昭和四年十月十五日　發行
毎月一回十五日　發行

W.E.C. NUMBER

PART I.

昭和四年十月刊行

造船協會雜纂

第九十一號

造　船　協　會

(非　賣　品)

造 船 協 會 雜 纂

昭和四年十月刊行 第九十一號 内 容 目 次

W.E.C. NUMBER

PART I.

Contents.

Paper Number

雜 錄

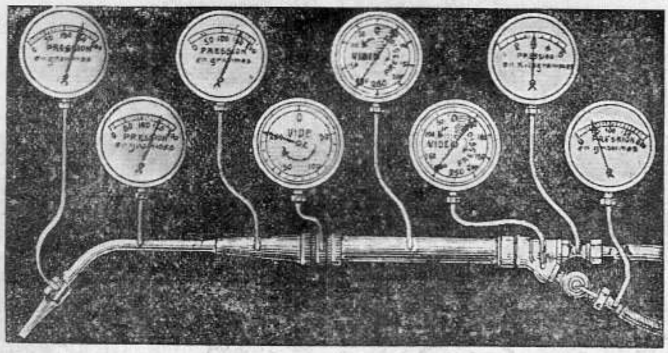
頁

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徴するも明かであります。

吹管各部の壓力檢査裝置



上揭の寫眞は弊社の製造場に於ける吹管各部の瓦斯壓力の**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。

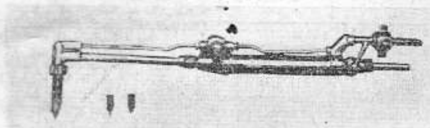
弊社の**吹管**の機能が理想的に確實で而も作業の成績と經濟とに最も重大なる關係を有する酸素アセチレーヌ**兩瓦斯の比**が極めて學理的數字に接近して居り斯界に嘖々たる好評を博しつゝあるは此の如き綿密なる檢査裝置に依り確實なる性能を與へてあるからであります。

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。

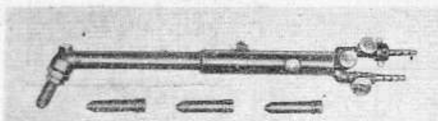
下揭の寫眞は弊社最近の製造に係る鎔接用及び切斷用吹管で**名實共に斯界最良**のものであります。

型錄進呈

切斷用

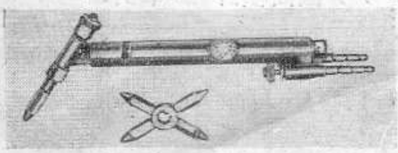


ピロコツプ式小型吹管

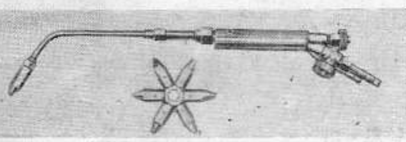


ピロコツプ式AS第一號

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(二)一八七九 三七六三 市外專一三

液体空氣會社

東京支社 東京丸ノ内三菱二十一號館 電話丸ノ内(23)一一六六 一七六二

支社

神戸支社 神戸市明石町三八 電話三宮一八七九

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 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七 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 電話下四六三四

新潟出張所 新潟市本町通九番丁 電話二三三七 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 電話六九二一六

直江津出張所 新潟縣直江津町 信越窒素肥料株式會社内

武生出張所 福井縣武生町 大同肥料株式會社内

水俣出張所 熊本縣葦北郡水俣町字馬刀潟

工場

兵庫工場 神戸市兵庫高松町 電話兵庫二七〇四 七七

大阪工場 大阪市此花區櫻島町三七 電話土佐堀三七四六

神奈川工場 横濱市神奈川區子安町濱通 電話本局三〇六二 三二六六

京城工場 京城府外鷺梁津 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 電話南一一六二

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 電話三六ノ九 五及ビ六五

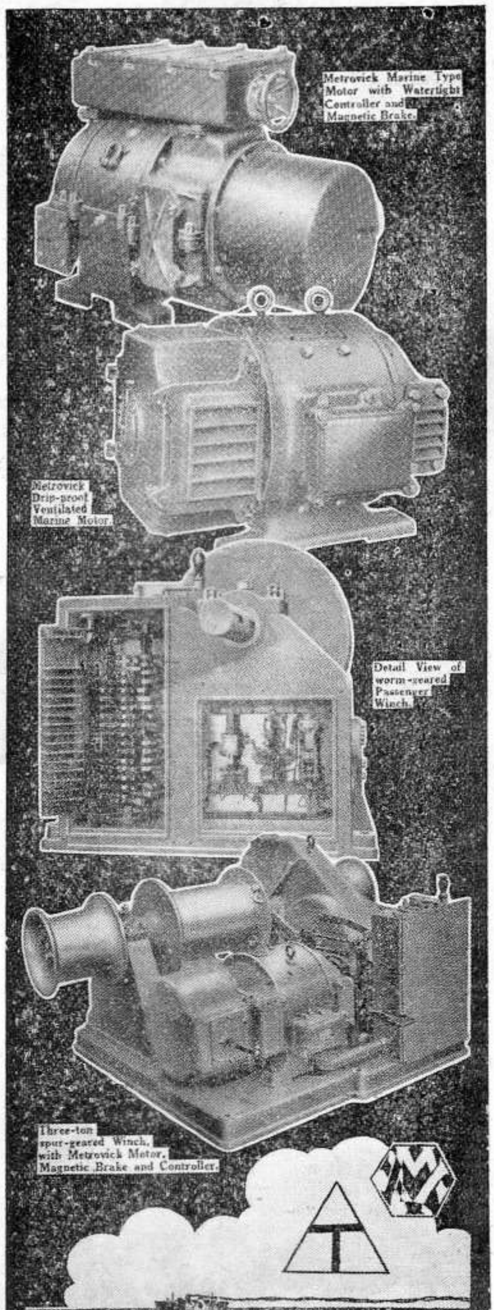
水俣工場 熊本縣水俣町字馬刀潟 電話一一六

直江津工場 新潟縣直江津町 電話直江津三五五

METROPOLITAN Vickers
ELECTRICAL CO. LTD.

英國Ｍ・Ｖ・社製船舶用・電氣機械・

構造堅牢！　能率良好！　取扱至便！　價格低廉！　優秀なる技術により製作せられたるＭＶ社製品は常に斯界の御好評を賜り居り目下横濱船渠株式會社殿及浦賀船渠株式會社殿を始め諸會社より御註文を拜受仕り居り候



英國メトロポリタン・ヴヰッカース電氣會社總代理店

株式會社　**高田商會**

東京市麹町區丸の內二丁目六番地（八重洲ビル）

支店・大阪・名古屋・門司・小樽・倫敦・紐育


ISODA METAL CO
ISODA
TOKYO JAPAN
株式會社
イソダメタル工場
本社 東京府大崎町桐ケ谷六七 電話高輪一四九九
工場 東京府大崎町桐ケ谷六八 電話高輪一四九九
研究所 東京府巣鴨町平松一五二二 電話大塚六八六
東京販賣所 東京府麴町區丸ビル四〇九區 電話丸ノ內一六九三
大阪出張所 大阪市東區本町一ノ二四 電話本町八九五
神戸出張所 神戸市北長狹通リ三ノ八 電話葺合二〇〇六
横濱出張所 横濱市中區住吉町二ノ一八 電話長者町七九三
小倉出張所 小倉市堺町三ノ五五 電話小倉一一〇一
北海道代理店 函館市東濱町一一 米森船具店
名古屋代理店 名古屋市中區裏門前町一三 ワシノ商店
吳代理店 吳市岩方通八丁目一 石村政一商店
九州代理店 小倉市室町四丁目八六 大澤商會小倉出張所
臺灣代理店 臺灣高雄市入船町五丁目 株式會社臺灣鐵工所
釜山代理店 釜山府埋立新町一四 忽那商店
京城代理店 京城府大和町一丁目四四 大倉商事株式會社
滿洲代理店 大連市山縣通大倉ビル內 大倉商事株式會社
青島代理店 青島堂邑路 菱田洋行
東北代理店 仙臺市南町通五 千代田組仙臺支店

シャープレス式油清淨機

シャープレス・スペシャリティ會社

東洋總代理店

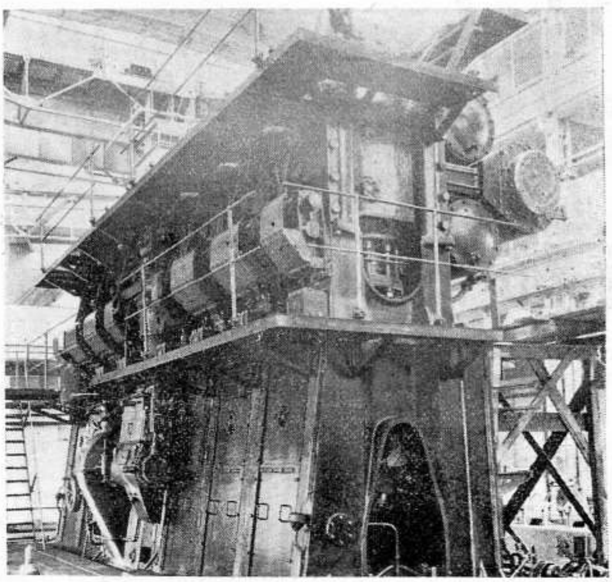
巽商事株式會社

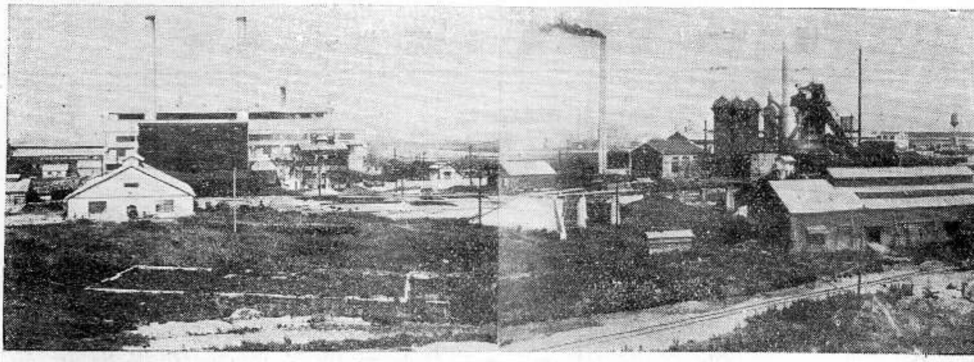
本　社　東京丸ノ内海上ビルディング内〔電丸ノ内二六〇三、二六〇四〕
出張所　大阪市西區梅本町小林ビルディング〔電、四、四三三二〕

〜〜型〜〜錄〜〜進〜〜呈〜〜

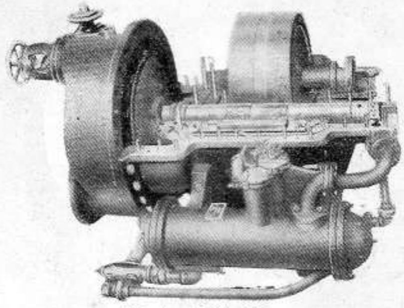

青寫眞界の革命
財團法人 理化學研究所發明
原畫がその儘紫紺色の
陽畫に現はる優良感光紙
理研陽畫感光紙
紙質優良
現像簡單
價格低廉
着色自在
水洗不要
保存永久
見本（カタログ）進呈
發賣元
東京・本郷・駒込富士前町四三
理化學研究所裏門前
理化學興業株式會社
振替口座東京六三九六六
代理店
東京 杉山商會
東京 櫻井商店
大阪 鈴鹿工務所
神戸 鈴木商店
吉村商會
特約店
名古屋 大島
德山 昭和商會


株式會社大阪製鎖所
海軍省指定
ロイド公認
營業課目
艦船用錨鎖及附屬品
特種兵器・チエンブロック
電氣鑄鋼製品
エレベーター・コンベヤー類
製作販賣
英国QA式電氣鎔接器具
エレクトロード
一手販賣
電氣鎔接水壓鉄管
一般電氣鎔接
製鑵工事請負
電話土佐堀
三二九二
三二九三
三二九四
三五五九
二八八二
大阪市此花區春日出町



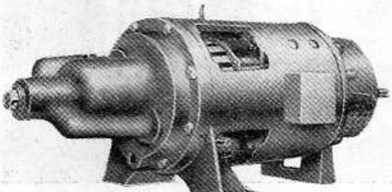


Westinghouse Turbo-Generators

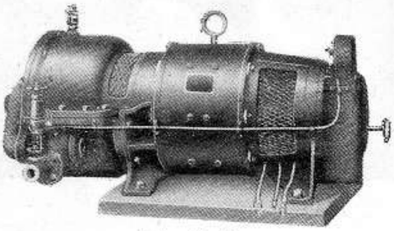


75 to 500 Kw.

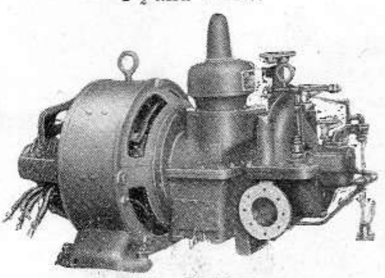
ウエスチングハウス
タービン發電機
各種



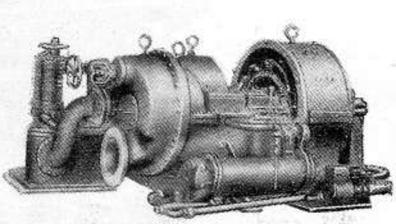
1 ½ and 3 Kw.



5 to 15 Kw.



25 to 50 Kw.



150 Kw. d-c.

Approximate Dimensions

K. W.	Length	Width	Height	Weight (lbs)
$1\frac{1}{2}$	33″	$14\frac{1}{2}$″	13″	270
3	$36\frac{1}{2}$″	17″	15″	375
5	3′—$9\frac{3}{4}$″	$24\frac{1}{2}$″	$20\frac{1}{4}$″	775
$7\frac{1}{2}$	3′—$9\frac{3}{4}$″	$24\frac{1}{2}$″	$20\frac{1}{4}$″	775
10	3′—$9\frac{3}{4}$″	$24\frac{1}{2}$″	$20\frac{1}{4}$″	775
15	3′—$9\frac{3}{4}$″	$26\frac{1}{2}$″	$20\frac{1}{4}$″	1,000
25	5′—$1\frac{5}{8}$″	3′—$4\frac{1}{2}$″	3′—$6\frac{1}{2}$″	1,590
35	5′—6″	3′—$5\frac{1}{4}$″	3′—8″	2,600
50	6′—2″	3′—$7\frac{3}{8}$″	3′—10″	3,000
75	7′—10″	4′—$7\frac{3}{8}$″	4′—$7\frac{3}{4}$″	6,570
100	8′—$4\frac{5}{8}$″	4′—$7\frac{5}{8}$″	4′—$8\frac{1}{2}$″	7,000
150	10′—$3\frac{1}{2}$″	4′—9″	5′—$1\frac{5}{8}$″	10,260
200	10′—$6\frac{1}{2}$″	4′—9″	5′—$1\frac{5}{8}$″	11,000

日本ウエスチングハウス電氣株式會社

東京丸之內郵船ビルディング

Westinghouse

PROGRAMME

WORLD ENGIEERING CONGRESS

Section 9. Ship Building and Marine Engineering.

Wednesday, October 30, 1929. Morning, 9 30 A.M.—Noon.

	Authers & Nationality.	Subjects.	Paper No.
1.	Mr. Y. Yamamoto (Japan), Mr. K. Inouye. (Japan)	The Recent Development of the Mercantile Shipbuilding in Japan and Other Matters relating to it.	225
2.	Mr. P.A. Hillhouse, Prof. (G.B.)	Shipbuilding in Great Britain.	425
3.	Associazione Nazionale Italiana Mecceni-cie Affini. (Italy)	Shipbuilding in Italy.	582
4.	Mr. Y. Hiraga, Vice-Admiral, KH., Prof. (Japan)	On After-War Development of the Ships of the Imperial Navy.	652
5.	Mr. A. Lamouche. (France)	The Limitation of Naval Armaments from a Technical Point of View.	691
6.	Mr. H. Hamada. (Japan)	Developments in Diesel Engine Shipbuilding in Japan.	85

Thursday, October 31, 1929. Morning, 9 30 A.M.—Noon.

1.	Baron C. Shiba, KH., Prof. (Japan)	Outline Sketch of the Development of Mercantile Marine Engines in Japan.	725
2.	Mr. F. Ushimaru, Rear Admiral. (Japan)	Development of Marine Engineering in Japanese Navy.	749
3.	Mr. H. L. Seward, Prof. (U.S.A.)	Marine Engineering and Design.	445
4.	Mr. G.R. Hutchinson. (G.B.)	Recent Development of the Marine Steam Reciprocating Engine.	269
5.	Sir. J.H. Biles. (G.B.)	The Outlook for Steam Engine Afloat.	256
6.	Mr. T. Ôno, Rear Admiral. (Japan)	Notes on the Recent Constructing Process of the Principal Parts of Marine Steam Turbines.	52
7.	Mr. T.B. Stillman. (U.S.A.)	Combustion in the Furnace of Marine Boilers.	447
8.	Mr. T. Inokuchi, KH., Prof. (Japan)	The Torsional Stress in the Hull of a Ship.	790

Friday, November 1, 1929. Morning, 9 30 A.M.—Noon.

1.	Mr. C.W. Dyson, Rear Admiral. (U.S.A.)	Marine Screw Propellers.	61
2.	Mr. D.W. Taylor, Rear Admiral. (U.S.A.)	Propeller Design Developments.	325

（昭和四年十月　造船協會）

會　　　告

(一) 地方委員の更迭

地方委員(播磨)就任　神保敏男君

同　　　　退任　三上英果君

正　　　誤

Errata of Paper No. 20.

Explanation { (denotes "from top" /) denotes "from bottom" }

Page	Line	False	Correct
2	(16	sponson	veranda
3	(17-(19	The bonito is not only cooked ......as katsuobushi.	The bonito is not only used as food when fresh, but it is universally used as a condiment, known as katsuobushi, which is suitable for storage.
5	(5, (9	sponson	veranda
6	(16	Height of bulwark.	Height of bulwark ...... (*to be written on left side*)
7	)19	4″×4′4″	4″×4″ two cyls.
7	)18-)16	connected by a main engine, with a special apparatus to keep the revolutions automatically constant.	connected to the main engine.
7	)15	Motor winch......	Power winch......
7	)15	The motor winch is driven	The winch for hauling net is driven
8	(1	2″dia. steel horse	2″dia. steell horse or $1^1/_2$″ steel wire

Note.

Figs. 20, 21, 22, 23, 24, 25 of Paper No. 269 are not printed.

Three Special Kinds of Fishing Boats recently developed in Japan.

(Paper No. 20)

By ***Seiichi Kato,*** *Chief Ship Inspector of Itozaki Marine Office.*

1. *Introduction :*

Japan is said to be poorly favoured with natural resources; but the author thinks there is one matter in which the country is rich, i.e., aquatic production. Actually, Japan has the largest output of aquatic products in the world, which is said to aggregate above 2,000,000 tons. Great Britain and Ireland are next in succession, their total being about 1,100,000 tons. But Japan has about 1,130,000 people who make a living by fishing and 350,000 fishing boats, whereas Great Britain is said to have only 90,000 fishermen and 20,000 fishing craft. In comparing the production per head or per boat of both countries, we find Japan's is only one-tenth. Therefore, Japan needs, evidently, further improvement in the methods used. When we look upon the condition of Japan of ten or twenty years ago, we realise her rapid development in the right direction. In the following table, it is shown that the catches have increased by 30% and the number of larger fishing boats and motor boats have also increased considerably, while the total number of fishing boats has decreased.

Number of Fishing Boats.

Years	20 tons and above		Below 20 tons		Catches in tons
	Steamers & Motor boats	Sailing boats	Motor boats	Sailing boats	
1914	222	310	2,300	387,000	1,600,000
1926	979	107	14,924	334,924	2,297,800

Number of Larger Fishing Motor Boats and Steamers in 1926, Classified by Kinds of Fishing.

Material of Hull	Trawlers	Whalers	Bonito fishing boats	Floating canneries for crab fishing	Fish carriers	Seiners, liners, drifters, &c.
Steel	70	30	20	14	53	38
Wood	—	—	179	—	73	502

The improvement of fishing boats has thus been remarkable during the last twenty years, contributing to the development of the fishing industry. This has been achieved not only by shipbuilders, but also by the aid of fishermen of progressive and enterprising spirit, and by the Government officers administering the Deep Sea Fisheries Encouragement Act.

Among other things, the three kinds of fishing boats mentioned here have been most noticeable and creative, and the author believes a short description would interest the menbers of this Congress. The author expresses his cordial thanks to the Government officers, shipbuilders and fishing companies who have helped him with useful drawings and other particulars for this paper.

2. *Bonito Fishing Boats :*

The bonito is a fish of fifteen to thirty inches in length, moving in shoals in warm currents. They are chiefly caught by the rod, though drift nets or purse seines are sometimes used. The boat should have sufficient stability and should be steady in a seaway, for a great number of men stand side by side on the sponson built outside of the bulwark at one side of the boat when fishing. The boat should have fish wells for live sardines, which should necessarily be used as bait though imitated bait might be used with it in the middle of fishing.

The boat should have good speed and quick maneuvering quality, in order to find and chase the best shoals and return to port quickly in order to sell the catches at the best price. Before the year 1905, the fishing boat was a Japanese junk and the fishing was carried on from April to September at every part of the south and east coasts of Japan. Even in this period, the fishing craft belonging to Shizuoka Prefecture went as far as the Izu Isles, several days being spent in one voyage. These were the bigger junks, the principal dimensions for instance being 50′ 0″ × 11′ 0″ × 4′ 6″, the crews numbering some thirty hands. When there was little or no wind, the men took to the oars. In 1906, the Fishery Experimental Bureau of Shizuoka Prefecture built a new boat on the European model, named " Fuji Maru ", under subsidy from the Government. She was a ketch-rigged, welled smack of 25 tons gross, with a 20 B.H.P. four cycle electric ignitioned petroleum motor made by the Union Iron Works of San Francisco as the auxiliary power. This was the pioneer motor fishing craft in Japan. Some fishermen thought such a boat might roll heavily, and not be suitable for fishing operations, or that the sound of the motor and the odour of kerosene might drive the fish away. But her performance was quite satisfactory. As the Government granted a subsidy to motor fishing boats strongly built after this model, many fishermen installed motors or built new hoats. Thus a special Eurasian type of craft appeared, having a broad keel to allow of hauling up easily on beaches, an angular knuckle on the bilge to check rolling, and masts capable of removal for the same purpose, as is done in Japanese junks; but she was stronger built with more frames than junks and had a complete deck, fixed rudder, clipper stem, counter and screw aperture. She could run above seven knots with the motor only and did not spread her sails except in case of a fair wind. Several petroleum motors for fishing boats were imported from abroad and some were built in Japan. The imported motors were "Union" "Dan," "Mitz and Weis", "Griffin", "Bolinder", "Scandia" &c.

A few petrol motors and suction gas engines were also adopted. But by and by, home made two cycle hot bulb petroleum motors of improved "Bolinder" pattern became most popular because of their small fuel consumption, simple handling and smooth running. A few years afterwards, high compression motors using heavy oil as "Fairbanks Morse" and improved "Bolinder" not using injection water were imported. In the meantime, several home makers improved their motors also in this direction.

Recently, for larger power, home made four cycle air injection Diesels have been generally adopted for the sake of fuel economy. On the other hand, sizes of boats became larger step by step, to 20 tons, 40 tons, 60 tons, 80 tons, even to 180 tons. Since 1922, some steel boats have been built. At present, the fishing grounds extend to the Bonin Isles, Marcus Island, and Formosa. The total amount of catches in Japan proper is now valued at Yen 35,000,000, while it amounted to only Yen 5,000,000 twenty years ago. The bonito is not only cooked when fresh, by boiling, smoking and drying, when it is suitable for storage, but it is universally used as a condiment, known as katsuobushi. There is no limit to the demand for katsuobushi, because it is essential as seasoning for soup and other domestic uses. Therefore this branch of fishery has great prospects of continued development.

Plates 1 and 2 are the general arrangement plan and midship section of the latest bonito fishing boat. Her dimensions are as follows:—

Principal dimensions	$80' \ 0'' \times 19' \ 0'' \times 9' \ 0''$	
Tonnage	Gross	102.02 tons
	Net	35.84 tons
Raised quarter deck	Height	9″
	Length	34′ 4¾″
Height of bulwark		4′ 2″
Sheers of the boat floating at even keel	Forward	3′ 3¾″
	Aft	3′ 5½″
Draft, displacement, &c. at loaded condition	Draft	7′ ¼″
	Displacement	195.6 tons
	Block coefficient	0.636
	Prismatic coefficient	0.669
	Gm	2.07 ft.
Main engine	"Niigata Diesel"	
	B.H.P.	200 at 350 revolutions
Propeller	Diameter	56″
	Pitch	38″
	No. of blades	3
Auxiliary machiner	1) Auxiliary compressor, coupled to the	

motor for wireless use.

2) General service pump driven by a belt connected to the main engine, the capacity being 250 gallons/min. This is used not only for wash deck, bilge suction, fire main &c., but also for spraying water from some nine hoses connected to the 2″ main fitted along the outside of bulwark, which is useful during fishing operation, and for discharging water from fish wells.

3) Motor for wireless telegraph.

4) Dynamo for lighting, driven by a belt from main engine

Trial results	Draft forward	2′ 2½″
	aft	7′ 0″
	mean	4′ 7¼″
	Displacement	116.2 tons
	Blockcoefficient	0.58
	Prismatic coefficient	0.63
	Speed	10.25 knots
	Revolutions	379
Number of crew		about 70
Tank capacities, &c.	Fuel oil tanks	14 tons
	Fresh water tanks	6.5 tons
	Ice stores	17 tons

Fish wells · · · · · · · · · · · Total capacity of water at loaded condition 1164 cubic feet

There are three compartments which are bordered by watertight wooden bulkheads and bottom ceilings. Each compartment has eight brass pipes of 4″ diameter and 6″ length on the bottom ceiling to communicate the sea water, gratings and guide lips being fitted on. In outward voyages, all compartments are used as wells for the baits; but in homeward voyages, the catches are stored in these compartments with iced salt water, all sea connections being closed by screwing down the plugs on pipes, water being discharged by pump suction and blocks of ice being dropped in.

Ice stores · · · · · · · · · · · · Both sides of fish wells are used as ice stores. They are insulated with cork dust or cork sheets by top and sides, as shown in midship section plan.

Sponson for fishing platform · · · Steel bracket plates are fitted outside of side and stern bulwark two feet below the top of rail, and $1\frac{1}{2}''$ thick wooden planks are laid on, making a sponson platform $2' 6''$ wide for fishing operation.

3. Seine Fishing Twin Motor Boats :

Seine fishing by junks was extensively carried on in several districts of Japan from olden days. In 1909, it was first undertaken in a steamer off Hokkaido. Next year, the Fishing Experimental Bureau of Kyoto and Fukui Prefectures made a joint experiment in seine fishing with a motor boat. In 1913, bonito fishing boats in Fukushima and Ibaraki Prefectures began to be adapted to seine fishing in the leisure season during the winter. On the other hand, many motor boats with winches worked by a belt and gear connected to the main engine were built especially for this fishing in Kyushu and western parts of Honshu. A few years later, some fishermen in the latter districts introduced successfully a method of dragging the seine by two motor boats about 600 yards apart as a set. This method soon prevailed everywhere. The boats used became larger and larger,—20 tons, 30 tons, and so on. The seine process is similar to trawling, though the net used is different. As in the case of trawling this method of fishing was violently opposed by the coastal fishermen and zoologists. In 1921, the Government proclaimed areas off the coasts of Japan to be prohibited for this kind of fishing and issued regulations, by which these boats should obtain the sanction of the local Governments, who restricted the gross tonnage of boats to 50 tons. This drove fishermen to the trawling grounds of the China Sea with boats having a maximum tonnage of 50 tons.

At present, there are about 200 pairs of seine motor boats in the trawl areas, while the number of trawlers is restricted to seventy by the Japanese Government. This prohibition was a serious blow to trawlers, because seine fishing boats use similar size of nets and get the same catches or even better earnings than trawlers, the kinds of fishes caught fetching higher prices, notwithstanding the smaller size of their boats and accordingly smaller expenses. Here it might be observed that the trawlers improved their nets by adopting the "Vigneron Dahl" patent which partly resembles the Japanese seine, and they increased their sizes and went further out to sea into other fishing grounds. Some of them have adopted Diesel engines for propelling, and installed refrigerating plants for the longer voyages. On the other hand, seine fishers demand boats larger than 50 tons. The Formosa Government approves the size of boats up to 100 tons which may be the most suitable size; but boats too big are not suitable for this fishing from the point of maneuvering and economy. At present, the seine fishing boat can not work so hard as the trawlers in winter on account of the weather, and generally do not work at night. Thus, the trawlers and the seine boats seem to be complementary each having its own merits and working capacity. The total fleet of seine fishing boats working in the China Sea is valued at about Yen 15,000,000. Seine fishing twin motor boats should be strongly built, seaworthy and speedy, for they work in rough seas through the whole year in spite of the small tonnage and have to return quickly to port.

Plates 3 and 4 are the plans of the most up-to-date vessel belonging to a fishing company in Formosa. Particulars are as follows:—

Principal dimensions · · · · · ·	85′ 0″ × 17′ 0″ × 9′ 0″	
Tonnage · · · · · · · · · · · ·	Gross	87.2 tons
	Under deck	78.0 tons
	Net	31.6 tons
Sunken forecastle · · · · · · · ·	Height	2′ 9″
	Height of bulwark	2′ 9″
Sheers of the boat floating at designed trim of 3′ 0″· · · · ·	Forward	4′ 0″
	Aft	1′ 0″
Draft, displacement, &c. at loaded condition	Draft	7′ 10½″
	Displacement	170.8 tons
	GM	1.44 ft.
Main engine · · · · · · · · · ·	"Niigata M 4 S Diesel"	
	B.H.P.	150 at 350 revolutions
Propeller · · · · · · · · · · · ·	Manganese bronze, three bladed right handed screw propeller	
	Diameter	4′ 4″
	Pitch	3′ 1″
	Developed area	750 sq. ins.
Auxiliary machinery · · · · · ·	1) Emergency air compressor, driven by 2 H.P. "Tobata oil motor"	
	2) 2″ centrifugal pump for general service	
	3) "York ammonia compressor"	
	4) 1½″ "Inokuti centrifugal pump" for the cooling water of the ammonia condenser	
Trial results· · · · · · · · · ·	Draft forward	3′ 11″
	aft	8′ 4″
	Displacement	115.5 tons
	Block coefficient	0.49
	Prismatic coefficient	0.591
	Speed	9.45 knots
	Revolution	355

Number of crew	10
Tank capacities	Fuel oil tanks 675.22 cub. ft. Fresh water tanks 7.8 tons
Fish hold	Total volume 2710 cub. ft. Fish hold is divided into six rooms and can store 1,100 boxes in total, kept at 30° fahr. by the aid of refrigerating plant. Each box contains crashed ice and 50 lbs. of fishes. The total length of ammonia cooling pipe is 731′ 0″, the diameter being 1½″.
Fish hold insulation	Sides:—cork dust between frames, two ¾″ boards inserted with insulating paper between them, 4″ cork dust, two ¾″ boards with insulating paper. Top:—3″ cork sheet between beams, two ¾″ boards with insulating paper, two 2″ cork sheets, two ¾″ boards with insulating paper. Bottom:—cork cement between floors, 1½″ wooden ceiling, two 1½″ cork sheets, 1″ asphalt.
Refrigerating plant	4½ refrigerating tons "York" machinery of direct expansion system. The size of ammonia compressor is 4″ × 4′ 4″. It is driven by a belt connected by main engine, with a special apparatus to keep the revolution automatically constant.
Motor winch	The motor winch is driven by a belt connected to main engine. The power about 30 H.P. and the designed number of revolutions of the warp heads is 90/min.
Special fittings	1) Acetylen lighting apparatus. 2) Two cast iron rollers for hauling tow rope, fitted on wooden frames strongly built on the stern bulwark. 3) Two wire reels fitted on the deck near foremast, capable of winding 1800 ft. of ⅝″ dia. tow wire rope. 4) Two bow rollers for anchoring warp, sometimes in rough weather being used for hauling tow rope.

5) 2″ dia. steel horse fitted on the stern bulwark to hook the end of tow rope.
6) Fish ponds at the sides of hatchway on deck, built of cast iron stanchions and shiftable boards.
7) Net dyeing tank of 1 ton volume fitted on the starboard deck aft of sunken forecastle.

4. Floating Canneries for Crab Fishing:

The crab canning export industry has been carried on since 1907 or so on the coasts of Hokkaido, Chishima and Saghalin. To prevent the extinction of crabs, the Government issued regulations to prohibit this fishing within certain distance from the coasts of these districts. On that occasion, cod fishing smacks working in the Okhotsk Sea reported the abundance of crabs on the west coast of Kamchatka.

In 1920, the Fishery Experimental Bureau of Toyama Prefecture sent a schooner of 175 tons to the Okhotsk, equipped as a crab fishing boat and floating cannery; and it was a great success. Many people then endeavoured to transform larger cargo steamers for use in this profitable business. The Government issued regulations requiring official sanction for this type of fishery in order to restrict the number of boats and check production. At that time, there were seventeen steamers of 700 to 4,000 tons engaged in this business and the total product was valued at Yen 20,000,000. In spite of the restriction policy, a depression in price has appeared, bringing financial difficulties in its train. Consequently, amalgamations having been accomplished the year before last, there are now two large firms, one having nine boats and the other five. The floating canneries are equipped at Hakodate in the middle of April to sail for the fishing grounds, where they stay till the middle of August.

To explain further, let us take a steamer of 3,000 tons for example. Her holds are loaded with about 2,000 tons coal, 20,000 empty cans, and about 800 tons of fresh water. Besides crew, some 450 fishermen and other workmen are quartered in the aft 'tween deck. On both sides, eight motor junks are carried in davits.

These are about 40 ft. in length, 10 ft. in breadth, and fitted with 10 H.P. petroleum motors. They are chiefly used for hauling nets and transporting catches to the mother ship. Besides these motor junks, the floating cannery is accompanied by one or two self navigable larger motor boats of 30 to 80 tons. They are used for casting nets and for communication between the junks and the mother ship. They cast nets within a circle of about five miles radius with the mother ship as the center. In proper intervals of time, the mother ship changes her anchorage. The catches together with their nets are hauled up on the after upper deck of the mother ship by the derricks of the after mast from the junks coming alongside. The crabs are taken off from the nets and boiled in the steam kettles on deck. By means of derrick booms and tackles, net bags containing boiled crabs are once dipped into the sea outboards from the after deck, and hauled up to the forward upper deck. On the fore upper deck, crabs are cut in pieces and classified. They are carried down in baskets to the fore 'tween deck where they are canned, weighed, sealed and steamed by several machines. The finished cans are boxed and stored in the holds. A floating cannery of this size can produce about 350 boxes per day, each containing eight dozens of half-pound cans.

Plate 5 is the elevation and deck arrangement plans of a floating cnnerya of 4,000 tons. This steamer was equipped with crashing, pressing and drying machinery this year, to get fish oil and fish meal from crab waste.

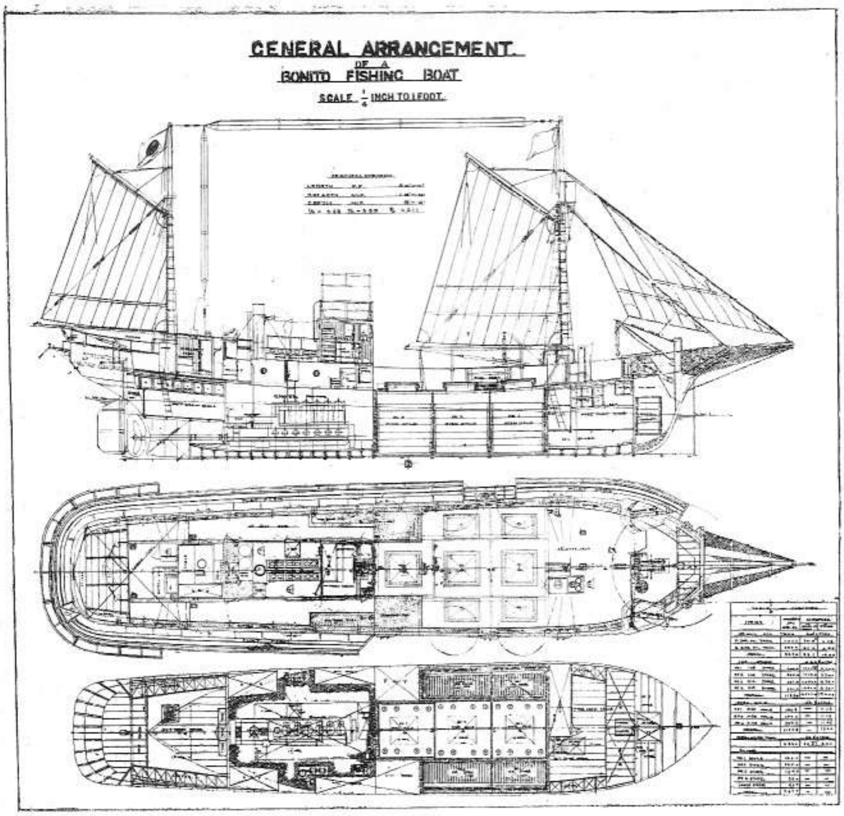


Plate 1

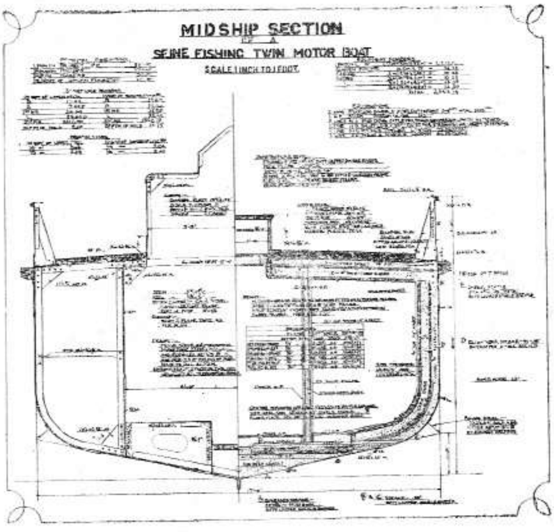


Plate 4

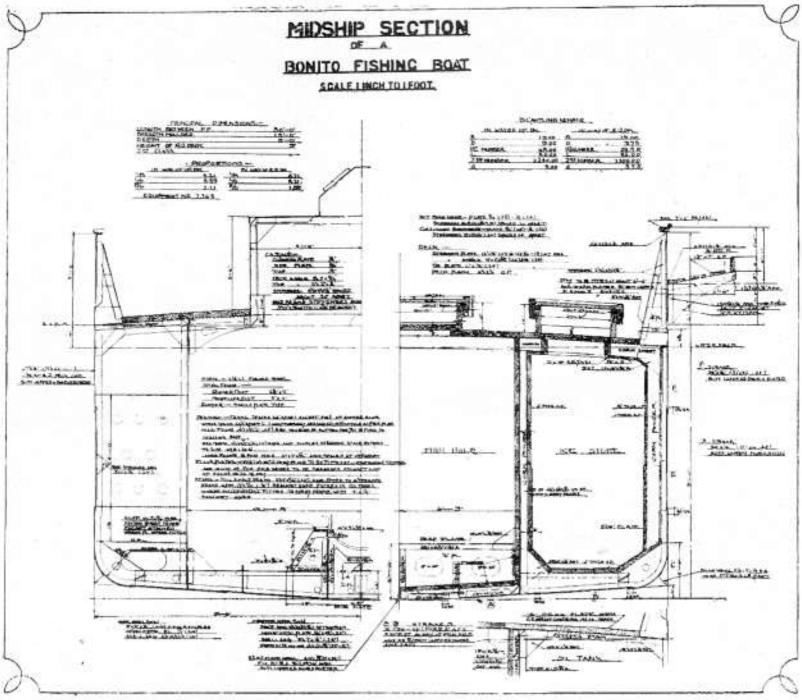


Plate 2

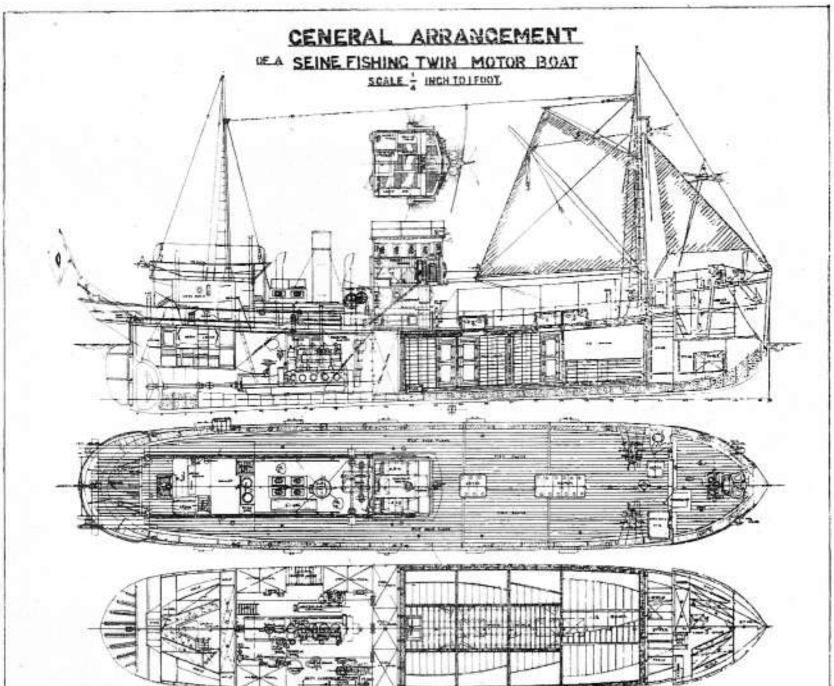
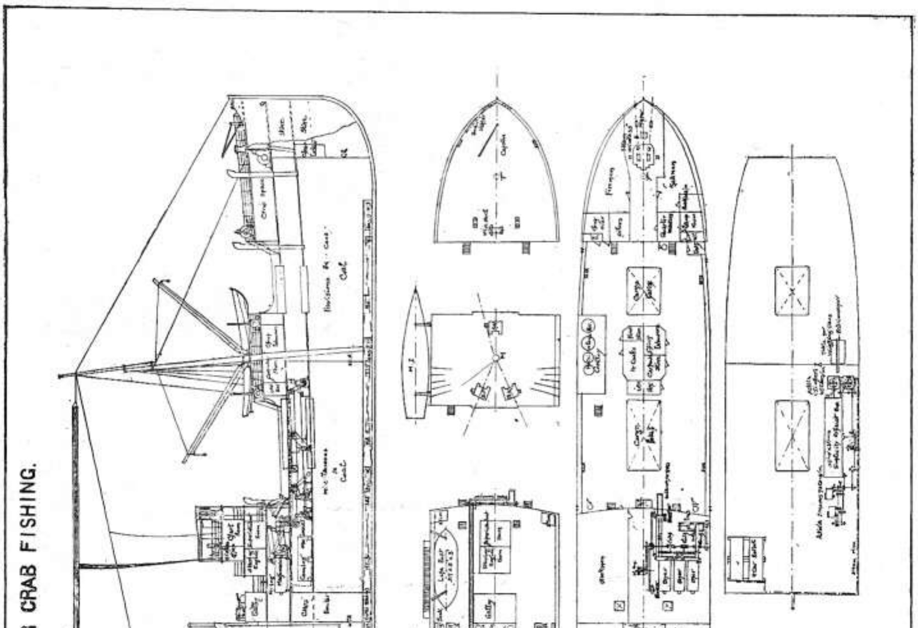


Plate 3



A FLOATING CANNERY FOR CRAB FISHING.

Plate 5

舶用蒸氣「タルビン」の主要構成部分の工作法に就て

(Notes on the Recent Constructing Process of the Principal Parts of Marine Steam Turbines.)

(Paper No. 52)

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〔一〕緒　　論

舶用機關として高速蒸氣「タルビン」の出現以來これが構成各部に要求せらるゝ材質の嚴選，工作の精密度は往日の比にあらず。特に軍用機關に於て然りとす。本論に於ては代表的のものとして衝動式蒸氣「タルビン」の翼車，翼，翼緣抑及噴口等の主要構成部分の代表的輓近工作法に就て述べんとす。

〔二〕「タルビン」翼車工作法

翼車材料として使用せらるゝものに炭素鋼あり，「マンガン」鋼あり，「ニツケルクローム」鋼あり，其の他「バイブラツク」鋼等種々あれ共，本項に於ては普通の硬質鋼即炭素鋼のみにつき，粗材選定の方法鍛造法熱處理法其の他に就き述べ，之れに關聯する諸檢査方法等を擧ぐるに止めんとす。

(イ)　粗材の選定

「タルビン」翼車は其の受くる應力比較的高からざるもの又は小型翼車にして「スタンプフオージ」を可とするもの等特種のものを除き，「インゴツト」より1個採りとするを原則とす。然して「インゴツト」は其の頂部より總重量の 40％ 以上を，又其の底部より5％ 以上を切り捨つるを要す。「インゴツト」は其の中央部の徑に於て凡そ 20/100 の外周鍛錬をなし，且つ翼車轂部鍛造後の高さの少くも $3\frac{1}{2}$ 倍なる長さを有する荒地を作り，之れを壓潰鍛造するものとす。

(ロ)　翼車鍛造法

翼車の鍛造の方法は汽鎚による場合，及水壓機による場合等あれ共，前者に於ては翼車の表面のみ鍛造せらるゝ嫌ひあるを以て後者によるを主とし，翼車鍛造後の「フローライン」を良好に保ち，鋼塊中心部の惡影響を少くし，且つ材料の組織の緻密なるを期するため，大凡鍛造に當り次の考慮を拂ふ。

(A)　粗材上下部は絞りて粗材の形狀を樽形（「セメント」樽の如き）に近からしめたるものを可とす。

(B)　仕上代は出來る丈け少なく鍛造す。

(C)　「ボス」部「リム」部「セギリ」用火造用具は充分丸味あるものを使用す。

鍛造用水壓機は出來る限り大力量のものを選び，鋼材を先づ外周に於て原徑の約

20/100 の外周鍛錬をしたる時，翼車鍛造後轂部の高さの少くも $3\frac{1}{2}$ 倍以上の長さを有する粗材とし，而も其の形狀は兩端部の徑を中央部より小さくせる樽形に近きものに作製するを可とす。而して之れを翼車轂部の高さに近く鍛造せる後「ハブ」部の「セギリ」を行ひ，次で「リム」部の鍛錬を行ひ續て「ハブ」部の鍛錬を行ふを普通とす。

鍛造に當りては上「タツプ」（ハンマー，ヘツド）は成る可く小なるものを選定し，普通 120 乃至 150 粍徑のものを以て「スクキーズ」するを可とす。鍛造中翼車は平等なる鍛錬を行ふため時々翼車を裏返す事必要にして，鍛錬溫度攝氏約 1150 度，鍛錬終期溫度攝氏約 800 度附近とするを普通とす。而して「リム」部鍛造後轂部を續て鍛造し，再加熱は極力之れを避けつゝあり。

加工用仕上代は翼車の大さによりて相違するは勿論なるも，大體直徑 1000 粍附近のものに對しては，片面約 10 粍，外徑は試驗片採取の關係もありて大凡直徑に於て 150 粍大きく鍛造するを普通とす。

（ハ） 翼車熱處理法

（A） 第 1 回熱處理

翼車は鍛造後，鍛造による組織の歪曲不同を匡正し，同時に適當なる組織的變化を與ふる趣旨に於て，硬質炭素鋼にありては左記の熱處理を行ふ。

燒鈍溫度 830 度（攝氏）

燒鈍溫度に達せしむる時間 7-8 時間 （翼車を均一加熱するため 700 度附近にて約 1 時間加熱を滯留するを良とす）

燒鈍溫度に保持時間 1.5 時間

冷却方法　大氣放冷

翼車は其の材質により加熱溫度及處理法に變化あり，又其の大さによりても加熱時間，加熱保留時間等も相違し，又冷却の方法もそれに應じ變更するの要あるは言を俟たざる所にして，上記記載の熱處理法は翼車外徑大凡 1000 粍にして翼車材質次の如きものに對する 1 實施例なりとす。

炭素	0.34	硅素	0.18
マンガン	0.4	燐	0.03
硫黄	0.027	「ニツケル」	1.49
「クローム」	0.25	銅	0.22

（B） 第 2 回熱處理

翼車は荒削後大凡下記により第 2 回の熱處理を行ひ，荒削時の內部應力及歪を除去するを例とす。

加熱溫度 50.0 度（攝氏）

加熱溫度に達する時間 5-6 時間

加熱溫度保留時間 $\frac{1}{2}$ 時間-1 時間

冷却方法　大氣放冷

（C） 翼車材料試驗

翼車は鍛造及第 1 回熱處理後に於て「リム」部及「ボス」部より夫々試驗片を採取し，良好なるものは機械加工に移し不良なるものは第 1 回熱處理を繰り返へし，更に不良なるものは廢却す。

(ニ) 翼車機械加工，組立及試驗法

翼車加工の順序は必ずしも一定のものにあらざるも其の1例を示せば，先づ轂部中心部を鑽通し各部の荒仕上を行ひたる後第2回熱處理を行ひ，次で翼車兩側面本仕上，軸車嵌合部本仕上，引拔用孔鑽通楔溝仕上と順を追て加工し，次て翼車は軸車に組立てられたる上翼溝の仕上に移り，然る後翼車を引拔き翼車個々につき翼植込作業をなすが如し。翼車は翼及緣抑裝備後，翼車個々の靜的釣合試驗及翼車組立の上軸車の靜的試驗を行ふものとす。然して翼車組立前各翼車の靜的自然振動數を計測し，要すれば自然振動數修正のため一部の加工をなすを例とす。

軸車靜的釣合試驗終了後は之れを動的釣合試驗機にかけ動的不釣合の有無を調査するものとし，是等の翼車及軸車の諸試驗終了後，艦內裝備と殆んど同一狀態に組立て蒸氣試驗を行ふものとす。

蒸氣試驗を行ふに當りては其の取付方法を成る可く艦內に於ける實情と同一ならしめ，前後進何れの回轉にありても振動少なく，又少くも全力回轉數の 1.2 倍の回轉數に於て約1時間の繼續運轉をなすものとし，是等の運轉中「タルビン」の振動が何れの方向にありても振幅小なるを要し，然らざる場合には其の振動原因を調査し，要すれば前記の諸試驗を再施行するものとす。蒸氣試驗に於ける許容振幅は「タルビン」の大さ，型式等により考慮す。

〔三〕「タルビン」翼工作法

「タルビン」翼材として使用せらるゝものに眞鍮あり，滿俺銅あり，燐青銅あり，又「ニツケルスチール」あり，不銹鋼等ありて，之れが工作法も其の材質により種々相違すと雖も，本論に於ては單に不銹鋼翼材の「スタンプフォージ」による工作法及之れが加工法等に就いて述べることゝす。

(イ) 粗材の選定

粗材は其の寸法大凡 40 粍角，長さ約 1500 粍のものを購入するを例とし，地疵，打疵，寄疵等なく，各鋼塊（インゴツト）毎に材料試驗片を採取し，之れに所定の熱處理を行ひ，所定の規格に合格せるものを採用す。

(ロ) 翼鍛造法

一、 鍛造用器具

「スタンプフオージ」用落下鎚は「スチームドリブン」式を使用し，其の力量 1/2 瓲附近のものを多く採用す。鍛造中の鰭張りは出來得る限り高溫度の間に於て，而も一擧動にして切り落すことの必要あるに付，右要求を滿足するため特種の構造の刄型を有する能力約 60 瓲の鰭張落し機械を使用す。

又翼材加熱爐は上下2段に區劃せられたる特種の構造を有するものを使用し，翼材は上段に於て攝氏約 800 度附近迄加熱し下段に移し，此處に於て攝氏約 1150 度附近に加熱するの方式にして，之れが加熱用燃料として重油を使用す。

以上の外鍛造作業中の地疵其の他の表面疵を完全に除去するため，研削砥石を設備すると共に，洗滌「タンク」を設け，「ドロツプフォージ」中の翼の表面に生ずる酸化皮を除去し，表面を美化すると共に地疵發見に便す。

洗滌「タンク」は 2 個を設備し其の1つには硫酸 2 と水 8，他の1つには鹽酸 5

と水 5 との混合液を入れ，兩者を併用するか，其の何れか 1 つのみを使用す。然れ共此等の「ピツクリング」のみにては「ドロツプフォージ」中の翼の表面の酸化層が悉く自然に落下する程度に到らざるを以て，回轉「ドラム」を設け其の中に「ピツクリング」せる翼材を入れ，金鋼砂及川砂の混合せるものと共に回轉せしむるを普通とす。

右回轉「ドラム」は毎分約 40 回轉程度のものにして，連續 30 分附近回轉せしめたる後翼材を取出し別に設備せる溫湯槽（溫度約攝氏 80 度）に入れ泥洗をなすものとす。

二、 金型の選定

金型用地金としては炭素鋼，「タングステン」鋼，又は「バナジウム」鋼等あれ共，普通に使用するものは「ニツケルクローム」鋼とす。

金型用「ダイス」は 3 瓲鋼塊より鍛錬製造するを普通とし，之れが鍛造は攝氏 1150 度乃至 800 度の範圍に於て繰り返へし所要の形狀となす。

斯くして鍛造せる金型は燒鈍爐に入れ，攝氏 800 度迄約 6 乃至 7 時間にて加熱し，同溫度に約 $1\frac{1}{2}$ 時間保留後爐內冷却を行ふものとす。抑も經濟的にして而も良好なる「スタンプフオージ」成品を得んとせば，適當なる金型材質を選定と共に適切なる金型の設計を要することは論を俟たざる所にして，然も金型の設計は鍛造品の材質形狀に左右せらるゝは勿論，鍛造品が鍛造後機械加工を要するや否や，又は熱處理を要するや否や等によりても影響せらるゝものにして，「タルビン」翼鍛造にありても，翼材質或は翼の形狀に從ひ金型も之れに應ずる特種の構造となすを要す。即ち「タルビン」翼にありては翼型の關係上上下「ダイス」の接觸面を單一平面のみとすることは殆んど不可能なるを以て，翼型により其の程度こそ異なれ，一般に翼鍛造に當りては上型が接觸面の傾斜面に沿ひ辷らんとする傾向を防ぐため，其の傾斜角の程度に應じ右の如き辷らんとする力に對抗すると共に必要に應し鍛造時の反動を吸收するため，特種の「バランス」すべき構造を必要とす。

「タルビン」翼「ダイス」設計に當り翼材の機械仕上代としては厚さに於て 2 粍，幅に於て 4 粍を限度とするを例とす。是れ翼の「エツヂ」の部に於ても充分なる鍛造を得せしめんとする趣旨によるものなり。而して鍛造時の翼の長さは加工上の便宜のため翼底部の仕上代，切代，及機械加工の際の掴み代等を考慮し，全長に對し約 25 粍を附加するを例とす。又實際鍛造品が加熱狀態と常溫狀態とに於て縮み代は，大凡長さ 100 粍につき 2.5 粍として設計するを普通とす。

金型は主として機械により所要の形狀に加工をなし，更に翼の「ゲージ」に合せ手仕上を行ふを普通とす。然して機械加工及手仕上を完了せる金型は燒入を行ふを普通とし，大凡攝氏 800 度に加熱後油燒入をなし，約 300 度に於て燒戾しを施行す。

三、 鍛造順序

40 粍角，1500 粍長の粗材は加熱爐に於て熱し，一旦約 250 粍又は 30 粍角に打延べ，次て角を落し適當の丸棒に打ち延ばし，翼 1 個の重量を見積り翼 1 個に對する長さを定め，然して翼の短きものは 2 個採り又は 4 個採りとし，其の他は 1 個採りとし，第 1 回の荒地取をなす。然る後洗滌槽に於て加熱鍛錬による酸化層を除去し，地疵の有無を檢査し，加熱後 1/4 瓲汽鎚に取付けられたる型「タツプ」により掴み部及植込部の「セギリ」及翼の長さを定む。

以上の作業を終りたる後「スタンプフォージ」に移るものにして，其の工程大凡次の如し。

(A) 第 1 回「ドロツプフォージ」，洗滌，表面檢査。

(B) 第 2 回「ドロツプフォージ」，鑄張落し，洗滌，表面檢査。

(C) 第 3 回「ドロツプフォージ」，鑄張落し，洗滌，表面檢査。

(D) 仕上「ドロツプフォージ」，「グラインダー」鑄張落し。

四、 熱處理法

翼材は右工程を終りたる後燒入を行ふものにして，其の燒入方法は豫熱爐に入れ約 25 分にして攝氏約 1000 度に達せしめ，同溫度に於て約 20 分保留後，豫め 1000 度の溫度に保留せる電氣爐に移し，材料の成分に應じ攝氏 1150 度又は 1200 度に加熱し，同溫度に 15 分以內保留後水燒入を行ふものとす。

然る後充分冷却せるを俟て取出し「サンドブラスト」にかけ，燒入のため生ぜる金垢及表面の「スケール」を除去し，精細なる表面檢査の後翼個々に就き其の硬度を測定し，硬度規格に合格のものゝみ機械加工に移すものとす。

不銹鋼にありても其の成分により熱處理法を異にするは當然にして前記のものは所謂不銹鋼に對するものゝ 1 例にして，不銹鐵に對する 1 例を示せば次の如し。

燒入溫度（攝氏）	950–1050
同溫度保熱時間	約 30 分
冷却方法	油
燒戻溫度（攝氏）	730–800
同溫度保留時間	30 分乃至 1 時間
冷却方法	大氣放冷

(ハ) 翼機械加工法

一、 加工順序

翼は殆んど總て「ミリリングマシン」による機械加工をなすものにして，其の加工の順序大凡次の如く，機械加工終了後翼の背面等の「ポリツシング」をなす。

加工順序（2 個採りの場合を示す）

一、 翼兩端取付部削	二、 翼幅削
三、 翼腹面仕上	四、 翼底削
五、 翼根背面仕上	六、 翼背面仕上
七、 翼根元本仕上	八、 中央部切斷
九、 翼根溝仕上	十、 翼尖端仕上

翼は右加工中歪み直しを數回行ひ，又仕上後翼の出入口緣等に充分なる丸味を與ふるものとす。

二、 加工工具燒入法

不銹鋼は一般普通炭素鋼に比し硬度高く切削困難なり。從て良好なる工具鋼の選定の必要なる事は勿論，之れが燒入法の如きに對しては特に深甚の注意を拂ふ必要あり。卽ち翼材の如く殆ど機械加工のみにより完成せらるゝものにして，然も加工度數多く且つ數量大なるものにありては，適當なる工具の選定が「タルビン」翼材として不銹鋼の適否を左右するに到るものにして，幸にして種々硏究の結果良好適切の工具燒入

法の成案を得たり。今之れが工具鋼につき簡單に其の燒入法を述ぶれば次の如し。

「カツター」地金としては適當の超高速度鋼を使用し,先づ「カツター」は瓦斯爐に於て漸次850度附近迄熱し次て之れを坩堝內に移す。坩堝內にては豫め木炭粉末を白熱狀態に熱し置き其の中に「カツター」を挿入し適當なる燒入溫度に達したる時迅速に採り出して油中に投入し,充分冷却せる後攝氏590度附近に於て燒戾を施行す。尙「カツター」は「サンドブラスト」にかけ其の表面を美化したる後研削機械又は油砥石により適當に仕上を行ひ實用に供するものとす。

斯くして得たる「カツター」は硬度「ショーア」の85以上にして其の表面頗る美しく且つ「ピツチング」等の痕跡なきを普通とす。

三、翼仕上公差

翼仕上寸法公差は總て「ゲージ」により嚴密に之れを守るものにして,翼根,翼幅,翼頂,翼尖端,翼長等緩急に應じ適當の寸法公差を附するものとす。

而して翼仕上重量公差は次の標準による。

(A) 翼の重量は翼10個宛10回重量を計測の上,各1個の平均重量を算出し,平均重量10回を平均し之を標準重量とす。

(B) 翼は1個毎に精密に重量を計測し,(A)項の標準重量に對し1%以上の增減を許さず。

(C) 同種類の翼にして1個の緣抑內に取付くる翼全部の重量は同數の標準重量の和に比し2/1000以上の增減なきを要す。

(ニ) 成品檢査の方法

翼は各個毎に其の硬度を計測し所定の規格內にあらしむると共に,同一熔解にて同時に最後に熱處理を施行せる1群の翼より必ず1個以上の試驗片を取り,抗張試驗冷質屈曲試驗を施行するを要し,試驗片の大さは規定のものを得難きにつき,抗張力試驗片は其の取り得る最大矩形の斷面を有するもの,又屈曲試驗にありては其の取り得る最大正方形のものとし試驗を行ふ。

機械加工を終りたる翼は曹達洗滌により工作中の油分を充分除去し,然る後海水函中に2晝夜浸し引き上げたる後,1晝夜空氣中に放置し疵の有無を詳細檢査するものとす。

(ホ) 翼及緣抑裝備法

翼は前記仕上公差及成品檢査に合格せるものゝみを取り,1個宛植込用具により翼車溝に植込むものにして,最後に止翼を入れ次て止金を挿入するものとす。止翼及止金の挿入を終りたる後緣抑を挿入す。緣抑取付は翼尖端を鋲締式に緊締する方法を採り大凡次の標準により之を裝備す。即緣抑は其の1個の含む翼數大凡10本を標準とし5本を最少限度とし之れを裝備し,而も其の1本の緣抑の長さを300粍以下とす。

緣抑の材質は翼材と同一のものを使用するを普通とす。而して不銹鋼緣抑の穿孔の方法は其の形狀に應じ「ドリル」又は「ポンチング」によるものにして,「ポンチ」せる場合には「ポンチング」による內部應力を除去するため攝氏約700度に於て約30分間燒鈍するを例とす。

緣抑絞締法用工具は平形のものゝみを使用し,約1听(0.5「キロ」)「ハンマー」により直上より打ち下すことなく四方より斜めに打ち下す方法を採用す。

〔四〕「タルビン」噴口工作法

「タルビン」噴口には蒸氣の性狀に應じ地金材質として鑄鐵を使用する場合あり。燐青銅其の他の銅合金を使用する場合あり。又噴口板にありても噴口の形式により使用材料を異にするものにして，其の種類廣汎に亘るも本論に於ては單に鑄鐵地金に對する平行噴口鑄込法，及燐青銅地金に對する膨脹噴口鑄込法のみにつき，簡單に其の工作法を述べんとす。

（イ） 鑄鐵地金に平行噴口鑄込の場合

噴口板材質としては「オープンファーネス」式又は坩堝式又は電氣爐により製作せられたる低炭素白銅鋼板を使用するを普通とし，鋼板の寸法公差は正負各 5％ 以下にして特定の試驗規格に合格せるものを採用す。

噴口板は所要の形狀に機械加工せる後之れを攝氏約700度に加熱し，熱間屈曲により完全に所要の形狀となす。然る後噴口板は乾燥鑄型に組立て溫度攝氏約 1250 度に於て鑄鐵を鑄込むを例とす。

鑄込の際に於て特に噴口板と地金との膚付きを良好ならしむるため，噴口板は鑄込前，噴口板鑄込部に半圓形切込を數個設くるを普通とす。

鑄込作業終了後噴口は正確に其の面積を計測し正 3％ 負 0％ の面積公差內にあるや否やを調査するものとす。然る後車室仕切は其の組立使用時受く可き最大壓力差の 1.5 倍を以て水壓試驗を施行し，各部共些の永久變形なきことを確むるものとす。

（ロ） 燐青銅地金に膨脹噴口鑄込の場合

噴口板及地金共燐青銅を使用し，唯前者即ち噴口板用に對しては地金材質より硬度高く「エロージョン」に耐へ得べき配合のものを使用す。

夫れ等の配合の 1 例を示せば次の如し。

噴口板		地金		
銅（％）	96.02	電氣分銅	（％）	86.0
燐（％）	0.27	錫	（％）	12.5
錫（％）	3.0	亞鉛	（％）	0.5
		燐銅（燐 15 ％）	（％）	1.0

模型用材料としては一般鑄物と同じく木材を使用するを普通とするも，膨脹噴口の如き特種の形狀にして而も精確なる鑄物を要求するものに對しては，木型に對しても特別の考慮を拂ふ必要あり。近來此等鑄物用模型に對し適切なる防濕塗裝法を完成し良好なる結果を收めつゝあり。即此等木型に對し先づ「アルミニウム」塗料を 1 回塗布し之れを乾燥せしめ，然る後「ラツク」塗料を 1 回刷毛を數回同一方向に擦過しつつ塗布し,「アルミニユーム」粉末をして木型の表面に鱗狀に配列せしめ防濕の目的を達成する方法にして，之れが塗裝法の應用により獨り木型のみならず噴口板駒として燒石膏を使用し得るに到り，益ゝ精巧なる膨脹噴口の鑄造を可能ならしむるに到れり。即ち燒石膏に適量の水（55％）及「セメント」の少量を加へ模型となし，之れに防濕塗料を施し鑄造用噴口の駒となすものにして，斯くの如くする時は型離れ一層良好にして仕上面美しく正確なる噴口面積を得らるゝものとす。鑄型は砂型を使用するを例とし鑄込溫度大凡攝氏 1050 度とす。

以上記載のものは今日一般に行はるゝ鑄造方法なりと雖も，尙噴口板と地金とに同一材質同一配合のものを使用せる例少なからず。此の場合に於ては噴口板に相當する部分には大凡錫 66 %，鉛 34 % の配合を有する可熔性合金を模型として使用し，之れを鑄型乾燥の際熔融流出せしめ（鑄型乾燥溫度約 250 度）然る後燐青銅を鑄込むものにして，工事費を節約し且つ正確なる鑄物を得らるゝ點に於て特徵を有す。

地金配合の 1 例を示せば次の如し。

銅 (%)	88.5	錫 (%)	10.0
ニッケル銅 (%)	0.5	燐銅 (%)	1.0
「マグネシウム」銅 (%)	0.02		

Developments in Diesel Engine Shipbuilding in Japan.

(Paper No. 85)

*By **Hyo Hamada.***

Foreword.

The development of the Diesel engine from a mere idea in 1892 up to the present time, is a most inspiring record to contemplate. The reasons for its phenomenal progress are simply explained in that it is the most economical source of power known today, and its economy, besides being independent of size of unit, remains constant through years of service.

That shipping circles in this country appreciate these advantages is amply confirmed by the great interest which is evidenced in this remarkable prime mover, and today it would seem that all other means of ship propulsion are likely to be overlooked in the enthusiasm displayed by ship owners in this engine.

In this paper, the author traces the history of the development in Japan as regards its application to marine propulsion, dating from the first installation fitted in a small fishing boat up to the present time, when the largest passenger liners flying the Japanese flag are Diesel engine driven. He also gives particulars of representative Diesel types and of the manufacture undertaken here.

General Description of Diesel-engined Vessels Built and Building.

Before proceeding to give details of Diesel engine ship construction in Japan, it may be well for me to refer briefly to the original development of this engine in Western countries immediately following the dawn of the Diesel epoch.

Upon Dr. Rudolf Diesel securing his first patent in Germany in 1892, and a second in the following year, for the engine which bears his renowned name, it will be remembered that the M.A.N. were quick to realise the possibilities of this novelty, and that, during the year 1893, they and Messrs. Fried Krupp, as well as Messrs. Sulzer Brothers, acquired manufacturing licenses from the inventor. Messrs. Carels Frères of Ghent in the following year also acquired similar rights, and produced their first engine of this type with a single cylinder. In the year 1898 Messrs. Burmeister & Wain entered into a license agreement and four years later, in 1902, completed their first unit.

From 1900 to 1914 the number of companies which took up licenses from those original builders, such as M.A.N., Carels Frères and Sulzer Brothers, was considerable, and the development achieved during the past 37 years since the engine was invented has been phenomenal, in the course of which its scope of use has been widened more and more from generator drive in power stations to rail car and locomotive drive, though without doubt its widest and most successful field of application has been for marine propulsion.

To turn from the West to the Far East, we find that oil engines first began to be used in Japan from comparatively early days, that is, from about the year 1905, for small fishing boats and harbour launches, and when I say that now fishing boats alone with these installations number as many as 15,000, it will be gathered that great strides have been made in its application, though, on the other hand, it must be admitted that in the manufacture of Diesel engines we have somewhat lagged behind European countries. It is a fact that an engine of this type was in 1919 utilized in a fishing boat, named the "Taiyo Maru No. 2" built in Shizuoka Prefecture, it being the pioneer installation for commercial purposes, though previous to that event, as from 1917, Diesel engines were used for propulsion purposes in submarine boats, being manufactured in Japan at the Naval Dockyards or by private engineering establishments, or imported from abroad. No merchant vessel of a large size, however, had at that time adopted the Diesel.

In the manufacture of oil engines and semi-Diesels, we find that the Niigata Engineering Works as well as the Ikegai Engineering Works have been producing them for many years, for use in fishing boats of small and medium sizes, and as main or auxiliary machinery for coasting cargo boats; and it was doubtless their association and long experience with such engines which influenced the former company to acquire first a license on the Mirrlees Diesel and later on similar rights covering the Nobel Diesel, and commence production of what is known today as the "Niigata" Diesel. Side by side with this new departure, the Ikegai Engineering Works invented a Diesel of their own design, on the basis of which manufacture was commenced.

In the meantime, the Department of Agriculture and Forestry of the Imperial Japanese Government, was ceaseless in its endeavours to emphasise throughout the country the Diesel advantages in its safer operation, easier starting and more economical fuel consumption than the oil engine or the semi-Diesel, and gave its strong support to influence the adoption of this type of machinery for fishing boats of large size. The Prefectural Authorities of Toyama took the lead in having a fishing flotilla leader built with Diesel propulsion, and the previous mentioned Government Department, with the idea of setting an example, placed an order in 1920 with the Ichikawa Shipyard for the construction of a fishing flotilla leader, named the "Kinshi Maru," and in the succeeding year entrusted the building of a similar vessel, the "Hakuho Maru" to the Hikoshima Works of the Mitsubishi Shipbuilding & Engineering Company. Prompted by this new departure, various Prefectural Authorities had their fishing guard boats and flotilla leaders installed with Diesel engine drive.

Though I have given particulars of the "Hakuho Maru" in Table I.

attached hereto, it might be convenient if I say here that she is a steel twin-screw vessel of 322-tons gross, and fitted with a four-stroke cycle single-acting Diesel of 300 B.H.P. capacity, manufactured by the Niigata Engineering Works. Again later on, we find the same Department of the Government ordering a Sulzer engined flotilla leader, the "Soyo Maru."

From about the years 1923 and 1924 interest had developed to such an extent in the Diesel engine that indications were visible that its application to larger passenger and cargo vessels was imminent. It fell to the lot of the Osaka Shosen Kaisha to be the first to fit this method of propulsion in a passenger vessel. The contract was placed with the Kobe Works of the Mitsubishi Shipbuilding & Engineering Company, and this motor boat, the "Ondo Maru," took the water in November 1923, and was placed on the Osaka-Sanyo service in February of the year following. She is indeed the pioneer motor passenger vessel in Japan. Her construction is of steel, twin screw, with a gross tonnage of 688-tons, and a length of 170 feet. The propelling machinery consists of 2 sets of Vickers four stroke cycle trunk piston airless-injection engines, developing 300 B.H.P. per set. The same steamship company added another such vessel, the "Kurenai Maru," which was launched in July 1924 from the shipyard of the Osaka Iron Works, and is fitted with two sets of 800 B.H.P. Burmeister & Wain's own engines. This boat is now on the run between Osaka and Beppu.

In view of the very satisfactory results obtained in these vessels, combined with the economy in fuel, the Osaka Shosen Kaisha decided to further increase their Diesel tonnage, and placed an order with the Kobe Works of the Mitsubishi Shipbuilding and Engineering Company, for two sister ships of the "Ondo Maru" type for their Sanyo service in the Seto Inland Sea. These coasting boats were followed by more vessels, namely, the "Nachi Maru" and "Muro Maru" for the Kishu service in 1926, and a larger sized "Midori Maru" and "Sumire Maru" in 1928 for the Beppu run, ordered also from the Kobe Works of the Mitsubishi Shipbuilding & Engineering Company. Details of these Beppu coasting liners are given in Tables I. and II.

If we turn to ocean going vessels, we find that the first to be installed with Diesel engine propulsion was the "Akagisan Maru," a cargo boat constructed at the Tama Dockyard of Mitsui & Company, and propelled by machinery supplied by Burmeister & Wain, which developed 2400 I.H.P. at the official trials and a speed of 12 knots at light draught. (Particulars are to be found in Tables I. and III). The auxiliary machinery in this vessel is eletric-driven and three Diesel engine-driven dynamos are installed to supply current.

Prior to this, the Osaka Shosen Kaisha had been planning an improvement in their South American round-the-world servie, and as a result of their investigations, decided to adopt the Sulzer type of Diesel for propulsion. Contracts were entered into with the Nagasaki Works of the Mitsubishi Shipbuilding & Engineering Company for the construction of the "Santos Maru" and two sister vessels, the "La Plata Maru" and "Monte Video Maru," the first of whih was launched in the year 1925, followed by the two others. It is interesting to note that, as a consequence of the splendid results obtained in actual service with these vessels, the steamship company in question was in 1928 influenced in its decision to build two more ships for the same run, and these are now in course of construction at the aforementioned Nagasaki Shipyard of the Mitsubishi Shipbuilding & Engineering Company.

The engines for the "Santos Maru" and the "La Plata Maru" were supplied by Messrs. Sulzer Brothers, Winterthur, but those for the "Monte Video Maru" are the product of the Mitsubishi, Nagasaki Works, and represent their maiden effort in the construction of Diesels. The unit of machinery, which was the biggest power ever produced in Japan at that time, consists of two sets of 6ST60 Sulzer type engines, giving a total normal output of 4600 B.H.P. Excellent results were obtained on the test bed and in actual service with these home-made engines, quite on a par with the splendid results Sulzer's own engines have given. For particulars of these engines, Tables I. and IV. may be referred to.

It must not be thought that other leading steamship companies operating under the Japanese flag were disposed to lag behind in this preference which was being shown for motor tonnage. It is to the credit of Japanese ship owners that they evidenced from the beginning a very keen interest in the progress which was being made in Western countries in the Dieselization programmes of contemporary European owners. The Nippon Yusen Kaisha, with the approach of the age limit of their San Francisco liners permitted under the provisions of the Ship Subsidy Acts of the Government, had for some years been conducting extensive investigations as to the type of ships they would order to replace the old craft. Expert engineers and naval achitects had been despatched to Europe to make exhaustive enquiries and obtain first hand information, and the placing of contracts in May 1927 with the Nagasaki Works of the Mitsubishi Shipbuilding & Engineering Company for the construction of two sister ships for their Orient–San Francisco line, was the outcome of those investigations. The keel of the first vessel was laid on September 10th and that of the second boat on December 10th 1927, the first vessel being launched on October 30th of the year following, and the other took the water at April 1929.

The first of these two units, which was christened "Asama Maru" was completed and put into commission in September this year. She is the largest passenger boat of her kind in Japan. Her main engines are of Sulzer type, designed for a sea speed of $17\frac{1}{2}$ knots with normal brake horse power of 16,000, operating quadruple screws.

Several months after the contract was signed for the above mentioned two vessels, the third sister ship was ordered by the Nippon Yusen Kaisha from the Yokohama Dock Company, and was launched at May of the present year. This boat is generally similar in design to the other two, but has two-propellers driven by four stroke cycle double-acting Diesel engines of the Burmeister & Wain type. Particulars of these three liners will be dealt with later.

Having satisfactorily settled the contracts covering the construction of its Orient–San Francisco liners, the Nippon Yusen Kaisha in 1928 turned its attention to another building programme which it has been contemplating. This programme called for three liners for the Seattle run, one for the South American service and two European Liners. All these vessels are now under construction. The Seattle Liners are to be fitted with main machinery of the B. & W. type and the European and South American boats with Sulzer type.

It will be gathered with what gratification these new building programmes were welcomed by the shipbuilders in Japan, as for a number of years the industry had suffered from depression, and whilst a similar condition prevailed in Europe after the World War, our Western contemporaries did not have to endure for so long the consequences of economic stagnation as fell to the lot of the shipbuilders in the Far East. It is doubtless true that a lot of Japanese tonnage was due for replacement about this time, or was approaching the age when economical operation could not be expected, but probably one of the leading factors in the revival in shipbuilding in this country may be attributed to the desire of owners to take advantage of the benefits accruing from Diesel propelled vessels.

The next large building programmes to be made public comprised three motor passenger and cargo boats which the Osaka Shosen Kaisha were in the market for, and the Dairen Steamship Company's plan covering four motor cargo vessels. These ships are all to be fitted with B. & W. engines and building contracts have already been concluded.

It might be of interest to record here, as an indication of the development made up to the end of December 1928, that the vessels under construction, excluing those of under 1000-tons gross, but not launched, at that time, in which Diesel engines were to be installed, numbered 21, of a total gross tonnage of 166,300 and a brake horse power of 130,300.

A glance at Table V. attached hereto, will reveal that Diesel boats launched during the period from 1920, when a vessel of over 100-tons was first built for Diesel drive, up to 1928, number 56, representing an aggregate tonnage of 146,774 and a brake horse power of 91,310. Those ships under construction at the end of December 1928, numbered 26 of a total tonnage of 167,740, and a power of 132,720 B.H.P. Besides, I should mention that two vessels should be added to this number, contracts for which were concluded at the time of the preparation of this paper.

When these figures are compared with those for Europe and America, it will be seen that they are far below those of our confreres in the west, but this in a measure may be due to the shortage of time which has elapsed since the engine began to be supplied for ship propulsion, and in view of what is taking place in our Empire shipping circles, we are confident that before long, we shall be able to contribute our due quota to the Diesel era in which we now live. It may not be without interest to refer to the Table VII. where a comparison of Diesel and steam vessels building during 1928 in Japan is given.

Perhaps I may now refer to the manufacturing side of the Diesel engine, and to give some information on the types generally adopted.

TYPES: Nine types of engines can be enumerated as being installed in this country in merchant vessels. For large units, pride of place must be given to the Sulzer system which has the most installations, while B. & W. comes second, with Vickers and the Niigata next in the order given. Several vessels have, however, been fitted with the Cammel--Laird-Fullager and the M.A.N. and Polar engine, but all these engines are of foreign make and none has been manufactured in this country. The Niigata engine is purely a home-made product, and most of the Sulzer and Vickers now in use were manufactured in Japanese engineering works. Tables VIII. and IX. show the number and some particulars of the domestic product and imported engines.

DIESEL ENGINE MANUFACTURING PLANTS IN JAPAN: Diesel engine makers in this country at the present time are the Imperial Naval Dockyards, the Niigata Engineering Works, the Ikegai Engineering Works, the Kobe Steel Works, the Mitsubishi Shipbuilding & Engineering Company, the Kawasaki Dockyard Company Limited and Mitsui & Company Limited. Though the Kawasaki Dockyard Company has a license, no production has yet been undertaken of Diesels for merchant vessels, and likewise the Yokohama Dock Company although possessing rights, have not so far commenced manufacture. The companies who have already experience in connection with these motors or have engine in course of construction, number five, and manufacture is undertaken at six different works.

In view of their Diesel engineering activities, I should now like to be permitted to give some information as to the origin and history of these private manufacturers.

1. *The Niigata Engineering Works Limited* has its Diesel shop situated at Kamata, a suburb of Tokyo. The company was established in the vicinity of Niigata City, in Niigata Prefecture, in the year 1895, as a subsidiary of the Nihon Petroleum Company, with the object of engaging in the manufacture of oil well boring machines and machine tools, as well as oil engines. It was in 1908 that it took up the production of oil engines for fishing boats, and in 1915 works for the exclusive production of these motors was erected by them at Tsukishima, Tokyo. Their connection with Diesel engine manufacture may be said to have started in 1917, when they acquired a license from Messrs. Mirrlees Bickerton & Day Limited of Stockport, England, for

the four stroke cycle engine. In 1920 one 100 B.H.P. engine for marine use and one 300 B.H.P. were completed for the first time. Simultaneously with the completion of these engines, the company removed its works to the present site at Kamata. They further entered into a manufacturing arrangement with Nobel Diesel Aktiebolaget of Sweden in 1922. A 1500 B.H.P. engine has already been turned out under this latter license, and is now operating in a floating cannery. Their marine four cycle standard engines range from 50 to 320 B.H.P. and are uniformly of the trunk-piston type, whilst their marine two stroke standard engines range from 400 B.H.P. up to 4,000 B.H.P.; for engines of an output below 1000 the trunk-piston type is adopted.

2. *The Kobe Steel Works Limited.* This company was founded in 1906 in the city of Kobe, and undertakes the manufacture of steel, cast steel, forged steel, machinery and various kinds of apparatus and tools. In 1912 they converted their organization into a joint stock company, and afterwards in 1921 amalgamated with the Harima Dockyard and the Toba Shipbuilding & Electrical Engineering Works. Their association with the Diesel dates from the time when they acquired a Sulzer license, at first devoting their attention to installations for warships, later on extending their activities to cover merchant vessels. The "Yahiko Maru" of 9,100-tons gross launched in July 1926 is fitted with two sets of 1500 B.H.P. Diesels produced by them, while the hull was constructed at their Harima Dockyard. Other propelling machinery turned out by this works comprises engines for two of the three Osaka Shosen Kaisha "Choan Maru" type cargo and passenger ships. One set for each boat develops 2300 B.H.P.

3. *The Mitsubishi Shipbuilding & Engineering Company, Ltd.* This company had its origin in a small shipyard founded at Nagasaki by the Tokugawa Shogunate in 1856, about 73 years ago, which plant was later leased to and afterwards completely transferred to Mitsubishi interests, who in 1917 converted their shipbuilding and engineering activities into a joint stock company. In addition to this Nagasaki Shipyard and Engine Works, they have a large similar plant at Kobe, as well as one at Hikoshima, near Shimonoseki, with a Research Laboratory in Tokyo and an Arms Works in Nagasaki. Their principal undertakings range from the building and repairing of warships, merchant vessels and machinery (for both marine and land purposes), armaments, iron and steel foundry work, &c. down to pumps, fans and small apparatus.

Their connection with the Diesel dates from 1917 when they secured a license for the Vickers four stroke airless-injection engine, to be manufactured at their Kobe Works for submarine boats built in their Shipyard there, and later the rights were acquired on this engine for merchant vessels. The engines they have produced under this systm both for warship and mercantile use from 1920 up to 1928 number 41 sets with 442 cylinders, of a total power of 49,000 B.H.P. The company now also manufacture land

Diesel engines on their own system for coupling to Mitsubishi dynamos, a product of their associated electrical engineering company.

In 1924 they also took up a Sulzer license, and this system is principally manufactured at its Nagasaki plant. The main machinery for the Osaka Shosen Kaisha South American passenger and cargo vessel, "Monte Video Maru," constructed in 1926, represents the first work there on Diesels, and eight more sets of engines of the 6ST60 type with an aggregate of 20,700 B.H.P. in 54 cylinders, have already been completed up to December 1928. Those engines in course of construction at the Nagasaki Works at the end of 1928 numbered 15 sets, of a total of 58,300 B.H.P. in 112 cylinders.

The Kobe Works of this company also manufactured one set of this type of a power of 1500 B.H.P.

4. *Mitsui & Company, Limited.* They have their shipyard and engine works at Tama, near the city of Okayama. The company itself is very old having been founded in 1876, though their shipbuilding activities date only from the year 1918. Since the completion of their first dock in 1919, they have engaged in the construction and repair of various types of warships and merchant vessels and machinery. In 1927 they acquired a license for the Burmeister & Wain four-cycle Diesel engine, and in the following year undertook the manufacture under this license, of the engines for the cargo boats "Takamisan Maru" and "Tatsutasan Maru," built for themselves. Both vessels carry a deadweight of 3,000 tons and are of single-screw type. Each engine develops 950 B.H.P. in 6 cylinders. At the end of December 1928 they had in hand the construction of the hull and machinery for one 9800 tons deadweight cargo boat, named the "Hakonesan Maru" for themselves, and the hulls and engines of two 4150-ton deadweight cargo motor vessels for the Dairen Steamship Company. The engines total three sets with an aggregate of 7000 B.H.P. Two more ships are under construction but their engines will be procured from Messrs. Burmeister & Wain.

5. *The Kawasaki Dockyard Company.* Their plant is located in Kobe, and their shipyard is one of the oldest and largest in the Empire, originating in the era of the Shogunate, similar to the Mitsubishi Dockyard at Nagasaki. The present works at Kobe was started in about 1871. Their principal undertakings are the construction and repair of warships, merchant vessels and machinery, as well as steel manufacture. They possess two Diesel engine licenses, namely, the M.A.N. and the Cammell-Laird-Fullager; the engines under the former license have been manufactured for warship installation, but so far no production is recorded under either of these licenses for merchant craft. They have once fitted a Fullager eigine of John Brown make in a vessel built in their shipyard.

6. *The Yokohama Dock Company* with its plant situated in that port, was founded in 1893. At the outset its activities were confined to docking and towing, but in 1917 a big expansion was planned and since the completion of that scheme in 1919, it embarked upon the construction of ships

and the manufacture of machinery. In regard to Diesel engines, they have licenses for the Polar and the M.A.N., but have not up to the present carried out any manufacture. Their shipbuilding berths at the present time are occupied with one Seattle liner for the Nippon Yusen Kaisha, and two cargo and passenger vessels for the Osaka Shosen Kaisha's Australian service. These ships when completed will all be among the typical Diesel engined boats in Japan. It should be mentioned that all the engines for these vessels are to be imported from Copenhagen.

7. *The Ikegai Engineering Works Limited* was founded in the year 1890 and is one of the oldest machine tool makers in the country. It was converted into a limited company in 1913, and besides manufacturing machine tools has been undertaking the production of oil engines from early times. Through the experience gained in the latter, they came to make a four stroke cycle Diesel. In 1923 they succeeded in the manufacture of the airless-injection system, and since then it has been adopted for all engines produced by them. Their standard marine engines range from 75 B.H.P. in three cylinders up to 1,000 B.H.P. in 8 cylinders, but the biggest unit actually undertaken is one devolping 320 B.H.P. in 6 cylinders of 280 mm. diameter. Their engines are principally used for propulsion purposes in fishing craft and fishing flotilla leaders.

Representative Marine Diesel Engines.

The Diesel tonnage at present (January 1929) under construction of over 10,000 B.H.P. per ship, is eight vessels. Above all, the Nippon Yusen Kaisha Orient–San Francisco Liners, "Asama Maru," "Tatsuta Maru" and "Chichibu Maru," may well be called representative motor ships in this country. As a matter of interest, a photograph of the "Asama Maru" and general arrangement drawings, as well as photographs of the Lounge and Smoking Room, of this vessel are attached hereto.

As was previously stated, the first two of these boats were built at the Nagasaki Works of the Mitsubishi Shipbuilding & Engineering Company, while the third is in hand at the Yokohama Dock Company. Particulars of the propelling machinery for these three vessels are given in Table X. The Sulzer system was adopted for the first two boats, and B. & W. for the other. The ships taking the Sulzer type engines are of quadruple screw and the B. & W. boat is twin-screw. The difference in the types of engines has naturally caused some changes to be made in the construction of the hull and interior arrangements, and especially in order to secure stability, the "Chichibu Maru" is designed to have an increase in width of two feet and in depth by six inches over the other two vessels.

At the commencement of the designing of the first two boats, when it had been decided that the Sulzer type engines would be installed, the builders in selecting the engine cylinder diameter and the number of cylinders, set about the study and investigation of the influence these would have on the propelling efficiency, on the arrangement of the engine room, hull construction, cabin arrangement, &c., while on the other hand, experiments were made at the builders' Experimental Tank in Nagasaki, with models to ascertain the hull resistance and hull and propeller efficiencies of twin and quadruple screws. As a result of these investigations, it was ascertained that about 13,500 B.H.P. is required to obtain a service speed of 17½ knots whether by twin or quadruple screws, and the latter was finally decided upon. Furthermore, in order to ensure the punctual fulfilment of sailing schedules, considering the government subsidy service for which the vessels are intended, it was decided, on the desire of the owners, that the engines should have a reserve power of about 2500, totalling 16,000 B.H.P. at normal condition with 15% overload.

In working out the design, it was arranged, after close consultation with Sulzer's engineers, that each engine should have 8 cylinders of 680 mm. According to the standard practice of Sulzer Brothers, a cylinder of 680 mm. should have a stroke of 1200 mm., but in order to have the engines housed under the second deck, the stroke had to be decreased to 1000 mm., yet without any anxiety that the efficiency would be decreased thereby, as was proved at the shop tests and sea-trials. The engines for the first boat were built at Sulzer's Works, Winterthur, and those for the second vessel at the Mitsubishi Works, Nagasaki.

The first vessel, the "Asama Maru," underwent sea-trials during the period from July 26th to August 10th at Nagasaki, and it may be appropriate to give here some particulars of the performance of this ship.

On the full power trial, she attained a speed of 20.7 knots, that is, the mean speed of three continuous double runs on the measured mile off Miye, the maximum speed reached on that trial being 21 knots. The engines developed on that occasion a collective B.H.P. of 18,900 at 126 r.p.m. In the fuel oil consumption test, the engines developed 13,756 B.H.P. at 113 r.p.m., the consumption being 172 grammes per B.H.P. per hour for the main machinery only. A trial of 24 hours' duration was also carried out at 19.65 knots with 15,450 B.H.P. at 119 r.p.m.

In all of the trials, the engines functioned with the utmost smoothness, and the results obtained exceeded all expectations, which occasioned the greatest satisfaction to the owners and builders alike. What is most gratifying to record was the singular absence of vibration in the ship.

The test results obtained on the engines of Mitsubishi make, were on the whole uniform with those of the Winterthur product. The main engines for the third boat have also completed their shop tests at the works of Burmeister & Wain. These latter engines are of the four stroke cycle 8 cylinder double-acting system, being duplicates of those installed in the Trans-Atlantic Liner "Kungsholm" of the Svenska-America Line, in Gothenburg, and the main air compressors are independently installed in the auxiliary engine room.

It is anticipated that the third vessel, the "Chichibu Maru," will be ready for trials about the same time as the second boat, the "Tatsuta Maru," and both will be commissioned in March next year.

Future Development of Diesel Engined Ships in Japan.

Japan being a sea-girt country, it will easily be realised that she has been interested and concerned in shipbuilding from remote times, but the foreign exclusion policy tenaciously enforced by the Tokugawa Shogunate, which banned foreign trade and the construction of big vessels, did a great deal to hamper development of the industry. At the time of the Meiji Restoration, however, the European practice of ship construction was adopted, and after the China-Japan War in 1894–5, rapid progress was witnessed, so much so that the warships and merchantt vessels using steam increased to such an extent as to place the country among the leading maritime nations of the world.

Japan being a coal producing country, and having little oil resources, the adoption of Diesel propulsion made very slow progress at first, but about the year 1920, engine makers here began to concentrate on the study of this prime mover, which was making such unprecedented progress in the West, and with this object constructed experimental engines themselves. At the same time extensive propaganda was undertaken so as to bring home to the shipping lines the great advantages which would accrue from the Dieselization of their tonnage. Shipowners too were anxious to make their own investigations and studies on Diesel engined vessels operating abroad, and it was not long before they were convinced that, notwithstanding even the disadvantage the country is confronted with in the matter of fuel for this prime mover, their future ship construction would have to be based on Diesel propulsion.

Thus the new ship construction during 1924 and 1925 showed a gradual increase in Diesel installations, and in 1928 the year presented such an aspect that practically almost all vessels from ocean going liners down to fishing craft called for Diesel engines for propulsion.

On the part of the engine makers, it must be stated that they showed untiring zeal in their efforts to bring about satisfactory results in their respective investigations and experiments, and the success relating to airless injection is the outcome of their labours. Their studies, it might be mentioned, cover methods of scavenging, constructional details of every part of the engine, materials, &c.

It must not be considered that engine makers here concentrated their efforts solely on engines for ship's use. Such manufacturers as the Niigata Engineering Works, the Ikegai Engineering Works and the Kobe Works of the Mitsubishi Shipbuilding & Engineering Company, came to manufacture Diesels for coupling to land and marine pumps, dynamos, &c. Such makers as the Niigata have made various Diesel machinery for marine and land purposes, and their accomplishment since they took up such manufacture, reaches the high mark of about 380 sets, with a total output of more than 48,000 B.H.P. The various units produced by the Ikegai Engineering Works during the years 1927 and 1928 alone count as many as 65 sets, developing about 6,400 B.H.P. It is most gratifying to know that all the manufactures of Diesels in Japan have given the utmost satisfaction, and now the home-made product, be it for main or auxiliary machinery, for men-of-war or merchant vessels, enjoys a high reputation for its reliability and for the workmanship put into it, which reputation it is worthy of mention is not confined within the limits of the Empire, and such being the case and bearing in mind the enthusiasm of ship owners to have their tonnage on the latest and most improved lines, who can doubt that there lies a great future for Diesel shipbuilding in Japan?

TABLE I (A). List of Diesel Engined Merchant Vessels Launched or Completed in Japan in the Past Nine Years (from 1920 to 1928).
Warships and Vessels Under 100-tons Gross are Excluded.

