昭和四年十一月十五日　發行
毎月一回十五日　發行

W.E.C. NUMBER

PART II.

昭和四年十一月刊行

造船協會雜纂

第九十二號

造船協會
The Society of Naval Architects.

(非賣品)

造船協會雜纂

昭和四年十一月刊行　第九十二號　内容目次

W.E.C. NUMBER

PART II.

Contents.

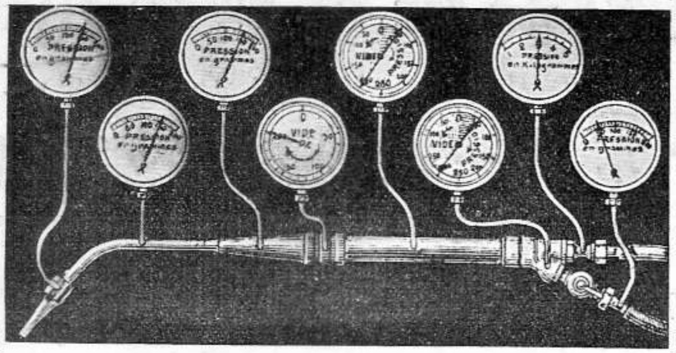
雜　　錄

時　　報

液体空氣會社の鎔接器具

弊社は酸素アセチレーヌ鎔接及切斷法の始めて出現せる以來即ち二十餘年來**吹管及調整器**の製作並に**鎔接法の研究**に專門的に從事せるものにして各種の鎔接及切斷作業を行ふに適する機能最も確實なる吹管及調整器を斯界に供給しつゝあるは一般の夙に認むる所であります。此の事實は弊社の製品**ピカール**式吹管及**ピロコツプ**式吹管が世界到る處に於てピカール又はピロコツプなる呼稱の下に愛用せられつゝある事實に徵するも明かであります。

吹管各部の壓力檢査裝置



上掲の寫眞は弊社の製造場に於ける吹管各部の瓦斯膨力の**檢査裝置**であるが機能正確にして瓦斯放出量の比の正しい吹管を得るには是非共此の如き裝置を必要とするのであります。而して此の如き精密なる檢査設備は獨り弊工場が有するのみで他の如何なる製作所にも有しないものであります。

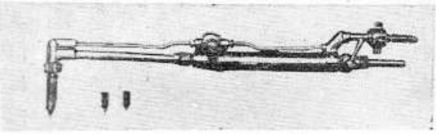
弊社の**吹管**の機能が理想的に確實で而も作業の成績と經濟とに最も重大なる關係を有する酸素アセチレーヌ**兩瓦斯の比**が極めて學理的數字に接近して居り斯界に嘖々たる好評を博しつゝあるは此の如き綿密なる檢査裝置に依り確實なる性能を與へてあるからであります。

此の點は特に需要家の注意を要する點であると同時に弊社の吹管と他の外形のみを似せた模造品との根本的差異をなす主要點であります。

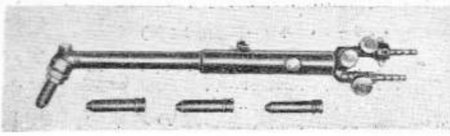
下掲の寫眞は弊社最近の製造に係る鎔接用及び切斷用吹管で**名實共に斯界最良**のものであります。

型錄進呈

切斷用

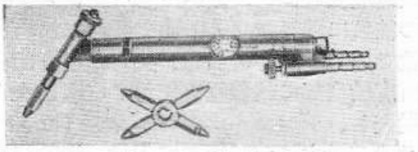


ピロコツプ式小型吹管

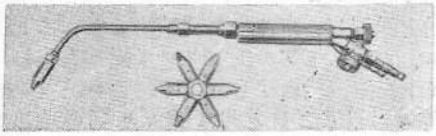


ピロコツプ式AS第一號

鎔接用



ピカール式第〇號



ピカール式AS第一號

本社 神戸市明石町三八
電話三宮（一八七九、三七六三）
市外專一三

液体空氣會社

東京支社 東京丸ノ內三菱二十一號館
電話丸ノ內(23)一一六六・七六二七

支社

神戸支社	神戸市明石町三八 電話宮八七九
長崎支社	長崎市松ケ枝町四二 電話一七七八
京城支社	京城府龍山漢江通一一 電話龍山九八五
名古屋支社	名古屋市中區御器所町江越二二 電話南一一・六二
小倉支社	小倉市大阪町五丁目七三 電話一三六四

出張所

大阪出張所	大阪市港區尻無川北通一ノ一 電話西二七九五
京都出張所	京都市大宮通四ツ塚下ル比永城町四 電話下六三四
新潟出張所	新潟市本町通九番丁二三三七 電話二五〇六
佐世保出張所	佐世保市三浦町一六二〇
大連出張所	大連市越後町一〇 電話六九二六
直江津出張所	新潟縣直江津町 信越窒素肥料株式會社內
武生出張所	福井縣武生町 大同肥料株式會社內
水俣出張所	熊本縣葦北郡水俣町字馬刀潟

工場

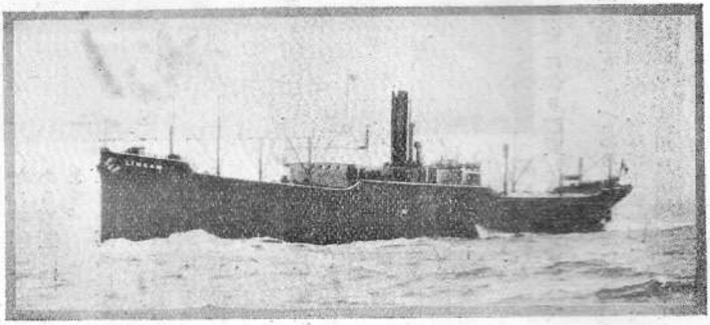
兵庫工場	神戸市高松町四七 電話兵庫七〇七
大阪工場	大阪市此花區櫻島町二七 電話土佐堀七四六
神奈川工場	橫濱市神奈川區子安町海岸通三〇六二 電話本局三二六六
京城工場	京城府外纛津 電話龍山一〇九二
名古屋工場	名古屋市中區御器所町江越二二 電話南一一六
小倉工場	小倉市下到津 電話四三五
武生工場	福井縣武生町北府三六ノ九 電話五及ビ六五
水俣工場	熊本縣水俣町字馬刀潟 電話一一六
直江津工場	新潟縣直江津町 電話直江津三五五

微粉炭燃燒船の出現

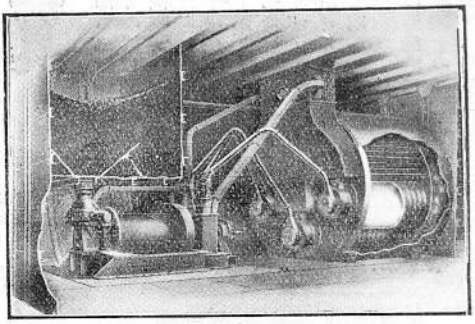
造船界に一大革命を劃す

ケネデー微粉炭燃燒裝置

微粉炭燃燒裝置
の完成と
優秀
汽船時代
再現の勢



米國リンガン汽船會社所有微粉炭燃燒船
「リンガン」號（4.700噸）



ケネデー舶用微粉炭燃燒裝置
（スコツチ・マリーン・ボキラーに應用せる例）

ケネデー微粉炭燃燒裝置は最も高度の粉末が得られ且最も機械を損傷しないケネデー獨特のチューブ・ミル・システムに汽罐の廢熱を利用した斬新な機構を具へ舶用汽罐用として幾多の特長を有す。

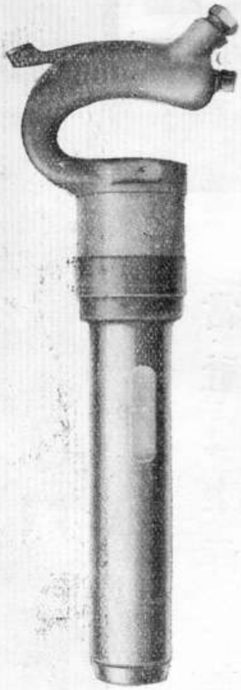
米國ケネデー・バンザン會社

昨春米國船舶院が燃油船マーサー號（9.500噸）を微粉炭燃燒裝置に改めニユーヨーク、ロツテルダム間の往航に燃料費 4000 弗を節約し、機關一部の改造で速力1浬を增加し、英國綠星汽船會社は近く所有船全部に該裝置を使用する事に決定した旨を發表し、デンマルクのフエンデイラ、トム、ハイダラの三船主は今回擧つて本式の造船に着手し、米國ミシシツピー遊覽汽船會社は進んで微粉炭燃燒船を採用し、更に米國船舶院はウエスト、アバウム、ウエストアルトゼツクス號を改裝し、又ホワイト炭礦汽船會社も數隻に對し本裝置を應用した等々の報導は日と共に續々と到着致します。而も微粉炭燃燒船は單に船舶デキーゼル化の過渡期の對策として出現したのでは無く實に其の造船費の低廉と安價なる粗悪炭利用の可能と速力積載能力等の增加を以て出現した時代の產物であります。詳細說明書御入用の方へ進呈す。

株式會社東洋工業社

東京市・丸の內・丸ビル
大阪・小倉・紐育・倫敦・漢堡

THE CLECO RIVETER

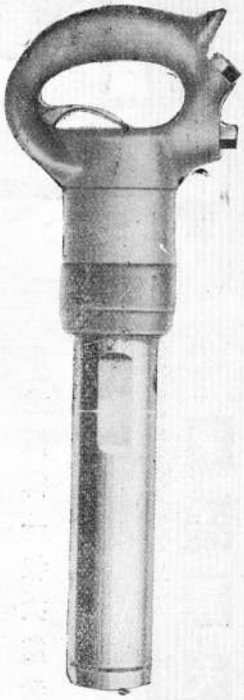


クレコ・リベッター各種

エアーツールの最も重要なる點は空氣の使用量如何？力及**スピード**は如何？取扱ひ容易なるや否やにあります。**クレコリベッター**には**エアーポツケツト**等の特種装置があり品質最も優秀、能率最も卓越せるものであります。

リベツター、**チツパー**、**クラインダー**、**エアードリル**、**ランマー**、等各種あり。

—（詳細御照會を乞ふ）



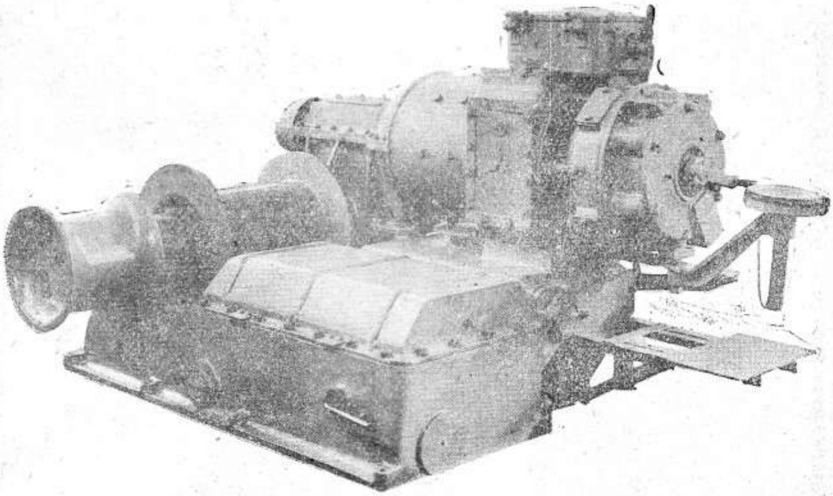
SIZE No	DIA PISTON (Inches)	STROKE (Inches)	WEIGHT (Lbs.)	BLOWS Per Min.	HOT RIVETS (Inches)	LENGTH Inches
40	1$\frac{1}{16}$	4	18	1380	⅝	15
50	1$\frac{1}{16}$	5	18¾	1300	¾	16
60	1$\frac{1}{16}$	6	19½	1200	1⅛	17
80	1$\frac{1}{16}$	8	21	1080	1⅛	19
90	1$\frac{1}{16}$	9	21¾	900	1⅜	20

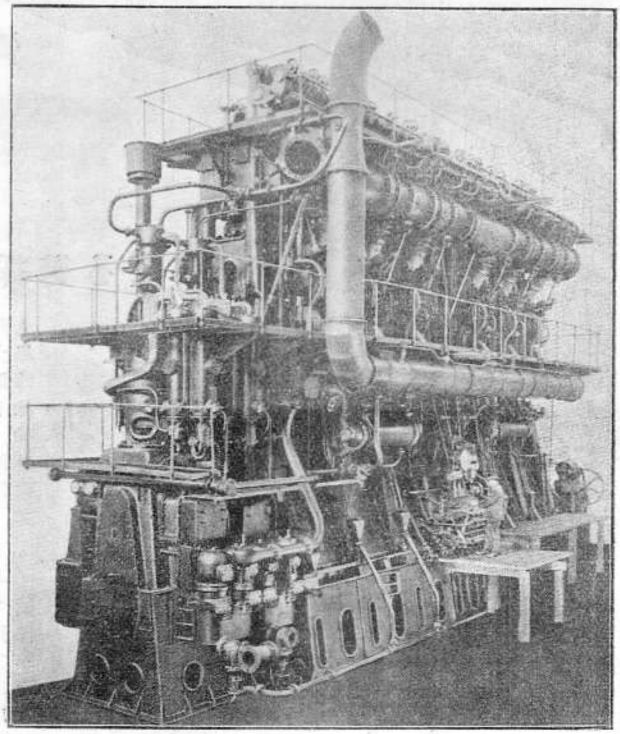
THE CLEVELAND PNEUMATIC TOOL Co.

東洋總代理店

株式會社 アンドリゥス商會

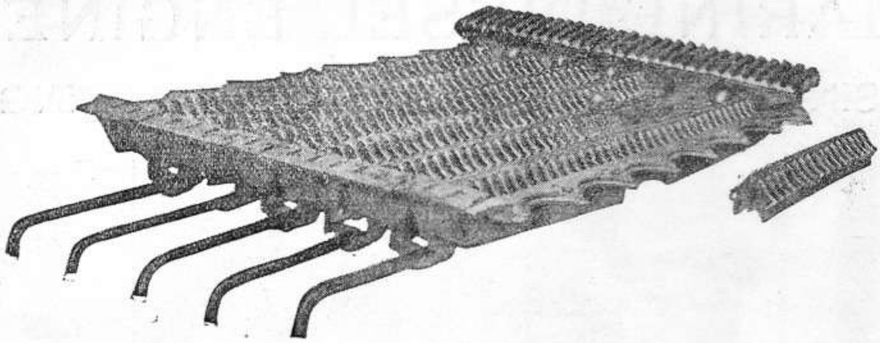
創立 明治二十七年
本店 東京市芝區芝公園五號地二ノ五
支店 大阪市西區江戸堀南通三ノ十八
名古屋・小倉・札幌・京城・臺北・大連・紐育





フアイヤーバー

特許 御法川二九式燃燒機



主なる御採用先

船舶

海軍省
鐵道省
農林省
大連汽船株式會社
國際汽船株式會社
朝鮮郵船株式會社
三井物産船舶部
共同漁業株式會社
樺太汽船株式會社
帝國汽船株式會社
日清汽船株式會社
攝陽商船株式會社
貝島商業株式會社
阿波共同汽船株式會社
政記公司
東洋捕鯨株式會社
九州汽船株式會社
樺太漁業株式會社
博多トロール株式會社
林汽船株式會社
林兼商店
日本トロール株式會社
山田漁業株式會社
其他數十ヶ所

造船所

橫須賀海軍工廠
浦賀船渠株式會社
三菱造船所
淺野造船所
神戶製鋼播磨造船所
大阪鐵工所櫻島工場
同　築港工場
石川島造船所

炭費節約

一割

東京市小石川區初音町

製造元 **御法川工場**

電話小石川二四一番

一手販賣 **三井物産株式會社機械部**

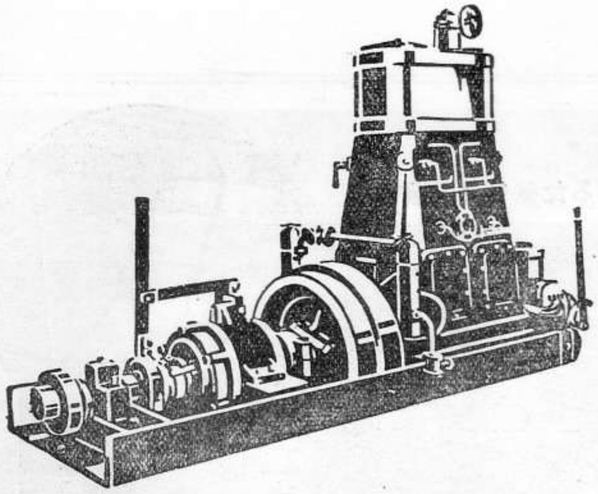
ユンケル式ディーゼル機關

日本及満洲總代理店

合資會社 **泰明商會**

東京市京橋區銀座二丁目十四番地
神戸市播磨町五十四番地
名古屋市中區大池町四丁目十番地
京城府黄金町一丁目百八十一番地

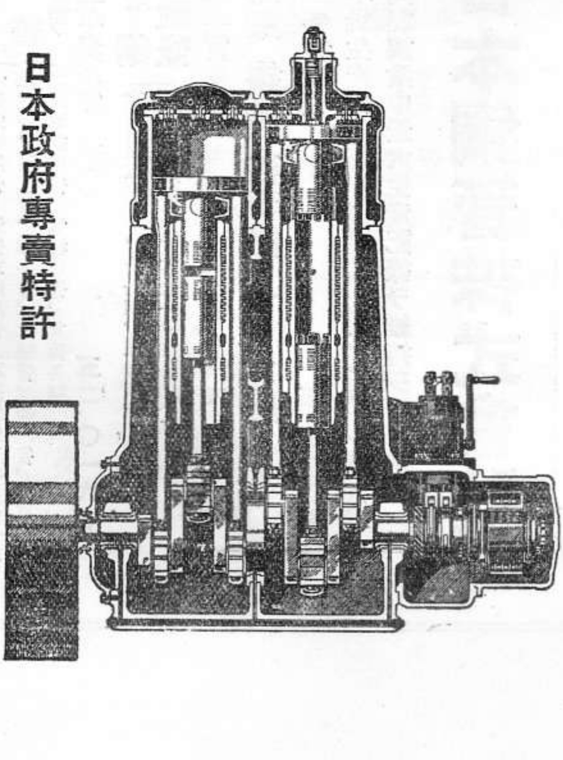
ダブルピストン型
無空氣噴油式
無弁無氣筩蓋式



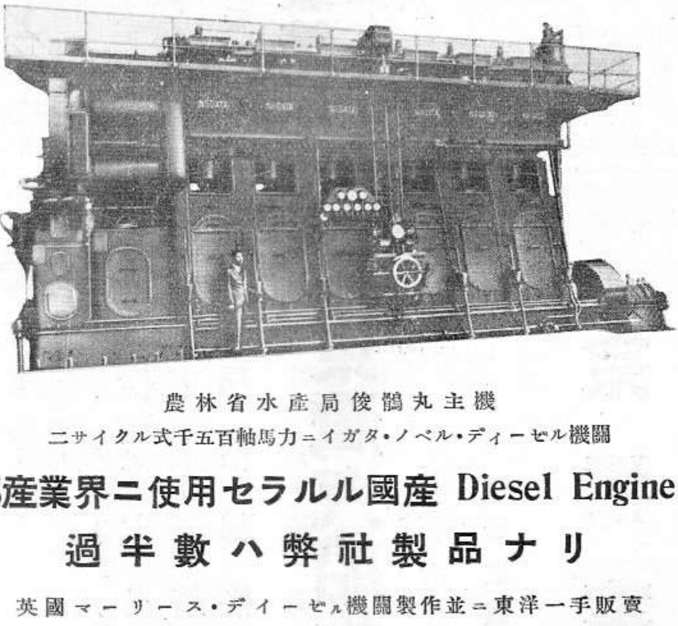
優秀なる舶用機關
としてユンケル式ダブル
ピストン型ディーゼル機
關を推賞す
卓越せる原理により
燃油消費量僅少
平衡完全にして振動絶無
容積重量輕少
構造簡單にして故障の憂
無く取扱極めて容易
八馬力より六〇〇馬力迄

型錄贈呈

日本政府專賣特許



飛行機上より見たる日本鋼管會社工場と其附近



株式會社

大阪鐵工所

海軍省指定工場

鐵道省指定工場

大阪市此花區櫻島南之町

電話土佐堀長三〇〇〇（9）五七〇〇（5）

造船　鐵橋
造機　構梁
車輛　鐵構

船渠

工場	番號	總長	Sill上ノ長サ	渠口上部ノ幅	渠口下部ノ幅	盤木上ニ於ケル滿潮面ノ深サ
櫻島	1	692′—0″	67[illegible]′—0″	75′—6″	71′—6″	21′—0″
築港	2	438′—0″	420′—0″	57′—0″	57′—0″	20′—3″
因島	3	346′—0″	338′—0″	46′—6″	43′—0″	17′—0″
同	4	459′—0″	446′—0″	56′—0″	55′—0″	20′—6″
同	5	154′—0″	148′—0″	32′—0″	28′—6″	18′—6″
同	6	421′—0″	414′—0″	57′—0″	51′—0″	20′—6″
同	7	300′—0″	291′—0″	42′—0″	38′—0″	16′—6″
彥島	8	223′—0″	218′—0″	37′—0″	29′—0″	15′—0″
同	9	294′—0″	287′—0″	55′—0″	50′—0″	20′—6″
笠戶島	10	484′—8″	481′—2″	74′—9″	70′—7″	21′—7″
同	11	319′—1″	310′—2″	54′—0″	50′—9″	17′—7″

神戶事務所

神戶市播磨町十七
電話三ノ宮長八七（2）一七七五（市外）一七（專用）

東京事務所

東京市丸ノ內仲通十五番館
電話丸ノ內長（23）八六六（2）

櫻島本社工場

大阪市此花區櫻島南之町
電話土佐堀長三〇〇〇（9）五七〇〇（5）

築港工場

大阪市港區船町
電話櫻川四長三三（2）九〇四 三〇六一

因島工場

廣島縣御調郡土生町
電話土生長一（7）

彥島工場

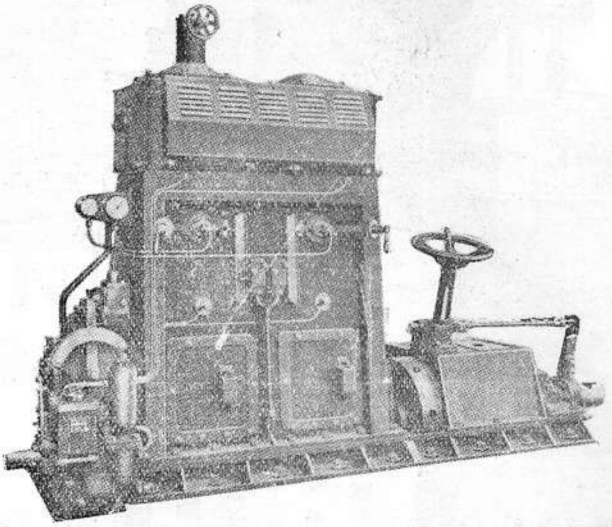
山口縣豐浦郡彥島町字江ノ浦
電話江ノ浦長二二四四

笠戶島工場

山口縣都濃郡末武南村大字笠戶島
電話下松長四七

NIPPATSU

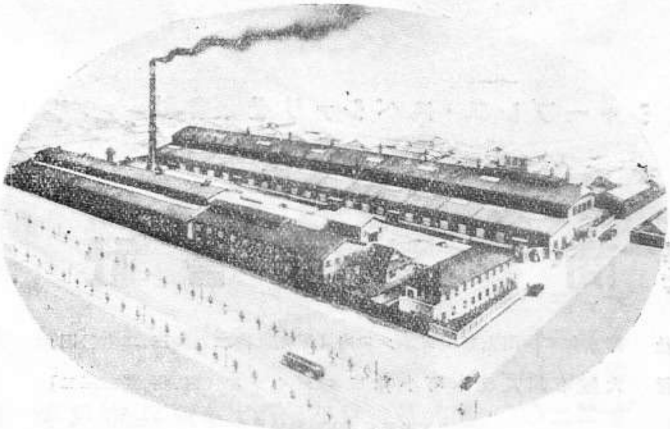
DOUBLE PISTON DIESEL ENGINE



内燃機界ノ新異彩

本機關ノ特長

(イ)換氣作用完全ナルコト(従來ノ二サイクルノ缺點ハ絶對的ニ除去セラル)
(ロ)熱効率尤モ優秀ナルコト(熱ノ漏洩面積ヲ極限シ得ルガタメナリ)
(ハ)回轉圓滑ナルコト(本式ノ特長ニシテ振動絶無)
(ニ)無空氣噴油ノ完全(本式ノ特長ニシテ燃料消費極少ナリ)
(ホ)機械油ノ經濟 従來ノ二サイクルノ缺點ハ容易ニ解決セラル)
(ヘ)シリンダーカバー及バルヴ不用本構造ノ本領ナリ)
(ト)機關据付面積及重量ノ小ナル事(本構造ノ本領ナリ)



農林省
遞信省
認定工場

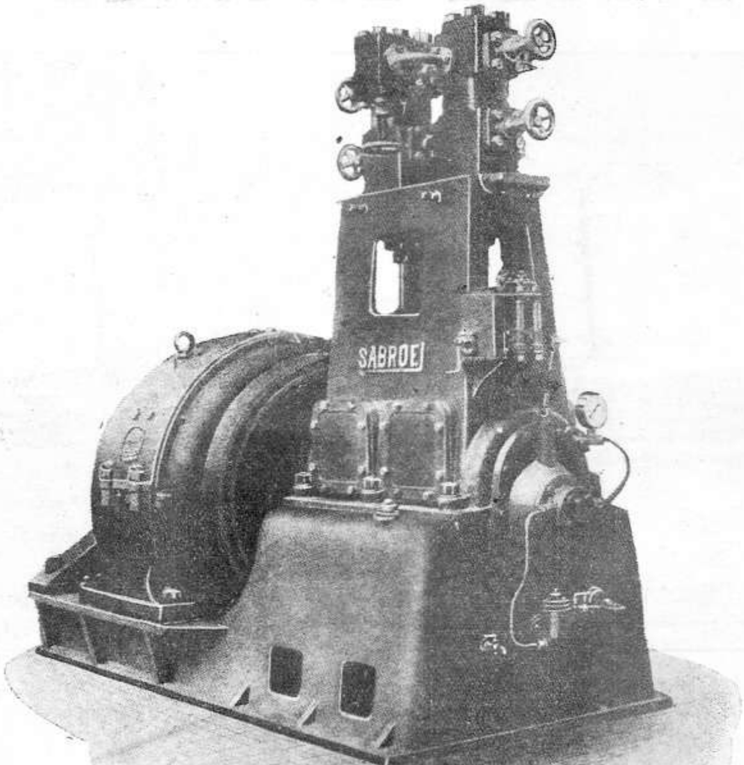
神戸日發

神戸市金平町二丁目三十五

日本發動機株式會社

發信電略(ニホ)又ハ(ニ)
受信電略(カウベシニツパツ)
振替口座大阪五六四九八番


LINOLEUM


SABROE

會告

(一) 播磨地方委員更迭

(新) 横尾龍君　(舊) 神保敏男君

正誤

Errata.

Explanation (denotes "from top"
) denotes "from bottom"

Paper	Page	Line	False	Correct
No. 20, by *Seiichi Kato.*	4	(1	motor	dynamo engine
	6	(5	fleet	amount of catches
No. 256, by *Sir J. H. Biles.*	3	(19	·307	·077
No. 257, by *J. Foster King.*	2	) 13	much discussed as the best multiple	much discussed as to which is the best multiple
	6	) 12	2/4ths	3/4ths
	8	(1	strains	strain

Wave Resistance of a Submerged Body in a Shallow Sea

(Paper No. 610)

By ***Katsutada Sezawa****, Kogakuhakushi,*

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In a recent paper⑴ I have described the formation of deep-water waves due to subaqueous shocks. The problem was important in connection with the submarine explosion. Since the publication of that paper, it occurred me that it is rather interesting to solve the similar problem of a submerged body moving horizontally. The problem of the resistance of a submerged body in a deep water has already been studied by H. Lamb⑵ and T. H. Havelock.⑶ The case I have considered specially is the wave resistance of a submerged body moving parallel to the free surface of a shallow sea. This is, I think, more important on the power and the speed of the submarine boats which are often occasioned to dive along the coast. Indeed, we have not yet found out the favourable depth of immersion at which the submerged moving body in a shallow sea should be placed.

As this question naturally presents a great deal of mathematical difficulties, it may be impossible to expect the rigorous solutions satisfying all the conditions. When some conditions which have no much bearings on the practical problem are disregarded, we obtain the approximate solutions suggesting some important natures of the wave-making resistance of the present case. The results are, thus, to give us a fairly complete view of the resistance of a submarine boat in a shallow sea in a qualitative sense. I may add that in mathematical calculations we may be obliged to take idealised cases; yet such cases will be rather important to find the characteristic natures of the problem. In the first place, I have taken a two-dimensional problem in which a submerged circular cylinder traverses perpendicular to its axis. The complex phenomena of diverging waves which occurs in a three-dimensional case has not entered the analysis and therefore the problem has been much simplified. Again, a certain effect of the image on the condition of the surface of the cylinder has been neglected, owing to very little effect of that condition on the wave-making resistance.

The present paper, after discussing the mathematical part (Appendix), goes on to obtain the general schema of the relation among the velocity of

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(1) "Formation of Deep-water Waves due to Subaqueous Shocks," *Jour. Soc. Nav. Arch., Tokyo,* **44** (1929).

(2) H. Lamb, *Ann. di math.,* **21** (1913).

(3) T. H. Havelock, *Proc. Roy. Soc.,* **93** (1917).

the motion of a cylinder in a shallow sea, the depth of its immersion and the resistance exerted on that body. It will be presently seen that in a shallow sea of a given depth the wave-making resistance increases as the immersion of the cylinder is decreased. A more important fact is that in an assigned immersion of the cylinder there is one maximum of the resistance at a certain speed of the body; this fact may be interpreted as the interference of the free surface on the cylinder. I have also obtained an apparently curious nature that there is no resistance at all speeds beyond the velocity of translation of the shallow-sea waves. This can be explained by the fact that in such speeds the surface elevation of the water becomes symmetrical about the vertical plane passing through the axis of the cylinder. This tendency comforms with the fact which is observed in the

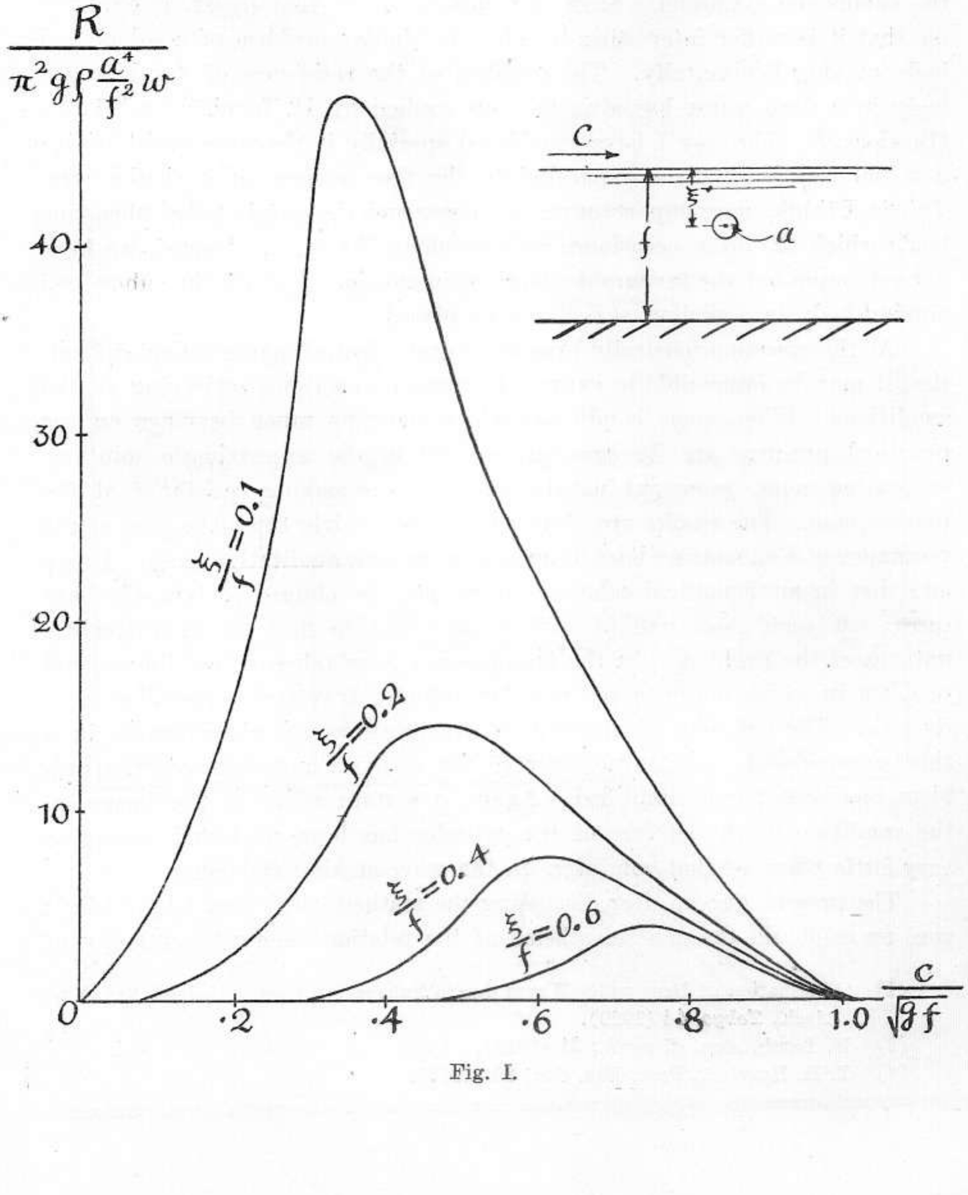

Fig. 1.

case of a voyage in a shallow canal. The above drawing is compiled from the result of the mathematical analysis in Appendix and illustrated as the confirmation of the variation of the resistance with the velocity and the immersion. In this c denotes the velocity, ξ and f are the immersion of the cylinder and the depth of the water respectively and a gives the radius of the cylinder, while w is its length which is supposed to be very large. The resistance R is taken as the ordinate in a dimensionless form by means of g, ρ, a^4 and f^2. (w: length of the cylinder.)

We may now take some numerical example to get the idea of the resistance more concretely. If we take $f=50$ m., $a=1$ m., $\xi=5$ m., $w=w$ cm., then we have the maximum resistance $R=570$ gr. weight. These values will be found to be of the same order of magnitude as those actually observed in model experiments. The object of my paper, however, is not to get the quantitative estimate of the resistance, but rather to know the pure nature of the resistance from the calculation of the idealised example.

In the actual investigation of a submerged body, the immersion ξ being kept constant, we have to determine a route in the sea of a suitable

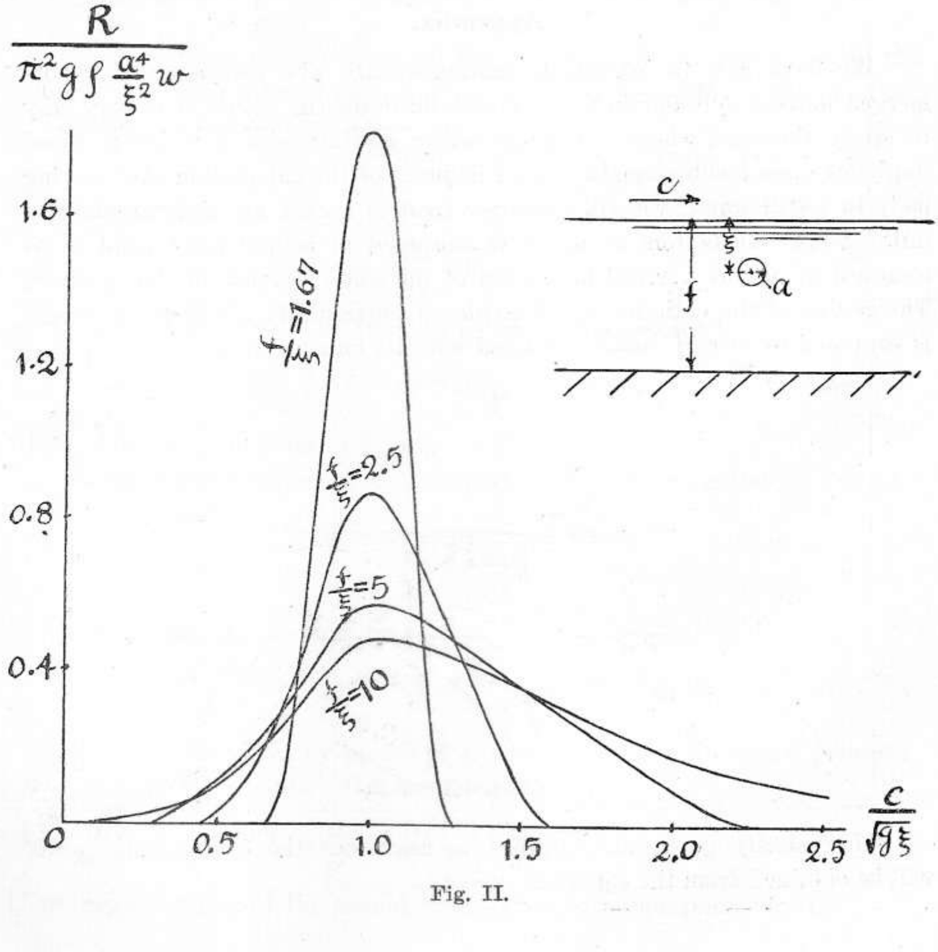

Fig. II.

depth where the resistance of the body at a given speed becomes minimum. I find that for such a purpose it is rather convenient to represent the resistance curve in the manner of Fig. II. The numerical value in this drawing has been taken from the preceding sketch.

It appears from this diagram that the resistance of the submerged body in all depth of the sea is maximum when the velocity of the horizontal motion is equal to $\sqrt{g\xi}$, ξ being the immersion of the body. It seems that at this speed a long trailing train of waves is considerably excited in the form of the translation waves and some interference phenomena between the free surface and the body takes place. W may also notice that for each velocity of the moving body there is a certain value of f which makes the wave resistance of the body minimum. This is not without importance on the selection of the course of a submarine boat at a submerged condition in a shallow sea.

The foregoing considerations have been, as referred to, on a very simple case of a submerged body. The more important problems such as the similar case in three dimensions and the resistance of a body of some other forms will appear in my future studies.

Appendix.

We have now to investigate mathematically the resistance of a submerged moving cylinder in a water of a finite depth. Since it is very easy to study the case where a shallow water moving with a uniform speed impinges upon a submerged cylinder in place of the calculation of a moving body in a still water, we will hereafter treat of such a running stream of a finite depth, the bottom of which is supposed to be perfectly rigid so as to admit of no any vertical movement of the water particle at that surface. The radius of the cylinder which is placed horizontally athwart the stream is supposed to be very small compared with its immersion ξ.

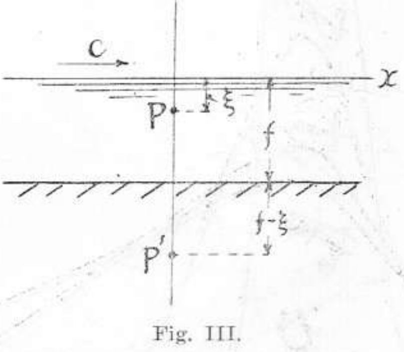

Fig. III.

The velocity potential, ϕ_1, when we leave out the free surface, $y=0$, will be obtained from the equation

$$\frac{\partial^2\phi_1}{\partial x^2}+\frac{\partial^2\phi_1}{\partial y^2}=0, \quad \ldots\ldots\ldots\ldots\ldots\ldots(1)$$

in the form

$$\phi_1=-cx\left(1+\frac{a^2}{r^2}+\frac{a^2}{r'^2}\right), \quad \ldots\ldots\ldots\ldots\ldots\ldots(2)$$

where c is the general velocity of the stream and r, r' denote the distances of any point from the cylinder and its image respectively, such that

$$r=\sqrt{x^2+(y+\xi)^2}, \qquad r'=\sqrt{x^2+(y+2f-\xi)^2}\ldots\ldots\ldots\ldots(3)$$

The expression in (2) satisfies the boundary conditions $\frac{\partial\phi_1}{\partial r}=0$ on $r=a$ and $\frac{\partial\phi_1}{\partial y}=0$ on $y=-f$.

Now we know the expressions

$$\frac{x}{r^2}=\int_0^\infty e^{-k(y+\xi)}\sin kx\,dk, \qquad \frac{x}{r'^2}=\int_0^\infty e^{-k(y+2f-\xi)}\sin kx\,dk. \quad ..(4)$$

Hence we write

$$\phi_1=-cx-a^2c\int_0^\infty e^{-k(y+\xi)}\sin kx\,dk-a^2c\int_0^\infty e^{-k(y+2f-\xi)}\sin kx\,dk. \quad ..(5)$$

In order to annul the stress on the surface $y=0$, we superpose on the solution (5) some other free waves ϕ_2 satisfying the differential equation

$$\frac{\partial^2\phi_2}{\partial x^2}+\frac{\partial^2\phi_2}{\partial y^2}=0, \quad \ldots\ldots\ldots\ldots\ldots\ldots(6)$$

and the condition

$$\frac{\partial\phi_2}{\partial y}=0 \quad \ldots\ldots\ldots\ldots\ldots\ldots(7)$$

at the bottom $y=-f$. The type of ϕ_2 may be expressed by

$$\phi_2=\int_0^\infty A(k)\cosh k(y+f)\sin kx\,dk, \quad \ldots\ldots\ldots\ldots(8)$$

where $A(k)$ is a function of k only.

As the surface elevation η is connected with the potential by the geometrical condition

$$-\frac{\partial\phi}{\partial y}=c\frac{d\eta}{dx}, \quad \ldots\ldots\ldots\ldots\ldots\ldots(9)$$

in which $\phi=\phi_1+\phi_2$, we find

$$\eta=a^2\int_0^\infty\left[e^{-k\xi}+e^{-k(2f-\xi)}+\frac{A(k)}{a^2c}\sinh kf\right]\cos kx\,dk. \quad \ldots\ldots(10)$$

To determine $A(k)$ by means of the condition of the surface pressure, let us introduce Bernoulli's theorem such that

$$\frac{p}{\rho}=-g\eta-\frac{1}{2}\left(\frac{\partial\phi}{\partial x}\right)^2. \quad \ldots\ldots\ldots\ldots\ldots\ldots(11)$$

If we neglect terms of the second order, the above equation gives us

$$\frac{p}{\rho}=-ga^2\int_0^\infty\left[e^{-k\xi}+e^{-k(2f-\xi)}+\frac{A(k)}{a^2c}\sinh kf\right]\cos kx\,dk$$
$$-\frac{c^2}{2}-a\,c^2\int_0^\infty[e^{-k\xi}+e^{-k(2f-\xi)}]\cos kx\,k\,dk$$
$$+c\int_0^\infty A(k)\cosh kf\cos kx\,k\,dk. \quad\ldots\ldots\ldots\ldots\ldots\ldots\ldots(12)$$

Hence the condition of the free surface is satisfied, provided

$$ga^2\left\{e^{-k\xi}+e^{-k(2f-\xi)}+\frac{A(k)}{a^2c}\sinh kf\right\}+ka^2c^2\{e^{-k\xi}+e^{-k(2f-\xi)}\}$$
$$+kc\,A(k)\cosh kf=0. \quad\ldots.(13)$$

This leads to

$$A(k)=\frac{(kc^2+g)\,a^2c}{kc^2\cosh kf-g\sinh kf}\{e^{-k\xi}+e^{-k(2f-\xi)}\}. \quad\ldots\ldots(14)$$

Thus we obtain the expression of the surface elevation in the form:

$$\eta=a^2c^2\int_0^\infty\frac{(\cosh kf+\sinh kf)(e^{-k\xi}+e^{-k(2f-\xi)})}{kc^2\cosh kf-g\sinh kf}\cos kx\,k\,dk. \quad ..(15)$$

Since this expression of η is even function of x, it will be sufficient to take the case of x-positive. If we write $kf=j$, this reduces to

$$\eta=\frac{a^2}{f}\int_0^\infty\frac{j}{j\coth j-\frac{gf}{c^2}}\left(e^{-\frac{j\xi}{f}}+e^{-j\frac{2f-\xi}{f}}\right)\cos\frac{jx}{f}\,dj$$
$$+\frac{a^2}{f}\int_0^\infty\frac{j}{j-\frac{gf}{c^2}\tanh j}\left(e^{-\frac{j\xi}{f}}+e^{-j\frac{2f-\xi}{f}}\right)\cos\frac{jx}{f}\,dj\ldots\ldots\ldots(16)$$

To integrate this expression, we consider the integrals,

$$\int_C\frac{Ze^{-mZ+\frac{ixZ}{f}}}{Z\coth Z-\frac{gf}{c^2}}dZ,\ldots\ldots\ldots\ldots\ldots\ldots(17)$$

$$\int_C\frac{Ze^{-mZ+\frac{ixZ}{f}}}{Z-\frac{gf}{c^2}\tanh Z}dZ,\ldots\ldots\ldots\ \ldots\ldots(18)$$

taken round the contours given in the next page.

In this h is the positive real root of $\frac{\tanh X}{X}=\frac{c^2}{gf}$ $(c^2<gf)$, while k_0, k_1, k_2,.... are the positive real roots of $\frac{\tan Y}{Y}=\frac{c^2}{gf}$ in the case $c^2>gf$; and k_1', k_2', k_3',.... are the similar roots in the case $c^2<gf$. The contour integrations in question then give us, when $c^2>gf$,

$$\int_0^\infty \frac{je^{-mj}\cos\frac{xj}{f}}{j\coth j-\frac{gf}{c^2}}\,dj=-\mathfrak{P}\int_0^\infty \frac{je^{-\frac{jx}{f}}\cos mj}{j\cot j-\frac{gf}{c^2}}\,dj+\pi\sum_{n=0}^{\infty}\frac{k_n^2e^{-\frac{k_nx}{f}}\sin mk_n}{k_n^2-\frac{gf}{c^2}\left(1-\frac{gf}{c^2}\right)}, \qquad \ldots\ldots(19)$$

$$\int_0^\infty \frac{je^{-mj}\cos\frac{xj}{f}}{j-\frac{gf}{c^2}\tanh j}\,dj=-\mathfrak{P}\int_0^\infty \frac{je^{-\frac{jx}{f}}\sin mj}{\frac{gf}{c^2}\tan j-j}\,dj-\pi\sum_{n=0}^{\infty}\frac{k_ne^{-\frac{k_nx}{f}}\cos mk_n}{1-\frac{gf}{c^2}-\frac{c^2}{gf}k_n^2} \qquad \ldots\ldots(20)$$

and, when $c^2<gf$,

$$\int_0^\infty \frac{je^{-mj}\cos\frac{xj}{f}}{j\coth j-\frac{gf}{c^2}}\,dj=-\mathfrak{P}\int_0^\infty \frac{je^{-\frac{jx}{f}}\cos mj}{j\cot j-\frac{gf}{c^2}}\,dj$$

$$+\pi\sum_{n=1}^{\infty}\frac{k'^2_ne^{-\frac{k'_nx}{f}}\sin mk'_n}{k'^2_n-\frac{gf}{c^2}\left(1-\frac{gf}{c^2}\right)}-\frac{\pi h^2e^{-mh}\sin\frac{xh}{f}}{\frac{gf}{c^2}\left(1-\frac{gf}{c^2}\right)+h^2}, \quad \ldots (21)$$

$$\int_0^\infty \frac{je^{-mj}\cos\frac{xj}{f}}{j-\frac{gf}{c^2}\tanh j}\,dj=-\mathfrak{P}\int_0^\infty \frac{je^{-\frac{jx}{f}}\sin mj}{\frac{gf}{c^2}\tan j-j}\,dj$$

$$-\pi\sum_{n=1}^{\infty}\frac{k'_ne^{-\frac{k'_nx}{f}}\cos mk'_n}{1-\frac{gf}{c^2}-\frac{c^2}{gf}k'^2_n}-\frac{\pi he^{-mh}\sin\frac{xh}{f}}{1-\frac{gf}{c^2}+\frac{c^2}{gf}h^2}. \quad \ldots\ldots(22)$$

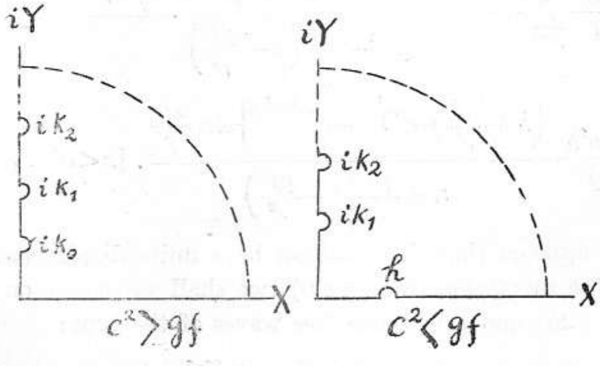

Fig. IV.

In this the principal values due to Cauchy are taken for the evaluation of the definite integrals. The indeterminateness may be avoided by inserting in the equation of motion small frictional terms and finally making the coefficients due to these terms vanish.

Applying now the above formulae, we have the surface displacements of the stream, in which $c^2 > gf$, as in the following forms:

$$\eta = \frac{\pi a^2}{f} \sum_{n=0}^{\infty} \frac{k_n e^{-\frac{k_n x}{f}} \left(k_n \sin \frac{k_n \xi}{f} + \frac{gf}{c^2} \cos \frac{k_n \xi}{f} \right)}{k_n^2 - \frac{gf}{c^2}\left(1 - \frac{gf}{c^2}\right)}, \quad [x>0] \ldots (23)$$

$$\eta = \frac{\pi a^2}{f} \sum_{n=0}^{\infty} \frac{k_n e^{\frac{k_n x}{f}} \left(k_n \sin \frac{k_n \xi}{f} + \frac{gf}{c^2} \cos \frac{k_n \xi}{f} \right)}{k_n^2 - \frac{gf}{c^2}\left(1 - \frac{gf}{c^2}\right)}. \quad [x<0] \ldots (24)$$

The forms of these expressions shew that the surface of the water is quiescent at a large distance from the point $x=0$. It follows, from the above fact and the fact the surface elevation is symmetrical about $x=0$, that no resistance is experienced by the submerged body at the speed c beyond $\sqrt{gf}$.

In a similar manner we find the surface displacement, in the case $c^2 < gf$, in the forms:

$$\eta = \frac{\pi a^2}{f} \sum_{n=1}^{\infty} \frac{k'_n e^{-\frac{k'_n x}{f}} \left(k'_n \sin \frac{k'_n \xi}{f} + \frac{gf}{c^2} \cos \frac{k'_n \xi}{f} \right)}{k'^2_n - \frac{gf}{c^2}\left(1 - \frac{gf}{c^2}\right)}$$

$$- \frac{\pi a^2 h}{f} \frac{\left(h + \frac{gf}{c^2}\right) \left\{ e^{-\frac{\xi}{f} h} + e^{-\frac{2f-\xi}{f} h} \right\} \sin \frac{xh}{f}}{h^2 + \frac{gf}{c^2}\left(1 - \frac{gf}{c^2}\right)}, \quad [x>0] \ \ldots\ldots (25)$$

$$\eta = \frac{\pi a^2}{f} \sum_{n=1}^{\infty} \frac{k'_n e^{\frac{n}{f}} \left(k'_n \sin \frac{k'_n \xi}{f} + \frac{gf}{c^2} \cos \frac{k'_n \xi}{f} \right)}{k'^2_n - \frac{gf}{c^2}\left(1 - \frac{gf}{c^2}\right)}$$

$$+ \frac{\pi a^2 h}{f} \frac{\left(h + \frac{gf}{c^2}\right) \left\{ e^{-\frac{\xi}{f} h} + e^{-\frac{2f-\xi}{f} h} \right\} \sin \frac{xh}{f}}{h^2 + \frac{gf}{c^2}\left(1 - \frac{gf}{c^2}\right)}. \quad [x<0] \ldots\ldots\ldots (26)$$

Since it appears that there cannot be a finite displacement at a large distance on the up-stream side $(x<0)$, we shall superpose on the solutions expressed by (25) and (26) some free waves of the type:

$$- \frac{\pi a^2 h}{f} \frac{\left(h + \frac{gf}{c^2}\right) \left\{ e^{\frac{\xi}{f} h} + e^{-\frac{2f-\xi}{f} h} \right\}}{h^2 + \frac{gf}{c^2}\left(1 - \frac{gf}{c^2}\right)} \sin \frac{xh}{f}. \quad \ldots\ldots (27)$$

Remembering that the first term of each right-hand side of (25) and (26) is insensible beyond a certain distance from the disturbance, we obtain

$$\left.\begin{aligned} \eta &= -\frac{2\pi a^2 h}{f} \frac{\left(h+\frac{gf}{c^2}\right)\left\{e^{-\frac{\xi}{f}h} - e^{-\frac{2f-\xi}{f}h}\right\}}{h^2+\frac{gf}{c^2}\left(1-\frac{gf}{c^2}\right)} \sin\frac{xh}{f} + \&\text{c.}, \ [x>0] \\ \eta &= \text{nil} + \&\text{c.}. \qquad [x<0] \end{aligned}\right\} \quad \ldots (28)$$

As the wave resistance of a body generating shallow-sea waves is expressed by

$$R = \frac{1}{4} g\rho\eta_0^2 \left(1 - \frac{\frac{2gf}{c^2}}{\sinh\frac{2gf}{c^2}}\right)^{(1)}, \quad \ldots\ldots\ldots\ldots (29)$$

where η_0 is the amplitude of the surface elevation, we get the resistance in this case easily by putting

$$\eta_0 = \frac{2\pi a^2 h}{f} \frac{\left(h+\frac{gf}{c^2}\right)\left\{e^{-\frac{\xi}{f}h} + e^{-\frac{2f-\xi}{f}h}\right\}}{h^2+\frac{gf}{c^2}\left(1-\frac{gf}{c^2}\right)}. \quad \ldots\ldots\ldots (30)$$

The compiled results are illustrated in the part of the general discussion.

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(1) Lamb, *Hydrodynamics*, § 249.

Ueber die Bestimmung der günstigsten Dicke des Warmeschutzmaterials für Dampfleitungen des Kriegsschiffes.

(Paper No. 611)

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I. Vorwort.

Man umhüllt die Dampfleitungen mit Isolierungsmaterial, um den Wärmeverlust möglichst klein zu halten. Dabei nimmt der Wärmeverlust mit der Zunahme der Isolierungsdicke zuerst merklich, dann aber nur wenig ab; und da das Gewicht des Isolierungsmaterials gleichzeitig zunimmt, ist eine zu grosse Dicke nicht zweckmässig. Es ist das Ziel der folgenden Berechnung, die günstigste Umhüllungsdicke für Rohrleitungen an Bord zu bestimmen.

II. Versuchseinrichtung und Versuchsverfahren.

Die in den Versuchen verwendeten schmiedeisernen Dampfleitungen sind in Fig. (I) durch (ab) bezeichnet. Es ist ein Wasserabscheider (A) angeordnet, in dem der Dampf vor Eintritt in die eigentliche Versuchsleitung entwässert wird. Die Leitungsstücke (ab) sind mit Gefälle in der Richtung des Dampfweges gelegt; ihre Hauptabmessungen und die Dicke der nacheinander umgehüllten Isolatoren sind wie folgt:

Länge der Srecken (cm)	Lichter Durchmesser (cm)	Wandstärke (cm)	Äusserer Durchmesser (cm)	Dicke von Isolatoren cm					Art von Isolatoren	Absolute Dampfdruck kg/cm²
361	6.35	0.294	6.94	0	1.27	1.91	2.54	3.81	Asbestosdecke	16
〃	12.70	0.488	13.68	0	1.27	2.54	3.81	5.10		
〃	20.30	0.635	21.57	0	1.91	3.81	5.10	6.35		
361	6.35	0.294	6.94	0	0.95	1.91	3.18	4.44	Hemische Asbestosfilz	3
〃	20.30	0.635	21.57	0	0.95	2.22	3.49	6.04		
〃	28.00	0.635	29.27	0	0.95	2.22	4.76	8.58		

In jeder Reihe der Versuche wurden je drei Leitungen von verschiedenen Durchmesser benutzt, und die Dicke des Isolierungsmaterials (Asbestosdecke und Asbestosfilz) war auch verschieden. Der Dampfdruck war zweierlei gewählt; 16 und 3 atm. entsprechend den Druckhöhen der Frisch- und Abdampfleitungen in Praxis.

Die Temperatur des gesättigten Dampfes wurde von dem durch den Manometer (M), Fig. (1), gezeigten Druck bestimmt.

Der Wärmeverlust in der Leitungen war durch Mengenbestimmung des entstandenen Niederschlagwassers gemessen, und zu diesem Zwecke hatte jede Versuchsleitung einen Wasserkollektor (W), Fig. (1); dabei war die Wassermenge mit dem zugehörigen Klingerschen Wassergehaltzeiger aus Glass in je zwanzig Minuten abgemessen.

Um die in der Versuchsleitung niedergeschlagene Wassermenge von Nebenumständen zu befreien, waren die Korrektion des Wasserkollektors und diejenige der nicht umgehüllten Fransche erforderlich. Von diesen zwei Korrektionen wurde die erste dadurch gemacht, dass ein Wasserkollektor unmittelbar hinter dem Wasserabscheider verbunden war, und das Niederschlagwasser desselben geeichnet wurde; die zweite war aus dem Wärmeverlust der nackten Leitung abgeschätzt, u.zw. durch Berechnung des verhältnismässigen Anteils der Oberfläche der Flansche.

Bei der Durchführung der Versuche musste der grösste Wert auf die Erreichung und Erhaltung des Beharrungszustandes gelegt werden. Ein Versuch wurde erst begonnen, als die Temperaturen der Umhüllung nicht mehr änderten. Dieser Zustand wurde in unserem Fall erst nach 2 bis 2½ st. erreicht; und während des Versuches wurde die Dampfspannung der Leitungen möglichst konstant erhalten.

III. Versuchsresultate.

Mit den vorstehenden Einrichtungen wurden die Versuche ausgeführt, hauptsächlich um die Abhängigkeit des Wärmeverlustes einer Rohrleitung von der Isolatordicke festzustellen. Die Resultate der Messungen sind in den Tafeln (1) bis (10) dargestellt.

Die vorstehenden Resultate lassen sich wie in den Tafeln (11) u. (12) zusammenfassen.

IV. Berechnung und Bestimmung der günstigsten Isolatordicke.

Es bezeichne:

ρ = specizifisches Gewicht des Isolierungsmaterials,
D_0 = äusserer Durchmesser der Rohrleitung in cm,
D = „ „ des Isolators in cm,
Q_0 = stündlicher Wärmeverlust in Kg cal. pro m. Länge der Rohrleitung bei 1°C Temperatur Unterschied zwischen der Oberfläche der Leitung und der Umgebungsluft,
η = mittelerer thermischer Wirkungsgrad von Schiffsmaschinen.

Frischdampfleitungen:—

Das Gewicht des Isolators ist

$$W_1=\rho\pi\frac{D^2-D_0^2}{4000}\times 100 \qquad \text{Kg/m.}$$

Dieses Gewicht ist nichts anders als Energieverlust der Schiffsmaschinen. Wenn bei Kriegsschiffe die von I Tonne Gewicht der Maschinen, Brennstoff usw. gelieferte Pferdestärke 7,5 angenommen wird, so berechnet sich der Energieverlust N_1 des Isolators zu

$$N_1=\rho\pi\frac{D^2-D_0^2}{4000}\times\frac{7.5}{1000}\times 100 \qquad \text{P.S/m.,}$$

oder mit $\rho=0.28$

$$N_1=0.000165(D^2-D_0^2) \qquad \text{P.S/m.}$$

Es sei die mittlere Temperaturdifferenz zwischen der inneren Seite des Isolators und dem Maschinenraum im Kriegsschiff 160°C, und ferner sei angenommen, dass der Wärmeverlust durch den Luftstrom etwa 10% mehr als bei ruhendem Zustand beträgt. Es verliert dann die Wärmemenge

$$Q'=1.1\times 160\times Q_0 \qquad \text{Kgcal/m, st.}$$

Dieser Energieverlust N_2 beträgt also

$$N_2=\frac{427}{75\times 3600}\times\eta\times Q' \qquad \text{P.S/m.}$$

When wir $\eta=0.18$ setzen, so wird es

$$N_2=0.000285\times Q' \qquad \text{P.S/m.}$$

Ferner ist das Gewicht des Lüftungsanlage auf den Wärmeverlust folglich auf die Lufterwärmung abhängig, um die Raumtemperatur konstant (höchstens etwa 35°C) zu halten. Die Luftmenge, welche von Kesseln- u. Maschinenräume abgeführt werden soll, ist also

$$w=\frac{Q'}{C_p(35-20)} \qquad \text{Kg/m, st.,}$$

C_p bedeutet die spezifische Wärme von Luft und ist gleich 0,237 gr. cal/gr.; und (35-20) ist das Temperaturgefälle zwischen Maschinenraum und äusserer Atmosphäre. Das Volumen derselben wird sonach

$$V=\frac{0.86}{0.237\times(35-20)}\times Q'$$

$$=0.242\times Q' \qquad \text{m}^3\text{/m., st.}$$

Also erhalten wir den Energieverlust für die Lüftungsanlage

$$N_3=0.242\times\frac{1}{2.21}\times\frac{7.5}{1000}\times Q'$$

$$=0.00082\times Q' \qquad \text{P.S/m.}$$

dabei bedeutet die Zahl 2,21 das 1 Kg. Gewicht von Gebläse, Brennstoff usw. entsprechende Luftvolumen.

Nun können wir die günstigste Dicke des Isolierungsmaterials so bestimmen, dass

$$N_1+N_2+N_3=min.,$$

wobei N_1, N_2, N_3 alle Funktionen von Umhüllungsdicke sind.[1]

Abdampfröhren:—

Wenn keine Wärmeenergie des Abdampfes wieder benutzt werden soll, so kann das Glied N_2 ausser Betracht stehen lassen. Sonst ist der Vorgang der Berechnung gleich wie beim obigen Fall. Also können wir die Bedingung für die günstigste Umhüllungsdicke so schreiben, wie folgt

$$N_1+N_3=\text{min.}$$

Wir haben dieselbe Dicke (h_0), die die obige Formel ein Minimum macht, durch die graphische Methode bestimmt, wie aus Fig. (2, 3, 4) u. (5, 6, 7) ersichtlich ist. Die hier in Betracht kommenden Werte sind in den Zahlentafeln (13) und (14) zusammengestellt.

In Fig. (8) und (9) sind die auf diese Weise bestimmte günstigste Umhüllungsdicke (h_0) u. zw. in Bezug auf verschiedenen Rohrabmessungen bezeichnet, sowie die Kurve, die die Beziehung zwischen dem Wärmeverlust und der Isolatordicke darstellt.

V. Versuchsergebnis und Schluss.

Aus den bisherigen Versuchserfahrungen können wir die günstigste Umhüllungsdicke für verschiedene Leitungsdurchmesser feststellen, wie Fig. (10) zeichnerisch darstellt.

Ohne Zweifel gilt der hier ermittelte Wert von Umhüllungsdicke für den Fall allein, wo die Annahme besteht, die wir mit unseren Kriegsschiffe gemacht haben; aber in anderen Fälle z.B. im Handelsschiff, kann man an dieser Bestimmungsweise der Isolatordicke beliebige Anwendungen vornehmen, wenn man jede geeignete Annahme macht.

Schliesslich ist der Verfasser Herrn Prof. Dr.-Ing. A. Ono, Hukuoka, für seinen wertvollen Rat zu besonderem Danke verpflichtet.

(April, 1929.)

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[1] Man kann auch so vorgehen, dass die Summe der Gewichte des Wärmeschutzmaterials und der Gebläse, so wie auch des Kessels, welcher zur Erzeugung des Dampfes von der Wärmemenge Q' nötig ist, ein Minimum wird. Aber das Ergebnis der Berechnung wird mit dem vorliegenden Fall gänzlich übereinstimmen.

(Paper No. 611)

TAF. (1)

				Niederschlagwasser Röhrendurchmesser		
Zeit der Messung	Absolute Dampfspannung	Dampftemperature	Mittlere Lufttemperatur	6.35^{cm}	12.70^{cm}	20.30^{cm}
				Dicke von Asbestosdecke		
				0	0	0
	kg/cm²	°c	°c	ltr/20min	ltr/20min	ltr/20min
Vorm. 10—0	16	200	41.0	2.55	3.70	4.40
10—20	〃	〃	42.1	1.52	3.60	4.90
10—40	〃	〃	42.4	1.65	3.75	4.80
11—0	〃	〃	43.5	1.97	——	——
11—20	〃	〃	43.9	1.92	3.50	4.90
11—40	〃	〃	——	——	——	——
12—0	〃	〃	——	——	——	——
Nachm 0 - 20	〃	〃	——	2 0	——	——
0—40	〃	〃	43.9	2.20	——	——
1—0	〃	〃	44.3	1.72	3.80	5.10
1—20	〃	〃	42.3	——	3.90	5.10
1—40	〃	〃	42.3	2.40	——	5.00
2—0	〃	〃	41.9	1.88	3.80	——
2—20	〃	〃	42.9	2.03	4.00	4.80
2—40	〃	〃	41.8	——	——	5.20
3—0	〃	〃	41.4	2.27	4.10	——
3—20	〃	〃	40.9	2.07	4.00	5.25
3—40	〃	〃	42.5	1.95	3.90	4.85
4—0	〃	〃	42.5	1.98	——	——
4—20	〃	〃	——	——	——	——
Mittlere Werte	16	200	42.5	2.04	3.82	4.94

TAF. (2)

				Niederschlagwasser Röhrendurchmesser		
Zeit der Messung	Absolute Dampfspannung	Dampftemperatur	Mittlere Lufttemperatur	6.35$^{\text{cm}}$	12.70$^{\text{cm}}$	20.30$^{\text{cm}}$
				Dicke von Asbestosdecke		
				1.27$^{\text{cm}}$	1.27$^{\text{cm}}$	1.91$^{\text{cm}}$
	kg/cm^2	°c	°c	ltr/20min	ltr/20min	ltr/20min
Vorm. 10—0	16	200	37.0	1.00	——	——
10—20	〃	〃	38.5	1.00	1.88	1.80
10—40	〃	〃	38.6	0.89	1.27	1.61
11—0	〃	〃	38.8	0.88	1.74	2.16
11—20	〃	〃	42.5	0.81	1.62	2.21
11—40	〃	〃	42.6	1.05	1.70	2.05
12—0	〃	〃	——	——	1.73	2.14
Nachm. 0—20	〃	〃	42.0	——	1.30	——
0—40	〃	〃	42.7	——	——	——
1—0	〃	〃	42.9	0.93	1.78	2.33
1—20	〃	〃	42.3	0.81	1.50	1.69
1—40	〃	〃	41.8	0.99	1.50	1.70
2—0	〃	〃	41.8	0.79	1.52	——
2—20	〃	〃	41.4	1.13	——	——
2—40	〃	〃	40.9	1.00	1.46	1.85
3—0	〃	〃	41.4	0.84	1.66	2.00
3—20	〃	〃	41.8	1.00	1.63	1.66
3—40	〃	〃	42.0	——	1.40	1.51
4—0	〃	〃	40.0	1.00	1.28	1.42
4—20	〃	〃	——	——	——	——
Mittlere Werte	16	200	41.1	0 94	1.56	1.85

TAF. (3)

Zeit der Messung	Absolute Dampfspannung	Dampftemperatur	Mittlere Lufttemperatur	Niederschlagwasser Röhrendurchmesser		
				6.35^{cm}	12.70^{cm}	20.30^{cm}
				Dicke von Asbestosdecke		
				1.91^{cm}	2.54^{cm}	3.81^{cm}
	kg/cm^2	°c	°c	ltr/20min	ltr/20min	ltr/20min
Vorm. 10—0	16	2.00	28.9	0.80	1.50	1.85
10—20	〃	〃	29.8	0.92	—	1.55
10—40	〃	〃	30.1	0.70	1.63	—
11—0	〃	〃	31.0	0.90	1.25	1.50
11—20	〃	〃	31.4	0.80	1.25	1.13
11—40	〃	〃	—	1.20	—	—
12—0	〃	〃	30.9	—	—	—
Nachm. 0—20	〃	〃	30.4	—	—	—
0—40	〃	〃	31.0	0.85	1.52	1.98
1—0	〃	〃	31.5	0.98	1.40	1.32
1—20	〃	〃	31.9	0.87	1.20	1.60
1—40	〃	〃	32.4	0.82	1.20	1.47
2—0	〃	〃	32.0	0.78	1.18	1.43
2—20	〃	〃	31.5	0.67	1.43	—
2—40	〃	〃	31.3	0.76	1.22	1.96
3—0	〃	〃	32.3	0.70	1.42	1.87
3—20	〃	〃	32.3	0.71	1.38	1.50
3—40	〃	〃	32.3	0.96	1.20	1.62
4—0	〃	〃	32.3	1.16	—	1.25
4—20	〃	〃	—	—	—	—
Mittlere Werte	16	2.00	31.2	0.85	1.34	1.57

TAF. (4)

				Niederschlagwasser Röhrendurchmesser		
Zeit der Messung	Absolute Dampfspannung	Dampftemperatur	Mittlere Lufttemperatur	6.35$^{\text{cm}}$	12.70$^{\text{cm}}$	20.30$^{\text{cm}}$
				Dicke von Asbestosdecke		
				2.54$^{\text{cm}}$	3.81$^{\text{cm}}$	5.10$^{\text{cm}}$
	kg/cm²	°c	°c	ltr/20min	ltr/20min	ltr/20min.
Vorm. 10–0	16	200	——	0.60	——	——
10–20	〃	〃	32.5	0.45	——	——
10–40	〃	〃	33.3	0.35	——	——
11–0	〃	〃	34.8	0.75	1.20	1.25
11–20	〃	〃	35.5	0.75	0.90	1.00
11–40	〃	〃	3.59	0.90	1.05	1.14
12–0	〃	〃	35.6	——	0.85	1.09
Nachm. 0–20	〃	〃	36.0	——	1.20	0.97
0–40	〃	〃	36.5	0.90	0.95	1.32
1–0	〃	〃	37.8	——	0.85	1.25
1–20	〃	〃	——	0.85	——	——
1–40	〃	〃	37.5	0.85	1.20	1.68
2–0	〃	〃	38.0	0.80	1.18	1.87
2–20	〃	〃	37.9	0.70	1.17	1.30
2–40	〃	〃	38.0	0.73	1.05	1.37
3–0	〃	〃	38.5	0.82	1.00	1.26
3–20	〃	〃	38.4	0.76	1.10	1.46
3–40	〃	〃	38.6	0.71	——	1.07
4–0	〃	〃	38.9	0.71	1.02	——
4–20	〃	〃	——			——
Mittlere Werte	16	200	36.7	0.73	1.05	1.29

TAF. (5)

				Niederschlagwasser Röhrendurchmesser		
Zeit der Messung	Absolute Dampfspannung	Dampftemperatur	Mittlere Lufttemperatur	6.35^{cm}	12.70^{cm}	20.30^{cm}
				Dicke von Asbestosdecke		
				3.81^{cm}	5.10^{cm}	6.35^{cm}
	kg/cm^2	°c	°c	ltr/20min.	ltr/20min	ltr/20min.
Vorm. 10—0	16	200	33.0	0.80	——	——
10—20	〃	〃	36.4	0.80	1.00	1.03
10—40	〃	〃	35.1	0.74	1.01	1.40
11—0	〃	〃	35.5	0.74	1.00	1.34
11—20	〃	〃	36.5	0.71	0.99	1.42
11—40	〃	〃	36.5	——	——	——
12—0	〃	〃	35.8	——	1.27	1.92
Nachm. 0—20	〃	〃	35.6	1.36	0.98	1.30
0—40	〃	〃	36.3	0.84	0.65	1.26
1—0	〃	〃	36.5	0.80	——	——
1—20	〃	〃	37.5	0.83	0.98	1.49
1—40	〃	〃	37.9	0.75	1.15	1.55
2—0	〃	〃	38.0	0.72	1.15	1.55
2—20	〃	〃	38.0	0.73	0.88	1.06
2—40	〃	〃	37.5	0.63	1.09	1.36
3—0	〃	〃	36.8	0.62	0.93	1.27
3—20	〃	〃	36.4	0.60	——	——
3—40	〃	〃	37.9	0.80	1.14	1.40
4—0	〃	〃	37.5	0.76	1.30	1.92
4—20	〃	〃	——	——	——	——
Mittlere Werte	16	200	36.6	0.78	1.03	1.42

TAF. (6)

				Niederschlagwasser Röhrendurchmesser		
Zeit der Messung	Absolute Dampfspanung	Dampftemperatur	Mittlere Lufttemperatur	6.35^{cm}	20.30^{cm}	28.00^{cm}
				Dicke von Asbestosfilz		
				0	0	0
	kgt/cm^2	°c	°c	ltr/20min	ltr/20min.	ltr/20min.
Vorm. 9—50	3	133	30.0	0	0	0
10—10	〃	〃	31.5	0.9	1.7	2.7
10—30	〃	〃	31.0	1.0	1.8	2.9
10—50	〃	〃	31.3	1.0	1.9	2.9
11—10	〃	〃	31.3	0.9	2.0	2.7
11—30	〃	〃	31.0	0.7	1.9	2.8
Nachm. 0—30	〃	〃	——	——	——	——
0—50	〃	〃	32.0	0.9	2.2	2.8
1—10	〃	〃	32.0	0.9	2.4	3.0
1—30	〃	〃	32.0	0.8	2.1	2.8
1—50	〃	〃	32.0	0.8	2.0	2.7
2—10	〃	〃	32.5	1.0	2.1	2.9
2—30	〃	〃	32.5	0.9	2.2	2.8
2—50	〃	〃	33.0	1.1	2.3	3.0
3—10	〃	〃	33.0	1.2	2.1	2.9
3—30	〃	〃	33.0	0.9	1.9	2.6
Mittlere Werte	3	133	32.4	0.92	2.12	2.83

TAF. (7)

				Niederschlagwasser Röhrendurchmesser		
Zeit der Messung	Absolute Dampfspannung	Dampf temperatur	Mittlere Luft-temperatur	6.35^{cm}	20.30^{cm}	28.00^{cm}
				Dicke von Asbestosfilz		
				0.95^{cm}	0.95^{cm}	0.95^{cm}
	kg/cm^2	°c	°c	ltr/20min.	ltr/20min.	ltr/20min.
Vorm. 10—0	3	133	26.5	0	0	0
10—20	〃	〃	27.0	0.6	1.0	0.9
10—40	〃	〃	26.5	0.5	1.0	1.0
11—0	〃	〃	27.0	0.5	0.8	1.1
11—20	〃	〃	26.5	0.5	0.8	1.0
Nachm. 0—20	〃	〃	25.5	—	—	—
0—40	〃	〃	26.0	0.5	0.75	1.0
1—0	〃	〃	27.0	0.4	0 55	1.1
1—20	〃	〃	27.0	0.6	0.80	1.2
1—40	〃	〃	27.5	0.5	1.00	1.2
2—0	〃	〃	27.5	0.7	0.9	1.2
2—20	〃	〃	27.0	0.5	0.9	1.1
2—40	〃	〃	27.0	0.6	0.75	1.1
3—0	〃	〃	27.0	0.4	0.7	1.1
3—20	〃	〃	27.0	0.5	0.9	1.2
3—40	〃	〃	27.0	0.4	0.7	1.0
Mittlere Werte	3	1.33	27.0	0.51	0.80	1.12

TAF. (8)

				Niederschlagwasser Röhrendurchmesser		
Zeit der Messung	Absolute Dampfspannung	Dampftemperatur	Mittlere Lufttemperatur	6.35$^{\text{cm}}$	20.30$^{\text{cm}}$	28.00$^{\text{cm}}$
				Dicke von Asbestosfilz		
				1.91$^{\text{cm}}$	2.22$^{\text{cm}}$	2.22$^{\text{cm}}$
	kg/cm^2	°c	°c	ltr/20min.	ltr/20min.	ltr/20min.
Vorm. 10—20	3	133	27.0	0	0	0
10—40	〃	〃	27.0	0.4	0.4	0.65
11—0	〃	〃	28.0	0.4	0.5	0.75
11—20	〃	〃	28.0	0.75	0.8	1.85
Nachm. 0—20	〃	〃	27.0	—	—	—
0—40	〃	〃	27.0	0.70	0.8	0.8
1—0	〃	〃	27.0	0.60	0.8	0.8
1—20	〃	〃	27.0	0.50	0.6	0.7
1—40	〃	〃	38.0	0.40	0.5	0.9
2—0	〃	〃	27.5	0.50	0.8	0.7
2—20	〃	〃	27.0	0.40	0.7	0.8
2—40	〃	〃	27.0	0.40	0.8	0.7
3—0	〃	〃	27.0	0.40	0.7	1.0
3—20	〃	〃	27.0	0.50	0.8	0.8
3—40	〃	〃	28.0	0.50	0.7	0.8
4—0	〃	〃	28.0	0.50	0.8	1.0
Mittlere Werte	3	133	27.4	0.47	0.72	0.82

TAF. (9)

				Niederschlagwasser Röhrendurchmesser		
Zeit der Messung	Absolute Dampf- spannung	Dampf- temperatur	Mittlere Luft- temperatur	6.35$^{\text{cm}}$	20.30$^{\text{cm}}$	28.00$^{\text{cm}}$
				Dicke von Asbestosfilz		
				3.18$^{\text{cm}}$	3.49$^{\text{cm}}$	4.76$^{\text{cm}}$
	kg/cm^2	°c	°c	ltr/20min.	ltr/20min.	ltr/20min.
Vorm. 9—20	3	133	27.0	0	0	0
9—40	〃	〃	27.5	0.4	0.5	0.7
10—0	〃	〃	27.5	0.4	0.6	0.6
10—20	〃	〃	28.0	0.4	0.5	0,6
10—40	〃	〃	28.0	0.4	0.6	0.6
11—0	〃	〃	28.0	0·4	0 6	0.6
11—20	〃	〃	29.0	0.4	0.6	0.7
Nachm. 1—0	〃	〃	29.0	0.3	0.5	0.6
1—20	〃	〃	29.0	0.5	0.6	0.7
1—40	〃	〃	29.0	0.4	0.7	0.6
2—0	〃	〃	29.5	0.4	0.5	0.5
2—20	〃	〃	29.0	0.5	0.6	0.6
2—40	〃	〃	29.0	0.5	0.7	0.7
3—0	〃	〃	29.0	0.4	0.6	0.7
3—20	〃	〃	29·0	0.5	0.7	0,7
3—40	〃	〃	29.0	0·3	0.6	0.7
Mittlere Werte	3	133	29.0	0.42	0.61	0.65

TAF. (10)

				Niederschlagwasser Röhrendurchmesser		
Zeit der Messung	Absolute Dampfspannung	Dampftemperatur	Mittlere Lufttemperatur	6.35$^{\mathrm{cm}}$	20.30$^{\mathrm{cm}}$	28·00$^{\mathrm{cm}}$
				Dicke von Asbestosfilz		
				4.44$^{\mathrm{cm}}$	6.04$^{\mathrm{cm}}$	8.58$^{\mathrm{cm}}$
	kg/cm^2	°c	°c	ltr/20min.	ltr/20min.	ltr/20min.
Vorm. 9−40	3	133	24.5	0	0	0
10—0	〃	〃	24.5	0.4	0.5	0.4
10—20	〃	〃	25.0	0.3	0.6	0.6
10—40	〃	〃	25.0	0.4	0.7	0.7
11—0	〃	〃	25.0	0.3	0.5	0.5
11—20	〃	〃	25.0	0.5	0.8	0.9
Nachm. 0—40	〃	〃	27.0	0.2	0.2	0.1
1—0	〃	〃	27.5	0.3	0.3	0.3
1—20	〃	〃	27.5	0.5	0.5	0.5
1—40	〃	〃	27.0	0.4	0.6	0.5
2—0	〃	〃	27.0	0.3	0.6	0.7
2--20	〃	〃	27.0	0.4	0,7	0.7
2—40	〃	〃	27.0	0.3	0.6	0.7
3—0	〃	〃	27.0	0.4	0.6	0.7
3—20	〃	〃	27.0	0.4	0.7	0.7
3—40	〃	〃	27.0	0.3	0.5	0.7
4—0	〃	〃	27.0	0.4	0.5	0.6
Mittlere Werte	3	133	27.1	0,37	0.56	0,61

TAF. (11)

Durchmesser der Röhre	Dicke der Asbestosdecke	Niederschlagwasser	Wärmeverlust	Dampftemperatur	Mittlere Lufttemperatur
cm	cm	ltr/20min.	kg.cal/m.st.	°c	°c
6.35	0	2.04	674	200	42.5
〃	1.27	0.94	310	〃	41.1
〃	1.91	0.85	280	〃	31.2
〃	2.54	0.73	241	〃	36.7
〃	3.81	0.78	257	〃	36.6
12.70	0	3.82	1259	200	42.5
〃	1.27	1.56	515	〃	41.1
〃	2.54	1.34	443	〃	31.2
〃	3.81	1.05	347	〃	36.7
〃	5.10	1.03	340	〃	36.6
20.30	0	4.94	1630	200	42.5
〃	1.91	1.85	611	〃	41.1
〃	3.81	1.57	518	〃	31.2
〃	5.10	1.29	426	〃	36.7
〃	6.35	1.42	469	〃	36.6

TAF. (12)

Durchmesser der Röhre	Dicke der Hemischen Asbestosfilz	Niederschlagwasser	Wärmeverlust	Dampftemperatur	Mittlere Lufttemperatur
cm	cm	ltr/20min.	kg.cal/m.st.	°c	°c
6.35	0	0.92	368	133	32.4
〃	0.95	0.51	204	〃	27.0
〃	1.91	0.47	188	〃	27.4
〃	3.18	0.42	168	〃	29.0
〃	4.44	0.37	148	〃	27.1
20.30	0	2.12	849	133	32.4
〃	0.95	0.80	320	〃	27.0
〃	2.22	0.72	288	〃	27.4
〃	3.49	0.61	244	〃	29.0
〃	6.04	0.56	224	〃	27.1
28.00	0	2.83	1164	133	32.4
〃	0.95	1.12	448	〃	27.0
〃	2.22	0.82	328	〃	27.4
〃	4.76	0.65	260	〃	29.0
〃	8.58	0.61	244	〃	27.1

ZAHLENTAFEL (13)

Innerer Durchmesser der Leitung	Dicke der Asbestosdecke	Temperaturgefälle zwischen der innerenseite der Isolierung und der mittleren Luft-temp.	Wärmeverlust bei 1°c Unterschied	Desgl. bei 160°c Unterschied	$1.1 \times Q$	$0{\cdot}000165 \times (da^2 - di^2)$	$0.001105 \times Q'$
di	$\frac{da-di}{2}$	$t_1 - tr$	Q_0	Q	Q'		
cm	cm	°c	kg.cal/m.st°c	kg.cal./m.st.160°c	kg. cal./m.st. 160°c	p.s/m	p.s/m
6.94	0	152.5	4.42	571.0	629·0	0	0·6930
〃	1.27	153.9	2.02	126.0	139.0	0·00689	0·1538
〃	1.91	163.8	1.71	81.2	89.4	0·01120	0·0986
〃	2.54	158.3	1.52	59.7	65.7	0·01600	0·0726
〃	3.81	158.4	1.62	50.2	55.4	0·02720	0·0612
13.68	0	152.5	8.25	1151.0	1269.0	0	1·3990
〃	1.27	153.9	3.35	302.0	332.0	0·0125	0·3668
〃	2.54	163.8	2.71	166.4	183.1	0·0272	0·2020
〃	3.81	158.3	2.19	115.0	126.8	0·0441	0·1390
〃	5.10	158.4	2.15	97.5	107·3	0·0630	0·1180
21.57	0	152.5	10.70	1518.0	1670·0	0	1·8400
〃	1.91	153.9	3.98	355.0	391·0	0·0296	0·4330
〃	3.81	163.8	3.17	209.0	230·0	0·0640	0·2540
〃	5.10	158.3	2.69	173.0	190·5	0·0899	0·2105
〃	6.35	158.4	2.96	157.0	173·0	0·1170	0·1910

Bemerkungen:

1. Absolute Dampfspannung = 16 kg/cm²
2. Länge der Leilungs-strecken = 361 cm
3. Korrektion der Wasserableitung = 0.39 ltr/20min
4. Dergl. der Flaschen = 0·191 ; 0·364 ; 0·478 ltr/20min, entsprechend den drei verschiedenen Grössen der Rohrleitung.

