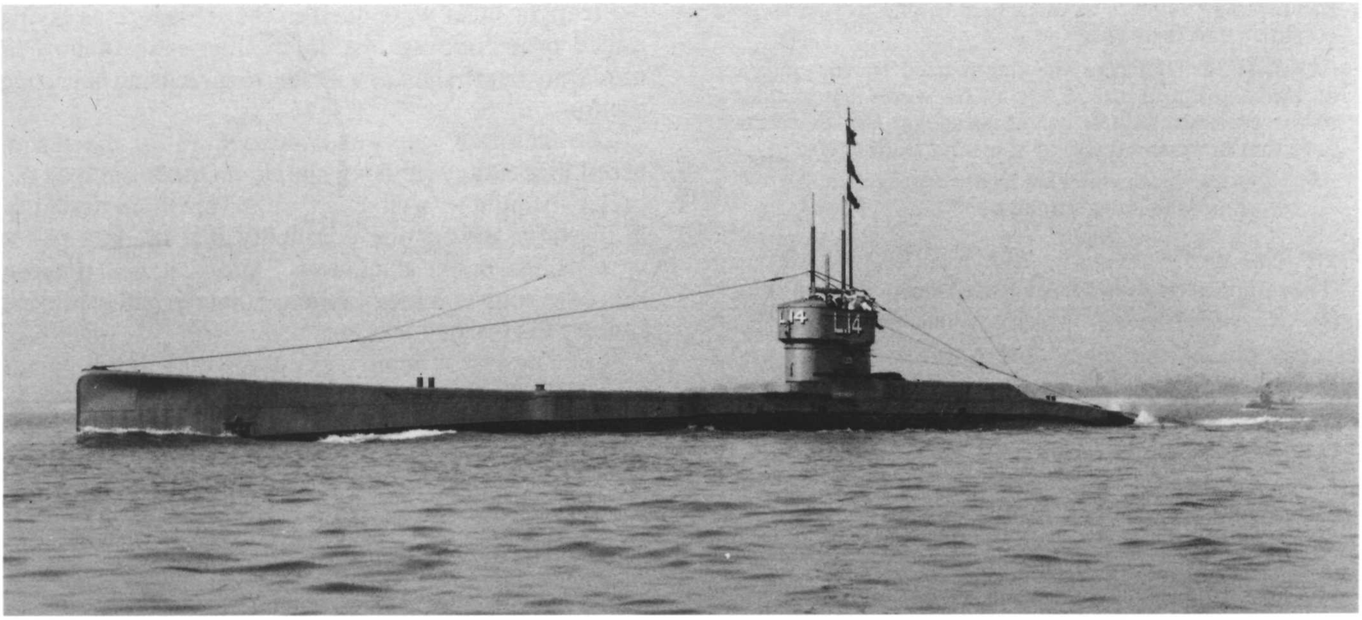


H. M. Submarine, L 14 *similar to* L 2 *which dove to a depth of 300 ft in WW I.* **Courtesy of the author.**

Submarine Pressure Hull Design & Diving Depths Between the Wars

by D. K. Brown RCNC

"The clearing up of the many doubtful points which still await complete theoretical and practical solution must be left to the future. These, however, are mostly of a minor description; the general problem of the submarine boat may be regarded as completely mastered."

—K Dietz
International Marine Engineering
October 1911

Introduction

Until the end of World War I, a submarine was invisible and could not be detected once submerged. There was little need to go much below periscope depth though boats were usually designed with a capability of 100–150 ft to keep them clear of the keel of the biggest ships and to permit them to rest on the bottom. During the twenties and thirties the introduction of asdic (sonar) and improved depth charge arrangements gave advantages to deeper diving boats which could use temperature layers to avoid distant detection and go below the reach of standard depth charge settings at that time.

Diving Depths

Diving depths were usually secret and even now are not easily found for World War II submarines. Even the meaning of "diving depth" can be unclear as these words can be used to represent at least three very different figures. Is is usually possible to distinguish the following depths.

OPERATIONAL DEPTH. This was the maximum depth which could be used in normal operation. There was a considerable margin of safety at this depth and, as will be seen, many boats exceeded this figure by a wide margin.

TEST DEPTH. A new submarine would always carry out a test deep dive, usually to the operational depth. Many commanding officers would go some 10% deeper to give confidence to their crews.

COLLAPSE DEPTH. The depth used by the designer at which the pressure of the water would cause failure of the pressure hull. It would seem that British practice (and that of most navies) by the early thirties was to take

Collapse depth = Twice Operational depth
(with pessimistic assumptions)

Modes of Failure

There are many ways in which a complicated structure, such as that of a submarine, can fail. Brief descriptions of the main failure modes follow; for a fuller description see Daniel and other papers in Reference 1.