Name of Vessel		Owners	Built by	Engined by	Gross Tonnage, Tons.	Engine total Output, B.H.P.	Hull Lpp. × Bm. × Dm. Ft.-Ins.	Designed Speed, Knots	No. and Type of Engines		No. of Cylinders per Engine and Bore × Stroke mm.	Type of Vessel	When Launched
Kureha Maru	呉羽丸	Toyama Prefecture	Tsukishima Shipbuilding Co.	Kobe Steel Works, Ltd.	174	150	104'-0"×26'-6"×13'-0"		1-Sulzer	2 Cycle S.A.	4- 310 ×500	Fishing Flotilla Leader	April, 1920
Kinshi Maru	金鵄丸	Fisheries Bureau	Ishikawa Shipbuilding Co.	Niigata Engineering Works,	163	200	96'-2"×26'-0"×10'-8"		1-Niigata	4 Cycle S.A.	4- 228.6×304.8	do.	July, 1921
Hakubō Maru	白鳳丸	do.	Hikoshima Works, Mitsubishi Zosen Kaisha, Ltd.	do.	332	600	130'-0"×24'-9"×13'-9"	12	2-Niigata	4 Cycle S.A.	6- 304.8×457.2	do.	February, 1922
Ondo Maru	音戸丸	Osaka Shosen Kaisha, Ltd.	Kobe Works, Mitsubishi Zosen Kaisha, Ltd.	Vickers, Ltd.	688	"	170'-0"×28'-0"×18'-3"	12	1-Vickers	4 Cycle S.A.	6- 463.5×685.8	Inland Sea, Cargo & Passenger	November, 1923
Sōyō Maru	蒼鷹丸	Fisheries Bureau	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	Kobe Steel Works, Ltd.	202	330	112'-0"×21'-3"×12'-3"	10	1-Sulzer	2 Cycle S.A.	4- 310 ×500	Fishing Flotilla Leader	January, 1923
Akagisan Maru	赤城山丸	Mitsui Bussan Kaisha, Ltd.	Tama Dockyard, Mitsui Bussan Kaisha, Ltd.	Burmeister and Wain	4,634	1,800	375'-0"×50'-0"×30'-0"	10½	1-B. & W.	4 Cycle S.A.	6- 740 ×1500	Transpacific Cargo	March, 1924
Banshū Maru, No. 26	第二十六播州丸	Hayashi Kane Co.	Kobe Works, Mitsubishi Zosen Kaisha, Ltd.	Sulzer Brothers	225	410	115'-0"×21'-0"×11'-3"	10	1-Sulzer	2 Cycle S.A.	4- 340 ×540	Fish Cold Storage	July, 1924
" " No. 27	第二十七播州丸	do.	do.	do.	"	"	"	"	"	"	"	do.	" "
Kurenai Maru	紅丸	Osaka Shosen Kaisha, Ltd.	Osaka Iron Works, Ltd.	Burmeister and Wain	1,540	1,600	238'-0"×38'-0"×19'-6"	13½	2-B. & W.	4 Cycle S.A.	6- 500 ×900 Combined	Osaka-Beppu Line, Passenger	" "
Florida Maru	ふろりだ丸	Kawasaki Kisen Kaisha, Ltd.	Kawasaki Dockyard Co., Ltd.	John Brown Co., Ltd.	5,845	2,500	405'-0"×53'-0"×34'-0"	13	1-Camellaird Fullaga	2-Cycle S.A.	6- 558.8×1676.4	K-Line, Cargo	October, 1924
Fukko Maru	復興丸	Suzuki Co., Ltd.	Harima Dockyard	Sulzer Brothers	3,834	1,600	350'-0"×50'-0"×27'-0"	11	2-Sulzer	2 Cycle S.A.	4- 470 ×740	Transpacific Cargo	September, 1924
Hayatomo Maru	早鞆丸	Osaka Shosen Kaisha, Ltd.	Kobe Works, Mitsubishi Zosen Kaisha, Ltd.	Vickers, Ltd.	697	600	170'-0"×28'-8"×18'-0"	12	1-Vickers	4 Cycle S.A.	6- 403.5×685.8	Inland Sea, Cargo & Passenger	December, 1924
Mihara Maru	三原丸	do.	do.	do.	"	"	"	"	"	"	"	do.	January, 1925
Chikubushima Maru	竹生島丸	Taiko Kisen Kaisha, Ltd.	Tama Dockyard, Mitsui Bussan Kaisha, Ltd.	Sulzer Brothers	262	500	110'-0"×24'-0"× 7'-3"	"	2-Sulzer	2 Cycle S.A.	4- 310 ×420	Lake Biwa Excursion	June, 1925
Urashima Maru	浦島丸	Tokyowan Kisen Kaisha, Ltd.	Yokohama Dock Co., Ltd.	Atlas Diesel Akt.	195	220	100'-0"×20'-6"×13'-0"	9	1-Polar	2 Cycle S.A.	4- 290 ×430	Coastal Passenger	August, 1925
Santos Maru	さんとす丸	Osaka Shosen Kaisha, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	Sulzer Brothers	7,266	4,600	430'-0"×56'-0"×36'-0"	14¾	2-Sulzer	2 Cycle S.A.	6- 600 ×1060	S. American Line, Cargo and Passenger	September, 1925
Ebisu Maru	えびす丸	Tokyowan Kisen Kaisha, Ltd.	Yokohama Dock Co., Ltd.	Atlas Diesel Akt.	195	220	100'-0"×20'-6"×13'-0"	9	1-Polar	2 Cycle S.A.	4- 290 ×430	Coastal Passenger	October, 1925
La Plata Maru	らぷらた丸	Osaka Shosen Kaisha, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	Sulzer Brothers	7,266	4,600	430'-0"×56'-0"×36'-0"	14¾	2-Sulzer	2 Cycle S.A.	6- 600 ×1060	S. American Line, Cargo and Passenger	December, 1925
Urupp Maru	得撫丸	Fisheries Bureau	do.	Niigata Engineering Works, Ltd.	224	320	112'-0"×21'-9"×12'-0"	10	1-Niigata	4 Cycle S.A.	6- 304.8×457.2	Fishing Flotilla Leader	January, 1926
Shizuoka Maru	静岡丸	Hayashi Kane Co.	Kanesashi Shipbuilding Co.	Kobe Steel Works, Ltd.	138	220	92'-3"×19'-3"×9'-11"	"	1-Sulzer	2 Cycle S.A.	4- 250 ×400	Fish Cold Storage	March, 1926
Monte Video Maru	もんてびでお丸	Osaka Shosen Kaisha, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	7,266	4,600	430'-0"×56'-0"×36'-0"	14¾	2-Mitsubishi-Sulzer	2 Cycle S.A.	6- 600 ×1060 Combined	S. American Line, Cargo and Passenger	April, 1926
Cuba Maru	玖馬丸	Kawasaki Kisen Kaisha, Ltd.	Kawasaki Dockyard Co., Ltd.	John Brown Co., Ltd.	5,950	2,500	405'-0"×53'-0"×34'-0"	13	1-Camellaird Fullagar	2 Cycle S.A.	6- 558.8×1676.4	K-Line, Cargo	" "
Yahiko Maru	彌彦丸	Miyakichi Itaya	Harima Dockyard	Kobe Steel Works, Ltd.	5,742	3,000	416'-0"×54'-6"×31'-6"	11	2-Sulzer	2 Cycle S.A.	4- 630 ×1060	Transpacific Cargo (Lumber)	July, 1926
Unsen Maru, No. 1	第一運銑丸	Imperial Steel Works	Kobe Works, Mitsubishi Zosen Kaisha, Ltd.	Sulzer Brothers	106	120	80'-0"×22'-0"× 8'-0"	6	1-Sulzer	2 Cycle S.A.	4- 240 ×340	Ore Carrier	August, 1926
Nachi Maru	那智丸	Osaka Shosen Kaisha, Ltd.	do.	Kobe Works, Mitsubishi Zosen Kaisha, Ltd.	1,600	1,200	230'-0"×37'-0"×20'-0"	13½	2-Mitsubishi-Vickers	4 Cycle S.A.	6- 433.5×685.8	Osaka-Kishu Line, Cargo and Passenger	October, 1926
Muro Maru	牟婁丸	do.	do.	do.	"	"	"	"	"	"	"	do.	December, 1926
Chōan Maru	長安丸	do.	Harima Dockyard	Kobe Steel Works, Ltd.	2,611	2,300	284'-6"×43'-6"×23'-0"	14¼	1-Sulzer	2 Cycle S.A.	6- 100 ×1060	Japan-China Line, Cargo and Passenger	" "
Chōjō Maru	長城丸	do.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	2,594	"	"	"	"	"	"	do.	" "
Kōyasan Maru	高野山丸	Mitsui Bussan Kaisha, Ltd.	Tama Dockyard, Mitsui Bussan Kaisha, Ltd.	Burmeister and Wain	1,998	950	273'-6"×40'-9"×21'-0"	11½	1-B. & W.	4 Cycle S.A.	6- 500 ×900	Transpacific Cargo	February, 1924
Shunpū Maru	春風丸	Ocean Meteorological Observatory	Kobe Works, Mitsubishi Zosen Kaisha, Ltd.	M. A. N.	124	150	90'-0"×18'-0"× 9'-6"	9	1-M. A. N.	4 Cycle S.A.	6- 210 ×300	Meteorological Observation	" "
Shōhō Maru	翔鳳丸	Fisheries Bureau	Osaka Iron Works, Ltd.	Niigata Engineering Works, Ltd.	173	300	110'-0"×21'-0"×10'-0"	10½	1-Niigata	4 Cycle S.A.	6- 304.8×457.2	Fishing Flotilla Leader	March, 1927
Oshiro Maru	忍路丸	Imperial University of Hokkaido	Toba Dockyard	Kobe Steel Works, Ltd.	471	500	135'-0"×30'-0"×15'-6"		1-Sulzer	2 Cycle S.A.	4- 380 ×600	do.	" "
Momotarō Maru	桃太郎丸	Taiko Kisen Kaisha, Ltd.	Harima Dockyard	Sulzer Brothers	215	350	110'-0"×22'-0"× 7'-3"	12	"	"	4- 300 ×400	Lake Biwa Excursion	" "
Chōkō Maru	長江丸	Osaka Shosen Kaisha, Ltd.	Osaka Iron Works, Ltd.	Kobe Steel Works, Ltd.	2,613	2,300	284'-6"×43'-6"×23'-0"	14¼	"	"	6- 600 ×1060	Japan-China Line, Cargo	April, 1927
San Pedro Maru	さんぺどろ丸	Mitsubishi Shoji Kaisha, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	7,268	"	430'-0"×57'-0"×34'-0"	12½	"	"	"	Transpacific Tanker	May, 1927
Kuramasan Maru	鞍馬山丸	Mitsui Bussan Kaisha, Ltd.	Tama Dockyard, Mitsui Bussan Kaisha, Ltd.	Burmeister and Wain	1,995	950	273'-6"×40'-9"×21'-0"	"	1-B. & W.	4 Cycle S.A.	6- 500 ×900	Transpacific Cargo	" "
Columbia Maru	ころんびあ丸	Mitsubishi Shoji Kaisha, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	5,611	2,300	405'-0"×55'-0"×32'-0"	13	1-Sulzer	2 Cycle S.A.	6- 600 ×1060	do.	June, 127
Fuji Maru	富士丸	Shizuoka Prefecture	Osaka Iron Works, Ltd.	Niigata Engineering Works, Ltd.	179	300	110'-0"×23'-0"×10'-6"		1-Niigata	4 Cycle S.A.	6- 304.8×457.2	Experimental Fishing	" "
Olympia Maru	おりんぴあ丸	Mitsubishi Shoji Kaisha, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	5,611	2,300	405'-0"×55'-0"×32'-0"	13	1-Sulzer	2 Cycle S.A.	6- 600 ×1060	Transpacific Cargo	" "
Kushiro Maru	釧路丸	Kyōdo Gyogyō Kaisha, Ltd.	do.	Niigata Engineering Works, Ltd.	311	750	135'-0"×24'-0"×13'-6"	11	1-Niigata-Nobel	2 Cycle S.A.	6- 350 ×490	Trawler	September, 1927
Yone Maru, No. 1	第一米丸	Fujimura Shosen Kaisha, Ltd.	Tochigi Shipbuilding Co.	do.	136	150	92'-0"×17'-6"× 8'-0"	10	1-Niigata	4 Cycle S.A.	6- 228.6×304.8	Coastal Cargo and Passenger	November, 1927
San Diego Maru	さんぢえご丸	Mitsubishi Shoji Kaisha, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	7,268	2,300	430'-0"×57'-0"×34'-6"	12½	1-Sulzer	2 Cycle S.A.	6- 600 ×1060	Transpacific Tanker	January, 1928
Asakaze Maru	朝風丸	Government of Chōsen	Hikoshima Works, Mitsubishi Zosen Kaisha, Ltd.	Niigata Engineering Works, Ltd.	124	320	100'-0"×19'-0"× 9'-6"	11	1-Niigata	4 Cycle S.A.	6- 304.8×406.4	Fishing Guard	February, 1928
Keihan Maru	京阪丸	Konan Kisen Kaisha, Ltd.	Osaka Iron Works, Ltd.	M. A. N.	350	700	160'-0"×25'-3"×12'-0"	13	2-M. A. N.	4 Cycle S.A.	6- 275 ×420	Lake Biwa xcursion	" "
Takamisan Maru	高見山丸	Mitsui Bussan Kaisha, Ltd.	Tama Dockyard, Mitsui Bussan Kaisha, Ltd.	Tama Dockyard, Mitsui Bussan Kaisha, Ltd.	1,992	950	275'-6"×40'-9"×21'-0"	11½	1-B. & W.	4 Cycle S.A.	6- 500 ×900	Transpacific Cargo	" "
Tatsutasan Maru	龍田山丸	do.	do.	do.	"	"	"	"	"	"	"	do.	March, 1928
Yone Maru, No. 3	第三米丸	Fujimura Shosen Kaisha, Ltd.	Tochigi Shipbuilding Co.	Sulzer Brothers	144	160	92'-0"×17'-6"× 9'-0"		1-Sulzer	2 Cycle S.A.	4- 270 ×370	Coastal Cargo and Passenger	May, 1928
Shunten Maru	春天丸	Yamamoto Shoji Kaisha	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	5,623	2,300	405 -0"×55'-0"×32'-0"	13	"	"	6- 600 ×1060	Transpacific Cargo	June, 1928
Taihei Maru	太平丸	Shimatani Kisen Kaisha, Ltd.	Tama Dockyard, Mitsui Bussan Kaisha, Ltd.	Burmeister and Wain	5,900	2,150	412'-0"×55'-6"×32'-0"	12¾	1-B. & W.	4 Cycle S.A.	8- 630 ×1300	do.	July, 1928
Shunkotsu Maru	俊鶻丸	Fisheries Bureau	Yokohama Dock Co., Ltd.	Niigata Engineering Works, Ltd.	550	1,500	155'-0"×29'-6"×15'-6"	13	1-Niigata-Nobel	2 Cycle S.A.	6- 516 ×760	Crab Floating Cannery	" "
Midori Maru	緑丸	Osaka Shosen Kaisha, Ltd.	Kobe Works, Mitsubishi Zosen Kaisha, Ltd.	Kobe Works, Mitsubishi Zosen Kaisha, Ltd.	1,719	1,840	243'-0"×38'-0"×19'-6"	14¾	2-Mitsubishi-Vickers	4 Cycle S.A.	8- 425 ×620	Osaka-Beppu Line, Passenger	September, 1928
Hakubasan Maru	白馬山丸	Mitsui Bussan Kaisha, Ltd.	Tama Dockyard, Mitsui Bussan Kaisha, Ltd.	Burmeister and Wain	6,500	4,200	345'-0"×56'-6"×33'-0"	13½	2-B. & W.	4 Cycle S.A.	8- 630 ×1100	Transpacific Cargo	" "
Kanzaki Maru	神崎丸	Osaka Municipality	Yokohama Dock Co., Ltd.	M. A. N.	175	900	90'-0"×24'-0"×12'-0"		2-M. A. N.	4 Cycle S.A.	6- 345 ×500	Tug	" "
San Luis Maru	さんるいす丸	Mitsubishi Shoji Kaisha, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	7,268	2,300	430'-0"×57'-0"×34'-6"	12½	1-Sulzer	2 Cycle S.A.	6- 600 ×1060	Transpacific Tanker	October, 1928
Asama Maru	浅間丸	Nippon Yusen Kaisha, Ltd.	do.	Sulzer Brothers	16,000	16,000	560'-0"×72'-0"×42'-6"	19	4-Sulzer	2 Cycle S.A.	6- 680 ×1000	San Francisco Line, Passenger	" "
Sumire Maru	菫丸	Osaka Shosen Kaisha, Ltd.	Kobe Works, Mitsubishi Zosen Kaisha, Ltd.	Kobe Works, Mitsubishi	1,700	1,840	243'-0"×38'-0"×19'-6"	14¾	2-Mitsubishi-Vickers	4 Cycle S.A.	8- 425 ×620	Osaka-Beppu Line, Passenger	" "

TABLE I(B). List of Diesel Engined Merchant Vessels under Construction in Japan at the End of 1928.

Warships and Vessels under 100-tons Gross are Excluded.

Name of Vessel		Owners	Built by	Engined by	Gross Tonnage. Tons.	Engine total Output. B.H.P.	Hull Lpp. × Bm. × Dm. Ft.-Ins.	Designed Speed. Knots	No. and Type of Engines		No. of Cylinders per Engine and Bore × Stroke mm.	Type of Vessel	Launching expected
Chichibu Maru	秩父丸	Nippon Yusen Kaisha, Ltd.	Yokohama Dock Co., Ltd.	Burmeister and Wain	16,750	16,000	560'-0" × 74'-0" × 42'-6"	19	2-B. & W.	4 Cycle D.A.	8-840 × 1500	San Fransisco Line, Transpacific Passenger	May, 1929
Hikawa Maru	氷川丸	do.	do.	do.	11,000	11,000	510'-0" × 66'-0" × 41'-0"	18	"	"	8-640 × 1600	Seattle Line, Passenger	September, 1929
No. 178		do.	do.	do.	"	"	"	"	"	"	"	do.	February, 1930
Sydney Maru	しどにい丸	Osaka Shosen Kaisha, Ltd.	do.	do.	5,300	3,000	380'-0" × 54'-6" × 34'-3"	14¾	1-B. & W.	4 Cycle S.A.	6-740 × 1500	Austrarian Line, Cargo and Passenger	August, 1929
No. 175		do.	do.	do.	"	"	"	"	"	"	"	do.	December, 1929
No. 176		do.	do.	do.	"	"	"	"	"	"	"	do.	March, 1930
Kowa Maru	幸和丸	Shōwa Kisen Kaisha, Ltd.	Uraga Dock Co., Ltd.	M. A. N.	5,600	3,200	415'-0" × 56'-0" × 31'-9"	12	1-M. A. N.	2 Cycle D.A.	6-600 × 900	Transpacific Cargo	March, 1929
Iwate Maru	岩手丸	Iwate Prefecture	do.	Niigata Engineering Works, Ltd.	130	250	99'-0" × 20'-6" × 9'-9"	"	1-Niigata	4 Cycle S.A.	6-270 × 430	Fishing Flotilla Leader	
Heiyo Maru	平洋丸	Nippon Yusen Kaisha, Ltd.	Osaka Iron Works, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	9,500	7,500	460'-0" × 60'-0" × 40'-6"	16	2-Mitsubishi-Sulzer	2 Cycle S.A.	8-680 × 1000	S. American Line, Cargo and Passenger	October, 1929
Heian Maru	平安丸	do.	do.	Burmeister and Wain	11,000	11,000	510'-0" × 66'-0" × 41'-0"	18	2-B. & W.	4 Cycle D.A.	8-680 × 1600	Seattle Line, Passenger	
Hino Maru	日の丸	Nippon Shokuen Kaiso Co.	Kobe Works, Mitsubishi Zosen Kaisha, Ltd.	Kobe Works, Mitsubishi Zosen Kaisha, Ltd.	2,650	1,500	315'-0" × 46'-0" × 25'-0"	12	2-Mitsubishi-Sulzer	2 Cycle S.A.	4-600 × 1060	Coastal Cargo	May, 1929
Kiku Maru	菊丸	Tokyowan Kisen Kaisha, Ltd.	do.	do.	750	750	180'-0" × 30'-0" × 16'-6"	12¼	1-Mitsubishi-Vickers	4 Cycle S.A.	6-425 × 620	Coastal Passenger	April, 1929
Hakonesan Maru	箱根山丸	Mitsui Bussan Kaisha, Ltd.	Tama Dockyard, Mitsui Bussan Kaisha, Ltd.	Tama Dockyard Mitsui Bussan Kaisha, Ltd.	6,500	4,200	435'-0" × 56'-6" × 33'-0"	13½	2-B. & W.	4 Cycle S.A.	8-630 × 1100	Transpacific Cargo	
Tensan Maru	天山丸	Dairen Kisen Kaisha, Ltd.	do.	Burmeister and Wain	2,600	1,400	325'-9" × 46'-6" × 21'-6"	12	1-B. & W.	4 Cycle S.A.	6-550 × 1000	China Sea Cargo	May, 1929
Konsan Maru	崑山丸	do.	do.	do.	"	"	"	"	"	"	"	do.	July, 1929
Ronsan Maru	崙山丸	do.	do.	Tama Dockyard, Mitsui Bussan Kaisha, Ltd.	"	"	"	"	"	"	"	do.	
		do.	do.	do.	"	"	"	"	"	"		do.	
Tatsuta Maru	龍田丸	Nippon Yusen Kaisha, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	16,000	16,000	560'-0" × 72'-0" × 42'-6"	19	4-Sulzer	2 Cycle S.A.	8-680 × 1000	San Fransisco Line, Passenger	April, 1929
Terukuni Maru	照國丸	do.	do.	do.	11,800	11,000	505'-0" × 64'-0" × 37'-0"	17	2-Sulzer	2 Cycle S.A.	10-680 × 1200	European Line, Cargo and Passenger	December, 1929
Yasukuni Maru	靖國丸	do.	do.	do.	"	"	"	"	"	"	"	do.	April, 1930
Buenos Aires Maru	ぶえのすあいれす丸	Osaka Shosen Kaisha, Ltd.	do.	do.	9,500	6,000	460'-0" × 62'-0" × 39'-0"	16	"	"	6-680 × 1000	S. American Line, Cargo and Passenger	May, 1929
Rio de Janeiro Maru	りおてじあねろ丸	do.	do.	do.	"	"	"	"	"	"	"	do.	November, 1929
Ogura Maru	小倉丸	Ogura Sekiyu Kaisha, Ltd.	do.	do.	7,200	2,300	430'-0" × 57'-0" × 34'-6"	12½	1-Sulzer	2 Cycle S.A.	6-600 × 1060	Transpacific Tanker	June, 1929
Fusa Maru	ふさ丸	Chiba Prefecture	Hikoshima Works, Mitsubishi Zosen Kaisha, Ltd.	Ikegai Engineering Works, Ltd.	160	320	105'-0" × 22'-3" × 10'-3"	9	1-Ikegai	4 Cycle S.A.	6-280 × 440	Fishing Guard	February, 1929
Heiun Maru	飛隼丸	Fisheries Bureau	Harima Dockyard	Niigata Engineering Works, Ltd.	300	800	135'-0" × 24'-3" × 13'-0"		1-Niigata-Nobel	2 Cycle S.A.	6-380 × 610	Fishing Flotilla Leader	
Uko Maru No. 1.	第一宇高丸	Department of Government Railways	Kawasaki Dockyard Co., Ltd	Ikegai Engineering Works, Ltd.	"	300	150'-0" × 30'-0" × 8'-6"		2-Ikegai	4 Cycle S.A.	4-240 ×	Uno-Takamatsu Line, Goods Train Ferry	

TABLE II.

Particulars of Diesel Engined Passenger Ferries, O.S.K.'s Inland Sea Osaka-Beppu Service.

Name	Kurenai Maru.	Sumire Maru. Midori Maru.
Hull builders	Osaka Iron Works	Mitsubishi Kobe Works
Engine builders	Burmeister and Wain	" " "
L × B × D	238′-0″ × 38′-0″ × 19′-6″	243′-0″ × 38′-0″ × 19′-6″
Gross tonnage	1540 tons	1700 tons
Draught, loaded		
Service speed	13.3 knots	14 knots
Passenger accomodation	1st 38 2nd 108 3rd 452 total 598	1st 46 2nd 148 3rd 535 total 729
Engines	2-Burmeister and Wain 4 stroke cycle trunk piston type	2-Mitsubishi-Vickers 4 stroke cycle trunk piston airless injection type.
Normal B.H.P.	Each 800	Each 920
R. P. P.	150	210
Cylinder No. and bore × stroke.	6-500 mm. × 90 mm.	8-425 mm. × 620 mm.
Dynamo engines	3-Single cylinder B. & W. Diesel Engine driven 33 K.W. Dynamos.	3-M.A.N. 3 cylinders Diesel Engine driven 35 K.W. Dynamos.

TABLE III.

Particulars of the 7000 tons Single-screw "Akagisan Maru."

Owners	Mitsui & Co.
Hull builders	Mitsui & Co., Tama Works.
Engine builders	Burmeister and Wain.
Length, over all	389′-0″
Length, P. P.	375′-0″
Breadth, moulded	50′-0″
Depth, moulded to up. dk.	30′-0″
Draught, loaded	24′-4″
Gross tonnage	4,638 tons
Deadweight	6,981 tons
Speed, designed	10¾ knots
Engines	1-Burmeister and Wain 4 stroke cycle single acting.
B. H. P.	1,800 at 87 R.P.M.
Cylinder No. and bore × stroke	6-740 mm. × 1,500 mm.
Dynamo engines	3-Diesel Engine driven, output each 50 K.W.

TABLE IV.

Particulars of Diesel Engined Passenger and Cargo Ships
O.S.K.'s Round-World S.A. Service.

Particulars	Santos Maru, La Plata Maru, Monte Video Maru	Two Sister Ships building
Owners	Osaka Shosen Kaisha, Ltd.	Osaka Shosen Kaisha, Ltd.
Hull builders	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.	Nagasaki Works, Mitsubishi Zosen Kaisha, Ltd.
Engine builders	Santos, La Plata—Sulzer Bros; Monte Video—Nagasaki Works.	do.
Lpp. × Bm. × Dm.	43′-0″ × 56′0″ × 36′-0″	460′-0″ × 62′-″ × 39′-″
Draught, loaded	25′-0″	26′-1″
Deadweight	7,266 tons.	9,500 tons
Gross tonnage	6,800 tons.	8,200 tons
Passengers accommodation	1st 40 3rd 700	1st 60 3rd 1,000
Speed, designed	14¾ knots	16 knots
Engines	2-Sulzer 6 ST 60	2-Mitsubishi-Sulzer 6 ST 68
Output, normal	6 Cyl.-600 mm. × 1,060 mm.	6 cyl.-680 mm. × 1,000 mm.
No. and bore × stroke of cylinders	2,300 B.H.P. each, at 112 R.P.M.	3,000 B.H.P. each, at 120 R.P.M.
Generating Sets	3 Sets—Engine—Sulzer 4RVPH31 Diesel engine, Each 250 B.H.P. at 300 R.P.M. 4 cyl.-310 mm. × 420 mm. Dynamo—B.B.C. 150 K.W. 225 volts. 1 Set—Engine—Sulzer 2RVH24 Diesel engine, 60 B.H.P. at 350 R.P.M. 2 cyl.-240 mm. × 340 mm. Dynamo—B.B.C. 37.5 K.W. 225 volts.	3 Sets—Engine—Mitsubishi M45C6 Diesel engine, each 350 B.H.P. at 310 R.P.M. 6 cyl.-300 mm. × 450 mm. Dynamo—Mitsubishi Denki 230 K.W. 225 volts.
Donkey, boiler	1-Cochran type H.S. 300 sq. ft.; W. P. 100 lb./□″	1-Scotch type H.S. 464 sq. ft.; W.P. 120 lb./□″
Official trial	"Monte Video Maru"; Half load	
Date	31st July, 1926	
Speed of ship	16.19 knots	
Main Eng. R.P.M.	P. 120.8 S.121.3	
Mean ind. pressure	P. 7.01 kg/cm² S. 6.98 kg/cm²	
I. H. P.	P. 3,384 S. 3,385. Collected 6,769	
Mechanical eff.	P. 80.5 S. 80.5	
B. H. P.	P. 2,725 S. 2,726. Collected 5,451	
Mile post distance	3 Miles	
Runs	6 Runs	

TABLE V.

Diesel Engined Ships Launched in Japanese Shipbuilding Yards.

Year	No. of Ships	Gross Tonnage	B. H. P. Normal		
			Home Made	Foreign Made	
1920	1	174	150		150
1921	1	163	200		200
1922	1	332	600		600
1923	1	688		600	600
1924	6	17,202	330	8,920	9,250
1925	10	15,881		10,640	10,740
1926	13	27,831	15,140	2,620	17,760
1927	13	26,705	11,200	2,400	13,600
1928	15	57,298	14,300	24,110	38,410
Total	56	146,774	41,940	49,390	91,310
1928 (December) Building	26	167,740	69,220	63,500	132,720
Order Placed	2	1,500			1,000

TABLE VI. The World's Diesel Engine Ships Building (From "The Motorship")

Ships under 1,000 tons Gross or Deadweight Capacity are Excluded.
(Passenger Ships—Gross tonnage; Cargo Ships—Deadweight.)

	Great Britain	Germany	America	Denmark	Sweden	Holland	Spain	France	Italy	Norway	Russia	Japan
1924—No. of Ships	42	13	9	7	8	3	—	1	1	1	—	2
Total Tonnage Completed	314,440	111,650	54,275	47,600	44,150	13,000	—	11,700	8,400	6,000	—	8,521
Total I.H.P.	125,800	48,900	25,450	18,150	18,090	7,750	—	4,000	3,200	1,600	—	4,250
1925—No. of Ships	33	34	4	11	8	4	—	—	7	2	—	3
Total Tonnage Completed	296,700	287,950	16,715	70,260	56,630	42,500	—	—	60,400	2,200	—	22,420
Total I.H.P.	129,000	113,200	6,560	29,880	21,060	15,360	—	—	23,070	1,240	—	10,800
1926—No. of Ships	38	26	9	17	9	5	—	6	12	2	2	4
Total Tonnage Completed	234,000	246,500	77,500	132,260	68,200	29,600	—	49,500	106,400	10,100	13,400	32,760
Total I.H.P.	184,350	111,900	28,250	53,100	22,800	17,900		27,250	40,250	4,900	3,800	18,250
1927—No. of Ships	39	14	17	13	11	11	3	5	12	—	—	10
Total Tonnage Completed	237,630	84,420	99,310	54,795	53,580	86,600	8,300	26,350	120,750	—	—	46,390
Total I.H.P.	181,750	57,550	59,700	38,800	29,900	63,000	8,400	25,050	103,800	—	—	22,660
1928—No. of Ships	73	28	8	21	15	10	2	4	19	1	3	8
Total Tonnage Completed	470,405	190,660	51,090	128,340	98,880	75,700	5,040	27,070	89,050	4,300	12,160	57,080
Total I.H.P.	310,450	16,770	26,500	66,300	55,500	56,060	4,600	14,900	78,300	3,000	7,500	21,260

TABLE VII.

Steam and Diesel Engined Ships Launched during 1928.
(Over 100 tons Gross Tonnage.)

Month	Steamer		Diesel Engine Ships	
	No.	Gross Tonnage	No.	Gross Tonnage
Jan.	4	7,960	1	7,268
Feb.	2	6,805	3	2,460
Mar.	2	5,258	1	1,992
Apr.	2	1,740	—	—
May	5	4,807	1	144
June	3	706	1	5,622
July	3	3,677	2	6,450
Aug.	4	1,550	—	—
Sep.	3	5,368	3	8,394
Oct.	2	230	3	24,979
Nov.	1	2,100	—	—
Dec.	7	12,538	—	—
Total	28	52,739	15	57,309

TABLE VIII. Types of Diesel Engines Fitted.

(A) Four Stroke Cycle Engines (1920 to 1928, Launched.)

Niigata	Vickers	M. A. N.	B. & W.	Ikegai
200 B.H.P.	*600 B.H.P.	*150 B.H.P.	*1,800 B.H.P.	
300	*600	*900 (2 × 450)	*1,600 (2 × 800)	
600 (2 × 300)	*600	*700	*950	
320	1,200 (2 × 600)	3 ships	*950	
300	1,200 (2 × 600)	4 sets	950	
750	1,840 (2 × 920)	*1,750 B.H.P.	950	
150	1,840 (2 × 920)		*2,150	
320	7 ships		*4,200 (2 × 2,100)	
8 ships	11 sets		8 ships	
9 sets	7,800 B.H.P.		10 sets	
2,940 B.H.P.	(6,080 / *1,800)		13,550 B.H.P.	
			(1,900 / *11,650)	

(B) Two Stroke Cycle Engines (1922 to 1928, Launched.)

Fullagar	Polar	Niigata	Sulzer	M. A. N.
*2,500 B.H.P.	*200 B.H.P.	1,500 B.H.P.	150 B.H.P.	
*2,500	*220	1 ship	330	
2 ships	2 ships	1 set	*410	
2 sets	2 sets	1,500 B.H.P.	*410	
*5,000 B.H.P.	*440 B.H.P.		*1,600 (2 × 800)	
			*500	
			*4,600 (2 × 2,300)	
			*4,600 (2 × 2,300)	
			220	
			4,600 (2 × 2,300)	
			3,000 (2 × 1,500)	
			*120	
			2,300	
			2,300	
			500	
			*350	
			2,300	
			2,300	
			2,300	
			2,300	
			2,300	
			*160	
			2,300	
			2,300	
			*16,600 (4 × 4,000)	
			25 ships	
			33 sets	
			58,250 B.H.P.	
			(29,500 / *28,750)	

* Marked thus—Foreign Make.

TABLE IX.

Types of Diesel Engines to be Fitted on Board the Ships Building.

(A) Four Stroke Cycle Engines.

Niigata	Vickers	M. A. N.	B. & W.	Ikegai
250 B.H.P.	750 B.H.P.		B.H.P. D.A.*16,000 (2 × 8,000)	320 B.H.P.
800	1 Ship		D.A.*11,000 (2 × 5,500)	300
2 ships	1 set		D.A.*11,000 (2 × 5,500)	2 ships
2 sets	750 B.H.P.		*3,000	2 sets
1050 B.H.P.			*3,000	620 B.H.P.
			*3,000	
			D.A.*11,000 (2 × 5,500)	
			4,200 (2 × 2,100)	
			*1,400	
			*1,400	
			1,400	
			1,400	
			12 ships	
			17 sets	
			67,800 B.H.P.	
			(7,000 / *60,800)	

(B) Two Stroke Cycle Engines.

Fullagar	Polar	Niigata	Sulzer	M. A. N.
			7,500 (2 × 3,750) B.H.P.	D.A.*3,200 B.H.P.
			1,500	1 ship
			16,000 (4 × 4,000)	1 set
			10,000 (2 × 5,000)	*3,200 B.H.P.
			10,000 (2 × 5,000)	
			6,000 (2 × 3,000)	
			6,000 (2 × 3,000)	
			2,300	
			8 ships	
			16 sets	
			59,3000 B.H.P.	

* Marked thus—Foreign Make.

TABLE X. N.Y.K. Transpacific Liners.

	Asama Maru	Tatsuta Maru	Chichibu Maru
Built by	Mitsubishi Nagasaki Works	Mitsubishi Nagasaki Works	Yokohama Dock & Co., Ltd.
Engined by	Sulzer Bros.	Mitsubishi Nagasaki Works	Burmeister and Wain
Lpp. × B mould × D mould	560′-0″ × 72′-0″ × 42′-0″	560′-0″ × 72′-0″ × 42′-0″	560′-0″ × 74′-0″ × 42′-6″
Draught	28′-6″	28′-6″	28′-6″
Gross Tonnage	16,000 tons	16,000 tons	16,750 tons
Deadweight	8,000 tons	8,000 tons	8,000 tons
Oil Fuel Capacity	3,000 tons at 39 cub. ft.	3,000 tons at 39 Cub. Ft.	3,000 tons at 39 Cub. Ft.
Service Speed	17½ knots	17½ knots	17½ knots
Trial Speed	19 knots, half loaded	19 knots, half loaded	19 knots, half loaded
Accommodation	1st Class 180 Interchangeable 1st 34 or 2nd 51 2nd Class 95 3rd Class 500	1st Class 180 Interchangeable 2nd 51 2nd Class 95 3rd Class 500	1st Class 183 Interchangeable 2nd 56 2nd Class 96 3rd Class 500
Main Engines			
No. of Engines	4	4	2
Type of Engines	Sulzer 2 stroke cycle s.a.	Mitsubishi-Sulzers 2 stroke cycle s.a.	B. & W. 4 stroke cycle d.a.
No. of Cylinders, each set	8	8	8
Bore and Stroke	680 mm. × 1,000 mm.	680 mm. × 1,000 mm.	840 mm. × 1,500 mm.
Normal Power, total	16,000 B.H.P. at 120 R.P.M.	16,000 B.H.P. at 120 R.P.M.	16,000 B.H.P. at 120 R.P.M.
Main Diesel Generating Sets			
No. of Sets	4	4	3
Diesel Builders	Allen and Sons	Allen and Sons	Burmeister and Wain
Generator Builders	Allen and Sons	Allen and Sons	Allen and Sons
No. of Cylinders	6	6	6
Bore and Stroke	410 mm. × 600 mm.	410 mm. × 600 mm.	400 mm. × 600 mm.
B.H.P. per Engine	675 at 250 R.P.M.	675 at 250 R.P.M.	600 at 250 R.P.M.
Generator Output	450 K.W.	450 K.W.	400 K.W.
Auxiliary Diesel Generating Sets			
No. of Sets	1	1	1
Diesel Builders	Niigata Engineering Works	Niigata Engineering Works	Niigata Engineering Works
Generator Builders	Mitsubishi Denki Kaisha, Ltd. Nagasaki Works	Mitsubishi Denki Kaisha, Ltd. Nagasaki Works	Mitsubishi Denki Kaisha, Ltd. Nagasaki Works
No. of Cylinders	6	6	6
Bore and Stroke	9″ × 12″	9″ × 12″	9″ × 12″
B.H.P.	150 at 400 R.P.M.	150 at 400 R.P.M.	150 at 400 R.P.M.
Generator Output	100 K.W.	100 K.W.	100 K.W.
Main Engine Main Air Compressors	Driven from main engine shaft	Driven from main engine shaft	Independent

TABLE XI.

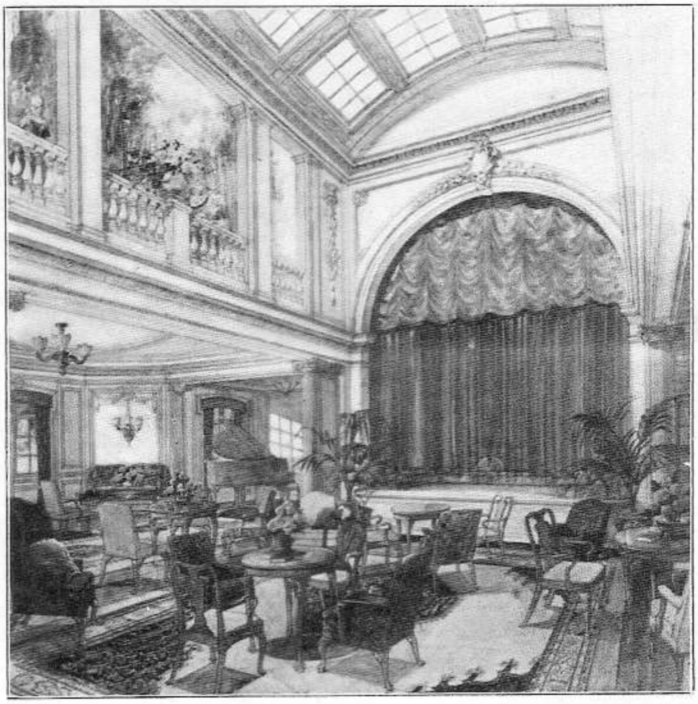
Shop Trial Results.
M.S. Tatsuta Maru Port Wing Engine.

Date—	November 5th 1928
Load—	4/4
Duration—	6 hours
Barometer—	767 mm. of Hg.
R. P. M.—	121.4
Brake Load, Kgs.—	6,667
B.H.P.—	4,047
Mean Indicated Pressure, kgs./cm^2—	6.13
I.H.P.—	4,808
Mechanical Efficiency, %—	842
Exhaust Gas Temperature, Deg.C.—	205
Fuel Consumption, Grs./B.H.P./Hr.—	172
Exhaust Gas Analysis—	O_2-15.0; CO-0; CO_2-4.3
Exhaust Gas Pressure—	394.8 mm. of H_2O

(Paper No. 85)

The Motor Vessel "Asama Maru"

Motor Vessels "Asama Maru" and "Tatsuta Maru"



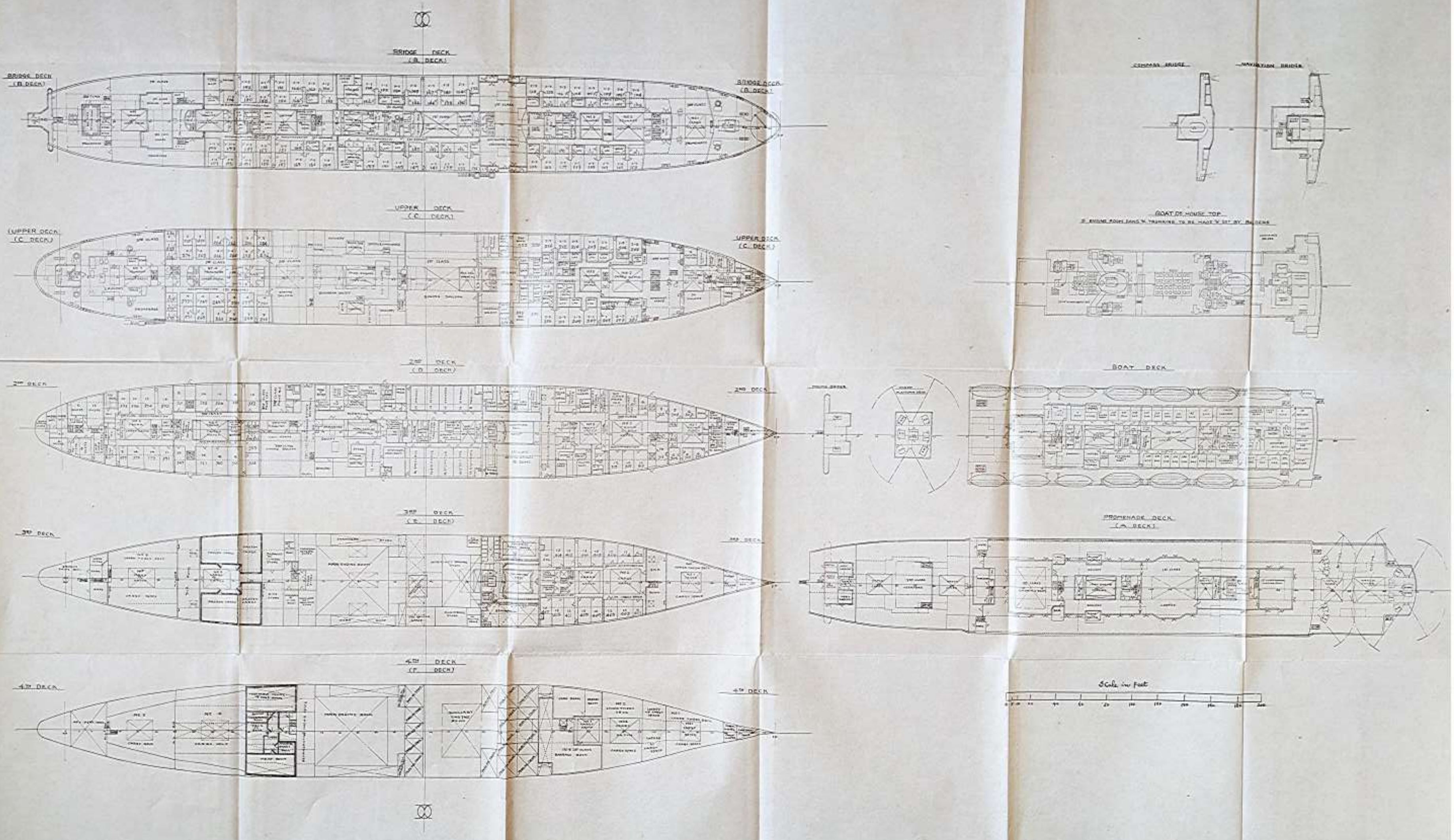
The Lounge.

The Smoking Room.

ASAMA MARU

PLAN

(Plan No. 85)



The Recent Development of the Mercantile Shipbuilding in Japan and Other Matters Relating to It.

(Paper No. 225)

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and
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Expert of the same Department.

I. Introduction.

We have been asked by the Papers Committee of the World Engineering Congress to prepare a paper for the Congress and we have selected this subject, partly because the subject is in our line of profession, and partly because we thought it more or less our duty to publish these matters to the Congress.

The conspicuous development of the modern mercantile steel-shipbuilding in Japan commenced practically 34 years ago when the s.s. "Suma Maru," 1500 tons gross, was launched in 1895. Three years after, (1898), the s.s. "Hitachi Maru," 6100 tons gross, and nine years thence, (1907), the s.s. "Tenyo Maru," 13400 tons gross were launched. Last year (1928) there was the launching of the motor ship "Asama Maru," 17,000 tons gross.

The above is the outline of the development as regards the size of merchant ships built in this country, but we are sure with the present shipyards, their equipments, experts and workmen, it is possible to build even the world's biggest ship, 60,000 tons gross and 1,000 feet long.

II. The Historical Description of Typical Merchant Ships of Various Kinds Built in Japan.

All kinds of merchant ships have been built in this country. We will describe the typical ones under classified headings, mentioning the tonnage of each ship as at the time of build.

A. Passenger ships and passenger-&-cargo ships.

(1) In 1898 the before mentioned s.s. "Hitachi Maru," a passenger-&-cargo ship of 6100 tons gross and 14 knots maximum speed, was launched at the Mitsubishi Shipyard, Nagasaki, for the Nippon Yusen Kaisha's new

Japan-Europe Line. From 1899 to 1902 the same shipyard launched four passenger-&-cargo steamers of 6300 to 6400 tons gross for the same line.

(2) In 1903 the same shipyard launched the s.s. "Nikko Maru," 5500 tons gross and 17 knots maximum speed, for the same owners' Japan-Australia Line. This was indeed the first high class large passenger ship built in this country.

(3) In 1904 the same shipyard launched the s.s. "Tango Maru," a passenger-&-cargo ship of 7400 tons gross, for the same owners who intended to put her in their Hongkong-Japan-Seattle Line.

(4) From 1907 to 1909, for the improvement of the Japan-Europe Line, the same owners had the s.s. "Kamo Maru" and five other passenger-&-cargo steamers, 8500 tons gross each, four of them launched at the Mitsubishi Shipyard, Nagasaki, and two at the Kawasaki Dockyard, Kobe.

In the same period, the Mitsubishi Shipyard, Nagasaki, launched for the Teikoku Kaiji Kyokai the two steamers "Sakura Maru" and "Umegaka Maru," 3200 tons gross each, which had steam turbines and Miyabara water tube boilers and were high speed passenger ships of 21 knots maximum speed, forming a volunteer fleet.

(5) From 1907 to 1911, for the improvement of the Hongkong-Japan-San Francisco Line, the Toyo Kisen Kaisha had three steamers, the "Tenyo Maru," "Chiyo Maru" and "Shinyo Maru," 13400 tons gross and 20 knots maximum speed each, launched at the Mitsubishi Shipyard, Nagasaki. These ships had steam turbines and were the biggest and most excellent transpacific passenger liners at that time.

(6) From 1909 to 1911, the Osaka Shosen Kaisha had, for their new Hongkong-Japan-Puget Sound Line, the s.s. "Seattle Maru" and five other passenger-&-cargo steamers, about 6000 tons gross each, three of them launched at the Kawasaki Dockyard, Kobe, and three at the Mitsubishi Shipyard, Nagasaki.

(7) In 1913 the Teikoku Kaiji Kyokai added to their volunteer fleet a passenger turbine-steamer, "Sakaki Maru," 3800 tons gross and 19 knots maximum speed, launched at the Kawasaki Dockyard, Kobe. This ship was sold to the South Manchurian Railway in 1923 when the fleet was dissolved.

(8) From 1913 to 1923, for the improvement of the Japan-Europe Line, the Nippon Yusen Kaisha had nine high class passenger-&-cargo steamers, the "Katori Maru" to the "Hakusan Maru," about 10500 tons gross and 16 knots maximum speed each, seven of them launched at the Mitsubishi Shipyard, Nagasaki, and two at the Kawasaki Dockyard, Kobe. Among them the "Katori Maru" had combined reciprocating and turbine engines, and the "Hakusan Maru" and four other steamers double reduction geared turbines.

(9) From 1923 to 1928, the Osaka Shosen Kaisha had several fine passenger motor ships of 600 to 1600 tons gross launched at various ship-

yards. These ships are now running the scenic Inland Sea and other coasting routes.

(10) From 1925 to 1926 the same owners had, for the Japan-South America (East Coast) Line, three emigrant passenger motor ships, viz. the "Santos Maru," "La Plata Maru" and "Monte Video Maru," 7200 tons gross and 15 knots maximum speed each, launched at the Mitsubishi Shipyard, Nagasiki.

(11) Last year (1928) the same owners had the passenger-&-cargo steamer "Uraru Maru" with geared turbine engines, 6100 tons gross and 17 knots maximum speed, for the Kobe-Dairen Line, launched at the Mitsubishi Shipyard, Nagasaki.

(12) From last year (1928) to this year (1929), the Nippon Yusen Kaisha had three passenger motor ships, viz. the "Asama Maru," "Tatsuta Maru" and "Chichibu Maru," 17000 tons gross and 20 knots maximum speed each, the former two launched at the Mitsubishi Shipyard, Nagasaki, and the latter one at the Yokohama Dockyard, Yokohama. They are the latest and finest transpacific liners with the most complete and elaborate accommodations to run on the Hongkong-Japan-San Francisco Line which was transferred to the present owners from the Toyo Kisen Kaisha in 1926.

(13) This year (1929), the same owners had the passenger-&-cargo motor ship "Hikawa Maru," 11000 tons gross and 18 knots maximum speed, for the Hongkong-Japan-Seattle Line, launched at the Yokohama Dockyard, Yokohama, and next year (1930) two more sister ships for the same line will be launched, one at the same yard and the other at the Osaka Iron Works, Osaka.

(14) This year (1929) the Osaka Shosen Kaisha had the emigrant passenger motor ship "Buenos Ayres Maru," 9500 tons gross and 17 knots maximum speed, for the Japan-South America (East Coast) Line, launched at the Mitsubishi Shipyard, Nagasaki, and this year (1929) a sister ship for the same line will be launched at the same yard.

(15) This year (1929) the Nippon Yusen Kaisha had the passenger-&-cargo motor ship "Heiyo Maru," 9500 tons gross and 16 knots maximum speed, for the Japan-South America (West Coast) Line, launched at the Osaka Iron Works, Osaka.

(16) Near the end of this year (1929) the same owners will have a passenger-&-cargo motor ship, 11800 tons gross and 17 knots maximum speed, for the Japan-Europe Line, launched at the Mitsubishi Shipyard, Nagasaki, and next year (1930) a sister ship for the same line will be launched at the same yard.

B. Shallow draught river steamers.

Shallow draught steamers, of the Nisshin Kisen Kaisha or of their predecessors with light hulls adapted for use in Yan Tsee River or thereabouts and with fine passenger accommodations, were launched in Japan as follows:—

(1) 1899, s.s. "Ta-Yueng Maru," 1700 tons gross, at the Kawasaki Dockyard, Kobe.

(2) 1900, s.s. "Ta-Lee Maru," 2200 tons gross, at the Kawasaki Dockyard, Kobe, and s.s. "Ta-Foo Maru," 2200 tons gross, at the Mitsubishi Shipyard, Nagasaki.

(3) 1901, s.s. "Ta-Chang Maru," 2700 tons gross, at the Mitsubishi Shipyard, Nagasaki, and s.s. "Ta-Chi Maru," 2000 tons gross, at the Osaka Iron Works, Osaka.

(4) 1903, s.s. "Siang-Kiang Maru" and s.s. "Yuen-Kiang Maru," 900 tons gross each, at the Osaka Iron Works, Osaka.

(5) 1905, s.s. "Ta-Hung Maru," 1700 tons gross at the Mitsubishi Shipyard, Nagasaki.

(6) 1906 to 1907, s.s. "Yoh-Yang Maru," s.s. "Nang-Yang Maru" and s.s. "Shang-Yang Maru," 3500 tons gross each, at the Kawasaki Dockyard, Kobe.

(7) 1915, s.s. "Feng-Yang Maru," 3900 tons gross, at the Osaka Iron Works, Osaka.

C. Cross channel passenger and cargo steamers for railway connections.

(1) In 1905 the steamers "Iki Maru" and "Tsushima Maru," 1600 tons gross and 16 knots maximum speed each, for the Tsushima Channel Line, were launched at the Mitsubishi Shipyard, Nagasaki.

(2) In 1912 the steamers "Koma Maru" and "Shiragi Maru," 3100 tons gross and 16 knots maximum speed each, for the same line, were launched at the Kawasaki Dockyard, Kobe.

(3) In 1921 and 1922 the steamers "Keifuku Maru," "Tokuju Maru" and "Shokei Maru," 3600 tons gross and 20 knots maximum speed each, for the same line, were launched at the Mitsubishi Shipyard, Kobe. These three excellent ships with geared turbine engines are now in the service, crossing the distance of 122 miles between Shimonoseki and Fusan in about eight hours, and connecting the trunk line of Japanese main land with Siberian railway via. Korean and South Manchurian railways.

D. Railway car ferry ships.

(1) In 1923 and 1924 the steamers "Shoho Maru," "Hiran Maru" "Tsugaru Maru" and "Matsumaye Maru," 3400 tons gross and 17 knots maximum speed each, were launched, the former two with Rateaux turbines at the Uraga Dockyard, Uraga near Yokohama, and the latter two with geared turbines and Babcock-and-Wilcox water tube boilers at the Mitsubishi Shipyard, Nagasaki. These ships are now in the service to connect Aomori at the north end of Japanese main land with Hakodate, traversing the distance of 60 miles of the Tsugaru Channel in four hours, and each

of them carries 25 railway cars of 15 ton capacity as well as 900 passengers.

(2) In 1926 the s.s. "Seikan Maru No. 1," 2300 tons gross with geared turbines and Ikeda water tube boilers, for solely carrying railway cars on the same channel, was launched at the Yokohama Dockyard, Yokohama.

E. Oil tankers.

Large ships carrying oil in bulk were launched in Japan as follows:—

(1) 1909, s.s. "Kiyo Maru," 9200 tons gross, of the Toyo Kisen Kaisha, at the Mitsubishi Shipyard, Nagasaki.

(2) 1921 and 1922, steamers "Tachibana Maru," "Manju Maru" and "Kanju Maru," 6500 tons gross each of the Teikoku Petroleum Co., the predecessors of the Asahi Petroleum Co., at the Harima Shipyard near Kobe, all having hulls built on Isherwood system.

(3) 1927 and 1928, motor ship "San Pedro Maru," "Santiago Maru" and "San Luis Maru," 7200 tons gross each, of the Mitsubishi Trading Co., at the Mitsubishi Shipyard, Nagasaki, all having hulls built on F.K. system.

(4) 1928, s.s. "Shoyo Maru," 7400 tons gross, of the Nippon Tanker Co., at the Yokohama Dockyard, Yokohama, and 1929, s.s. "Yeiyo Maru," 8500 tons gross, of the same owners at the same yard, both having hulls built on Blythwood system.

(5) 1929, motor ship "Ogura Maru" with hull built on F.K. system 7200 tons gross, of the Ogura Petreleum Co., at the Mitsubishi Shipyard, Nagasaki.

F. Dredgers.

In 1911 and 1912 the Uraga Dockyard, Uraga, launched, for the Home Department, the steamers "Kizu Maru No. 1" and "Noda Maru," 1300 tons gross each, with Babcock-and-Wilcox water tube boilers. The Osaka Iron Works, Osaka, the Ishikawajima Shipyard, Tokyo, and other shipyards have built several dredgers.

G. Submarine telegraphic cable layers.

In 1906 the Mitsubishi Shipyard, Nagasaki, launched the s.s. "Ogasawara Maru," 1400 tons gross, and in 1923 the Osaka Iron Works, Osaka, launched the s.s. "Nanyo Maru," 3500 tons gross, both for the Communication Department.

H. Nautical training ships.

(1) In 1898, the Mitsubishi Shipyard, Nagasaki, launched the three-masted sailing barque "Tsukishima Maru," 1500 tons gross, with an auxiliary steam engine, for the Tokyo Higher Nautical School.

(2) In 1903, the Kawasaki Dockyard, Kobe, launched the four-

masted sailing barque "Taisei Maru," 2200 tons gross, with auxiliary steam engines, for the same school. This ship was the substitute for "Tsukishima Maru" which had been missed in 1900.

(3) In 1923, the Mitsubishi Shipyard, Kobe, launched the four-masted sailing barkentine "Shintoku Maru," 2500 tons gross, with auxiliary steam engines, for the Kobe Higher Nautical School.

(4) By the end of this year (1929) or the beginning of next year (1930), the Kawasaki Dockyard, Kobe, will launch two ships for the Education Department. Both of them are to be four-masted sailing barques of 2200 tons gross with auxiliary motor engines and are intended for the training of the boys of eleven local nautical schools.

I. Refrigerated cargo ships.

In 1922 and 1923 the Yokohama Dockyard, Yokohama, launched the s.s. "Kaiko Maru" and six other ships, about 1500 tons gross each, and in 1923 the Mitsubishi Shipyard, Nagasaki, launched the s.s. "Choko Maru," 1600 tons gross.

In the same period several other shipyards launched 15 ships of under 1000 tons gross and also converted three other ships, and later they have launched six small ships.

Carrying cargoes in the holds kept cold by regrigerating engines has developed quite recently in Japan, in addition to carrying fresh fishes in the usual ice chambers of fishing boats and catches carriers. Such ships as mentioned above are specialized in carrying refrigerated cargoes which are frozen or fresh fishes. Nearly 30 other Japanese large ships have a part of holds refrigerated and are carrying meat, fruits etc. as cargo to and from Japan.

J. Fishing boats.

A great number of fishing boats, large and small, of all varieties, such as trawlers, whalers, drifters, long liners, different kinds of seine boats and angling boats, training ships, instruction ships, experiment ships, catches carrying ships etc. were and are being built at several places around the coast. The recent rapid development of the fishing industry has been side by side with the building of numerous fishing boats with motors. At present the number of existing fishing boats with engines—most of them being motors—is about 20,000 including about 300 steel ships, and the number of existing fishing boats of 5 tons gross and upwards without engines is about 13,000.

As small wooden ships and small motors in the severe services of fishing have very short lives, they must be frequently replenished by building new ones. For this and for the steady expansion of the fishing enterprises, fishing boat builders are flourishing and many specialized motor builders are kept always busy.

Besides the building of new ships, it may be interesting to note that several cargo steamers and other steamers, about 2000 or 3000 tons gross each, have recently been converted and are employed as floating canneries at the very spots of catching crabs with seines in the open seas in the north of Japan, to produce the freshest canned-flesh. This new enterprise employing large steamers as canneries in open sea, which is in addition to the usual land canneries along the coast of Kamchatka for the crab fishing near the coast, was started some eight years ago and employs about 16 cannery-steamers with a small motor vessel or two and a number of seine junks attached to each cannery-steamer, and a few catches carrying steam tenders of about 1000 tons gross during the season every year and makes a great amount of products which are supplied everywhere, the exports for Europe and America being remarkable.

K. Cargo Ships.

The building of cargo ships has been so numerous that we would better omit to describe about particular ships but generally say that, owing to the progress of the art of shipbuilding, new cargo ships are becoming more and more efficient and economical, reducing their weights, increasing their speeds and carrying capacities, being provided with innovated machineries of small fuel consumption, comfortable accommodations, bettered safety appliances and improved equipments—especially cargo handling appliances—and having constructions adapted for specialized cargo carrying. Oil tankers and refrigerated cargo ships as mentioned before are the examples of specialized cargo ships. For another example it may be added here that the building of ships, mostly motor ships, with long holds, clear upper deck and special fittings adapted for lumber trade, has become remarkable owing to the specially increased supply of lumber from Hokkaido, Karafuto, Maritime Province of Siberia and the Pacific Coast of North America after the Japanese great earthquake of 1923.

The following list shows the cargo motor ships of 1000 tons gross and upwards launched in Japan besides those four oil tank motor ships already mentioned in E (3) and (5).

Name of Owners	Name of cargo motor ship	Gross tonnage	Launched at	in
Mitsui Bussan Kaisha	"Akagisan Maru"	4600	Mitsui Shipyard, Tama	1924
Kobe Steel Works	"Fukko Maru"	3800	Harima Shipyard, Aioi	,,
Kawasaki Kisen Kaisha	"Florida Maru"	5800	Kawasaki Dockyard, Kobe	,,
,, ,, ,,	"Cuba Maru"	5900	,, ,, ,,	1926
Itaya Shosen Kaisha	"Yahiko Maru"	5700	Harima Shipyard, Aioi	,,
Mitsubishi Shoji Kaisha	"Columbia Maru"	5600	Mitsubishi Dockyard, Nagasaki	1927
,, ,, ,,	"Olympia Maru"	,,	,, ,, ,, ,,	,,
Yamamoto Shoji Kaisha	"Shunten Maru"	,,	,, ,, ,, ,,	,,
Mitsui Bussan Kaisha	"Koyasan Maru"	2000	Mitsui Shipyard, Tama	,,
,, ,, ,,	"Tatsutasan Maru"	,,	,, ,, ,,	,,
,, ,, ,,	"Kuramasan Maru"	,,	,, ,, ,,	,,

,, ,, ,,	"Takamisan Maru"	,,	,, ,, ,,	1928
Shimatani Kisen Kaisha	"Taihei Maru"	6200	,, ,, ,,	,,
Mitsui Bussan Kaisha	"Hakubasan Maru"	6600	,, ,, ,,	,,
,, ,, ,,	"Hakonesan Maru"	,,	,, ,, ,,	1929
Showa Shosen Kaisha	"Kowa Maru"	5800	Uraga Dorkyard, Uraga	,,
Nippon Shokuyen Kaisha	"Hino Maru"	2600	Mitsubishi Shipyard, Kobe	,,
Tairen Kisen Kaisha	"Tenzan Maru" and three other ships	2700 each	Mitsui Shipyard, Tama	,,
Osaka Shosen Kaisha	"Sydney Maru"	5300	Yokohama Dockyard, Yokohama	,,

Besides those in the above list, the following cargo motor ships will be launched in the near future.

Name of Owners	Number of cargo motor ships	Gross tonnage	at	in
Osaka Shosen Kaisha	one	5300	Yokohama Dockyard, Yokohama	1929
,, ,, ,,	one	,,	,, ,, ,,	1930
,, ,, ,,	four	8300	Mitsubishi Shipyard, Nagasaki	,,
Kishimoto-Kisen Kaisha	two	8600	Yokohama Dockyard, Yokohama	,,
Yamashita Kisen Kaisha	one	7500	Uraga Dockyard, Uraga	,,

III. The Annual Outputs of the Mercantile Shipbuilding in Japan and the Comparison with Those in Other Countries.

Although the Japanese mercantile shipbuilding became conspicuous from about 34 years ago as mentioned already, the launching of merchant ships of 100 tons gross and upwards for each year was about 7,000 to 60,000 tons gross until 1913, but over 50,000 tons gross in a year were seldom launched.

Since 1914, however, the annual launching never dropped below 50,000 tons gross as shown in Table 1.

TABLE 1.

Ships of 100 tons gross and upwards launched in Japan.

Year	Number	Gross tonnage
1914	39	85,125.
1915	31	50,104.
1916	68	157,196.
1917	196	403,016.
1918	396	641,056.
1919	190	646,344.
1920	146	452,688.
1921	69	226,081.
1922	57	71,076.
1923	56	74,284.
1924	39	71,440.
1925	33	55,086.
1926	27	51,303.
1927	35	52,473.
1928	48	109,058.

Note. Sailing ships are excluded, but the launching of large sailing ships has been very rare.

This Table tells how much Japanese shipyards contributed to fill up the shortage of the world's shipping tonnage during the European War. The maximum record of the annual outputs was 646,300 tons in 1919 and it would have exceeded 1,000,000 tons if the supply of shipbuilding materials had been easy enough to let the shipyards work in full efficiency.

Before the War, Japanese private shipyards had built some gun-boats and some merchant ships including light-house tenders, steam yachts etc. to the orders from foreign countries in the orient. In the time of the War they exported their newly built merchant steamers about 70 in number and 460,000 in gross tonnage, for Great Britain, United States, France, Norway, Italy, Spain, Denmark and Russia, while the then total export of merchant steamers, old and new, from Japan for those countries was about 200 in number and 560,000 in gross tonnage.

The maximum record of the annual launching of merchant ships of 100 tons gross and upwards since 1914 in each of the chief shipbuilding countries of the world was as shown in Table 2, the Japanese record having been in the third position.

TABLE 2.

Order according to tonnage	Name of country	Record gross tonnage of ships launched in a year	The record Year
	The world total	7,144,549. exclusive of German	1919
1	U.S.A., excluding Great Lakes	3,579,826.	1919
2	Great Britain and Ireland	2,055,624.	1920
3	Japan (see Note under Table 1)	646,344.	1919
4	Germany	525,829.	1922
5	British Dominions, excluding Canadian Lake Ports	298,495.	1919
6	Holland	232,402.	1921
7	Italy	220,021.	1926
8	France	210,663.	1921
9	Denmark	143,918.	1928
10	Sweden	105,339.	1928
11	Norway	57,578.	1919
12	Spain	52,609.	1919

Thus most of the above records were made in the time of the War.

After the War the annual launching of merchant ships in Japan was about 50,000 to 70,000 tons gross as already shown in Table 1, and the Japanese launching for 1927 descended to the eighth position in the world as shown in Table 3.

TABLE 3.

Name of country	For 1927 Order according to tonnage	For 1927 Gross tonnage of ships launched	For 1928 Order according to tonnage	For 1928 Gross tonnage of ships launched
The world total		2,285,679.		2,699,239.
Great Britain and Ireland	1	1,225,873.	1	1,445,920.
Germany	2	289,622.	2	376,416.
U.S.A., excluding Great Lakes	3	124,270.	7	86,092.
Holland	4	119,790.	3	166,754.
Italy	5	101,076.	9	58,640.
Denmark	6	72,038.	4	138,712.
Sweden	7	67,361.	6	106,912.
Japan (see Note under Table 1)	8	52,473.	5	109,058.
France	9	44,335.	8	81,416.
Russia	10	43,917.	11	24,714.
Danzig	11	30,910.	10	39,597.
Spain	12	22,899.	14	11,852.
British Dominions, excluding Canadian Lake Ports	13	20,119.	12	22,950.
Norway	14	5,363.	15	10,401.
Belgium	15	4,693.	13	16,243.

Returning to Table 1, one will see that last year (1928) the figure of the Japanese launching abnormally increased to 109,000 tons gross, which took fifth position in the world for that year as shown in Table 3. Further, this year's figure will probably exceed 165,000 tons gross. However, this latest activity of the shipbuilding is not likely to continue long, as it has been caused chiefly by a contingent concurrence of the replacements of the subsidized liners on the expiration of their age limits. Still the recent progressive tendency of building cargo motor ships is worthy of paying attention—see II. *K*.

IV. The Economical Conditions of the Mercantile Shipbuilding in Japan and the Governmental Policy for It.

The aggregate gross tonnage of Japanese merchant ships of 100 tons and upwards was 4,425,645 tons in June this year (1929). For several years the Japanese merchant shipping has been the third in tonnage among the world countries, and such a tonnage is naturally essential for the home and foreign trades which support this insular country.

However, the normal position of the Japanese mercantile shipbuilding has been considerably lower than that of the Japanese shipping, that is to say, the shipbuilding has not been keeping pace with the shipping. This is chiefly due to the fact that while the shipbuilding is really a synthesizing industry in which practically all kinds of products of other industries are assembled together as the materials of ships, the development of other industries in this country has not been so perfectly harmonious as to give a satisfactory and adequate supply of materials to the shipbuilding industry and this industry has had to rely very much on the imports of materials with the unavoidable accretion of the import duties, freights, insurance premiums, and other charges, which is at least a handicap to the shipbuilding cost.

The subsidies given by the State for the mercantile shipbuilding under the Law of 1896 were intended to compensate for the above mentioned handicap, and were, for ships built, 20 Yen per gross ton and 5 Yen per indicated horse power of machineries simultaneously built, the rate for hulls having been altered by 1909 and become 11 to 22 Yen according to the class of ship and the criterion of service. This law, together with other laws for the subvention for shipping voyages discriminating foreign-built ships, had caused a remarkable increase of home-built passenger liners but had not been so effective as to induce any considerable output of cargo ships, and the annual launching of merchant ships had seldom exceeded 50,000 tons gross until the time of the European War, when a sudden shortage of the world's shipping tonnage urged the old and new shipyards in Japan, as well as in other countries, to produce a great many cargo ships, most of them standardized, with the result that the profits of the shipyards became so great and the burden on the State so increased that at last the subsidies were stopped in 1917 and ceased ever since.

While no measure was yet taken to substitute for the shipbuilding subsidy system the depression of the shipping and shipbuilding after the cessation of the War not only struck out most of the new shipyards but also afflicted even the well founded shipyards with overexpanded installations. Therefore in 1921 the present system of indirect encouragement of the shipbuilding was laid down. In this system the import duties on steel materials in general and the special equipments and machineries which are difficult to make in Japan are made free so far as they are used in building or repairing ships, and the home steel makers in their turns are to receive bounties of so much Yen per ton of steel materials of their make so far as they are used in building or repairing ships, the rates being equal to those of the import duties. Thus the disadvantage to the shipbuilding cost due to the import duties on such materials as mentioned above which mostly have to be imported from Europe has been relieved of, but there still remains the disadvantage due to the freights and other charges on those materials. We may add here that there is a disadvantage which is due to somewhat high overhead charges to be imposed on the shipbuilding cost in consequence of the general deficiency of orders as experienced in normal years.

Another indirect system for the encouragement of the shipbuilding at present is the subvention for four main Japan-America lines in which foreign built ships are prohobited to be employed except under special permission. However, for all other Japanese liners in the services by the

Government orders including those under mail contracts there is no discrimination, wherever they may have been built.

Although the tonnage of Japanese shipping is in the third position among the world countries as mentioned already, it has been expanded largely with the imports of foreign second hand cargo ships to the discouragement of the home shipbuilding, so that there are quite a number of old ships. Their replacement with new and efficient ones is hoped from the point of view of economy and safety but it is not an easy matter for many owners of insufficient means, who are the very owners of most of old ships in this country. Thereupon much was argued as to the necessity of a measure by the State to aid in breaking-up over-aged ships on condition that new ships are to be built here in substitution. Such a measure, whether in the form of bounties or in that of loans it may be, would be another desirable indirect system of encouraging the shipbuilding for the time being, but it has not been realized yet.

Formerly the customs duties on imported ships had been 15 Yen per gross ton for ships not exceeding 10 years in age and 10 Yen for ships exceeding 10 years. After the War these rates as well as those for all other commodities were reconsidered with a view to the entirely changed economical conditions. If then a policy to invigorate the shipbuilding had been introduced or if, at least, the rates had been generally increased in proportion to the average rise of the prices of commodities, the rates would have become so high that the shipping interest would have suffered too much even in the cases of the imports of young ships. So that in 1926, with an intension to suppress the import of old ships, the rates were revised to 15 Yen for ships not exceeding 20 years in age and 20 Yen for ships exceeding 20 years. Since this revision, there has been almost no import of very old ships except for breaking-up purposes.

Before the War, the wages of Japanese shipyard workers were very low, but now they are earning as high wages as in the time of the War, as shown at the bottom of the following Table 4. Still now, their wage rates are considerably lower than those in other chief shipbuilding countries, but this advantage to the shipbuilding cost is nearly counterbalanced by the inefficiency of labour owing mainly to the fact that very often materials can not be acquired in good order so as to meet the progress of works and a number of idle workers are usually kept employed even while shipbuilding orders are short as it is difficult to replenish skilled ones when required.

V. Ship Building Yards and Ship Repairing Docks in Japan.

Table 4 shows the changes of conditions of the Japanese private shipyards which can build merchant ships of 1,000 tons gross and upwards, before, in and after the European War.

TABLE 4.

Year	1913	1918	1928
No. of firms	5	53 including 12 firms for building wooden ships	18
Authorized capital	About 25,550,000 Yen	About 162,050,000 Yen	About 235,950,000 Yen
Paid-up capital	23,150,000 „	22,000,000 „	104,659,000 „
Debentures	3,600,000 „	22,000,000 „	104,659,000 „
No. of yards for building ships of 1000 tons gross and upwards	6	57 including 12 yards for building wooden ships	21
No. of berths for building ships of 1000 tons gross and upwards	17	157 including 22 berths for building wooden ships	75
No. of dry docks for repairing ships of 1000 tons gross and upwards	30	37	45
No. of workmen (contingent labourers excluded)	26,100	107,200	40,409
No. of other persons permanently employed	?	7,300	5,800
Regular daily working hours	Mostly 10 hours	Mostly 10 hours	Nearly half of yards 8 hours, about half of yards 9 hours, & a few yards 9½ hours
Average daily wage earning of a workman	.97 Yen (1914)	2.17 Yen (1918) 2.75 „ (1919)	2.81 Yen

Most of the firms perform shipbuilding, shiprepairing and certain other works, and the figures given in Table 4 are for the whole works. The extraordinary numbers of shipbuilding firms, yards and berths in the war time were due to the contingent establishment of numerous small yards, most of which vanished after the War. At present, there are 22 private shipyards which can build merchant ships of 1,000 tons gross and upwards.

The principal private shipyards are as shown in Table 5 in which only dry docks not shorter than 350 feet, 29 in total number, are given, while in this country there are 46 more dry docks of shorter lengths and about 180 slipways for mercantile shiprepairing.

TABLE 5.

Name of firm	Situation of Shipyard	Dry docks
Mitsubishi Shipbuilding Co., Ltd.	Nagasaki	728 ft. 523 ft. 375 ft.
	Kobe	16,000 ton floating dock 12,000 ton „ „ 7,000 ton „ „
	Hikoshima, near Moji	460 ft. 369 ft.
Kawasaki Dockyard Ltd.	Kobe	428 ft.
	Osaka	476 ft. 438 ft.

Osaka Iron Works Ltd.	Innoshima	469 ft. 407 ft. 355 ft.
	Kasadojima, near Hiroshima	470 ft. 365 ft.
Yokohama Dock Co., Ltd.	Yokohama	640 ft. 495 ft. 400 ft.
Asano Shipyard, Ltd.	Yokohama	659 ft. 497 ft.
Uraga Dock Co., Ltd.	Uraga, near Yokohama	497 ft. 456 ft.
Harima Shipyard of Kobe Steel Works, Ltd.	Aioi, near Kobe	420 ft.
Mitsui Shipyard of Mitsui Bussan Kaisha, Ltd.	Tama, near Okayama	485 ft. 350 ft.
Fujinagata Shipyard, Ltd.	Osaka	483 ft.
Ishikawajima Shipyard, Ltd.	Tokyo	
Hakodate Dock Co., Ltd.	Hakodate	531 ft.
Mukaijima Dock Co., Ltd. (repair works only)	Mukaijima, near Onomichi	410 ft.

VI. Various Engineering Plants in Japan, Allied or Auxiliary to the Shipbuilding.

This subject belongs to the separate branches of engineering, so that we will only state here the names of the Japanese firms and their products chiefly concerned with the shipbuilding.

(*a*) *Engines, boilers, and auxiliary and deck machineries.*

In Japan, every large shipyard has its own department of marine engineering to build machineries, but there are also many other firms specialized in marine engineering, especially motor engineering.

We omit to state here the names of the marine enginering plants and the details of their products, making all such matters over to another writer.

(*b*) *Rolled steel materials.*

Government Steel Works, Yawata, have the largest mills and manufacture all kinds of rolled steel materials.

Asano Steel Works, Yokohama, manufacture steel plates; Kobe Steel Works, Kobe, Steel shapes; and Tokai Steel Works, Dairi near Moji, steel plates and shapes.

Kawasaki Sheet Steel Mills, Kobe, manufacture steel plates also.

Tokuyama Steel Plate Mills, Tokuyama, manufacture steel plates.

Nippon Steel Pipe Co., Kawasaki near Yokohama, manufacture steel shapes and all kinds of steel pipes.

(*c*) *Large steel forgings, large steel castings and anchors.*

Kobe Steel Works, Kobe.

Sumitomo Steel Casting Works, Osaka.

Nippon Steel Works, Muroran.

Oshima Steel Works, Tokyo.

Hyogo Branch of Kawasaki Dockyard, Kobe.

(*d*) *Chains.*

Osaka Chain Manufacturing Co., Osaka, manufacture all sizes of chains with Hingley iron bars and have a branch plant in Yokohama. There are several other chain makers in Osaka and Tokyo.

(*e*) *Wire and Manila ropes.*

Tokyo Seiko Kaisha, Kawasaki near Yokohama, manufacture all kinds of wire and Manila ropes and have branch plants in Kobe and Kokura. There are several other rope-makers in the environs of Osaka.

(*f*) *Paint.*

Nippon Paint Co., Tokyo.

Fuji Paint Co., Kawasaki near Yokohama.

Kawakami Paint Co., Osaka.

Several other paint companies in Tokyo, Osaka and Hiroshima.

(*g*) *Compasses, telegraphs, search lights, pressure gauges, revolution meters, thermometers etc.*

Tokyo Keiki Seisakusho, Tokyo.

(*h*) *Binoculars and other optical instruments.*

Nippon Optical Engineering Co., Tokyo.

(*i*) *Ship's lamps, signal lights, life buoys, life jackets etc.*

Several specialized makers in Tokyo and Osaka.

VII. Colleges and Schools in Japan for Naval Architecture.

Naval architecture is taught in the following colleges and schools in Japan.

Tokyo Imperial University, Tokyo.

Kyushu Imperial University, Fukuoka.

Osaka Engineering College, Osaka.

Yokohama High School of Engineering, Yokohama.

Kogakuin, Tokyo, a night school.

Mitsubishi Shipyard, Nagasaki, has its own apprentice school.

At several local centers of wooden ship building, there are schools for shipwrights' apprentices.

Naval dockyards, have a few training schools for junior naval architects as well as for junior engineers.

VIII. Ship Model Experimental Tanks in Japan (Navy tanks being excluded).

The tanks—i.e. basins—for the model experiments to study and determine the resistance of ships, the efficiency of propellers, etc. are provided in Japan as follows:—

Mitsubishi Tank, Nagasaki, for the private use of Mitsubishi Shipyards,

length 400 feet, breadth 20 feet and depth 11 feet.

Teishinsho (the Communication Department) Tank, Tokyo, for public use,

length 140 metres, breadth 10 metres and depth 6.1 metres.

Fisheries Institute Tank, Tokyo, for the researches of fishing boats, seines and other fishing instruments,

length 263 feet, breadth 20 feet, and depth 10 feet.

There is also a small tank in each of Kyushu Imperial University and Osaka Engineering College.

IX. Government and Private Competent Authorities in Japan, Supervising Merchant Ships and Mercantile Shipbuilding.

(a) The Government supervises merchant ships and mercantile shipbuilding by laws regarding the safety of life and ship, the hygiene and accommodation of persons on board, etc.

Teishinsho (the Communication Department) has the central bureau called Kansenkyoku (Mercantile Marine Bureau) in Tokyo and there are 5 District Directions of Communications under which the inspectors are stationed at 23 Kaijibu (Local Marine Offices) at important ports and maritime centers.

(b) Classification societies supervise merchant ships and mercantile shipbuilding upon the request of shipowners, and classify and register ships for the purpose of giving references which may be useful in the cases of insuring, buying, chartering or otherwise utilizing ships.

At present, Teikoku Kaiji Kyokai (Nippon Marine Corporation) is the sole Japanese classification society, and has the head office in Tokyo and the local offices at six principal Japanese ports. Also at four principal Japanese ports, there are stationed the surveyors to Lloyd's Register of Shipping.

X. Final Remarks.

In concluding this paper we are to add that we have omitted the details of the progressive changes in the machineries of ships built in Japan, from reciprocating steam engines and coal burning boilers to oil burning boilers, turbine engines, electric motors and Diesel motors, and also the details of the marine engineering departments of shipyards and other specialized marine engineering plants, and of their products. We expect another writer will present a paper to the Congress, treating such matters minutely.

The end

Steam Turbine Driven Auxiliaries for Ship Work.

(Paper No. 254)

By R. W. Allen, C.B.E., M. Inst. C.E., M.I. Mech. E., M. Inst. N.A.

Following the rapid development of the Steam Turbine for marine main propulsion there has occurred a remarkable development of the utilisation of the steam turbine for the driving of auxiliary machinery.

It is not surprising that having experienced the advantages of the steam turbine as a prime mover for driving the ship, attention should be given to the driving of all important auxiliary machinery in the same efficient manner. Further, the main turbines themselves demand a greater efficiency in their auxiliary machinery, for it is well-known that whereas the reciprocating steam engine will function satisfactorily with a moderate vacuum, the steam turbine depends for its characteristic efficiency upon the utilisation of highly efficient condensing apparatus, hence the auxiliary requirements of the turbine in the matter of condensing plant are of a highly exacting nature.

It may now be said that turbine driven auxiliaries have become standard practice not only in ships of war but also in the Mercantile Marine and this is evidenced by practically all recent large steam ships being equipped with steam turbine driven circulating pumps, generators and in many cases forced draught fans.

It is unnecessary to recount the advantages which Turbine drive has over other forms of propulsion. Suffice it to say that the same arguments which have led to the adoption of turbines for main propulsive purposes are in a great measure applicable to the auxiliary machinery: evenness of torque, flexibility, cleanliness and high thermal efficiency are characteristics which are generally credited to the steam turbine and these all carry considerable weight in relation to the use of turbine driven auxiliaries.

While the size and output of the auxiliary machines may be somewhat insignificant compared with the main propulsion turbines their duties are no less important and the design and construction of these auxiliary turbines for various duties has received great attention during the past few years with a view to ensuring the maximum economy and efficiency. At this point it will be convenient to classify the types of marine auxiliaries to which turbine drive has been applied:

1. Electric generating sets for ship lighting and power purposes.
2. Centrifugal circulating pumps for circulating water for main condensers.
3. Bilge Pumps.
4. Boiler Feed Pumps.
5. Boiler Room Fans for forced draught air supply.

Turbine Driven Electric Generating Sets.

One of the most important auxiliaries is the plant which provides the electric lighting of the ship and power for various purposes. The importance of the electric power supply on board ship need not be enlarged upon, and may almost be said to be primary and not an auxiliary part of the equipment.

Apart from the lighting, innumerable motor-driven auxiliaries are dependent upon the power supplied by the turbo-generators, and the demand for more power for heating and other uses is steadily growing, so that the modern dynamo-room is in itself a moderate-sized electric power station. As an example, ships have lately been constructed having turbo-generators with a total capacity of 2,000 k.w., and larger capacities still are under consideration. It is the invariable rule to instal direct current machinery for this purpose and the rating of these machines varies widely both as to output and also to a less extent as to voltage. But all such machines must be of the robust, simple and reliable character demanded of all marine auxiliaries.

A typical installation might consist of three or four Generators, each of about 400 k.w. output, and a 10% overload would probably be required for a period of a few hours. Turbine speeds range from 3,000 to 6,000 revs. per minute, while the speed of the Generator ranges from 500 to 1,000 revs. per minute.

Quietness of the reduction gearing is invariably required as no engineer is inclined to nullify the advantages of the smooth running turbine by permitting a noisy or vibrating transmission.

Steam is generally supplied at from 150 to 250 lbs. per square inch with 150 to 200° F. superheat, and the maximum economy is required when the turbines are running under normal sea-going conditions. These conditions are frequently specified as exhaust to feed heaters against a small back pressure; occasionally exhaust is to the L. P. Stages of the main turbines. The sea-going conditions are those for which the turbines are designed to give their best efficiency, while dock conditions are generally given as exhausting to about 20 inches vacuum. Maximum steam consumptions are usually specified under both dock and sea-going conditions.

The design of turbines for marine auxiliaries probably makes greater demands on the designer than is generally appreciated. The production of a turbine to work efficiently under the severe and varying conditions found in ship-work, to give good and reliable service both on the high seas and in harbour requires the utmost co-operation between designer, metallurgist and the constructor; in addition the experience of the engine room staff should be embodied in every machine.

The following details are given to indicate some of the main points in design.—

Turbine Rotor.

With the steam conditions usually specified it is found possible to expand the steam in a single impulse wheel, this being of the velocity compounded type. Where the back pressures are less than usual, e. g. 5 lbs., it may be advisable to add a second stage single row wheel to secure the best economy. In spite of the apparent advantages of a number of single row wheels expanding the steam in a number of stages, the present speeds of revolution would tend to greater losses due to wheel friction and this would probably more than counter-balance the economy obtained by higher blade efficiency. The wheels for these high speeds are sufficiently small to justify the rotor being forged in one piece, thus making for greater strength and rigidity. In any case, even when the wheels are larger and are shrunk on to the shaft, the rotor is designed so that its normal speed is below its first critical speed. The upper limit of speed for such units may be taken as about 8,000 r.p.m., a more common speed being 6,000 r.p.m.

The material of the rotor shaft is mild steel having a tensile strength of 34 to 38 tons per square inch with a minimum elongation of 23%.

Great care is necessary in the machining of these rotor discs in order to reduce wheel friction to a minimum and to avoid risk due to the effects of tool marks left in the metal.

Turbine Blades.

In the case of blading of steam turbines, there has been a constant search after materials to cope with the exacting conditions of modern practice, copper and brass being replaced by phosphor bronze, and the latter itself, in general use until quite recently, being then largely replaced by high-chrome stainless steel. Phosphor bronze, although considerably stronger than copper or brass, is a comparatively soft material. Its Brinell hardness is approximately 120 as compared with 230 for stainless steel, and it is therefore not surprising to find that it sometimes suffers unduly from erosion, a form of attack from which stainless steel is comparatively immune. In a particular case where stainless steel blades were fitted, some phosphor bronze blades were also put into the last wheel, and it was found that after running, whilst the steel blades were still in good condition, erosion had occurred to a considerable extent on the inlet edges of the phosphor bronze blades. Even stainless steel is now being discarded, chiefly by reason of the difficulties in manufacture, particularly in riveting, caused by its hardness. High-chrome stainless iron possesses the requisite properties, a ductility lacking in stainless steel and an excellent resistant to erosion in spite of its slightly lower Brinell figure of 200.

Turbine Nozzles.

The nozzles upon which so much of the efficiency of any turbine depends must be designed so that they may retain their true shape under severe conditions of heat and pressure. High pressure nozzles usually consist of stainless steel plates cast into cast-iron diaphragms.

Material which will completely withstand the erosive and corrosive action of steam under all conditions found in steam turbine practice and which are also mechanically satisfactory under heavy temperature and pressure stresses have yet to be evolved by the metallurgist, but if the working conditions are estimated correctly it is possible to provide materials for turbine parts which will give satisfaction over a wide working range.

Turbine Governing.

Steam turbines when driving pumps and fans usually have no control governors fitted as the load is a constant one and there is no particular tendency for the turbines to get out of control. All sets are however fitted with emergency governors which operate should the set exceed a pre-determined speed, say 10% above the normal running speed.

A further reason for not using governors on fans and pumps is that the speed range has to be extremely wide. On war ships when cruising the fan speed is reduced considerably as very much less air is required for the furnaces. The speed control is more easily obtained by controlling on the stop valve than by having a speed governor arranged to cover such a wide range of speeds. With the electric generating sets the case is different and each set must have a sensitive speed governor.

It is most essential that the electric generating sets should run in parallel with each other. The governing is very frequently specified for dual steam conditions viz. to exhaust against a back pressure when at sea and into a condenser when in port.

The governor gear must therefore be designed to pass the comparatively large quantity of steam which is necessary when running on back pressure conditions and further it must successfully control the turbine when it is only passing a sufficient quantity to run the set on no-load when under vacuum. The two steam quantities are, as will be appreciated, very different, and the designing of the governing gear needs very special consideration for these rather onerous conditions.

The present tendency towards high pressures makes the controlling problem possibly more difficult, as when the set is running at a vacuum it takes a very small steam quantity indeed to run the set up to full speed on no-load.

Turbines of small output using throttle valve of not more than 2″ in diameter are sometimes fitted with direct acting governor; thus the power of the governor itself is used directly to move the valve and vary the amount of steam admitted to the machine. For the larger valves the invariable practice is to have the governor used in conjunction with an oil relay. This oil relay does not affect the essentials of governing in the least, but it gives one very much more power to move the valve.

The method of control is quite simple, the supply of lubricating oil under pressure is supplied either from the lubricating oil pump or from a separate pump. The governor controls a small pilot valve which admits or allows oil to escape from under a piston. This piston is spring loaded and the oil pressure compresses the spring. The throttle valve is operated by a lever or direct from the oil control piston.

The selection of suitable metals for the valves, spindles and spindle glands calls for much experience, particularly now that temperatures are becoming very high.

Valve material must be sufficiently hard so that it is not easily cut or eroded by the steam.

The spindle must as far as possible be free from a tendency to rust and it must work satisfactorily in the gland bush through which it passes. Plain glands are always used, as packed glands are open to abuse by screwing up too hard, and this would have deleterious effects on the governor.

The necessity of clean feed is important. As the make up is nearly always distilled water little trouble is likely to be experienced due to impure and dirty steam with a consequent tendency to stick up the moving parts of the control valve gear.

Turbine Balancing.

The steam turbine being essentially a high speed machine, accurate balance of the rotating parts is vital if good running and satisfactory service are to be obtained.

Two kinds of balance are possible, viz. static or standing balance, and dynamic or running balance. The first is obtained by rolling the rotor on suitable knife edges or scraped strips. The knife edges are carefully set up parallel to each other and level, the rotor placed thereon and given a tendency to roll. The heavy part eventually settles at the bottom, and by removing weight at the heavy portion a condition is obtained where the rotor will stay in any position on the strips.

For small rotors with single wheels very good balance indeed can be obtained in this way. Further, where the rotor consists of a number of separate wheels the balance may be obtained very accurately by the static method by balancing after each separate wheel has been mounted on the shaft.

Multi-stage rotors which are not balanced after each wheel has been mounted on the shaft cannot be statically balanced with entire certainty as there is always a possibility that a couple may exist, and this may have serious effects on the running.

It will be realised that it is quite possible to have a rotating member in perfect balance statically and at the same time badly out of dynamic balance. It is becoming more and more general therefore to balance all high speed machinery in a special dynamic balancing machine.

These machines vary considerably in details of design, but essentially they consist of bearings in which the rotor is mounted, a driving medium for running the rotor up to a certain speed and springs to indicate vibration should there be out of balance.

In addition to the necessity of balancing accurately, other factors may affect the running of a turbine such as loose wheels, critical speeds of shafts and blades. Dealing with the first item, wheels must be fitted by shrinking or forging on to the shaft so that they do not come loose when running. A loose wheel will cause very bad vibration. It is further inadvisable to run the rotating shaft on its critical speed. When running at its critical speed very small forces can cause very big deflections of the shaft and cause serious vibration. Should this vibration be allowed to continue there is a danger of the shaft fracturing.

Blades must be so proportioned that the critical speed is well removed from the running speed.

High temperatures are liable to cause distortion in castings and the effect of this has to be carefully considered in the design, as should the turbine get out of line due to this distortion it may lead to most unsatisfactory running.

Reduction gearing should be carefully balanced and the clearances at the bearings kept down as small as possible in order to ensure good running.

Reduction Gearing.

The main turbines in the early days of marine practice suffered in efficiency owing to the fact that the speed had to be kept sufficiently low to suit the propeller. To give the best efficiency the turbine must run at a fairly high speed, conversely for the best efficiency of the propeller a low speed is best. The introduction of mechanical reduction gearing has enabled both the driver and the driven member to run at their best economical speed.

What is true of the main turbine applies with equal force to the auxiliaries, and it is now almost the universal practice for them to be driven from reduction gears. This general statement applies whether the turbine be driving the generating sets, the fans or the circulating pump. Further, for a given output the sets can be made more efficient and more compact when gearing is introduced.

The need for gear is more manifest than ever today where steam conditions are getting higher and higher, and consequently the heat available in the turbine is also increasing. The driven member speed remains substantially constant, and in order to keep pace with the member greater heat drop it is necessary for the prime mover to have a higher peripheral speed, or alternatively more stages.

Here gearing is obviously of great assistance as the ratio can be easily varied and therefore the prime-mover can be run at a speed to suit the particular conditions, whereas the driven member speed can remain at its economic value.

Pitch line speeds of gears may run up to 150 feet or more per second, and in order to obtain noiseless running great accuracy must be exercised in cutting the teeth of the gears. The hobbing machines must be periodically overhauled and every care taken to ensure a high degree of accuracy. Further the hobs must be correctly shaped and kept sharp.

The materials generally used for the gear wheel rim are mild steel and for the pinion 5% nickel steel. This harder material is used for the pinion on account of the higher number of impacts which take place per tooth.

Condensers and Condensing Plant.

In the development of machinery on board ship generally the condenser and condensing plant has played an important part.

In the past marine installations generally consisted of a surface condenser with cast iron shell with air pump of the piston direct-acting plunger type with foot, bucket and head valves.

This type of plant has now disappeared, and frequently for auxiliaries the condenser has been made of steel and mounted alongside the turbine, being connected to it by an overhead exhaust pipe. But in recent installations, due to the demand for lighter weights for the turbo-generating sets and the need for economy in space a new type of plant has been developed in which the condenser forms part of the baseplate for the turbine.

In this case the circulating pump is driven from the end of the electric generator and the extraction pump is driven through gearing from the end of the turbine shaft.

The air and vapour is removed by a 2-stage steam-jet air-pump, and this is also mounted on the baseplate so that the whole plant is self-contained.

The condenser is of the regenerative type, that the condensate temperature depression has disappeared, and the water as handled by the extraction pump has the same temperature as the steam entering the condenser. Further by a suitable arrangement of steam lanes etc. the pressure drop across the steam side of the condenser has been eliminated.

Incidentally to this improved design the heat transmission per sq. foot per degree F. temperature difference has been greatly increased owing to the more efficient steam distribution, and lastly the oxygen contained in the condensate leaving the condenser is now as low as 0.02 cubic centimeters per cubic meter. This latter is of great importance as it is well known that oxygen has a very detrimental effect on boiler tubes and may cause corrosion.

Centrifugal Circulating Pumps.

Turbine drive has now been successfully applied to all the larger circulating water pumps used in ship-work under normal circumstances. As mentioned before, the demands of the main turbines for high vacuum adds considerably to the responsibilities of the condensing plant, and both air and water extraction and circulating water pumps must be driven by a particularly reliable source of power. It is here that Turbine drive excels, for it is evident that an engine room staff which has become used to turbine conditions will be more at home with turbine driven auxiliaries.

Two points to which attention should be drawn and which are of considerable importance in connection with turbine driven pumps are the possibility of using vertical spindle pumps where there is little floor space but plenty of head room; and the absence of noise and pipe shock due to uneven velocity of the water.

The steam turbines may be directly coupled to the pump, but the tendency is to introduce a reducing gear between the turbine and the circulating pumps in order to obtain better economy in steam consumption. In this connection it may be mentioned that a recent example of a geared turbine pump delivering 22,000 G.P.M. against a head of 22 ft. had a steam consumption only half that of a direct turbine-driven pump of similar output constructed a few years ago. It might be of interest to compare the steam consumptions of these two types of circulating pumps under the same steam conditions. In the case of the direct-driven turbine pump the particulars are as follows:—

Revolutions per minute.	770
Quantity of water delivered per min.	22,000 galls.
Head.	22 feet.
Steam Consumption, lb. per hour	19,870
Lb. of steam per W.H.P.	135

With regard to the gear-driven set, which has the same output as the direct-driven set, the following are the figures:—

Revolutions per minute (pump)	600
Revolutions of steam turbine	4,000
Quantity of water delivered per min.	22,000 galls.
Head.	22 ft.
Steam consumption, lb. per hour	9,744
Lb. of steam per W.H.P.	66

These figures indicate very clearly the saving in steam consumption of the geared set as compared with the direct-driven set.

The pump speed of the geared set was somewhat lower than that of the direct-driven pump, a very desirable feature from the point of view of the design of the pump, whilst the turbine of the geared set ran about six times the speed of the direct driven set permitting the use of a single turbine wheel of 18 in. diameter as against two wheels of 33½ in. diameter for the direct-driven set.

The space occupied by the geared set was very little more than the direct-driven set, and being vertical spindle pumps this difference was in a vertical direction only.

Bilge Pumps.

An interesting example of the value of the adaptability of the turbine to the driving of Vertical Spindle pumps is the pipeless Bilge Pump. This pump is driven by a steam turbine by means of a vertical intermediate shaft which can be made of any reasonable length and consequently the steam turbine can be placed in a convenient and accessible position well above the highest possible level of water in the vessel.

To compensate for any lack of alignment between the turbine and Pump the vertical intermediate shaft is connected to the turbine spindle by a flexible coupling while the whole weight of the rotating parts is carried by Michell thrust bearings one above and one below the flexible coupling.

The turbine which is of the single stage impulse type already described runs at 900 revs. per minute, is specially designed with a view to securing great reliability and ease of inspection. Both pump and turbine are automatically lubricated, the pump bearings being of lignum vitae, water lubricated from the pump itself, the turbine lubrication being provided by a gear driven oil pump. As this pump is mounted directly on the tank top or inner bottom of the vessel there is no pipework and consequently the usual risks of pipe damage and leakage are avoided. The entire absence of pump pipework also reduces the weight of the installation to a minimum.