ZAHLENTAFEL (14)

Innerer Durchmesser der Leitung	Dicke des Hemische Asbestosfilz	Temperaturgefälle zwischen der inneren-seite der Isolierung und der mittleren Luft-temp.	Wärme-verlust bei 1° Unterschied	Desgl. bei 100°c Unterschied	$1.1 \times Q$	$0.000165 \times (da^2 - di^2)$	$0.00082 \times Q'$
di	$\frac{d_a - d_i}{2}$	$t_1 - t_r$	Q_0	Q	Q'		
cm	cm	°c	kg.cal/m.st°c	kg.cal/m.st.100°c	kg.cal/m.st.100	P.S/m	P.3/m
6.94	0	95.6	3.85	253.0	278.1	0	0.2280
〃	0.95	101.0	2.02	94.0	107.0	0.00495	0.0849
〃	1.91	100·6	1.87	60.0	66.0	0.01120	0.0541
〃	3.18	99.0	1.70	40.0	44.0	0.02120	0.0362
〃	4.44	100.9	1.47	35.0	38.5	0.03340	0.0316
21.57	0	95.6	8.78	705.0	776.0	0	0.6370
〃	0.95	101.0	3.17	148.0	163.0	0.01420	0.1338
〃	2.22	100.6	2.86	97.5	107.1	0.03481	0.0880
〃	3.49	99.0	2.46	55.0	60.5	0.05780	0.0496
〃	6.04	100.9	2.22	45.0	49.5	0.11510	0.0406
29.27	0	95.6	12.18	964.0	1060.0	0	0.8700
〃	0·95	101.0	4.44	248.0	273.0	0.01930	0.2240
〃	2·22	100.6	3.26	109.0	120.0	0.04670	0.0984
〃	4.76	99.0	2.62	55·5	61.0	0.10780	0.0500
〃	8.58	100.9	2.42	40.0	44.0	0.22000	0.0361

Bemerkungen :

1. Absolute Dampfspannung = 3 kg/cm²
2. Länge der Leitungs-strecken = 361 cm
3. Korrektion der Wasserableitung = 0.24 ltr/20min
4. Dergl. der Flaschen = 0.075 ; 0.191 ; 0.290 ltr/20 min. entsprechend den drei verschiedenen Grössen der Rohrleitung.

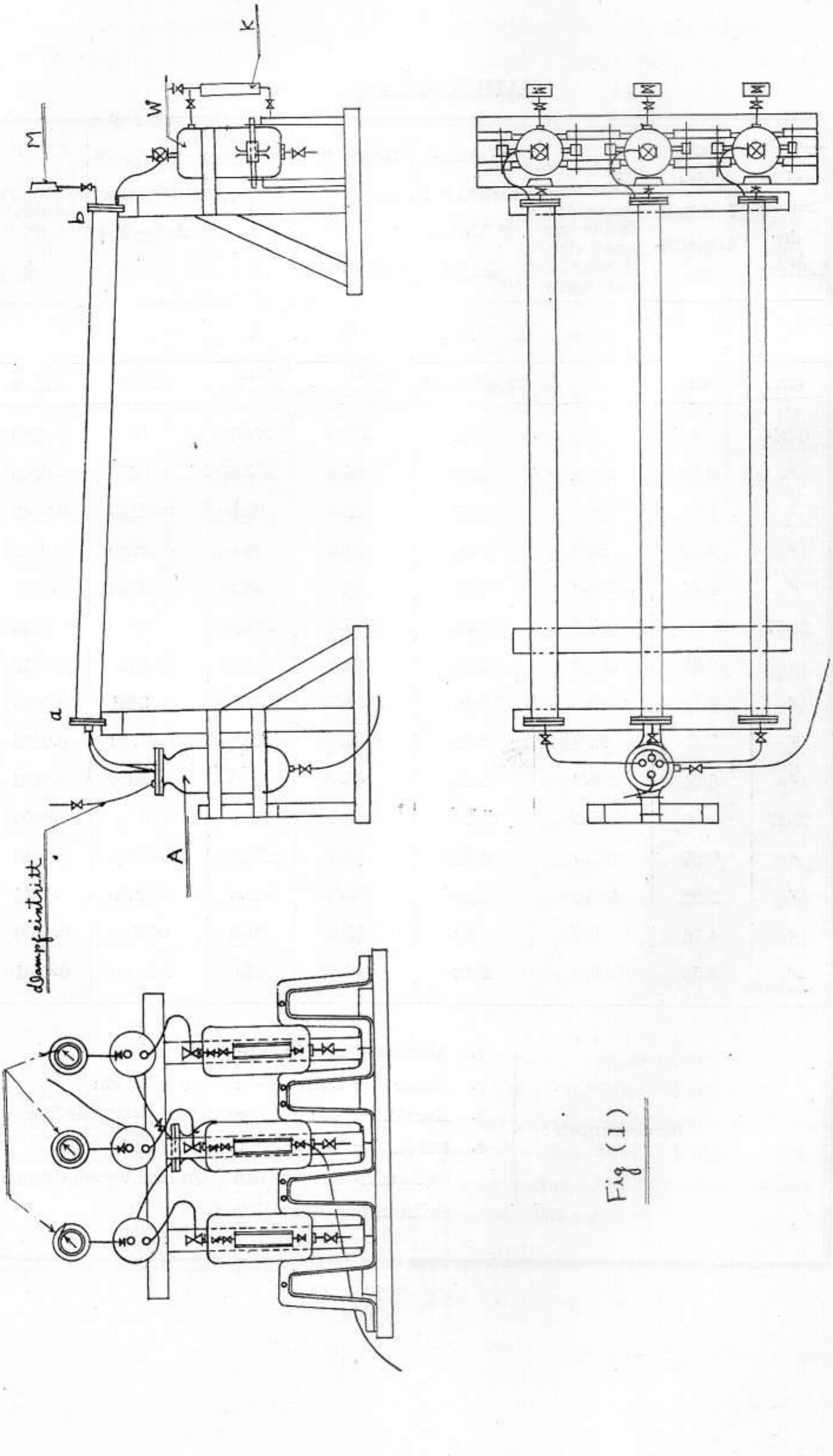


Fig (1)

Fig. (2)

(Frischdampfleitung)

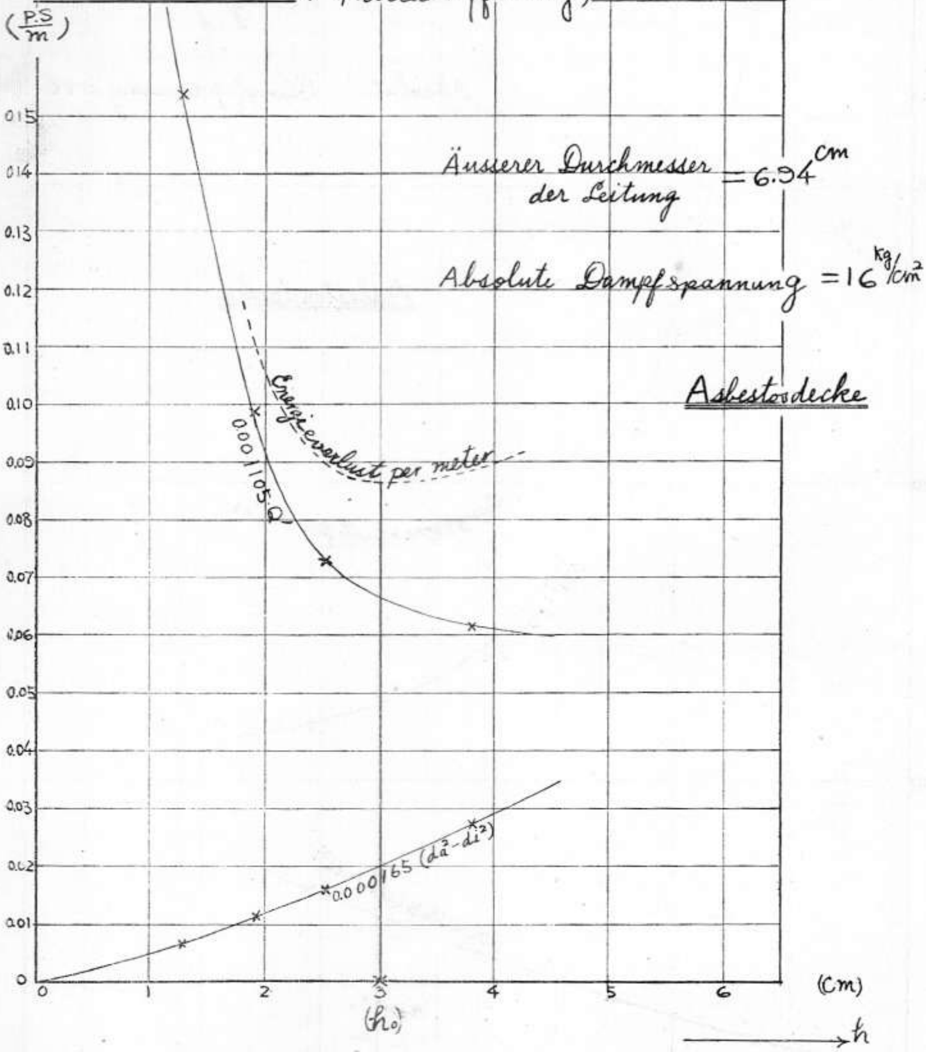


Fig. (3)

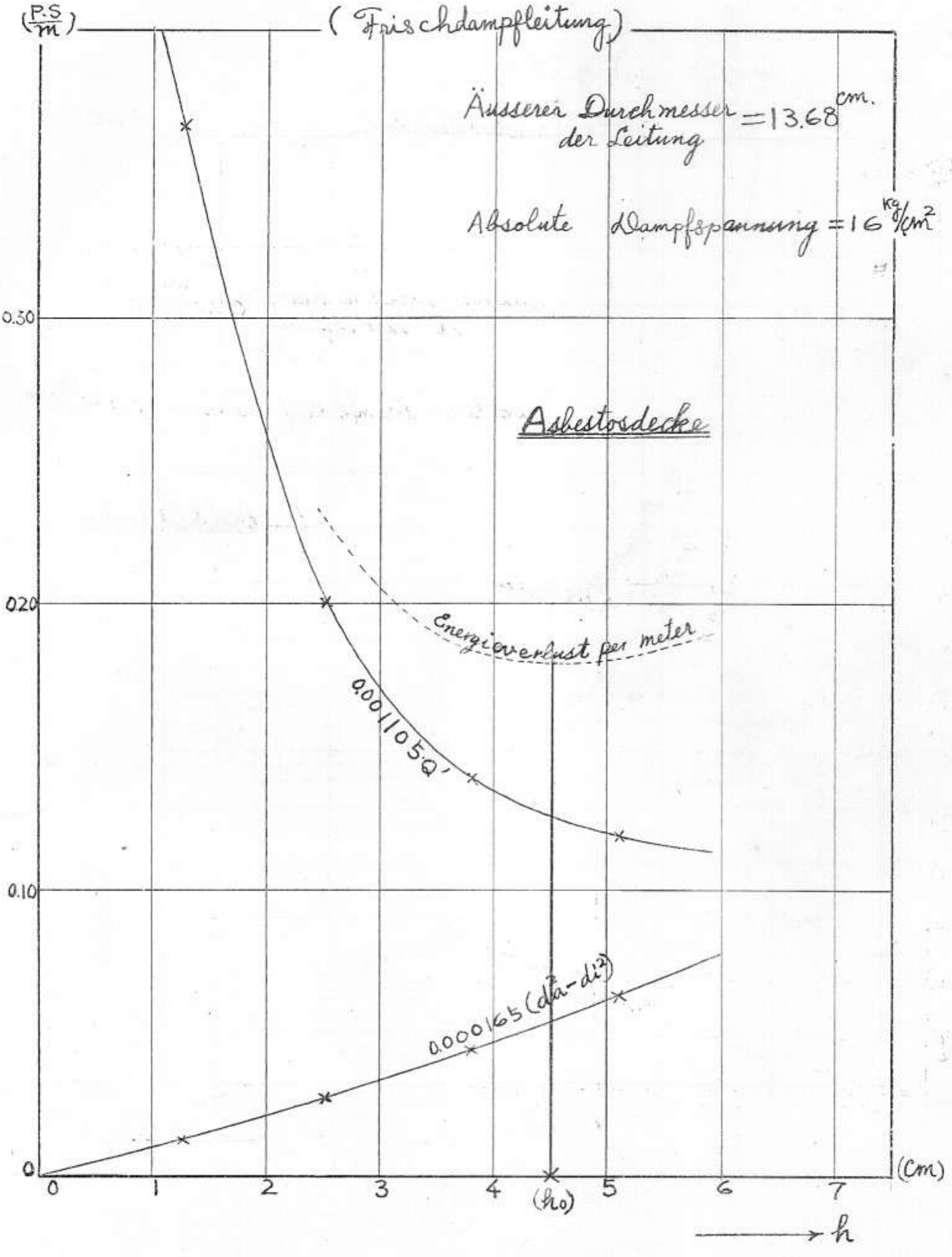

(Frischdampfleitung)
$(\frac{P.S}{m})$
Äusserer Durchmesser der Leitung $= 13.68^{cm.}$
Absolute Dampfspannung $= 16\ {}^{kg}/_{cm^2}$
Asbestosdecke
Energieverlust per meter
0.001105·Q'
$0.000165(d_a^2 - d_i^2)$
0.30
0.20
0.10
0
0
1
2
3
4
(h_0)
5
6
7
(cm)
→ h

Fig (4)

(Frischdampfleitung)

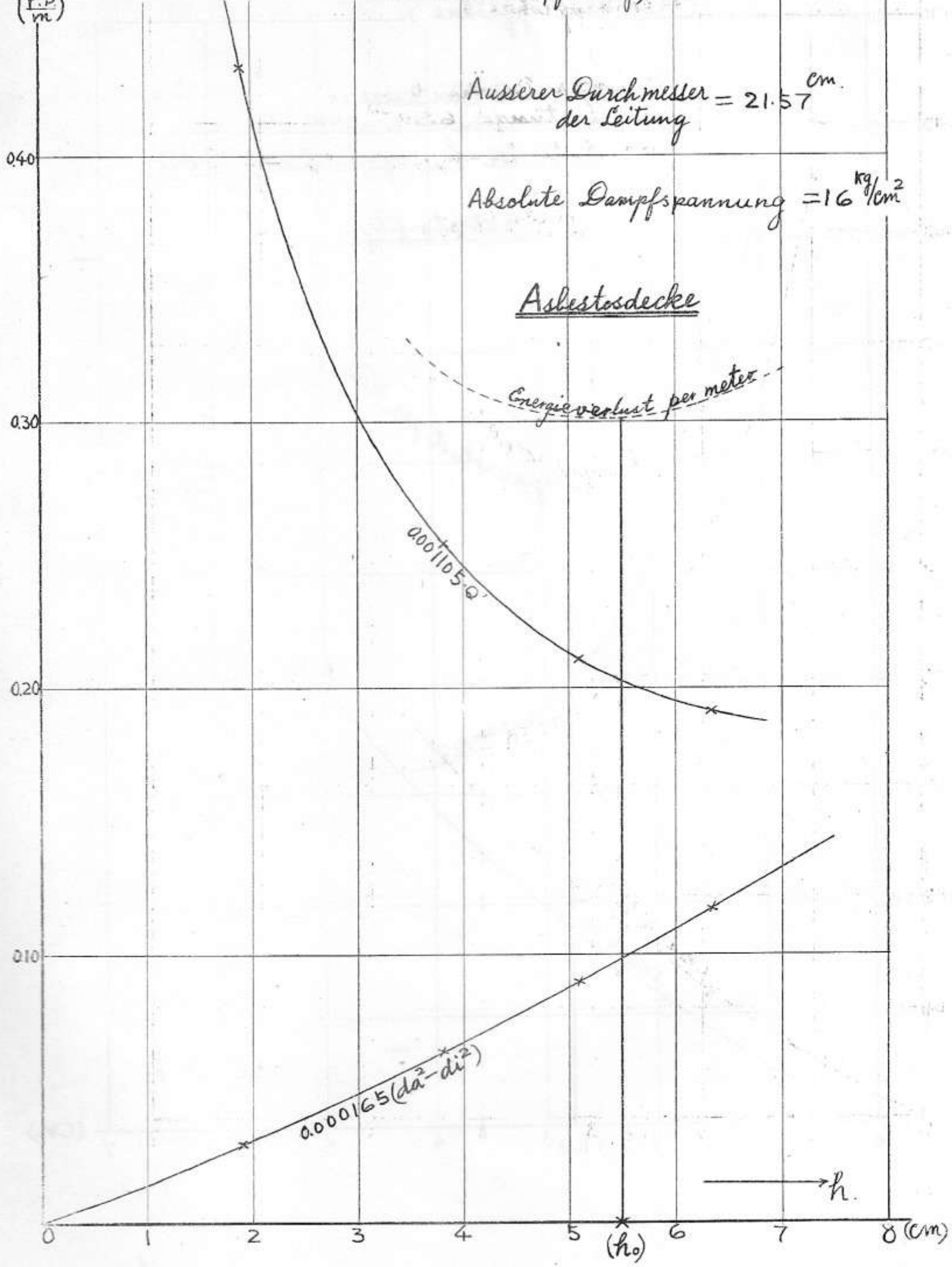
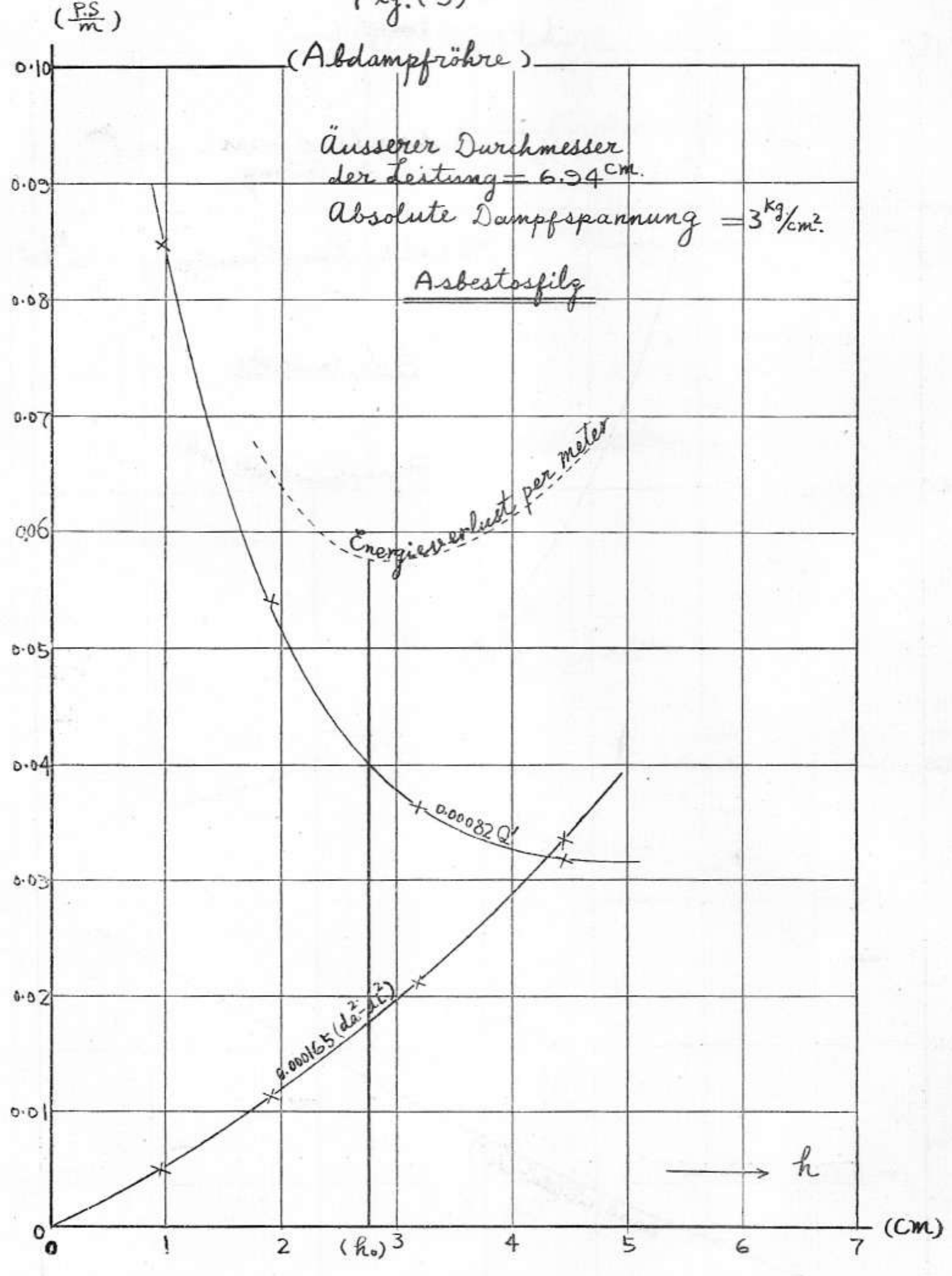


Fig. (5)


$(\frac{P.S}{m})$
(Abdampfröhre)
äusserer Durchmesser der Leitung = 6.94 cm.
absolute Dampfspannung = $3^{kg}/_{cm^2}$
Asbestosfilz
Energieverlust per meter
0.00082 Q'
0.000165 $(d_a^2 - d_i^2)$
0·10
0·09
0·08
0·07
0·06
0·05
0·04
0·03
0·02
0·01
0
0
1
2
(h_0) 3
4
5
6
7
(cm)
→ h

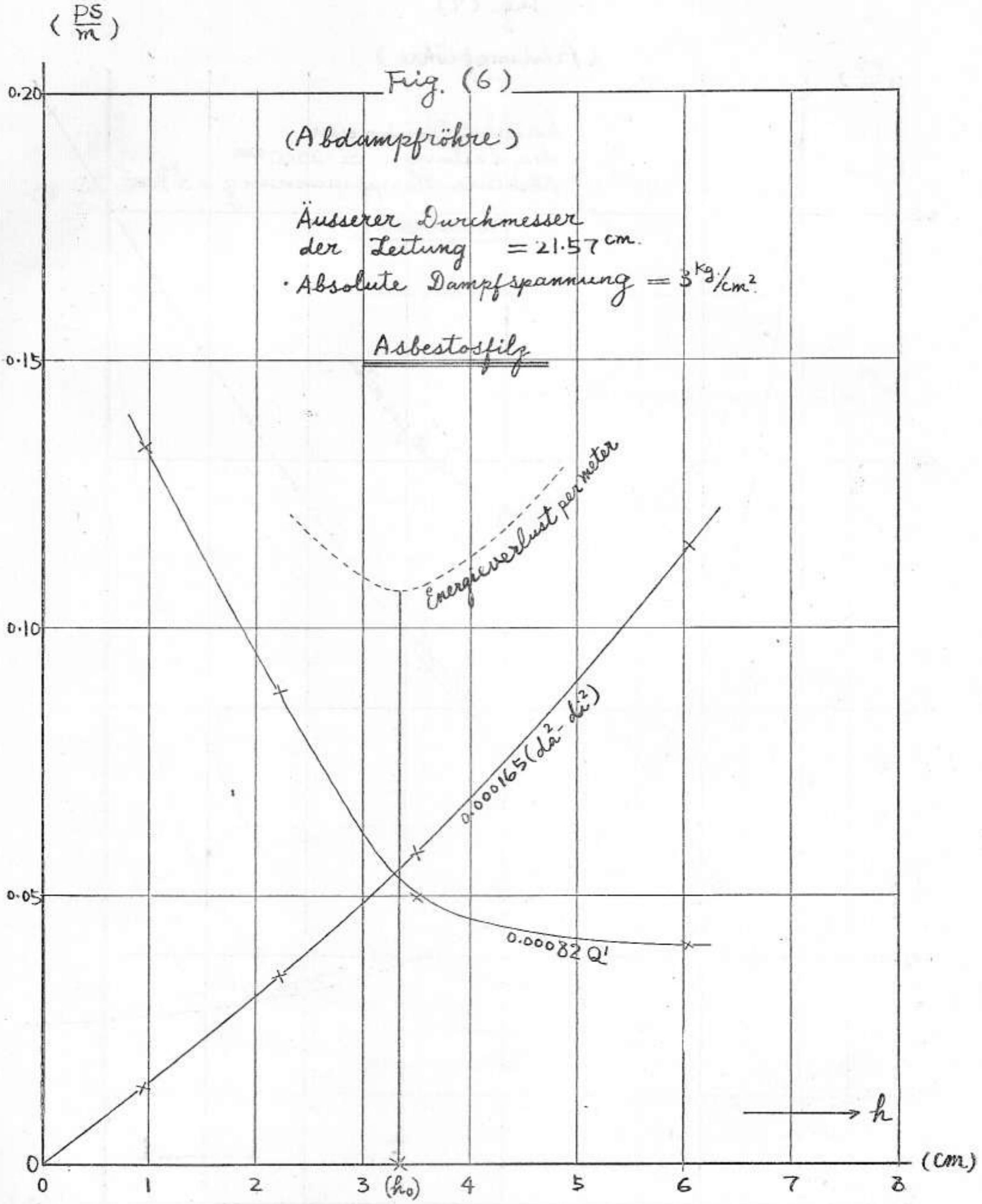


Fig. (6)

(PS/m)
(Abdampfröhre)
Äusserer Durchmesser der Leitung = 21.57 cm.
· Absolute Dampfspannung = 3 kg/cm²
Asbestosfilz
Energieverlust per meter
0.000165(da² - di²)
0.00082 Q'
0.20
0.15
0.10
0.05
0
1
2
3 (h₀)
4
5
6
7
8
(cm)
h

Fig. (7)

(Abdampfröhre)

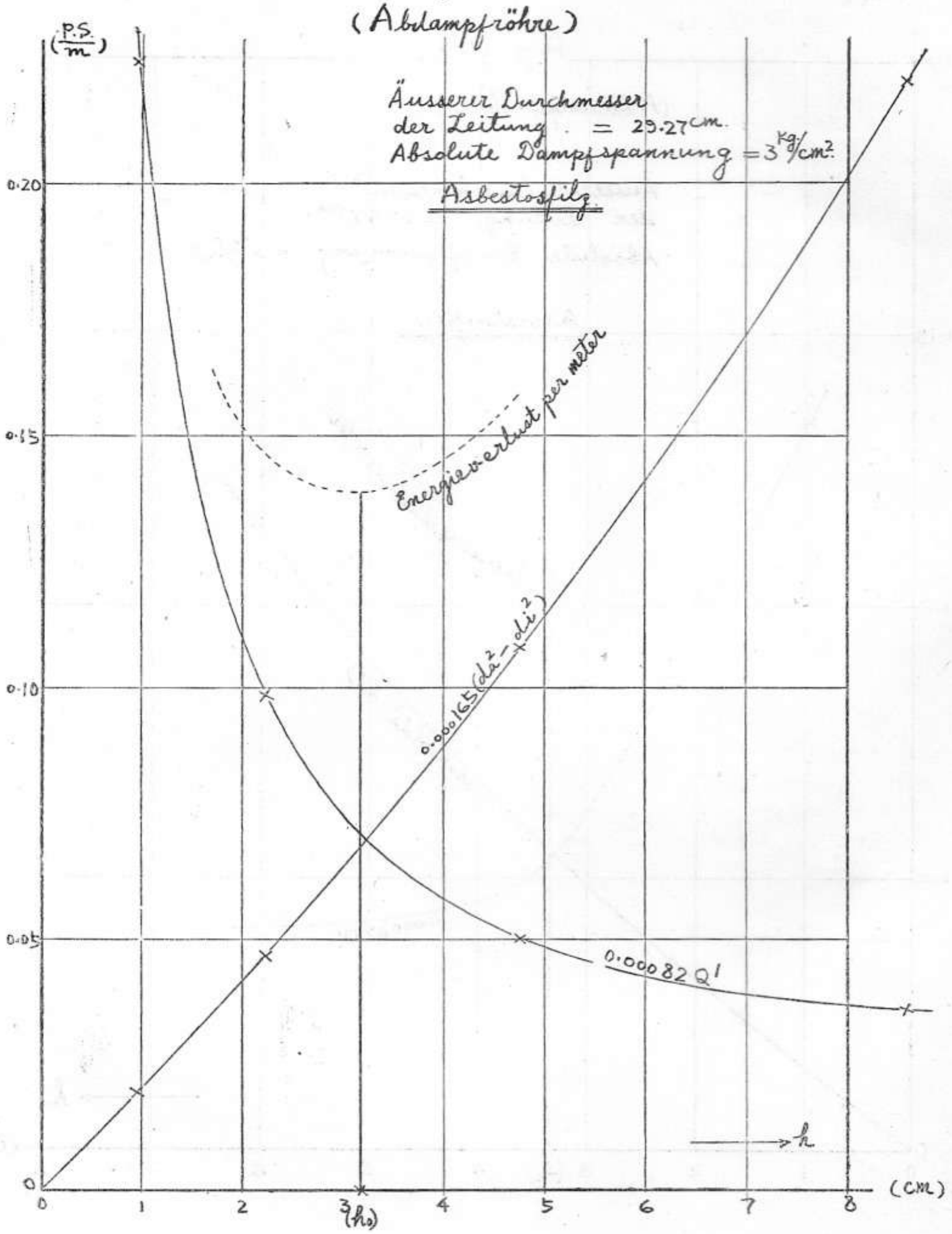


Fig. (8)

(Frischdampfleitung)

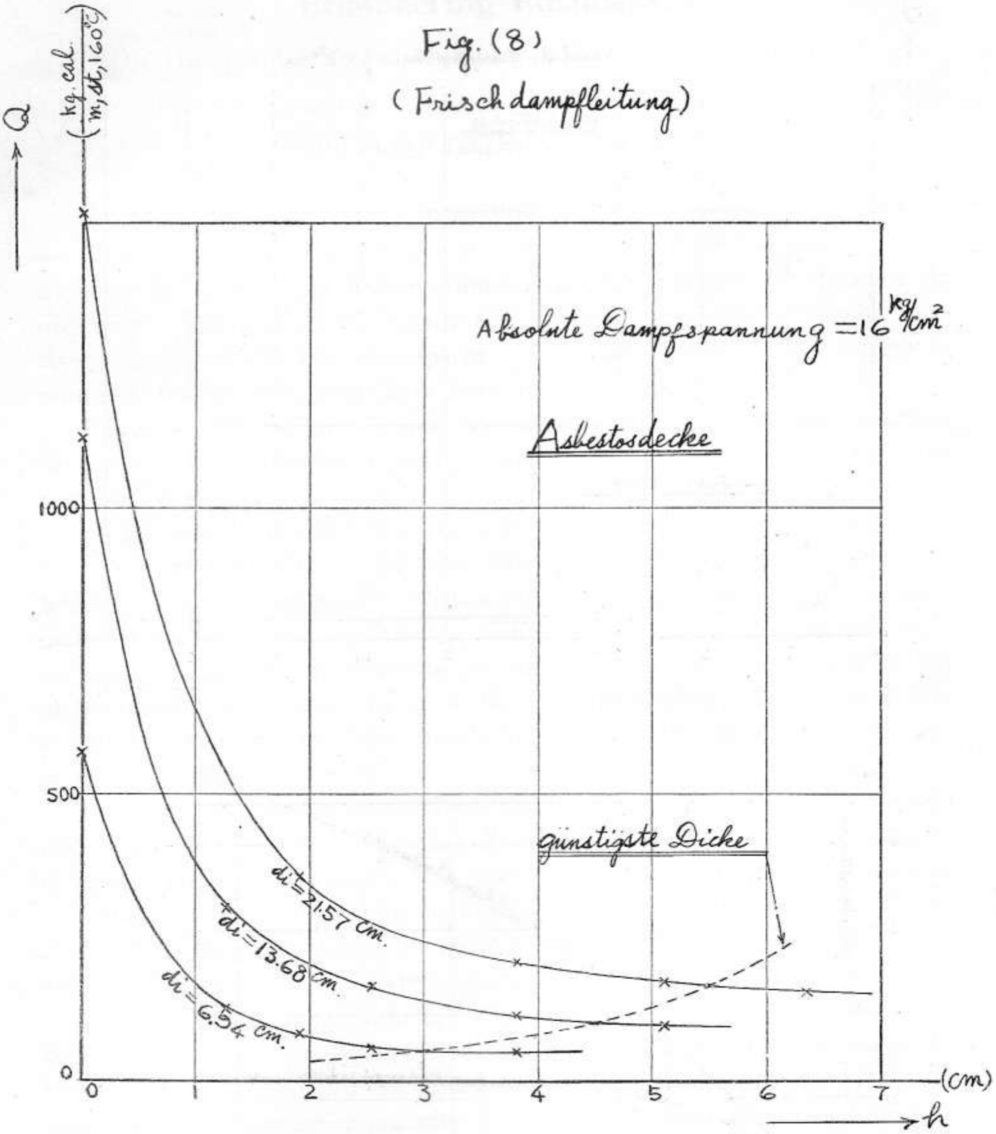
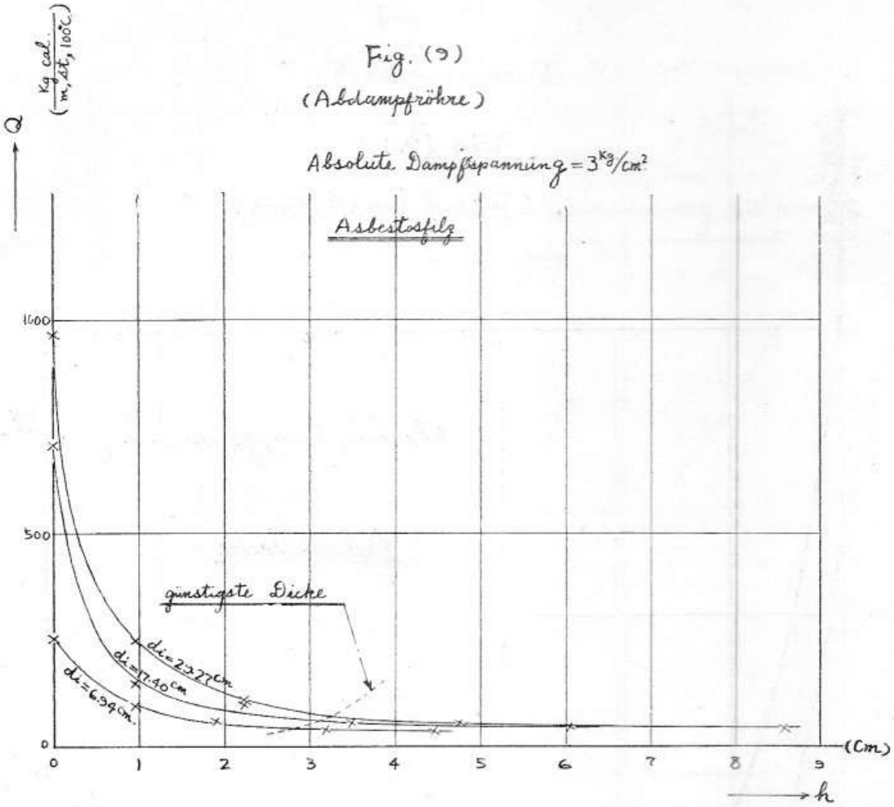
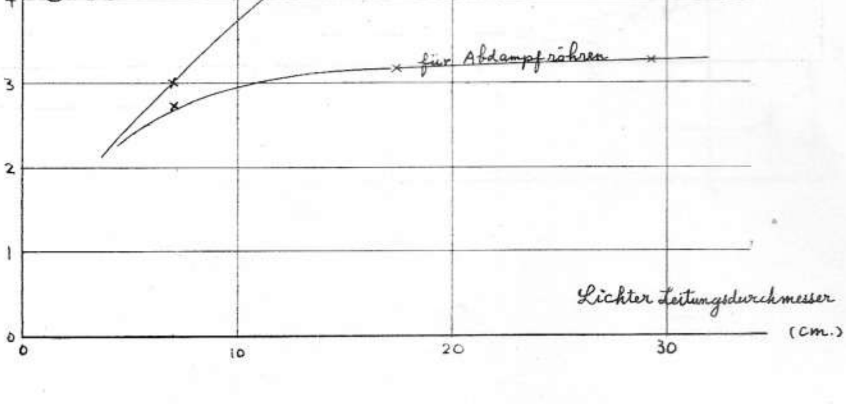


Fig. (9)

(Abdampfröhre)





Researches on Non-Ferrous Alloys for Marine Engineering Purposes.

(On the properties of the new alloy " Silzin Bronze ").

(Paper No. 613)

*By **Tokiji Ishikawa**, D. Eng.*

Introduction.

The recent development of manufacturing high tension bronzes, such as manganese-, nickel-, and aluminium-bronzes, has increased the demand for non-ferrous alloys as marine engineering materials; and they are now extensively employed for the manufacture of articles where great strength is required, for instance, propellers, turbine blades, and others.

However, the ordinary bronze (the so-called gun-metal) is still favoured as material for complicated castings such as steam or sea-water valves or varieties of pumps.

It is a well-known fact that the ordinary bronze not only has low tensile strength, but also contains large quantities of tin, which is rather expensive; besides, in bronze castings the phenomenon known as "inverse segregation" is liable to occur, and the casting lack the uniformity of structural constituents throughout; in consequence, unsound portions are usually found at the interior of the casting. It is also a matter of great difficulty to obtain a sound bronze casting which stands high hydrostatic pressures after it is properly machined.

In spite of these various deffects above mentioned, the ordinary bronze still predominates in many branches of casting industry, because of its facility of casting and of the comparatively great reliability for hydrostatic pressure provided the outer skin of the casting is not machined off, and especially of its considerable resistance to the corrosive action of sea-water; further, it is suitable for machine parts exposed to steam.

On the other hand, although the constituent metals of high tension bronzes are cheap, the articles made of these alloys may become expensive in the end, owing to the great difficulties of casting. This is the reason, at least up to the present time, why they have not been substituted for the ordinary bronze. Indeed, as no one has yet discovered an excellent alloy which deserves as a substitute for the ordinary bronze, so the latter, irrespective of the great loss both in weight and in cost, is still in general use.

Light aluminium alloys have recently received a considerable attention of the naval engineer for the weight saving of warships to comply with the Washington treaty, and their use is steadily increasing. But unfortunately they are inferior in tensile strength and liable to rapid corrosion in the usual service condition as compared with the ordinary bronze, and also are not suitable for the parts exposed to steam. So they cannot, of course, be substituted for bronze in anything and everything.

With the present state of things in view, an attempt is made to find some new copper-base alloys which meet the following requirements:—

(a) Mechanical properties should be 50 percent higher than those of gun-metal, viz.,

Tensile strength,	over 32 kg/mm^2,
Yield-point,	over 19 kg/mm^2,
Elongation,	not less than 10 percent.

The other properties determined by various dynamic tests should also be higher than those of gun-metal, at least by 50 percent or even more.

(b) The resistance to the corrosive actions of sea-water and steam should be higher than that of gun-metal.

(c) Facility of casting so as to give reliability of articles.

(d) Not to be inferior to the ordinary bronze in hardness, smallness of expansion-coefficient, and other physico-chemical properties.

(e) To avoid the use of expensive constituent metals such as tin, nickel, etc., and the alloys should be comparatively light and, at the same time, most economical for marine engineering purposes.

(I) Preliminary Experiments.

At the outset, preliminary experiments on a small scale were carried out with a view to ascertaining approximately the effect of several third elements on the alloys of copper and zinc.

The ternary alloys containing 10, 20, 30, 40 percent of zinc respectively with either silicon, or aluminium, or iron, or manganese, or antimony, or calcium, or cadmium, or lead, or phosphorus, or chromium, as a third element, were prepared for hardness and bending tests first of all.

The heats for this purpose were made in a small electric tube-furnace, in the laboratory, the weight of each charge being 70 grammes approximately. This quantity was found sufficient to allow one chill-casting (10 mm diameter, 20 mm long) for hardness test, and one sand-casting (5 mm diameter, 100 mm long) for bending test to be made. The test pieces for hardness test were polished smoothly after filing and tested by Vickers' diamond hardness tester. On the other hand, the test pieces for bending test were not machined and bent over a radius not greater than 5 mm.

The diagrams representing the compositions and the results of hardness and bending tests of the ternary alloys are given in Figs. 1 to 12, in which H10, H20, H30, H40 and A10, A20, A30, A40 denote the hardness-numbers and the bending angles for fracture of the alloys containing zinc 10, 20, 30, 40 percent respectively.

(II) Exploratory Heats.

The compositions of the alloys for exploratory heats were decided from the results obtained in the preliminary tests carried out for the purpose of a general exploration to determine the range of useful alloys. The heats for this purpose were made at the small foundry attached to the Metallurgical Department in the Naval Technical Research Institute, the weight of alloy made per heat being 9 kilogrammes approximately. This quantity was sufficient to allow three small chill-castings and three sand-castings for test pieces to be made. The casting moulds used in these exploratory heats are shown in Figs. 13 and 14.

The grades of the materials used for small heats were as follows:

Name.	*Grade.*	*Purity.* %
Copper	cathode copper	99.98
Tin	electrolytic tin	99.98
Manganese	metallic manganese	99.71
Zinc	electrolytic zinc	99.97
Silicon	metallic silicon	98.00
Nickel	nickel shot	99.75
Chromium	metallic chromium	96.14
Iron	low carbon iron	T.C., 0.08; Si, 0.06; P, 0.05; S, 0.033.

Method of melting and casting of the alloys:—The crucibles employed were of "Standard" No. 10 and the process of melting adopted was as follows:—

Copper was first melted under a layer of charcoal, next the third metal was added and heating was continued until the added metal was completely dissolved, and lastly zinc was added. The crucible was then taken out of the furnace and the molten alloy was mixed by stirring with a graphite rod. The crucible with its contents was then allowed to cool to the desired casting temperature, the surface being skimmed after a while. The chill-castings were first cast then the sand-castings.

Mechanical tests on exploratory alloys:—The castings were all machined to the dimensions of the standard tensile test piece, 50 mm in gauge length and 14 mm in diameter.

In tables or diagrams, the higher values of the duplicate tests are given; there were no considerable discrepancies between duplicate results. Where these discripancies did occur, they were found to be ascribed to small local defects in the castings.

(III) General Results of the Ternary Alloys Cu-Zn-X.

(1) *The alloys of ternary system with* 10 *percent of zinc.*

a) Cu–Zn–Si system. In Fig. 15 (Table 1), with increasing silicon and decreasing copper, a decided and steady rise in ultimate stress, yield-point, and hardness accompanied by a fall in shrinkage will be observed. With regard to elongation and ductility, they rise rapidly at first, pass a maximum point, and then fall with a further increase of silicon (above 2 percent in elongation, above 4 percent in ductility). The practical limit appears to lie in the neighbourhood of 5 percent of silicon where an elangation of 22 percent is still retained. The chill-castings of the same group of alloys have no distinguished difference from the sand-castings. The alloy with 5 percent of silicon appears to fulfil the required properties mentioned in the introduction and seems to suggest a field for further careful study.

b) Cu–Zn–Al system. In Fig. 16 (Table 1) it will be seen that the variation in mechanical properties of the alloys with increasing aluminium shows nearly a similar tendency to that of the alloys containing silicon except the fact that the shrinkage does not fall at all with increasing aluminium up to 5 percent, which leads to the difficulty of casting. The chill-castings of these alloys present always considerably better results than the sand-castings.

c) Cu–Zn–Fe system, Cu–Zn–Mn system and others.

The alloys were improved very slightly by increasing iron up to 5 percent, and also by increasing manganese up to 20 percent, as shown in Figs. 17 and 18 (Table 1).

The addition of nickel increases both the yield-point and ultimate stress, and decreases the elongation very slightly; tin and phosphorus also increase the ultimate stress but decrease the elongation rapidly. The addition of lead, or cadmium, or calcium decreases both the ultimate stress and elongation.

(2) *Ternary alloys with* 20 *percent of zinc.*

The effect of various third elements on the copper-zinc alloy with 20 percent of zinc was found to be qualitatively very similar to that of the alloy with 10 percent of zinc, but generally being more sensitive in increasing hardness.

a) Cu–Zn–Si system.

The addition of silicon to the Cu–Zn alloy with 20 percent of zinc gives so good mechanical properties that it nearly satisfies the riquirements in both the sand- and chill-cast conditions without much difference between them. The diagrams representing the curves of the ultimate stress, yield-point, elongation, hardness, and shrinkage of the sand-castings of these alloys are exhibited in Fig. 19 (Table 2).

b) Cu–Zn–Al system and others.

The results of the mechanical tests on sand-castings of the alloys with aluminium, or iron, or manganese are shown in Figs. 20, 21, and 22 (Table 2) respectively. The chill-castings of these alloys, however, present considerably higher results, and some alloys containing aluminium or manganese show such high strength and ductility as they easily satisfy the requirements except a rapid increase of solidification shrinkage.

(3) *Ternary alloys with* 30 *percent of zinc.*

In Figs. 23, 24, 25, and 26 (Table 3), the results of mechanical tests on sand- castings of the ternary alloy with silicon, or aluminium, or iron, or manganese respectively are shown. The chill-castings of these alloys give always higher results than the sand-castings and the difference between them is more remarkable than in the cases of the ternary alloys with 20 percent of zinc.

(4) *Ternary alloys with* 40 *percent of zinc.*

The addition of silicon, or aluminium, or iron, or manganese to the Cu–Zn alloy with 40 percent of zinc raises the ultimate stress and diminishes the elongation. In Figs. 27, 28, 29, and 30 (Table 4), the results of the mechanical tests on sand-castings of these alloys are shown. The chill-castings of the same group give somewhat higher results. The shrinkage of these alloys increases with the increasing percentage of the added metals.

(5) *The effect of aluminium and iron on the ternary alloys of* Cu–Zn–Si *system.*

The addition of aluminium or iron to the Cu–Zn alloys always increases their strength; hence, it may be expected to obtain better mechanical properties if they are added to the ternary alloys of Cu–Zn–Si system, so the quaternary alloys have been examined with this end in view. In Tables 5 and 6, the results of the tests on sand- and chill-castings of these alloys are given. It is to be seen that the addition of aluminium or iron produces a great difference between sand- and chill-castings in the results of tests. The properties of the sand-castings were rather spoiled by the addition of them, while the strength of the chill-castings were considerably improved.

(IV) Further Study of Selected Alloys.

Looking back upon the principal results obtained from the present investigation, the ternary alloys of Cu–Zn–Si system which have great advantages, chiefly consisting of a high strength and a low solidification shrinkage, may be considered as most useful casting alloys; this invite a more careful study of these selected alloys. In Tables 7 and 8, the results of the tests on these serviceable alloys of the same system are given. Among them, the alloys with less than 20 percent of zinc, which are called "SILZIN BRONZE" by the author, will be most suitable for an alternative to gunmetal, and alloys Nos. 11 and 17 are specially recommended for important parts.

The influence of pouring temperature:—The pouring of alloy No. 11

(Cu, 85; Zn, 10; Si, 5), was made at four different pouring temperatures. The results of mechanical tests on these castings are given in Table 9. Some of the specimens were cast at as low temperature as possible, e.g., 915°C which is only about 15°C above the freezing-point. Its mechanical properties are superior in every respect to those of the specimens cast at considerably higher temperatures.

Bars forged to 25 mm diameter:—The ingots for forging were 80 mm diameter and on an average 500 mm long and were forged to 25 mm by a 1/6-ton air hammer. The results of tests are given in Table 10. Up to 5.5 percent of silicon in the base alloys up to 15 percent of zinc, the mechanical properties are decidedly superior to those of the castings. The yield-point and ultimate stress reach very high values while the elongation and resilience are also higher than those of the chill-castings.

Specific gravity:—The specific gravities of the selected alloys were measured at the ordinary temperature on sand-castings, chill-castings, and bars forged to 25 mm. The results are given in grammes per cubic centimeter in Table 11.

Elastic modulus of the selected alloys:—These tests were carried out on turned specimens, 14 mm in diameter and with a parallel portion allowing 200 mm between gauge marks. The extensions of the specimens were read by Marten's extensometer and readings were taken at every increase of 500 kg in the actual load. The results of these tests are given in Table 12.

It is noteworthy that the elastic moduli of these alloys are much higher than the other ternary alloys of copper-zinc base.

Bending tests on the castings:—Test pieces for bending test were machined to the size 10 mm thick, 20 mm broad. The test piece was placed on two supports 80 mm apart and pressed down at the centre by a die of a radius equal to the thickness of the test piece. The test pieces of gun-mtal also were tested by the same manner. The results of these comparison tests are given in Table 13.

Repeated-bending impact tests on notched specimens:—

These were carried out on a machine specially designed for this purpose by Dr. T. Matsumura (Matsumura's alternating stress testing machine). The specimens were turned to 15 mm diameter with a circular grooved notch at the centre, the diameter at the bottom of the notch being 12 mm.

They were placed on knife-edges 120 mm apart and struck over the notch by a falling tup alternately at each end of a diameter. The reversals of the specimen between successive blows were performed by means of the motion of a link attached to the machine. The results of the test are given in Table 14.

The resistance of these ternary alloys to repeated-bending impact is remarkably high while that of gun-metal is very low.

Tensile tests at high temperatures:—Tensile tests at high temperatures were carried out on the sand-castings of alloy No. 11, gun-metal, and forged bars of the former. A small electric tube-furnace was used and the specimens were kept at a desired constant temperature during one hour so as to obtain a uniform distribution of heat along the length. The results of these tests are given in Tables 15 and 16. The tensile strength of these alloys appears to be well retained up to a temperature of about 300°C. The elongation of alloy No. 11 appears to decrease as temperature rises, attaining a minimum at a point between 400°C and 450°C, and then increases very rapidly.

Sea-water corrosion tests (single) :—The tests were carried out on alloys Nos. 11 and 17. Each alloy was cast into round bars of 60 mm diameter, 80 mm long. These were machined accurately to a disk form of 50 mm diameter, 5 mm thick, and a small hole was drilled at a point near the periphery. These specimens were subjected to corrosion by sea-water. Moreover, in order to compare the behaviour of these alloys with that of other copper alloys, each specimen of two kinds of gun-metal was prepared.

Each specimen was separately suspended in each beaker containing 500 c.c. of sea-water by means of a glass hook. In order to keep the specimens at a constant temperature of 30°C, these beakers were kept in a large common hot-water bath which was maintained at a desired temperature by an electric heater with an automatic regulator.

The specimens were taken out for examination and weighed at the interval of a week and the sea-water was changed at the same time.

The results of the experiments are tabulated in Table 17. Each figure denotes the loss of weight in grammes per square cm at the end of every 2 weeks.

In Fig. 31, the actual loss of weight of the specimens is plotted against the time of immersion. The rate of loss in weight for alloy No. 11 rapidly decreases after 4 weeks, and alloy No. 17 shows a very small rate of loss from the beginning, while two gun-metals appear to lose more steadily.

It is also to be seen that the amount of loss of alloys Nos. 11 and 17 is very small as compared with gun-metal.

The constitution and microstructure of the alloys :—

Although the author hopes in the near future to make the systematic study of the entire ternary system of Cu–Zn–Si, yet for the time being his investigation of the constitution of the alloys will be confined to the determination of the freezing points and the microstructures of a few useful alloys. In Figs. 32 and 33 the inverse rate cooling curves of the two series of the alloys with 10 percent and 15 percent of zinc are shown.

Also, the changes in the maximum solubility of silicon into the α-solution of the Cu–Zn base alloys with increasing content of zinc, were determined at the temperatures of 400°C and 800°C respectively. Those solubility curves are shown in Fig. 34.

The microstructures of a few useful alloys are shown in Figs. 35 to 46.

Conclusion.

From the results of the present investigation on the properties of ternary alloys produced by adding various third elements to the binary copper-zinc system, it has been shown that metallic silicon and aluminium have the most remarkable influence upon increasing the strength of the base alloys. But the addition of metallic aluminium results in a high solodification shrinkage, hence the castings would not be reliable, in other words, while the chill-castings bring about comparatively good tensile properties, the sand-castings show rather inconsistent results, and very seldom give excellent properties.

On the contrary, the addition of silicon decidedly reduces the shrinkage and increases the fluidity, except in the case of the base alloy of 60 : 40 brass, and gives very favourable properties for the production of good castings. Both the chill- and sand-castings also give excellent results under tensile or other dynamic tests without much difference between them. The facility of casting and high malleability are the most remarkable features of this kind of alloys. It may well be said that, in all respects, the ternary alloys obtained by adding metallic silicon to the copper-zinc alloys are the most serviceable bronze.

Those alloys whose compositions lie in the region of zinc less than 20 percent with from 4 to 5.5 percent of silicon are exceedingly suitable for both forging and casting purposes. Particularly the results of examination on various physical and chemical properties of these two alloys, viz., the one containing 10 percent of zinc, 5 percent of silicon, and the other 15 percent of zinc, 4.5 percent of silicon, show that they are exceedingly superior to gun-metal in strength, lightness, and resistances both to sea-water corrosion and to high pressure, etc.

They can, therefore, not only be substituted for gun-metal in naval design, but also will find a much wider field of practical application than gun-metal because of their high strength.

Taking into consideration the cheapness of contituent metals, the low density, and the high strength by using these new alloys we can save safely over 20 percent of weight and not less than 30 percent of manufacturing cost as compared with gun-metal castings.

The outstanding characteristics of the new alloy, "Silzin Bronze," may conveniently be summarized as follows:—

(a) Low solidification shrinkage and the facility of both casting and forging.

(b) High tensile strength and high elastic limit together with a good deal of elongation.

(c) High elastic modulus.

(d) High toughness and high resistance to shock.

(e) High resistance to hydraulic pressure.

(f) Stability when exposed to steam at high temperatures.
(g) Facility of machining and brightness of its highly finished surface.
(h) High resistance to sea-water corrosion (the cast valve-body and the forged valve-spindle of these alloys withstand better the corrosion).
(i) Relatively low density.
(j) Cheapness of the articles made of these alloys due to the substitution of silicon and zinc for expensive metals such as nickel or tin.

H 10, 20, 30, 40...Hardness Nos. of the Alloys with 10, 20, 30, 40%Zn.
A 10, 20, 30, 40...Bending Angles of the Alloys with 10, 20, 30, 40%Zn.

Fig. 1.

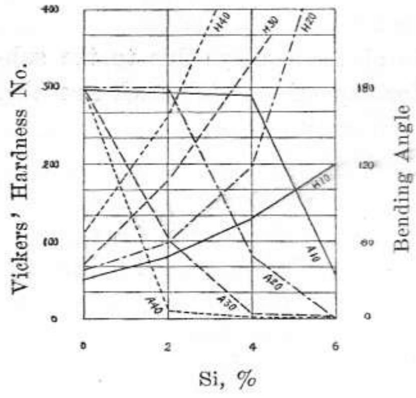


Fig. 4.

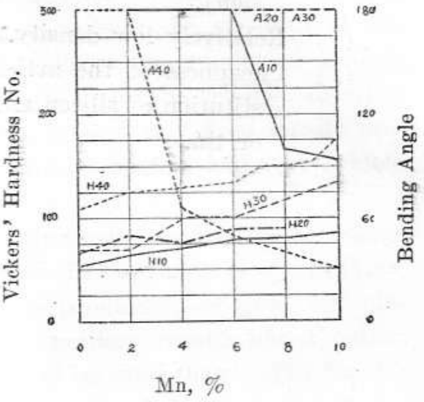


Fig. 2.

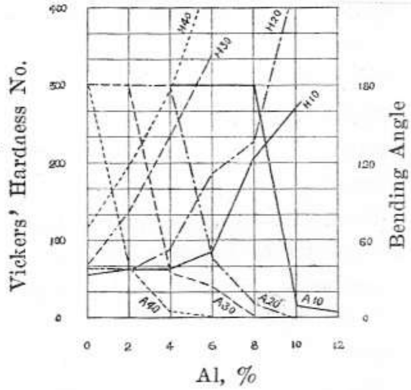


Fig. 5.

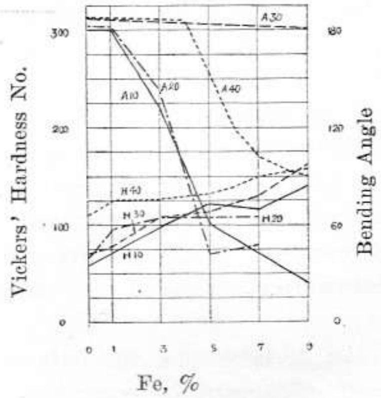


Fig. 3.

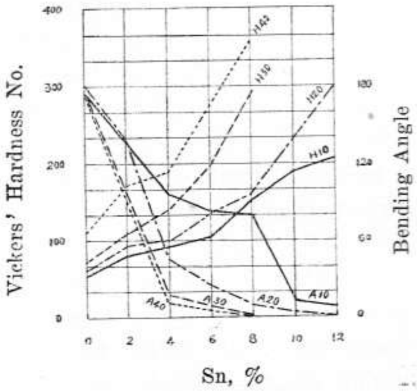


Fig. 6.

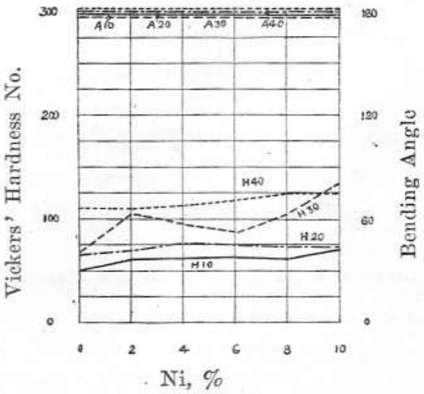


Fig. 7.

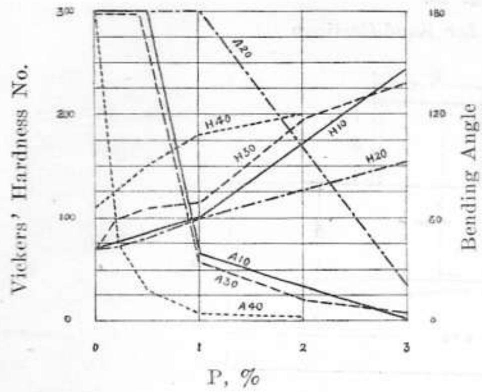


Fig. 10.

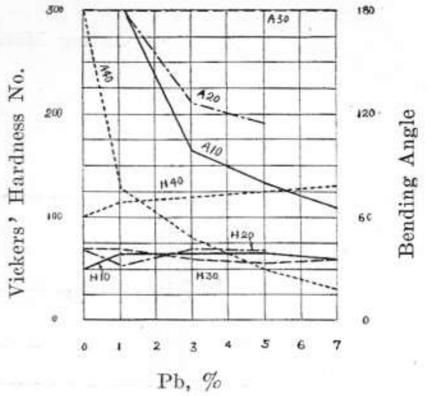


Fig. 8.

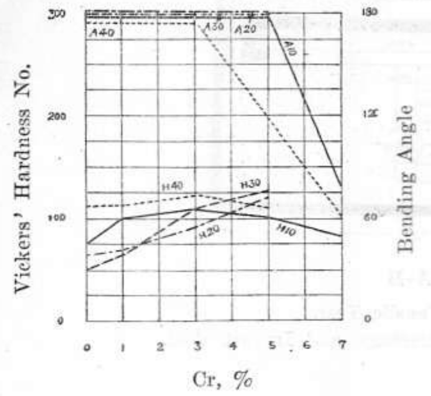


Fig. 11.

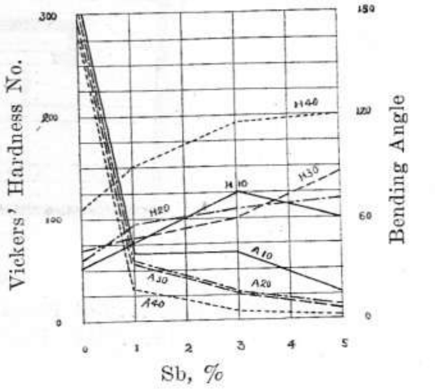


Fig. 9.

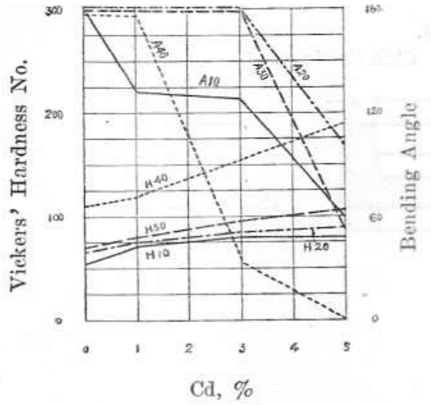


Fig. 12.

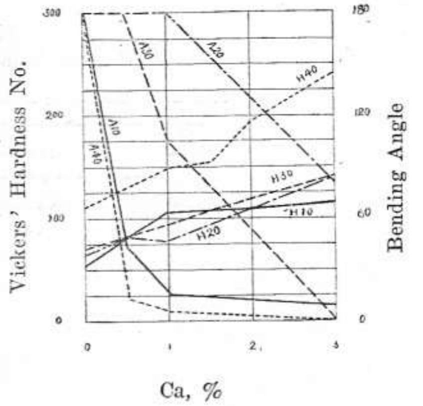
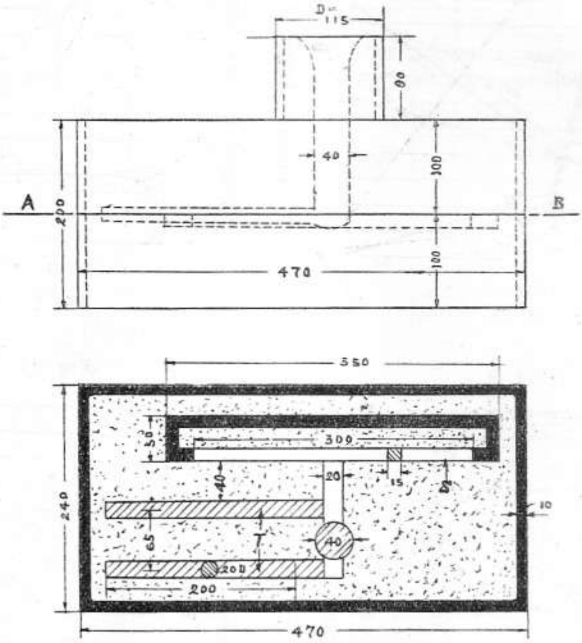


Fig. 13.
Casting Mould for Sand-Castings.



Section A–B

T... Test Pieces for Tensile Test.
S... Test Piece for Shrinkage and Impact Test.

Fig. 14.
Casting Mould for Chill-Castings.

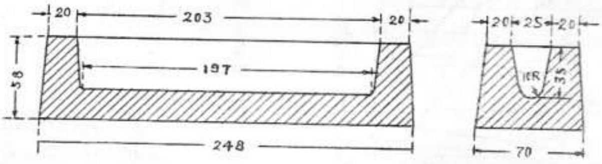


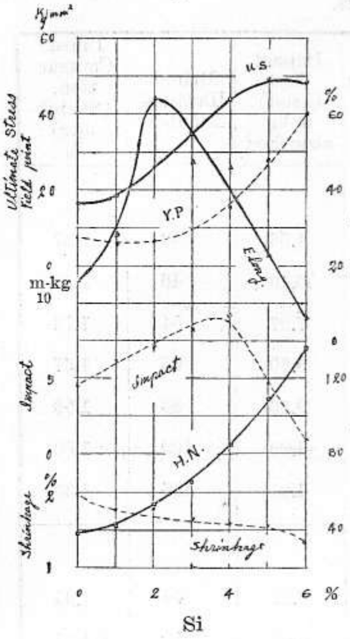
TABLE 1

Tests on Sand-Castings.

Composition. %			Yield-Point. kg/mm²	Ultimate Stress. kg/mm²	Elongation on 50 mm. %	Reduction of Area. %	Impact Test (Izod). m-kg absorbed	Brinell Hardness Number.	Linear Contraction. (Shrinkage) %
Cu	Zn								
		Si							
90	10	0	7.6	16.2	16.4	43.5	4.55	42	1.97
89	,,	1	5.5	18.4	28.2	46.2	6.16	46	1.78
88	,,	2	6.2	26.4	63.4	52.2	7.27	54	1.78
87	,,	3	9.4	34.7	47.4	52.2	9.20	65	1.67
86	,,	4	14.8	43.8	46.4	50.8	9.36	85	1.53
85	,,	5	28.2	49.0	22.6	22.4	4.90	109	1.50
84	,,	6	39.7	48.6	6.0	7.9	1.05	136	1.33
		Al							
89	10	1	5.5	22.6	31.0	45.2	6.17	44	1.95
87	,,	3	6.1	24.7	48.0	50.6	6.04	46	1.93
85	,,	5	7.7	24.3	50.0	52.0	6.64	49	1.90
83	,,	7	19.2	34.8	9.0	9.3	1.22	100	1.67
82	,,	8	31.3	41.3	1.4	2.0	0.25	143	1.52
81	,,	9	30.9	32.8	0.4	0.6	0.20	281	1.50
		Fe							
89	10	1	10.0	21.0	17.6	38.0	3.81	51	1.93
87	,,	3	11.2	25.9	15.0	34.6	3.25	57	2.10
85	,,	5	11.8	29.8	24.8	35.9	2.86	65	2.11
83	,,	7	12.1	28.0	15.2	21.2	2.60	69	1.93
80	,,	10	12.8	22.2	6.8	12.9	2.27	66	2.00
		Mn							
86	10	4	8.4	20.5	18.6	31.5	5.74	52	1.83
80	,,	10	9.9	24.9	23.0	32.0	6.00	54	1.60
75	,,	15	10.8	29.9	36.0	39.5	6.04	57	1.77
70	,,	20	13.1	31.5	23.4	29.7	3.60	61	1.77

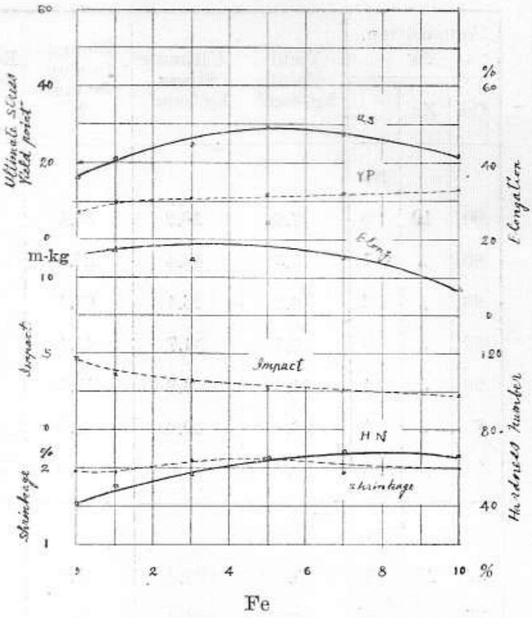
Alloys with 10 percent Zinc. (See Table 1).
Sand-castings.

Fig. 15.



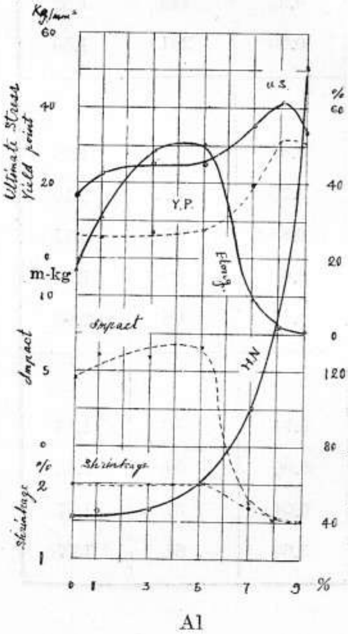
Alloys with 10 percent Zinc. (See Table 1).
Sand-castings.

Fig. 17.



Alloys with 10 percent Zinc. (See Table 1).
Sand-castings.

Fig. 16.



Alloys with 10 percent Zinc. (See Table 1).
Sand-castings.

Fig. 18.

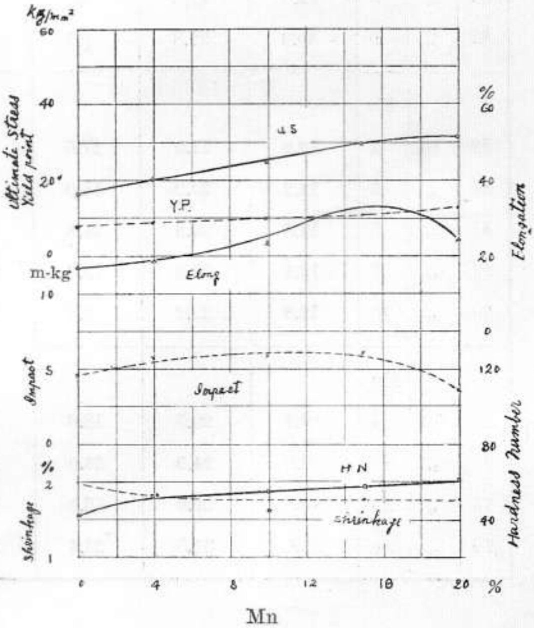


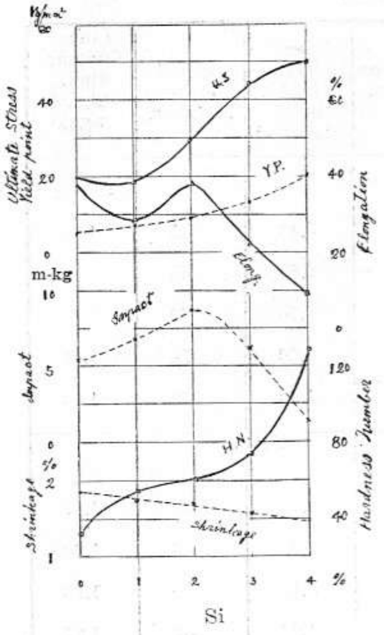
TABLE 2

Tests on Sand-Castings.

Composition. %			Yield-Point. kg/mm²	Ultimate Stress. kg/mm²	Elongation on 50 mm. %	Reduction of Area. %	Impact Test (Izod). m-kg absorbed	Brinell Hardness Number.	Linear Contraction. (Shrinkage) %
Cu	Zn								
		Si							
80	20	0	5.1	19.3	38.0	52.5	5.43	31	1.82
79	,,	1	6.9	18.4	28.0	34.9	6.80	55	1.72
78	,,	2	9.2	29.6	38.0	44.9	8.70	60	1.70
77	,,	3	12.7	43.9	21.6	33.4	6.20	73	1.53
76	,,	4	20.4	49.6	9.6	10.8	1.45	128	1.43
		Al							
79	20	1	5.7	18.7	36.0	51.0	6.43	38	1.77
77	,,	3	7.0	20.7	33.2	52.0	7.30	43	1.90
76	,,	4	9.0	30.1	26.0	32.5	8.48	50	1.78
75	,,	5	20.1	42.4	8.0	18.4	4.90	89	1.67
74	,,	6	27.6	43.8	8.2	19.6	1.40	128	1.43
		Fe							
79	20	1	9.7	19.8	15.0	31.3	3.85	58	1.95
77	,,	3	11.5	24.7	12.6	23.5	3.95	68	2.00
75	,,	5	11.5	22.9	8.8	13.7	1.90	56	2.03
70	,,	10	11.9	23.1	7.0	13.0	2.15	56	2.05
		Mn							
76	20	4	7.6	20.4	29.0	30.8	3.67	44	1.77
72	,,	8	9.7	22.4	30.0	36.0	4.67	64	1.72
70	,,	10	11.7	26.2	32.0	32.4	5.05	61	1.76
65	,,	15	14.6	27.0	14.0	22.2	6.04	70	1.78
60	,,	20	17.2	27.8	11.0	21.6	4.65	85	1.93

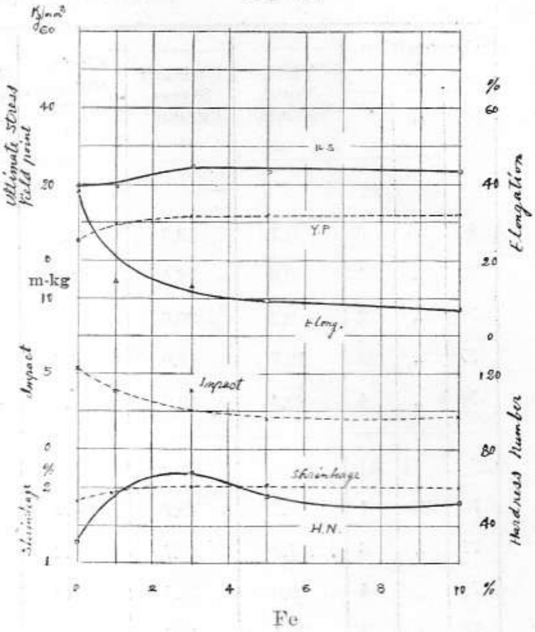
Alloys with 20 percent Zinc.
Sand-castings.

Fig. 19.



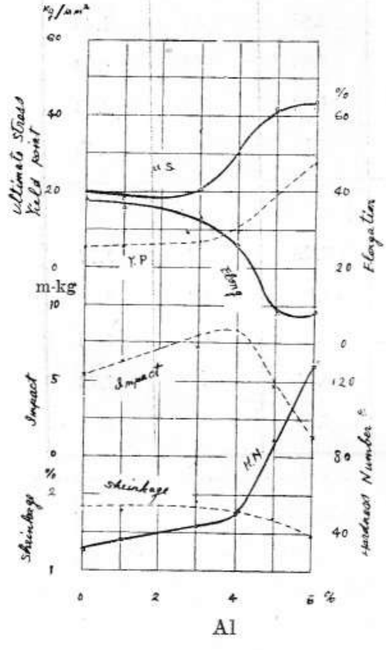
Alloys with 20 percent Zinc.
Sand-castings.

Fig. 21.



Alloys with 20 percent Zinc.
Sand-castings.

Fig. 20.



Alloys with 20 percent Zinc.
Sand-castings.

Fig. 22.

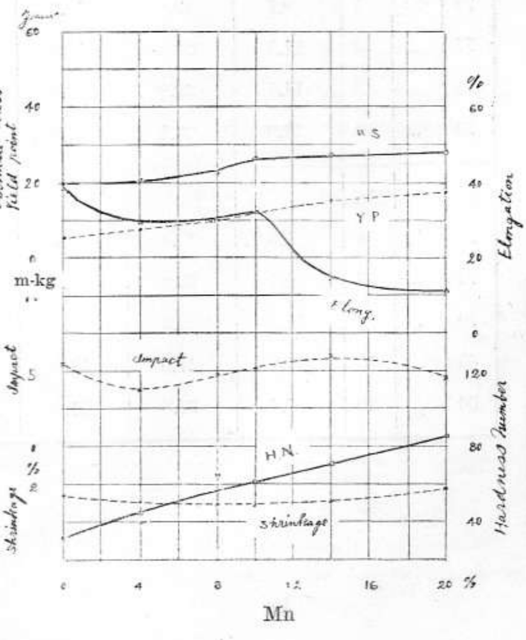
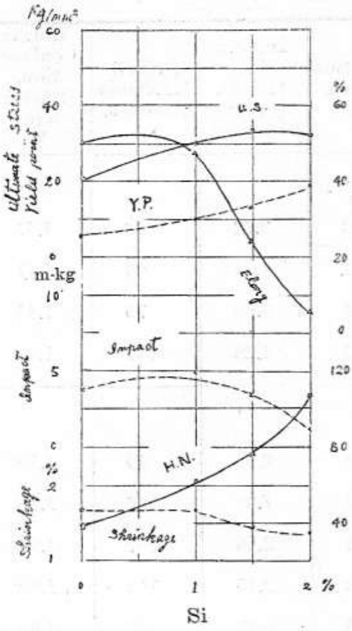


TABLE 3

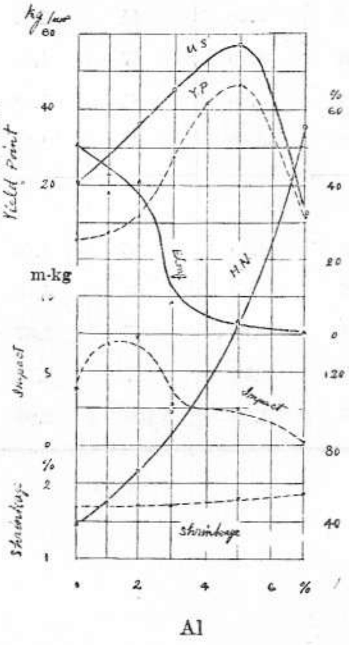
Tests on Sand-Castings.

Composition. %			Yield-Point. kg/mm²	Ultimate Stress. kg/mm²	Elongation on 50 mm. %	Reduction of Area. %	Impact Test (Izod). m-kg absorbed	Brinell Hardness Number.	Linear Contraction. (Shrinkage) %
Cu	Zn								
		Si							
70	30	0	5.6	20.7	50.8	50.3	3.72	42	1.67
69	,,	1	9.3	30.3	48.0	54.7	4.95	63	1.67
68.5	,,	1.5	12.5	33.9	24.0	28.4	3.60	76	1.43
68	,,	2	18.6	32.1	5.6	9.3	1.25	107	1.33
		Al							
69	30	1	7.5	22.4	38.0	40.0	6.90	49	1.70
68	,,	2	10.4	36.3	40.6	67.0	7.30	67	1.70
67	,,	3	27.1	44.3	8.0	16.4	2.46	107	1.70
65	,,	5	46.6	56.7	2.6	7.3	2.45	144	1.80
63	,,	7	11.0	11.5	0.2	2.8	0.15	251	1.83
		Fe							
69	30	1	10.7	28.5	27.6	31.0	1.37	64	1.67
65	,,	5	12.6	27.4	12.0	25.0	3.20	41	1.77
63	,,	7	11.8	25.9	20.0	21.6	4.90	67	1.60
61	,,	9	12.8	35.2	26.0	31.1	5.60	81	1.58
59	,,	11	14.9	38.5	41.2	43.3	5.80	75	1.57
		Mn							
66	30	4	8.3	23.5	41.0	43.8	4.35	46	1.75
64	,,	6	9.5	22.9	26.6	39.6	5.85	54	1.70
60	,,	10	13.9	29.1	15.0	25.4	3.81	65	1.67
56	,,	14	19.5	32.7	5.6	12.7	1.40	87	1.77
54	,,	16	23.4	30.7	3.0	4.3	1.30	100	2.00

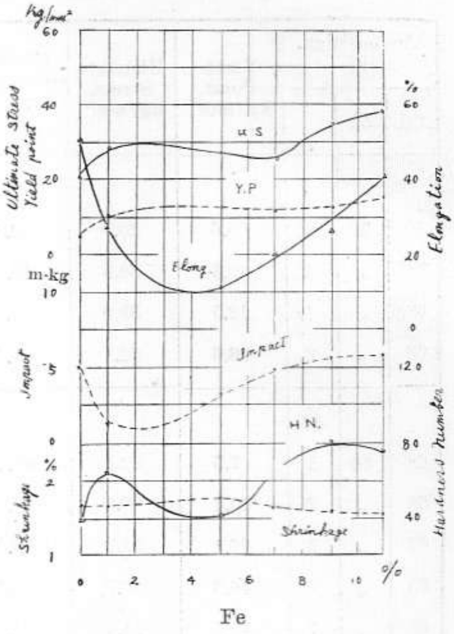
Alloys with 30 percent Zinc.
Sand-castings.
Fig. 23.



Alloys with 30 percent Zinc.
Sand-castings.
Fig. 24.



Alloys with 30 percent Zinc.
Sand-castings.
Fig. 25.



Alloys with 30 percent Zinc.
Sand-castings.
Fig. 26.

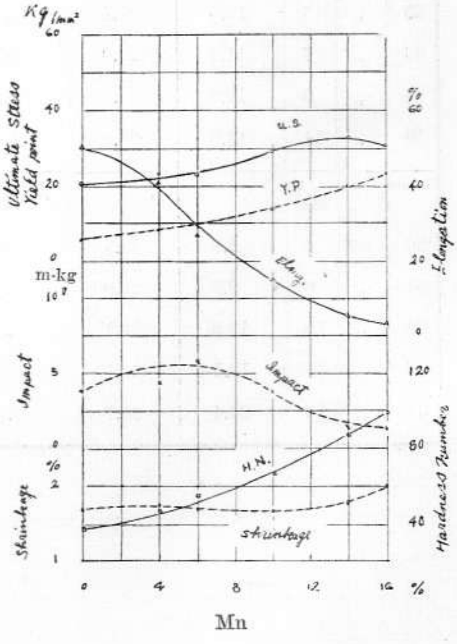


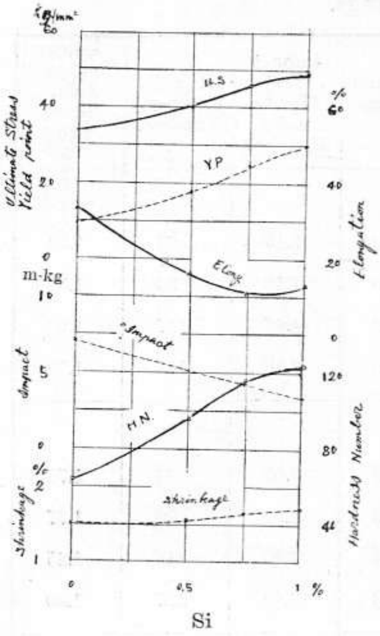
TABLE 4

Tests on Sand-Castings.

Composition. %			Yield-Point. kg/mm²	Ultimate Stress. kg/mm²	Elongation on 50 mm. %	Reduction of Area. %	Impact Test (Izod). m-kg absorbed	Brinell Hardness Number.	Linear Contraction. (Shrinkage) %
Cu	Zn								
		Si							
60	40	0	9.3	33.8	33.0	35.0	7.10	63	1.52
59.5	"	0.5	17.5	40.6	16.0	30.0	5.10	96	1.56
59.25	"	0.75	24.7	46.3	11.0	19.6	4.45	117	1.66
59	"	1	29.6	49.0	13.0	22.2	3.46	123	1.70
		Al							
59.5	40	0.5	16.0	40.3	22.6	23.9	5.00	65	1.70
59	"	1	23.6	48.7	13.5	16.2	3.90	98	1.77
58.5	"	1.5	25.1	50.0	8.6	25.5	4.20	114	1.85
58	"	2	29.8	52.2	7.8	10.3	3.80	117	1.90
		Fe							
58	40	2	13.3	41.7	35.0	36.6	8.20	78	1.67
56	"	4	13.2	42.0	22.0	26.4	6.60	80	1.67
52	"	8	15.6	46.1	16.0	22.8	5.10	88	1.73
50	"	10	15.2	49.4	15.0	19.2	4.90	96	1.83
48	"	12	14.4	41.0	11.0	7.3	4.20	105	1.70
46	"	14	16.5	47.0	14.0	11.3	4.70	100	1.67
		Mn							
58	40	2	11.4	37.0	23.2	29.3	7.10	68	1.48
56	"	4	17.3	39.2	17.0	19.9	4.27	84	1.67
54.5	"	5.5	19.9	44.9	15.2	19.6	3.60	100	1.70
53	"	7	29.3	46.0	7.8	12.6	2.55	110	1.89
50	"	10	32.0	39.6	3.4	3.1	1.90	136	1.93
48	"	12	29.9	32.6	2.0	2.6	1.35	140	2.16
46	"	14	23.7	25.0	1.6	—	0.46	193	2.50

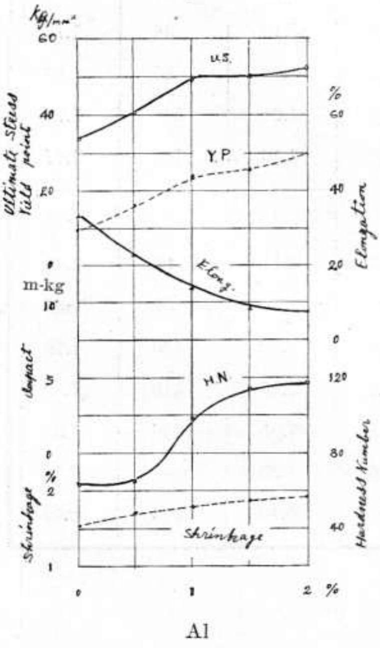
Alloys with 40 percent Zinc.
Sand-castings.

Fig. 27.



