{4}$	$\frac{1}{2}$	17·9	8·7	...	...	...
$5 \times 3\frac{1}{4} \times 3\frac{1}{4}$	$\frac{11}{16}$	23·7	10·7	...	...	...
$6 \times 3\frac{1}{2} \times 3\frac{1}{2}$	$\frac{3}{8}$	15·6	9·5	3 × 3	7·2	14·1
$6 \times 3\frac{1}{2} \times 3\frac{1}{2}$	$\frac{9}{16}$	22·7	12·9	3 × 3	7·2	17·0
$6 \times 3\frac{1}{2} \times 3\frac{1}{2}$	$\frac{3}{4}$	29·3	15·5	3 × 3	7·2	19·4

Note.—A strip of bulkhead plating of 1·25 sq. in. in sectional area is assumed to work with the stiffeners.

TABLE XX.—CARNEGIE STEEL COMPANY'S SHAPES.

Shipbuilding Channels.

Size.	Thickness of Web.	Weight per Foot Run.	Section Modulus.	
			Without Face-Plate.	With Face-Plate.
in.	in.	lb.	sq. in. × in.	sq. in. × in.
6 × 3·50 × 3·50	·35	15·0	9·5	15·4
7 × 3·45 × 3·45	·45	20·9	14·7	23·3
8 × 3·50 × 3·50	·50	23·8	18·3	28·2
9 × 3·80 × 3·80	·45	28·6	26·4	42·4
10 × 3·38 × 3·38	·38	21·8	20·8	31·7
10 × 3·50 × 3·50	·50	27·2	25·1	37·4

Note.—A strip of bulkhead plating of sectional area equal to that of the flange attached to the bulkhead is assumed to work with the stiffeners. Face-plates are assumed to have the same width and thickness as the flange.

TABLE XXI.—BRITISH STANDARD SECTIONS.

Lloyd's Rules, 1909–10.

Channels.

Correct Standard Profile.					Maximum Web Thickness.				
Size.	Thickness.		Section Modulus.		Size.	Thickness.		Section Modulus.	
	Web.	Flange.	Without Face-Plate.	With Face-Plate.		Web.	Flange.	Without Face-Plate.	With Face-Plate.
in.	in.	in.	sq. in. × in.	sq. in. × in.	in.	in.	in.	sq. in. × in.	sq. in. × in.
6·0 × 3·0 × 3·0	·375	·475	10·0	16·0	6·0 × 3·2 × 3·2	·575	·475	11·5	17·6
6·0 × 3·5 × 3·5	·375	·475	11·3	18·5	6·0 × 3·5 × 3·5	·575	·475	12·8	20·1
7·0 × 3·0 × 3·0	·375	·475	12·3	19·4	7·0 × 3·2 × 3·2	·575	·475	14·3	21·4
7·0 × 3·5 × 3·5	·400	·500	14·3	23·1	7·0 × 3·7 × 3·7	·600	·500	16·4	25·3
7·5 × 3·0 × 3·0	·400	·500	14·1	23·5	7·5 × 3·2 × 3·2	·600	·500	16·5	26·1
7·5 × 3·5 × 3·5	·400	·500	15·9	25·4	7·5 × 3·7 × 3·7	·600	·500	18·3	27·8
8·0 × 3·0 × 3·0	·400	·500	15·0	23·5	8·0 × 3·2 × 3·2	·600	·500	18·1	26·7
8·0 × 3·5 × 3·5	·425	·525	18·2	28·7	8·0 × 3·7 × 3·7	·625	·525	20·9	31·4
8·0 × 4·0 × 4·0	·450	·550	21·0	33·8	8·0 × 4·2 × 4·2	·650	·550	23·7	36·6
8·5 × 3·0 × 3·0	·425	·525	17·6	27·0	8·5 × 3·2 × 3·2	·625	·525	20·5	30·0
8·5 × 3·5 × 3·5	·450	·550	20·3	32·1	8·5 × 3·7 × 3·7	·650	·550	23·3	35·1
9·0 × 3·5 × 3·5	·450	·550	22·2	32·5	9·0 × 3·7 × 3·7	·650	·550	25·6	38·1
9·0 × 4·0 × 4·0	·475	·575	23·4	38·6	9·0 × 4·2 × 4·2	·675	·575	26·8	42·1
9·5 × 3·5 × 3·5	·450	·550	23·9	37·1	9·5 × 3·7 × 3·7	·650	·550	27·6	40·9
10·0 × 3·5 × 3·5	·475	·575	26·8	41·1	10·0 × 3·7 × 3·7	·675	·575	30·9	45·3
10·0 × 4·0 × 4·0	·475	·575	29·6	46·3	10·0 × 4·2 × 4·2	·675	·575	33·8	50·5
10·5 × 3·5 × 3·5	·475	·575	28·7	43·7	10·5 × 3·7 × 3·7	·675	·575	33·2	48·3
11·0 × 3·5 × 3·5	·475	·575	30·6	46·4	11·0 × 3·7 × 3·7	·675	·575	35·6	51·0
11·0 × 4·0 × 4·0	·500	·600	35·2	54·0	11·0 × 4·2 × 4·2	·700	·600	40·1	59·0
11·5 × 3·5 × 3·5	·500	·600	34·0	51·0	11·5 × 3·7 × 3·7	·700	·600	39·3	56·0
12·0 × 3·5 × 3·5	·500	·600	36·1	54·0	12·0 × 3·7 × 3·7	·700	·600	41·9	60·0
12·0 × 4·0 × 4·0	·525	·625	41·2	63·0	12·0 × 4·2 × 4·2	·725	·625	47·1	68·0