Boiler Feed Pumps.

The turbine driven Centrifugal Feed Pump may be cited as a further example of the value of turbine drive for reducing the space occupied by auxiliary machinery. The duty of boiler feed pumps demands reliability and robustness and this can be provided in a Turbine driven centrifugal pumping installation without recourse to the massive structures necessitated by feed pumps of the reciprocating type.

The great perfection which the reciprocating boiler feed pump has attained is responsible for the slow headway made by the turbine driven auxiliaries of this type, but it may be confidently affirmed that the rotary type of pump has the advantages of smoother running and consequently a more steady delivery, less weight and space occupied, and equal steam economy.

Centrifugal pumps for boiler feed purposes are generally of the 2 or 3 stage type, thus making it possible for the turbine to run at speeds not greater than 2,500 to 3,000 revs. per minute.

The possibility of gearing down to quite low speeds is an advantage possessed by turbine drive of pumps which it is difficult to over estimate. On the one hand the mechanical losses become excessive at high speeds as these losses depend upon the square of the velocity imparted to the water. But while this would in itself put a definite upper limit on pump speeds, the questions of erosion and corrosion are of even greater moment. The erosion, or mechanical tearing action of the water on the pump blades rapidly increases with increase of speed. This in its turn renders the pump a much easier prey to the ravages of chemical and electro chemical actions which are commonly grouped under the heading of corrosion. Hence, the high speed pump is not only mechanically inferior to one of slower speed, but its cost of maintenance will be considerably greater. Gearing solves this problem exactly as it solves the parallel problem of the drive of Continuous Current Electric Generator.

Boiler Room Fans.

Much more attention is nowadays paid to the efficient air supply to boiler rooms than in the past, and the fact that for a small cost a highly efficient supply of air can be obtained if the fans are turbine driven has resulted in the installation of many turbine driven fans especially on ships of war.

The ultimate efficiency of any fan service depends in the first place upon the economy of its prime mover and this is assured if the steam turbine is employed. It is also necessary however, to provide for adequate and properly designed fan entrances and air channels. This is in itself a large subject and can only be mentioned in passing. It may be pointed out, however, that by careful design of the air intakes and channels, and the removal of blockages and awkward junctions, it has been found possible to effect a reduction of 8% in the steam consumption of fans and give the same air supply.

One of the advantages of turbine drive for fans is that the turbine may be coupled to the fan through gearing whether the fan is to work vertically, horizontally, or at some intermediate inclination. This is probably more important in the case of fan work than in any other case as it is frequently necessary to instal fans at awkward angles in order to give the necessary air conditions.

Among the other advantages of turbo-driven fans may be noted (1) they may be smaller and lighter for a given output than reciprocating engine fans, (2) they may be bolted up to the deck and thus dispense with the fan flats, (3) the steam consumption does not increase with running wear and tear as is the case with the Steam Engine.

Conclusion.

It has been impossible in the course of this short paper to do more than indicate the general lines of development of the steam turbo-driven auxiliary, but it can be confidently affirmed that the turbine has found an assured place as an auxiliary drive and this may be attributed to a wide variety of causes.

The turbine provides a range of speed unequalled by the engine. For fast running machinery it will give 6,000 or 8,000 r.p.m. and it may be geared down satisfactorily to give 200 or 300 r.p.m. It will give vertical or inclined, direct or geared drive and has therefore the advantage of flexibility when laying out the ship's machinery. It also gives an exhaust free from oil and is the prime mover possessing the greatest freedom from noise and vibration. So long as steam retains its place in marine propulsion it may be safely prophesied that the turbine will continue to grow in popularity as the most satisfactory drive for marine auxiliary machinery.

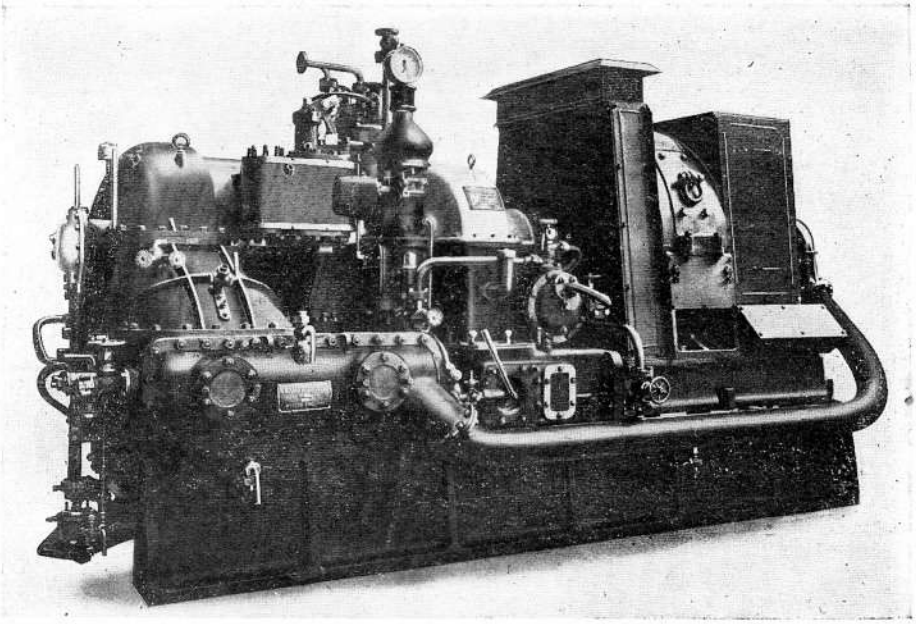

Fig. 1. 300 KW. Turbine Driven Auxiliary Generator complete with built-in Condenser.

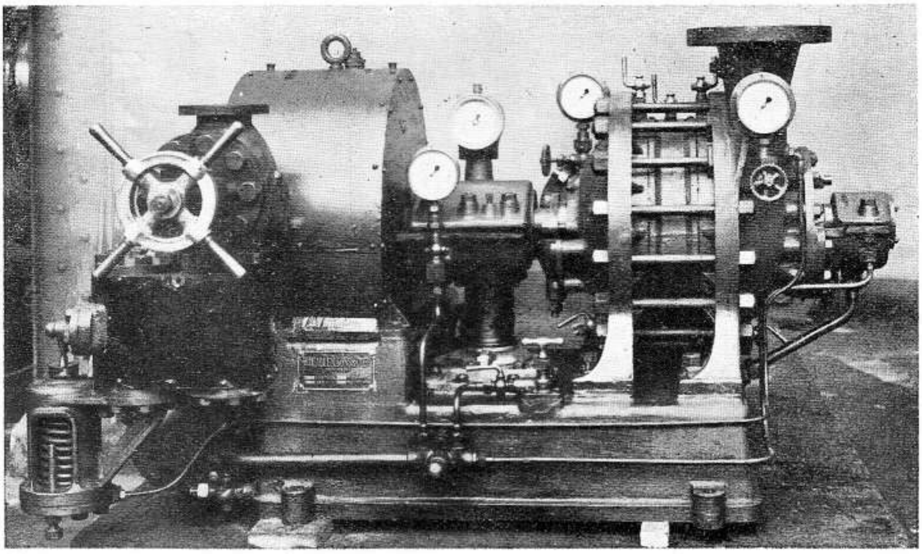

Fig. 3. Direct Steam Turbine Driven Multi-stage Turbine Boiler Feed Pump.

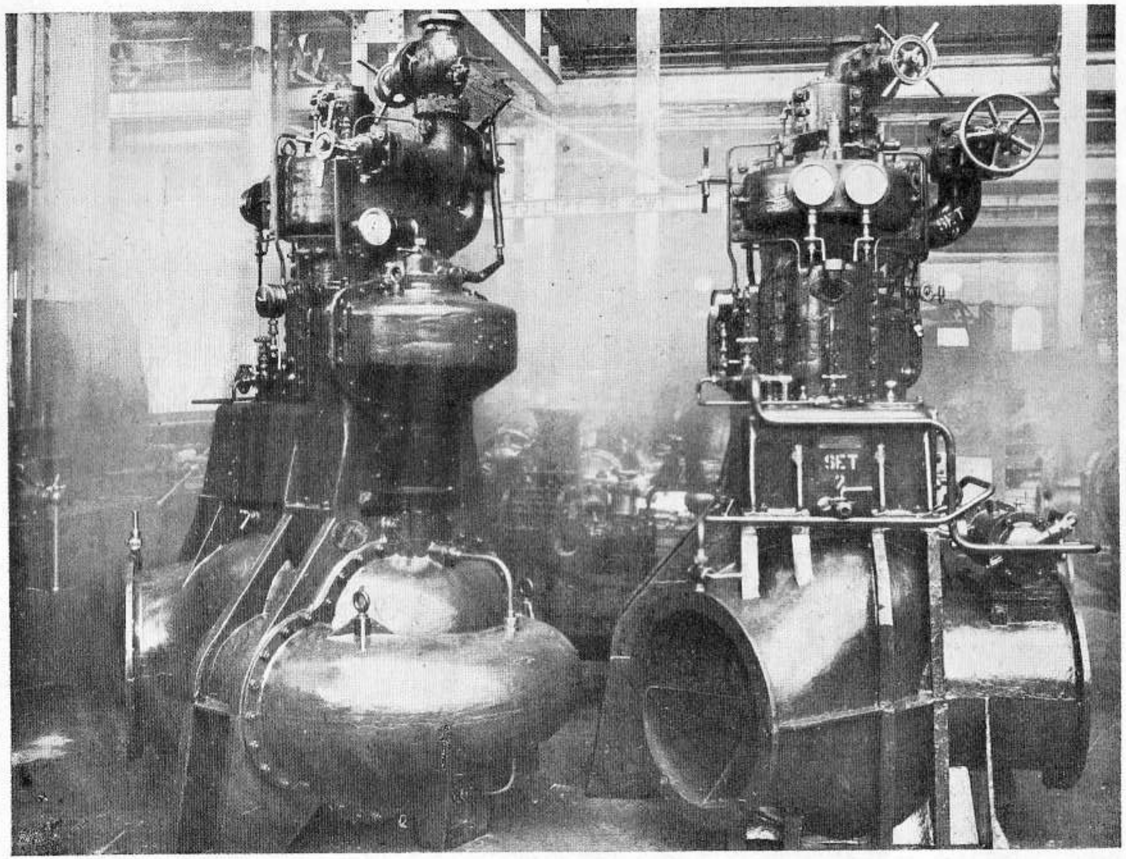

Fig. 2. Two 300 BHP. Vertical Turbine Driven Geared Circulating Pumps.

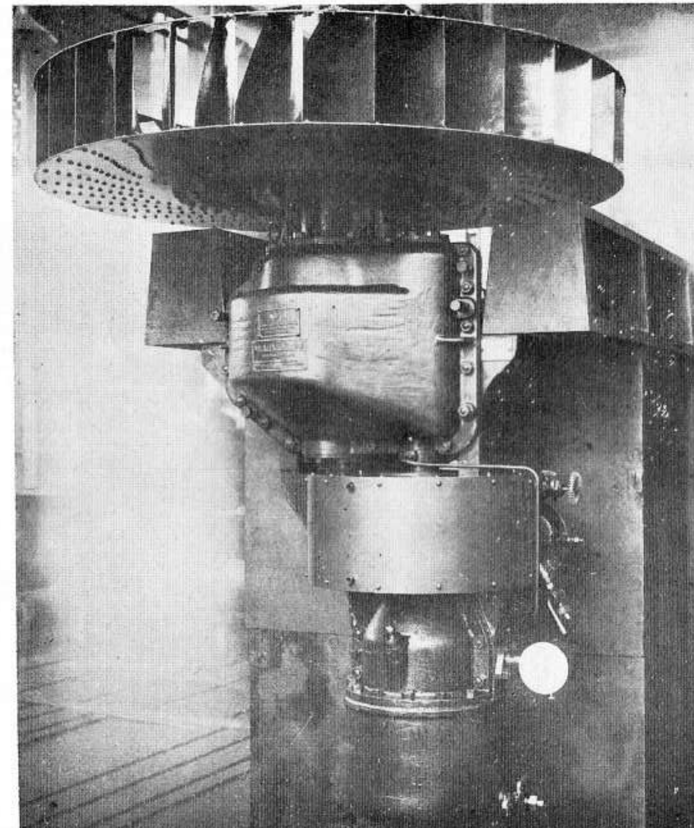

Fig. 4. Vertical Geared Turbine Driven Fan for air supply to Boiler Room.

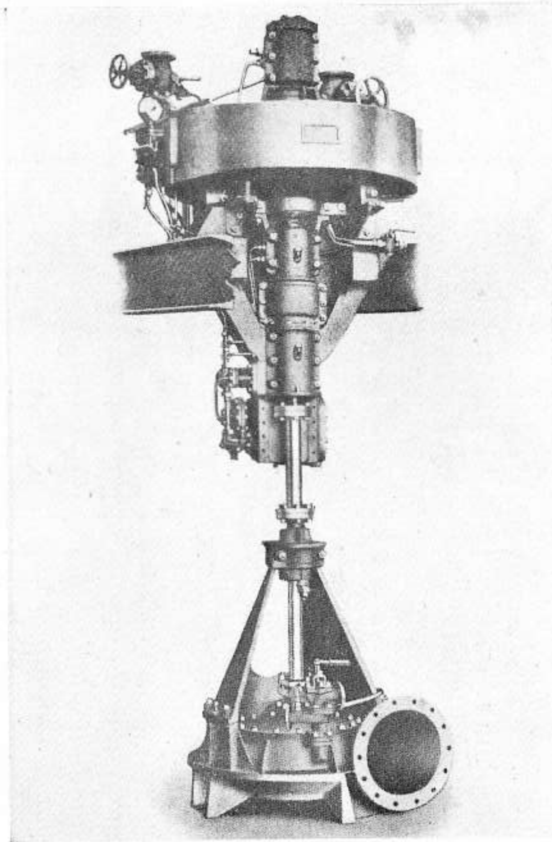

Fig. 5. Vertic l Steam Turbine Driven Pipeless Bilge Pump.

Diesel Auxiliary Generating Machinery.

(Paper No. 255)

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It is interesting to look back over a period of more than 40 years and to record the developments which have taken place in the supply of machinery for the production of electric light and power.

The first record is of the small single cylinder open steam engine driving directly a continuous current dynamo. This type was used for some considerable time, then developed the twin open compound engine.

The next step was the introduction of the enclosed high speed engine where forced lubrication was adopted, and subsequently owing to the increase in the size and power of the generating plants, the steam turbine gear driven dynamos were introduced.

This gives us three very interesting records incorporating three different types of machinery, and today we have to record the fourth, i.e. the oil engine driven generator.

The present tendency towards electric drive of marine auxiliaries and particularly of those in motor ships has led to the rapid growth of the ship's power station. In some cases the power required for the various auxiliary services is 15 or 20% of the power of the main engines and since the auxiliary services are essential to the running of the main engines, the importance of the units employed for this purpose can hardly be exaggerated.

It may be pointed out that whereas in the old type of ship there were independent supplies of power for steering and other essential services, in the modern motor ship the idea of a centralised power station has been carried into practice with exceptional thoroughness. In general the whole of the following services are provided by the ship's power station, viz.—supply of power to steering gear, fuel and cooling water pumps for main engines, bilge pumps, refrigerating plant, hoists, winches and capstans, lighting, heating, cooking, laundry and other domestic functions. When the importance of these vital services is considered, it will be realised that the auxiliary installation must be a paramount consideration.

Therefore, reliability must be placed above everything in the design of auxiliary Diesel Engines and the type and number of the engines per ship must be such as to ensure an absolutely continuous supply of electric power for these essential services and at the same time permit of the most economical running under both sea-going and port conditions.

The single acting 4-cycle Diesel Engine has proved its reliability in service and has been adopted in by far the largest number of auxiliary Diesel installations both in passenger and fast cargo-carrying motor ships. This type of engine has held favour for many years but there has been a natural evolution of this engine, resulting from the experience of both engine builders and ship owners, which gives us an engine which today has reached a high state of efficiency.

I propose in this paper to indicate some of the main lines of development of this type of engine as they have occurred during the past few years.

Such engines are placed in ships in units ranging from 100 H.P. to 700 H. P. in groups of twos, threes or fours. The total H.P. of the auxiliaries in a motor ship may thus range from about 200 H.P. to 2800 H.P.

While many features of the Diesel Engine have been subject to considerable development it will be realised that certain features remain unchanged over long periods, and in the case of the 4-cycle engine this may be said of the fuel injection system which has been found to be entirely satisfactory. It is in fact chiefly in mechanical construction rather than in thermo-dynamic principles that changes have been made. In other words, the cycle of operations remains unchanged but the material and mechanism involved in these operations have been subject to far-reaching improvements.

Crankshaft.

The tendency in modern Diesel Engines is to relieve the crankshaft of as much bending stress as possible by increasing the rigidity of the frame and reducing unsupported lengths of crankshaft to a minimum. For the same reason flywheel overhang is also reduced to a minimum without impairing the accessibility of coupling bolts, bearing caps, etc. The dynamic balance and freedom from torsional vibration are now determined with great precision before an engine is put into production. As the engine passes through the Shops the actual weights are obtained in order to recheck these calculations and to prevent any departure from the design.

It is necessary to make a clear distinction between questions of (1) Balance and (2) Torsional Vibration. As to (1), it is hardly possible to make a reciprocating Diesel Engine of any kind absolutely free from unbalance. Even in a Six cylinder Diesel Engine, one has to take into account the unbalance of the compressor with its resulting free forces and couples, but experience has clearly shewn that given suitable foundations a certain amount of unbalance can be allowed without having any effect on the structure of the ship. In other words, with this certain amount of unbalance it is hardly possible to say whether the auxiliary engines are running. Even if the periodicity of this remainder of unbalance coincides with the natural frequency of the ship no serious consequences are to be felt. This latter point is of importance to the Diesel Auxiliary Engine builder, because he may not know any of the natural frequencies of the hull structures which in many cases ought to be investigated by the Main Engine builders. Therefore, the only course open is to keep the absolute figures of unbalanced forces or couples so low as to have no effect, even in the case of the coincidence of the periodicities of both ship and auxiliaries.

As to (2), the problem which one has to face is not only to design the crankshaft free of major critical speeds for torsional oscillation within the range of one given running speed, but also to build the engine so as to meet the requirements of various shipbuilders who want to run a certain design of engine at speeds differing from the standard. As is well known, it is very difficult to change the natural frequency of a given design, the whole of which has been worked out from the point of view of compactness, economy of space, and economy of weight. The practical solution of this problem forcibly leads the designer to simplify his whole torsional system as much as possible. This is mainly achieved by attempting to have a single noded vibrational system only, or in other words to have one major mass only. This leads to the uniting of the two major masses in the system, namely, the flywheel and the armature, when, from the point of view of the calculation of the elastic length, the torsional rigidity between these two masses becomes practically infinite as compared with the elastic length between the minor masses.

Instead of a shaft with a bearing between flywheel and armature, the spider carrying the armature core is carried right against the flange, which couples the armature to the flywheel. As the elastic length varies with the polar inertia or with the fourth power of the shaft diameter, it will easily be seen that this armature construction is very rigid.

This consideration has resulted in a most important development, the elimination of the intermediate bearing, the additional bearing pressure resulting from this being adequately provided for by additional bearing surface at the flywheel end of the engine. Moreover, this stiffening of the whole installation by the omission of the intermediate bearing has the effect of reducing the overall length of the plant, a valuable consideration in marine work.

In order to simplify again the torsional system, care must be taken that the major criticals resulting from the straight line addition of all the components out of the swinging form are put well above the maximum running speed.

If this is achieved, one has to deal with the second majors only, which have a comparatively narrow range of running speeds to be avoided. However, it must always be remembered that this question of torsional vibrations on auxiliaries requires the utmost consideration, mainly because the damping effects on such comparatively high speed engines are relatively much smaller than, for instance, in the main Diesel Engine. There is no propeller turning in a soft medium capable of taking up the damping and there is no long shafting of uniform diameter with water cooled bearings capable of dissipating a great amount of heat energy created by the molecular work of torsional vibrations.

One other point should be mentioned namely, when the engine is finished and careful torsiograph records have been made over a wide range of running speeds, the damping effect which tends to lower its vibrating stresses will have considerably decreased after a certain period of running. This is only to be expected in view of the fact that the damping effect is due to those friction parts such as piston rings and bearings which after a period of say, three months of service will have become thoroughly well run in. For this reason the torsional stresses shown by an engine after a few months of service are always higher than those indicated by the new engine when running on a test bed.

Fig. 1. shows the Shaft of a 5-Cylinder 210 K.W. Set and of a 6-Cylinder 300 K.W. Set, both engines running at 300 r.p.m. At the right hand side are shown the flanges on which a flywheel and a dynamo shaft are attached by means of reamered bolts.

The end of the crank webs are chamfered away to avoid unnecessary rotating masses which would lower the natural frequency of the shaft.

At the left hand side of the 5-cylinder shaft is shown a flange which carries a small flywheel, which has been designed in order to lower the natural frequency in a special case.

Bedplate Trunk and Cylinders.

The original Diesel Engine was designed to generate power for land purposes, and was for many years of the A frame type. The introduction of the Diesel Engines into marine practice called for pressure lubrication throughout and consequently an enclosed engine. This led to the engine being designed with a trunk for all cylinders and separate cylinders on top of the trunk. With improvements in Foundry Practice it was found convenient to cast cylinders together thus giving a more stable and at the same time a lighter engine. This was followed by combination of the cylinder block with the trunk in one casting. It was also found that with this combination casting it was advantageous from the point of view of general rigidity to eliminate the through bolts which hitherto held the cylinders down to the bedplate. This could be done the more easily as in this size of Diesel Engine the consideration of foundry requirements called for a wall thickness which would easily carry the stresses from the cylinders to the bedplate.

The effects of this change have been to greatly increase the rigidity of the engine, and also to facilitate manufacture. The latter is effected by a simplification of machining operations, one setting-up serving for all cuts instead of necessitating separate settings for each casting.

Bedplate with Dynamo Baseplate.

Fig. 2. These two form a rigid structure wherein the weakest point, that is to say, the attachment flange in the middle, has been strengthened up by increasing the depth so that the whole forms one beam of as uniform strength as possible.

The bedplate itself carries strong seams of rows of bolts for the attachment of the cylinders. It should be noted that these bolts are very lowly stressed and the allowable stresses would necessitate only one-third of the number of bolts shown. The purpose of this exceedingly low stressing is, so to speak, to weld the two main castings together and to avoid any working of the joint between the bedplate and the cylinders.

The engines in actual service do not show any movement between these two main castings.

The bedplate in its main section is necessarily very deep so that the fly-wheel shall not protrude below the bottom machined face. The engine can be put on any level foundation provided with suitable bolts for fixing, and being rigid in itself is independent of special supports from the main transverse structure of the foundation.

Fig. 3. gives a view of the beam construction for the support of the main bearings. On each side of the main bearings are three rows of bolts, to each one of which corresponds a girder form.

The inner four bolts carry the stress flux through a rib cast round the bearing periphery. The middle four have a special hemispherical web and the outer seam carries the flux to the outer wall and through the webs shewn at the bottom to the bottom itself.

The main bearings are attached by means of bolts of ample dimension which have bolt heads underneath the second radial web.

Cylinders.

Fig. 4. illustrates the Cylinders as cast together in blocks not exceeding three for any given number of cylinders. A connection between the two blocks is carried nearly to the top of the cylinders so as to increase the depth of longitudinal beam.

The cylinder heads have each an individual expansion length and they are not cast together at the top. Therefore, should for any reason one cylinder be cut out for any length of time and so get cold, and its neighbours continue working at a correspondingly higher load, undue stresses will not be set up as a result of the different temperatures.

Brackets are cast on to the cylinders which support the camshaft bedplate; the same planed top surface which carries the cylinder heads forming a rigid support for the camshaft baseplates.

At the bottom of each water jacket is a cleaning door on each side of the cylinders, and in addition to this on the other side there is a cleaning door on top, which is not shewn in this illustration.

The inspection doors on each side of each cylinder are of ample dimension so as to ensure easy access to the main bearing bolts and the big end bolts, and also to the inside studs which bolt together the bedplate and cylinders.

The bottom flange for the attachment of the trunk to the bedplate has been elevated and is hollow. This gives the bottom flange additional strength and also additional stretching length for the studs referred to above.

At the right hand side is shewn the chain casing cast together with the last cylinder which entirely encloses the chain, a feature which is necessary on account of the chain running through an oil bath.

Connecting Rods.

In order to reduce the reciprocating and rotating masses to satisfy the conditions of balance and freedom from torsional vibration to which reference has been made, great care is taken to reduce the weights of such parts as connecting rods, and this is achieved by approaching the mathematically ideal contours so far as manufacturing methods will allow.

Fig. 5. illustrates the Rod itself which is made hollow and the lower half of the big end bearing is ribbed instead of full. For the latter, cast steel or stampings are found to give the best results.

Pistons and Liners.

The question of clearances is of primary importance.

It is in the interests of the customer to have these clearances cut down to the allowable minimum which ensures running without seizing up at overload. By the right choice of material and as a result of carefully thought out methods of manufacturing such as grinding of cylinders and honing of liners, the clearances have been brought down to what we believe to be very low figures.

The question of smooth, so to speak " steam engine-like " running is very much involved with these clearances. Clearances such as ¾ of a thousandth of an inch per inch diameter are about the minimum.

It is absolutely necessary to have liners and pistons round, both when the liner is fitted into position and when the piston is in the cylinder. If every care and precaution is taken by everybody concerned in this most important item in the manufacture of a Diesel Engine, then we arrive at what I referred to as " steam engine-like " running.

It will be seen from the picture Fig. 6 that the Piston is relieved on both sides where the gudgeon pin comes through, so as to allow for expansion when heated up. The gudgeon pin itself is made hollow which gives a considerable reduction in weight, and the boss which is formed round the gudgeon pin has a circular rib at the outside as to have a maximum strength with minimum of material.

The piston crown is hollowed out to a great extent, the advantage of which is to form a more compact combustion chamber than is possible with a more flat shape of piston crown.

Cylinder Heads.

The chief improvements in the design of the cylinder heads have been made with a view to reducing heat stresses and combinations of thermal and mechanical stresses. This has been done by shaping the wall against the combustion chamber so as to have no localisation of heat and an easy heat transmission. It should also be noted that the combustion wall is made thinner than the upper wall; the latter serves to support the combustion wall by means of the necessary channels for the valves.

Further changes in the design of cylinder heads have been made with a view to reducing the resistance to water flow and providing easy removal of sediment. This has been done by simpler coring of the cylinder heads thus providing open water passages, and the provision of large doors for sediment removal.

Fig. 7. shews two Cylinder Heads, the one on the left hand side taken from the side of the camshaft bedplate and the other from the inlet and exhaust pipes.

There are a number of valves which have to be inserted for the service of a 4-stroke engine, viz. inlet and exhaust, fuel, and starting air. Necessary provisions have to be made for indicating, and also a safety valve for excessive pressure in the combustion chamber. Some of the nuts giving access to the valves have been raised on longer bolts so as to facilitate dismantling.

It is in this casting especially where the benefit of the use of Perlit shows itself. This material can stand a large amount of combined mechanical and heat stresses and gives continuous service throughout the life of the engine.

Valves.

In order to obtain long service it was found advantageous to adhere to conservative figures in regard to air and gas speeds for inlet and exhaust valves. Therefore the cross sections of the valve passages were considerably increased. The material for exhaust valves is a point which has been under careful observation over a long period. Such material demands qualities which are sometimes mutually contradictory. These qualities are:

1. High tensile strength, because the sections of the valves must be kept low in order to avoid big inertia loads on the valve gear.
2. First class fatigue qualities against repeated impact. A broken valve may lead to great destruction through jamming between piston and cylinder head while the engine is still running.
3. High heat resisting qualities. The mean temperature under which the exhaust valve operates is high and it is necessary that in addition to resisting this high temperature, the tensile and fatigue properties of the material shall be maintained under the high temperature at which the exhaust valve works.
4. The material must be resistant against chemical action resulting from impurities in the fuel.
5. The Brinell hardness must be high so that small solid impurities do not indent the valve seating.

Fig. 8. shews a Fuel Valve with the inward opening valve. The conical valve seat ensures a good spreading of the fuel particles throughout the combustion chamber.

The ensuring of good mixing of air and fuel are the results of long experience with different grades, even the heaviest ones, of the fuels available on the marine market.

The Valve Levers illustrated in Fig. 9 are of cast steel and made in H. section so as to keep them as slight as possible, while the pins are hollowed out for the same purpose. The ends of the levers are detachable so that it is not necessary to take the fulcrum shaft away when one of the valves is dismantled for grinding its seat.

Camshaft Governor and Drive.

The camshaft has been subject to no important modifications, but instead of the worm drive the camshaft is now chain driven, giving greatly improved accessibility. The chain as an element for reliable sea-going machinery was at first regarded with very great suspicion by marine engineers generally. It has however proved its absolute reliability against many prejudices, and no example is known of a Marine Engineer who has had Diesel Engine machinery with chain drive under his charge, not to have been converted in favour of the chain. The chain which is forced lubricated runs from the camshaft to a lay shaft upon which the governor runs at a high speed, giving ample governor power and freedom from hunting. The governor in this arrangement is in the same chamber with the chain drive, and therefore benefits from its ample lubrication.

Camshaft Suspended above its Bedplate.

Fig. 10. The shaft with all its cams runs in a deep oil-tight trough which is combined with the one containing the governor shewn in Fig. 11.

Governor.

Fig. 11 shews a Governor on its lay shaft which is operated by the chain. The governor links run in ball bearings and the governor weights themselves are machined all over. This latter feature tends to keep the centre of gravity of the weights always at the same place on all governors and thus eliminates one of the troubles involved in the paralleling of different sets.

With all the governors on which the essential parts are carefully and uniformly made and with governor springs calibrated within narrow limits, the paralleling of quite a number of sets becomes an easy matter.

Compressors.

Air compressors for all sizes are now being made 3-stage. Fig. 12 shows a Compressor Piston and as a matter of detail it may be noted that the H.P. piston is made floating at the bottom so that it can find its own position.

Fuel Supply.

One of the main features of the Fuel Supply system is the fitting of individual pumps for each cylinder, the object of which is to ensure the delivery of fuel at the same crank position for each cylinder. This helps considerably towards equalising the indicator cards, and also enables us to choose the best position of fuel delivery to give the quickest reaction to differences of load and speed. A diagram of the Fuel Pump is shewn in Fig. 13.

Lubrication.

The natural tendency in the matter of lubrication has been to obtain an entirely forced lubricated engine. This has been achieved by supplying lubrication oil under pressure to all the moving parts taking load. An important development has been the enclosure of the camshaft gear by easily removed splash guards and the consequent efficient lubrication of this important mechanism. The return oil drains back into the camshaft bedplate and returns to the sump via the chain casing.

Extractor Fan.

Fig. 14 shows an Extractor Fan with two oil drains to recover the condensed oil in the pipe bend. The suction side of the fan is connected with the highest part of the oil chamber which is formed by the bedplate, cylinders and chain drive.

All the Diesel Auxiliaries in a ship might be coupled up to one fan and the vapour extractor therefore functions also for the sets which are shut down without detriment to the lubricating oil consumption.

It has been found that a simple arrangement is very efficient and if the pipe line from the extractor fan to the outlet on the top of the engine room hatch is long enough, then the whole of the oil vapour which is carried through will be condensed and flow backwards along the pipe walls to the oil drains where it can be recovered and led to the camshaft bedplate.

The speed of the air through the pipe must be such as to starve the oil vapour mixture in the crankcase below the inflammation range, and on the other hand must not be so violent as to carry the oil fumes away.

Materials.

Having now outlined the main features in the devolopment of the various parts of the Diesel Engine, I wish to refer to some of the changes which we owe entirely to the growth of metallurgical science.

In the very earliest days of the Diesel Engine it was realised that special grades of cast iron were necessary to withstand the high temperature and pressure conditions. Some valuable special mixtures were thus developed for the purpose. Practically all the ordinary castings included in a Diesel Engine are now of a special homogeneous grade of cast iron. Further than this, we now have " Perlit," which has taken what appears to be a permanent place in Diesel Engine construction, and the most vulnerable parts of the engine, which are subject to combined thermal and mechanical stresses, are now constructed of this material.

Manufacturing Methods.

Changes in design and materials have naturally brought with them changes in manufacturing methods. I have already referred to the simplification of machining operations by the adoption of the composite trunk and cylinder design.

Such machines as the Plano-Miller can be utilised with a very high degree of usefulness on parts such as this. The multiple operations previously conducted on planing machines with a large number of settings being reduced to a few operations with a single setting.

To secure perfection of liner shape and surface has also involed the introduction of special machine tools including Liner Borers and Honing Machines.

In connection with the latter a certain advance with regard to improving the liner surface seems possible by honing the bore by means of special machinery. This practice has also the advantage of enabling us to work to closer clearances and ensures the perfect interchangeability of spare parts.

Dynamos for Diesel Engine Drive.

Having now dealt with the prime mover it remains to deal with the electrical end of the auxiliary Diesel generator. In common with other electrical machines the requirements of these dynamos have the necessity for first class electrical properties, i.e. sparkless running from no load up to at least the maximum capacity of the engines, ample margin in volts and good inherent voltage regulation. By this it is meant that the machines should be so compounded to keep the volts as nearly as possible steady with the varying load and with the engine speed as controlled by its governor. They must also have ample capacity in order that the temperature rise of no part shall be excessive for the duties they are called upon to perform in the positions in which they are placed.

For vessels trading through tropical waters, it is desirable that the temperature rise shall not exceed say 63 degrees Fahr. after 6 hours on full load, but for vessels working in cooler climates a temperature rise up to 72 degrees Fahr. is permissible. It is particularly important to stress the limitations of temperature rise when considering the shunt field windings, as these are always working on full load, whereas the load in the armature and series windings of the machines varies according to the demand.

In considering the temperature rise of a machine, it is desirable that not only should the surface temperature be moderate, but the internal temperature should also be within safe limits, and for this reason field coils mounted on the ventilated system whereby they are subjected to a cooling draught inside and outside are strongly to be preferred.

In general the ventilation of a machine should be as good as possible to prevent the formation of hot pockets due to stagnant air, but in machines of the usual open type the fan effect of the armature windings is sufficient to provide ventilation without the use of additional fans.

Apart from these thermal and electrical properties special consideration must be given to mechanical construction and accessibility.

Mechanical Construction of Dynamos.

In order to cope with any irregularities of turning moment, sudden applications and removal of load, the armature construction should be of specially robust type, and for this reason it is recommended that the armature spider shall be of cast steel of ample section formed at the driving end into half coupling for direct connection to flywheel, the arms of the spider being extended the full length up to the face of the coupling. A steel shaft is pressed into the spider and formed into the bearing at the outboard end.

The core plates are mounted on the armature spider and driven from it by means of a suitable number of keys of specially large surface area designed to give minimum practicable pressure per square inch between the keys and the core plates. The core plates are a tight press fit on all the keys and are clamped together by means of endplates of very heavy construction.

The windings which are first insulated by Micanite of special grade moulded upon them are pressed into the slots and retained in place by wedges and bands with a large factor of safety to withstand the stresses to which they are subjected.

The commutator construction follows the usual lines, the segments being insulated by a special grade of Mica or Micanite and held between the "V" rings, the pressure on the segments being given by a large number of bolts. The whole commutator structure is mounted directly on to the cast steel spider positively driven from it by means of keys. The connections from the commutator to the armature windings are made by copper strips which are riveted to the segments and are of sufficient size or are independently damped to prevent any dangerous vibrations occurring in them. A special point to be observed in this construction is that the armature is built up of a number of parts in such a way as to virtually form a solid mass which is directly coupled to the flywheel.

The bearing is preferably of the forced lubricated type, lubrication being supplied from the Diesel engine lubricating system through pipes of sufficiently large size to ensure that there shall be no blockage and interruption to the bearing lubrication. Ample capacity in journal area and wide base to the pedestals are the two essential features to provide in this bearing design.

The brush gear cannot receive too much consideration. It should preferably be carried from substantial brackets on the magnet yoke and comprise a cast iron ring from which the brush arms are themselves insulated and carried, the brush boxes being at the commutator end of these brush arms and of such a type as to permit of adjustment of pressure on the brushes themselves. Also movement of the brush boxes as a whole to follow up any possible wear of the commutators and provision for staggering to ensure that the commutator surface is swept as near as is practicable by an equal number of brushes should be allowed for. It is desirable that the brush arms if removed from their carrier should be replaced in exactly the same position as that from which they were taken and for this reason it is necessary that a registering device be fitted to each brush arm to ensure that it is always fitted in its proper place. Although the brush gear as a whole should be rotatable by means of a worm and nut, this worm should not be fitted with a permanent handle because once the brushes have been set there should be no necessity to move them again.

Accessibility.

Accessibility of the generator is of great importance and for this reason protecting guards are necessary to ensure that falling moisture does not get on to the windings. The mere fact of the machine being open is an inducement for it to be kept clean, one of the most important considerations to bear in mind in regard to generators in any positions and particularly in situations where there is liable to be a certain amount of oily deposit. With the machine well open there is less tendency for it to be allowed to get in a dirty condition than with a machine that is more closed up. In designing these machines facilities for cleaning them should receive special consideration and for this purpose long creepage surfaces of insulation should be provided between the current carrying parts and earthed metal and these surfaces should be smooth to prevent the adhension of oil and dirt and to permit of them being wiped down with the greatest possible ease.

With machines constructed generally on the lines mentioned, provided due attention is given to cleanliness, maintenance costs are very low.

The only accessory which may be considered as a definite part of the dynamo and to which attention need be called is the shunt regulator. Failure of this comparatively insignificant piece of apparatus is sufficient to put the whole set out of commission and therefore great care should be taken to ensure that the shunt regulator supplied with these generators is constructed on equally robust lines to the main generator, the mechanical properties being considered as much as the electrical properties. The use of thin slate bases, unsupported fine wire resistances, hygroscopic insulating materials, soldered joints and other features of kindred type should be rigidly tabooed. The resistance wire should be insulated from earth by Micanite, should be supported on metal tubes or frames and all connections should eb bolted. Contacts and contact brushes should be of large size, being designed from the point of view of mechanical considerations rather than for the electrical properties. In fact with generators etc. for Diesel engine drive the mechanical properties of the apparatus should receive prior consideration to the electrical properties, as if the latter are not up to standard the failure is easily observed on the initial test, whereas the mechanical shortcomings would only be disclosed after a period of running and possibly at a time when the effects of the breakdown may be very serious.

Conclusion.

In dealing with this subject only the main points relating to the design, construction and running of one type of Diesel Engine have been described but doubtless these brief notes will serve as a basis for discussion, which will in turn possibly lead to further research and improvements.

Fig. 19. shows a Complete Section of a 6-Cylinder Auxiliary Engine and Generator embodying all the points which have been previously referred to, while Figs. 15, 16, 17 and 18 illustrate typical examples of completed engines and generators.

Fig. 17. shows a Two Cylinder 100 H.P. Engine, Fig. 16 a Three Cylinder 200 H.P. Engine, Fig. 18 a Six Cylinder 450 H.P. Engine and Fig. 15 a Six Cylinder 700 H.P. Engine. Fig. 22 illustrates a set of Curves taken on the official trials of a 6-Cylinder 450 K.W. set running at 250 r.p.m. while Fig. 21 is a graph showing the calculated stresses in the crankshaft due to Torsional vibrations compared with the results given by measurement with the Geiger Torsiograph.

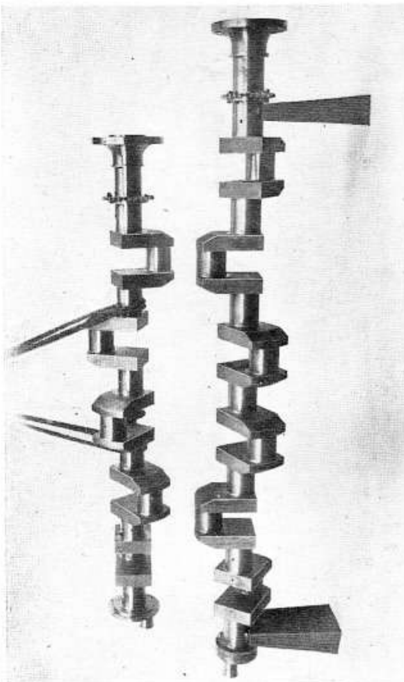


Fig. 1

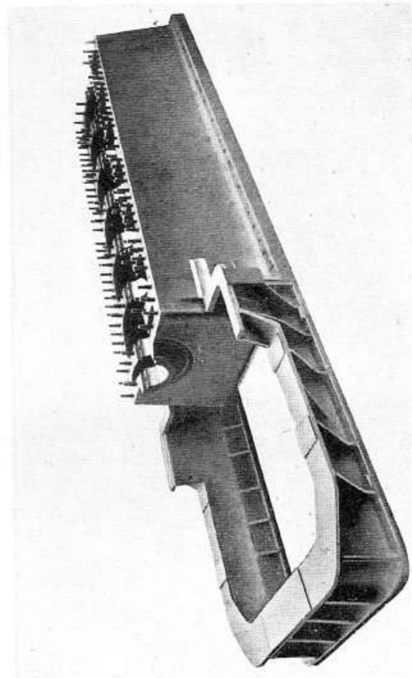


Fig. 2

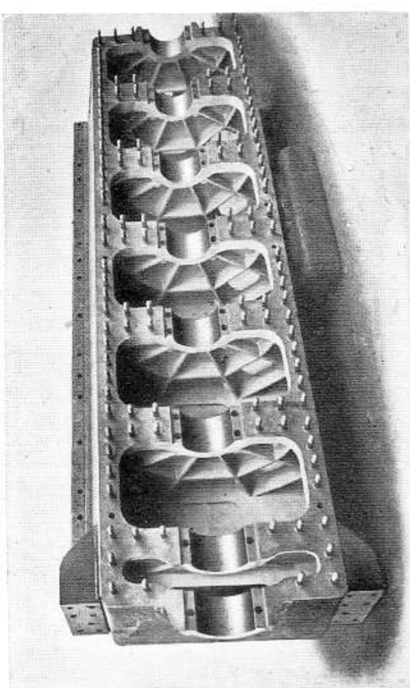


Fig. 3

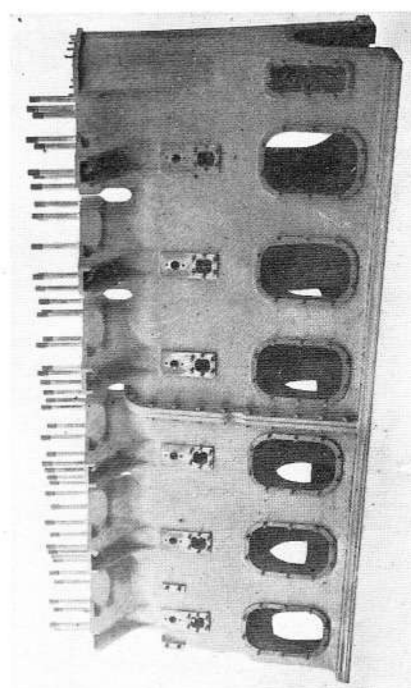


Fig. 4

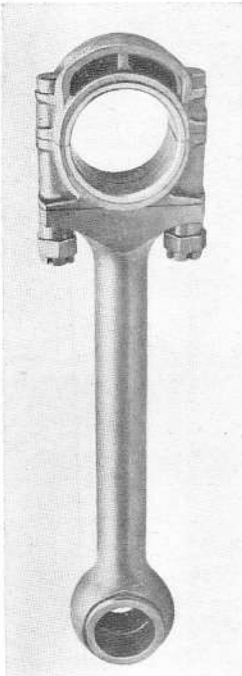


Fig. 5

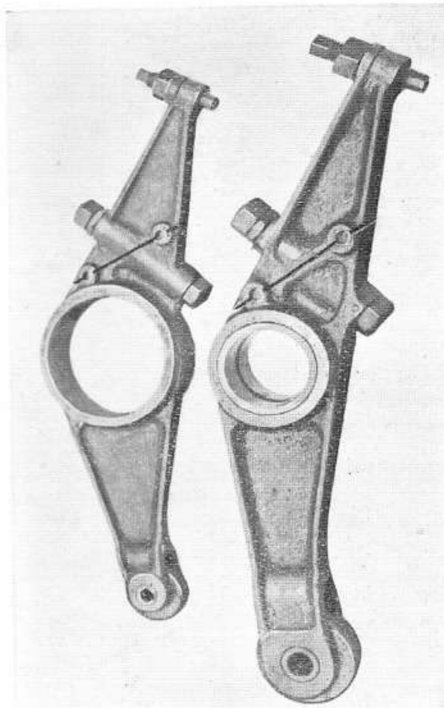


Fig. 9

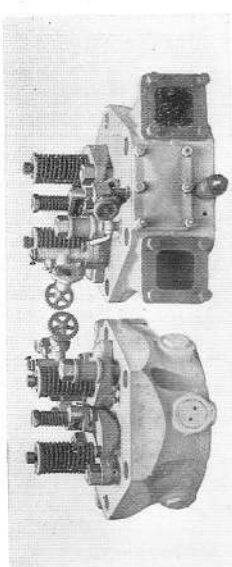


Fig. 7

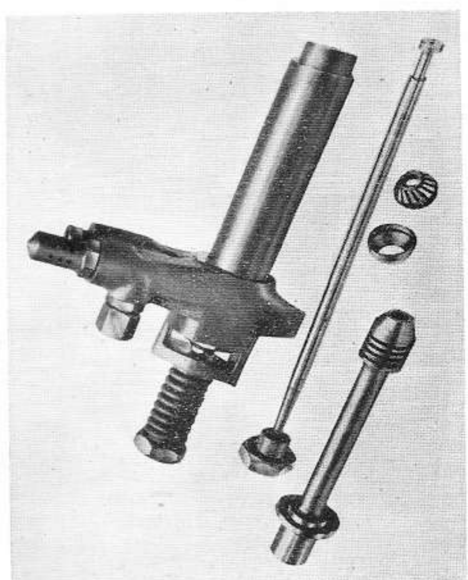


Fig. 18

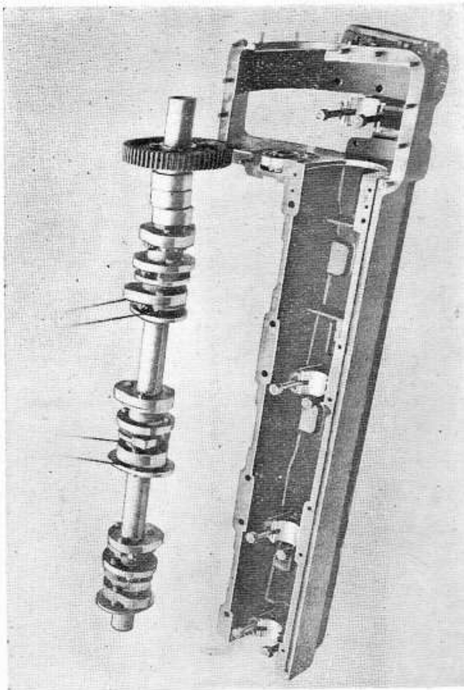


Fig. 10

Fig. 6

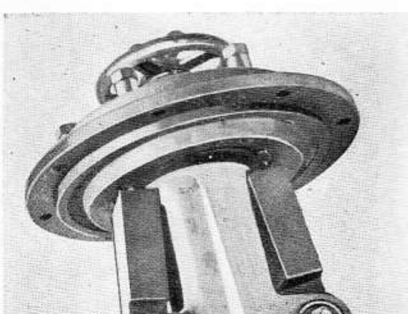


Fig. 11

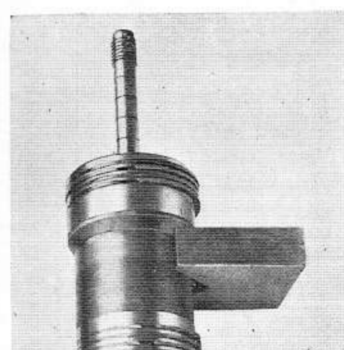


Fig. 12

(Paper No. 255)

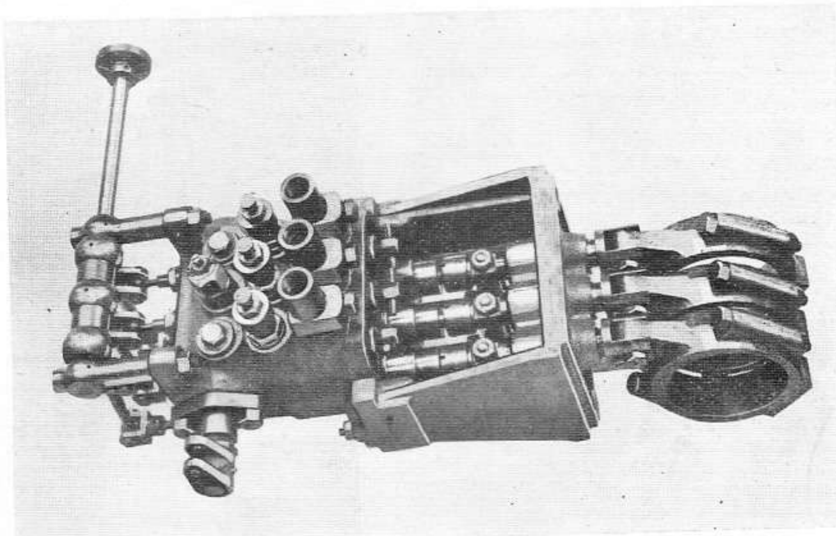


Fig. 13

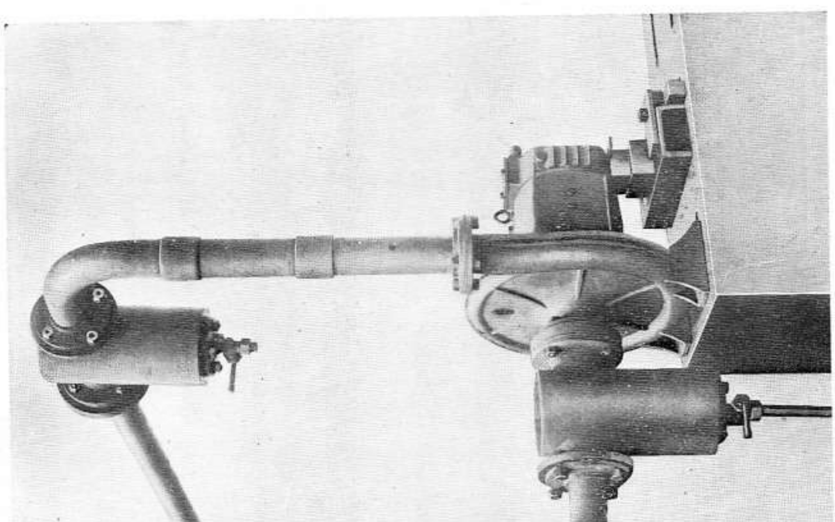


Fig. 14

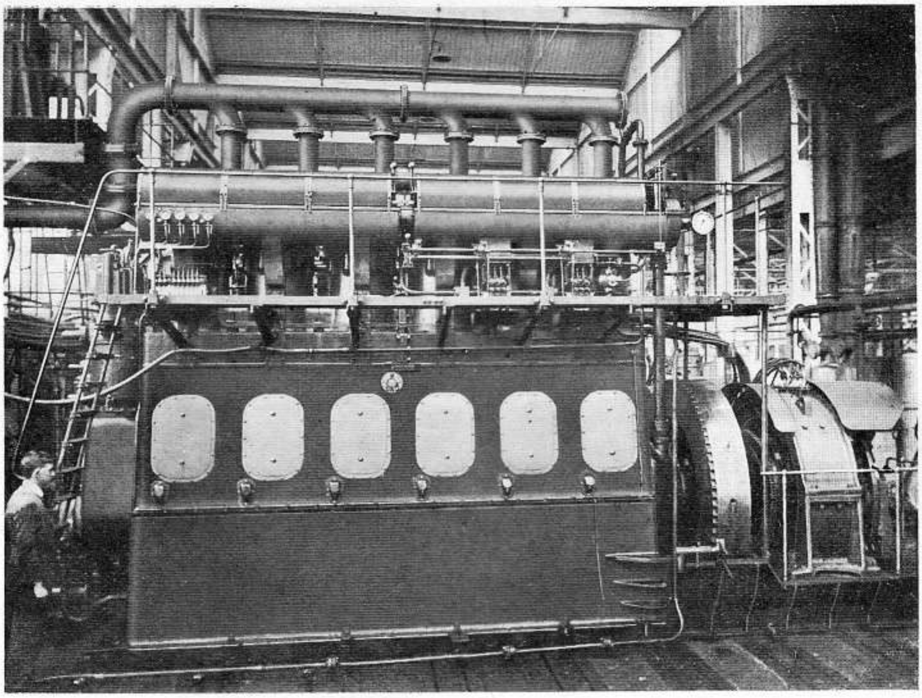


Fig. 15

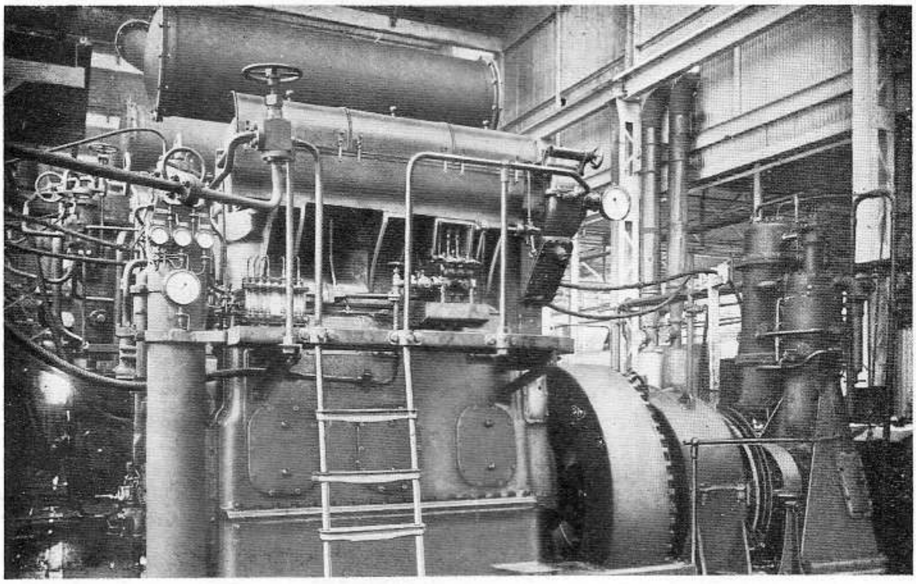


Fig. 16

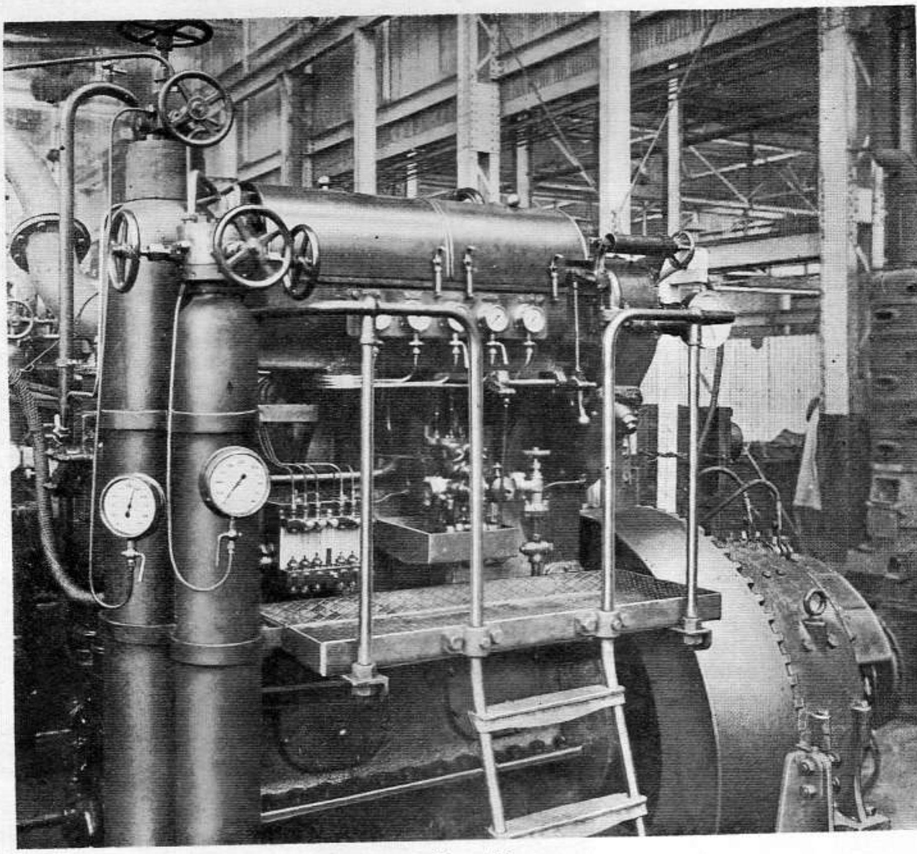


Fig. 17

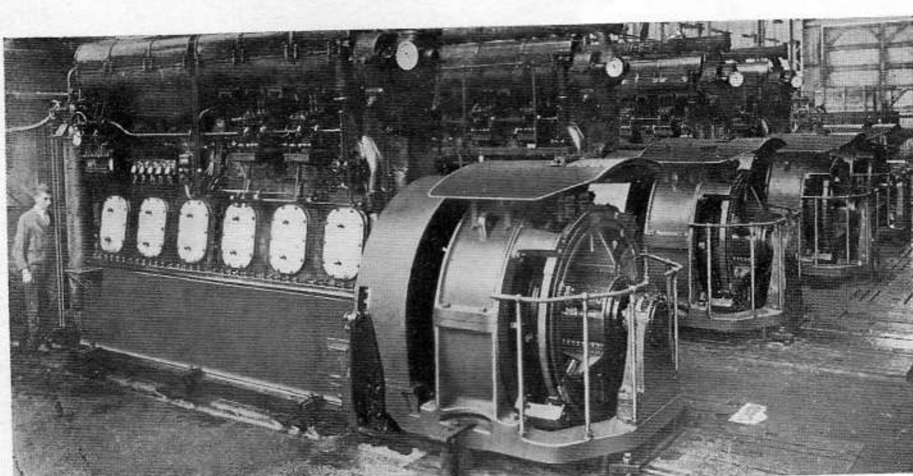
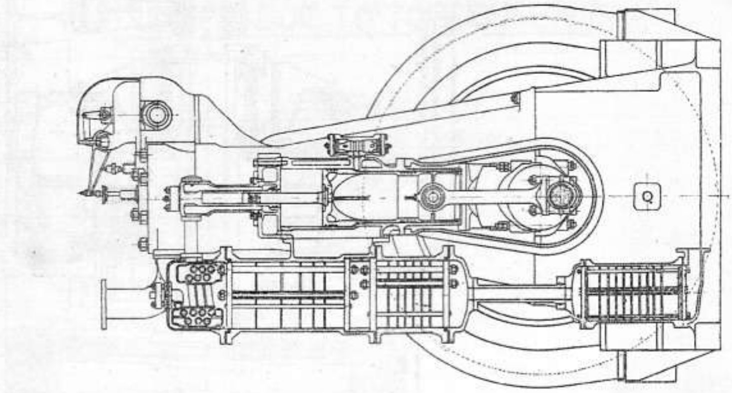
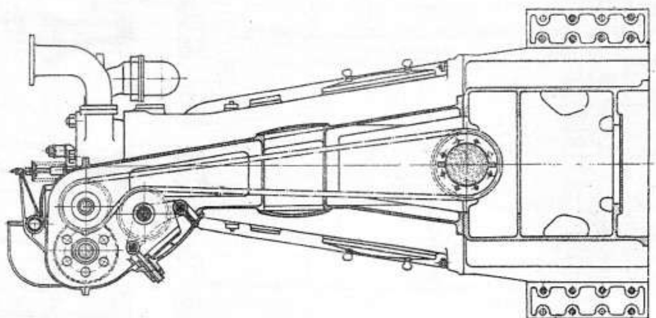


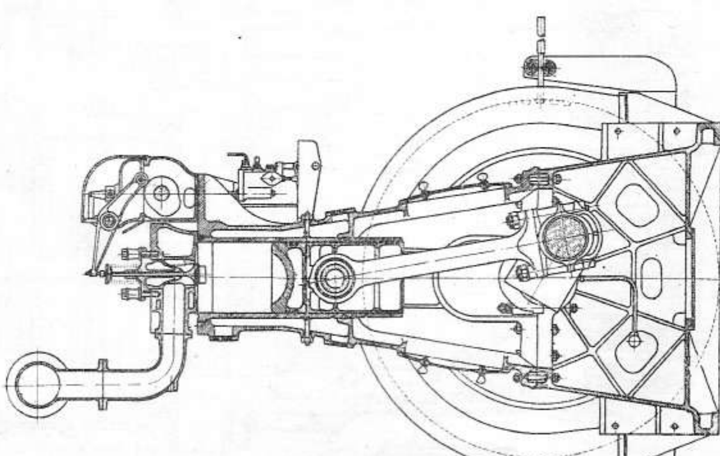
Fig. 18





Section thro' B.B

Fig. 19



Section thro' A.A

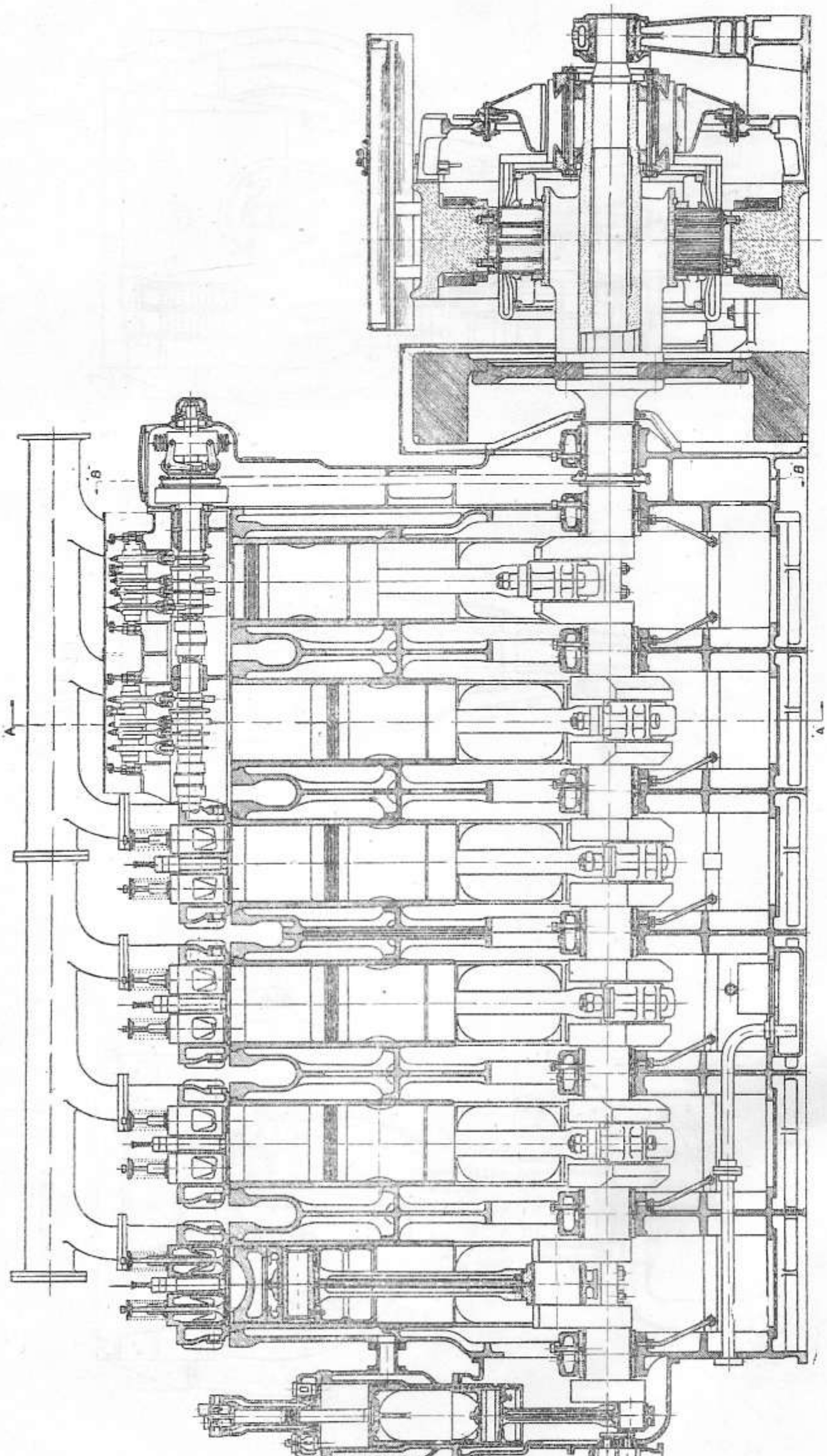


Fig. 20

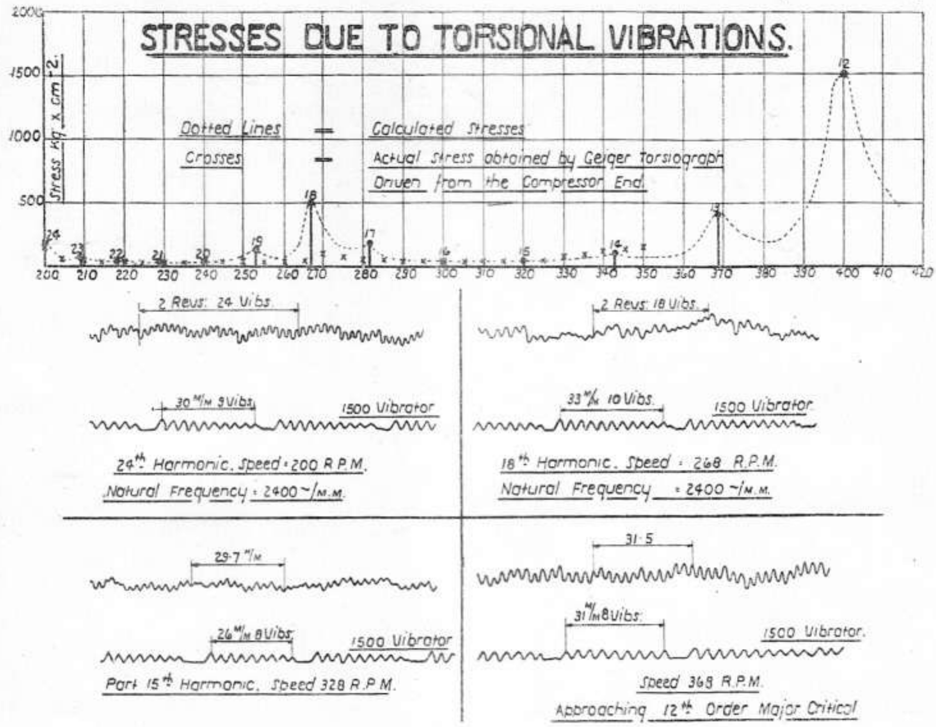

STRESSES DUE TO TORSIONAL VIBRATIONS.
Stress Kg x cm-2.
Dotted Lines = Calculated Stresses
Crosses = Actual Stress obtained by Geiger Torsiograph
Driven from the Compressor End.
2 Revs: 24 Vibs.
30 M/M 9 Vibs.
1500 Vibrator
24th Harmonic. Speed = 200 R.P.M.
Natural Frequency = 2400 ~/M.M.
2 Revs: 18 Vibs.
33 M/M 10 Vibs.
1500 Vibrator
18th Harmonic. Speed = 268 R.P.M.
Natural Frequency = 2400 ~/M.M.
29·7 M/M
26 M/M 8 Vibs.
1500 Vibrator
Part 15th Harmonic, Speed 328 R.P.M.
31·5
31 M/M 8 Vibs.
1500 Vibrator.
Speed 368 R.P.M.
Approaching 12th Order Major Critical

Fig. 21

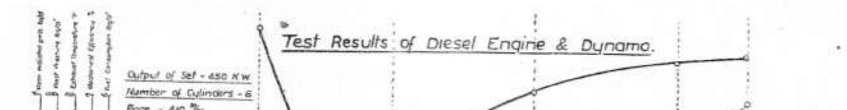

Test Results of Diesel Engine & Dynamo.
Output of Set - 450 KW
Number of Cylinders - 6
Bore - 410 M/M
Stroke - 600 M/M
R.P.M. - 250
Mechanical Efficiency
Exhaust Temperature
Blast Air Pressure
Mean Indicated Pressure
Output of Set in KW
Fuel Consumption in lbs per B.H.P.
Dynamo Efficiency
Temperature of Piston Cooling Oil
Fuel Consumption in lbs per I.H.P.
Output in Brake H.P.
One Quarter Load 173.7 B.H.P.
Half Load 330.5 B.H.P.
Three Quarter Load
Full Load
Overload

Fig. 22

The Outlook for the Steam Engine Afloat.

(Paper No. 256)

By Sir John H. Biles, K. C. I. E., LL. D., D. Sc.

In responding to the invitation to offer a paper to be read before this important gathering, which I am unfortunately unable to attend in person, I am led to review the three papers which I read before the Institution of Naval Architects in the years 1925, 1926, 1928. Their respective titles were:—

(1) Relative commercial efficiency of internal combustion and steam engines, for high speed passenger vessels.

(2) Relative commercial efficiency of steam turbine and Diesel machinery for cargo vessels.

(3) The present position of the question of fuel for ships.

The upshot of these three papers was to call attention to the tendency, not uncommon in the world's history, of jumping to the conclusion that some new invention in a particular line had made all former practice obsolete. Later events have often proved this conclusion to be wrong, and in fact the stimulus of discovery often points the way to improvements in old methods; and generally, that there is a place in the world for both new and old methods, and "Life in the old dog yet".

The "old dog" in this case is "steam propulsion for ships", and the fashionable trend towards the Diesel engine, implying exclusive use of oil, has been a blow to the coal industry, especially so in Great Britain, where the export of coal enables vessels otherwise sailing in ballast, to carry freight on their outward journey, and it has had accumulative effect on the cost of all goods and materials which we import, and consequently upon the cost of living and production costs.

This rush to Diesel machinery was aided by troubles in the mining industry, with uncertainties of coal prices and deliveries.

In the valuable and exhaustive paper read by Mr. John Johnson at the Institution of Naval Architects in March 1929, we have evidence that this rush has spent itself, and that we shall in future see the steam engine more firmly established than ever, as the means of propulsion of nearly all ships of large powers and of an increasing proportion of those of smaller powers.

The retention of the steam engine does not necessarily involve the use of coal, but with every new Diesel engine ship put into service, a possible coal consumer is definitely written off. Dr. Diesel's invention arose from efforts to burn coal in a powdered form in a cylinder, but though attempts are still being made to achieve this, there is little likelihood of its becoming practicable in the near future.

At this international congress I may appear to be stressing a point that is primarily of British, and therefore local interest. But it is in effect of world-wide interest, though Great Britain is the first to experience the effect. To the best of our knowledge by far the largest quantity of stored energy on this planet exists in the form of coal, and the oil supplies, despite widespread discoveries, are almost certainly very much less in quantity.

When we become sufficiently conscious of our own interests in conserving the stores of energy provided in the earth there will probably be restrictions placed on the uses which may be made of the available oil supplies, while it may be generations before similar steps are rendered necessary for the conservation of the coal measure.

I quoted in 1925 a statement of Sir Fortescue Flannery:—" He believed that the use of Diesel machinery for navigation would become so common that only for special purposes would the steam engine be able to hold its place." Lord Bearsted, the Chairman at the meeting of the Royal Society of Arts, at which Sir Fortescue's paper was read, delivered the funeral oration, and Sir Robert Dixon, while excusing the Admiralty for continuing the use of a supposed dead instrument, pointed out that for some relatively unimportant purposes, such as obtaining the highest speed in warships there was still a flicker of life in the steam engine.

Commenting on this I remarked. " It is premature to talk of the Diesel completely superseding the steam engine. The upper limit of power of the Diesel is much below the upper limit of the steam engine, so that at present there can be no supersession of steam machinery above the Diesel's upper limit of power. High speed warships of all types cannot adopt the Diesel engine for this reason. Oil, which has made the Diesel a possibility, has much increased the output and the economy of the steam engine. The advantage of the Diesel over the steam engine is that the oil has been more efficiently used in the cylinders of the Diesel than in the furnaces of the boilers which supply the steam for the older type of machinery, and the steam so produced has not had the same range of temperatures to work in that the oil in the Diesel has had. But developments are taking place in boilers and steam turbines which promise to remove some of these differences in economic output."

It seems probable too that for some time to come oil fuel for the steam engine will be cheaper than for the Diesel.

An analysis followed of costs, weight, and fuel consumption, of Diesel, steam, and high pressure turbine installations respectively, which the passage of four years has not greatly altered.

At that time the excess cost of a Diesel installation of 5,200 S.H.P. over an H.P.T. installation was taken as £35,000. In a similar comparison in my paper in 1928, the excess was £21,500 for 3600 S.H.P. machinery, while Mr. Johnson in 1929 gives £30,000 as the difference on 7000 S.H.P. machinery. He also stated that at the time the contract for the Duchess of Bedford 16,000 S.H.P. was placed quotations showed that the cost of a motor-ship would have been £100,000 more than the steam ship. Hence we may say that the difference of cost to be faced is certainly not less than £4 per S.H.P.

It is evident therefore that the relative first cost is not coming down at a rate which is likely to level costs for many years, and Mr. Johnson's figures, which were not contested, may be taken as based on information derived from competitive tenders, as were those given by me.

There is a point of view as to weight which has not been much emphasized when comparing Diesel and steam installations. However light the unit Diesel engine may be its weight must increase directly with the increase of horsepower, that is, the power weight curve is a straight line. In fact there must be a slight additional weight to cover extra shafting or electric cable, required to combine several engines in a single duty.

On the other hand with the increasing powers in high pressure steam ships the weight per S.H.P. continually decreases. For example, in my paper last year I gave the figure of .2 for a small high pressure steam installation of 3600 horsepower. The steam installation of the Duchess of Bedford working at 350 lbs. steam pressure weighs .126 tons per S.H.P.

When full advantage is taken of the high pressures now practicable it seems certain that this figure of .126 will be reduced considerably further. At least one engine builder will give 55,000 S.H.P. 4000 tons or .037 tons per S.H.P.

H.M.S. Hood with a horsepower of 144,000 and only moderate steam pressure is shown in the paper by Sir Eustace Tennyson d'Eyncourt at the Institution of Naval Architects, in 1920, to have machinery weighing only 5376 tons or .037 tons per S.H.P. This of course is a figure which is only practicable under naval conditions, and if two-thirds of her power is taken as that required to produce the service speed it may be considered as equivalent to .055 tons per S.H.P. in merchant service practice.

At to fuel consumption, in 1925 .7 pounds of oil per S.H.P. was held out as the practicable figure for machinery with H.P. steam.

Mr. Johnson in 1929 gives the actual service consumption in a vessel of 1.03 lbs. for coal, or .63 lbs. for oil, and this in a ship with only 350 lbs. pressure in boilers. We therefore can expect a further substantial reduction with higher steam pressure.

In a paper the following year –1926– an analysis was made of the uses of the two types of machinery in cargo vessels, in which I endeavoured to show the length of voyage at which " dollar " efficiency becomes favourable to the Diesel engine. The data on which the tables in this paper are based are no longer correct in the light of present day prices of fuel, which are still more in favour of the coal-fired steam ship.

In that paper coal prices were taken from 23/– to 33/– a ton, boiler oil 65/– to 34/– and Diesel oil 80/– to 34/–.

On April 10th this year corresponding figures are 15/– to 33/– for coal, 67/6 to 35/– for boiler oil, and 82/6 to 35/– for Diesel oil.

On the question of pure water, I cannot do better than again quote Mr. Johnson, who says:—" Having regard to the fact that water-tube boilers are used exclusively in the British and other navies, and in all modern power stations throughout the world, at pressures ranging from 250 lbs. to 1,300, it cannot be seriously maintained that there is to-day any insuperable obstacle to their more general adoption in mercantile vessels. It is, however, recognised by all engineers experienced in the best water-tube boiler practice that de-aerated pure feed is essential for success, more particularly with high pressure. The positive exclusion of oil and scaleforming matter should be regarded as an indispensable condition if thoroughly satisfactory boiler performance is to be secured, and on this—perhaps more than on any other factor—does the success of marine high pressure steam practice depend. It must not be supposed that this is a counsel of perfection, for it can be attained with a modicum of prevision and operating care. But apart from questions of safety and efficiency, scale in boilers has always been a nuisance, and its elimination is greatly to be desired on the score of cleaning expenses. Leaving aside the time required in ports, the cost of cleaning boilers in a large fleet may reach a formidable annual total, and constitutes one of the disabilities under which steam plant has hitherto laboured. Nothing less, of course, than the best grade of condenser tube procurable should be employed, and practical measures taken to minimize risk of leakage at the tube plates."

In Mr. Johnson's ships, Scotch boilers have been fitted through which all water taken from the shore or distilled from the sea must pass before reaching the water-tube boilers. This is undoubtedly a great safeguard and a concession to prejudice, but an extra first cost and an extra complication.

I am strongly of the opinion, however, that the further step of the abolition of the Scotch boiler is the true line of progress. Evaporators outside the Navy have always been supplied of very meagre capacities, and have only been able when perfectly clean to produce water in quantities that may be required. Forcing is therefore inevitable and the fact that they are seldom kept in continuous use has led to priming and established the prejudice that undoubtedly exists against them in the merchant service. Given low pressure evaporators of large capacity there is no reason why the carrying of large quantities of fresh water should not be dispensed with as it has been so for many years in all war ships. The associated step of driving all possible small auxiliaries electrically from a Turbo or a Diesel generator is the other factor which counts most in maintaining a pure supply.

I notice that an evaporator of 60 tons capacity provided in the Duchess of Bedford, presumably is only for use in an emergency, which could be doubled at the cost of £600, thus providing a distilled water capacity of 120 tons daily. Taking Mr. Johnson's figure as 35 tons make up feed for the 16,000 horsepower of the Duchess of Bedford this would amply cover the 50 tons required daily for a 20,000 horsepower ship.

In table VII page 7 of Mr. Johnson's paper he shows a Scotch boiler costing £3,000 (probably not including pumps and accessories) producing 11,000 lbs. of steam. This 11,000 lbs. of steam,—an increase of 6½% on the output of the water-tube boilers—could be produced without proportionate increase of cost of these boilers. Suppose it proportionate, it would amount to £2568 making with the £ 600 for additional evaporators a total of £ 3168. The true figure, however, is more likely £2000. As a result of this substitution there would be an economy of stowage of fresh water of 50 tons for each day of the voyage; a saving of the space and weight of the Scotch boiler and its accessories; no increase in personnel; a saving in the cost of shore water; and a condition of entire independence of the shore so long as the fuel lasted.

My paper of 1928 took as a basis the relative values translated into commercial efficiency of available fuels, and the methods of obtaining the energy of propulsion from them. The figures on which they were based again showed that by the use of coal as fuel a very definite " dollar " economy is possible.

Objection was taken in the discussion to the rate—18½%—taken for charges, and a rate varying with the type of machinery proposed. Since, however, this—18½%—includes depreciation, insurance, and interest on capital, which must be the same for all types, I do not agree that such variation is justified. Mr. Johnson, who has a close knowledge of the commercial situation, also takes a uniform figure—15%—, for interest and depreciation. He claims " that in an 18,000 S.H.P. installation the annual charge on the extra first cost of the motor-ship, viz £15,000, is sufficient to defray the entire cost of repairs (running and periodical) in the steam installation, with £8,000 to spare. As the fuel costs of the steam plant in the two services quoted are not greater than the Diesel, it is not even necessary to inquire what the cost of repairs is in the latter. It may remain as a surplus of indeterminate magnitude. As it has been suggested, however, that there are some ports where the difference between the price of Diesel and Boiler oil is small—say as little as 3s. per ton—it may be of value to examine the 18,000 S.H.P. case on the basis that the difference is nil. The nearest approach to this appears to be San Francisco, where the market rates for Diesel and Boiler oil are 33s. and 30s. respectively, and further that if the economics of high-pressure steamships and motor-ships of this class be fully explored, it will be found that in the majority of cases the advantage lies with the former, this condition holding when the steamship is using oil as fuel, the extra cost of boiler oil per annum only being slightly in excess of the annual charge on the higher cost of the motor-ship. It is evident, therefore, that if the steam-ship is also able to use coal in zones where its price is attractive a strong superiority obtains with that type of installation."

The only point brought out in the discussion upon his paper which is adverse to the steam engine is that there is certain falling off in the vacuum, and consequently the efficiency, where the temperature of the sea water is high. Looking back, however, I feel able to claim that the point of view taken in 1925 and since maintained, has been justified by experience and even more firmly established. New factors have come clearly to the front, for example reliable material for condenser tubes, and a great advance in the technique of the use of powdered coal afloat. In this advance, notwithstanding that it is the greatest oil-producing country, the United States of America has done great pioneer work, and progress is now rapid. Since the use of powdered coal promises to reduce costs of labour and of fuel even in ships with triple expansion engines and cylindrical boilers, it will do much to retain the use of coal on the seas. With the extension of its use we may yet see steam ships capable of burning either coal or oil, as commercial considerations may direct, as burners for powdered coal can be made so as to be easily adapted for burning oil.

Mr. Johnson also brings to notice the fact that for powers above 20,000 S.H.P. water-tube boilers are now actually less costly than those of the familiar Scotch type. The adaptation of the water-tube boiler to the use of pulverised fuel has been sufficiently developed to ensure the use of this fuel in high-pressure boilers. In Scotch boilers the use of pulverised fuel has passed the experimental stage and is accompanied by a gain of 15% over hand firing. The gain in a water-tube boiler will not be less. In the King George V which has been built for the Parsons Company, to prove the advantage of H.P. steam of 500 lbs. pressure, hand-fired water-tube boilers have produced one S.H.P. for 1.08 lbs. of coal of 13,500 B.T.Us. With pulverised fuel installation this machinery should get an S.H.P. for .93 lbs. of coal. With oil the figure for fuel consumption will be .62 lbs. This installation is a low-powered one of only 3500 S.H.P.

The efficiency of a larger power will be greater and the figure for fuel consumption may well be still lower.

There appears every indication that the present relation between the prices of coal and oil is likely to continue for a long time to come. In the light of advance in powdered coal methods one can safely reduce the figure given by me for daily consumption, in column seven of table one, to 36 tons. This gives a further saving of 100 tons in a 200 days steaming, and brings the total savings per year of the high pressure, water tube, powdered coal installation over the Diesel to £6795. Appended will be found a repetition of a part of this table, with a new column 7a. The prices of coal and oil have been retained at 20/- and 80/-, for it is understood that the actual costs in different lines and services are never identical.

Extract from Table I. of Sir John Biles' Paper—1928.

Table showing savings in running costs of various types of machinery of 3,600 S.H.P. on continuous sea service if they are used instead of the Ordinary Triple-Expansion Engine and Low-Pressure Cylindrical Boiler.

	1. Ordinary Triple Steam	7. Turbine with high Pressure Water-tube Boilers & Pulverizers	7a. 1927 as 7.	8. Oil Engines
Boiler pressure, lbs. per sq. in.	200	525	525	—
Temperature, steam.	390°F.	750°F.	750°F.	—
Cost of machinery.	£36,000	£68,000	£68,000	£85,000
Consumption, tons per day.	62	36½	36	17½
Pounds per S.H.P. per hour.	1.6	0.95	.93	0.45
Weight of machinery, tons.	850	770	825	825
Diff. in weight in tons carried due to diff. in fuel carried for 200 days.	Zero	5,100	5,200	8,900
Diff. in weight in tons carried due to diff. in weight of machinery for ten voyages per year.	Zero	800	800	250
Diff. in cargo earnings at £1 per ton per voyage for ten voyages per year.	Zero	£5,900	£6,000	£9,150
Savings in fuel costs in 200 days at £1 per day.	Zero	£5,100	£5,200	−£1,600
Cargo, net earnings in excess of triple steam Column 1.	Zero	£11,000	£11,200	£7,550
Charges, 18½% of cost of machinery in excess of triple steam Column 1.	Zero	£5,920	£5,920	£5,065
Net savings per year over triple engines.	Zero	£5,080	£5,280	−£1,515

Oil is taken at 80s. per ton.

Ship Experiments and Theories.

(Paper No. 257)

By J. Foster King, C. B. E., Vice-President of the Institution of Naval Architects, London.

Throughout unnumbered centuries of wood shipbuilding, construction and scantlings were the pure product of skill and experience unalloyed with mathematical theories. Within the past century, at the beginning of the iron era, ships of colossal dimensions in relation to all past experience, were successfully built without the aid of precedent but with that of imagination, of practical genius and of theories of applied mechanics which were then new. These achievements seem to have been forgotten or ignored when iron and steel became a common material for ordinary merchant ships. Construction under classification rules then reverted to a curious condition under which iron ships became imitations of wood ships, innovations in structural design and arrangement of material were stoutly resisted, scantlings were not considered in relation to theories of mechanics, and structural efficiency was again the product of practice or full scale experiment.

In 1893 the British Corporation Register published construction rules in which analysed experience was associated with ordinary theories and fixed standards of stress. The value of this advance in method was immediately demonstrated by experience, and although it took years to reach that stage, it may safely by said that modern shipbuilding is now governed by experiment guided by theory.

Commercial success in the structural design of ships is conditional upon the use of the smallest possible quantity of material to withstand unknown stresses which are induced by unknown loads and affected by unknown forces. Structural success is, therefor, limited in its ambit to success under some average of conditions and the Societies can justly claim that their methods have steadily improved the commercial and structural efficiency of classed ships under increasingly severe conditions. Advance is conditional upon close observation of the experiments in hull structure, which are made in almost every ship that is built, and upon close analysis of the bearing of observed facts upon working theories. The "facts" themselves may be no more than probabilities to be added to the assumptions with which we pursue knowledge, but every fact is worth its proverbial ton weight in the adjustment of theory to practice, and every observed experiment which helps to explain the relation between known load and known stresses is of inestimable value.

For these reasons, and because of their intrinsic value, I asked Mr. Lawrence Holt for permission to publish particulars of certain experiments made by his firm, in connection with the first ship built of a mild steel of higher qualities than ordinary ship steel. He was good enough to allow me free use of the information at my disposal and thus enable me to record the biggest thing of the sort ever done by private enterprise.

These experiments were carried out some three years ago in connection with the sister ships "EURYMEDON" and "PROMETHEUS" of the Blue Funnel Line. The first was built of ordinary 28/32 ton mild steel to the standard scantlings adopted by Messrs. Holt for their 430 ft. ships; the second was built of 31/35 ton mild steel having an elastic limit (or limit of proportionality) not less than 15 tons per square inch, and made to the specification of Mr. Martin, the firm's metallurgist. The "EURYMEDON" was built with transverse framing throughout; the "PROMETHEUS" had the bridge deck plating supported by longitudinal beams carried on transverse girders, with the object of providing effective resistance to compression with lighter deck plating than in the "EURYMEDON". The saving in weight of steel in the "PROMETHEUS" due to the use of the superior steel, is estimated to have been not less than 250 tons.

The qualities of the new steel were most carefully compared with those of ordinary ship steel by means of tests made under modern accurate conditions. The bearing which these tests have upon my subject is the fact that they raised a suspicion in my mind that, under supremely accurate observation, it may be found that the average stress-strain curve is not a straight line from zero as is assumed in evolving Young's modulus. In the case of ordinary 28/32 ton ship steel, at least, there were many diagrams where as an ex-ship-draughtsman, I would have put a batten through the spots instead of a straight edge, while the first spot was seldom in a position appropriate to a straight line from zero.

The average extension of both steels, derived from the test results, was .006 inch on 8" at a stress of 10 tons per square inch, which gives an average value of 13,300 tons per square inch for Young's Modulus (E); but the range in value was, at least 7½ percent on each side of the average or 15 percent in all. Text Books always give a range in E values, but so wide a variation as 15 percent, coupled with the possibility that the derived E at the lower or working stresses is below a given average, does add to the uncertainties of beam formulae.

The respective merits of elastic limit, yield point and ultimate strength, are much discussed as the best multiple of the figure to be adopted for working stress, and we of the Classification Societies are deeply concerned in the right answer to this question because we know that the whole structure and most of its parts are continually subject to alternating stresses. Speaking as an observer of tests and user of material, it seems to me that the resistance to shear, elastic limit and yield point of mild steel fresh from the rolls (which conforms to standard physical tests) bear proportions to the ultimate strength which are sufficiently constant for use, and that strains which pass the yield point do not involve destruction of power to recover elasticity. The factor of safety puzzle is made more puzzling by the commonplace shipyard practice of straining much of the material beyond its yield point by joggling, flanging, &c., before it becomes part of the hull, without causing apparent detriment to its ability to withstand service stresses.

I wonder whether at the root of the matter there lies the possibility that beam theories which are mathematically accurate on the assumptions made, may require amendment because steel is a cooled liquid and, therefor, may not conform to the " laminae " theory.

The first bending experiment arranged by Messrs. Holt for the purpose of comparing the behaviour of ordinary steel with that of the special steel, was made with a beam of each material, each made of two 10" channels of identical size, riveted back to back. Each beam was 25 ft. long, was supported on knife edges 20 ft. apart, and was subjected to concentrated loads, which were increased by a ton at a time. The deflections were carefully measured, both when under load and after release from load. Values of E were then calculated from the deflections at different loads, and it was found that at the lower loads those obtained were much less than at higher loads. The beam of ordinary steel gave fairly constant values of about 10,500 tons per square inch for calculated stresses of 5 tons per square inch and above, up to loads where failure began. The special steel beam did not reach a steady E value of 10,500 until the calculated stress was about 8 tons per square inch.

The second experiment was made with other two beams of 10" channels in pairs, each having the same scantlings and modulus of resistance as the first two but, instead of being riveted back to back, they were held 10" apart by seven 10" channel distance pieces riveted to them, and were kept from endwise movement by end gusset plates riveted to their top flanges. The comparative results were of similar character to those obtained with the first pair of beams but E as calculated from the deflections, was 14,100 instead of 10,500 and was approximately constant over a wide range of load. The deflections in this case may have been reduced by a more stable disposition of material but theoretically they should have been the same.

Observation of the experiments confirmed that after permanent sett has apparently taken place and the material is on the point of failure, beams can and do recover straightness after a sufficient lapse of time. The beams of ordinary steel deflected less at low loads, took permanent sett sooner, showed signs of distress at an earlier stage and failed sooner than those of special steel, results which seem to point towards my contemplated interpretation of stress-strain diagrams.

It is difficult to explain the differences between E values obtained in the test house and calculated values derived from simple bending experiments of the order on which the elastic theory is based. The conditions, moreover, were the same for both beams in each experiment and each shows disagreement with accepted theory at such relatively low stresses as concern constructors.

The third experiment was made upon a watertight box about 19.5 ft. high, 10 ft. wide and 3.6 ft. in depth, one side being of ordinary and the other of special steel, while the stiffeners represented frames 30 inches apart with solid brackets which were expected to give fixed-end support. Unfortunately, the diaphragm influence of the plating, when subjected to water pressure, made it impossible to deduce practical relationship between beam theories and experimental results; but the results did confirm the contention that it is impossible to achieve fixed ends for a beam in a ship, or to ignore evidence that, at small loads, the modulus of resistance for stiffeners is increased by some proportion of the plating which is riveted to the stiffener.

The fourth and principal experiment was undertaken in order to make a direct comparison between the behaviour under load of the " PROMETHEUS " and that of the heavier sister ship " EURYMEDON ". When the " EURYMEDON " was completed and lying in still water, the Builders, Messrs. The Caledon Company of Dundee, after having made the end holds practically watertight, induced a hog of rather less than $2\frac{1}{2}''$ amidships, by means of a load of about 1200 tons of water at each end of the ship. Attempts were made to measure strain but wind and weather reduced delicate strain meters to an unreliable state. The water was then pumped out and it was found that the ship showed no measurable sett.

A sagging test was made by filling some 1800 tons of water into midship deep tanks &c. and this load had the effect of raising the sights on each side of the vessel amidships, $1\frac{7}{16}''$ above their initial position.

Mr. Lawrence Holt consulted me with regard to the subsequent experiment on the " PROMETHEUS " and readily agreed to the most severe load conditions that could safely be applied, and to the measurement of strain as well as deflection. The experiment, therefor, developed into an attempt to associate strength calculations with observed effects of known loads upon a large merchant steamer. They were carried out most carefully and deliberately in the presence of technical representatives of Owners, Builders, Board of Trade, Lloyd's Register and the British Corporation Register, and the stated facts are agreed facts.

As past experience had shown the incompatibility of delicate strain meters and working conditions, I devised a simple strain meter. This consisted of two steel bars, approximately $60'' \times 3'' \times \frac{3}{8}''$, each having a brass stud tapped into one end. The bars were placed end to end at the allotted positions and the outer end of each bar was secured to the plating by a steel stud, the distance between the two steel studs being 10 feet. Any strain over that distance of 10 feet was then directly measurable by gauging the gap between the brass studs at the free inner ends of the bars.

No adjacent hammering, vibration, &c., in fact, nothing short of direct action, had any disturbing effect upon these strain meters, and no one expressed the least doubt as to the accuracy of the strain measurements, after having seen them made. Twenty of these meters were placed in pairs, one on each side of the ship, and were located on the upper deck stringer abreast of the foremast and main mast, immediately before and abaft the bridge (or centrecastle) and amidships; on the bridge sheerstrake amidships, and at the ends of the bridge; on the heavy upper deck girders just forward of the bridge; and on the tank top margin plate just below that position, all as shown in Fig. 1. The arrangements for sighting the deflections on two lines of sights, were very complete and reduced human error to a minimum.

The initial readings were taken immediately after inclining experiments had been completed by the Builders, Messrs. Scott & Co. of Greenock, so that the ship was as free from initial deflection and stress, as is practicable under ordinary circumstances. The ship was then filled with sufficient water to induce the strains and calculated hogging moments recorded in No. 1 condition, Fig. 1 and Table A., and a maximum deflection, of 2½ inches. (In each case the stated deflections are means of those on each line of sights). Water was then added at each end to an amount which induced the strains and calculated hogging moments recorded in No. 2 condition, Fig. 1 and Table A., and a maximum deflection of 3.6 inches. When the water was pumped out, there was an apparent "sag" of about 1/16th of an inch. The strain meters showed changes from initial readings which are indicated by No. 3 condition, Fig. 1. (It should be noted that the strain readings have been joined by curves on this diagram not as representing curves of strain but for convenience and effect).

The double bottom and midship deep tanks were then filled until the reverse deflection was 13/16ths of an inch; the strains and calculated sagging moments being those for No. 4 condition, Fig. 1 and Table A. The water in the tanks was then manipulated to limits governed by the risk of endangering the ship's stability, until the measured deflection was 1.3 inches and the strains and calculated bending moments were those for No. 5 condition, Fig. 1 and Table A. When the ship was brought back to the initial condition it showed no measurable deflection from the original position and the strain meter readings corresponded to No. 6 condition, Fig. 1, or speaking generally, to a plus reading of 1/1000th of an inch.

Comparison between the deflections on the "EURYMEDON" and "PROMETHEUS" shows in "hogging" an apparent difference of the order of 7 per cent in favour of the "EURYMEDON", a figure which corresponds to the comparative values of the moments of inertia of the material in the two ships. The relative deflections in "sagging" are distinctly in favour of the "PROMETHEUS", which is probably a consequence of the fact that the latter has longitudinal beams on the bridge deck.