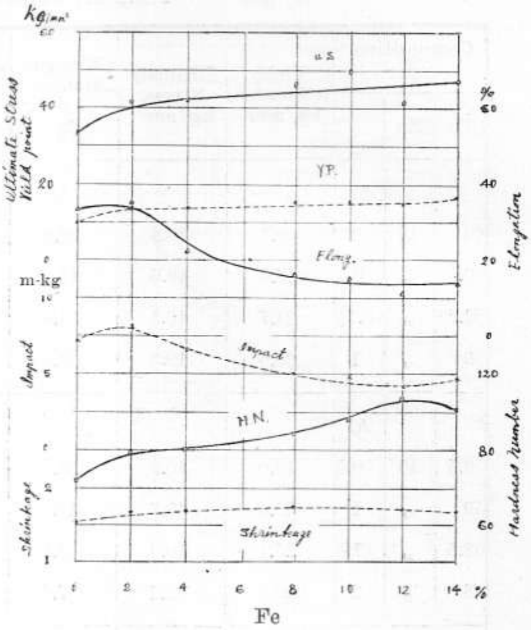
Alloys with 40 percent Zinc.
Sand-castings.

Fig. 28.



Alloys with 40 percent Zinc.
Sand-castings.

Fig. 29.



Alloys with 40 percent Zinc.
Sand-castings.

Fig. 30.

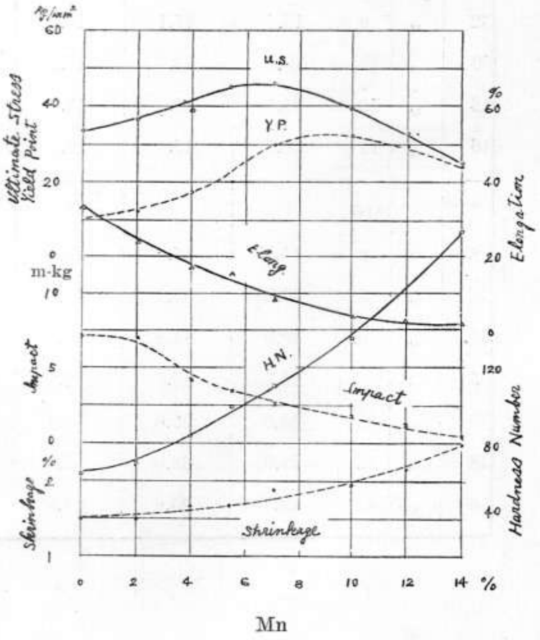


TABLE 5 Tests on Sand-Castings.

Composition. %				Yield-Point. kg/mm²	Ultimate Stress. kg/mm²	Elongation on 50 mm. %	Reduction of Area. %	Impact Test (Izod). m-kg absorbed	Brinell Hardness Number.	Linear Contraction (Shrinkage) %
Cu	Zn	Si								
			Al							
85.5	10	4	0.5	16.9	38.3	24.6	34.3	5.30	84	1.66
85	,,	,,	1	16.6	42.6	23.0	30.8	5.74	96	1.66
84.5	,,	,,	1.5	17.8	44.7	19.0	22.2	5.20	102	1.56
84	,,	,,	2	17.0	39.0	8.8	17.1	3.35	112	1.55
85	,,	4.5	0.5	14.9	33.1	11.0	21.1	4.06	80	1.50
84.5	,,	,,	1	29.4	43.5	6.6	15.1	3.50	145	1.50
84	,,	,,	1.5	31.8	41.1	6.0	12.4	3.80	130	1.44
83.5	,,	,,	2	24.4	42.6	7.4	14.4	2.90	116	1.33
84.5	,,	5	0.5	20.6	44.2	6.8	11.1	2.20	122	1.66
84	,,	,,	1	27.7	44.5	6.6	11.1	3.00	122	1.63
			Fe							
85	10	5	0	25.0	47.0	15.4	19.0		123	
84.5	,,	,,	0.5	30.9	52.6	13.0	11.1		129	
84	,,	,,	1	30.5	49.7	13.6	11.1		122	
83.5	,,	,,	1.5	27.1	46.1	11.0	8.6		109	
83	,,	,,	2	26.7	40.4	5.6	4.8		102	

TABLE 6 Tests on Chill-Castings

Composition. %				Yield-Point. kg/mm²	Ultimate Stress. kg/mm²	Elongation on 50 mm. %	Reduction of Area. %	Impact Test (Izod). m-kg absorbed	Brinell Hardness Number.
Cu	Zn	Si							
			Al						
85.5	10	4	0.5	15.9	51.0	55.0	42.0	7.75	106
85	,,	,,	1	18.5	53.8	36.6	31.6	6.43	100
84.5	,,	,,	1.5	18.6	53.5	35.4	31.1	6.15	114
84	,,	,,	2	25.0	54.9	25.6	29.7	5.05	116
85	,,	4.5	0.5	17.3	50.5	47.8	30.7	7.10	102
84.5	,,	,,	1	34.7	57.4	24.0	26.5	4.35	133
84	,,	,,	1.5	30.0	57.4	20.0	23.1	4.10	130
83.5	,,	,,	2	22.0	57.0	22.0	25.5	4.15	128
84.5	,,	5	0.5	32.9	54.7	16.0	17.0	2.73	135
84	,,	,,	1	31.5	56.8	18.0	19.6	3.45	132
			Fe						
85	10	5	0	26.4	46.1	12.0	13.4		114
84.5	,,	,,	0.5	33.1	57.4	22.0	18.5		133
84	,,	,,	1	30.2	54.2	17.0	12.6		128
83.5	,,	,,	1.5	29.2	50.5	8.0	7.8		122
83	,,	,,	2	29.3	45.9	6.4	5.5		114

TABLE 7

Tests on Sand-Castings of Selected Alloys.

Alloy No.	Composition. %			Yield-Point. kg/mm²	Ultimate Stress. kg/mm²	Elongation on 50 mm. %	Reduction of Area. %	Impact Test (Izod). m-kg absorbed	Brinell Hardness Number.	Linear Contraction (Shrinkage) %
	Cu	Zn	Si							
1	93	3	4	14.4	31.2	32.0	26.5	7.43	74	1.73
2	92	„	5	18.0	42.2	22.4	32.4	2.20	117	1.66
3	92	4	4	16.3	36.2	29.0	24.1	6.93	79	1.66
4	91	„	5	21.3	41.0	20.6	25.8	4.60	95	1.60
5	91.5	5	4.5	16.7	40.7	21.2	12.4	4.75	100	1.55
6	90	„	5	17.5	44.2	22.0	30.5	2.40	104	1.53
7	89	„	6	27.6	39.8	5.6	8.1	2.30	121	1.43
8	87.5	7.5	5	21.3	46.3	24.0	27.9	2.70	109	1.53
9	86	10	4	14.8	43.8	46.4	50.8	9.30	85	1.55
10	85.5	„	4.5	18.3	45.7	31.0	25.0	6.95	94	1.50
11	85	„	5	28.2	49.0	22.6	22.4	4.90	109	1.50
12	84.5	„	5.5	32.0	51.7	12.4	16.1	2.35	136	1.50
13	83.5	12.5	4	17.7	45.6	35.4	31.9	6.30	94	1.43
14	82.5	„	5	34.2	54.9	16.0	20.9	3.95	120	1.43
15	82	„	5.5	32.7	44.6	4.6	3.8	1.26	135	1.20
16	81	15	4	19.9	50.4	27.0	35.9	6.55	100	1.43
17	80.5	„	4.5	31.3	53.2	22.8	19.4	5.25	122	1.33
18	80	„	5	34.0	50.5	11.0	14.0	3.05	124	1.30
19	77	20	3	12.7	41.9	44.0	35.1	7.55	73	1.53
20	76	„	4	28.4	49.6	9.6	10.8	1.45	128	1.43
21	68.5	30	1.5	12.7	34.3	23.6	30.0	3.60	77	1.43
22	68	„	2	18.6	32.5	6.0	10.6	1.28	107	1.33
23	59.5	40	0.5	17.4	40.1	16.0	30.1	5.10	96	1.56
24	59.25	„	0.75	24.7	46.3	11.0	19.6	4.45	110	1.66
25	59	„	1	29.6	48.7	13.0	22.2	3.46	123	1.70

TABLE 8

Tests on Chill-Castings of Selected Alloys.

Alloy No.	Composition. %			Yield-Point. kg/mm^2	Ultimate Stress. kg/mm^2	Elongation on 50 mm. %	Reduction of Area. %	Impact Test (Izod). m-kg absorbed	Brinell Hardness Number.
	Cu	Zn	Si						
1	93	3	4	12.1	42.2	54.0	42.5	9.40	78
2	92	„	5	20.1	40.9	15.4	19.8	3.00	109
3	92	4	4	14.9	43.8	51.0	35.0	7.95	77
4	91	„	5	19.4	46.5	25.6	31.0	4.70	100
5	90.5	5	4.5	15.8	40.6	31.0	19.1	6.97	87
6	90	„	5	20.7	43.6	18.0	20.3	3.25	109
7	89	„	6	21.0	33.2	11.0	10.8	3.26	114
8	87.5	7.5	5	21.4	46.3	23.4	20.3	4.24	107
9	86	10	4	14.7	45.6	51.2	44.1	9.70	87
10	85.5	„	4.5	17.9	50.3	36.1	37.1	7.60	99
11	85	„	5	23.7	51.9	27.0	29.8	4.55	118
12	84.5	„	5.5	32.5	52.3	14.0	13.7	2.10	136
13	83.5	12.5	4	19.6	42.5	22.0	19.8	6.15	87
14	82.5	„	5	34.4	52.6	14.0	13.8	0.90	125
15	82	„	5.5	41.9	55.8	12.0	13.8	1.35	136
16	81	15	4	20.5	52.2	38.0	25.5	6.50	104
17	80.5	„	4.5	34.9	56.0	24.0	21.8	5.93	123
18	80	„	5	33.1	53.9	14.0	13.8	2.80	128
19	77	20	3	15.0	46.6	48.7	44.8	6.45	85
20	76	„	4	34.6	51.1	6.0	8.5	0.75	139
21	68.5	30	1.5	12.6	41.5	29.8	30.5	3.69	81
22	68	„	2	16.6	48.5	16.6	20.6	1.48	102
23	59.5	40	0.5	20.5	49.2	30.0	27.1	5.90	109
24	59.25	„	0.75	29.8	52.2	12.0	19.0	4.66	119
25	59	„	1	30.5	49.3	12.0	16.7	4.10	121

TABLE 9

Mechanical Tests on the Casting of Alloy No. 11 (Cu 85; Zn 10; Si 5) Poured at different temperatures.

Pouring Temp. °C.	Yield-Point. kg/mm²	Ultimate Stress. kg/mm²	Elongation on 50 mm. %	Reduction of Area. %	Impact Test (Izod). m-kg absorbed	Brinell Hardness Number.	Condition
1,030	25.7	48.3	21.2	27.0	4.80	117	Sand-Casting
970	25.8	48.9	20.8	24.1	4.25	122	,,
945	27.3	49.2	21.2	30.7	4.60	125	,,
915	28.4	49.0	17.0	23.4	4.30	125	,,
1,030	20.3	53.9	26.2	31.3	4.60	124	Chill-Casting
970	24.4	52.4	22.4	24.5	4.80	124	,,
945	21.6	52.6	21.2	22.8	4.60	125	,,
915	21.1	56.1	28.0	26.0	5.00	125	,,

TABLE 10

Tests on Forged Bars (25 mm dia.)

Alloy No.	Composition. %			Yield-Point. kg/mm²	Ultimate Stress. kg/mm²	Elongation on 50 mm. %	Brinell Hardness Number.	Impact Test (Izod). m-kg absorbed
	Cu	Zn	Si					
	88	7.5	4.5	12.9	40.7	56.8	72	8.57
8	87.5	,,	5	32.2	57.1	32.6	128	5.02
	87	,,	5.5	37.8	63.0	18.7	154	3.12
	86.5	,,	6	41.1	61.2	9.6	159	1.17
9	86	10	4	18.7	40.6	60.2	89	8.51
10	85.5	,,	4.5	29.4	47.7	42.8	93	7.80
11	85	,,	5	38.3	66.8	28.0	148	5.70
12	84.5	,,	5.5	34.4	60.9	17.4	147	3.50
16	81	15	4	18.7	54.1	48.0	100	8.47
17	80.5	,,	4.5	36.2	66.5	29.7	135	5.32
18	80	,,	5	38.0	61.0	20.6	154	3.12
	79.5	,,	5.5	40.3	52.4	13.0	178	0.93
	78	20	2	14.6	41.5	69.4	76	10.89
19	77	,,	3	15.9	51.6	51.6	100	6.60
	76.5	,,	3.5	18.1	52.7	32.0	101	6.00
20	76	,,	4	25.1	52.6	12.6	138	1.70

TABLE 11

Specific Gravities.

Alloy No.	Composition. %			Specific Gravity.		
	Cu	Zn		Sand-Casting	Chill-Casting	Forged Bars
			Si			
9	86	10	4	8.39	8.38	8.39
10	85.5	”	4.5	8.28	8.32	8.33
11	85	”	5	8.25	8.27	8.28
12	84.5	”	5.5	8.24	8.24	8.25
16	81	15	4	8.30	8.29	8.33
17	80.5	”	4.5	8.26	8.26	8.28
18	80	”	5	8.23	8.23	82.4
19	77	20	3	8.34	8.34	8.36
20	76	”	4	8.25	8.23	8.26
			Sn			
G.M.	88	2	10	8.70	8.71	

TABLE 12

No.	Composition. %			Elastic Modulus in kg/cm^2.	
	Cu	Zn	Si	Sand-Casting	Chill-Casting
9	86	10	4	1,012,500	873,000
10	85.5	”	4.5	1,065,500	997,600
11	85	”	5	1,086.000	1,033,000
12	84.5	”	5.5	1,105,700	1,066,000
17	80.5	15	4.5	1,121,121	1,037,333
	Cu	Zn	Sn		
G.M.	88	2	10	961,000	943,000

TABLE 13

Bending Tests on the Castings.

Alloy No.	Composition. %			Max. load. kg	Deflection. mm	Bending angle at max. load in degrees.	Angle to be bent in degrees.	
	Cu	Zn						
			Si					
9	86	10	4	1,290	25.3	88	88	Sand-Casting
10	85.5	„	4.5	1,650	19.6	74	98	„
11	85	„	5	1,460	13.2	47	47	„
12	84.5	„	5.5	2,840	8.2	24	24	„
16	81	15	4	1,780	26.6	67	113	„
17	80.5	„	4.5	1,825	23.6	90	90	„
18	80	„	5	1,270	5.6	17	17	„
			Sn					
G.M.	88	2	10	530	10.6	33	33	„
			Si					
9	86	10	4	1,650	30.0	70	180	Chill-Casting
10	85.5	„	4.5	1,890	17.0	68	125	„
11	85	„	5	1,810	20.7	78	78	„
12	8.45	„	5.5	1,940	14.4	43	43	„
16	81	15	4	2,250	30.0	70	158	„
17	80.5	„	4.5	1,850	24.8	90	90	„
18	80	„	5	1,650	12.2	32	32	„
			Sn					
G.M.	88	2	10	1,000	15.3	36	36	„

TABLE 14

Repeated-Bending Impact Tests on Notched Specimens of Selected Alloys, & Gun-Metal.
(No. of blow per min., 60)

Alloy No.	Composition. %			Energy of each blow. kg-cm	Number of Blows for Fracture.	
	Cu	Zn			Sand-Casting	Chill-Casting
			Si			
9	86	10	4	20	7,398	5,046
	”	”	”	25	6,993	4,342
	”	”	”	30	5,595	3,832
10	85.5	”	4.5	20	11,738	8,512
	”	”	”	25	7,212	5,281
	”	”	”	30	5,700	3,628
11	85	”	5	20	11,816	10,568
	”	”	”	25	6,444	6,417
	”	”	”	30	2,593	5,708
12	84.5	”	5.5	20	8,766	9,876
	”	”	”	25	4,923	5,405
	”	”	”	30	1,211	2,802
16	81	15	4	20	8,678	7,887
	”	”	”	25	5,773	2,949
	”	”	”	30	3,811	2,403
17	80.5	”	4.5	20	13,007	11,472
	”	”	”	25	8,251	8,920
	”	”	”	30	5,237	5,314
18	80	”	5	20	8,937	7,918
	”	”	”	25	5,916	3,976
	”	”	”	30	3,333	3,044
			Sn			
G.M.	88	2	10	20	949	51
	”	”	”	25	243	43
	”	”	”	30	107	33

TABLE 15

Tensile Tests on Sand-Casting at High Temperatures.

Alloy No.	Composition. %			Temperature (One hour). °C.	Ultimate Stress. kg/mm²	Elongation on 50 mm. %	Reduction of Area. %
	Cu	Zn					
			Si				
11	85	10	5	100	49.0	22.6	20.3
"	"	"	"	150	45.3	22.4	22.0
"	"	"	"	200	44.6	21.2	17.7
"	"	"	"	250	42.0	16.5	13.7
"	"	"	"	300	36.8	13.8	8.5
"	"	"	"	350	33.0	8.8	7.6
"	"	"	"	400	23.2	7.2	6.1
"	"	"	"	450	20.6	7.2	4.7
			Si				
G.M.	88	2	10	150	27.0	25.2	16.9
"	"	"	"	200	26.5	22.2	18.2
"	"	"	"	250	22.6	12.7	8.1
"	"	"	"	300	20.2	16.0	17.7
"	"	"	"	350	19.5	12.0	11.0
"	"	"	"	400	17.5	10.6	7.0
"	"	"	"	450	15.0	9.4	4.9

TABLE 16

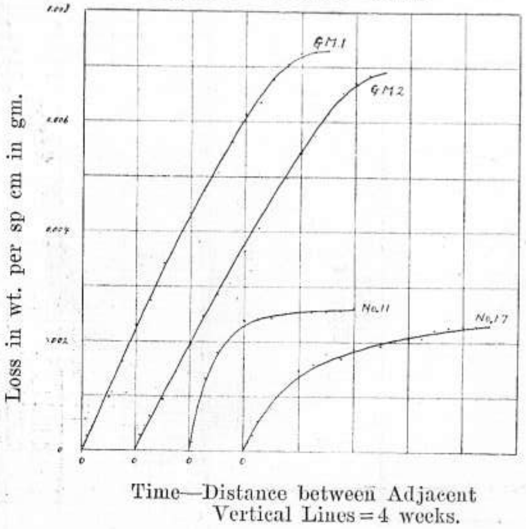
Tensile Tests on Forged Bars of Alloy No. 11 at High Temperatures.

Composition. %			Temperature (One hour). °C.	Yield-Point. kg/mm²	Ultimate Stress. kg/mm²	Elongation on 50 mm. %	Reduction of Area. %
Cu	Zn	Si					
85	10	5	100	39.0	63.00	30.8	28.0
"	"	"	150	44.2	61.00	28.4	22.7
"	"	"	200	41.5	57.10	22.2	19.0
"	"	"	300	39.0	46.25	12.0	13.7
"	"	"	400	24.8	29.60	9.8	12.5
"	"	"	500	8.5	11.80	56.0	53.5
"	"	"	600	2.6	5.43	62.0	80.0
"	"	"	700	1.5	2.42	65.0	80.6
"	"	"	800	0.3	0.50	56.2	56.7

TABLE 17

Sample	Composition, %				Loss in wt. per sq cm in gm.								
	Cu	Zn	Sn	Si	After 2 weeks	After 4 weeks	After 6 weeks	After 8 weeks	After 10 weeks	After 12 weeks	After 14 weeks	After 16 weeks	After 18 weeks
G.M. 1	88	2	10		0.0009910	0.0022982	0.0034293	0.0043248	0.0050866	0.0061455	0.0068012	0.0072065	0.0073275
G.M. 2	87	5	8		0.0009146	0.0019714	0.0028627	0.0036351	0.0044012	0.0054261	0.0061476	0.0067057	0.0069052
No. 11	85	10		5	0.0018059	0.0023958	0.0024467	0.0025571	0.0025719	0.0026526			
No. 17	80.5	15		4.5	0.0007894	0.0014600	0.0017995	0.0018823	0.0019162	0.0020584	0.0022006	0.0022961	0.0023385

Fig. 31. Corrosion Curves.



(Paper No. 613)

Fig. 32.

Inverse Rate Cooling Curve.

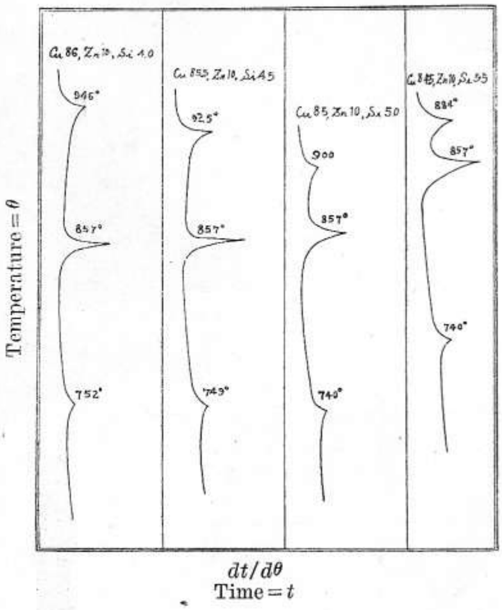


Fig. 33.

Inverse Rate Cooling Curve.

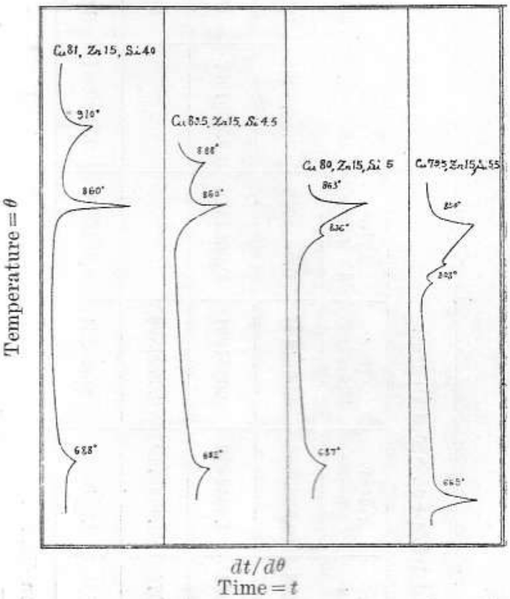
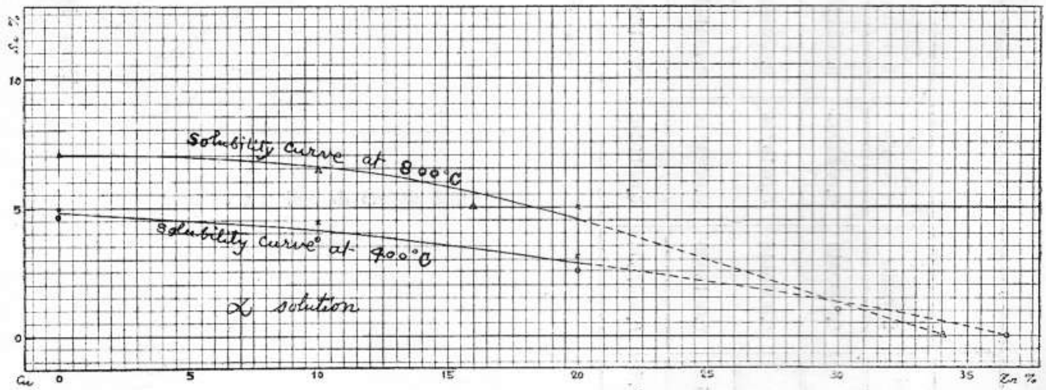


Fig. 34.

Maximum Solubility Curves of Silicon into α- Solution of Cu–Zn–Si System.



×100 diams.

Alloy No. 10 (10%Zn, 4.5%Si)

Fig. 35. Sand-casting.

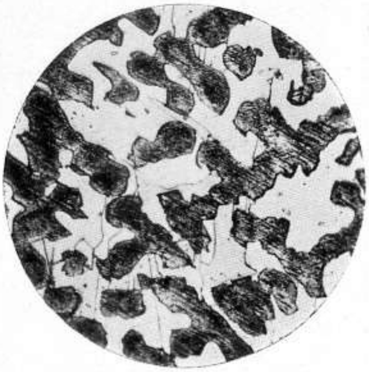


Fig. 36. Chill-casting.

Alloy No. 11 (10%Zn, 5.0%Si)

Fig. 37. Sand-casting.

Fig. 38. Chill-casting.

Alloy No. 12 (10%Zn, 5.5%Si)

Fig. 39. Sand-casting.

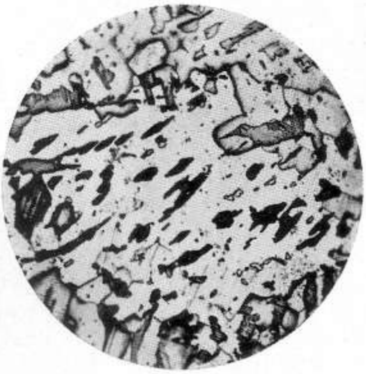
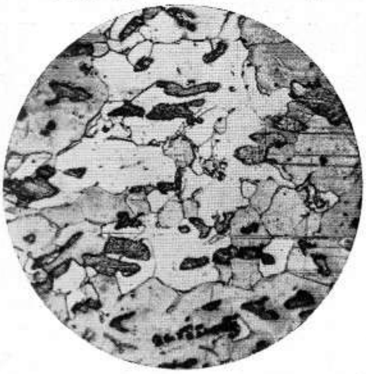


Fig. 40. Chill-casting



×100 diams.

Alloy No. 16 (15%Zn, 4.0%Si)

Fig. 41. Sand-casting.

Fig. 42. Chill-casting.

Alloy No. 17 (15%Zn, 4.5%Si)

Fig. 43. Sand-casting.

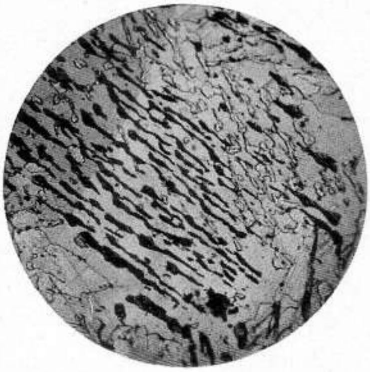


Fig. 44. Chill-casting.

Alloy No. 18 (15%Zn, 5.0%Si)

Fig. 45. Sand-casting.

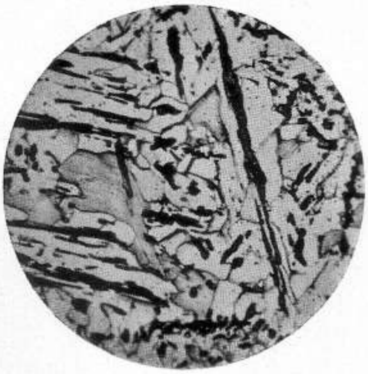
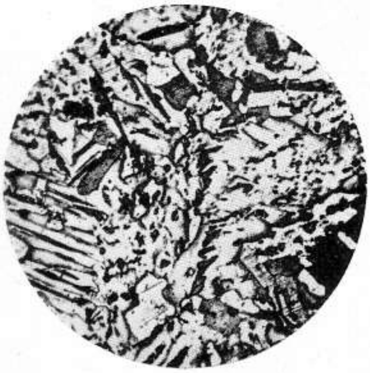


Fig. 46. Chill-casting.



On the Scantling of Hull Materials of Rolled Steel

(Paper No. 621)

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Introduction.

The structural members of the hull of a ship may be classified into the following three groups.

(1) Longitudinal Member, which contributes chiefly to the longitudinal strength of the hull structure.

(2) Transverse Member, which contributes chiefly to the transverse strength of the hull structure.

(3) Local Member, which lies either longitudinally or transversely and acts chiefly for the purpose of stiffening or supporting a part of the hull. Structural members of this group are assumed to have no direct connection with the strengths of the hull structure as a whole.

The scantling of the structural member belonging to Group (1) is usually determined by experience on the bases of the length and draught of the ship. The scantling of the structural members belonging to Groups (2) and (3) are usually determined on the bases of loads and stresses. The scantling of forgings and castings such as stem, bar keel and stern frame, and that of the structural members fitted for strengthening against panting, blow of ice etc. should be better determined by experience.

In cases where the scantling of structural members is determined on the bases of loads and stresses, it is usual to assume same value of the working stress so far as the structural use of the material remains same. This does not, however, hold good in all cases. If there is no need to make any provosion for the wasting of materials, then same value of the working stress may undoubtedly be adopted in all cases. In actual cases, account should be taken more or less of the effect of the wasting of materials. It is here intended to examine the extent of the wasting of rolled steel experienced in actual ships, and to consider the necessary extent of provision for the wasting, and further to investigate how the scantling of structural members of the hull of a ship is thereby affected.

Factor of Safety.

The working stress of a material is dependent on the factor of safety. Although the factor of safety is defined in different ways, its value varies according to,

(1) Nature of stresses,
(2) Quality of workmanship,
(3) Accuracy of calculations and judgments,
(4) Reliability of material,
(5) Liability to wasting on account of corrosion as well as wear and tear.

When the size of structural members is decided on the basis of same value of the working stress, the cross section of the structural members contains over the whole area considerations for all the elements enumerated above. The elements (1), (2), (3) and (4) are inseparable in their nature and it is right to fix the value of the factor of safety aggregately for them. But the element (5) has quite a different aspect from the others. Really corrosion and wear and tear of materials take place simply on the surface and therefore the effect of this element has to be determind in a different way.

Extent of the Wasting of Rolled Steel experienced in Actual Ships, and the Preparatory Thicknesses required in order to provide for the Wasting.

Here as a thing of course, structural members are assumed to have been carefully protected by some sorts of coating.

The wasting can be measured by the reduction of the thickness of the material. The extent of the reduction of the thickness of rolled steel experienced in actual ships is shown in Table I. The Table is founded on the data obtained from the test hole results on occasion of the survey for freeboard assignment. Test holes were drilled in the hull material of various ships of 10 to 28 years of age. Unfortunately the reduction of the thickness in plates and sectional materials other than those mentioned in the Table could not be ascertained owing to the lack of data. But it can be deduced without great fault from the data of the plates and sectional materials for similar use and in similar locality.

The word "Average" in the Table is used to mean "the Average Extent of the Probable Reduction in 25 years."

TABLE I.

The Reduction of the Thickness of Rolled Steel on account of Wasting.

Item	Reduction of Thickness in inch		
	Least	Greatest	Average
Sheer strake	.02	.07	.05
Strake below Sheer strake	0	.10	.08
Side plating	.02	.07	.06
Bottom plating except Keel plate	.01	.07	.05
Inner bottom plating	.02	.05	.04
Centre girder plate	.01	.05	.05
Exposed deck platings;			
When unsheathed	0	.10	.05
When sheathed	0	.02	.01
Unsheathed lower deck plating and Watertight bulkhead plating	0	.10	.04
Hold frames;			
Bulb angle	.05	.08	.06
Channel	?	.10	.08
Built frame;			
Frame	.04	.12	.06
Reverse frame	.09	.12	.10

Hereafter the term "PREPARATORY THICKNESS" is used to mean "the Thickness to be provided beforehand for the Wasting on account of Corrosion and Wear and Tear."

It is very difficult in general to distinguish between the reduction of thickness on account of corrosion and that on account of wear and tear, and therefore the provision is considered for both of them inclusively. Further provision, however, should be made for materials which are located in boiler space, coal bunker or bilge etc., as the corrosion in such quarters are excessive.

Of course the extent of the wasting varies according to the kind of cargoes, the service of ships and the method of maintainance and also depends on the quality of the material and the locality of the structural members. The structural members, such as frames, beams etc. which are compact in size, accessible and visible, may often be inspected, and their thickness be examined when deemed necessary. On the other hand, the reduction of the thickness of the structural members, such as side platings, watertight bulkhead platings etc. which are wide spreading in area and are not easily accessible although visible, is apt to escape the eye of the surveyor. The extent of the provision for wasting may differ according to the authors.

The author's view is developed in the following Table. The preparatory thickness for plates has been assumed equal to three times the average value of the reduction of the thickness shown in Table I, while as that for sectional materials the average value of the reduction set forth in the same table has been taken in its entirety.

TABLE II.

Preparatory Thicknesses.

Item	Preparatory thickness in inch	Further provision in inch, for materials in boiler space etc.
Flat plate keel, and Intercostal plate of deck girder	.27	—
Side plating in the vicinity of the load water line	.24	—
Inner bottom plating under hatch when not protected by ceiling	.20	—
Bottom plating except flat plate keel	.18 (may be .16 if desired)	—
Side plating other than that in the vicinity of the load water line	.18	.06
Keelson plate, Single bottom floor, Plate of web frame, and Intercostal plates of side stringer and horizontal girder	.18	.08
Sheer strake of the superstructure deck, and Centre girder plate of double bottom	.15	.08
Unsheathed exposed deck plating	.15	.02
Solid floor plate, Side girder plate and Bracket plate in the double bottom	.15	.16
Inner bottom plating	.12	.16
Unsheathed lower deck plating, and Bracket plate outside the double bottom	.12	.06
Watertight bulkhead plating	.12	.10 (lowest strake only)
Sheathed deck plating	.04	.02
Angle for reverse frame of side frame	.10	.10
Angles and Bulb angles for main frame of side frame, and for reverse frame and strut of skeleton floor	.06	.06
Angle and Bulb angle for frame of skeleton floor	.06	.04
Channels for side frame, reverse frame and strut of skeleton floor, face bars on web frame and side stringer	.08	.08
Channel for frame of skeleton floor	.08	.06
Angles and Bulb angles for deck beam and bulkhead stiffener	.04	.02
Channels for deck beam and bulkhead stiffener	.06	.02
Face bars on deck girder and horizontal girder	.04	.02
Plate and Section of built pillar	.04	—

Ratio of the Moduli of Resistance of Two Sets of Compound Girders which are of Similar Form in Cross Section but have Composing Members of Different Thicknesses.

Here the percentage ratio of the moduli of resistance of the cross section of the following two sets of compound girders is considered.

Girder (A).

In the case of the girder of this set, it is assumed that both the plate and sectional material are provided with the preparatory thickness.

Girder (B).

In the case of the girder of this set, it is assumed that the plate is provided with two thirds of the preparatory thickness, and the sectional material with no preparatory thickness at all. In short, the girder (B) is lesser in thickness than the girder (A) by an amount equal to the average value of the reduction of the thickness set forth in Table I.

When calculating the modulus of resistance of girders, the following assumptions have been made.

1. The effective width of platings which belong to the "Longitudinal Member" and serve at the same time as the belt of the transverse ring of the hull structure, is assumed as follows.

a. In the case of girders such as side frame, web frame, floor girder of single bottom, and deck beam,

Width equal to the standard frame spacing.

The standard frame spacing for full scantling ship is generally expressed by the following formula;

$s=0.025L+17$, which need not be less than 21 inches in case of ships which are less than 160 feet in length,

where s=standard frame spacing in inches.

L=length of ship in feet.

b. In the case of girders such as deck girder, side stringer etc.,

Width equal to 80 times the thickness of the plating including the preparatory thickness.

2. The effective width of watertight bulkhead plating etc. is assumed as follows.

c. In the case of stiffeners,

Width equal to the spacing of the stiffener or 30 inches whichever is the less.

d. In the case of supporting girders,

Width equal to 80 times the thickness of the plating including the preparatory thickness.

3. No account is taken of the effect of rivet holes.

4. Short attachment angles of the slotted girder is totally disregarded

in the calculation, and no account is taken of the effect of the slot.

The avarage values of the percentage ratio of the modulus of resistance of the girder (B) to that of the girder (A) are shown in Tables III A to C.

TABLE III.

Average Values of the Percentage Ratio of the Moduli of Resistance.

A. Hold frame, Frame and Reverse frame of skeleton floor, and Beam of exposed unsheathed strength deck.*

Item		Percentage Ratio
Ordinary angles		89
Bulb angles	3 inch flange	92
	3½ inch flange	94
Channels	3 inch flanges	85
	3½ inch flanges	84
	4 inch flanges	83
Built frames of two angles of different thicknesses	3 inch flanges	77
	3½ inch flanges	82
	4 inch flanges	86

---

* The thickness of the deck plating is assumed to be not less than what is necessary in way of deck opening.

B. Bulkhead stiffener, and Deck beam other than that given in A.

Item		Percentage Ratio
Ordinary angles	Thin plating	86
	Thick plating	93
Bulb angles	Thin plating	93
	Thick plating	97
Channels	Thin plating	91
	Thick plating	93

C. Web frame, Side stringer, Horizontal girder and Deck girder.

Item		Percentage Ratio
Web frame		90
Side stringer and Horizontal girder		85
Deck girder	Thin deck plating	90
	Thick deck plating	95

Examples of Modified Scantlings.

As has been shown in Tables III A to C, the percentage ratio of the modulus of resistance of the girder (B) to that of the girder (A) varies roughly from 75 to 95. It is of an extreme importance to decide the lower limit of the ratio, or reciprocally to estimate the upper limit of the working stress of rolled steel allowable in the case of the girder (B). It should be noted that the majority of the hold frames, which happened to be examined with respect to the reduction of the thickness and showed remarkable reductions, were those of the built type of the moderate depth of girder consisting of two angles of equal thickness, and therefore 83 may safely be taken as the lower limit of the percentage ratio.

The ultimate tensile strength of rolled steel for ship construction may in general be assumed at 28 tons per square inch and a factor of safety of 4 may without delay be used for this value of the ultimate strength. These give a working stress of 7 tons per square inch. Then the working stress of rolled steel allowable in the case of the girder (B) may be assumed at $7 \div .83 = 8.40$ tons per square inch. This value of the working stress should be modified in the case of the girder (A) in accordance with the following formula.

Modified working stress of rolled steel allowable in the case of the girder $(A) = 8.40 \times v$ in tons per square inch,

where v = percentage ratio given in Table III A to C.

It is not quite clear what would be the working stress of rolled steel assumed by Classification Societies, although it can be guessed to a certain extent, in determining the table scantlings of the structural members of the hull. In the following examples, the scantlings of structural members in accordance with the requirements of the Rules of the two well known Classifications Societies are added for the sake of reference and comparison.

Abbreviations "B. C." and "L. R." are used to denote "The British Corporation Register" and "Lloyd's Register" respectively.

EXAMPLE 1. A seagoing cargo steamer built in 1928.

Principal dimensions = 430.3′ × 57.5′ × 39.5′ to shelter deck.

Moulded load draught = 27.0′

'Tween deck heights { Shelter deck to Upper deck = 8.0′
Upper deck to Second deck = 10.0′

Spacings of frame and beams amidships = 36″

Item.			Scantlings by Calculation.*		Scantlings in accordance with the Rules of Classification Societies.	
			Working stress in tons per sq. in.			
			7	8.40 × v	B. C.	L. R.
Hold frames	Thickness of side plating = .70″ Height of the top of tankside bracket above the base line = 7.00′ Depth of the second deck beam knee = 3.00′		11 × 3½ × .60 B.A. 10½ { 7½ × 3½ × .44 6½ × 3½ × .48	11 × 3½ × .46 B.A. 10½ { 7½ × 3½ × .46 6½ × 3½ × .50	12 × 3½ × .56 B.A. 12 { 7½ × 3½ × .42 8 × 3½ × .52	11 × 3½ × .58 B.A. 10 { 7 × 3½ × .48 6 × 3½ × .48
Skeleton floor frames	Length of span = 10.88′, one strut being fitted at the middle.		9½ × 3½ × .50 B.A.	9 × 3½ × .50 B.A.	9 × 3½ × .52 B.A.	8 × 3½ × .40 B.A.
	Length of span = 8.00′, no strut being fitted at the middle.		10 × 3½ × .50 B.A.	9½ × 3½ × .50 B.A.	9½ × 3½ × .50 B.A.	10½ × 3½ × .52 B.A.
Shelter deck beams	Thickness of deck plate = .42″	(1) Length of span = 20.25′, height of load = 5.35′	9 × 3½ × .46 B.A.	8½ × 3½ × .46 B.A.	9 × 3½ × .45 B.A.	10 × 3½ × .46 B.A.
		(2) Same span, height of load = 6.00′	9½ × 3½ × .44 B.A.	9 × 3½ × .46 B.A.		
	Thickness of deck plate = .56″	(1′) Length of span = 16.00′, height of load = 5.35′	7 × 3½ × .50 B.A.	7 × 3½ × .40 B.A.	8 × 3½ × .40 B.A.	8 × 3½ × .50 B.A.
		(2′) Same span, height of load = 6.00′	7½ × 3½ × .48 B.A.	7½ × 3½ × .38 B.A.		
Upper deck beams	Length of span = 20.25′, thickness of deck plate = .30″		11 × 3½ × .46 B.A.	10 × 3½ × .50 B.A.	10 × 3½ × .45 B.A.	10 × 3½ × .46 B.A.
Second deck beams	(1) Length of span = 20.25′, thickness of deck plate = .32″		12 × 3½ × .51 B.A.	11½ × 3½ × .48 B.A.	11 × 3½ × .48 B.A.	10½ × 3½ × .52 B.A.
	(2) Length of span = 16.00′, thickness of deck plate = .38″		9½ × 3½ × .50 B.A.	9½ × 3½ × .40 B.A.	9 × 3½ × .43 B.A.	9½ × 3½ × .46 B.A.

* See the footnote at end of Example 2.

EXAMPLE 2.

A seagoing cargo steamer built in 1927.

Principal dimensions=300.00′ × 43.50′ × 22.25′, with one deck.
Moulded load draught=18.50′
Spacings of frame and beam amidships=24″

Item.			Scantlings by Calaulation.*		Scantlings in accordance with the Rules of Classification Societies.	
			Working stress in tons per sq. in.			
			7	8.40 × v	B. C.	L. R.
Hold frames	Thickness of side plating = .48″ Height of the top of tankside bracket above the base line = 5.50′ Depth of the beam knee = 2.04′		8½ × 3 × .40 B.A. 7½ { 5½ × 3 × .36 5 × 3 × .40	8 × 3 × .48 B.A. 7½ { 5½ × 3 × .38 5 × 3 × .42	8 × 3 × .42 B.A. 7½ { 5½ × 3 × .34 5 × 3 × .40	9 × 3 × .40 B.A. 7½ { 5½ × 3 × .42 5 × 3 × .42
Upper deck beams	(1) Length of span = 11.00′, thickness of deck plate = .36″		5½ × 3 × .42 B.A.	5½ × 3 × .34 B.A.	5½ × 3 × .32 B.A.	5½ × 3 × .42 O.A.
	(2) Length of span = 8.75′, thickness of deck place = .50″		5½ × 3 × .37 O.A.	5 × 3 × .40 O.A.	5 × 3 × .38 O.A.	5 × 3 × .38 O.A.
Deck girders of Upper deck	Length of span = 22.00′, mean breadth of deck supported = 11.00′	(1) Thickness of deck plate = .36″	{ 18 × .45 8½ × 3 × .46 B.A.	{ 18 × .45 7½ × 3 × .46 B.A.	{ 16½ × .46 7 × 3½ × .54 B.A.	{ 12 × .40 9 × 3 × .50 B.A.
		(2) Thickness of deck plate = .48″	{ 18 × .45 8 × 3 × .46 B.A.	{ 18 × .45 7 × 3 × .48 B.A.		

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* Scantlings determined by the author; the methods are fully described in the author's work "Kosen Kozo Ron" (Steel Ship Construction) published in 1928.

Conclusion.

1. The preparatory thickness for hull materials of rolled steel should be suitably decided in accordance with the structural use of the materials and their locality in the hull, as well as with the kind of cargoes, and the predestinated service and life of the ship.

Experience shows that the preparatory thickness for the bottom plating may be somewhat less, while that for the side platings situated in the vicinity of the load water line and in the boiler space, coal bunker, bilge etc. should be considerably greater than that for the side plating in general.

2. In cases where the scantlings of structural members are determined on the bases of loads and stresses, the preparatory thickness should be dealt with as an independent element different from the thickness which is necessary in order to afford the requisite strength of the structures. This will inevitably cause certain deviation of the scantling of structural members, when compared with that which has been determined using same value of the working stress irrespective of the size of structural members.

3. As the result of observing the performance of various structural members in old ships found to stand still well after more than half a generation from the date of build, it is gathered that the value of the working stress of rolled steel may be assumed at a higher standard, probably 10% to 20% higher, than that usually assumed. But this does not necessarily result in the reduction of the scantlings of structural members of rolled steel in general.

4. It is usual in the practical design of hull construction to assume the value of the working stress of materials for the sections which include the preparatory thickness. In such cases, the value of the working stress should be suitably modified in proportion to the thickness included.

The end.

Model Experiments on the Elastic Stability of Closed and Cross-Stiffened Circular Cylinders under Uniform External Pressure.

(Paper No. 651)

By Viscount Takesada Tokugawa,

Constructor-Commander, I.J.N.

Abstract.

In this paper a method is developed for calculating the collapsing pressure of closed and cross-stiffened circular cylinders under uniform external pressure. Evidently, there are two distinct cases; viz.—(I) where the frames remain intact and the shell alone collapses; (II) where both the shell and frames collapse bodily. So far as I am aware, the latter case has not yet been dealt with. The following are the formulae deduced.

For case (I):—

$$p_k=\frac{E\frac{s}{D}}{n^2-1+\frac{1}{2}\alpha^2}\left[\frac{2\alpha^4}{(n^2+\alpha^2)^2}+\frac{2}{3}\frac{1}{1-\sigma^2}\left(\frac{s}{D}\right)^2\left\{(n^2+\alpha^2)^2-\frac{n^4(2n^2-1)}{(n^2+\alpha^2)^2}\right\}\right]$$

in which p_k is the collapsing pressure, s the thickness, D the mean diameter, l the unsupported axial length of the cylinder, E Young's modulus, σ Poisson's ratio, n the number of lobes,

$$\alpha=k\frac{\pi}{2}\frac{D}{l},$$

k is an empirical constant and is called the "frame factor." In the forgoing expression the value of n is to be taken so that the whole number which, for given values of s, D, and l, will make p_k a minimum.

For case (II):—

$$p_k=\frac{E\frac{s}{D}}{n^2-1+\frac{1}{2}\alpha^2}\left[\frac{2\alpha^4}{(n^2+\alpha^2)^2}+\frac{2}{3}\frac{1}{1-\sigma^2}\left(\frac{s}{D}\right)^2\beta\left\{(n^2+\alpha^2)^2-\frac{n^4(2n^2-1)}{(n^2+\alpha^2)^2}\right\}\right]+\frac{8EI}{D^3l'}\gamma(n^2-1),$$

where l' is the length of a bay, n the number of lobes into which the circumference of the shell in common with frames divides, β and γ are the factors dependent on the built-up section, I is the moment of inertia of the section of a frame proper,

$$\alpha=\frac{\pi}{2}\frac{D}{L},$$

L is the total length of the cylinder.

These formulae have been confirmed to agree very well with experiment and the results obtained are collected together in the form of curves; in addition a nomogram ready for immediate application to any particular problem is appended.

Introduction.

The study described below is a sequel to, and an extension of, my report "Model Experiments on the Elastic Stability of Submarine Pressure-

Hulls" submitted to the Japanese Naval Department on Nov. 10, 1928* The main object of this paper is twofold—first, to investigate the nature of the so-called "frame factor" involved in my formula; second, to obtain the formula for the critical pressure when both the shell and frames collapse as a body. So far as I am aware practically nothing is known about the latter case. It should be emphasized that all statement and conclusions are advanced on my sole responsibility, and carry no official sanction.

Taking this opportunity, I express my gratitude for:—

(i) That all the expenses of this research were defrayed by a grant from the Naval Department.

(ii) That full facilities for the execution of this research have been most kindly afforded by the Naval Technical Research Institute.

(iii) The unfailing collaboration and suggestion of my colleague officers, as well as the faithful assistance of the men in my charge.

Part I. Theoretical.

1. *An Application of the Formula for Short Cylinders to the calculation of the Collapsing Pressure of the Shell between Frames of Closed and Cross-stiffened Cylinders.* According to the size of the stiffener, the closed and cross-stiffened cylinder will be collapsed in two ways: if stiffeners are large and strong enough, the collapse occurs at the shell between frames while the frames remain intact; and if stiffeners are slender, both the shell and frames collapse bodily. Moreover, the shape of stiffeners and the mode of fixing the stiffeners to the shell will affect the value of the collapsing pressure.

In this section the case where the shell alone collapse will be considered, and as a starting point the following formula** is taken:—

$$(1)\qquad p_k=\frac{E\dfrac{s}{D}}{n^2-1+\frac{1}{2}\alpha^2}\left[\frac{2\alpha^4}{(n^2+\alpha^2)^2}+\frac{2}{3}\,\frac{1}{1-\sigma^2}\left(\frac{s}{D}\right)^2\left\{(n^2+\alpha^2)^2-\frac{n^4(2n^2-1)}{(n^2+\alpha^2)^2}\right\}\right]$$

the notation has already been explained in the abstract.

It is to be noted that the introduction of the "frame factor" k, which is to be determined by experiment, into

$$\alpha=k\frac{\pi}{2}\cdot\frac{D}{l}$$

enables us to apply the formula to the case of cross-stiffened cylinders.

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* From Oct., 1925 till March, 1926, an extended experiment with models, 1.2～1.6 m. in dia., was continually carried out at the Kure Naval Dockyard.

** I deduced this formula a few years ago and fully explained in my report cited before, but as this report is not published, the mathematical analysis is given in Note I.

Any rational formula, however, is based on certain ideal assumption which can never be realized in practice. For this reason a purely theoretical formula is likely to give results in excess of those determined by experiment. As a correction for the reduction in strength due to imperfections which violate our ideal assumptions, we have introduced the "frame factor" k. For the practical application of the formula, the cases may be divided as follows:—

(a) When α is comparable to n, and n is as large as 5 or even larger, unity may be neglected against n^2 or $\frac{1}{2}\alpha^2$ in the denominator $n^2-1+\frac{1}{2}\alpha^2$; further $(s/D)^2$ being very small, the second term in the brackets [ ] is of less importance, so that the second term $n^4(2n^2-1)/(n^2+\alpha^2)^2$ in the braces { } may be neglected against the first term $(n^2+\alpha^2)^2$. Thus, taking $\sigma=0.3$ for mild steel [$\sigma=0.4$ for brass], consequently $\frac{2}{3}(1-\sigma^2)^{-1}=0.73$ [0.794 for brass], we have

$$p_k=\frac{E\dfrac{s}{D}}{n^2+\frac{1}{2}\alpha^2}\left[\frac{2\alpha^4}{(n^2+\alpha^2)^2}+0.73\left(\frac{s}{D}\right)^2(n^2+\alpha^2)^2\right]^* \tag{2}$$

(b) When α is rather small compared with n, and $(s/D)^2$ being very small, α^2 in the braces may be neglected against unity or n^2. Thus, we have

$$p_k=\frac{E\dfrac{s}{D}}{n^2-1+\frac{1}{2}\alpha^2}\left[\frac{2\alpha^4}{(n^2+\alpha^2)^2}+0.73\left(\frac{s}{D}\right)^2(n^2-1)^2\right], \tag{3}$$

The former is especially important for the practical application such as the design of submarine pressure-hulls, so that a nomogram of (2) is prepared for ready use as shown in Pl. I.

---

[*] Since having developed my theoretical formula I noticed that Prof. J. Prescott, in his "Appplied Elasticity," 1924, dealt with the problem taking the end-thrust into account. His formula runs as follows:—

$$p_k=\frac{E\dfrac{s}{D}}{n^2-1+\dfrac{\alpha^2}{2}\left(1+\dfrac{1}{n^2}\right)}\left[\frac{2\alpha^4}{n^4}+\frac{2}{3}\,\frac{1}{1-\sigma^2}\left(\frac{s}{D}\right)^2(n^2-1)^2\right],$$

where $\alpha=\frac{1}{2}\pi D/l$. But it is to be noted that his treatment is different in details from mine; moreover, the main object of the present work is to determine the correction-factor in the deduced formula which fits closely experimental data.

In the next place, since presenting my report to the Naval Department, Nov. 10, 1928, I find that the same problem introducing the effect of end thrust has been considered by von Mises in the Paper "Der kritische Aussendruck fuer allseits belastete zylindrische Roehre," 'Festschrift Prof. Dr. A. Stodola zum 70 Geburtstag,' 1929. pp. 418–430. His formula incidently agrees in form with (2) except that his α does not involve the "frame factor" k. Then, he proceeds to apply his formula to the results obtained in the Danzig experiment, 1918, with models of submarine pressure-hulls. But I cannot quite agree with his way of application. To the best of my belief it is more reasonable to introduce a constant k into the formula.

First, if there is no end thrust, the term $\frac{1}{2}\alpha^2$ in $n^2-1+\frac{1}{2}\alpha^2$ disappears and k in α becomes 1; the formula then becomes quite similar to that appeared in the previous paper of R. von Mises,* except that the latter including one more term of less importance.

Second, in (3) if we neglect moreover α^2 and $\frac{1}{2}\alpha^2$ against unity or n^2, and put $k=1$, viz., no end thrust and no distorsion at ends, we can arrive at Southwell's** or Love's† formula.

Third, if the cylinder is infinitely long, then putting $l=\infty$ in α and $n=2$,‡ we arrive at the classical formula due to Bryan§

$$p_k=3\frac{C}{r^3} \tag{4}$$

where $C=\frac{2}{3}h^2\frac{E}{1-\sigma^2}$, i.e., the so-called "cylindrical rigidity." Again, for the "flexual rigidity," putting $E\,I$ in place of C, in which I is the moment of inertia of the cross-section, we arrive at Lévy's formula*** which is used to calculate the critical pressure of a thin circular ring, i.e.,

$$p_k=3\frac{EI}{r^3} \tag{5}$$

or more generally,#

$$p_k=\frac{EI}{r^3}(n^2-1). \tag{6}$$

---

* "Die kritische Aussendruck der zylindrische Roehre," Z.V.D.I., vol. 58, No. 19, 1914, pp. 750–755. The formula:—

$$p_k=\frac{E\frac{s}{D}}{n^2-1}\left[\frac{2\alpha^4}{(n^2+\alpha^2)^2}+\frac{2}{3}\frac{1}{1-\sigma^2}\left(\frac{s}{D}\right)^2\left\{(n^2-1)^2+\frac{\alpha^2(2n^2-1.3)(n^2-1)}{n^2+\alpha^2}\right\}\right].$$

** Southwell, R. V., "On the Collapse of Tubes by External Pressure," Phil. Mag., vol. 25, 1913, pp. 689–698.

$$p_k=\frac{E\frac{s}{D}}{n^2-1}\left[\frac{2C}{n^4}\left(\frac{D}{l}\right)^4+\frac{2}{3}\frac{1}{1-\sigma^2}\left(\frac{s}{D}\right)^2(n^2-1)^2\right],$$

where C is a constant depending on the type of constraints at the ends of the cylinder.

† Love, A. E. H., "The Mathematical Theory of Elasticity," 4th. Ed., p. 574.

$$p_k=\frac{E\frac{s}{D}}{n^2-1}\left[\frac{2\alpha^4}{n^4}+\frac{2}{3}\frac{1}{1-\sigma^2}\left(\frac{s}{D}\right)^2(n^2-1)^2\right].$$

This is a slight extension of the above formula, by putting $C=\frac{1}{16}\pi^4$, which corresponds to only the type of constraints; if the ends are kept circular without restricting the slope of the cylinder wall, C may be determined theoretically.

‡ It is well known that very long cylinders tend to collapse into the two-lobed form.

§ Bryan, G. H., "Application of the Energy Test to the Collapse of a Long Thin Pipe under External Pressure," Cambridge Phil. Soc. Proc., vol. VI, Part V, 1889, pp. 287–292.

*** Lévy, M., "Mémoire sur un nouveau cas intégrable du problème de l'élastique et l'une de ses application," J. de Math. (Liouville), Ser. 3, to 10, 1884, pp. 5–42.

\# See my earlier paper, Jour. of the Jap. Soc. of N. A., No. 45, Oct., 1925.

2. *Bodily Collapse of Closed and Cross-stiffened Circular Cylinders under External Pressure.* The bays at both ends are subjected to an unknown amount of load through the end covers. Except these two bays, any one bay may be considered as a built-up structure consisting of the shell and the frame as shown in Fig. 1.

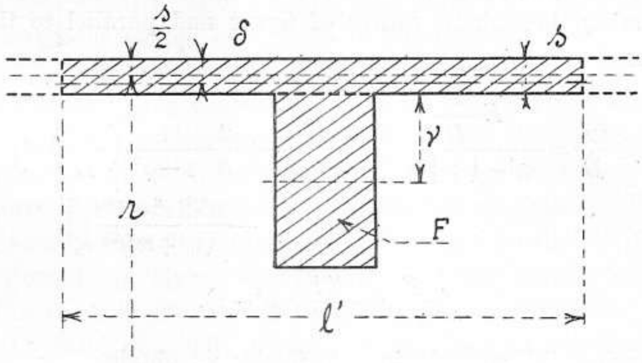

FIG. I.

Referring to the figure, let r be the mean radius of the cylinder, δ the distance from the outer surface of the shell to the neutral axis of the section of the built-up structure, we have

$$\delta = \frac{\frac{1}{2}\, l' s^2 + F(\nu + s)}{s l' + F},$$

where F is the sectional area of the frame proper, ν the distance from the inner surface of the shell to the neutral axis of the section of the frame proper. On the other hand, as the thickness is small compared with the diameter, the equation of equilibrium may at once be obtained from the figure as follows:—

$$p_k r l' = f_1 s l' + f_2 F,$$

where f_1 is the average hoop-stress produced in the shell, f_2 that in the frame. Hence

$$p_k = f_1 \frac{2s}{D} + f_2 \frac{2F}{D l'} \tag{7}$$

where $f_1 = E\eta_1/(1-\sigma^2)$, $f_2 = E\eta_2$. If η_1 and η_2 be the critical peripheral contractions at instant of bodily collapse, it follows that

$$\eta_1 = \frac{1}{n^2 - 1 + \frac{1}{2}\,\alpha^2} \left[\frac{\alpha^4 (1-\sigma^2)}{(n^2+\alpha^2)^2} + 4\left(\frac{i_1}{D}\right)^2 \left\{ (n^2+\alpha^2)^2 - \frac{n^4(2n^2-1)}{(n^2+\alpha^2)^2} \right\} \right],$$

where $\alpha = \pi r / L$, and that

$$i_1^2 = \frac{s^2}{12} \left\{ 1 + \frac{12}{s^2} \left(\delta - \frac{s}{2} \right)^2 \right\} = \frac{s^2}{12} \beta, \text{ say.}$$

$$\eta_2=\left(\frac{i_2}{r}\right)^2(n^2-1),$$

$$i_2^{\,2}=i_0^{\,2}+(s-\delta+\nu)^2=i_0^{\,2}\gamma,\ \text{say},$$

$$\gamma=1+\frac{1}{i_0^{\,2}}(s-\delta+\nu)^2.$$

where i_0 is the radius of gyration of the section of the frame proper about the axis passing through its centre of figure and parallel to the cylinder axis. Hence

$$(8)\qquad f_1\frac{2s}{D}=\frac{E\frac{s}{D}}{n^2-1+\frac{1}{2}\alpha^2}\left[\frac{2\alpha^4}{(n^2+\alpha^2)^2}+\frac{2}{3}\frac{1}{1-\sigma^2}\left(\frac{s}{D}\right)^2\beta\left\{(n^2+\alpha^2)^2-\frac{n^4(2n^2-1)}{(n^2+\alpha^2)^2}\right\}\right],$$

$$(9)\qquad f_2\frac{F}{rl'}=\frac{8EI}{D^3l'}\gamma(n^2-1).$$

Substituting (8) and (9) in (7), finally we obtain

$$(10)\qquad p_k=\frac{E\frac{s}{D}}{n^2-1+\frac{1}{2}\alpha^2}\left[\frac{2\alpha^4}{(n^2+\alpha^2)^2}+\frac{2}{3}\frac{1}{1-\sigma^2}\left(\frac{s}{D}\right)^2\beta\left\{(n^2+\alpha^2)^2-\frac{n^4(2n^2-1)}{(n^2+\alpha^2)^2}\right\}\right]+\frac{8EI}{D^3l'}\gamma(n^2-1),$$

where

$$\begin{cases}\alpha=\dfrac{\pi r}{L},\\[2mm] \delta=\dfrac{l'\frac{s^2}{2}+F(\nu+s)}{sl'+F},\\[2mm] \beta=1+\dfrac{12}{s^2}\left(\delta-\dfrac{s}{2}\right)^2,\\[2mm] \gamma=1+\dfrac{1}{i_0^{\,2}}(s-\delta+\nu)^2.\end{cases}$$

Part II. Model Experiment.

3. *Outline of a Large-Scale Experiment with Steel Models.* From Oct., 1925 till March, 1926, an extended experiment with steel models of large-scale was continually carried out at the Kure Naval Dockyard with a view to obtaining the data necessary for designing the submarine pressure-hull. The total number of models tested was 45, which was divided into 6 series. The influence on the collapsing pressure of steel models has been studied, (1) when the frame space is varied, (2) when three different sizes are chosen, but similarly constructed, (3) when the shell joint is welded or riveted with double straps, (4) when the frame scantlings are slightly

varied, (5) when the shell is of oval section, (6) when frames of special dimensions are fitted, etc. Unfortunately, I am not at liberty to disclose the details of the experimental results, but the conclusions arrived at with reference to the formula already cited may be summarized as follows:—

(i) For frameless cylinders both ends of which are kept circular without restricting the slope of the cylinder wall, the calculated collapsing pressure agrees exactly with the observed provided that the hoop stress does not exceed the proportional limit of the material.

(ii) For cross-stiffened cylinders, the "frame factor" k is introduced in the formula and k is calculated reversely by inserting actual data; it is found that if the models are similar, the value of k is practically constant independent of the frame space, of the absolute size of the models, and, to a certain extent, independent of the size of frames. The observed number of lobes is found to be in good agreement with that of the calculated from the formula involving k. Therefore, if the actual mean thickness of the cylinder shell of standard workmanship is measured prior to the test, then the value of k will probably be obtained as practically constant. However, in the similarity test carried out in 1918 at Danzig by the German Navy, it was observed that the smaller was the model, the smaller was the collapsing pressure compared with the calculated. This discrepancy has been attributed to the fact that the imprecision of construction will weaken more its resistance in the case of smaller models. Also by R. von Mises* it is attributed to the fact that the smaller is the model, the more is impaired its circularity at the time when the shell is welded, and the less is attained the uniformity of material; consequently the collapsing pressure falls short of that calculated. In my belief, however, the law of similitude holds even in this case if the thickness of the shell be carefully measured and k be calculated inversely. In the published data† of Danzig experiment, the thicknesses of plates are given in round number; but this will never be the case with commercial thin steel plates if carefully measured. Moreover, considering the mode of manufacture, the proportional limit of the thin plate is comparatively high, so that the collapsing pressure of the small models which frames closely spaced must comparatively be higher.

(iii) According to the experiment, the ratio of the collapsing pressure observed to the calculated decreases linearly from the point where the corresponding hoop stress exceeds the proportional limit of the material to the point where the stress-failure takes place.

(iv) The formula for calculating the collapsing pressures of oval cylinders is also obtained.

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* In his recent paper already cited.

† Sanden, K. V. & K. Guenther, "Ueber des Festigkeitsproblem quersteifter Hohlzylinder unter allseitig gleichmaessigen Aeussendruck." Werft—Reederei—Hafen, Vol. I, No. 8, 9, 10; Vol. II. No. 17. (1920–'21).

(v) There were a few instances observed in which the cylinder collapsed bodily, i.e., both the shell and frames as a whole, but at time I could not succeed in accounting for this case.

4. *A Small-Scale Experiment with Brass Models conducted at the Naval Technical Research Institute.* Having obtained the necessary data for designing submarine pressure-hulls from the Kure experiment, I have proceeded to carry out the experiment with small-scale models to obtain more detailed informations at the Naval Technical Research Institute. Brass models were used in the present experiment because of the difficulty in obtaining small-scale steel models of homogeneous quality. The standard dimensions of the models are as follows:—10 cm. in diameter, 0.5 mm. in thickness, stiffeners are generally of rectangular section, its breath being 3 mm. Cold drawn brass tubes of ordinary commercial thickness were carefully machined down to the dimensions above prescribed with frames of rectangular section, so that the shell and frames are of the body. As the measurement of thickness of the shell plays an important part, each shell is measured at 8 points on the circumference equal distance apart, 4 points each lengthwise, i.e., 32 points in all, and the average value of these measurements is used in the calculation; the variation in thickness from the average did not exceed 5 per cent. The general arrangement of experimental apparatus may easily be gathered from Plate III. The pressure was raised at a rate of 2 kg. per square cm. per minute, and the following are the mean value of the physical qualities of brass tubes:—

Young's modulus	1,070,000 kg./cm.^{2}
Proportional limit	1,200 kg./cm.^{2}
Ultimate strength	4,360 kg./cm.^{2}
Yield-point	not observed clearly
Elongation over 5 cm.	38%

The particulars of models are as follows:—

(i) Series H.*—These are frameless models and the object of the experiment with this series is to precisely measure the fall of the ratio of the observed collapsing pressure to the calculated upwards from the corresponding hoop-stress exceeds the proportional limit of the material. The ends were kept circular without restricting the slope of the cylinder wall, full particulars are given in Pl. IV.

(ii) Series J.—These are cross-stiffened cylinders and of the particulars as follows:—

Frame space in mm.	35, 30, 25, 20, 15.
Free span in mm.	32, 27, 22, 17, 12.
Mark of series	J_1, J_2, J_3, J_4, J_5.

Nine models were made for each series, and keeping the breadth of stiffener

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* The numbering of the series has been made to run continuously from the previous experiment.

constant, 3 mm., and the depth d was varied as 5, 4, 3, 2, 1.5, 1.25, 1.0, .75, .5 mm. from Model 1 to 9. These models were intended to find the criterion which indicates up to what size of frames the shall alone collapses and the frames remain intact, below what size of frames both the shall and frames collapse bodily; and in addition, to determine the effect of frame space on this criterion.

(iii) Series K.—These are intended to study the effect of the frames on the collapsing pressure of the shell by comparing the models having stiffeners of 4 mm. deep with H 7, 8, 12, 13, 14, 15, 16, 18, 20, which have rigid ends alone and no stiffener. Most of the models were not made specially, the data were used from the results of the models which belong to Model No. 2 of J_1 to J_5, and in addition, K7, K8, K12, K20 were made afresh. They are called K7.8, 12, 13, 14, 16, 18, 20, the unsupported length between frames of each model being made correspondingly equal to that of H (see column 11 Pl. IV, cf. Models No. 2, J_1-J_5, Pl. VI).

5. *Conclusions derived from the Results obtained in the Experiment with Series* H, J *and others.* Looking over the obtained results given in Pl. IV—XII, we may conclude as follows:—

(i) Unless the hoop stress exceeds the proportional limit of the material, the collapsing pressure and the number of lobes observed are in good agreement with those calculated from (2) for frameless cylinders, even in the experiment with brass models.

(ii) Over the range upwards from where the hoop stress reaches the proportional limit of the material to where the axial stress reaches the yield-point and the so-called pure elastic breakdown takes place, the ratio of the collapsing pressure observed to that of calculated is plotted against the length of the cylinder, and it is found that they hold a linear relation as shown in Pl. V.*

(iii) When pure elastic break-down takes place, the cylinder is bulged in as one groove along the circumference.

(iv) In the region where the hoop stress exceeds the proportional limit, yet does not reach the pure elastic breakdown, the form of lobe becomes irregular and presents some wavy form.

(v) Within the limits of experiment, up to $(i_0/r)=0.0072$ for $(s/D)=0.005$, the shell alone collapses, below that value the model collapses bodily; and according as the decrease in frame space collapsion tends to occur in the case where deeper frames are provided (see Pl. VIII).

(vi) The value of the collapsing pressure of the shell between frames agrees very well with that of the frameless model fixed rigidly at both ends if the total length of which is made equal to the frame spaces of the former. In other words, when shell alone collapses, $k=1$ for the model whose shell and the frames are of one body, as is observed in the case of

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* Handbuch der Physik, Bd. VI, (1928) p. 301., and Geckeler, J. W., ZS. f. angew. Math. u. Mech., 1928, Oct. Bd. VII. p. 341.

frameless models; that is to say, such frames reinforce the shell as equally as rigid ends before the limiting condition, in which bodily collapse takes place, is reached.

vii) It is manifest from the results of Series J, that the formula for bodily collapse (10) is exactly in consistence with experiment before the hoop stress reaches the proportional limit of the material. If the said stress exceeds the proportional limit, a correction is needed (see Pl. IX).

(viii) In Series J, it is found that $k=1$; that is to say, the ideal conditions assumed in theory are practically realized. This fact is also observed to be realized in the case of small-scale cylinder with stiffeners, when the surface of contact of both is exactly fitted and then riveted. On the other hand, the results of the experiment with large-scale cross-stiffened models conducted at Kure show that always $k<1$; namely, the reduction in strength due to imperfections which violate the ideal assumptions is observed. A cylinder consisting of the shell and frames of separate pieces must necessarily be fitted together by some means or other. In this course of manufacture, the surfaces of contact of both being never found to be geometrically true and similar,, they are bound to conform each other and adapt themselves and in consequence the true circularity of the model along frame lines is impaired. This fact probably accounts for the variation of k.

Some instances of riveted cylinders in which $k<1$ may be cited as follows:—

(Compare to Pl. XI & XII.)

Mode of Collapse	D cm.	s cm.	l cm.	L cm.	p_k (n)			k	Frame $b \times d$ cm.
					Observed kg./cm.^{2}	Calculated kg./cm.^{2}	Ratio		
Shell Alone Collapses	10	.0261	3.2	13.7	2.46 (11)	3.46 (12)	.712	.73	$b=0.3$, $d=0.4$
	〃	.0254	2.7	11.7	2.39 (11)	3.90 (13)	.613	.635	
	〃	.0288	2.2	9.7	4.43 (12)	6.78 (14)	.653	.68	
	〃	.0252	1.7	7.7	4.36 (12)	6.43 (15)	.677	.705	
	〃	.0253	1.2	5.7	6.40(14-15)	9.82 (17)	.693	.73	
	〃	.0225	0.7	3.7	6.68(no clear)	14.42(19-20)	.552	.65	
	10	.0504	1.2	5.7	28.83(not clear)	60.35 (13)	.797	.83	
Bodily Collapse	10	.0258	2.7	11.7	2.04	5.39	.378		$b=0.3$, $d=0.75$
	〃	0.252	1.7	7.7	3.23	12.01	.269		
	〃	.0259	1.2	5.7	5.55	18.05	.308		
Remarks	These models are consisted of the shell and frames of separate pieces fitted together by 1.5 mm. tack rivets spaced some 10 dia. apart, and the frames are finished 0.50 mm. less than the shell.								

But we have yet to ascertain whether k is dependent on the absolute value of s by future experiment. Nevertheless, according to the results of the large-scale model-experiment, it is found that the value of k is at least independent of s/D, l, and frame size to a certain extent.

(ix) In the case where bodily collapse takes place, the further research whether the effect of frame factor is more marked for riveted cross-stiffened cylinder than for the case above cited is still wanting.

The investigation on the mode of collapse of cross-stiffened cylinders from the standpoint of vibration, may be of interest. In connexion with this, I have observed a phenomenon in the cource of experiment that if a cross stiffened cylinder filled with water is lightly hammered from outside, the number of sets of waves on the water surface is found to be twice as many that of the lobes into which the shell collapses under uniform external pressure, Fig. 4, Pl. II, shows an instance.

Note I. Derivation of the Formula.

In what follows, an attempt will be made to derive a formula which may be a better guide than any other existing formula for practical design of submarine pressure-hulls.

In the first place, we assume, that the end constraint tends to maintain the circularity of the cylinder at the sections reinforced by stiffening frames, without restricting the slope of the cylinder wall; but the combined effect of radial and end pressures will be taken into account. In the second place, we distinguish, as usual, two configurations: (1) the position of equilibrium, in which the cylinder remain circular and merely suffers contractions; this is the configuration of investigating the stability; and (2) a configuration of infinitesimal displacement from this.

It is evident, from the nature of the problem, that all the stress-resultants and stress-couples except that axial and hoop stresses are zero for the first configuration. Adopting the notation shown in Fig. 2, the equations of equilibrium are

$$r\frac{\partial[T_1]}{\partial x}=0,$$

$$\frac{\partial[T_2]}{\partial x}=0,$$

$$[T_2]+pr=0,$$

where the symboles in brackets indicate the stress-resultants for the first configuration: $[T_1]$ is equal to the constant end thrust P'; $[T_2]$ the hoop compression pr:—viz.,

$$[T_1]=-P',$$

$$[T_2]=-pr.$$

In proceeding to the second configuration, we have now to suppose that a small additional displacement is superposed upon the displacement answering to the first configuration. Let T_1, N_1, S_1, T_2, N_2, S_2 be the stress resultants, and G_1, H_1, G_2, H_2 be the stress-couples calculated from this displacement as shown in Fig. 2. Then we have

$$[T_1]+T_1=-P'+T_1,$$
$$[T_2]+T_2=-pr+T_2,$$

where T_1, T_2 are infinitesimal increments due to distortion.

In a similar manner to the method which Prof. Love* adopted, we have for the stress-resultants,

$$\text{(i)}\qquad \left\{\begin{aligned} &\frac{\partial T_1}{\partial x}-\frac{\partial S_2}{r\partial\phi}+p\left(\frac{\partial^2 v}{\partial x\partial\phi}-\frac{\partial w}{\partial x}\right)=0,\\ &\frac{\partial S_1}{\partial x}+\frac{\partial T_2}{r\partial\phi}-\frac{N_2}{r}-\frac{\partial^2 v}{\partial x^2}P'=0,\\ &\frac{\partial N_1}{\partial x}+\frac{\partial N_2}{r\partial\phi}+\frac{T_2}{r}-\frac{\partial^2 w}{\partial x^2}P'-\frac{p}{r}\left(w+\frac{\partial^2 w}{\partial\phi^2}\right)=0\end{aligned}\right.$$

and for the stress-couples,

$$\text{(ii)}\qquad \left\{\begin{aligned} &\frac{\partial H_1}{\partial x}-\frac{\partial G_2}{r\partial\phi}+N_2=0,\\ &\frac{\partial G_1}{\partial x}+\frac{\partial H_2}{r\partial\phi}-N_1=0,\\ &H_2+(S_1+S_2)r=0,\end{aligned}\right.$$

where v, w are displacement-components of a point on the middle surface of the cylinder wall in the directions of ϕ, r (positive if inward).

Further, from the third equation of (ii), we have

$$\text{(iii)}\qquad S_2=-\frac{H_2}{r}-S_1\ ;$$

from the condition of symmetrical loading,

$$\text{(iv)}\qquad -H_2=H_1=H,\ \text{say,}^{**}$$

from the first and second equation of (ii),

$$N_1=\frac{\partial G_1}{\partial x}+\frac{\partial H_2}{r\partial\phi}=\frac{\partial G_1}{\partial x}-\frac{\partial H}{r\partial\phi},$$
$$N_2=-\frac{\partial H_1}{\partial x}+\frac{\partial G_2}{r\partial\phi}=-\frac{\partial H}{\partial x}+\frac{\partial G_2}{r\partial\phi},$$

Now, differentiate N_1, N_2 with respect to x and ϕ respectively, and substitute these values and (iv) in (i), then it follows:—

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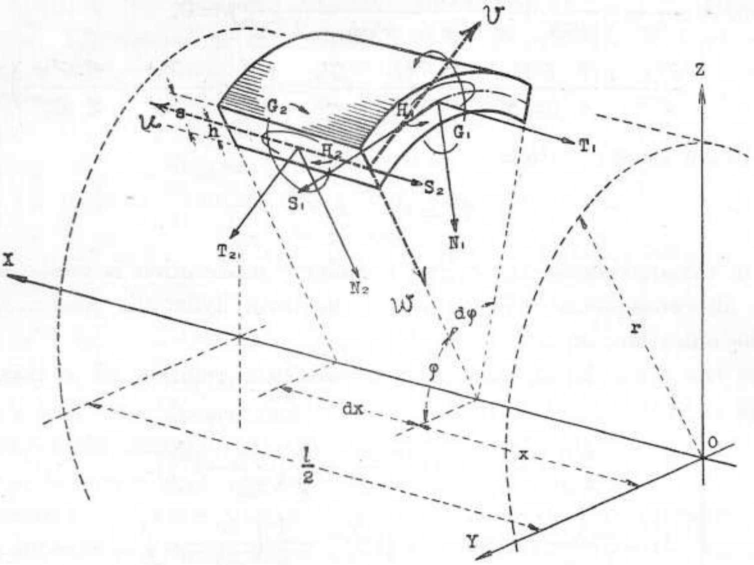
* Love, A. E. H., Mathematical Theory of Elasticity, 4th Ed., pp. 571 et seq.

** Love, l.c., p. 530.

Fig. 2.



STRESS SYSTEM IN THE SECOND CONFIGURATION OF EQUILIBRIUM.

Fig. 3.

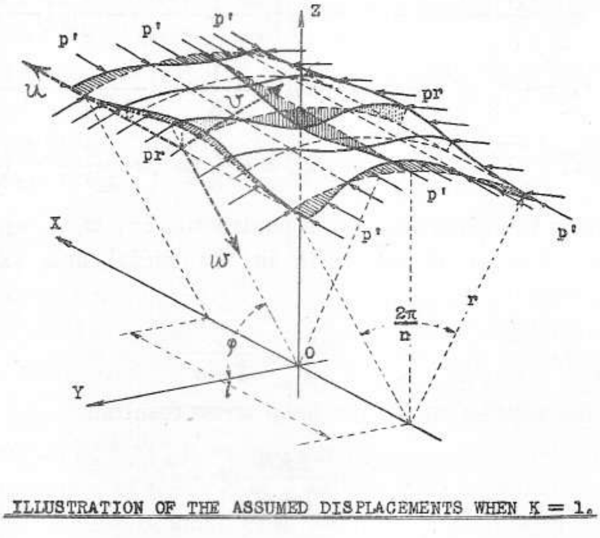


ILLUSTRATION OF THE ASSUMED DISPLACEMENTS WHEN K = 1.

$$(\mathrm{v})\qquad \left\{\begin{aligned} &\frac{\partial T_1}{\partial x}-\frac{\partial S_2}{r\partial\phi}-p\left(\frac{\partial^2 v}{\partial x\partial\phi}-\frac{\partial w}{\partial x}\right)=0,\\ &\frac{\partial S_1}{\partial x}+\frac{\partial T_2}{r\partial\phi}+\frac{1}{r}\frac{\partial H}{\partial x}-\frac{\partial G_2}{r^2\partial\phi}-\frac{r}{2}\frac{\partial^2 v}{\partial x^2}p=0,\\ &\frac{\partial^2 G_1}{\partial x^2}+\frac{2}{r}\frac{\partial^2 H}{\partial x\partial\phi}+\frac{1}{r^2}\frac{\partial^2 G_2}{\partial\phi^2}+\frac{T_2}{r}=\frac{p}{r}\left(w+\frac{\partial^2 w}{\partial\phi^2}+\frac{r^2}{2}\frac{\partial^2 w}{\partial x^2}\right), \end{aligned}\right.$$

In obtaining these equations,

$$P'=\frac{p\pi r^2}{2\pi r}=\frac{pr}{2}$$

has been taken; because the cylinder under consideration is closed at ends and is, in consequence, subjected to a uniform hydraulic pressure, p all over the outer surfaces.

On the other hand, from the stress-strain relation of a thin shell, we have

$$(\mathrm{vi})\qquad \left\{\begin{aligned} &T_1=\frac{2Eh}{1-\sigma^2}(\epsilon_1+\sigma\epsilon_2)=c\left\{\frac{\partial u}{\partial x}+\frac{\sigma}{r}\left(\frac{\partial v}{\partial\phi}-w\right)\right\},\\ &T_2=\frac{2Eh}{1-\sigma^2}(\epsilon_2+\sigma\epsilon_1)=c\left\{\sigma\frac{\partial u}{\partial x}+\frac{1}{r}\left(\frac{\partial v}{\partial\phi}-w\right)\right\},\\ &H=\frac{2}{3}\frac{Eh^3}{1+\sigma}\tau=c(1-\sigma)\frac{h^2}{3}\cdot\frac{1}{r}\left(\frac{\partial^2 w}{\partial x\partial\phi}+\frac{\partial v}{\partial x}\right),\\ &S_1=\frac{Eh}{1+\sigma}\varpi=\frac{(1-\sigma)}{2}c\left\{\frac{\partial v}{\partial x}+\frac{1}{r}\frac{\partial u}{\partial\phi}+\frac{h^2}{3r^2}\left(\frac{\partial^2 w}{\partial x\partial\phi}+\frac{\partial v}{\delta x}\right)\right\},\\ &S_2=\frac{H}{r}-S_1=-\frac{(1-\sigma)}{2}c\left\{\frac{\partial v}{\partial x}+\frac{1}{r}\frac{\partial u}{\partial\phi}-\frac{h^2}{3r^2}\left(\frac{\partial^2 w}{\partial x\partial\phi}+\frac{\partial v}{\partial x}\right)\right\},\\ &G_1=-\frac{2}{3}\frac{Eh^3}{1-\sigma^2}(\kappa_1+\sigma\kappa_2)=-\frac{1}{3}ch^2\left\{\frac{\partial^2 w}{\partial x^2}+\frac{\sigma}{r^2}\left(\frac{\partial^2 w}{\partial\phi^2}+\frac{\partial v}{\partial\phi}\right)\right\},\\ &G_2=-\frac{2}{3}\frac{Eh^3}{1-\sigma^2}(\kappa_2+\sigma\kappa_1)=-\frac{1}{3}ch^2\left\{\sigma\frac{\partial^2 w}{\partial x^2}+\frac{1}{r^2}\left(\frac{\partial^2 w}{\partial\phi^2}+\frac{\partial v}{\partial\phi}\right)\right\}, \end{aligned}\right.$$

where ϵ_1, ϵ_2 are the axial and radial contraction; τ, ϖ, κ_1, κ_2, the slide, the rotation, the changes of curvature in the radial and axial directions respectively; and

$$c=\frac{2Eh}{1-\sigma^2}=\frac{Es}{1-\sigma^2}.$$

Further, in the expression for the hoop stress-resultant

$$T_2=pr=\frac{2Eh}{1-\sigma^2}(\epsilon_2+\sigma\epsilon_1),$$

if we put

$$\epsilon_2+\sigma\epsilon_1=\eta,$$

which can be regarded as the "peripheral contraction" of the shell, then we have

(vii) $$p=\frac{2Eh}{1-\sigma^2}\frac{\eta}{r}=c\frac{\eta}{r}.$$

In equation (v) we are to substitute the values given by equation (vi) and (vii). We assume as expressions for the components of displacement of the forms (see Fig. 3.)

$$\begin{aligned} u&=V\sin m\,x\sin n\phi,\\ v&=V\cos m\,x\cos n\phi,\\ w&=W\cos m\,x\sin n\phi, \end{aligned}$$

where n is the number of lobes into which the circumference divides at the moment of collapse,

$$m=k\frac{\pi}{l},$$

in which k is the "frame-factor" that is to be determined from the experiments, l the unsupported axial length of the cylinder.

It is, however, apparent that circularity of the cylinder at the edges of stiffeners i.e., at $x=\pm\frac{1}{2}l$, cannot be realized in the built-up construction such as submarine pressure-hulls. Thus the value of k must be other than unity (in the present case $k<1$); nevertheless, in all other theories, it is usually assumed that $k=1$ and $v=w=0$ at $x=\pm\frac{1}{2}l$:—viz., both ends maintain the perfect circularity of the same radius under pressure.

Adopting this modification and putting $\xi=h^2/3r^2$, equations (v) become

(viii) $$\left\{\begin{aligned} &-\left(m^2+\frac{1-\sigma}{2}\frac{n^2}{r^2}\right)U+\left(\frac{1+\sigma}{2}+\eta-\frac{1-\sigma}{2}n^2\xi\right)\frac{mn}{r}V\\ &\qquad\qquad+\left(\sigma+\eta-\frac{1-\sigma}{2}n^2\xi\right)\frac{m}{r}W=0,\\ &-\frac{1+\sigma}{2}\frac{mn}{r}U+\left\{\frac{1-\sigma}{2}m^2+\frac{n^2}{r^2}+\frac{m^2}{2}\eta+\frac{3(1-\sigma)}{2}m^2\xi+\frac{n^2}{r^2}\xi\right\}V\\ &\qquad\qquad+\left(\frac{n}{r^2}+\frac{3-\sigma}{2}m^2n\xi+\frac{n^3}{r^2}\xi\right)W=0,\\ &-\frac{\sigma m}{r}U+\left(\frac{n}{r^2}+\frac{2-\sigma}{3}m^2n\xi+\frac{n^3}{r^2}\xi\right)V+\left\{\frac{1}{r^2}-\frac{1}{r^2}(n^2-1)\eta-\frac{m^2}{2}\eta\right.\\ &\qquad\qquad\left.+m^4r^2\xi+2m^2n^2\xi+\frac{n^4}{r}\right\}W=0. \end{aligned}\right.$$

The elimination of U, V, W from these equations lead to a determinantal equation determining η in terms of r, ξ, m, n. This equation may simplified by observing that η must be small of order $(r/h)\ (p/E)$, and we may therefore approximate by omitting terms of the order η^2 or $\eta\xi$. We may also omit terms containing ξ^2; but, instead of the case treated by Love, we retain the terms containing $m\xi$, in which m is of the order$1/l$; because l may sometimes be rather small in the present case.