OVERALL COLLAPSE by general instability. This is associated with frames of inadequate strength and would involve collapse of a whole compartment between bulkheads. This mode is very susceptible to out of circularity and hence strong frames are needed to preserve shape.

INTERFRAME BUCKLING is a condition in which the plating between frames buckles in a large number of nodes around the circumference. Static pressure did not normally lead to this failure mode in World War II boats but it could be provoked by depth charging. This mode becomes a greater problem when high yield strength steels are used. Riveted construction with butt straps etc. was less likely to develop a buckling failure node and more likely to fail by shearing of the rivets.

YIELDING OF PLATING between frames produces pleats around the circumference of the boat.

There was general recognition by designers of the different ways in which a hull could fail from fairly early days. However, not until well after World War II was new and advanced theory wedded to the computer to give useful results, due to work by Kendrick and others at the Naval Construction Research Establishment, Dunfermline.

Design practice was to overdesign the framing by judgment, so that overall collapse, the most intractable calculation, could be ignored. Inter frame failure could then be calculated, fairly accurately, using a very simple equation known as the "boiler formula."

$$\text{Stress} = \frac{\text{Pressure x Radius (hull)}}{\text{Thickness (pressure hull plating)}}$$

This formula, and the preceding discussion, relate to circular pressure hulls. Many submarines designed before World War II had oval sections forward to facilitate the arrangement of torpedo tubes and aft to suit a twin shaft propulsion system. The circular hull was also broken by hatches, and, in particular, by the torpedo loading hatch. Local structure was usually over strong in way of such discontinuities, though tests to destruction after the war suggested that the torpedo loading hatch area was often the point at which collapse started. In deep diving trials deflection of these oval sections was always measured and formed a guide to the safety of the boat. The paint on the webs of pressure hull frames would crack on 45° shear lines as another indicator of shear yield.

Stronger hulls were seen in the RN more as giving added protection against depth charges at shallow or moderate depth than as a means to increasing operating depths.

The factor of safety also accounted for the recognized inaccuracy in using simple methods such as the boiler formula as well as for minor errors in design or in building and for the possibility that the hull plates were rolled under thickness. Finally, it was realized that corrosion was inevitable and that the hull would get thinner as it rusted away.

When new, test depth was usually the same as operational depth. On some older boats reduced test depths were applied, though operational depth was not always reduced to the same extent.

TABLE 1

BRITISH DIVING DEPTHS

Class	*Operational Depth (ft)*		
L	150		
O, P, R	300		
"River"	200		
Porpoise	200		
Sunfish	200		
1940 S	300	Riveted boats	350 Welded boats
T	300	Riveted boats	350 Welded boats
U	200		
V	300		
A	500		

TABLE 2

SOME OVERSEAS DIVING DEPTHS

(mainly based on Reference 2)

Country	*Class*	*Date*	*Operational Depth (ft)*
France	*Requin*	1923	256
	L'Aurore	1935	328
Germany	1A	1936	472
	VII C	1940	472–590*
	IX	1938	492
	XXI	1944	492–656*
Italy	*Balilla*	1925	288
	Archimedes	1932	288
	Brin	1936	288
Japan	*I 153*		200
	I 9	1934	328
	RO 100	1941	246
USA	*Barracuda*	1920	200
	Salmon	1936	256
	Gato		300-(400 later boats)
USSR	L	1931	288
	K	1938	229

With the exception of Germany, other navies required operational depths very similar to those of the RN.

* Figures vary.

PRESSURE HULL DESIGN
TABLE 3—BRITISH HULL PARTICULARS
(Submarine Museum records)

Class	*Plating Thickness (ins)*	*Pressure Hull Diameter (ft-ins)*
L	0.5	15-7
O, P, R	0.875	16-1¾
"River"	0.625	18-5
Porpoise	0.625	17-2
Sunfish	0.375	14-11
1940 S	0.55	14-11
T	0.625	16-3
U	0.5	16-3
V	0.625	16-3
A	0.85	16-0

There is still some uncertainty in the understanding of design methods used in calculating the strength of pressure hulls before World War II. The basis was an informed comparison with previous successful designs and it would seem that much importance was attached to *L2*'s dive to 300 ft (twice the design depth) during the first world war. Throughout the twenties the design aim seems to have been to keep the stresses in plating and framing to about those of the earlier E class. With hindsight, this approach is seen as over cautious since the Collapse Depth of the main hull of *L2* was about 500 ft. The boiler formula was used as a basis of comparison and not as a design criticism.