Table A. illustrates one of the difficulties which attend endeavours to connect theoretical with actual stress figures—the figures for which constructors ever seek. It is obvious that a ship's hull must contain material which is only effective at low stresses and becomes less and less effective as stress increases. I think this is the first published endeavour to relate experiment to this unknown variable, by associating the strains with four calculated values for the modulus of resistance. It will be seen that on the bridge deck amidships in No. 2 condition the stress of 4.8 tons per square inch derived from the observed strain, agrees rather well with the calculated maximum hogging stress S_3 (upper line) in computing which all longitudinal material is taken into consideration, subject to a reduction of 1/7th for rivet holes from material in tension. Consideration of the figures within the squares and the probable effects of unconsidered local material, gives good support to present belief that under tension, where the bending moment and the effective quantity of material are known, the stress derived from $p=\frac{My}{I}$ is sufficiently accurate to be confidently used as a basis for the scantlings of a box girder such as a ship. In other words the use of $\frac{I}{Y}$ as a measure of resistance to strain has been satisfactorily supported by this experiment for this type of girder.

The position is different when one seeks experimental proof of $p=\frac{Ey}{R}$, if only because of the difficulty of determining the neutral axis of a ship, not to speak of determining the radii of curvature to which it is bent at the neutral axis or the value of E. at different strains.

The lower lines of stress figures in Table A. were obtained by reducing the calculated stress figures in the ratio of 11,500 to 13,300. The former figure is, I think, suggested by Sir John Biles paper on the "WOLFF" experiments as a value for E. and I understand some such figure is used by bridge engineers for riveted girders. I fail to see an advantage in inventing a fictitious E. value when Young's modulus represents an understandable fact in relation to measured strains, which has hitherto been accepted as true. It seems unwise to prejudice theory in the minds of constructors who are trying to derive strains from calculated stresses, by suggesting that outside the test house, stress is not proportional to the measured strain upon material.

Evaluations of E. from the maximum deflections were made for each ship by the respective Builders, using the highest and lowest Moment of Inertia values given in Table A. for the "PROMETHEUS" and corresponding figures for the "EURYMEDON", both sets of figures having been provided by the British Corporation. This method of procedure does not altogether eliminate the personal equation, as the evaluations are extremely complicated calculations which were necessarily made by different people, nor the fact that deflections measured on the top of the ship can not be accurate in a mathematical sense, and do not lie on a fair curve. The derived values of E, ranged from 7,300 to 17,100, but the figures give qualitative information for each ship, in regard to their behaviour when the deck is under tension and compression respectively; the E. value in compression being 2/3rds of that in tension in the case of the "EURYMEDON", which has transverse beams throughout, and nearly 2/4ths in the case of the "PROMETHEUS", which has longitudinal beams in the bridge deck. It also happens that the E. value calculated from the "EURYMEDON" hogging deflection, on the assumption that all longitudinal material is effective in compression and 6/7ths in tension, practically conforms to 13,300 tons per square inch, a result which compares with that obtained from the strain readings on the "PROMETHEUS" in No. 2 condition, Table A.

These figures emphasise the need for correct assessment of resistance values of all material subject to strain and, especially, the need for further experiments which will tell constructors how to assess the effective resistance value of the same material under tension and compression, and which will teach how why and when they differ.

A ship is not a perfect beam and does not behave as such but there is an element of surprise in seeing that even in still water conditions, one side of a symmetrical girder does not behave in the same way as the other. If twist is normal in such conditions, what must it be under the dynamic forces of a seaway? The type of problem which still confronts structural designers is indicated by the evidence that the powerful deck girder at the break of the bridge, was relieved from strain in tension but not in compression.

The full scale experiment has, however, satisfied me of the wisdom of continuing to base calculated ability to resist unknown, alternating stresses, upon the minimum effective area of material after deducting a corrosion margin therefrom. I have already said, in Japan and elsewhere, that working stresses on a well-designed ship under average conditions can seldom rise much above 6 tons per square inch, and this assertion receives confirmation from the fact that under static conditions and an observed stress of 4.8 tons per square inch, the "PROMETHEUS" showed some slight signs of disturbance at the connections of light material, the kind of evidence usually quoted in support of statements about "working" and "straining". In these circumstances, few would risk sea service under the experimental load conditions, without doubling or more than doubling the ship's resistance modulus to cover very "live" loads. Under the classic wave crest calculations the abnormal disposition of weight added 75 percent to the still water bending moment, but the static comparison of the classification standard added 10 percent thereto. In this respect alone the opportunity of testing the effect of shifting centres of gravity of end weights 25 feet instead of the ordinary 3 or 4 feet from the "ideal" position,* has proved the value of the Holt experiments, but they have done much more, they have given facts which bear upon the relationship between experiment and theory.

The forces which act upon a ship under average service conditions are so complex and variable that they can only be approximately estimated and on particular occasions they may prove irresistible, so that the art of shipbuilding is chiefly the product of experience or full scale experiments. Modern applications of beam theories to this art are, however, of inestimable value as has been shown by the release of the modern ship from misconceptions inherited from wood shipbuilding, and by their successful use in the comparative analysis of structural design as well as in the design of ships for which no precedent existed. That use, however, is coloured by the fact that the standard formulae which are the practical residue of mathematical expression of the assumptions in which they are founded, always require modification in order to adapt them to the evidence of experience.

Having regard to the facts that elastic beam theories are based on laboratory experiments, that text books warn the student that beams of rectangular section might show experimental evidence of ability to withstand nearly double the calculated maximum load, that I can recall no technical paper descriptive of experimental research on behalf of naval architecture where the author has

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* See 'Bending and Loading of Ships" Trans. I. N. A. Vol. LXX. p. 37.

been able to correlate observed strains with calculated stress, it seems to me that a case has been made out for an overhaul of past assumptions by mathematicians.

Much has been done by intelligent use of existing thories but much more could be done if mathematicians would provide formulae which give certainty that under known loads and known conditions of support, calculated stresses would correspond to induced strains in material as built into steel ships. It is almost essential to complete observation that strains shall be measured and we await the necessary, simple strain meter which can be attached to any beam and will be affected by nothing but the strains to be measured.

The "Holt" experiments have given a lead and it is to be hoped that the example may induce others to follow the same road. If the interested public realised the economic gain to be derived from the immediate acquisition of facts which enable constructors to make better use of material subject to stress, it would surely seek for light from observed experiments of the character described, rather than suffer the delay and cost of countless experiments with ships in service.

— T.S.M.S. "PROMETHEUS." —

— Comparison of Observed and Calculated Stresses. —

Position of Strain Meter	Frame	Strain Meter	Moments of Inertia, Hogging: I_A, I_B, I_C, I_D	y_A, y_B, y_C, y_D	Moments of Inertia, Sagging: I_A, I_B, I_C, I_D	y_A, y_B, y_C, y_D	Bending Moments, Hogging: BM_1	Hogging: BM_2	Sagging: BM_3	Sagging: BM_4
Upper Dk. Stringer Angle For^d	135	AP AS	377920 350650 316840 267530	16·4 17·45 18·53 20·38	377920 347620 316100 285700	16·4 15·36 16·47 17·78	31175	40788	5119	1945
Upper Dk. Stringer Angle Bridge Front	118	BP BS	417460 385470 326460 295430	16·3 17·3 18·84 19·88	417460 385520 331450 317750	16·3 15·28 16·88 17·4	61232	81222	17844	13262
No. 2 Hatch Coaming	118	CP CS	417460 385470 326460 295430	17·4 18·4 19·94 20·98	417460 385520 331450 317750	17·4 16·38 17·98 18·5	61232	81222	17844	13262
Tank Top inside Gutter Bar	118	DP DS	417460 385470 326460 295430	15·3 12·3 10·76 9·72	417460 385520 331450 317750	15·3 14·32 12·72 12·2	61232	81222	17844	13262
Upper Dk. in Centrecastle	94	KP KS	542240 499090 486610 380000	14·17 15·26 13·49 17·48	542240 504320 492780 415710	14·17 13·1 13·33 14·67	83431	103491	38080	37839
Centrecastle For^d	112	EP ES	606970 565710 565710 516940	20·35 21·46 21·46 22·54	606970 559460 559600 547420	20·35 19·27 19·27 19·65	69372	90682	24398	20814
D° D°	112	EP ES	460080	23·98	489050	21·16	69372	90682	24398	20814
D° Amidships	94	FP FS	542240 498090 486610 380000	22·17 23·26 23·49 25·48	542240 504320 492780 415710	22·17 21·1 21·33 22·67	83431	103491	38089	37839
D° Aft	68	GP GS	606240 560920 539740 456830	20·46 21·54 21·99 23·15	606240 558630 537730 480410	20·46 19·4 19·44 20·39	77244	86564	37786	39396
Upper Dk. Stringer Angle Abaft Bridge	63	HP HS	382610 350790 289160 275790	17·09 18·05 19·59 19·16	382610 355830 293960 296380	17·09 16·13 17·8 16·7	72325	79845	35555	36780
Upper Dk. Stringer Angle Aft	34	IP IS	330460 303760 272690 228900	15·48 16·43 17·5 19·18	330460 304760 273550 246480	15·48 14·54 15·63 16·25	24067	24552	6093	5554

Strain Meter	No. 1 Condition (Hogging): Strain Meter Stress*	Mean of P. & S.	ϕ S_A	ϕ S_B	ϕ S_C	ϕ S_D (Normal + Modified, see Notes)	No. 2 Condition (Max^m Hog): Strain Meter Stress*	Mean of P. & S.	ϕ S_A	ϕ S_B	ϕ S_C	ϕ S_D (Normal + Modified, see Notes)
AP AS	1·35 1·44	1·38	1·35 1·56	1·55 1·79	1·82 2·1	2·37 2·74	1·56 1·56	1·56	1·77 2·05	2·03 2·35	2·38 2·75	3·11 3·6
BP BS	2·67 2·5	2·58	2·39 2·76	2·75 3·18	3·51 4·06	4·12 4·76	3·83 3·33	3·58	3·17 3·67	3·64 4·21	4·66 5·39	5·46 6·31
CP CS	1·56 1·78	1·67	2·55 2·95	2·92 3·38	3·72 4·3	4·35 5·03	2·56 2·5	2·53	3·38 3·91	3·58 4·42	4·93 5·7	5·77 6·67
DP DS	-2·11 -1·94	-2·02	-1·95 -2·26	-1·95 -2·26	-2·01 -2·32	-2·01 -2·32	-2·8 -2·8	-2·8	-2·59 -3·0	-2·59 -3·0	-2·66 -3·08	-2·67 -3·09
KP KS	2·55 2·45	2·5	2·12 2·52	2·56 2·96	2·64 3·03	3·84 4·44	3·61 3·78	3·7	2·71 3·13	3·17 3·67	3·29 3·8	4·76 5·51
EP ES	1·78 3·0	2·39	2·32 2·68	2·63 3·04	2·63 3·04	3·02 3·49	2·22 4·0	3·11	3·03 3·5	3·44 3·98	3·44 3·98	3·95 4·57
EP ES	1·78 3·0	2·39				3·62 4·19	2·22 4·0	3·11				4·73 5·47
FP FS	3·78 3·78	3·78	3·41 3·94	3·90 4·51	4·03 4·66	5·59 6·46	5·11 4·5	4·8	4·24 4·9	4·83 5·59	5·0 5·78	6·94 8·03
GP GS	3·11 1·56	2·33	2·61 3·02	2·97 3·43	2·98 3·45	[illegible] 4·57	4·0 1·78	2·89	2·92 3·38	3·32 3·84	3·34 3·86	4·39 5·08
HP HS	3·0 2·95	2·97	3·23 3·74	3·72 4·3	4·90 5·67	5·02 5·81	4·1 3·2	3·65	3·57 4·13	4·11 4·75	5·41 6·26	5·55 6·42
IP IS	·78 ·33	·55	1·13 1·31	1·29 1·49	1·54 1·78	2·02 2·34	1·1 —	1·1	1·16 1·33	1·32 1·53	1·58 1·83	2·06 2·38

No. 4 Condition (Sagging): Strain Meter Stress*	Mean of P. & S.	ϕ S_A	ϕ S_B	ϕ S_C	ϕ S_D (Normal + Modified, see Notes)	No. 5 Condition (Max^m Sag): Strain Meter Stress*	Mean of P. & S.	ϕ S_A	ϕ S_B	ϕ S_C	ϕ S_D (Normal + Modified, see Notes)	Strain Meter	Frame	Position of Strain Meter
-·11 -·44	-·27	-·22 -·25	-·23 -·27	-·27 -·31	-·32 -·37	·22 0	·11	-·08 -·09	-·09 -·10	-·10 -·12	-·12 -·14	AP AS	135	Upper Dk. Stringer Angle For^d
-·55 -·44	-·5	-·7 -·81	-·71 -·82	-·91 -1·05	-·98 -1·15	-·55 -·67	-·61	-·52 -·60	-·53 -·61	-·67 -·77	-·73 -·84	BP BS	118	Upper Dk. Stringer Angle Bridge Front
-·67 -·67	-·67	-·74 -·86	-·76 -·88	-·97 -1·12	-1·06 -1·20	-·89 -·55	-·72	-·55 -·64	-·56 -·65	-·72 -·83	-·77 -·89	CP CS	118	No. 2 Hatch Coaming
·55 ·44	·5	·57 ·66	·66 ·76	·63 ·73	·60 ·70	·25 ·44	·5	·42 ·49	·49 ·57	·51 ·59	·51 ·59	DP DS	118	Tank Top inside Gutter Bar
-1·00 -1·55	-1·27	-1·0 -1·16	-·99 -1·14	-1·03 -1·19	-1·34 -1·55	-·89 -1·33	-1·11	-·99 -1·14	-·98 -1·13	-1·02 -1·18	-1·34 -1·55	KP KS	94	Upper Dk. in Centrecastle
-·78 -·89	-·83	-·82 -·95	-·84 -·97	-·84 -·97	-·88 -1·02	-·67 -1·22	-·94	-·69 -·80	-·72 -·83	-·72 -·83	-·75 -·87	EP ES	112	Centrecastle For^d
-·78 -·89	-·83				-1·05 -1·21	-·67 -1·22	-·94				-·9 -1·04	EP ES	112	D° D°
-1·56 -2·0	-1·78	-1·56 -1·8	-1·59 -1·84	-1·65 -1·91	-2·08 -2·41	-1·78 -2·0	-1·89	-1·55 -1·79	-1·58 -1·83	-1·64 -1·9	-2·06 -2·38	FP FS	94	D° Amidships
-1·22 -1·22	-1·22	-1·28 -1·48	-1·31 -1·51	-1·32 -1·53	-1·6 -1·85	-1·44 -1·33	-1·38	-1·33 -1·54	-1·37 -1·58	-1·37 -1·58	-1·67 -1·93	GP GS	68	D° Aft
-1·56 -2·0	-1·78	-1·59 -1·84	-1·61 -1·86	-2·14 -2·47	-2·0 -2·31	-1·78 -1·89	-1·83	-1·64 -1·9	-1·67 -1·93	-2·21 -2·56	-2·07 -2·39	HP HS	63	Upper Dk. Stringer Angle abaft Bridge
-·22 -·11	-·16	-·29 -·34	-·29 -·34	-·35 -·4	-·42 -·49	-·11 0	-·05	-·26 -·30	-·26 -·30	-·32 -·37	-·38 -·44	IP IS	34	Upper Dk. Stringer Angle Aft

* Stresses based on ·005" extension in 8" at 10 Tons per Sq. in.

ϕ Upper figure shows calculated stress
  Lower " " " " increased × $\frac{13300}{11500}$

ϕ S_A Calculated stress using I_A (I_A = Moment of Inertia of all longitudinal material including bulwarks & 1/5 area of wood deck)

ϕ S_B " " " I_B (I_B = " " " I_A under reduction of 1/5 from all material in tension.)

ϕ S_C " " " I_C (I_C = " " " Effective longitudinal steel excluding bulwarks under reduction of 1/5 from all material in tension.)

ϕ S_D " " " I_D (I_D = " " " all longitudinal material including, in compression only, 1/5 area of wood deck in wells and longitudinal beams on weather decks, and excluding bulwarks, deck girders, intercostals, deckhouses, connecting bars, and all material within line of deck openings, [See †], and under reduction of 1/5 from all material in tension.

† This calculation assumes material within line of deck openings effective.

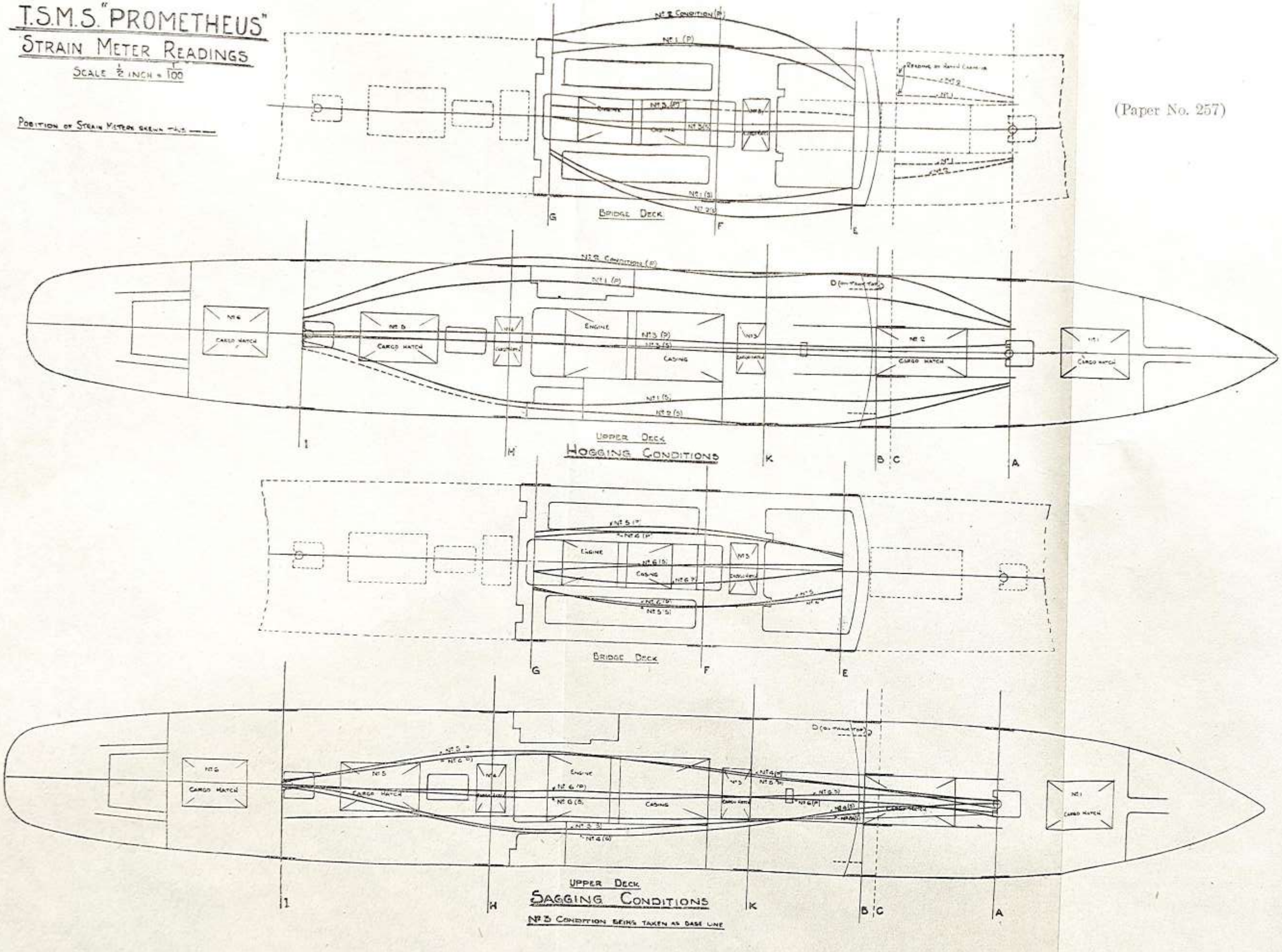
(Paper No. 257)

T.S.M.S. "PROMETHEUS"

STRAIN METER READINGS

SCALE ½ INCH = 1/100

POSITION OF STRAIN METERS SHEWN THUS ——



Recent Developments of the Marine Steam Reciprocating Engine.

(Paper No. 269)

By G. R. Hutchinson.

When the writer was invited to contribute a paper on modern marine steam machinery developments to the World Engineering Congress, through the Institute of Marine Engineers, he felt that the task was too difficult to accomplish in a single communication of limited length. To treat adequately the subject one must consider the recent important advances which have been made with (1) high-pressure, high-temperature geared turbine installations, (2) turbo-electric propulsion, (3) the tendency towards the wider use of water-tube boilers in mercantile steamships, (4) pulverised fuel and mechanical stoker firing for both water-tube and Scotch type boilers, (5) developments in feed water circuits (e. g., closed feed systems with multi-stage feed heating by "bled" steam, along lines similar to those which have proved successful on land), (6) condensing practice progress, (7) poppet valve reciprocating engines, (8) uniflow type engines, (9) exhaust steam turbines used in conjunction with a reciprocating engine driving the same shaft, and (10) auxiliary machinery improvements (e.g., uniflow engines for pump and fan driving, electric motor-driven auxiliaries, etc.).

From the foregoing brief survey it will be seen that a useful paper, calculated to be acceptable to engineers attending the Congress, can only be produced by confining attention to certain of these lines of progress. For this reason, the present paper is confined to the interesting and far-reaching developments which the reciprocating steam marine engine has undergone within recent years. The other branches of marine steam propulsion progress have been adequately and authoritatively discussed in several recent papers before the leading British, German, American and other technical institutions. In view of the importance and technical interest of the various developments which the steam reciprocating marine engine has undergone during the past decade or thereabouts the attention which it has received by technical societies is inadequate; and, so far as the present writer is aware, only one previous attempt[1] has been made to adequately review progress in this field.

Prior to 1914, the steam reciprocating marine engine had reached a stage of development when progress in design had ceased, apart from the use of newer materials and detail alterations from standard practice which certain engine builders saw fit to adopt. For many years the triple-expansion engine continued to be built in substantially the form which Dr. A. C. Kirk had

1) Trans. Inst. of Marine Engineers, London, Vol. XL. May 1928.

shown to be successful in the "Aberdeen" in 1881, although during the period up to 1914 numerous satisfactory patent metallic rod packings were introduced, piston type valves and balanced slide valves came into use, and the practice of casting the condenser shell with the back columns was abandoned in favour of a separate cast iron or steel plate shell either of cylindrical or pear shape. During the same period the use of the separately-driven centrifugal circulating pump gradually came into favour until it is, to-day, universal on good-class engines of moderate and large powers. The increasing use of the Edwards type air pump is another development of some importance to which reference might be made in this survey. What has been said in connection with the triple-expansion engine applies, naturally, to the more recent quadruple-expansion engine. As in the case of the "triple," various cylinder and crank arrangements were tried. Gradually, certain arrangements became standardised and have continued in use until the present time. The most notable development in this direction since the "standard" forms of triple-expansion and quadruple-expansion engine were evolved was the intoroduction, some 30 years ago, of the Yarrow-Schlick and Tweedy balancing system for four-crank engines, a refinement, if one may call it such, that has stood the test of time.

It will be appreciated that up to 1914, technical development in the field under review had not been very noteworthy. From 1914 to 1919 "production" rather than "progress" was the watchword in the industry and it was not until the postwar intensive development of the heavy-oil engine for ship propulsion began seriously to threaten the hitherto dominant position of steam that attention was devoted to the further improvement of the well-tried steam reciprocating engine, which had been, and still was, the type of propelling machinery adopted in the bulk of the world's mercantile tonnage.

Although the conventional slide-valve triple-expansion or quadruple-expansion engine possesses several excellent recommendations for marine work, than which there is no more arduous service, it is capable of being improved upon so far as performance is concerned. Improvement can be effected in several ways, and it may not be out of place to briefly examine the losses which occur in the cylinders of such an engine and discuss means of reducing them. Fig. 1. represents the indicator diagrams for each cylinder of a small quadruple-expansion engine combined on a single diagram of "ideal" characteristics, i.e., a diagram having a theoretical expansion line for the steam conditions, high-pressure cylinder cut-off, etc., obtaining in the actual engine. A study of this diagram will show that the combined areas of the four actual indicator cards is considerably less than the area of the "ideal" diagram which circumscribes them. The loss of mean indicated pressure may be classified under three heads, as follow: (1) Losses due to initial condensation and wire-drawing; (2) clearance losses; and (3) losses due to the expansion of the steam not being fully utilised in the low-pressure cylinder of the engine.

If Fig. 1. be examined the losses under heading (1) are clearly shown by the area between the theoretical expansion curve and the actual expansion curves for the four cylinders. Clearance losses (2) are shown as the area between the vertical pressure line of the theoretical card and the compression lines of the diagrams taken from the actual engine. The loss under (3) is characterised by the area between the horizontal volume line of the "ideal" diagram and the exhaust line of the low-pressure cylinder card. Fig. 1. serves to illustrate these points and is not put forward as showing a representative set of indicator diagrams with average percentage losses under the three headings given. The diagrams refer to a special type of engine with jet condenser, and will be referred to again later in the paper. Before passing on to a discussion of how the various types of loss suffered in a marine multiple-expansion reciprocating steam engine may be reduced it may not be out of place to examine the reasons for these losses.

Losses due to wire-drawing and initial condensation are greatest in an engine which makes use of the ordinary slide valve actuated by the Stephenson link motion or similar eccentric-driven gear. The ordinary slide valve has a gradual opening and closing action which wire-draws the steam as it passes through the ports. The cross-sectional area of the steam inlet ports is much less than the area of the piston and the steam cannot enter the cylinder quickly enough to maintain the full initial pressure on the piston up to the point of cut-off. A certain lose of pressure is suffered as the valve opens, due to the valve's acceleration being relatively slow, although the piston velocity at this stage is also low. As the piston moves away from the cover and its velocity increases the effective inlet port area increases; but because this area, even for conditions of maximum free ingress of the steam, is insufficient, wire-drawing or loss of initial pressure is suffered. As the valve closes and the effective inlet area decreases the wire-drawing effect is aggravated because the velocity of the piston is then increasing while the speed of the closing valve is decreasing. Clearly then, ample port areas, together with rapidly-opening and closing valves, are desirable if wire-drawing losses are to be minimised and the familiar rounded "corners" of the indicator diagram squared up, with a consequent small, but acceptable, gain in mean effective pressure. Although inlet ports were specifically mentioned above in connection with the phenomenon of wire-drawing, the remarks, obviously, hold good when considering the exhaust process.

The second loss coming into the first category is that due to "initial" condensation. Much has been written on this subject and many attempts made to minimise it. While it is established that steam jacketing of a cylinder is beneficial in minimising losses due to cylinder condensation this practice has now fallen into disfavour, principally because the cost of steam jacketing is not worth the saving obtained. One must go further to obtain any appreciable reduction of initial condensation. The obvious course, perhaps, is to adopt superheating, a practice which brings other advantages in its train, too well known to justify mentioning in this paper.

While it is true that superheating steam will minimise initial condensation, such benefit is only obtained at some sacrifice. Low superheat is better than saturated steam from the economy standpoint. Steam temperatures of 600/750° Fahr. are, however, now successfully carried in marine installations, and although high temperature steam can be safely used in a reciprocating engine, its use in an engine fitted with any form of slide valve is, the author thinks, inadvisable, with present materials. Whether a high, moderate or low degree of superheat is carried in a slide valve engine its benefit is partially lost in the passages and ports. When saturated steam is used condensation occurs, as is generally known, through the entering steam giving up some of its heat to the surfaces of the passages, valve surfaces and cylinder walls in the region of the port. The reason for this is that these parts are at a lower temperature than the admitted steam, due to their having just previously been cooled, comparatively, by the outgoing expanded steam of the last cycle. This effect is still seen when superheated steam is used, although radiation losses are decreased; but in this case the temperature fluctuation of the metal walls serves to reduce the degree of superheat which is usefully available. It is probable that the percentage loss of superheat is smaller the higher the degree of superheat carried, but any gain obtained in this direction is probably more than counter-balanced after a period of service by a falling off in performance due to valve leakage caused by pitting and grooving of the working faces. This is brought about by the carrying over with the steam of hard insoluble foreign matter which, moving at high velocity through the passages and ports, erodes and adheres to the lubricated slide faces, thus causing leakage. Although this erosive action can and often does manifest itself in the cylinder, by accelerating liner and piston ring wear, it is not unreasonable to believe that most of the trouble occurs before the steam reaches the cylinder. The use of an efficient steam drier and purifier is considered essential if the best results are to be obtained with superheated steam in a slide valve engine, while balanced slide valves, such as the Andrews and Cameron type, are desirable because of the reduction of valve face pressure which they effect. Another probable reason is softening of the springs in the piston rings by the high-temperature steam. Either different spring materials or ramsbottom type piston rings are the obvious remedy. The neglect of these facts is, it is believed, largely responsible for the full economy of superheat not being realised in many installations.

The losses suffered in a reciprocating steam engine which come under heading (2) are greatest in a slide valve engine of conventional tppe. If a general arrangement drawing of a typical marine triple-expansion engine be studied it will be seen that clearance spaces are relatively great. Receivers of ample capacity and considerable superficial area are provided and the effect of these receivers is to reduce the temperature and pressure available for performing useful work in the cylinder concerned. The necessarily long ports between the valve and the cylinder represent a volume of perhaps 12 per cent. or more of the cylinder capacity, which is entirely lost from the power-producing standpoint. The steam which fills this space after cut off is effected falls in pressure at approximately the same rate as the steam in the cylinder and does not fulfil the same amount of work as does the equivalent weight of steam in the main cylinder. After the exhaust process is completed the steam which fills the clearance space is at a relatively low temperature and this reduces the temperature of the next incoming charge of steam. Naturally, the longer the passage, and the greater its volume between valve and cylinder, the greater will be the temperature and pressure drop suffered by the steam before it reaches the cylinder, other things being equal; and hence the desirability of long valves and of minimum clearance volumes for all cylinders.

The losses under the third head are due to the practical necessity of keeping the low-pressure cylinder or cylinders of an engine within reasonable dimensions. This limitation does not allow of a reciprocating engine utilising the "toe" of the indicator diagram, as can be done in the steam turbine.

Minimising Losses.

Initial condensation and wire-drawing losses can conveniently be considered alongside clearance losses when discussing the manner in which they may be minimised in a marine steam engine. From our brief discussion of the losses due to initial condensation it is clear that the employment of separate inlet and exhaust ports and valves is desirable. This fact was recognised many years ago and was applied to land type steam engines in the well-known Corliss valve gear and later in the Sulzer "drop" valve mechanism. Marine engineers were somewhat tardy in adopting any type of valve gear giving separate inlet and exhaust ports. The first practical gear of this type was the Lentz poppet-valve system, which is really a modification of the Sulzer (and similar) drop-valve with a valve gear of simple design, reasonable first cost and which allows of reversing and manoeuvring being carried out expeditiously and with an absence of mechanical complication. The Caprotti poppet-valve system is somewhat similar thermodynamically to the Lentz gear in that both employ separate inlet and exhaust passages controlled by cam-operated poppet type double-beat valves. By adopting balanced double-beat valves, the power absorbed by the valve gear is reduced and the mechanical efficiency improved slightly.

Both the Lentz and the Caprotti poppet-valve gears reduce the losses due to wire-drawing of the steam in the ports, because the valves, being cam-operated, can be given very rapid openings and closings. In the Lentz engine closing of the valve is effected by a spring, as in a four-stroke cycle heavy-oil engine, while with the Caprotti system positive closing of the valve is accomplished by means of a cam. With both these systems wire-drawing losses are reduced and the indicator diagrams have sharp corners.

A promising line of attack is the Meier-Mattern hydraulic system for operating poppet valves. This system has been fitted to the triple-expansion engines of two vessels belonging to the Nederlandsche Stoomboot Mattschappij, the "Moena" and the "Borneo." It is interesting to note that both ships have water-tube boilers fitted with mechanical stokers. As a means of reducing condensation clearance and wire-drawing losses, this system has much to recommend it provided it is not too costly and is as reliable and as easily maintained in service as a mechanical valve gear. The Meier-Mattern system gives very rapid valve opening and closing characteristics, while the hydraulic valve actuation lends itself to the placing of the valve chests as near their respective cylinders as is practicable, thereby minimising clearances. The Meier-Mattern system is dealt with more fully in another part of the paper.

The best means of reducing initial condensation in a reciprocating steam engine is, as is well known, to adopt the thermo-dynamically desirable uniflow or central exhaust system, often credited to Prof. Stumpf, but actually the invention, many years earlier, of an Englishman named Todd. An examination of Todd's patent specification shows that he was fully alive to the advantages of the uniflow principle, but it was left to Prof. Stumpf to produce a commercially successful design. Uniflow engines have proved very economical on land but their application to marine work has been restricted to a few installations. One or two uniflow marine installations have been designed and built in Germany[1] and an interesting little single-acting engine is manufactured in Holland (Fig. 2), but beyond this nothing has been done by marine engineers with the uniflow principle for marine propulsion, although one or two auxiliary engines of this type are made. The cost and weight of the uniflow engine are against it in comparison with the equivalent triple-expansion engine, but the author feels that this type of marine steam engine may yet become popular. A double-compound, Wolff principle poppet-valve engine with a uniflow low-pressure cylinder should prove very economical and could doubtless be built at a competitive price if designed along straight-forward lines. The use of poppet valves in an engine of the type suggested would minimise the clearance, wire-drawing and condensation losses in the high-pressure cylinders, while the Wolf principle would eliminate the need for receivers between H.P. and L.P. cylinders, thus making the engine more compact and slightly reducing radiation and condensation losses. Apart from the reduction of cylinder condensation losses realised with uniflow low-pressure cylinders, a certain gain in economy would be obtained by the reduction of back pressure, which is possible in these engines. A well designed uniflow engine should operate with a very low back pressure, owing to the ease and rapidity with which the exhaust process may be completed. This feature is not, however, taken full advantage of in some uniflow engines, where the exhaust ports are somewhat restricted. The maximum possible exhaust port area should be aimed at, while at the same time the depth of the ports should not be too great, otherwise the percentage loss of effective stroke due to the presence of the ports will be too great. In this respect the Sulzer uniflow engine described elsewhere in this paper is commendable, the port area being

1) See "Der Schiffsmaschinenbau" by Dr. Ing G. Bauer.

considerable, the height of the ports small, and the path of the steam from cylinder to condenser short and easy.

Prof. Stumpf has recently produced designs for a double-compound uniflow marine steam engine. This engine, which is illustrated in Fig. 3, has a piston valve between each high and low-pressure cylinder, and the valve gear is of the Marshall type. The Wolff principle is used but the cranks are arranged at 150° instead of 180° to one another. This allows of a reduction of the compression period and so reduces throttling losses. It has been claimed for this engine that with a 1 to 7 cylinder ratio, 32 expansions, 29 in. vacuum, and an initial steam pressure of 470 lbs. per sq. in., the economy of the equivalent ordinary triple-expansion engine in combination with a Bauer-Wach exhaust turbine will be surpassed. The first engine of this type has been fitted in a small vessel, but no performance figures have been published.

Until quite recently no satisfactory solution of this problem bound up with the losses under heading (3), had been forthcoming so far as the ordinary single-screw cargo ship was concerned. Some years before the war the three-shaft "combination" system—wing reciprocators exhausting, usually, into a low-pressure turbine on the centre shaft—was introduced with excellent results, but naturally this system was not applicable to many vessels of moderate power. Lately we have seen the adaptation of the "combination" system to a single shaft, in the Bauer-Wach system of propulsion.

Although the "combination" system, working on a common shaft, was proposed by Sir Charles A. Parsons as long ago as 1906, the invention was not developed and it was left to Drs. Bauer and Wach to apply the system commercially some two years ago. The success which has attended the Bauer-Wach system since its introduction is well known and it need only be said here that over 100 installations have been built or ordered in about two years. Recently, the Parsons Marine Steam Turbine Co. Ltd. have evolved an exhaust turbine system for single screw vessels and the Brown Boveri Company, Baden, Switzerland, have also designed such a plant.

These developments are discussed later in the paper, and it is only necessary to state here that the Bauer-Wach and similar systems recover a high proportion of the energy still remaining in the steam after it leaves the low-pressure cylinder of the reciprocator. "Something for nothing" appeals to all of us and for this reason the exhaust turbine system, which offers a really substantial economy of fuel, at given power, as compared with the reciprocator alone, is certain of continued success. The arrangement minimises the losses under heading (3) in the only way possible, to the writer's knowledge, and as the saving possible under this head is important the development can be regarded as noteworthy.

The second part of the paper is purely descriptive and gives drawings, illustrations and descriptions of the various new forms of marine steam reciprocating engine which have been evolved in recent years. Before passing on to this section of the paper it might not be out of place to offer a few comments on the possible course of development in the future.

High-pressure, high-temperature steam yields substantial economies and its use makes for a reduction in the initial cost, weight and space occupied by the propelling machinery. For moderate powers—up to about 5,000 S.H.P. —the reciprocating engine is preferable to the "straight" turbine for various reasons, particularly when high pressures are used. High steam pressures, i.e., anything over about 300 lbs. per sq. in. gauge, necessitates the use of water-tube boilers; and water-tube boilers and reciprocating engines are not a happy combination for marine work. The author is aware that several shipping companies have made a success of this combination for several years but it seems an unlikely development for general adoption, despite its claims. Reciprocating engines require internal lubrication, and however efficient the separator used may be, oil is bound to find its way into the boilers, when trouble will result. If an exhaust steam turbine is fitted, the turbine will also suffer from oily deposits, as experience has shown. For these reasons the writer considers it unlikely that water-tube boilers will become very popular for reciprocating-engined steamships, unless internal lubrication can be greatly reduced, a possibility of which we must not lose sight.

So far as engine design is concerned, it is felt that the exhaust turbine will continue in popularity, although it is probable that reciprocating engine and turbine will eventually be designed as an entity which is calculated to obtain the best results in respect of economy, weight, cost and space from both units. As regards the reciprocating engine itself, it is felt that the value of poppet valves will become better appreciated as time passes, but which type of gear and which system of poppet valve will become most widely used is problematical at the present stage. Uniflow low-pressure cylinders will also become more popular, it is considered.

The flat slide valve is considered by many engineers to be obsolete for all but low-pressure cylinder work. The writer believes, however, that the balanced slide valve, which has undoubted merits where the steam conditions are not too severe, will continue to be widely used for many years. The possibility of using separate inlet and exhaust valves of this type has not been explored. Such an arrangement should prove economical and cheap, and a quadruple-expansion slide valve engine of this type using steam at 300 lbs. per sq. in. pressure, superheated to a moderate degree, should prove an excellent proposition in every way for the propulsion of ordinary tramp tonnage.

The field covered by this paper is a wide and interesting one and it is hoped that this paper will serve to bring to the notice of a wider circle the work which has been done in recent years.

Much of this development is dealt with in the next section of the paper.

Arnhem Quadruple-Expansion Engine.

A somewhat novel small quadruple-expansion engine was introduced some time ago by the Arnhemsche Stoomsleephelling Maatschappig, Arnhem, Holland which justifies consideration in this paper because of the ingenious way in which the valve gear, of conventional type, has been simplified and fore-and-aft length reduced. With a boiler pressure of 235 lbs. per square inch, the engine develops 320 I.H.P. at 190 r.p.m., the diameter of the cylinders being $10\frac{1}{2}$ ins., $14\frac{1}{2}$ ins., 21 ins., $30\frac{1}{8}$ ins. by 14–in. stroke. The successive stages of expansion are arranged with the H.P. cylinder at the extreme forward end of the engine, the first I.P. cylinder immediately aft of it, and so on. (Fig. 4).

The principal feature of this engine is the arrangement of the various valves and valve chests in relation to the cylinders which they serve. The high-pressure and first intermediate-pressure valve chests are placed athwartships on the same transverse centre line, while the flat slide valve for the second I.P. engine is arranged on the front of the engine with its working face at right angles to that of the low-pressure engine slide valve, which is placed at the extreme after end of the engine. The cranks for the H.P. and first I.P. engines are arranged at 180 deg. to one another, as are the cranks for the second I.P. and L.P. cylinders. The after pair of cranks, however, are turned through 90 deg. relative to the forward pair, an arrangement which gives four cranks equally spaced at 90 deg. around the crankpin circle.

The high-pressure piston valve is of the inside-steam pattern, while that of the first I.P. cylinder takes its steam from the ends of the valve. The valve rods for these two valves are coupled up to a common quadrant block and are driven by a common set of Stephenson link gear. Thus, by utilising for one cylinder an inside-steam piston valve and for the second cylinder an outside-steam piston valve, with the two main cranks placed relatively at 180 deg., the simplification of the valve gear is made possible without any compromises in respect to valve setting being suffered. The valves for the second I.P. and L.P. engines are ordinary flat slide valves, and a single pair of eccentrics and set of link motion has been adapted to operate both valves. The second I.P. valve rod is on the same transverse centre line as the second I.P. piston rod, and the main air pump. The L.P. slide valve is actuated directly through the link gear shown at the extreme after end of the engine, the second I.P. valve being worked by means of a rocking shaft, placed at about the same height as the wyper shaft, along the front of the engine. This rocking shaft derives its motion from a beam lever driven off the quadrant block by means of a pair of links; a similar short beam lever, but arranged at 180 deg. to the first lever, being keyed to the forward end of this rocking shaft. Motion is given to the second I.P. valve rod by means of a pair of links attached at one end of the rocking lever and at the other end to a yoke-piece fitted to the valve spindle. This arrangement permits of the motion of the second I.P. valve being exactly opposed to that of the L.P. valve, it being recalled that the cranks of these two engines are arranged at 180 deg. to one another. Inside receiver passages are cast in the cylinder block of the first three stages, but an outside receiver pipe conveys the steam from the second I.P. valve chest to that of the last stage of the engine.

It will thus be appreciated that the length of this modified quadruple-expansion engine is rather less than that of the normal engine of this type. In the case of the engine of this type fitted in the steam tug "Teuna III," the 320 i.h.p. unit referred to, it is claimed that the overall length is only about 7 per cent. more than that of the equivalent three-crank triple-expansion engine having the normal arrangement of valve gear. Not only is the length of the engine reduced, but the simplification of the valve gear probably effects a small gain in mechanical efficiency. The rocking shaft through which the second I.P. slide valve is driven is held in two substantial brackets bolted at either end to the bottom of the cylinders. The shaft works in large adjustable brasses and the L.P. and second I.P. valve spindle guides are bolted, respectively, to the after and forward brackets of the rocking shaft. The link motion, drag links, wyper shaft, reversing mechanism, etc. of the engine follow normal practice.

An interesting feature of the design is the lubrication system employed, the usual oil boxes on the front of the engine being dispensed with. The bedplate of the engine is closed in to form an oil-tight crank chamber, and an oil sump is arranged in the after bedplate between the end of the engine and the thrust block. After passing through a filter the oil from the crank chamber flows into this sump. A small oil pump, which is driven by means of rocking levers from one of the crosshead pins, draws the oil from this sump and delivers it through a pressure filter to a main distribution pipe running along the bottom of the cylinders. From here the oil is led to a number of small boxes on the front of the engine, these being provided with adjusting needle valves and sight glasses. In this manner a positive supply of lubricant, at a pressure of from 15 to 25 lbs. per square inch, is assured to all the working parts, while ready regulation of any or all of the oil feeds is possible. A stand-by lubricating oil pump, similar to the service pump, is provided. Relief valves with overflow pipes to the crank chamber are placed at different positions in the lubrication system, so that damage due to pressure accumulation is impossible. The pressure filter in the lubrication circuit is provided with a sight glass so that the collection of water or foreign matter may at all times be observed. A small steam pipe is connected to the lubrication system so that any oil pipe may be blown through should it become clogged. Arrangements are also provided for warming up the oil in the sump during cold weather so as to aid its initial circulation, a steam-heating pipe being led to the bottom of the sump. The lubricating oil consumption of the engine of the "Teuna III." during 100 hours' running worked out at rather less than one gallon, which is very economical.

Accompanying Fig. 1 are some performance data from this engine. Having regard to its small size and to the fact that a jet condenser was used, the results are very creditable.

Sulzer Uniflow Engine with Hydraulic Valve Gear.

A uniflow marine steam engine of very interesting design was produced some time ago by Sulzer Bros., Winterthur, Switzerland, for the propulsion of a small paddle steamer, the "Hèlvétie," for service on Lake Geneva. The engine, see Figs. 5 and 6 is of the horizontal type and has three uniflow cylinders 850 mm. in diameter by 1,200 mm. stroke. With a boiler pressure of 120 lbs. per square inch 1,500 B.H.P. is developed at 46 r.p.m., the cut-off in each cylinder being, at this power, 14 per cent. and the vacuum in the condenser 27/27½ ins. The engine is noteworthy in having oil-operated valve gear instead of any form of mechanical gear, the mechanism being very similar to that used on certain Sulzer land type uniflow steam engines. The engine is enclosed and forced lubrication is applied to the principal moving parts. A sectional arrangement drawing of this interesting engine is shown in Fig. 5. Fig. 6 is a longitudinal section of the Sulzer uniflow engine cylinder showing the principle on which the cylinders are constructed. The three cylinders are carried by a central casting which also serves as an exhaust steam receiver. Each of the cylinder liners is divided into two parts, fitted into the central casting from the front and rear respectively and suitably secured. As can be seen from the drawing, the ends of the liners lying in the centre piece are sinuous, so that a projection on one half fits into a recess in the other, without, however, touching. In this manner one half-cylinder has already started to guide the piston rings when the other half leaves off, while the area bounded by these two sinuous edges serves as exhaust ports, which are uncovered alternately by the front and rear of the piston. This construction prevents the cylinder walls being subjected to dangerous expansion stresses.

The single-bear poppet admission valves are fitted direct in the covers and are parallel to the cylinder axis, thus reducing the clearance to a minimum. Since the engine works with high vacuum, two sets of packing are fitted in each stuffing box in order to keep it tight against pressure in one direction and against vacuum in the other. Both parts are arranged symmetrically and each consists of two rings of four packing segments pressed against the piston rod. The inner packing, which renders the stuffing box steam-tight, is also pressed outwards by spring and steam pressure in the axial direction against a spherical ring piece, while the outer packing, which keeps the stuffing box air-tight, is pressed inwards. The pressing together of two pairs of rings makes it possible to obtain a reliable packing which is tight in either direction. Lubrication of the stuffing box is effected from outside through an oil wiper, to which lubricating oil is introduced from the force pump lubricating the cylinder walls.

The length of the piston is about 90 per cent. of the stroke and consists of four main parts. The steam pressure on both sides is transmitted by the two piston heads direct to the piston rod. The piston rod is supported on a slipper piece bearing, on the bottom half of the liner only, by means of a central piece. In order to compensate for expansion due to heat and to avoid consequential deformation, the central piece is carried in a spherical socket on the slipper. Any wear of the rubbing surfaces can be taken up by shims. Lubricating oil is introduced to the cylinder at six points by the cylinder lubricating pump.

Another departure from normal practice is in the provision of a spherical joint between crosshead and slipper. Here, as in other parts of the engine, spherical joints are employed to ensure uniform stressing of material and less liability to deformation due to heat or working stresses or to alterations in the vessel's hull during rough weather.

The big ends of the connecting rods are of the standard marine type. To obtain a part of uniform strength, and also uniform expansion of the bolts, the bodies of the bolts are drilled out over the plain portion of their length so that they have approximately the same cross-section as at the root of the thread. Here also the advantages of a spherical bearing have been made use of, in that the nuts and the bolt heads are supported by spherical working faces bearing in spherical washers. In this manner, it is made as certain as possible that the connecting-rod bolts are uniformly and centrally stressed and that expansion is uniformly distributed over the whole length, thus reducing the risk of breakage to a minimum. The main bearings for the crankshaft are built on the same principle as are the big-end bearings. All four main bearings are fitted in one common bearing frame. The cap bolts serve at the same time to hold the connecting pieces between the cylinder block and the bearing-frame, thus giving direct transmission of the forces from the cylinders to the bearings.

The admission valves, seen in Fig. 7, are provided with a single seat and are operated by oil under pressure. The forces tending to close the valves are the pressure of the incoming steam on the back surfaces of the valve members V and the force of the spring F; on the other hand, the forces tending to open the valves are the oil pressure from the pipe O acting on the piston K and the compression pressure in the steam cylinder Z on the underside of the valve V. The valve only begins to open when the compression pressure has reached its maximum value, since the oil pressure is until then not sufficient to open the valve against the admission steam pressure. Because of this, the oil control valve has, during normal running, only to effect the closing of the valve when cut-off takes place, while the amount of opening before dead centre is determined by the compression in the cylinder. When the engine is starting, the conditions are different, since the compression pressure is then comparatively low, and in order that the engine may be capable of starting, it is necessary to reduce the admission steam pressure until the pressure of the oil alone is sufficient to open the inlet valve. As soon as the engine has made a few revolutions and the compression pressure has reached a normal figure, the steam admission pressure is raised to full boiler pressure. In order that the valve may return quietly to its seat when closing and also not be violently arrested after opening, the piston K (Fig. 8) with the cylinder M has been designed as an oil brake in such a way that the piston closes in succession helically-arranged ports in the cylinder, until finally a certain quantity of oil is trapped and consequently acts as a brake to the piston. In order to make the braking position conform exactly with the closing position of the valve, the cylinder M can be moved axially by means of the screw S, and worm wheel C and the thread G, thus obtaining very smooth motion of the valve. During opening and closing, the oil enters at first through the valves W_1 and W_2 until a sufficient number of the helically arranged escape ports has been uncovered in the wall of the cylinder M.

No information regarding the performance of this interesting uniflow engine is available, but in view of the low steam consumption of land type Sulzer uniflow engines with hydraulic valve gear, it is probable that the results obtained are in no way inferior to those of the equivalent triple-expansion engine of normal design.

Meier-Mattern Hydraulic Valve Gear.

The Meier Mattern poppet valve gear has been referred to in the first part of the paper and as it is of considerable technical interest and practical promise a short description of the system, which is a Dutch invention, might prove of general interest. Briefly stated, hydraulic pressure generated in plunger pumps driven off the engine is utilised to actuate the double-beat poppet valves, ready means being provided for altering the point of cut off, reversing the engine, etc. Reversing of even large engines is accomplished by hand.

Referring to Fig. 9, (1) is the engine valve, to the stem of which a plunger (3), known as the passive plunger, is attached. This moves in a cylinder (4), which is connected through a space (5) and the pipe (6) to the cylinder (7), in which the so-called active plunger (8) moves. The latter is actuated directly off the crankshaft by means of an eccentric, the plunger rod (14) being able to rotate in the crosshead (15). The active plunger is hollow and there are holes in its walls, one of which is shown at (13). When the plunger rises from its lowest position and all cylinder spaces and pipes are filled with oil the latter can pass either of two ways: (a) through (6) to the passive plunger, which, however, is springloaded (12); or (b) through (10) to the vessel (11), which is not under pressure. The oil will take the latter direction until the position shown at A is reached, when the plunger shuts off the passage (9). The oil is then forced under the passive plunger and the valve is opened with a velocity which can be made as high as desired by reducing the size of the passive plunger relative to that of the active one. The passive plunger rises until it exposes the opening (26), when it will remain balanced between the spring and oil pressures. Thus full valve opening for all practical cut-offs is obtained, since this maximum lift is reached in a very short time. The active plunger continues rising until it reaches the position B, when the opening (9) is exposed once more, this time via the opening (13). The pressure drops immediately and the spring rapidly closes the valve. A dash pot (19) is fitted to prevent the valve hammering the seat. Further upward motion of the active plunger has no influence on the valve. Position C shows the top dead centre.

Regulation of the moment of cut-off is obtained by turning the active plunger about its axis by means of the handle (22). In the position shown the effective length of stroke is the distance b. Obviously, therefore, this length will be altered by turning the plunger, and any desired cut-off can thus be obtained. The point of admission can be altered by changing the shape of the top of the plunger (20), while exhaust and compression are regulated by the exhaust plungers. The lever (22) can be actuated by hand or governor. On the return stroke of the active plunger oil is drawn in via the openings (9) and (13) until the former is again shut off. The automatic non-return valve (23) then opens and oil flows through the passages (24) and (6) to the cylinder (7) until the upper edge of the active plunger exposes the opening (9) again. Thus the oil system is automatically kept filled and any oil which may have leaked away is at once replaced by oil from the low-pressure vessel. In practice the leakage has proved to be nearly negligible. For a marine engine the valve gear must be reversible. It is obviously possible to arrange the angles between the crank of one cylinder and the pump eccentric for any of its valves so that when the engine is running in the reverse direction the same pump is timed correctly for actuating the corresponding valve on another cylinder. Therefore, the only mechanism needed is a suitable cock placed in the delivery pipes of the pumps. The passages drilled through this cock direct the oil to either the one valve or the other, the direction taken depending on whether the cock is in the ahead or the astern position.

For starting up the engine one or more valves have to be opened by some outward means, as the valve gear can only work when in motion. A separate small pump, an accumulator, or any other similar means can be used for this purpose.

Fig. 10 shows the steam admission passages and the steam velocities for various cut-offs for the high-pressure cylinder of the Borneo before and after conversion, and clearly indicates the great advantages of quick-acting poppet valves. The active plungers can be so designed that the points of admission, exhaust, and compression are correct for giving the lowest possible steam consumption for every cut-off. The result is an indicator diagram which is practically ideal. Fig. 11 shows the comparison between the high-pressure indicator diagrams of the Moena before and after conversion, the amount of visible steam consumption being equal. To obtain this, a cut-off of 56 per cent. suffices with hydraulically-operated valves, as compared to 70 per cent. formerly. The area of the new diagram is considerably larger, due to a wide inlet valve opening, a sharp out-off, a rapid exhaust, and a compression of 8 per cent. instead of 13 per cent., which is possible because of a quick-shutting exhaust valve.

Bauer-Wach Exhaust Steam Turbine.

In the Bauer-Wach system the losses inherent to the low pressure cylinder of the reciprocating engine are overcome by continuing expansion of the steam in a high-speed, low pressure turbine, which transmits its power through double-reduction gearing on to the shafting of the reciprocating engine. In other words, the economy of the triple-screw "combination" system is obtainable with a single-screw or twin-screw installation. The power of the Bauer-Wach exhaust steam turbine is transmitted to the gearing through a

Föttinger hydraulic coupling or clutch, so arranged that the turbine can pick up its load gradually and without shock, while means are provided for quickly filling or emptying the coupling, so that the turbine may be easily and automatically out cut when it is desired to manoeuvre, which is done with the reciprocating engine alone. A change-over valve is fitted for by-passing the exhaust steam direct to the condenser, under such conditions. Owing to its high speed, the turbine is possessed of considerable momentum, and a quick and easy means of isolating the turbine from the system is essential. The change-over valve and the hydraulic coupling are operated by a common oil-pressure system, and when it is desired to cut in the turbine, the oil coupling is gradually filled by means of an oil-regulating valve. As the coupling fills, the turbine commences slowly to revolve, and when the coupling is full, the oil develops pressure, which, in turn, operates the change-over valve, which automatically cuts the exhaust off from the condenser and admits it to the turbine. When it is desired to disengage the turbine the change-over valve is turned to direct the exhaust direct to the condenser, when the pressure in the oil coupling rapidly becomes automatically reduced, and the latter is emptied. The turbine is now freely disconnected, and gradually comes to rest under the influence of friction. In order to render the system foolproof the oil-regulating valve lever is mechanically connected to the wyper shaft of the reciprocating engine in such a way that the engine cannot be reversed without first by-passing the steam through the change-over valve, direct to the condenser, thereby cutting out the turbine. The mechanical connection of the oil-regulating valve lever to the wyper shaft, although positively locked against any astern movement, is so arranged that the turbine may be engaged or disengaged by working the lever by hand at any time when the reciprocating engine is running ahead.

The principle of utilising the exhaust steam turbine and reciprocating engine on a common line of shafting was first proposed by Sir Charles A. Parsons, although the credit for producing a practical unit belongs to the German engineer, Dr. G. Bauer, until recently head of the engineering section of the Vulcan Werke A. G., Hamburg; and Dr. Wach, technical director of the firm of Joh. C. Techlenborg A. G., Geestemunde, Bremerhaven. The Techlenborg firm built the first experimental engine on the Bauer-Wach system, about two years ago, and this unit was subjected to an exhaustive series of trials, before being fitted into the large trawler "Sirius." The engine is of the triple-expansion type, with cylinders 13¾ ins., 20½ ins. and 31½ ins. in diameter, by 23¾ ins. stroke, and is capable of developing about 450–500 I.H.P. at 110 r.p.m. Fig. 12 shows the general layout of the "Sirius" combination unit. The exhaust turbine is mounted above the main engine shafting and immediately abaft the low pressure cylinder. The multiple-collar horse-shoe thrust block, usually found in engines of this class, was discarded in favour of the single-collar Michell thrust block, and the longitudinal space required for the normal multiple-collar thrust block taken up by the turbine and double-reduction gearing.

In this pioneer unit the turbine runs at a speed of 6,000 r.p.m., which is reduced by means of double-reduction mechanical gearing, having a 50 to 1 ratio. In order to relieve the gear teeth of any inaccuracies of mal-alignment, due to the wear down of the main engine shafting, the gear wheel is not mounted directly on the engine shaft, but is fixed to a special hollow shaft, concentric with the main shafting, between the after main bearing and the Michell thrust block.

On the test-bed, the combination unit was coupled to an electric generator, and when the engine was tested with the turbine idle, a power of 500 I.H.P. was recorded at 110 r.p.m., with a vacuum of 25½ ins. With the exhaust turbine in action, the output at constant specific steam consumption was increased by 20 per cent. at 28¼ ins. vacuum. Fig. 13 shows the comparative results obtained on the trials of this installation.

Service results of the "Sirius" have shown that with a cut-off of 55 per cent. in the high pressure cylinder, the same sea speed has been attained as with a 71 per cent. cutoff when the turbine was not is use, from which it is estimated that the saving in fuel due to the turbine is about 28 per cent., the boiler pressure being 220 lbs. per square inch, and the total superheated steam temperature about 600° Fahr.

The question of the effect which the increased total power developed, due to the addition of the turbine, will have upon the shafting, was given careful consideration by the sponsors of the Bauer-Wach system. The turning moment diagram of a normal steam reciprocating engine shows a succession of peaks which are greatly in excess of the mean average driving torque on the shaft. When the Bauer-Wach exhaust steam turbine is applied as a conversion, it is usual to increase the back pressure in the L.P. cylinder, the equivalent to reducing its useful work, and to apply this higher pressure through the turbine. In other words, the peak on the diagram for the L.P. cylinder is reduced, and although the average driving torque on the shaft is increased due to the turbine, the effective maximum torque still falls well below the peak torque for which the shafting was designed. The intermittent impulse of the reciprocating engine is therefore partially replaced by the even torque of the turbine. This consideration has led certain of the classification societies to approve of the addition of the Bauer-Wach turbine, with an increase of power of perhaps 30 per cent., without any increase of scantling of the line shafting. As the conversion of existing ships has proved to be an equally important line of business to new construction on the system, this point is of great importance.

Although a comparatively new development, upwards of 110 ships have already either been fitted with new Bauer-Wach machinery or have been converted or are being converted to the system. Fig. 12 shows a typical arrangment of Bauer-Wach turbine gearing and hydraulic coupling.

Fuller details of this very promising line of development were recently given in a lengthy paper before the Institution of Engineers and Shipbuilders in Scotland by Dr. Gustav Bauer, to which source reference should be made for details of the various methods of applying the Bauer-Wach exhaust steam turbine to steam reciprocating engines. The cost of the exhaust steam turbine of this type is said to be sufficiently low to enable the cost of conversion to be written off in from two to about six years, according to the efficiency of the original machinery. Naturally, the more economical the vessel, the longer will it take to wipe off the first cost in the fuel saving obtained at a given hourly steam consumption. The writer regards this development as the most promising one made to improve the efficiency of the reciprocating-engined steamer. It is an invention which can reasonably claim to approach finality in the development of the reciprocating-engined steamship, the small complication of gearing, coupling, vacuum augmentor, etc., being more than balanced by the economy realised.

Brown-Boveri Exhaust Steam Turbine.

Another interesting design of exhaust steam turbine for use in conjunction with a reciprocating engine driving a common line of shafting is that which Brown Boveri of Baden, Switzerland have developed. At the time of writing no actual application of this system has been made but as it possesses several interesting features, a description of it might not be out of place. The general arrangement of the Brown Boveri exhaust steam turbine will be found in Fig. 14 and 15 and this is practically self-explanatory. The principal difference between this system and the Bauer-Wach arrangement is the substitution of a mechanical flexible coupling in place of the Föttinger fluid coupling and the adoption of an astern section in the exhaust turbine. During manoeuvring the exhaust turbine is not uncoupled, as it is considered by the designers that should any of the control mechanism of the Bauer-Wach system fail or should the hydraulic coupling fail to fill the uncoupled turbine will run away with disastrous results. The astern blading in the Brown Boveri system is really provided to absorb the high kinetic energy of the turbine during reversing. When the control lever is placed in the astern position live steam is simultaneously admitted to the astern blades and this has the effect of checking the speed of the turbine. When changing back to the ahead position live steam is again admitted to the ahead turbine through special connections. The valves which control the admission of boiler pressure steam to both sides of the turbine as well as the valves controlling the admission of exhaust steam either to the condenser or to the turbine are positively operated by an oil pressure system. The flexible coupling is oil damped and prevents the torque irregularities of the reciprocating engine from affecting the reliability and wearing qualities of the gearing. It is also arranged so that it can allow of slight mal-alignment of gearwheel shaft and crankshaft which might occur due to wear-down of the crankshaft bearings of the reciprocating engine.

Parsons Exhaust Steam Turbine.

The arrangement of the Parsons exhaust steam turbine differs in several respects from either the Bauer-Wach or the Brown Boveri designs. In this system as in the Brown Boveri design a mechanical flexible coupling is used in preference to a fluid clutch, double reduction gearing is employed and the general arrangement may be gathered from Fig. 16. The control valve which directs the steam either to the exhaust turbine or to the condenser is operated, as in other systems, by a connection to the wyper shaft of the reciprocating engine. The elastic coupling which is utilised to absorb the uneven torque of the reciprocating engine is interesting inasmuch as it is in the nature of a multiple plate friction clutch which transmits the power through a series of stiff springs, the slipping of the friction plates being limited by the extent of the compression of these springs. The stress in the springs does not exceed 3 to 4 tons per square inch, and the construction is such that it is not necessary to disconnect the turbine when reversing the main engine. During reversing the exhaust steam from the reciprocating engine is passed directly to the condenser by the automatically-operated change-over valve, a feature which, it is interesting to note, was incorporated in the original Parsons single-screw combination machinery patent of 1906. It is claimed that the turbine rotating in vacuum can be reversed with the reciprocating engine without any undue stress being imposed upon the teeth of the gearing. Whether these claims can be substantiated in practice has yet to be shown for no exhaust turbine installation of this design had been built up to April of this year.

Jaffa Single-Acting Uniflow Engine.

A somewhat unusual uniflow engine is manufactured by the Jaffa Engineering Co Ltd., Utrecht, Holland, in various sizes up to about 1,000 I.H.P. The engine is notable in being single-acting, its appearance and general design closely resembling a two-stroke cycle marine oil engine. Steam distribution is effected by means of a small piston valve, carried centrally in the cylinder cover. Steam is admitted to the cylinder cover, around which it flows before passing into the admission valve casing, thereby steamjacketing the cover in accordance with true uniflow principles. The engine shown in Figs. 17 and 18 is a 4-cylinder unit of 200 I.H.P., the cylinders having a bore of 11.8 ins. and a stroke of 15.7 ins. As 200 I.H.P. is produced at 240 r.p.m., the piston speed is thus 580 ft. per minute and the mean indicated pressure 47 lbs. per sq. in. The four cylinders are supported on a hollow cast-iron beam, which acts as an exhaust steam receiver, and which also carries the piston rod gland. Cylinders and hollow beam are mounted on the tops of four sets of cast-iron "A" frames of conventional type, white metal-faced crosshead guides being secured to the backs of the frames. The engine is provided with a closed crankcase, the bottom portion of the crankcase serving as a sump for the forced lubrication system. The forced lubrication pump is driven by a crank off the forward end of the crankshaft, the pump drawing oil from the sump through a fine copper gauze filter and suppling it to the main bearings at a pressure of about 30 lbs. per square inch. From here the oil passes along the drilled journals and crank webs to the bottom end bearings of the connecting rod, up the connecting rod to the small ends, and from there to the crosshead guides. The oil passes down the guides to the sump and so is again drawn through the filter and circulated. The steam admission valves are operated by eccentrics keyed to the crankshaft, the motion of the eccentric being transmitted to the valve by means of long push rods, as shown in the drawing. The valve motion is of the well-known Klug-Hackworth pattern, and it will be seen that about midway along the eccentric rod a slot is formed. The fulcrum pin carried in this slot is controlled by the handwheel shown on the front of the engine, rotation of the screwed control spindle varying the angle of advance of the eccentric, and so the point of cut-off.

The 200 I.H.P. engine shown is stated to have a specific steam consumption of $10\frac{1}{4}$ lbs. per I.H.P. per hour when using steam at 190 lbs. per sq. in. boiler pressure superheated to 570°Fahr.

The Lentz Poppet Valve Double-Compound Engine.

The Lentz engine is the best-known of the various new types of marine reciprocating steam engine which have been developed of late years. For this reason, detailed description is not necessary in this paper; the drawings reproduced in Figs. 19 and 20 convey, moreover, more than could any written description. The Lentz engine, designed by the German engineer H. Lentz, is made in Germany in several standard sizes ranging from about 300 to 5,200 i.h.p., all being of the four-crank double-compound type, utilising the Wolff principle. At the outset Lentz poppet valve gear was applied to several triple- and quadruple-expansion engines before the standard double-compound engines were evolved. The experience with the former showed that the reduction of cylinder condensation achieved, by virtue of using separate inlet and exhaust ports, made it unnecessary to have three or four stages of expansion and so the double-compound system was adopted.

The Lentz double-compound engine is virtually two Wolff compound engines so coupled together that the two H.P. cranks are at 90° to one another. Six steam distribution valves are required top and bottom, the valves being directly actuated from an oscillating camshaft at half cylinder height. The valve gear, as Fig. 19 shows, is of the Hackworth type and control and reversing is carried out by a single hand wheel which operates the gear for both " halves " of the engine. The starting valve wheel is adjacent to the reversing wheel. Usually all the pumps necessary for running the engine are separately driven, an arrangement which naturally facilitates rapid starting and manoeuvring, although somewhat more costly and less efficient than the usual practice.

The writer can testify, from personal experience, as to the rapid and easy manoeuvring of a Lentz engine. He has seen an engine of 3,000 I.H.P. manoeuvred by hand in a most impressive manner, while the general running of the engine (which was provided with forced lubrication) was very satisfactory indeed.

The standard 1,500 I.H.P. at 95 r.p.m. Lentz engine is said to have a specific steam consumption of 10.1 lbs. per I.H.P. per hour, with H.P. cut-off of 39 per cent, when using steam at 200 lbs. per sq. in. boiler pressure superheated to 620°F. The condenser vacuum for this result is 27 ins. This performance is very good and should be capable of improvement by increasing the superheat to 700°F. or thereabouts. This latter temperature is higher than it is advisable to attempt to go in a slide valve engine, but as valve lubrication troubles are eliminated in a poppet valve engine the Lentz and similar types are able to utilise in full the benefits of really high superheats.

Caprotti Poppet Valve Gear.

The Caprotti poppet valve gear is another promising development, which is being sponsored in Britain by Wm. Beardmore & Co. Ltd. While the double-beat poppet valves used are somewhat similar to those employed in the Lentz design, the operating mechanism is radically different. In the first place a rotary camshaft is employed, whereas that used in the Lentz engine is of the semi-rotary pattern. Incidentally, Lentz patents have recently been taken out which embrace the use of rotary, totally enclosed camshafts, but to the best of the author's belief no marine engines of this modified type have been constructed.

With the Caprotti system three rotary cams are employed to operate the four valves of a cylinder—one inlet and one exhaust valve per cylinder end. The steam admission valves are provided with two cams, one controlling admission and the other cut off. One valve controls both functions in the case of the exhaust valve. An ingenious mechanism, utilising quick-threads for altering the angular position of the cams, is provided for reversing the engine and altering the point of cut off, the whole of the cam gear running in an oil-tight box. The camshaft is driven by means of bevel gearing and a vertical shaft in much the same manner as is employed in Diesel engine practice. This feature is clearly shown in Fig. 21.

Two small British vessels have been fitted with triple-expansion engines having poppet valve gear of the Caprotti type, Wm. Beardmore & Co. Ltd. supplying the machinery. Performance data from these ships have been published in the technical press[1], and it need only be said that the results obtained were satisfactory. Fig. 22 shows indicator diagrams obtained from the triple-expansion Beardmore-Caprotti engine of a small steamship, while Fig. 23 shows the combined diagrams for the same engine. The diagram factor of 0.78 will be noted. It is worthy of note that the vessel uses saturated steam, which is somewhat extraordinary in view of the fact that poppet valves are particularly adapted to the utilisation of superheated steam. In fact, the writer

1) The Marine Engineer & Motorship Builder, August 1928.

considers that the extra cost of the poppet valve engine is not justified, having regard to the results obtained, when saturated or slightly superheated steam is used. Figs. 24 and 25 show different views of a Beardmore-Caprotti poppet valve triple-expansion engine. The compact cylinder arrangement and neat valve gear mechanism will be noted. The employment of double-beat poppet valves, using separate valves for the inlet and exhaust processes, has certain thermodynamical advantages apart from those discussed earlier in the paper The periods of release and compression are maintained constant for varying degrees of cut-off. In this way clearances can be small and more work is obtained from the expanding steam than when slide valves are employed.

Finally, the writer wishes to acknowledge his indebtedness to the Editor of The Marine Engineer for permission to use certain of the illustrations which accompany this paper.

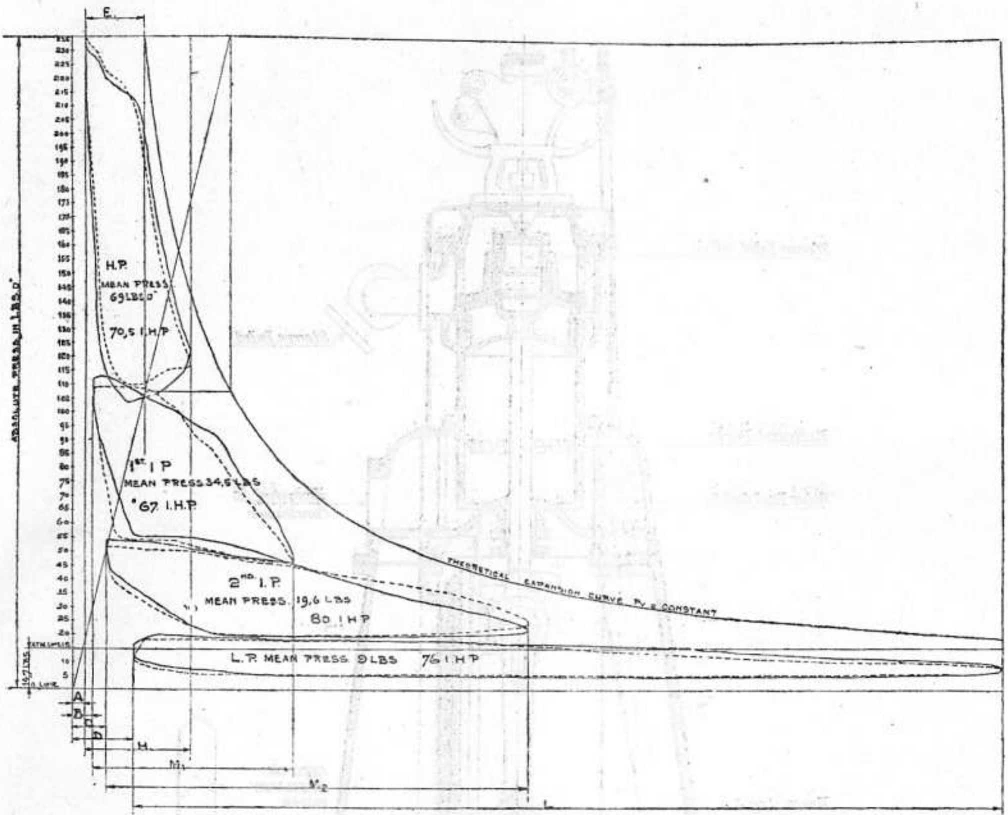

E
H.P.
MEAN PRESS
70,5 I.H.P
1st I P
MEAN PRESS 34,5 LBS
67 I.H.P.
2nd I.P.
MEAN PRESS. 19,6 LBS
80 I H P
L.P. MEAN PRESS 9 LBS
76 I.H.P
THEORETICAL EXPANSION CURVE PV = CONSTANT
A
B
C
D
H
M1
M2
L

Fig. 1.

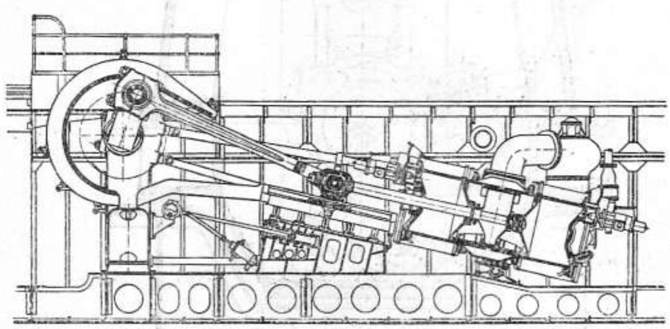


Fig. 3.

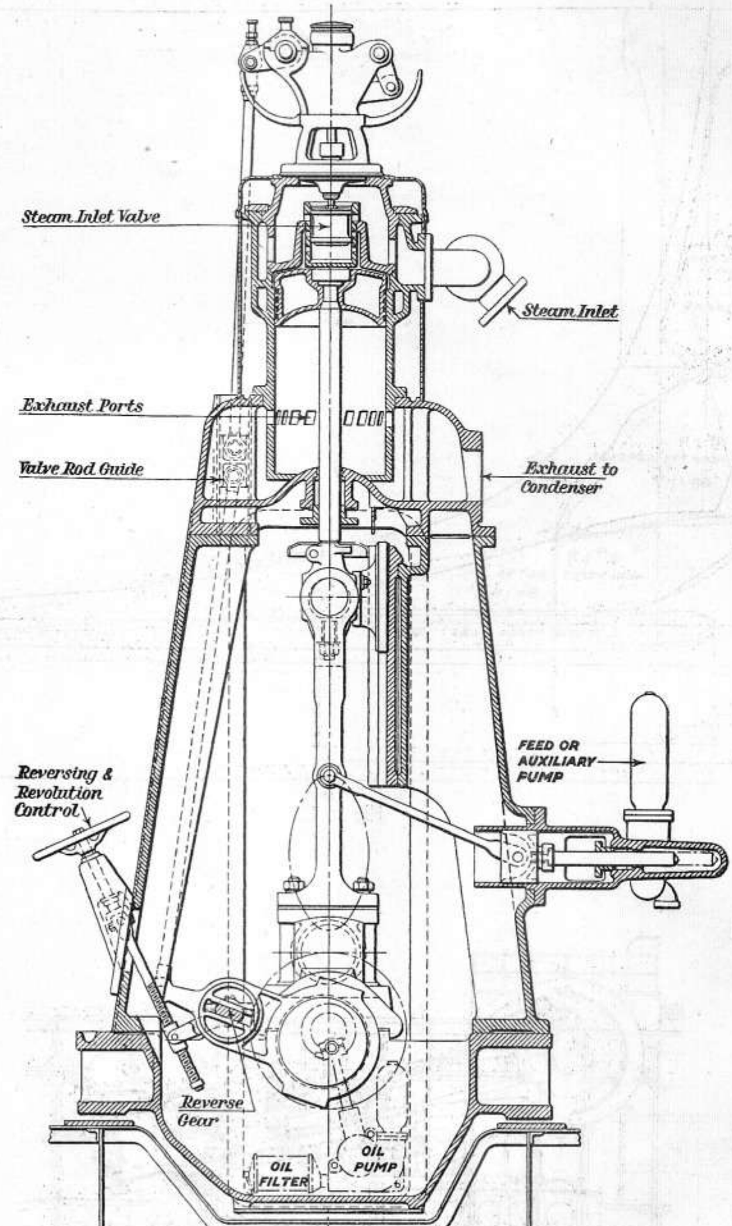

Steam Inlet Valve
Steam Inlet
Exhaust Ports
Valve Rod Guide
Exhaust to Condenser
FEED OR AUXILIARY PUMP
Reversing & Revolution Control
Reverse Gear
OIL FILTER
OIL PUMP

Fig. 2.

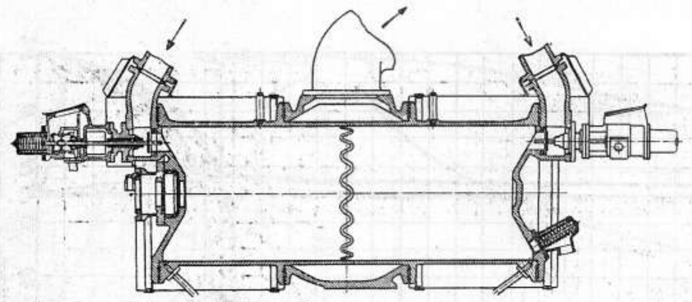


Fig. 4.

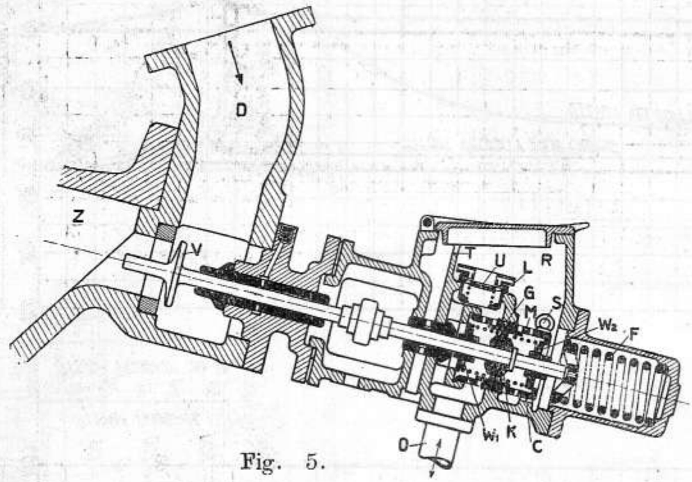


Fig. 5.

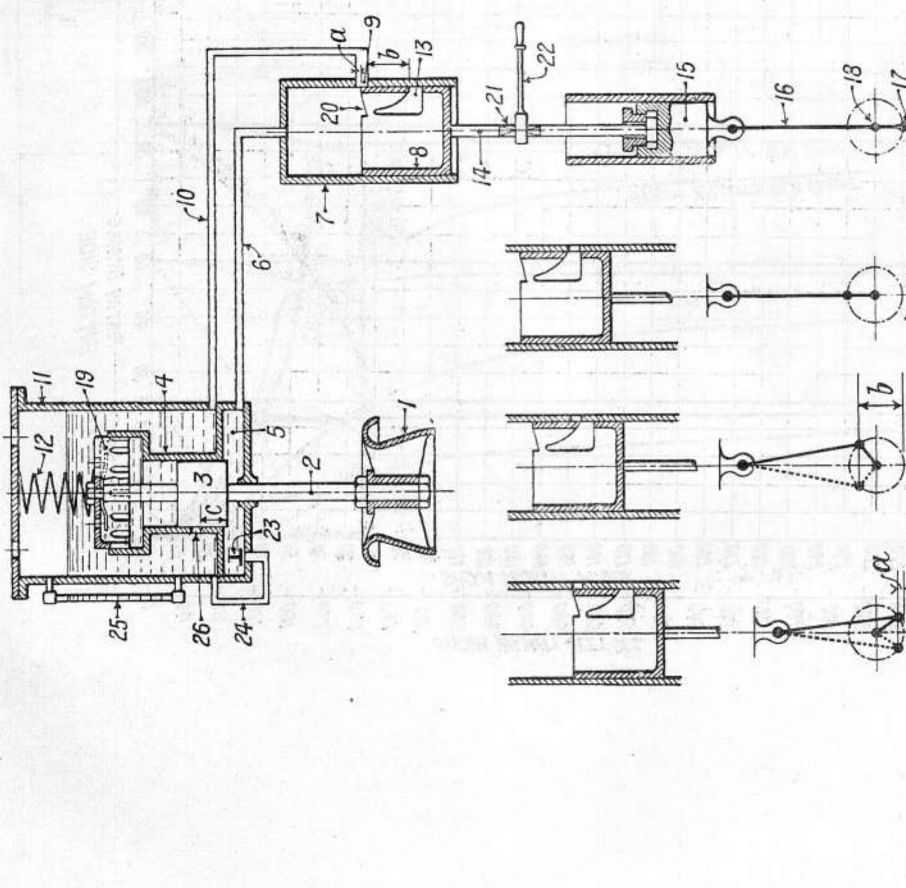


Fig. 6.

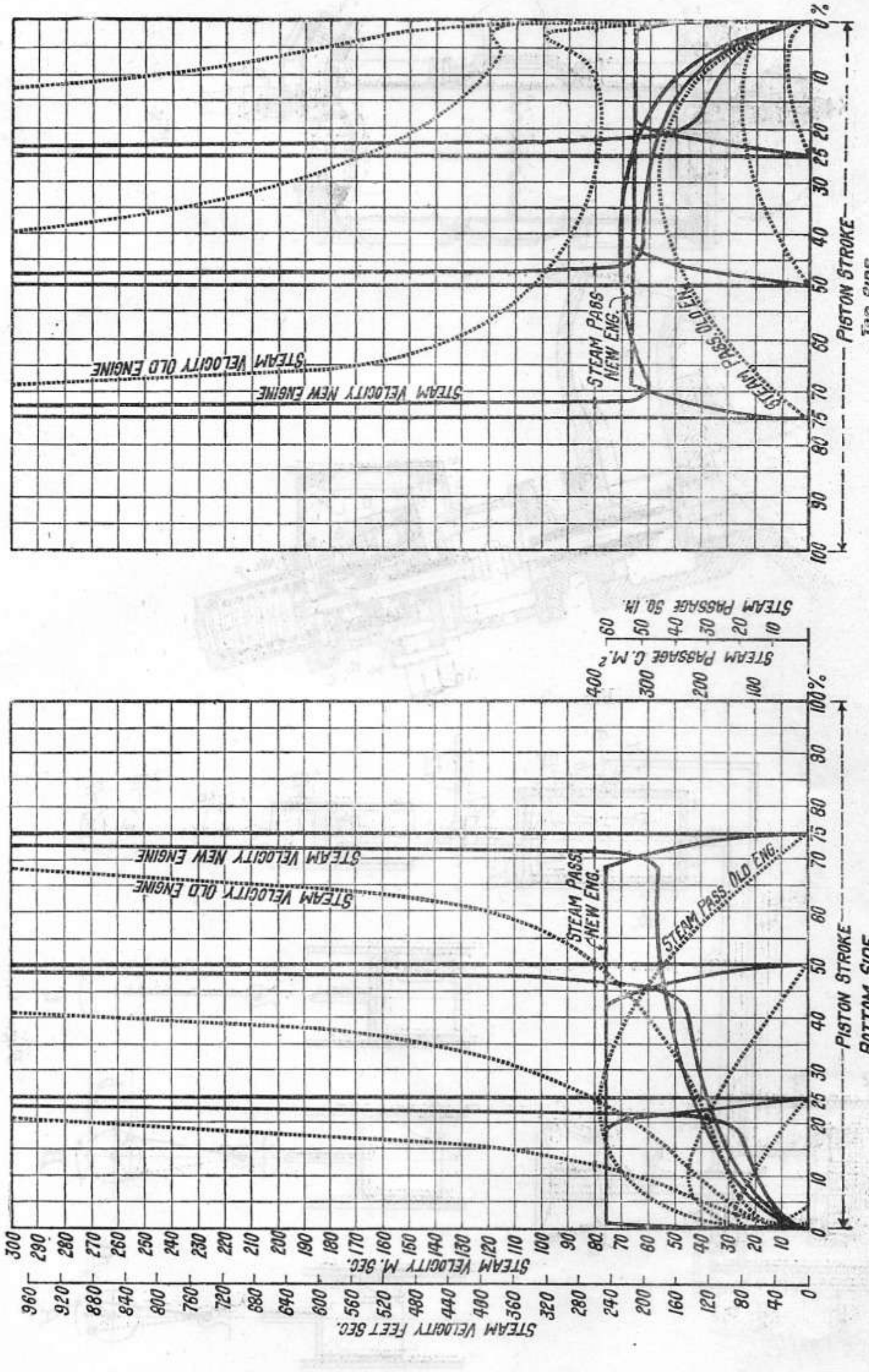

STEAM VELOCITY OLD ENGINE
STEAM VELOCITY NEW ENGINE
STEAM PASS. NEW ENG.
STEAM PASS. OLD ENG.
PISTON STROKE
TOP SIDE
BOTTOM SIDE
STEAM PASSAGE SQ. IN.
STEAM PASSAGE C.M.²
STEAM VELOCITY M. SEC.
STEAM VELOCITY FEET SEC.

Fig. 7.

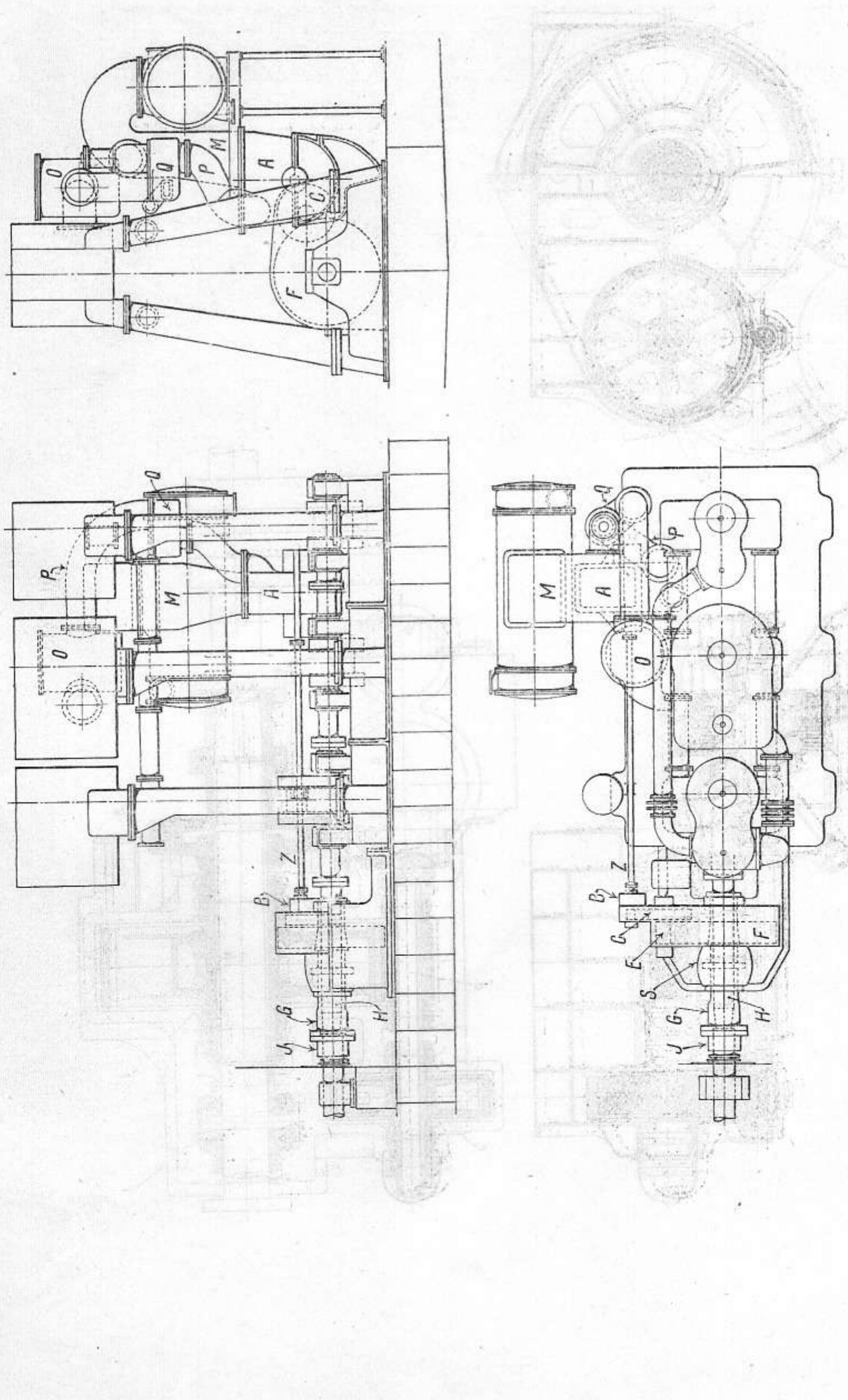


Fig. 9.

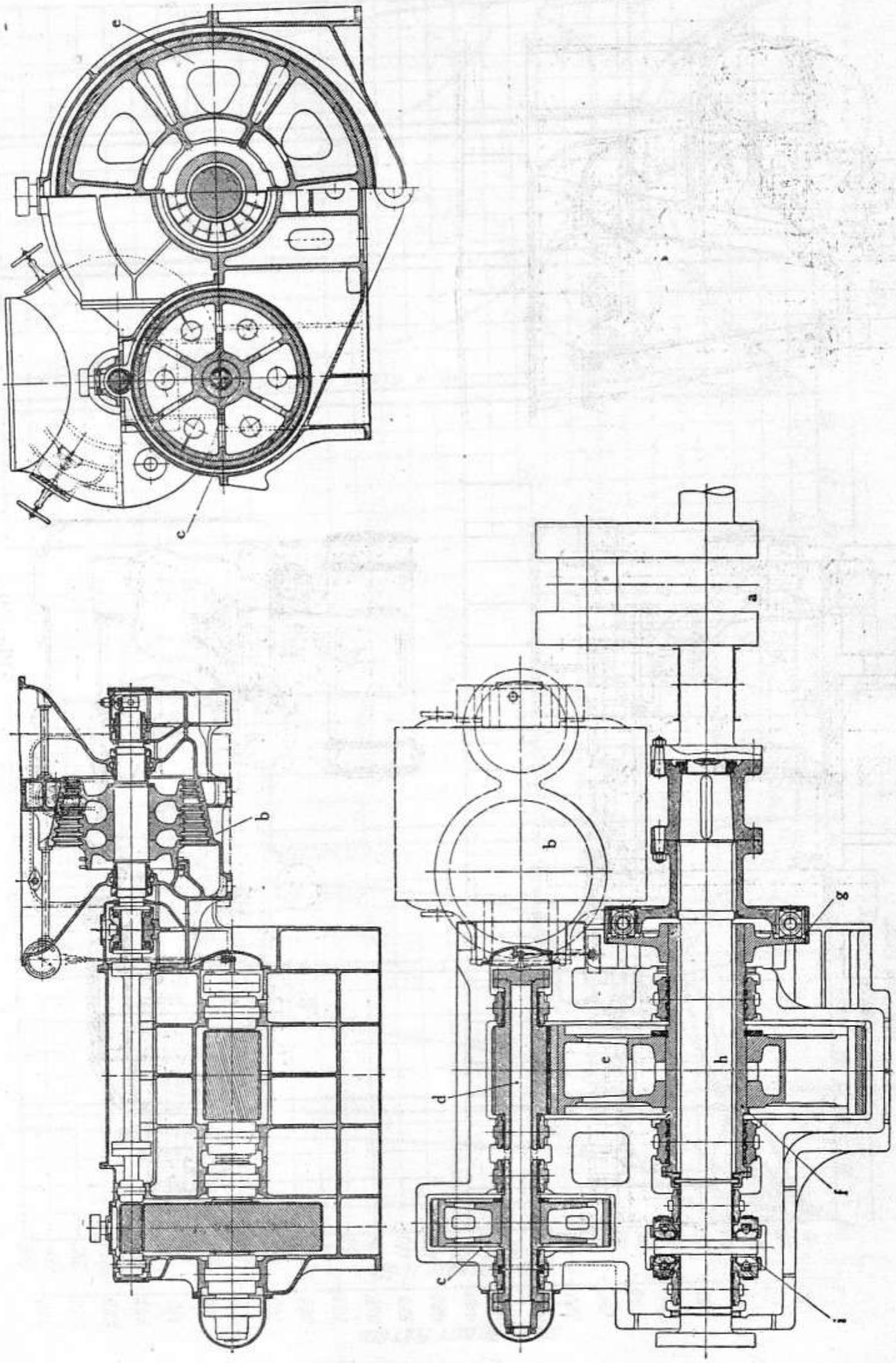


Fig. 10.

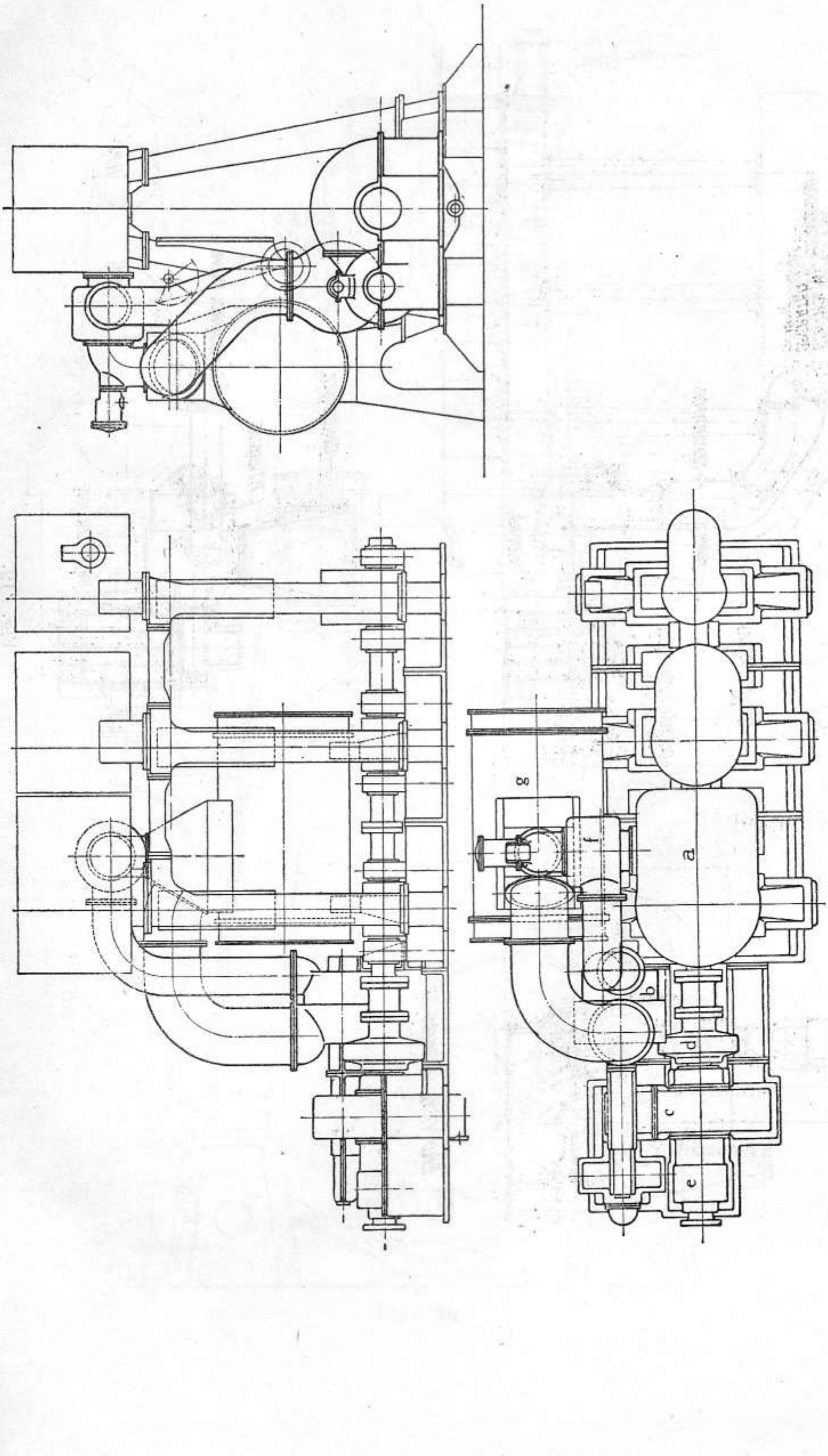


Fig. 11.