The determinantal equation is then

$$\text{(ix)}\quad \begin{vmatrix} -\left(m^2-\frac{1-\sigma}{2}\frac{n^2}{r^2}\right) & \frac{(1+\sigma)}{2}\frac{mn}{r}-\frac{(1-\sigma)}{2}\frac{mn^3}{r}\xi+\frac{mn}{r}y & \sigma\frac{m}{r}-\frac{(1-\sigma)}{2}\frac{mn^2}{r}\xi+\frac{m}{r}\eta \\ -\frac{(1+\sigma)}{2}\frac{mn}{r} & \left(\frac{1-\sigma}{2}m^2+\frac{n^2}{r^2}\right)+\left\{\frac{3(1-\sigma)}{2}m^2+\frac{n^2}{r^2}\right\}\xi+\frac{m^2}{2}\eta & \frac{n}{r^2}+\left(\frac{3-\sigma}{2}m^2n+\frac{n^3}{r^2}\right)\xi \\ -\frac{\sigma m}{r} & \frac{n}{r^2}+\left\{(2-\sigma)m^2n+\frac{n^3}{r^2}\right\}\xi & \frac{1}{r^2}+\left(m^4r^2+2m^2n^2+\frac{n^4}{r^2}\right)\xi-\left\{\frac{(n^2-1)}{r}+\frac{m^2}{2}\right\}\eta \end{vmatrix}=0,$$

and, when we develope the determinant, neglecting terms of the orders mentioned above, and dividing throughout by $(1-\sigma)m^4/2r^2$, and putting $mr=\alpha$, we find

$$-\eta\left\{\left(\frac{n^2+\alpha^2}{\alpha^2}\right)^2\left(n^2-1+\frac{1}{2}\alpha^2\right)-\frac{n^2+2\alpha^2}{2\alpha^2}\right\}+(1-\sigma^2)$$
$$+\xi\left\{\frac{(n^2+\alpha^2)^4}{\alpha^4}-\frac{n^4(2n^2-1)}{\alpha^4}+\frac{(\sigma+7)n^2}{2\alpha^2}-\frac{(\sigma+15)n^4}{2\alpha^2}+\frac{(2\sigma^2-14)}{2}+3(1-\sigma)^2\right\}=0,$$

or approximately

$$\text{(x)}\qquad \eta=\frac{1}{n^2-1+\frac{1}{2}\alpha^2}\left[\frac{(1-\sigma^2)\alpha^4}{(n^2+\alpha^2)^2}+\xi\left\{(n^2+\alpha^2)^2-\frac{n^4(2n^2-1)}{(n^2+\alpha^2)^2}\right\}\right].$$

By the help of (vii), we have

$$p=\frac{2Eh}{1-\sigma^2}\frac{\eta}{r}\quad\text{and}\quad \xi=\frac{h^2}{3r^2}=\frac{s^2}{3D^2}$$

Substituting these values in (x), we finally obtain the equation giving the critical value of p consistent with stability, in terms of the thickness, the mean diameter, the unsupported axial length, and elastic constants of the material as follows:—

$$\text{(xi)}\qquad p_\kappa=\frac{E\frac{s}{D}}{n^2-1+\frac{1}{2}\alpha^2}\left[\frac{2\alpha}{(n^2+\alpha^2)^2}+\frac{2}{3}\frac{1}{1-\sigma^2}\left(\frac{s}{D}\right)^2\left\{(n^2+\alpha^2)^2-\frac{n^4(2n^2-1)}{(n^2+\alpha^2)^2}\right\}\right].$$

Note II. The Use of the Nomogram.

Referring to the nomogram in Pl. I, we see a net work $(n,\ kD/l)$, composed of two systems of figured curves (n), (kD/l) crossing each other. If we take any point on this net work, a curve of both systems will pass through this point, and we may assign to the point a value of both (n) and of (kD/l) taking the values from the curves of the systems (n), (kD/l) which intersect in the point. The point has thus in a sense two values and is usually termed a binary point.

Suppose that we want to know the collapsing pressure of a cylinder whose s, D, l are given. First of all, taking a suitable value of the "frame-factor" k and we calculate kD/l, s/D.

For instance assume:—

$$s/D=0.0025,\ kD/l=5.0.$$

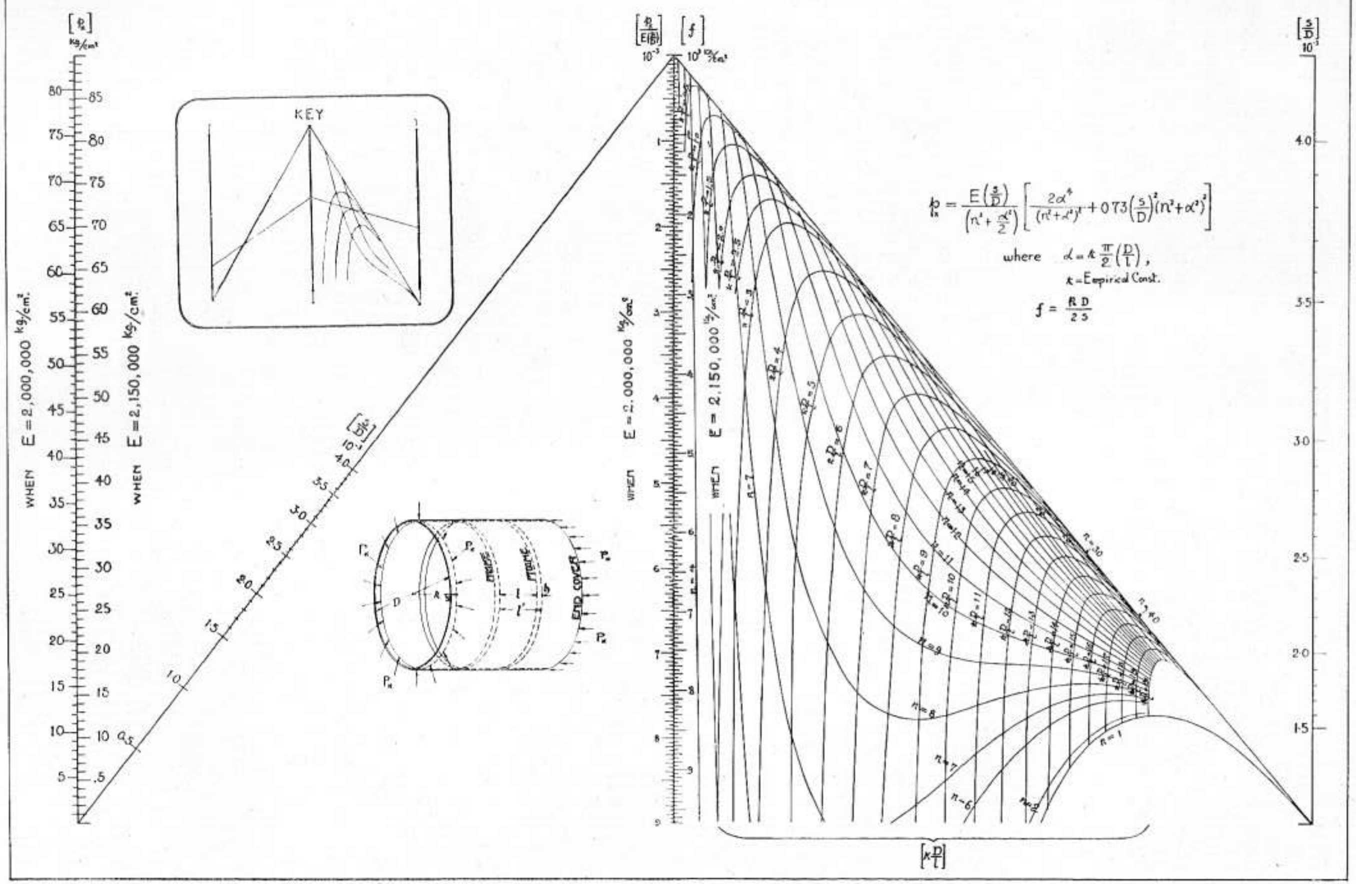
Now, the tangent passing through 0.0025 on the right-hand (s/D) scale to the (kD/l) curve marked 5.0, will cut the (f) scale at 1950, which gives the hoop stress corresponding to the required pressure. In reading the (f) scale, if Young's modulus of the material is 2,150,000 kg./cm.^{2}, then the figure on the right-hand side of the scale must be taken as is done in the present problem; but if $E=2{,}000{,}000$ kg./cm.^{2}, the left-hand side. As has already been stated, the touching point of the (kD/l) curve is a binary point, so that the (n) curve marked 15 crossing this point gives the value of number of lobes.

On the other hand, if the straight line joining the point 1950 obtained on the (f) scale and 0.0025 on the oblique (s/D) scale is produced, then the straight line will cut the (p_k) scale at 10 kg./cm.^{2}, the required pressure. In reading this scale, a similar attention has to be paid for the value of E.

Strictly speaking, the method explained above is an approximation; theoretically the binary point must be so chosen as to give the least value of p for the (kD/l) curve assigned. For practical purpose, however, the approximation method is quite accurate enough.

PLATE—I.

NOMOCRAM FOR CALCULATINC COLLAPSINC PRESSURES.



$$p_k = \frac{E\left(\frac{s}{D}\right)}{\left(n^2+\frac{\alpha^2}{2}\right)}\left[\frac{2\alpha^4}{(n^2+\alpha^2)^2}+0{\cdot}73\left(\frac{s}{D}\right)^2(n^2+\alpha^2)^2\right]$$

where $\alpha = k\frac{\pi}{2}\left(\frac{D}{l}\right)$,

k = Empirical Const.

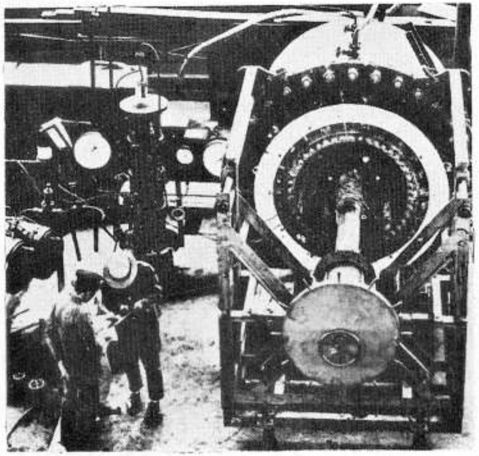
$$f = \frac{p_k D}{2s}$$

(Paper No. 651)

PLATE II.

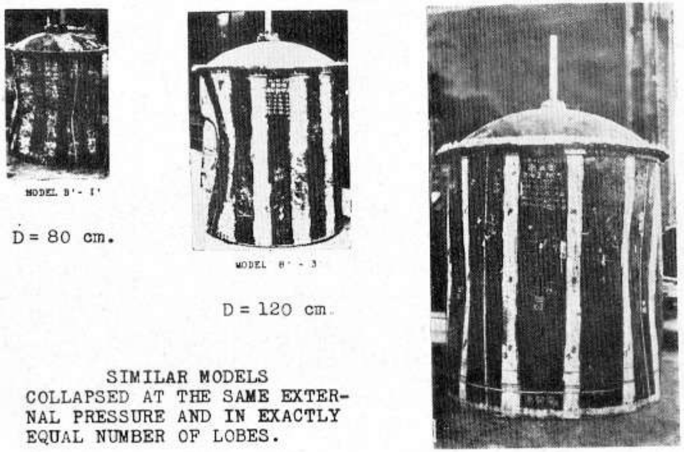
PHOTOGRAPHIC VIEWS OF THE APPARATUS USED IN THE LARGE-SCALE EXPERIMENT AND SEVERAL MODELS AFTER COLLAPSE.

FIG. 1.



A BIRD'S-EYE VIEW OF EXPERIMENTAL APPARATUS.

FIG. 2.



MODEL B'-1'

D = 80 cm.

MODEL B'-3

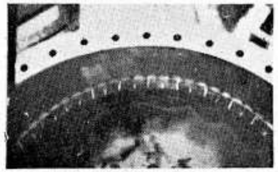
D = 120 cm.

MODEL B'-5'

D = 160 cm.

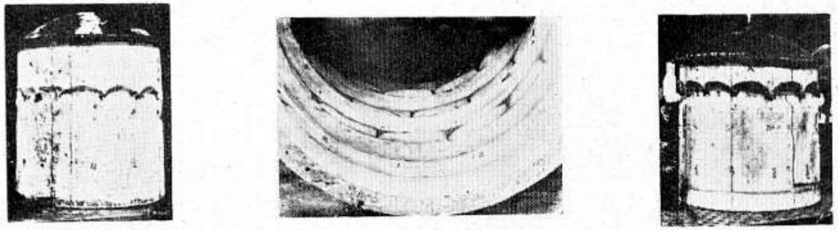
SIMILAR MODELS
COLLAPSED AT THE SAME EXTERNAL PRESSURE AND IN EXACTLY EQUAL NUMBER OF LOBES.

FIG. 4.



SETS OF WAVES ON THE WATER SURFACE WHEN THE SHELL HAMMERED FROM OUTSIDE.

FIG. 3.



MODEL B-5

MODEL B-4

MODEL B-6

TYPICAL EXAMPLES OF VIEWS OF THE MODELS AFTER COLLAPSE.

PHOTOGRAPHIC VIEWS AND DIAGRAMMATIC SKETCHES OF TESTING ARRANGEMENT USED IN THE PRESENT EXPERIMENT.

PLATE III.

FIG. I.

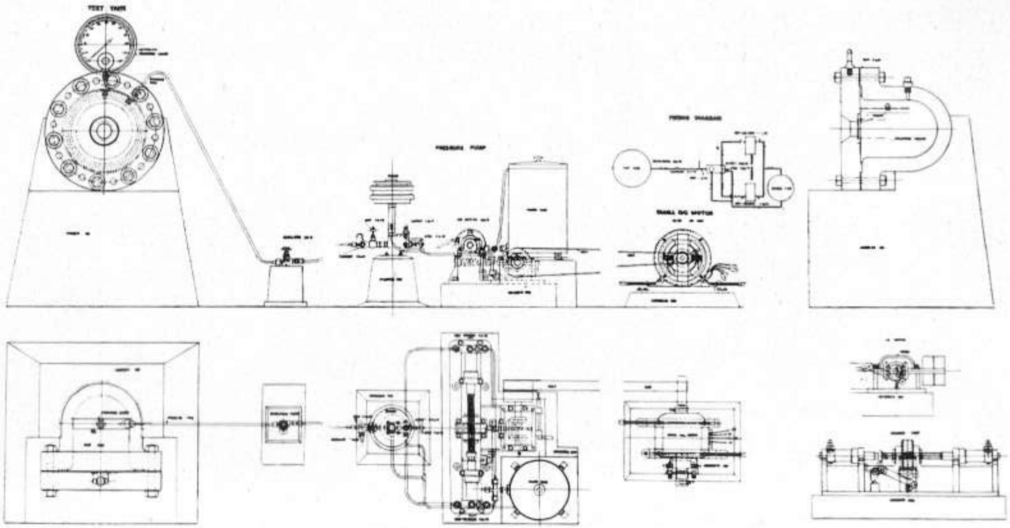


FIG. 2.

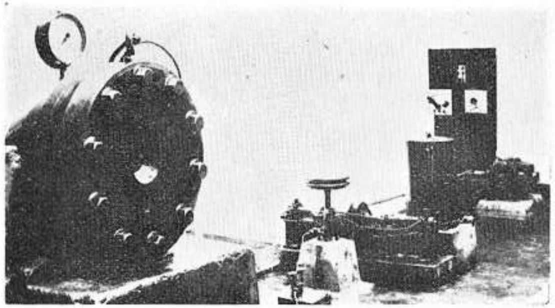


FIG. 3.

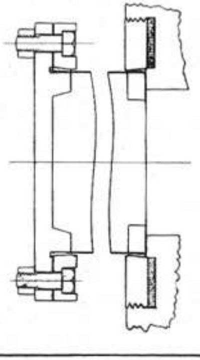


FIG. 4.

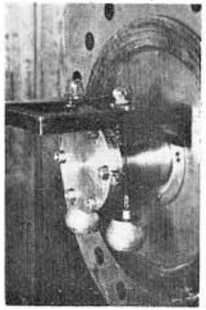


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PLATE IV.

ANALYSIS OF THE RESULTS OF THE EXPERIMENTS - I.

LENGTH OF MODEL OR ONE BAY. cm.	SERIES - H. (FRAMELESS MODELS)									SERIES - K. (CROSS-STIFFENED MODELS)						
	SERIES & NO. OF MODEL.	ACTUAL MEAN THICKNESS. mm.	COLLAPSING PRESSURE			NO. OF LOBES		STRESS		SERIES & NO. OF MODEL.	ACTUAL MEAN THICKNESS. mm.	COLLAPSING PRESSURE			NO. OF LOBES	
			OBSERVED Kg./cm².	CALCULATED Kg./cm².	RATIO	ACTUAL	CALCULATED	AMOUNT Kg./cm².	KIND			OBSERVED Kg./cm².	CALCULATED Kg./cm².	RATIO	ACTUAL	CALCULATED
15	H - 1	.513	3.87	3.82	1.015	5	5	377	HOOP STRESS							
12.5	H - 2	.505	4.04	4.48	.901	6	5	400								
10	H - 3	.503	5.41	5.51	.984	5-6	6	538								
7.5	H - 4	.515	7.87	7.90	.996	6	7	764								
6	H - 5	.498	9.35	9.32	1.004	8	7-8	937								
5.5	H - 6	.503	11.32	10.30	1.099	8	8	1125								
5	H - 7	.489	10.41	10.60	.984	8	8	1065		K - 7	.476	9.14	9.85	.927	8	8
4.7	H - 8	.501	12.37	12.05	1.026	9	8	1235		K - 8	.476	9.703	10.64	.911	8	8-9
4.5	H - 9	.474	10.195	11.03	.926	9	8-9	1075								
4.2	H-10	.488	12.16	12.80	.950	9	9	1245								
4.0	H-11	.501	13.64	14.26	.957	9	9	1362								
3.7	H-12	.498	13.57	15.36	.885	9	9	1360		K-12	.490	13.148	14.76	.890	9-10	9
3.2	H-13	.505	15.40	18.70	.823	10	10	1525		K-13 (J_1-2)	.512	15.046	19.42	.775	9	10
2.7	H-14	.485	14.98	20.46	.733	11	10-11	1545		K-14 (J_2-2)	.510	17.718	23.20	.763	9	10
2.2	H-15	.503	19.19	28.30	.677	12	11	1905		K-15 (J_3-2)	.515	20.53	30.06	.682	9	11
1.7	H-16	.511	24.12	40.20	.599	13-14	12	2360		K-16 (J_4-2)	.517	25.522	41.20	.620	10	12
1.5	H-17	.505	25.59	45.60	.563	14	12-13	2530								
1.2	H-18	.503	33.61	60.70	.553	13	13-14	3335		K-18 (J_5-2)	.515	34.521	63.70	.542	12	13
1.0	H-19	.504	33.89	77.20	.438	NOT CLEAR	13-14	3360								
.7	H-20	.501	42.89	128.70	.333	"		4286		K-20	.505	44.153	132.00	.334	NOT CLEAR	13
.5	H-21	.505	45.77	233.00	.198	"		4150	AXIAL STRESS#							
.3	H-22	.502	63.70	590.00	.108	"		4160								
.2	H-23	.498	74.80	1297.00	.058	"		4275								

CALCULATED FROM SANDEN'S FORMULA :- $f' = \{0.5 + 1.55\,L\}\frac{R_A}{\delta}$. (SEE JOHOW'S HILFSBUCH FUER DEN SCHIFFBAU, 4th. ED., pp. 1057-1058.)

PLATE V.

CURVES SHOWING THE EFFECT OF VARYING THE LENGTH OF BRASS MODEL ON THE COLLAPSING PRESSURE.

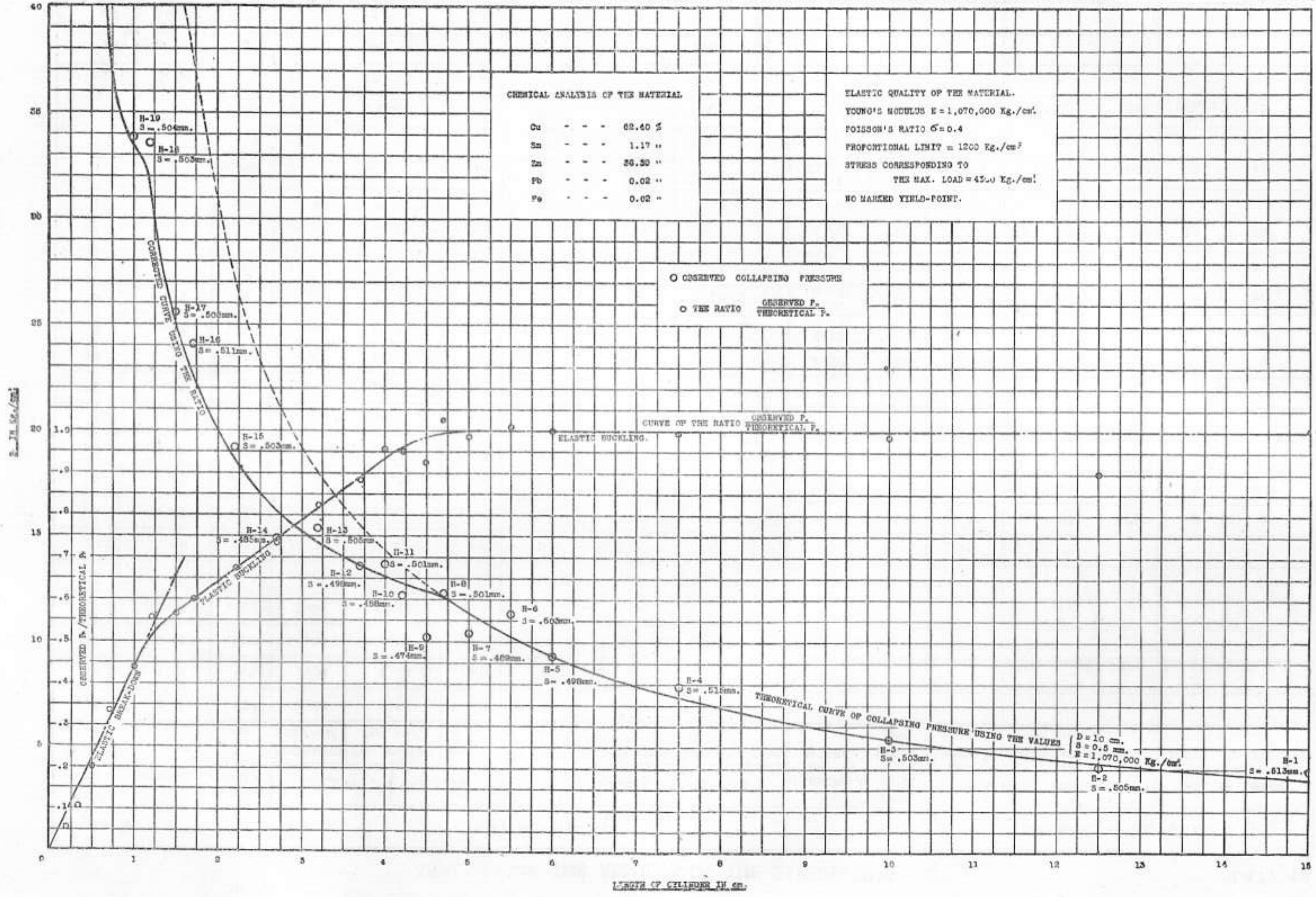

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PLATE VI.

PHOTOGRAPHIC VIEWS OF THE MODELS AFTER COLLAPSE - I.

MODEL NO.	s mm.	L cm.		OBSERVED P_K Kg./cm².	n
H-22	.502	0.3	D = 10 cm. FOR ALL THE MODELS. 	63.70	NOT CLEAR
H-21	.505	0.5		45.77	"
H-20	.501	0.7		42.89	"
H-19	.504	1.0		33.89	"
H-18	.503	1.2		33.61	13
H-17	.505	1.5		25.59	14
H-16	.511	1.7		24.12	13-14
H-13	.505	3.2		15.40	10
H-7	.489	5.0		10.41	8
H-4	.515	7.5		7.87	6

ANALYSIS OF THE RESULTS OF THE EXPERIMENTS – II. PLATE VII.

MODEL NUMBER.	d DEPTH OF RIBS. mm.	SERIES	J_1	J_2	J_3	J_4	J_5	MODE OF COLLAPSE.
		FREE SPAN l mm.	32	27	22	17	12	
		TOTAL LENGTH cm.	13.7	11.7	12.2	11.7	11.7	
1	5	s mm.	.495	.509	.515	.511	.506	SHELL BETWEEN FRAMES ALONE COLLAPSED.
		P_K kg./cm^2	15.61	17.58	20.04	28.83	34.80	
		n	9 (10)	10 (10)	9 (11)	9 (12)	11 (13)	
		RATIO[#]	.880	.758	.666	.718	.573	
2	4	s mm.	.512	.510	.515	.517	.515	
		P_K kg./cm^2	15.05	17.72	20.53	25.52	34.52	
		n	9 (10)	9 (10)	9 (11)	10 (12)	12 (13)	
		RATIO	.775	.763	.682	.620	.542	
3	3	s mm.	.511	.487	.515	.496	.473	
		P_K kg./cm^2	16.03	16.94	20.53	25.80	31.85	
		n	9 (10)	11 (10-11)	9 (11)	9 (12)	13 (13-14)	
		RATIO	.832	.820	.682	.698	.625	
4	2	s mm.	.504	.496	.528	.503	.513	
		P_K kg./cm^2	15.89	19.33	20.95	25.38	35.01	
		n	9 (10)	11 (10-11)	9 (11)	9 (12)	13 (13)	
		RATIO	.854	.891	.647	.661	.554	
5	1.5	s mm.	.503	.508	.524	.514	.507	BODILY COLLAPSED.
		P_K kg./cm^2	15.05	18.21	20.04	27.35	32.69	
		n	9 (10)	10-11 (10-11)	9-10 (11)	12 (12)	5 (4)	
		RATIO	.812	.792	.630	.571	.902	
6	1.25	s mm.	.490	.489	.503	.499	.494	
		P_K kg./cm^2	15.75	18.21	22.01	26.93	31.36	
		n	5 (4)	5 (4)	5 (4)	5 (4)	4-5 (4)	
		RATIO	.962	.873	.969	1.080	1.131	
7	1.0	s mm.	.521	.503	.520	.482	.495	
		P_K kg./cm^2	13.96	15.54	17.51	19.47	23.06	
		n	5 (4)	6 (4)	5 (4)	6 (4)	6 (4)	
		RATIO	1.096	.967	1.017	1.044	1.090	
8	.75	s mm.	.498	.484	.486	.493	.487	
		P_K kg./cm^2	9.85	12.16	12.80	14.90	14.96	
		n	6 (4)	5-6 (5)	5-6 (5)	5 (5)	5 (4)	
		RATIO	1.097	1.151	1.122	1.103	.978	
9	.5	s mm.	.514	.510	.518	.499	.516	
		P_K kg./cm^2	6.96	7.66	8.79	9.98	11.88	
		n	5 (5)	6 (5)	5 (5)	6 (5)	6 (5)	
		RATIO	1.070	.993	1.076	1.160	1.153	

s = MEAN THICKNESS ACTUALLY MEASURED. P_K = COLLAPSING PRESSURE OBSERVED.
n = NUMBER OF LOBES OBSERVED (CALCULATED).
[#] THE RATIO OF THE PRESSURE OBSERVED TO CALCULATED.

PLATE X - A.

PHOTOGRAPHIC VIEWS OF THE MODELS AFTER COLLAPSE - II. (A)

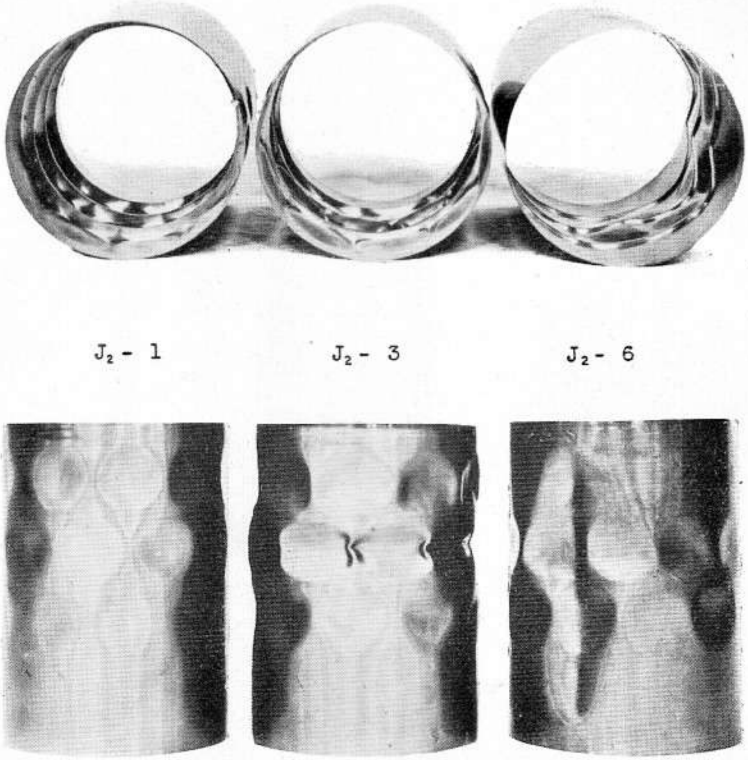
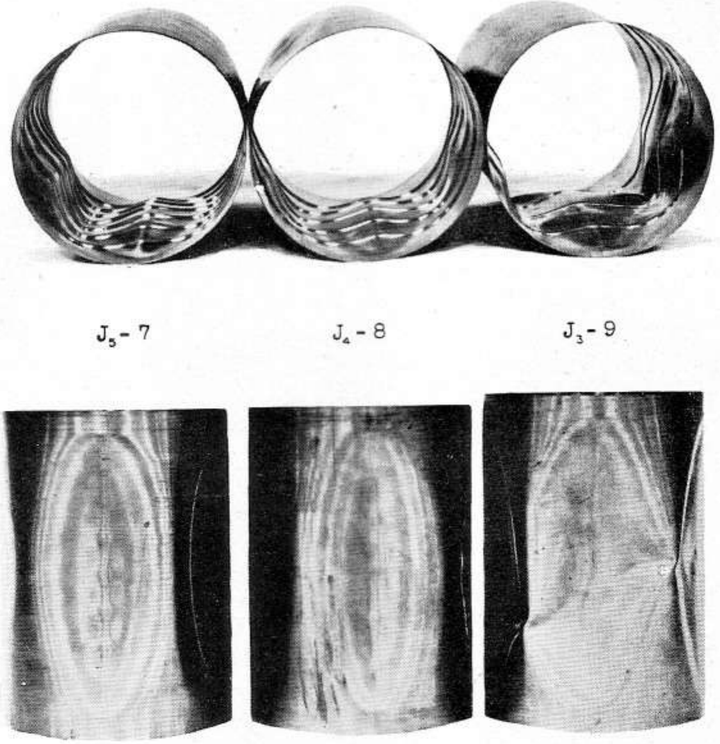


PLATE X-B.

PHOTOGRAPHIC VIEWS OF THE MODELS AFTER COLLAPSE - II.(B)



J_5- 7 J_4- 8 J_3- 9

PLATE XI.

CURVES SHOWING THE STIFFENING EFFECT OF RIVETED FRAMES COMPARED WITH THAT OF SOLID RIBS.

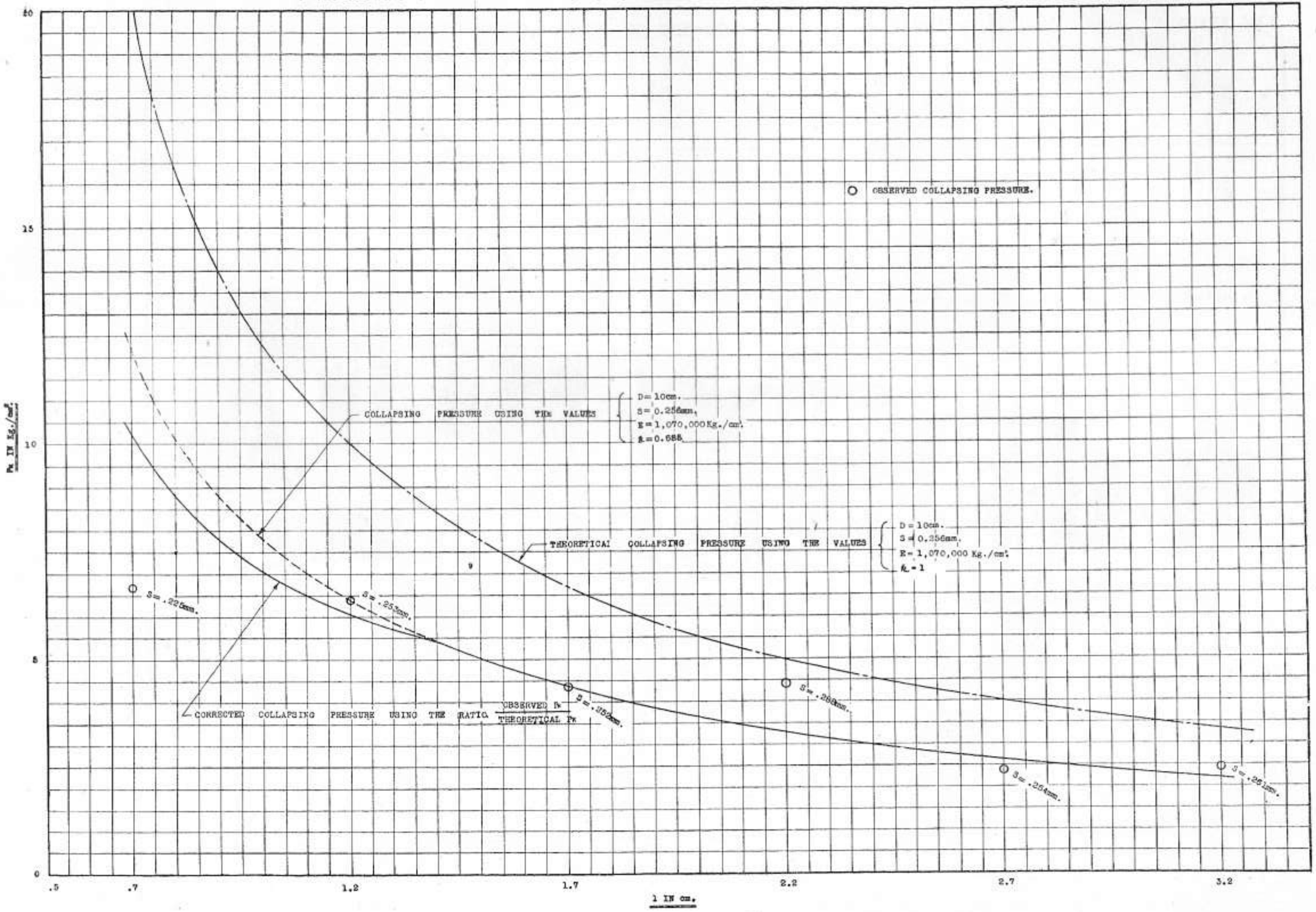
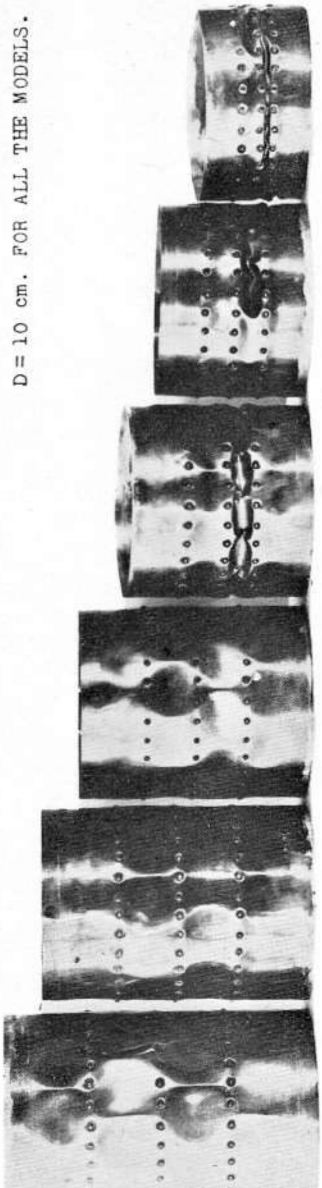


PLATE XII.

PHOTOGRAPHIC VIEWS OF THE MODELS AFTER COLLAPSE - III.

s mm.	l cm.	D = 10 cm. FOR ALL THE MODELS.	OBSERVED P_{κ} Kg./cm^2	n
.225	0.7		6.68	NOT CLEAR
.253	1.2		6.40	14
.252	1.7		4.36	12
.288	2.2		4.43	12
.254	2.7		2.39	11
.261	3.2		2.46	11

On the After-War Development of the Ships of the Imperial Navy

(Paper No. 652)

By ***Yuzuru Hiraga,*** *D. Eng.,*
Constructor Vice-Admiral, I.J.N.

Introduction

It was after the two wars of 1894–1895 and 1904–1905 that the Empire of Japan was, for the first time, ranked with the Great Powers of Europe and America, and since then the patriotism and *esprit de corps* of the nation has brought about the prosperity of the present day. During this period, our shipbuilding industry has also undergone several changes, that is to say, prior to the 1904–1905 War the principal warships of our navy were all built at the leading countries in Europe and America, and we were merely on the stage of learning the shipbuilding technic from these ships.

Indeed, it was during the 1904–1905 War the design and construction of capital ships were, for the first time, undertaken in this country and since then a great stride was made, and during the period between the end of that War and the beginning of the Great-War, battleships of Kawachi and Fuso classes, cruisers of Chikuma class and destroyers of Umikaze and Sakura classes were designed and constructed with satisfactory results, exhibiting a decided advance in our warship design and construction.

During this period, the battle-cruiser Kongo was ordered from the Vickers', England, by way of introducing the latest technic to our ships, which was followed by the orders for two ships placed on the Mitsubishi yard and Kawasaki yard to augment the resources for building capital ships in this country. The experience gained in building these ships no doubt helped in a large measure to establish a new era in the general shipbuilding history of this country. The outbreak of the Great-War was an opportunity to realize our warship design on various types of ships which may be looked upon as our navy-type, and by the time of Armistice, the battleships Nagato class, Kaga class, the cruisers Tenryu and Kuma classes, and the aircraft carrier Hosho had been designed and constructed. After the Great-War, the designs for the battle-cruisers Amagi class, the battleships Kii class were completed, also the designs for the cruiser Yubari and cruisers Furutaka class were completed. At the same time many specialities were brought in the designs of destroyers and submarines.

As a matter of fact, all these advances in design and construction are the fruit of the instructions and advices given by the technical men of the leading countries of Europe and America on various occasions and different opportunities, for which the author and his colleagues are always more than gratified, and which enabled them to undertake that great shipbuilding programme known as the Eight-Eight Programme through the knowledge and experiences thus gained.

Truly, it was this programme that influenced a great advance in design and construction, and both navy and private yards in co-operation exerted their best effort for carrying out this great programme.

Table 1 shows the condition of the navy of that time in a concrete form. It will be seen that in October, 1921, i.e., just before the Washington Conference, the number of the warships under construction was 53 and the aggregate displacement amounted to 338,187 tons including 5 capital ships of 196,000 tons, showing the great activity of warship building, Further, from the fact that the greater part of materials with the exception of a small percentage of steel materials, armour plates, corticene, etc. which was obtained from abroad, were of home productions, it will not be difficult to judge how greatly this shipbuilding programme has contributed towards the development of general engineering industry of this country.

Again, it should be particularly noted that all who were concerned in carrying out this programme whether of navy or private yards, bearing in mind the rising prices brought about by the Great-War and the increase of building cost due to the great dimensions of ships, exercised a possible economy in both design and construction, and the designs of the Yubari and Furutaka were, in a way, the results of these efforts, that is to say, the designs for a least possible building cost by adopting a smallest possible ship in place of a costly large ship, but with equally good tactical units.

The signing of the Washington Treaty (1922), however, suddenly cancelled the Eight-Eight Programme, and the building of capital ships was stopped. When an active step was taken for the proper allotment of aircraft carriers, none of which was completed at that time and auxiliary ships which became deficient due to a heavy depletion of capital ships, the effort already exercised in obtaining a powerful ship in smallest possible dimensions for the reduction of expenditure came nicely in for the tonnage limit of the Washington Treaty. The result was that the cruisers Yubari of 3,100 tons and Furutaka class of 7,100 tons for which the designs were completed before the Treaty, were the first cruisers built after the Treaty.

Following the Furutaka class, cruisers of 10,000-ton class, destroyers of 1,700-ton class, submarines of various classes, as well as aircraft carriers converted from discarded capital ships, etc. were completed, all these ships were designed with the same idea as the Furutaka class in economy and tactical quality, therefore, in general they may be rightly accepted as comparatively powerful ships for their dimensions.

Table 2 is the list of warships under construction at present (March, 1929). It will be seen that there are cruisers and other auxiliary ships, 30 in all, and of an aggregate displacement of 104,837 tons. In comparing these with 53 ships of 338,187-ton displacement in Table 1, it will be easy to see the large curtailment of the shipbuilding programme and the great reduction of the national expenditure.

So far the course of development of the ships of the Imperial Navy has been outlined, and it remains now to describe the history of the design and special features of ships of different classes under each heading.

I. Capital Ships

Before going into the description of the outline history of capital ship construction, an attention might be drawn to the fact that the Tsukuba which was designed during the 1904–1905 War, was the first battle-cruiser carrying 12-inch guns in the world. Also the Aki with four 12-inch guns and twelve 10-inch guns was one of the most powerful battleships of that time. After a few years, the Kongo which was built at the Vickers', England, carried 14-inch guns while 13½ inches were the greatest calibre then adopted in other Naval Powers. Further, the Fuso class which was designed in the navy after the Kawachi class carried, in addition to an adequate defensive power and a speed of 22½–23 knots, twelve 14-inch guns.

Thus it will be seen that even before the Great-War, our navy was always following a progressive course in the design of capital ships.

Nagato Class

(Plate I, Fig. 1, Table 4)

In 1915, the second year of the Great-War, the first ship of the four Fuso class having been completed, it was the high time to prepare for the design of next battleships and it was decided to adopt the 16-inch guns against the 15-inch guns of other Naval Powers. About that time, the report of the Jutland Engagement (July, 1916) was received and the result of a careful study of the lessons obtained in that great sea battle was to alter the design by increasing the displacement, speed and defensive power, and this new design was completed in September of the same year and the building was commenced in the next year (1917).

The principal features of the battleship Nagato are:—

(1) Displacement .... 33,800 tons
(2) Speed .... 23 knots
(3) Guns .... Eight 16 inches
(4) First capital ship of the Imperial Navy engined with all-geared turbines and with boilers for both oil burning and mixed burning.

Also improvements were made in fire-control, including the adoption of the tower mast in place of the usual tripod mast.

In these and many other respects, the ship may be said to have made an epoch in the annals of the capital ship building of the Imperial Navy.

The Nagato was completed at the Kure yard in November, 1920, and is the first capital ship carrying 16-inch guns in the world.

Her sister ship Mutsu was completed at the Yokosuka yard in November of the next year. The Washington Conference was then sitting and her destiny was an object of important debatement, however the fact that her retention was ultimately accepted on condition that United States was authorized to complete two ships of the Maryland class and Great Britain to build two new ships (the Nelson class), formed one of the notable records in the naval history of the world.

These sister ships were satisfactory in their power, speed, and gun trials, as well as in their seaworthiness, and may be taken as successful ships of this class.

Kaga Class

In 1918, that is eighteen months after the design for the Nagato class was made, the design for the battleships Kaga and Tosa was completed. These are the third and fourth ships of the Eight-Eight Programme, and are of 39,900 tons against 33,800 tons of the Nagato class.

The particulars of these ships are:—

Length p.p.		715′–0″
Max. Breadth (W.L.)		100′–0″
Draft		30′–9″
Displacement		39,900 tons
Speed		23 knots
Armament	Main guns	10 × 16″
	Secondary guns	20 × 5.5″ 4 × 3″ (A.A.G.)
	Torpedo tubes	8 × 21″

and they were provided with efficient side and deck armours, along with sufficient torpedo protections; therefore, although the speed is the same, these ships are more powerful than the Nagato class.

It was, while this design was going on or thereabout, that, judging from the designs of the recent capital ships, the necessity and advantage of increasing the ship's beam became a predominant question, and further, there was every possibility of this being realized much sooner than expected. On the other hand, while it is not a difficult matter to lengthen docks or building berths, the extension of breadth means a radical change or it may mean an entire reconstruction of docks and building berths. This is one of the important factors to be borne in mind to meet the increased dimensions, and sufficient and proper considerations were taken in the facilities of the naval yards at that time.

In 1920, the building of the Kaga was commenced at the Kawasaki yard and the Tosa at the Mitsubishi yard, Nagasaki.

The time when the preparations for building of these two ships were made was about the busiest time of merchant ship building as the effect of the Great-War, and great difficulty was experienced in obtaining the supply of both material and labour. In order to follow the pre-arranged course of work, both naval and private authorities grudged no pains and exerted their best efforts for the progress of the work.

Indeed, in looking back over that time, whosoever had anything to do with the work then would still recall a certain thrill on the striving and struggling conditions of the shipbuilding yards of those days.

In spite of these difficulties, these two ships were launched (November and December, 1921) when the Washington Conference was sitting, and as is well known, the decision of the Conference cancelled these two ships, so that directly after launching, they were placed on the surplus list, the same procedure as being taken in the battleship Washington of the U. S. Navy.

Amagi Class, etc.

The drafting of the design for the new battle-cruisers was commenced after that of the Kaga and the Tosa, and was completed in 1919, the year after the completion of the battleship design.

These new ships were named Amagi, Akagi, Takao, and Atago, and were the quartet battle-cruisers of the Eight-Eight Programme. The first two were laid down in 1920, but by the Washington Treaty they were rendered surplus before launching, and their existence as would-be battle-cruisers was thus abruptly terminated.

The design particulars of these ships were:—

Length p.p.		770′–0″
Max. Breadth (W.L.)		101′–0″
Draft		31′–0″
Displacement		41,200 tons
Speed		28.5 knots
Armament	Main guns	10 × 16″
	Secondary guns	16 × 5.5″ 4 × 3″ (A.A.G.)
	Torpedo tubes	8 × 21″

Besides those already mentioned, the design for the battleships Kii and Owari was completed prior to the Washington Conference, these ships were of 42,600-ton displacement and were the 9th and 10th ships of the Eight-Eight Programme.

There were still six more capital ships of the Eight-Eight Programme, but these were only under consideration then, and if the Washington Treaty were not signed and these six capital ships materialized, it is not difficult to surmise that the displacement would have been still further increased with corresponding increase in their building cost.

The Imperial Navy after the Washington Conference, so far as the capital ships are concerned, is practically in the state of complete intermission, only a preparatory investigation for the 35,000-ton ship which is to be laid down towards the end of 1931, being proceeding.

Reconstruction of Capital Ships

Directly after the Washington Conference, an investigation for the reconstruction of the capital ships then in commission was taken up within the restriction of 3,000-ton increase of displacement and other treaty-limits.

The battle-cruiser Haruna is the first ship reconstructed. In her case, the under-water protection was increased by adding blisters for the defence of submarine-attacks, and the above-water protection by adding suitable provisions to the decks for the defence of air-attacks.

The Kirishima is the second ship, now under reconstruction, and will be followed by other battle-cruisers and battleships one after another as the circumstances permit.

II. Aircraft Carriers

Hosho

(Plate II, Fig. 2, Table 4)

The Hosho is the first aircraft carrier in our navy. Her design was made in 1918 and she was completed in 1922.

Notwithstanding her displacement of only 9,500 tons, and though this was our first attempt at this class of ships, she might be regarded as a successful ship in design and construction. The special features are that the length and breadth of the landing deck are made as long and broad as possible, and in order to give a clear deck, the funnels are led up along the starboard side with a further provision for folding them down outboard when necessary, and also the Sperry-stabilizer is installed.

Regarding the Sperry-stabilizer fitted in this ship, there were minor troubles now and then for a few years after the installation, but this was wholly due to the unfamiliarity of men with this appliance. It is now in a good working order, answering well in using it in fairly rough seas, and from the experiences obtained in this ship, it is confirmed that the installation of the Sperry-stabilizer in the small type of aircraft carrier such as the Hosho would allow aviation without any difficulty under prevailing rough weather. The weight of the gyro for the Hosho is 136 tons all and the total weight of the installation is 190 tons, that is, about 2% of the displacement.

Akagi

(Plate II, Fig. 3, Table 4)

The battle-cruisers Amagi and Akagi of 41,200 tons rendered surplus by the Washington Treaty, were converted into aircraft carriers.

In both ships as a battle-cruiser, the greater part of the protective deck was finished. Leaving the lower portion untouched, the protective deck and its adjacent parts were reconstructed. The upper part was entirely altered in design for conversion into an aircraft carrier.

The principal features taken into consideration in this conversion were:—

(1) To ensure a suitable metacentric height due to alteration of the draft from 31 ft. to 22 ft., moreover, to reduce the weight in the reconstruction of the protective deck and its adjacent parts, the breadth was reduced from the original 101 ft. to about 92 ft.; further, not to impair the propeller efficiency, as much trim by the stern as practicable was given.

(2) The facility for the landing of aircraft was the primary aim in designing the upper arrangement, for which purpose, the funnels were carried outside on the starboard side, their height being kept below the level of the deck, so as to have the landing deck entirely free from any projections or obstacles.

(3) Space forward of the landing deck, and also space forward of the forecastle deck were arranged as flying-off decks.

(4) Ten 8-inch guns and twelve 4.7-inch A.A. guns were suitably disposed so as not to interfere with landing and flying off.

(5) Tendency for the greater height of the hull above the water line from the nature of the design was restrained as far as practicable.

The design was completed in the summer of 1923.

The Amagi which was on the building berth at the Yokosuka yard was practically destroyed by the great earthquake in the autumn of the same year, by the collapse of her building berth, and as it was hopeless to restore her, she was scrapped, and only the work of the Akagi then under reconstruction at the Kure yard was proceeded with. At the same time, it was decided to replace the Amagi by converting the Kaga on the scrap-list into an aircraft carrier. The Akagi was completed in 1927 and was directly put in commission, and the results have been found very satisfactory.

Kaga

(Table 4)

The Kaga was designed entirely on the same principle as the Akagi and there is no marked difference between them, except that only the Kaga is slower by a few knots and her landing deck is a little shorter. The lead of the funnel was afterward altered and led to go astern along the underside of the landing deck. Further consideration will, however, be necessary as to the best position and lead of the funnel in this class of ships.

The Kaga, in all her trials, was satisfactory, as was the case with the Akagi. In these two ships there is pratically nothing wanting as to their value in the qualities of the aircraft carrier; the size and arrangement of the future aircraft carrier will, nevertheless, require further considerations.

III. Cruisers

Tenryu Class

(Plate III, Fig. 4, Table 4)

The Tenryu and Tatsuta class which appeared after the cruiser Chikuma (4,950-ton displacement) designed in 1909, is, as already stated in the Introduction, the pioneer class of recent cruisers of the Imperial Navy.

In their outside appearance, they may come under the category of the protected flotilla leader. The design was made in 1915 and the Tenryu was completed in 1919. The structures of these ships are of lighter scantlings compared with the Chikuma.

They are light cruisers of 3,500-ton displacement and a high speed of 31 knots, with a suitable steaming radius and an adequate defensive power, carrying four 5.5-inch guns and six torpedo tubes.

Kuma Class

(Plate III, Fig. 5, Table 4)

After the Tenryu class, the Kuma class was designed in 1916. They are of 5,500-ton displacement and 33-knot speed, with an armament of seven 5.5-inch guns and eight torpedo tubes, and also with adequate radius of action and defensive power.

They are high speed cruisers of, so to speak, a magnified form of the Tenryu class.

Fourteen ships of this design were completed during 1920–1925, giving the same satisfactory results as in the Tenryu class.

After this class, the design of the light cruiser Yubari was completed in August, 1921, and further that of the Furutaka class was completed in October, 1921.

Yubari

(Plate III, Fig. 6, Table 4)

During the years 1920–1921, it was necessary to curtail the expenditure to an extreme limit to ensure the completion of the Eight-Eight Programme.

The Yubari is of a special type cruiser designed under this circumstance with a view to coping with the Kuma class in speed, offensive and defensive power, and radius of action, in spite of a smaller ship.

Notwithstanding her displacement of only 3,100 tons, that is, less than 60 per cent. of 5,500 tons of the Kuma class and correspondingly with the building cost of a little over 60 per cent. only of the latter ship, she carries six 5.5-inch guns and four torpedo tubes in the middle line making her offensive power equal to that of the Kuma class, and not in the least inferior in speed, radius of action, and defensive power.

At the beginning of this design there was much discussion as to the practical possibility of enabling the ship of this small dimension to have sufficient seaworthiness and habitability, and also to provide a sufficiently strong working platform for guns and torpedo tubes, though theoretically there might be no question as to the possibility of the design. However it was decided to have one ship built, more or less, by way of experiment.

With this object in view, the construction of the Yubari was commenced in 1922, directly after the Washington Conference. The work was urged on and she was completed in only one year. Judging from the results of severe official trials and of actual service in commission, there is no denial that she has satisfactorily fulfilled the object of the design in her speed, as well as her offensive and defensive power, and also in seaworthiness and habitability.

Furutaka Class

(Plate IV, Figs. 7–8, Table 4)

The design subsequent to that of the Yubari is the cruiser Furutaka class which was commenced in 1921, before the Washington Conference.

Although the tactical requirement then was to have cruisers of about 10,000-ton displacement with 8-inch guns, such ships were not embodied in the Eight-Eight Programme; besides, on account of economy their realization was regarded as nearly impossible. But from a technical point of view, it was considered that there is no reason whatever to take 10,000-ton displacement as a necessary minimum limit for carrying 8-inch guns and a study was taken up for a medium-sized cruiser having displacement as small as practicable and carrying a suitable number of 8-inch guns, at the same time possessing all the necessary qualities for a high speed cruiser. In the autumn of the same year, a definite plan was decided upon and a ship with 7,500-ton displacement (corresponding to the standard displacement of 7,100 tons of the Washington Treaty) was found to fulfil all the requirements in the form of the Furutaka class.

In the next year, directly after the Washington Conference, on the formation of the Cruiser Repletion Programme, it was decided to build four ships of this class, and their construction were directly commenced. The Furutaka, the first ship of these quartet cruisers, was completed in 1926 and was followed by the Kako, Aoba, and Kinukasa.

An armament of six 8-inch guns and twelve torpedo tubes, a speed of 33 knots, efficient side and deck protections, a watertight compartment for each set of engines, and a large radius of action are the noteworthy features of the class.

Bearing in mind that the displacement is only 7,500 tons, the design was made before the Washington Conference, and both trial and service results have been satisfactory, perhaps this class may be regarded without prejudice as the pioneer of the modern light cruisers carrying 8-inch guns.

Myoko Class

(Plate IV, Fig. 9, Table 4)

The Myoko class is the first cruiser designed to the "standard" of 10,000 tons directly after the Washington Treaty.

The Myoko was designed in 1923, and her construction was commenced at the Yokosuka yard in 1924; since then, closely one after another, 8 ships of this class were laid down at the Yokosuka, Kure, Kobe, and Nagasaki yards. The first ship completed was the Nachi and was commissioned in November, 1928, and three of her sister ships will be completed very soon.

These ships have an armament of ten 8-inch guns and twelve torpedo tubes (eight tubes for later four ships), a speed of 33 knots, and a radius of action almost as extensive as a large cruiser. They have efficient side and deck protections. The watertight subdivisions are particularly reliable by the partition of a separate compartment for each set of engines and practically one for each boiler. On the whole, judging from the details of the design, it will be quite proper to look upon this class as the magnified Furutaka.

The first ship Nachi has already completed a series of strict trials, and, from the results obtained up to the date, she has proved satisfactorily to have all the requisite qualities of a large sized cruiser.

IV. Summary of the Present-Day Cruiser Design

(Table 3)

The designs of the cruisers Yubari, Furutaka, Myoko, etc. are almost on the same basic line, and, as the description of principal features of these designs will serve to show the general trend of warship construction of the Imperial Navy, that of the Furutaka will be taken as an example.

Length.—Needless to say, of the principal dimensions and form of a ship, the length is the most important factor in the design of a ship. The extension of length means the reduction in the power and weight of machinery for the same displacement and the same speed, but it will increase the weight of hull and fittings, and further, the corresponding decrease of breadth and draft will obviously impair the seagoing quality. Also in all other respects, except the advantage in propulsion there will be nothing gained by increasing the length; therefore the selection of a suitable length plays the most important part of the design. In designing the Furutaka as a high speed cruiser of the latest type, much greater length than the general practice was selected, that is, for the displacement of 7,500 tons in the normal condition, the length p.p. is 580′–0″, breadth in W.L. 50′–9.5″, draft 15′–3.7″ with the ratio of length to breadth 11.4, and also the ratio of length to a cube root of volume of displacement 9 against about 10 and 8.5 respectively in the cruisers of other Naval Powers (Furutaka–the depth at side being 33′–0½″, the length 17.5 times the depth).

Therefore, leaving out the questions of advantage and disadvantage of long length to other qualities, but speaking only on the propulsion, the fact that the required speed was attained with a comparatively low power well justifies us in the selection of these proportions in the Furutaka. Moreover, by the effort of engineer-designers, the weight of machinery was reduced to about 53 S.H.P. per ton of the machinery which made the design more favourable to high speed.

Although it is a matter of minor importance, it should be stated here that the mixed burning system was adopted in some of the boilers in this case owing to the peculiar situation of this country.

Weight of Hull and Fittings.—The difference in the weight of hull and fittings between the ships of the same type and the same displacement is quite unavoidable according to the principal dimensions, particularly to the length and the details of the internal arrangement, but practically there are general standards in respect of the displacement according to the class of ships, and 48–50% for present-day light cruisers, 38–40% for destroyers and 33–35% for capital ships may be taken as general averages.

The causes of these differences in percentage according to the class of warships are of exceedingly complicated nature and cannot be defined in a simple form; however they may be principally due to:—

(a) Great variation in speed according to the class of warships.
(b) Relation between the speed and displacement, principal dimensions, particularly the length.
(c) Mutual relation between the displacement and principal dimensions, also freeboard.
(d) Existence or non-existence of protective plates, their thickness, their arrangement, and their utilization to the structures.
(e) Quality of material used in construction.
(f) Power and size of armament and machinery, and amount of fuels.
(g) Area of habitable division.
(h) Factors of safety in the design of structure and fittings, the time-limit of utility, care and attention of men on board, etc.

But the most important question is that, while the displacement of a light cruiser comes about between those of a destroyer and a capital ship, yet the ratio of hull and fittings to displacement assumes the abnormal figure of 48–50%, quite outside the range of 40% of destroyers and 35% of capital ships. This was quite sufficient to form a strong belief that there must still be an ample room left for an improvement in this direction. As the result of thorough investigations to verify this belief, a great reduction in weight was effectively carried out in the design of the Furutaka, without imparing the integrity of the structural strength, for in spite of her great length and heavy armament of six 8-inch guns, etc. which are all against the general principle of the ship design, the percentage was brought down nearer to that of destroyers, and enabled the ship to have high speed, efficient armament and protection at a moderate displacement.

Although it is not possible here to go into all the details of reduction in the weight of hull and fittings, yet one or two points which could be judged from her outside appearance will be explained; on looking at the broadside of this ship, it will be noticed that there is much freedom taken in dealing with the shape of the upper deck. This was done to avoid the unnecessary height for main guns, as well as to reduce the hull weight without impairing the requisite height of the upper deck in relation to the equipment of the ship. The greater percentage of the material used for the side and deck protections is non-cemented armour plates, which were, in addition to their main function of protection, freely and efficiently utilized in the structural strength of the hull.

The materials for the hull construction are as usual of high tensile and mild steels, and no special material was used, the average cost per ton of material being thus of usual average, but the labour per ton of material was slightly higher on account of the careful workmanship and complicated nature of the work. On the other hand, if the percentage of weight to displacement had not been reduced by the reduction of the hull weight, the displacement of this ship with her speed, armament, and protection would have been increased by quite 1,000 tons over above her displacement of 7,500 tons, and a slight increase in the labour would be only a fraction in comparison with the increase in cost due to a greater displacement.

Corrosion.—A considerable attention was also given to the corrosion in this ship; for instance, a certain percentage of the structural materials was galvanized.

Vibration.—It is well known that, of recent years, the problem to prevent the vibration in a high speed ship forms one of the important studies in the ship design.

In one of the former cruisers, the vibration of the lower deck directly above the propellers was in such an extent as to cause much discomfort to the men in that vicinity. It was impossible to use a vibrograph on that deck.

In the Furutaka, the vibration is practically nil, so that men off duty on the lower deck directly above the propellers were observed reading newspapers with ease. There is still less vibration on the structure of hull as a whole.

Generally speaking, the maximum amplitude is not more than about 0.5 mm., as is shown in Table 3(c), that is, about ⅓–¼ of those of the former cruisers.

Oscillation.—There is no doubt that a superiority of seaworthiness is one of the essential qualities in cruisers. It is also well known that the standard to judge the seaworthiness is the periodic time of rolling.

Table 3 (A) shows the data relating to the rolling of the present-day principal warships of the Imperial Navy.

Again, from this table and other facts, the period of oscillation on the basis of the displacement is shown by a curve in the Table 3 (B). On account of the variation in classes from capital ships to destroyers, there is no reason to hold that the curve should be a fair one, however it will serve to show the general trend of the change of period with the change of displacement in the present-day principal warships.

Further, according to this curve, the period for a present-day cruiser having the displacement of the Furutaka may be taken as somewhere about 12.5 seconds, but the actual period obtained from the rolling experiment carried out with the Furutaka with 3.25 ft. metacentric height is as much as 13.7 seconds. Again, the Yubari, according to this curve, should have about 11 seconds for her displacement, but actually her period is as good as 12.7 seconds.

For these ships, Table 3 (B) is prepared and in comparing it with the Table 3 (A), it will easily be seen that the periods of the ships of the later design are all longer, the reason for attaining these results, despite their large metacentric height, is due to large radius of gyration, that is to say, satisfactory effects of bilge keels and ship's form, proper distributions of protection and armament and several other matters in the design.

In fact, the ships of the Furutaka class have an exceptionally long period of oscillation on the sea, so that they are unanimously pronounced as very easy ships and as good as large warships in a seaway, and moreover their upper decks are always kept dry. They have also given satisfaction in easiness of steering and turning qualities.

Before the Furutaka class, the Yubari of 3,100 tons has given already every satisfaction in her hull and equipment weight, vibration, oscillation, seaworthiness, etc. After the Furutaka class, the Nachi of 10,000 tons was already completed, and the actual records of power, speed, gun trials, etc. show every satisfaction in all these qualities.

V. Destroyers

Second Class Destroyers

(Plate V, Fig. 10, Table 4)

The outbreak of the Great-War and our participation in the War as one of the allied Powers necessitated the hurried construction of the second class destroyers (1914). Ten of the Sakura class built in 1911 were then constructed (these are the Kaba class, of 665-ton displacement and 31-knot speed) and followed by four of the Momo class of 835-ton displacement and 31.5-knot speed, and six of the Nara class of 850-ton displacement and 31.5-knot speed. These were constructed closely one after another and in 1918 the construction of the Momi class was commenced. It may be mentioned that, in passing, the twelve destroyers of the Kaba class above referred to were built, during the War, in this country for an allied Power.

The Momi class of 850$\sim$900-ton displacement are the first medium-sized destroyers of our navy fitted with oil burning boilers and all-geared turbines, and twenty-one of them were constructed one after another. Further, eight of the improved type the Wakatake class were constructed. Of these, the Momi class was taken up as the standard of our second class destroyers and was acknowledged to have all the qualities appropriate to this class.

After 1924, the construction of the second class destroyers was brought to an end, and only the first class destroyers were thereafter constructed.

First Class Destroyers

(Plate V, Fig. 11, Table 4)

Following the Umikaze and Yamakaze which were constructed in 1911, the Isokaze class (displacement 1,227 tons and speed 34 knots) and the Kawakaze class (displecement 1,300 tons and speed 34 knots) were designed during 1915–1916, and four of the former class and two of the latter were constructed, these being followed by the design of the Minekaze class.

The ships of the Minekaze class were the pioneer first class destroyers of our navy fitted with oil burning boilers and all-geared turbines, as in the case of the Momi class of the second class, and this class was taken up as the standard type of the first class destroyers and fifteen of them were constructed.

By 1927, with a little improvement, nine of the Kamikaze class and twelve of the Mutsuki class were constructed.

Large Destroyers, Fubuki Class

(Plate V, Fig. 12, Table 4)

The latest design since that of the 1,445-ton first class destroyer is the Fubuki class. Already nine of this class were completed and fifteen are about to be completed. The displacement is 1,850 tons, the armament consists of six 4.7-inch guns and nine torpedo tubes, and the speed is 34 knots.

The first ship of this class was completed in October, 1928, and her trial results and actual behaviour up to the present have fully justified the design.

VI. Submarines

Second Class Submarines

(Plate VI, Fig. 13, Table 4)

The submarines are at present classified as the "Ha" class (old type), "Ro" class (medium type), and "I" class (larger type) ; those of the "Ro" class are the second class submarines.

In 1915, the first submarine boat of medium type was designed in the navy, and the "Ro" 11 and "Ro" 12 (old 19 and 20, surface displacement 720 tons, surface speed 18 knots) were constructed after this design in 1919.

Since, by 1927, as many as twenty-two boats of this type and improved type were completed. Of these, eighteen were built at the naval yards and four at the Kawasaki yard.

It has already been mentioned here and there in this paper regarding the valuable services rendered by several private yards in the construction of warships, but their services are particularly prominent in the submarine constructions for working in parallel with those of the medium type of the navy; during 1920–1922 the Kawasaki yard completed five of their so-called "F"-type modelled after the Italian Fiat type, and during 1920–1927 the Mitsubishi yard, Kobe, completed eighteen of their so-called "L"-type modelled after the English Vickers' type, and all of them are in service now as the second class submarines.

First Class Submarines

(Plate VI, Figs. 14–19, Table 4)

All of "I" Submarines are the first class submarines, they are all large ones of more than 1,000 tons, including cruiser-submarines, fleet-submarines, minelayer submarines, etc. and truly they form a group of the best submarines of the Imperial Navy.

The technical lessons of the Great-War are apparent in this line too, particularly in the design "I" 1–4 (surface displacement 1,970 tons) and "I" 21–24 (surface displacement 1,150 tons). All of these were constructed at the Kawasaki yard, and they are considered as the most practical types of this class. The "I" 51 (one), "I" 52 (one), "I" 53 (four completed and five under construction), "I" 61 (three completed and two under construction) are the high speed submarines of surface displacement 1,400–1,650 tons. These may be regarded as of the direct lineage of submarines after the designs of our navy. All of these, except two which were built at the Mitsubishi yard, Kobe, were completed at the naval yards, and in respect of their structures, armament, qualities above and below water, they are considered as successful submarines of the day.

VII. Special Service Ships

(Plate VII, Figs. 20–22, Table 4)

From the experiences obtained by the Gerat-War, a proper allotment of so-called naval auxiliary vessels became an urgent necessity, and as a temporary means, a considerable number of old warships were converted into minelayers or minesweepers or submarine depôt ships.

Recently, the following ships were specially designed for the purpose, and some were already completed and some are under construction.

Class	*Completed*	*Under Construction*
Gun boat	Ataka	
Gun boat (shallow draft)	Four of Seta class	Atami & Futami
Submarine depôt ship	Jingei & Chogei	
Minelayer	Katsuriki	Itsukushima, Shirataka, Tsubame, & Kamome
Minesweeper	Nos. 1–6	

Minesweepers are generally similar in their structures to destroyers, armed suitably and each fitted with a pair of paravanes.

Conclusion

In the foregoings, the author has attempted to describe the general course of the After-War Development of the Ships of the Imperial Navy, as well as to explain the characteristic features of various classes of warships in detail.

In short, the warship technic, shipbuilding and kindered industries of this country were progressing with a great stride on account of the great Eight-Eight Shipbuilding Programme, but as the direct consequence of the Washington Treaty, a sudden intermission of capital ship building appeared, for a time, to impede their progressive course. The necessity of proper allotment of auxiliary warships has, however, allowed them to continue their original course of development.

Lastly, the author desires to acknowledge his great indebtedness to Constructor Commander, Viscount T. Tokugawa, I.J.N. for his painstaking assistance in preparing this paper.

TABLE 1. WARSHIPS UNDER CONSTRUCTION, AS REPORTED OCT. 1921.

TYPE OF SHIPS. / BUILDER	BATTLE-SHIPS.	BATTLE CRUISERS.	LIGHT CRUISERS	DESTROYERS.	AEROPLANE CRUISERS.	SUBMARINES	GUN BOATS & SPECIAL SERVICE VESSELS.	TOTAL	
								NUMBER.	DISPLACEMENT. (NORMAL)
YOKOSUKA D.Y.	1	1				5		7	78,710
KOBE D.Y.		1				5		6	45,560
SASEHO D.Y.			2			2		4	12,630
MAIZURU D.Y				4				4	5,380
							IN DOCKYARDS.	21	142,280
MITSUBISHI S.Y. (NAGASAKI)	1		1					2	45,470
MITSUBISHI S.Y. (KOBE)						5		5	4.500
KAWASAKI S.Y.	1		2	1		7	1	12	59,247
YOKOHAMA S Y.							2	2	16,220
ASANO S.Y.					1			1	9,500
URAGA S.Y.			1	2				3	7,270
ISHIKAWAJIMA S.Y.				2				2	1,700
HARIMA S.Y.									
TAMA S.Y.									
OSAKA S.Y.							2 ✓	2 ✓	30,800 ✓
FUJINAGATA S.Y.				2				2	1,700
NEW YORK S.C. U.S.A.							1 ✓	1 ✓	19,500 ✓
							IN PRIVATE SHIPYARDS	32	195,907
TOTAL NUMBERS.	3	2	6	11	1	24	6	53	
TOTAL DISPLACEMENT.	113,600	82,400	33,350	11,330	9,500	18,787	69,220		338,187

✓ ..... OILERS

(Paper No. 652)

(Paper No. 652)

TABLE 2. WARSHIPS UNDER CONSTRUCTION, AS REPORTED MAR. 1929.

TYPE OF SHIPS / BUILDER	LIGHT CRUISERS.	DESTROYERS.	SUBMARINES.	GUN BOATS & SPECIAL SERVICE VESSELS.	TOTAL	
					NUMBER.	STANDARD DISPLACEMENT.
YOKOSUKA D.Y.	2		1		3	21,650
KURE D.Y.	1		3		4	14,950
SASEHO D.Y.		2	1		3	5,050
MAIZURU D.Y.		2			2	3,400
				IN DOCKYARDS.	12	45,050
MITSUBISHI S.Y. (NAGASAKI)	2				2	20,000
MITSUBISHI S.Y. (KOBE)			2		2	3,300
KAWASAKI S.Y.	2		1		3	21,970
YOKOHAMA S.Y.		1		1	2	2,143
ASANO S.Y.						
URAGA S.Y.		2		1	3	5,340
ISHIKAWAJIMA S.Y.		1		1	2	3,024
HARIMA S.Y.						
TAMA S.Y.				1	1	167
OSAKA S.Y.				1	1	443
FUJINAGATA S.Y.		2			2	3,400
				IN PRIVATE SHIPYARDS	18	59,787
TOTAL NUMBER	7	10	8	5	30	
DISPLACEMENT	70,000	17,000	13,520	4,317		104,837

TABLE 3.

(A) ROLLING PERIODS, ETC. OF RECENT WARSHIPS

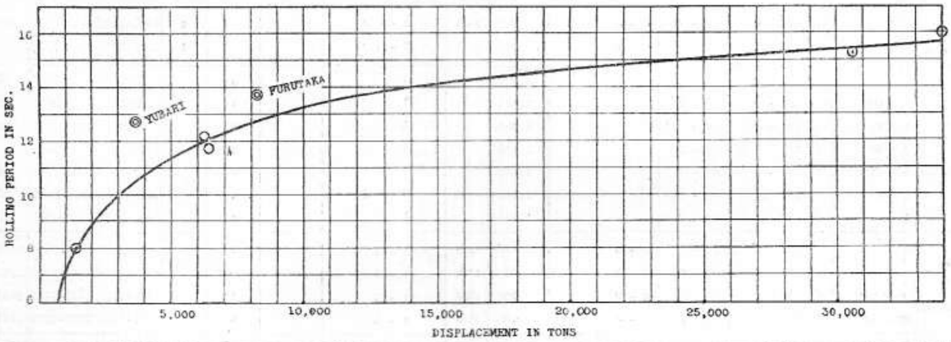
TYPE OF SHIPS.		DISPLACEMENT NORMAL CONDITION (... .) AS EXPERIMENTED.	NATURAL PERIOD. (COMPLETE) 2 T	METACENTRIC HEIGHT. GM	RADIUS OF GYRATION "K" OBTAINED FROM $2T = 2\pi\sqrt{\frac{K^2}{gGM}}$	SHIP'S HALF BREADTH.	"C" IN K = C x (HALF BREADTH)
		Tons	Seconds	Ft.	Ft.	Ft.	
CAPITAL SHIP	✝	30,600 - 33,800	15 - 16	5.0 about	30.3 - 32.3	47 - 47.5	0.65 0.68
LIGHT CRUISER.	×	5,570 (6,475)	11.7	2.9	18.0	23.4	0.77
BILGE KEEL ENLARGED.	×	" (6,256)	12.1	3.1	19.2	23.4	0.82
DESTROYER.	×	1,345 (1,328)	8	2.7	11.9	14.65	0.81

(B) ROLLING PERIODS, ETC. OF THE LATEST CRUISERS

TYPE OF SHIPS.		DISPLACEMENT NORMAL CONDITION (... .) AS EXPERIMENTED.	NATURAL PERIOD. (COMPLETE) 2 T	METACENTRIC HEIGHT. GM	RADIUS OF GYRATION "K"	SHIP'S HALF BREADTH.	"C"
YUBARI.	×	3,100 (3,564)	12.7	2.4	17.8	19.75	0.90
FURUTAKA.	×	7,500 (8,201)	13,7	3.25	22.3	25.39	0.88

✝ OBSERVED, APPROXIMATE.
× BY ROLLING EXPERIMENT IN STILL WATER

CURVE SHOWING THE PERIODS OF ROLLING FOR VARIOUS WARSHIPS.



(C) AMPLITUDE AND FREQUENCY OF VIBRATION IN THE STERN-PART OF "FURUTAKA."

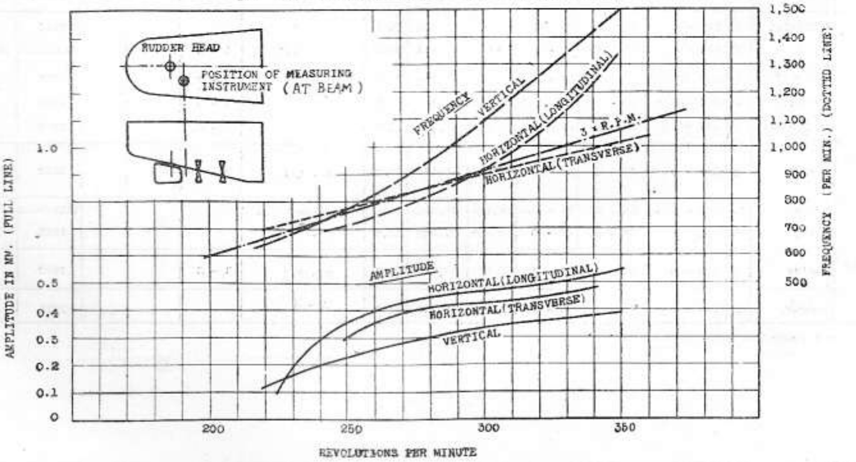


TABLE 4. PARTICULARS OF RECENT WARSHIPS. MAR.1929.

TYPE	CLASS	NO.	PRINCIPAL DIMENSIONS (NORMAL CONDITION)			DISPLACEMENT		SPEED	GUNS			TORPEDO TUBES		DATE OF LAYING DOWN	DATE OF LAUNCHING	DATE OF COMPLETION
			LENGTH P.P.	BREADTH (W.L.)	DRAUGHT MEAN	NORMAL	STANDARD		MAIN	SECONDARY	ANTIAIR-CRAFT	SUBMER-GED	ABOVE WATER			
			FT. IN.	FT. IN.	FT. IN.	TONS	TONS	K.	NO. IN.	NO. IN.	NO. IN.					
BATTLE-SHIPS	FUSO	2	630 - 0	94 - 0	28 - 6	30.600	29.330	22.5	12 14	16 - 6	4 - 3	6		1912-1913	1914-1915	1915-1917
	ISE	2	640 - 0	94 - 0	28 - 8	31,260	29.990	23.0	12 - 14	20 - 5.5	4 - 3	6		1915	1916-1917	1917-1918
	NAGATO	2	660 - 7	95 - 0	30 - 0	33.800	32,720	23.0	8 - 16	20 - 5.5	4 - 3	4	4	1917-1918	1919-1920	1920-1921
BATTLE-CRUISER	KONGO	4	653 - 6	92 - 0	27 - 6	27,500	26,330	27.5	8 - 14	16 - 6	4 - 3	8		1911-1912	1912-1913	1913-1915
AIRCRAFT CARRIERS	HOSHO	1	510 - 0	62 - 0	20 - 3	9,500	7,470	25.0	4 - 5.5		2 - 3			1919	1921	1922
	AKAGI	1	770 - 0	92 - 3	22 - 2.4	28,100	26.900	28.5	10 - 8		12 - 4.7			1920	1925	1927
	KAGA	1	715 - 0	102 - 0	22 - 1	28.100	26,900	23.0	10 - 8		12 - 4.7			1920	1921	1928
LIGHT CRUISERS	TENRYU	2	440 - 0	40 - 9	13 - 0	3,500	3,230	31.0	4 - 5.5		1 - 3		6	1917	1918	1919
	KUMA	5	500 - 0	46 - 9	15 - 9	5.500	5.100	33.0	7 - 5.5		2 - 3		8	1918-1919	1919-1920	1920-1921
	NAGARA	6	500 - 0	46 - 9	15 - 10.5	5.570	5.170	33.0	7 - 5.5		2 - 3		8	1920-1921	1921-1923	1922-1924
	NAKA	3	500 - 0	46 - 9	15 - 10.5	5.595	5.195	33.0	7 - 5.5		2 - 3		8	1922	1923-1925	1924-1925
	YUBARI	1	435 - 0	39 - 6	11 - 9	3,100	2,890	33.0	6 - 5.5		1 - 3		4	1922	1923	1923
	FURUTAKA	2	580 - 0	50 - 9.5	15 - 3.7	7,500	7,100	33.0	6 - 8		4 - 3		12	1922	1925	1926
	AOBA	2	580 - 0	50 - 9.5	15 - 3.7	7,500	7.100	33.0	6 - 8		4 - 4.7		12	1924	1926	1927
	MYOKO	8	630 - 0	57 - 5	17 - 3.6	10,600	10.000	33.0	10 - 8		*6 - 4.7		*12	1924-1929	1927-	1928-
FIRST CLASS DESTROYERS	MINEKAZE	15	320 - 0	29 - 3	9 - 6	1,345	1,215	34.0	4 - 4.7				6	1918-1921	1919-1922	1920-1922
	KAMIKAZE	9	320 - 0	30 - 0	9 - 7	1.400	1,270	34.0	4 - 4.7				6	1921-1923	1922-1924	1922-1925
	MUTSUKI	12	320 - 0	30 - 0	9 - 8.5	1,445	1,315	34.0	4 - 4.7				6	1924-1926	1925-1927	1925-1927
	FUBUKI	24	367 - 5.5	34 - 0	10 - 6	1,850	1,700	34.0	6 - 4.7				9	1926-	1927-	1928-
SECOND CLASS DESTROYERS	MOMI	21	275 - 0	26 - 0.8	8 - 0	850	770	31.5	3 - 4.7				4	1918-1921	1919-1922	1919-1923
	WAKATAKE	8	275 - 0	26 6	8 3	900	820	31.5	3 - 4.7				4	1921-1922	1922-1923	1922-1924
FIRST CLASS SUBMARINES	I - 51	1	300 - 0	28 - 11			SURFACE 1.400		1			8		1921	1921	1924
	I - 52	1	330 - 0	25 - 1			1,400		1			8		1922	1922	1925
	I - 53	3	330 - 0	26 - 1.5			1,650		1			8		1924	1925-1926	1927
	I - 56	6	331 - 4.4	25 - 11			1,650		1			8		1924-1927	1925-	
	I - 61	2	320 - 6	25 - 7			1,650		1			6		1926-1927	1927-1928	1928-
	I - 1	4	319 - 10.6	30 - 3.3			1,970		2			6		1923-1926	1924-1928	1926-
	I - 21	4	279 - 6.3	24 - 7.2			1,150		1			4		1924-1926	1926-1927	1927-
SECOND CLASS SUBMARINES	RO - 26	3	230 - 0	20 - 1	12 - 3		750	16.0	1 - 3			4		1921	1921-1922	1923-1924
	RO - 51	6	231 - 7	23 - 6	12 - 9		900	17.0	1 - 3			4 or 6		1918-1920	1919-1921	1920-1922
	RO - 57	3	250 - 0	23 - 6	12 - 9		900	17.0	1 - 3			4		1920-1921	1921-1922	1922-1923
	RO - 60	9	250 - 0	24 - 2.6	12 - 4.4		998	16.0	1 - 3			6		1921-1925	1922-1926	1923-1927
GUN-BOATS	ATAKA	1	222 - 0	32 - 0	7 - 6	820	725	16.0	2 - 4.7		2 - 3			1921	1922	1922
	SETA	4	180 - 0	27 - 0	3 - 4	338	305	16.0	2 - 3					1922	1922-1923	1923
	ATAMI	2	148 - 7.5	22 - 3.6	3 - 3.4	188	167	16.0	1 - 3					1928-	1929-	
MINE SWEEPERS & MINE LAYERS	KATSURIKI	1	240 - 0	39 - 1	13 - 6	2,000	1,540	13.0	3 - 3					1916	1916	1917
	NO. 1	6	235 - 0	26 - 4	7 - 6	700	615	20.0	2 - 4.7		1 - 3			1922-1928	1923-1928	1923-1929
	SHIRATAKA	1	259 - 10.1	37 - 8.8	10 - 2	1,516	1,324	16.0			3 - 4.7			1927	1929	1929
	TSUBAME	2	206 - 8.3	23 - 7.5	6 - 10.8	503	443	19.0			1 - 3			1928	1929	
	ITSUKUSHIMA	1	328 - 1	41 - 10	10 - 6.8	2,047	1,940	17.0			3 - 4.7			1928	1929	
SUBMARINE DEPOT SHIP	JINGEI	2	380 - 0	53 - 0	22 - 8	8,500	5,160	16.0	4 - 5.5		2 - 3			1922	1923-1924	1923-1924
OILER	NOTORO	10	455 - 0	58 - 0	26 - 6.1	15,400	14,050	12.0	2-4.7 or 2-5.5					1919-1922	1920-1923	1920-1924
	KAMOI	1	496 - 0	67 - 0	28 - 0	19,500	17,000	15.0	2 - 5.5					1921	1922	1922
PROVISION SHIP	MAMIYA	1	475 - 0	61 - 0	27 - 8	17,500	15,820	14.0	2 - 5.5		1 - 3			1922	1923	1924
ICE BREAKER	ODOMARI	1	200 - 0	50 - 0	21 - 0	2,830	2,330	13.0	1 - 3					1921	1921	1921

*4 - 4.7 & 8 TUBES FOR LATER FOUR SHIPS

PLATE I.

FIG. 1. BATTLESHIP "MUTSU."