By 1929, J. H. B. Chapman (later DNC), had marshalled a considerable body of theoretical and empirical methods and data relating to the design of stiffened cylinders. These formulae were then related to *L2*'s dive and used in the design of the *Sunfish* (and probably *Thames*). He studied the effect on overall hull weight of varying the design collapse depth. For *Odin*, a reduction from 500 ft to 300 ft gave a saving of 35 tons, a saving which was used by Bailey (lost in *Thetis*) in his design for *Thames* to allow more powerful machinery.

It seems that the boats which formed the bulk of the wartime fleet—1940 S, T and U—were designed in much the same way as just described, though with a more consistent factor of safety. Knowledge of von Mises' work reached the designers in the late thirties, just too late.

The first deep diving trials were carried out in an O class vessel circa 1927–8 with A. N. Harrison as the trials officer. Battens were rigged across the hull at various places to measure deflections. Despite additional support from pillars, the oval frames aft showed excessive deflections as did the oval gun access trunk. The trial was abandoned due to a leak at the forward torpedo loading hatch later found to be caused by defective welding. The after end and gun trunk were given additional stiffening and no further trouble was experienced (based on a letter to the author from A.N. Harrison, later DNC), In these riveted boats, the deep dive was a noisy affair as rivets would slip and pop.

Welding

The Admiralty had been a pioneer in the welding of ship structures with the publication in 1920 of the "Portsmouth Rules," the first UK standards for welding. Progress was slow. The depressed industry of the inter-war years lacked the money to develop weldable steels, electrodes, equipment and to retrain designers, managers and men.

During the thirties considerable progress was made in welding surface ships but only in 1940 did a weldable, high strength steel, suitable for submarines become available. This was 'S' quality, fairly similar in mechanical properties to the 'HTS' which it replaced, though with a yield stress of 18.5 instead of 17 tons/in^2.

TABLE 5—S QUALITY

Composition			*Strength tons/in^2*		*Elongation*
C	Mn	Si	*Yield*	*Ultimate*	%
0.21	0.8	0.3	18.5	30-34	18
Max	Min	Max	Min		

It was not easy to weld and great care was needed.

Weight was saved in the welded boats due to elimination of connecting flanges, butt straps, etc., and this enabled slightly thicker plating to be used.

Plating thickness (inches)

	Welded	*Riveted*
S	.55	.375
T	.75	.625
U/V	.5	.625

In turn, this led to an increase in diving depths of about 50 ft. The first T class with an all welded hull was tested to 400 ft on completion and *Amphion*, the first A, to 600 ft. The hulls were given a high pressure air test before launch to ensure that there were no leaks.

Diving Depths: Recorded, Design and Tests

The table below compares the operational depth and that calculated from the boiler formula with the maximum recorded (usually inadvertently) during the war. The table shows plating thickness and dimensions measured on the boat, which allowing for tolerances varied from the design figures given earlier. A test on an XT midget and on a large scale test section have been added.

[See following page for Table 5]

By the time these tests were carried out, more advanced design methods were in use and, by these methods, calculated collapse depth was within 5% of the actual depth at which failure occurred. The failure

TABLE 5

TESTS TO COLLAPSE

Name	*Material*	*Yield Strength Tons/in²*	*Diameter*	*Thickness (in)*	*"Formula" Depth (aft)*	*Actual (ft)*
Stoic	HTS	17	14-11	0.54	534	527-537
Supreme	S	18.5	14-11	0.625	700	647
Varne	S	18.5	16-7	0.614	614	576
Achates	S	18.5	16-0	0.875	860	877
"XT"		20.5	5-9	.227	702	565
Test Section	S	18.5	5-9	.25	698	563

usually started at a discontinuity, often the torpedo loading hatch, but there was clear evidence that predicted failure modes were close or had started. It is interesting to note that *Stubborn*'s wartime excursion to 540 ft was greater than the depth at which *Stoic* collapsed. The margin of safety was less than some commanding officers believed.

There was concern over the variable quality and thickness of S quality plates. This standard deviation on yield strength was 3.1 tons/in^2 with a mean less than the specified 18.5 tons/in^2. Plates were normally under the specified thickness, though usually within permitted tolerance. (Calculations ignored rolling tolerance as lying within the overall accuracy).

TABLE 4

COLLAPSE DEPTH, BOILER FORMULA, 18.5 tons/in² YIELD

Class	*Operational Depth*	*Formula Depth*	*Maximum Recorded*
L	150	520	152
O, P, R	300	880	400
Clyde	200	550	300 (distorted)
T	300	626	400
U	200	500	400
S	200	407	300
1940 S (riveted)	300	596	540
V	300	616	380
Porpoise	200	598	—
A	500	840	600 (*Amphion* Trial)

Of these only *Clyde* reported any damage—to the oval section aft. With the exception of *Stubborn* (540) none approached collapse depth.