TABLE XXII.—BRITISH STANDARD SECTIONS.

Lloyd's Rules, 1909–10.

Bulb Angles.

Correct Standard Profile.				Maximum Web Thickness.			
Size.	Thickness.		Section Modulus.	Size.	Thickness.		Section Modulus.
	Web.	Flange.			Web.	Flange.	
in.	in.	in.	sq. in. × in.	in.	in.	in.	sq. in. × in.
5·5 × 3·0	·350	·350	4·8	5·5 × 3·2	·550	·450	6·6
6·0 × 3·0	·375	·375	5·8	6·0 × 3·2	·575	·475	8·0
6·5 × 3·0	·375	·375	6·8	6·5 × 3·2	·575	·475	9·2
6·5 × 3·5	·400	·400	7·7	6·5 × 3·7	·600	·500	9·8
7·0 × 3·0	·400	·400	8·4	7·0 × 3·2	·600	·500	11·1
7·0 × 3·5	·425	·425	8·9	7·0 × 3·7	·625	·525	11·8
7·5 × 3·0	·425	·425	10·0	7·5 × 3·2	·625	·525	13·5
7·5 × 3·5	·425	·425	10·5	7·5 × 3·7	·625	·525	13·7

TABLE XXII.—BRITISH STANDARD SECTIONS—*Bulb Angles—continued.*

Correct Standard Profile.				Maximum Web Thickness.			
Size.	Thickness.		Section Modulus.	Size.	Thickness.		Section Modulus.
	Web.	Flange.			Web.	Flange.	
in.	in.	in.	sq. in. × in.	in.	in.	in.	sq. in. × in.
8·0 × 3·0	·425	·425	11·6	8·0 × 3·2	·625	·525	14·9
8·0 × 3·5	·450	·450	12·2	8·0 × 3·7	·650	·550	15·8
8·5 × 3·0	·450	·450	13·8	8·5 × 3·2	·650	·550	17·3
8·5 × 3·5	·475	·475	14·4	8·5 × 3·7	·675	·575	18·5
9·0 × 3·0	·475	·475	15·6	9·0 × 3·2	·675	·575	20·5
9·0 × 3·5	·475	·475	16·2	9·0 × 3·7	·675	·575	20·5
9·5 × 3·5	·500	·500	19·0	9·5 × 3·7	·700	·600	23·7
10·0 × 3·5	·525	·525	21·5	10·0 × 3·7	·725	·625	27·0
10·5 × 3·5	·550	·550	24·6	10·5 × 3·7	·750	·650	31·3
11·0 × 3·5	·550	·550	26·7	11·0 × 3·7	·750	·650	33·4
11·5 × 3·5	·600	·600	31·9	11·5 × 3·7	·800	·700	39·0
12·0 × 4·0	·600	·600	35·3	12·0 × 4·2	·800	·700	42·8

TABLE XXIII.—BRITISH STANDARD SECTIONS.

Lloyd's Rules, 1909–10.

Common Angles.

Size.	Minimum Thickness.		Maximum Thickness.	
	Thickness.	Section Modulus.	Thickness.	Section Modulus.
in.	in.	sq. in. × in.	in.	sq. in. × in.
3·0 × 2·0	·250	·65	·375	1·0
3·0 × 2·5	·250	·68	·375	1·0
3·0 × 3·0	·250	·69	·375	1·1
3·5 × 3·0	·250	·89	·375	1·4
3·5 × 3·5	·250	·92	·375	1·4
4·0 × 2·5	·250	1·2	·375	1·7
4·0 × 3·0	·250	1·2	·425	2·0
4·0 × 4·0	·300	1·5	·425	2·1
4·5 × 3·0	·300	1·8	·425	2·5
5·0 × 3·0	·300	2·2	·425	3·0
5·0 × 3·5	·375	2·7	·450	3·3
5·5 × 3·0	·375	3·2	·500	4·3
5·5 × 3·5	·375	3·3	·500	4·4

Note.—In all British Standard Sections (channels, bulbs, and angles), a strip of bulkhead plating of sectional area equal to that of the flange attached to the bulkhead has been included in calculating the section modulus. Face plates are assumed to have the same width and thickness as the flanges to which they are attached.

INDEX.

H

I

J

K

L

M

T

U

V

W

Y

Z

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