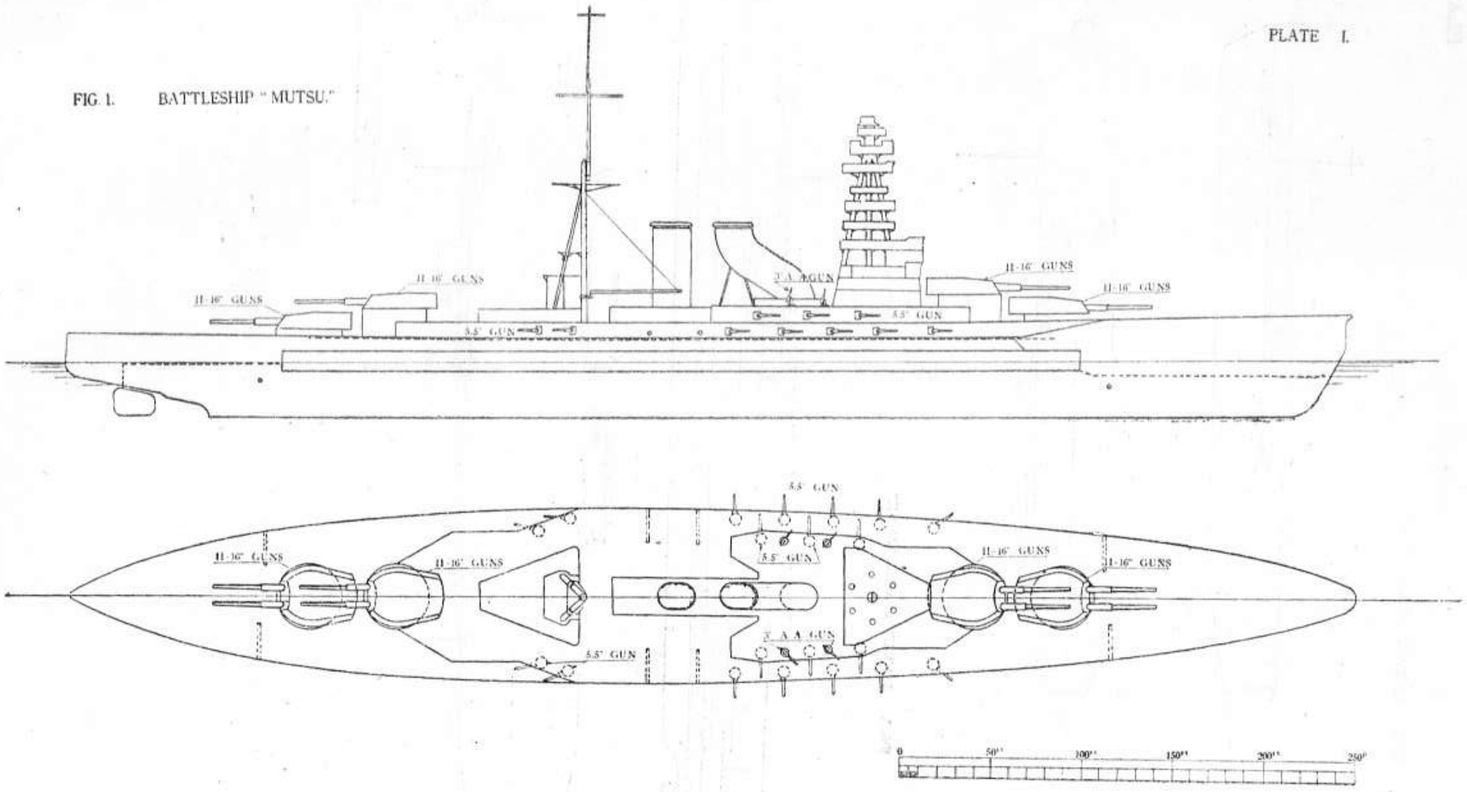


PLATE II.

FIG. 2. AIRCRAFT CARRIER "HOSHO."

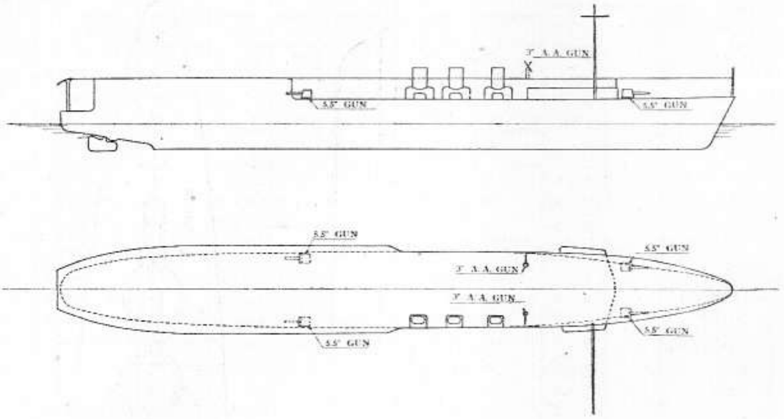


FIG. 3. AIRCRAFT CARRIER "AKAGI."

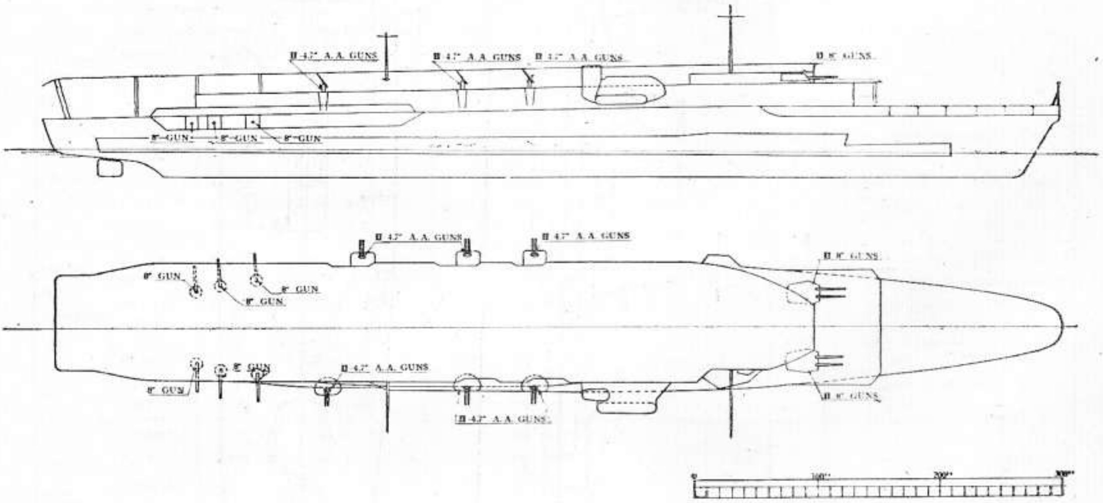


PLATE III.

FIG. 4. LIGHT CRUISER "TENRYU."

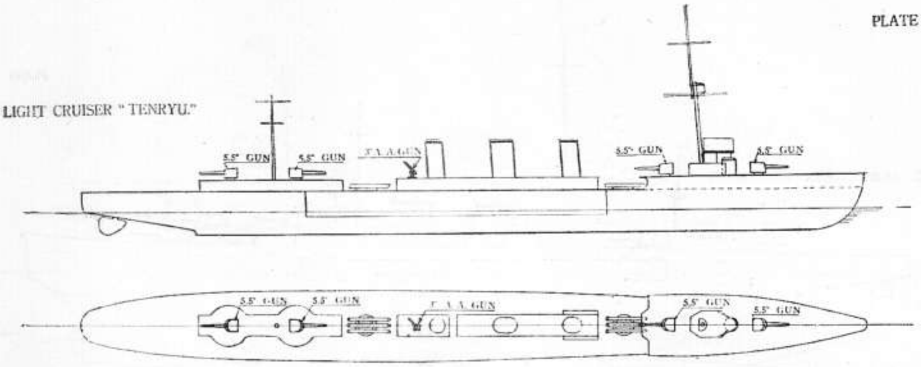


FIG. 5. LIGHT CRUISER "KUMA."

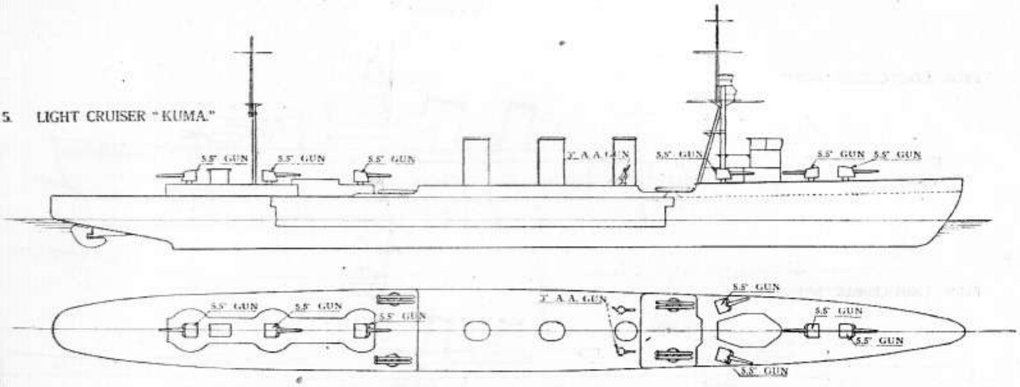


FIG. 6. LIGHT CRUISER "YUBARI."

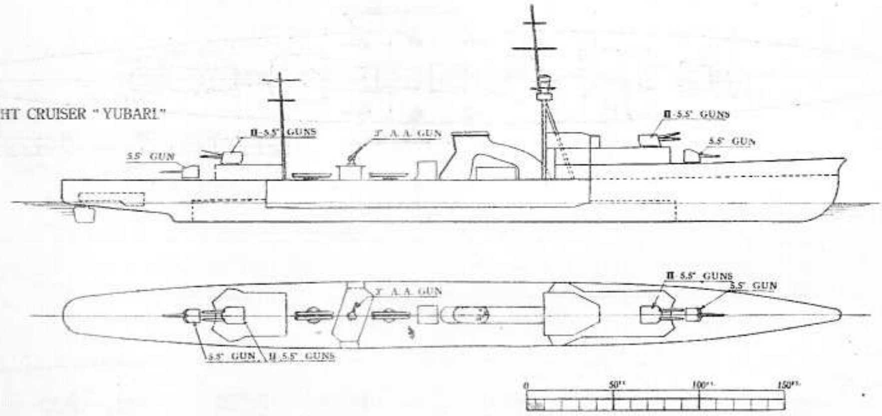


PLATE IV.

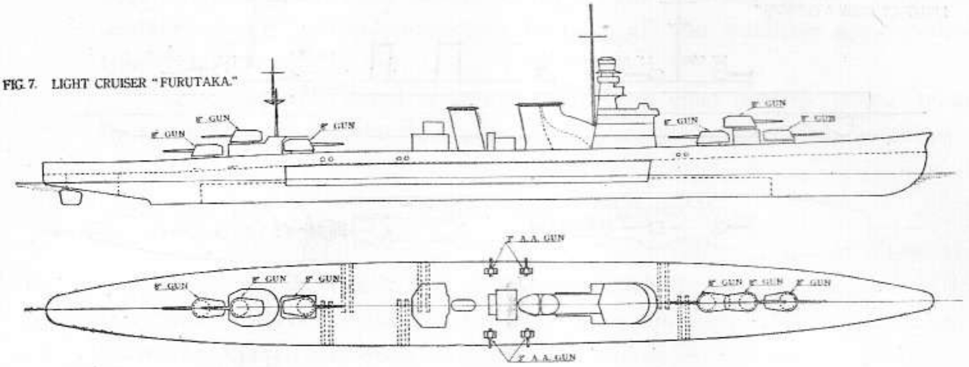


FIG. 7. LIGHT CRUISER "FURUTAKA."

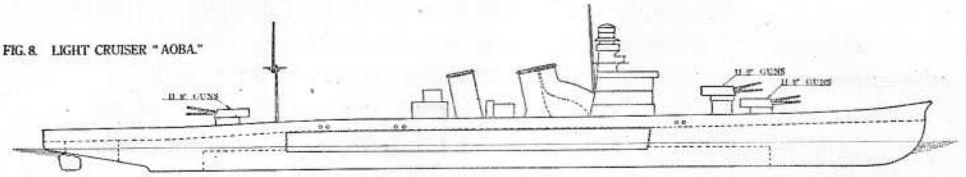


FIG. 8. LIGHT CRUISER "AOBA."

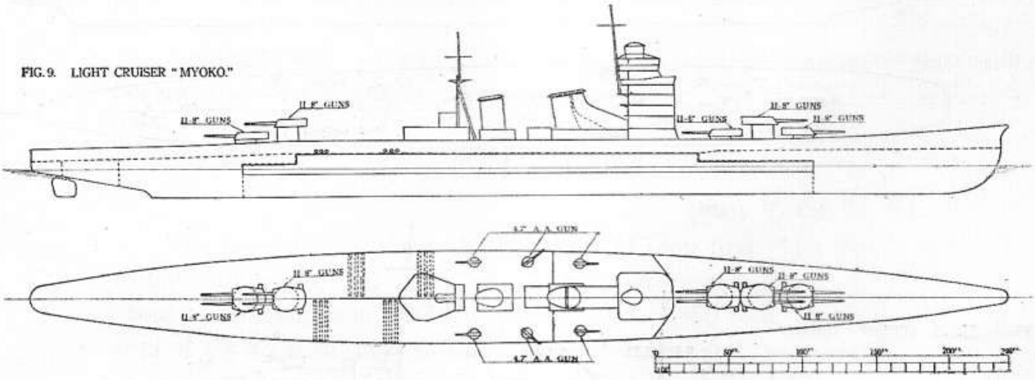


FIG. 9. LIGHT CRUISER "MYOKO."

PLATE V.

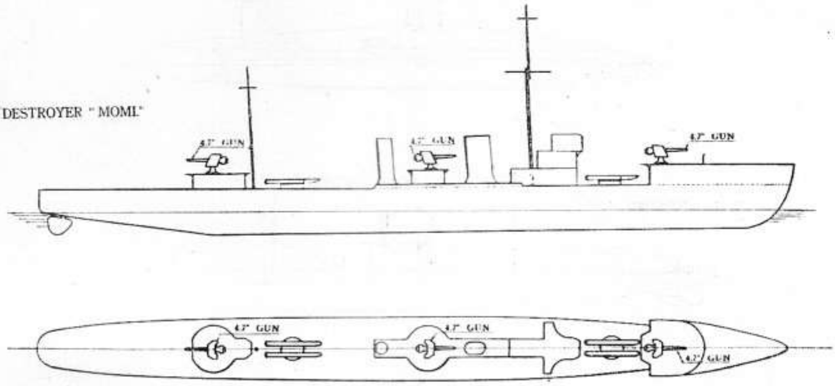


FIG 10 DESTROYER " MOMI."

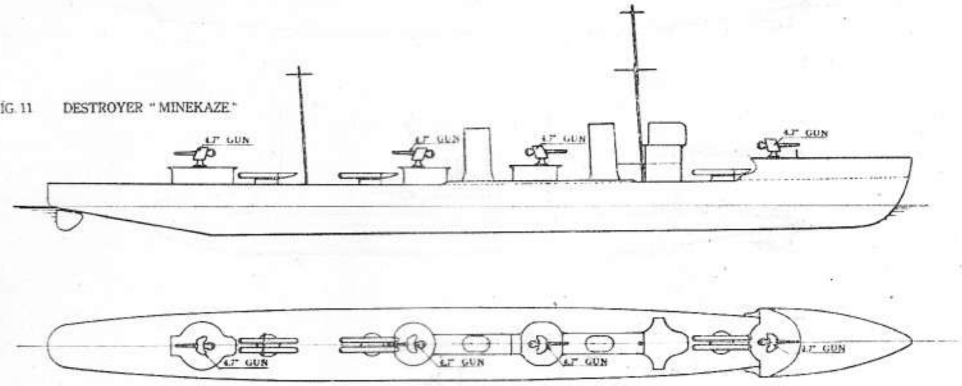


FIG. 11 DESTROYER " MINEKAZE."

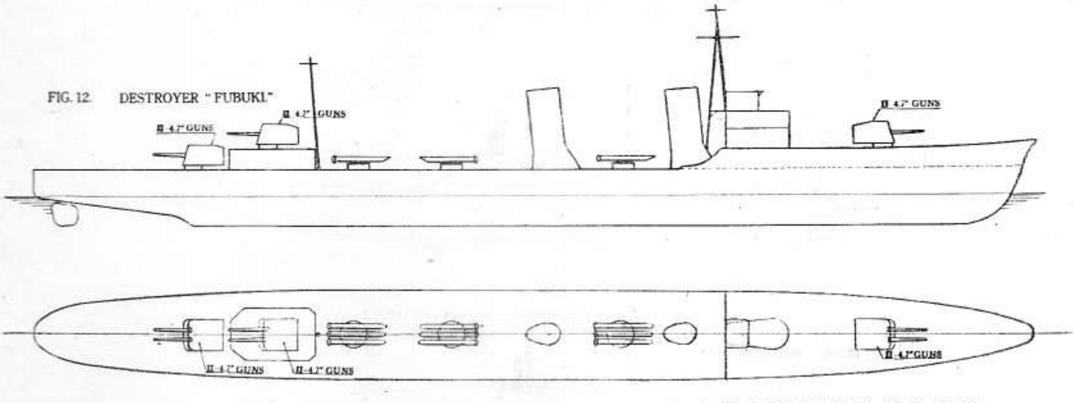
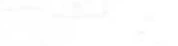


FIG. 12 DESTROYER " FUBUKI."



(Paper No. 652)

PLATE VI.

FIG. 13. SUBMARINE "RO. 60."

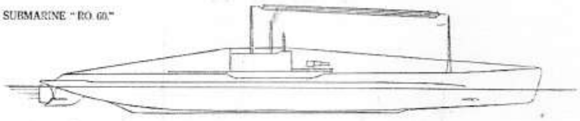


FIG. 14. SUBMARINE "I. 51."

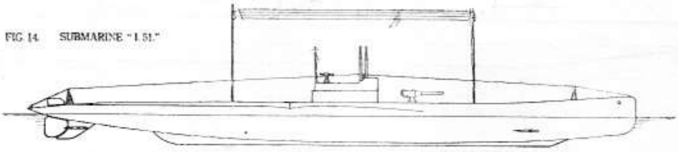


FIG. 15. SUBMARINE "I. 52."

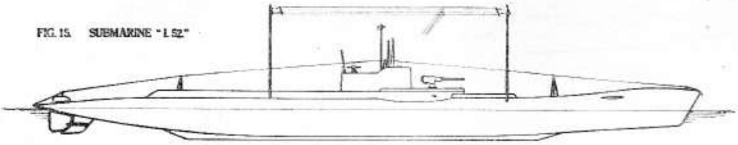


FIG. 16. SUBMARINE "I. 53."

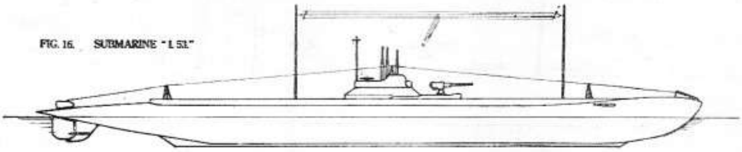


FIG. 17. SUBMARINE "I. 56."

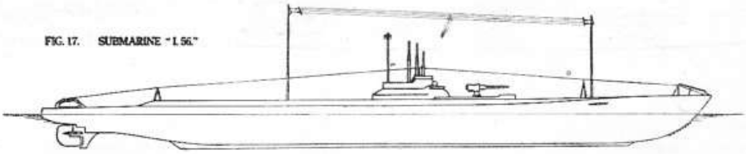


FIG. 18. SUBMARINE "I. 1."

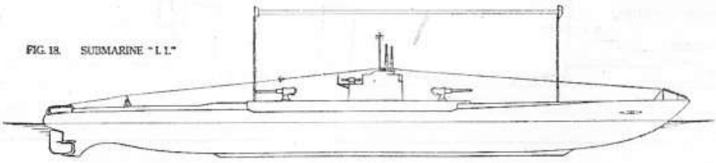
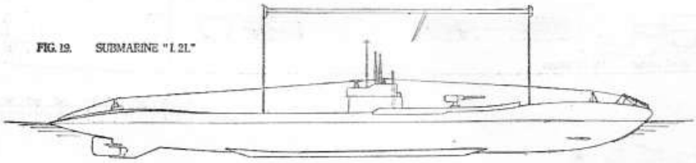


FIG. 19. SUBMARINE "I. 21."



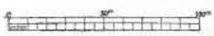


PLATE VII.

FIG. 20 GUNBOAT "SETA."

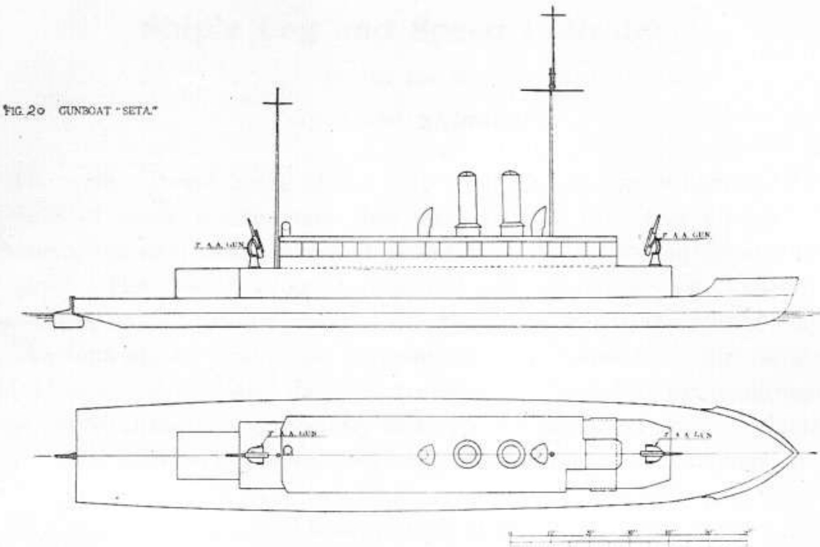


FIG. 21 SUBMARINE DEPÔT SHIP "JINGEI"

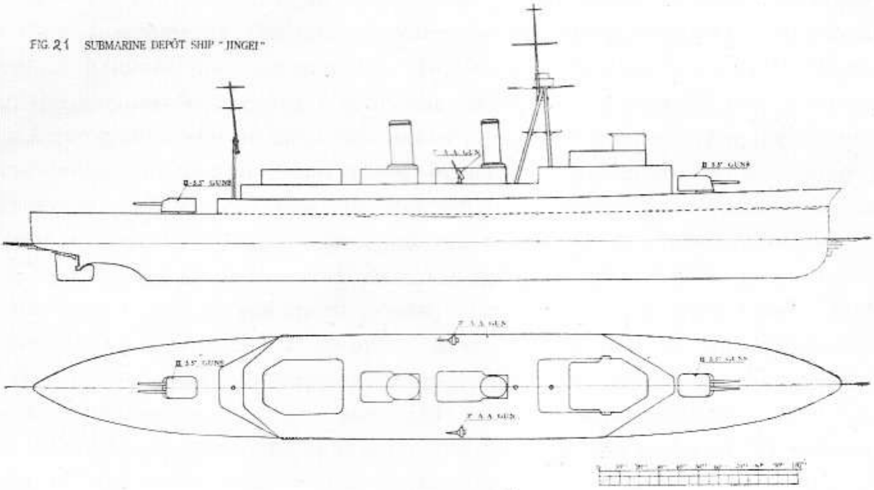
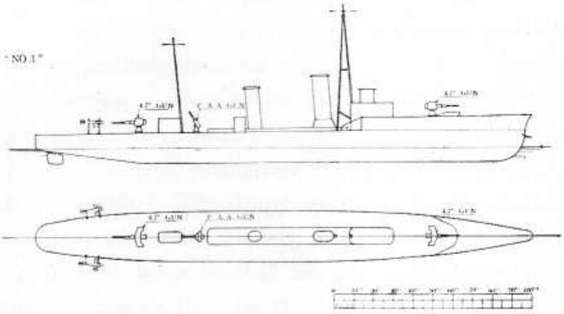


FIG. 22 MINESWEEPER "NO. 1"



Ship's Log and Speed Indicator.

(Paper No. 666)

By Seizo Shimizu.

There are several kinds of the ship's log and speed indicator. By the principle of these instruments they are divided into two classes. The mechanical log such as the common patent log and Holbus log belongs to the first class. The pressure log such as Sal log, Navigator log, Pouline log (in Sweden) and Pitot meter log (in U. S. A.) belong to the second class.

The logs of the first class indicate the ship's speed by the rotational speed of a screw and the distance trevelled of the ship by counting the number of rotations of the screw by means of a mechanical or of an electrical device. But it is well known fact that the indication of the logs in this class are not accurate.

The logs of the second class measure the ship's speed by the pressure in the Pitot tube installed at the turning centre of the bottom of the ship. In these instruments, the counter force to the pressure is eiher an elastic force of a metal plate or gravity. In Sal log, Pouline log and Navigator log, the counter force being elastic, the speed of ship is indicated by means of a lever system and an electrical device; and in Pitot meter log the counter force being the gravity, the speed of ship is indicated by means of a mechanical and an electrical devices which is regulated by the rising of a bouy in mercury trough corresponding the speed pressure of the ship.

In the logs of the second class the speed of the ship is proportional to the square root of the speed pressure produced in the Pitot tube. Then above all the logs in this class are so deviced as to indicate the square root of the speed pressure mechanically or electrically. Hence the speed meter has not uniform scale and is unreliable at low speed for some one.

When the speed of the ship is not indicated by the meter of the uniform scale its integration with respect to the time or the indication of the length of the passage of the ship which has run with variable speed of wide range from low to high will be not accurate.

The present instrument is made by an application of the principle of Dines Anemometer.

It comprises an enclosed container having a specially shaped buoy floating in mercury therein. The buoy rises in an amount which is proportional to the square root of the speed pressure. It is dynamically correct. The speed of the ship may be indicated by an electric current or an electric potential difference corresponding to the motion of the buoy. Hence the speed is indicated with an uniform scale and the indication of the log is correct whatever its speed may be.

Construction

The construction of the ship's log and speed indicator and the arrangement with its accessories are shown in Fig. 1. In the figure Q is a Pitot tube projecting from the bottom of the ship as near as possible to the ship's turning centre and is provided with two separate passageways, one adapted to be acted upon by the static pressure of the water, and the other by the kinetic pressure of the water.

These passageways respectively communicate through water pipes P_2 and P_1 with different enclosed vessels A constructed of metal or glass and partly filled with water. In each of these vessels A, the space above the water contains air, and air pipes are provided leading from these air spaces in the vessels A to different parts of an enclosed cylindrical container B in which there is Dines buoy C floating in a definite quantity of mercury.

The mercury container B is supported in any desired position in the hull of the ship by means of gimbals or a spring suspension, so that the container B is always vertical.

The air pipes leading from the vessels A are arranged so that one communicates with the space above the mercury in the container B, while the other communicates with the inside of the buoy C.

The water in the vessels A is initially adjusted as hereinafter described, so that it will come to a certain definite height at the time when the pressure in the two passageways leading from the Pitot tube is the same.

The buoy C is a cylinder which has a float G of an iverted bell shape at its upper part and all the parts are constructed of iron plates, and the air space of the float is filled with mercury. At its top an additional weight is attached so that it just sinks to the upper verge of the curved part of the float in the mercury.

The surface of the float is a surface of rotation of a curve AP shown in accompanying diagram about the axis OX and the equation of the curve is

$$x=\frac{K\pi}{4}(y^2-d_0^2)\ldots\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

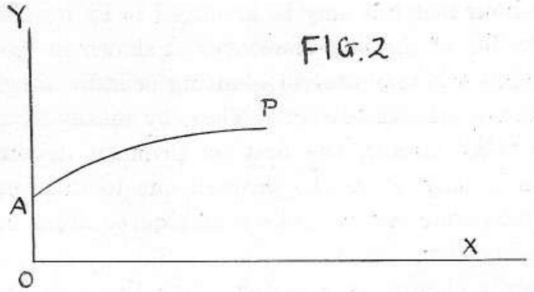
where d_0 is the inner radius of the cylinder of the buoy, and K is a constant which determines a rising amount of the buoy proportional to the ship's speed.

The length of the float is determined according to the maximum speed of the ship.

The full explanation of the principle of the buoy is given by "Mathematische untersuchung und verbesserung der winddrückregistrier apparate system "Dines" von Aurel V. Bücky, Physikalische Zeitschrift 10 Jahrgang No. 25, 1909, pp. 1008."

This buoy will be raised from its initial position through a height proportional to the square root of the pressure difference between the separate passageways in the Pitot tube Q, the initial position being understood to

FIG.2



be that which corresponds to the pressures in the two separate passageways being the same.

Hence the dimension of the apparatus and the rising amount of the buoy due to the ship's speed can be designed arbitrarily by taking the proper values of the constants K and d_0 in the equation (1).

The upper end of the buoy C carries an electrically insulated rod E, having a metalic contact T which makes sliding contact with an electrical resistance S.

A cylindrical casing J is attached to the top of the mercury container B to accommodate the rod E and resistance S, said casing being made airtight to prevent leakage of air from or into it, which would affect the container B. The resistance S is insulated from the casing J and from the container B.

The movable contact T and the resistance S are electrically connected so that they form a Ayrton & Mather's universal shunt. Maintaing the main current from the source as a constant an ammeter M can read such a current that will vary directly as the height through which the rod E is raised from its initial position.

When there is no pressure difference between the inside and outside of the buoy C the contact T is located at the lowest end of the electrical resistance S so that no current passes through ammeter M. Hence it will be seen that the current through the ammeter M is proportional to the square root of the pressure difference between the two passageways of the Pitot tube Q, and therefore this current through the ammeter M is proportional to the velocity of the sea water relative to Pitot tube Q, i.e., the current through M is proportional to the speed of the ship.

An ampere-hour meter R may be inserted in series with the ammeter M so as to directly indicate the distance travelled by the ship, i.e., it serves as log. The electrical connection may be indicated in Fig. 1.

In the figure V is an electric source of constant voltage, while W may be an electrical resistance to regulate the main current.

The motion of the contact T changes the electrical resistance of the circuit and this must be avoided by compensating it or by other device so that the main current remains unchanged for any position of the contact T.

The ampere-hour meter R may be arranged to by itself directly indicate the distance travelled of the ship more over as shown in the diagram Fig. 3 the ampere-hour meter R may control counting or indicating train L through a relay and separate driving battery. Thus, by means of an electromagnet inserted in the relay circuit, the first or primary driving wheel of the integrating train L may be moved through one tooth every time that the rotor of the ampere-hour meter causes a making or breaking of the electric current of the said relay circuit.

K′ represents a change over switch, when the instrument is in use to indicate the ship's velocity, this switch K′ is connected to the terminal 1. But when it is required to adjust the instrument for instance when the current is thought to be incorrect, the switch K′ is connected to the terminal 2 and the resistance W is so adjusted as to give a correct indication to the ammeter M.

N is a thin wooden plate which prevents the disturbance of the water surface can be caused by the oscillation of the ship.

A number of stop cocks D_1 are provided within the water pipe and air pipe lines for facilitating repair in case of damage.

Also for convenience of adjustment and repair, the two water pipes are connected by a branch pipe which is normally closed by a stop cock D, and the two air pipes are also connected by a similar branch pipe normally closed by another stop cock D.

Each of these branch pipes is communicated with outside through openings H′ and H and stop cocks D_2' and D_2. These stop cocks D_2' and D_2 are also normally closed.

To adjust the water level in the vessels A when the log is in action, four of the stop cocks D_1 are closed so as to cut off the communication of the vessels A from the Pitot tube Q and also from the container B.

The stop cocks D are then opened and also the stop cocks D_2 and D_1' so that air passes in through D_2 and water escapes through D_2'. Stop cocks D_2 and D_2' are closed after the water in the vessels A is adjusted to a certain predetermined level.

Also the two stop cocks D_1 which are in air pipes are then opened, the stop cocks D_1 in the water pipes remaining closed, while the stop cocks D in the branch pipes remain open. The quantity of mercury in the container B is then adjusted so that the buoy C is brought to a certain predetermined level, which level corresponds to the case of no current through the ammeter M. Mercury is added or withdrawn as required through the side pipe K which is provided with a stop cock.

In order to reduce the quantity of mercury required, the container B is provided with a hollow double wall F (see Fig. 1) in its lower part at which the buoy has a smaller diameter, while guide pulleys are provided therewith for facilitating the vertical movement of the buoy C within the container B.

Experiments

The relation between ship's speed and pressure difference between the separate passagways in the Pitot tube is represented by the equation

$$P=CV^2 \quad \ldots\ldots\ldots\ldots \quad (2)$$

where P, Pressure difference in the Pitot tube;

V, Ship's speed;

C, A constant depending on to the Pitot tube.

An apparatus of the above principle was made for the experiment by assuming $C=0.88$ in equation (2). The measuring range was from zero to 22 knots, and the buoy of it was designed to float up one centimeter per knot.

The dimension of the log was shown in Fig. 4. The arrangement of the experiments was shown in Fig. 5. The water tank S was designed so as to move vertically up and down by means of rope H and qulley Q to give any amount of hydraulic pressure to the air chamber D from the tank. The pressure being introduced to the container B from the said chamber, the buoy F in the container B is raised. Consequently, by shifting over the coil of universal shunt P the contact point T attached at the head of the rod E, gives an electric current in the ammeter A (speed indicator) and ampere-hour meter R (travelled distance indicator) which were connected in series. The electric source to supply the current was 100 volts. W is the regulating resistance and K junction box. A coil used as the universal shunt was made of a copper wire of 2 mm. in dia. and the length and diameter of the coil were 22 cm. and 1.5 cm. respectively for which whole range can be read to 22 knots.

Now, in this experiment, inspection was carefully made as to whether or not the current and speed hold a linear relation. For this purpose the mercury pressure gauge G was connected to the container B so as to be able to take direct reading of the speed by means of speed scale attached to the gauge, besides the pressure scale as shown in the Fig. 5. The pressure was given in the container B by raising the water tank S and readings of the speed in the pressure gauge and an electric current in the ammeter were taken simultaneously. In the other experiment, keeping the pressure in the log as constant for some time interval, the readings of the ampere-hour in the ampere-hour meter were taken.

Results of the Experiments

1. Relation between change of speed pressure and shifting amount of the buoy is shown in the following Table 1.

TABLE 1.

Speed in Knot	Raising of buoy in mm.	Falling of buoy in mm.
2.4	24.3	25.0
4.3	45.3	44.7
7.1	73.0	72.7
9.0	92.7	93.0
10.6	109.7	109.0
12.1	124.7	123.0
13.9	142.0	142.0
16.0	163.0	163.0

2. Relations between change of speed pressure and readings of ammeter and ampere-hour meter were carefully tested with an ammeter of max. current 1.5 amperes of The Weston Co., as speed indicator, and an ampere-hour meter of max. current 1.5 amperes of The Ferranti Co., as a log. Changes of speed pressure were given by the water tank S to the container B and readings of the ammeter and ampere-hour meter at every time were taken. This process was repeated several times. The relation between speed pressure and ampere meter reading was found so that it holds good a linear relation. Testing by ampere-hour meter requires a long time on account of time integral of the speed, hence, readings of the log for several speeds were taken under different conditions, viz., at normal condition, 30° inclined condition, and in oscillating state, for an hour at each case. The results of these experiments were obtained as shown in the Table 2, and in Fig. 6 and 7, and readings of ampere-hour meter were found to be held unexpectedly accurate as in the case of normal condition even the cases of bad conditions of the log in the ship.

TABLE 2.

Speed	6.8 Knots		12 Knots		18.1 Knots	
Position of Log	Oscillating Amplitude 30°, with a Ship's Period Period = 20 sec.		30° Inclined		Normal	
Time in minute	Ampere-hour meter reading	Diff.	Ampere-hour meter reading	Diff.	Ampere-hour meter reading	Diff.
0	73256	7	73294	13	73353	19
15	73263	7	73307	12	73374	18
30	73270	7	73319	13	73392	19
45	73277	8	73332	12	73411	19
60	73285		73344		73430	
		Sum 29		Sum 50		Sum 75

Comparison of the Present Log with Sal-log by Means of Volt-meter.

The experiments were carried out by an application of a volt meter as the speed indicator in accordance with the Sal-log in which the volt meter is used. In the present case, the speed pressure are given in common to both logs, but independent speed pressure was given to only the Sal-log above 20-knots and at each case volt meter reading was taken. The results of the experiments for the present log were obtained as shown in Fig. 8 and those for the Sal-log as shown by a curve in Fig. 9. The curve remarkably deviates from a straight line, while the results of the author's present log are represented by an exact straight line through the origin showing that there accurately holds the linear relation even at low speed near zero.

Summary

1. Speed of the ship causes a motion of a buoy in a mercury trough and the buoy being made by the principle of "Dines" Anemometer its displacement is proportional to the ship's speed.

2. Pressures of sea water caused by ship's speeds are transmitted with air into the log body.

3. Motion of the buoy gives an electric current or volt proportional to the ship's speed by means of Ayrton & Mather's universal shunt.

4. Ship's speed is indicated by any number of ammeters connected in series or voltmeters in parallel simultaneously.

5. Distance travelled by ship is indicated by an ampere-hour meter exactly for all range of speed is guaranteed to the log.

6. All adjustments of the log are made simply by means of an electric regulating rheostat in the main current and the change over switch arranged in the circuit.

The author wishes to express his sincere thanks to Prof. T. Okada, Director of the central Meterological Observatory, Tokyo, for his valuable advice for the research of this apparatus and to Captain C. Kato for his kindness in carrying out the experiments under his direction.

The End.

(Paper No. 666)

PLATE I.

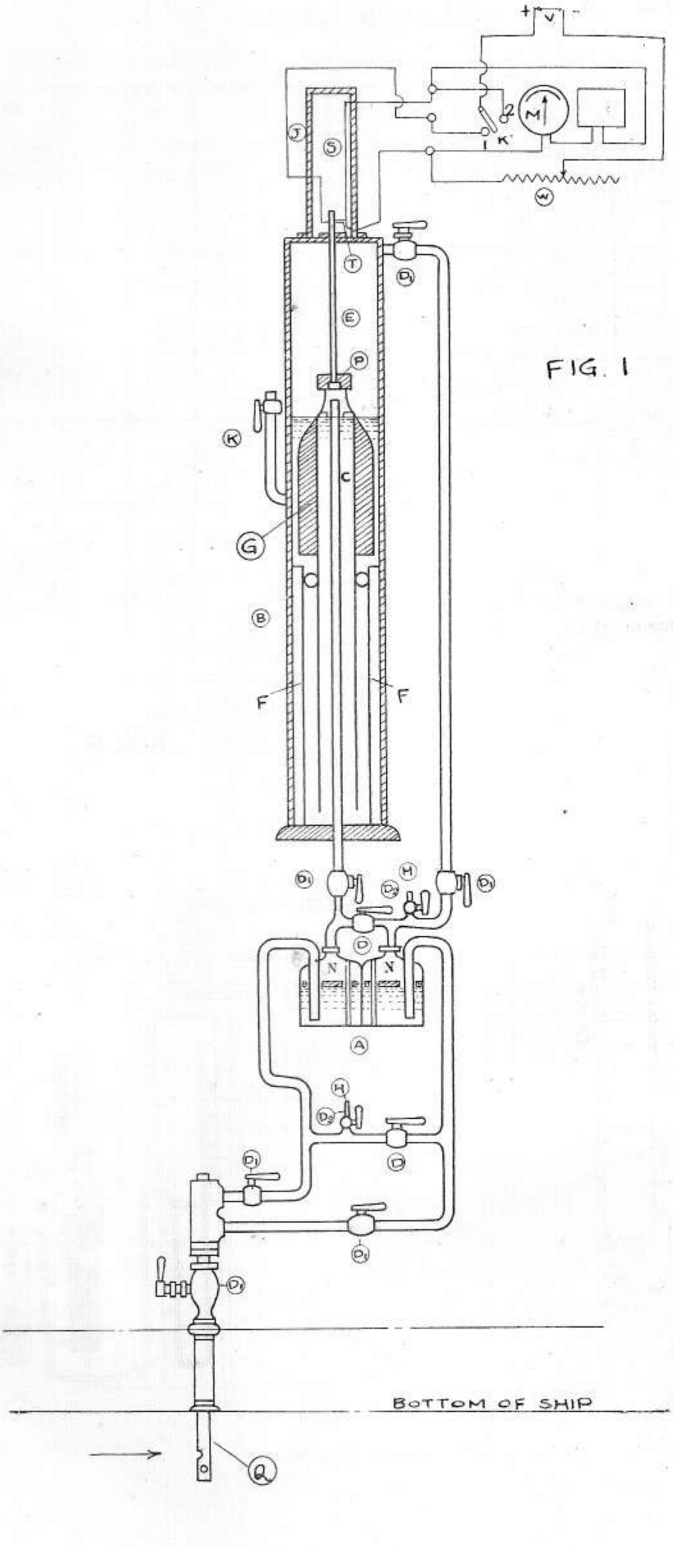


FIG. 1

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PLATE II.

FIG. 3

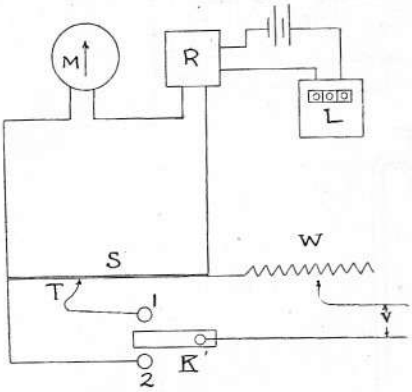


FIG. 4
Dimension of Log Used
in the Experiments.

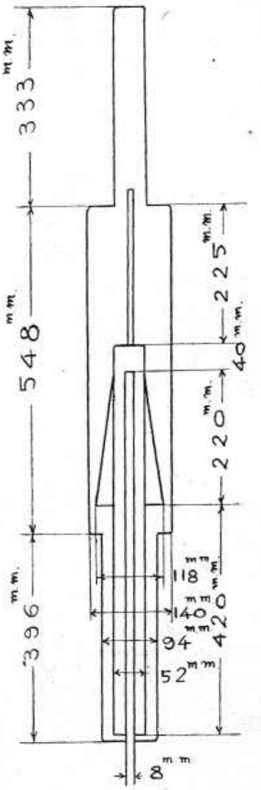


FIG. 5

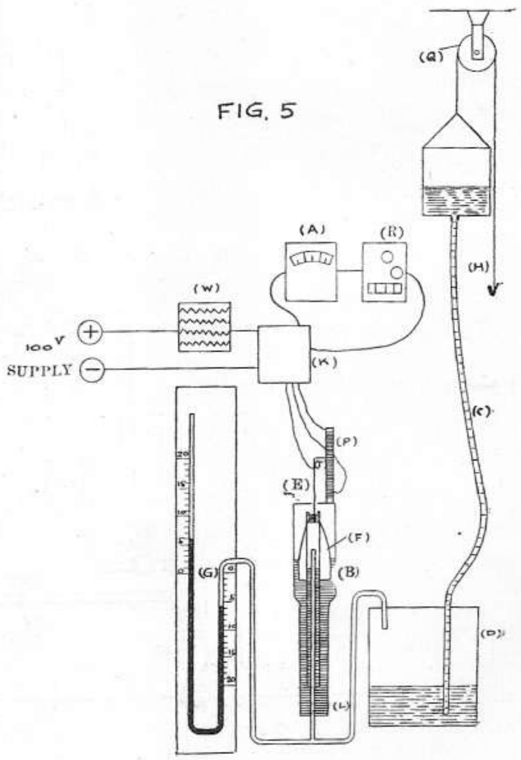


PLATE III.

FIG. 6

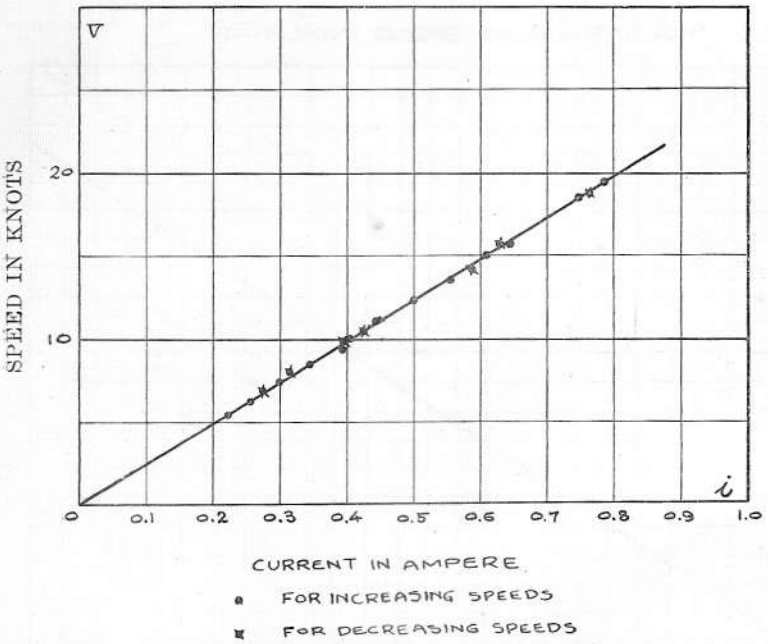


FIG 7

TRAVELLED DISTANCE METER (AMPERE-HOUR METER)
ERROR UNDER 2%

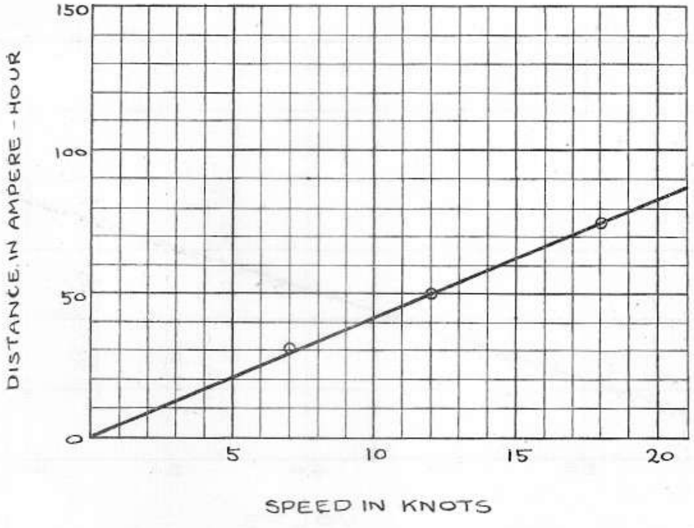


PLATE IV.

FIG. 8

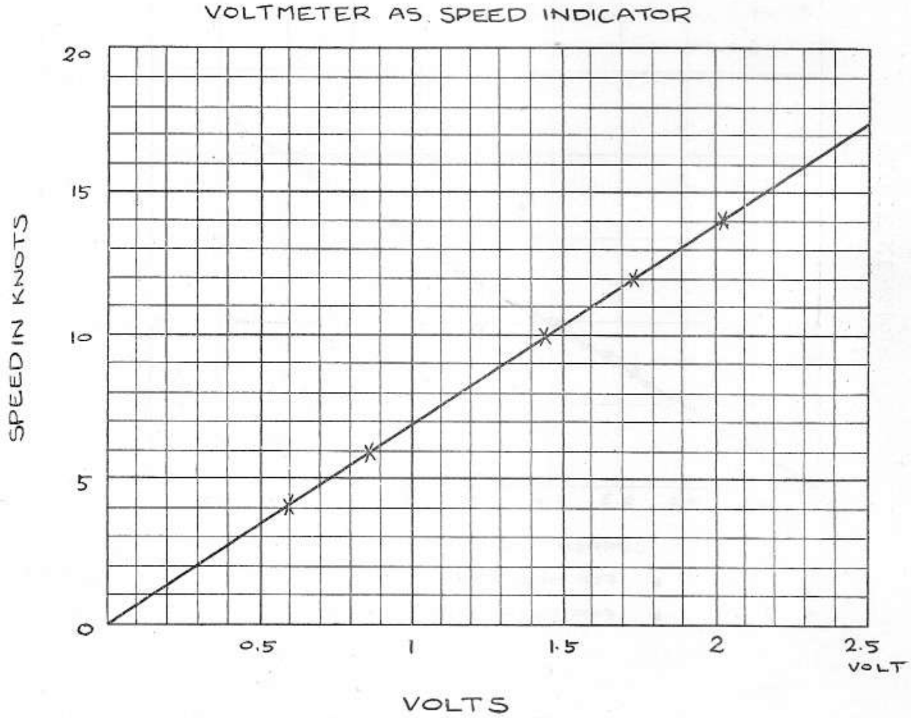
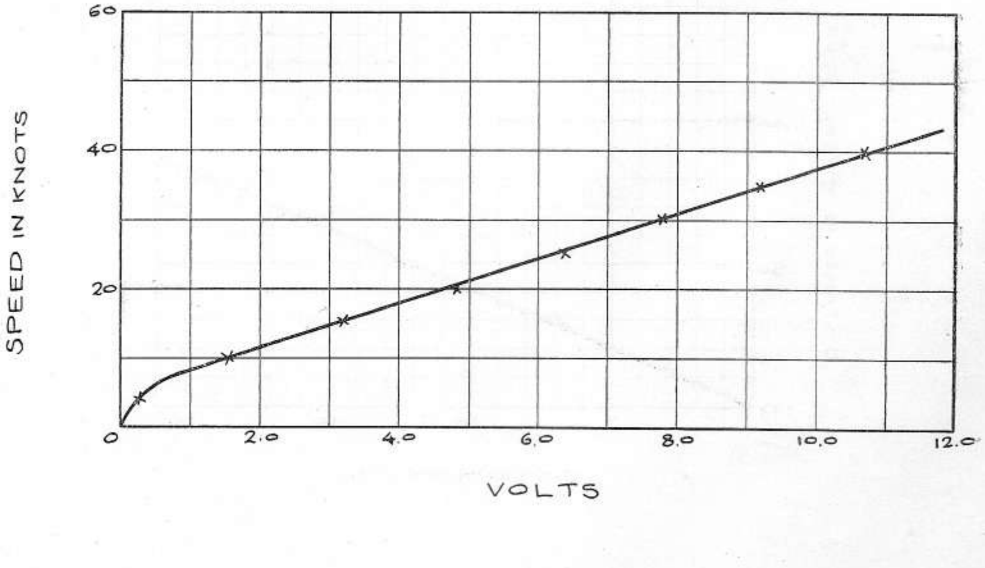


FIG. 9



La Limitation des Armements Envisagée du Point de Vue Technique.

(Paper No. 691)

*Par **André Lamouche,** Ingénieur en Chef du Génie Maritime, France.*

1.—Préambule.

Le problème de la limitation des armements navals est assez important et assez complexe pour exiger la collaboration de tous ceux qui, par leur connaissances ou par leurs fonctions, peuvent contribuer à faciliter la solution rationnelle et équitable de ce problème.

Le technicien, en particulier, ne peut se désintéresser de cette question. Son devoir est de déblayer le plus possible le terrain: d'apporter aux experts navals et aux hommes politiques des éléments aussi clairs et aussi bien classés que possible, pour éviter que la complexité technique du problème vienne encore aggraver les difficultés d'ordre militaire et politique.

La première limitation, fixée par la Convention de Washington, était basée sur des caractères quantitatifs bruts: tonnage et calibre de l'artillerie. De l'avis à peu près unanime, des bases nouvelles doivent être trouvées, pour tenir compte de certains facteurs qualitatifs dont dépend la valeur technique et militaire du bâtiment de combat.

La question que l'expert et le négociateur peuvent actuellement poser au technicien est donc la suivante:

Avez-vous un moyen de tenir compte, dans l'appréciation générale d'un navire de guerre, de toutes ses qualités principales ? Existe-t-il, entre ces diverses qalités, une commune mesure qui puisse servir de base à une comparaison scientifique et à une classification rationnelle ?

La présente étude n'est qu'une contribution à la solution de ce problème.

2.—Bases d'une classification rationnelle.

Une classification est dite " rationnelle " lorsqu'elle fait appel à un petit nombre de caractères communs à l'ensemble des éléments qu'on se propose de classer, et de telle manière que chacun de ces éléments puisse être défini par une combination ou une proportion déterminée entre ces caractères fondamentaux.

Ces caractères seront ici les qualités principales du bâtiment de combat. On a classé autrefois ces qualités en qualités offensives et qualités défensives.

Ces deux notions ont pu se présenter sous une forme assez simple, avant la guerre, car le duel était alors considéré comme circonscrit, à peu près, entre le cannon et la cuirasse. Si l'on essaie de l'approfondir aujourd'hui, on constate au contraire qu'elle se complique: 1°) du fait de la multiplication rapide, en variété et en puissance, des engins offensifs dirigés contre le bâtiment de ligne, et des sujétions qui en résultent pour sa protection; 2°) du fait que ces notions un peu simplistes ne tiennent pas explicitement compte de facteurs stratégiques et tactiques aussi importants que la vitesse, le rayon d'action, les qualités nautiques, la résistance de la coque, etc.

Si l'on veut aboutir à une classification rationnelle des navires de guerre, il est nécessaire de faire intervenir toutes les qualités essentielles, militaires et techniques du bâtiment. Et l'on s'aperçoit alors qu'elles peuvent se grouper en deux catégories, qu'on peut appeler qualités actives d'une part, et qualités résistantes d'autre part. Les premières comprendront notamment l'armement, la vitesse, les qualités manœuvrières, les qualités de puissance et de souplesse des différentes installations. Les autres comprennent la résistance de la coque, la protection, la flottabilité et la stabilité, les qualités nautiques, le rayon d'action, l'approvisionnement en munitions, les qualités d'endurance des diverses installations.

Les premières représentent, aux points de vue technique et militaire à la fois des qualités d'intensité; les dernières des qualités de capacité ou de durée.

Cette division à d'ailleurs une base scientifique: les qualités actives correspondent à de l'énergie cinétique, les qualités résistantes à de l'énergie potentielle.

Cette classification des qualités étant supposée admise, les bâtiments de combat seront classés à leur tour d'après la part faite, dans leur conception, aux qualités actives d'une part, aux qualités résistantes d'autre part.

On voit immédiatement, alors: 1°) que le capital-ship est précisément le type de bâtiment dans la conception duquel on s'est toujours efforcé de réaliser l'équilibre le plus harmonieux entre les qualités actives et les qualités résistantes. 2°) que les bâtiments de combat se rangent très régulièrement, suivant une proportion croissante entre les qualités actives et les qualités résistantes, dans l'ordre suivant:

Monitor,
Cuirassé lent,
Cuirassé rapide,
Croiseur de bataille,
Croiseur cuirassé,
Croiseur protégé,
Croiseur léger, éclaireur,
Contre-torpilleur, destroyer,
Torpilleur,
Vedette rapide, M.A.S.[1]

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(1) La liste ci-dessus ne fait pas expressément mention des types particuliers de bâtiments de combat nés de la spécialisation de certaines qualités actives ou résistantes, notamment des porte-avions et des sous-marines. On peut cependant les introduire dans cette classification, en constituant des sous-groupes.

Les divers Porte-avions construits ou aménagés jusqu'à ce jour ne sont en effet que des croiseurs de bataille, des cuirassés, des croiseurs-cuirassés, des croiseurs légers, sur lesquels l'artillerie principale a été remplacée par des avions. Il paraît rationnel (au moins à titre provisoire) de les répartir dans celles des catégories susdites auxqelles les apparente

3.—Le corollaire technique du principe d'Archimède.

Il reste, pour arriver à une définition précise d'un type de bâtiment déterminé, et à une classification rationnelle de ces types, à exprimer numériquement, en fonction d'une même unité, l'importance qui a été donnée, dans sa constitution, à chacune des grandes qualités actives ou résistantes.

C'est ce que tend à réaliser, dans une certaine mesure, l'expression de ces différents facteurs en unités de poids (bilan des poids), ou encore, pour permettre la comparaison, en pourcentages du déplacement. Mais à l'heure actuelle ces données numériques ne se prêtent pas, en général, à une comparaison suffisamment précise, en raison d'une part des types très variables des installations; en raison d'autre part de l'arbitraire qui règne sur le mode de classification ou de groupement des poids. Et cette difficulté s'accentuera à mesure que la technique ira se différenciant devantage.

Il est donc désirable d'arriver à exprimer les différentes qualités actives résistantes d'un bâtiment en fonction d'une unité telle, que cette expression résume plus exactement et plus complètement que ne peut le faire un simple poids l'importance ou l'efficacité militaire et technique, à la fois, qu'on a voulu attribuer à chaque installation.

Or ce mode d'évaluation nous sera fourni par la loi très générale qui a régi toute l'évolution des constructions navales, et qui n'est en définitive que le corollaire technique du principe d'Archimède.

Qu'il s'agisse de l'énergie résistante de la coque, de l'énergie résistante des blindages; qu'il s'agisse de la puissance propulsive de l'appareil moteur, de la puissance destructive de l'artillerie, toute l'évolution des constructions navales a été régie, techniquement, par cette loi générale, qui est la conséquence directe du principe d'Archimède: la réduction systématique du poids de l'unité d'énergie (ou de l'unité de puissance) concentrée dans les diverses installations.

Les progrès successifs de l'architecture navale, en effet, qu'ils aient consisté dans l'invention de procédés nouveaux ou dans le perfectionnement des solutions déjà appliquées, ont toujours été caractérisés par la tendance à faire, pour toutes les

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leur type général, en constituant seulement dans chacune d'elles un sous-groupe caractérisé par cette importante particularité d'ordre militaire et technique à la fois.

Quant aux sous-marins, leur arme principale étant la torpille, ils se trouvent logiquement rattachés au groupe des torpilleurs, la vitesse plus faible étant compensée dans ce cas, aux points de vue offensif et défensif à la fois, par l'effet de surprise et par l'immunité spécial que confère aux sous-marins la possibilité de s'immerger. Dans ce groupe général des torpilleurs, ils constitueront seulement un ou plusieurs sous-groupes bien distincts. (Les grands sous-marins de croisière dans lesquels l'artillerie reprendrait la prépondérance formeraient un sous-groupe des croiseurs légers).

Le fait de parvenir ainsi, par des conventions raisonnées, à faire entrer dans une classification unique tous les types de bâtiments de combat, n'empêcherait bien entendu nullement, lors d'une nouvelle limitation internationale, de détailler cette limitation par groupes et sous-groupes, selon les types que l'on croirait devoir à ce moment-là contingenter séparément.

catégories de navires, des bâtiments dans lesquels, après avoir amélioré le plus possible les qualités de la matière et le rendement des appareils, on tendait fatalement à réduire progressivement les coefficients de sécurité, qui ne représentent autre chose que des marges de puissance ou de résistance. Or ces deux étapes de l'évolution générale des constructions navales se ramenaient bien l'une et l'autre à la diminution systématique, pour les diverses installations, du poids de l'unité d'énergie active ou résistante effectivement utilisée.

Les limitations de tonnages et de calibres fixées à Washington n'ont fait qu'intensifier cette tendance générale. La Convention de Washington a transformé la course à la quantité en course à la qualité. Mais il est bien évident qu'il ne faut pas prendre ici l'opposition entre "quantité" et "qualité" au sens absolu que lui donnent les philosophes. Cette opposition signifie simplement que la valeur technique et militaire des diverses installations d'un bâtiment est de moins en moins fidèlement exprimée par des chiffres bruts, tels que déplacements ou calibres, échantillons de coque ou épaisseur de cuirasse, poids totaux d'appareils moteurs ou de combustible. Il faut, par delà ces caractéristiques purement nominales, connaître le poids unitaire (le poids par unité d'énergie ou de puissance) des installations correspondantes. La course à la qualité, c'est finalement la course au rendement de l'unité de poids de matière: la course à l'utilisation énergétique du kilogramme de navire.

Il était inévitable, d'ailleurs, qu'une telle tendandance fût portée au plus haut degré par les Marines les plus strictement limitées quant à la quantité seule. C'est bien ce que confirme la publication des caractéristiques des bâtiments récemment mis en chantier en Allemagne. Cet accroissent extrême du rendement énergétique de chaque tonne de navire, qui fait le plus grand honneur à la science et à l'industrie allemande, n'est d'ailleurs obtenu, dans ce cas, qu'aux prix de sacrifices financiers qui ne peuvent être considérés que comme un luxe isolé. La course à la qualité, si elle se généralisait à ce degré exceptionnel d'intensité, devrait être limitée comme la course à la quantité.

Quoi qu'il en soit, le corollaire technique due principe d'Archimède, dont l'importance et la généralité ont été mises en évidence ci-dessus, va nous fournir les éléments nécessaires pour tenir compte avec précision du facteur "qualité," dans la classification rationnelle des bâtiments de combat. Il peut permettre d'apporter plus de clarté, tant dans l'étude des projets de bâtiment, que dans les discussions relatives aux limitations futures des armements navals.

4.—Bilan des poids et bilan énergétique.

Ainsi qu'on l'a rappelé plus haut, le bilan des poids est insuffisant, non seulement pour caractériser la valeur technique et militaire du bâtiment, mais même pour permettre de définir avec précision la classe à laquelle il appartient. Il faut connaître en outre, non seulement le mode de groupement des poids partiels, qui diffère dans les diverses Marines, mais aussi le type des différentes installations.

Au point de vue technique, ces diverses installations se trouvent définies de façon beaucoup plus précise par les poids unitaires qui leur correspondent. Si l'on rapporte ces poids unitaires, d'après ce qui a été vu plus haut, à des caractéristiques de puissance ou d'énergie, le mode de définition complet ainsi adopté pour les bâtiments de combat fournira finalement une expression de leur valeur technique et militaire à la fois.

Pour cela, il faut établir, concurremment avec le bilan des poids, le bilan énergétique complet des bâtiments de guerre.

Ce bilan énergétique sera établi d'abord a posteriori, pour les bâtiments antérieurement construits. On déduira de là, par la voie statistique, des coefficients qui ultérieurement permettront d'établir approximativement à l'avance le bilan estimé des bâtiments nouveaux à construire, exactement comme cela s'est fait pour le bilan des poids.

Ce bilan énergétique ne sera nullement destiné à remplacer le bilan des poids, ni le difficile balancement des caractéristiques de toutes sortes auquel donne lieu, plus généralement, l'élaboration progressive d'un projet de navire. Il apportera seulement un élément de plus à cette discussion si difficile, en explicitant certains facteurs qui demeurent à l'heure actuelle trop implicites, trop purement qualitatifs surtout, échappant ainsi à une discussion quantitative précise de l'ensemble du projet.

Il fournira une traduction mutuelle des caractéristiques militaires d'utilisation d'une part, et des caractéristiques techniques de construction d'autre part. Il précisera numériquement le mécanisme des échanges (bénéfices d'un côté, sacrifices de l'autre) auxquels on doit procéder, au cours de la préparation et de la discussion d'un projet de navire.

A titre d'indication, voici une des formes sous lesquelles il semble que ce bilan énergétique puisse se présenter utilement.

On partira de la même relation fondamentale que pour l'équation des poids:

$$D = \Sigma p,$$

qui exprime simplement que le déplacement est égal à la somme des poids élémentaires entrant dans la construction du bâtiment.

Mais au lieu d'exprimer ces poids p en fonction du déplacement, on les décomposera en deux facteurs: l'un représentant, une donnée militaire d'utilisation, l'autre un coefficient technique de construction.

L'identité d'où l'on est parti se présentera donc sous la forme:

$$D = \Sigma \lambda \mu.$$

Les quantités μ représenteront les données du programme (vitesse, rayon d'action, armement, approvisionnement en munitions, protection, etc.). Ces données seront, ainsi qu'il a été dit plus haut, traduites sous la forme qui résume le plus complètement leur efficacité, c'est-à-dire en unités de puissance ou d'énergie.

Les quantités λ ne seront pas autre chose dès lors que les poids unitaires, selon la conception générale qui en a été donnée ci-dessus: poids par unité de puissance ou d'énergie, active ou résistante selon le cas.

On sera conduit, en définitive, à une forme nouvelle de l'équation de déplacement, qui pourra être discutée en elle-même si le déplacement est fixé a priori. Dans le cas contraire, elle sera établie et discutée concurremment avec l'équation de déplacement sous sa forme habituelle.

On peut remarquer d'ailleurs que, dans la pratique, c'est bien de cette manière que l'on procède, mais en considérant isolément les installations sur lesquelles on veut faire porter tel sacrifice ou tel bénéfice, et en rapportant les " poids unitaires " à des unités différentes selon les installations envisagées.

La présente étude tend seulement à préciser et à unifier la méthode, en substituant à ces balancements isolés un bilan d'ensemble.

Il reste à donner quelques exemples de la manière dont on peut former explicitement les différents termes de ce bilan énergétique. Notons, à ce propos, qu'il y a un avantage important à présenter ce bilan sous la forme indiquée ci-dessus, c'est-à-dire en partant de termes exprimés en unité de poids: c'est que l'on peut se contenter de décomposer en facteurs d'énergie ou de puissance ceux de ces termes pour lesquels on sait effectuer cette opération avec une précision suffisante, à l'époque considérée, Pour les termes, au contraire, qui, soit définitivement par leur nature (personnel, vivres, etc.), soit provisoirement du fait que l'étude de l'utilisation énergétique moyenne de l'unité de poids de matière est encore insuffisamment avancée, ces termes peuvent rester exprimés en unités de poids, sans détruire l'homogénéité de la relation qui traduit le bilan énergétique.

La vitesse et le rayon d'action sont évidemment les caractéristiques auxquelles s'applique de la façon la plus immédiate la conception présentée ci-dessus.

Les poids unitaires sont ici respectivement, le poids par cheval de l'appareil moteur et évaporatoire, et le poids par cheval-heure du combustible consommé par cet appareil à une allure donnée. L'accroissement ininterrompu de la puissance massique des appareils d'une part, de leur rendement énergétique d'autre part pour un combustible donné, le choix même de ce combustible enfin, ont entrainé la réduction progressive de ces deux poids unitaires, par unité de puissance et par unité d'énergie.

Si V_1 est la vitesse maxima prévue au programme ou résultant d'une première estimation; V_2, la vitesse tactique principale (celle qui détermine l'approvisionnement total en combustible), et Δ_2 la distance franchissable correspondante; F_1 et F_2, les puissances qui correspondent à V_1 et V_2, on aura ici:

$$\mu_1 = F_1, \quad \mu_2 = \frac{F_2 \Delta_2}{V_2}.$$

Quant aux facteurs λ_1 et λ_2, le premier sera le poids par cheval de l'appareil moteur et évaporatoire, le second le poids de combustible consommé par cheval-heure à la vitesse V_2.

Dans le cas de l'armement, et de l'artillerie notamment, il est aisé de mettre en évidence deux caractéristiques de puissance et d'énergie aussi importantes que dans le cas de la propulsion, et de les décomposer en facteurs d'utilisation et facteurs de construction, comme ci-dessus.

Si, pour l'artillerie principale par exemple, on appelle:

N, le nombre de pièces;

p, le poids du projectile;

v, la vitesse initiale;

n_1, le nombre de coups tiré par pièce et par minute;

n_2, le nombre de coups par pièce correspondant à l'approvisionnement total.

Le facteur de puissance correspondant sera:

$$\mu_3 = \frac{Nn_1\, pv^2}{2g}.$$

Et le facteur d'énergie (qui correspondrait exactement, au point de vue militaire et au point de vue technique à la fois, au rayon d'action dans le cas de la propulsion) sera:

$$\mu_4 = \frac{Nn_2\; pv^2}{2g}$$

Les coefficients de construction correspondants λ_3 et λ_4 seront ici encore des poids unitaires, rapportés à l'unité de puissance dans le premier cas, à l'unité d'énergie dans le second cas.

On aurait à faire des calculs analogues pour la coque, pour la protection, etc.

On peut encore présenter le bilan énergétique sous une forme différente, se prêtant à un résumé historique très condensé de l'évolution des principaux types de navires de guerre.

Cette nouvelle présentation ne sera donnée ici qu'à titre purement indicatif et approximatif. Il serait nécessaire, pour lui conférer une précision suffisante, d'affecter à ce travail un nombreux personnel dessinateur ou calculateur. On pourrait concevoir que ce travail fût exécuté et tenu à jour par un Bureau international permanent d'étude et de contrôle, qu'il faudra sans doute créer un jour, si l'on veut faire en matière d'armements navals de la limitation rationnelle et effective.

Ce mode de représentation consiste, pour chacune des classes définies (ainsi qu'il a été dit plus haut) par une certaine proportion moyenne entre les qualités actives et les qualités résistantes, à figurer graphiquement cette répartition, ou ce bilan des qualités actives et des qualités résistantes. On pourra prendre pour abscisses les temps. Quant aux ordonnées, si l'on prend les déplacements et leur répartition déduite du bilan des poids, on aura une première approximation (graphique I); celle-ci se trouvera précisée davantage si l'on porte au contraire en ordonnées la répartition déduite du bilan énergétique (graphique II). La connaissance des poids unitaires est nécessaire pour passer du premier au second (ou encore la connaissance directe des caractéristiques de puissance ou de résistance énergétique des principales installations).

Si d'ailleurs, dans chacun de ces deux cas, on rapproche les courbes correspondant aux diverses classes de bâtiments, en les disposant suivant une troisième coordonnée, on aura un diagramme résumant la classification conçue sur ces bases. Ce dernier mode de représentation a, par surcroit, l'avantage de mettre en évidence la loi de continuité qui règne entre les différents types (loi à laquelle recourt forcément tout auteur d'un project nouveau, au cours des interpolations et des extrapolations auxquelles il est obligé de se livrer). Il permettrait de situer éventuellement, sur la surface définie par cette série de courbes de niveau (si l'on peut s'exprimer ainsi), un type hybride qui n'aurait pu être rattaché exactement à aucune des classes précédemment définies.

On n'a pas fait figurer dans cette représentation (donnée, il faut le redire, à titre purement indicatif et approximatif) les sous-groupes relatifs aux porte-avions et aux sousmarins. Ceux-ci constitueraient simplement des arcs de courbes spéciaux; ou, si l'on veut, des nappes particulières plus ou moins étendues, dans certaines régions de la surface générale définie par cette représentation.[1]

Enfin, il serait possible, bien entendu, dans un graphique plus précis et plus détaillé, de décomposer chacune des fractions correspondant aux qualités actives et aux qualités résistantes en leurs éléments essentiels (la première en armement et vitesse, etc.).

5.—Conclusions.

L'évolution contenue de l'Architecture Navale a tendu, non seulement à l'accroissement quantitatif (déplacement, calibres, etc.) mais aussi et surtout au perfectionnement qualitatif. Or ceci signifie simplement, pour le technicien: accroissement incessant de la quantité d'énergie ou de puissance concentrée dans l'unité de poids des diverses installations.

Cette tendance générale a été intensifiée par les récents traités ou conventions fixant des limitations quantitatives (Convention de Washington, Traité de Versailles). Tant que cette tendance suivra le rythme général du progrès industriel, on ne pourra que se féliciter de la contribution qu'elle apportera à ce progrès (comme elle le fit toujours dans le passé), ou du bénéfice qu'elle pourra inversement en tirer.

Mais il pourrait arriver qu'il se produisit exactement dans l'ordre qualitatif ce qui a eu lieu dans l'ordre quantitatif avant la Conférence de Washington. C'est-à-dire que la concurrence internationale portât cette tendance à un tel degré, que les dépenses spéciales faites pour le perfectionnement intensif du matériel naval fassent croitre à nouveau démesurément les budgets des différentes Marines.

A ce moment-là on devrait superposer à la limitation quantitative une limitation qualitative, c'est-à-dire (d'après ce qui a été dit plus haut) une limitation d'ordre énergétique; ou encore une limitation budgétaire, qui remplirait indirectement le même but. Car ce ne serait plus alors le prix par tonne qui déterminerait le coût de construction d'une flotte, mais plutôt le prix par unité d'énergie ou de puissance.

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1) On a pris en général, pour établir ces graphiques, les bâtiments de la Marine française. Pour les grands bâtiments cuirassés postérieurs à 1914, on a pris ceux des Marines anglaise et américane.

En résume, on peut retenir de ce qui précède:

1°) Que la valeur militaire (the fighting power) des bâtiments de combat ne peut être définie uniquement par le poids total d'un flotteur ou le diamètre d'un tube;

2° Que pour définir cette valeur militaire (comme d'ailleurs pour établir une classification rationnell) il faut tenir compte au moins autant des quantités d'énergie ou de puissance que des déplacements ou des calibres nominaux. On y parviendra en établissant un bilan énergétique.

3°) Que pour éviter de porter atteinte à l'ingéniosité des techniciens, comme à la liberté industrielle et politique des différentes nations, on pourrait par exemple faire intervenir cette condition dans les limitations futures des armements navals, en affectant simplement d'un coefficient de majoration, pour le calcul des tonnages alloués à chaque nation, les déplacements des bâtiments d'après la valeur énergétique moyenne de leurs installations militaires.[1] Ce coefficient spécial devrait, dans tous les cas, être appliqué spécialement aux bâtiments munis d'installations dont le rendement énergétique par unité de poids serait nettement disproportionné avec les possibilités industrielles moyennes des grandes nations contractantes. Un autre procédé pourrait consister (puisque cette disproportion entraînerait forcément des prix de revient par tonne aussi disproportionnés), à superposer à la limitation des tonnages et des calibres par classes, une limitation budgétaire globale,-celle-ci étant d'ailleurs calculée assez largement pour respecter l'indépendance de chaque nation dans toute la mesure compatible avec la sauvegarde de la paix universelle.

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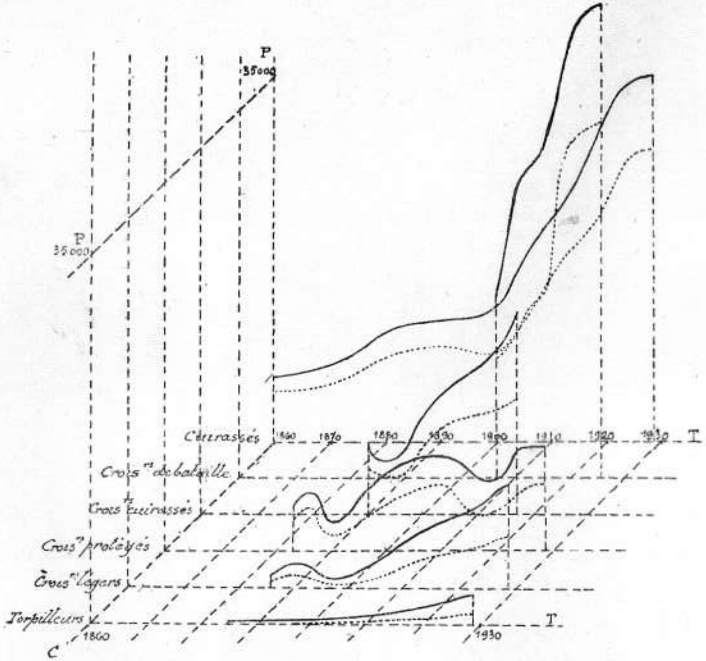
[1] Il serait d'ailleurs aisé de faire intervenir dans ce coefficient l'âge du bâtiment

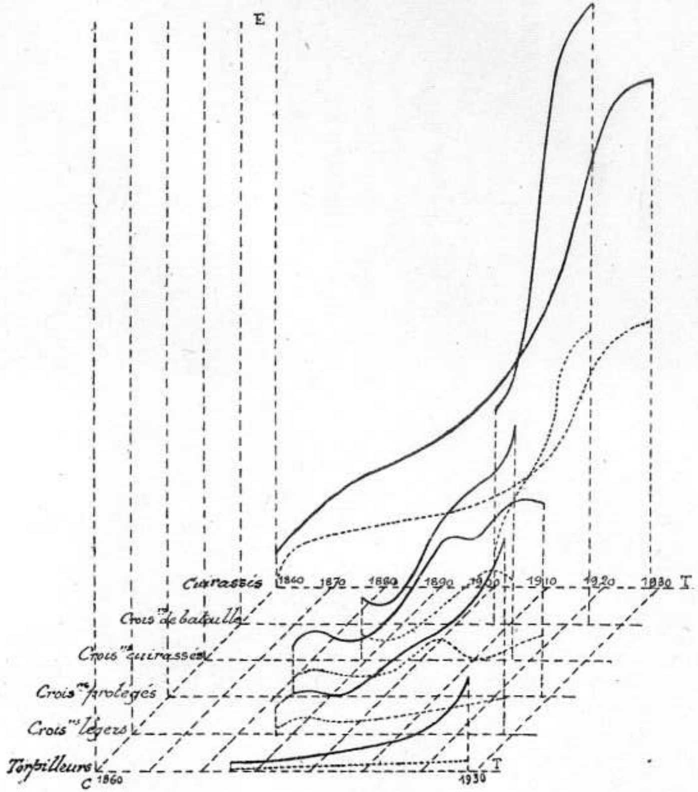


Graphique I
(établi d'après le bilan des poids)

Légende

┈┈┈┈ Courbe des poids correspondant aux qualités resistantes

━━━ ———do.——— totaux (la portion d'ordonnée comprise entre les 2 courbes représente donc les poids consacrés aux qualités actives).



Graphique II
(établi d'après un bilan énergétique approx.)

帝國海軍機關の發達

(Development of Marine Engines in Imperial Japanese Navy.)