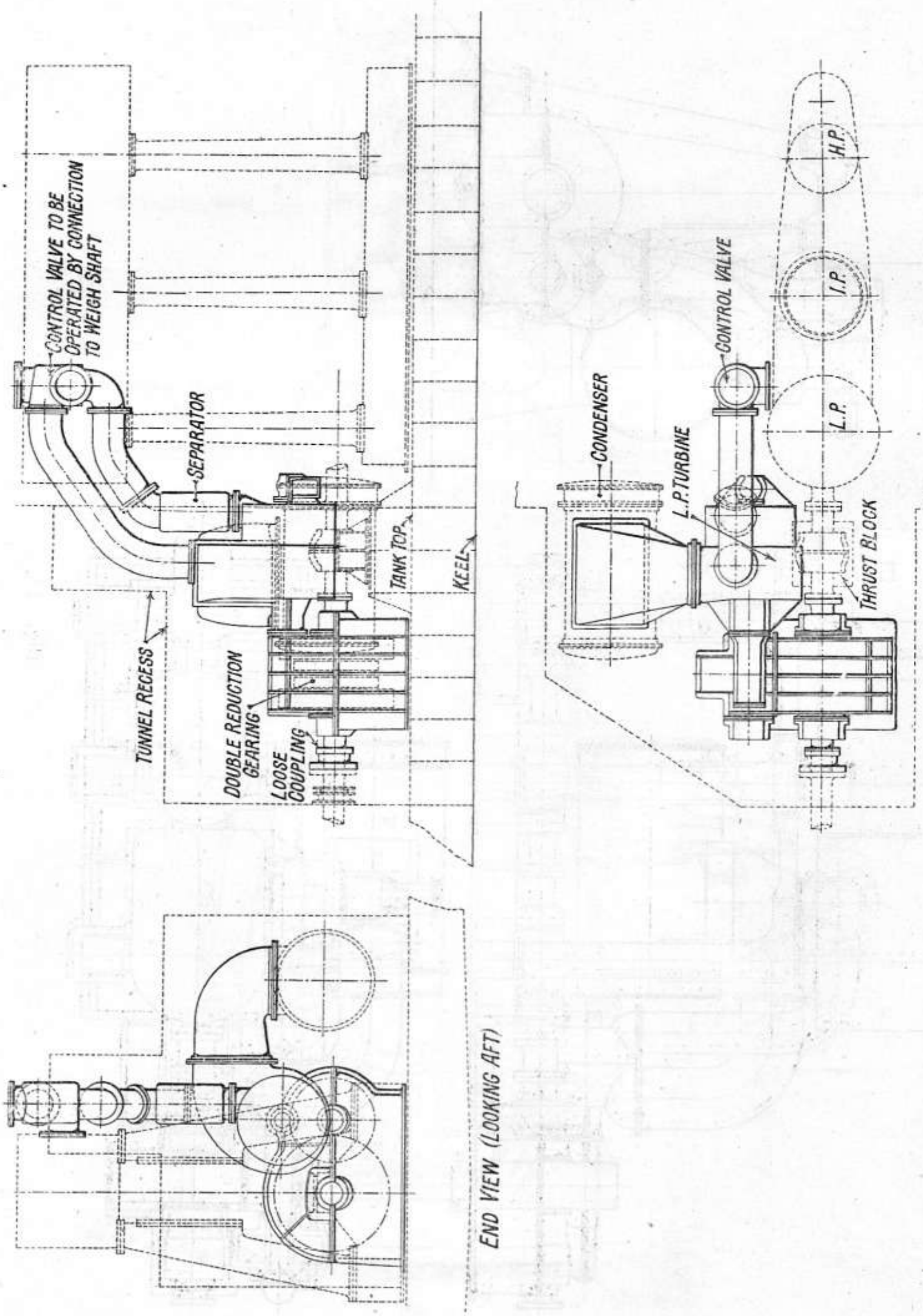

CONTROL VALVE TO BE OPERATED BY CONNECTION TO WEIGH SHAFT
SEPARATOR
TANK TOP
KEEL
TUNNEL RECESS
DOUBLE REDUCTION GEARING
LOOSE COUPLING
END VIEW (LOOKING AFT)
CONDENSER
L.P. TURBINE
CONTROL VALVE
H.P.
I.P.
L.P.
THRUST BLOCK

Fig. 12.

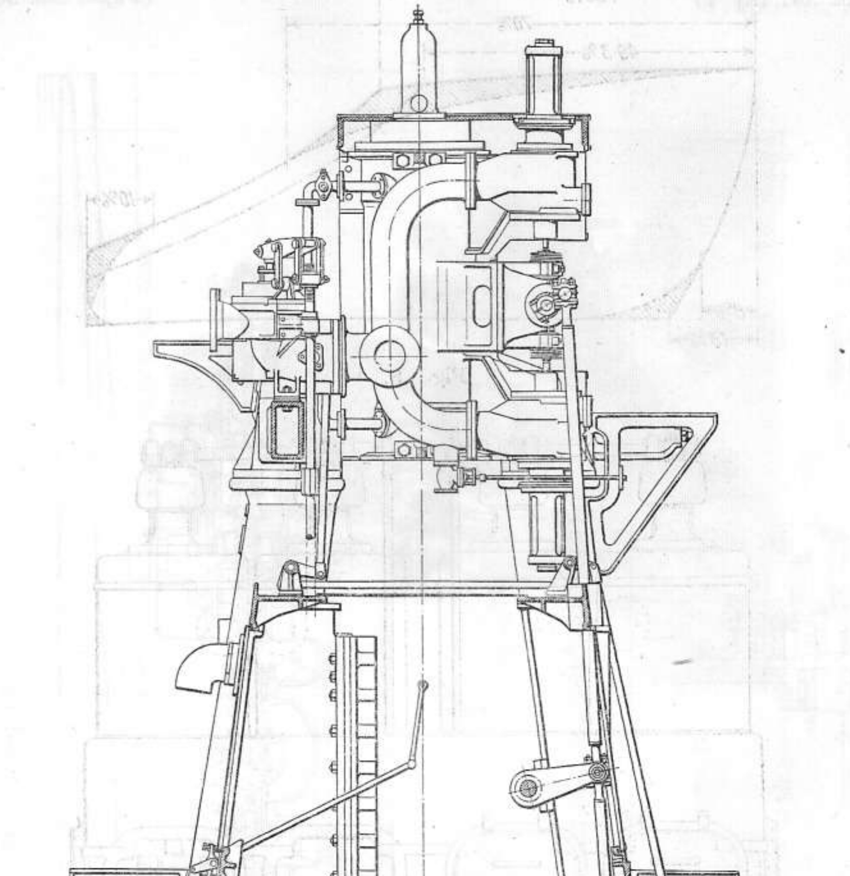


Fig. 13.

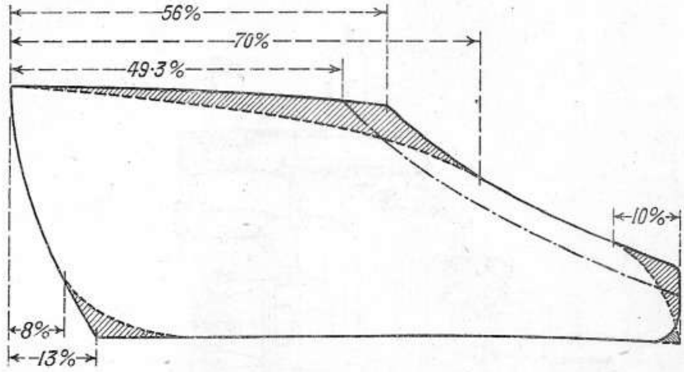


Fig. 8.

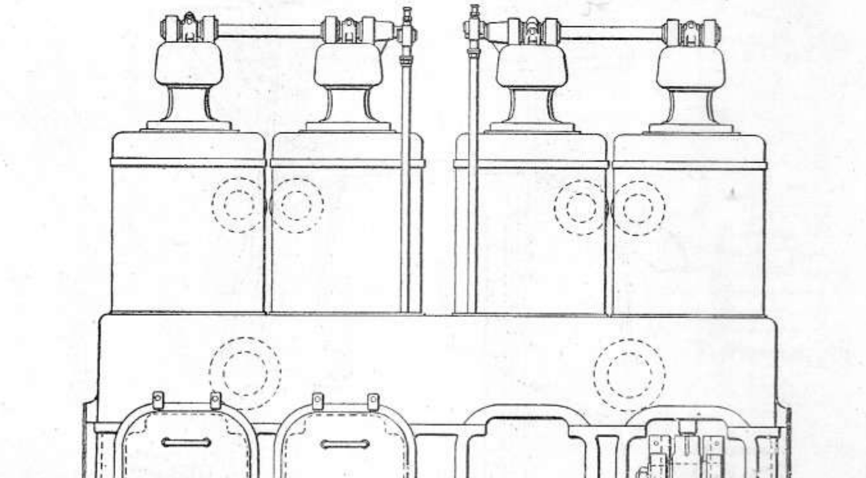


Fig. 14.

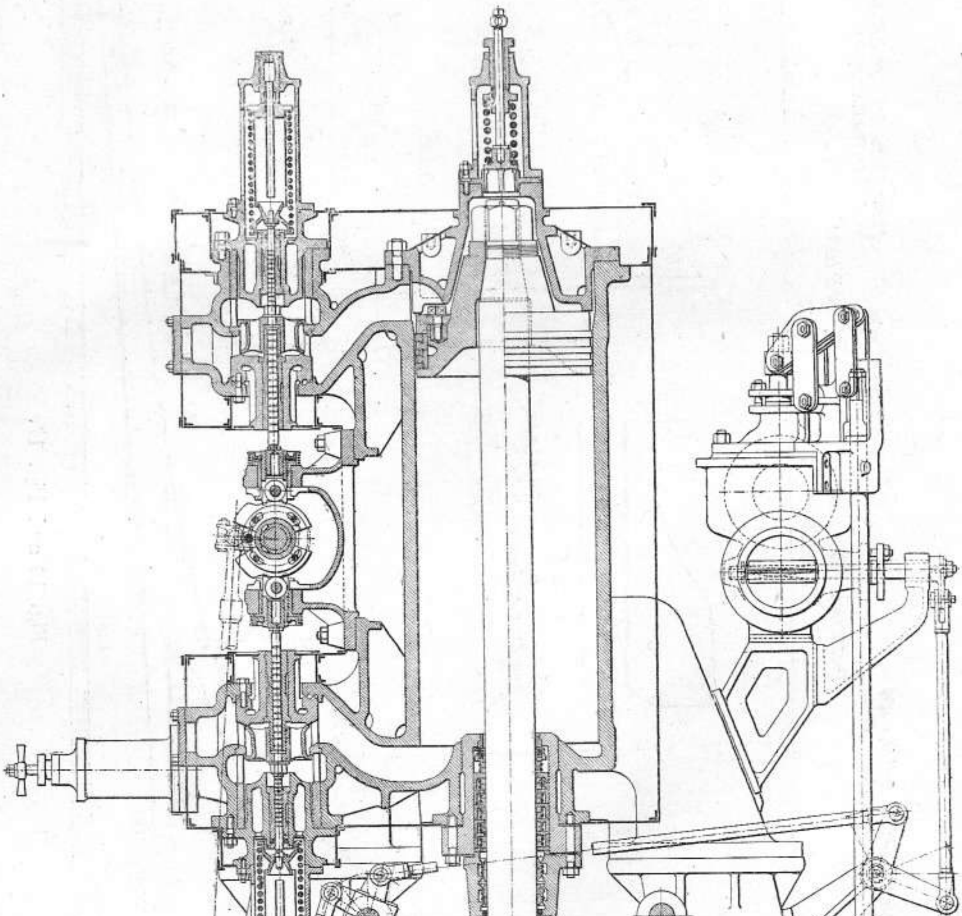


Fig. 15.

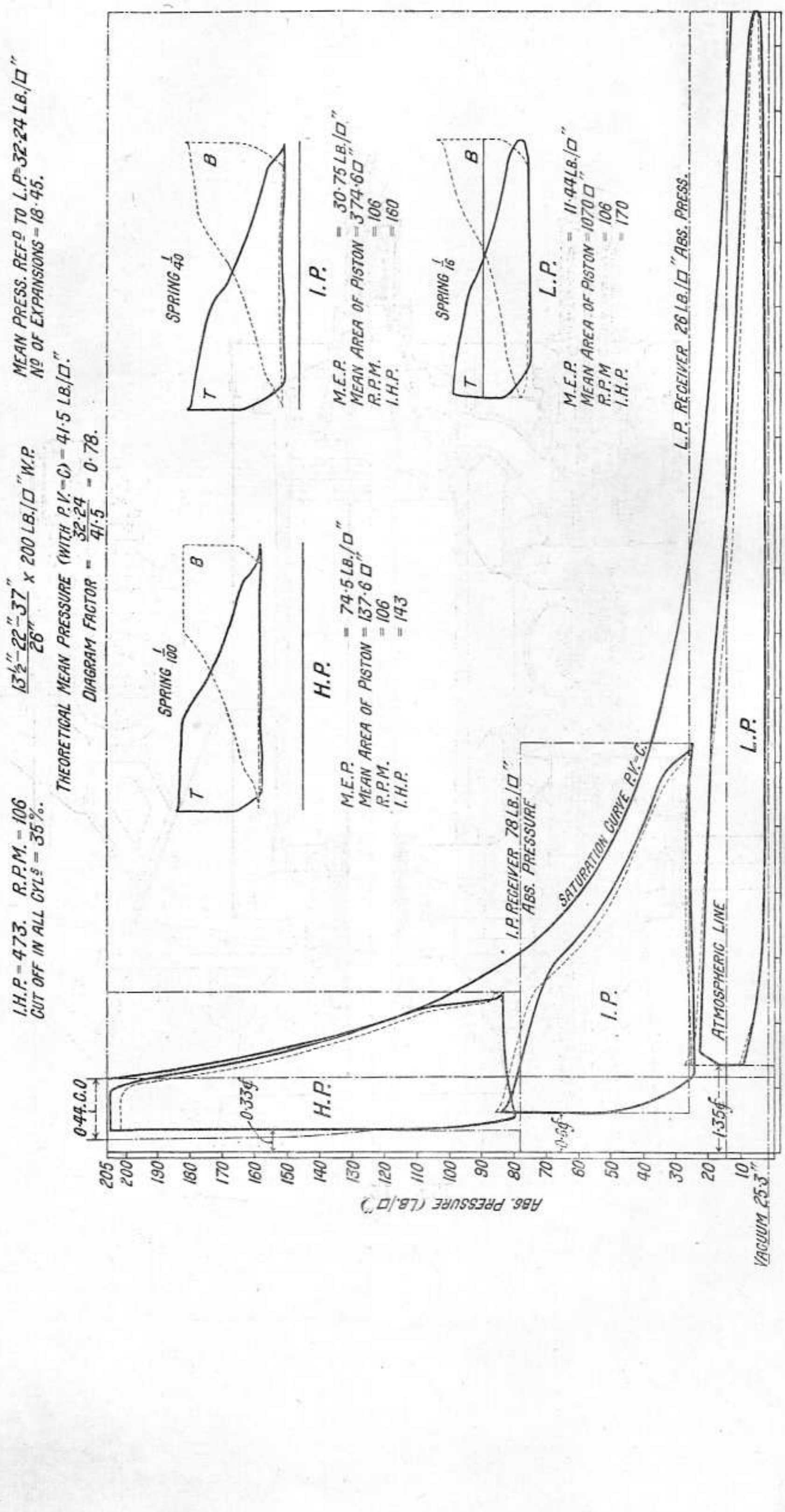


Fig. 16 and Fig. 17.

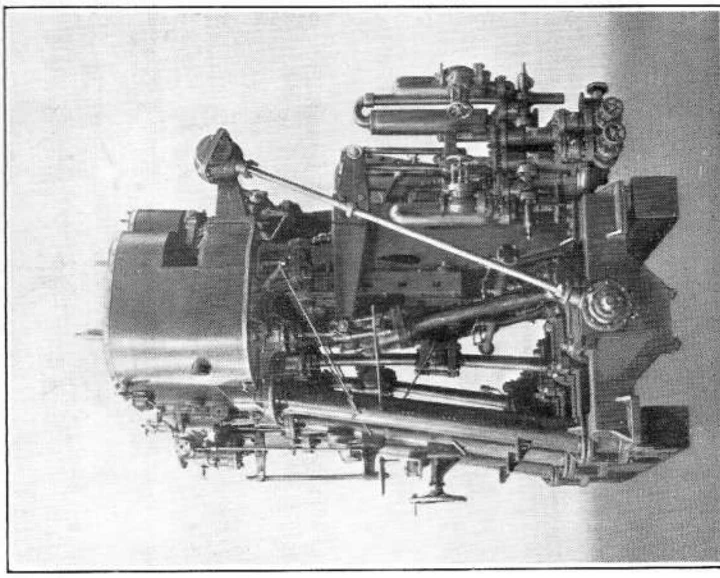


Fig. 18.

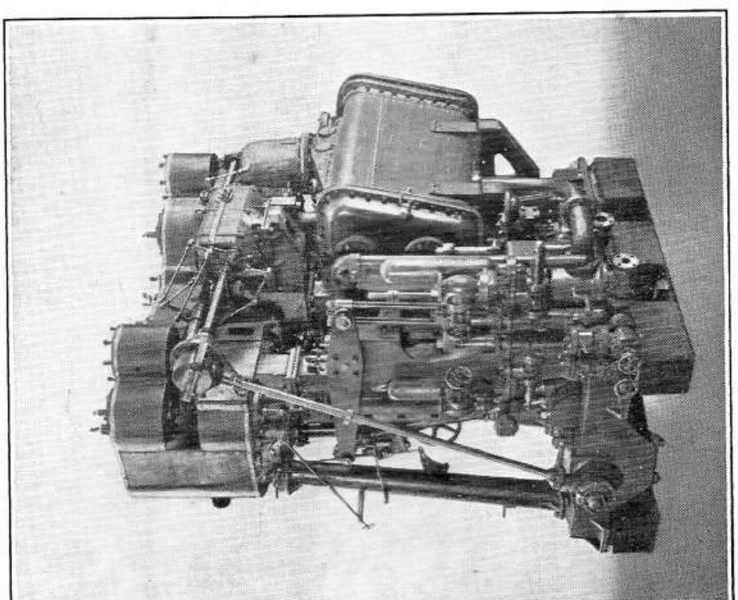


Fig. 19.

Propeller Design Developments.

(Paper No. 325)

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1. Introduction.

During the present century, progress in propeller design has resulted mainly from two lines of investigation.

In the first place, the method of experiment with models of propellers initiated by Froude something like fifty years ago has been pursued by a number of experiment tanks, so that there is now available much model data dealing with propellers.

In the second place, the development of the airplane has perforce resulted in a rapid development of airplane propellers and shed a good deal of light upon propellers in general.

No engineer can regard without deep respect the insignificant-looking instrument which directly propels an airplane at from 100 to more than 200 miles per hour and indirectly sustains in the air from a few hundred pounds to many tons.

2. Methods of Reduction of Model Experiment Results.

Let us consider first methods by which results of experiments with models of propellers are reduced to a form enabling them to be utilized for the design of full-sized propellers.

Such methods are all based upon the application of Froude's Law, or more generally, the Law of Comparison, to the results of model tests. This is too well known to require detailed discussion, but it should be pointed out that in some cases the Law of Comparison requires modification from the form used by Froude by reason of what has, of late years, come to be called "scale effect," which is the effect of viscosity. While scale effect does apply to propellers, it does not seem to have much influence for cases that occur in practice, except at slips below those commonly used and for models smaller than should be used.

Fortunately for present purposes, as we are dealing with results from models of reasonable size under conditions corresponding to those under which full-sized propellers operate, scale effect, while doubtless present and of great scientific interest, does not seem sufficiently important quantitatively to materially affect engineering conclusions drawn from the Law of Comparison in the Froude form.

3. *Importance of Reactions between Ship and Propeller.*

It has become more and more evident of late years that in marine work the mutual reaction of ship and propeller are vital factors in a given case.

We may know exactly how a propeller would perform in undisturbed water, but when propelling a ship it is working in water disturbed by the ship, and moreover, in addition to developing thrust at the thrust block, it acts through the water upon the ship's hull.

Then, in practical cases, the action of a ship's propeller in undisturbed water is modified by the ship's wake and by its action through the water upon the ship's after body. The latter effect is a virtual increase of resistance and is called Augment of Resistance or Thrust Deduction.

4. *Consideration of Propeller apart from Ship Coefficients.*

The problem is made easier if attacked in detail, hence consideration will first be given to the propeller proper apart from the ship. For this there is available a mass of model experiment data.

The first step is to express the results of model experiment in dimensionless form or by dimensionless coefficients, that is to say, by coefficients that do not change with size.

There are a large number of such coefficients and it is very important for practical purposes that we make a wise choice among them. There is one dimensionless quantity which we must constantly use. It is "Pitch Ratio," or the value of Pitch divided by Diameter for the propeller, and will be denoted by a. For propellers of uniform face pitch, the pitch ratio is constant. For others it may vary at every radius or every point, but even for such propellers we may adopt a single pitch (and hence pitch ratio) which for that propeller adequately expresses the pitch concept.

Aeronautical propeller designers seem to prefer to express the pitch concept by a pitch angle. This varies with radius, so it is customary in aeronautical work to express the pitch concept by the pitch angle at some definite fraction of radius, usually somewhere between $\frac{2}{3}$ and $\frac{3}{4}$ of the radius.

A second dimensionless quantity which results from pitch is slip ratio, slip per cent, or slip.

This ratio is obtained thus. From the advance, V_P, which the propeller would make in one revolution or one unit of time if it were revolving in a solid nut, deduct the corresponding actual advance of the propeller, V_A. Then $\frac{V_P - V_A}{V_P}$ is slip ratio.

In model experiment work it is usually convenient and best to plot results of test of a single propeller upon slip, but when we wish to plot group or family results for use in design, slip proper is not nearly so desirable as the basic coefficient B deduced below and the coefficient δ which is a compound coefficient involving slip.

In any given case once the propeller is designed the slip for any condition is readily calculated but to avoid further complicating rather complicated diagrams I have not introduced it directly.

For design purposes, we need a basic dimensionless coefficient which follows at once from the data of a given case.

Now, for the most common case of design, the given data comprise:

1. The horsepower (P) to be absorbed or the horsepower (U) to be utilized.
2. The number of revolutions per minute (N) of the propeller.
3. The volocity of advance (V_A) of the propeller in knots with reference to the water in which it works.

Assuming density constant, the well-known function of the three variables above for a dimensionless coefficient is $\frac{N^2P}{V_A{}^5}$. This then is in theory a suitable basic variable but in practice it has certain objections. Its numerical values are so large and vary so from ship to ship that it is difficult to use a satisfactory and convenient scale in plotting. The first objection is cured by adopting as our basic coefficient the square root of the preceding expression, writing Basic Coefficient $= B = \frac{N\sqrt{P}}{V_A{}^{2.5}}$ or $= \frac{N\sqrt{U}}{V_A{}^{2.5}}$.

This form, has, too, the advantage of using N in the first power, a desirable feature.

The significant numerical values of B range from about unity to something under 50, which is satisfactory. However, if we use as abscissae B values upon a uniform scale, we still have the objection that if we use a scale which is as open as desirable over the range most common in practice, we have a diagram which is quite large, of which the major portion is seldom needed.

This difficulty could be cured by adopting $\sqrt{B}$ as our base, but then we should have to employ $P^{\frac{1}{4}}$ and $V_A{}^{\frac{5}{4}}$, expressions awkward to calculate. Instead of this, the diagrams in this paper are plotted upon a variable scale, the actual abscissae being $\sqrt{B}$. When this is done, we are still able to interpolate with accuracy ample for practical purposes.

Having settled upon B as denoting our basic coefficient, using subscripts as necessary to indicate whether it refers to P or U, to 3-bladed or 4-bladed propellers, we will need to plot upon B various curves of dimensionless quantities. One such quantity, of course, is efficiency. There are three other quantities that are more or less useful.

Our basic variable, B, involves N, P, and V_A. There is a fourth quantity of primary importance, namely, size, most conveniently expressed by diameter in feet, denoted by d. Our basic variable B does not involve d. The three other dimensionless coefficients we need each involve d, and also two out of the three quantities, N, P, and V_A. They are:

$$\delta = \frac{dN}{V_A}$$

$$\varDelta = \frac{1000\, a\, P \text{ (or } U)}{d^2 V_A{}^3}$$

$$C=\frac{aP\,(\text{or }U)}{d^2\left(\frac{pN}{1000}\right)^3}\quad \text{where } p=\text{pitch}=ad$$

It will be observed that δ is independent of P, while A and C are respectively independent of N and V_A. Also it will be found that different as they seem, A and C are functions of B and δ. Thus except for the numerical coefficients which are arbitrarily given convenient values, A is $\frac{B^2}{\delta^2}$ and C is $\frac{B^2}{\delta^5}$.

Also δ is really compounded of the two dimensionless quantities, Pitch Ratio, a, and Slip Ratio, s. For it will be found that $\delta=\frac{101^{\frac{1}{3}}}{a(1-s)}$

5. *Design Diagrams deduced from Model Experiments.*

Having determined upon the basic variable and the other dimensionless variables that are useful and all readily determined in the case of a given model propeller from results of its test, we need next to derive diagrams that deal, not with single propellers, but with groups of families of propellers. For this result, there is a good deal of data available. The form of the diagram, however, depends somewhat upon its purpose.

Some years ago (in a paper before the Society of Naval Architects and Marine Engineers, entitled "Comparison of Model Experiments in Three Nations"), I took results of published experiments with large numbers of models and showed that they were in sufficient agreement to enable diagrams (somewhat arbitrarily averaged) to be derived, representing what were called standard models for 3-bladed and for 4-bladed propellers.

For these models, there were plotted upon B as a base curves of efficiency, δ etc., for successive values of pitch ratio. These comparative curves were readily applied for design purposes, but if we are dealing with design only, it is found better to transform the data to a form even more readily utilized.

Upon our basic coefficients B as abscissae and values of pitch ratio—a—as ordinates, there are plotted in one diagram contours of values of efficiency of propeller e_p and of δ and in a second diagram contours of A and C.

Then for each family there are four contour figures, since we plot upon B_p for power P absorbed and B_u for power U utilized.

Three-bladed propellers constitute one family and 4-bladed propellers another, so for these standard or composite models we have in all eight diagrams numbered Fig. 1 to 8, inclusive.

Fig. 1 to 8 in addition to scales of B and a each have two sets of contours. They are thus somewhat more difficult to use in practice than if sixteen figures were used each with one set of contours. However, there are distinct advantage in compactness. The most important and useful figures are Nos. 1 3, 5, and 7. The coefficients of Figs. 2, 4, 6, and 8 could be derived from the others, but it is more convenient to calculate and plot them once for all.

It will be observed, however, in Figs. 4 and 8 that for large B values the determination of C_{u_3} and C_{u_4} is not close. In the rare cases for which close values might be needed they can be calculated from Figs. 3 and 7, since as already stated, C is proportional to $\frac{B^2}{\delta^5}$ and we have in Figs. 3 and 7 contours of δ for all values of B.

It should be remembered that Figs. 1 to 8 are derived entirely from model experiment by the Law of Comparison. Now the Law of Comparison is not exact, but experience has shown it to be a reasonable working approximation.

Further, even assuming the exactness of the Law of Comparison, Figs. 1–8 refer to propellers of certain characteristics, and, strictly speaking, for different characteristics we should have different sets of diagrams. However, the characteristics used for Figs. 1–8 are reasonably close to the average characteristics of marine propellers most commonly used in service so that they form a reasonable approximation to usual propellers. Large departures of course from average characteristics involve corrections, but the departures must be very large before the corrections become serious.

I give below some values of B for typical ships covering a wide range of characteristics and the pitch ratios of their propellers which are all good:

Ship	B Value	Pitch Ratio
1200 ton 35 knot destroyer—twin screws	7.5	1.11
500 ton 18 knot patrol vessel—single screw	20.5	0.72
20000 ton 21 knot battleship—twin screws	8.7	1.08
33000 ton 21 knot battleship—four screws	10.6	1.13
16000 ton 18 knot passenger and cargo—twin screws	9.2	1.03
11000 ton 13 knot collier—twin screws	17.5	0.97
14000 ton 11½ knot tanker—single screw	18.5	0.90

On comparing the actual pitch ratios of the above with the best pitch ratios indicated by Figs. 1 and 5, it will be found that the agreement is excellent. The maximum departure is in the case of the 4-screw battleship. Fig. 1 indicates 1.00 as the best pitch ratio for the B value of 10.6 while the actual pitch ratio is 1.13. The best propeller efficiency indicated in Fig. 1 is .678 and that for a pitch ratio of 1.13 is .669. This reduction is readily admissible if for instance it is desirable to reduce diameter. It is seen from Fig. 1 that for the B value involved the diameter for 1.13 pitch ratio is nearly 5 per cent less than for 1.00 pitch ratio.

Assuming for the present the correctness for practical purposes of Figs. 1–8, attention may be called to a few deductions from them.

Once we have fixed power, revolutions, and speed of advance, we have fixed B and hence the upper limit of possible propeller efficiency. We can approach this limit for a fairly wide range of pitch ratio and diameter, but we cannot exceed it.

This upper limit depends upon B and falls off steadily as B increases. In practical cases B increases with N, and the upper limit of propeller efficiency decreases with N.

6. Comparison between 3-Bladed and 4-Bladed Propellers.

It will be observed that Figs. 1-8 indicate a slightly greater efficiency for 3-bladed propellers than for 4-bladed. The difference is not material for the B values prevalent for cargo vessels, but is appreciable for the lower B values found for many naval and other fast vessels.

Naval propellers are nearly always 3-bladed and cargo vessel propellers 4-bladed, hence Figs. 1-8 are in reasonable accord with practice, but remembering the approximate and somewhat arbitrary methods of deriving Figs. 1-8, confirmation of their indications as regards this question of numbers of blades is desirable. In this connection attention is invited to Fig. 9, which shows curves of efficiency and δ (proportional to diameter) for 3-bladed and 4-bladed models tested at the U. S. Model Basin some time ago. Curves from Figs. 1 and 5 for propellers of the same pitch ratios and nearly the same other characteristics are also given.

It is seen that the special experiments confirm the composite model diagrams. They indicate a slightly greater efficiency for 3-bladed propellers, for low values of B and a very slight advantage for the 4-bladed propellers at high B values.

Moreover the agreement between the composite diagrams and the special models is good except for values of B too low to be encountered in practice. Particular attention is invited to the almost negligible effect upon efficiency and diameter of the 33⅓ per cent increase of area of propellers D and I over E and K.

Not very many years ago marine propeller designers were apt to attach much importance to small variations of area and shape of blade.

Such ideas have never been confirmed by model experiment and there seems no reason to suppose that full-sized propellers do not agree with models in this regard.

Figure 9 sustains present practice under which 3-bladed propellers are usually preferred by designers under conditions where B values are low and 4-bladed under conditions where B values are high. However, the differences for practical cases beyond B values of 10 or so are too small to be of serious practical importance. For four bladed propellers the turning moment variation has four cycles per revolution instead of three. If we run into synchronism with the ship's period and hence excessive vibration with 4 blades, we can usually cure it by using 3 blades and vice versa. We are not likely to encounter synchronism on a given ship with both 4 and 3 blades.

7. Propellers at Excessive Slip.

Before passing from the subject of diagrams derived from model propeller experiment, attention is invited to two special cases.

Figs. 1–8 do not extend beyond about 50 per cent slip, whereas sometimes, as in the case of tugboats, we find B values far beyond 50 corresponding to much greater slips.

Some years ago, Schaffran published results of experiments with rather small 3-bladed models at excessive slips, and upon converting these to the B δ basis, data is obtained to plot Figs. 10 and 11. These begin where Figs. 1–8 end as regards B values.

A uniform scale is used for B which accounts for the concavity of the curves of efficiency. In Fig. 10 is seen a special curve of $\dfrac{1000\ P}{a^2\, d^5\left(\dfrac{N}{100}\right)^3}$. It happens that this coefficient varies but little for each of the four pitch ratios and the curve plotted in Fig. 10 is an average for the four.

8. *Propellers above Surface of Water.*

A second special case is that of the propeller with blade tips out of water.

Fig. 12 shows curves of efficiency and δ for a few propellers of the characteristics indicated tested in an extreme condition with only $\frac{4}{10}$ of their diameter in the water.

While the maximum efficiencies indicated compare favorably with the maximum efficiencies of similar submerged propellers, they occur at much smaller B values (or much lighter total loads) and for each model propeller there was a breakdown point at which there was a sudden drop in efficiency and power absorbed.

There seems reason to think that for full-sized propellers the break would occur relatively earlier, and efficiencies realized would be somewhat less, but even so, the comparatively small loss of efficiency of the surface-breaking model propeller is remarkable.

9. *Reactions between Propeller and Ship.*

In all of the preceding, no account has been taken of cavitation which has not, so far as published results go, been successfully investigated by model experiment.

Deferring discussion of cavitation, I will take up now the reactions between propeller and ship.

The propeller works in the ship's wake, which has three components, the frictional wake, the streamline wake, and the wave-produced wake. The latter is sometimes negative, but never for single screws and rarely for twin screws strong enough to overcome the two former.

The result is that actual propellers usually work in a forward current or wake which varies irregularly over the propeller disk.

For the purposes of propeller design we undertake to deal with the wake by determining a uniform current which will produce the same effect upon the propeller as the actual disturbed wake.

This presents two difficulties. The uniform wake deduced from the effect of the actual wake upon the torque of the propeller sometimes differs from the wake deduced from the effect of the actual wake upon the thrust of the propeller. This results in a virtual change of efficiency of propeller and sometimes model experiments have indicated that this is material.

However, there seems no way to estimate this except by model experiment in each case, and without that we can only assume a uniform wake affecting torque and thrust alike.

The second difficulty arises from the fact that, owing to the different frictional coefficients of model and ship, there is no experimental method available for determining the ship wake.

However, we shall see that exact knowledge of wake is not essential to the design of a suitable propeller. Here again, as in so many other respects, a reasonable approximation is adequate for practical purposes.

If we denote the speed of our assumed uniform wake by wV where V is the speed of the ship and w the wake fraction, we have $V_A = V - wV = V(1-w)$.

Now V_A is the speed of advance of the propeller with reference to the water in which it is working, a vital factor, of course, in the performance of the propeller and in our dimensionless coefficients characterizing the propeller. Hence the importance of w is obvious.

The thrust deduction or suction of the propeller upon the after part of the ship is usually expressed as a fraction of the thrust delivered to the thrust block, the fraction being denoted by t. It will have been observed that in dealing with propeller efficiency in Figs. 3 and 7 it was based upon U, the "power utilized." This is different from E, the effective or net or towrope horsepower absorbed in overcoming the resistance of the ship unaffected by the propeller.

U and E are connected by the well-known relation

$$U = E\frac{1-w}{1-t}$$

In general when we undertake to design a propeller we must know N, the desired revolutions per minute and we must have an estimate of P, the shaft horsepower, or E, the effective horsepower, of the ship for the expected speed of ship, V.

If P is the given power, we must estimate V_A from known V and estimated w. Then we can calculate B_p and enter the appropriate diagram.

If we start with E, we must estimate w and t and then U by the formula $U = E\frac{1-w}{1-t}$, and V_A by the formula $V_A = V(1-w)$.

This enables us to calculate B_u and then enter the appropriate diagram.

The above indicates the necessity for satisfactory estimate of w and t.

There are so many features of a vessel which affect the wake and thrust deduction coefficients that so far there have been put forward no satisfactory formulae dealing with these fractions with accuracy. Fortunately, however, in this case, as in others, for the purposes of propeller design minute accuracy is not essential.

Taking ships as they are, it would seem that the major determinant of both wake and thrust deduction is fullness or block coefficient. The accepted method of determining the wake of the full-sized ship is to test the model self-propelled, estimate from the results (in comparison with the propeller or propellers tested in the open) the average wake for the model, and then apply this model wake, or model wake slightly reduced, to the full-sized ship.

There are various sources of error in this process, the method itself not being exact in theory. Until the matter is put upon a more nearly exact basis, it would seem as satisfactory as any other method to estimate the wake from the characteristic curves for the composite propeller models and the power of the full-sized ship. It is easy to determine thus a nominal wake which, assuming that the actual ship's propeller has the same characteristics as the composite model propellers, would cause the propeller to absorb the correct horsepower at the actual revolutions.

Thus in any given case we have for the full-sized propeller the diameter d, the pitch p, the pitch ratio a, the power delivered to the propeller P, the revolutions per minute N, and the speed of the ship V. We need to determine V_A.

From the known data, calculate $C = \dfrac{aP}{d^2\left(\dfrac{pN}{1000}\right)^3}$

Figs. 2 (3-bladed) and 6 (4-bladed) give contours of C. Enter the appropriate figure on the line corresponding to the known a, and follow across until we reach the point giving the calculated value of C. This determines the appropriate value of B, δ, and A. Now $\delta = \dfrac{Nd}{V_A}$ and when δ is determined, since N and d are already known, V_A follows at once, and $V_A = V\,(1-w)$; hence w is determined.

This method has been applied to more than 150 vessels, single- and twin-screw, and the resulting nominal wakes, or composite model wakes, while somewhat erratic, are found to vary broadly with the block coefficient. These composite model wakes are tabulated below:

Block Coefficient	Wake Fraction for Composite Propeller Curves	
	Twin Screw	Single Screw
.50	−.038	.230
.55	−.021	.234
.60	.007	.243
.65	.045	.260
.70	.091	.283
.75	.143	.314
.80	.200	.354
.85		.400
.90		.447

We need to estimate t before we can determine in a given case the value of $\dfrac{1-t}{1-w}$ connecting E and U and usually called the Hull Efficiency. This is a misnomer because it is not an efficiency, being frequently greater than unity. Hull Coefficient would be a more correct expression.

Now, t, the thrust deduction coefficient, varies generally in the same way as w. For a reasonable approximation to its value for ships without special stern arrangements we may take $t=w$ for twin screw vessels and $t=.3b$ for single screw arrangements. During the last few years however various patent sterns and special designs have come into use whose principal value seems to be in a reduction of t and to a lesser degree an increase of w. If we neglect in the design these charges we provide a margin. We can determine by model experiment values of w and t for the model with special stern, but there is some reason to doubt whether the favorable model results are always fully borne out for the full-sized ship. Pending complete clarification of this subject, it seems advisable, if we assume the model indications will be fully borne out, to allow a margin somewhat larger than otherwise.

10. Approximation of Results to Exactness.

In what has gone before, a distinction has repeatedly been made between exact results and the results sufficiently exact for engineering purposes, and it has been stated repeatedly that certain approximations were sufficiently close.

Let us consider now in more detail a few of these approximations. We have seen that the power absorbed by a given propeller varies as the coefficient independent of size, as the fifth power of the diameter, and as the cube of the revolutions. The fact that the horsepower varies as the fifth power of the diameter means that if we are in error as much as 10 per cent with respect to the power absorbed by a given propeller, the corresponding error in the diameter is about 2 per cent. Moreover, it is evident from Figs. 1, 3, 5, and 7, that if the propeller as originally designed is somewhere near the best diameter for the conditions, a material variation above or below this best diameter affects the efficiency very little. A 2 per cent variation is immaterial as regards efficiency.

The composite model diagrams are for a fixed mean-width ratio and a fixed blade thickness fraction. In dealing with Fig. 9, attention was called to the slight effect of variation of area in the case of the 4–bladed propellers. This is general. The effect of an increase of area, for instance, beyond, say 25 per cent mean-width ratio, is to slightly increase the power absorbed and to slightly decrease the efficiency. The differences in most practical cases are within the limits of error of tests.

When we come to blade thickness, we find that model experiments indicate that the thicker the blade, the greater the power absorbed, but as a rule this somewhat reduces efficiency. The effect of blade thickness is greater the narrower the blades, and for fine pitch ratios, if we narrow the blades and increase the thickness as necessary to maintain the same strength, we find the narrow blades in spite of their reduced area actually absorb more power in the model. However, such narrow thick blades cannot safely be used in full-sized marine work without danger of breakdown.

Model experiment appears to indicate a mean-width ratio of .25 for a blade thickness fraction of .05 is about the most desirable combination for 3-bladed propellers. For 4-bladed propellers, the combination of mean-width ratio .20 and blade thickness fraction .04 produces similar blades, but there is some question whether, even for 4-bladed propellers, it would not be desirable to use blade widths nearer .25 than .20.

Coming now to the effect of variation of blade thickness upon diameter, analysis of experiments made at the Washington Model Basin about 20 years ago appears to indicate that, on the average, the effect of increasing the blade thickness fraction 20 per cent for a blade of .25 mean-width ratio and .05 blade thickness fraction would be to enable it to absorb about 3 per cent more possible power for coarse pitches and 5 per cent for fine pitches. This would indicate for such thickening of blades a reduction of diameter of 1 per cent at most. It follows that variations of blade thickness from the values of Figs. 1-8 do not seriously affect conclusions drawn from them.

The question of blade thickness to be used in a given case is readily dealt with by well-known methods. There appears to be some tendency on the part of classification societies to deal with this matter by rules rather than by passing on individual cases. In Appendix II, I give the current rules of the American Bureau of Shipping, which are about the most comprehensive of any classification society.

11. Cavitation.

I come now to the one circumstance which does not enable us to use the composite diagrams Figs. 1-8 with confidence in every case. This is cavitation, a phenomenon known for 30 years but not yet fully understood. So far, it has never been dealt with satisfactorily by model experiment, though there is reason to believe that this reproach to the research engineer will be removed at an early date. Barnaby, in the early 90's, found in connection with a destroyer that beyond a certain speed the propellers appeared to lose their grip, as it were, a result which he ascribed to cavities forming in the water about them. We know now that these cavities are vacua of a high order, and it is reasonably certain that they occur on the driving face of the blade as well as the back. Barnaby cured the difficulty by using wider blades and laid down the dictum that thrust upon the propeller per square inch of projected area greater than 11¼ pounds was liable to produce cavitation.

Most designers have believed also that a tip speed should not be exceeded, and some years ago a tip speed of 10,000 or 11,000 feet per minute was regarded as about the limit. Barnaby's wide blades still form the only accepted preventative of cavitation, although his limits have been exceeded. In a recent case of four screws absorbing about 50,000 horsepower each at more than 34 knots, the thrust per square inch of projected area was over 17½ pounds, and the tip speed nearly 15,000 feet per minute.

The adoption of wide blades, while it may prevent cavitation and hence breakdown of the propeller, necessarily reduces propeller efficiency owing to the additional friction of the extra wide blades. It is to be hoped that when this field can be explored by model experiment non-cavitating blades of less abnormal surface may be developed. Probably too it will be found that cavitation or breakdown is liable to occur under some conditions well below the Barnaby limits.

12. Comparison between Airplane and Marine Propellers.

Analysis of some of the very extensive research which has been carried on during the last few years in connection with models of air propellers, and recently (by the National Advisory Committee for Aeronautics), upon full-sized propellers is not only interesting of itself, but sheds light upon some problems of marine propellers. However, there is one radical difference between propellers working in air and in water. The former work under such a head of air that the behavior of the fluid around them agrees much more closely with theoretical streamline motion than can possibly be the case with marine propellers. Airplane propellers work under a fluid head of between 5 and 6 statute miles, whereas the marine propeller works under a head, including the head due to air pressure, of from 40 to 60 feet only. It is a result of the difference of conditions above that we find air propellers usually 2-bladed with narrow blades, even when they are relatively quite thick. These are more efficient than the 3- and 4-bladed marine propellers found in practice, but it is not practicable to use 2-bladed propellers for marine work, not only because they would promptly break down if made narrow bladed and of the thickness requisite for strength, but because 2-bladed propellers of any width working in the irregular ship's wake are very bad from the point of view of vibration. However, model experiment on 2-bladed propellers working in water has shown that *in the model* where questions of cavitation, vibration, and strength are not involved, the maximum efficiency of a narrow, thin 2-bladed propeller is well above 80 per cent, as for air propellers.

13. Durand and Lesley's Model Experiments.

A most extensive series of experiments with air propeller models have been made by Dr. W. F. Durand and Professor E. P. Lesley, at Stanford University, in California, under the auspices of the National Advisory Committee for Aeronautics. This work extended over several years and was covered in a number of reports, but is summarized and reviewed in Report No. 141 of the National Advisory Committee for Aeronautics, published some years ago. Something like 150 models were tested in all, but Report No. 141 deals only with 88 of the most significant ones. These models were all 3 feet in diameter, tested in a 5.5 foot circular open jet wind tunnel, usually at a speed of about 40 statute miles per hour though lower speeds were used for excessive slips.

Figure 13 shows on a small scale the features of the various propellers tested.

There were two forms of blade contour used, designated F_1 and F_2, and two blade areas for each contour, designated A_1 and A_2. This gave four families of propellers, each of which was tested with six values of uniform pitch ratio of the driving face, these values being .3, .5, .7, .9, 1.1, and 1.3. Moreover, for models of each form and area, a number of different forms of section of blade were used, as indicated in Figure 13.

The S_1 section had a straight face, so that the driving face was of the ordinary helicoidal form. The S_2 had a hollow face, S_2 designating the maximum hollow. The S_3 section also had a hollow face, midway between S_1 and S_2. The S_4 had a convex driving face, as shown. These were the main sectional variations. There were four more sections, S_5 to S_8, all used with F_2A_1 models and nominal pitch ratio of .7, which differed from the S_1 section only in the position of the maximum thickness. For the S_1 section, the maximum thickness occurred at ⅓ of the width from the leading edge. For S_5, S_6, S_7, and S_8, the maximum thickness occurred respectively at .17, .25, .41, and .49 of the width from the leading edge.

Durand and Lesley reduced their results upon the basis of various coefficients. One of their coefficients, C_3, is very close in form to the B coefficient used herein for marine work. If we denote by ρ the density of the air or weight per cubic foot divided by g the acceleration due to gravity, by n the revolutions per second, by v the speed in feet per second, and by F the foot pounds per second, $C_3 = \frac{n^2F}{\rho v^5}$.

The units in this are not engineering units, but were adopted by Durand and Lesley because the numerical values of the resulting coefficients would be the same in any constant set of units, metric or English.

When considering Durand and Lesley's results, it is convenient to plot them upon their coefficients. But the coefficient C_3 is even worse in its variation than the coefficient B, used for marine purposes. However, this feature is controlled by plotting results upon log C_3 as abscissa. Of course, efficiency is plotted as before.

For study and comparison of Durand and Lesley's results, the only other quantity we need to take up is the diameter coefficient. Durand used for this $\frac{v}{nd}$, where v is speed in feet per second, d is diameter in feet, and n is revolutions per second. This coefficient, as we have already pointed out, is a compound coefficient, depending upon slip and speed, and is very commonly used in aeronautic work. However, it has the objection that the diameter occurs in the denominator, so in plotting Durand and Lesley's results, the reciprocal, or $\frac{nd}{v}$, will be used.

14. Effect of Variation of Blade Types.

When on the basis above described we plot the results given by Durand and Lesley for the four families of S_1 section shown in Figure 13, we find that, with minor eccentricities and variations, the curves for the four families are so close together that when we are regarding them from the engineering point of view, there seems no advantage in considering them separately. Accordingly, in Figure 14, there is shown the average of Durand and Lesley's results for these four families. It will be observed that for some reason, the curve for .7 pitch ratio is somewhat out of place, the efficiency being somewhat higher than it should be. However, broadly speaking, these curves are quite consistent and may be regarded as applicable with reasonable accuracy to the type of propeller experimented with.

In Figures 15 to 19, the standard curves of Figure 14 are reproduced in broken lines and curves for other propellers are shown in full lines, thus enabling visual comparison to be readily made. The standard curves, as stated, all refer to the S_1 type of section. Figure 15 shows some results for the S_2 type, and Figure 16 for the S_3 type, the S_2 having a driving fact of maximum concavity and the S_3 of moderate concavity. It is seen that the departures from the standard due to concavity of face are not very great and involve no material change of efficiency or diameter if applied in practice.

Figure 17 shows a comparison between models with convex faces and the standard. This, too, does not show any radical departure or material gain. Broadly speaking, the convex faces are more efficient than the concave.

Figure 18, for propeller model No. 96, which had blades capable of being twisted, is interesting. No. 96 had F_1 form, A_1 area, uniform pitch at .7 pitch ratio, and the S_1 type of section. In these respects it reproduced No. 5, one of the F_1A_1 family. However, it will be seen in Figure 18 that it does not agree closely with .7 pitch ratio in the standard curves. No. 5 and No. 96 were experimented with at entirely different times, which may account for some of the difference, but No. 96 differed from No. 5 in that, in order to enable the blades to be twisted, it had a different and quite large hub which would materially reduce the efficiency. However, Figure 18 indicates that twisted blades, which are sometimes advocated as very desirable, great importance being attached to minute approximations to certain angles, are not particularly objectionable and not particularly desirable.

Of late, we often find marine propellers with a maximum thickness of cross section forward of the center, and sometimes these are stated to be based upon aeronautical research. This type of section is frequently used for sections of wings of airplanes and with success for airplane propellers, but Figure 19, after making allowance for the fact that the standard curve for pitch ratio .7 is somewhat out of place, being too high in efficiency, does not appear to indicate that for airplane propellers shifting the position of the maximum thickness results in material improvement of efficiency. The location $\frac{1}{3}$ of the width from the leading edge seems a little the best.

15. Reduction of Durand and Lesley's Result to Design Form.

In dealing with airplane propellers, many designers prefer to express the pitch concept by pitch angle, rather than pitch ratio. This is almost meaningless, unless the radius for the pitch angle is given, so, when following this practice, the pitch angle at some point between ⅔ and ¾ of the radius is usually adopted as the characteristic angle.

In this connection, Figure 20 shows, over the range of nominal pitch ratio liable to be used in practice, the relation between pitch ratio a and pitch angle over the propeller radius. The contours for pitch angle are straight lines, but, as is obvious from the figure, the pitch angles themselves depend upon the pitch proper and the fraction of radius at which the pitch angle is taken.

The figures heretofore given, dealing with Durand and Lesley's results, are not in engineering units. When we undertake to plot these results for purposes of design, it is necessary to base them upon engineering units. For this purpose, a suitable basic coefficient appears to be

$$B = \frac{N\sqrt{P}}{\rho^{\frac{1}{2}} V^{2.5}}$$

This is identical with the basic coefficient B, used for marine work, except that the density is included. This is necessary on account of the varying density of air in which airplanes operate.

As regards efficiency, there is no change needed.

For diameter, substitute $\delta = \dfrac{Nd}{V}$, this being exactly the same as in marine work, except that now V refers to statute miles of 5,280 feet per hour.

In order to deal fully with the problem, we need another coefficient which will enable us to estimate performance. The airplane propeller has to deal with a very much wider range of resistance or speed than marine propellers. For this third coefficient, a suitable expression would seem to be $\dfrac{Td}{1000\ P/N}$, where T denotes thrust in pounds, d denotes diameter in feet, P denotes horsepower absorbed, and N denotes revolutions per minute.

Taking airplane engines as they are, for a first approximation it is generally possible to estimate a value of $\dfrac{P}{N}$ at full throttle which varies comparatively little with speed. Using these new engineering coefficients and constants, Figure 21 shows the result when the standard curves for Durand's four families of Figure 14 are reduced to the form used in the preceding for marine propellers, B being plotted upon a logarithmic scale. It is evident that, given the power, the revolutions, and the expected speed, Figure 21 can be used at once to determine diameter and pitch and efficiency, just as in marine work.

It will be seen that, broadly speaking, Figure 21 follows in its general features the general features of Figures 1 and 5, the comparable diagrams for 3-bladed and 4-bladed marine propellers.

16. Results of Five Full-Sized Propellers.

In the 20-foot wind tunnel recently completed by the National Advisory Committee for Aeronautics at the Langley Memorial Aeronautical Laboratory, it is possible to test full-sized propellers up to a speed of 100 miles per hour, attached to that part of the fuselage which comes within the propeller wake. Recently, 5 full-sized 2-bladed 9-foot propellers were tested thus in front of a partial fuselage. These had metal blades much thinner than Durand and Lesley's wooden models. They had a variable pitch, the blades being capable of being twisted. With the blades set at 13 degrees at the 42-inch radius, the pitch from the 36-inch radius to the tip had the approximately uniform value of 5 feet. From a 36-inch radius, it was gradually reduced toward the hub so that at the 18-inch radius it was only 4½ feet. The particulars and results of these tests are given in Report No. 306 of the National Advisory Committee for Aeronautics, published this year.

Figure 22 shows these results reduced to the same form as in Figure 21, except that instead of pitch, the values of pitch angle at the 42-inch radius are used. For this propeller, 42 inches is .778 of the radius. It should be borne in mind, however, that Figure 22 deals with the full-sized propeller in front of the fuselage, not in the open.

These metal propellers with thin blades would naturally be expected to show a higher efficiency than Durand and Lesley's models of thicker-bladed wooden propellers, and they do show materially higher efficiency, in spite of a slight reduction in efficiency resulting from the fuselage reactions.

Figure 22, like Figure 21, is readily applicable to propeller design or setting. In considering these figures, it may be as well to point out some of the values of B found in practice in aeronautical work. The table below gives for 6 actual airplanes the values of B at maximum speed in level flight near the ground and during maximum climb.

Table of B Values.

Type	Maximum Speed	Values of B	
		Level Flight Max. Speed	At Maximum Climb
Torpedo	110	6.0	15.5
Training	115	4.0	10.5
Observation	120	5.5	12.9
Flying Boat	130	4.3	11.3
Fighter	150	3.0	9.0
Racer	230	1.2	4.8

It is well known that a propeller best suited to level flight is not best suited to climb, and vice versa.

From Figures 21 and 22, it is possible to readily evaluate the differences in this connection.

These figures may also be used to estimate the performance of a given propeller of the family under various conditions. In dealing with the given propeller, we know the diameter and pitch ratio, or pitch setting. For a given case we may estimate the thrust for given conditions in which we know V. While we do not know P or N, we can make a close estimate to $\frac{P}{N}$ for full throttle, as already stated. Hence, we can estimate $\frac{Td}{1000\ P/N}$. Entering the diagram at the proper level for the pitch ratio or pitch setting, we follow the horizontal line until we reach the point where $\frac{Td}{1000\ P/N}$ has the estimated value. From this we can determine δ and B. We can determine N at once, because $N = \frac{\delta\ V_A}{d}$. Having determined N, we can recur to the value of $\frac{Td}{1000\ P/N}$ and evaluate P.

17. *Conclusion.*

The problem of the airplane propeller is essentially simpler than that of the marine propeller because the former works under such a head that breakdown or cavitation is not to be feared at tip speeds below that of sound in air and the reactions between propeller and airplane, while present, are seldom important. It is as regards cavitation and the influential reactions between propeller and ship that in the marine field we need to extend our knowledge.

As regards the propeller apart from the ship or airplane, once we have fixed power, speed, and revolutions, we have fixed the limit of propeller efficiency. We may fall short of it, but no practicable variation of dimensions or shape will enable us to materially improve propeller efficiency, which is dependent primarily upon diameter and pitch ratio. If cavitation or breakdown is absent the skillful designer may pick up one or two points of propeller efficiency—no more. Propulsive efficiency is susceptible of many times this improvement if by skillful design helpful reactions between propeller and ship are fostered and harmful reactions minimized.

Appendix I.

Symbols, Formulae and Definitions.

There is given below a list of symbols, formulae, etc., which, unless it is specifically stated otherwise in the text (as in Appendix II), are used throughout. There is one complication, namely, that following established practice in each field the symbol V denoting speed is expressed numerically by the knots for marine propellers and by the statute miles per hour for airplane propellers. A knot is a speed in sea miles of 6080 feet per hour while a statute mile is 5280 feet.

a.—Denotes Pitch Ratio or P. R. It is the ratio between the pitch (p) of the propeller at any point and its diameter (d).

A.—Denotes a coefficient independent of revolutions. Subscripts as necessary characterize it more closely.

Thus $$A_p = \frac{1000\, aP}{d^2 V_A^3} \qquad A_u = \frac{1000\, aU}{d^2 V_A^3}$$

Further subscripts are used to indicate whether we are dealing with 3-bladed or 4-bladed propellers.

B.—Denotes the basic coefficient used which is independent of diameter. The formula is for marine propellers $B_P = \frac{N\sqrt{P}}{V_A^{2.5}}$, $B_U = \frac{N\sqrt{U}}{V_A^{2.5}}$ with appropriate further subscripts. In air work we must bring in the density and for air the formulae for our basic B are $B_P = \frac{N\sqrt{P}}{\rho^{\frac{1}{2}} V^{2.5}}$ and $B_U = \frac{N\sqrt{U}}{\rho^{\frac{1}{2}} V^{2.5}}$

For Figs. 21 and 22 since we have no figures based upon U, B is used without subscript and V is speed of plane, since wake is of small effect for usual air propellers.

b.—Denotes block coefficient of a ship

B.T.F.—Denotes Blade Thickness Fraction. It is the ratio between the maximum intercept of face and back of blade upon the axis (if extended to axis, and the diameter of the propeller.

C.—Denotes a coefficient independent of speed. Formulae are $$C_P = \frac{a\, P}{d^2 \left(\frac{pN}{1000}\right)^3} \qquad C_U = \frac{a\, U}{d^2 \left(\frac{pN}{1000}\right)^3}$$

Coefficients C and A are very similar.

d.—Denotes diameter of propeller in feet.

δ.—Delta denotes a coefficient independent of power. The formulae is $\delta = \frac{dN}{V_A}$.

As is seen from the text, δ is a very important coefficient. It is exceeded in importance only by B the basic coefficient used.

h.—Denotes Mean Width Ratio (M.W.R.) for a propeller. It is the mean width of the blade divided by the diameter.

e.—Denotes efficiency of propulsion or $\frac{E}{P}$.

e_p.—Denotes efficiency of propeller.

E.—Denotes the Effective or Tow Rope horsepower of the ship. It is proportional to the resistance of the ship without propeller multiplied by its speed with reference to undisturbed water.

n.—Denotes revolutions per second. It is used usually for models only. Also n is used below to denote number of propeller blades.

N.—Denotes revolutions per minute. It is used usually for full-sized propellers.

p.—Denotes pitch in feet.

P.A.—Denotes pitch angle in degrees. It is the angle between a tangent to the radial section and (at its point of tangency) a line through the same point parallel to the axis.

ρ.—Rho denotes density of the air or the weight of a cubic foot in pounds divided by the acceleration of gravity in feet per second. Standard dry air is taken at 15° C. temperature and 760 mm. of mercury preasure. Its weight per cubic foot is .07651 lbs. and its ρ value is .002378.

s.—Denotes slip or slip ratio. Its formula is

$$s = \frac{V_p - V_A}{VP}$$

t.—Denotes Thrust Deduction Fraction. It is the ratio between the suction of the propeller upon the after part of the ship and the thrust of the propeller upon thrust block.

T.—Denotes thrust in pounds.

U.—Denotes power utilized. It is proportional to the thrust multiplied by V_A the speed of advance.

V.—Denotes speed in knots for ships and in miles per hour for airplanes.

V_p.—Denotes speed of propeller in knots or speed with which it would advance if working as a screw in a solid nut.

V_A.—Denotes speed in knots with which the propeller advances with reference to the water in which it is working. In a disturbed wake V_A is calculated with reference to an average uniform wake.

w.—Denotes Wake Fraction. It is the ratio between the velocity of wake (usually forward involving w position) averaged and the speed of the ship V.

For propellers with blades of normal elliptical type with hubs .2 the propeller diameter, there are several useful relations given below (from Speed and Power). In these formulae l denotes the ratio between maximum blade width and diameter and n denotes number of blades.

Mean Width Ratio—$h = .842\frac{l}{d}$ or $l = 1.188\ h\ d$

Developed Area $= .4\ n\ d^2\ h$

Projected Area (with close approximation) $= (0.4267 - 0.0916\ a)\ n\ d^2\ h$

Projected Area ÷ Developed Area $= 1.067 - .229\ a$

Developed Area ÷ Disc. Area $= .509\ n\ h$

Projected Area ÷ Disc. Area $= (.543 - .1166\ a)\ n\ h$.

Appendix II.

The rules of the American Bureau of Shipping for propeller blade thickness, quoted below, are based upon the use of material having the following characteristics:

> Cast iron, tensile strength 20,000 lbs. per square inch;
>
> Semi-steel, tensile strength 28,000 lbs. per square inch;
>
> Cast steel, tensile strength 58,000 lbs. per square inch;
>
> Manganese bronze, or similar bronze, tensile strength 60,000 lbs. per square inch.

Cast steel blades are made relatively thicker than bronze because they are more subject to corrosion.

The rules are quoted below:

" (3) Blades.—Where the propeller blades are of Standard design the thickness of the blades shall be not less than determined by the formula:

$$t = C\sqrt{\frac{D \times d^3}{P \times w \times N}}$$

Where t = Thickness of blade in inches at radius of .125 D.

w = Width of blade in inches, minimum $4t$, at radius of .125 D.

d = Diameter of propeller shaft under liner in inches.

D = Diameter of propeller in feet.

P = Pitch of propeller in feet.

N = Number of blades.

C = Constant from the following table.

Rake of Blade in Inches per Ft.	Cast Iron		Semi-Steel		Cast Steel		Bronze	
	Turbines, Motors	Recipr. Engines	Turbines, Motors	Recipr. Engines	Turbines, Motors	Recipr. Engines	Turbines, Motors	Recipr. Engines
0	1.85	1.66	1.61	1.45	1.29	1.17	1.19	1.07
½	1.88	1.69	1.64	1.48	1.32	1.19	1.21	1.09
1	1.91	1.72	1.67	1.50	1.34	1.21	1.23	1.11
1½	1.94	1.75	1.70	1.53	1.36	1.23	1.25	1.12
2	1.97	1.77	1.72	1.55	1.39	1.25	1.27	1.14
2½	2.00	1.80	1.75	1.57	1.41	1.27	1.29	1.16
3	2.03	1.83	1.78	1.60	1.43	1.29	1.31	1.18

For blades of special design the value of "C" will be determined by the Committee upon submittal of the drawing of the Propeller.

"Fillets at the root of the blades should not be considered in the determination of blade thickness.

"The required blade section must not be reduced in order to provide clearance for nuts.

"Detachable blades should be recessed into the propeller hub; the side clearance should be as small as possible and should be filled with a protective mixture to prevent corrosion and galvanic action.

"(4) Studs.—The sectional area of the studs at the bottom of the thread is to be determined by the formula:

$$a = \frac{d^3 \times .55\,D}{P \times N \times n \times r}$$

Where a = Area of one stud at bottom of thread in square inches.

n = Number of studs on driving side of blade.

r = Pitch diameter of the studs in inches.

The remaining symbols are noted under 'Blades,' Par. 3.

"Studs should be fitted tightly into the hub and a shoulder on the same under the flange is recommended. The nuts should also have a tight-fitting thread and be secured by stop-screws."

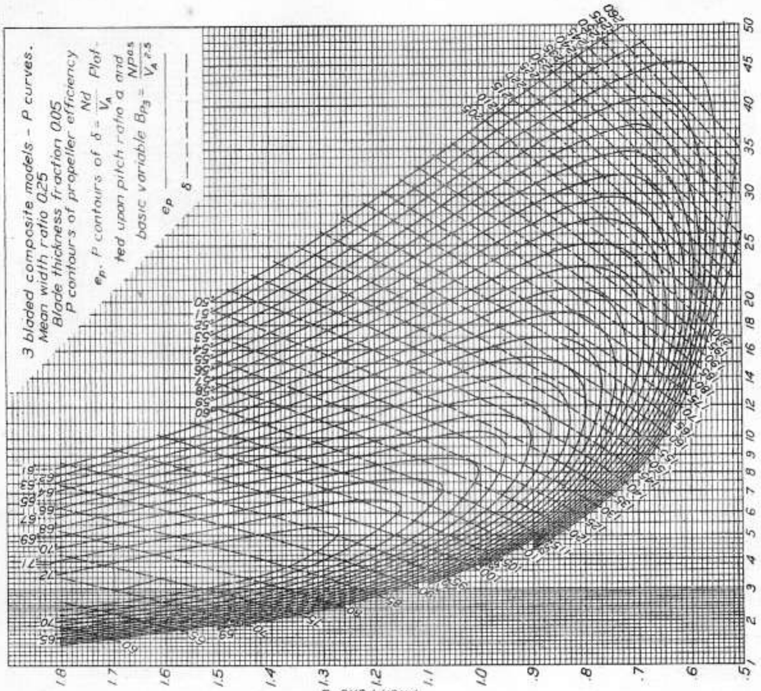

3 bladed composite models - P curves.
Mean width ratio 0.25
Blade thickness fraction 0.05
P contours of propeller efficiency
e_P. P contours of $\delta = \frac{Nd}{V_A}$ Plotted upon pitch ratio a and basic variable $B_{P3} = \frac{NP^{0.5}}{V_A^{2.5}}$
e_P
δ
Pitch ratio a
Basic variable B_{P3}

Fig. 1

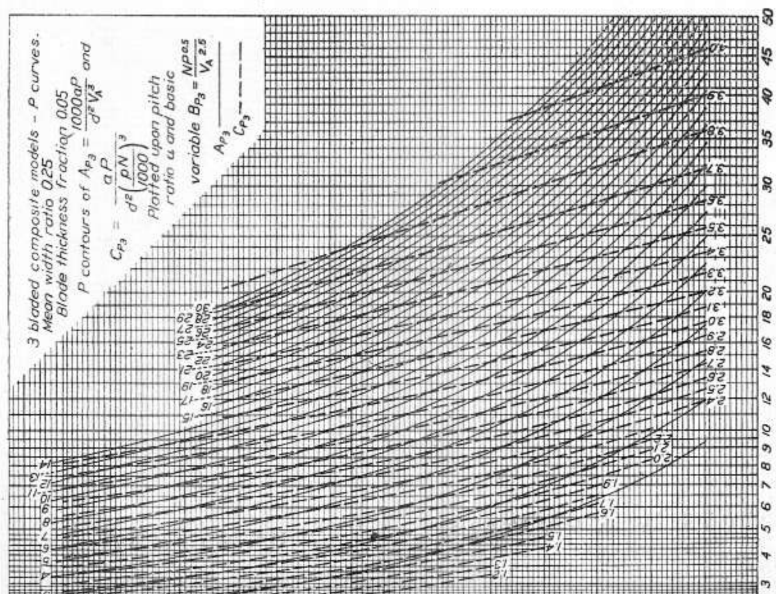

3 bladed composite models - P curves.
Mean width ratio 0.25
Blade thickness fraction 0.05
P contours of $A_{P3} = \frac{1000aP}{d^2V_A^3}$ and
$C_{P3} = \frac{aP}{d^2\left(\frac{pN}{1000}\right)^3}$
Plotted upon pitch ratio a and basic variable $B_{P3} = \frac{NP^{0.5}}{V_A^{2.5}}$
A_{P3}
C_{P3}
Pitch ratio, a
Basic variable B_{P3}

Fig. 2

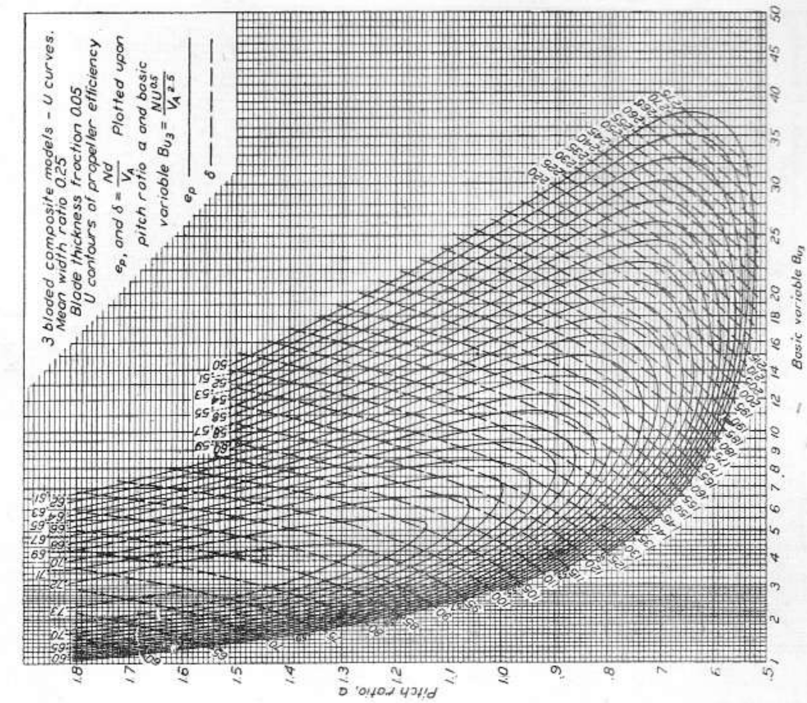

3 bladed composite models - U curves.
Mean width ratio 0.25
Blade thickness fraction 0.05
U contours of propeller efficiency
e_P, and $\delta = \frac{Nd}{V_A}$ Plotted upon
pitch ratio a and basic
variable $B_{U3} = \frac{NU^{0.5}}{V_A^{2.5}}$
e_P
δ
Pitch ratio, a
Basic variable B_{U3}

Fig. 3

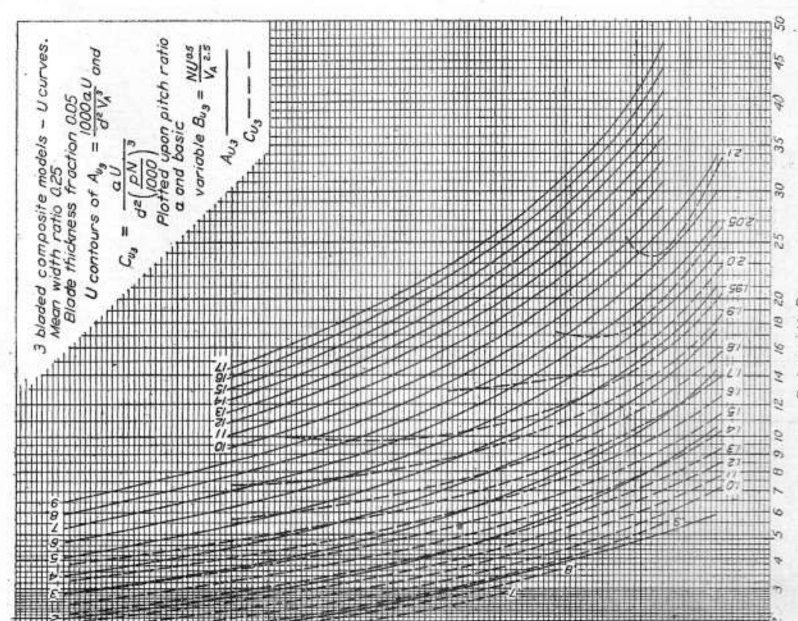

3 bladed composite models - U curves.
Mean width ratio 0.25
Blade thickness fraction 0.05
U contours of $A_{U3} = \frac{1000aU}{d^2 V_A^3}$ and
$C_{U3} = \frac{aU}{d^2\left(\frac{pN}{1000}\right)^3}$
Plotted upon pitch ratio
a and basic
variable $B_{U3} = \frac{NU^{0.5}}{V_A^{2.5}}$
A_{U3}
C_{U3}
Pitch ratio, a
Basic variable B_{U3}

Fig. 4

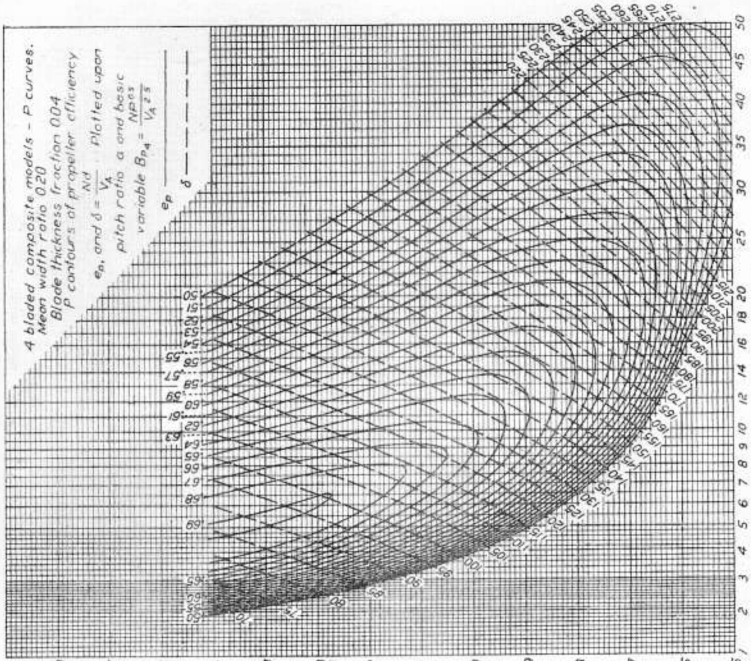

4 bladed composite models - P curves.
Mean width ratio 0.20
Blade thickness fraction 0.04
P contours of propeller efficiency
e_p, and $\delta = \frac{Nd}{V_A}$. Plotted upon
pitch ratio a and basic
variable $B_{P_4} = \frac{Np^{0.5}}{V_A^{2.5}}$
e_p
δ
Pitch ratio, a
Basic variable B_{P_4}

Fig. 5

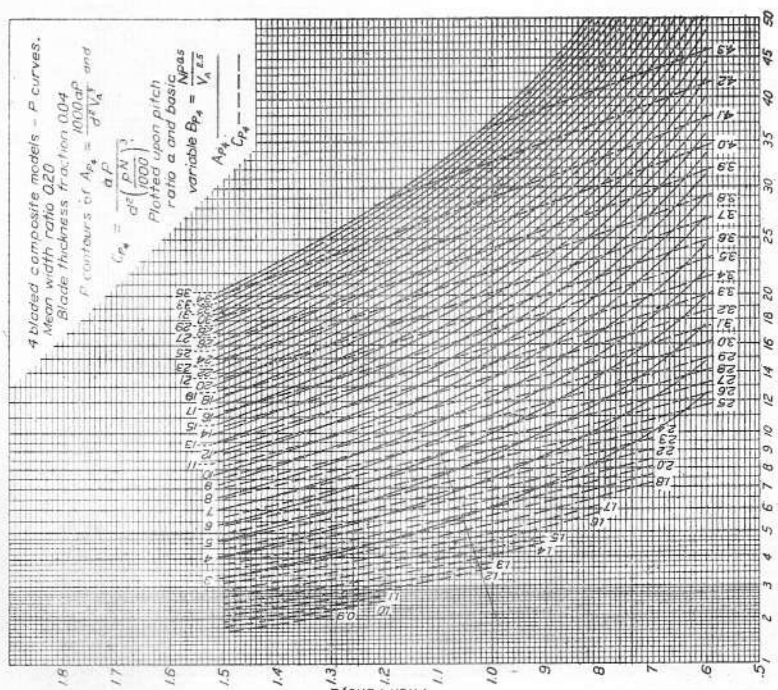

4 bladed composite models - P curves.
Mean width ratio 0.20
Blade thickness fraction 0.04
P contours of $A_{P_4} = \frac{1000\,aP}{d^2 V_A^3}$ and
$C_{P_4} = \frac{aP}{d^2\left(\frac{pN}{1000}\right)^3}$
Plotted upon pitch
ratio a and basic
variable $B_{P_4} = \frac{Np^{0.5}}{V_A^{2.5}}$
A_{P_4}
C_{P_4}
Pitch ratio, a
Basic variable B_{P_4}

Fig. 6

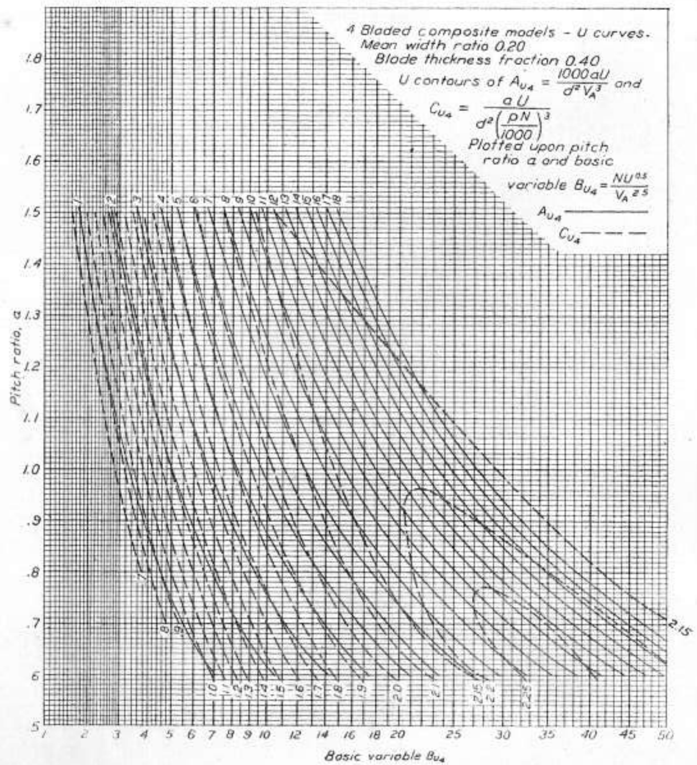

4 Bladed composite models - U curves.
Mean width ratio 0.20
Blade thickness fraction 0.40
U contours of $A_{U4} = \frac{1000aU}{d^2 V_A^3}$ and
$C_{U4} = \frac{aU}{d^2\left(\frac{pN}{1000}\right)^3}$
Plotted upon pitch ratio a and basic
variable $B_{U4} = \frac{NU^{0.5}}{V_A^{2.5}}$
A_{U4}
C_{U4}
Pitch ratio, a
Basic variable B_{U4}

Fig. 8

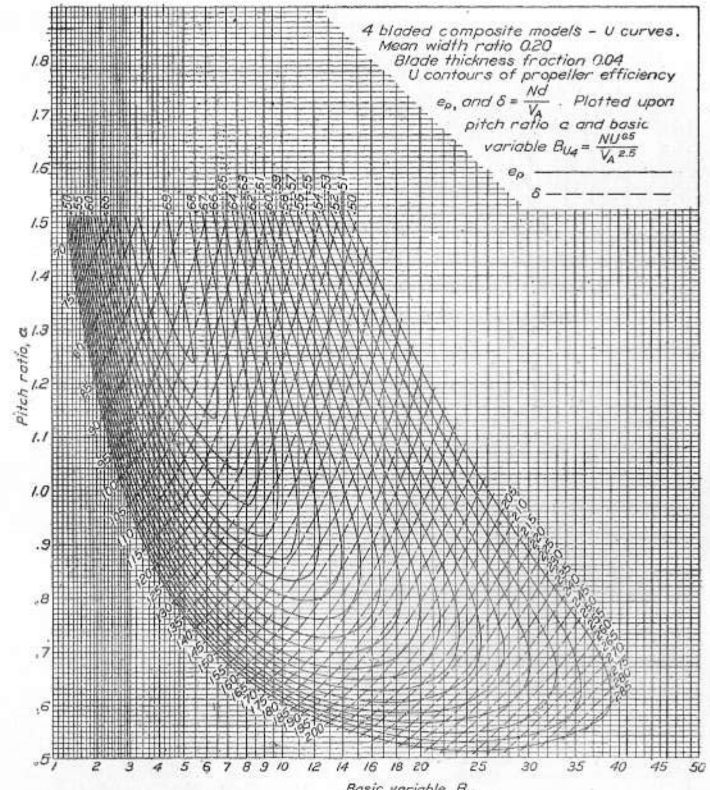
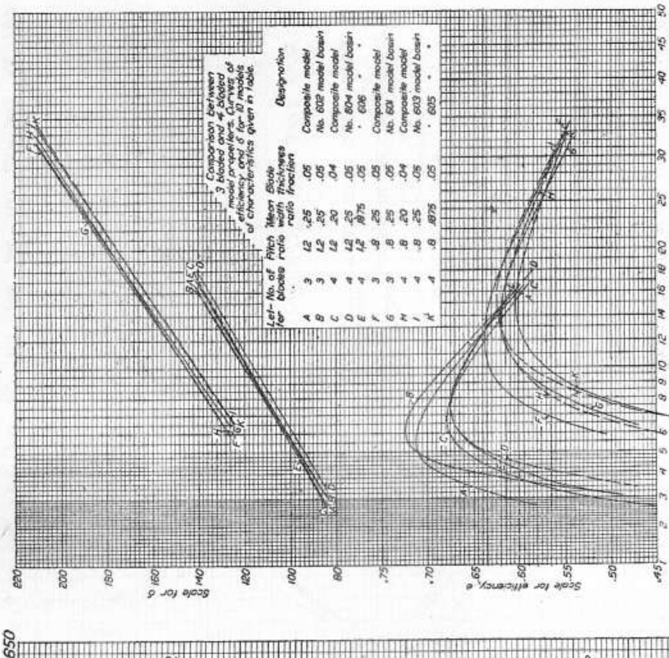
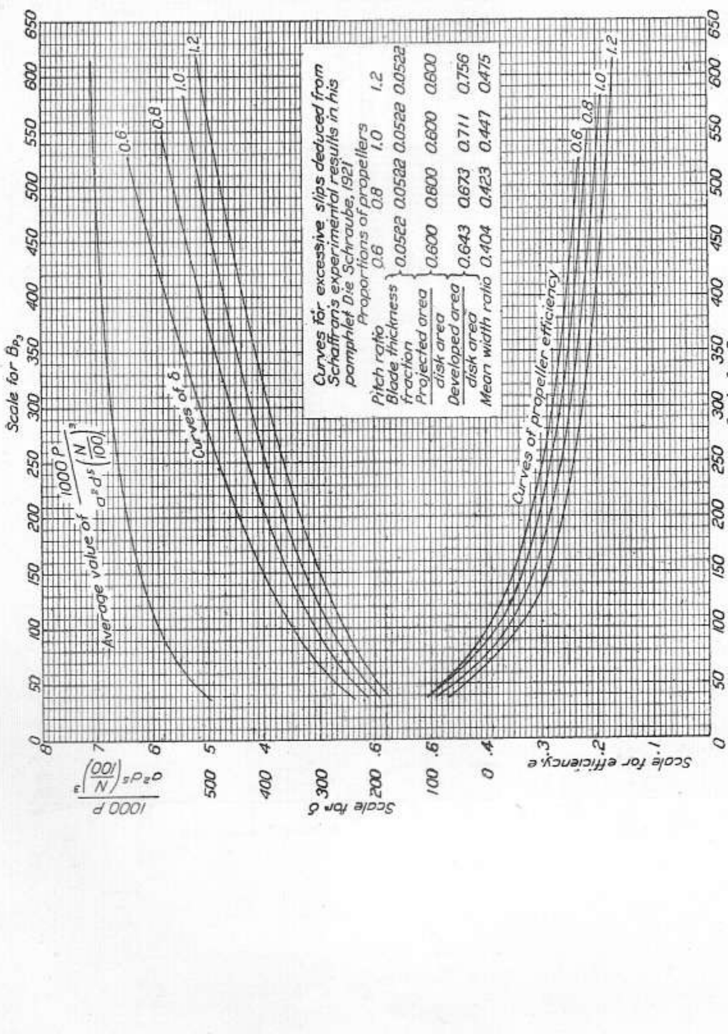

4 bladed composite models - U curves.
Mean width ratio 0.20
Blade thickness fraction 0.04
U contours of propeller efficiency
e_p, and $\delta = \frac{Nd}{V_A}$. Plotted upon
pitch ratio a and basic
variable $B_{U4} = \frac{NU^{0.5}}{V_A^{2.5}}$
e_p
δ
Pitch ratio, a
Basic variable, B_{U4}

Fig. 7



Comparison between 3 bladed and 4 bladed model propellers. Curves of efficiency and δ for 10 models of characteristics given in table.
Basic variable B_p
Scale for δ
Scale for efficiency, e

Fig. 9



Scale for B_{p_3}
Average value of $\frac{1000\,P}{a^2 d^5 \left(\frac{N}{100}\right)^3}$
Curves of δ
Curves for excessive slips deduced from Schaffran's experimental results in his pamphlet Die Schraube, 1921
Proportions of propellers
Pitch ratio 0.6 0.8 1.0 1.2
Blade thickness fraction 0.0522 0.0522 0.0522 0.0522
Projected area / disk area 0.600 0.600 0.600 0.600
Developed area / disk area 0.643 0.673 0.711 0.756
Mean width ratio 0.404 0.423 0.447 0.475
Curves of propeller efficiency
Scale for δ
Scale for efficiency, e

Fig. 10

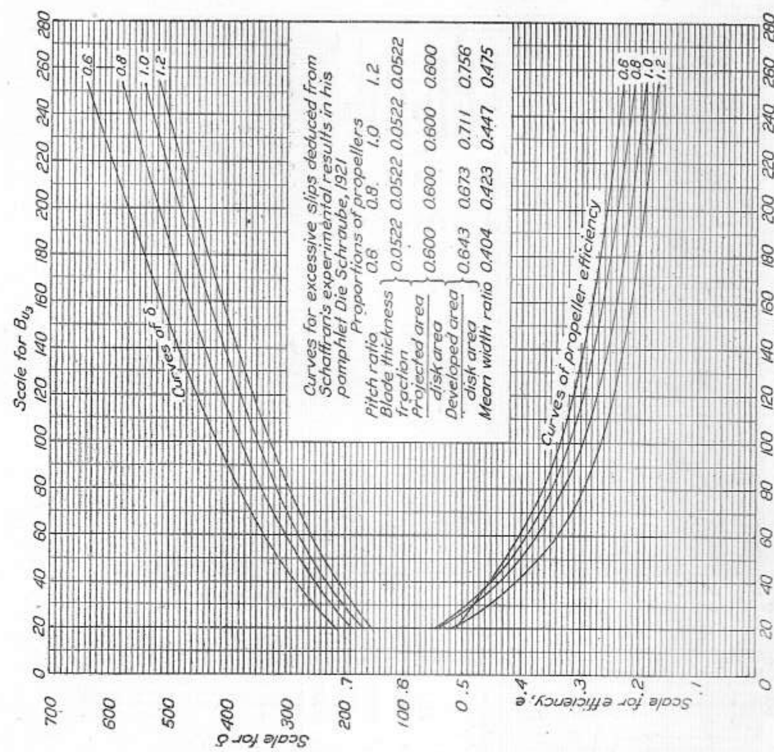

Scale for B_{u_3}
Scale for δ
Scale for efficiency, e
Curves of δ
Curves of propeller efficiency
Curves for excessive slips deduced from Schaffran's experimental results in his pamphlet Die Schraube, 1921
Proportions of propellers
Pitch ratio 0.6 0.8 1.0 1.2
Blade thickness fraction 0.0522 0.0522 0.0522 0.0522
Projected area / disk area 0.600 0.600 0.600 0.600
Developed area / disk area 0.643 0.673 0.711 0.756
Mean width ratio 0.404 0.423 0.447 0.475

Fig. 11

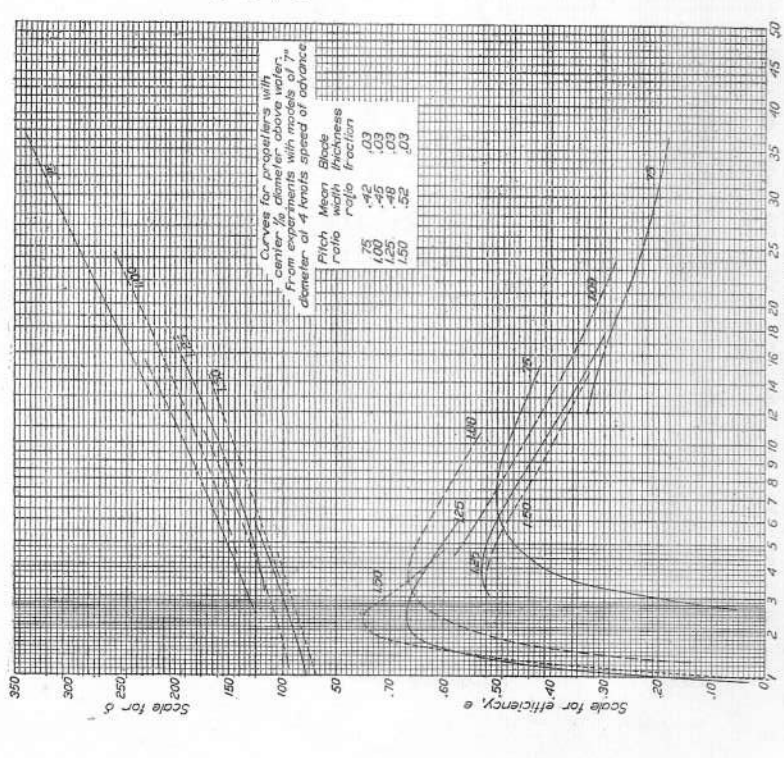

Scale for δ
Scale for efficiency, e
Basic variable B_{u_3}
Curves for propellers with center ½ diameter above water. From experiments with models of 7" diameter at 4 knots speed of advance
Pitch ratio / Mean width ratio / Blade thickness fraction
.75 .42 .03
1.00 .45 .03
1.25 .48 .03
1.50 .52 .03

Fig. 12

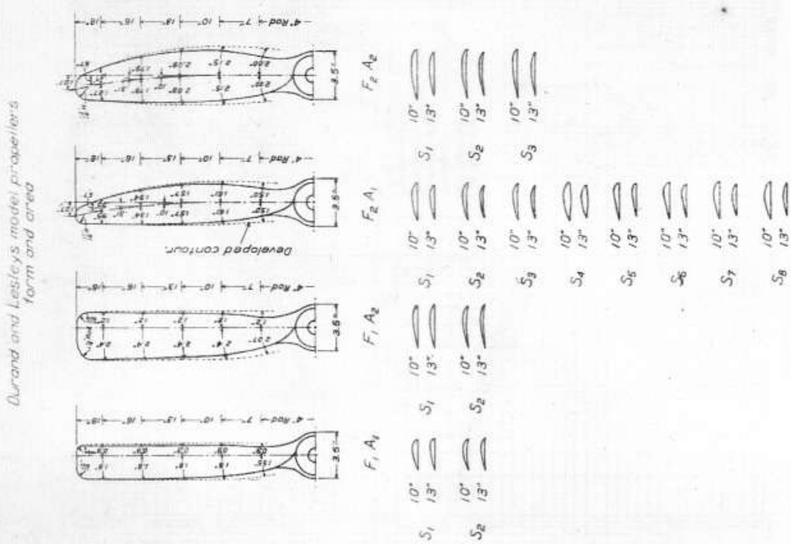

Durand and Lesley's model propellers form and area
Developed contour
$F_1 A_1$
$F_1 A_2$
$F_2 A_1$
$F_2 A_2$

Fig. 13

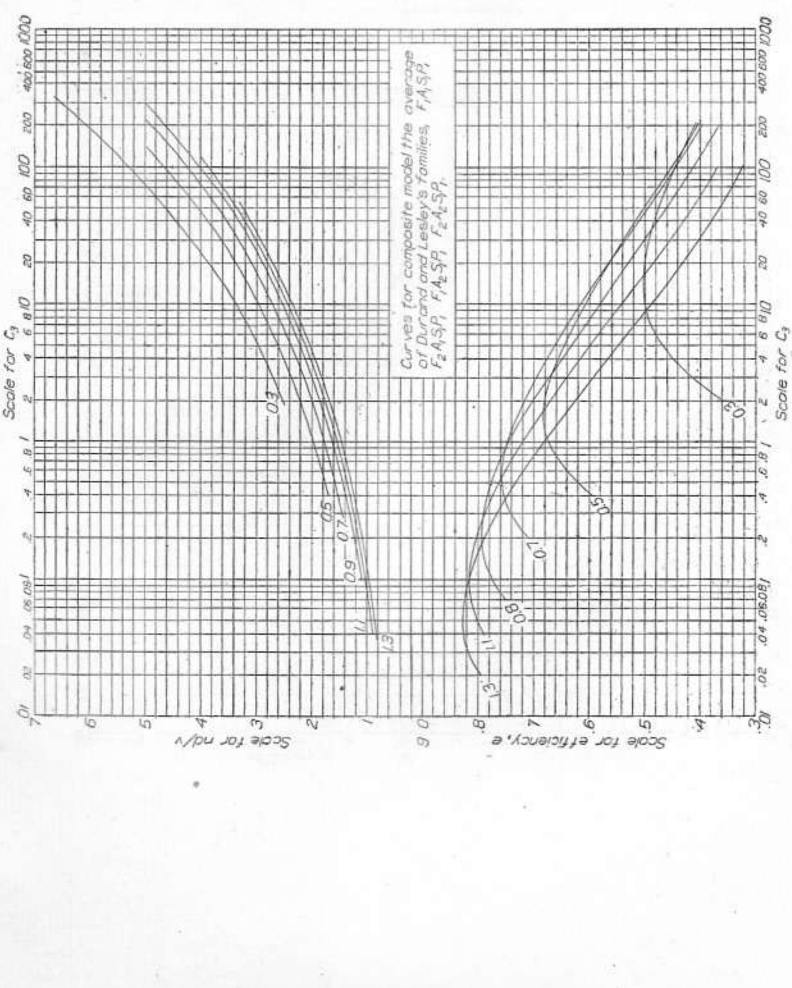

Scale for C_3
Scale for nd/v
Scale for efficiency, e
Curves for composite model the average of Durand and Lesley's families $F_1A_1S_1P_1$ $F_1A_2S_1P_1$ $F_2A_1S_1P_1$ $F_2A_2S_1P_1$

Fig. 14

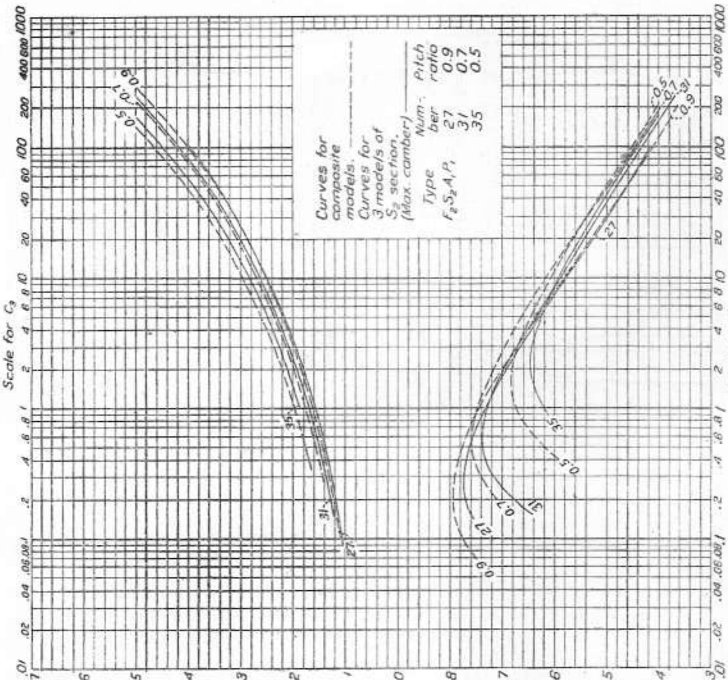

Curves for composite models.
Curves for 3 models of S_2 section, (Max. camber)
Type $F_2S_2A_1P_1$
Num-ber 27 31 35
Pitch ratio 0.9 0.7 0.5
Scale for C_3
Scale for nd/v
Scale for efficiency, e

Fig. 15

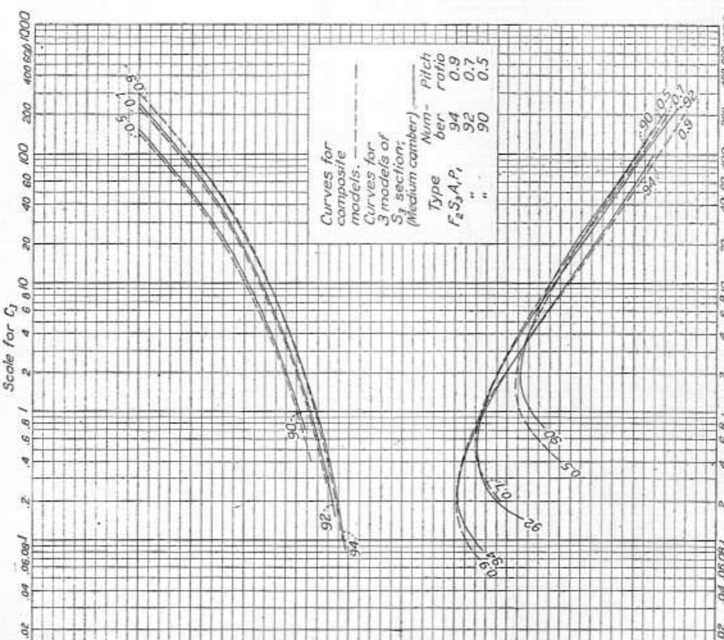

Curves for composite models.
Curves for 3 models of S_3 section; (Medium camber)
Type $F_2S_3A_1P_1$
Num-ber 94 92 90
Pitch ratio 0.9 0.7 0.5
Scale for C_3
Scale for nd/v
Scale for efficiency, e

Fig. 16

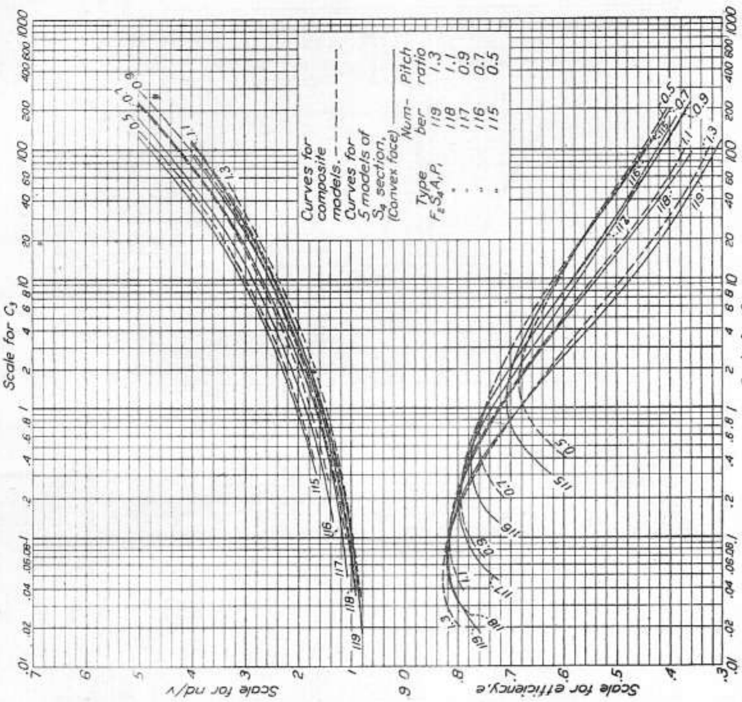

Scale for C_3
Scale for nd/v
Scale for efficiency, e
Curves for composite models. -----
Curves for 5 models of S_4 section, (Convex face) ———
Type $F_2S_4A.P.$
Number 119 118 117 116 115
Pitch ratio 1.3 1.1 0.9 0.7 0.5

Fig. 17

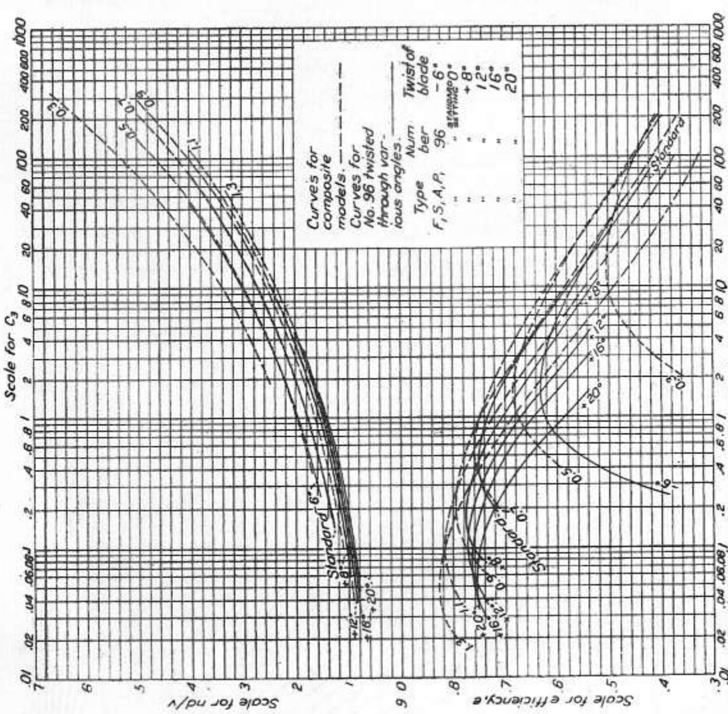
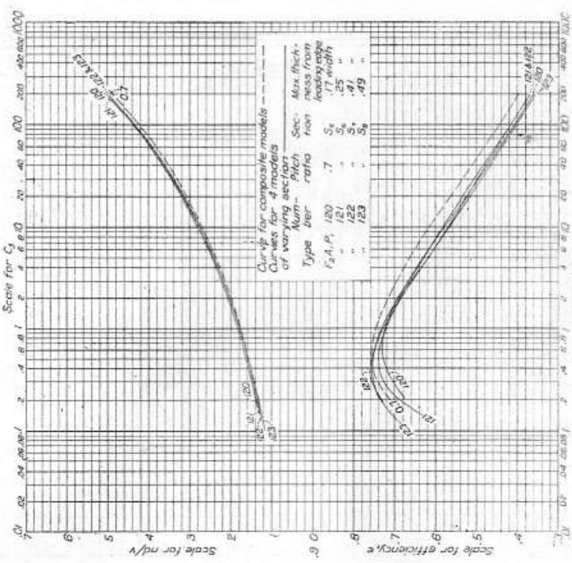
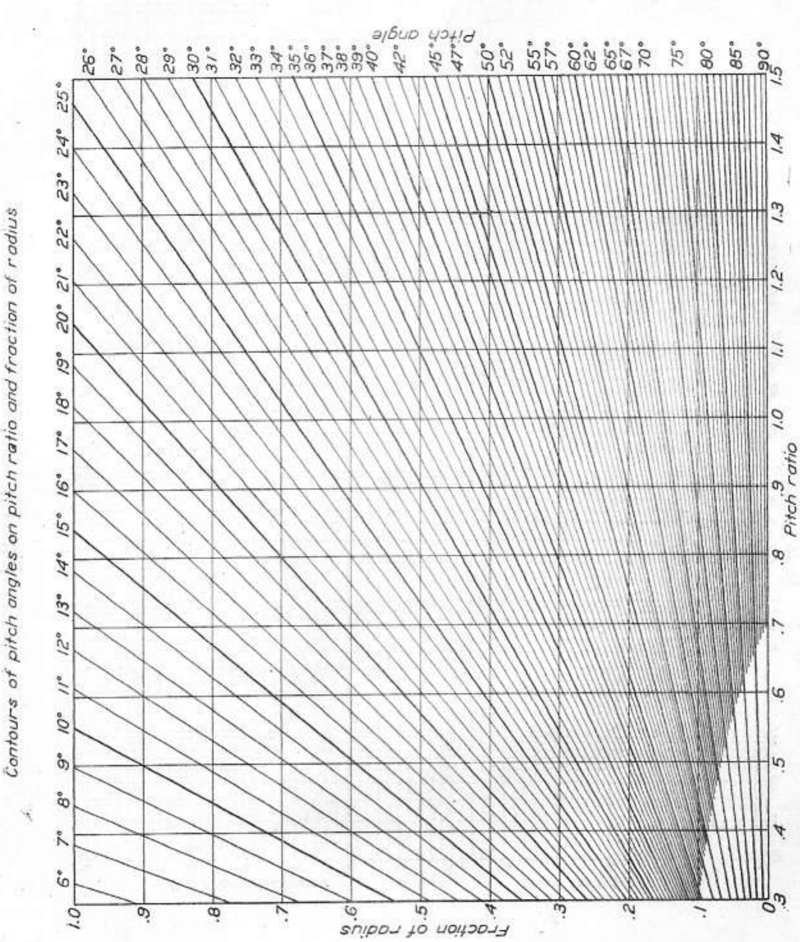

Scale for C_3
Scale for nd/v
Scale for efficiency, e
Curves for composite models. -----
Curves for No. 96 twisted through various angles. ———
Type $F_1S_1A.P.$
Number 96
Twist of blade −6° Standard setting 0° +8° 12° 16° 20°
Standard

Fig. 18



Scale for C_3
Scale for nd/v
Scale for efficiency, e
Curve for composite models
Curves for 4 models of varying section

Fig. 19



Contours of pitch angles on pitch ratio and fraction of radius
Fraction of radius
Pitch ratio
Pitch angle

Fig. 20

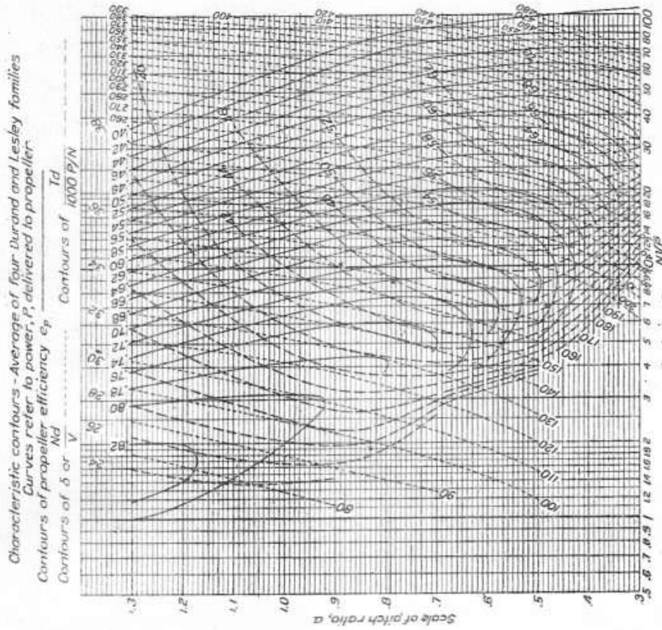

Characteristic contours - Average of four Durand and Lesley families
Curves refer to power, P, delivered to propeller
Contours of propeller efficiency e_p
Contours of δ or $\frac{Nd}{V}$
Contours of $\frac{Td}{1000\,P/N}$
Scale of pitch ratio, a
Scale for $B = \frac{N\sqrt{P}}{\rho^{1/2}V^{5/2}}$

Fig. 21

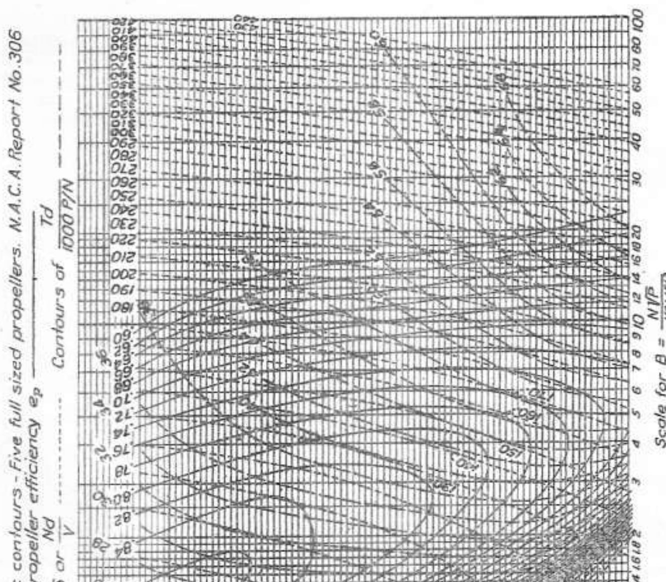

Characteristic contours - Five full sized propellers. N.A.C.A. Report No.306
Contours of propeller efficiency e_p
Contours of δ or $\frac{Nd}{V}$
Contours of $\frac{Td}{1000\,P/N}$
Pitch angles at .78 radius
Scale for $B = \frac{N\sqrt{P}}{\rho^{1/2}V^{5/2}}$

Fig. 22

Shipbuilding in Great Britain.