The Cost of Deep Diving

Increased diving depths could be obtained by using a higher strength steel, by a smaller hull diameter or by thicker plating. For the Royal Navy the first two options were not available and increased plating thickness was the only possible route. For the *Oberon* the hull and equipment weight was 794 tons out of a total surface displacement of 1480 tons (54%). An increase in plating thickness would increase the hull weight almost pro rata. Either the submarine size would increase or the weight of something else must be reduced. The converse of this situation was seen in the *Thames* Class where, to make weight available for more powerful diesels, the thickness of the hull plating was reduced and, with it, the diving depth.

When the DNC, Sir Stanley Goodall, was asked in June 1941 to explain the surprising performance of the "500 ton" U Boats (VII C) he replied "I believe his battery is lighter, engines run to death and welding saves more than we do, reserve buoyancy is less and perhaps 500 tons is an under statement" (it was). Later, in October 1941, Sir Stanley was able to inspect the captured *U 570* and commented "Very interesting. A clean hull and welding good. But why such a thick pressure hull in association with comparatively flimsy frames . . . ." In fact, the U boats frames were quite adequate but small in comparison with the over large British structure.

Since there was no great demand from the Staff for deeper diving boats, design features that restricted diving were often adopted. British batteries were reliable and long lasting compared with German batteries but were much heavier, and to some extent, these remarks apply to the diesel engines and electric motors. The Vickers engines were heavy and fairly reliable while the Admiralty engines were about 20 tons heavier and much less reliable. There was, perhaps, an excessive use of non magnetic bronze plating around the magnetic compass which, because it was high up, added further weight in the form of ballast low down. Escape trunks, sealed in war, were another heavy feature, using weight which could have gone into the hull.

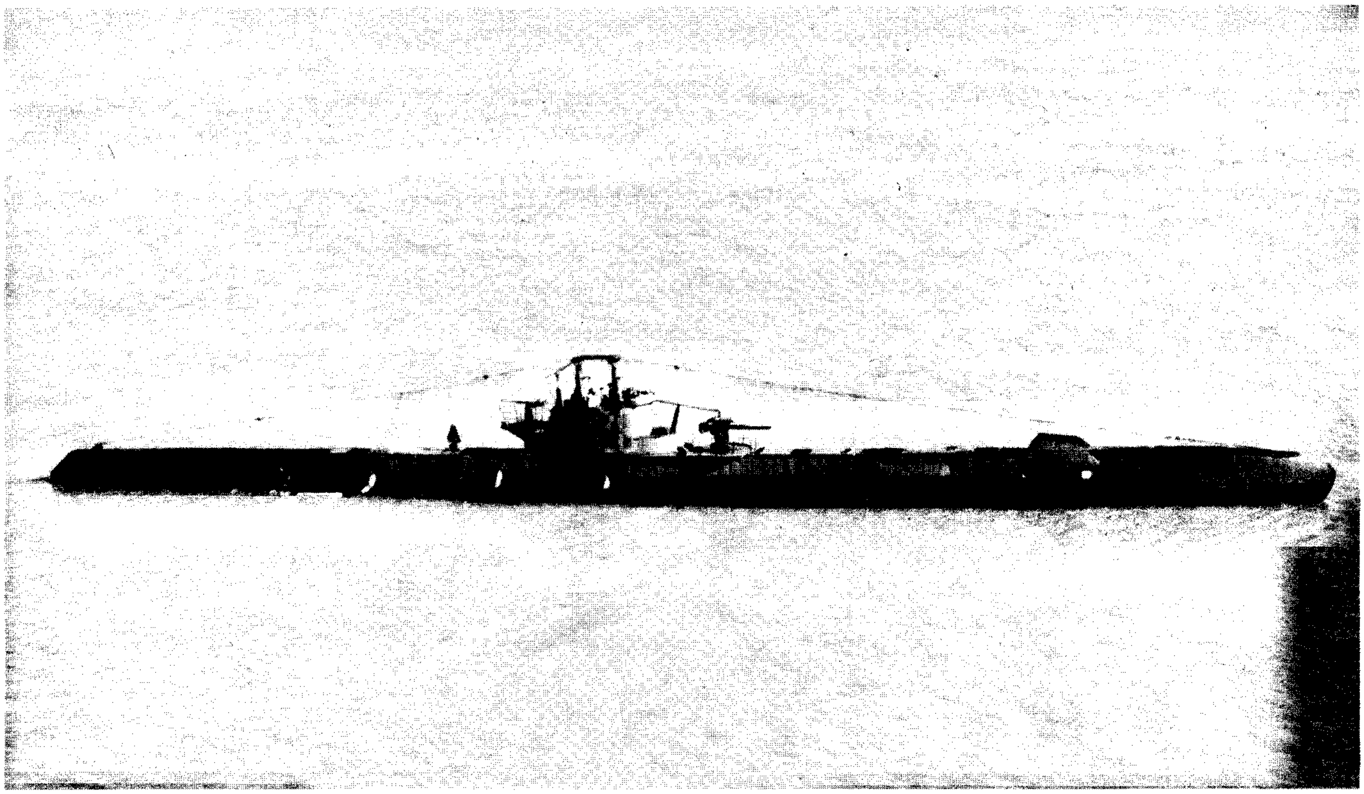
The requirement for six internal bow tubes enforced the use of oval sections forward on small submarines until the much deeper diving A class had to accept a four tube internal battery.