(Paper No. 749)

海軍少將　牛丸福作

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我國内に於ける舶用機關の製造は1863年（文久3年）石川島造船所が幕府の命により千代田を建造したる時を以て初めとする。爾來年を經ること67年其間我海軍が英國を師とし，佛國に學び，米國に習ひ，叉諸種の研究を重ねて今日の域に達するを得たのであるが1894年乃至1895年の日清戰役及1904年乃至1905年の日露戰役に於ける我主力艦の全部及補助艦艇の大部分は，猶歐米諸國（主として英國）で建造したものであつた。然るに此兩戰役を經て我造船技術は急激なる發達を遂げ機關の製造技術も亦全く其面目を一新し得たのである。以下各項に亘り其發達の大要を述べようと思ふ。

第一章　蒸汽機關の發達

（1）主機械

我國最初の建造艦千代田の主機械は60馬力のもの1臺で其の制式は横置直働機關である。これが我國人の手によりて造られた舶用機關の嚆矢である。1875年（明治8年）には木製砲艦清輝に443馬力の横置式三筩機關を採用し叉外輪式の御召艦迅鯨に1,400馬力の斜動式機關を裝備した。1891年（明治24年）には砲艦大島に初めて1,200馬力の直立三聯式機關を裝備し，1895年（明治28年）には巡洋艦須磨に，2軸8,500馬力の同式機關を裝備した。

日清戰役後に至り戰艦三笠以下12隻の主力艦及巡洋艦，驅逐艦の多數を一時に英，米，獨，佛の諸國に註文し技術上大に得る處があつた。1901年（明治34年）には2軸9,500馬力の直立四筩三聯式機關を巡洋艦新高に裝備し，1905年（明治38年）には一躍して2軸19,000馬力のものを巡洋艦筑波及生駒に，叉2軸17,000馬力のものを戰艦薩摩に，翌年には2軸21,000馬力のものを巡洋艦鞍馬に裝備して何れも好成績を得たのであるが，爾來蒸氣「タルビン」の發達に從ひ大力量の吸鎒式機關は製造せられぬやうになつた。

是れより先き，1900年（明治33年）驅逐艦霞及曉の2艦を英國に註文の當時，其の中の1艦に「パーソンス・タルビン」を裝備せんとして製造會社との交渉も纒まつたのであつたが，輿論の反對に遇ひ遂に沙汰止みとなつたことがある。これ我海軍が蒸氣「タルビン」を採用せんとした最初である。其後1905年（明治38年）に至り，英國海軍が「ドレッドノート」に「タルビン」を搭載することが一般に明かとなる

や，漸次我國に於ても「タルビン」を採用せざるべからずとの議論が盛となり，遂に1905年（明治38年）11月巡洋艦伊吹の計畫を變更し，吸鍔式に換ゆるに「タルビン」を以てすることとなつた。其の制式に關しては「パーソンス・タルビン」を可とするものと「カーチス・タルビン」を可とするものとの2派を生じ議論區々であつたが時の當局者宮原機關中將は反動「タルビン」も衝動「タルビン」も各利點はある，併し自己の經驗より得たる Engineering Commonsense により判斷すれば「カーチス」式の方が優良であると斷定し，1906年（明治39年）匆々米國「フオワーリバー」會社に2軸23,000馬力のものを註文した。本機關は翌年8月完成し9月艦内に搭載された。これは斯る大力量の「カーチス・タルビン」を軍艦に裝備した世界の最初である。同年引續き戰艦安藝にも同會社製2軸21,600馬力の「カーチス」式が採用せられ，此兩艦の公試成績は吸鍔式機關裝備の姉妹艦に比べて優良なる成績を擧げ得たのである。

翌1907年（明治40年）には通報艦最上に初めて2軸8,000馬力の「パーソンス・タルビン」が採用された。爾後建造の軍艦及驅逐艦には總て「タルビン」が採用せられ，1915年に至る約8年間は「パーソンス」,「カーチス」,「インプルーブト・パーソンス」及「ブラウン・カーチス」等が採用せられた。此間に4軸64,000馬力の巡洋戰艦金剛が英國「ビ」社に註文せられ，又同型艦3隻が内地にて建造せられた。此4隻は今猶高速主力艦として我海軍の中堅となつて居る。此等の「タルビン」には低力に於ける效率を增進する爲，主「タルビン」内に巡航段落を設けられたものが多い。此期間に於ける「タルビン」の製造所は海軍工廠及三菱，川崎の兩造船所であつた。

1913年（大正2年）度の戰艦伊勢，日向には齒車減速裝置附の獨立巡航「タルビン」が採用せられて主「タルビン」軸に連結せられた。これ我海軍に於ける齒車減速裝置採用の初めである。

1915年（大正4年）度の驅逐艦桃級の主「タルビン」は2軸16,000馬力の比較的小型ではあるが，齒車減速裝置附の獨立巡航「タルビン」を有し，且つ主「タルビン」と共に我海軍にて初めて計畫せられたものである。これ迄我海軍工廠及三菱，川崎の兩造船所にては既に多數の艦艇用「タルビン」を製造し，當時製造上の技術に於ては歐米に比し，毫も遜色なき自信を有して居たが，其の計畫は常に歐米諸會社の手で爲されたものであつた。驅逐艦桃級に至つて始めて計畫の獨立を實現し得たのである。

1916年（大正5年）度の驅逐艦谷風級及輕巡洋艦天龍級の主機械は共に我海軍に於ける「オールギヤード・タルビン」採用の嚆矢である。此等の機關は内地にて製造せられ，其の公試運轉には頗る優秀なる成績を得た。爾後は「タルビン」の蒸氣消費量も大に減ずることが出來た。

1917年（大正6年）起工の巡洋艦球磨級には技本式「タルビン」が裝備せられた。此技本式「タルビン」は海軍技術本部の計畫で，重量容積小さく且つ總ての力量を通じて經濟的なる點に於て特に良好なる成績を擧げ得たのである。又私立造船所建造の驅逐艦には「ツェリー」式及 M.V.式「タルビン」等を裝備したものもあるが，最近建造の驅逐艦には總て艦本式「タルビン」が裝備せられて居る。本「タルビン」は「インパルス」式で，其の翼及噴口の形狀は艦政本部考案のもので，將來の各種艦艇

には總て此式が採用せらるることと信ずる。

「オールギヤード」式が採用せられた結果，翼の周速度は每秒 500 呎乃至 600 呎にも上り，從て段落及翼の數は著しく減ぜられた。これが爲翼の受くる應力は急激に增大し，金剛級の「タルビン」翼に比すれば實に數倍にも達し，從て材質，形狀及工作法に大なる進步改良を生じた。翼の材料は初め眞鍮又は燐靑銅を主として使用し，又時としては 5 %「ニツケル」鋼又は「モネール・メタル」等が使用せられたが，最近に至りては總て不銹鋼が採用せらるるに至つた。翼の形狀は私立造船所製造のものに限り其の製造權を有する型式を採用して居るものもあるが，海軍工廠で製造するものは總て艦本型の翼を採用し，工作法に於ても植込法に於ても極めて丁寧で且つ確實なる方法を採用して居る。又高速「タルビン」の採用以來翼の故障を生じた實例も屢々あつたが，是等は主として翼車の振動に起因するものと認められ，其の研究は充分微細なる點まで進で居ると確信する。

現在我海軍で採用して居る減速裝置は，米國「ウェスチングハウス」社の遊動齒車式と英國「パーソンス」社の固定齒車式とである。前者は主として大艦に，後者は重に小艦に採用せられて居る。齒車の周速度は每秒 140 呎を普通とし，齒幅の每 1 吋に對する壓力は 1,000 听乃至 1,200 听に達して居る。又齒の角度は初期のものは 45 度であつたが次で 30 度に改められた。我海軍が齒車減速裝置の採用以來過去十數年に亘り嘗て齒車の故障を生じた實例なきは，計畫及工作の確實なると，取扱の優れたる結果と考へらるるのであつて，吾人の大に意を强ふする處である。

往復式機械を裝備せる時代の推力軸承は密閉式即ち推力鍔を密閉せるものと露出式即ち馬蹄形受推片を備ふるものとの兩式が採用せられた。而して力量の增大するに連れ其形體も益々增大したが潤滑の理論上より言へば何等の發達を見なかつた。又「パーソンス」型調整用軸承が採用せられ，且つ「タルビン」が主機械として採用せらるるに至つてよりは，「タルビン」內に於ける蒸氣の壓力と推進器の推力とを相平均せしむる裝置が採用せられて，推力が增加する割合に軸承の形體が尨大とはならなかつたが，直結「タルビン」の晩年に至ると隨分大きなものとなり，取扱上にも諸種な困難が伴つて來た。然るに 1918 年（大正 7 年）初めて「ミツチェル」式の高壓推力軸承が我海軍に採用せられ驅逐艦に裝備せられて以來一大革命を生じたと云ひ得ると思ふ。爾來我海軍にて採用せられた型式は「キングスベリー」式，「パーソンス」式及「ヱッシヤーウヰッス」式等があるが，主として「パーソンス」式が大部分を占めて居たのである。數年前よりは特許海軍型と名づくる鋼球應用の推力承も採用せられ、頗る優秀なる成績を擧げつつある。

米國に於て電氣推進が軍艦に採用せられて以來短時日間に異常なる發達を遂げたるに鑑み，我海軍に於ても相當早き時代より試驗的に採用せんとの議があつたが，容易に其實現を見ることが出來なかつた。然るに 1921 年（大正 10 年）に至り米國に特務艦神威が註文せられ，其主機械として G.E 社製造の電氣推進が採用せられた。同機械は 2 軸 8,000 馬力で各軸は 4,000 馬力の電動機を以て回轉せられ，電流は 1 臺の「ターボ」發電機によりて供給せられて居る。同艦は 1923 年（大正十二年）竣工して以來頗る良態に作動して居る。

（2） 罐

1863 年（文久 3 年）我國に於ける最初の建造艦千代田には 2 個の汽車罐が裝備せられた。其後 1899 年（明治 32 年）迄に進水せる軍艦には何れも汽車罐若くは圓罐を裝備し，水雷艇には「ノルマン」「ソーニークロフト」又は「ヤーロー」罐等が裝備せられて居る。1898 年（明治 31 年）に至り「ベルヴヰル」罐が採用せられた。本式罐は日露戰役當時の主力艦の大多數に搭載せられたものである。1901 年（明治 34 年）には橋立の圓罐を宮原式に換裝せられた。同罐は宮原機關中將の發明に係るもので，軍艦に本罐を採用したのはこれを以て嚆矢とする。橋立の宮原罐は日露戰役に一大試練を經て其優秀なるを認められ，同戰役後の新造軍艦の大多數に此罐が採用せられた。

1902 年（明治 35 年）八重山の圓罐は「ニクロース」罐を以て換裝せられた。又新高にも新たに採用せられ，次で日露戰役の末期英國にて建造せられた戰艦鹿島及香取にも裝備せられたが，「ベルヴヰル」罐と同じく其後の艦には採用せられたものがない。

1902 年（明治 35 年）驅逐艦春雨に「イ」號艦本式罐を裝備した。初めは海軍型と稱せられ，今日廣く海軍に於て採用せらるる「ロ」號艦本式罐の前身である。同罐は成績が良好なので當時建造の驅逐艦には一般に採用せられた。併し此罐は水「ドラム」が楕圓形であるため數年後に至りて故障を生ずる樣になつたので之れを圓形に改め，又罐管の配列法を改良して「ロ」號艦本式となつた。

宮原罐は戰艦扶桑，山城時代まで採用せられたが，漸次高き燃焼度が要求せらるるに至つては不適當なため，其後の建造艦には總て「ロ」號艦本式罐を採用せらるゝ樣になつた。同罐は 1914 年（大正 3 年）初めて驅逐艦樺級に採用せられ，其後幾多の改良が施されて現在各種の艦船に最も多く使用せられて居る。

石炭と重油の混燒を行つたのは 1906 年（明治 39 年）通報艦八重山に重油噴燃裝置を採用せられたのが初まりで，同年進水の巡洋艦筑波，生駒にも採用せられた。重油專燒罐は 1909 年（明治 42 年）起工の驅逐艦海風及山風に採用せられたのが初めである。其後は混燒罐と重油專燒罐とを混用し，或は重油專燒罐のみを裝備せる幾多の艦船が建造されたが，最近は全部重油專燒罐のみが使用せられて居る。

過熱器は相當以前より宮原罐及「イ」號艦本式罐にも使用せられたが，稍々高く過熱したのは一九一三年（大正 2 年）「ヤーロー」會社の推擧せるもので，同年英國に註文せられた驅逐艦浦風に之れを採用せられた。又 1916 年（大正 5 年）度の巡洋艦天龍及驅逐艦谷風には技術本部の計畫に係る新型過熱器が採用せられ，過熱度は何れも華氏 100 度である。次で戰艦長門及陸奥にも同型過熱器が採用せられた。

初期の「ロ」號艦本式罐に於ては給水は水「ドラム」內の一隅に設けられたる「ポケット」內に入り，外側 2 列の蒸發管を給水加熱に利用したことがある。併し罐管の腐蝕が早いので再び蒸氣「ドラム」內に導く樣改正せられた。又降路管は一般に取付られて居る。

罐の力量も漸次大きくなり。今日は 1 罐にて 12,000 馬力を超過するに至つたと同時に，罐の燃燒度及効率も高くなり，$\frac{1}{3}$ 燃燒度てに 84% を超え，全力にて約 80% に達するに至つた。

（3） 復 水 裝 置

往復式機械採用時代に於ける主復水器は其型式區々であつて，直立管內蒸氣式，橫

置管內蒸氣式及橫置管外蒸氣式等がある。其形狀も四角形のもの，楕圓形のもの又眞圓等があつて一様ではないが，冷却管は殆ど總て外徑 5/8 吋，厚さ 0.048 吋の眞鍮管を使用し，管の節は「ユニフォーム」であつた。1906 年（明治 39 年）起工の戰艦安藝に至つて初めて壺形の「ユニフラックス・プリンシプル」のものが採用せられたが，管の節は總て 3/32 吋である。次で 1909 年（明治42年）起工の戰艦河內に至り「ヴァリヤブル・ピッチ」の「ユニフラックス」復水器が採用せられた。

1916 年（大正 5 年）度の驅逐艦谷風及巡洋艦天龍よりは「オールギヤード・タルビン」を採用し，其裝備の關係上，低壓「タルビン」の下に懸垂したる半圓形のものを採用せられた。此裝置によると重量及容積を減ずる利益はあるが，軸の傾斜を大ならしむる不利を伴ふので，機械室配備の都合により橫置式或は懸垂式が共に使用せられて居る。

旋轉式抽氣喞筒が採用せらるるに至つた時代より，復水器の一部を仕切りこれを「ドライエヤー」の吸入管に連結する裝置が設けられて，復水器の效率が高められた。爾後建造の艦船には多く此式が採用せられて居る。

（4） 送水喞筒

喞筒は何れも遠心喞筒で，吸鍔式主機械を裝備せる時代にありては其力量は艦種によりて略一定して居た。即ち其當時は主送水喞筒たると同時に艦の大排水用の目的であつたのである。それであるから其力量も每時 1,600 噸を超ゆるものがなかつた。主機械として「タルビン」が採用せられて以來は喞筒の力量も著しく增大して一臺の力量が每時 8,500 噸に達するものがある。又喞筒の吐出水頭も漸次高くなつて種々の改良が施され，或る艦には「タルビン」を用ひ減速裝置を採用したものもある。又高速の航走には送水喞筒を使用せず，艦の速力に依る自然流入式を採用した艦船もある。

（5） 抽氣喞筒

往復式主機械を使用せる時代に於ける抽氣喞筒は主機械の滑頭より天秤によりて作働するもの，及曲肱軸に連結せるもの等が採用せられたが，又時としては獨立喞筒も使用せられた。併し曲肱によりて作働するものが多かつた。戰艦朝日時代よりは獨立せる且つ曲肱を有せざる型式が採用せらるるに至つた，即ち「ウエヤー」式，「ブレーキ」式，「ワトソン」式及「ウオーシンヶトン」式等の種類があるが，これ等は何れも復水と空氣とを混合して抽出するものである。1907 年（明治40年）起工の伊吹には復水抽出用としては「タルビン・ドリブン」の旋轉式喞筒を，又空氣抽出用として吸鍔式喞筒が採用せられた。斯の如く乾濕喞筒を別々に分けたのは，眞空喞筒の力量を增大して高き眞空を得んが爲であつた。軍艦安藝には初めて「ウエヤー・デウアル」喞筒を採用し，又 1913 年（大正 2 年）起工の潛水艦母艦駒橋には初めて「ルブラン」抽氣喞筒を採用して，何れも著しく良好なる成績を得たので，1915 年（大正 4 年）度建造の一等驅逐艦に各「ウエヤー・デウアル」式，「ルブラン」式及 カイネチック」式抽氣喞筒を裝備して比較實驗が行はれた。其成績によると，旋轉式のものは眞空度や重量容積では仲々優良であるが，低速力になると蒸氣消費量が多く又取扱上も困難だと言ふので，「ウエヤー」式が一般に歡迎せられた。隨つて其後建造の艦船には總て「ウエヤー」式が採用せられて居る。併し一層其效果を高むる爲 1920 年（大正 9 年）頃よりは蒸氣「エゼクター」を使用する特殊抽氣裝置が採用せられて好成績を得たの

で，今日は一般に此式が採用せらるるに至つた。

（6） 强壓注油裝置

巡洋艦筑波以前の往復式機關裝備の軍艦及驅逐艦は一般に油滴の落下による自然注油法が用ひられたが，1906 年（明治 39 年）巡洋艦筑波に至り初めて强壓注油法が採用せられた。此裝置は獨立の注油喞筒を有し曲肱坑より油を汲み上げて主機械の滑頭，曲肱軸，偏心器，「リンク」裝置及諸軸承等の主要部に注油し再び曲肱坑に還らしむるもので，其詳細は比較的幼稚のものであつたが，自然注油法に比して著しく好結果が得られ，長途の航海を終へた後でも磨面の耗りは殆んど認められない位となつたので，爾後建造の艦船には總て此種の注油裝置を採用し而かも簡單なる油冷却裝置を施したものもある。1907 年（明治 40 年）軍艦伊吹に初めて「タルビン」を裝備するに至つて注油法も一段の進步を來し，獨立の注油喞筒，油冷却喞筒及油溜「タンク」等を有する現在の注油法となつた。又「タルビン」裝置に於ける各軸承の遊隙量，油の入口及出口の位置及齒車減速裝置に於ける噴油器の構造等に關しては幾多の實驗を經て最良の方法が採用せられた。注油喞筒は總て「ウエヤー」式を採用し又油冷却喞筒は從來大艦には「ウエヤー」式を採用し，驅逐艦の如き小艦には主送水喞筒より支管を設けて海水を導き以て冷却用としたが，近來は排氣「タルビン」若くは蒸氣「タルビン」による旋轉式喞筒が採用せられて居る。

油吟却器は最初は横置式にて油は管の內部を循環し冷却水は管の外部を通過する型式を採用し，其管內には熱の傳導を良好ならしむる爲め「レターデー」を裝備した。又二重管板に二重管を取付けたる「ウエヤー」式を採用せるものもあるが，漸次良型が計畫せられ最近にては直立式となり，管の內部に海水を通じ外部に油を循環せしむる型式が採用せらるるに至つた。管の材料は復水器管と同樣なる眞鍮製とし，管板との取付法は一般に擴管器が使用せられて居る。

（7） 罐室强壓通風裝置

一般艦船の罐室は總て吸鍔式機械による兩面又は片面の扇車により强壓通風を行ふを例とした。圓罐時代には時に「ハウデン」式の强壓通風裝置のものもあるが，水管式罐が裝備せられた時代に至つて多くは焚火室密閉裝置となつた。

「ターボ・フワン」は 1913 年（大正 2 年）「ヤーロー」會社にて建造の驅逐艦浦風に始めて採用せられた。其型式は「ヤーロー・テリー」式で其構造は極めて簡單且つ强固であつた爲め大に好評を博したが，其後一層蒸氣の經濟なる且つ重量容積共に小なる艦本式「タルビン」が採用せられ，1915 年（大正 4 年）起工の驅逐艦桃級に裝備せられた。本「タルビン」は 2 段落よりなる輻流「インパルス」式で其成績が頗る良好であつたので，最近に至るまで各種の軍艦，驅逐艦に博く採用せられた。扇車は「ヤーロー・テリー」式に對しては，「キース」型が採用せられたが，其後のものは新考案なる艦本式扇車が採用せられ良好なる成績を擧げて居る。1924 年（大正 13 年）以後の驅逐艦に至り，「マーグ」式齒車減速裝置を有するもの，又は「アレン」社減速裝置附の「ターボ」送風機械が採用せられて良好なる成績を擧げつつある。

（8） 給水裝置

一般に主抽氣喞筒の吐出側には自然濾過式又は强壓濾過式の給水濾を裝備し，補助抽氣喞筒の吐出側には別に補助給水濾を裝備するを例とする。給水加熱器は蒸氣及排

氣を使用する觸面加熱器が採用せられ，給水喞筒の吐出側に裝備せられたが，1911年(明治 44 年) 度の戰艦扶桑に初めて混合式加熱器を補助に用ひ，給水「タンク」に入れる前に加熱するものが採用せられた。此混合式加熱器は重量容積小なると，且つ效率が良好なるを以て，爾後建造の驅逐艦及輕巡洋艦に多く採用せられたるも，高溫度に加熱するときは機械室に裝備する關係上室內溫度を高むる不利があるのと，又給水喞筒の作働を不確實ならしむる惧れあるが爲め，加熱溫度に制限を受くる缺點がある。且つ低速力に於ては充分加熱し得ざる等の缺點があるので，最近に至り再び觸面加熱器が採用せらるるに至つた。

給水喞筒は最初は主として「ウォシンットン」式が採用せられたが 1900 年頃よりは一般に「ウェヤー」式直働喞筒が採用せられて來た。1918 年 (大正 7 年)起工の驅逐艦矢風には「ターボ」給水喞筒を採用した。本喞筒は重量容積に於ては頗る有利であるが，低速力の場合に於ては蒸氣消費量が多い缺點があつた。然し漸次改良が加えられ，又優良なる型式のものを採用し，近來は軍艦驅逐艦共主として旋轉式喞筒が裝備せらるることとなつた。

自働給水加減裝置としては「ソーニクロフト」式，「ベルヴォル」式等が各裝備罐に附隨して使用せられたが，共後は「マンフォード」式自働給水加減器が艦本罐に取付けられ現在に及んで居る。

(9) 重油噴燃裝置

1906 年初めて重油が燃料として我海軍に採用せられた時代より總て喞筒によつて噴射せしむる型式を採用して居る。其當時の重油噴燃喞筒は「ブラッドフォード」式が使用せられたが，漸次「ウエヤー」喞筒が採用せられた。

重油加熱器は其初期時代には普通の復水器に似たる形狀で，眞直なる管を管板に取付けたものであるが，其後種々なる改良型が案出された。加熱器は何れも喞筒の吐出側に裝備せられ，又喞筒の吸入側及吐出側には油濾器が取付けられてある。

重油噴燃器は驅逐艦浦風に於ける「ヤーロー」式及特務艦神威に於ける「フキッシヤー」式を除きては總て海軍式を採用して居る。 而して噴燃器 1 個の力量は漸次增大して來た。

「エヤーコーン」は噴燃器の各力量に適應する海軍式を採用して居る。

(10) 造水裝置

蒸化器の型式は種々であるが，古くより主として「ケヤード・レーナー」式及「ウエヤー」式が採用せられた，併し1904 年頃よりは「ウエヤー」式が重に使用せられて居る。其力量も以前には比較的小さかつたので，日露戰役に於ては力量の不足なることが痛感せられ，力量が急に增大せられた。又細管式の艦本罐が採用せられてよりは補給水は蒸溜水のみを要する樣になり，一層力量の增大を要求する樣になつた。即ち日露戰役當時の主力艦たる三笠にては總力量は 66 噸に過ぎないが，金剛級にては 400 噸となり，最近の戰艦にては一層增大せられた。

蒸化器には高壓式及低壓式がある。高壓式は生蒸氣を使用し，低壓式は排氣を使用するのが普通であるが，蒸氣排汽何れでも使用し得るのである。高壓式は主として驅逐艦に，低壓式は重に大艦に使用せられて居る。

蒸溜器は以前は常に蒸化器より容量の小なるものが使用せられたが，軍艦扶桑時代

よりは常に同一容量のものが採用せられて居る。

(11) 舵取裝置

從來各種の型式が採用せられたが近代の軍艦は總て「ネピヤー」式「デフェレンシァル・スクリュー」裝置を用ひ，驅逐艦には「クォードランド」式が採用せられて居る、主力艦には直立 2 筩吸鍔式機械を兩舷機械室に各 1 臺宛備へて主用とし，別に補助舵取裝置として，後部揚錨機の軸によりて作動せしむる方法をも裝備するを常とす。又輕巡洋艦，驅逐艦には直立又は倒置吸鍔式機械を機械室に 1 臺裝備せるものが多い。電働油壓式の舵取裝置は 1913 年驅逐艦浦風に初めて採用せられた。其制式は「ヘルショー」式で，其後特務艦に「ジョンネー」式及「ヘルショー」式が採用せられ，吸鍔式機械に比し優秀なる成績を擧げた。又 1923 年（大正 12年)以降の巡洋艦にも此兩式が裝備せられて居る。一般に 2 臺の電働喞筒を裝備し外に豫備裝置として吸鍔式機械の喞筒を裝備するを常とする。

(12) 揚錨裝置

揚錨機械は一般に直立吸鍔式機械が採用せられて居る。揚錨裝置に於て起る主なる故障は推力軸承及螺齒車の擦熱である。前者は急激なる荷重の變化と，潤滑の不充分なるに起因する場合多く，後者は垂直軸の下降により自然齒車の嚙合を不良ならしむるによるものが多いので，近來は此等の部分に球入軸承を用ひて居るが其成績は頗ぶる良好である。

(13) 推進器

1910 年頃までの大艦の推進器は轂と翼とは別體に製造せられたが,其後のものは總て一體に製造せられて居る。但し驅逐艦以下の小艦には以前より總て一體に製造せられて來た 。又河川用砲艦には「タルビン」推進器を採用したものもある。

推進器の材質には從來 Manganese Bronze, Nickel Bronze, N.M. Bronze, Monel Metal, Stone Bronze, Turbadium Bronze 等が使用せられたが，此內最も多く使用せられつゝあるものは Manganese Bronze で，之に次ぐは Nickel Bronze である。此 2 つのものは「エロージョン」に對しても相當に强く，鑄造も容易で，且つ鑄掛も自由に出來る利益がある、翼の形狀は吸鍔式機械にありては多くは楕圓形であつたが,「タルビン」機械となるに及んでは何れも「ワイドチップ」のものが使用せられて居る。

(14) 現在に於ける 3 問題

第 1 は航空母艦である。1919 年（大正 8 年）建造の航空母艦鳳翔は我國に於て航空母艦として建造せられた最初のものである。主機械及罐は何等の特長として述ぶる點はないが，本艦には米國「スペリー」會社製造の「スタビライザー」が裝備せられて頗る好評を博した。又航空母艦として最も困難なる問題は煙突であらう。これは今日各國が同樣に苦心を拂ひつゝあることゝ信ずる。航空母艦として其着甲板は「フラッシュ」なることが飛行家にとりては最も希望せらるゝ所であるが，煙突の位置及形狀(勿論艦橋も)と相關聯して考ふるときは困難なる問題に逢着するのである。其後華府會議の結果赤城及加賀の兩艦が航空母艦に改造せらたが，我海軍としては常に同一見解の下に種々多大の犧牲を拂つて，此煙突問題を解決したのである。併し吾人は內火機關が一層發達して高速航空母艦に裝備し得る日の早からんことを希望するものである。

第 2 には 1 萬噸巡洋艦の機關である。これも各國海軍が其計畫に苦心を拂ひつゝある問題であつて，其國情により計畫の根本に相違する點があることゝ信ずる。隨て我海軍も亦我國情に適應し得る樣最善の努力の下に計畫し得られたことゝ信ずるのである。

第 3 に目下最も考慮を拂はれつゝあるは，海軍休日明けの後 1931 年以降に建造に着手せらるべき主力艦の代艦であらう。10 年間の海軍休日は世界海軍が最も智腦を搾りつゝある時代であつて，許されたる最大排水量 35,000 噸の內幾何の重量と容積とが機關に割き得らるかは，各國の國情によりて異る所があらうとは言へ，其重量を最も有效に使用することが，當事者の技倆比べの點であらう。吾人は數年後の完成時を興味を以て見んと欲するものである。

以上は我海軍に於ける蒸氣機關の各項目につきて其發達の大要を述べたものである。今之を綜合して機關全體が如何に變遷し改良せられたかを考ふるに，一目して其狀況を知り得ることは重量と容積である。即ち機關の 1 單位重量によりて發生し得る力量及機關室の單位面積により發生し得る力量である。之に關し最も大なる影響を及ぼしたものは蒸氣「タルビン」の發達，齒車減速裝置の出現及重油專燒罐の採用である。今日驅逐艦の如き高速艦にありては機關の重量の 1 噸に付き發生する力量は優に 70 馬力に達し，機關室面積の每平方呎につき 11 馬力を發生して居る 1917 年（大正 6 年）起工の球磨級巡洋艦は 5,500 噸の排水量なるに係はらず相當大馬力の主機械を搭載して居ることは，我海軍の特色とする所である。

次に主機械效率の增進及補助機械の改良によりて蒸氣消費量を減じ，重油專燒罐の採用によりて效率を增し，斯くして兩々相俟て機關全體の進步を遂げ得たのである。即ち往復式機械使用の時代にありては最も進步したるものと雖も補助蒸氣量を含みて每時每馬力につき 18 听を普通となしたるも，今日に於ては每時每軸馬力につき 11 听を以て足る狀態になつた。近來「デイゼル」機械の發達に連れ其刺激を受けて，蒸氣機關も高壓高溫蒸氣の使用となり，又罐裝置の改良を促しつゝあるので，軍用蒸氣機關の發達は停止する所なく，層一層微に入り細に亘つて進步を續けることゝ信ずるのである。

第二章　內火機關

（1）　潛水艦用主機械

我海軍に於ける內火機關は主として潛水艦用主機械として發達を遂げたものである。我國最初の潛水艦は日露戰役當時 1904 年（明治 37 年）米國 Electric Boat Co. より購入せる排水量 106 噸のもので，本艦には同社製造の直立 4 筩 180 馬力の 4 衝式「ガソリン」機械 1 臺を裝備せられた。是れ我海軍に於ける舶用內火機械の嚆矢である

同年起工の舊第 6 及舊第 7 潛水艦は我國に於て計畫せられたもので，米國 New Jercy Standard Motor Co. 製造の直立 6 筩 300 馬力の 4 衝式「ガソリン」機械 1 臺を裝備せられた。1907 年（明治 40 年）には橫置 16 筩 600 馬力の Vickers 社 4 衝式「ガソリン」機械 1 臺を有する波號第 1 及第 2 潛水艦が起工せられた。本機械の 12 筩のものは橫須賀海軍工廠に於て製造せられた。これ我國に於ける潛水艦

主機械製造の最初のものである。

1910 年（明治 43 年）には米國 New Jercy Standard Motor Co. より直立 6 筩 1,000 馬力の 4 衝式の「ガソリン」機械を購入し，川崎造船所にて建造の波號第六潜水艦に装備せられた。

其後 1917 年（大正 6 年）に至り初めて「デイゼル」機械を装備する 450 噸速力 17 節の潜水艦が竣工した。本艦は佛國 Schneider 社にて建造せられ，主機械は直立 8 筩 2 衝式 1,000 馬力のもので，同機械が 2 臺装備せられた 本機械は燃料として石油を使用したもので 1918 年横須賀工廠で 1 隻分製造せられた。

1917 年（大正 6 年）より 1920 年（大正 9 年）までの[illegible]に起工せられた呂號第 1 第 2, 第 3, 第 4 及第 5 潜水艦には伊國 Fiat 社の直立 6 筩 1,000 馬力の「デイゼル」機械が 2 基装備せられた。本機械は燃料として初めて重油が使用せられたもので Fiat 社及川崎造船所に於て製造せられ，其後種々の改良を施されて，先づ實用上遺憾なき迄に完成せられたが，今日の機械とは到底比較にならぬものであつた。

其後我海軍に採用せられた機械は Sulzer 社 2 衝式單働空氣噴射式のものである。同機械は我國に於て最も多數に製造せられ，幾多の實驗研究を經て改良が加へられ極めて滿足すべき程度に發達した。前記の Schneider, Fiat, 及 Sulzer 式とも何れも 2 衝式であつて，主として此の方針にて進まれたが，4 衝式のものも亦相前後して採用せられた。即ち Vickers 社の 4 衝式單働無空氣式及 Rauschenbach 社の 4 衝式單働空氣噴射式等である。而して Sulzer 式のものは海軍及神戸製鋼所に於て製造し，Vickers 式は三菱造船所に於て，又 Rauschenbach 式のものは重に川崎造船所に於て製造せられて居る。

潜水艦の型式が漸次増大され速力も大なるものを必要とせらるゝに至つたので，機關の發生力量も亦漸次増大して來た。前述の 3 種の「デイゼル」機關は各其の装備されて居る船體の型式と共に特徴を有し，必要に應じて種々なる改良が施され，今日に於ては頗る優秀なる機關にまで發達を遂げた。

機關の力量が増大するに從ひ，軸の捩振動に對する問題も漸次重大視せらるゝに至つた。本問題につきては理論上及實驗上に種々の研究が行はれ，機械に對して諸種の改良が施された。

（2） 潜水艦装備以外の內火機關

我海軍に於て潜水艦以外の艦船に舶用「デイゼル」機械が採用せられたのは，1912 年（大正元年）第三横須賀丸に装備せられた Sulzer 社製 2 衝式單働 300 馬力の空氣噴射式機械を以て嚆矢とする。次で翌 1913 年驅逐艦浦風の巡航用機械として Burmeister and wain 社製の 4 衝式單働 500 馬力空氣噴射式のもの 2 臺を採用した。本機械は Föttinger 式水壓式傳導装置によりて主「タルビン」軸に連結せられたものであるが。其後之れを取外して「デイゼル」機械のみは特務艦劍崎の主機械として装備せられた。其後 Sulzer 社の 2 衝式單働無空氣 600 馬力の「デイゼル」機械 2 臺を交通船に装備して諸種の實驗が行はれた。其他には池貝式及新潟式の 4 衝式單働の 150 馬力乃至 250 馬力の機械を装備する機動艇が漸次其の數を増した。

「ガソリン」機關を装備する艦載機動艇は 1912 年頃より漸次多數に建造されたが，近來は「ガソリン」に代ふるに石油を使用するものに改められた。

C.M.B. として知らるる高速内火艇は世界大戰の末期頃より漸次製造せらるゝに至り，此等の機動艇には「ガソリン」機關が採用せられ Thornycroft 式 Meibach 式等が採用せられて居る。

。一般艦船用の發電機に内火機關を用ゆることは 1912 年頃より初まつた。其の初期には 15 乃至 25 K.W. の「セミデイゼル」機械が採用せられたが,「デイゼル」機械の發達に從ひ最近に於ては各種艦船に必要に應じて 300 K.W. 位までの「デイゼル」發電機が裝備せられて居る。

海軍に於ける内火機械の發達は大略上記の如き有樣で，一般舶用「デイゼル」機械の發達と相俟て，唯に潛水艦用に止まらず大型特務艦用としても益々其の使用範圍が增大せられつつある。

上述せる所により帝國海軍舶用機關發達の大要を窺知することが出來ると思ふ。今や我海軍は世界 3 大海軍國の 1 となり，蒸氣機關に於ても内火機關に於ても列强諸海軍と比肩し得るに至つたと信ずるのである。之は主として歷代の當局者が連綿不斷の研究と努力の結果であるとは言へ，又一面に於ては今日に至る迄永年の間先進國より受けたる懇篤なる指導，並に其の駸々として止まざる發達の刺激に負ふ所極めて大なるものあるを痛感するものである。之に對し玆に深甚の謝意を表するものである。(終)

Descriptions of some Life-Saving Devices for Crews of Submarine Boats in the Imperial Japanese Navy.

(Paper No. 769)

*By **R. Hodzumi,** Constructor Captain, I. J. N.*

Abstract.

By accidents of submarine boats, the navies of the world have lost in time of peace, hundreds of picked officers and sailors. In almost all cases, rescues were not in time to save the crew entrapped alive in the deep sea.

In successive disasters of our submarine boats Nos. 70 (1922) and 43 (1923), in spite of every possible effort of divers, it was found impossible to save the entrapped crew from a depth of about 30 fathoms.

Through those bitter experiences, we learned that salvage devices prepared before that period had been far from being perfect to meet the aim of life-saving.

To improve these defects, naval constructors in co-operation with some experts in salvage work, investigated the following items:—

I. Deep-sea diving.
II. Locating the sunken boat.
III. Communication with the boat.
IV. Preservation of the lives of the crew.
V. Escape of the crew.
VI. Hauling up the boat.

Those problems being now nearly solved, the Naval Department authorized me to disclase the results of investigations on these items, to the members of the World Engineering Congress, for the welfare of crew in submarine service of all navies.

I. Improvement on the Deep Sea Diving Method.

Deep sea diving of common practice, with air supply from hand pump, is very inconvenient and sometimes rather dangerous. Below a depth of 20 fathoms usually two sets of pumps are connected tandem to keep the high pressure of supply air; but even with the utmost effort of several groups of pumpers working alternately, the air supply is not sufficient. Air heated by compression and fouled by the smell of lubricating oil is very disagreeable to divers.

Under such bad conditions, divers get rapidly tired, and it is impossible for them to stay more than 15 minutes at a depth of 30 fathoms.

When the sea is not calm, the pump barge will naturally be rolled more or less, and it is hopeless to keep the necessary pressure for the diver. In consequence a steady platform is required in order to work the pump efficiently.

After we experienced such troubles in case of our two accidents, we decided to adopt the "Compressed air system" for deep sea diving. In this system air pumps are replaced by high pressure air chambers as shown in Figs. I and II.

The merits of this new system are remarkable. Divers are supplied with an ample quantity of fresh air, cooled automatically by expansion at a reducing valve. Only one man is required at the reducing valve board in stead of many pumpers, and the rolling of the barge in no way affect the efficiency of the system.

With this novel method designed by Constructor Commander T. Hata, I.J.N. our experienced diver, after some period of training, achieved to dive 50 fathoms in an ordinary diving dress, and could stay there about 10 minutes.

The method of "Scientific Diving", or the rate of stepped ascending from the bottom of the sea was carefully studied, and the standard time of stop at every stage was fixed as shown in Table I.

All precautions for the safety of divers are taken, and "Decompression Chambers" are always in readiness to set on the salvage ships.

Two sets of "Deep sea diving apparatus" of the Neufeldt und Kuhnke system were recently purchased from Germany. In case of accident, officers in charge of salvage work are expected to dive and to inspect the damages, so that the proper instructions may be given to lead the diver's work.

II. Locating the Sunken Boat.

The first step of submarine boat salvage work is to locate quickly the spot of accident.

In our submarine boats, following arrangements are prepared for that purpose.

(*a*) *Indicator buoys.*

Two indicator buoys are stored in the superstructure, one is set free from the fore end compartment, and the other from the aft end compartment of the boat. Each buoy is equipped with a telephone connection and a pilot light, and also a card for writing messages is attached on the top of the buoy with the name of the boat.

(*b*) *Fuel oil discharge pipes.*

Several discharge pipes of small diameter, with stop valves inside the boat are arranged from a fuel oil tank to the deck. In the case of accident, a small quantity of fuel oil discharged at regular intervals, will make distinct spots of oil film on the sea-surface; and those spots carried by current will help to trace the position of the boat as a guide. This method proves to be the best for aeroplane scouting.

(c) *Message discharge tube.*

Several tubes about 6 inches in diameter, with a similar construction to torpedo tubes, are fitted vertically in the superstructure or on the top of the conning tower as shown in Fig. III.

Many small metal cases, strong enough to resist the deep sea pressure, will be discharged from the tube by compressed air with messages enclosed in them. Those cases floated to the surface are distributed in some area so that they may probably catch a lucky chance of being found by passing steamers.

When sound of propeller is detected, a Holmes light tube will be shot to indicate the position of the boat.

In one occasion a carrier pigeon was tried in a special tube so arranged as to float after a while, then the cover will be automatically opened and simultaneously a piston pushes the pigeon out of the tube. The idea was tentatively realized but was found very difficult to improve the detail design, as the pigeon did not start to fly freely after it had been enclosed several minutes in a narrow tube. (Fig. IV.)

III. Communication with the Boat.

Besides by the ordinary means of submarine signal and telephone buoy, above mentioned, a "Message discharge tube" is found to be a unique promising method of communication when those delicate apparatus are out of order.

The upper end cover of the tube can be opened from the outside of the boat by a diver. A pressure-tight tube used as an envelope and letters may be exchanged between the submarine boat and the salvage ship. (Fig. III.)

IV. Preservation of the Lives of Crew.

A well-known method of air and liquid food supply through flexible pipes, connected by divers, is also adopted in our submarine boats, however a special care is taken in arranging the pipe connections and valve handles.

They are gathered on the walls of the bridge structure under the red and green lamps, so that divers may easily locate the position in the dark bottom of the sea. Another consideration is payed to select the position of those gears to allow the diver's work keeping their helmets upright. In a deep sea it is very difficult for the divers to bend over on deck, i.e., to work with their helmets inclined.

Here the message discharge tubes are again utilized to pass some food and medicine, or small tools and materials required by the enclosed crew.

Every boat is provided with an arrangement of regenerating fresh air, and when the ventilation system is out of order, it is so devised that each crew could breathe through a soda box, applying a special mouth piece to it.

V. Escape of Crew.

If there is no hope of quick lifting a sunken submarine, the only method of life-saving is the measure to let the crew escape out of the boat.

Escape trunk.

In the modern submarine of our navy, every hatch makes an escape trunk; especially the fore and aft hatches are of the form of dome, and large enough to allow four men stand inside. Each escape trunk has, at the top and bottom, water-tight covers, and forms a lock.

To escape out of the boat, the crew put on special life-jackets and place them inside the locks. Keeping top and bottom covers tightly closed, water is slowly let in through a cock, then the air in the lock will be gradually compressed until the pressures inside and outside are balanced. Then the top cover is slowly opened and the crew get out of the boat.

The top cover is then closed by divers and the water inside of the lock is blown out by compressed air, the next group repeat the same method of escape.

This method is easy to explain but very hard to perform, and a good deal of practice in a training tank is necessary for the submarine crew.

Escape suit.

1. Life-jacket of "Dräger" type.

In our navy several jackets of this type were bought from Germany. The result of experiments were excellent, but even in the improved type, the size and weight are too much to store the jackets for all hands of a boat.

2. The "Lungs."

Some of this type were quite recently bought from the United States of America; and we are now preparing to try them. Simple construction and small weight are the merits of this type, and if the result of the trial is so good as it is published, it will be applicable for the practical use in the submarine boat.

3. Japanese "Mask."

The fact that simple masks of Japanese civil divers are very efficient for deep sea diving, has been testified by the salvage works of Mr. Y. Kataoka, in the Mediterranean. He accomplished to lift many cases containing gold ingots from the hold of the Yasakamaru, a passenger boat sunken by a certain German submarine boat.

Adopting the principle of those masks, an assistant engineer in the Saseho Dock Yord designed an escape mask and the result of its trial was very satisfactory.

Diving bell.

The escape trunk, combined with an escape jacket or a mask, will probably make the individual escape possible; but it is found impossible that, all crew who are almost exhausted under trying conditions in an unfortunate submarine, may rise safely to the surface.

To take in those crew just after they have got out of the escape trunks, I have proposed a diving bell as shown in Fig. V. The bell is of light construction, and may be lowered from the deck to the sea-surface by means of an ordinary derrick of ship. Then the bell is ballasted just to reserve a little buoyancy.

After the air pipes and electric wires are connected, the bell will be pulled down by two wire ropes towards the escape trunk of the sunken boat. The wire ropes, passing the rollers on deck of submarine boat, will be wound by deck winches on the salvage ship.

Air is constantly supplied from compressed air bottles, and adjusted to keep the inside pressure a little higher than the corresponding water pressure. When some of crew from the submarine are received in the bell, it will be hauled up slowly to the surface.

It was often proposed to lower the pressure of the resisting bell and fix the lower end flange on the hatch submarine; but water-tight work with many bolts and nuts in the deep sea is not an easy matter, and it will take long time to attach and detach the connection. Also the pressure-tight bell will be too heavy to handle with an ordinary cargo derrick.

VI. Hauling up of the Boat.

A device was proposed by Constructor Admiral J. Fukui, I.J.N., expert of salvage work in our navy, to haul up a sunken submarine boat quickly above water. The principle of this novel idea is to use an ex-service submarine boat as a counter weight of hauling to prevent the heeling of the salvage ship and to economize the power of lifting gear. Further, this device will easily be applied to any ship of ordinary size by a simple alteration with a small expense.

The proposal was at once taken up by the Naval Department, and an order was given to the Dock Yard of Yokosuka to try this device on the Asahi with a temporary arrangement. The Asahi is an old battle ship of 15,000 Tons, disarmed in compliance with the treaty of Washington Conference.

General design and details were studied by the constructors of the Dock Yard and the trial was completed with all expected results of the system.

The Asahi was then removed to the Dock Yard of Kure, where she was equipped as a salvage ship of submarine boat, with all improvements found necessary in the preliminary trial.

In June 1929, an official trial was conducted at a spot near Miyajima and on that occasion the Naval Attaches of all nations were invited to inspect the experiment. Every detail of design and work was openly explained to them.

The result of the trial was quite satisfactory, and following items are the principal records.

1. Weight of sunken submarine....500 Tons in water.
2. Depth of water....30 fathoms at high tide.
3. Time required
    a. to moor the Asahi in position ....2 hours.
    b. to bring allongside and to secure the counter-weight boat ....1 hour.
    c. to complete the wire connection with the sunken submarine ....1.5 hour.
    d. to adjust the weight of the counter-weight boat......1 hour.
    e. to haul up the sunken boat ....1 hour.
    Total ....6.5 hours.

Space is limited here to describe all the details of process, however, Figs. VI ~ VIII will give the general idea about the arrangements and mechanisms of the whole system.

Conclusion.

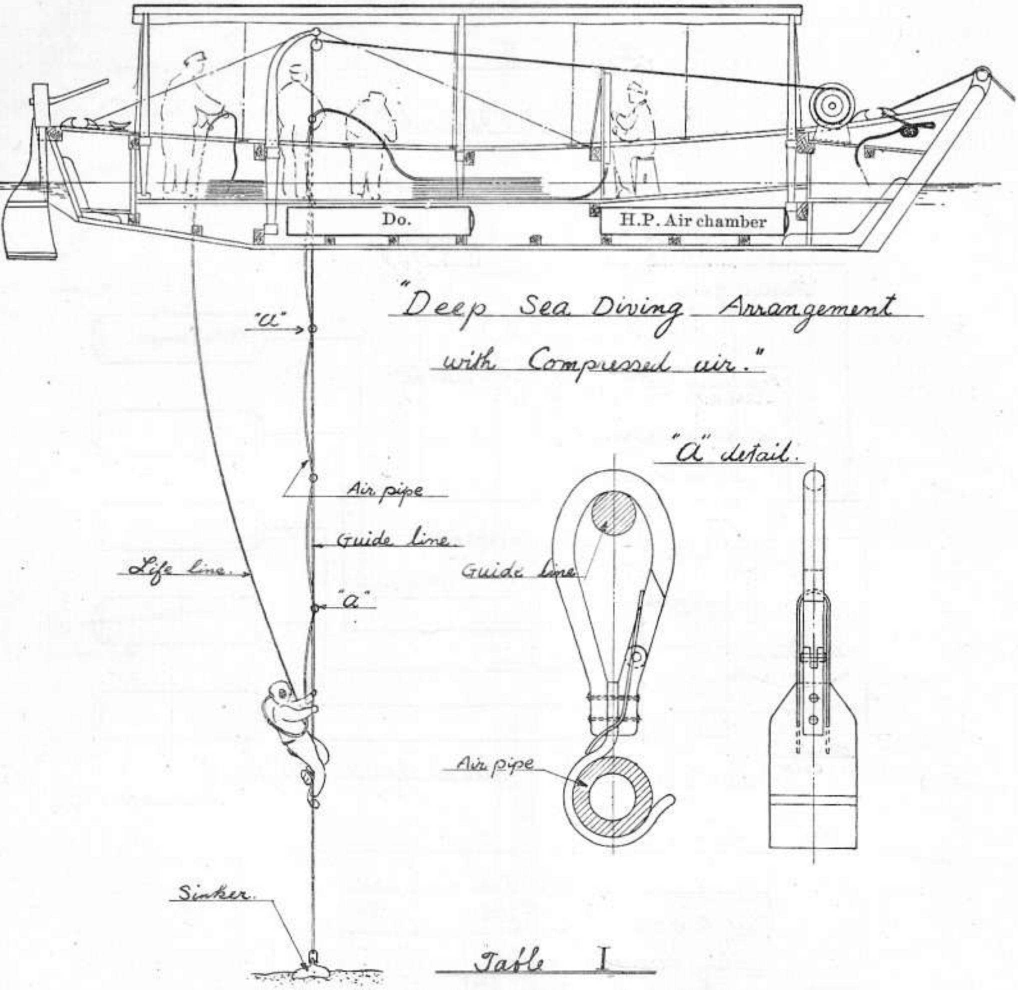
It is impossible to find "a master key" to the difficult problem of "Life-saving of submarine crew in the deep sea." We have to contribe suitable devices to meet different conditions.

The devices described in this paper may not surely be sufficient to meet all kinds of mishaps, yet we must always bear in mind that safety devices must not greatly influence the efficiency of active service; as all officers and sailors in submarine service persistently object to add anything that may sacrifice the fighting value of their boat.

Taking this opportunity, I ask the congress which composed of the leading members of all branches of the engineering profession to co-operate with us for the improvement of this subject in the cause of humanity.

(Paper No. 769)

Fig. I.



"Deep Sea Diving Arrangement with Compressed air."

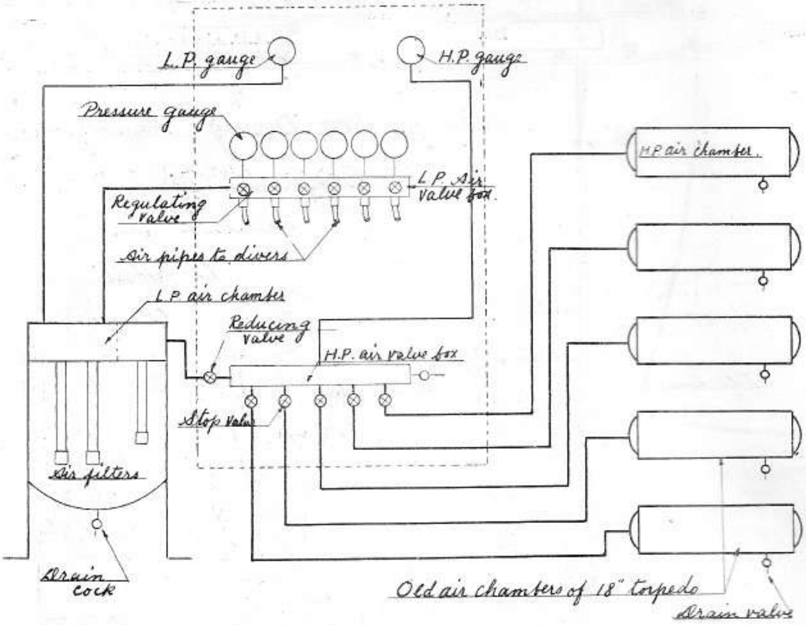
Table I

Standard Time of Stops
In Stepped Ascending from Deep Sea Diving.

Depth	Time Stayed at Bottom	Time to Stop									Total Time of Stop
		From Surface 108 feet	" 96"	" 84"	" 72"	" 60"	" 48"	" 36"	" 24"	" 12"	
Fatham	Second	"	"	"	"	"	"	"	"	"	"
20	20								4	6	10
25	20							3	4	5	12
30	15						2	3	4	7	16
35	10					2	3	4	5	8	22
40	10				2	2	3	5	7	10	29
45	10			2	3	4	5	6	7	10	37
50	10		2	3	4	5	6	7	8	10	45

Fig. II

Reducing Valve Board.



	High pres.	Low pres.
Test pressure	2800 lbs/in²	300 lbs/in²
Working Pressure.	1800 "	200 "
Volume	510 litre	150 litre.

Fig. IV

"Carrier pigeon case"

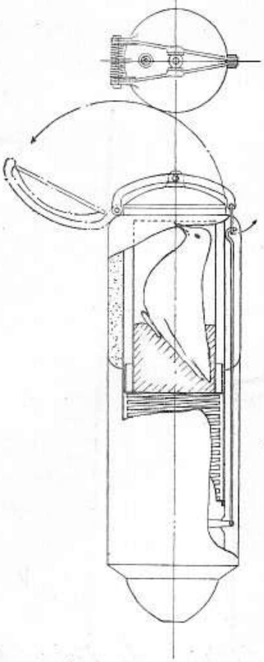


Fig. III

"Arrangement of The Message Discharge tube."

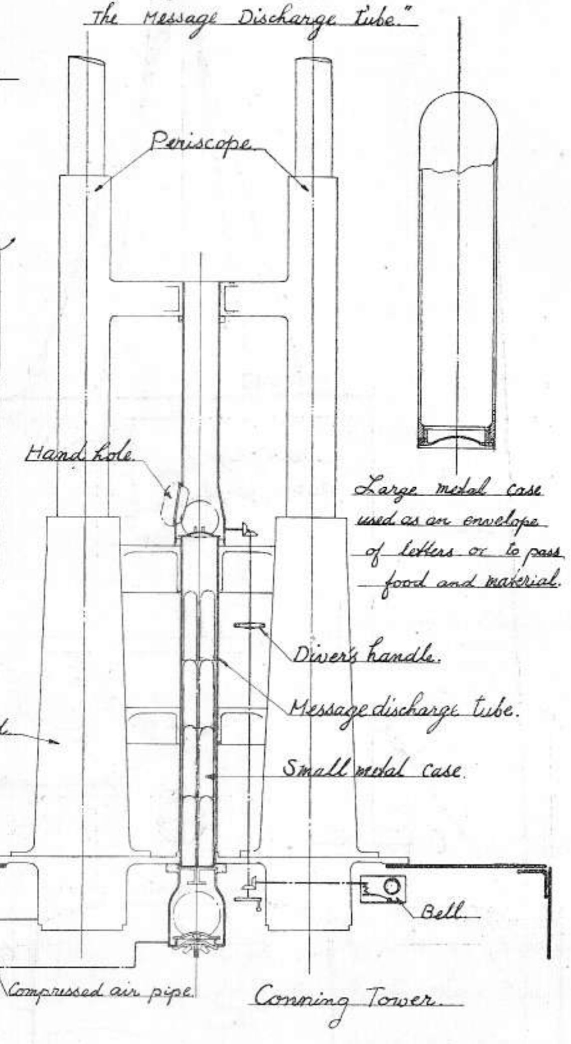


Fig V

"Diving Bell"

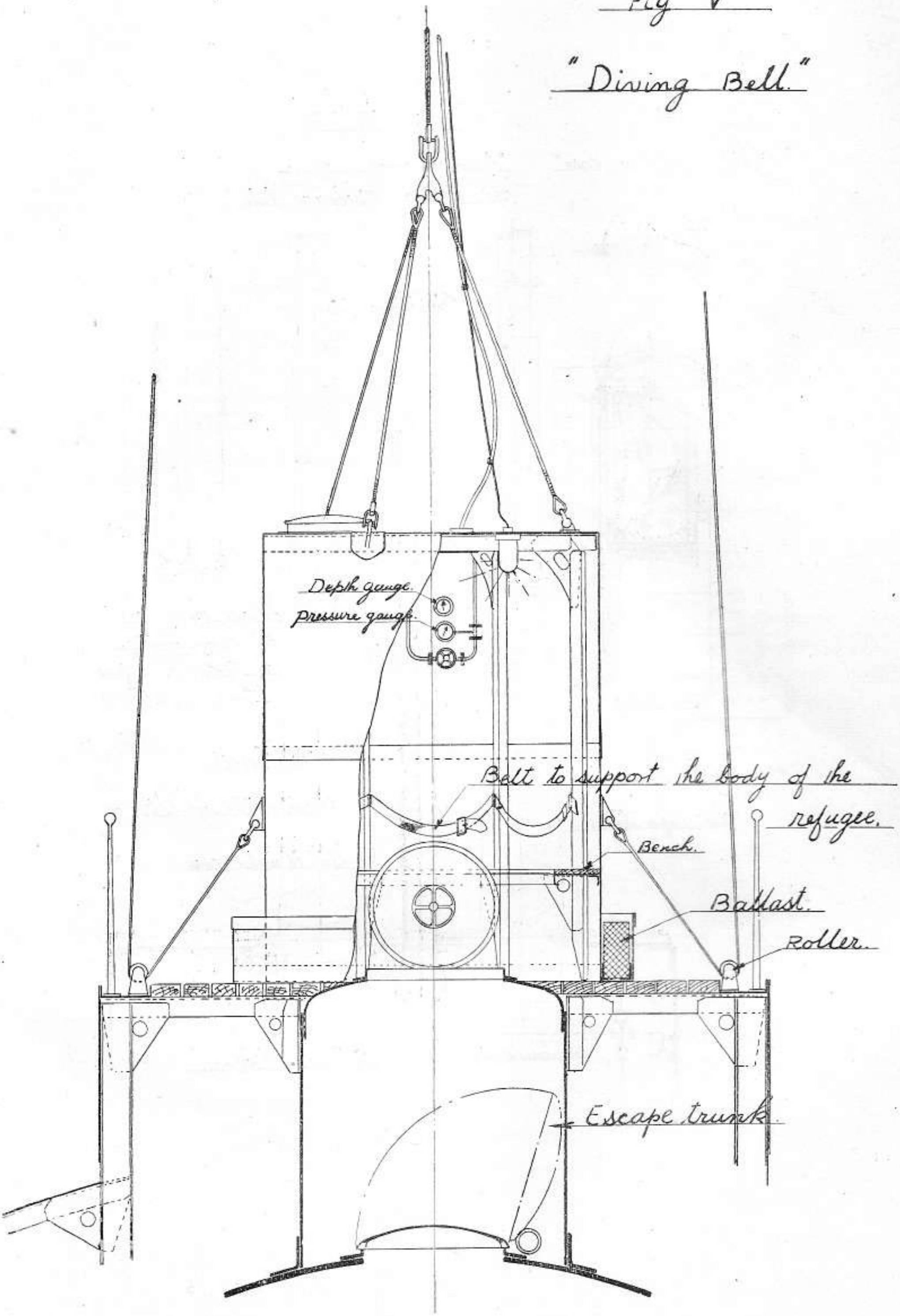
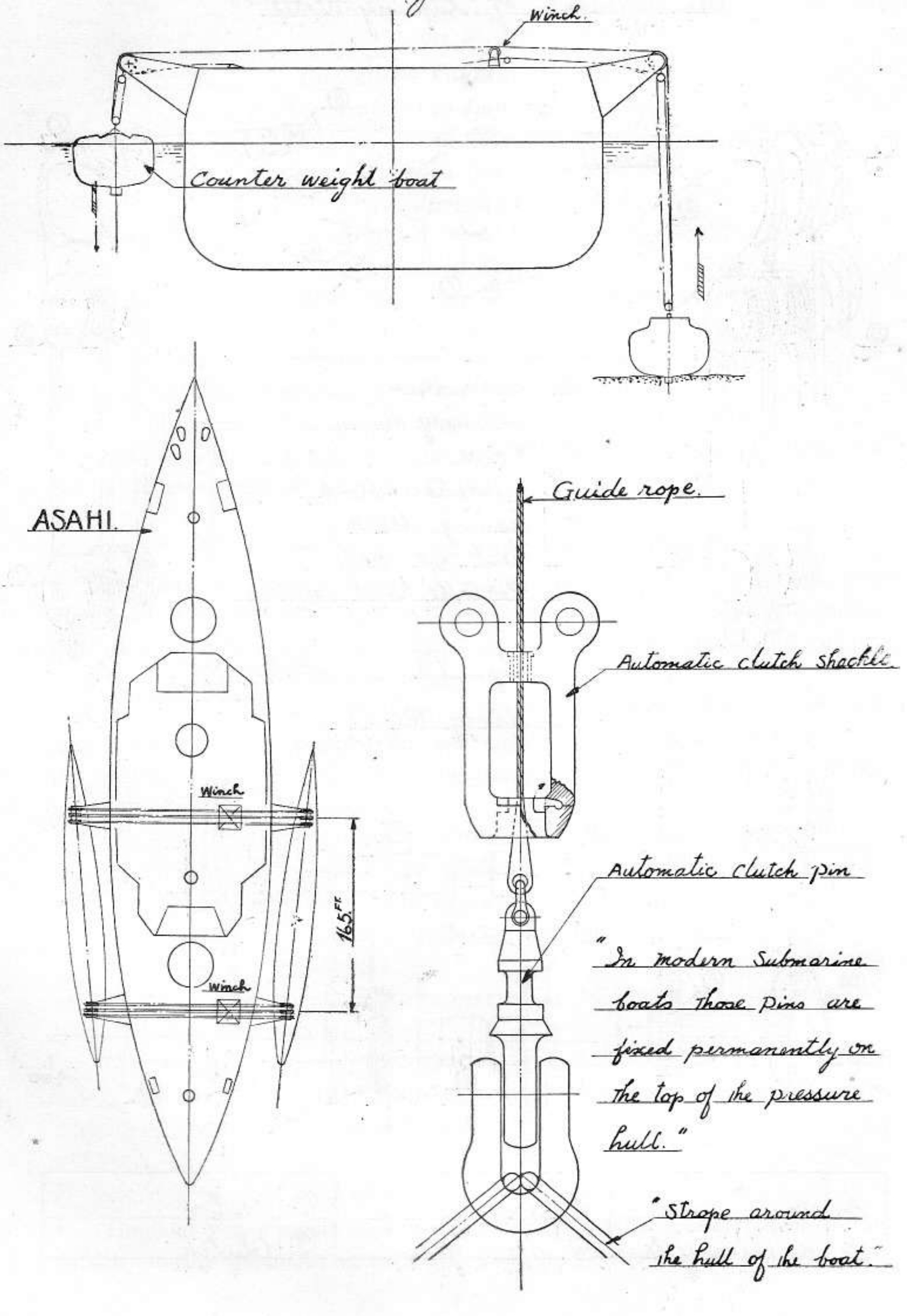


Fig VI

General Arrangement

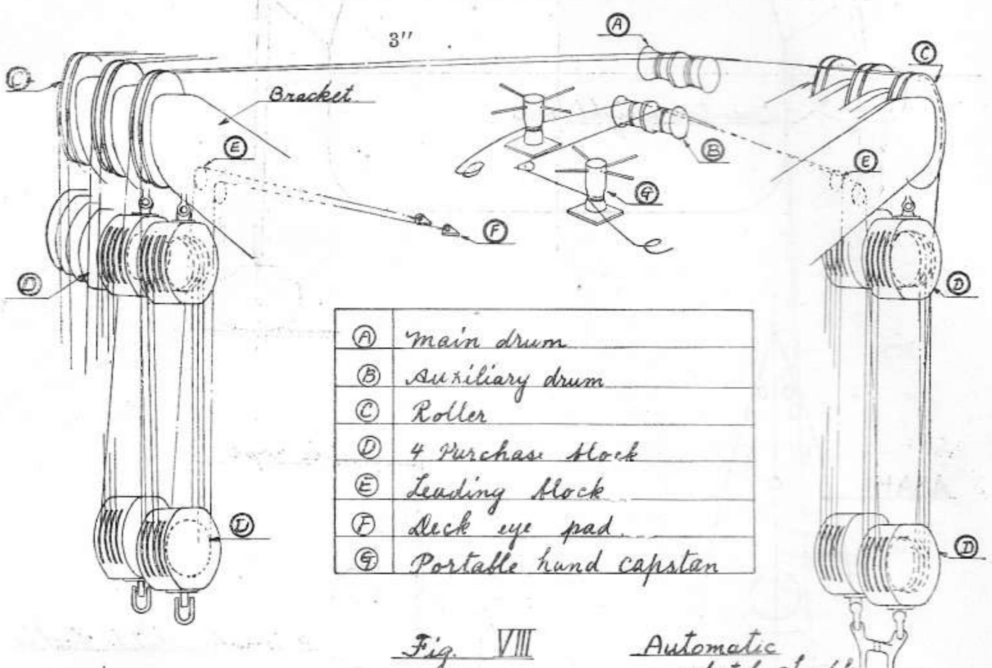


"In modern Submarine boats those pins are fixed permanently on the top of the pressure hull."

"Strope around the hull of the boat."

Fig. VII

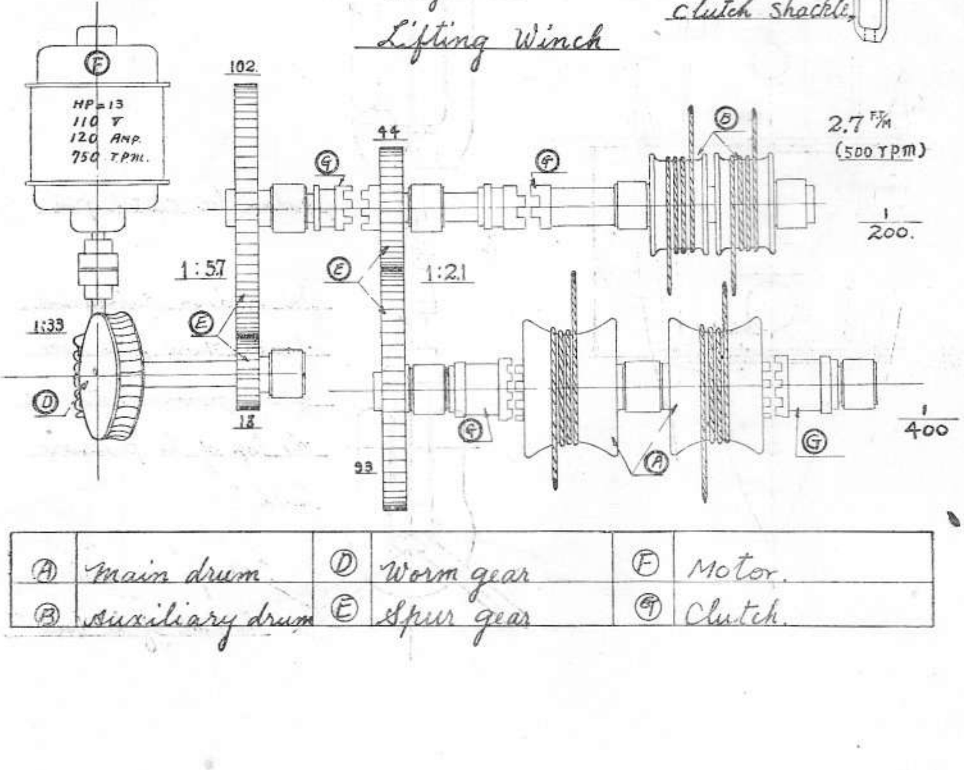
Arrangement of Lifting Wires



Ⓐ	Main drum
Ⓑ	Auxiliary drum
Ⓒ	Roller
Ⓓ	4 Purchase block
Ⓔ	Leading block
Ⓕ	Deck eye pad.
Ⓖ	Portable hand capstan

Fig. VIII

Lifting Winch



Ⓐ	Main drum	Ⓓ	Worm gear	Ⓕ	Motor.
Ⓑ	Auxiliary drum	Ⓔ	Spur gear	Ⓖ	Clutch.

Pressure Distribution over the Surface of a Ship and its Effect on Resistance.

(Paper No. 789)

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This paper is the report of the experiments conducted by us, about the pressure distribution and the total resistance of a ship.

The object is to find out the resistance due to the former, to investigate its relation with the latter, and finally to obtain a fundamental conception for the improvement of the ship's form.

The Course before Beginning the Experiments.

The plan of the experiments to be conducted was carefully thought out during the 13th year of Taishō. As a first step, an apparatus for measuring the pressure was designed and fitted to a motor boat. The result of its repeated preliminary tests was promising.

On February of the next year, a pecuniary support offered by Kondō Kinen Kaiji Zaidan prompted the carrying out of the experiments, and on the beginning of April the following procedure was agreed upon and the necessary preparations began.

(1) The experimental ship is to be a steamboat borrowed from our Navy.
(2) The pressure measuring apparatus used in the preliminary tests is to be adopted with necessary modifications.
(3) The total resistance is to be determined by measuring the thrust of the propeller.
(4) The experiments are to be carried out both with her own propeller and an aero-propeller.
(5) The place of the experiments is to be Arakawa Hōsuiro, where the depth of water is almost constant and disturbances from currents and waves are small.

The Experimental Boat.

The boat was a small steel steamer, decayed with age, corrosion of the outside plating pronounced, and with indentations at several places. But, after close examination, it was finally decided to use her for the purpose, and the execution of the following modifications was entrusted to Yokosuka Navy Yard officials and there completed.

(1) The bilge keels are to be taken away.

(2) The indentations and the seams and the lap butts of the outside plating are to be carefully faired with putty or other stuffing materials, and the whole surface to be finished as smoothly as possible.

(3) Pressure measuring holes are to be bored at the appointed places. In places at both ends of the boat, where the borings of the holes are difficult, the outside of the steel plating is to be sheathed with wood; the measuring holes to be worked therein and led with copper pipes to positions above the water line.

(4) Rudder post and rudder are to be sheathed with wood; the measuring holes to be provided at the allotted places and led to the above water positions with copper pipes.

The form of the hull and the principal dimensions after the modifications are shown in Fig. 1.

Measurement of the Pressure.

The number of the measuring holes on the surface of the boat totals 289, and, as in Fig. 1, almost all of them are arranged symmetrically with regard to the centre line plane. This symmetrical arrangement simplifies the process of obtaining the mean values of the pressures on both sides of the boat.

Along the centre line of the stem, the propeller post, &c., holes are provided, but none along the under side of the keel.

The diameter of the measuring hole is 1 m/m throughout, and all holes are bored normally to the outside surface. In these places, the surface was faired with special care.

Referring to Fig. 2, the measuring hole A is connected through the rubber tube D with the glass bottle B, which is placed directly above it and above water level. The glass bottle B and the glass tube C, of which the latter acts as the pressure indicator, are connected with the rubber tube E.

When the air in the bottle B is sucked out through the pipe F, water rises into it, and its level may be kept higher than that in the tube C and that at the outside of the boat, by closing the pipe F.

The bottle B is purposely placed directly above the hole A, to let the air bubbles, which during a run might enter through the hole A and cause a false reading at the indicator C, rise and accumulate in it. Also the tube D is, for the same reason, fitted possibly straight and vertically, to avoid the air bubbles from lodging in it.

The groups of the pressure indicators as C were arranged in the forward cabin under the deck, but the height of the room was too low for the measurement of high pressures. For such holes, U-tube indicators as shown in Fig. 3 were used. The rubber tube E is connected with the bottle H, and the water level in H coincides with that on the outside of the boat, when the tube J is left open. When the air space in the bottle H is connected with the U-tube indicator K through the rubber tube I and when the tube J is closed, the change of pressure at the point A may be measured by the change of level at the indicator K, the correction due to the compressibility of air being slight and negligible in the present case.

The most difficult task during the preparation was the exclusion of air bubbles which adhered obstinately on the inside of almost all the water tubes. Each rubber tube was carefully washed by connecting its one end with a city-water cock and left running for several hours. Then it was brought into its place and carefully connected. But, some of them were very long and ranged from stern to bow, and others suffered unavoidable bendings.

When the connections were made, the water levels in the group of indicators would by no means coincide with the outside water level; each indicator showed its reading in its own way and quite at random.

After trying several methods of excluding the air bubbles which occurred to us, and after spending nearly three months, we succeeded at last in the complete expulsion of the formidable enemy.

Fig. 4 is a photograph of indicators taken during the experiments, and shows the coincidence of the initial readings conclusively.

The water levels in the indicators, which lie in a straight line as long as the ship stands still, begin to move as she gathers speed and settle down at definite positions when her speed reaches a steady state, as in Figs. 5 and 6.

By reading the height of the water column of each indicator from these photographs, the water pressure at the corresponding hole may be determined.

Measurement of the Speed of the Boat.

The speed of the boat was measured by connecting the Pitot-tube in Fig. 7 with a U-tube, as shown in Fig. 8.

The relation between the speed and the Pitot-tube reading was determined experimentally at Arakawa Hōsuiro on the 26th of December, in the 14th year of Taishō.

A distance of 100 metres was run up and down with her marine propeller, and the corresponding time and the Pitot-tube reading were measured. The draught of the boat was about 120 c/m, both forward and aft.

The result obtained by analysing the measurements is shown in Fig. 9. With this diagram, the relative speed of the boat and the surrounding water may be read off from the Pitot-tube reading.

Measurement of the Thrust of the Aero-propeller.

The diameter of the aero-propeller used in the experiment is 2.5 metres and its pitch 1.00 metre. It is directly coupled to a 100 HP aero-engine, which is fixed on a thrust measuring stand. This stand is, in turn, fixed on a rigid wooden frame-work built upon the after part of the deck, as shown in Fig. 10.

The thrust measuring stand, specially designed for the present purpose, is of cast iron and its side view is rectangular. The base CD is fixed, but the other three sides, being hinged at their corners, are movable.

When the propeller is rotating, its thrust pushes the upper side AB toward the fixed point E, and actuates a dynamometer fitted between them. A damper and its counter weight are provided to obviate disturbing vibrations.

The result of calibration of the dynamometer, made with statical loads, are shown in Fig. 11. The thrusts of the propeller were read off with this diagram from the dial indicator readings.

Measurement of the Thrust of the Marine Propeller.

The diameter of the marine propeller is 0.953 metre and its pitch 1.276 metres. The intermediate shaft between the engine and the tail shaft was replaced by a thrust-meter specially designed for the present experiments. Refer Figs. 12 and 13.

The new intermediate shaft consists of two parts, front and rear. The front shaft has collars which fit the thrust block of the boat. Its forward end is directly coupled with the crank shaft; its aft end is fork shaped and supported by a shaft bearing. The fork transmits the turning moment of the engine to the rear shaft.

The rear shaft is rigidly coupled with the tail shaft. The forward part of the rear shaft is supported with a ball bearing, which is, in turn, mounted on a special bed and is free to move 4 m/m either way from its normal position in the forward and aft directions.

Two springs on both sides of the shaft and parallel to it connect the ball bearing and two movable screws at their rear ends. The motion of the screws is actuated by a handle through two worms and worm wheels; the boss of each wheel forms a stationary nut and acts to shift the screw in the fore and aft direction. When the screws are moved aft, the tension of the springs balances the thrust of the propeller, and the ball bearing is in a floating state at its normal position.

The elongation of the springs in this state is measured and their corresponding tension gives the amount of the thrust.

The foremost part of the rear shaft is also fork shaped and is coupled with the fork of the front shaft, and, between their faces of contact, rollers are provided to allow a slight motion of the shaft in the forward and aft directions.

The calibration of the thrust-meter was effected by substituting for the thrust the standard weights, the shaft being rotated all the while. The result of the calibration is shown in Fig. 14.

The Experiments.

The experiments were carried out at Arakawa Hōsuiro on December in the 14th year of Taishö. The date, the depth of water, the draught, &c. are as follow:

Date	Mode of Propulsion	Draught (Both forward and aft)		Depth of Water	Temper. of Water	Density of Water
		Before experim.	After experim.			
		m	m	m m		
27th	With marine prop.	1.223	1.212	3.90 to 4.20	6°C	1.0008
29th	With aero-prop.	1.220	—	3.38	7.5C	1.0005

During the experiments with the marine propeller, the aero-engine and the aero-propeller were not removed. But, for the experiments with the aero-propeller, the marine propeller was taken away.

Throughout the whole experiments, care was taken to keep the boat on even keel and at no transverse inclination. For this purpose, a selected staff devoted themselves to the adjustments, guided by two clinometers one indicating the longitudinal and the other the transverse inclination.

The wave form during the run was photographed with a kinematograph from another ship running parallel with the experimental boat. But it was found afterwards that the wave form thus photographed was the projection on the boat's side of the highest points of the waves as seen from the position of the kinematograph, and not the actual form of profile along her side.

As will be observed later in this report, the wave form plays a predominant part in deducing the value of the resistance, and it is absolutely necessary to have its accurate data for each experiment. In default of better means, it was decided to make tank experiments to obtain the necessary informations from a model and adopt them as these of the experimental boat.

During the two days, we made 32 runs and every time recorded the speed, the thrust and the pressure distribution, with methods before mentioned.

On the second day, when the experiments were nearly finished, we struck on a sunken boat, completely concealed under water and perfectly invisible. Our boat sprang a leak and was filling fast, compelling us to cut short the experiments and hurry up to a place of safety.

The relation between the speed and the thrust derived from the present experiments is shown diagramatically in Figs. 15 and 16.

The Model Experiments.

A model of the experimental boat was at first tested at Nagasaki Shipbuilding Yard of Mitsubishi Company. When the measurement of the wave profile became necessary, another test was carried out at the experimental tank belonging to the Department of Communication.

The establishment, being newly founded, had at the time its building barely finished and was in the stage of fitting up and equipping. But, in view of the pressing desire to complete our experiments, the remaining work was hurried with all speed. Thus, the model hull and the model propeller for our experimental boat are the very first ones executed in the establishment.

The experiments executed at the government tank are as follow:

The scale of the model hull and the model propeller is one third. As shown in Fig. 17, the hull is fitted with the keel, the rudder, the fender, the Pitot-tube in front of the stem, and is provided with a propeller dynamometer for self propulsion.

The object of the experiments is to measure the wave profile along the side of the model, when towed and when self propelled. The experiments were finished on the beginning of September, this year.

In the towing experiments, the model propeller was removed. Throughout the whole experiment, a suitable trim was given to the model before a run, so that it might keep its keel horizontal during the run.

The result of the measurement is given in Fig. 18.

Analysis of the Experimental Results.

As described above, the present experiments may be grouped into those by the aero-propeller and those by the marine propeller. In the model experiments, the former corresponds to the towing, and the latter to the self-propelling tests.

As the mode of analysis is quite the same in the two cases, we shall describe it mainly with regard to the former and add remarks proper to the latter.

(1) Measured Pressures.

The relation between the reading of a pressure indicator and the corresponding pressure which acts upon the ship is as shown in Fig. 19. APB is a transverse section of the boat and P a measuring hole. C is the water level when she is standing still on her even keel. The water pressure p_0, in this case, is the straight line CE when plotted with the vertical line AD as the base. This corresponds to the case when the water columns of all indicators in Fig. 4 maintain a constant height.

When she is running, p_0 becomes changed into p, as shown in Figs. 5

and 6. If we plot p on the diagram in Fig. 19, we get the curve FG, G being the water surface when running. The reading from the photograph is $QR = p - p_0$, the difference between the dynamic and the static pressure.

Table I is the tabulated list of the measured pressures.

(2) Adjustment of the measured pressures.

The positions of the measuring holes were, as far as practicable, selected at the intersecting points of water lines and square stations. There were some, however, where the borings were impossible from the point of construction.

Among the holes, also, there were some, which, owing to subsequent obstructions, would not show their pressures.

In order to deduce the pressures at these positions, the measured values in Table I were plotted in terms of the girth of each transverse section, and a fair curve drawn through the spots. Fig. 20 is an example of this process.

The forward cabin, where the pressure indicators were equipped, was narrow and a simultaneous photographing of all the indicators was utterly impossible. We were obliged, thus, to divide the whole indicators into four groups and to photograph them successively one after the other, during a single run.

The unavoidable fluctuations of the speed during this interval demanded a graphical interpolation of the pressure of each hole, in order to deduce it to the basis of a constant common speed. Fig. 21 shows a few examples of the process, the square of the speed being conveniently used for the base.

(3) Deduction of the pressure near the water surface.

The draught of the experimental boat was about 122 c/m in still water, but when she was running, the water surface took wavy form, and some of the upper holes emerged out of the water. When once emerged, the holes would no longer give correct indications in the succeeding experiments, even if they remained submerged. The correct indications of the pressures were, thus, practically limited to the holes under the 100 c/m water line.

Hence, the necessity of deducing the pressure near the water line arose. In Fig. 19, the curve FG must be fair and continuous. Its lower part is determined by the measured pressures, while its upper end G is given by the wave profile. Consequently, if we prolong the lower part of the curve up to the point G, we will be able to deduce the intermediate pressures without much error.

Fig. 22 shows some examples of the curves thus drawn. The correctness of the above deduction may further be ascertained by examining the case where the point G is low and falls in a fair curve with the measured points quite near by.

(4) Distribution of pressure.

Denoting the speed of the boat in metres per second by V, the pressures corresponding to the four values of

$$V^2=3,\ 5,\ 7 \text{ and } 9,$$

as determined by the methods (2) and (3), are given in Table II.

As an example, the longitudinal distribution of pressure under the 100 c/m water line is shown in Fig. 23.

(5) Determination of the resistance due to the pressure.

The resultant of p_0 in the forward and aft direction of the boat is evidently zero. Hence we get the resistance due to the pressure by finding the resultant of $p-p_0$ in the same direction.

If dA denotes an elementary surface of the boat, the effective pressure acting on this element is $(p-p_0)$ dA and is normal to it. Its component in the longitudinal direction is

$$(p-p_0)\ dA.\cos\theta,$$

where θ is the angle between the normal to the element dA and the longitudinal direction. But, dA. cos θ is the projection of dA on the body plan of the boat. Now, imagine a fictive surface above the body plan, which passes through the upper ends of the perpendiculars, erected on the plan at the measuring points and of lengths equal to the corresponding values of $p-p_0$. Then, the volume bounded by this surface on the forward body plan diminished by that on the aft gives the resistance due to the pressure.

The cases selected in Fig. 24 are for the water lines of 80 and 125 c/m draughts and $V^2=7$. Curves in full lines are the values of $p-p_0$ for the forward body and those in dotted lines are for the aft body, both in terms of the distance of the measuring point from the centre line plane. The relation of the area between the full and the dotted curves and the draught is shown in Fig. 25.

The resistance due to the pressure, or the form resistance, which is nothing other than the area of the above curve, is shown in Fig. 15, on the base of V^2.

(6) The experimental results with the marine propeller.

The method of analysis is quite similar with that above described, but in this case the result for $V^2=3$ is not calculated. The experimental boat was unfit to maintain such low speed with her engine.

List of Tables and Drawings, showing the correspondence between the two kinds of the experiments.

Experiments with the aero-propeller	Experiments with the marine propeller
Table I	Table III
Fig. 20	Fig. 26
Fig. 21	Fig. 27
Table II	Table IV
Fig. 23	Fig. 28
Fig. 24	Fig. 29
Fig. 25	Fig. 30
Fig. 15	Fig. 16

Considerations on the Experimental Results.

As mentioned before, the experiments were forced to terminate abruptly by an accident. But they provide useful and important materials for the greater part of our object.

Measuring holes were not provided along the sixth square station (mark J). This omission was unavoidable on account of the boiler extending from side to side of the ship and no space for fittings.

The omission may throw a shade on the determination of the resistance, but the station being situated close by the midship section, errors in the deduced values of the pressures along this station would have small or no effect on the determination.

Moreover, the displacement, calculated by using the vertical component of the deduced pressures at the above station, agreed within a few millimetres of draught, in the case of the aero-propeller.

The displacement similarly calculated for the case of the marine propeller showed a difference of about 12 millimetres, but as the change of draught caused by the fuel consumption amounted to 11 millimetres, the agreement was again fully within the experimental errors.

As the vertical component of pressures at the station under consideration greatly influences the value of the displacement, the calculation is a very severe check on the validity of the deduced values. Yet, as mentioned above, the agreement is quite satisfactory, so that we may safely use the deduced values for the estimation of the resistance.

If we compare the wave form in Fig. 18 with the pressure distribution in Figs. 23 and 28, the similarity of the general form may be noticed, up to a tolerable depth. But if we compare this with a pressure distribution of an airship, some differences are recognized. This shows the importance of considering the surface disturbance of a ship, which moves on the surface of water, apart from that which travels within a medium, as an airship or a submarine. In the region near the keel, both have a common tendency.

The curves in Figs. 25 and 30 are wavy. The hollow about the 20 c/m draught is caused by low pressure at MNOP in Figs. 23 and 28, agreeing with the generally accepted idea of the concentration of stream lines at that part.

The humps and the hollows of the upper half are mainly caused by the waves. The area of this part occupies a substancial part of the whole area, and shows the important influence of the waves upon the resistance. The increase of the influence with the speed is, also, clearly noticeable.

The upper end of each curve coincides with the highest point of the wave profile at the bow, and the top of the hump is a little lower than the highest point of the stern wave. This hump shows that the elevation of the water surface at the bow more than cancels the good effect of the elevation at the stern, and goes on to increase the resistance.

If we were able to diminish the elevation at the bow, the resistance would be considerably diminished. In other words, this fact shows the possibility of diminishing the resistance by a suitable improvement of the bow form.

The general tendency in the pressure distribution for the two kinds of propulsion is similar, and in particular agrees well at the bow. But, a definite difference exists at the stern where the action of the marine propeller has influence. This is so-called suction phenomena, and is a cause of the increase of resistance.

This increase of resistance may be seen by comparing Figs. 25 and 30. The ordinates in Fig. 30 is larger than in the other, except in the vicinity of the 130 c/m draught.

By comparing the form resistances in Figs. 15 and 16, we get,

TABLE A.

V^2		5	7	9	
Resistance due to pressure	Fig. 16 (marine)	kg 47.6	kg 79.4	kg 138.0	a
	Fig. 15 (aero-)	28.1	48.4	86.1	b
Difference of resistance due to pressure...		19.5	31.0	51.9	c
Above difference in percentage of resistance in the first row (marine)		% 41.0	% 39.0	% 37.6	d

The above is the increase of resistance caused by the suction phenomena.

The increase of the total resistance due to the suction effect may be deduced by comparing the thrusts of the propellers in Figs. 15 and 16, as follows:

TABLE B.

V^2	5	7	9	
Thrust of marine propeller	kg 80.5	kg 126	kg 200	a
Thrust of aero-propeller	68.5	106.5	165	b
Difference	12	19.5	35	c
Above difference in percentage of thrust of marine propeller (row a)	% 14.9	% 15.5	% 17.5	d

In considering the above figures, we must bear in mind that a certain amount in the thrust of the aero-propeller counterbalances the increased resistance caused by the induced air current from striking the upper works on the deck.

Also, the frictional resistances for the two kinds of propulsion can not be definitely concluded to be the same, in consideration of the different velocity distributions in the two cases.

If we subtract the third row figures of Table B from those of Table A, we get

TABLE C.

V^2	5	7	9
Difference of resistance due to the pressure	kg 19.5	kg 31.0	kg 51.9
Difference of thrusts	12.0	19.5	35
Second difference	7.5	11.5	16.9

This second difference may be supposed, for the moment, to be the increase of thrust proper to the aero-propeller.

The thrust deduction factor for the case of marine propeller, deduced from the third row of Table A and the first row of Table B, are

TABLE D.

V^2	5	7	9
Thrust deduction factor in percentage	% 24.2	% 24.6	% 26.0

The above values seem reasonable for a small vessel of this class.

Besides the resistance due to the pressure acting normally to a ship's surface, there is another which is the longitudinal component of the friction acting tangentially along it. The former is the form, and the latter the frictional, resistance.

The thrust of a propeller must be equal to the sum of these two kinds of resistances, if they were measured without a single omission. But, in the present experiments, the form resistance was deduced from the pressure distribution on the hull proper only, and no account was taken of the resistances of the rudder and the Pitot-tube, while the thrust of the propeller included the air resistance of upper works, besides those above mentioned.

Hence, it is difficult to deduce the correct value of the frictional resistance by our experiments.

If we calculate the frictional resistance by using the O-coefficient of

R. E. Froude and add it to the form resistance deduced above, we get a line directly below the thrust curve, as shown in Figs. 15 and 16.

From this, we get the results:

TABLE E.

V^2	5		7		9	
	Air	Water	Air	Water	Air	Water
Thrust of propeller*	kg 61.0	kg 80.5	kg 95.0	kg 126	kg 148.1	kg 200
Sum of form and frictional resistances...	58.8	78.3	90.2	121.2	138.7	190.6
Difference	2.2		4.8		9.4	

* For the aero-propeller, the second difference in Table C is deducted out of the measured thrust.

As the differences in the last row are small, we may infer that the frictional resistance calculated by the Froude's formula is applicable for this kind of ships.

The recent controversy about the validity of the formula is for ships of high Reynolds numbers, and not for ships, small and of low speed as was used in the present experiments. The above deduction intrudes, by no means, upon the applicability of the formula for a large and speedy ship. The experimental research on the frictional resistance of a large and speedy ship is of pressing necessity and forms one of the most important subject in the science of naval architecture.

In finishing this report, we wish to express our thanks for a liberal gift made by Kondō Kinen Zaidan to aid the expense of our experiments. Vice admirals Noda, Hiraga and Isozaki of our Navy showed us the utmost good will and consulted our convenience during the whole experiments.

Our thanks are, also, due to Dr. Motora, who made a model experiment of the boat at Nagasaki Shipbuilding Yard, and to Messrs. Naoshi Satō, Masao Yamagata, Yoshio Kikuchi, Yoshio Matsubara and Torao Takehara, who assisted us in various ways during the experiments.

— End —

TABLE I.