(Paper No. 425)

By Professor ***Percy A. Hillhouse, D.Sc.,***
M.I.N.A., M.I.E.S.

Great Britain, like Japan, is an island state, and shipping is a factor of prime importance to its life and progress. It is natural therefore that both should be shipbuilding countries and that much of their thought and energy should be devoted to the design, construction, and operation of ships. The two countries are almost equal in size, the United Kingdom having an area of about 120,000, and Japan about 140,000 square miles. They are alike in that both lie adjacent to a large continent and are not entirely self-contained. Japan, though somewhat later than Great Britain in entering the field of ocean shipbuilding, has now a vigorous and well equipped shipbuilding industry and her mercantile fleets rank among the largest in the world.

In Britain to-day there are some 80 first class shipbuilding firms whose names are well known to the shipping community and in many cases are almost "household words." As "first class" I am counting all firms capable of building vessels 350 ft. in length and over. The firms referred to can be classified as follows:—.

Maximum Length of Ship which can be built.	Number of Firms.
350′	10
350′ to 400′	13
400′ to 450′	14
450′ to 500′	13
500′ to 550′	4
600′	6
650′	3
700′	7
750′	2
900′	2
1000′	6
	80

Or by tonnage capability, thus:—

Maximum Gross Tonnage of Mercantile Vessel actually built.	Number of Firms.
over 40000	2
„ 30000	1
„ 20000	3
„ 10000	11
„ 5000	35
under 5000	28
	80

There are in all about 570 berths, an average of over seven berths per yard. The average maximum possible annual output per firm is about 50,000 tons gross, so that if all these eighty first class establishments were fully employed, about four million tons of shipping could be produced in one year, about two million tons of steel converted into ships, and employment given to about 750,000 men in the shipyards and engine shops. The total employment would of course be very much greater, since the materials and fittings required in the construction of a modern ship have to be brought in from many widely distributed sources. For example, no fewer than 625 different firms contributed to the building of the "Empress of Canada" and these again without doubt were dependent on many subcontractors for supplies. It would therefore not be an exaggeration to say that every industry in the country shares to some extent directly or indirectly in the production of a liner.

The actual maximum output of mercantile vessels was just under 2,000,000 tons in the year 1913, and just over that figure in 1920. Last year the total was a little over 1,500,000 tons, which was nearly one half of the total world production. Of the total about 40% emanated from Clyde Yards.

The vessels were of all the types, from liner and yacht to cargo tramp and oil tanker, the proportions being somewhat as follows:—

Type	Steam	Motor	Electric	Total Tonnage	Number of Ships	Average Tonnage per Ship
Cargo carriers	692000	187000		879000	210	4190
Tankers	108000	176000	9000	293000	50	5860
Liners	101000	37000	19000	157000	14	11210
Channel steamers	17000			17000	7	2420
War vessels	90000			90000	20	4500
				1436000	301	4770
Various				137000	638	215
	TOTAL			1573000	939	

The 301 large vessels listed above were built by some 60 firms, the average output being about 24,000 tons per yard: 50 smaller firms, averaging 2,750 tons each, contributing the remainder. It is evident therefore that the various "first class" firms were not by any means fully employed even in the year of maximum national output, and that the shipbuilding capacity of the country is largely in excess of the demand for its services. Broadly speaking, the Clyde could, if put to it, produce as much tonnage as the whole Kingdom does, and Great Britain as much as the whole world.

A notable feature of modern shipbuilding practice in Great Britain, is the ever increasing resort to model experiments. These are by no means confined to the liner or cross-channel type, as much attention being paid to the cargo carrier as to her more magnificent sisters. The principal tanks are:—

Situation	Owners	Dimensions
Clydebank,	John Brown & Company,	400′ × 20′ × 9′-6″
Dumbarton,	Denny Brothers,	275′ × 22′ × 10′
Haslar,	British Admiralty,	400′ × 20′ × 9′
St. Albans,	Vickers,	420′ × 20′
Teddington,	National Physical Laboratory,	493′ × 30′ × 12′-3″

All these establishments are kept continuously busy. Three belong to private firms, but are occasionally employed by others when available. Messrs. Stephens have a small tank at Linthouse, and a new tank, 375′ × 20′ × 9′-6″ was built in 1925 by Mr. R. Reid at Hull. The William Froude National Tank at Teddington, presented to the nation by Sir Alfred Yarrow, is available to any shipbuilder or naval architect and is a very hard-pressed establishment, the hours commonly worked being from 7 a.m. till 9 p.m. daily, the total staff numbering 36. In addition to work for private firms, a considerable amount of general research work is carried out and the results freely given to the world in the form of papers read to the principal technical societies. The privately owned tanks also carry out and publish the results of research work, notable among such contributions to our general knowledge being Mr. Luke's well known papers on Wake and Thrust Deduction based on experiments made in the Clydebank tank.

It frequently happens that parties to a law case desire to test disputed matters by means of experiments with models. At Teddington this can only be done if both parties in the case make a joint application, but private tank owners do not insist on this preliminary. The Teddington models are from 15 to 20 ft. in length and are made of wax. The charge for making a "naked" model (hull form, but without bossing, bilge keels, rudder, or other appendages), measuring its resistance in the tank at various speeds, and reporting the results in the from of curves of effective horse power (E.H.P.) and of resistance constant (C), is £140. The report discusses the form proposed, points out its good and bad qualities and suggests improvements. The charge for each modification tested is £40 and the same additional charge is made for each propeller or pair of propellers tested, analysed, and reported on in connection with the model hull.

In 1928, 81 model hulls were tested, representing 59 designs and the maximum saving resulting from modifications suggested by the tank staff was a 16% reduction in power. Over 55% of the work done related to cargo carriers, while 20% concerned vessels of the liner class, and 10% channel and coasting steamers. Self-propelled models can now be tested with one, two, or four propellers, and the best lines for bossings determined by exploring the stream-lines by means of small "flags" or vanes projecting outside of the model and indicating within it.

A second tank could be built at Teddington and kept fully occupied if funds were available. The Admiralty are at present constructing a new tank which will without doubt be designed and equipped in accordance with the teachings of accumulated experience.

The budding naval architect or marine engineer in Great Britain in now well catered for in the matter of education, his subjects being taught in many schools and colleges. Three universities, Durham, Glasgow, and Liverpool, have Chairs of Naval Architecture and Marine Engineering, and confer Degrees in Engineering Science, Naval Architecture side, upon successful students. The average number of students attending each University yearly is about 20, though at Glasgow University 81 were enrolled during the Winter Session of 1921. It is customary for shipyard apprentices to be sent to the University, working in the shipyard during the Summer months and attending classes in Winter. This method of combining practical work with theoretical instruction is known as the "sandwich system," the shipyard work being, I presume, the bread and the University the beef and mustard. Part of the University time is counted as a period of apprenticeship. The normal apprenticeship extends over five years of shipyard work, divided it may be as follows:—

Joiners' shop	1 year
Mould loft and shipwrights	1 "
Platers	1 "
Mechanics	3/4 "
Drawing office	1¼ "
Total	5 years

On the sandwich system six years would be occupied thus:—

Joiners' shop	1 year
Mould loft and shipwrights	1 „
University	1/2 „
Mechanics	1/2 ,
University	1/2 „
Platers	1/2 „
University	1/2 „
Drawing office	1/2 „
University	1/2 „
Drawing office	1/2 „
Total	6 years

The shipyard time is thus reduced to four years, while two years, or four Winter Sessions, are spent at the University.

If the Engineering Degree is obtained before entering the shipyard the time required to complete education and apprenticeship is still longer:—

University winter work and Summer vacations	4 years
Mould loft and shipwrights	1 „
Platers	1 „
Mechanics	3/4 „
Drawing office	1¼ „
Total	8 years

the shipyard time being again four years.

For many years the degree course at all three universities could be completed in three years. Recently however the University of Glasgow extended the course to four years, the first two years' subjects being common to all engineering students, and the last two specialised in the various branches of engineering. Liverpool University is about to make a similar extension.

A large number of Scholarships are now available to intending students of naval architecture during their period of study and to encourage post-graduate research. Most of these are administered by the Institution of Naval Architects, Lloyd's having recently discontinued their Scholarships owing to lack of competition. The I. N. A. Scholarships are:—

*Elgar	£ 130 per annum
*Martell	£ 130 ,,
*I. N. A.	£ 130 ,,
Armstrong	£ 150 ,,
Vickers	£ 150 ,,
Fairfield	£ 150 ,,
*White	£ 100 ,,
Denny	£ 75 ,,

For marine engineering students:—

*Parsons	£ 150 per annum
*Yarrow	£ 100 ,,
Denny	£ 75 ,,

There are two post-graduate scholarships:—

1851 Exhibition	£ 250 per annum for 2 years
Sir William White	£ 150 ,, ,,

The scholarships marked * are endowed, but the renewal of the others every three or four years lies in the option of the donors and may be discontinued at any time when the current holder has completed his course.

In spite of the very material assistance which such scholarships afford to young men intending to enter the shipbuilding industry, competition is not keen and difficulty has been found in recent years in obtaining a sufficient number of candidates for the awards.

In addition to University teaching of naval architecture, the subject occupies a place in the curriculum of many technical colleges and secondary schools and in connection with these the Institution of Naval Architects and the Worshipful Company of Shipwrights in conjunction with the Education Department of the Government have recently instituted a system of National Certificates in Naval Architecture. "Ordinary" and "Higher" certificates are granted, the courses of instruction qualifying candidates extending over three and five years respectively. The certificates are awarded on the results of special examinations as well as on the students records of attendance, home work, class work, and laboratory work during his course of instruction. In addition to the certificates, the Worshipful Company of Shipwrights offers for competition each year among holders of Higher Certificates a gold, a silver, and a bronze medal. These are awarded on the results of special examination separate from and in addition to those set for the certificate.

The youth who is considering naval architecture or shipbuilding as a possible career not unnaturally wants to know what prospects lie ahead. Having taken his degree and duly served his apprenticeship, what chances has a young man of obtaining remunerative employment, a competence, or a fortune ? His apprenticeship completed he becomes a " journeyman " draughtsman, and as in all other walks of life, his future depends very largely upon himself, his abilities and personality tempered to some extent perhaps by chance and opportunity. He may become " leading hand " or chief draughtsman with the firm to which he was apprenticed. He may progress to be under-manager, manager, director, managing-director, or chairman. He may elect to strike out on his own account as " consulting " naval architect and surveyor, or join at first some already established firm of consultants. He may enter the service of one of the Classification Societies—Lloyd's Register of Shipping or the British Corporation Register of Shipping and Aircraft—or become enrolled as a Board of Trade surveyor. In any of these he may rise to the position of Chief Surveyor, and in any case is fairly certain of steady employment and a pension. Honest pennies can also be earned by contributing to the Technical Press and by acting as Arbitrator, Assessor, or expert witness. He may find his sphere of usefulness as naval architect and technical adviser to a firm of shipowners. There is no lack of opportunity; it remains to be ready when opportunity beckons.

The design of a ship is the product of many brains but begins naturally in the mind of her prospective owner. It is he who first decides that a new vessel is desirable and who determines generally the type and qualities that are required. Very often the matter is simply one of asking for a " repeat " of some existing vessel with such modifications as experience dictates. If an entirely new design is necessary the owner may have an expert staff of his own capable of putting his ideas on paper and preparing the necessary plans and specifications. Failing this he may lay his case before some firm of consultants who will either undertake the design and invite various builders to tender on his plans and specifications, or ask several builders to prepare competitive designs and prices. Or the owner may deal directly with the builders and place his order with the firm whose proposals most nearly accord with his requirements.

There are in Great Britain over 200 firms who advertise themselves as Consulting Naval Architects, Marine Engineers, and Surveyors. Many of these undertake design work and will prepare the plans and specifications for proposed new tonnage and superintend the issue of enquiries and the placing of contracts. In most cases however the shipbuilder is asked to guarantee all the techinical qualities of the ship under heavy penalties in case of shortcoming. In other cases the shipowner and his technical adviser pool their experience with that of the selected builder and all work together to produce the best possible vessel, and the contract is placed without penalty clauses.

The design of any vessel is of course profoundly affected by the type of propelling machinery adopted and it is in means of propulsion that the most marked progress has been made in recent years. The marine engineer is steadily producing machinery of less and less weight and lower and lower fuel consumption per horse power developed. The naval architect endeavours to make the best use of this power by seeking the least resistful form of hull and by particular attention to the propeller and its surroundings. Many forms of guide blades and of rudders designed to assist the propeller and reduce eddy losses aft, are being experimented with. Messrs. Denny have successfully revived "Vane Wheel" propulsion in which the shaft centres are above the water surface and only the lower portion of the propeller disc is immersed. Very remarkable propulsive efficiencies have been obtained by this method in small vessels for service in smooth water.

Progress in marine engineering can perhaps best be appreciated by considering some of the notable vessels built within recent years. As examples of the large powers that may now be installed in a single hull, we have the 32 knot battle-cruiser "Hood" with turbines capable of developing no less than 150,000 shaft horse power, and the American air-plane carriers "Saratoga" and "Lexington" with turbo-electrical machinery producing 180,000 horse power and $33\frac{1}{2}$ knots speed. The internal combustion engine, with its low fuel demand, has made a tremendous appeal and the output of motorships now very closely approaches that of steamers. Last year the total output of vessels using oil fuel was double that of coal users. "Aorangi" $580' \times 72'$ built in 1924, was the first large passenger liner propelled by internal combustion engines. Her success blazed the trail for many larger vessels with Diesel machinery, notably the Italian "Augustus" $665 \times 82'$-$6''$ of 28,000 B. H. P., and a speed of $20\frac{3}{4}$ knots. The Clyde pleasure steamer "King George V" which began service in September 1926, had a pioneer installation of high-pressure steam turbines, a boiler pressure of 550 lbs. per square inch and steam superheated to 750°F. She consumed slightly over one pound of coal per S.H.P. per hour. The H.P. turbine measuring little over $3'$-$6''$ in length and one ft. in diameter developed 600 S. H. P. In the turbine steamer "Orama" $630' \times 75'$, the oil consumption was only .785 of a pound per S. H. P. hour. A notable advance was made in the C. P. R. "Duchess" liners, $580' \times 75'$, in which the installation of Yarrow boilers of 370 lbs. pressure resulted in an oil fuel rate of .64 of a lb. for all purposes. These were pioneer installations of high pressure steam in large liner work. In the "Empress of Japan" $662' \times 83'$-$6''$, and "Empress of Britain" $730' \times 97'$, now building, the boiler pressures will be slightly higher and it is expected that the fuel consumption rate for all purposes will be .6 of a lb. or less. In the C.P.R. coal burning cargo carriers of the "Beaver" class, having watertube boilers and superheated steam, the fuel consumption rate is 1.1 lbs. with mechanical stokers, and 1.15 with hand firing. Pulverised fuel was tried out successfully on one boiler of the Blue Star Liner "Stuartstar" and has now been applied to a second boiler. The "Berwindlea" launched in the Clyde in May last, is the first vessel whose boilers have been designed *ab initio* to use pulverised fuel. Diesel-electric propulsion has been installed in the oil-tanker "Brunswick" 469′×63′, 13,000 tons deadweight, and the turbo-electric drive on the large P. and O. passenger liner "Viceroy of India" 612′×78′. The latter vessel has Yarrow boilers of 350 lbs. pressure and steam superheated to 700°F., the installation being similar to those of the "Duchesses" already referred to. Her propelling motors develop 17,000 S.H.P., and she has passed through very successful trials and is now on service. The Anchor Liner "Britannia" has recently had a Bauer-Wach exhaust steam turbine added to her quadruple expansion reciprocating engine and reduced her fuel consumption by about 20%. The engine of the single screw cargo steamer "Boniface" 407′×53′-6″, has been specially designed to include an exhaust turbine and used 24% less fuel when using the turbine than when running without it. The Fairfield Company accomplished a notable feat when they replaced the machinery of the "Empress of Australia" by a more modern installation and reduced the daily consumption from 200 tons at 16½ knots to 145 tons at 18 knots. A similar conversion has recently been carried out on "Empress of Canada."

These are some of the things which have been done. For the future there are rumours of 1000 ft. liners to be propelled by electric motors and there can be no doubt that the resources, skill, and enterprise of the British shipowner and shipbuilder, of which I have endeavoured to give an outline, will bring such projects to a successful issue.

Pulverized Coal for Marine Boilers.

(Paper No. 426)

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An attempt will be made in the following pages to sketch briefly the events leading up to the recent successful introduction of pulverized coal for fuel on sea-going vessels and to give a rough outline of the present status of this important development.

The author feels that this new art is still in an experimental stage. It is making rapid progress under the stimulus of wide interest focussed on the subject and is fast approaching the point of becoming a definite science. The time is not ripe, however, to advance definite conclusions concerning various important factors which go to make up a sea-going pulverized coal installation. Certain fundamental principles of course apply, such as, for example, the necessity of designing the piping for the carrier air so that sufficient velocity will be maintained to prevent the pulverized coal from settling out. The velocity must in general exceed 2800 ft. per minute. Such things as this and the importance of using draft gauges and CO_2 recorders have come down to us from the earlier development of pulverized fuel equipment in the stationary field.

To go further than this and to say positively, for example, that "one pulverizer per boiler gives the best results" is not yet justified by the state of the art. There are already a number of systems for burning pulverized coal on board ship which differ quite radically in their fundamental approach to the subject. They vary between the widest possible limits and cover all sorts of ideas from that of bunkering the vessel with coal already pulverized on shore, to the use of an individual pulverizer for every fire door of every boiler in the ship. It will be some years before the merits and relative value of these various systems can be known and time alone will settle the question of which is best, if in fact a number of different systems do not finally survive. It will be time enough then for superlatives.

The New Day.

On Thursday, December 8th, 1927 there appeared in the ancient harbor of Rotterdam a freight steamer bearing the house flag of the American Diamond Line and known as the S. S. Mercer. This vessel had come across the Atlantic from America burning pulverized coal under her boilers, thus demonstrating for the first time in history that this fuel could be used commercially in the restricted furnaces of the boilers of a sea-going vessel.

There were on board the Mercer on this important occasion a number of people who had been identified with bringing to a successful conclusion this extraordinary demonstration.

Mr. Carl J. Jefferson, representing the U. S. Shipping Board, the owners of the vessel, had had an important part in carrying out the tests on shore which had led to the Shipping Board's decision to try pulverized coal at sea. It was he and his assistants who had planned the arrangement of the pulverized coal equipment on the Mercer and supervised the installation thereof and who represented the Shipping Board during the preliminary trial trips which had taken place in America.

Commander Joseph S. Evans, U. S. N., had had charge of the Fuel Oil Testing Plant at the Philadelphia Navy Yard and in cooperation with Mr. Jefferson had directed the taking of the data and the supervision of the evaporative tests made there. Commander Evans had participated in the preparations for sending the Mercer to sea and represented the U. S. Navy during the vessel's first epoch making voyage across the Atlantic.

Mr. Robert C. Vroom, M. E., represented the Peabody Engineering Corporation, inventors and builders of the short flame turbulent burners which were the outstanding feature of the successful attempt to burn pulverized coal under marine conditions. Mr. Vroom had participated in the design and installation of the equipment for the shore tests on the Mercer, and had been in charge of the operation of the burners throughout all of the preliminary work extending over a period of many months. He participated in all the trial trips and was on board the Mercer during the first complete round trip from New York to Rotterdam and return.

Mr. John V. Pyle, represented the Kennedy-Van Saun Manufacturing & Engineering Corporation, builders of the Kennedy-Van Saun pulverizers which were installed on the Mercer and which had been tested during the preliminary experiments on shore at Philadelphia. Mr. Pyle had charge of the operation of the pulverizers during the trial trips and the first voyage of the Mercer from New York to Rotterdam.

Mr. Charles H. Jack of the Fuel Conservation Section of the Shipping Board had gained a valuable experience during the pulverized coal experiments prior to and including the tests of the equipment later installed on the Mercer and it was quite fitting that he should be selected as an observer for the Board during the first trans-Atlantic round trip of this vessel.

Mr. E. L. Johnson and Mr. W. Richards also made the first round trip as observers for the Board. Mr. C. J. Pierce was the Chief Engineer of the vessel, in charge of all machinery.

Through the courtesy of the owners of the vessel, (the U. S. Shipping Board) and the operators (the American Diamond Line), the vessel was thrown open for inspection almost immediately upon her arrival at Rotterdam. She was visited by some 300 engineers and others interested in this new development and in this group were representatives from seven different nations. This indicates clearly the immense interest aroused by the long heralded reports of the use of pulverized coal on the Mercer and her final arrival in a Continental port.

The very frankness with which the boiler room of the Mercer was thrown open to visitors without any attempt to cover up the results of certain slight mechanical defects which had developed, notably difficulty in keeping the distributors in operation, rendered it easy for those who were disposed to make adverse criticism, to point out unimportant defects and to prophesy utter failure for this American innovation. It is rather natural that the most severe criticism came from certain individuals who had been more or less prominently identified with the burning of pulverized coal in stationary practice on shore. It seemed to them impossible that the combustion rate per cubic foot of furnace volume could be increased to three or four times the combustion rates they were familiar with. And this had been just what had been done in order to adapt pulverized coal for marine use. There were, however, others who graciously acknowledged the success of this experiment and who saw through the little cloud of small troubles and were able to visualize the enormous possibilities of the new system and its importance to the Merchant Marine of all countries. The author wishes to acknowledge the many complimentary messages which he himself received at this time.

Tests at the Philadelphia Navy Yard.

In 1920 Mr. F. W. Dean of the U. S. Shipping Board while supervising some tests with oil fuel on a water tube boiler at Erie, Pa., had occasion to witness the burning of pulverized coal under some stationary boilers. It occurred to him that pulverized coal might be applied to marine use and he suggested that experiments be made with that end in view. A Scotch marine boiler was then being used at Chester, Pa. for testing various oil burners and under the authority of Mr. Angelo Conti, Chief Engineer of the Shipping Board, this boiler was designated for use with pulverized coal burning equipment. Mr. Dean resigned from the service before this boiler was ready for the tests and Mr. C. J. Jefferson was appointed to carry out the actual experimental work.

The pulverizers and burners were furnished by the Aero Pulverizer Company. The pulverizer was representative of the best type of impact machine on the market at that time. The burners were of the so-called "stream line" type such as prior to 1925 were commonly used in stationary practice where a long flame is permissible. The tests were not a success owing to unsuitable burner design and the consequent impossibility of burning the necessary amount of coal in the restricted furnace volume of the Scotch boiler. The tests were therefore abandoned.

In 1925 Capt. Charles A. McAllister, who had a few years before been appointed chairman of a special committee of the U. S. Shipping Board to study methods of reducing fuel costs on government vessels was attracted by reports of an improvement in burning pulverized coal on shore which had originated with Mr. Gustave Kaemmerling, Jr. of the Fuller Lehigh Company. This was known as the " Well " furnace and had demonstrated that high combustion rates of fuel per cubic foot of furnace volume might be secured by creating high turbulence in the furnace. Although the Well type furnace as then constructed had not been a practical success on shore on account of high maintenance costs and as later admitted by Mr. H. W. Brooks, Consulting Engineer of the Fuller Lehigh Company, the modification of the design proposed for use under the Scotch boiler was not suitable for practical use at sea, the Fuller Lehigh Company at the request of the Shipping Board installed their burners at the U. S. Fuel Oil Testing Plant at the Philadelphia Navy Yard under a Scotch boiler which had been used at that plant for tests on oil fuel. The cooperation of the Navy Department was secured and for several months during 1926 experiments were carried out which while unsuccessful for the purposes intended, clearly demonstrated the value of high turbulence in the furnace.

About two years previous to that time, the Peabody Engineering Corporation had made some modifications in the design of a combined oil and gas burner manufactured by them, whereby pulverized coal suspended in a current of air was substituted for the gas—so that it became possible to burn either coal or oil in the same burner. This equipment had been installed in 1925 in connection with stationary boilers at a large power plant in New York. After witnessing the performance of this burner and finding that it gave a short turbulent flame, the Fuel Conservation Committee, under the leadership of Capt. McAllister, invited the Peabody Engineering Corporation to furnish a set of burners for testing at the Philadelphia Navy Yard. In the interim while the Fuller Lehigh burner was being tested the Peabody burner was tried out in a single furnace Scotch boiler through the courtesy of the Continental Iron Works, builders of the well known Morrison suspension furnace. The test proved successful and on the completion of the Fuller Lehigh experiment, the Peabody equipment was installed at Philadelphia.

The burners were put into operation on December 22nd, 1926, just about one year before the Mercer arrived at Rotterdam. The demonstration was highly satisfactory and it is believed that this was the first occasion when a Scotch boiler of the type used on board ship, without an enlarged or extended furnace, had been successfully operated with pulverized coal. The results of these tests have been so often published that they will not be reproduced here. They included, however, a ten-day continuous test under varying conditions which might be encountered in a vessel at sea in so far as these could be simulated in a shore plant. The successful outcome of the tests encouraged the Shipping Board to proceed with an installation of pulverized coal burning equipment on board ship and the Mercer was selected for the purpose.

Burner Design.

The burner which had thus solved the difficult problem of burning pulverized coal at very high rates of combustion (sixty thousand BTU per cubic foot of furnace volume per hour) in the entirely water cooled furnace of the Scotch boiler, resulted from an adaptation of the principles used in burning oil fuel at high rates such as pertain to naval practice. The Peabody Oil Burner together with its special form of Air Register had been developed for just this purpose and combustion rates had been reached with oil approaching a heat release of three hundred thousand British thermal units per cubic foot of furnace volume per hour. In 1924 this burner had been used in combination with a new design of gas burner whereby gas also was burned at very high combustion rates. The production of a turbulent effect in the furnace was one of the chief characteristics of this burner and it was believed that this same principle might be applied to the burning of pulverized coal inasmuch as this fuel suspended in air acts in many respects not unlike a gas. Experiments proved this to be the case.

The burner has often been described and will not be enlarged on here. The principle of its operation, however, is well stated by Mr. Jefferson in one of his articles which was quoted by Engineer Captain Brand in his paper before the Institute of Naval Architects at Cambridge, England in 1927 as follows:—"The Peabody Burner . . . is a turbulent burner using both primary and secondary air. The primary air carries the coal to the furnace under comparatively low air pressure and it enters the furnace with the coal through an annular slot in a small . . . casting located at the mouth of the furnace. This arrangement is in effect an infinite number of nozzles directing a coal stream against the stream from another infinite number of nozzles. These infinite number of streams cutting each other produce a violent turbulence which is further increased by the admission of the secondary air at the center of the burner."

Although the Peabody Combined Oil and Pulverized Coal Burner came into existence as a separate development through the intermediary of the gas burner, it should particularly be stated that Admiral Charles W. Dyson, U.S.N., retired, was the first to perceive the advantages of using the air register such as is used with oil burners, for burning pulverized ocal. He patented a device for this purpose in 1922, two or three years before the effect of turbulence on the combustion of pulverized coal became generally appreciated. The design was in several respects like the final development.

As the years go on and various methods of applying this principle to pulverized coal burners are developed, the importance of the burner in solving the problem of obtaining high combustion rates in a Scotch boiler furnace must not be forgotten. Admiral Dyson's far visioned insight into an undeveloped field and the application of his ideas to the first practicable burner for burning coal for this service stands as an engineering achievement of the highest quality.

Limitations of a Republic.

A widespread belief exists not only in America but in England, Japan and in other civilized nations, that the United States Shipping Board in cooperation with the U.S. Navy together with the personal efforts of certain government officials were the sole channels through which this long anticipated application of pulverized coal to the Scotch boiler furnace was financed and accomplished. This is not the case. Admiral Dyson's participation in the development was as a private citizen and it has already been pointed out that the development of the burner which had such a vital effect in bringing about the success of the experiment was developed by private enterprise and private capital. The Government furnished the boiler, etc., the feed pump, scales for weighing coal and water and the majority of the men for conducting the tests and tabulating the data. The burners, boiler fronts, piping, distributors, pulverizers and conveyors for delivering the coal from the weighing apparatus into the pulverizer hopper were all supplied and installed at private expense. Even the fuel for all the preliminary work and for the final test was furnished free of cost by private companies engaged in the business of marketing coal.

The Shipping Board indeed did a great work in initiating and financing the fitting up of the Mercer, albeit they were able to buy equipment at cost, and for many months the services of skilled engineers were furnished gratis by the manufacturers. It is not too much to say that without the enthusiasm of Captain McAllister and his associates, and a moderate expenditure by the U. S. Government, this advent of pulverized coal on the sea would have been indefinitely delayed. But citizens of other countries who feel that their governments have been backward in promoting important developments of this kind may like to know that the Limitations of a Republic make it quite as difficult to secure funds for carrying on development work, as is the case with the governments of their own countries.

Advantages of Pulverized Coal.

The benefits of using coal in the pulverized form have often been enumerated and will only be briefly summarized here. First; the saving in labor brought about by reducing the operation of burning coal on board ship to the same category as that of burning oil. Second; continuous steaming at full pressure and elimination of the loss in speed due to cleaning fires. Third; probable increase in net efficiency compared with any other method of burning coal, and in consequence lower operating cost. This is partly brought about by the use of highly preheated air for combustion, thus reducing waste heat losses. Fourth; possibility of using slack coal of an inferior grade which could not be fired by any other method.

The writer feels that the first three items are vital factors. The last item while having considerable appeal due to low first cost of fuel must be carefully evaluated against the economic losses comprised in transporting, bunkering, pulverizing and burning fuel which contains a large amount of material which has no value in producing heat and which on the contrary results in other economic losses besides those mentioned. It is to be remembered also that any great increase in use of a low priced article has the effect of increasing the market price of that article. The use of inferior fuel in pulverized form for marine service therefore appears to possess advantages of rather narrow and questionable limits.

Type of Pulverizer.

A great variety of pulverizing mills is available from which choice may be made for Marine purposes. In some form or other and with varying effect they all utilize the principles exemplified in the time honored mortar and pestle of the apothecary, that is a combination of crushing and attrition. Crushing may be brought about by impact and "impact pulverizers," so called, are provided with revolving blades or hammers which strike the coal and shatter it into small particles. These particles are then further reduced in size by attrition produced by eddies of air within the mill whereby the coal particles rub against each other or against the metal surfaces. "Roller mills" crush the material and supplement this by a sliding or grinding effect. The "cannon ball" type consisting of large metal balls (10″ to 20″ in diameter) revolved within the casing, accomplishes the purpose in much the same way, centrifugal force also entering into the process. The so called "ball tube" type operates like the old fashioned "rumbler" for cleaning castings. A large quantity of steel balls 1″ to 3″ in diameter (several tons in weight) are placed in a horizontal cylinder which by its rotation causes the balls to be lifted up and "cascaded" down again by gravity, thus pulverizing the fuel which is fed into the cylinder and mixed in among the balls. Practically all mills now operate on the "air swept" principle, whereby a slow air current removes the dust when it has become sufficiently fine.

The pulverizer installed by the Peabody Engineering Corporation for the test at Philadelphia, was of the impact type, loaned for the purpose by the Furnace Engineering Company. Unfortunately, this particular pulverizer was not representative of the latest development in this type of machine, which is noiseless and reliable and easily gives proper pulverization. Hurried attempts to improve the operation of the pulverizer were made but these were unsuccessful and public and somewhat extravagant accounts have been given of the tremendous vibration experienced after fitting new hammers and the excessive amount of work required to get the machine properly balanced.

It happened therefore, that the impact type of pulverizer came to be viewed with suspicion by the officials of the Shipping Board and at their instigation the impact type of machine was removed and a pulverizer of the so-called "ball tube type" was installed by the Kennedy-Van Saun Manufacturing & Engineering Corporation. This machine gave excellent results in the matter of pulverization. Remarkable fineness was continuously and steadily obtained.

Much import at the time was made of the fact that the Kennedy mill operated at 35 revolutions a minute whereas the Furnace Engineering mill operated at 1500, and it was freely prophesied that a machine operating at 1500 revolutions per minute was entirely unsuited to use on board ship. It is a commentary on this view that we are now anticipating the early addition of a vessel to the pulverized coal burning fleet, which will be fitted with pulverizers of the impact type operating at 3600 revolutions per minute. Furthermore, it is interesting to note that the first English vessel fitted with pulverized coal, namely, the Stuartstar, is equipped with pulverizers of the impact design. Thus while the early antagonism displayed toward the impact pulverizer undoubtedly reacted for a time against this type of machine, it is now rapidly being recognized as a formidable contender for first honors in the marine field.

Power for Pulverizing.

It is the author's opinion that the type of pulverizer which will finally survive for marine work will be that which is able to properly prepare the fuel with the least consumption of power and lowest maintenance charges. It is obvious that if certain economies may be secured by the use of powdered fuel in the actual evaporation of the water into steam, these may be partly dissipated or entirely wiped out by increased use of steam for preparing the fuel and delivering it to the burners. Interest on the cost of additional equipment must also be considered, and in the last analysis it is the "net economy" produced by the pulverized coal burning plant which will regulate its use. Coupled with lightness, compactness and dependability (the latter item of course being of utmost importance) the coal pulverizer, if it is to remain on the ocean, must prepare the fuel at low cost of power. And not only must the power be low but when we speak of the cost of power, we must consider the method of its generation. Three or four per cent of the power of the main engine for driving the pulverized coal burning auxiliaries may very well represent what is required, but if this power is developed by simple slide valve engines or small uneconomical turbines, the steam consumption may easily run up to 15% or 20% of the total steam consumption of the ship. Such figures as these may very well jeopardize the commercial success of pulverized fuel for marine use.

Fineness.

Coupled with this factor of power consumed for grinding, is that of fineness of grind. Much has been written and said about the necessity of exceptionally fine grinding and the need of a large amount of so-called "superfines" for successful operation of pulverized coal in Scotch boilers. Mr. Jefferson, the foremost advocate of this theory describes "superfines" as follows:

"You can't keep fires on 40 mesh coal. You must have what we call superfines, and we don't know just how fine superfines are. . . . Superfines are things that go through 300. Maybe they go through 500 or 1000, I don't know, but they are so fine that the only way you can determine their percentage is by a microscopic study. There is a very definite limit that it is necessary to observe if you are going to maintain flame propagation, and if you are going to do it in a small furnace you must jumble that stuff up and jumble it up fast. Superfines and turbulence are both essential." Another author goes so far as to point out the alleged advantages of coal pulverized so finely that all of it will pass through a "500 mesh" screen.

These arguments in favor of "superfine" grinding leave out of consideration the extra cost of obtaining this exceptionally fine product, or else they touch upon it in a more or less perfunctory way. Finer grinding than is required in stationary practice is undoubtedly necessary for high combustion rates in small water cooled furnaces. The question is:—just how fine! If finer grinding required no additional power, that would settle the matter once and for all, but considering the way in which the power costs run up as the fineness of the grind is increased, it is obvious that there must be a balance somewhere, an economic balance between loss in unconsumed carbon at the stack (resulting in lower boiler efficiency) and loss in power consumed in the grinding mechanism.

The author believes that it is not so important that there be present a large percentage of "superfines" as that there be absent the large particles which would say, be caught upon a 40 mesh (420 micron) screen. The first tests in Philadelphia on the Peabody equipment when only about 92% of the coal passed a 420 micron screen indicated that the coarser ground material would easily ignite in the furnace. Superfines might assist ignition but when this occurs promptly, it is obvious that the coal need only be ground to a size where the particles will be consumed in their passage through the furnace. No further grinding is necessary and in fact it is a waste. Present opinion seems to be drifting toward an agreement that if the fuel be ground to a point where 100% will pass a 420 micron screen it will give satisfactory results. Probably coarser coal than this will be satisfactory, for it is well understood that mixed with these coarser particles there must be a large percentage of very much finer material. This question of fineness is one which cannot be definitely answered until we have further reports from operators of pulverized coal burning vessels. We do know now, however, that increase in evaporative efficiency must be balanced against increased cost of grinding whenever we attempt to correctly specify the fineness.

Screens for Classifying Pulverized Coal.

It is well known that the term "mesh to the inch" signifies the number of openings in a screen per linear inch. If the wire used in making a screen is the same size as the opening between the wires, this results in a very definite scale by which to measure and classify pulverized material. Such a screen, however, will have but 25% screening area and it has come to be the practice of manufacturers of screens to depart from this 50:50 ratio. The moment that this is done, the term "mesh to the inch" means nothing at all that can be used as a standard.

The American Society of Testing Materials has recommended a classification whereby the screen is designated by the size of the opening measured in microns. Obviously it is the size of the opening which is important in any comparison, and this nomenclature at once provides an international standard whereby we may all speak the same language when referring to pulverized coal. The substitution of the term "micron" instead of "mesh to the inch" and designating the screen by the number of microns comprised in the openings in the screen is heartily endorsed by the present author. It is believed the custom would have the effect of standardizing screens all over the world or if not so effective as that, would at least give us a means of comparing various reports on fineness.

Distributors.

The division of a stream of carrier air and pulverized coal into more than two streams, each carrying the same amount of fuel, is a rather difficult problem. The device furnished for distributing the fuel to the three furnaces of the Scotch boiler used in the tests of the Peabody equipment at Philadelphia, consisted of a series of blades rotated within a portion of the pipe delivering the pulverized coal and carrier air from the mill. These blades created a homogeneous mixture which passed to an expanding chamber from which the necessary branch connections led to the burners. This was the first distributor of the type and was therefore of an experimental nature. Its operation was improved by changing the number and shape of the blades.

This distributor was found quite satisfactory in the matter of distribution both during the test in Philadelphia and later on in the Mercer, but great difficulty was experienced in keeping the rotor from stopping due to plugging of the stuffing boxes. After considerable experimenting with packing, success in keeping the rotor in operation was finally achieved by delivering a small amount of air under pressure to the stuffiing boxes, thus preventing the pulverized coal from penetrating into the packing.

This arrangement was installed on the Mercer but used for only one trip as the Shipping Board decided to install a distributor which operated on a pneumatic principle. In this device streams of air under pressure from an outside source are delivered tangentially to a chamber in the piping containing the carrier air and coal. This arrangement causes the fuel to rotate within the chamber and thoroughly mix before being separated into the desired number of streams. The distribution thus obtained was about the same as that produced by the mechanical rotor, but the device possesses some other advantages as for example, reduction in pressure loss in passing the point of mixing, no moving parts, simple construction, etc.

The distribution problem has been met on the S.S. Stuartstar by use of the "loop system" passing entirely around the boiler room, and returning to the pulverizer. The desired amount of fuel fed to each furnace is taken off this pipe as it passes the furnace. This is not a new development except in its application to marine use. The system has been used with more or less success in stationary practice.

The distribution of pulverized coal is avoided in the S.S. West Alsek now being equipped by the Todd Shipbuilding and Drydock Company by using individual pulverizers for each furnace. This means that the unground fuel is distributed to each of the pulverizers and the entire product of each pulverizer delivered to its corresponding burner. A full account of the system is given in Mr. Brierly's article.

A very attractive plan for securing an even supply of fuel to each burner is not by means of a "distributor" as a separate part of the operation but by placing the pulverizer under pressure and taking the pulverized coal away by individual pipes leading direct from the pulverizer to each burner. The coal is thus "distributed" before its mixture with the carrier air is permitted to lose its homogeneity. This simple and logical method of splitting up the coal stream is well suited to the arrangement whereby one pulverizer is provided per boiler and is also particularly desirable in view of the fact that no coal dust passes through the fans, and the serious item of wear on the fan blades is eliminated. But the plan possesses an inherent element of danger which must not be overlooked, namely the possibility of leakage of coal dust from the pulverizer. It is not difficult to keep tight the joints in a pipe line, but the experiments on distributors have shown the difficulties which may be encountered with stuffing boxes and the idea of putting the pulverizer itself under pressure is certainly going to require ingenuity in design and great care in operation if it is to be made successful. The experiments now under way will be watched with great interest.

Driers.

Coal driers are frequently employed on land for drying the coal preparatory to delivering it to the pulverizer. Marine experience so far, has not indicated that such driers are necessary on board ship. It is common practice however, to divert a certain proportion of heated air from the Howden heater for sweeping the pulverizing mill. This hot air from the heater in passing through the pulverizer is effective in absorbing a very considerable amount of moisture and promoting the pulverizing process. It has the disadvantage that the moisture so taken from the coal is delivered to the furnace where it has to be evaporated and creates some loss in efficiency. The external drier on the other hand although complicated and liable to produce spontaneous fires, and rather expensive to install and operate, has the effect of feeding only dry coal to the pulverizer thus reducing cost of power for pulverizing and preventing moisture in the coal from entering the furnace. It is not impossible that some such device will sooner or later be introduced on board ship.

Air Preheaters.

The use of heated air for combustion possesses many advantages in burning pulverized coal and may be considered as almost an essential part of a Marine installation. If waste heat is utilized for the source of heating, as is usually the case, there is a direct saving of fuel. The author has long felt that the Howden heater was one of the important inventions of the 19th century. Without it, it is probable that the Scotch boiler could not have maintained its time honored popularity in the merchant marine, because of sheer lack of efficiency. The recovery of waste heat and improved combustion conditions due to the Howden heater have certainly been important factors in its survival.

So far, the introduction of pulverized coal has taken the Scotch boiler and the Howden heater as it found them, but it is probable that not only will boiler and furnace design be improved but special attention will be given to air preheaters. Highly heated combustion air possesses special advantages in connection with pulverized coal. There is almost no limit to the temperature which may be utilized and the very fact that preheated air speeds the rate of combustion will make it possible to burn coarser fuel and thus save waste in the grinding process. It is also likely that hot combustion air may make possible the use of fuel of low volatile content and other grades of coal which could not otherwise be burned satisfactorily. We have probably only touched as yet on the economies to be brought about by further development of air preheaters.

Disposal of Ash.

It is customary in stationary practice to assume that about 25% of the ash in the pulverized coal with remain in the furnace, the remainder being carried through the boiler and uptake to the chimney, or collected in pockets from which it may be periodically removed. Prior to the actual experimenting at Philadelphia it was a moot question therefore, whether or not trouble might be experienced due to ash collecting in the furnace when burning pulverized coal in the Scotch boiler. Mr. Jefferson advanced the theory that as soon as refuse in the furnace had built up to a certain height the velocity of the gases would thereafter sweep the particles of ash away from the furnace and out through the boiler. This turned out to be true to a very considerable extent and in fact the clearing process improved as the rate of combustion increased. So that during the 240 hour test at Philadelphia, the ash gave very little trouble, the accumulation in the furnaces at the end of the test not being at all excessive. In building the equipment for the S. S. Mercer, it was thought desirable to install an ash door below the burner and this was found to be useful from time to time.

In this connection, it is believed that the removal of the ash from the furnace by the so-called "wet" method, i. e., in a molten state, possesses possibilities for marine work. In this way, the furnaces could be kept almost entirely clear of slag without any interference to operation whatever. The degree of success obtained by this process will depend to some extent on the fusing temperature of the ash.

The removal of the fine ash which settles in the combustion chamber and in the tubes, presents no very serious difficulties from a physical standpoint. The consumption of steam in the "soot" blowers however, is a matter which should be given careful study as it is a direct percentage of loss to be computed on the cost of fuel.

Passenger Ships—Dust Removal.

Cinder catchers or dust collectors which are frequently used on shore for removing the fine ash from the waste gases escaping from the chimney are, as designed at present, cumbersome and difficult to install on board ship. Even with high funnels, however, there are times when with a following wind of the right velocity, the ash escaping from the top of the funnel must inevitably be deposited on deck. This may not be of vital importance in connection with a freight vessel, but would be a serious handicap in connection with passenger ships. It is probable, therefore, that passenger ships will continue to use fuel oil in place of pulverized coal for a considerable time to come. In the meantime, the development of the proper dust collector for marine use offers a wide field for research.

Water Tube Boilers.

The water tube boiler offers a somewhat easier problem in combustion than does the entirely water cooled furnace of the Scotch Boiler. On the other hand, the difficulty of removing the ash from the tubes in the water tube boiler is somewhat accentuated as compared with the Scotch boiler. Already, however, the burning of pulverized coal in the marine water tube boiler is making progress and the Babcock & Wilcox tests have shown that excellent combustion and high efficiency can be obtained. It is probable that the innovation of high steam pressures which is making more and more rapid progress as competition spurs on the shipowner to seek for reduced cost of operation, will stimulate the use of water tube boilers. Pulverized coal will therefore not be limited to Scotch boiler ships but will be used in conjunction with the economic advantages of higher working pressures.

Types of Installation.

Several arrangements of pulverized coal burning equipment for steam vessels have been suggested, only the first of which has as yet been given a thorough test in sea-going service.

1. Delivering coal to the bunkers; crushing it (if necessary) and delivering the crushed coal to a ready use bin; feeding by gravity to the hopper of the unit pulverizer and delivering the pulverized fuel by means of carrier air direct to the burners.
2. Delivering coal to the bunkers; crushing; pulverizing and delivering the pulverized fuel to a ready use bin (or so-called "unit bin") of comparatively small size (say sufficiently large for four hours steaming at maximum capacity); feeding the pulverized coal by means of individual feeders to pipes leading to the individual burners; and delivery to the burners by means of carrier air.
3. Pulverizing on shore (or possibly floating pulverizing plant); delivering to the ships' bunkers in pulverized form; removing as needed to a ready use bin by means of carrier air or a suitable conveyor and feeding to the individual burners as in Arrangement 2.

The first arrangement is known as the "Unit" system in distinction to the "Bin" or "Storage" system. It was used on the Mercer, Lingan, Stuartstar, Hororota and will be used on several vessels now being fitted for burning pulverized coal.

It has the advantage of being self contained and thus capable of world wide use, also it is simple of application and with proper conveying machinery the labor of bunkering and "trimming" may be reduced to a minimum. The power required for pulverizing has however, in existing installations of this system, proved to be excessive.

The second arrangement was suggested by Mr. William E. Pearson of the Bethlehem Steel Corporation and was described and illustrated in the author's paper on the "Burning of Hydrocarbons Under Marine Boilers," presented before the Society of Naval Architects and Marine Engineers, November, 1928. It contemplates the use of a large size pulverizer capable of great economy of power in the grinding process, the idea being that the pulverizer would be operated at maximum capacity for a part of the time only and then shut down. It has the advantage of being flexible as to the number of burners used, and for operation in port at low loads without running the pulverizer.

The third arrangement is known as the Brand System and was one of the earliest suggested. It has been widely advertised but the general opinion in shipping circles appears to be very well described in the following remarks made by Sir Eustace Tennyson D'Eyncourt before the Royal Society of Arts in London on January 11, 1929.

"In the adoption of such a system there is first of all the question of danger of explosion when powdered fuel is used in this manner, and Captain Brand proposes to eliminate this danger by the introduction of inert gases from the funnel, thus guarding against the existence of an explosive mixture. I cannot help thinking that there will be great difficulty in making any such arrangement practicable. On the other hand, I am doubtful whether the danger is very great, and am inclined to class it with the danger bogey which was so prevalent in the minds of many who feared the adoption of oil fuel; and which in the end proved to be easily provided for. The chief objection to the use of powdered fuel in bulk appears to be the difficulty of storage, involving not only arrangements in the ship's bunkers, but also the provision of tanks on shore for various coaling stations; with special provision in both cases to keep the powdered fuel absolutely dry. This provision would involve an enormous outlay and take a long time to establish—just as the arrangements for oil fuel have been gradually developed with storage tanks at various ports all over the world.

"In the meantime, until we learn more about the use and storage of powdered fuel, I cannot help thinking that the most practical method of using pulverized coal for ships is to carry the pulverizing plant on board and to pulverize the coal as it is required."

It appears to the present author that the high first cost of gas tight bunkers and the difficulty and expense of maintaining them in service on board ship may be a serious handicap for this system, especially as in the last analysis the ship owner must pay for the pulverizing plant on shore. If, however, the enthusiasm of the advocates of this system prevails and the by-no-means-impossible featof constructing pulverizing coaling stations over the face of the globe becomes an accomplished fact, the Brand System will be a strong contender for first honors in cost of pulverizing. This is no mean advantage and coupled with this the system provides the possibility of "bunkering" through pipe lines as in the case of oil fuel.

Range in Combustion Rate.

Load conditions at sea (in merchant ships at least) are likely to be much more uniform than in land requirements. In docking however, a severe test is placed on flexibility of combustion to meet the fluctuating demand for steam. It has been stated that the Unit System (Arrangement 1 above) is not adapted to high flexibility. Experience shows however that it is very flexible. Commander Evans reported that when the Mercer first arrived at New York the operating staff in the boiler room answered 32 bells in 62 minutes without resort to the oil burners, without blowing safety valves, and without trouble of any kind.

Inland-River Tow Boats.

The use of pulverized coal on river steamers preceded by a few months its introduction to sea service. This was due to the initiative of Mr. A. R. Wurtele, Chief Engineer of the U. S. Inland Waterways Commission. While the Shipping Board was experimenting on land, he began experimenting on the Mississippi River.

The Tow Boat "Illinois" was converted to pulverized coal in 1927 and made her appearance in the Fall of that year, using water tube boilers and Kennedy-Van Saun pulverizers and burners.

While the author knows of no reliable comparative figures concerning coal consumption or conditions of operation on this boat, the experiment was sufficiently encouraging to warrant further use of pulverized coal.

The Tow Boat—" Dwight F. Davis " had a successful trial on the Kanawa River, on April 17, 1929. This is the first vessel to be specially designed and built to burn pulverized coal. She is to operate on the Warrior River in direct comparison with two sister vessels, one fitted with a Diesel Engine and the other with Electric Drive. The Davis which is fitted with two triple expansion engines, Babcock & Wilcox boilers and pulverized coal burners, and the newly developed rotating table type of pulverizer, is destined therefore to play a conspicuous part in demonstrating the relative value of burning coal in pulverized form.

One feature of the equipment of interest is that the air for " sweeping " the mill is introduced under pressure so that no coal passes through the primary air fan. This also permits the division of the pulverized coal and carrier air directly at the pulverizer and eliminates a separate distributor.

Truth about the " Lingan."—The Collier Lingan owned by the Dominion Iron and Steel Company Ltd., of Canada, a subsidiary of the British Empire Steel Companyard Ltd. was converted from hand firing to pulverized coal by the Halifax Shipyards Ltd. in the Spring of 1928, in accordance with plans prepared by Mr. Edwin C. Bennett. The vessel is fitted with three Scotch boilers each having three furnaces, the arrangement followed in regard to pulverized coal equipment being similar to that on the S.S. Mercer.

In addition to the nine Peabody Burners the following pulverized coal burning equipment was installed:

Two Kennedy-Van Saun pulverizers each with a thirty H.P. vertical slide valve reciprocating driving engine.

Two pulverizer feeders each with a one half H.P. two speed driving turbine.

Two primary air fans each driven by a seven and one half H.P. turbine.

Three Peabody distributors each driven by a one and one half H.P. turbine.

After a very satisfactory sea trial off the Harbor of Halifax the Lingan proceded to Sydney, N.S., burning Pictou coal containing (on dry coal basis) 30% volatile matter and 13% ash, Btu. per lb. 12875, fusing point of ash 2300° F.

The ship arrived at Sydney on schedule time all machinery working perfectly. At Sydney the vessel was bunkered with another grade of coal as follows (on dry basis), volatile matter 35.0%, ash 7.8%, Btu. per lb. 13650, fusing point of ash 2000° F. This coal immediately began to make trouble due to premature ignition in the furnace throat which caused overheating and coking at that point. The furnaces had been fitted with fire brick linings extending 18" back from the fire brick wall, no other refractory material being installed. This was the same arrangement which had been used with success on the Mercer.

Mr. R. C. Vroom and Mr. O. Milne, the Engineers representing the manufacturers of the burners, at once began changing the fire brick lining and modifying the furnace throat to delay slightly the ignition of the fuel and permit the coal to pass into the furnace proper before taking fire. In a surprisingly short time they eliminated the trouble, the final arrangement consisting merely of leaving a 9" water cooled space between the front furnace wall and the 18" refractory lining.

The vessel proceded to Montreal without any trouble notwithstanding some intermittent action in the turning of the distributor rotors. It was evident however that the two-speed feeders on the pulverizers, neither of which was correct for the load, caused an improper supply of fuel which resulted in excess air at one rate of feed and heavy smoke at the other, a condition not conducive to economy.

The trouble with the distributors could not be overcome although every conceivable kind of packing was tried to prevent the coal dust from penetrating the bearings. Mr. Bennett had refused to permit the use of a small blower for delivering a slight amount of air to the packing, as had been done with complete success on the Mercer.

The Lingan continued in service for some months and operated between Sydney and Montreal on schedule time. It became apparent however, that it was impossible to reduce the coal consumption below 35 tons per day which, it was stated, had been the normal consumption when firing the same coal by hand before the pulverized coal equipment was installed.

It should be noted here that the power of the various auxiliary engines which had been installed with the pulverizing machinery aggregated more than 80 H.P. The power developed on the main engines was something less than 2000 H.P. so that the power of the coal burning auxiliaries was 4% or more of the main engines. But considering the steam used by these auxiliaries the estimated consumption is 16% or 18% of that of the main engine.

During the summer the manufacturers of the pulverizers attempted to improve the results by installing Kennedy-Van Saun burners on one of the boilers in place of the Peabody burners. In a short time the Kennedy burners were removed and the Peabody burners replaced.

After operating the vessel for about six months as a pulverized coal burner, the owners decided to remove the pulverized coal burning equipment and reconvert the boilers and boiler room to hand firing. Thus the second sea-going ship converted to pulverized coal burning showed that under some conditions pulverized coal on shipboard is a failure.

Review of Papers on Pulverized Coal for Marine Purposes.

During the last two years much has been written concerning the subject of pulverized coal for marine use. Among the important papers presented, the following may be noted.

PULVERIZED FUEL FOR MARINE PURPOSES
by Engineer Captain J.C. Brand, R.A.N.; Institution of Naval Architects; Cambridge, England, July 12, 1927.

This paper recounts the important pioneer work performed by Captain Brand in Australia 1915–18 culminating in the dock trial of H.M.A.S. "Sealark" fitted with Scotch boilers. Drawings are included showing the furnaces fitted with refractory lined "extensions" which projected into the firerooms and up in front of the tube doors, and apparently provided for about 8 ft. longer flame travel than the regular furnaces. The coal was pulverized on shore and dried before being placed in the ship's bunkers. The fineness of grind and power required for grinding do not seem to be included with the data relative to these tests, the curves which are given apparently referring to a later period.

The paper advocates the use of the Brand System and cites advantages from burning the pulverized residue obtained by the low temperature distillation of coal, particularly by the L and N System.

The pulverized coal burning experiments made by Clarke Chapman in England are referred to and illustrations given of the various extension furnaces employed by them.

The solving of the many problems involved in delivering pulverized coal to a ship's bunkers, getting it out again and feeding it to burners of his own invention, and burning it at a rate resulting in the liberation of nearly thirty thousand BTU per cubic feet of furnace volume per hour with a boiler efficiency of 72.4%, was a most noteworthy achievement. The writer knows of no one who even attempted such a thing prior to 1915 and the degree of success obtained by Captain Brand so early as 1918 is remarkable.

TESTS OF PULVERIZED COAL AS APPLIED TO SCOTCH MARINE BOILERS by Mr. Carl J. Jefferson, U. S. Shipping Board and Commander Jos. S. Evans, U.S. Navy; Society of Naval Architects and Marine Engineers; New York, November 11, 1927.

This paper includes an account of the tests made on the equipment installed at the Philadelphia Navy Yard by the Peabody Engineering Corporation; criticism of the impact pulverizer; substitution of the Kennedy ball tube mill; and a detailed report of the tests then carried out including the final 240 hour continuous test under conditions which would be obtained at sea.

No data are given for power consumed in grinding by the impact type mill. That consumed by the Kennedy mill, feeders, distributors and primary air fan, is stated in itemized form, from which the following figures are obtained.

Test number	I	II	III	IV	V
Coal per hour-lbs.	1205	1383	1599	1801	1975
Total K.W. per hour	20.57	20.73	20.83	20.71	21.18
K.W. per ton (2000 lbs.)	34.14	29.97	26.04	23.00	21.44
H.P. per ton	45.8	40.2	34.9	30.8	28.7

The "fineness" results are given as follows:

Test number	I	II	III	IV	V
Through 50 mesh—(297 microns) %	98.6	99.7	99.2	99.3	99.3
Through 100 " —(149 ")	91.2	94.6	92.0	92.9	93.1
Through 200 " —(74 ")	77.4	80.9	78.1	79.8	79.2
Through 300 " —(49 ")	68.0	70.4	68.5	69.3	69.9

PULVERIZED COAL TESTS OF A MARINE WATER TUBE BOILER
by Thomas B. Stillman, Jr; Society of Naval Architects and Marine Engineers; New York, November 11, 1927.

Record of tests on Babcock & Wilcox marine boiler using both impact and ball tube pulverizers, and burners adapted for use with pulverized coal using as a basis the Babcock & Wilcox oil burner. Curves are given which show the power consumption of mill, primary air fan, and feeder, at various rates of grinding, for the two types of pulverizers. These may be analysed as follows:

Coal pulverized per hour	800 lbs.	1200 lbs.	1500 lbs.
H.P. per ton (2000 lbs.) Ball tube mill	62 5	50.0	65.3
H.P. per ton (2000 lbs.) Impact mill	53.7	39.5	37.3

PULVERIZED COAL AFLOAT
by R. C. Vroom; Stevens Indicator (Stevens Institute of Technology); March 15, 1928.

Detailed description of the pulverized coal burning equipment on the S.S. Mercer.

SEA GOING PULVERIZED FUEL
by C. J. Jefferson; Journal American Society of Naval Engineers; Washington, D.C., May, 1928.

Describes experimental nature of the installation on the Mercer. Gives the author's views on the unsuitability of the impact pulverizer for marine use. Explains that a vessel which previously burned oil fuel was chosen for conversion to pulverized coal rather than a hand fired coal burning vessel because the combined oil and coal burners enabled either fuel to be used and insured the safety of the vessel if the pulverized coal experiment failed. States that only 100 bls. of oil were used on the first east-bound voyage of the Mercer and 90 bls. on the first west-bound voyage.

MARINE DEVELOPMENT OF PULVERIZED COAL
by Commander J.S. Evans, U.S.N. and Lt. Commander R.C. Brierly, U.S.N.R.; Journal American Society of Naval Engineers; August, 1928.

Greatly elaborated report of the tests at the Fuel Oil Testing Plant previously described; supplemented by an account of distributor experiments. Also description of the first three voyages of the Mercer. Reference also to preparation for the tests of the Babcock and Wilcox coal burning equipment.

The same copy of the Journal contains a "Discussion" by Rear Admiral C. W. Dyson, U.S.N. Retired, pointing out the failure of the paper to explain the importance of the burner which is treated incidentally, emphasis being placed by the authors only on the results of the tests and their own experiments.

PULVERIZED FUEL IN MARINE WORK
by Lt. Commander H. W. Brooks U. S. N. R.; World Power Conference; London, October, 1928.

Notes on history of pulverized coal and exhaustive resume of the papers read at the meeting of the Society of Naval Architects and Marine Engineers in 1927. Also an account of the first voyage of the S. S. Mercer.

DEVELOPMENT OF PULVERIZED FUEL FOR MARINE PURPOSES DURING 1927–1928

by Carl J. Jefferson, Commander Jos. S. Evans and Commander Joseph Broshek, U.S.N.; Society of Naval Architects and Marine Engineers; New York, November 16, 1928.

This paper is a continuation of the paper presented before the same Society the previous year by Mr. Jefferson and Commander Evans. The equipment as originally installed on the S.S. Mercer is described and the changes made therein. An account is given of the first five trans-Atlantic voyages of the Mercer and figures submitted showing that the net saving of coal on the first voyage to Rotterdam and return was 9.2% of the average fuel consumption of five sister vessels using coal fired by hand. This does not allow for oil fuel used, which reduces the saving to about 6.5%.

Low cost of oil fuel is the reason given why the Shipping Board made no further trials of pulverized coal burning equipment in its fleet. Installations on the S.S. Lingan and S.S. Stuartstar and the Mississippi Tow Boat "Illinois" are referred to. This is followed by a review of the work done at the Fuel Oil Testing Plant including experiments on distributors, development of the "blo-gun" whereby wear on fan blades may be eliminated, tests of the Babcock & Wilcox "Lodi" coal burner and finally the Todd System.

THE BURNING OF HYDROCARBONS UNDER MARINE BOILERS

by Ernest H. Peabody; Society of Naval Architects and Marine Engineers; New York, November 16, 1928.

Explains analogy between pulverized coal and oil fuel sprayed by mechanical atomizers, and shows that the successful attempt to burn pulverized coal in the small furnaces of marine boilers came about through the application of the same principles which had been applied to the burning of liquid fuel at high rates of combustion. Novel photographs are shown of various oil sprays produced by mechanical atomizers.

Emphasis is laid on the effect of steam and power consumption of crushers, pulverizers, primary air fans and soot blowers on the economy resulting from the use of pulverized coal, and the paramount importance of reducing the cost of operating these auxiliaries. Fineness of grind in its application to power consumption is discussed and whether or not superfines are required if the coarser particles passing a 40 mesh screen are absent.

A summary is given of the events leading up to the successful tests of the Peabody burners at Philadelphia and on the Mercer.

PULVERIZED FUEL AND ITS APPLICATION TO SHIPS BOILERS

by W. E. Woodeson; Society of Consulting Marine Engineers and Ships Surveyors; Liverpool, November 16th, 1928.

This paper describes the author's impressions gained on the initial voyage of the S.S. Stuartstar. Points out the savings possible by using cheap coal in the pulverized form and some of the disadvantages due to transporting the poorer fuel. Emphasises the revolutionary results produced by the "Woodeson Patent Burner," and refers to some other experiments, but does not mention the Mercer.

ADAPTATION OF PULVERIZED COAL TO MARINE BOILERS

by C. J. Jefferson and Commander J. J. Broshek, U.S.N.; Second International Conference on Bituminous Coal; Pittsburgh, Pa. November 21, 1928.

This is a land version of the paper presented by the same authors at the Society of Naval Architects and Marine Engineers on November 16, 1928.

FUEL FOR SHIPS

by Sir Eustace Tennyson D'Eyncourt; Royal Society of Arts; London, January 11th, 1929.

This paper treats the subject in general terms and emphasises the importance of "economy for the ship owner" as the outstanding factor in selecting the best fuel for the particular service in question. The use of oil fuel is referred to as the most convenient in every way and later in discussing pulverized coal the point is made that its use is an approximation to the use of liquid fuel.

PULVERIZED COAL FOR MARINE BOILERS ASHORE AND AFLOAT

by Commander J. J. Broshek and C. J. Jefferson; Washington Society of Engineers; February 20, 1929.

Description of the Fuel Oil Testing Plant at the U. S. Navy Yard, Philadelphia, and a review of the tests made with pulverized coal, from the first Fuller-Lehigh equipment to the Todd equipment. Also reference to the first Shipping Board test. The paper is profusely illustrated and includes an account of the Mercer and a list of marine pulverized coal burning installations to date.

Perhaps the most illuminating portion of this paper is the statement that the average coal consumption on the five hand-fired sister ships of the Mercer is 414 lbs. per mile, while the corresponding figure on the first voyage of the Mercer was 364 lbs. This shows a net gain of more than 12% over hand-firing which is somewhat better than elsewhere reported.

PULVERIZED FUEL TODAY

by Engineer-Captain J. C. Brand, R.A.N. (retired); Institution of Engineers and Shipbuilders in Scotland; Glasgow, March 12, 1929.

Exhaustive discussion including comments on fuels suitable for use in pulverized form; effect of moisture; effect of sulphur; various arrangements of equipment; theory of combustion; removal of slag; distribution, etc. States that no pulverizer is ideal for use on shipboard and gives requirements. Gives cost of marine pulverized coal installations varying from £7800 for 2500 H.P. to £20000 for 15000 H.P. and estimates power for operating the equipment as 2½% to 3% of main engines. Cites figures for power for grinding to 95% through 200 mesh screen as 19 S.H.P. for 700 lbs. per hour (54.3 S.H.P. for 2000 lbs. per hour) and for grinding to 92.4% through 200 mesh as 17 S.H.P. for 800 lbs. per hour (32.5 S.H.P. for 2000 lbs.). Emphasizes importance of Brand System and describes equipment. States that the first turbulent burner was developed in Australia in 1918. The paper contains many tables including test results and illustrations of furnaces for marine, stationary and industrial purposes.

POWDERED COAL FOR SHIPS

by Engineer Rear-Admiral W. Scott Hill, R.N. (retired); Institution of Naval Architects; London, March 22, 1929.

A comprehensive account of the subject to date by one who takes the position of an "outsider" but whose interest in and general approval of the use of powdered coal for ships, has lead him to give it very careful study and who therefore speaks with authority.

THE MODERN DEVELOPMENTS OF THE WATER TUBE BOILER FOR MARINE PURPOSES

by Arthur Spyer, Esq.; Institution of Naval Architects; London, March 22, 1929.

Contains pertinent notes on pulverized coal from the Babcock and Wilcox point of view. Four of the ten illustrations refer to pulverized coal.

THE TODD PULVERIZED COAL BURNER TESTS ON SCOTCH MARINE BOILERS

by R. C. Brierly, Lt. Commander U.S.N.R.; Journal Society of Naval Engineers; Washington, May 1929.

Description and illustrations of the Todd pulverized coal burning system, and detailed account of development work and tests carried out at the U.S. Fuel Oil Testing Plant at the Philadelphia Navy Yard. Mr. Brierly is assistant mechanical engineer at this plant, and has had authoritative contact with all the pulverized coal experiments made there to date.

The Todd System is unique in that a small high speed impact pulverizer with primary air fan is provided for each burner, making a complete self contained unit for each furnace. The "distribution" question is reduced to the problem of feeding the coal from the bunkers to the individual pulverizers, and maintaining a uniform and regular supply to the mill and from the mill to the furnace. Mr. Brierly explains the steps that led to the successful accomplishment of this result.

The pulverizers operated at 3600 R.P.M., the grind was very fine, and it is stated that an overall gross efficiency of 84.19% was obtained at an evaporation of 5.13 lbs. of water per square foot of heating surface from and at 212° F per hour, and a heat release of about 37000 Btu per cubic foot of furnace volume per hour.

Power requirements of the pulverizers varied from 40 to 60 H.P. per ton (2000 lbs.) depending on the load. This includes the primary air fan and feeder.

The paper is particularly valuable in view of the fact that coal from North and South America, the Philippines etc. covering a wide range in quality was tested experimentally, most of which was burned successfully.

Vessels fitted for Burning Pulverized Coal.

The following vessels have been fitted or are now being fitted for burning pulverized coal

River Tow Boat "Illinois"

Water tube boilers, Kennedy (ball tube) pulverizers and burners.

S.S. "Mercer"

Scotch boilers, Kennedy pulverizers, Peabody burners.

S.S. "Lingan"

Scotch boilers, Kennedy pulverizers, Peabody burners.

S.S. "Stuartstar"

Scotch boilers, Clarke-Chapman (impact) pulverizers, Woodeson burners.

Ocean Tug "Tamaqua"

Scotch boilers, Aero (impact) pulverizers, Peabody burners.

S.S. "Hororata"

Scotch boilers, Fuller-Bonnot (ball tube) pulverizers, Buell burners.

Tow Boat "Dwight F. Davis"

Water tube boilers, Fuller-Lehigh (rotating table) pulverizers. Babcock and Wilcox burners.

S.S. "West Alsek"

Scotch boilers, Todd combined (impact) pulverizers and burners.

Collier building in America.

Water tube boilers, Fuller-Lehigh (table) pulverizers. Babcock and Wilcox burners.

Collier building in America.

Scotch boilers, Fuller-Lehigh (table) pulverizers, Babcock and Wilcox burners.

Collier building in England.

Scotch boilers, Clarke-Chapman pulverizers, Woodeson burners.

S.S. "Swiftpool"

Scotch boilers, Rema (impact) pulverizers, Brand burners.

Conclusion.

Australia, the United States of America, England and Canada are the only countries which have as yet actively participated in the experiment of using pulverized coal at sea. Other countries are eagerly watching the results. Sweden, Germany, Holland, France, Italy and Japan stand ready to convert their merchant vessels to this new method of burning coal and to try it under their own flags, if there is even a remote promise of economic advantage. There will be failures. Other vessels will follow the Lingan. But no one who has been a witness to the easily operated, smokeless fires produced with pulverized coal, resembling in many respects the furnace and fireroom conditions obtained with liquid fuel, can believe that this remarkable new process can finally result in anything but unqualified success.

(Paper No. 426)

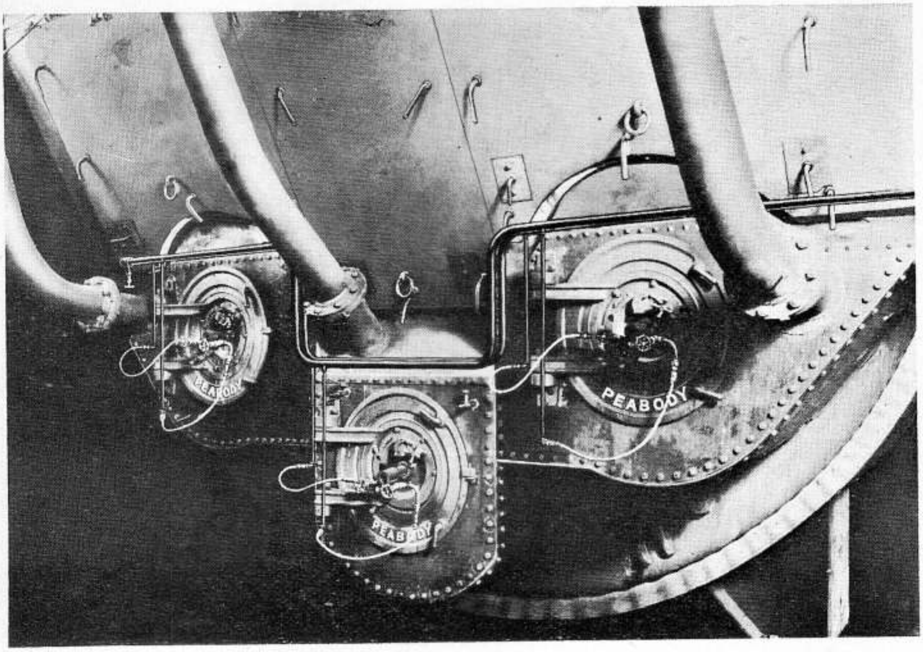

Fig. 1—Peabody Pulverized Coal System, Scotch Marine Boiler combined with Peabody-Fisher Wide-Range Oil Burners.

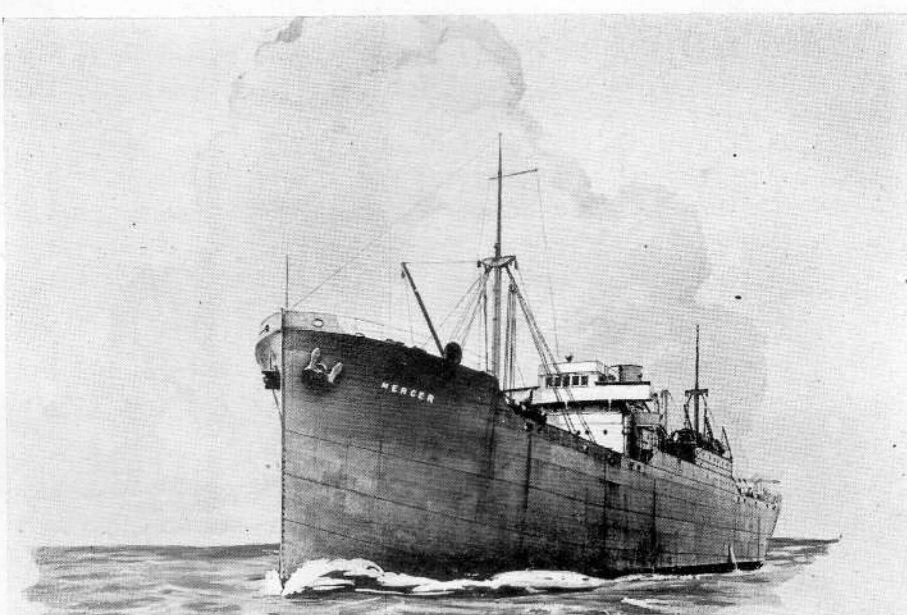

Fig. 2—Steam Ship "Mercer."

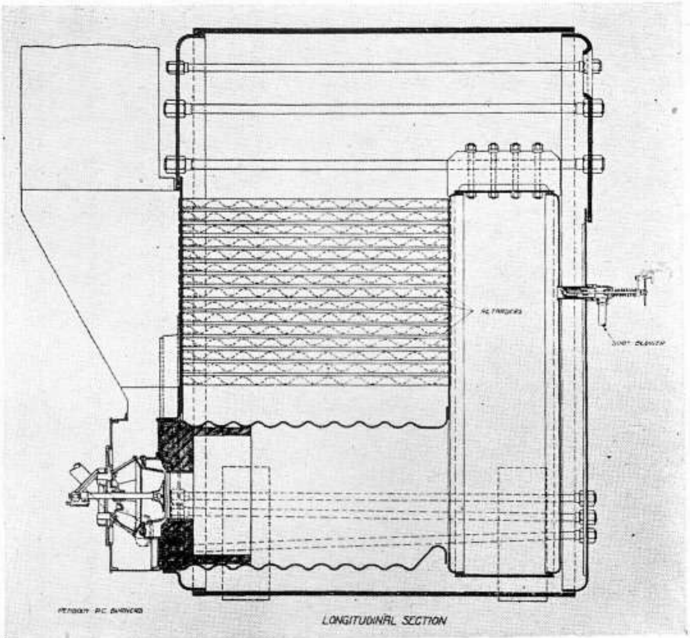

Fig. 3—Arrangement of Peabody Pulverized Coal Burner, Scotch Marine Boiler patented in the United States and Foreign Countries.

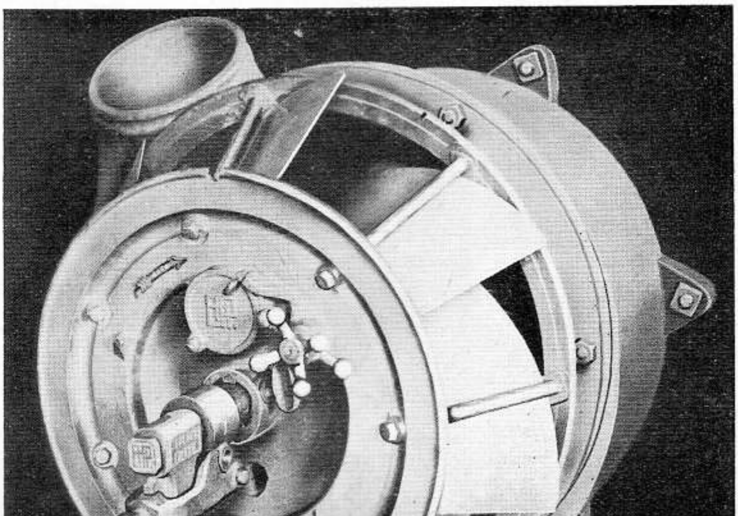

Fig. 4—Peabody Pulverized Coal Burner.

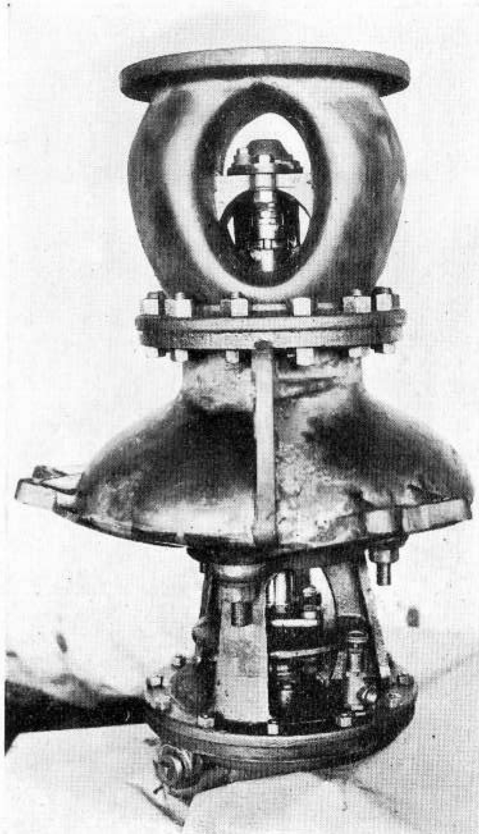

Fig. 6—Distributer.

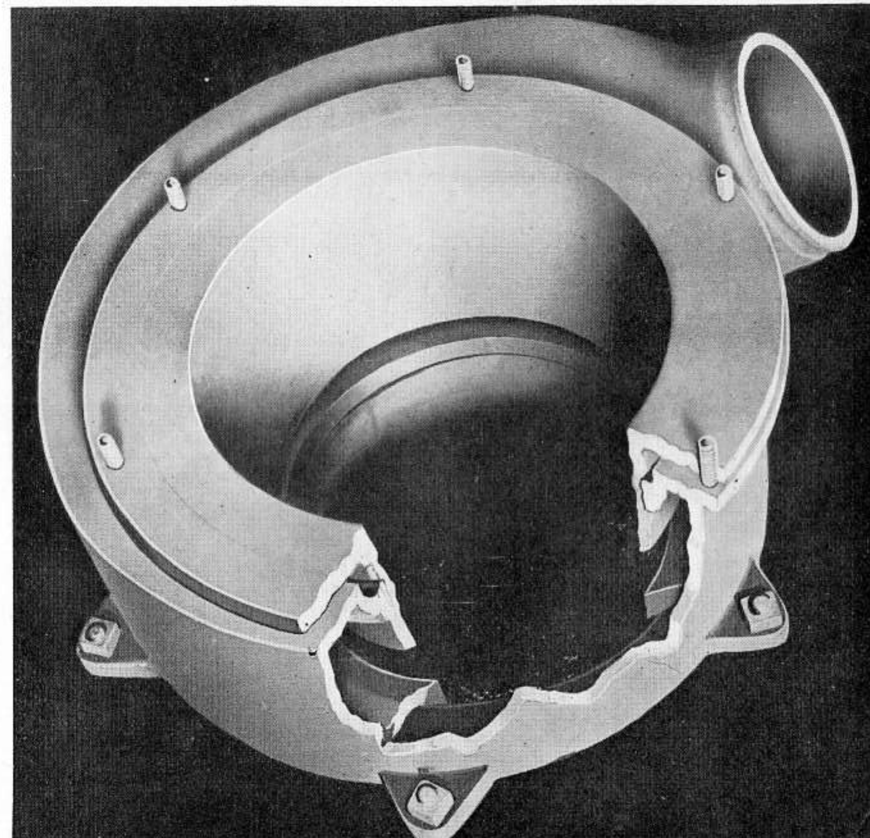

Fig. 5—Peabody Pulverized Coal Burner showing Coal Chamber.

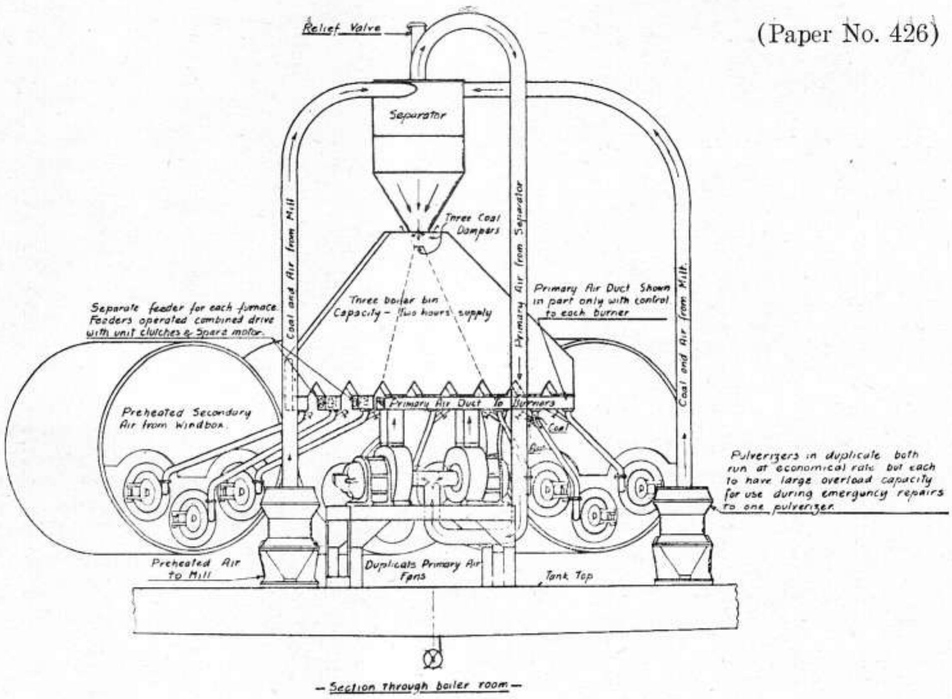

Fig. 7—Unit Bin System of Pulverized Coal Installation.

OPERATION WITH PEABODY BURNERS.

Coal from bunkers suitably conveyed by gravity or conveyor to controllable mill feeders.

Preheated air fed from Howden or other type air heater by fan to both mills. Preheated air is also used from this source to supply the burners with secondary air necessary to complete combustion.

Primary air fans in duplicate draw this preheated air through the mills and carry the pulverized coal to the separator common to both mills.

In separator the coal drops from the air stream by gravity into one or more small capacity bins and the clean air passes on to the primary air fans.

The primary air fans deliver this clean air to a common primary air duct from which there is a dampered connection to each coal pipe leading to each furnace. This air enters the pipe just below the coal feeder from the bin.

Coal feeder for each furnace with common driving unit but with individual control feed coal from the bins to the furnace through a pipe into which is introduced the air from the primary air duct.

The installation is fitted throughout with dampers and controls so that any condition of firing may be met and controlled at any capacity between the maximum and minimum.

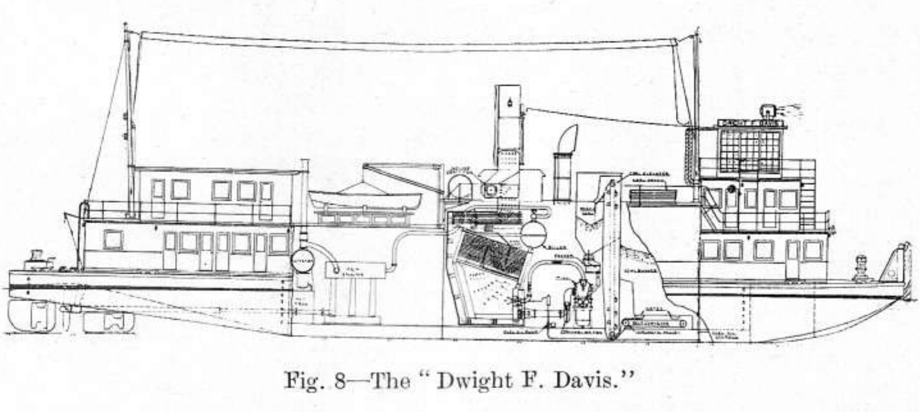

Fig. 8—The "Dwight F. Davis."