The Value of Deep Diving

In World War II deep diving could help the submarine in the following ways:

Rest on the bottom in deeper waters
Reduced risk of asdic detection
Reduced risk of a lethal depth charge attack

Of these, the first is self evident. World War II sonars were not very effective and were even further degraded



H.M.S. Sirdar, ***a 1940 riveted S class sub.*** **Photo courtesy of the author.**

by the effects of layers of water of different density. The deeper diving boat had more chance to hide under a layer than one limited to shallow depths.

The standard depth charge used by most navies at the outbreak of war would rupture a pressure hull at a distance of about 30 ft. Since the precise position and depth of a submarine was not known, patterns of 5, 10 or 14 charges were dropped to straddle the likely positions of the target. Non lethal hits could still cause small leaks and damage equipment making the submarine more susceptible to a further attack or even force it to the surface.

Since the asdic beam looked ahead and was relatively narrow, contact was lost with the submarine well before the bow of the escort was above the target. Depth charges were dropped from the stern and fell at about 10 ft/sec (16½ for the later, heavy charge). This lengthy dead time gave the submarine moving at some 2–3 knots (3–5 ft/sec) a chance to evade the attack. Furthermore, depth could not be measured by Asdic until late in the war and rapid depth changing was an effective way of avoiding a slow falling charge. The resistance of the hull increased a little with thicker plating.

To sum up, the Staff failed to recognize the value of deep diving submarines during the thirties but even had they asked for more depth, the means for significant improvement were not available in the UK. Strong steels suitable for welding were not available and both batteries and engines were heavy, reducing the weight available for hull strength.

The assistance of R. J. Daniel, A. N. Harrison, J. H. B. Chapman, Cdr Compton Hall, the staff of the Submarine Museum and of several of my colleagues is acknowledged.

References:

1. RINA Symposium on Submarine Design, 1983, particularly paper by R. J. Daniel "Considerations Influencing Submarine Design."
2. E. Bagnasco, *Submarines of WWII*, Arms & Armour Press 1977.
3. A. N. Harrison. *The Development of HM Submarines from Holland to Porpoise*. BR3043 HMSO.

NOTE: Copies of this paper, with a lengthy Annex summarizing J. H. B. Chapman's work in 1929 and with copies of the letters referred to from Daniel, Chapman & Harrison have been placed in the following libraries.

Submarine Museum
National Maritime Museum
RINA
Naval Library
Science Museum
DCW Library

Annex 1

Pressure Hull Design—1929

The following notes are abstracted from the work books of J. H. B. Chapman (Ref 294/7) who later became Director of Naval Construction. The entries start on 13 February 1929.

"The various points to be considered are:

1. Crushing pressure
2. Hoop stress
3. Panel stress and frame spacing"

Chapman then lists the formulae which were then available to him.

Definitions

E Modules of elasticity
t Thickness of plating
d Diameter of hull
σ Poisson's ratio (approx ¼)
l Length (of compartment)
s Frame spacing

1. a. from Marley's "Strength of Materials" (p. 343 in 8th Edition 1934).

$$\text{Crushing pressure} = \frac{8E}{3}\left(\frac{t}{d}\right)^3$$

$$\text{or, more exactly } \frac{m^2}{m^2-1}\cdot 2E\left(\frac{t}{d}\right)^3$$

$$= c\left(\frac{t}{d}\right)^3 \qquad m = \frac{1}{\sigma}$$

where $c = 50 \times 10^6$ for steel tubes
25×10^6 for brass

(b) from 'Engineering' of 4·1·07

$$= \frac{2E}{(1-\sigma^2)}\ \frac{t^3}{d^3}$$

(c) Carman and Carr

$$= \frac{t^2}{d^2} \times 10^6 \text{ for long tubes}$$

$$= 1.25 \times \frac{t^2}{d^2} \text{ (lap welded tubes)}$$

(d) Stewart $= 1000\left(1-\sqrt{1-1600\, t^2/d^2}\right)$

(e) Fairbairn $= 9.675 \times 10^6\ t^2/ld$ (empirical result)