$\left\{\begin{array}{l}\text{p–p}_0 \text{ in cm.}\\ \text{V in metres per sec.}\end{array}\right.$

Exp. No.		29	30	31	32
Speed V=		1.49	2.03	2.68	2.875
(Holes in front of stem) Marks of Holes.	g_2	6.2	12.0	21.5	24.0
	g_3	11.0	20.7	36.5	41.8
	g_4	13.2	24.7	——	——
	g_5	13.8	24.8	42.4	48.2
Marks of Holes	g_2	3.3	6.9	12.5	14.0
	g_3	3.0	5.8	10.2	11.5
	g_4	5.2	9.5	16.4	18.5
	g_5	6.8	13.0	22.4	27.4
	g_6	7.8	15.0	25.2	28.3
	g_7	8.7	16.6	27.5	30.7

Exp. No.		17′	17	18	19	20
Speed V=		2.025	1.645	2.365	2.555	2.78
Marks of Holes	R_3	8.4	5.4	11.5	13.7	16.0
	R_4	8.6	5.9	13.4	16.2	19.2
	R_5	11.6	7.1	16.1	19.5	23.1
	R_6	12.8	7.6	18.0	21.5	25.7
	R_7	13.2	7.8	18.8	22.7	27.0
Marks of Holes	Q_1	4.6	3.4	7.2	8.3	9.6
	Q_3	6.4	4.2	8.7	10.7	12.6
	Q_4	8.0	4.9	10.8	13.4	15.8
	Q_5	8.6	5.5	12.1	15.1	18.2
	Q_6	9.6	6.0	13.6	17.1	20.9
	Q_7	9.8	6.1	14.5	18.6	22.8
Marks of Holes	P_0	3.3	1.9	3.6	4.6	5.2
	P_2	4.3	3.1	5.7	6.8	7.9
	P_3	5.5	3.8	7.2	8.9	10.3
	P_4	6.6	4.4	8.5	10.6	12.9
	P_5	7.6	5.2	10.3	13.0	16.0
	P_6	8.0	5.7	11.1	14.6	18.0
	P_7	8.3	5.9	11.4	15.5	19.0

Table I, *Continued.*

Exp. No.		21	22	23	24
Speed V=		2.03	2.60	2.84	3.07
Marks of Holes	O_0	0.6	0.9	1.9	2.6
	O_3	0.7	1.2	2.3	3.5
	O_4	2.0	3.2	5.1	7.0
	O_5	3.3	5.3	7.8	10.3
Marks of Holes	N_0	−0.1	0	0.3	0.4
	N_1	−0.5	−0.6	−0.4	−0.1
	N_2	−1.6	−2.6	−2.9	−3.2
	N_3	−0.1	−0.8	−0.9	−0.6
	N_4	2.6	2.9	4.0	5.4
	N_5	3.5	3.2	4.4	6.4
	N_6	4.4	3.9	5.7	8.0
Marks of Holes	M_0	0.4	0.5	0.7	0.8
	M_1	−0.7	−1.2	−1.5	−1.9
	M_2	−2.0	−3.6	−4.6	−6.2
	M_3	−1.0	−2.8	−3.9	−5.4
	M_4	−0.4	−1.2	−2.3	−3.3
	M_5	1.4	0.9	0.6	0.5
	M_6	0.8	−0.2	−1.1	−1.0
Marks of Holes	L_0	0.6	0.9	0.8	0.3
	L_1	−0.7	−1.0	−1.5	−2.7
	L_2	−1.6	−2.8	−3.7	−5.3
	L_3	−1.6	−2.9	−4.6	−6.5
	L_4	−2.0	−3.2	−5.0	−8.0
	L_5	−1.5	−2.3	−4.3	−6.5
	L_6	−0.8	−0.8	−2.9	−5.5
Marks of Holes	K_0	1.3	2.0	2.1	1.2
	K_1	0.2	0.7	0.5	−0.2
	K_2	−2.2	−4.0	−5.4	−7.5
	K_3	−3.7	−5.3	−7.3	−11.4
	K_4	−2.8	−4.2	−6.7	−10.4
	K_5	−3.5	−5.1	−8.4	−11.9
	K_6	−2.2	−1.5	−4.3	−9.9
Marks of Holes	I_0	0.8	1.2	1.4	2.1
	I_1	0.7	1.6	1.8	1.3
	I_3	0.6	0.8	1.5	1.3
	I_4	0.1	0.2	0.2	−0.1
	I_5	0.1	0.5	1.2	0.2
	I_6	−0.2	0.4	1.2	0.8

Table I, *Continued.*

Exp. No.		21	22	23	24
Speed V=		2.03	2.60	2.84	3.07
Marks of Holes	H_0	1.7	2.9	3.5	3.5
	H_1	0.9	1.6	2.1	1.5
	H_2	−0.7	−0.5	−0.5	−0.5
	H_3	−0.1	0.35	1.0	1.2
	H_4	−1.2	−1.0	−0·7	0.2

Exp. No.		25	26	27	28
Speed V=		2.115	2.615	2.88	3.05
Marks of Holes	G_0	1.2	1.3	2.4	3.4
	G_1	−0.5	−0.8	−1.0	−0.3
	G_2	−0.9	−1.9	−1.9	−1.2
	G_3	−0.6	−1.2	−1.5	−0.2
	G_4	−0.5	−1.5	−1.7	−0.3
	G_5	−0.7	−3.0	−3.0	−1.1
	G_6	−0.4	−3.9	−3.9	−1.5
Marks of Holes	F_0	−0.8	−1.7	−1.7	−1.3
	F_1	−1.6	−2.7	−3.7	−3.6
	F_2	−2.3	−3.7	−5.1	−4.4
	F_3	−1.6	−3.1	−4.2	−4.0
	F_4	0	−0.7	−0.9	−0.2
	F_5	0.1	−1.3	−2.0	−1.6
	F_6	−0.2	−2.5	−4.1	−3.5
Marks of Holes	E_0	0.7	1.3	1.1	0.6
	E_2	−0.9	−1.6	−1.8	−1.9
	E_3	−0.4	−1.1	−0.7	−0.6
	E_4	0.7	0.8	0.7	0.6
	E_5	−0.2	−0.4	−1.3	−2.3
	E_6	0.6	1.2	−0.5	−1.9
Marks of Holes	D_1	0.8	1.0	1.8	2.7
	D_2	1.3	2.1	2.2	2.2
	D_3	−0.6	−1.0	−1.0	−1.7
	D_4	1.6	2.7	4.2	3.6
	D_5	2.3	3.8	4.6	4.4
	D_6	3.0	4.8	5.8	4.8

Table I, *Continued.*

Exp. No.		25	26	27	28
Speed V=		2.115	2.615	2.88	3.05
Marks of Holes	C_2'	5.1	6.9	7.5	7.75
	C_3'	6.05	10.05	10.85	12.3
	C_4'	4.2	5.8	8.0	8.6
	C_5'	6.0	9.2	11.3	12.4
	C_6'	6.0	9.2	11.3	12·6
Marks of Holes	B_2	6.1	9.6	11.7	12.7
	B_3	5.4	8.5	10.3	11.1
	B_4	7.1	10.9	13.4	14.3
	B_5	7.2	11.3	13.6	14.6
	B_6	6.9	11.1	13.4	14.4
Marks of Holes	A_2	6.0	9.5	11.3	11.6
	A_3	5.6	8.8	10.9	11.2
	A_4	7.7	11.5	14.7	15.5
	A_5	8.0	12.6	14.9	15.6
	A_6	8.2	12.8	15.3	16.4

Exp. No.		29	30	31	32
Speed V=		1.49	2.03	2.68	2.875
Marks of Holes	f_0	3.1	4.6	7.8	9.1
	f_1	2.8	3.0	5.1	5.7
	f_2	3.4	5.3	8.9	10.4
	f_3	3.3	5.8	10.2	12.0
	f_4	4.1	7.3	12.6	14.6
	f_6	4.1	7.5	12.9	14.5
Holes at back of Propeller Post	f_0	3.0	4.6	7.9	10.3
	f_1	3.3	4.4	7.1	7.5
	f_2	2.7	4.6	8.3	10.1
	f_3	3.1	5.8	10.4	11.6
	f_4	4.4	7.3	12.4	14.0
	f_6	5.3	7.7	12.8	14.5

TABLE II.

$p-p_0$ in cm.
V in metres per sec.

$V^2=3$

MARK	DRAUGHT IN CM.						
	10	20	40	60	80	100	105
g_c	6.50	9.55	15.60	17.60	18.00	17.60	17.6
g_s	4.35	4.80	4.20	7.00	9.40	10.80	11.4
R	5.70	5.80	6.00	6.60	8.00	8.65	8.7
Q	3.95	4.10	4.55	5.70	6.17	6.70	6.8
P	2.75	3.35	4.20	4.90	5.75	6.15	6.4
O	0.22	0.20	0.60	1.65	2.70	3.70	4.1
N	−0.35	−1.10	0.10	2.30	3.22	3.95	3.5
M	−0.51	−1.45	−0.65	−0.25	1.30	0.80	0.4
L	−0.60	−1.20	−1.20	−1.67	−1.35	−0.90	−1.1
K	−0.17	−1.80	−3.30	−2.70	−3.20	−2.30	−2.4
J	0.05*	−1.60*	−3.70*	−1.65*	−2.65*	−0.90*	
I	0.30	0.70	0.45	0.05	0.05	−0.18	−0.5
H	0.70	−0.70	−0.10*	−1.00*	−0.50*	−0.20*	−0.2
G	−0.46	−0.40	−0.30	−0.20	0.10	0.55	−0.2
F	−1.14	−1.60	−0.85	0.20	0.40	0.35	0
E	0.22	−0.20	−0.05	0.50	−0.10	0.10	0
D	0.45	0.75	−0.30	1.20	1.40	1.80	2.3
C′	2.46*	2.85*	3.80*	3.00	4.05	4.10	4.4
B	4.24	4.00	3.60	4.85	4.85	4.55	4.8
A	3.67	3.90	3.80	5.20	5.30	5.50	5.8
f_s	2.98	4.20	4.30	5.20	5.83	5.50	5.3
f_c	3.66	3.50	4.20	5.35	6.15	6.27	6.1

* Obtained by Fairing.

$V^2=3$

MARK	DRAUGHT IN CM.					
	110	115	120	125	130	135
g_c	17.5	17.4	17.3	14.1	9.0	3.8
g_s	11.9	12.5	13.5	11.3	7.2	3.2
R	8.9	8.5	8.3	5.0	≑0	
Q	6.8	6.3	5.6	1.5		
P	6.5	6.5	6.2	3.1		
O	4.4	4.5	4.7	1.8		
N	3.3	3.1	2.5			
M	−0.2	−0.4	−0.4			
L	−1.6	−2.8	−2.0			
K	−3.1	−4.5	−2.0			
J						
I	−0.7	−1.0	−1.5			
H	−0.6	−1.0	−2.0			
G	−0.5	−1.7	−2.0			
F	−0.3	−1.0	−1.8			
E	0	0	0			
D	2.4	2.5	2.5	≑0		
C′	4.3	4.3	4.5	1.2		
B	4.6	4.5	4.5	1.2		
A	6.0	6.1	6.1	3.2		
f_s	5.0	4.6	4.4	0.7		
f_c	5.9	5.2	4.7	0.8		

Table II, *Continued.*

$V^2=5$

MARK	DRAUGHT IN CM.							
	10	20	40	60	80	100	105	110
g_c	10.80	15.70	25.80	29.00	29.70	29.30	29.20	29.0
g_s	7.20	8.60	7.15	11.45	15.95	18.00	19.0	19.6
R	9.90	10.00	10.35	11.95	14.45	15.90	16.3	16.6
Q	6.50	6.70	7.85	9.80	10.90	12.20	12.6	12.9
P	4.15	5.15	6.55	7.75	9.35	9.95	9.7	9.5
O	0.3	0.10	0.85	2.20	3.50	4.75	5.2	4.8
N	−0.65	−2.00	−0.35	2.40	3.40	4.05	5.1	4.7
M	−0.85	−2.50	−1.40	−0.60	1.30	0.60	0	0
L	−0.80	−1.95	−1.95	−2.25	−1.50	−0.60	−0.7	−1.3
K	0.49	−2.70	−4.00	−2.95	−3.60	−1.70	−2.1	−4.0
J	0.55*	−3.62*	−7.30	−3.10*	−2.60*	−1.15*		
I	1.00	0.90	0.55	0.10	0.10	−0.17	−0.2	−0.2
H	1.10	−0.70	−0.05*	−1.20*	−1.10*	−0.85*	−1.1	−1.6
G	−0.69	−1.10	−0.75	−0.80	−1.25	−1.15	−1.5	−2.5
F	−1.94	−2.65	−1.80	−0.10	−0.15	−0.60	−1.0	−1.3
E	0.65	−0.85	−0.50	0.75	−0.15	0.90	1.1	1.4
D	0.60	1.55	−0.70	1.80	2.65	3.40	3.5	3.6
C′	3.71*	4.45*	7.10*	4.50	6.65	6.60	6.6	6.6
B	7.15	6.80	6.10	7.90	8.05	7.80	8.3	8.4
A	7.00	6.70	6.20	8.40	9.00	9.25	9.1	9.3
f_s	3.32	6.35	7.00	8.37	9.40	9.15	9.3	9.3
f_c	5.17	5.50	7.15	8.55	9.40	9.10	9.2	9.5

* Obtained by Fairing.

$V^2=5$

MARK	DRAUGHT IN CM.						
	115	120	125	130	135	140	145
g_c	28.5	28.1	24.4	18.8	13.2	7.7	2.0
g_s	20.4	21.1	18.7	14.5	10.3	6.1	1.8
R	17.0	17.2	14.7	10.4	6.0	1.8	
Q	12.9	12.6	9.3	4.0			
P	8.8	8.1	4.3				
O	4.0	3.0					
N	4.0	2.9					
M	0	0					
L	−2.3	−2.0					
K	−7.0	−2.0					
J							
I	0	0					
H	−2.4	−1.9					
G	−4.8	−2.0					
F	−1.7	−1.9					
E	1.6	1.6					
D	3.6	3.8	0.7				
C′	6.5	6.5	3.3				
B	8.5	8.6	5.5	0.5			
A	9.3	9.3	6.4	1.5			
f_s	9.4	9.7	6.6	1.7			
f_c	9.4	9.6	6.5	1.6			

Table II, *Continued.*

$V^2 = 7$

MARK	DRAUGHT IN CM.								
	10	20	40	60	80	100	105	110	115
g_c	15.10	22.00	36.10	40.50	41.40	41.50	41.6	41.6	41.4
g_s	9.90	12.20	9.90	16.00	22.80	24.60	25.8	26.4	27.3
R	14.05	14.15	14.70	17.30	20.90	23·17	23.9	24.4	24.9
Q	8.85	9.25	11.40	14.30	16.30	18.45	19.0	19.5	20.1
P	5.70	7.22	9.40	11.40	14.10	15.80	15.9	16.0	15.6
O	0.60	0.22	1.50	3.60	5.70	8.35	8.6	8.3	7.5
N	−0.55	−2.75	−0.88	3.05	3.45	4.15	4.1	3.5	2.5
M	−1.27	−3.80	−3.00	−1.40	0.85	−0.40	−1.1	−1.9	−2.2
L	−1.08	−2.95	−3.15	−3.43	−2.60	−1.08	−1.4	−2.5	−5.4
K	0.66	−4.25	−5.55	−4.55	−5.60	−1.80	−2.2	−4.4	−6.9
J	1.40*	−4.05*	−5.60*	−4.80*	−3.55*	−2.75*			
I	1.66	1.45	0.90	0.25	0.60	0.55	0.5	0	−0.5
H	1.72	−0.50	0.45*	−1.00*	−1.20*	−1.40*	−1.5	−1.5	−1.6
G	−1.08	−1.95	−1.25	−1.70	−3.05	−4.00	−4.2	−4.7	−5.6
F	−3.04	−3.85	−3.15	−0.75	−1.40	−2.70	−3.5	−4.1	−4.9
E	1.05	−1.72	−1.00	0.80	−0.45	1.10	0.9	0.5	0.2
D	0.77	2.15	−0.90	3.00	3.90	5.00	4.9	4.8	4.8
C′	5.08*	6.00*	10.10*	6.00	9.40	9.33	9.5	9.4	9.4
B	10.20	9.80	8.75	11.20	11.55	11.40	11.5	11.5	11.5
A	10.30	9.65	9.10	11.75	12.90	13.15	13.2	13.3	13.4
f_s	4.98	8.85	9.92	11.80	12.90	12.65	13.5	13.7	14.1
f_c	6.99	8.10	10.15	11.80	12.85	12.55	13.0	13.2	13.7

* Obtained by Fairing.

$V^2 = 7$

MARK	DRAUGHT IN CM.							
	120	125	130	135	140	145	150	155
g_c	41.1	37.8	32.5	27.0	21.4	15.5	10.0	
g_s	28.2	26.0	21.7	18.3	14.5	10.7	6.8	
R	25.5	23.2	18.8	14.6	10.2	5.8	1.4	
Q	21.0	18.8	14.6	10.4	6.1	1·8		
P	15.4	11.8	6.2	0.5				
O	6.3	1.3						
N	0.5							
M	−2.0							
L	−2.0							
K	−2.0							
J								
I	−1.5							
H	−1.5							
G	−2·0							
F	−1.9							
E	0							
D	4.6	1·6						
C′	9.3	6.0	1.0					
B	11.7	8.6	3.6					
A	13.5	10.4	5.5	0.5				
f_s	14.4	11.7	7.1	2.5				
f_c	14.0	11.5	7.0	2.5				

Table II, *Continued.*

$V^2=9$

MARK	DRAUGHT IN CM.								
	10	20	40	60	80	100	105	110	115
g_c	19.40	28.20	46.30	52.00	53.10	53.00	52.5	52.4	52.2
g_s	13.40	14.65	12.20	19.50	30.00	30.20	31.9	32.7	33.4
R	18.10	18.20	19.00	22.75	27.45	30.50	31.3	32.1	33.0
Q	11.00	11.95	15.40	19·20	22.60	25.65	26.5	27.1	27.8
P	7.55	9·65	12.90	16.20	20.40	23.35	24.0	24.3	24.3
O	1.65	0.60	3.25	6.50	9.55	12.35	13.1	13.7	14.4
N	−0.20	−3.20	−0.75	5.00	5.75	7.25	6.6	5.5	3.6
M	−1.76	−5.60	−4.85	−3.00	0.50	−1.15	−2.5	−3.6	−4.5
L	−2.26	−4.80	−5.90	−6.90	−5.85	−4.70	−5.5	−8.0	−6.8
K	−0.02	−6.85	−9.80	−9.20	−10.85	−7.90	−8.6	−12.1	−7.0
J	1.80*	−5.70*	−6.55*	−5.55*	−5.85*	−4.70*			
I	1.43	1.75	1.45	0	0.60	1.00	1.1	1.0	0.7
H	1.84	−0.50	1.20*	−0.20*	−0.10*	0*	0.2	0.6	1.4
G	−0.80	−1.45	−0.70	−0.95	−1.75	−2.40	−2.8	−3.5	−5.0
F	−3.80	−4.80	−4.10	−0.50	−1.80	−3.80	−5.7	−9.3	−7.0
E	0.50	−1.95	−0.50	0.65	−2.00	−1.50	−3.0	−5.0	−7.2
D	2.35	2.12	−1.40	4.00	4.52	5.20	5.2	5.4	5.5
C′	6.87	7.75	11.80	8.50	12.10	12.20	12.2	12.2	12.0
B	12.95	12.35	10.90	14.15	14.35	14.20	14.1	14.1	14.1
A	13.20	11.50	11.40	15.20	15.40	16.10	16.3	16.5	16.8
f_s	5.58	11.47	13.20	15.70	16.20	15.58	15.7	16.0	16.3
f_c	7.77	11.55	12.20	14.50	16.05	15.60	16.1	16.5	17.1

* Obtained by Fairing.

$V^2=9$

MARK	DRAUGHT IN CM.							
	120	125	130	135	140	145	155	160
g_c	51.7	48.3	42.8	37.3	31.7	25.8	13.8	8.1
g_s	34.5	32.3	28.4	24.6	20.7	17.0	9.4	5.5
R	33.9	31.7	27.5	23.3	19.2	15.0	6.6	2.4
Q	28.4	26.2	22.0	17.6	13.3	8.9		
P	24.3	20.9	15.7	10.3	5.0	≑0		
O	15.0	12.8	8.5	4.1	≑0			
N	0.9							
M	−2.0							
L	−2.0							
K	−2.0							
J								
I	0.5							
H	2.2	0.4						
G	−2.0							
F	−2.0							
E	−2.1							
D	5.6	2.7						
C′	12.0	9.1	4.1					
B	14.2	11.1	6.2	1.2				
A	17.1	14.4	9.6	4.9	0			
f_s	16.8	14.3	9.9	5.5	1.2			
f_c	17.5	14.9	10.3	5.7	1.2			

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TABLE III.

$p-p_0$ in cm.
V in metres per sec.

Exp. No.		13	14	15	16
Speed V=		2.10	2.33	2.99	3.142
(Holes in front of stem) Marks of Holes	g_1	4.7	6.6	11.2	12.0
	g_2	12.3	15.6	28.6	34.6
	g_3	20.1	25.0	43.6	51.3
	g_5	24.0	29.5	51.0	59.9
	g_6	25.4	32.0	52.2	57.8
	g_7	25.3	32.0	52.4	56.2
Marks of Holes	g_1	4.7	5.8	12.2	15.1
	g_2	4.8	6.5	14.4	17.8
	g_3	4.3	5.4	11.9	15.0
	g_4	8.0	10.3	19.4	23.4
	g_5	12.2	15.4	28.2	31.8
	g_6	13.5	16.6	29.6	35.2
	g_7	15.3	18.1	31.7	36.8

Exp. No.		1	3	4
Speed V=		2.39	2.76	3.065
Marks of Holes	R_3	10.5	14.2	18.6
	R_4	12.1	16.5	22.1
	R_5	15.6	21.2	27.4
	R_6	17.6	23.6	30.7
	R_7	18.4	25.0	32.4
Marks of Holes	Q_1	5.1	7.0	9.6
	Q_3	7.7	10.7	14.4
	Q_4	10.0	13.9	18.6
	Q_5	11.5	16.2	21.7
	Q_6	13.3	19.2	25.8
	Q_7	14.1	20.5	27.5
Marks of Holes	P_0	2.2	3.3	5.9
	P_2	3.9	5.3	8.4
	P_3	5.8	8.0	11.7
	P_4	7.5	10.6	14.9
	P_5	9.5	13.7	18.9
	P_6	10.4	15.8	21.5
	P_7	11.0	17.0	22.7

Exp. No.		6	7	8
Speed V=		2.53	2.70	3.27
Marks of Holes	O_0	1.6	2.3	4.3
	O_3	2.1	2.8	5.4
	O_4	4.1	5.2	10.1
	O_5	6.2	7.5	14.5
Marks of Holes	N_0	0	0.4	1.2
	N_1	−0.4	0	0.3
	N_2	−2.2	−2.0	−3.4
	N_3	−0.1	0	0.3
	N_4	3.7	4.3	7.6
	N_5	4.0	4.6	8.4
	N_6	4.8	5.3	11.7

Exp. No.		6	7	8
Speed V=		2.53	2.70	3.27
Marks of Holes	M_0	0.1	0.2	0.6
	M_1	−1.1	−1.0	−1.3
	M_2	−3.5	−4.0	−6.5
	M_3	−2.5	−3.2	−5.8
	M_4	−0.7	−1.2	−2.4
	M_5	1.6	1.3	2.1
	M_6	0.7	0.1	0.4
Marks of Holes	L_0	0.1	0.1	0.8
	L_1	−2.1	−2.5	−3.8
	L_2	−3.0	−3.4	−5.7
	L_3	−3.3	−4.1	−7.7
	L_4	−3.2	−4.2	−8.9
	L_5	−2.1	−2.7	−6.8
	L_6	−0.5	−1.5	−5.3
Marks of Holes	K_0	0.9	0.7	1.8
	K_1	0.1	−0.1	0.3
	K_2	−4.7	−5.4	− 9.2
	K_3	−6.2	−7.3	−12.3
	K_4	−4.8	−6.0	−12.2
	K_5	−5.3	−6.5	−14.2
	K_6	−2.7	−4.1	−10.3
Marks of Holes	I_0	0.6	0.6	1.4
	I_1	0.1	−0.1	1.0
	I_3	0.4	0.3	0.6
	I_4	−1.0	−1.1	−2.3
	I_5	−0.1	−0.2	−0.9
	I_6	0.5	0.5	−2.4
Marks of Holes	H_0	1.2	1.1	4.6
	H_1	−0.2	−0.5	2.5
	H_2	−1.7	−1.9	1.0
	H_3	−0.7	−0.9	1.9
	H_4	−2.0	−2.4	1.3

Table III, *Continued.*

Exp. No.		9	10	11	12
Speed V=		2.14	2.75	3.01	3.13
Marks of Holes	G_0	0.8	2.2	2.3	3.2
	G_1	−1.3	−0.8	−1.2	−0.8
	G_2	−1.5	−1.65	−2.3	−1.5
	G_3	−1.2	−1.1	−1.3	−0.35
	G_4	−1.2	−1.4	−1.5	−0.15
	G_5	−1.4	−2.55	−2.9	−0.8
	G_6	−1.2	−3.05	−2.8	0.1
Marks of Holes	F_0	−1.8	−2.0	−2.8	−2.75
	F_1	−2.7	−3.3	−4.6	−4.55
	F_2	−3.3	−4.3	−5.6	−6.0
	F_3	−2.9	−3.9	−5.3	−4.8
	F_4	−0.8	−1.0	−2.1	−0.9
	F_5	0	−1.6	−3.6	−2.1
	F_6	−0.6	−3.05	−5.8	−4.2
Marks of Holes	E_0	−0.3	0.6	−0.3	0.5
	E_2	−0.9	−0.6	−1.7	−1.1
	E_3	−1.3	−0.9	−2.3	−2.0
	E_4	−0.3	0.2	−1.9	−1.2
	E_5	−1.1	−1.4	−5.1	−4.7
	E_6	−0.3	−0.3	−4.2	−4.3
Marks of Holes	D_0	−1.0	0.1	−1.3	−1.3
	D_2	−0.3	0.9	−0.5	−0.1
	D_3	−0.9	−0·7	−3.3	−2.6
	D_4	0.5	1.8	0.3	0.7
	D_5	1.0	3.5	1.7	1.7
	D_6	1.1	3.8	0.7	−0.1
Marks of Holes	C_2'	3.2	6.0	6.0	8.6
	C_3'	3.3	7.2	7.3	8.6
	C_4'	3.7	6.0	5.6	5.6
	C_5'	4.1	8.0	8.1	9.1
	C_6'	4.0	8.3	8.3	8.8
Marks of Holes	B_2	4.2	8.1	8.6	9.6
	B_3	3.3	7.1	7.3	8.6
	B_4	4.7	9.1	9.6	11.1
	B_5	5.9	10.2	11.9	12.6
	B_6	5.8	10.5	12.5	13.1
Marks of Holes	A_2	3.2	6.4	6.5	7.1
	A_3	3.6	7.3	7.6	8.3
	A_4	4.2	9.2	9.5	12.4
	A_5	5.1	9.9	11.0	12.0
	A_6	6.5	11.3	13.1	13.4

Exp. No.		13	14	15	16
Speed V=		2.10	2.33	2.99	3.142
Marks of Holes	f_0	−1.8	−2.5	−5.3	8.4
	f_1	−2.9	−4.3	−6.9	−9.4
	f_2	1.6	1.8	4.7	4.9
	f_3	1.9	2.0	4.3	3.7
	f_4	3.1	3.5	7.5	6.0
	f_6	−1.5	−1.0	−6.8	−7.3
(Holes at Back of Propeller Post) Marks of Holes	f_0	−2.0	−3.3	−5.1	−7.8
	f_1	−1.8	−3.1	−5.1	−8.8
	f_2	−0.5	−1.0	−1.0	−2.2
	f_3	2.8	3.2	6.2	5.3
	f_4	2.6	2.6	5.9	4.0
	f_6	0	1.8	−3.0	−2.2

TABLE IV.

$p-p_0$ in cm.
V in metres per sec.

$V^2=5$

MARK	DRAUGHT IN CM.						
	10	20	40	60	80	100	110
g_c	10.1	15.7	23.3	25.7	27.3	29.1	29.0
g_s	5.5	5.7	4.9	9.3	14.0	15.2	17.0
R	8.75	8.85	9.2	10.4	13.2	15.0	15.5
Q	4.5	4.95	6.7	8.6	9.9	11.2	11.7
P	2.5	3.45	5.0	6.5	8.1	8.7	8.7
O	0.35	0.2	0.9	2.6	4.3	3.1	3.5
N	−0.85	−2.7	−0.5	2.6	2.8	3.6	3.5
M	−1.1	−2.85	−1.7	−0.5	1.5	0.5	0.7
L	−1.9	−2.6	−2.6	−2.4	−1.7	0.1	−0.2
K	−0.1	−4.1	−5.0	−3.5	−4,0	−1.6	−2.5
J	−0.1*	−5.1*	−6.4*	−4.4*	−3.8*	−2.2*	
I	0.3	0	0.9	−0.7	0.4	0.5	1.5
H	0.55	−1.0	0.2*	−0.8*	−0.2*	−0.7*	−0.5
G	−1.1	−1.5	−1.0	−1.1	−1.3	−1.4	−1.0
F	−2.85	−3.5	−2.8	−0.7	−0.1	−0.7	−0.2
E	−0.1	−0.8	−1.0	−0.1	−1.0	−0.2	0.5
D	−1.2	−0.1	−0.7	0.7	1.5	1.6	2.1
C′	3.8*	3.9*	4.0*	4.3	4.9	4.8	5.4
B	5.4	4.75	3.9	5.4	6.5	6.4	6.8
A	3.5	3.8	4.2	5.0	5.7	7.1	7.5
f_s	−3.6	1.8	1.9	3.3	1.5	−1.0	0
f_c	−2.5	−0.65	3.0	3.1	1.4	1.2	2.5

* Obtained by Fairing.

$V^2=5$

MARK	DRAUGHT IN CM.						
	115	120	125	130	135	140	145
g_c	28.5	28.0	24.7	19.3	13.8	8.3	2.6
g_s	18.0	19.0	16.7	13.0	9.3	5.5	1.7
R	15.9	16.1	13.6	9.2	4.7	0	
Q	11.5	11.4	8.4	3.0			
P	8.7	8.0	4.4				
O	3.5	3.5	0.6				
N	5.0	4.2					
M	2.2	2.1					
L	0	0					
K	−4.0	−2.0					
J							
I	1.8	2.5					
H	0	0.5					
G	−0.5	0					
F	0.5	1.0					
E	1.0	1.3					
D	2.2	2.3					
C′	5.4	5.5	2.6				
B	6.8	7.0	4.0				
A	7.9	8.2	5.5	0.8			
f_s	0.8	2.3	1.0				
f_c	3.0	3.6	1.5				

Table IV, *Continued.*

$V^2=7$

MARK	DRAUGHT IN CM.							
	10	20	40	60	80	100	110	115
g_c	13.5	23.4	33.5	36.4	38.6	41.3	40.8	40.2
g_s	8.4	9.4	7.8	14.0	21.1	21.8	23.8	24.9
R	12.1	12.2	12.9	15.0	19.4	21.6	23.3	24.0
Q	6.3	7.05	9.6	12.5	14.6	17.2	18.5	19.1
P	3.4	4.75	7.2	9.4	12.2	14.0	14.2	13.8
O	1.3	0.55	2.7	4.9	7.0	6.2	5.4	4.6
N	−0.1	−2.15	−0.1	4.1	4.4	5.1	4.2	3.2
M	−1.0	−3.8	−2.9	−1.0	1.5	0.3	0.5	0.6
L	−2.35	−3.25	−3.8	−3.8	−2.5	−1.1	−1.6	−1.7
K	0	−5.15	−6.9	−5.5	−6.1	−3.6	−4.0	−4.7
J	0.7*	−6.85*	−5.0*	−3.8*	−3.1*	−2.8*		
I	−0.2	−0.7	0.4	−1.0	−0.2	0.5	1.0	1.5
H	−0.45	−1.9	−0.9*	−2.3*	−2.4*	−0.7*	1.0	1.7
G	−0.85	−1.35	−0.9	−1.3	−2.1	−2.7	−2.5	−2.2
F	−3.15	−4.1	−3.4	−0.5	−0.9	−2.0	−2.5	−2.5
E	0.7	−0.4	−0.6	0.5	−0.6	0.4	0.5	0.5
D	−0.1	1.0	−0.4	1.9	3.6	3.9	4.7	5.0
C′	5.5*	5.9*	6.9*	5.9	7.7	7.9	8.8	8.8
B	8.1	7.65	6.8	8.7	9.4	9.6	10.3	10.5
A	5.9	6.15	6.9	8.6	9.3	10.4	11.2	11.5
f_s	−5.45	3.25	3.3	5.3	3.8	−2.8	−1.3	0.2
f_c	−4.0	−0.85	4.7	4.9	2.5	0.2	1.3	2.4

* Obtained by Fairing.

$V^2=7$

MARK	DRAUGHT IN CM.							
	120	125	130	135	140	145	150	155
g_c	40.0	36.6	31.3	25.6	20.0	14.2	8.4	2.6
g_s	25.6	23.8	20.0	16.3	12.6	9.0	5.2	1.7
R	24.6	22.4	18.2	14.1	10.0	5.8	1.7	
Q	19.6	17.4	13.2	8.8	4.5	0.2		
P	13.5	10.0	4.5					
O	4.1	0						
N	2.5							
M	0.6							
L	−2.0							
K	−2.0							
J								
I	2.0							
H	2.5							
G	−1.8							
F	−2.0							
E	0							
D	5.4	2.6						
C′	9.0	6.1	1.5					
B	11.0	8.1	3.6					
A	11.8	9.2	4,8	0.4				
f_s	2.3	2.2	0.8	0.1				
f_c	4.2	3.5	1.5	0.2				

Table IV, *Continued.*

$V^2=9$

MARK	DRAUGHT IN CM.								
	10	20	40	60	80	100	110	115	120
g_c	18.8	31.9	44.8	48.8	51.5	52.5	51.3	51.0	50.4
g_s	12.8	14.6	12.2	19.6	28.4	29.9	31.8	32.7	33.5
R	16.75	16.9	17.6	20.7	25.7	28.9	30.3	31.0	31.9
Q	8.95	10.05	13.5	17.5	20.3	24.2	25.5	26.0	26.5
P	6.15	7.6	10.7	13.9	17.6	20.2	21.0	21.3	21.4
O	2.1	1.25	3.8	7.6	10.6	11.3	13.0	13.5	14.0
N	0.1	−2.55	−0.3	5.1	5.7	6.5	6.5	5.7	4.4
M	−1.8	−5.75	−5.2	−2.6	0.7	−1.4	−2.5	−3.0	−2.0
L	−3.5	−4.55	−6.2	−6.9	−5.1	−4.15	−5.3	−6.8	−2.0
K	−0.4	−7.5	−10.1	−9.3	−10.3	−8.0	−10.2	−7.0	−2.0
J	0.2*	−10.40*	−9.7*	−9.7*	−10.0*	−8.85*			
I	0.25	−0.8	0.8	−1.4	−0.1	−0.2	0.5	0.9	1.5
H	0.65	−0.85	0.4*	−0.7*	0.2*	0.7*	2.1	2.9	3.7
G	−1.5	−2.25	−1.3	−1.6	−2.9	−2.8	−1.5	−0.7	0.5
F	−4.7	−5.6	−5.1	−2.0	−3.6	−5.7	−5.8	−5.8	−2.0
E	−0.55	−1.75	−2.2	−1.9	−5.0	−4.1	−4.5	−5.2	−2.0
D	−1.7	−0.5	−3.0	0.4	1.8	0.8	1.5	1.5	1.8
C′	5.45*	5.9*	7.3*	5.6	8.1	8.3	9.8	10.0	10.5
B	9.0	8.5	7.3	9.6	11.8	12.4	13.0	13.2	13.5
A	6.2	6.5	7.6	9.5	11.0	13.1	14.0	14.5	15.0
f_s	−6.9	4.8	4.3	7.1	4.5	−6.7	−3.6	−1.0	2.5
f_c	−5.25	−0.75	6.2	6.5	2.5	−2.9	−1.3	1.0	4.0

* Obtained by Fairing.

$V^2=9$

MARK	DRAUGHT IN CM.							
	125	130	135	140	145	150	155	160
g_c	46.8	41.2	35.5	29.5	23.5	17.2	11.2	5.0
g_s	31.2	27.3	23.2	19.2	15.0	11.0	7.0	3.3
R	29.7	25.5	21.5	17.6	13.9	10.1	6.5	3.1
Q	24.2	19.7	15.2	10.7	6.1	1.6		
P	18.3	13.1	8.0	2.7				
O	11.5	7.1	2.7					
N								
M								
L								
K								
J								
I								
H								
G								
F								
E								
D								
C′	7.7	3.0						
B	11.0	6.4	2.0					
A	12.9	8.8	4.5	0.2				
f_s	2.8	1.7	0.8	0.2				
f_c	4.0	2.5	1.4	0.5				

FIG. 1

POSITION OF PRESSURE MEASURING HOLES

PRINCIPAL DIMENSIONS

LENGTH BETWEEN PERPENDICULARS	12.000 m
BREADTH EXTREME	2.984
DEPTH (From bottom of keel & upper deck side line)	1.750

REMARK :—
THE LINES REPRESENTS OUTSIDE OF SHELL PLATING

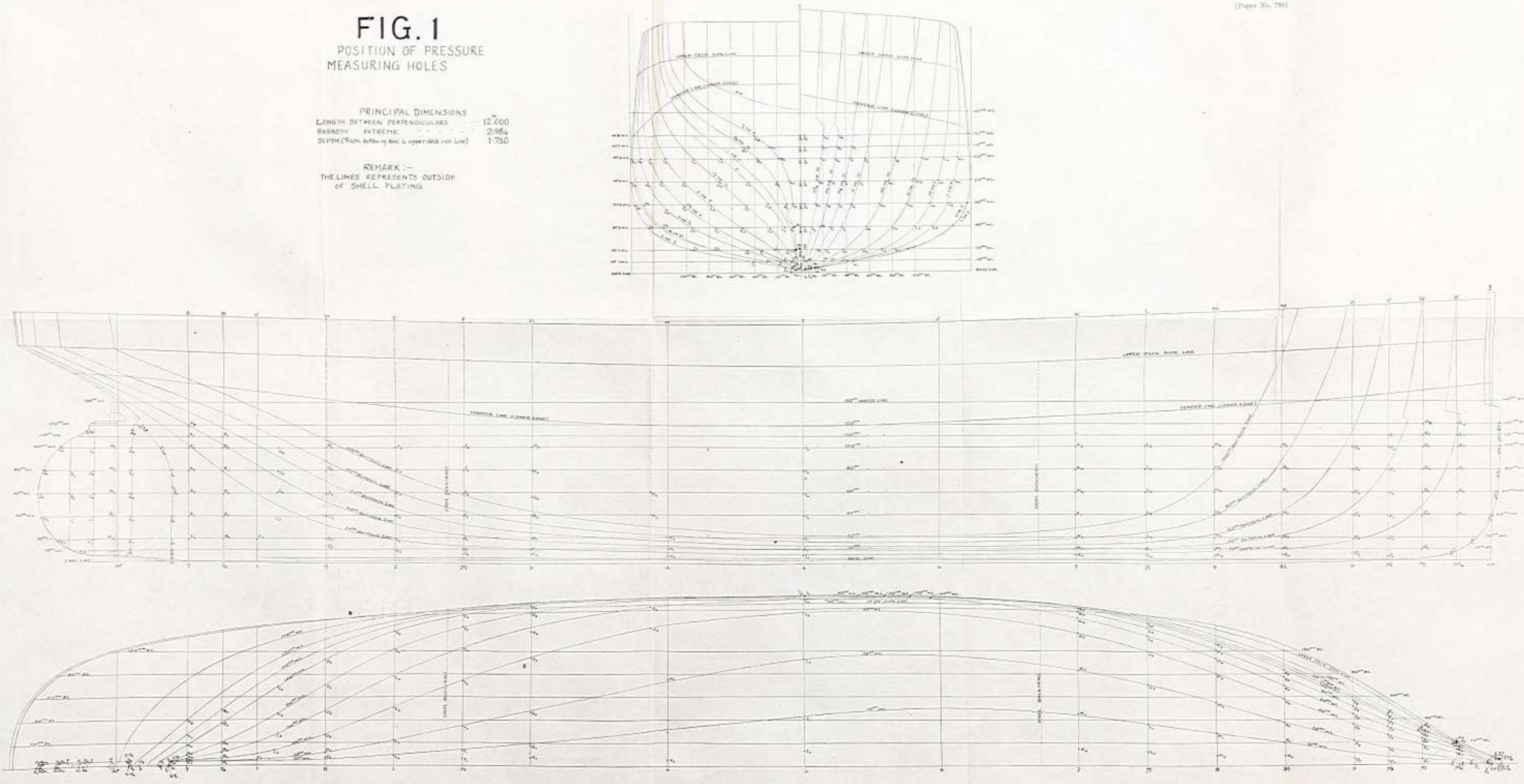


FIG. 3.

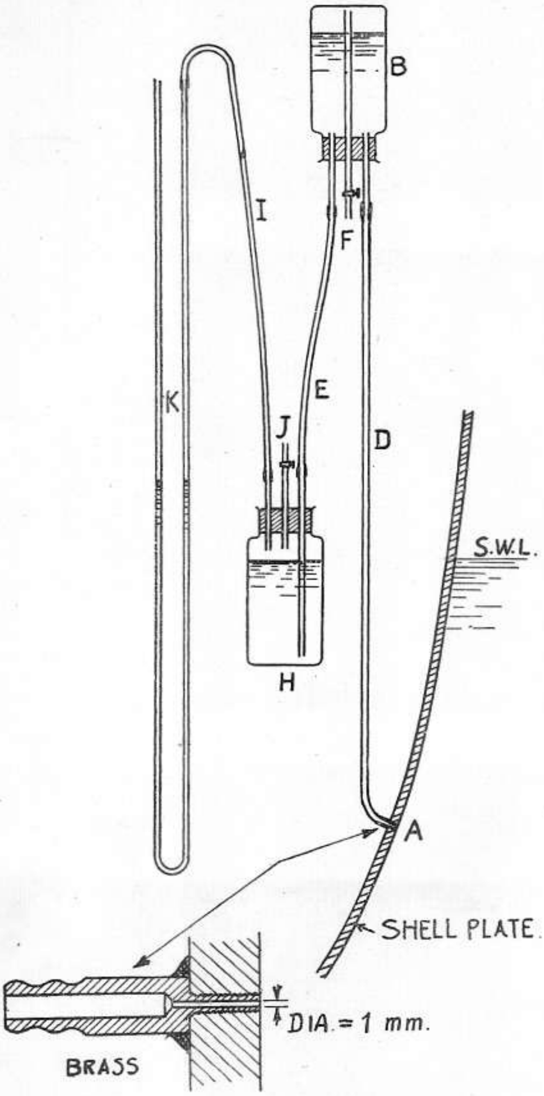

B
I
F
E
J
D
K
S.W.L.
H
A
SHELL PLATE.
BRASS
DIA. = 1 mm.

FIG. 2.

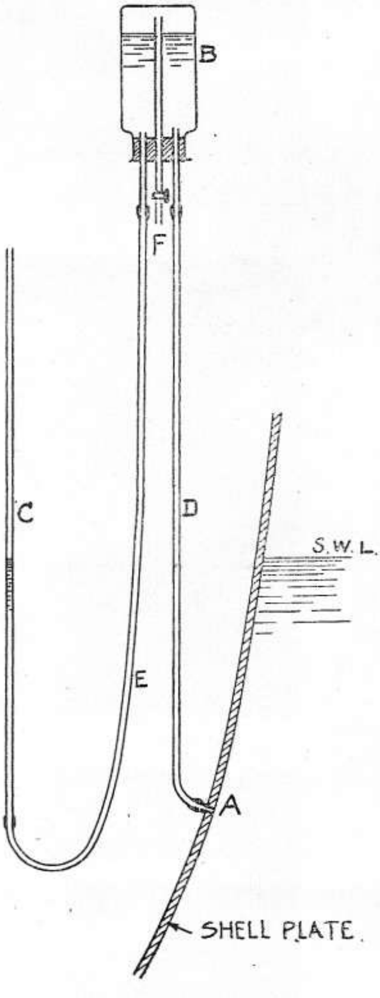

B
F
C
D
S.W.L.
E
A
SHELL PLATE.

Fig. 6

AERO PROPELLER

Experiment No. 26, 29th Dec.

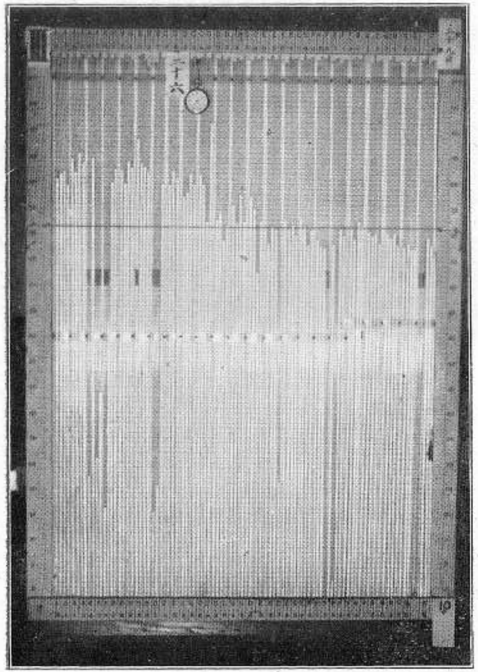


Fig. 5

MARINE PROPELLER

Experiment No. 11, 27th Dec.

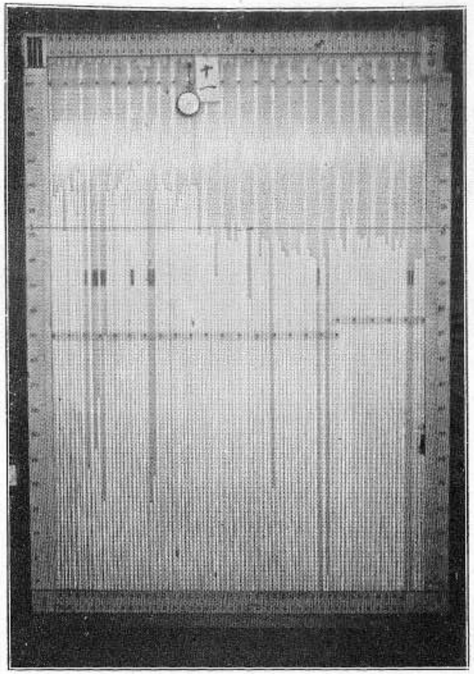
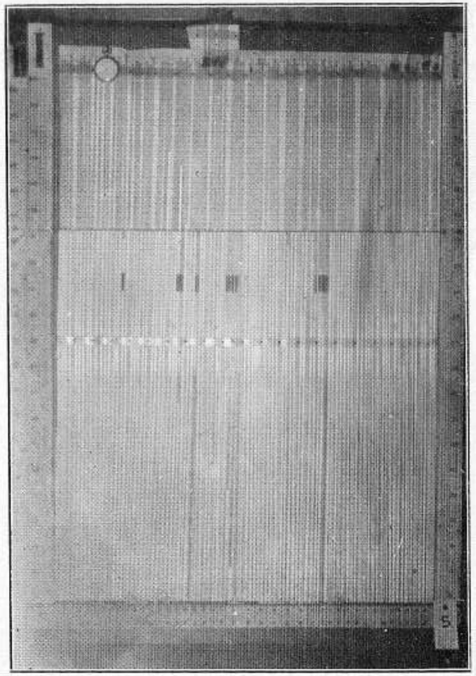


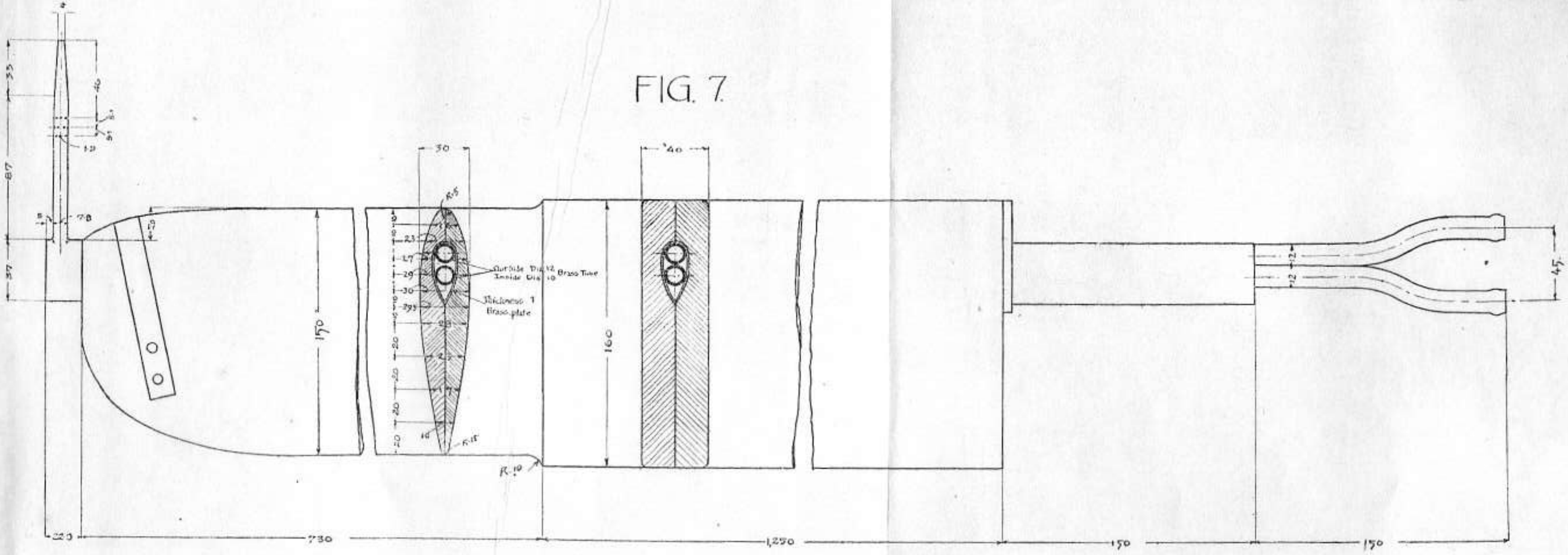
Fig. 4

Experiment No. 5, 29th Dec.



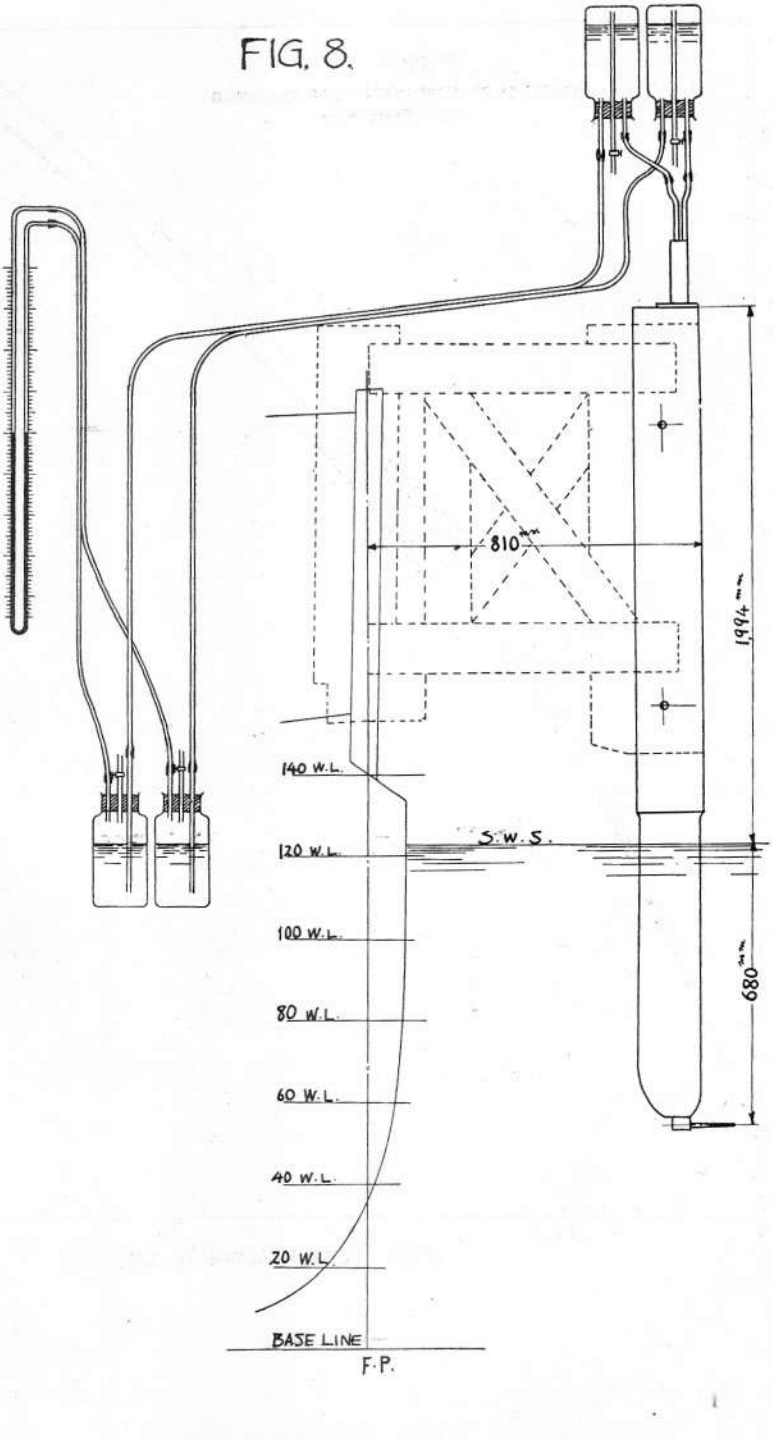
(Paper No. 789)

FIG. 7



PITOT TUBE.

FIG. 8.


810 mm
1994 mm
680 mm
S.W.S.
140 W.L.
120 W.L.
100 W.L.
80 W.L.
60 W.L.
40 W.L.
20 W.L.
BASE LINE
F.P.

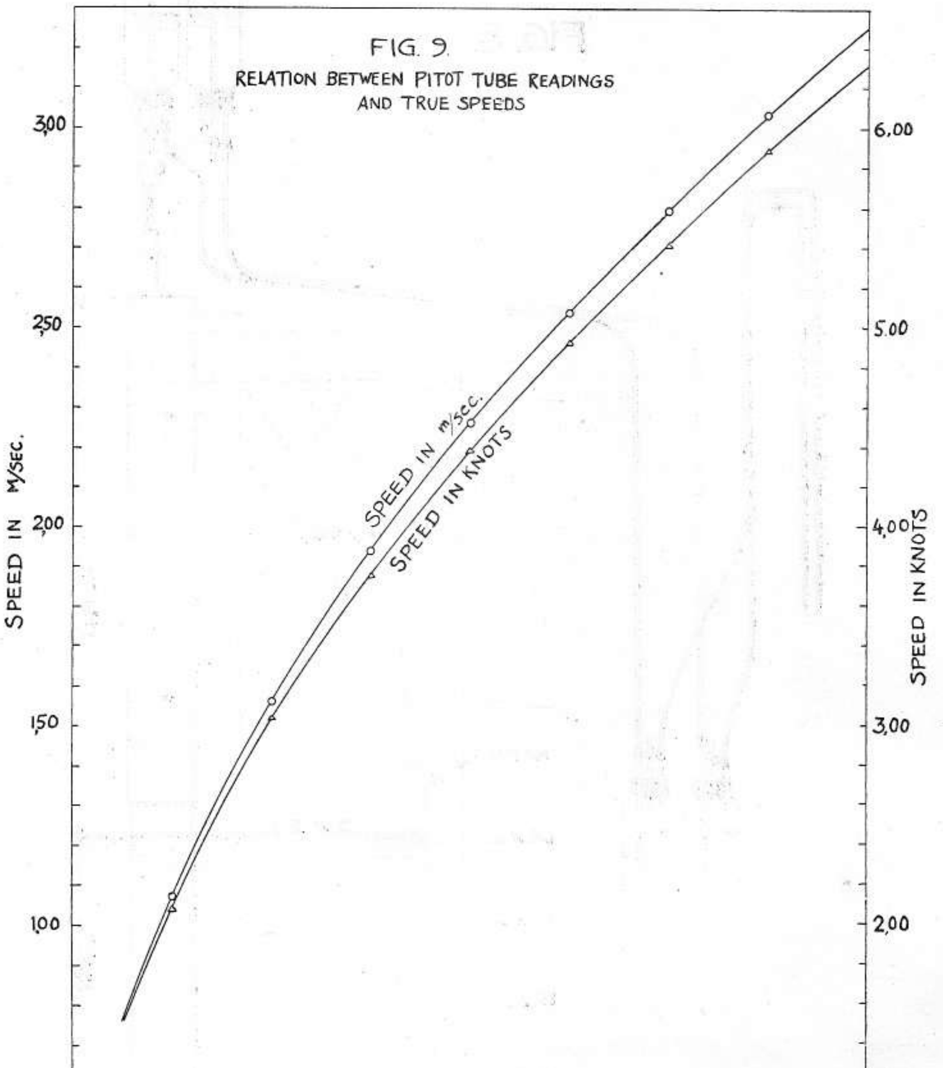
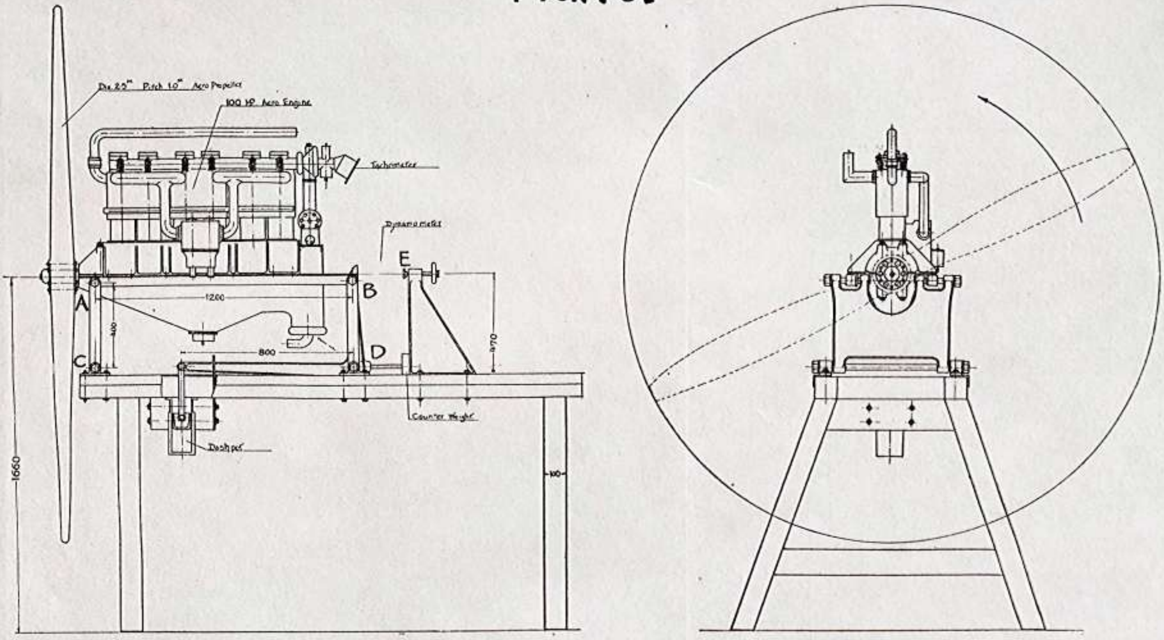
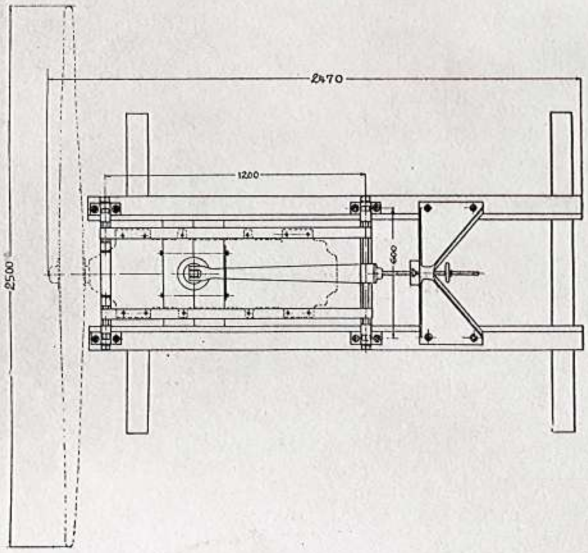


FIG. 9.
RELATION BETWEEN PITOT TUBE READINGS AND TRUE SPEEDS

FIG. 10.





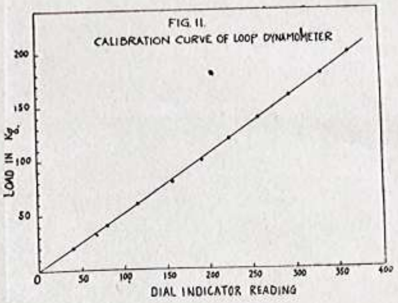
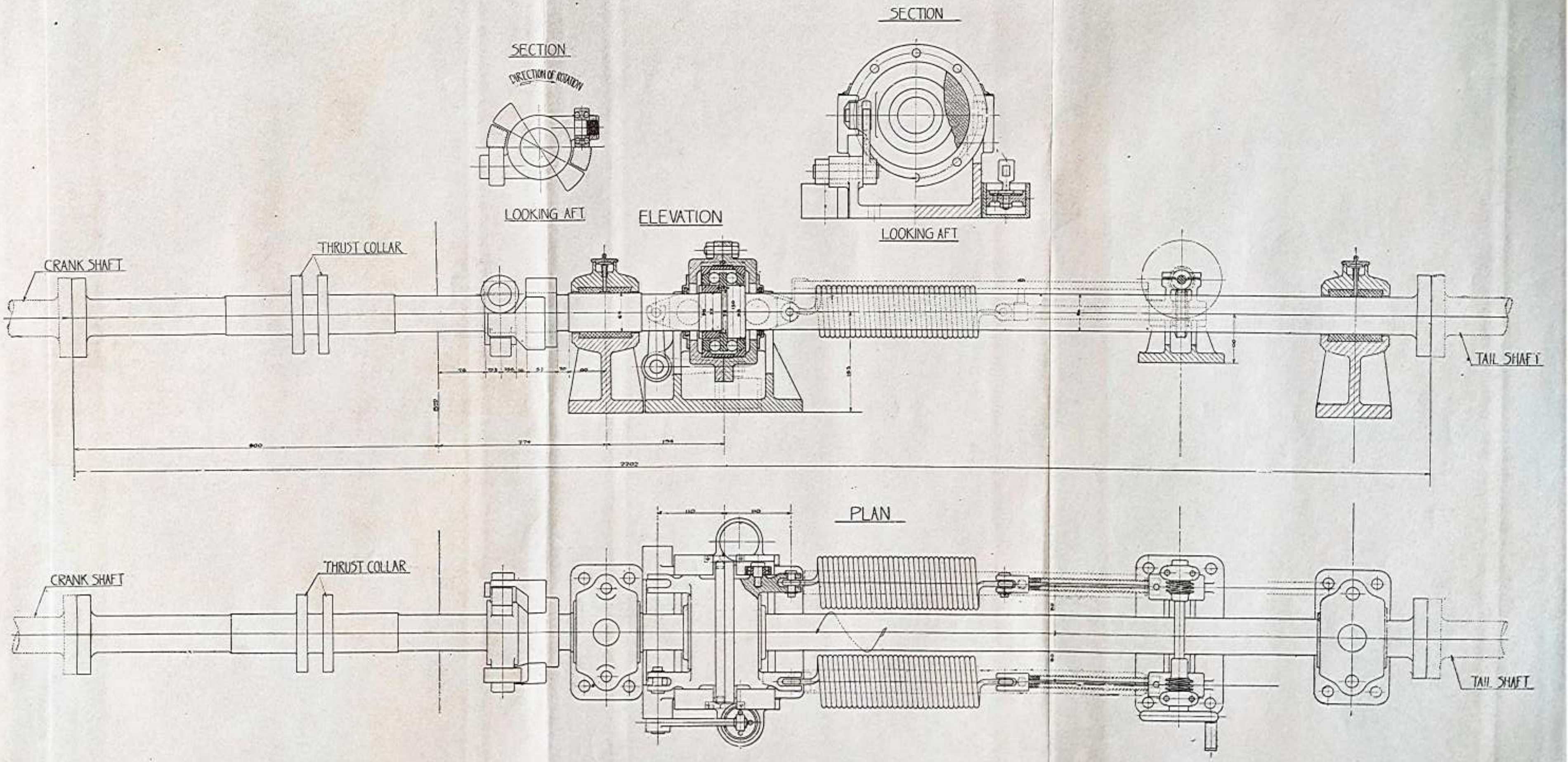


FIG. 11.

CALIBRATION CURVE OF LOOP DYNAMOMETER

FIG. 12. THRUST METER FOR MARINE PROPELLER.



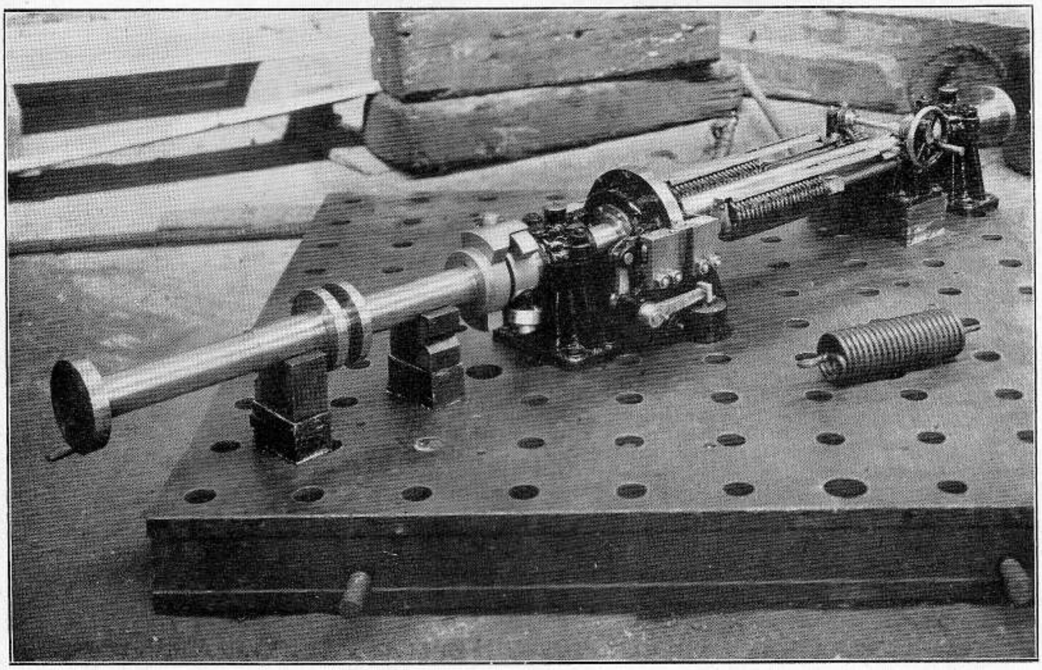


Fig. 13

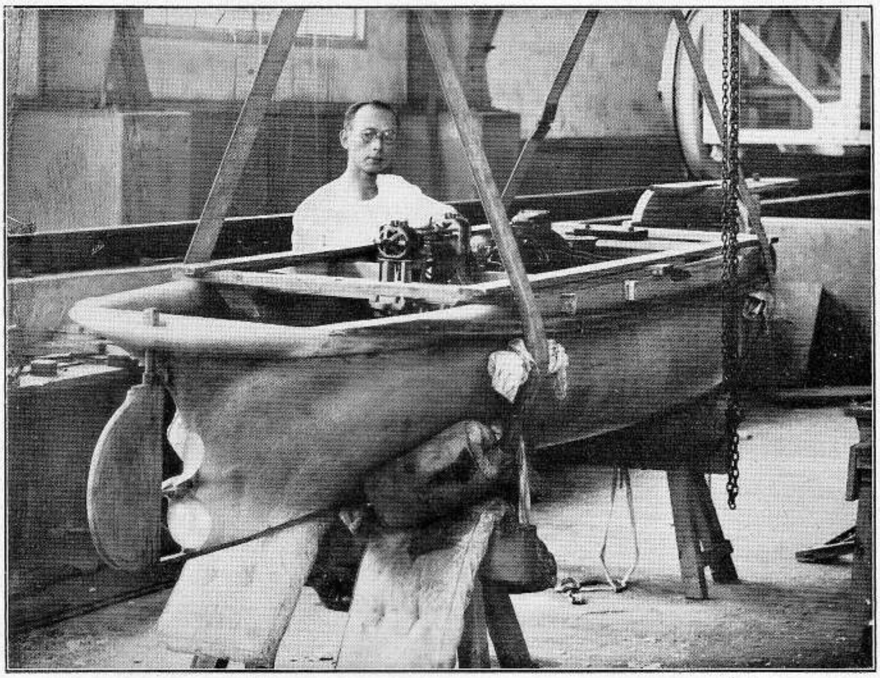


Fig. 17

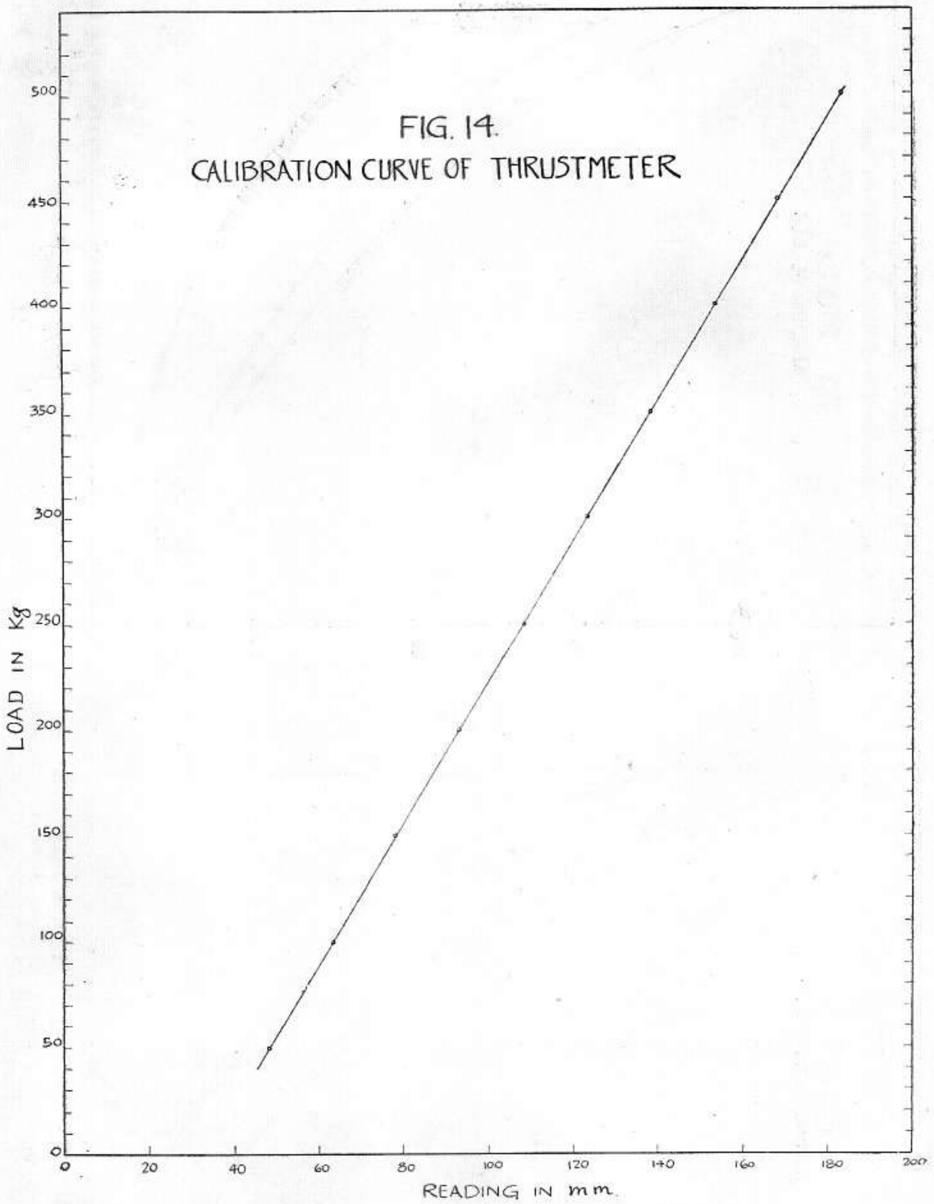


FIG. 14.
CALIBRATION CURVE OF THRUSTMETER
LOAD IN Kg
0
50
100
150
200
250
300
350
400
450
500
0
20
40
60
80
100
120
140
160
180
200
READING IN mm

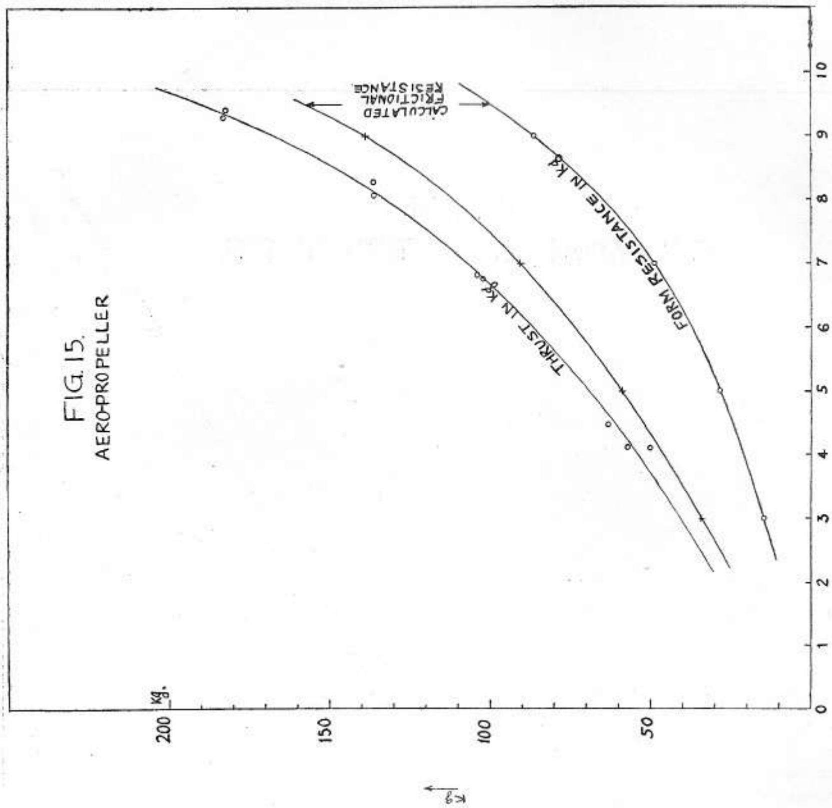


FIG. 15.
AERO-PROPELLER

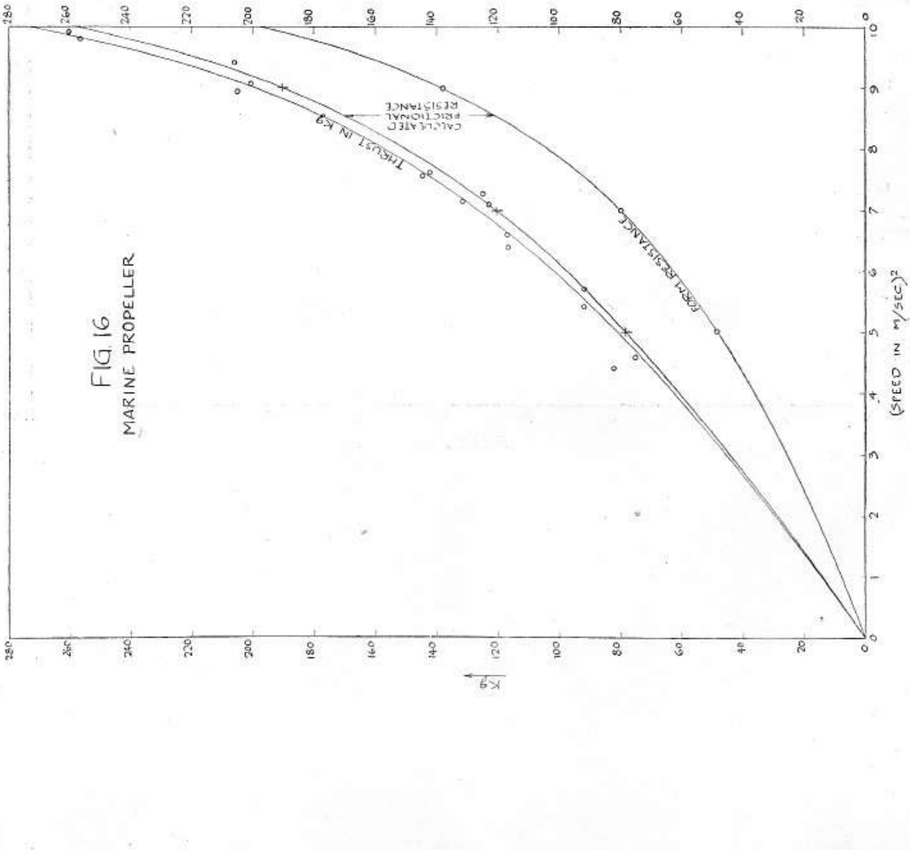


FIG. 16
MARINE PROPELLER

FIG. 18

RISE OR FALL OF WATER SURFACE RELATIVE TO THE WATER LINE Dm = .4067 M MARKED ON MODEL

UNIT CM

	SQUARE STATION / Vm IN M/sec	AP	½	¾	1	1½	2	2½	3	4	5	6	7	7½	8	8½	9	9¼	9½	9¾	F.P
			A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	
MARINE PROPELLER	1278	1.68	29	24	22	075	05	048	0.13	0.1	1.3	-068	-17	-009	168	1.9	1.2	22	355	60	85
	1516	48	45	40	38	20	0	-09	-065	12	092	-1.2	-17	-0.7	02	04	093	395	96	10.15	11.72
	1747	7.1	61	521	47	0.5	-1.8	-19	04	23	0.6	-251	-368	-246	-1.1	081	54	684	99	1425	140
AERO-PROPELLER	1020	124	21	1.38	165	084	0	-058	-083	-063	-053	-107	-177	-1.23	-07	065	165	204	144	248	569
	1309	329	3.15	283	247	118	052	-083	-193	-102	0	-208	-267	-1.13	0.11	06	063	213	378	680	850
	1535	525	452	393	342	154	0	-175	-198	-056	-053	-220	-298	-262	-075	-012	138	452	827	10.06	124
	1740	644	597	472	458	189	-235	-353	-178	148	0	-395	-468	-3.3	-1.62	-008	597	747	10.92	13.73	150

REMARKS:- —o— AERO-PROPELLER. —•— MARINE PROPELLER

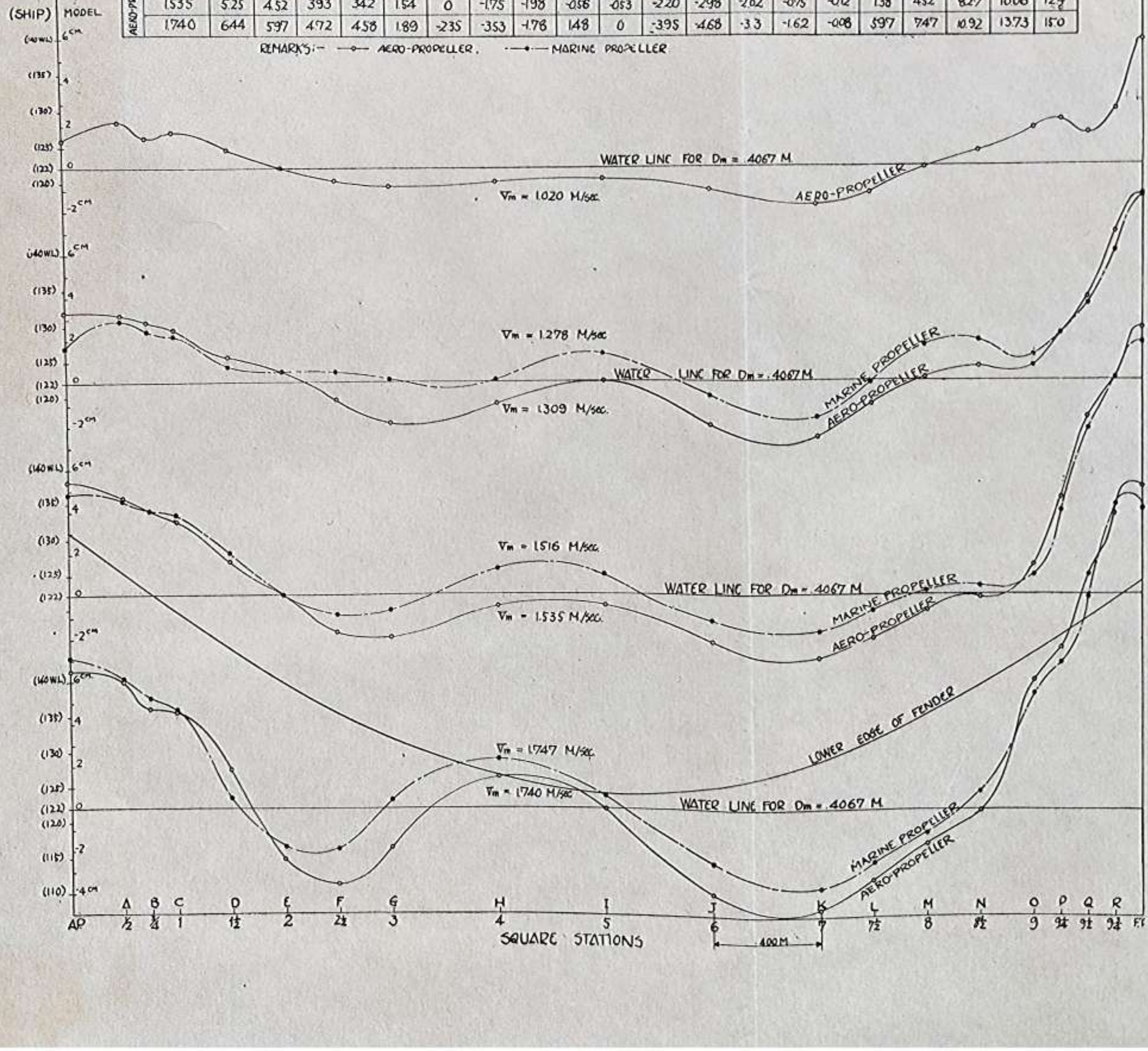
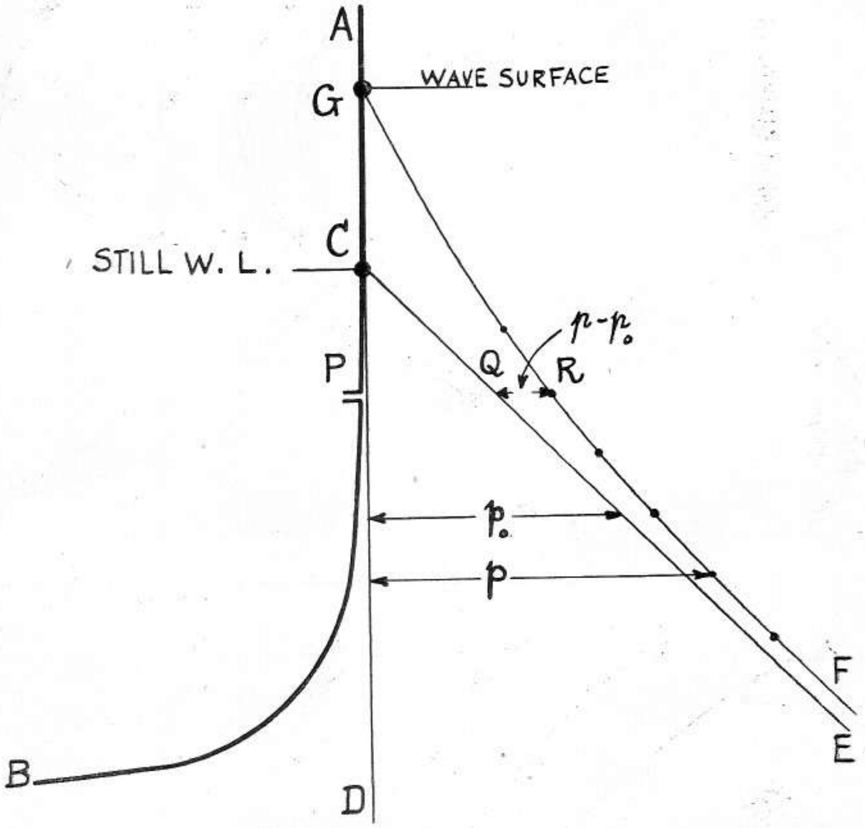


FIG. 19



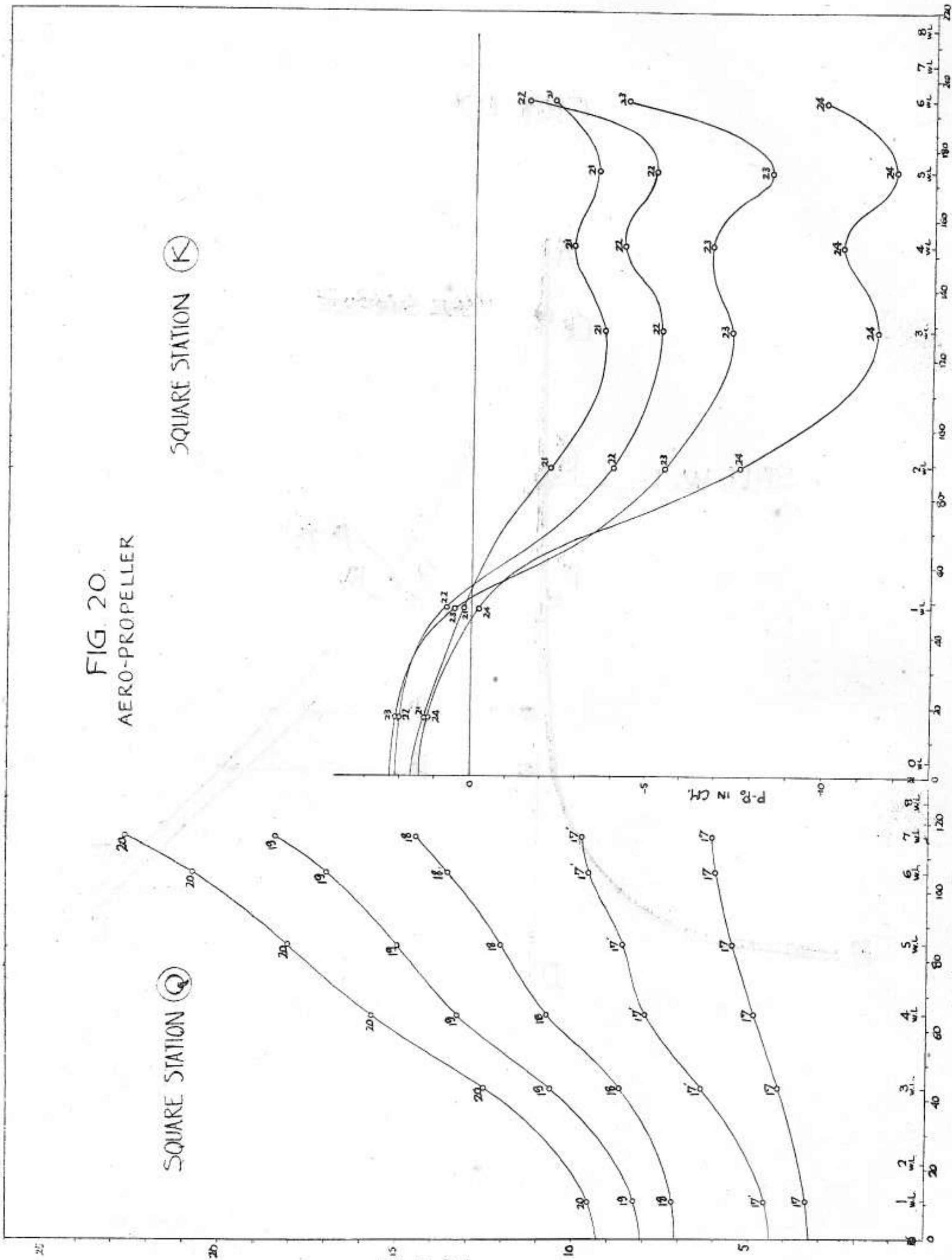


FIG 20
AERO-PROPELLER

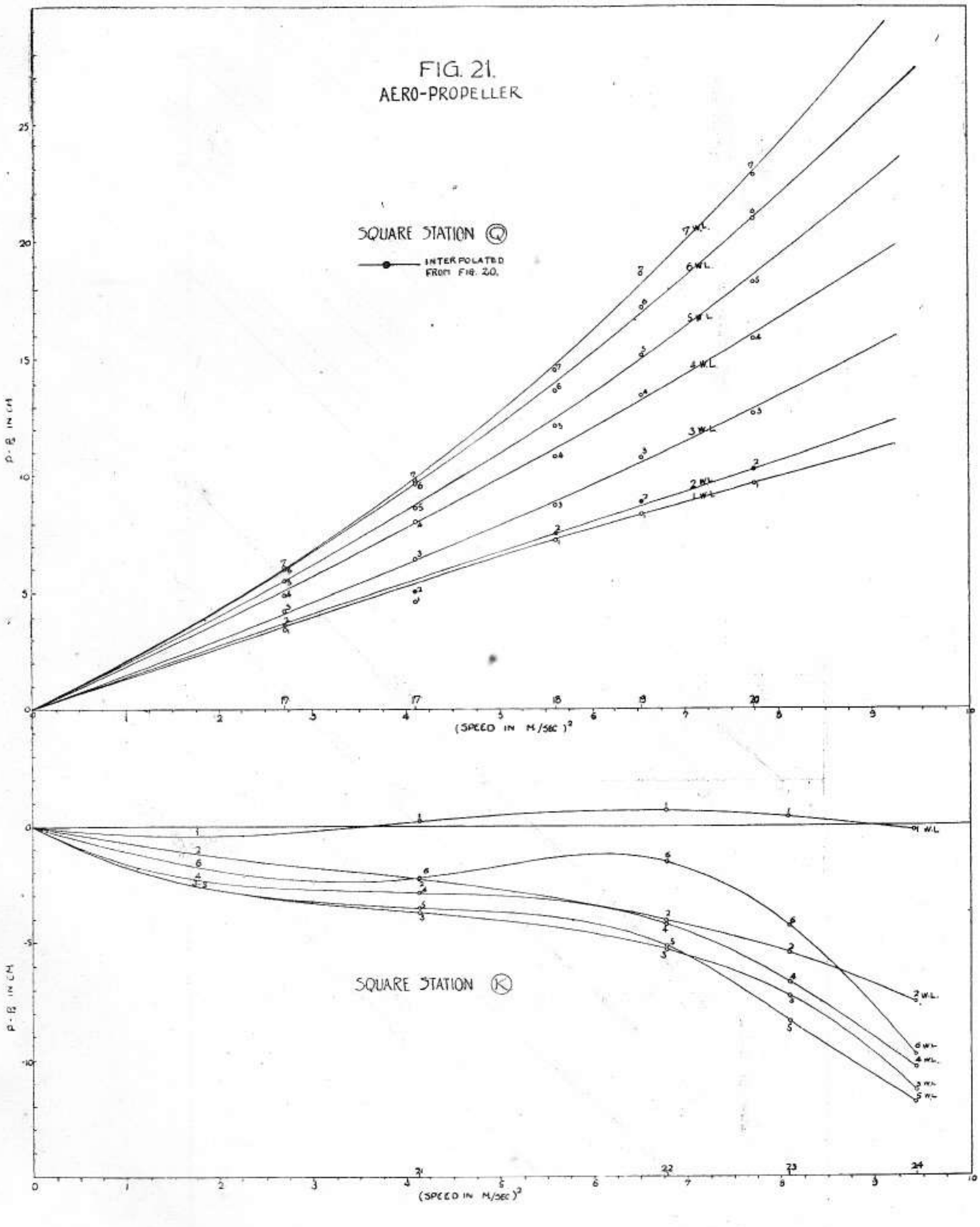


FIG. 21.
AERO-PROPELLER

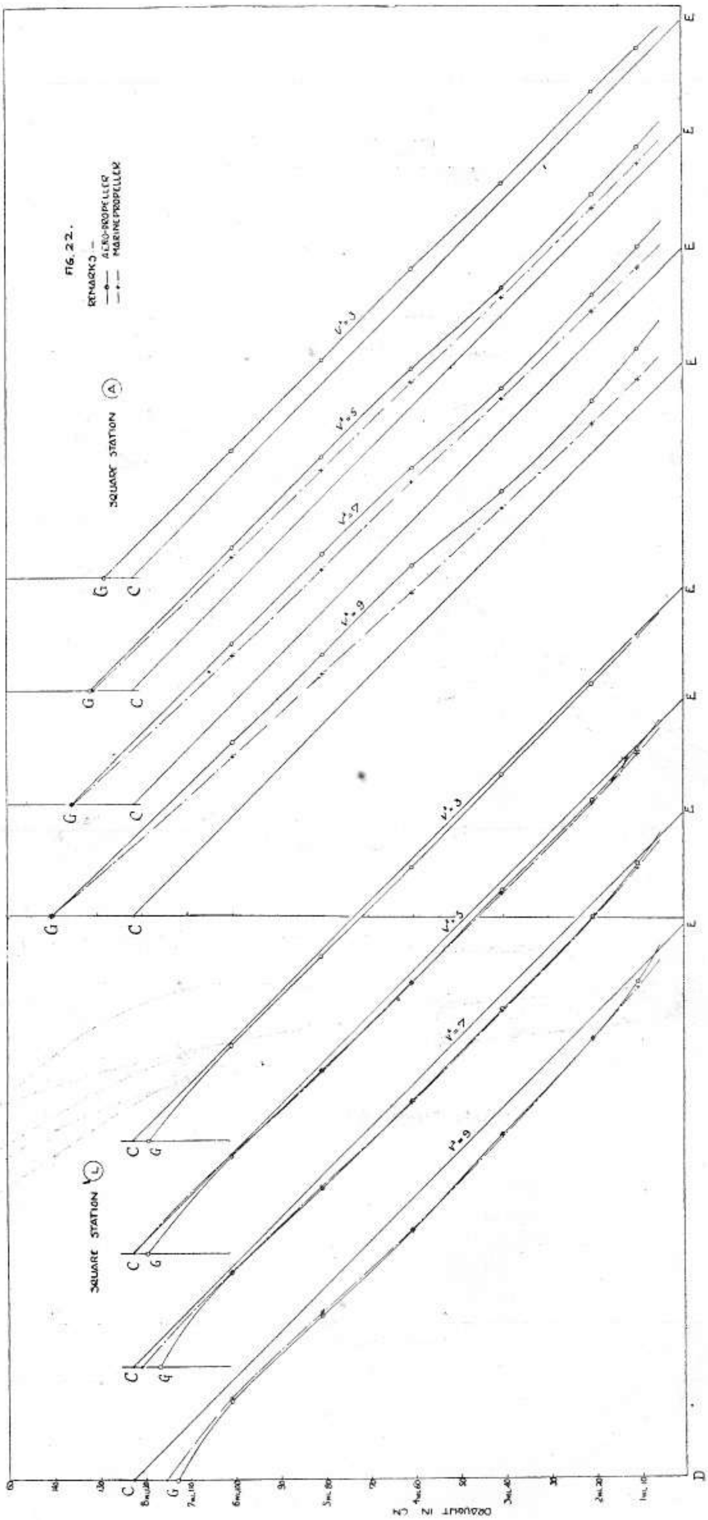


FIG. 22.