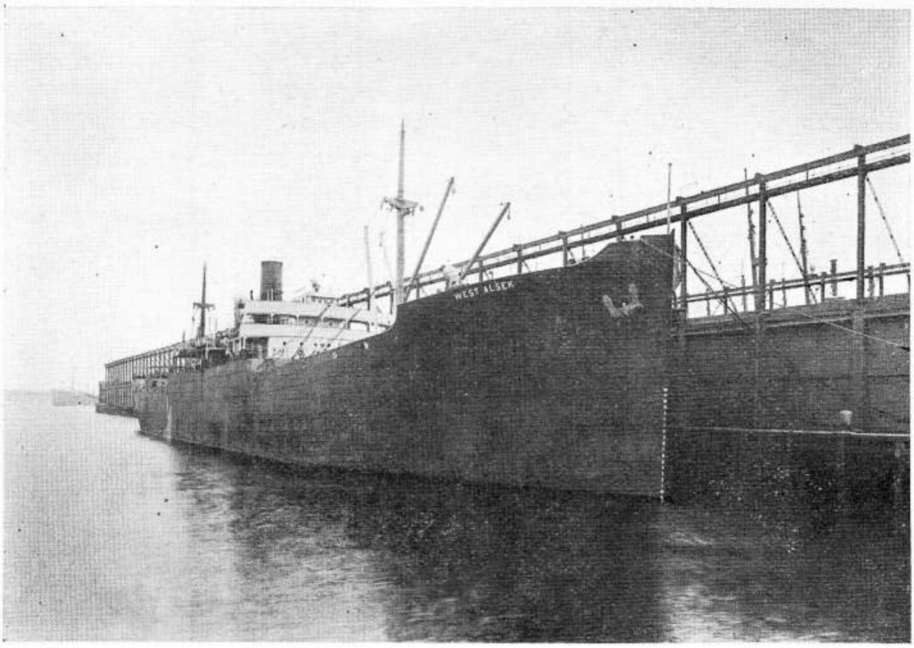

Fig. 9—" West Alsek."

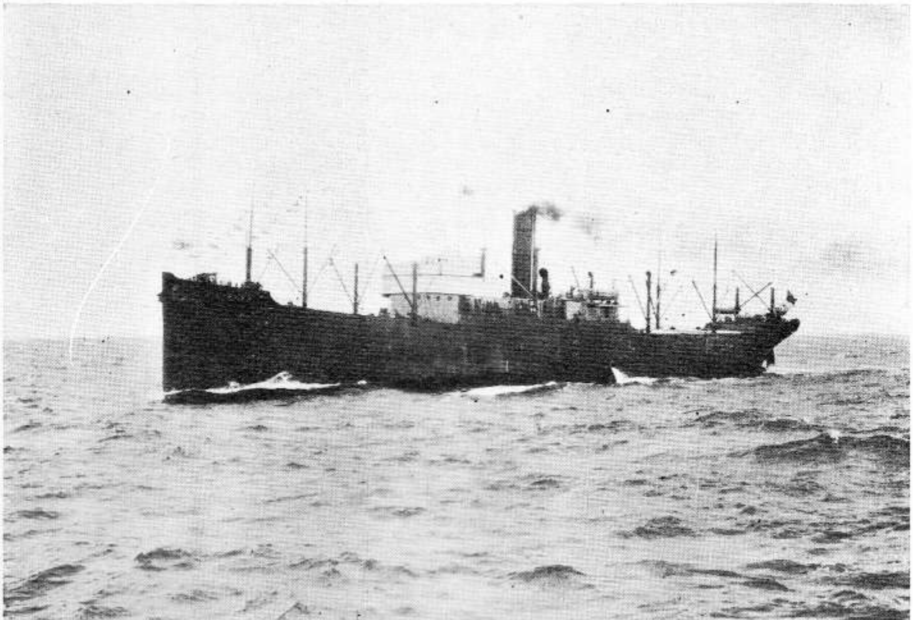

Fig. 10—" Lingan."

Marine Engineering and Design.

(Paper No. 445)

By ***Herbert L. Seward,*** *Professor of Mechanical Engineering, Sheffield Scientific School, Yale University; Member, Fuel Conservation Committee, United States Shipping Board.*

The attention of our modern marine engineers is now concentrated on the problem of evolving the most reliable form of power plant which will convert fuel energy into propulsive effort plus the other necessary services at the least overall cost consistent with the special conditions imposed in any given case. If the statement of the problem includes a curve of horse-powers at all knots and revolutions, limiting weights, expected auxiliary loads, cruising radius, type of service, available operating personnel, capital limitations and other information with its limiting characteristics, an opportunity is then offered for a study of the various possible types of equipment and arrangements to meet the conditions imposed. Further refinement in design based upon experience in like situations or upon development work and experience with the equipment under definite conditions then becomes possible. The designer must know all of the elements of which a total or unit cost is composed; he must work for economy but not at the expense of reliability or safety; he must come within certain limitations of space and weight but not at the expense of accessibility or facility of control, inspection, repair or adjustment. He must realize the limitations he is imposing on the operating personnel as well as the flexibility of the design to meet unusual or special conditions.

Much use has been made of the empirical method of design wherein certain established arrangements, such as Scotch Boilers, Reciprocating engines, steam driven auxiliaries, held permanent position in the field and variations occurred only because of the application of certain "variable constants" in the established formulas. Many of these relations were based on fundamental mathematical laws but their application became possible only when someone applied a coefficient which was largely experimental and included a wide margin of safety. A reluctance on the part of the marine engineer to give up practices which have proved to be good for those claimed to be better is but natural, although the comparison between advances made in power plants ashore and power plants afloat makes it appear that considerable inertia has to be overcome in the marine fraternity today. Equipment which has performed satisfactorily ashore cannot always be sent to sea because of the peculiarity of conditions under which it must operate at sea. Questions of corrosion, temperature, ventilation, the fact that foundations may be tilted in every direction at steep angles, as well as the complete isolation of the plant at sea from immediate possible help and the resourcefulness of the personnel in emergencies, all tend to introduce a note of caution in the designer's mind which has made marine engineering practice appear to lag far behind stationary practice. One modern authority has said that there are now building in various shipyards throughout the world, seagoing ships whose machinery would have been no particular credit to a designer of thirty years ago. Scotch boilers with an efficiency, as fired, of 50 to 70 per cent; triple expansion propelling engines with a steam consumption rate of 18 to 25 pounds per indicated horsepower per hour (mechanical efficiency unknown and apparently a matter of indifference); single cylinder steam engines or single wheel steam turbines, with a rate of 50 to 100 pounds, for driving auxiliary machinery; and reciprocating steam pumps with a rate up to 150 pounds. One line of steam and exhaust pipes will go all the way to the anchor engine in the bow, and another to the steering engine in the stern.

It is now apparent that a change is taking place in the attitude of marine engineers and designers towards the use of the fundamental sciences as well as in the application of more modern apparatus in marine power plants. Such items as Heat Balances, cost per ton mile, quick turn around, CO_2 percentages, port efficiency and the like are now being discussed. An application of the more analytical type of study is being made to the records of operation and an explanation is being sought for all discrepancies, variations or differences which are noticed when comparisons to established standards are made. At the last meeting of the Society of Naval Architects and Marine Engineers in New York it was pointed out that the topics discussed in the papers presented before the society included an interesting range of engineering subjects, many of which would not have been considered possible on the agenda of ten years ago. Some of these subjects were, pulverized coal, air preheaters, stokers, Diesel engines, electric auxiliaries, turboelectric drive and the use of welding.

In the present paper there can be included mention of only a few of the outstanding developments with an attempt to point out significant tendencies. Descriptions of new marine power plants have been well published in current literature. Reports of accomplishments in performance and economy have not been so regularly made in the press except for commercial advertising purposes. It is obviously more difficult to secure reliable operating data from a plant which includes some radically new designs because of a natural desire of the owners to wait until conditions have steadied themselves, adjustments made and operating personnel have been accustomed to its characteristics, before publishing the data even before technical societies.

Boiler practice is making substantial use of central station experience, although the space and weight limitations are obvious. Many years ago there was a feeling amongst marine operating engineers that the boiler room was a necessary evil and a place to keep away from, but today there is a realization that economy is made or lost in the control of combustion and in the care and attention given to apparatus in this department. Present designs are well illustrated in such plants as the S.S. Dixie and the S.S. Virginia. Proposals and designs of current work are apparently based on the experience gained with these ships which represent a moderate rise in steam pressure. Some designs are being laid down with air heaters, superheaters and a pressure of 375 pounds.

The question of higher steam pressures is directly related to the size of the power plant. If the plant is of low horsepower, it is obviously inadvisable to go to such a high steam pressure as to reduce the size of the turbine blades to a point where they become so small as to be inefficient. In the higher powered plants the boiler designer and the turbine designer must get together in order to work out the most reasonable steam pressure and the necessary supplementary conditions. It is hoped that among those who are considering installations of around 70,000 h.p. or more there will be someone with courage enough to go to 1200 pounds initial pressure. The initial cost should not be much different from present costs, judging by shore experience in building high pressure boilers.

The application of equipment to pulverize coal and burn it in Scotch boilers on the S.S. Mercer is well known. This is one activity of the Fuel Conservation Committee of the United States Shipping Board, whose outstanding contributions to marine development have been well known since its organization in 1922. From a paper by Messrs. Jefferson, Evans and Broshek entitled "Development of Pulverized Fuel for Marine Purposes, 1927-1928" before the Society of Naval Architects and Marine Engineers in November 1928, the following quotations are taken. The authors are pioneers in this field.

"The year's operating service of the S.S. Mercer has brought out several salient points that must be observed in effecting a satisfactory installation of pulverized fuel in Scotch marine boilers, which may be summarized as follows:

(a) Considerable finer pulverization of fuel is required than that necessary with large refractory lined furnaces such as found in shore plants.

(b) High turbulence of flame is required to produce complete and rapid combustion necessary in the comparatively short flame travel characteristic of a Scotch boiler.

(c) Consistently uniform distribution of pulverized fuel between various furnaces of Scotch boiler is necessary to maintain efficient performance.

(d) The power plant incident to producing pulverized fuel must be of an economical type, which may be secured either by installation of efficient low water rate units, or by useful employment of exhaust if less efficient units be used.

(e) Extra precaution must be taken to secure dust-tight smoke box doors and uptakes; otherwise, when using soot blowers it will be impossible to maintain a satisfactory fireroom condition.

(f) The velocities of coal stream throughout cycle requires special consideration to prevent overloading of fans and coal precipitation in pipe lines and to secure satisfactory selection of superfine grind from mill and turbulence at burner.

(g) Personnel must be trained to give the same close supervision to "velocity" as is required in regard to viscosity and pressure when using fuel oil.

(h) The usual fireroom instruments, such as orsat and pyrometer, are important but of greater importance are the draft gauge and smoke indicator.

(i) Sized coal, or 100 per cent slack coal, gives more uniform operating conditions than will run-of-mine, in fact, bunkering with run-of-mine will require use of crushers to produce fuel having a limited maximum size, otherwise, feed to mill will be very irregular, a condition that will be reflected throughout the whole operation of boiler plant."

Several other applications of pulverized coal to seagoing plants have been made and its reliability and safety are well established. The ability to use satisfactory fuels which could not be used otherwise in a Scotch marine boiler has likewise been definitely demonstrated. One of the most difficult problems in firing a three furnace boiler with pulverized coal has been that of evenly dividing the air-coal stream; i.e., proper distribution. Many types of pulverizers have been tried and have failed but better progress is now being made. The solution to the problem as offered by the Todd Dry Dock and Engineering Corporation consists in providing individual mills for each furnace of the two-stage impact type (3600 r. p. m., motor driven). As each mill requires about 10 h.p. the auxiliary electrical load is rather high and development work is now being done on a plan to drive the three mills of each boiler by a steam turbine through clutches. It is hoped that by the time this paper is published a trial installation will have been made and be in operation.

Mechanical reduction gears were given their first real opportunity for service afloat in 1910 and have made a very satisfactory reputation for themselves. Few other mechanical devices have been built with efficiency curves sustained so flatly at such high levels. When properly built and cared for, a set of reduction gears will go several hundreds of thousands of miles. They have been installed in almost every type and size of naval and merchant vessel in both the United States and foreign countries. There are many types and sizes of ships where the geared turbine form of drive is the logical installation because it has the lowest first cost, good economy, reasonably low weight, does not occupy excessive space, is simple and reliable and has a low maintenance cost. Much effort has been spent in investigating tooth curves, lubrication, flexible forms of pinion support or gear design so that at the present time a turbine and gear built by one of the recognized manufacturers can be installed with the assurance that it will operate successfully. There is so much latitude possible in the relative arrangements of centerlines that the geared turbine has been applied to single or twin screw plants, single, double, triple, or quadruple pinions per gear, and in jobs where a particular arrangement of turbine and condenser is desired. In the S. S. Malolo there are two gears on each of the twin screw shafts, two turbine pinions on each gear, eight turbines in all. Marine geared sets have been built up as high as 32800 s.h.p. per gear with 1000 lb. tooth pressure per inch of face at 3000 r.p.m. which corresponds to 105 lb. per inch face per inch of pinion pitch circle diameter.

Excessive noise in a geared set seems so be traceable to inaccurate work in cutting the gear teeth. If the hobbing machines are properly built and the work carefully done, a reasonably quiet set can be built. Operating personnel have now had sufficient experience with reduction gears to know how to take care of them and know what to look for in inspections. Matters of pitting, scoring, lubrication or alignment now receive intelligent and prompt attention.

When the United States Navy brought out the three colliers Cyclops, Neptune and Jupiter, it was the Neptune which proved the case for mechanical reduction gears (in addition to work done by Parsons in England) but the Jupiter (now Langley) proved the feasibility of the turboelectric drive. A real step forward was taken when the decision was reached to apply the system to the six battleships New Mexico, California, Tennessee, Maryland, Colorado, and West Virginia, with approximately 35,000 s.h.p. Their maneuverability and flexibility in operation is truly remarkable. The experience gained by American manufacturers in connection with the 180,000 s.h.p. power plants of the U. S. Airplane Carriers Lexington and Saratoga has been especially valuable and it is hoped that the possibility of applying this experience to passenger vessels demanding horsepowers on the order of 100,000 s.h.p. will soon be an accomplished fact. The S.S. Virginia of the Panama Pacific Line on her first trip from New York to San Francisco had an average fuel consumption of 0.74 lb. oil per s.h.p.-hr. for all purposes including a very heavy refrigerator load. Considering that a large part of the voyage was made through sea water at 78° to 85° fahr., and that this was her first trip, make this record worthy of praise. The smoothness of operation and the absence of vibration have been frequently made the subject of comment by passengers.

Table I shows the characteristic analysis of a turboelectric oilburning plant with steam turbine driven D.C. auxiliary power sets.

Tables I-V inc. have been prepared by Mr. Frank V. Smith, Marine Department, General Electric Company. Particular attention is called to the last line of Table I and of Table II.

TABLE I.

Summary of Steam and Fuel Consumption.

(Based on Using 4-350 K-W D-C Generating Sets for Auxiliary Power)

Operating Conditions:				
Shaft horse power	10,000	12,500	14,000	16,000
Propeller r.p.m.	123	134	136	144
Steam pressure at turbine throttle, lb. g.	275	275	275	275
Superheat, deg. F.	200	200	200	200
Vacuum, In. Hg. referred to 30″ Bar.	28.25	28.25	28.25	28.25
W/R-Lb. Steam per propeller S.H.P.-Hr.	8.77	8.53	8.52	8.62
Steam Consumption:	Lb.-Hr.	Lb.-Hr.	Lb.-Hr.	Lb.-Hr.
Main turbines, without extraction	87,700	106,625	119,280	137,920
Admit in lieu of extraction	3,900	5,830	7,800	11,100
Total to Main Turbine Throttle	91,600	112,455	127,080	149,020
Auxiliary Turbine generators—d.c.	8,680	9,350	10,000	10,900
Steam auxiliaries	6,625	7,195	7,625	8,175
Steam for other purposes	1,320	1,430	1,505	1,620
Total steam consumption	108,225	130,430	146,210	169,715
Lb. Steam per S.H.P.-Hr. all purposes	10.82	10.45	10.45	10.6
Fuel Consumption:				
Evaporation per lb. fuel oil based on a boiler efficiency of 82.0%; heat value of fuel oil 18,500 B.T.U.-Lb.; and feedwater temperature of	228°F	237°F	245°F	254°F
Evaporation, lb. fuel =	13.4	13.55	13.65	13.75
Lb. fuel oil per hr.	8,070	9,625	10,720	12,350
Tons fuel oil per day	86.5	103.5	114.5	132.5
Lb. fuel per S.H.P.-Hr. all purposes	.807	.77	.765	.772

For a larger plant such as might be used in a liner of approximately 50,000 tons displacement with a maximum speed of 26 knots but with a normal sea speed of 23.5 knots with an average of 55,000 s.h.p., the following analysis is applicable.

TABLE II.

Main Machinery:		
Four turbogenerators 15,200 KW each		
Four synchronous induction double motors 10,000 S.H.P. each, 240 r.p.m.		
Four auxiliary D.C. generators 1000 KW each, 230 Volts.		
Steam Conditions:		
S.H.P.	55,000	80,000
Pressure, lb. per sq. in. at turbine throttle	275	275
Superheat, deg. F. at turbine throttle	200	200
Vacuum, In. Hg. referred to 30″ Barometer	29.0	29.0
W/R–Lb. steam per propeller S.H.P./Hr.	8.2	7.75
Steam Consumption:	Lb. Hr.	Lb. Hr.
Main turbines without extraction	451,000	620,000
Admit in lieu of steam extraction	22,500	40,000
Total steam to main turbine throttle	473,500	660,000
Auxiliary Turbine D.C. Generator sets	26,400	37,200
Steam Auxiliaries	20,250	25,230
Steam used for other purposes	17,930	19,090
Total Steam Consumption	538,080	741,520
Lb. Steam per S.H.P./Hr. for all purposes	9.78	9.27
Fuel Consumption:		
Lb. steam evaporated per lb. fuel burned, based on a boiler efficiency of 82 percent; fuel oil having a heat value of 18,500 B.T.U. per lb.; and a feed-water temperature of 230°F.	13.5	13.5
Lb. Fuel Oil per hour	39,900	54,900
Tons of Fuel Oil per Day	427	588
Lb. Fuel Oil per S.H.P./Hr. all purposes	.725	.687

In further analysis of three of the items listed under Steam Consumption, Table II, there follow Tables III, IV and V.

TABLE III.

Electric Auxiliaries.

	S.H.P. 55,000	S.H.P. 80,000
	K.W.	K.W.
Excitation	272.0	300.0
Main Motor Ventilating Fans	198.0	288.0
Main Condenser Circulating Pumps	260.0	688.0
Main Condenser Condensate Pumps	55.0	63.5
Auxiliary Condenser Circulating Pumps	50.0	74.5
Auxiliary Condenser Condensate Pumps	13.0	19.5
Forced Draft Blowers	200.0	270.0
Lubricating Oil Pumps, 4-10 H.P., 2 in oper.	17.0	17.0
Oil Cooler Circulating Pumps, 2-7½ H.P.	13.0	13.0
Bilge Pumps, intermittent, average	20.0	20.0
Hull Ventilation, estimated average	125.0	125.0
Steering Gear, " "	50.0	75.0
Refrigeration, " "	75.0	75.0
Evaporator Feed Pump, estimated average	5.0	5.0
Distiller Circulating Pump, " "	10.0	10.0
Sanitary Pumps, 3-35 H.P., 2 in oper.	58.0	58.0
Fresh Water Pumps, hot and cold, estimated	10.0	10.0
Lights and bracket fans, estimated average	150.0	150.0
Electric Ranges and Bake Ovens, estimated average	150.0	150.0
Miscellaneous Items, including elevators, interior communication, wireless, laundry and tailor shop, machine shops, etc.	35.0	35.0
	1766.0	2446.5
No. of Aux. turbine generators in operation	2	3
W/R, Lb. steam per K.W./Hr. at given load	14.95	15.2
Lb. Steam per Hour	26400	37200

TABLE IV.

Steam Auxiliaries.

	S.H.P. 55,000	S.H.P. 80,000
Feed pumps, turbine centrifugal	9,150	12,500
Fuel Oil service pumps	1,600	2,230
Fuel oil transfer pumps	500	750
Standby fire and bilge pumps. av.	1,500	1,500
Air ejectors, main condensers	6,000	6,000
Air ejectors, auxiliary condensers	1,500	2,250
Total	20,250	25,230

TABLE V.

Steam Used for Other Purposes.

	S.H.P. 55,000	S.H.P. 80,000
Steam for heating fuel oil, calc.	2,930	4,090
Steam for heating vessel, estimated	4,000	4,000
Steam for hotwater service, estimated	2,500	2,500
Steam for galley, estimated	1,500	1,500
Steam for evaporators, estimated	7,000	7,000
Total	17,930	19,090

A Dieselization Fund of $25,000,000 has been set aside by the Congress of the United States for the purpose of developing, in the government merchant marine vessels, the art of applying Diesel engines. Dieselization has been applied to three series of ships, one series now being in full commercial operation. Full reports of the program as well as such operating data as are available have been thoroughly presented at meetings of the Society of Naval Architects and Marine Engineers by Capt. R.D. Gatewood, and reference is made to those reports for detailed information. The first series of installations were in twelve single screw cargo ships of about 3000 s.h.p. each. Four different builders provided the engines, of which two-thirds are two-cycle, one-third are four-cycle; some are double-acting, others are single-acting, four cylinder or six cylinder, but all are direct connected. Some have independent auxiliaries while others have direct driven air compressors and pumps. Engine builders in the United States have gained valuable experience in building these engines. The initial costs have been excessive. Although mechanical difficulties have been experienced, they are such as would normally be expected in an extensive program of this nature, which is purely pioneering, and to a certain extent experimental.

The Diesel had a decided advantage over the steam-driven cargo ship in port performance, but is at a great disadvantage in the use of lubricating oil, which has cost about 6 cents per mile. It is but natural that no chances would be taken by the operating personnel in the use of lubricating oil on these new installations. Some of the series of eight more ships to be fitted out with direct drive and three ships with Diesel electric drive will perhaps be in operation by the time this paper is read. Space does not permit a discussion of the Diesel engine in yachts, electric ferry boats and tugs nor of its fine history in submarines. Some developments looking towards a simpler and less expensive form of Diesel engine in the smaller sizes appear to have some promise of success.

Millions of dollars of government money have become available to test and develop the Diesel engine, but no such sums have been applied for experimentation in the development of the steam plant. Manufacturers of steam equipment have done much valuable research work and they hope to further develop the uses of higher pressures, higher superheat, and air heaters in accordance with best shore practice, and they expect to still lower their fuel costs as well. Unsubstantial claims of superiority and some high pressure salesmanship have occasionally

Table VI. Distribution of Steam. Coast Guard Cutters Modoc (1923) and Pontchartrain (1928).

Item		MODOC				PONTCHARTRAIN											
		1		2		3		4		5		6		7		8	
1	Run No.	5		8		1		2		3		4		5		6	
2	% of Rated Full Speed	101.0		78.2		101.7		100.0		99.1		78.1		75.5		76.9	
3	% of Rated Full Power	102.1		38.8		106.6		97.8		94.6		39.8		35.5		38.5	
		Steam Lbs/Hr	% of Total	Steam Lbs/Hr	% of Total	Steam Lbs/Hr	% of Total	Steam Lbs/Hr	% of Total	Steam Lbs/Hr	% of Total	Steam Lbs/Hr	% of Total	Steam Lbs/Hr	% of Total	Steam Lbs/Hr	% of Total
4	Feed Pump	1000	2.32	475	2.29	1340	3.72	1045	3.57	1045	3.46	730	4.35	571	4.05	710	4.18
5	Fuel Oil Pump	560	1.29	595	2.88	87	.24	43	.15	39	.13	42	.25	23	.16	24	.14
6	Fuel Oil Heater			120	.58	165	.46	150	.51	140	.47	126	.75	100	.71	113	.67
7	Forced Draft Blower	1540	3.56			244	.68	115	.39	115	.38						
8	Total Boiler Group	3100	7.17	1190	5.75	1836	5.10	1353	4.62	1339	4.44	898	5.35	694	4.92	847	4.99
9	Main Circulator	2960	6.85	1370	6.61	389	1.08	215	.73	209	.69	185	1.10	101	.72	123	.72
10	Main Air or Cond. Pump	975	2.25	730	3.52	93	.26	52	.18	52	.17	84	.50	56	.39	64	.38
11	Main Air Ejector					374	1.04	368	1.26	368	1.22	345	2.05	353	2.50	350	2.07
12	Total Vacuum Group	3935	9.10	2100	10.13	856	2.38	635	2.17	629	2.08	614	3.65	510	3.61	537	3.17
13	Lub. Oil Pump					66	.18	41	.14	42	.14	73	.43	52	.37	51	.30
14	Motor Vent. Fan					205	.57	114	.39	108	.36	82	.47	72	.51	80	.47
15	Lighting, etc.	1440	3.30	1460	7.05	301	.84	170	.58	160	.53	272	1.62	115	.82	114	.69
16	Steering Gear					119	.33	55	.19	56	.18	127	.76	58	.41	63	.37
17	Sanitary Pump	580	1.34	710	3.43	91	.25	59	.20	65	.21	109	.65	77	.54	79	.47
18	Bilge Pump	200	.46	320	1.55												
19	Steam Kettles			100	.48												
20	Transfer Pump	245	.57	165	.80	83	.23	48	.16	51	.17	75	.44	48	.34	51	.30
21	Quarters Vent. Fan					150	.42	76	.21	78	.26	189	1.13	88	.62	95	.56
22	Aux. Condensate Pump					7	.02	7	.02	12	.04	24	.14	8	.06	10	.05
23	Total Miscellaneous	2465	5.67	2755	13.31	1022	2.84	570	1.94	572	1.89	951	5.66	518	3.67	543	3.19
24	Total Boiler Group	3100	7.17	1190	5.75	1836	5.10	1353	4.62	1339	4.44	898	5.35	694	4.92	847	4.99
25	Total Vacuum Group	3935	9.10	2100	10.13	856	2.37	635	2.17	629	2.08	614	3.65	510	3.61	537	3.17
26	Total Miscellaneous	2465	5.67	2755	13.31	1022	2.84	570	1.94	572	1.89	951	5.66	518	3.67	543	3.19
27	Excitation, Main Mach.	4260	9.86	3690	17.81	1275	3.54	789	2.69	808	2.65	1342	8.00	648	4.60	982	5.78
28	Stm. Used on Acct. of Aux.	13760	31.80	9735	47.10	4989	13.85	3347	11.46	3348	11.06	3805	22.66	2370	16.80	2909	17.13
29	Bleeder Stm. for Feed Heater					1822	5.03			2530	8.38	1128	6.72			1435	8.45
30	Stm. Main Turb., Prop. Only	29520	68.20	10983	52.90	29230	81.12	25983	88.54	24332	80.56	11867	70.62	11726	83.20	12636	74.42
31	Total Steam Used	43280	100.00	20718	100.00	36031	100.00	29330	100.00	30210	100.00	16800	100.00	14096	100.00	16980	100.00
32	Leakage	860		767		310		378		320		2249		268		45	
33	Total Steam Evaporated	44140		21485		36341		29708		30530		19049		14364		17025	
34	Oil/S.H.P./Hr., all Purposes, lbs.	1.195		1.535		.890		.840		.823		1.183		1.079		1.11	

tended to hurt the cause of the Diesel engine in its marine applications, and so the subject of Steam versus Diesel has become a highly controversial one. When the operating records of Diesel ships become available over a reasonable period of time so that the overall cost of operation including capital and maintenance charges can be definitely known, then the class of work to which each type of plant can best contribute in its own way will be known. Until such data are available, one must maintain an open mind and welcome the development work so nobly begun.

The United States Coast Guard Service, through its Chief Engineer, Commander Q. B. Newman, is making some very important contributions in the art of marine engineering and design. The ships of this service are small cruising gunboats about 250 ft. long, making 17 knots with 3000 s.h.p. The storms which keep other vessels in port are often the cause of the Coast Guard Cutters putting to sea in answer to calls of distress. They go looking for the icebergs which other ships hope to avoid because of the information faithfully furnished by the cutters. In the forgotten seas of the North these vessels are the whole United States Government. They must always keep to the sea—and keep within limited appropriations not inflated by war enthuiasm.

In 1920 the first turboelectric installations in the Coast Guard were made by the General Electric Company on the cutters Tampa, Haida, Mojave and Modoc. These ships were the first for which a synchronous motor was adopted to drive the propeller. While their machinery was in process of development it was proposed by Commander Newman that some of the auxiliary machinery be driven by current from the main generator. It is of interest to note that the contractors for the machinery refused to develop this idea for two reasons, which at the time were valid; turboelectric propulsion in as low power as 2600 s.h.p. had never been used and the synchronous motor was being employed for propulsion for the first time. It was considered inadvisable to allow unrelated problems to influence the success or failure of the new style of main machinery.

All of the auxiliary machinery for the four cutters was therefore of conventional steam driven types, known to be extravagant, but not even suspected of being bad enough to have a marked effect on the whole plant. And so when the four ships of the Tampa class failed to live up to expectations as to fuel consumption, it was decided to conduct thorough performance trials of all machinery separately and then of the plant as a whole.

The Modoc was selected for the tests and the trials were run in August 1923. The results were astounding. At full power 32 per cent of all the steam produced was being used by auxiliary machinery, and more at lower powers. In fairness to the auxiliaries it should be stated that they were exhausting against eight pounds back pressure, and that some of the heat in the exhaust steam was recovered in the feedwater heater and in the low pressure stages of the main turbine. The results of the Modoc trials were presented as a paper before the Society of Naval Architects and Marine Engineers in November 1923.

In 1925 the Great Lakes ore ship T. W. Robinson, of the Bradley Transportation Company, was fitted with a partial application of the idea that electric auxiliaries should receive power from the main central power plant and thus became the pioneer ship with this type of machinery. It has been largely due to the efforts of Commander Newman that the standardized electrical units of the shore central power house are being used in marine power plants.

An appropriation in 1926 made possible the signing of contracts in 1927 with the Westinghouse Electric and Manufacturing Company for the machinery of the five newest Coast Guard Cutters, Chelan, Pontchartrain, Tahoe, Mendota and Champlain. These vessels resemble as much as possible the central power house idea afloat in that they are electric-drive with as much of the auxiliary power as is at present possible being derived from the main turbogenerator. In accomplishing this, the chief difficulty to be overcome was in the varying frequency of the main generator which corresponds to the speed of the ship. The fluctuation in speed of an alternating current motor while driving some auxiliaries, such as the main circulator, or fuel oil pump, may be desirable as its speed fluctuates with the speed of the ship, and may not be objectionable within certain limits for blowers, fans and a few other auxiliaries. It would be entirely unsuitable for lighting, radio, high pressure centrifugal pumps and refrigeration.

Since there is an absolute need of direct current for excitation, searchlights, battery charging and other d.c. power requirements there must needs be installed a motor generator set. The a.c. end of this set may be either motor or

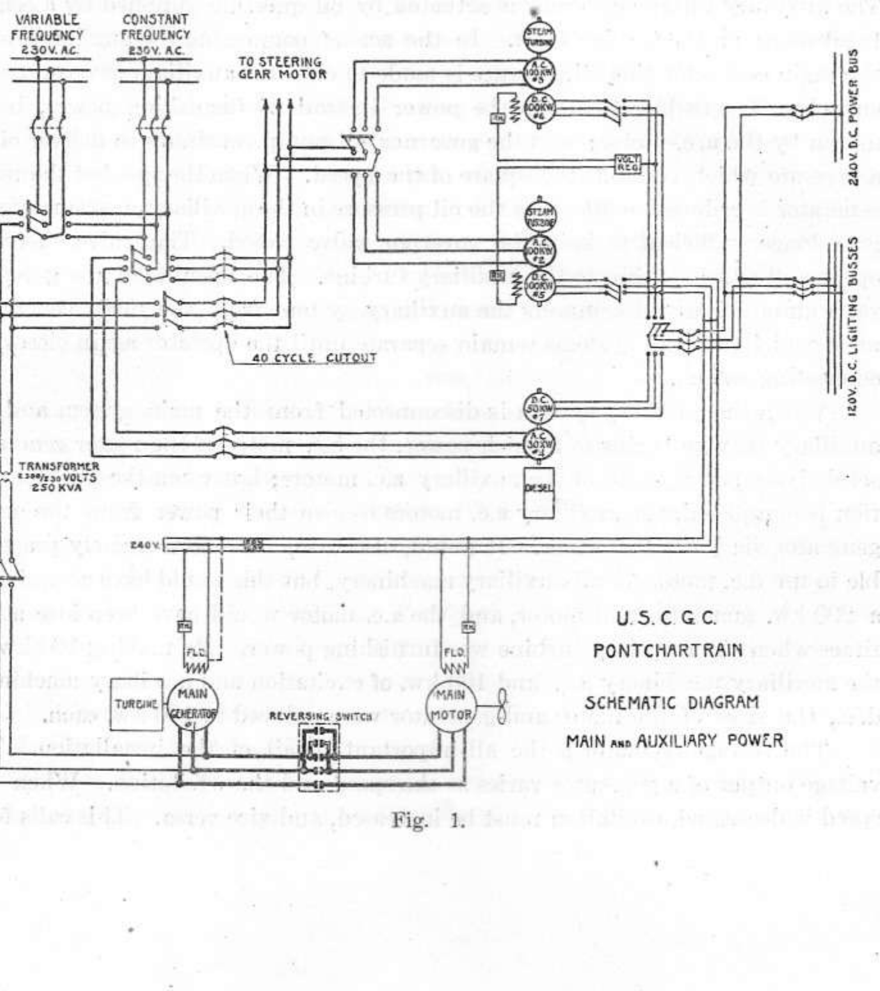

Fig. 1.

generator, depending on conditions, but the d.c. end must give a constant voltage.

The propelling machinery of these five Coast Guard Cutters consists of the main turbogenerator, 2500 kilowatts at 3600 r.p.m.; the synchronous propelling motor which develops 3000 shaft horsepower at $163\frac{1}{2}$ r.p.m.; and the necessary control panel which contains the maneuvering switches and speed control. The auxiliary circuit is tapped off from the high voltage terminal of the main generator and is led directly to a power transformer which reduces the voltage to 230. The circuit is led from the low voltage terminal of the transformer to a disconnecting switch and a circuit breaker on the auxiliary switchboard, and from the circuit breaker to the a.c. motor of one of the generator sets previously referred to. A diagram of the electrical connection is shown in Fig. (1). An auxiliary steam turbine is connected through a speed reduction gear to the motor generator set and furnishes power to the generator at all times when the circuit between the main and auxiliary systems is open.

The auxiliary turbo-motor generator can be connected to the main generator at any frequency between 40 cycles and 60 cycles. It is, of course, necessary that the two machnes be synchronized and in phase at the instant of closing the connecting switch, and a simple means is provided on the synchronizing panel for varying the speed of the auxiliary turbine to suit main generator frequency. The auxiliary turbine governor is actuated by oil pressure supplied by a centrifugal pump on the turbine shaft. In the act of connecting the auxiliary set to the main generator this oil pressure is made to close the auxiliary governor valve entirely. The turbine then absorbs power instead of furnishing power, being driven by the a. c. motor; and the governor oil pump continues to deliver oil at a pressure which varies as the square of the speed. When the speed of the main generator is reduced to 40 cycles the oil pressure in the auxiliary governor system is no longer sufficient to keep the governor valve closed. The valve therefore opens and admits steam to the auxiliary turbine. The opening of the governor valve automatically disconnects the auxiliary system from the main generator, after which the two systems remain separate until the operator again closes the connecting switch.

When the auxiliary system is disconnected from the main system and the auxiliary turbine begins to furnish power, the a.c. motor of the motor generator set delivers power to all of the auxiliary a.c. motors; but when the interconnection is completed, the auxiliary a.c. motors receive their power from the main generator via the transformer. It would, of course, have been entirely practicable to use d.c. motor for all auxiliary machinery, but this would have necessitated a 200 kw. generator and motor, and the a.c. motor would have been idle at all times when the auxiliary turbine was furnishing power. By making 100 kw. of the auxiliary machinery a.c., and 100 kw. of excitation and auxiliary machinery d. c., the sizes of the motor and generator were reduced to 100 kw. each.

The voltage regulator is the all-important detail of the installation. The voltage output of a generator varies as the speed and the excitation. When the speed is decreased, excitation must be increased, and vice versa. This calls for a generator which will deliver full voltage at 40 cycles with full excitation and the same voltage at 60 cycles with reduced excitation. The function of the voltage regulator is to fit the excitation to the speed requirements. On the Coast Guard cutters the regulator operates by cutting resistances in or out of the field and is a modification of the well known Tirrill regulator. The operation is dependent upon a pair of vibrating contacts in parallel with a bank of resistance, the coil of the vibrator being connected across the bus bars. When the contacts are closed the resistance is short circuited. If the voltage is low the contacts remain closed for a relatively long time, thereby increasing the excitation. If the voltage is high the contacts remain closed a relatively short time, thereby decreasing the excitation. The frequency of the vibrator is 60 cycles and there is therefore no perceptible flicker of lamps. Variation from normal voltage does not ordinarily exceed two per cent.

The acid test of all engineering matters is supplied by experience and it is still too early for final judgment. The Chelan, the first of the new ships, had her trials in July 1928 and since then has cruised upwards of 20,000 miles with complete success and with a marked improvement in fuel consumption as compared with earlier ships.

Elaborate trials were run on the Pontchartrain, which was the second of the new ships, and the performance of all her machinery was measured with a high degree of precision. A complete report of her trials was presented before the Society of Naval Architects and Marine Engineers in November 1928, by Captain Newman. Table (VI) shows some of the results obtained on the Pontchartrain and in parallel columns the results obtained on the Modoc's trials in 1923. An examination of the tabulation will show the extent to which expectations have been realized. On the first and fourth runs of the Pontchartrain auxiliary power was furnished by the auxiliary generator. On the second and fifth runs auxiliary power was furnished by the main generator. On these two runs the main turbine was not bled for feed heating. On the third and sixth runs auxiliary power was furnished by the main generator and steam was extracted from low stages for heating feedwater.

The two ships are comparable as to size, tonnage, power, speed, boilers and propelling machinery. Refinements have been made in hull design of the Pontchartrain, and she is fitted with a contra-propeller for rectifying the propeller stream. She operates on steam at 250 pounds (gage) pressure and 250 degrees F., superheat, whereas the Modoc has 200 pounds and 90 degress. All these are very real improvements and make for reduced operating costs. But the outstanding fact of all is that the Pontchartrain auxiliaries take only 25 per cent as much steam as those on the Modoc. Provision is made for the comfort of the crew, in case the vessel lies in port in the tropics, by the installation of a 100 kw. Diesel auxiliary which will provide the necessary d.c. and a.c. power for port use while both boilers may be secured.

In the comparison of machinery of ships it is customary to consider fuel consumption for all purposes in pounds per shaft horsepower per hour referred to the propeller shaft. This figure for full operation is Modoc 1.195, Pontchartrain 0.823; for cruising speed Modoc 1.535, Pontchartrain 1.079.

Capital investment is always a matter of much concern to ship-builders. For the new Coast Guard ships the total cost of machinery and boilers f.o.b. factory was $ 173,000 per ship, which included all machinery except windlass capstan, steering gear and quarters ventilating fans.

The statement can be made that the system used on these ships is not applicable to ships of large power. This is of course true, but one has only to stand in the engine room of one of our numerous freighters of the same horsepower, or study the performance records of this fleet, to realize that many applications of this system can be made with lower initial cost, less space occupied, and much lower operating costs. If auxiliary power requirements are in excess of 250 kilowatts it might be more advantageous to supply this power from an auxiliary turbine having at least five stages and exhausting against not more than one pound back pressure absolute. If the requirements for auxiliary power are below 250 kilowatts the difficulty is greater and the advantage of connecting up to the main plant is increased.

The greatest lesson learned from a study of most modern marine plants is that there is at last being developed a realization of the importance of using engineering methods of analysis in applying practices developed ashore and in the use of higher standards of operation not only to conserve the elusive B.t.u. but in all phases of the business of producing a ton-mile of marine transportation.

Combustion in the Furnaces of Marine Boilers.

(Paper No. 447)

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Introduction.

Because of the limitations of space and weight prevailing in marine work, the generation of power on shipboard is limited to the two fuels—oil and coal. Wood, gas, bagasse, coke, etc., which are used extensively in different parts of the world for the generation of power on shore are eliminated from marine practice because of their large volume for a given weight and B.T.U. value, an item of great importance in the case of any fuel used at sea. It is the fact that oil is much more compact for a given B.T.U. value than any other commercial fuel, which makes it so desirable aboard ship, especially in the case of passenger vessels where space is at such a premium. The fact that oil is liquid at ordinary temperatures and may be readily pumped from one part of the ship to another is also an item of importance in its popularity for marine use, and were it not for the cost incurred by its use at times, this paper would be devoted exclusively to the combustion of oil in the furnaces of marine boilers.

Although from the chemical aspect, the combustion of coal and oil are very similar, consisting in both cases of a combination of hydrogen and carbon with oxygen, the methods employed in securing this combustion are so different with the two fuels that they will be treated separately, a number of the more usual methods used with each fuel being given.

Oil.

A. Air Atomizers:—When oil was first introduced into marine work for power generation purposes, air atomizers were sometimes used in salt water ships, as in this way a very fine atomization was readily secured without the loss of any fresh water. It was soon learned, however, that the cost of compressing the air for atomization was very high, and also that the weight of the compressors was an undesirable feature for marine work. Roughly, it costs eight times as much to atomize a thousand gallons of oil with compressed air as it does with a modern pressure atomizing system. This combination of excessive weight and high cost of operation of the air atomizing type of oil burners gradually reduced their use in the marine field until they are practically unknown there at the present time.

B. Steam Atomizers:—As in the case of the air atomizers, these were carried into the marine power plant field when oil was first introduced there, because of the knowledge that had been gained of their use in stationary installations. They had one serious drawback for salt water ships, and that was the loss of fresh water entailed from their use with the result that they were never used extensively except on fresh water, and for short runs (around harbors, etc.) on salt water.

The flat type of flame which was usually employed in conjunction with a checkerwork of firebrick for admission of the air to the furnace was limited in capacity in the small furnaces available under marine boilers, any attempt to force this combustion to high rates resulting in excessive smoke and high hydrocarbon and other combustible losses. These two features, loss of fresh water and limited capacity soon began to eliminate the steam atomizers from marine work, until at the present time they are never seen except in a few old installations on harbors or rivers. Steam atomizers never had anything to recommend them for marine work except their low first cost, and that saving was so trifling compared to the losses their use involved that in modern marine work they are never employed.

C. *Pressure or Mechanical Atomizers:*—Oil burners using this type of atomizer are almost universally installed in modern oil burning ships. Although there is a great number of this type of burner made throughout the world, in general principle they are all alike. An atomizer located in the center of an air register, discharges oil into the furnace in the form of a hollow conical spray, which is produced by the centrifugal action of the oil under pressure passing through suitable passages located near the tip of the atomizer. The air for combustion is brought to the register so as to surround and mix with the conical oil spray to produce the combustion desired. Various designs of tips are employed to produce the hollow conical spray, and the fineness and angularity of the cone of this spray varies accordingly in different types. In a similar manner, the different designs of air registers use different means of directing the air so as to have it mix with the oil spray, some producing a very rapid and complete mixture; some a delayed mixing with slow burning flame, and some never giving a complete mixing at any time.

It is of course essential that the oil, when it reaches the tip of the atomizer, be sufficiently fluid so that it will readily break up into a spray, due to centrifugal action, as soon as it is released from the orifice. This requires that its viscosity be reduced to at least 3 to 5 degrees Engler, or 100 to 120 degrees Saybolt. In general, it may be said that the wider the angle of oil spray, the finer the atomization and the greater the speed of combustion will be. It should not be inferred from this that wide angle sprays are always the best to use, even in the small furnaces available with marine boilers.

In the case of Scotch boiler furnaces it is impossible to use very wide angles because of the tendency of sprays of this type to build rings of carbon in the furnaces, these rapidly growing to a point where it is necessary to shut down the burner. Fortunately, the length of travel available for the flame in the furnace and combustion chamber before the tubes are reached in this type of boiler is sufficient, so that with the relatively small amounts of oil burned in these furnaces, fairly satisfactory results may be obtained with the narrower angles of spray, provided a well designed air register is used. Because of the relatively low furnace temperatures existing in Scotch boilers, it is very easy for a poorly designed burner (combination of atomizer and air register) to cause big fuel losses, either due to the necessity of using large amounts of excess air to prevent smoke, or, if large amounts of excess air are not used, still greater losses due to smoke and unconsumed hydro-carbons passing up the stack.

In furnaces under water tube boilers of the straight tube, header type, where the tubes are fairly long and reasonable travel is given to the gases before they enter the tube bank, moderately wide angles of oil spray are desirable, so that the flame reaches the full length of the furnace. Care should be taken not to have angles of spray too narrow in furnaces of this type, or the combustion will be slowed down to the point where it will be incomplete before the tubes are reached, and, conversely, the very wide angles are to be avoided, as, if the combustion is too rapid and concentrated near the front wall maximum use is not made of the radiation effect from the flame into the furnace row of tubes. Unless correct use is made of the furnace radiation by means of proper flame distribution in the furnaces of this type, a distinct loss in boiler efficiency will result.

In the case of water tube boilers of the three drum express type where it is necessary for the flames to pass directly to the tube bank from burners located close to it, wide angle sprays are absolutely essential for best results. Also, as it is usual to operate this type of boiler at higher rates (especially in Naval installations) than the heavier types of boilers, the speed of combustion becomes a very important factor to prevent smoke and unconsumed hydro-carbon losses, and the widest angles of oil spray combined with the proper types of registers are necessary to give the desired results.

From the above it may be noted that no one size or design of burner is best for use in all types of marine furnaces. It is only by careful experimental investigation that the best combination of oil sprays and registers can be developed to meet the widely varying furnace conditions encountered in marine practice, and the different cases as they arise should be handled accordingly, if the efficient results sought are to be obtained.

Tables No. 1, 2 and 3 give representative boiler efficiency tests which show what may be expected with properly designed and efficiently operated oil burners used under Scotch, Babcock & Wilcox Marine and Babcock & Wilcox Express type boilers respectively. None of the boilers with which these tests were run was fitted with air heaters or economizers. The water and oil in each case was carefully weighed on calibrated scales, and the readings throughout were taken with the greatest accuracy. In Plate No. 1 the three upper curves show the efficiencies and the three lower curves the total radiation and unaccounted for losses of the heat balances of the tests shown on Tables No. 1, 2 and 3. The total radiation and unaccounted for losses were obtained by deducting the sum of all the accounted for losses shown in the heat balances plus the boiler efficiencies from 100%.

In studying these latter curves it is interesting to note the form these losses take under different conditions and at different rates of operation. Per cent radiation by itself is always greatest at the lower rates, and lowest at the higher rates of operation of the boiler, whereas the hydro-carbon and other combustible losses usually increase with rating. The curve at the bottom of Plate No. 1 of The Babcock & Wilcox Express boiler from the 4.0 lb. to the 12.0 lb. of water per sq. ft. rate represents almost pure radiation with a very small hydro-carbon or other combustible loss up the stack. From that point on the curve turns upward, due to the combustible losses increasing at the higher rates to a point where their increase is appreciably greater than the corresponding reduction in radiation. The Scotch boiler curve snows this same characteristic to a marked degree at the relatively low rates of operation. If the reduction in radiation is about counterbalanced by the " unaccounted for " losses, a horizontal line for the total of these losses will be secured over practically the entire range of operation of the boiler, as shown in The Babcock & Wilcox curve at the bottom of plate No. 1. In addition to the radiation and unconsumed combustible losses mentioned above, there are other slight "unaccounted for" losses from boilers, which vary with different installations and which cannot be accurately segregated, such as heated air escaping from the fireroom and air leakage through the boiler casing, this latter seldom being properly determined by the flue gas analysis, and hence not being properly accounted for in the heat balance.

Coal.

A. Hand-fired:—In the marine field practically all coal used is hand-fired, the exceptions to this in actual service being trifling in number and relatively negligible. However there has lately been a marked tendency shown by up-to-date ship operators to start installing apparatus which will handle this fuel more efficiently, and without the heavy manual labor required for hand-firing. As time goes on the practice of hand-firing will undoubtedly grow less and less as the more modern methods are introduced, and there will be as little chance of a new ship being built to operate hand-fired, as there is at the present time for a modern stationary steam power plant to be so equipped.

The efficiency results that may be expected with hand-firing are so dependent upon the firemen and of the grade of coal used that definite tests are of little use in making anything but general statements of what may be expected with a given piece of apparatus. Changing firemen may make a difference of 10 percent or more in the efficiency results secured with a given boiler, and the same may also be said for a radical change in the coal. A great many hand-fired tests are on record with which reasonable care was taken to insure proper attention to the fires and the coal used well adapted to this method of firing. From these an approximate generalization may be made as follows—

Scotch boilers without air heaters	—60 to 65% efficiency;
Scotch boilers with air heaters	—70 to 75% efficiency;

Water tube boilers without air heaters—65 to 70% efficiency;
Water tube boilers with air heaters —75 to 80% efficiency;

It is realized in making these general statements that there are many installations in service that fall wide of the figures given, but in the average ship, they should be approached, with proper attention to the firing, reasonably suitable coal, and properly designed and clean boilers. It is of interest to note that the air heaters are responsible for an increase of approximately 10 percent in the average efficiency figures, even though there are practically no air heaters in service in marine work that would give this increase because of the lowering of the temperature of the uptake gases alone. It is necessary to cool the uptake gases coming from the average hand-fired boiler approximately 35 degs. F. for each 1% gain in boiler efficiency, and the average air heater in marine service does not cool these much more than enough to account for about 5 percent in efficiency. The remainder of the gain shown is obtained by improving the combustion conditions in the furnaces due to the use of the pre-heated air, which has the effect of reducing both the carbon loss in the ashpit and the unconsumed carbon and hydro-carbon losses up the stack, these losses being very much in evidence in the average hand-fired installation. The speed of combustion necessary in the relatively small furnaces available in marine work, again is the important factor, the preheated air assisting materially in speeding up the combustion, increasing the furnace temperature, and reducing appreciably the combustible losses which occur without its use.

B. *Pulverized Coal:*—During the last year or two a good deal of attention has been given to the use of pulverized coal on shipboard as one of the means of eliminating the labor of hand-firing and improving the efficiency of the ship. A few installations have been made and are in service, and more are being put in at the present time. Although this method of firing coal under boilers has experienced a very rapidly increasing use in the stationary power plant field for some time, its application to the small furnaces available in marine work was not successfully accomplished until lately.

In the first attempts to do this, pulverized coal burners of the "stream line" type, similar to those utilized in the majority of stationary installations, were tried, and the results secured impractical for commercial installations. The loss of unconsumed carbon up the stack was so great that efficiency results as low as those obtained by hand-firing on the same boiler were secured, and trouble was also experienced from excessive slag formation caused by the delayed combustion conditions existing. It was not until burners using registers giving a turbulent action to the air, similar to those used with the more efficient designs of mechanical atomizing oil burners, were emploped, that the combustion was speeded up sufficiently to give satisfactory results. Also the "stream line" of coal discharged into the furnace had to be broken up, so the particles of coal were enabled to mix thoroughly at the burners with the turbulent entering air. With this combination of a properly distributed supply of coal and a turbulent mixture of the coal with the secondary air, satisfactory results were secured and commercial installations on shipboard became possible. Tables No. 4 and 5 give test data showing the efficiency results secured with turbulent types of pulverized coal burners on a Scotch boiler and a Babcock & Wilcox marine boiler, respectively. The Scotch boiler was the same one used for the tests reported in Table No. 1, and the data are reproduced herewith from a paper by Mr. Carl J. Jefferson, presented before the 35th Annual Meeting of the Society of Naval Architects and Marine Engineers in November, 1927.

It may be noted that for the pulverized coal tests this Scotch boiler was fitted with an air heater, a very desirable adjunct for speeding up the combustion to a satisfactory extent with this type of fuel in these relatively cold furnaces and also serving to preheat the supply of primary air going to the mill. The Babcock & Wilcox boiler used for these pulverized coal tests was the same as that used in the tests reported in Table No. 2, no air heater being used with this boiler for either the oil or pulverized coal tests. The furnace used for pulverized coal under this boiler is shown in Plate No. 2. A small steam air heater was utilized to preheat the supply of primary air to the mill for the pulverized coal tests run with this boiler. The two upper curves of Plate No. 3 show graphically the efficiencies reported in Tables No. 4 and No. 5 and the two lower curves, corresponding total radiation and unaccounted for losses.

To date it has not been practical to use pulverized coal with small tube Express type boilers because of slag filling the small gaps existing between the tubes in this type of boiler and preventing the escape of the gases from the furnace. The problem with this type of boiler is also complicated by the fact that most of these boilers in service are operated at rates considerably in excess of 100,000 B.T.U. release per cu. ft. of furnace volume per hour, and at these rates the carbon losses up the stack would be excessive, even if the slagging of the tubes did not prevent operation.

Pulverized coal is a much more difficult fuel with which to obtain relatively complete combustion in the small furnaces available in marine work than is the case with oil. The more the furnaces are cooled, the more difficult it becomes, the losses increasing rapidly with increased rates of B.T.U. released per cu. ft. of furnace volume. The curves shown on Plate No. 3 clearly emphasize this point, the Scotch boiler furnaces being completely water cooled except for a distance of about 2 ft. near the burners where a cylindrical refractory lining was used. The furnace under the Babcock & Wilcox boiler was lined with insulated brick throughout except for the furnace row of boiler tubes. It may be noted by comparing the data in Tables No. 4 and No. 5 that when releasing 60,000 B.T.U. per cu. ft. in the Babcock & Wilcox brick lined furnace, the total radiation and unaccounted for losses were approximately 7 percent, and when releasing 60,000 B.T.U. per cu. ft. in the Scotch boiler furnaces this loss rose to approximately 16 percent. This in spite of the fact that the pulverization of the coal used at this rate was a little finer with the Scotch boiler than with the water tube type. As may be noted, the same burners were not used for the Scotch Boiler tests and the Babcock & Wilcox Boiler tests.

Following the completion of the brick lined furnace tests with the Babcock & Wilcox boiler, 15 sq. ft. of water cooled surface was used in each side wall, this resulting in an increase of nearly 2 percent in the unaccounted for losses at the 60,000 B.T.U. per cu. ft. rate. At low rates of B.T.U. released per cu. ft. of furnace volume (about 30,000 B.T.U. per cu. ft. or less), there was no loss in boiler efficiency, due to the use of the increased water cooled surface in the furnace, the increased heat absorption of this surface counter-balancing the slight increase in carbon loss. These rates are too low, however, to be considered for practical operation in marine installations.

From the above it may be noted that the higher the furnace temperatures used with pulverized coal in small marine furnaces, the more complete the combustion at a given rate of operation. Care must be taken in these small furnaces to keep the fusing temperature of the ash in the coal above the furnace temperatures existing near the walls and tubes, or slagging will cause operating difficulties. Also the pulverization of the coal should be as fine as it is commercially possible to have it, as the finer the pulverization of the coal, the lower the carbon losses, and the less the slag trouble in the furnace when operating at a given rate with a given coal.

To sum the matter up, to secure the best results with pulverized coal on shipboard, careful investigation should be made of the particular installation under consideration, to insure the best compromise between furnace temperatures, carbon losses, fineness of pulverization, slag difficulties, and boiler efficiencies. All of these must receive due consideration if the advantages to be expected from the use of pulverized coal on shipboard are to be realized to their fullest extent.

C. Stokers :—The application of stokers to marine boilers is not new, but it was only during the last few years that their use may be said to have become commercially desirable. Previously the relatively low price of fuel and labor counteracted to a considerable extent the expense of installing stokers, their weight, and the operating difficulties of some of the earlier designs. This picture is now radically changed. The price of coal in the bunkers is higher than it was before the war, and the difficulty of securing competent firemen who will remain with a ship that is hand-fired is becoming a real problem. Also the experience gained in the stationary field in building reliable stokers is of the greatest value when applied to marine practice, where reliability is absolutely essential. A number of ships are now in regular service fitted with modern stokers, and the results obtained fully justify their use.

The principal application of stokers in the marine field has been under water tube boilers, their application to the Scotch boilers not having been commercially successful; the shape of the furnace and the amount of cooling surface present in a furnace of this type of boiler being primarily responsible. Where high furnace settings are available under water tube boilers, the Underfeed type of stoker may be used with good results, especially if there is room below the stoker for a deep ashpit and clinker grinder. For the low set water tube boilers, such as are usually encountered in marine work, the Chain Grate type of stoker is undoubtedly the most suitable. The design is such that a refractory arch may be used over the coal bed, insuring high furnace temperatures, and a correspondingly rapid, efficient combustion, so necessary for these small furnaces. By using a chain grate stoker of the Forced Blast type, with the air supplied from a number of separately controlled compartments under the grate, excellent control of combustion is secured and correspondingly high boiler efficiencies are obtained. The power required to operate this type of stoker is small, an item of importance on shipboard, where auxiliary power is usually generated in small and relatively inefficient apparatus.

Plate No. 4 shows a Babcock & Wilcox Forced Blast Chain Grate stoker installed under a Marine Water Tube Boiler of that make, used for test purposes in the Bayonne, N. J. plant of that company. This is the same boiler that was used for the oil burning tests reported in Table No. 2 and the Pulverized Coal tests reported in Table No. 5. The standard setting height was not changed, this leaving a furnace volume of 167 cu. ft. above the stoker. An ash conveyor manufactured and supplied by the Allen-Sherman-Hoff Company of Philadelphia, Pa. was installed at the discharge end of the stoker to crush and move the ashes from the stoker to a convenient spot for disposal.

Table No. 6 shows the evaporation results secured with this Forced Blast Chain Grate Stoker, and Plate No. 5 shows graphically the efficiencies and total radiation and unaccounted for losses recorded in Table No. 6. From an analysis of these results it may be noted that even at very high rates of B.T.U. released per cu. ft. of furnace volume, excellent combustion results were obtained with high CO_2 values and negligible smoke in the products of combustion.

A number of different coals were tried on this stoker, including high and low fusing temperatures ash coals and high and low volatile coals. Excellent operating results were obtained with all of these coals, although the boiler efficiencies obtained with them varied, depending upon the coking properties of the coal, and the size of the coal fed to the stoker. At no time was difficulty experienced because of low fusing ash in the coal.

In closing it is gratifying to note the increased interest operators of modern steamships are taking in the movement to obtain more efficient combustion conditions in their furnaces. Present day competition has developed the realization that it is not only necessary to employ higher pressure and higher temperature steam, but that it is equally important that modern combustion equipment be used in generating this steam, to insure the most economical use of the fuel when it is fired in the small furnaces available under marine boilers.

TABLE No. 1

Evaporation Tests of a Scotch Boiler fitted with Babcock & Wilcox 13″ Mayflower Oil Burners League Island Navy Yard, Philadelphia, Pa, Tests made by Commander H.H. Norton, U.S.N. and Mr. C. J. Jefferson, U.S.S.B. Boiler Heating Surface 2555 sq. ft. Total Furnace Volume 483 cu. ft.

Test Number		1	2	3	4
Date	1924	3–14	3–14	3–18	3–18
Duration	Hours	2	2-1/2	3	2
Furnace Volume	cu. ft.	482.89	482.89	482.89	482 89
Number of burners used		3	3	3	3
Sprayer Plate Number		5520	5220	5020	5020
PRESSURES:					
Steam in boiler	lbs/sq. in.	145.80	145.84	147.10	149.45
Oil at the burners	"	219.52	206.69	241.20	207.55
TEMPERATURES:					
Room	°F	64.0	68.57	70.08	77.05
Oil at the burners	"	160.52	176.15	185.40	173.65
Gases in the uptake	"	499.57	533.84	581.50	635.75
Feed Water	"	220.28	205.42	201.26	205.70
Steam at calorimeter	"	296.04	296.46	298.00	298.10
DRAFTS:					
Base of stack	inches of H_2O	.056	.095	.094	.271
Gas Analysis-Uptake CO_2	%	10.2	12.25	13.86	13.21
Gas Analysis-Uptake O_2	"	6.91	4.49	2.18	2.99
Gas Analysis-Uptake CO	"	0	0	0	0.03
Gas Analysis-Uptake N_2	"	82.89	83.26	83.96	83.77
Quality of Smoke (Ringelman scale)		Haze	Haze	0	0
Total oil burned	lbs.	1478	2667	4025	3249
Total water fed to boiler	"	21720.5	39190.0	58785.0	45247.5

(Continued)

(Continued)

Test Number		1	2	3	4
CALCULATED RESULTS:					
Quality of steam leaving boiler	%	99.46	99.48	99.55	99.43
Oil burned per hour	lbs.	739.0	1066.8	1341.66	1624.50
Oil burned per burner per hour.	"	246.33	355.6	447.22	541.50
Oil burned per sq. ft. H.S. per hour.	"	0.289	0.417	0.525	0.635
Oil burned per cu. ft. Fur. Vol. per hour.	"	1.53	2.20	2.77	3.36
Btu released per cu. ft. of Fur. Vol.		28370.8	40794.6	51364.1	62304.5
Actual evaporation per hour	lbs.	10860.20	15676.0	19595.0	22623.75
Evaporation per hour (corrected for moisture)	"	10801.55	15594.48	19506.8	22517.4
Water evaporated per lb. of oil.	"	14.695	16.634	14.604	13 920
Factor of evaporation		1.0369	1.0522	1.0566	1.0523
Water per hour F & A 212°F	lbs.	11200.13	16408.51	20610.9	23695.08
Water evaporated per sq. ft. H.S. per hour	"	4.25	6.13	7.66	8.85
Water per sq. ft. of H.S. F & A 212°F. per hour	"	4.38	6.4	8.06	9.27
Equivalent evaporation per lb. of oil	"	15.155	15.381	15.362	14.586
Btu per lb. of oil		18543.0	18543.0	18543.0	18543.0
Efficiency of boiler		79.31	80.49	80.39	76.33
HEAT BALANCE:	%				
Heat absorbed by boiler	"	79.31	80.5	80.4	76.3
Heat loss due to combustion of hydrogen	"	6.2	6.2	6.2	6.4
Heat loss due to dry chimney gases	"	10.9	9.8	9.6	11.0
Heat loss due to CO	"	0	0	0	0.1
Radiation and unaccounted for losses	"	3.6	3.5	3.8	6.2
Total		100.0	100.0	100.0	100.0
OIL ANALYSIS:					
Moisture	"	0	0	0	0
Hydrogen	"	10.97	10.97	10.97	10.97
Carbon	"	85.47	85.47	85.47	85.47
Sulphur	"	2.15	2.15	2.15	2.15
Gravity-Baume'	Degrees	17.25	17.25	17.25	17.25
Heat value per lb.	B. t. u.	18543	18543	18543	18543

TABLE No. 2

Evaporation Tests of a Babcock & Wilcox Marine Boiler Fitted with 3 Babcock & Wilcox 13″ Mayflower Type Oil Burners Tests made by Mr. T. B. Stillmant and E. L. Boland-B. & W. Co. Boiler Heating Surface 2755 sq. ft. Superheating Surface 256 sq. ft. Furnace Volume-230 cu. ft.

Test No.			A la	A 4 ack	A 7
Date		1926	8-27	9-2	7-30
Duration		Hours	7	2	4-1/2
Type of burner				13″-Mayflower	
PRESSURES					
Setam:					
Drum		lbs/sq. in.	207.0	209.9	213.4
Calorimeter		”	206.0	208.9	212.4
Superheater outlet		”	205.0	206.0	206.7
Burner turbines		”	—	124.0	155.7
Oil at burners		”	109.8	216.1	174.5
TEMPERATURES					
Room		°F	126.3	91.9	105.7
Oil at burners		”	246.1	249.3	256.1
Uptake gas		”	434.0	497.4	537.7
Feed Water		”	79.3	76.0	75.0
Steam at calorimeter		”	305.7	304.0	307.7
Steam at superheater outlet		”	436.1	461.5	483.7
DRAFTS					
Room pressure		in. of H_2O	—	—	3.188
Furnace		” ” ”	.057	.081	1.675
Uptake		” ” ”	.284	.992	4.307
GAS ANALYSIS					
Top of 1st pass	CO_2	%	13.85	14.02	13.82
	O_2	”	2.72	2.52	2.82
	CO	”	0	0	0
	N_2	”	83.43	83.46	83.36
Uptake	CO_2	”	13.36	13.60	12.50
	O_2	”	3.34	3.03	4.66
	CO	”	0	0	0
	N_2	”	83.30	83.37	82.84

(Continued)

(Continued)

Test No.		A 1a	A 4 ack	A 7
Quality of smoke. Ringelman scale		0–1/4	0–1/4	0–1/4
Total oil burned	lbs.	5014.7	2687.4	8365.0
Total water evaporated	"	65677.0	33856.0	102374.0
CALCULATED RESULTS				
Quality of steam at S.H. inlet	%	99.70	99.58	99.76
Oil burned per hour	lbs.	716.4	1343.7	1858.9
Oil burned per burner per hour	"	238.8	447.9	619.6
Oil burned per sq. ft. H. S. per hour	"	.260	.488	.675
Oil burned per cu. ft. F. V. per hour	"	3.115	5.842	8.082
Btu released per cu. ft. of Fur. Vol. per hour	"	57449.9	107837.5	150519.2
Water evaporated per hour	"	9382.4	16928.0	22749.8
Water evaporated per lb. of oil	"	13.097	12.598	12.238
Calculated sat. steam temp.	°F	389.78	390.18	390.46
Degrees superheat at S. H. outlet	"	46.32	71.32	93.24
Factor of evaporation		1.2187	1.2369	1.2502
Total water evap. F & A 212° F	lbs.	80041.2	41876.1	127983.9
Water per hour F & A 212° F	"	11434.4	20938.1	28440.9
Water per sq. ft. HS F & A 212° F per hour	"	4.150	7.600	10.323
Equivalent evaporation per lb. of oil	"	15.961	15.582	15.299
Btu per lb. of oil		18443	18459	18624
Boiler efficiency	%	83.98	81.92	79.71
HEAT BALANCE				
Heat absorbed by boiler & superheater	%	83.98	81.92	79.71
Heat loss due to combustion of hydrogen	"	6.22	6.56	7.07
Heat loss due to dry chimney gases	"	6.49	8.40	9.70
Radiation & unaccounted for losses	"	3.31	3.12	3.52
Total		100.00	100.00	100.00
ULTIMATE ANALYSIS of the OIL				
Carbon	%	84.92	84.92	85.82
Hydrogen	"	10.96	10.96	11.39
Sulphur	"	1.72	1 72	1.40
Oxygen & Nitrogen	"	2.40	2.40	1.39

TABLE No. 3

Ev. poration Tests of a Babcock & Wilcox Express Type Boiler, Fitted with 11 Babcock & Wilcox Cuyama Type Oil Burners. League Island Navy Yard- Philadelphia, Pa. Tests made by Commander A. M. Penn, U. S. N. Boiler Heating Surface-7565 sq. ft. Superheating Surface-753 sq. ft. Furnace Volume-751 cu. ft.

Test Number		30–14	30–15	30–16	30–17	30–18	30–19	30–20
Date	1912	6–9	6–9	6–10	6–10	6–13	6–13	6–13
Duration	Hours	3	3	3	2	3	$2\frac{1}{2}$	2
Number of burners used		4	6	11	11	4	9	11
Sprayer plate size.		104	104	104	104	103	103	103
PRESSURES								
Steam								
Drum	lbs/sq. in.	296.1	295.8	295.2	295.6	297.1	296.0	295.9
Calorimeter	„	294.1	292.3	283.2	281.6	291.6	—	—
Below boiler stop	„	290.1	285.5	254.2	245.5	294.9	278.3	272.9
Oil at burners	„	197.2	197.6	197.3	245.9	169.4	170.6	169.3
Air in fireroom	inches of H_2O	4.037	5.987	8.752	9.548	2.739	4.952	5.827
Gas in furnace	„	.370	1.011	3.306	3.707	.116	1.188	1.881
Gas in uptake	„	−.093	−.046	+.192	+.242	−.115	−.035	+.046
TEMPERATURES								
Room	°F	101.7	99.3	97.8	109.1	115.1	115.5	114.6
Oil at burners	„	119.6	119.6	120.0	120.0	119.9	120.0	120.0
Uptake	„	532.0	614.0	715.0	729.0	490.0	617.0	648.0
Feedwater	„	197.6	196.9	196.9	198.0	195.1	194.7	197.2
Calorimeter	„	315.5	314.7	302.4	296.7	315.3	—	—
Steam leaving superheater	„	465.3	480.6	486.5	484.5	459.5	482.9	483.6
GAS ANALYSIS								
CO_2	%	13.02	12.07	12.76	13.03	12.90	12.42	12.44
O_2	„	3.27	4.70	3.71	3.32	3.53	4.27	4.40
CO	„	.11	.02	.00	.09	.04	.03	.00
N	„	83.60	83.21	83.53	83.56	83.53	83.28	83.16
Quality of smoke-Ringelmann scale		$0-\frac{1}{4}$	$0-\frac{3}{4}$	$\frac{1}{2}-\frac{3}{4}$	$1\frac{1}{2}-2$	$0-\frac{1}{4}$	$0-\frac{1}{4}$	$0-\frac{1}{4}$
Total oil burned	lbs.	11178.0	16752.0	30693.0	22710.0	8517.0	15975.0	15620.0
Total water evaporated	„	165969.0	238896.0	418256.0	306934.0	127560.0	230520.0	221326.0
CALCULATED RESULTS								
Quality of steam at S. H. inlet	%	99.28	99.26	98.60	98.29	99.26	—	—
Oil burned per hour	lbs.	3726.0	5584.0	10231.0	11355.0	2839.0	6390.0	7810.0
Oil burned per burner per hour.	„	931.5	930.7	930.1	1032.3	709.8	710.0	710.0
Oil burned per sq. ft. of H.S. per hour	„	.493	.738	1.352	1.501	.375	.845	1.032
Oil burned per cu. ft. of Fur. Vol. per hour	„	4.961	7.435	13.623	15.120	3.780	8.509	10.399

(Continued)

(Continued)

Test Number		30–14	30–15	30–16	30–17	30–18	30–19	30–20
Btu released per cu. ft. of Fur. Vol. per hour	lbs.	95747.3	143495.5	262923.9	291816.0	72954	164223.7	200700.7
Water evaporated per hour	"	55323.0	79632.0	139452.0	153467.0	42523.0	92208.0	110663.0
Water evaporated per sq. ft. of H. S. per hour	"	7.313	10.526	18.434	20.286	5.621	12.189	14.628
Water evaporated per lb. of oil	"	14.848	14.261	13.630	13.515	14.978	14.430	14.169
Calculated sat. steam temp.	of	419.0	417.7	407.8	403.7	420.2	415.5	413.8
Degrees superheat at S. H. outlet	"	46.3	62.9	78.7	80.8	39.3	67.4	69.8
Factor of evaporation		1.1056	1.1161	1.1222	1.1206	1.1041	1.1204	1.1187
Water per hour F & A 212° F.	lbs.	61165.1	88877.3	156493.0	171975.1	46949.6	103309.8	123798.7
Water per sq. ft. of H. S. F & A 212° F per hour	"	8.085	11.748	20.687	22.732	6.206	13.657	16.364
Equivalent evaporation per lb. of oil	"	16.416	15.917	15.296	15.145	16.537	14.167	15.851
Weight of dry gases per lb. of oil	"	16.49	17.85	16.94	16.49	16.71	17.34	17.36
Weight of dry air per lb. of oil	"	16.59	17.94	17.04	16.58	16.80	17.41	17.40
Btu per lb. of oil		19300	19300	19300	19300	19280	19280	19280
Boiler efficiency	%	82.54	80.03	76.91	76.15	83.24	81.37	79.78
HEAT BALANCE								
Heat absorbed by boiler & superheater	"	82.54	80.03	76.91	76.15	83.24	81.37	79.78
Heat loss due to combustion of hydrogen	"	6.72	6.94	7.15	7.12	6.37	6.69	6.77
Heat loss due to dry chimney gases	"	8.82	11.43	13.00	12.71	7.80	10.83	11.53
Heat loss due to CO	"	.38	.07	.00	.31	.14	.11	.00
Radiation and unaccounted for losses	"	1.54	1.53	2.94	3.71	2.45	1.00	1.92
Total								
Analysis of the Oil		100.00	100.00	100.00	100.00	100.00	100.00	100.00
Carbon	"	86.00	86.00	85.90	85.90	85.90	85.90	85.90
Hydrogen	"	11.70	11.70	11.60	11.60	11.40	11.40	11.40
Sulphur	"	.65	.65	.65	.65	.96	.96	.96
Nitrogen	"	.24	.24	.24	.24	.29	.29	.29
Oxygen and undetermined	"	1.41	1.41	1.61	1.61	1.45	1.45	1.45
Gravity-Baume	Degrees	26.1	26.1	25.55	25.55	25.55	25.55	25.55

TABLE No. 4

Pulverized Coal Evaporation Tests of a Scotch Boiler Fitted with Peabody Burners and using Kennedy-Van Saun Mill-League Island Navy Yard- Philadelphia, Pennsylvania; Tests made by Commander J. S. Evans, U. S. N. and Mr. C. J. Jefferson U. S. S. B.

Boiler Heating Surface 2555 sq. ft. Total Furnace Volume 470 cu. ft. Air Heating Surface 638 sq. ft.

Test No.		1	2	3	4	5
Date started	1927	7–27	7–29	7–31	8–2	8–4
Date terminated	1927	7–29	7–31	8–2	8–4	8–6
Duration	Hours	48	48	48	48	48
PRESSURES						
Steam in boiler (absolute)	lbs/sq. in.	190.3	190.0	190.8	191.5	192.1
Air-Primary						
Discharge from fan	in. of H_2O	1.60	1.63	1.97	2.19	2.20
Burner Tuyere	" " "	.95	.93	1.11	1.26	1.33
Air-Secondary	" " "					
In double front	" " "	.53	.43	.56	1.03	1.23
TEMPERATURES						
Room	°F	101.0	96.0	98.0	91.0	95·0
Primary air-						
At feeder end of mill	"	145.0	134.0	132.0	133.0	108.0
Inlet to fan	"	150.0	142.0	142.0	147.0	148.0
At distributor	"	144.0	147.0	140.0	140.0	138.0
Secondary air-						
Entrance to air heater	"	89.0	88.0	86.0	80.0	75.0
Air ducts leaving air heater	"	197.0	199.0	202.0	201.0	198.0
At burner	"	214.0	217.0	215.0	211.0	206.0
Gas leaving boiler	"	501.	551	599	628	640
" leaving air heater	"	415	452	502	532	538
DRAFTS						
Above air heater	in. of H_2O	.20	.348	.342	.328	.342
Below air heater	" " "	.20	.345	.337	.324	.333
Rear of furnace	" "	.176	.297	.246	.209	.190
GAS ANALYSIS						
Uptake CO_2	%	16.18	16.71	17.01	17.10	16.26
Uptake O_2	"	2.65	1.97	1.72	1.57	2.78
Uptake CO	"	.005	0	.037	.079	.013
Uptake N_2		81.17	81.32	81.23	81.25	80.95
CALCULATED RESULTS						
Total water fed to boiler	lbs.	615,315.	718,873.	811,266.	906,293	935,762
Total coal as fired	"	57859.0	66395.0	76767.0	86422.0	94808.0
Total refuse	"	3298.0	4050.0	4376.0	4321.0	5783.0
Moisture in the coal as fired	%	2.1	2.9	2.6	2.4	2.8
Coal burned per hour-as fired	lbs.	1205.4	1383.3	15 9.4	1800.5	1975.2
Coal per burner per hour-as fierd	"	401.8	461.2	533.2	600.2	658.2
Coal per hour-dry	"	1180.1	1354.2	1558.2	1757.2	1919.8
Coal per burner per hour-dry	"	393.4	451.4	5194	585.7	639.9
Coal per sq. ft. of H. S. per hour as fired	"	.471	.541	.626	.705	.773
Coal per sq. ft of H. S. per hour-dry	"	.461	.530	.610	.688	.751

(Continued)

(Continued)

Test No.		1	2	3	4	5
Coal per cu. ft. of Fur. Vol. per hour as fired	"	2.57	2.94	3.40	3.83	4.20
Coal per cu. ft. of Fur. Vol. per hour-dry-	"	2.51	2.88	3.32	3.74	4.09
Btu released per cu. ft. of Fur. Vol. per hour	"	37162.0	42057.0	48878.0	55841.0	59984.0
Actual evaporation per hour	"	12820.0	14977.0	16901.0	18881.0	19495.0
Quality of steam leaving boiler	%	99.80	99.81	99.81	99.82	99.84
Evaporation per hour (corrected for moisture moisture.)	lbs.	12794.0	14949.0	16869.0	18847.0	19464.0
Water per hour per sq. ft. of H.S.	"	5.01	5.86	6.61	7.38	7.63
Water per lb. of fuel-as fired	"	10.61	10.81	10.54	10.46	9.86
Factor of evaporation	"	1.052	1.052	1.050	1.054	1.050
Water per hour F & A 212° F.	"	13461.0	15720.0	17709.0	19859.0	20437.0
Water per sq. ft. H.S. F & A 212° F per hour	"	5.26	6.15	6.93	7.77	7.99
Equivalent evaporation per lb. of coal-as fired	"	11.16	11.36	11.07	11.02	10.34
Equivalent evaporation per lb. of coal-dry	"	11.41	11.61	11.37	11.30	10.65
Btu per lb. of coal-as fired	"	14460	14305	14376	14580	14282
Efficiency of boiler and air heater	%	74.89	77.06	74.72	73.35	70.26
HEAT BALANCE						
Heat absorbed by boiler	%	74.89	77.06	74.72	73.35	70.26
Heat loss due to moisture in coal	"	.17	.24	.22	.21	.25
Heat loss due to combustion of hydrogen	"	3.62	3.64	3.78	3.71	3.85
Heat loss due to dry chimney gases	"	6.95	7.53	8.43	9.10	9.84
Heat loss due to CO	"	.02	.0	.18	.26	.04
Radiation and unaccounted	"	14.35	12.53	12.67	13.37	15.76
TOTAL	"	100.00	100.00	100.00	100.00	100.00
COAL ANALYSIS-PROXIMATE						
Moisture	%	2.1	2.9	2.6	2.4	2.8
Fixed carbon	"	72.1	70.9	71.7	72.2	70.0
Volatile matter	"	20.1	20.1	20.0	20.4	21.1
Ash	"	5.7	6.1	5.7	5.0	6.1
COAL ANALYSIS-ULTIMATE						
Hydrogen	"	4.9	4.8	4.9	4.8	4.8
Carbon	"	81.8	80.7	81.3	82.2	80.6
Nitrogen	"	1.4	1.2	1.3	1.3	1.3
Oxygen	"	5.0	5.6	5.2	5.3	5.5
Sulphur	"	1.2	1.6	1.6	1.4	1.7
Ash	"	5.7	6.1	5.7	5.0	6.1
SCREENING.						
Thru 50 mesh	"	98.6	99.7	99.2	99.3	99.3
Thru 10 mesh	"	95.9	98.4	96.6	96.8	97.0
Thru 100 mesh	"	91.2	94.6	92.0	92.9	93.1
Thru 140 mesh	"	86.4	89.9	87.1	88.2	88.1
Thru 200 mesh	"	77.4	80.9	78.1	79.8	79.2
Thru 300 mesh	"	68.0	70.4	68.5	69.3	69.9
ASH AND REFUSE DATA						
Fusion temperature of ash	°F	<········ 2370 ········>				
Combustible in ash and flue dust	%	<········ 35 (approx) ········>				
Non-combustible in ash and flue dust	"	<········ 65 (approx) ········>				
Heat value of ash and flue dust	B.T.U.	<········ 5180 ········>				

TABLE No. 5.

Pulverized Coal Evaporation Tests of A Babock & Wilcox Marine Type Boiler Fitted with babcock & Wilcox Lodi Pulverized Coal Burners and using A Fuller-Bonnot Ball Mill.

Bayonne-N. J

Tests Made by T. B. Stillman and E. L. Boland of the Babcock & Wilcox Co. Boiler Heating Surface 2755 Sq. Ft. Furance volume 250 Cu. Ft.

1	Date of test, 1927	7/25	7/26	7/27	7/28	7/29	8/4	8/5	8/11	8/16	8/17
2	Duration (hours)	4½	7	5	6	7	7	6	5	6	6
3	Type of coal used	Dundon gas slack—West Virginia								Pocahontas	
4	Speed of mill (R.P.M)	32	32	32	32	32	32	32	32	32	32
5	Speed of exhauster (R.P.M.)	1,670.3	1,819.3	2,258.1	1,878.4	2,247.0	2,’31.0	1,842.0	2,350.0	2,336.8	2,336.0
	Steam Pressure (lbs.-sq. in.):										
6	Drum	205.1	192.3	191.0	187.1	191.0	185.1	205.3	176.3	183.3	187.0
7	Superheater inlet	203.3	189.5	188.5	186.2	189.3	184.5	205.0	178.0	182.2	186.4
8	Superheater outlet	202.0	187.0	187.3	185.6	187.6	183.9	204.7	180.2	180.7	184.7
	Temperatures (°F.):										
9	Furnace	2,126.1	2,237.3	2,302.2	2,252.1	2,291.1	2,255.9	2,078.0	2,483.5	2 450.6	2 535.3
10	Gas-uptake	420.8	424 2	465.4	442.7	479.6	440.2	424.2	471.3	468.1	454.4
11	Feed water	170.6	196.0	197.9	80.9	80.4	197.6	201.7	189.5	195.5	186.4
12	Superheated steam	438·3	431.0	455.6	444.5	458.6	428.1	429.9	442.7	451.7	449.8
13	Room	92.5	101.3	102.0	103.0	102.8	93.5	99.2	98.3	89.5	94.6
14	Air entering mill	166.5	169.0	183.8	185.9	188.5	183.9	187.2	147.6	166.5	
15	Calorimeter—superheater inlet	312.6	310.5	310.1	310.1	309.9	308.6	313.0	307.3	308.5	308.3
	Drafts (in H_2O):										
16	Furnace	.069	.086	.364	.053	.283	.049	.019	.167	.339	.568
17	Uptake	.269	.324	1.321	.538	1.299	.213	.076	.543	.875	1.206
18	Air pressure in burner pipe	2.20	2.59	4.36	2.99	4.27	2.01	1.46	2.61	2.435	2.35
	Gas Analysis (%):										
19	Top of 1st pass CO_2	13.63	13.72	13.17	13.54	13.70	13.24	13.34	13.09	13.45	13.82
20	Top of 1st pass O_2	5·53	5.29	5.89	5.51	5.22	5.66	5.66	5.89	5.55	5.28
21	Top of 1st pass CO	0 00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
22	Top of 1st pass N_2	80.84	80.99	80.94	80.95	81.08	81.10	81.00	81.02	81.00	80.90
23	Uptake CO_2	12.57	13.22	12.40	12.35	12.39	11.25	12.68	12.00	12.16	12.57
24	Uptake O_2	6.48	6.17	6.52	6.53	6.68	7.65	6.30	6.94	7.30	6.67
25	Uptake CO	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
26	Uptake N_2	80.95	80.61	81·08	81.12	80.93	81.10	81.02	81.06	80.54	80.76

(Continued)

(Continued)

27	Quality of smoke					Light gray ash					
28	Total water evaporated lbs.)	31,443.0	60,812.0	68,531.0	53,108.0	84,156.0	55,490.0	32,885.0	50,160.0	64,465.0	70,830.0
29	Total wet coal to burners (lbs.)	3,377.0	6,063.0	7,189.0	6,230.0	9,865.0	5,744.0	3,365.0	5,262.0	6,330.0	7,050.0
30	Moisture in coal entering mill (%)	3.55	2.48	2.84	4.82	4.27	5.06	6.22	3.53	4.55	3.93
31	Moisture in coal leaving mill (%)	1.07	.94	1.08	1.22	1.35	1.27	1.47	1.23	.63	.76
32	Quality of steam S. H. inlet (%)	99.93	99.91	99.90	99.92	99.88	99.87	99.93	99.84	99.81	99.86
33	Calculated saturated steam temperature at S. H. outlet (°F.)	388.68	382.58	382.70	382.02	382.82	381.24	389.66	379.75	379.96	381.60
34	Degrees superheat at S. H. outlet (°F.)	49.42	48.02	73.00	62.38	75.98	46.86	40.24	62.95	75.94	69.10
35	Factor of evaporation	1.1262	1.0973	1.1099	1.2242	1.2326	1.0946	1.0888	1.1119	1.1130	1.1192
36	Total water F & A 212° F. (lbs.)	35,399.8	66,729.0	76,062.6	65,014.8	103,730.7	60,739.4	35,805.2	55,772.9	71,749.5	79,272.9
37	Water evaporated per hour, actual (lbs.)	6,985.1	8,687.4	13,706.2	8,851.3	12,022.3	7,927.1	5,480.8	10,032.0	10,744.2	11,805.0
38	Water per hour F & A 212° F. (lbs.)	7,866.6	9,532.7	15,212.5	10,835.8	14,818.7	8,677.1	5,967.5	11,154.5	11,958.3	13,212.2
39	Water per hour F & A 212° F. per sq. ft. of H. S. (lbs.)	2.86	3.46	5.52	3.93	5.38	3.15	2.17	4.05	4.341	4.[illegible]96
40	Total dry coal (lbs.)	3,257.17	5,912.6	6,984.8	5,9[illegible]9.7	9,443.8	5,453.4	3,155.7	5,076.3	6,042.0	6,773.0
41	Wet coal per hour (lbs.)	750.4	866.1	1,437.8	1,038.3	1,409.3	820.6	560.8	1,052.4	1,055.0	1,175.0
42	Dry coal per hour (lbs.)	723.8	844.7	1,397.0	988.3	1,349.1	779.1	526.0	1,015.3	1,007.0	1,128.8
43	Dry coal burned per cu. ft. of F. V. per hour (lbs.)	2.90	3.38	5.59	3.95	5.40	3.12	2.10	4.06	4.03	4.52
44 45	Dry coal burned per sq. ft. of H. S. per hour (lbs.)	.263	.307	.507	.359	.490	.283	.191	.369	.366	.410
46	Wet coal per burner per hour (lbs.)	375.2	433.05	718.9	519.2	704.7	410.3	280.4	526.2	527.5	587.5
47	Dry coal per burner per hour (lbs.)	361.9	422.4	698.5	494.2	674.6	389.6	263.0	507.7	5[illegible]3.5	564.4
48	B. T. U. liberated per cu. ft. of F. V. per hour	37,642.0	44,632.9	75,062.5	52,507.4	71,366.4	41,939.[illegible]	27,680.1	52,897.7	57,036.6	63,876.6
	Actual evaporation per lb. of dry coal (lbs.)	9.65	10.28	9.81	8.96	8.91	10.17	10.42	9.88	10.67	10.46
49	Equivalent evaporation per lb. of dry coal (lbs.)	10.87	11.29	10.89	10.96	10.98	11.14	11.35	10.99	11.88	11.70
50	Total air supplied per hour (lbs.)	9,880.6	11,561.4	20,199.2	13,806.6	18,389.6	1,099.8	7,3[illegible]3.5	14,224.4	15,014.4	16,559.5
51	Primary air per hour (lbs.)	. . . .	7,372.9	8,825.4	7,117.3	7,598.2	7,0[illegible]8.8	4,327.2	. . . .	7,612.0	7,574.4
52	Per cent primary air	. . . .	63.77	43.69	51.55	41.32	63.14	58.85	. . . .	50.70	45.74
53	Per cent excess air used for combustion	34.98	32.88	38.05	34.74	32.24	35.94	36.01	37.88	35.06	32.84
54	Horsepower input, mill	20.7	20.8	24.2	21.0	23.6	20.8	20.8	20.9	20.8	21.0
55	Horsepower input, exhauster	3.8	4.5	18.7	6.1	17.2	4.2	3.2	6.2	7.1	7.3
56	Total H. P. of mill and exhauster	24.5	25.3	42.9	27.1	40.8	25.0	24.0	27.1	27.9	28.3
57	Total H. P. of mill and exhauster per 1,000 lb. wet coal per hour	32.64	29.29	29.83	26.10	28.91	30.47	42.79	25.75	26.45	24.09
58	B. T. U. per lb. of dry coal	12,980	13,205	13,428	13,293	13,216	13,442	13,181	13,029	14,153	14,132
59	Efficiency—boiler, superheater and furnace (%)	81.27	82.97	78.70	80.01	80.62	80.42	83.56	81.85	81.46	80.34
	Heat Balance (%):										
60	Heat absorbed by boiler	81.27	82.97	78.70	80.01	80.62	80.42	83.56	81.85	81.46	80.34

61	Loss due to moisture in coal	0.34	0.23	0.26	0.45	0.41	0.68	0.60	0.34	0.41	0.35
62	Loss due to water from combustion of H_2	4.11	4.01	4,01	4.01	4.09	3.99	4.03	4.15	3.27	3.24
63	Loss due to dry chimney gases	9.10	8.40	9.87	9.35	10.41	10.31	8.81	10.77	10.85	10.01
64	Radiation and unaccounted for losses	5.18	4.39	7.16	6.18	4.47	4.80	3.00	2.89	4.01	6.06
65	Total	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00
	Fineness of Coal (%):										
66	Through 48 mesh	99.8	99.8	99.6	99.6	99.6	....	99.8	99.8	99.8	99.8
67	Through 100 mesh	97.4	94.8	93.4	93.0	89.4	....	98.2	93.4	97.0	96.0
68	Through 200 mesh	83.0	78.0	75.6	72.0	66.2	....	89.0	74.6	72.6	77.6
	Furnace Ash Analysis (%):										
69	Combustible	....	0.15	0.05	0.80	0.00	....	0.36	0.50	0.07	0.00
70	Ash	....	99.85	99.95	99.20	100.00	....	99.64	99.50	99.93	100.00
	Uptake Dust Analysis (%):										
71	Combustible	7.47	6.39	15.18	9.05	23.84	9.93	4.42	6.23	16.22	14.88
62	Ash	92.53	93.61	84.82	90.95	76.16	90.07	95.58	93.77	83.78	85.12

Coal Analyses

ULTIMATE ANALYSIS OF DRY COAL

	Dundon gas slack Dundon, W. Va.	Pocahontas screened coal West Virginia
Carbon	74.75	81.59
Hydrogen	4.97	4.24
Nitrogen	1.39	1.18
Sulphur	.78	.59
Ash	11.79	10.20
Oxygen	6.32	2.20

PROXIMATE ANALYSIS

Moisture	4.94	4.24
Volatile matter	34.50	16.23
Fixed Carbon	53.71	73.57
Ash	11.79	10.20
Sulphur	.78	.59
Fusing point of ash F	2,710	2,705

(Continued)

Weights and Percentages of Furnace Ash

Date.	7/21–7/22	7/25–7/29	8/2–8/6	8/10–8/12	8/16–8/19
Number of hours of operation.	12.60	106.33	77.33	53.78	77.00
Total wet coal fired, lbs.	14,245	95,464	62,684	50,617	74,505
Average per cent of ash in coal.	14.76	11.66	12.17	13.98	9.23
Total ash in coal, lbs.	2,102.6	11,131.1	7,650.5	7,076.3	6,876.8
Total slag removed from furnace, lbs.	450	2,070	1,060	904	1,373
Average per cent of ash in slag.	99.75	99.75	99.72	99.55	99.98
Total ash in slag removed from furnace, lbs.	448.9	2,064.8	1,057.0	899.9	1,372.7
Percentage of total ash fired remaining in furnace.	21.34	18.55	13.82	12.72	19.96

TABLE No. 6

Evaporation Tests of a Babcock & Wilcox Marine Type Watertube Boiler Fitted With Babcock & Wilcox Forced Blast Chain Grate Stoker Tests By E. L. Boland of the Babcock & Wilcox Co.

Boiler Heating Surface 2706 SQ. FT. Furnace Volume 167 CU. FT. Grate Surface 58 SQ. FT.