(f) Mackrow 'Handbook of Naval Architecture' $= \dfrac{2150\, t^2}{l\ d/2}$ (for short tubes at least 3/8" thick)

(g) Prescott's "Applied Elasticity" p554

$$= t E'\left[\frac{1-\sigma^2}{m^4(m^2-1)}\ \frac{\pi^4 r^3}{l^4} + \frac{1}{3}(m^2-1)\frac{t^2}{4r^3}\right]$$

where $E' = E/(1-\sigma^2)$

Known as the Southwell formula (Morley p 344)
& m is number of nodes from

$$\frac{2}{m^6(m^2-1)} + \frac{1}{m^4(m^2-1)} = \frac{1}{3(1-\sigma^2)}\cdot\frac{t^2 l^4}{4\pi r^6}$$

2. Hoop stress (Boiler formula)

$$f = \frac{bd}{2t}$$

3. Panel stress & frame spacing

$$\text{Morley } \frac{S}{d} = 1.73\frac{E}{f}\sqrt{\frac{t^3}{d^3}}$$

Morley p 347

b
a

$$\text{Panel stress } f_{ps} = \frac{n^2}{2n^2+6.5}\ \frac{pS^2}{t^2} \qquad n = \frac{a}{b}$$

p = unit load

(from Law p369 - Engineering Vol CXIII 7-12 /21

Chapman then applied some of these formulae to existing submarines.

Class	"L"	*XI*	*Odin*
Hull plating t(ins)	0.5	1.0	0.875
Stiffener	6x3½x3x15Z	7x3½x3½x20.2Z	6x3½x3x15Z
Frame Spacing S"	21	18	21
Diameter d(ft)	15.75	19.75	16.75
Sectional area for frame space in²	14.95	23.94	22.8
y of plating + frame (inches)	5.25	6.5	5.74
I/y	11.4	18.0	12.1

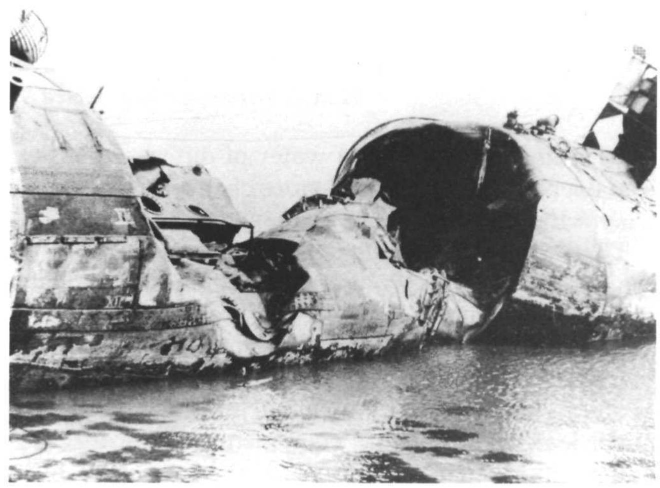
"Propose now to examine the strength assuming the frames to remain rigid and the plating to be rigidly held at the frames,

use formula a,
ie collapsing pressure $p = 50 \times 10^6 \dfrac{(t)^3}{(d)}$

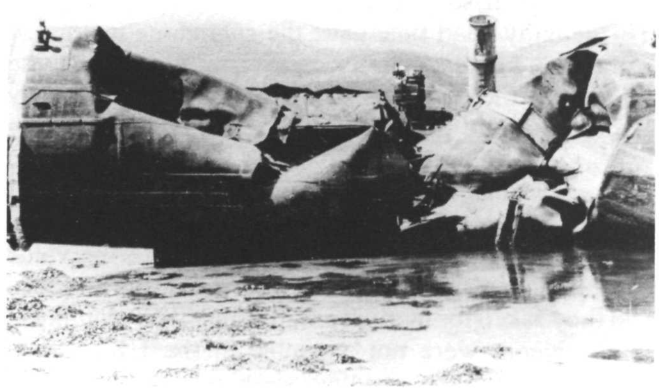
According to Morley—"Strength of Materials"—tubes below a critical length L—dependent upon the diameter and the thickness—offer a greater resistance to collapse than long tube.

The relation may be written, for a tube of length l' less than L, the crushing pressure.

$p_1 = \dfrac{pL}{l}$, p being the crushing pressure for a length L.