SQUARE STATION (A)
SQUARE STATION (L)
REMARKS –
AERO-PROPELLER
MARINE PROPELLER
V² = 3
V² = 5
V² = 7
V² = 9
G
C
E
D
DRAUGHT IN CM
150
140
130
8WL 120
7WL 110
6WL 100
90
5WL 80
70
4WL 60
50
3WL 40
30
2WL 20
1WL 10

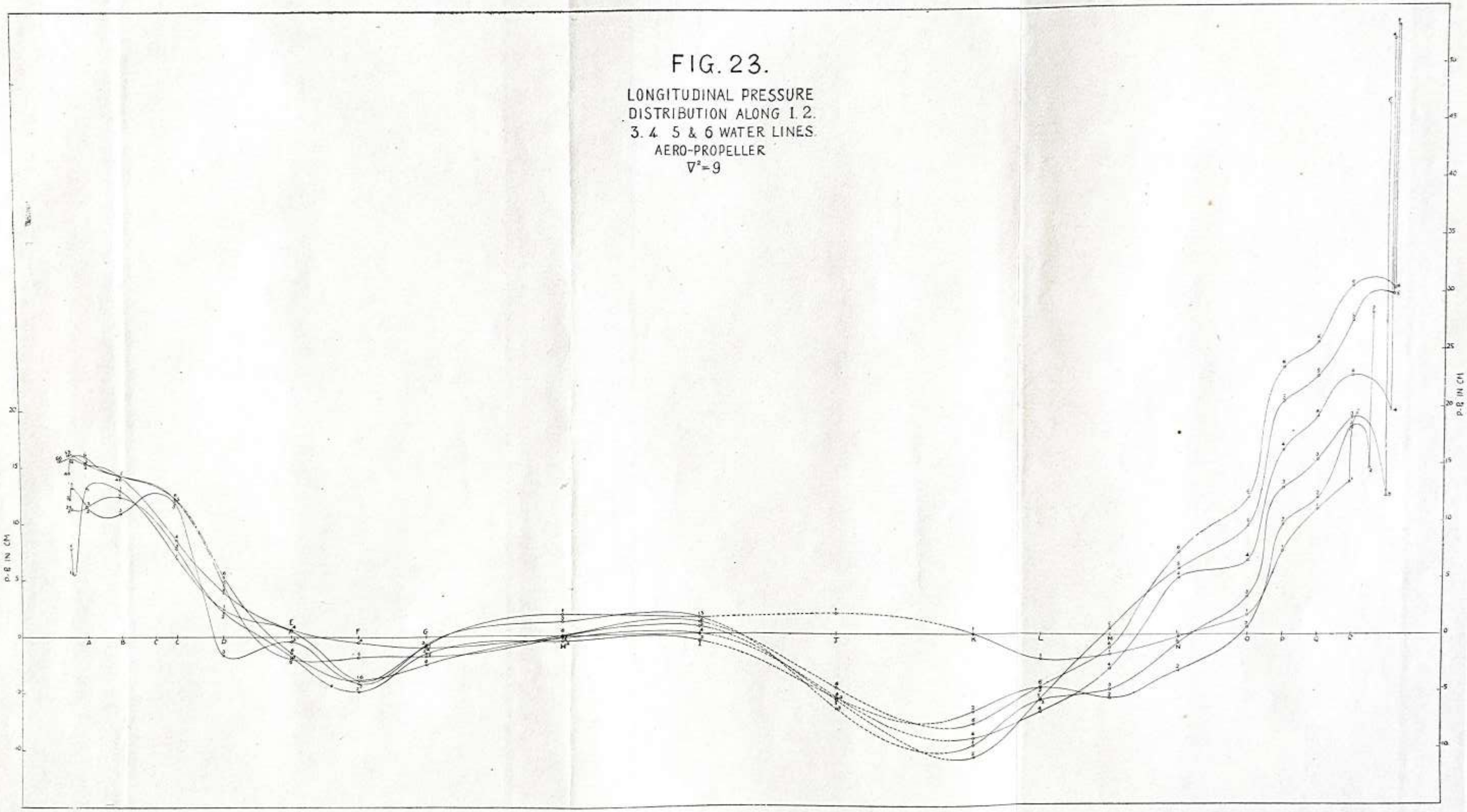


FIG. 23.

LONGITUDINAL PRESSURE
DISTRIBUTION ALONG 1. 2.
3. 4. 5 & 6 WATER LINES.
AERO-PROPELLER
$V^2 = 9$

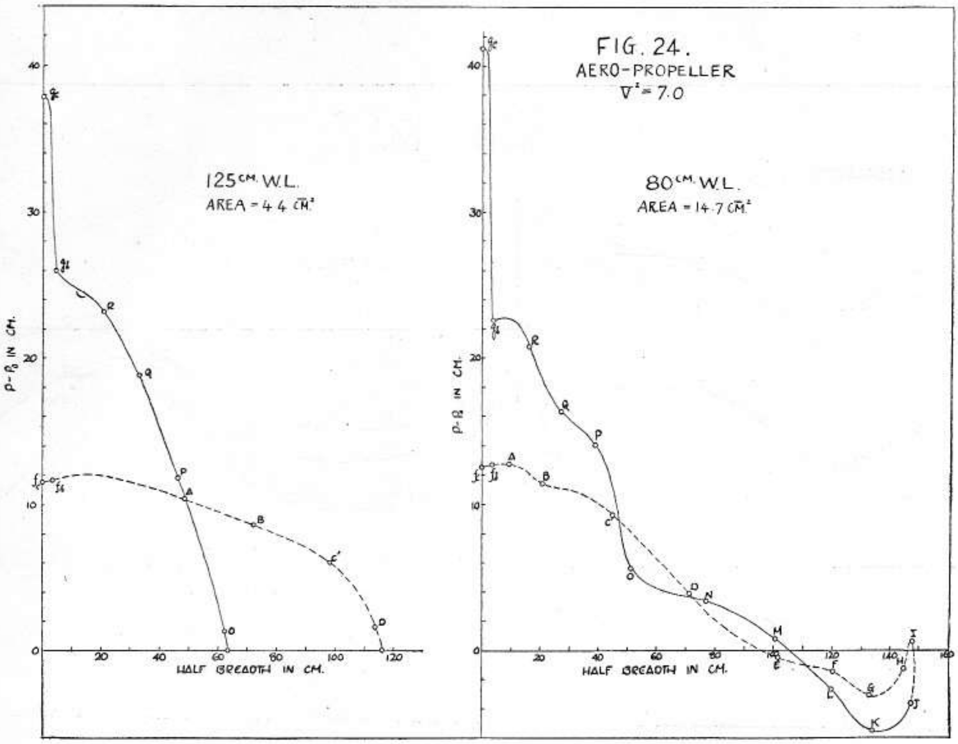


FIG. 24.

AERO-PROPELLER

$V^2 = 7.0$

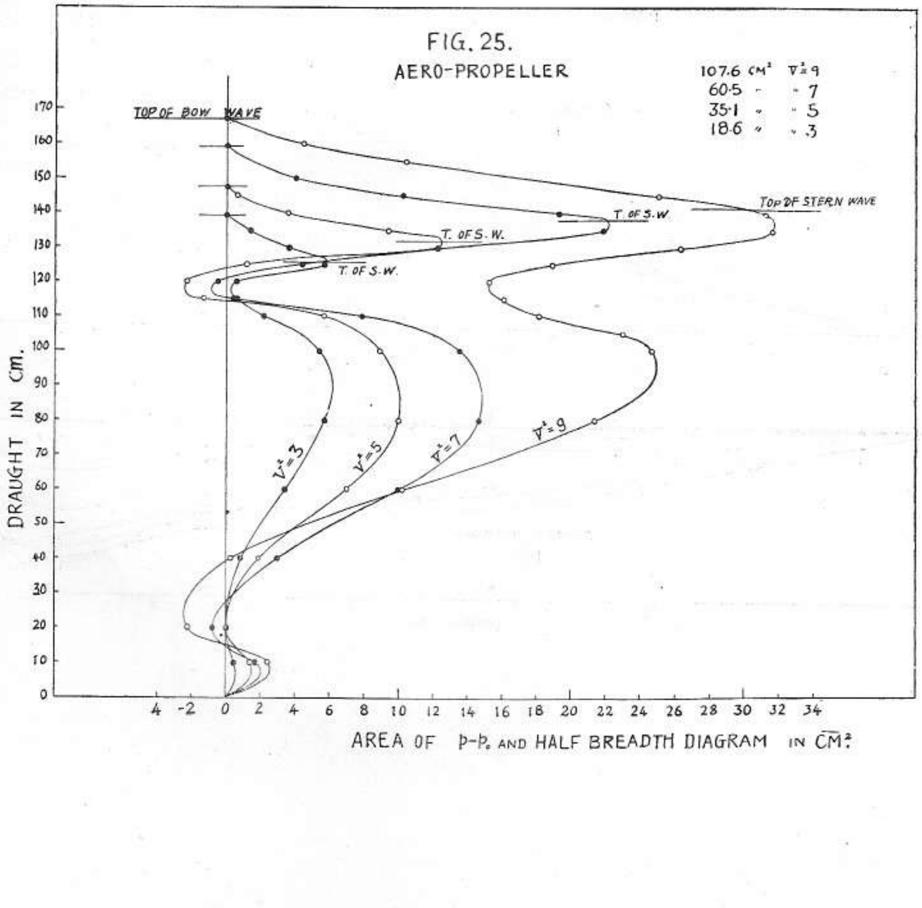


FIG. 25.

AERO-PROPELLER

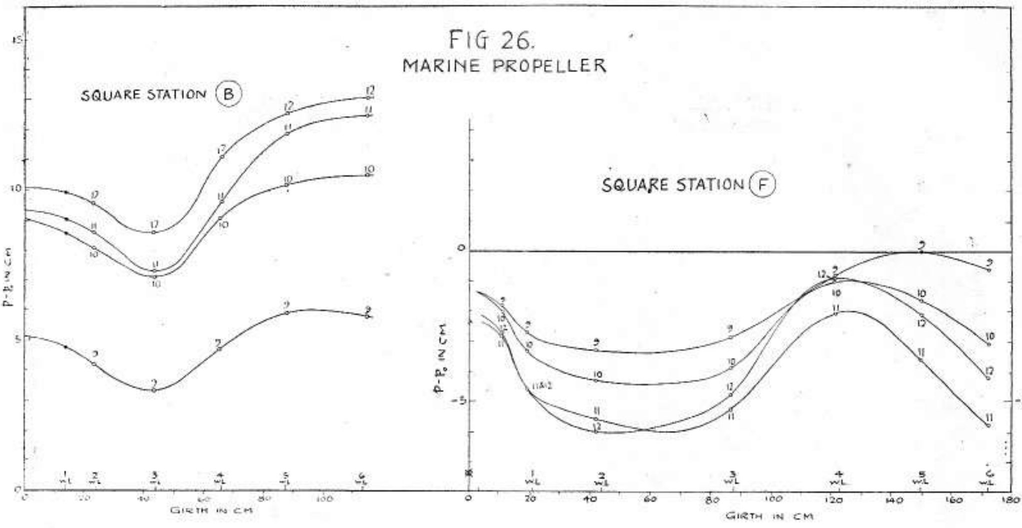


FIG 26.
MARINE PROPELLER

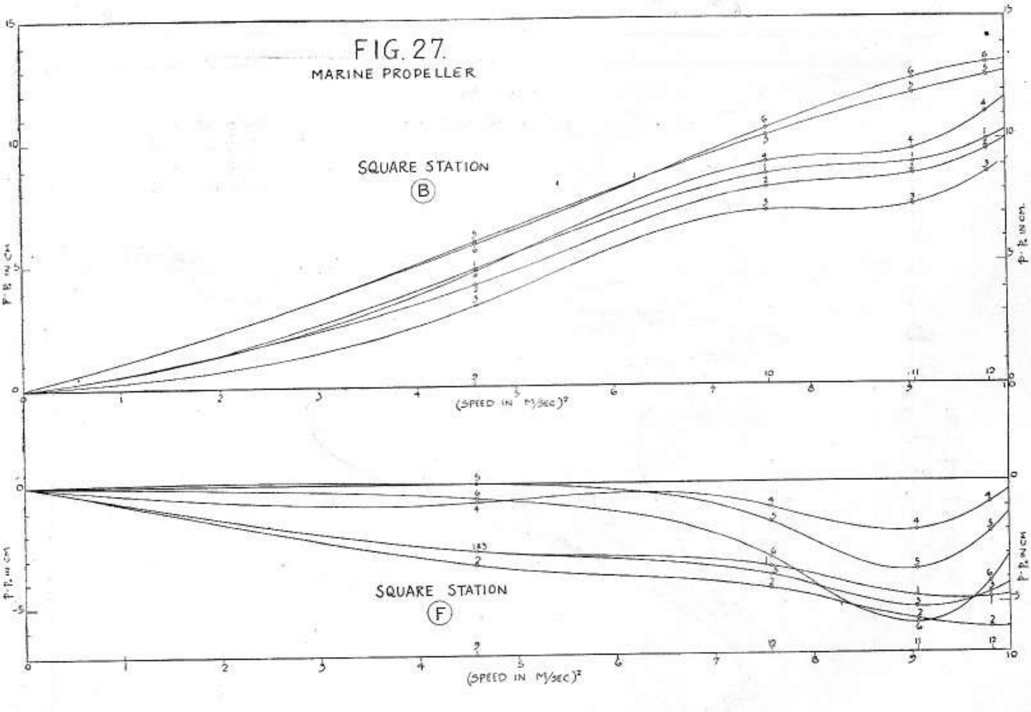


FIG. 27.
MARINE PROPELLER

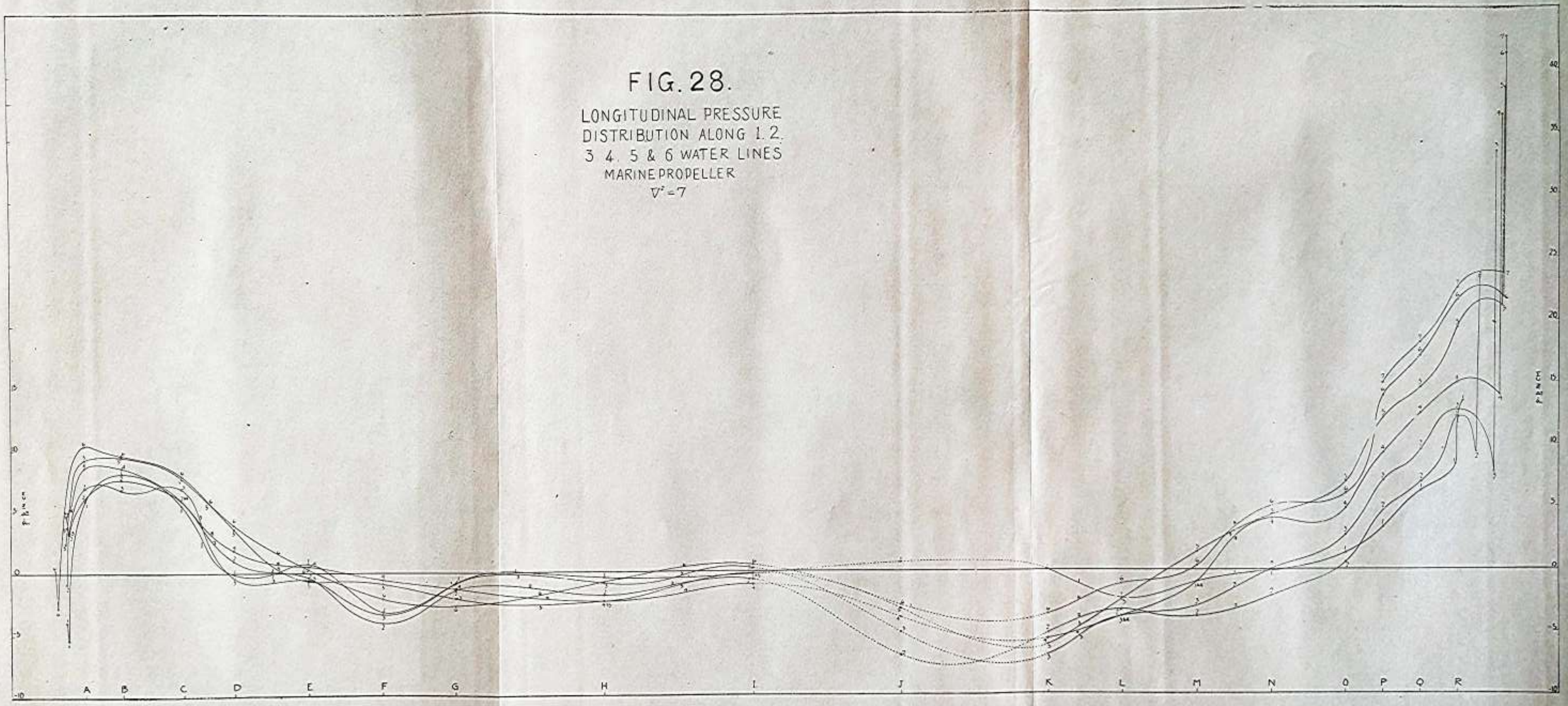


FIG. 28.

LONGITUDINAL PRESSURE
DISTRIBUTION ALONG 1. 2.
3 4. 5 & 6 WATER LINES
MARINE PROPELLER
$V^2 = 7$

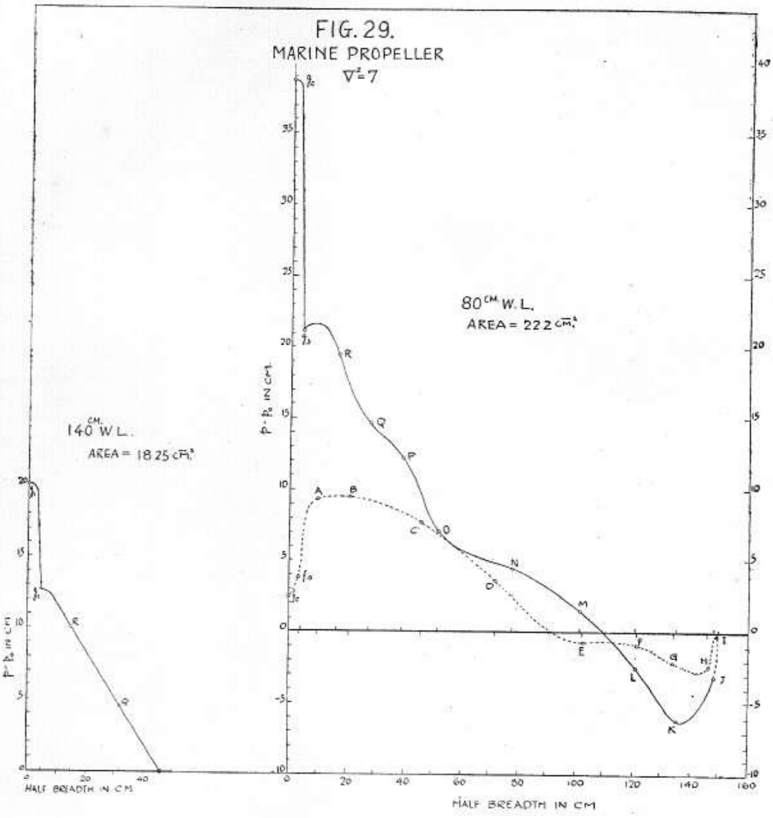


FIG. 29.
MARINE PROPELLER
$V^2 = 7$

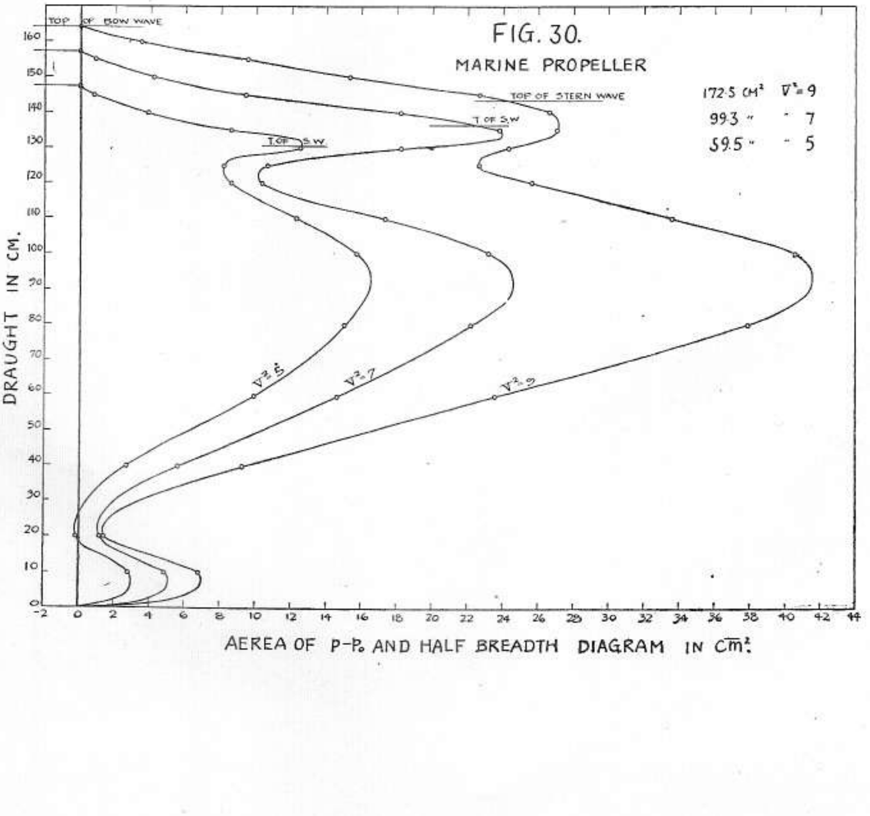


FIG. 30.
MARINE PROPELLER

The Torsional Stress in the Hull of a Ship.

(Paper No. 790)

By ***Tuneo Inokuty****, Kg., KH.,*

Professor of Naval Architecture in the
Tokyo Imperial University.

Introduction.

The problem on the torsional stress in a ship's hull has been already discussed by some authors[1] for ships of simple cross-sections such as single or two decked vessels without double bottom, however, so far as the author is aware, the method of computing the torsional stress in the structure of an ordinary ship with many decks and double bottom has not yet been given. The object of the present paper is to show the general method of calculating the torsional stress for any complicated cross-section, basing upon the fundamental theory of torsion.

The results of an example given for the illustration show that the torsional stress in a ship's hull under ordinary circumstance appears to be not a matter of much importance, however, we may expect a considerable stress-concentration at the corner of deck opening due to the discontinuity of the structure at that part. It is another object of this paper to show the existence of such stress-concentration in ship's structure.

1. Torsion of Thin Hollow Tube.

When a thin hollow tube is subjected to torsion, the variation of shearing stress on the cross-section across the thickness of the wall is very small, and we may assume it to be uniform and equal to the mean shearing stress q across the thickness of the wall. The direction of this stress is, of course, parallel to the tangent to the wall at that point. Let t be the thickness of the wall. It is shown in books on the theory of elasticity[2] that

$$qt = \text{const.} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

The ship's structures which are consisted of numerous walls in the form of decks, side plating, bottom and inner bottom plating may be regarded as a thin hollow tube having several hollow parts in the cross-section,

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[1] Vedeler, On the Torsion of Ships, T.I.N.A., 1924.
Dr. Sezawa, On the Torsion of Ships among Oblique Waves, Graduation Essay, Depart. of N.A., Tokyo Imp. Univ., 1921. Unpublished.
Dr. Dahlmann, Festigkeit der Schiffe, pp. 119–124.
Prof. Lienau, Versuchseinrichtungen und Ergebnisse des Institutes für Schiffsfestigkeit an der Technischen Hochschule Danzig, Jahrbuch d. Schiffbautechnischen Gesellschaft, 1928.

[2] Prescott, Applied Elasticity, p. 165.

and this tube may be considered as one having the cross-section surrounded by n number of closed boundaries and forming $n-1$ number of hollow parts. For instance, in Fig. 1, there are three closed boundaries and two hollow parts, and the cross-section is consisted of three walls, EABC, EC, and CDE. In such case, the above expression (1) holds good for each wall. Take a transverse ring of a very short length δx, and consider a portion where three walls meet at one point cut by longitudinal planes normal to respective walls at any point in each wall. Fig. 2 represents the portion thus obtained. If, now, the shearing stresses and thicknesses of the walls CP, CQ and CR are q_1, q_2, q_3, t_1, t_2, t_3, there are longitudinal shearing stresses of equal intensity q_1, q_2, and q_3 on the section PP′, QQ′ and RR′ respectively. For the equilibrium of this portion, the algebraic sum of these longitudinal stresses must be zero, i.e.

$$q_1t_1\,\delta x+q_2t_2\,\delta x-q_3t_3\,\delta x=0,$$

or

$$q_1t_1+q_2t_2-q_3t_3=0. \qquad\qquad (2)$$

This relation holds good for all three walls meeting at other points. When the number of closed boundaries is n, the number of points where three walls are meeting is $2(n-2)$. Therefore we have $2(n-2)$ equations of the type (2) in all. But one of these equations can be obtained by combining the others. Hence we have $2n-5$ independent equations of the type (2).

Our next step is to find the relation between the shearing stress and the twist of the tube. Take the axes of x and y in the plane of the cross-section, and let ϕ be the torsion function, and ψ a conjugate function such that $\phi+i\psi$ is a function of $x+iy$. Then the shearing stress and the lines of shearing stress are given by the following equations.[3]

If we put

$$\Psi=\psi-\tfrac{1}{2}(x^2+y^2), \qquad\qquad (3)$$

the curves $\Psi=$const. is the lines of shearing stress. The shearing stress q is given by

$$q=-G\tau\frac{\partial\Psi}{\partial\nu},$$

where $d\nu$ is the element of the normal to the curve, τ the twist, and G the modulus of rigidity.

Substituting (3) in the above equation, we have

$$q=-G\tau\Big[\frac{\partial\psi}{\partial\nu}-\{x\cos(x,\nu)+y\cos(y,\nu)\}\Big].$$

Therefore we have

$$q=G\tau\frac{\partial\phi}{\partial s}+G\tau p, \qquad\qquad (4)$$

where ds is the element of the lines of shearing stress, and p the perpen-

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[3] See Love, Mathematical Theory of Elasticity, Chapter XIV.

dicular distance from the origin to the tangent to the curve.

The axial displacement is given by

$$w=\tau\phi.$$

Therefore, the equation (4) becomes

$$q=G\frac{\partial w}{\partial s}+G\tau p. \qquad \text{(5)}$$

Now it is obvious that

$$\int\frac{\partial w}{\partial s}\,ds=0,$$

the integration being taken round any closed curve in the cross-section. Hence, if we integrate both sides of the equation (5) round any closed line of shearing stress, we have

$$\int q\,ds=2G\tau A, \qquad \text{(6)}$$

where A represents the area bounded by this closed curve.

Taking any inner boundary, the shearing stresses and thicknesses of the walls, which have this boundary as their side, are represented by

$$q_r,\ q_s,\ \ldots,\ t_r,\ t_s,\ \ldots\ .$$

As the boundary is one of the lines of shearing stress, the relation (6) holds good for this boundary. Changing the left-hand side of (6) by integrating for each wall, we have

$$q_r\,t_r\int\frac{ds}{t_r}+q_s\,t_s\int\frac{ds}{t_s}+\ldots=2G\tau A_{rs\ldots}, \qquad \text{(7)}$$

where $A_{rs\ldots}$ is the area enclosed by this boundary. In the above equation, the sign of shearing stress must be taken into account, and, when the direction of the shearing stress in any wall is opposite to the direction of integration round the boundary, the minus sign shall be put before the term for that wall. If this integration is made for all inner boundaries, we shall have $n-1$ equations of the type (7). These equations are independent to each other, and another equation given by integrating round the outer boundary can be obtained if we combine the equations for the inner boundaries.

The last equation we should have is one for the relation between the torque Q acting on the tube and the shearing stress. If, for each wall, the perpendicular distances from the axis of twist to the tangent of the wall are represented by $p_1, p_2, \ldots$, then, the moment of shearing stress on the element $t\,ds$ being $p\,q\,t\,ds$, we shall have by integrating for all walls

$$=q_1t_1\int p_1\,ds+q_2\,t_2\int p_2\,ds+\ \ldots \qquad \text{(8)}$$

Using the relations of the type (2), the above equation can be transformed into the form having the terms of some of q_1t_1, $q_2t_2, \ldots\ldots$, the coefficients of which are the areas enclosed by some boundaries.

We have now $2n-5$ equations of the type (2), $n-1$ equations of the type (7), and the transformed equation of (8). Therefore there are, in all, $3n-5$ equations, which contain $3n-4$ number of quantities, q_1t_1, q_2t_2, $\ldots\ldots q_{3(n-2)}\,t_{3(n-2)}$, Q, and τ, the number of walls being $3(n-2)$. Hence we have obtained necessary and sufficient simultaneous equations of the 1st degree for solving the above quantities by making one of them known and the remainder unknown.

So far we have treated the cross-section in which not more than three walls are meeting at one point. If there are any points where more than three walls are meeting, the number of the equations of the type (2) will be reduced, but, at the same time, the number of walls will also be reduced by the same number. Therefore we have still necessary and sufficient equations left for solving the required quantities.

2. Calculation of Torsional Stress in Ship's Hull.

Applying the above method, the torsional stress was calculated for an ordinary cargo ship of the following dimensions, having double bottom and three decks including bridge deck, the midship section of which is shown in Fig. 3.

Length between perpendiculars	400′–0″
Breadth moulded	54′–6″
Depth moulded	30′–0″

Vedeler[4] suggests that the static torque acting on a ship shall be proportional to $L\cdot B^3$, where L and B are the length and breadth of the ship. From the results of calculation of static torque made by Vedeler[5] and Dr. Sezawa,[6] the ratio of Q to $L\cdot B^3/35$ was calculated, the results of which are given in Table I.

TABLE I.

Authors	L × B × D	Q in ft-tons	$\frac{Q}{\frac{L\cdot B^3}{35}} \times 10^2$
Vedeler	360′–0″ × 50′–0″ × 26′–1″	3,075	.239
Sezawa	345′–0″ × 50′–0″ × 29′–1″	3,420	.278
”	400′–0″ × 54′–6″ × 30′–0″	6,190	.334
”	445′–0″ × 58′–0″ × 34′–0″	8,290	.334
”	403′–0″ × 53′–0″ × 31′–0″	5,780	.337

In the present paper, it is assumed that the static torque is 6,190 ft-tons. In addition to static torque, there will be dynamic torque due to the oscillation of the ship, but at present the stresses due to the static torque only were calculated. The results obtained are given in Table II, and are also shown graphically in Fig. 3, in which the intensity of stress is drawn perpendicularly to each member on one side, irrespective of the sign of stresses.

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[4] and [5] Vedeler, loc. cit.

[6] Dr. Sezawa, loc. cit.

TABLE II.

	qt in ton/in.	Thickness of Thinnest Plate in inch	Shearing Stress in ton/□″
Bridge deck plating and bridge side plating	.0849	.40 (Br. dk. plating)	.212
Upper deck plating	.0427	.36	.119
Tween deck side plating	.1275	.70	.182
2nd deck plating	.0142	.36	.039
Side plating below 2nd deck	.1420	.70	.203
Bottom plating	.1042	.70	.149
Inner bottom plating	.0378	.44	.086

At the centre girder, the hull being symmetrical on both sides, the shearing stress becomes zero. As will be seen in the above table, the greatest stress is at the bridge deck, and the smallest at the second deck.

The shearing stresses on the seam riveting due to the above torsional stresses were computed, but they did not amount to anything, even at the bridge deck plating was a little less than 1 ton/□″.

3. Discontinuity of Structure at Corner of Deck Opening.

Judging from the above results, the torsional stress in the ship's hull under ordinary circumstance appears to be not a matter of much importance, however there is one part which requires some considerations, that is the discontinuity of structure at the corner of the deck opening. The part of deck opening may be regarded as an open tube and other part as closed tube.

The shearing stress q and the twist τ of a thin hollow circular tube having uniform thickness t and radius r, when subjected to the torque Q, are given by

$$q=\frac{Q}{2\pi r^2 t},$$

$$\tau=\frac{Q}{2\pi G r^3 t}.$$

And, with a tube of the same radius and thickness having a split cut parallel to the axis, the shearing stress q' and the twist τ' for the same torque[7] are represented by

$$q'=\frac{3Q}{2\pi r t^2},$$

$$\tau'=\frac{3Q}{2\pi G r t^3}.$$

Therefore the ratio of the stress in the split tube to that in the closed tube is $3(r/t)$ for the same torque, and the ratio of the twist is $3(r/t)^2$. Thus the split tube is much weaker than the closed tube, and very much less rigid, and we may anticipate a considerable stress-concentration at the

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[7] Prescott, Applied Elasticity, p. 171.

corner of the deck opening making the question somewhat important from the strength point of view in ship's structure.

Further the same question may bear an important effect on aircraft construction in which the study of torsion is of greatest importance in design when an opening is cut on the metallic fuselage.

4. Model Experiments.

The foregoing theory holds good provided the cross-section of the tube has no sharp bent of the wall and no sudden change in the thickness of the wall. In the torsion of a solid shaft, if the boundary of the cross-section has a sharp re-entrant, the shearing stress near the corner is very great, and, if the boundary has a sharp projection, the shearing stress is zero at such projection. The same effect will be observed in the torsion of a thin tube having discontinuities in the boundaries of the cross-section. If there is a sharp bent of the wall in the cross-section, the shearing stress at this part is not uniform across this bent, and the shearing stress near the inner corner will be very great and that at the outer corner will be zero.

As there are several sharp bents and other discontinuities in the cross-section of a ship, the theory in the foregoing article does not hold for the torsion of a ship's hull. To investigate the effect of the stress-concentration at such parts on the torsion of a tube as a whole, some thin hollow rectangular tubes of celluloid which have decks and double bottom as in a ship's hull were subjected to torsion, and the angle of twist was measured. The results of this experiment show that the observed angle of twist agrees with the value calculated by the foregoing method disregarding the stress-concentration at the discontinuities within the difference of $\pm 10\%$. Therefore we may conclude that the effect of the sharp bents and other discontinuities on the torsion of the tube as a whole is relatively small, and the method of calculating the torsional stresses, &c., in the preceding article may be applied to the ship's hull without any considerable error.

Another experiment carried out was to examine the effect of opening on the torsion of hollow tube. A hollow circular tube of india-rubber which has a rectangular opening was twisted and the strains were observed. The length of the part of the tube which was subjected to torsion is 50.2 cm, the inner diameter of the tube 10.0 cm, and its thickness .42 cm. The length of the rectangular opening is 6.0 cm and the width 3.6 cm. Figs. 4 to 6 represent the distributions of the longitudinal and circumferential strains, and the shear strain. It will be seen from the figures, that there are considerable direct and shear strains near the corner of the opening, and, moreover, the shear strain at the part of the opening is far greater than that at the part of closed tube. The experiment was not accurate enough to obtain a quantitative results, but the results show fairly well the general nature of the torsion of tube having an opening.

The above two experiments were carried out by the students of the

department of Naval Architecture in the Tokyo Imperial University under the instruction of the author, and hearty thanks of the author are due to them for supplying the useful data.

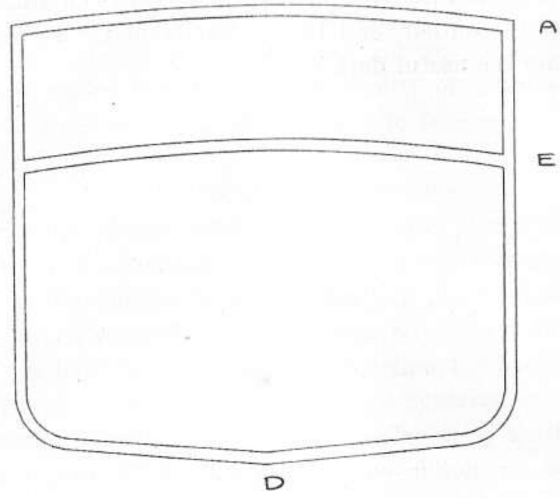


Fig. 1

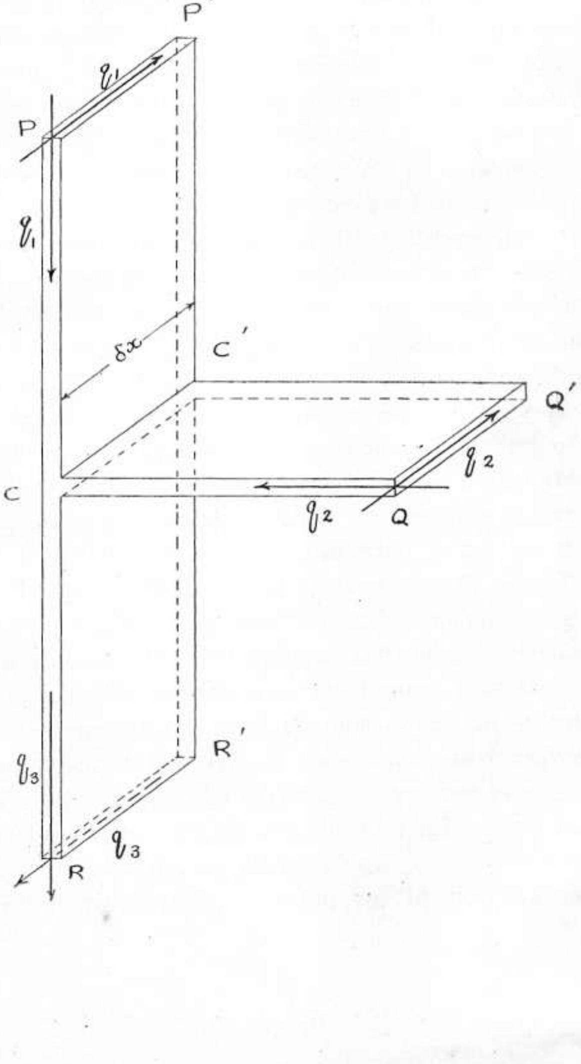


Fig. 2

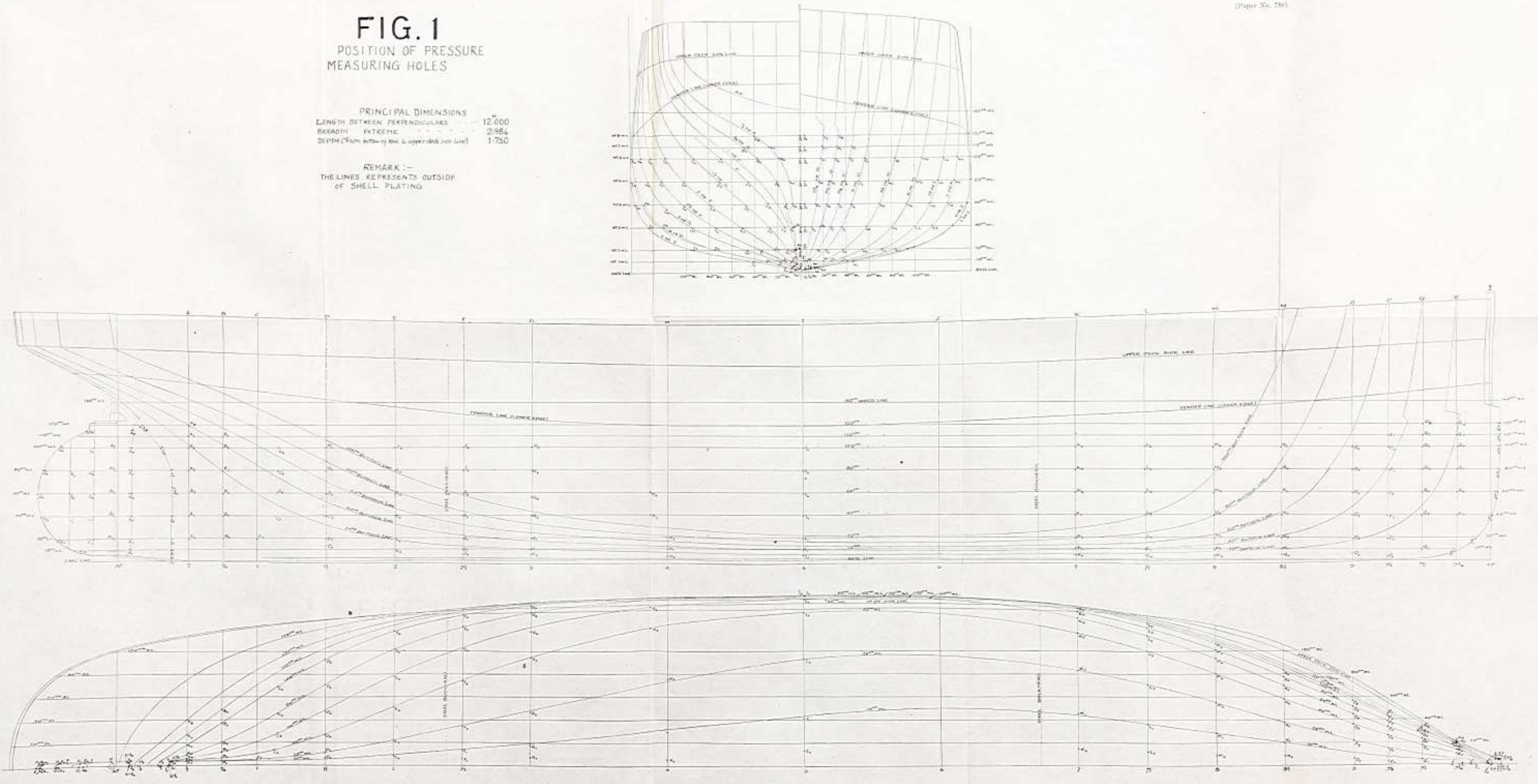


FIG. 1
POSITION OF PRESSURE
MEASURING HOLES

PRINCIPAL DIMENSIONS

LENGTH BETWEEN PERPENDICULARS	12.000 m
BREADTH EXTREME	2.984
DEPTH (From bottom of keel & upper deck side line)	1.750

REMARK :—
THE LINES REPRESENTS OUTSIDE
OF SHELL PLATING

(Paper No. 790)

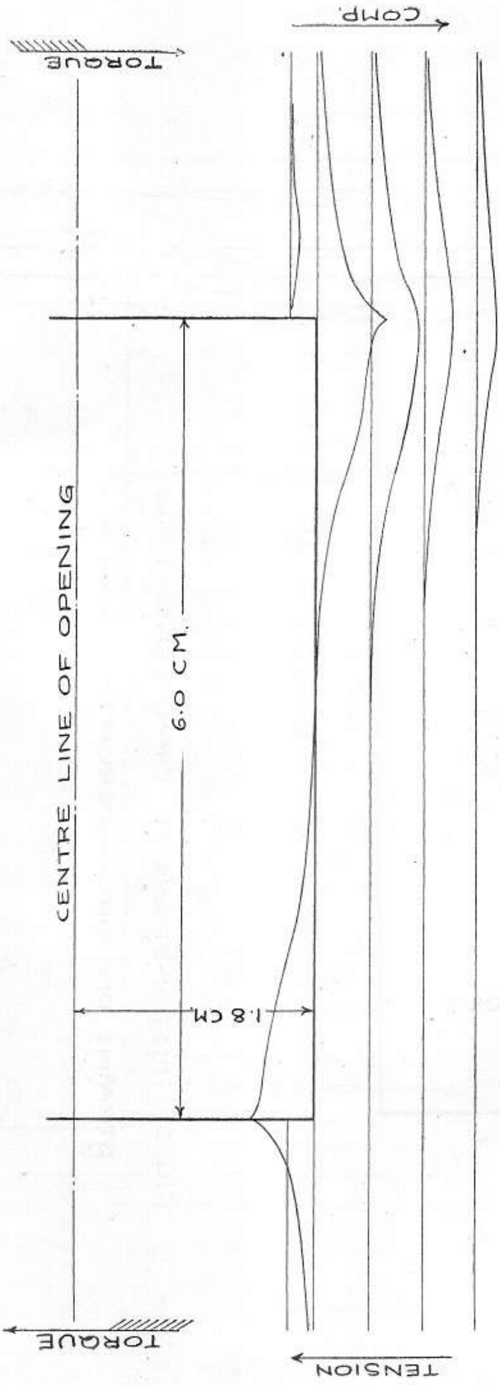


FIG. 4. DISTRIBUTIONS OF LONGITUDINAL STRAINS ON LONGITUDINAL LINES.

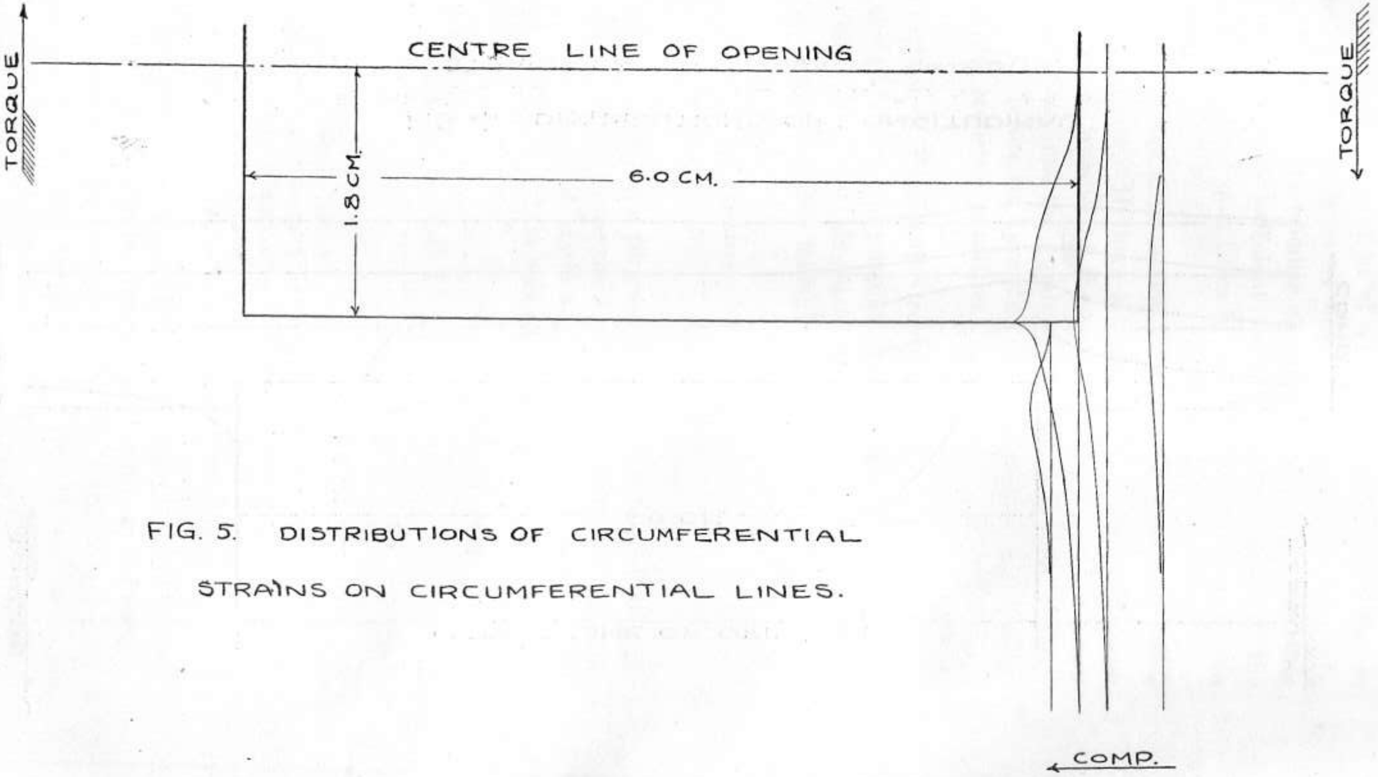


FIG. 5. DISTRIBUTIONS OF CIRCUMFERENTIAL STRAINS ON CIRCUMFERENTIAL LINES.

(Paper No. 790)

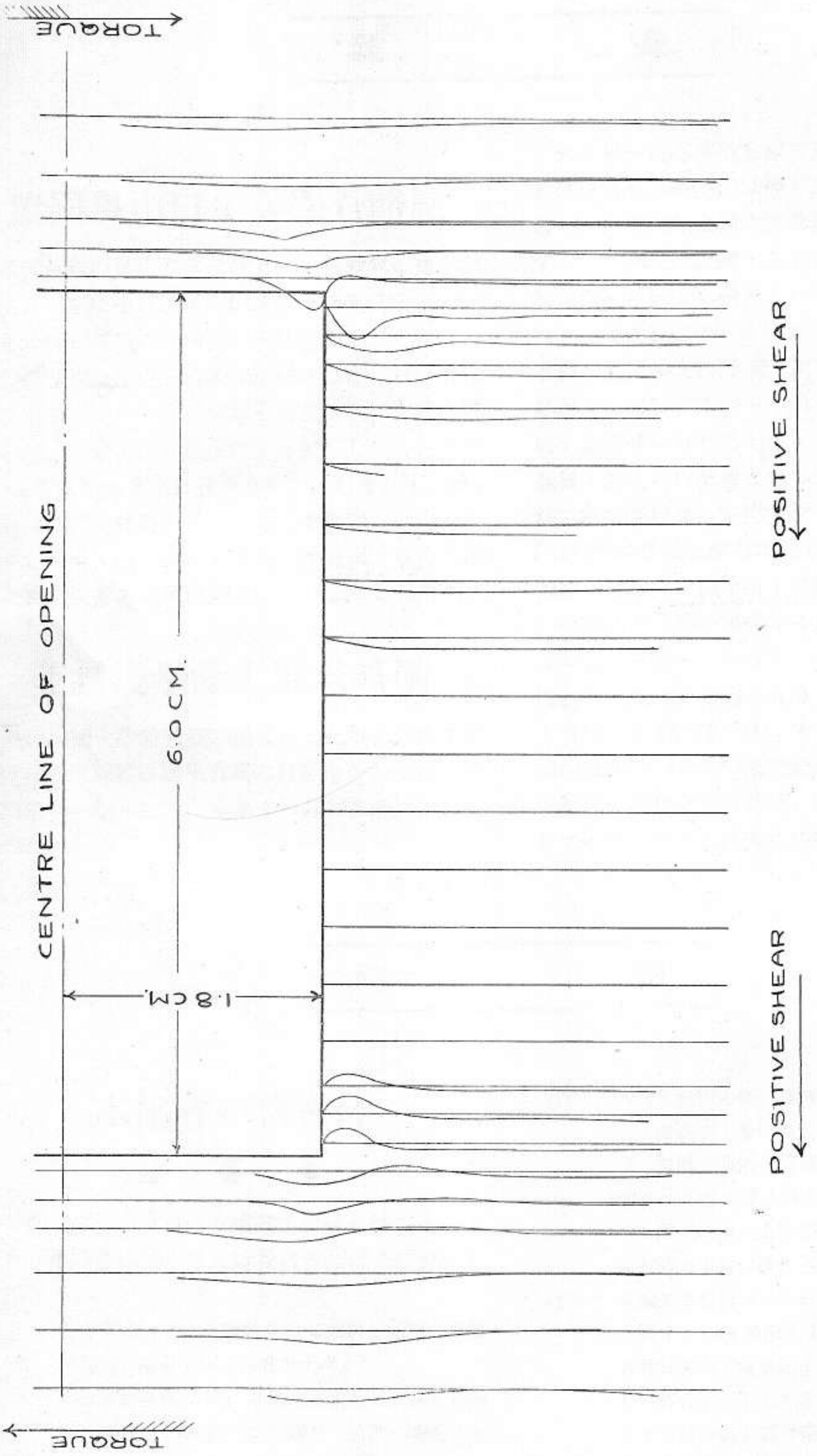


FIG. 6. CIRCUMFERENTIAL DISTRIBUTIONS OF SHEAR STRAINS FOR LONGITUDINAL AND TANGENTIAL LINES

雜　　　　錄

協同員山口半兵衛君略歷

本協會協同員山口半兵衛君は大阪財界の重鎭たる山口一家に生れ、早稻田大學卒業後、大正十二年まで山口銀行に勤務し、本店營業部長となり、その後塚口土地株式會社取締役に選任され、更に東洋リノリユーム株式會社取締役に就任在職中胃癌を患ひ、豫てより療養に手を盡くしつゝあつたが、遂に本年九月三日長逝された。享年五十二歲。

君は溫厚な好紳士で、そしてまた頗る活動家であり、繪畫に深い趣味を持ち、讀書家でもあり立派な實業家であつたが洵に惜しいことをした。

正員淺羽隆太郎君略歷

九州帝國大學助敎授の職に在つた本會正員工學士淺羽隆太郎君は篤學有爲の材を抱きながら、三十二歲を一期として、本年十一月二日溘焉として逝つた。

君は第一高等學校を經て東京帝國大學工學部船舶科に入り、優良の成績を以て大正十二年卒業するや、直に九州帝國大學助敎授に任ぜられ、孜々すとして研究に沒頭し、每に同大學發行の工學彙報に自己の硏究を發表しつゝあり、大正十三年には九州帝大特設の「タンク」によつて實驗を行ひ、本會の懸賞論文に應募して、「元良式の船舶動搖制式裝置に關する一二の硏究」と題する論文を提出し當選せられしことあり、殊に昭和二年十一月五日工學大會の部會として東京帝國大學工學部新館に於て開催せる本協會講演會に於て講演された「有孔板に於ける應力に就て」と題する（昭和三年四月刊行本會會報第四十二號掲載）論文は役員會の決議により特に優秀なるものと認められ本協會より賞牌を贈呈したることあり、當時某氏の如きは、この論文を激稱して博士論文たる價値があると言はれた程であつた。この外にも、尙抵抗理論の實驗的のもので未發表のものが二三あつたといふに、神經衰弱の結果、はかない最後を遂げられてしまつたことは洵に痛惜の至りである。

時　　　　報

本協會の諸會合

役　員　會

昭和四年十月九日（水曜日）午後五時三十分より本會事務所に於いて臨時役員會開催次の諸件を審議す。

（一）入會者承認の件。團體員（第四級）滿洲船渠株式會社、准員上野敬三君外十名。

（二）地方委員推薦の件。播磨地方委員三上英果君神戸に轉任の爲め辭任に付横尾　龍君へ委囑すること。

（三）昭和五年度豫算案の件。原案承認總會に附議すること。

（四）外國人招待の件。本年度通常總會に引續き開催する當協會主催の晩餐會に萬國工業會議に出席せる當協會關係の外國人招待に關し、招待狀は邦文にて認め可成早く發送すること、夫人同伴の外國人は本人丈を招待すること、招待の範圍は越智主事竝に湊評議員に一任すること。

（五）萬國工業會議論文の件　萬國工業會議第九部會に屬する論文外國人十七篇日本人一篇は各貳千百部宛萬國工業會議より無償にて本會へ提供を受け其代償として本會に於て印刷する邦人の論文十六篇を各六百部宛同會議へ提供する事となれり。從つて前回の役員會にて可決せし論文印刷費豫算は相當減額する見込なり。

（六）萬國工業會議論文前刷其他の件。本會會員に限

り第九部會に屬する論文前刷は無代にて出席者へ頒布すること。論文掲載の「雜纂」は從來會報を寄贈しつゝある外國の學會へも送付する事。

（七）米國機械學會創立五十年會幷に「ベルギー」鐵鋼構造に關する萬國會議の件につき工學會より交渉の件。米國機械學會に對しては本會より祝賀狀を差出す外、なるべく論文を提出する方可なるにより一應勸誘を試みること。又「ベルギー」萬國會議へは當協會よりは出席者及論文提出なき旨工學會へ回答すること。

出　席　者

會長	末廣恭二君	理事（主事）	越智誠二君
理事	平賀讓君	監事	山本幸男君
評議員	斯波孝四郎君	評議員	田原得三君
評議員	山本開藏君	評議員	永村清君
評議員	湊一磨君	評議員	島谷敏郎君
評議員	山本武藏君	會務委員	陰山金四郎君

造船史編纂委員會

昭和四年十月十六日（水曜日）午後五時三十分より本會事務所に於いて松長委員長司會の下に第三十七回の會合をなし次の報告竝に議事を諮り午後九時三十分散會す。

（一）委員異動の報告。

退任、中根經三君

新任、海軍造船中佐渡邊武夫君

新任、遞信技師板部成雄君

（二）議事。曩に配付せる原稿第八七號乃至第九〇號原稿につき研究の末一部修正之が査定を了す。

昭和四年九月末日現在に於ける未提出の原稿項目に關し審議の末、造船業に稍關係薄きものにして材料蒐集にも亦困難を感ずるもの數項目は此際目次より削除することゝし、結局昭和五年貳月末日迄に原稿全部提出することに申合せ、之が實現を期すること。

當日出席者（順序不同）

松長規一郎君	山内不二雄君
牛尾平之助君	横山要三君
湊一磨君	高島三郎君
朝永研一郎君（代理）	板部成雄君
渡邊武夫君	松山武秀君

總噸數百噸以上 工事中、進水及竣工船舶毎月合計調

月別	工事中船舶		進水船舶				竣工船舶			
			合計		累計		合計		累計	
	隻數	總噸數	隻數	總噸數	隻數	總噸數	隻數	總噸數	隻數	總噸數
昭和四年一月	50	156,061	5	1,906	5	1,906	2	1.832	2	1,833
〃 二月	49	159,705	4	9,076	9	10,982	9	9,774	11	11,606
〃 三月	49	165,105	7	15,806	16	26,838	4	9,606	15	21.212
〃 四月	49	176,455	13	22,173	29	49.011	6	6,033	21	27,245
〃 五月	42	173,724	9	32,778	38	81,789	11	14,619	32	41,864
〃 六月	36	181,345	7	16,770	45	98,559	10	10,842	42	52,706
〃 七月	38	182,035	3	3,800	48	102,359	1	233	43	52,939
〃 八月	34	177,530	6	11,470	54	113,829	10	22,804	53	75,743
〃 九月	33	158,740	6	12,507	60	126,336	6	20,603	59	96,346

昭和四年九月末 總噸數百噸以上の工事中船舶調

造船所	船種	船名	船質	計畫總噸數	進水年月	進水豫定年月	船舶工事進捗の模樣	註文者又は所有者
横濱船渠會社	發	秩父丸	鋼	16,750	4. 5		艤裝中	日本郵船會社
〃	〃	氷川丸	〃	11,000	4. 9		〃	〃
〃	〃	日枝丸	〃	11,000		5. 1	內底板取付中	〃
〃	〃	しどにい丸	〃	5,300	4. 8		艤裝中	大阪商船會社
〃	〃	未定	〃	5,300		4. 12	フレーム取付中	〃
〃	〃	〃	〃	5,300		5. 2	外板加工中	〃
浦賀船渠會社	〃	〃	〃	7,500		未定	30%	山下汽船會社
原田造船所	汽	〃	〃	175		4. 10	37%	大阪商船會社
〃	發	〃	〃	140		4. 11	15%	松尾八三郎
大阪鐵工所	〃	平洋丸	〃	9,500		4. 10	73%	日本郵船會社
〃	〃	平安丸	〃	11,000		未定	35%	〃
遠藤造船所	汽	第十二清貞丸	〃	170		4. 10	70%	藤岡船舶部
大塚造船所	帆	和興丸	〃	400	4. 9		艤裝中	野村政一
川崎造船所	汽	第三十六共同丸	〃	1,500	4. 8		〃	阿波共同汽船會社
〃	發	白鷹丸	〃	1,200	4. 8		〃	水產講習所
〃	〃	第一宇高丸	〃	300	4. 9		〃	鐵道省
〃	帆	未定	〃	2,250		4. 12	2%	文部省
〃	〃	〃	〃	2,250		5. 1	2%	〃
播磨造船所	發	第二八重丸	〃	105		未定	80%	笹岡鐵男
〃	〃	紀洋丸	〃	110		〃	80%	和歌山縣
〃	〃	沖島丸	〃	110		〃	80%	門司宗太郎
〃	汽	未定	〃	5,000		〃	20%	朽木商事會社
〃	〃	〃	〃	1,100		〃	15%	北日本汽船會社
三井玉工場	〃	〃	〃	2,000		〃	80%	山九運輸會社
松浦造船所	發	〃	〃	120		〃	四年九月龍骨据付	北九州商船會社
脇本造船所	帆	吉德丸	木	100		〃	肋骨作成中	相澤吉藏
三菱彥島造船所	發	妙義丸	鋼	325	4. 9		艤裝中	共同漁業會社
三菱長崎造船所	〃	龍田丸	〃	16,000	4. 4		〃	日本郵船會社
〃	〃	ブエノスアイレス丸	〃	9,500	4. 5		〃	大阪商船會社
〃	〃	リオデジヤネイロ丸	〃	9,500		4. 11	43%	〃
〃	〃	照國丸	〃	11,800		4. 12	24%	日本郵船會社
〃	〃	靖國丸	〃	11,800		5. 3	14%	〃
堀常造	帆	新寶丸	木	135	4. 9		艤裝中	岩野英吉

登簿船調 昭和四年九月末現在

	積量		內地	朝鮮	臺灣	關東州	合計
汽船	20噸以上 100噸	隻 噸	1,646 68,606	144 5,784	21 790	24 1,128	1,832 76,308
	100 〃 300〃	隻 噸	415 74,674	12 2,274	8 1,247	15 2,342	450 80,537
	300 〃 500〃	隻 噸	142 56,493	6 2,301	1 499	7 3,014	156 62,217
	500 〃 1,000〃	隻 噸	228 172.601	8 6,209	— —	7 5,531	243 184,341
	1,000 〃 2,000〃	隻 噸	237 347,025	14 17,497	— —	12 17,434	263 381,956
	2,000 〃 3,000〃	隻 噸	198 481,420	6 13,107	— —	14 36,363	218 530,890
	3,000 〃 4,000〃	隻 噸	146 489,065	— —	— —	14 50,659	160 539,724
	4,000 〃 5,000〃	隻 噸	78 348,206	— —	— —	24 108,390	102 456,596
	5,000 〃 6,000〃	隻 噸	136 765,281	— —	— —	16 87,190	152 852.471
	6,000 〃 7,000〃	隻 噸	53 345,497	— —	— —	5 31,276	58 376,773
	7,000 〃 8,000〃	隻 噸	40 293,532	— —	— —	2 14,307	42 307,839
	8,000 〃 9,000〃	隻 噸	8 67,885	— —	— —	1 8,230	9 76,115
	9,000 〃 10,000〃	隻 噸	14 132,628	— —	— —	— —	14 132,628
	10,000 〃	隻 噸	12 144,669	— —	— —	— —	12 144,669
	計	隻 噸	3,353 3,787,492	187 47,172	30 2,536	141 365,864	3,711 4,203,064
	內 100噸以上	隻 噸	1,707 3,718,886	46 41,388	9 1,746	117 364,736	1,879 4,126,756
	內 1,000噸以上	隻 噸	922 3,415,208	20 30,604	— —	88 353,849	1,030 3,799,661

	船種	積量		內地	朝鮮	臺灣	關東州	合計
帆船	噸數船	20噸以上 100噸	隻 噸	13,166 585,296	678 21,324	121 6,780	48 1,934	14,013 615,334
		100 〃 300 〃	隻 噸	2,087 292,401	2 384	3 360	— —	2,092 293,145
		300 〃 500 〃	隻 噸	32 12,303	3 983	— —	— —	35 13,286
		500 〃 1,000 〃	隻 噸	2 1,193	— —	— —	— —	2 1,193
		1,000 〃 2,000 〃	隻 噸	— —	— —	— —	— —	— —
		2,000 〃 3,000 〃	隻 噸	2 4,941	— —	— —	— —	2 4,941
		計	隻 噸	15,289 893,134	683 22,691	124 7,140	48 1,934	16,144 927,899
	石數船	200石以上 300石	隻 石	258 64,666	— —	10 2,399	25 6,284	293 73,349
		300 〃 400 〃	隻 石	133 45,651	— —	2 759	4 1,474	139 47,884
		400 〃 500 〃	隻 石	52 22,902	— —	1 416	2 874	55 24,192
		500 〃 1,000 〃	隻 石	13 7,880	— —	2 1,402	— —	15 9,282
		1,000 〃		— —	— —	— —	— —	— —
		計	隻 石	456 141,099	— —	15 4,976	31 8.632	502 154,707
	合計		隻 噸	15,745 910,244	683 22,691	139 7,638	79 2,797	16,646 943,370

10 石を 1 噸に換算し合計に算入す

		內地	朝鮮	臺灣	關東州	合計
總計	隻 噸	19,098 4,697,736	870 69,863	169 10,174	220 368,661	20,357 5,146,434

最近本邦海上運賃及傭船料

		9月中	10月中	11月下旬
運賃	石炭（單位噸）			
	九州―橫濱間	円 円 1.10-1.20	円 1.10	円 円 1.05-1.10
	九州―伊勢灣間	1.10-1.20	1.10	1.05-1.10
	九州―上海間	1.30-1.40	1.50	1.30-1.50
	九州―香港間	2.20-2.30	2.20	2.20
	九州―新嘉坡間	3.00	3.00	3.00
	豆粕（單位擔）			
	大連―橫濱間	.09- .10	.09-.10	.09- .10
	大連―伊勢灣間	.09- .10	.09-.10	.08- .10
	大連―神戶間	.07- .09	.07-.08	.07- .08
	小麥（單位噸）			
	北米（太平洋岸）―日本間	弗 3.50	弗 3.50	弗 3.50
	木材（單位 樺太―內地間百石 北米―日本間千呎B.M.）			
	樺太―內地間（丸材）	円 円 85.00-90.00	円 円 95.00-105.00	円 80.00
	北米（太平洋岸）―日本間	弗 7.00	弗 弗 7.25-7.50	弗 弗 6.75-7.25
	鐵（單位噸）			
	北米（太平洋岸）―日本間	——	——	——
	紐育――日本間	弗 12.00	弗 12.00	弗 12.00
傭船料	大型	円 1.30	円 円 1.50	円 円 1.40-1.50
	中型	1.50	1.80-2.00	1.60-1.80
	小型	一區 4.00 二區 2.80	4.00 2.50-2.80	2.80-4.00 2.00-.250

最近世界海上運賃

(1) 英國方面向（1噸當）

發航地	到達地	貨物	9月中	10月中
亞歷山	英本國	棉實	志片 志片 13.06	志片 志片 11.06-13.06
濠洲	英本國 歐大陸	小麥	30.00	30.00-30.06
ビルバオ	カーデイフ	鑛石	7.00	6.06- 7.00
孟買	英本國 歐大陸	雜貨	21.00-22.03	21.06-22.06
ビルマ	〃	米	——	——
ダニユーブ河	〃	穀類	——	——
リヴアプレート	〃	〃	16.06-18.00	13.00-15.00
北米大西洋岸	〃	〃	× ——	× ——
メキシコ灣	〃	〃	× 3.00	× 2.06

備考 ×印は標準を480封度とす

(2) 英國發（1噸當）

カーデイフ	坡西土	石炭	志片 志片 10.00-10.06	志片 志片 9.00-10.00
同	リヴアプレート	〃	16.00-17.06	16.00-18.00
同	セントヴインセント	〃	10.03	9.00-10.06

會員動靜

○入會

		職名、勤務先	住所
村木友吉	正員	三菱造船株式會社長崎造船所技師	長崎市今博多町五一
上野敬三	准員	九州帝國大學工學部造船學科學生	福岡市春吉六番町井上時枝方
田中利一	同	三菱造船株式會社長崎造船所技師（造機設計）	長崎市城山町北一條一五
島哲夫	同	同上	長崎市今町一八
角家悟一	同	同上	長崎市片淵町五七、中路育雄方
中川英一郎	同	三菱造船株式會社神戸造船所技師	神戸市西須磨一の谷八本松一三四ノ五
飯島英夫	同	遞信省管船局船舶試驗所	東京府下、碑衾町衾二二三九

株式會社石川島飛行機製作所　代表者 吉原四郎　團體員（第四級）　東京市京橋區月島西仲通九ノ七

○准員より正員に會員資格變更者

正員　山本一雄　　正員　水上五六　　正員　田邊龜雄

○轉居轉任

大原勇平	東京府下、南足立郡綾瀨村綱五郎新田二六
合田秀雄	東京市小石川區八千代町四一森松方
谷川正次	東京府下、下落合府營住宅二號地第十六號澁谷方
植原文内	神戸市須磨西寺田町六四
瀧野愛之助	神戸市都由乃町二丁目二三ノ二
畑敏男	海軍造船少佐、英國出張ヲ免シ獨國ヘ出張ヲ命セラル
山本一雄	京都市外、深草町稻荷福稻字六反田一ノ四田尻庄藏方
有馬孝	長崎市本尾町一五
本儀正	戸畑鑄物株式會社常務取締役（住所東京市外、高田町目白上リ屋敷三五七〇、電、牛込〔34〕4348番）
庭田尙三	横須賀市稻岡官舍第十二號
矢ケ崎正經	米國ヨリ歸朝、(住所、東京市四谷區簞笥町九四)

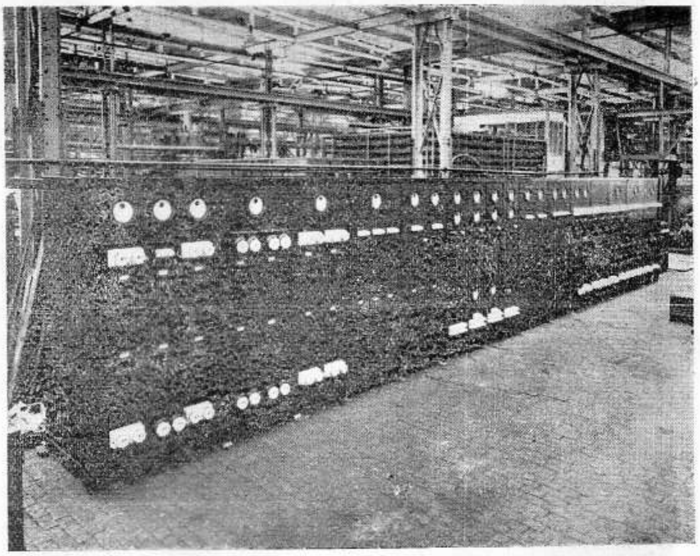
八幡順之吉	横濱ニ於ケル事務所移轉、横濱市中區日本大通十一、横濱商工奬勵館（中二階別室）内外海事鑑定社
吉村秀人	神戸市氷室町一丁目一一七神谷政治郎方
渡邊浩	岡山縣兒島郡日比町玉海事部官舍
大山一兵衛	中野電信第一聯隊第四中隊退營、昭和四年十二月ヨリ三井物産株式會社造船部玉工場造船設計課ニ復歸
赤崎繁	東京市麻布區笄町一七六
齋藤眞	東京市麴町區下六番町二四（電、九段〔33〕1165番）
三井高塞	岡山市國富町一一八
小澤貞二	兵庫縣武庫郡精道村芦屋伊勢講田五四九
三上英果	株式會社播磨造船所　常務取締役
横尾龍	〃　〃　〃
浦田鐵六	〃　〃　取締役
神保敏男	〃　〃　〃

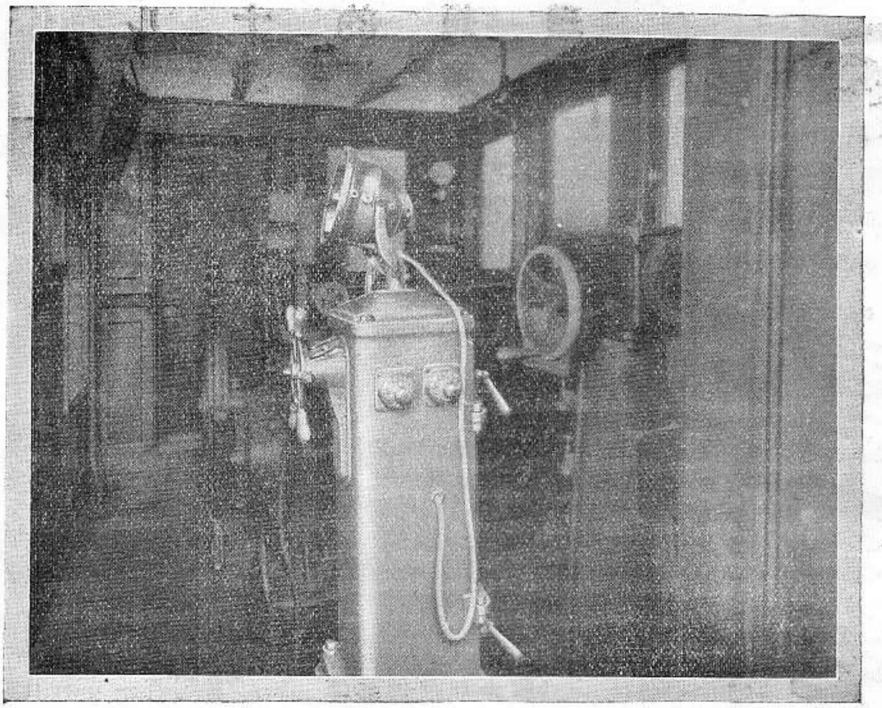
○死亡會員

正員　工學士　淺羽隆太郎君　昭和四年十一月二日死去

協同員　下村耕次郎君　昭和四年十一月十六日死去

本會は此の訃音に接し謹みて哀悼の意を表す





左圖は米國**デーゼル船コウラジアス**號操舵室に於ける**スペリー**式自働操舵機を示す。
本自働操舵機では「手働による電氣的操舵」「自動操舵」又は「水壓**テレモーター**」何れの方式によつても操舵し得らるゝものである。

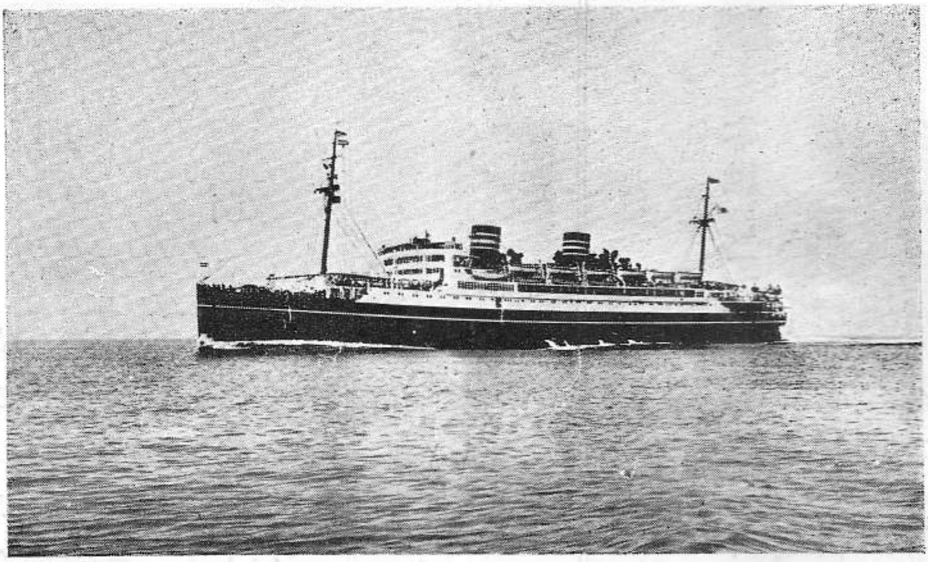
九度の操舵角を
一度で濟ますには

西諺に「綻の最初に直ぐ一針縫はゞ後九針の手間を省く」と云ふ事があるが**スペリー**式自働操舵機の機能程此諺を具體的に立證してるものは無い。

進路のふれを起した最初なら操舵角は僅々一二度ですむ、が、うつちやつとけば遂に十度或は夫れ以上の更正を要する。大角度の操舵は船足を遲くし動力の消費を増し結局不經濟となる。

然るに我**スペリー**式自働操舵機は推進と補助機關の動力とを最經濟的ならしめる、のみならず適當に之れを利用すれば三人以上の人手を省く事が出來る。

日本一手販賣代理店
三井物產株式會社
機械部
東京市日本橋本町二丁目一番地



長崎造船所建造　日本郵船桑港航路用　淺間丸（一六、九二〇噸）

昭和四年十一月十三日印刷
昭和四年十一月十五日發行

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印刷所　東京市神田[illegible]美土代町二丁目一番地　三秀舍

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電話丸ノ内（二三）一〇六九番
振替貯金口座東京一三七五〇番

廣告取扱所　東京[illegible]京橋區上柳原町八番地（電話京橋八七三番、振替東京三〇六九番）　東京第一通信社