HMS Stoic. ***The remains after collapse at about 530 ft.*** **Photo courtesy of the author.**



A welded S class which collapsed at 647 feet. **Photo courtesy of the author.**

From experiments $L = 1.73 \sqrt{d^3t}$"

(*NOTE:* Von Mises *assumes* simple support)
(1 is taken as frame spacing)

Class	*L*	*X1*	*Odin*
Crushing pressure p′ (ft of water)	640	2940	2300
Design diving depth	200	500	500
Factor of Safety	3.2	5.88	4.6

Hoop Stress at design depths (as above)

Class	*Hoop Stress*
L	5.2
X1	8.9
Odin	9.25

Panel Stress

Chapman then calculated panel stress from

$$f = \frac{n^2}{2n^2 + 6.5} \quad \frac{p.S^2}{t^2}$$

but noted "These figures are of little practical value, the formulae really applying to flat plates between rigid parallel supports of infinite length. They do, however, offer a comparison."

Class	*Stress (tons/in²)*
L	35
X1	16.2
Odin	28.8

Frame Spacing

$$\text{Morley—}\frac{S}{d} = 1.73 \frac{E}{f}\left(\sqrt{\frac{t3)}{d^3)}}\right)$$

with $E = 30 \times 10^6$ lbs/in²
with $f = 34000$ lbs/in²—elastic limit

"and that practical shows that the value of S should be about halved."

Class	*Theoretical Spacing*	*Actual Spacing*	*Facts of Safety*
L	39.6	21	1.9
X1	100	18	5.5
Odin	88.5	21	4.2

"*Conclusions*

These calculations indicate that, provided the assumptions that the frame bar remains rigid is correct, then *Odin* and X1 are better able to withstand a 500 ft depth than is an L class C/M able to withstand a depth of 200 ft.

L2 has been submerged to 300 ft and but for minor defects withstood the pressure."

There is then a revised approach.

"Crushing pressure $p = \frac{4EI}{Sr^3} = \frac{32EI}{Sd^3}$

This result is arrived at theoretically but not by a rigid method. For plates (see formula a1) the constant is reduced by ⅝ to form an empirical formula

ie $p = \frac{5}{8} \times 32 \times 30 \times 10^6 \frac{I}{Sd^3}$

$= 600 \times 10^6 \frac{I}{Sd^3}$"

Class	*Crushing pressure lbs/in²*	*Factor of Safety*
L	590	2.95
X1	660	1.32
Odin	550	1.1

"This neglects end conditions at bulkheads and the effect of flats, external structure etc.

Chapman then tries a final approach

a. As given earlier

$$p\,S = p'\,L \qquad \text{(page 5)}$$

$$p \propto \frac{p'\,L}{S}$$

$$p' = 50 \times 10^6 \left(\frac{t}{d}\right)^3$$

$$\propto = t^3/d^3$$

$$\text{and} \quad L = 1.73 \sqrt{d^3/t}$$

$$\propto \sqrt{d^3/t}$$

$$\therefore p \propto \frac{t^3/d^3 \sqrt{d^3/t}}{S}$$

$$\propto \frac{t^{5/2}}{S\,d^{3/2}}$$

Chapman assumes 2 node buckling.

b. Assume frame and one frame space of plating to form a separate circular ring.

Morley and as amplified above give

$$p = 600 \times 10^6 \frac{I}{Sd^3}$$

c. Assuming plating between frames is flat

$$f = \frac{pS^2}{2t^2}$$

Max pressure $\propto t^2/S^1$

"The formula in a. gives, I think, an exaggerated importance to the thickness of the plating. The limiting depths assessed by this method are higher than obtained from formula b. This formula indicates that the frame bar is the most important factor.

Formula c. is similar to formula a.

I suggest that in using these formulae the actual figures are not of much use but that they are valuable for purposes of comparison.

The method proposed is:

To determine scantlings = use $p \propto \frac{I}{Sd^3}$

To determine the effect of small change of thickness of plating

use $p \propto \frac{t^{5/2}}{Sd^{3/2}}$

(or $P \propto t^2/S^2$)

The actual figures obtained are

	L	X1	*Odin*
Max depth assuming frame remains rigid	640	2940	2300
Max depth assuming frame and associated plating form a separate ring	590	660	550"

Effect of reducing diving depth

Odin from 500 ft to 300 ft

S, td = constant

p reduced from 225 to 135 lbs/in^2

$\therefore$ t = <u>¾ in</u>, say

I = <u>41.8</u> in^4

"Therefore, for a 300 ft diving depth boat of similar dimensions to the *Odin* we can reduce the plating to ¾ in and the framing to 5 x 3 x 3 x 14.17Z instead of ⅞ in and 6 x 3½ x 3 x 15.16

Wt of pressure hull in *Odin* = 240 Tons

Wt of framing in *Odin* = 240 Tons

Saving = 34.3 + 2.6 = 36.94T

= 35 Tons, say"

b. Increase diving depth of L to 300 ft leads to ⅝ in plating and 6 x 15 lbs Z

Annex 2

Diving Depth and Pressure Hull Weight 1929

Later in 1929, Chapman used the formula of Annex 1 in the pressure hull design of a new small submarine.

The original requirements were:

"Geneva" displacement	600 tons
Speed	17 kts
Torpedo tubes	4—21 inch
Diving depths	200 ft
Endurance	3000 @9 + allowance for charging equivalent to 1 week or 4500 @ 9
W/T range	600 miles
Gun	1=3 inch, HA, if possible

Later these were modified to:

Endurance "S/M to operate 1200 miles from base, passage speed to be 9 knots. Allow 6 days each way + 8 day patrol ie 20 days = 4500 @ 9

Torpedoes 6 tubes + 6 reloads bow

2 tubes stern, in preference to speed

Speed Minimum 14 kts

Displacement 600–760

3 in HA + 2 mg

300 ft diving depth

Submerged endurance as L

Diving Depth

Plating 20 lbs (½ inch) as E, L and L50 classes

Diameter of pressure hull 15 ft (as E)

Frame spacing 21 in as E & L

Frame bars 5 x 3 x 3 x14.17Z

Then if frame remains rigid

Crushing pressure = 300 lbs/in^2 = 667 ft

Plating & frame as ring leads to 420 ft

Hoop stress at 300 ft = 7.8 $tons/in^2$

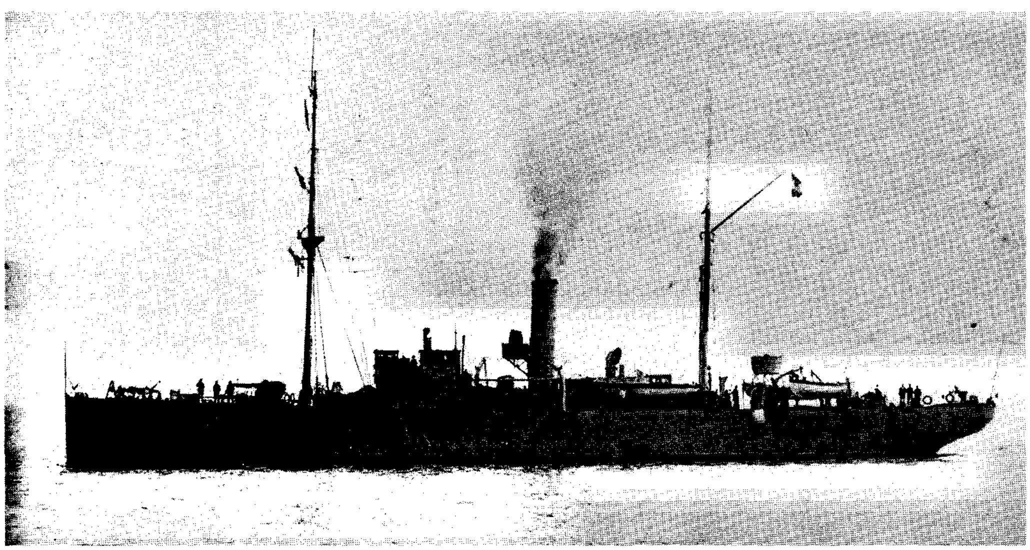
"Summary

1. Assuming frame bars rigid, crushing depth = 667 ft
2. Assuming frame bar and plating as continuous unsupported ring, crushing depth = 420 ft (Increased to 650 ft if 6 in Z are used)
3. Hoop stress at 300 ft = 7.8 ton/in^2

The effect of varying frame size, spacing and plating thickness was then considered. Weight savings relative to the E class were calculated

E class Plating, butt struts	62.6 tons
Frames	15.8 tons

Design	E	I	II	III	IV	V	VI
Plating t lbs	20	17½	17½	17½	15	15	15
Frames	5″Z	6″Z	5″Z	5″Z	6″Z	5″Z	5″Z
Frame spacing	21	21	21	18	21	21	18
Crushing depth* I	667	475	475	557	326	326	380
Crushing depth II	420	635	407	458	605	392	440
Hoop stress at 300 ft	7.8	8.4	8.6	8.1	9.25	9.5	8.96
Estimated weight reduction of E, tons	—	7.25	8.5	4	15.75	17	12.5



Captain's Scrapbook

***The German Surveying Ship* Meteor *as completed, photographed by K. Steinhauser of Wilhelmshaven. This ship was begun before World War I as a gunboat but completed postwar as an auxiliary. She commenced her first overseas cruise, to the Canary Islands, on 20 January 1925.* Photograph from C. C. Wright Collection.**