1	Date of test, 1928.	7/2	7/5	7/6	9/11	9/12	9/28	9/14	9/18	9/25	9/26	9/27
2	Duration of test (hours).	6	5	6	5	6	6.5	6	6	6.5	6	6
3	Type of coal used.			Dundon Gas Slack						Pocahontas		
4	Speed of stoker (inches per min.).	2.22	3.39	2.27	3.22	2.82	2.25	2 82	2.82	2.82	3.32	2.27
5	Height of gate (inches).	4.0	4.5	4.0	4.5	4.0	4.0	4.0	4.0	4.0	4.5	4.0
	Steam pressure (lbs.—sq. in):											
6	Drum.	196.7	201.4	191.4	191.5	193.2	194.9	196.8	189.6	193.9	198.8	195.2
7	Superheater inlet.	195.7	198.8	189.3	187.9	191.0	193.7	195.0	187.8	191.7	196.2	193.6
8	Superheater outlet.	194.7	196.6	188.3	186.5	189.5	192.6	193.7	186.4	190.9	194.4	192.3
	Temperatures (°F.):											
9	Furnace.				2554	2617	2350	2637	2568	2457	2497	2389
10	Gas uptake.	464.8	541.0	462.9	555.9	518.4	467.7	524.2	522.7	495.2	527.3	473.9
11	Feed water.	207.3	187.8	200.8	187.3	176.1	187.2	193.4	193.4	180.9	187.5	176.9
12	Superheated steam.	445.3	466.7	441.6	462.8	455.1	445.8	461.5	459.9	452.3	464.0	441.8
13	Calorimeter superheater inlet.	311.5	311.8	310.8	310.6	310.7	310.8	311.0	310.0	311.0	311.7	310.3
14	Room.	108.9	109.6	99.6	96.1	95.9	88.5	96.3	86.9	90.3	101.2	95.4
15	Air entering stoker.	94.1	95.6	90.0	111.5	111.5	109.2	113.6	105.2	114.3	134.2	117.3
	Air pressures (in H_2O):											
16	Main duct.	2.39	3.52	2.97	3.54	3.50	2.27	2.97	3.25	2.65	3.17	1.99
17	Compartment No. 1.	0.057	0.217	0.179	0.179	0.138	0.100	0.128	0.141	0.231	0.369	0.238
18	Compartment No. 2.	0.319	0.636	0.289	0.776	0.541	0.459	0.616	0.627	0.736	1.145	0.701
19	Compartment No. 3.	0.158	0.386	0.155	0.444	0.304	0.419	0.343	0.316	0.666	0.987	0.627
20	Compartment No. 4.	−.086	−.075	−.072	−.012	0.000	0.014	0.000	0.000	0.334	0.497	0.351
21	Air duct over fire.	0.525	1.917	0.575	2.139	1.557	0.300	1.782	1.958	0.700	0.419	0.137
	Drafts (in. H_2O):											
22	Furnace.	0.153	0.203	0.176	0.139	0.147	0.087	0.093	0.131	0.050	0.061	0.052
23	Uptake.	0.630	1.349	0.612	1.259	1.036	0.598	1.002	1.001	0.992	1.377	0.730
	Gas analysis (%):											
24	Top 1st Pass. CO_2	15.06	14.57	15.10	15.71	15.89	14.22	16.11	15.92	14.13	14.78	13.74
25	Top 1st Pass. O_2	3.25	3.23	4 15	2.43	2.76	4.29	2.20	2.90	4.68	4.15	5.24
26	Top 1st Pass. CO	0.01	0.69	0.00	0.67	0.13	0.00	0.08	0.07	0.00	0.00	0.00
27	Top 1st Pass. N_2	81.68	81.51	80.75	81.19	81.22	81.49	81.61	81.11	81.19	81.07	81.02
28	Top 3rd Pass. CO_2	13.07	13 64	13.25	13.28	13.89	13.04	14.40	13.74	12.90	14.61	13.18
29	Top 3rd Pass. O_2	6.14	5.36	6.10	5.23	5.35	5.61	4.32	5.15	6.17	4.10	5.99
30	Top 3rd Pass. CO	0.00	0.35	0.00	0.33	0.00	0.00	0.00	0.00	0.00	0.00	0.00
31	Top 3rd Pass. N_2	80.79	80.65	80 65	81.16	80.76	81.35	81.28	81.11	80.93	81.29	80.83

(Continued)

(Continued)

32	Quality of smoke.	0 to .25	.50	Trace	50	.25	0 to .25	.25	0 to .25	Trace	0	0
33	Total water evaporated (lbs).	61245	83109	60353	75808	76305	61064	79954	73591	86114	101608	67908
34	Total wet coal fired (lbs.).	6959	9917	6968	9209	8840	7191	9009	8560	8772	10494	6836
35	Total refuse, dry (lbs.).	1014.9	1205.3	877.0	1165.1	1266.0	1037.0	1004.1	918	760	987	580
36	Moisture in coal (%).	8.58	8.42	7.60	11.15	7.07	7.31	7.32	8.06	4.30	3.21	2.74
37	Quality of steam S.H. inlet (%).	99.90	99.88	99.90	99.90	99.88	99.87	99.87	99.87	99.89	99.90	99.84
38	Calculated saturated steam temperature at S. H. outlet (°F.).	385.76	386.52	383.10	382.38	383.60	384.92	385.36	382.34	384.24	385.64	384.80
39	Degrees superheat at S. H. outl t (°F.).	59.54	80.18	58.50	80.42	71.50	60.88	76.14	77.56	68.06	78 36	57.00
40	Factor of evaporation.	1.0940	1.1255	1.0987	1.1248	1.1318	1.1146	1.1171	1.1170	1.1251	1.1245	1.1229
41	Total water F & A, 212°F. (lbs.).	66965.3	93539.2	66309.8	85268.8	86362.0	68061 9	89316.6	82201.1	96886.9	114258.2	76253.9
42	Water evaporated per hr. actual (lbs.).	11207.5	16621.8	10058.8	15161.6	12717.5	9394.5	13325.7	12265.2	13248.3	16934.7	11318.0
43	Water per hour F & A, 212°F. (lbs.).	11160.9	18707.8	11051.6	17053.8	14393.7	10471.1	14886.1	13700.2	1 905.7	19043.0	12709.0
44	Water per hour F & A, 212°F. per sq. ft. of H.S. (lbs.).	4.125	6.913	4.084	6.302	5 320	3.870	5.500	5.060	5.508	7.037	4.697
45	Total dry coal (lbs.).	6361.9	9082.0	6438.4	8182.2	8215 0	6665.3	8349.5	7870.1	8394.8	10157.1	6648.7
46	Wet coal per hour (lbs.).	1159.8	1983.4	1161.3	1841.8	1473.3	1106.3	1501.5	1426.7	1349.5	1749.0	1139.3
47	Dry coal per hour (lbs.).	1060.3	1816.4	1073.1	1636.4	1369.2	1025.4	1391.6	1311.7	1291.5	1692.9	1108.1
48	Dry coal burned per cu. ft. of F. V. per hour (lbs.).	6.35	10.88	6.43	9.80	8.20	6.14	8.33	7.85	7.73	10.14	6.64
49	Dry coal burned per sq. ft. of H. S. per hour (lbs.).	0.392	0.671	0.397	0.605	0.506	0.379	0.514	0.485	0.477	0.626	0.409
50	Wet coal burned per sq. ft. of grate surface per hour (lbs.).	20.00	34.20	20.02	32.89	25.40	19.07	26.81	24.59	23.27	30.15	19.64
51	Dry coal burned per sq. ft. of grate surface per hour (lbs.).	18.28	31.32	18.50	28.21	23.61	17.68	23.99	22.62	22.27	29.19	19.11
52	Actual evaporation per lb. dry coal (lbs.).	9.63	9.15	9.37	9.27	9.29	9.16	9.58	9.35	10.26	10.00	10.21
53	Equivalent evaporation per lb. dry coal (lbs.).	10.53	10.30	10.30	10.42	10.51	10.21	10.70	10.44	11.54	11.31	11.47
54	B. t. u. per lb. dry coal.	13072	13277	12867	13180	13206	12875	13310	13222	14642	14548	14572
55	B. t. u. liberated per cu. ft. F. V. per hour.	82994.1	144413.9	82734.8	129150.8	108289.2	79052.5	110912.2	103845.6	113182.7	147516.7	96758.1
56	Total air supplied per hour (lbs.).	12730.0	21633.3	13381.6	18982.2	16186.7	12858.5	16086.9	15643.3	19750.9	23837.7	16937.3
57	Per cent excess air used for combustion.	17.75	15.01	24.17	12.80	14.78	24.91	11.37	15.67	33.91	24.06	32.45
58	Average Horsepower per hour—stoker.	. . .	. . .	. . .	0.704	0.706	0.693	0 699	0.709	0.707	0.699	0.708
59	Average Horsepower per hour—ash conveyor.	. . .	. . .	. . .	0.247	0.206	0.248	0.184	0.333	0.152	0.295	0.174
60	Average total H. P. per hour—stoker and ash conveyor.	. . .	. . .	. . .	0.951	0.912	0.941	0.883	1.042	0.859	0.994	0.882
61	Efficiency—boiler, superheater and furnace (%).	78.17	75.28	77.68	76.72	77.23	76.95	78.01	76.62	76.48	75.44	76.38
	Heat balance (%):											
62	Heat absorbed by boiler.	78.17	75.28	77.68	76.72	77.23	76.95	78.01	76.62	76.48	75.44	76.38
63	Loss due to moisture in coal.	0.90	0.85	0.77	1.19	0.71	0.75	0.73	0.83	0.38	0.28	0.23
64	Loss due to water from combustion of hydrogen.	3.88	3.93	3.94	4.19	4.12	4.17	4.10	4.15	3.39	3.42	3.36
65	Loss due to dry chimney gases.	9.16	10.12	9.07	11.24	10.03	9.78	9.84	10.63	10.80	10.13	9.92
66	Loss due to incomplete combustion of carbon.	0.00	1.36	0.00	1.35	0.00	0.00	0.00	0.00	0.00	0.00	0.00

67	Loss due to unburned carbon in refuse	2.61	1.33	2.33	2.74	3.34	3.81	2.72	2.06	2.71	3.43	3.39
68	Radiation and unaccounted for	5.28	7.13	6 21	2.57	4.57	4.54	4.60	5.71	6.24	7.30	6.72
69	Total	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00
70	Refuse—per cent of dry fuel	15.95	13.27	13.62	14.24	15.41	15.60	12.03	11.06	8.63	9.72	8.72
	Refuse analysis:											
71	Combustible	14.68	15.65	15.09	17.35	21.07	21.57	20.58	16.25	34.18	35.14	33.78
72	Ash	85.32	84 35	84.91	82.65	78.93	78.43	79.42	83.75	65.82	64.86	61.22

Coal Analysis.

	Ultimate Analysis of Dry Coal	
	Dundon gas slack Dundon W. Va.	Pocahontas Screened coal West Virginia
Carbon.	74.47	85.97
Hydrogen.	4.90	4.49
Nitrogen.	1.42	1.24
Sulphur.	.66	.68
Ash.	11.94	5.74
Oxygen.	6.61	1.88
	Proximate Analysis	
Moisture.	7.36	2.41
Volatile matter.	33.85	18.40
Fixed carbon.	54.21	75.86
Ash.	11.94	5.74
Sulphur.	.66	.68
Fusing point of ash (°F.)*	2600	2410

* Defined as softening temperature in A. S. T. M. Specification D-22-24.

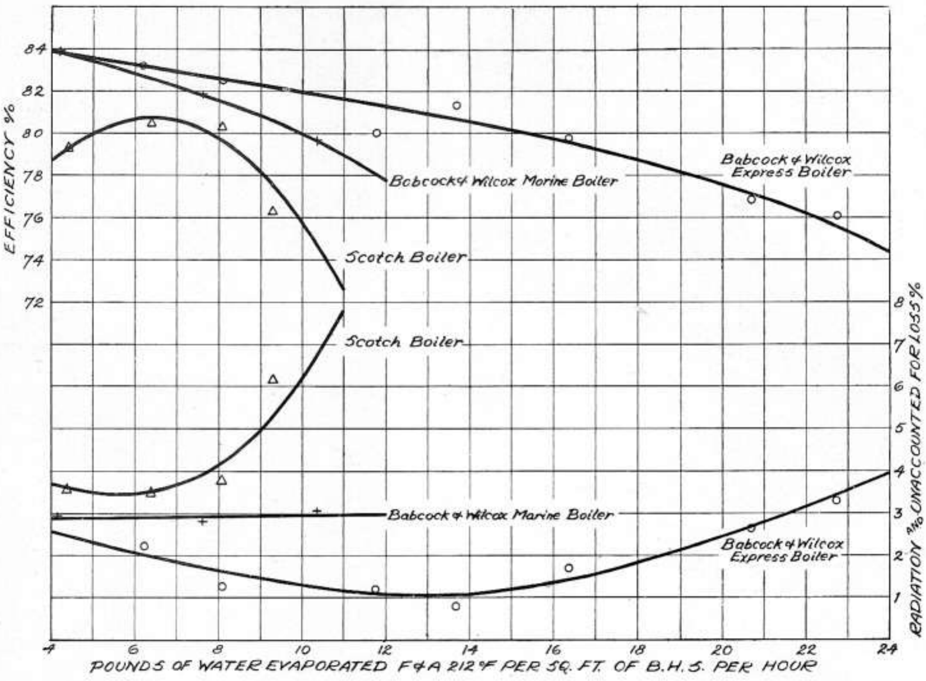


Plate 1.

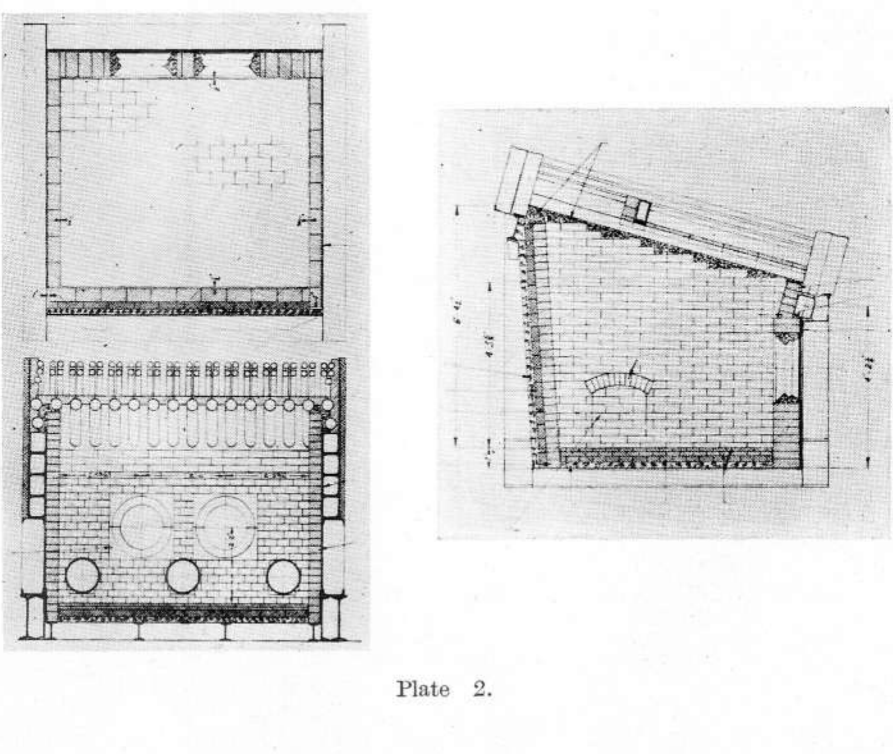


Plate 2.

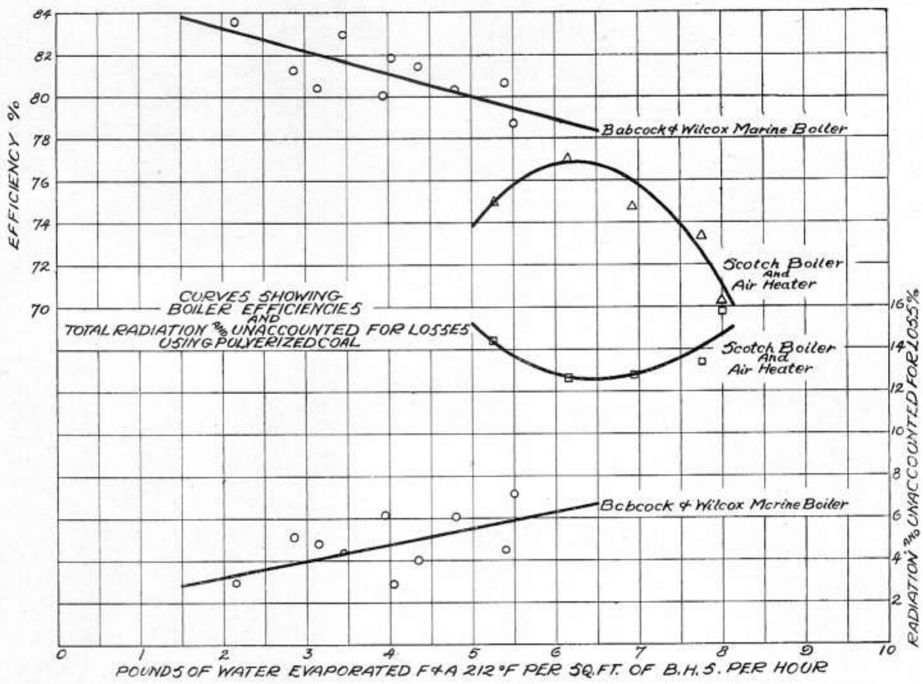
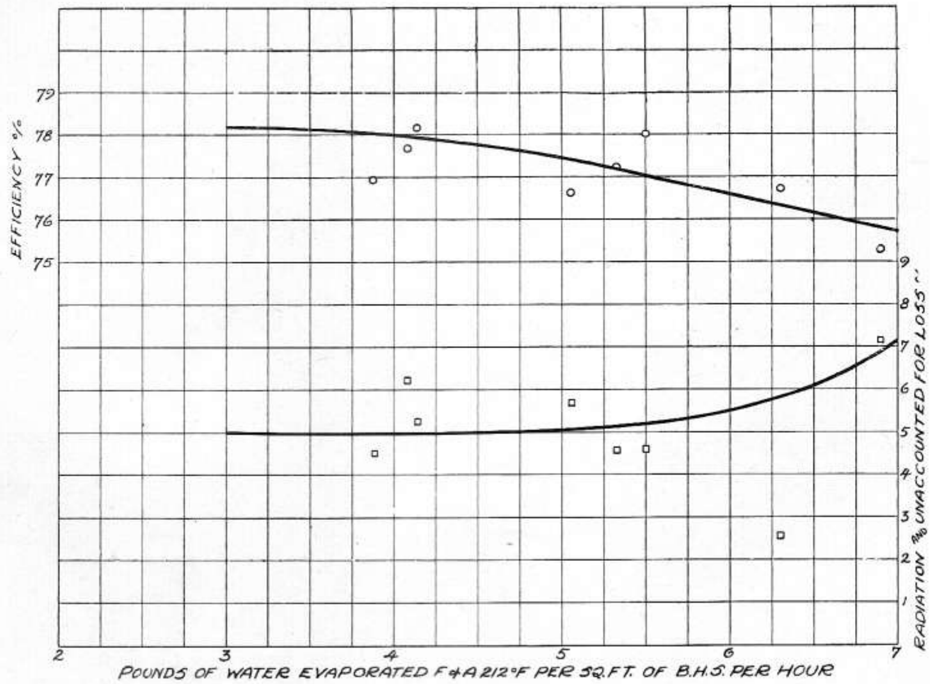


Plate 3.



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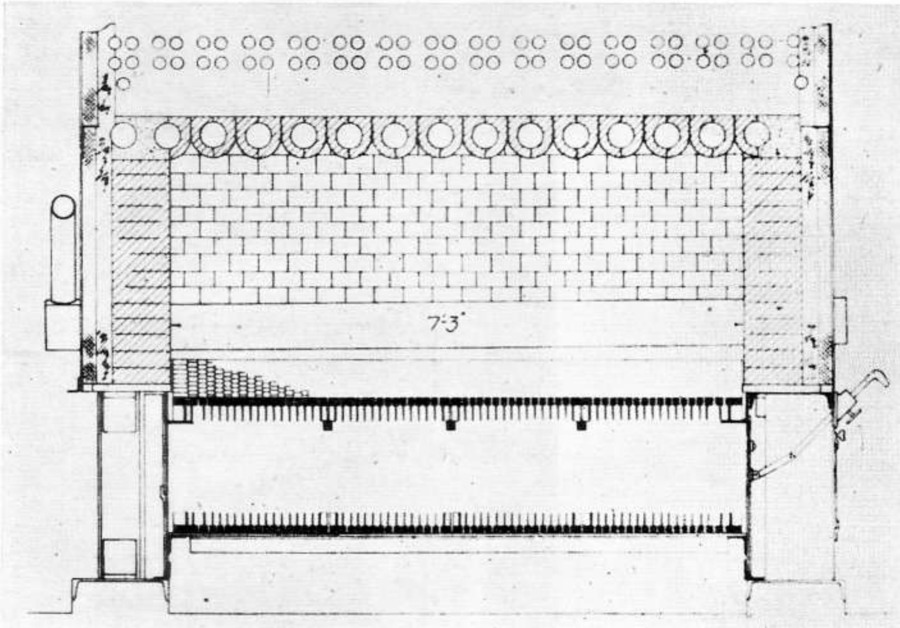
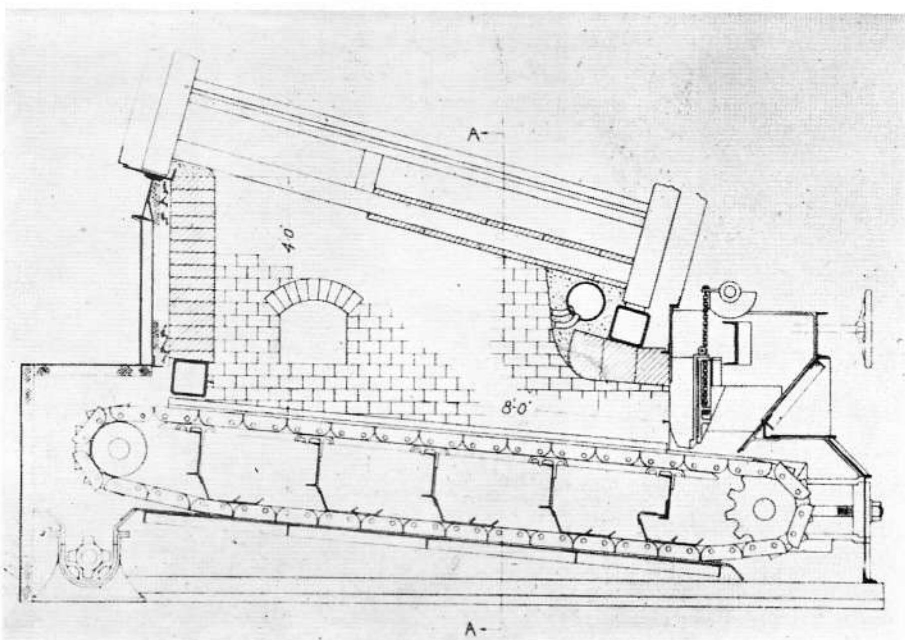


Front View Section A-A.



Longitudinal Section.

Plate 4.

Ship Vibration.

(Paper No. 457)

By ***Frank M. Lewis,*** *Prof. of Eng., Webb Institute of Naval Architecture, New York City, M. Soc. of Naval Architects & Marine Engs., M. Amer. Soc. of Mech. Engrs.*

Introduction.

Vibration has been a vexing problem to the Naval Architect since the advent of the steamship. The desire for ease and comfort in travel, and the use of the Diesel Engine, a prime mover having greater potential vibration producing qualities than either the reciprocating steam engine or the turbine, makes its study at the present time of the greatest importance.

It is the intention in this paper to summarize briefly existing knowledge on ship vibration, and to indicate those fields in which further research would be useful.

We distinguish between vibration of the hull structure proper and vibration which is associated exclusively with the machinery, such as torsional vibration of the Diesel, and the various forms of disk and blade vibration peculiar to the turbine. In this paper we deal primarily with the hull vibration.

Serious vibration of either the hull or its machinery is, in the great majority of cases, a resonance phenomenon; that is, vibration becomes serious when the frequency of some periodic force acting upon an elastic structure approaches the natural frequency of one of the normal modes of vibration of the elastic structure. Vibration at frequencies removed from the natural frequency of the structure is termed forced vibration.

Vibration may thus be classified as to its exciting causes and as to the normal modes of vibration of the hull structure. The exciting periodic forces may be classified:

Propeller forces.

Forces generated by the main or auxiliary engines; unbalance, torque reaction, minor forces, and noise.

Forces generated by the reaction of the surrounding water in which the ship moves.

The principal normal modes of vibration of a ship's hull can be classified as:

Vertical flexural vibration of two or more nodes:

Horizontal flexural vibration of two or more nodes:

Torsional vibration:

The tuning fork and rocking vibrations between engines and between engines and hull.

All other forms are designated as local vibrations.

The vertical and horizontal flexural vibrations are very similar in character and are illustrated in Fig. 1. The 2-noded vibration is the commonest, but vibration of 3, 4, or an even higher number of nodes may at times be serious. This type of vibration may be excited by unbalanced engines, or propellers unbalanced in mass or pitch. It should be noted that for points remote from the neutral axis there will be a certain amount of longitudinal motion due to the bending. This is sometimes designated as longitudinal vibration or "galloping", but in reality it is only a part of the flexural vibration.

In torsional vibration the ship vibrates as a tube about a longitudinal axis. There may be any number of nodes. In single screw vessels torsional vibration could be excited only by torque reaction forces, except possibly by unbalanced auxiliaries. In twin screw vessels torsional vibration can be excited by torque reaction forces, or by unbalanced engine or propeller forces, when the forces on the two sides of the ship are 180° in phase, Fig. (2).

The tuning fork and rocking vibrations are considered in connection with reciprocation engines.

The designation " local vibration " is merely a matter of convenience. Each one of these vibrations is merely one of the infinite possible number of modes of normal free vibration of the hull. Some of the commonest forms of local vibration are the vertical vibration of decks, and of pilot houses and other light superstructure.

Before considering in more detail the various phases of the subject mentioned, it will be of interest to consider, in a general way, the possible methods of avoiding vibration:

(a) Elimination of the exciting periodic forces;

(b) Avoidance of synchronism between the frequencies of the periodic forces and the elastic structure;

(c) So placing the periodic force that it acts upon the elastic body near a node;

(d) Damping and other special devices.

It is apparent as regards hull vibration that (a), elimination of the exciting forces is by far the most desirable method. If, from the viewpoint of vibration, the ideal ship is the entirely vibrationless ship, such an ideal is possible of attainment only by the complete elimination of exciting periodic forces. As long as periodic forces act upon the elastic hull, forced vibration will exist in it at all speeds, and even if one of the principal forms of vibration is avoided, one of the innumerable forms of local vibration is liable to be excited.

As has often been pointed out before, synchronism between the fundamental hull frequencies and the engine revolutions is a matter over which the designer has but limited control, for the dimensions of the hull and its scantlings, and the disposition of the weights are fixed by considerations quite apart from vibration. Likewise, the revolutions of the engines are fixed by considerations of engine and propeller design sufficiently important in themselves. If then synchronism occurs, it can be avoided only by serious sacrifice in other features of the design.

In many instances the avoidance of synchronism is unnecessary, for such perfection of balance can be attained in engine and propeller, that it is impossible to detect whether the ship is running near its synchronous speed or not.

Subsequent damage to the propeller might then cause vibration. While the avoidance of this possibility would be desirable, it is doubtful whether it would be worth while to sacrifice other advantageous features of design in order to obtain it.

Method (c) while interesting, is of no assistance to the designer, as the positions both of the nodes and the machinery are fixed by the general design of the ship and cannot be altered without great sacrifice.

Likewise (d) damping is of little use. While the amplitude of the vibration attained in any case is determined by the amount of damping present, this feature is not open to control except in special cases where dampers may be constructed to reduce certain forms of local vibration.

We now consider in more detail the exciting periodic forces and the possibility of their elimination.

Propellers.

The vibration producing forces of a marine propeller have long been known. We list them as:

Unbalance;

Variation in pitch between the blades;

Variation in the velocity of flow over the disk area, or close approach of the propeller tips to the water surface or parts of the ship.

At the present time practically all propellers are balanced. Since the mass of the propeller lies so nearly in one plane a static balance suffices. Dynamic balancing of a large diameter propeller would be a difficult matter.

Consider Fig. (6) and assume only blade (a) to be attached. If a tangential force A, assumed to be concentrated, is exerted by the blade upon the water, then an equal and opposite force A^1 is exerted by the shaft upon the hull. The forces due to the other blades are B, B^1, C, C^1. If all these blades are precisely equal, $A^1=B^1=C^1$, the resultant force upon the hull is zero.

But if the pitch of the blades is not equal, A^1 B^1 C^1 have a resultant, a force of once a revolution frequency tending to cause vertical or horizontal vibration in the hull. Since slip angles are normally small, even a slight variation in pitch may cause considerable variation in thrust. With machined blades vibration from this cause is not likely unless they become damaged. Rough cast blades are more difficult, but if made adjustable it should be possible to bring them all to nearly the same pitch.

Assuming the propeller to be balanced and the blades of equal pitch, vibration may still arise from variation in the wake stream over the disk area, or too close approach of the tips to the water surface or hull. As a blade travels around the variations in stream velocity produce a varying thrust and torque on the blade. The resultant forces on the hull are of a frequency per revolution equal to the number of blades or a multiple thereof.

The variation in wake may be due to;

Frictional wake, the velocity increasing from the center plane of the ship out.

Variation due to stern post struts, bossing, etc.

Eddies forming to the rear of the hull.

Propeller vibration from these causes is the most difficult form to combat. It can probably never be entirely eliminated; but with careful attention to the lines of the hull at the stern so as to insure a free flow to the propeller, ample clearances between the blades and fixed parts, and rigid construction at the stern of the ship, it should be very slight.

Reciprocating Engines.

The reciprocating steam engine and the Diesel engine are so similar in their vibration producing characteristics that it is convenient to consider them together.

The vibration producing forces can be listed as:

Unbalance;

Torque reaction;

The irregular torque on the shaft, producing torsional vibration;

Minor effects and noise due to gearing, valves, etc., principally in the Diesel Engine.

The general subject of reciprocating engine balance has been so fully covered in various papers and texts that more than a brief discussion here is unnecessary. In accordance with the usual practice, we break up the unbalanced forces into 1st order or primary forces and moments, both vertical and horizontal; and 2nd order or secondary forces and moments, vertical only, due to the angularity of the connecting rod. Any engine in which all these six forces and moments cancel is considered a completely balanced engine; and while 4th and higher order forces may still exist, these are very small, and for practical purposes may be neglected.

The balance problem in the 3 or 4 cylinder multiple expansion steam engine differs somewhat from that in the Diesel; or in the newer uniflow engines of several cylinders.

In the steam engine we usually deal with 3 or 4 cylinders, the reciprocating parts are of unequal weight, the cylinders possibly of unequal spacing; and the angular spacing of the cranks may be made unequal also.

In the 4-cylinder engine advantage may be taken of these inequalities of weight and crank spacing to procure a very satisfactory balance. The moving parts of the engine can be divided into two systems; the rotating parts comprising the crankshaft and the lower part of the connecting rod, and the reciprocating parts comprising the pistons and attached masses and upper part of the connecting rod. The rotating and reciprocating parts are balanced separately. The rotating parts can always be balanced by suitable counterweights affixed to the shaft.

There remain unbalanced the primary and secondary vertical forces and moments, four types altogether. The unbalance of any one of these types given by a particular cylinder may be represented by a vector, and to nullify it the vector sum for all the cylinders must be 0. Since both the horizontal and vertical components of the resultant vector must be 0, two equations must be satisfied to nullify each of the four types of unbalance.

As variables we have the ratios of three weights, three crank angles, and the ratio of two cylinder spacings. The ratio of cylinder spacings is fixed to a large extent by the engine design and therefore must be excluded as an adjustable variable. From the remaining six variables it is evident that 6 of the equations can be satisfied, and consequently three of the types of unbalance nullified.

In this way all but the secondary moment, which is usually the least important type, can be nullified. The paper of Taylor (33)[1] gives the clearest explanation of this subject. Alternative solutions are given by Ingles (34) and Bennett (35).

Although this system of unequal weights and crank angles was never universally adopted, it is undoubtedly the best method devised for the balance of the 4-cylinder steam engine.

The balance of the 3-cylinder engine is not as satisfactory. It is possible to nullify only two of the four types of reciprocating balance, and usually the primary and secondary rocking moments are left unbalanced.

Diesel engines, except for auxiliaries, are rarely built with less than 4 cylinders. The balance of the 4-cylinder engine, with one exception, cannot be considered as satisfactory. In the Diesel engine we are restricted to equal, or nearly equal, reciprocating masses and equal crank angles.

The system of unequal weights and crank angles would require extremely heavy counterweights to be fitted to the already massive pistons and there is no space for these. Further, the adoption of unequal crank angles would complicate the problem of torsional vibration, which already is sufficiently difficult. If the four cranks are placed in one plane, as in the 4-cycle single-acting type, the secondary forces are left unbalanced, if they are placed at 90° as in 2-cycle or double-acting engines, the primary and secondary moments are left unbalanced. Because of the extremely heavy reciprocating masses in these engines, either arrangement is unsatisfactory and is liable to produce serious vibration in the hull.

There does not appear to be any way open by which these difficulties might be overcome. We would therefore consider it inadvisable to use a 4-cylinder engine (with one exception) in any vessel in connection with which vibration is an important consideration.

The exception is in the case of the 2-cycle opposed piston engine. So far as balance is concerned, this is equivalent to an 8-cylinder engine and a completely balanced engine can be obtained.

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1) The numbers in brackets refer to the references in the Bibliography.

5-cylinder engines are rarely used. Their balance is as unsatisfactory as that of 4-cylinder engines.

The 6 and 8-cylinder, single-acting, 4-cycle engines are the best from the viewpoint of vibration. With the usual symmetrical crank arrangements, and excluding the air compressor if fitted, a completely balanced engine is obtained. The inherently perfect balance of these engines is often ruined by the fitting of an air compressor crank, and this will sometimes lead to serious vibration in the vessel. We have shown (36) how the perfect balance of the engine may be restored by relatively slight adjustments in the main reciprocating weights.

The 6 and 8-cylinder 2-cycle engine, or double-acting engines, where no two cranks are in the same plane, are nearly but not quite as good. In general, there will be rocking moments unbalanced. However, there are a considerable number of crank arrangements possible with these engines; 60 with the 6-cylinder, and no less than 2520 with the 8, and by a proper choice of crank arrangement a very good, though not perfect, balance may be obtained.

We discuss these arrangements in (36). Engines of more than 8-cylinders are rare. On account of the multiplicity of possible crank arrangements excellent balance may be obtained.

The most serious vibration problems in connection with the Diesel engine arise because of the highly irregular torque. The pressure in a Diesel cylinder during firing is around 500 lbs./in.^{2}, while the mean pressure during the firing stroke is of the order of 100 lb./in.^{2}. The resultant turning effort curve is highly irregular; the minimum and maximum torque in 4-cycle engines of as many as 8-cylinders varying from 0 to double the mean turning effort. Further, the curve is rich in harmonics.

The effect of this irregular torque is twofold.

The torque acts upon the shaft, producing torsional vibration.

An equal and opposite torque, called torque reaction, acts upon the ship's hull through the engine framing and produces vibration in the hull. These effects were first noted in connection with reciprocating steam engines, but as the torque is smoother with these engines, the problem is much less difficult than with the Diesel.

Torque Reaction.

Torque reaction may excite the fundamental torsional vibration of the hull, or innumerable forms of local vibration. There are also two other types of vibration associated with it which have not yet been considered. These are illustrated in Figs. (4) and (5). The vibration shown in Fig. (4) is said to be of the tuning fork type, and of course can occur only in twin screw ships. The two engines vibrate against each other like the prongs of a tuning fork, the elastic connection being the bottom of the hull between the engines. In Fig. (5) the motion is called transverse rock, the two engines vibrate together against the hull of the ship.

It is evident that the tuning fork vibration can be entirely eliminated by the simple expedient of tying the engines together by bars at the top. Such cross connections should be considered as fundamentally necessary as the holding down bolts. By tying the engines to the ship at the sides the rocking type of vibration can be reduced, but we do not consider this correction advisable unless trouble actually developes.

The danger of torsional or local forms of vibration of the hull being excited by the torque reaction still exists and can not be prevented by any system of engine bracing. No entirely satisfactory solution of this problem has been reached. Torque reaction vibration would be eliminated in twin screw ships if the engines, turning in opposite directions, were held to the same speed and their cranks in proper phase relation. Mechanical gearing is obviously impractical for this purpose. It has been proposed to hold the engines in step by the use of a differential governor device. This cannot be considered practical either.

Instead of mechanical gearing electricals connections may be used, a synchronous generator being affixed to each shaft and the generators cross connected. The most obvious difficulty is that when the ship is turning the two engines normally do not keep to the same speed, but adjust their speeds to deliver the same power. If they are to be held to the same speed, considerable power must be taken through the cross connection.

By the use of geared counterweights as proposed in (36) one particular component of the torque reaction can be eliminated at a particular speed.

Torsional Vibration.

The problem to torsional shafting vibration first came to the attention of marine engineers in connection with certain breakages in the shafting of steamships having reciprocating engines. With the development of the Diesel engine it was found that the same form of vibration existed, and in an even more acute form.

The problem is not directly connected with hull vibration and has been discussed in great detail in many papers. We give only a brief outline.

The entire shafting system, which comprises the engine shafts and pistons, flywheel, propeller, and all driven shafting and attached masses, can be considered as a torsionally elastic system. Such an elastic system has one or more modes of normal free vibration, distinguished by the number of nodes associated with each and a natural frequency of vibration. The irregular torque of the engine acting upon this shaft system, sets it into torsional vibration.

The torque curve for each cylinder can be broken up into its harmonic components. When the frequency of one of these components synchronizes with the frequency of one of the normal modes of vibration of the shaft system, a critical speed occurs, and the stress in the shaft may build up to a dangerous extent. A great many of these criticals exist. Fortunately, in any given case only a few of them will be of importance.

The problem is to calculate the frequency of the normal modes of vibration and the positions of the dangerous critical speeds; and, if necessary, to rearrange the system so as to avoid them.

Dampers and other special devices to reduce vibration have been proposed and applied to small engines, but can not be considered practical for the large engines used for ship propulsion.

The natural frequencies of the shaft system can be calculated and the positions of the criticals predicted with an error of only 1 or 2%. The problem is mathematically simpler than that of the flexural ship vibrations and the data that enter into the computation can be accurately determined. Further, it is usually possible to adjust the elastic system so as to meet any reasonable requirements.

Fortunately, in one of the most numerous types of installation trouble is rarely experienced. This comprises engine and flywheel amidships, long lineshaft, and propeller aft. In such an installation the dangerous criticals in the lineshaft are invariably below the operating range and those in the engine flywheel system above the operating range. Further, as regards vibration in the lineshaft, the installation possesses certain self-damping qualities which keep down the amplitude in passing through criticals. With the reciprocating steam engine fractured shafts were often experienced with a very similar lineup. The difference lies in the provision of the heavy flywheel for the Diesel which was not usually fitted to the steam engine. Other types of installation have not been so free of trouble. Ships with engines aft, Diesel electric rigs, Diesel driving pumps, etc., can all be considered as potentially liable to torsional vibration.

In the geared Diesel ship the problem of torsional vibration becomes of transcendent importance.

For successful operation of a gear drive we must satisfy the condition of positive torque; that is, the torque of the gears must not pass through negative values, or separation and pounding at the teeth will occur with resultant destruction of the gears.

When we consider the highly irregular torque of the engines and the fact that this irregularity is magnified by resonance, it will be seen that the problem is not easy.

Successful drives of this type can be designed, however, and a number of German vessels have been built on this system. Usually two engines are geared to a single propeller. The only American example is the "Herman Falk" which has been in operation for about two years with an entire absence of any gear trouble.

Turbines.

The turbine, with its purely rotating parts, might be expected to be the ideal means of ship propulsion from the vibration viewpoint.

Yet the hope of the early days of the turbine, that its adoption would result in a vibrationless ship, was never quite realized, and very serious vibration has been experienced on many turbine ships. Nevertheless, the turbine ship has more inherent possibility of being free from vibration than any other type of drive.

The first turbine ships were direct drive ships and very high propeller speeds were needed in consequence. These high speeds accentuated vibration due to the propellers.

The geared drive has reduced propeller speeds to normal, with a consequent reduction in the amount of attendant vibration. The remaining causes of vibration are:

Unbalance;

Torsional vibration;

Whirling vibration, disk and blade vibration, and other forms peculiar to the turbine.

With modern dynamic balancing machines and the high degree of precision attained on them, unbalanced rotors need not be considered as a possible cause of ship vibration.

Very serious difficulties have been experienced due to torsional vibration in geared turbine drives. The forces necessary to excite such vibration may arise from the variable torque of the propeller which may exist if the velocity over the disk area is not uniform; or from variations in the tooth spacing of the gears. In the early gear drives, cut with inaccurate master wheels, the accumulated errors in tooth position were often considerable. These accumulated inaccuracies around the wheel may be divided into their harmonic components of 1st, 2nd, 3rd order, etc., and each component in conjunction with one of the normal modes of vibration of the system, of which several may exist, would produce a torsional critical speed. Altogether a large number of such critical speeds may exist and they may be so closely spaced that it is impossible to avoid them.

When in a critical speed the torque in the shaft goes through a variation several times per revolution. The torque of the shafts entering the gear case not being equal, the difference is carried by the gear case and transmitted by it to the hull. A torsional critical speed in the shaft will then result in periodic forces acting on the hull. A far more serious effect, however, is the pounding at the gear which will result if the torque variation exceeds the mean torque. This pounding will in a short time destroy any gear.

The positions of these criticals may be calculated with accuracy. The difficulty of avoiding them is because there are so many. In the so-called "nodal drive" devised by Dr. J. H. Smith (38) the frequency of the various component systems about the central gear wheel are made equal.

This reduces the possible number of natural frequencies to two, with a consequent reduction in the possible number of critical speeds, and it is then much easier to avoid them. But the ultimate answer to this problem of torsional vibration does not lie in avoiding the criticals which may theoretically exist, but in accurate gear cutting, and then, still higher accuracy. With the accuracy attainable with modern gear cutting machines, it is questionable whether any consideration need be given to those criticals arising from possible inaccuracies of tooth spacing. The possibility of criticals arising from the propeller still exists and cannot be entirely ignored, but nevertheless the majority of turbine ships are designed without consideration of these criticals and vibration difficulties are rare.

There exist several other forms of vibration peculiar to the turbine, such as whirling vibration, vibration of disk and blades, etc., but a discussion of these is beyond the scope of this paper.

Auxiliary Machinery.

Under the heading of auxiliary machinery we include all moving machinery on the vessel except the main propelling engines. In the design of a ship the vibration producing possibilities of this machinery usually receive but scant attention. While the frequencies are usually too high to excite the fundamental flexural or torsional vibration of the hull, exceedingly annoying local vibrations may be generated, and often appear at astonishing distances from the sources.

The auxiliaries may comprise purely rotating machinery, turbines, pumps, blowers, generator armatures, etc., or reciprocating machinery. Freedom from vibration with purely rotating machinery can usually be assured by static and dynamic balance. With reciprocating machinery the problem is more difficult.

On Diesel ships diesel driven auxiliaries are customary. For reasons of cost three and four cylinder engines are often fitted, and serious vibration can often be traced to the poor balance of these engines. As much care should be given to having proper balance for the auxiliary engines as for the main propelling machinery. Assuming that balanced diesel auxiliaries are used, there still remains the possibility of vibrations from their torque reaction, but trouble from this source rarely occurs.

Vibration from auxiliaries is often accentuated by their being mounted on relatively flexible platforms or decks, which may vibrate in resonance with the engine. It is obviously desirable to avoid this condition.

Packing of cork or other resilient material is often placed under the smaller auxiliaries. This is excellent in reducing the transmission of high frequency vibrations and noise from the engine to the hull. It is usual, however, to fit holding down bolts metal to metal, and with this construction the full possibilities of the packing are not utilized. If resilient packing is used, the entire connection from the engine to the hull should be resilient; that is, packing should be used under the bolt heads as well as under the engine.

Resilient packing will not counteract the low frequency forces of unbalance or torque reaction, the flexibility being insufficient. For these forces spring mountings offer interesting possibilities. In general, it can be stated that the frequency of the engine or its mounting must be considerably lower than that of the periodic force to be annulled. If the frequency of the engine spring system is too high, the periodic force acting on the hull may be considerably greater than if no elastic mounting were used at all. Such mountings must therefore be designed with great care.

Water Effects.

But little study has been given, as yet, to the influence of the water surrounding a ship, upon its vibration. We discuss this subject in a paper to be presented to the Society of Naval Architects and Marine Engineers in November, 1929.

The most important effect of the surrounding water is a virtual increase in the mass of the hull, for if the hull vibrates the water under it must vibrate with it. This increase in the mass of the hull may amount to 50% or more. The water inertia is of greatest importance at the flat bottom amidships, with the result that the amplitude of vibration amidships is relatively less and at the ends relatively greater, and the nodes nearer the center, than if the hull were vibrating without surrounding water. The water inertia is proportional more nearly to the water line area than to the displacement, with the result that frequencies do not vary inversely as the square root of the displacement, but at a much slower rate.

Vibration due to a series of waves striking the hull at proper intervals is unlikely, for any waves having a frequency sufficiently high would be too short to have an appreciable effect upon the hull.

Assuming a ship in motion in still water and free from any periodic forces due to its machinery, the question arises whether it is possible for vibration to be generated in some way in the hull from the motion of the water past it.

The instability of certain forms of fluid motion has long been known; for example, it is known that if two currents are moving over each other, the common surface is unstable, breaking up into a wave form of increasing amplitude. The flapping of flags and sails in a steady current of air may be ascribed to the same cause, and has been investigated by Rayleigh.

A solution for the case of a weighted elastic plate moving over the surface of a liquid may be easily obtained as it is found that if the speed is over a certain limit the motion becomes unstable. It is found, however, that if this plate is given the approximate characteristics of a vibrating ship the required velocity is far above any attained in even the fastest ships, being of the order of several hundred feet per second.

It would thus seem unlikely for vibration to be generated as long as purely stream line motion is involved.

The production of eddies would appear to be a more probable source of vibration. It is known that the eddies produced behind a blunt body are periodic in character with resultant periodic forces upon it, and in the case of a ship such eddies might be of correct frequency to produce hull vibration. It cannot be stated definitely, however, whether this is likely to occur in the case of a ship, and in the examination of the vibration on many vessels the writer has found no case which could not be traced to forces produced by the machinery or propellers.

It is a well-known fact that the vibration of a ship increases in shoal water, a vessel often vibrating violently in shoal water which is vibrationless in deep. No entirely satisfatory explanation of this phenomenon has been proposed. There are several possibilities.

Bottom irregularities. As the vessel passes over the irregular bottom the stream velocity of the water under it changes, with resultant changes of pressure over the hull. On this basis the bottom irregularities would need to come at just the right intervals, which would seem improbable.

Reduction of the natural frequency.

In shoal water the water inertia effect increases with consequent reduction of the natural frequency, so that vibration which was formerly above the operating speed might be brought down into it.

Eddy formation.

It would seem probable that the restricted flow under the hull would promote eddy formation at the stern, with consequent periodic forces as explained before.

No definite conclusion can be reached as to the relative merits of these theories, and others may be suggested. Fortunately, as the majority of ships operate but little in shoal water the matter is not of great practical importance.

Computation of the Natural Frequencies of a Hull.

Flexural Vibration.

The empirical formula given below was proposed independently by Norman and Schlick, and is generally known as Schlick's formula:

$$N=\Phi\sqrt{\frac{I}{\Delta L^3}}$$

N=natural frequency of 2-noded vibration, in vibrations per minute.

I=least moment of inertia of the midship section in feet^4.

Δ=displacement in tons of 2240 lbs.

L=length in feet.

Φ=a constant depending on type of ship, determined from ships of know frequency.

A few values of Φ are given below:

Cargo ship	1,533,000
Tanker	1,610,000
Yacht	1,710,000
Large passenger vessel	1,722,000
Destroyer	1,882,000

Schlick's formula can be strictly correct only for similar ships, and under other conditions will yield only approximate results. Nevertheless, it is the simplest and most widely used method of estimating the natural frequency, and in the present state of our knowledge at least, will probably yield as accurate results as calculations by far more elaborate methods.

By modifying the constant Φ Schlick's formula will apply to 3 or 4-noded vibration, but these higher frequencies are so much affected by the exact shape of the load curve that the result cannot be expected to be more than a rough approximation. In this connection, it may be noted that ratios of the frequencies of a free-free bar for 2, 3, and 4 nodes are as 3.01^2, 4.99^2, 7^2, etc., and the hull frequencies would be roughly in these ratios.

Various other methods of calculating the natural frequency of a hull have been devised; see papers 7, 9, 14, 15, 17, 18, etc.

The simplest mode of procedure and the one followed by a majority of these writers is equivalent, in a general way, to a method proposed by Rayleigh for calculating the natural frequency of bars.

The elastic curve is assumed, usually being taken as that of a free-free bar of uniform cross section and elasticity, with a base line correction applied so that the center of gravity of the vibrating ship remains fixed in position. This assumed flexural curve may be fixed with greater accuracy by successive approximations, if desired.

Then the natural frequency is calculated from the condition that the kinetic energy of the system at mean position is equal to its potential elastic energy at extreme position.

This condition can be expressed in the form of an energy balance:

Potential energy in extreme position	=	Kinetic energy in mean position
Bending energy		Kinetic energy due to vertical motion of ship
Shear energy		Kinetic energy of surrounding water
		Kinetic energy due to longitudinal or rotational motion of ship.

The items are listed in the order of their magnitude.

To calculate the bending energy, a knowledge is needed of the elasticity of the hull. This is a very uncertain factor. According to the common theory of bending, the rigidity of the hull at any point is proportional to I, the moment of inertia of its cross section. Assuming that the common theory is strictly applicable to such a complex structure, there is a considerable uncertainty as to which of the hull members should be included in the calculation of I.

The compression members, in particular, are a source of difficulty. As has been pointed out by Pietzker (24), it is probable that plates in compression remote from stiffening flex sideways in the manner of a column and therefore do not contribute their full area to I. It would even seem probable that due to this failure of the compression members, the moment of inertia is not a constant at all, but depends upon the amplitude of bending and is different for hogging and sagging conditions.

Experiments have been made to determine the rigidity of a hull, those on the "Wolf" in 1905, reported by Prof. Biles, being classic. It was found that the " Wolf " as much less rigid than expected. The moment of inertia I always occurs in conjunction with the modulus of elasticity E, and it is a matter of convenience to speak as if the value of E were reduced in the hull, rather than I, although of course the latter is actually the case. On this basis the virtual E for the " Wolf " was found to vary from 20,000,000 to 29,000,000 lbs. per sq. in., as against an actual E of 30,000,000 for steel. The " Wolf " results have been endlessly discussed, but no comparable experiments have since been made.

The " Wolf " was a naval vessel of very light construction, and it is doubtful to what extent the results can be applied to more modern types of construction. Further reliable experiments on this point are badly needed, and until such experiments have been made it is useless to attempt to estimate the rigidity of a hull.

The same uncertainties apply to the calculation of the shear energy as to the bending energy, but since this is a much smaller item than the bending energy an error in its calculation is not of so much importance.

The vertical kinetic energy of the ship can be easily calculated provided the load curve is known. Unfortunately this is a troublesome item in itself to calculate and is seldom known accurately, certainly not when the vessel is in its design stage; and this is when its natural frequency is needed. However, but little error will result if the form of the load curve is approximated, provided the total displacement and general distribution of the weight are correct.

The water inertia effect has been already discussed.

Any bending of the ship is necessarily accompanied by a certain amount of longitudinal motion, greatest at the nodes and at points remote from the neutral axis. The kinetic energy of this motion constitutes the third item of the kinetic energy column. Any exact calculation of this item would be a tedious matter, but since it is a relatively small item, high precision is not necessary and it may well be allowed for by increasing the vertical energy by a fixed percentage. See paper of Nicholls (15).

In view of the many uncertain factors involved, in particular those factors pertaining to the elasticity of the structure, close agreement between observed and calculated frequencies is hardly to be expected when the calculation is made by purely rational methods, as outlined above.

Good agreement may be obtained, however, if the calculation is first made for a ship of known frequency, and the results then applied to a somewhat similar ship, provided the same approximations and assumptions are made in each case. This is the method followed in practice; E, the modulus of elasticity being made to play the part of an empirical constant. In thus making the calculations the items of shear, rotational and water energy are often neglected, but the magnitude of the water inertia, and the fact that it considerably modifies the form of the vibration curve, would make it desirable to include this latter item.

Various methods have been proposed for calculating the torsional frequency of a hull, (11), (15), (17). Any of the methods will yield good results providing the data entering into the calculations are correct. This is the difficulty. It is an exceedingly tedious matter to calculate the mass polar moment of inertia of a ship at numerous points of its length, even for a completed ship, and still more difficult for one in the design stage. The torsional rigidity of a hull is an even more uncertain factor than its flexural rigidity and so far as we are aware no experiments have ever been made on this point. A rational formula similar to Schlick's has been proposed by Lockwood Taylor (17), but as yet the empirical constants cannot be fixed with any certainty as observations of a sufficient number of ships have not been made.

In addition to the bending and torsional vibration of a hull there are innumerable local forms of vibration which may exist. Some of the commonest local vibrations are those of decks and of light superstructures. To attempt to predict the frequency of these various local vibrations in the design stage would be an impossible task, not only because of the theoretic difficulties involved but because of the innumerable ways in which such a complex structure may vibrate.

We would again emphasize the point that the elimination of exciting periodic forces acting on the hull, rather than the avoidance of resonance with those forces, if they exist is the most desirable method of obtaining a vibrationless ship.

The Effect of Hull Structure upon Vibration.

In formulating any relations between hull structure and vibration it must be kept in mind that such vibration is usually a resonance phenomenon. Vibration is not necessarily due to weakness of the hull, but to chance coincidence of a hull frequency with the frequency of some periodic force. Such coincidence might be avoided by a more rigid and stronger hull or by a more flexible and weaker hull.

The fundamental flexural and torsional frequencies cannot be appreciably altered by any changes which it might be practical to make in the hull structure. For the frequencies vary only as the square root of the stiffness, and the general scantling sizes are fixed by considerations quite apart from vibration. Neither can the loading be altered sufficiently for appreciable effect.

The local vibrations, however, may be considerably affected by slight structural changes. Unfortunately these local vibrations cannot be predicted with the vessel in the design stage, so that little can be done but correct them after they develope.

The most obvious way of changing the frequency of local parts is by stiffening, but this is often difficult to carry out without expensive changes. Another useful method of controlling the frequency of local vibrations is by the addition of a heavy mass to the vibrating part. This treatment is often very effective in the case of decks, vibrating in drum head fashion.

Dampers and dynamic vibration absorbers may be useful at times in controlling local vibrations. The use of these devices was first proposed by

Frahm. They are illustrated (in principle only) in Fig. (3). C is a deck, on it is mounted an oil filled tank containing a weight A mounted on springs B. If the frequency of the weight on its springs is made considerably lower than that of the vibrating deck the device is a damper. The weight tends to stay fixed in space and the motion of the deck is damped by the oil friction in the tank. If the frequency of the weight is made higher, approximately equal to that of the deck, the device becomes the dynamic vibration absorber. With proper adjustment between the frequencies of deck and weight and the proper degree of damping the device is much more effective than a pure damper of the same size. The theory of this device is described in (21). The same principle has also been applied to the torsional vibration of crankshafts. The fundamental difficulty with these devices is that vibration cannot be damped unless some vibration exists.

In Diesel ships rigid construction of the engine foundations is of special importance. With balanced engines the only periodic force exerted by the engines on the hull is the torque reaction. In avoiding annoying local vibrations from this source, transverse rigidity of the foundations is of greater importance than longitudinal, and if any sacrifice of continuity is necessary it should be made in longitudinal rather than transverse members.

When unbalanced engines are used, especial attention should be given to the forces they may bring into play on the foundations and the general structure of the ship in their vicinity. With 4-cylinder engines, often used for cargo vessels, these forces may be very large and in consequence much stronger foundations are needed than for balanced engines. This is a problem of structural strength rather than vibration.

The structure at the stern should also receive especial attention. It is desirable to have this structure as rigid as possible in order to avoid local vibrations from the propeller. In twin screw ships the attachment of propeller struts should be carefully considered. These should be connected to a bulkhead or rigid transverse frame so that any irregular forces exerted by the propeller will be transmitted to the hull as a whole, and not set local parts vibrating.

Instruments.

The first instrument developed for the study of ship vibration was the Pallograph of Schlick (2), working on the principle of the sesmiograph. More convenient machines utilizing the same principle have since been developed, the Torsiograph and Vibrograph of Geiger being particularly useful. A good description of these, and other forms of vibration recording instruments, will be found in the recent work of Timoshenko (22).

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Abbreviations:

T.I.N.A.	Transactions of the Institution of Naval Architects, Great Britain.
S.N.A.M.E.	Society of Naval Architects and Marine Engineers, United States.
J.S.G.	Jahrbuch der Schiffbautechnischen Gesellschaft, Germany.

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FIG. 1

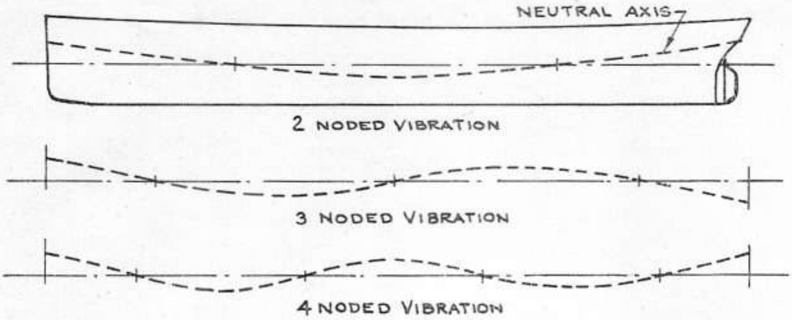


FIG. 2
TORSIONAL VIBRATION
IN HULL, FROM UNBALANCE

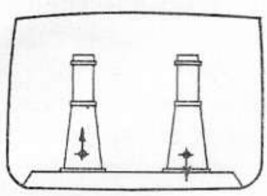


FIG. 3
DYNAMIC VIBRATION ABSORBER

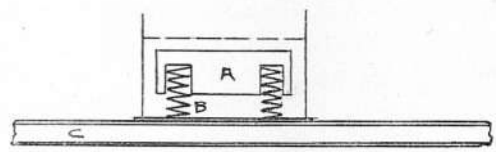


FIG. 4
TUNING FORK
MOTION

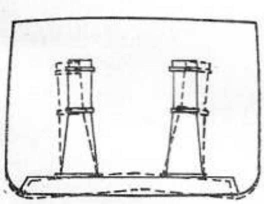


FIG. 5
ROCKING MOTION
OF TWO ENGINES

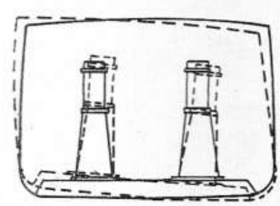
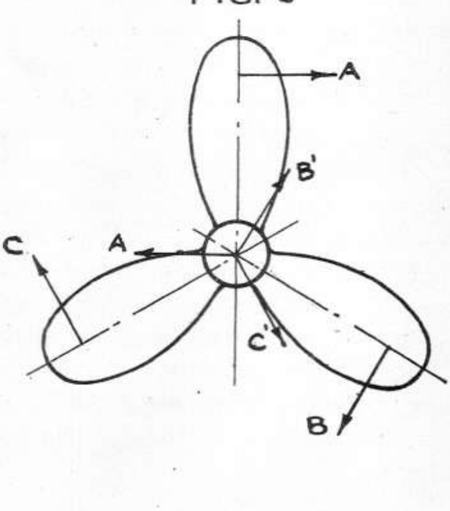


FIG. 6



Life Saving from Ships in Distress at Sea.

(Paper No. 502)

By Prof. ***Otto Lienau****, M. J. N. A., M. S. T. G.*

The methods of life-saving of those who are in danger through shipwreck have been completely changed and modernised since the discovery of the radio. Till this discovery was made the shipwrecked were dependent on chance or their own ingenuity to save their lives. Since the radio-discovery, ships equipped which radio-apparatus are capable of calling numerous ships to their assistance, which are certainly more succesful in bringing help with their crews, than the wrecked ship, even with best life equipment and their own limited crew. Today's task of navigation is to keep the sinking ship afloat as long as possible, so that help may reach the ship in time. Can this be accomplished, then there is no doubt that the ships which answer the call, can nearly always render necessary assistance, as long as there is the necessary equipment and a crew exercised in the same.

Should the ship sink before assistance arrives, then there must be most certainly not only "boats for all" but also the guarantee that the boats are properly launched and nautically served. Statistics of sea-accidents show with appealing evidence that success in rescue depends not only on the existence of useful and reliable gear, but more so still on experienced use and resolution.

The question of rescuing from shipwreck in the first place is a nautical one and seamen can judge which saving-gear ought to be applied in each case. For the naval architect however who has to technically design the saving-apparatus, it is necessary that he should know the requirements of navigation and at the same time the extent of means and strength for the rescue-service.

Only when seamen and naval architects work together in confidence, will it be possible to diminish the loss of life at sea.

Now I have set myself the task to describe the methods which according to statistics and nautical reports have surest led to that end under the different practical circumstances. I should like to try to draw a series of conclusions from this, which might be of service to navigation and ship-building and ease the solution of the serious and difficult question of rescue.

The methods of fifty years ago for fire-extinguishing on land have been completely changed through the telephone and telegraph and taken out of the hands of single individuals and transferred to fire-brigades, which attack the problem with quite different means and greater success; so the service of rescue from shipwreck will also take a similar turn since the radio enables the S.O.S. call to be heard at almost any distance. Today the question is how far is it possible to organize systematically the institution of self-help and under circumstances completed by a better equipped and trained professional organisation. In fire fighting the professional is able to see the possibilities and the necessities; quicker and clearer, the same now applies to rescue from shipwreck.

Evidence of the accuracy of this view is given in the statistics of sea accidents, out of which one distinctly can see the superiority of professional handling, and on the other hand there is the fact that on the coasts numerous trained organisations have been formed in the "Societies for Life Saving" and the "Salvage Companies", which have perfect equipments and trained staffs at their disposition.

There are two principal questions to be answered:

Firstly: To what extent can assistance at sea be given by professional organisations?

Secondly: How far can it be made possible without endangering economy to provide every ship in the world coming to aid with suitable equipment and a trained staff?

Regarding the necessity to introduce radio and also the demand from the radio-technic for cheap and small apparatus on small ships there will be no two opinions between marine engineer and ship-owner; likewise there will be no disagreement about the claim to keep a wrecked ship floating as long as possible. The expense and economy of navigation must here draw the necessary line. In answering the two questions it is necessary to gain a knowledge of statistics and nautical reports.

In the following Table A statistics of accidents at sea of the world fleet of merchant ships in the years 1927, 1928 show the accidents according to ship-sizes and kinds. The sacrifice demands about 500 ships yearly on which about 10000 people are ship-wrecked. This means about 1% of mercantile fleet tonnage and almost as much for the passengers and crews.

According to the class of accidents to ships there are about

40 % from stranding
40 % through storm
9 % through fire
11 % never heard of again.

According to tonnage vessels of 100-300 tons and of 1000-2000 tons are strikingly in the majority. An explanation may be found in the fact, that ships of 100-300 and 1000-2000 tons are very numerous, but otherwise they are less capable of resistance to storm because of their smaller size and smaller engine power.

It is almost impossible to obtain exact statistics of lost life, but numerous reports show that even in successful rescues many lives are lost.

Accidents at sea which arise from stranding, about 40%, are almost entirely free from loss of life because of the life saving-service on the coasts.

This service which is provided by private and public organisations makes use of the seaman or of almost equally good voluntary assistance from men who are especially trained for the work of rescue.

The equipment for rescue and the organisations are nearly the same in almost all civilised countries. At certain distances which vary with the extent of the equipment, stations are erected on dangerous coasts, which are in possession of either rocket apparatus, or life-saving boats with or without motors, and mostly there is a well-ordered coast guard service.

This coast guard service which has been maintained for many years by experienced seamen has developed standard lifeboats and saving-gear. Experience on the coast ought to be made use of in the general life-saving-service at sea, because in the surf the dangers are higher than in the open sea. Methods of life saving shall be described.

The rocket-apparatus is used to form a connection from land or from the life-boat with the ship by means of a line shot over, through which a whip-line is drawn as well a hawser. This apparatus is also for use on the high sea.

The hawser on which a hose-buoy is hanging may only be used from the land to a ship that is hove to. The buoy for the use of a single person is drawn over a pully to and fro by means of the whip-line. On the high sea this method can not be applied. There it is only possible to work with a loose hawser with which the fixed buoy or a loop may be moved to and fro by hand between the ships. This demands a well-trained crew.

It is also possible to use a " dinghy " with a boats-rope or the hawser, which is fixed in a similar way, as long it is possible to get near the ship in a small boat, while the big life-boat remains at a safe distance. Specially useful for this purpose are india-rubber boats.

The U. S. A. Coast Guards use for this purpose closed boats like swimbodies for 5-7 persons, which are made of tin-sheets and which are moved from ship to land by means of a rope.

The life-boat for oars is transported on land and moved through the water (Fig. 1) on wheels. At it must pass through the surf, so it always shows the swung form of surf-boats and requires for use a skilful and well-trained crew who not only understand the use of oar or sail but also the handling of the drift-anchor and oil-bags. If possible it goes to lee of the damaged ship to pick up those who jump into the water. To improve safety these life-boats are supplied with water-sight compartments; they are often self-emptying and carry on the outside big fenders of ropes. The handling of the boat by means of oars guarantees a safe journey through the surf by well-trained sailors. This applies also to the open sea. Nowadays boats often have a motor with a screw, through which their radius of action is greatly enlarged. The use of oars is abandoned in this case, as they are not suitable along with the motor.

The big motor-life-boat has developed out of the ordinary life-boat and is useable for both surf and coast. It may safely run on the strand, since its screw lies protected in a tunnel. Owing to its size it may however not be taken on wheels over land, but can only be stationed in safe harbours or bays. The motor is so large, that oars are no longer necessary. Through water-tight sections these boats are unsinkable and when in operation cannot capsize, are self-erecting and self-emptying. They are capable of overcoming any storm in skilful hands. In order to get a connection with the wreck, they have rocket-apparatus, hawsers, lines and a small boat on board, besides shine-lights and signal-apparatus. In shape they are like the best high sea tugs. One of the newest boats measures from $18{,}40 \times 4{,}00 \div 2{,}25$ m and has engines (Diesel-Motors) of 120 H.P. They are guarded on the outside against collision through specially strong fenders (Fig. 2 shows the Dutch motor life-boat "Jnsulinde" at sea).

Such motor life-boats, which have been proved in the heaviest sea are the best saving apparatus even on the open sea. However, they can be carried only on very large ships, as in the case on the newest ships "Europa" and "Bremen" of the "Norddeutscher Lloyd", which carry besides 6 smaller boats, 22 large boats of $11{,}50 \times 4{,}10 \times 1{,}70$ m in measure and which are each capable of carrying 145 persons. Every great ship, which is forced to help in case of need ought to have at least on board one big motor life-boat with a well trained crew.

Finally may be named another saving-apparatus, in development representing an improved type of boat: the inflatable india-rubber boat, which requires very little room, is quickly filled with air by a hand-pump or a bomb, and through its elasticity and light weight is totally indifferent to concussions. Subdivision of some of the sections makes the boat unsinkable, even when damaged. It can be used everywhere with or without a crew where it is impossible to keep in vicinity of the wreck with boats and where a rescue by swimming—as for women and children—may not be risked (See Fig. 10)

Following table B shows the extent of the "Coast-life-saving service" in some of the European countries. The radius of the stations is different according to circumstances between coasts.

Table B. Kind and sphere of action of the coast-life-saving in some European countries.

Country	Dangerous coast	Stations with				One station per km	One Motor-boat per km	Persons saved yearly
		rockets only	rowing boats	motor boats	together			
Denmark	400 km	13	50	11	74	5	40	61
Germany	1000 „	15	83	19	117	8,5	53	59
England	1500 „	15	135	70	220	7	21	572
Holland	360 „	20	29	7	56	6,7	52	87
France	1200 „	72	84	24	180	6,7	50	140
Sweden	1200 „	5	20	8	33	36	140	30
					Totals 680			Totals 949

With these 680 stations during the year 1927-1928 nearly 950 persons were rescued from the sea.

An increase of the life-saving service has been attained in some countries, for example Sweden, by instituting a steady service of patrols with bigger life-boats along the less populated coasts.

Fig. 3 shows such a strip on which a Motor life boat guards a district of about 40 sea miles in length. As the boats possess radio, they are capable of keeping a steady connection with the home stations or with ships in distress.

Here are taken the first steps for professional organisations with regular safety-service.

In the same directions run the enterprises of private Tug Salvage Companies, which have undertaken to establish besides regular tug-services in river-mouths and coast-districts with great traffic, a salvage and life-saving-service from certain harbour stations with large and technically greatly improved saving equipments. The radius of these boats, which are built like strong tugs with all the apparatus for salvage and towing, divers equipment etc. is very considerable. One of the largest, the Hamburg tug "Seefalke" (Fig. 4) has a radius of action of 500 sea-miles. The splendid success of these vessels was demonstrated in the saving of the "Pommern."

In Germany there exist two larger companies with together about 54 tugs and salvage-steamers.

In the U.S.A. the whole life-saving service is effectuated by public and professional organisations. Since 1915 the former private life-saving companies have united with the state organised marine-coast-guard. Thereby a great number of large life-boats, salvage-tug and motor-boats have been put at the disposal of the marine-life-service, who maintain a steady guard and patrol-service.

It is only a question of the cost of enlarging the radius of the ships which are used for stationary or patrol-service. Their radius of action between Europe and America ought to be at least 2000 sea miles, which is 5 times as much as at present with the largest saving tug.

The excellent suitability of such small boats for life-saving purposes on the high sea is shown in the following incident.

I. Rescue of the Crew of the "Pommern". (Fig. 5)

On November 24, 1928, the German three-masted bark "Pommern" was in the English Channel about 25 miles west of the northern coast of France, when she got into difficulties through springing a leak and losing her masts. She was being driving helplessly towards the French coast in a west-northwesterly gale (9-10 Beaufort). An S.O.S. immediately brought 4 large ships of 10-19000 tons register to the vicinity. At the same time, the 400-ton salvage tug "Heros", lying up at the Plymouth salvage station and the 570-ton motor salvage tug "Seefalke" from Penzance, put out to sea. The "Seefalke" was the first to arrive on the scene, but she lost her tow-rope in an attempt to take the "Pommern" in tow. The smaller "Heros" also tried, first of all, to get her tow-rope out, but was unsuccessful. The rocket apparatus also failed after one successful attempt.

As the " Pommern " was making water quickly and being rapidly driven on the coast, the captain asked for the crew to be saved. The " Heros " then reversed with her stern to the wind and manoeuvred astern the "Pommern", when it was possible to get a $2\frac{1}{2}''$ tow-line 640 ft. long, and having a noose in the middle. to the distressed vessel. One at a time, a number of the crew of the " Pommern " secured themselves in the noose and jumped overboard, and were hauled aboard the " Heros ". The large steamers that had come up in response to the S.O.S. ranged themselves in staggered formation on the weather side in front of the " Heros " to protect her, and calmed the sea by pouring oil in it. Of these ships, the British steamer " Lancastria " attempted to launch a boat, which, however, could make no progress and was dashed to pieces whilst being taken on board again. The " Heros " constantly steaming astern, remained on the weather side of the stern of the " Pommern ", and had already taken half of her crew aboard when a heavy breaker cast her broadside on to the sea. To prevent her being cast on to the wreck, the " Heros " had to steam full speed ahead round the stern of the " Pommern " and slip her cable. The two-line was again recovered, however, and finally, the full complement of the " Pommern " were got safely on board the " Heros ", the captain being the last to leave after 8 hours' strenuous work by the " Heros ". The entire crew of the " Pommern " were then safely landed in Plymouth; but the ship itself was lost.

This report reveals the following important facts. It is possible, even in fairly heavy seas, to get a life-line from a small and mobile high seas tug to a distressed vessel, and this method was adopted in the present case for special reasons. It was possible to get off the 79 men by means of a simple tow-line only because all the suitable lines and apparatus were ready on board the " Heros " and the whole of her crew were thoroughly trained seamen By skilfully manoeuvring the small steam tug astern and properly paying out the tow-line at the right moment, the line was prevented from tautening up too quickly when a heavy sea broke.

It was of course impossible to make the line fast at the ends. It is therefore impossible on the high seas to use girt-lines firmly secured to the mast or funnel and a breeches buoy, as the lines break due to the movements of the ships. An attempt by the large ships to effect a rescue by life-boats was unsuccessful, as the boat never reached the wreck, and if it had, it would probably have been battered to pieces.

The method adopted of rescuing by a life-line was the only one possible in a case of this kind, apart, of course, from the very doubtful method of the men of the " Pommern " jumping into the sea and being picked up one by one. This very trying work called for superhuman efforts from the crew of the " Heros ". The small crew of 12 men, wet through and cold all the time, worked a solid 36 hours at this rescue work without a murmur, and afterwards shared their small quarters with 79 men wet through to the skin. The large ships that came up in response to the S.O.S. helped considerably to calm the sea by pouring oil on and giving the rescue ship the leeward side, but even then they were not able to render such effective help as the small salvage vessel which, due to its carrying proper equipment (searchlights, rocket apparatus, life-lines, etc.) and its spendidly trained crew, was superior to the large ships. It is clear, then, that auxiliary ships stationed in the vicinity of much frequented and dangerous fairways, are able not only to afford valuable assistance, but in many cases are the only practicable and successful life-savers. Modern methods of taking bearings by radio enable the locality of the ship to be accurately ascertained, so that such ships can also go, on patrol duty, as they do in Sweden and in the United States of America, for instance, on sparsely populated coasts.

These observations abundantly demonstrate the important part which well-equipped salvage ships can play in life-saving operations at sea.

With regard to the question of whether it is possible to utilize the ordinary large sea-going ships in a similar way for rendering effectual assistance in life-saving and to equip them suitably for the purpose can best be answered in the light of a few practical examples, and we propose, then, to deal briefly with the " Laristan " and the " Herrenwijk " cases.

II. Salving the "Laristan". (Fig. 6)

On the evening of January 25, 1926, the large passenger and cargo steamer " Bremen " (10,800 Gross Reg. Tons, length 524 ft.) of the Norddeutscher Lloyd Line, while on the way from New York to Bremen received an S.O.S. call in the Atlantic from the English steamer " Laristan " (3600 Gross Reg. Tons, 345 ft. long, with a crew of 31) and arrived at the scene at 3. a. m. after steaming for 6 hours. She found the " Laristan " lying deep in the water with a heavy list to starboard and rolling badly.

The " Bremen " first of all attempted to establish a connection with the " Laristan " by means of life-buoys attached to lines which were thrown from the stern of the " Bremen ". This possibility was not, however, utilized by the crew of the " Laristan ", due, very likely, to their being very exhausted. On it being noticed that a boat was ready in the tackle on the lee side of the " Laristan ", a line that was got over to the latter ship was made fast to this boat; seven of the shipwrecked crew got into the boat and got it undamaged into the water, when with considerable effort they were hauled under the stern of the " Bremen ", whence they had to be hauled on board with ropes. One man perished during this attempt. Four times was communication established with the wreck with lines. The crew did not venture, however, to trust themselves to this primitive means of saving their lives and remained on board. Despite the fact that large quantities of oil were poured on the water, the " Bremen " found it impossible to get one of their own boats into the water, due to the height of the " Bremen " from the water and the high wind (10-11 Beaufort). Further rescues were therefore impossible, and the " Laristan " sank towards the evening in a heavy rain-squall with the remaining 25 members of the crew aboard her, and was not seen again. Not a soul was seen clinging to the bits of wreckage sighted next morning.

The tragic outcome of the "Laristan" catastrophe demonstrates most poignantly that the whole of the crew might have been saved if better life-saving appliances had been available. From the very start, the very heavy weather prevented the rescue ship using fixed boats, as it was impossible to get them into the water from the high ship. On the other hand it was found possible to get the "Laristan's" life-boat, suspended directly over the water on the leeward-side, safely into the water on the lee side and haul it by a line right to the stern of the "Bremen", despite the heavy seas running. But here again it was impossibe to get the boat on board of the high ship in view of the risk of its being smashed up, and consequently the men had to be hauled aboard singly by ropes. It might also have been possible to get the men off by lifebelts and life-lines carried direct between the two vessels, but the crew of the "Laristan" declined this method probably for fear of being hurled against the side of the ship by the sea. So 25 English seamen perished in the waves. Although the fixed type of the boat of wood or iron will stand up all right on the open sea, it is unsuitable for life saving purposes from shipwreck in heavy weather. What is wanted is an unbreakable flexible (rubber) boat which can be brought up alongside any ship without fear of being smashed. A boat of this type, paid out on a rope from the "Bremen" could have established connection with the disabled ship without fear of being smashed up, and even women and children or exhausted persons would have been able to be conveyed in such a boat.

III. Salving the "Herrenwijk." (Fig. 7)

On the afternoon of 22nd November 1928, the Lübeck steamer "Herrenwijk" (2,500 tons, 315 ft. long) was out in a heavy storm (wind strength 8 Beaufort) in the North Atlantic to the west of Ireland, when she sent out an S. O. S. call after the heavy seas had carried away her superstructure and hatches and she began to sink. The ship was being driven helplessly athwart the sea without her conning bridge and funnel. Of the ships that immediately rushed to her assistance, the "Transylvania" (16900 tons) was first to arrive on the scene, but she was unable either to launch a boat or lend other assistance. At 10 p. m., the Danish passenger ship "Estonia" (6300 Gross Reg. Tons), bound from New York to Copenhagen, arrived, and, as the weather threatened to grow worse, she immediately began rescue operations in the moonlight. With the assistance of the "Transylvania", which took up a position some distance from the "Estonia" on the weather side of the "Herrenwijk" it was possible to launch a boat with 5 men and 1 mate on the leeward side of the "Estonia", this boat fortunately reaching the leeward side of the wreck. Owing to the high seas, it was impossible to approach alongside the wreck without seriously jeopardising the boat; and the crew of the "Herrenwijk" were ordered to put on lifebelts and jump into the water. Seven members did so, and were safely picked up. The remainder of the crew remained on board. After 2½ hours' work, the boat returned to the "Estonia", where it was hoisted on board with great difficulty, bu undamaged. The next morning, the storm had increased so much that it was impossible to put a boat out, as the wind was 10-11 (Beaufort) and the breakers 30-45 ft. high. The " Estonia " therefore remained on the weather side of the sinking ship. About 9.30 a. m. the wreck went down. Of the 13 survivors, who were clinging to wood beams and boards, only 7 men could be saved. These men owed their lives to the skilful way in which the very seaworthy ship was handled, but, primarily, to the pluck of Fourth Officer Bach, who, without a lifebelt and with only a rope round his waist jumped overboard five times and had the survivors hauled on board the rescue ship by hauling tackle. Two men in the top-framing indicated where men were swimming about in the water; and the ship was always manoeuvred so that the survivors were always protected on the lee side.

In the case of the " Herrenwijk " we may see what can be done by a cautious but courageous captain who knows his ship from A to Z and has a skilled and thoroughly trained crew to call upon. That it was possible to get a boat out in the very high sea, to get to the lee of the wreck with this boat and to pick up the 7 seamen swimming about in the water there and bring them safely back, is indeed a remarkable achievement, and one which was possible only by excellently trained members of the crew. It was even possible to get the boat on board again practically undamaged. There is no doubt whatever that everything was done that was possible with a fixed boat, a wind strength of 8 and waves 24 to 32 ft. high. The operations were doubtless assisted by the presence of other large ships, which protected and calmed the site of the catastrophe by pouring water on the sea and giving the wreck and the rescue ship the leeward side. Any further rescues by boat were prevented by the storm increasing in intensity, but even after the disabled vessel sank the very brave action of the Fourth Officer in jumping into the water enabled 7 survivors to be taken, one by one, from the water. Here, too, 13 out of 27 men were lost, and they could very probably have been rescued had further rescue appliances been available.

The cases we have dealt with above show how very difficult it is to rescue a few men from a ship in very stormy weather with high seas running, even when there is plenty of time and there are several ships on the spot to aid in the rescue work.

Where it comes to rescuing, not a small handful of men, but hundreds and even thousands such as are carried nowadays by large passenger liners, the problem is a much more difficult one. Here, too, practical instances alone can teach us what is possible and how we can do still better in the future.

I intend dealing briefly at this point with three typical cases, viz., the " Volturno ", the " Vestris " and the " Titanic " disasters.

IV. Salving the " Volturno ". (Fig. 8)

On 9th October, 1913, the English passenger and cargo steamer " Volturno " (3600 Gross Reg. Tons, 340 ft. long, 690 passengers and crew), bound from the United States, got into difficulties in the North Atlantic in a north-westerly gale (10 Beaufort), fire having broken out on board due to the explosion of chemicals on the fore part of the vessel, and which it was impossible to put out. In response to the S. O. S. 10 large ships hurried to the scene in a few hours, but the high seas running made the work of rescue extremely difficult. First on the spot was the " Carmania " (20,000 Gross Reg. Tons, 650 ft. long, 19 knots). She attempted to launch a boat, but had to give up the idea. At 9. p. m. the electric light on the burning vessel give out and all the machinery stopped. Of the German ships that rushed to the assistance of the distressed vessel, the " Seydlitz " (8,000 Gross Reg. Tons) and somewhat later, the " Grosse Kurfürst " (13,000 tons) were first on the scene. An attempt made by the " Seydlitz " to get a boat out was also unsuccessful, as, although the boat got to the water, it could not stand up in the high seas. The " Grosse Kurfürst " then approached as close as possible to the leeward side of the " Volturno " with a view to launching boats, but she first of all took in tow the last boat of the " Volturno " containing 5 men, which was launched as an experiment, but which sank after the 5 men had been taken aboard. When the whole of the fore portion of the " Volturno " was in flames, two boats from the " Grosse Kurfürst " made an attempt at rescue, despite the darkness and the attempt just previously made. After mattresses had been suspended from the ship's side the boats were got to the water without mishap, and reached the burning " Volturno ". The first boat, which had got up very close to the " Volturno ", was almost capsized by 21 passengers foolishly jumping into it, and it returned with these 21 people aboard. The second boat kept a suitable distance it and took on board 11 people who had jumped into the water. It took 5 hours to return, and the sail had to be hoisted as the crew were almost paid out. The first boat then went alongside the " Volturno " again, but no one else ventured to jump into the sea, and the boat had to return to the ship without having accomplished its purpose. The " Seydlitz " also rescued 17 people with a boat in a similar way. These men had the courage to jump into the water. A frightful panic had broken out among the remaining survivors because, apart from the captain, none of the other officers remained on board of the " Volturno " and this made it impossible to approach the ship with boats. When day broke and the storm had abated, all the ships present again launched boats and rescued the remainder of those still on board. In this way, 541 persons were rescued out of a total of 630 passengers. About 150 people perished either by fire or drowning, or by the loss of two boats that had been put out from the " Volturno " in the morning, and for which the " Carmania " went in search on the first day. Up to the time the last of her passengers, etc. had been got off, the burning ship had kept afloat for 15 hours—sufficient time to carry out the work of rescue.

The case of the " Volturno " shows how dangerous it is in heavy weather to trust a boat to the sea with a large complement of passengers where there are not sufficient trained seamen in the boat who can keep it steady and prevent it being swamped or capsized. Presumably the first two boats which put out from the " Volturno " were lost through this cause. On the other hand, it is quite clear that an ordinary lifeboat is a very useful means of saving life when handled by a careful crew, especially as it enables people swimming about in a heavy sea to be rescued. This probably explains, too, why careful and experienced captains try to wait for the approach of outside rescue ships before ordering their passengers to the boats. It should also be noted that a panic may easily break out where large numbers of passengers are congregated, and that panic leads to still greater disaster. When the 21 passengers jumped into the first boat put out by the " Grosse Kurfürst ", a catastrophe was averted only by the prompt and decisive action of the boats crew in pushing off with the boat. In this particular case, the rescue of the majority of the shipwrecked passengers, etc. was due to the successful operations of the ships that came to the assistance of the " Volturno ". Had the latter ship had to rely upon its own means of rescue, it would have been impossible to save those on board.

These accidents demonstrate even most poignantly the necessity and the importance of a well ordered life saving service at sea.

The " Vestris " and the " Titanic " disaster, on the other hand, constitute two cases in which the ships went down before outside help came. They were faced with the extremely difficult task of getting passengers and crew off the doomed vessel in their own boats.

V. Salving the " Vestris " Passengers and Crew.

The large passenger and cargo steamer " Vestris " (10,500 Gross Reg. Tons, length 500 feet, carrying 129 passengers and 195 crew) bound from New York to Barbados, got into difficulties on 11th November 1928, about 300 English miles off the American coast, due to her cargo shifting and water getting into the vessel. After vain attempts had been made to avert the catastrophe by shifting the cargo, the captain ordered the S. O. S. to be sent out about 9 a. m. on 12th November. Unfortunately, the call was not heard by a ship steaming in the immediate vicinity as the latter was not equipped with radio. The " Vestris " gradually assumed a 30° list, making it a difficult matter to stand on the sloping deck, so that the boats could only be launched at great risk on the high side, which was also the weather side. The available boat capacity was sufficient to take every one off the ship. The sea was certainly rough, but not too rough, to prevent boats keeping up properly if skilfully handled. In this position, the ship sank deeper and deeper. Before one of the numerous ships that rushed to her aid had come on the scene, the captain of the " Vestris " was obliged to decide to abandon the sinking ship, which he did not do until 4 p. m., so that there were probably 6 hours in which to put out the boats. It is well to remember in this connection, however, that part of the time the work of rescue was rendered difficult by severe rain squalls. Numerous casualties occurred when the boats were put out, due to the panic that ensued, to the unequal distribution of the seamen in the boats and the apparent lack of nautical training. Two of the first boats, filled with women and children were either smashed while they were being let down or turned turtle when the rope was slipped, most of their occupants going down. Some of the other boats could not be lowered because the davits stuck, or they were damaged and made water. One boat dropped down because the tackle broke. During these manoeuvres, oil was poured on to the sea to calm it. Altogether, 8 lifeboats and a temporary raft were got away. The majority of the shipwrecked persons were then either picked up from the boats by different ships, or picked from the water. The latter factor shows that not all the boats were able to stand the sea. As darkness was coming on while the last of the boats were being launched and there were still no rescue ships in sight the boats were at the mercy of the waves the whole night before being sighted. Apparently no distress signals were given: an indication that no flares, etc. were provided in the boats, or, if they were, they were useless. Consequently the 8 ships which were searching around the scene of the disaster during the night could only rescue most of the shipwrecked passengers, etc. towards morning. Of the 129 passengers, 60 (or 47%) were saved; and of the crew of 195, 151 men (77%) were saved. Thus a total of 65% of those on board were rescued, while 113 persons (35%) were lost. A number of bodies recovered from the water had shark bites, so that presumably a number of people swimming about in the water had been killed by sharks.

The case of the "Vestris" is of particular significance, inasmuch as it shows how a catastrophe developes where the rescue work has to be performed by the lifeboats of the stricken vessel. It is perhaps regrettable that the captain delayed so long in sending out the S. O. S., and that a ship steaming in the immediate vicinity of the "Vestris" had no radio equipment on board and went on her way without hearing the call. Added to that is the dreadful fact that, due to the heavy list of the ship, it was extremely difficult, if not impossible, to lower boats on the weather side with the result that some of them were smashed up. A temporary raft also broke up. Some of the boat davits would not work, or stuck. Some of the boats appeared to be not watertight, and filled with water when lowered. Only a small portion of the passengers could be placed in the seaworthy boats—only 202 out of 324 people; and the rescue ships, which arrived on the scene too late, could only thoroughly search the site of the disaster, and after in some cases cruising around for 22 hours, save the boats, a raft and a few people swimming about in the water. It is also certain that some of the swimmers were killed by sharks.

Of the measures adopted on the Lloyd steamer "Bremen" for saving life may be mentioned the use of mattresses for making thick bolsters or cushions which were attached from the boat deck to the water line on the side of the ship to prevent the lifeboats being dashed to pieces while being lowered. This precaution shows the real significance of this particular danger. The loss of 113 lives may be attributed then partly to the delay in sending out the S. O. S., and to the fact that the big list which the "Vestris" assumed made it possible to utilise only a part of the lifeboats, and the available boat capacity was insufficient. Where an entire side of a ship, with its boats, is put out of commission, it must make the work of rescue extremely difficult indeed, and this is a point which will need very careful attention in future. The excellently planned arrangements on board the " Bremen " and " Europa " the new fast passenger ships of the Norddeutscher Lloyd are a very big step in the right direction, but are not in any way a final and satisfying solution of the problem. For even where it is possible to lower the boats successfully on the weather side, there is the danger in heavy seas of the boat being lifted out of the water by the next wave and being driven on to the ship. In cases like this where boats are put out of commission, there should be sufficient subsidiary boat space available in a compact form and kept in a suitable place and capable of being got ready quickly. These boats should to be got into the water in every case at the leeside of the ship.

Of the " Titanic " disaster, the general history of which is known, I shall only mention a few details.

The " Titanic ", of 46,000 Gr. R. Ts. one of the biggest Atlantic liners, fulfilled in its life-saving equipment all the requirements of the time. The safeguard was a well perfected system of bulkheads, from which it was expected that the ship, even when severely damaged, would float for a very long time. There was room in the boats for 1176 people, that is 33% of the full complement; there were 16 permanent lifeboats, and 4 semi-collapsible boats. On the voyage which led to the disaster there were altogether 2201 passengers and crew on board, that is about 66% of the full complement of 3500.

Late in the evening of the twelfth of April, 1912, the apparently impossible occurred. The ship ran against an iceberg, and one side of the ship's bottom was torn open for about a third of its length from the bows, so that water rushed into a number of rooms simultaneously. The ship sank completely in 2½ hours. The sea was fairly smooth, and the night was clear but impenetrable. The electric light soon went out, owing to flooding of the engine rooms. Since other ships, in spite of S.O.S. calls, could not arrive quickly enough, the boats had to be launched without external assistance. In spite of the unavoidable panic among the passsengers, 18 boats were floated; they were, however, in the dark, very irregularly loaded, some with only 20 to 30 people instead of 65. Two semi-collapsible boats, which stood on a deck-house, were left swimming when the boat sank, and so saved a number of the shipwrecked. In all, 712 persons were rescued. If all the boats had been utilisied in an orderly way, there would have been 466 more people saved, making 1178. A few saved themselves with life-belts. The survivors were all taken on board next day by rescue ships. Altogether, out of 2201 souls 1489, that is 68%, were drowned.

The question if it would have been possible to save all the passengers in the time available, is answered in that, owing to the small number of life-boats, this could only succeed to the extent mentioned above, but that considerably more could have been saved if the loading of the boats had been carried out in an orderly manner. This did not take place owing to the darkness and the panic, and, in fact, will very rarely do so completely.

In similar cases to-day, with the large modern life-boats in which there is enough room for everyone (' boat room for all ') and with modern Maclachlandavits, as on the newest liners "Europa" and "Bremen" (Fig. 9) everybody would be rescued if it were found possible, on such a huge ship, to bring everyone quickly enough to the boats on the upper deck, and to divide them without disturbance equally among the boats. It is to be feared however, that the engineers employed deep in the hold, the sleeping deck staff, who did not happen to be on watch, and some of the passengers, would not all come up on deck early enough. For them there ought to be enough supplementary boat room.

How great the losses are still, in spite of good life-saving equipment, when there are not enough sailors to man the boats properly, is shown by statistics for the year 1924. There were 11 big disasters, in which, out of 4137 persons only 1994, that is 48%, were rescued, in 33 boats, although 82 life-boats were available. It was just the same when the large 12,000 ton ship "Prinzipessa Mafalda" went down near Balnis on the 25th of October 1927. Although there were about 4½ hours for the rescue work and there was room in the boats for all, 314 people out of 1256 were drowned—that is 25%—partly owing to unequal loading, partly owing to overloading and capsizing of the boats.

The greater part therefore of these losses is nearly always to be attributed to badly proportioned manning or loading of the boats. The "Titanic" disaster, and numerous similar incidents have clearly shown that it is not enough merely to provide "boat room for all," but that there ought also to be sufficient "supplementary boat room", in view of unequal allotment of places and for individual late arrivals. In no circumstances therefore should this allowance be less than the hitherto regulation one of 25%.

These supplementary boats, in opposition to the previous practice (iron or wooden rafts of considerable weight were provided), ought to be of such a type that they can be thrown overboard or lowered from any chosen position on the ship, and by any of the passengers even. They must then be very light and portable, and at the same time offer a high degree of buoyancy, stability and safety. The only rafts which up to now satisfy these demands are the inflatable rubber raft boats, as made for instance by G. Winkler, Berlin, which have already proved themselves hundredfold as pontoon boats for military bridges. Such a boat is represented in Fig. 10.

It will bear 40 people, weighs only 135 kg, and when empty needs only 25 cm height in storing, so that there is room for six such boats, one above the other, where formerly one boat stood. They can thus be packed into a deckhouse on every promenade or upper deck, and do not decrease the deck space. The stability, even when fully inflated, is many times greater than that of lifeboats, and they are guaranteed against sinking by subdivision into cells. They can be inflated by hand in 5-8 minutes, or by means of a small compressed air cylinder in 30 seconds. They can be bound together on the water, and form then a very safe and not to be capsized platform for the survivors until help comes.

The utility of these rubber boats on the open sea has been shown in several sea-plane disasters, in which the fliers kept afloat for hours; and in one case the survivors were thrown high up on a rocky coast without the boat suffering any considerable damage.

After a thorough study of statistics of sea disasters and of seamen's reports, of which only a few can be presented here as examples, one can draw the following conclusions for the further development of life-saving at sea:

1) Stranding accidents (occurring to the extent of about 40%) are, as hitherto to be dealt with, first and foremost by the organization of the coast life-saving stations, whose present modern equipment must be further developed and propagated.

2) Accidents occurring in frequented sea-routes near land, in so far as they can not be dealt with from coast stations, should be handled by ships definitely provided for the purpose, such as those of the Tug and Salvage Companies, which have at their disposal excellent men and proved equipment. Full attention should be paid to the further development of this organization, with, possibly, state help.

3) All other accidents on the open sea, which, now as ever, depend on their own efforts or help from other ships should, especially in the range of the main sea-routes, be dealt with in the first place by an extended general world life-saving service provided with perfected life-saving equipment.

   a) The foundation of such a general world life-saving service to all ships must remain the old sailor's custom "Each one for the others". This must become the governing maxim of every sailor in the world. In order to bring to perfection this service of help dedicated to mankind, a thorough navigational education is to be demanded from every sailor, which is best obtained by means of voyage with sailing ships and which includes the use of all life-saving appliances.

   b) Radio-telegraphy should be installed even on small ships. Suitable cheap apparatus is to be developed.

   c) Maintenance of the buoyancy of ships for as long as possible increases to a large extent the chances of saving every life after an accident. This should be an object for extended and thorough research.

   d) Since the precept " boat room for all " has so far never guaranteed the rescue of everyone on board, a sufficient amount of supplementary boat room, not less than 25% of the main boat room is to be demanded. This must be so arranged that it is indifferent to concussions, that it can be easily launched from either side of the ship, and even by non-sailors.

   e) Every ship, in order to promote the general world life-saving service at sea, should carry at least one life-boat specially equipped for this service, and with a chosen crew as well as the most necessary rescue appliances, such as rocket apparatus, whip-line, hawsers, light signals, driving anchor, oil bags, and rubber raft boats. The life-boat should be made as unsinkable, as self-righting, and as self-emptying as possible, and be well protected by especially strong fenders. On big ships it should be a motor-boat.

f) The davits should be so chosen and constructed that the boats, in case of emergency, can be quickly and safely lowered, even on the weather side and in the case of a considerable list.

g) In cases where there is no outside help to be obtained, the disembarkation of the passengers is less dangerous when effected by life-boats properly manned by sailors, which then discharge the people to rafts, rather than with a large number of boats insufficiently manned. Thus, the goal to be aimed at is the concentration of the sailors schooled in navigation in a few well commanded boats. This point is worthy of particular attention.

I should be very happy if my remarks led to a stimulation of thorough-going discussions and to the co-operation of all sailors and ship-building engineers with the ship-owners of all lands, the great goal of which would be to cut down to an ever increasing extent the loss of human life and of goods resulting from distress at sea.

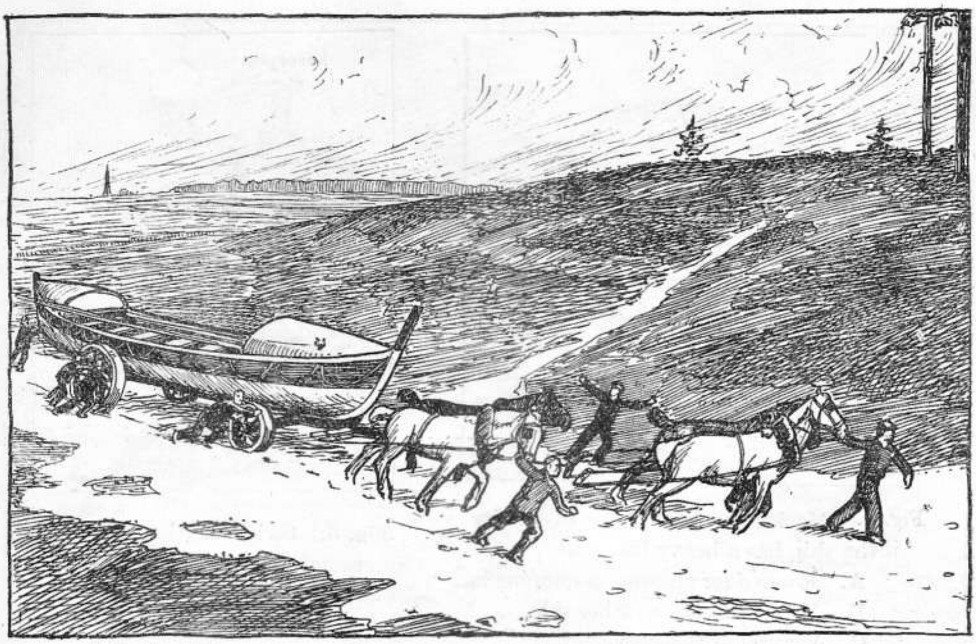


Fig. 1. Life saving service on the coast with surf boats carried on wheels into the water.

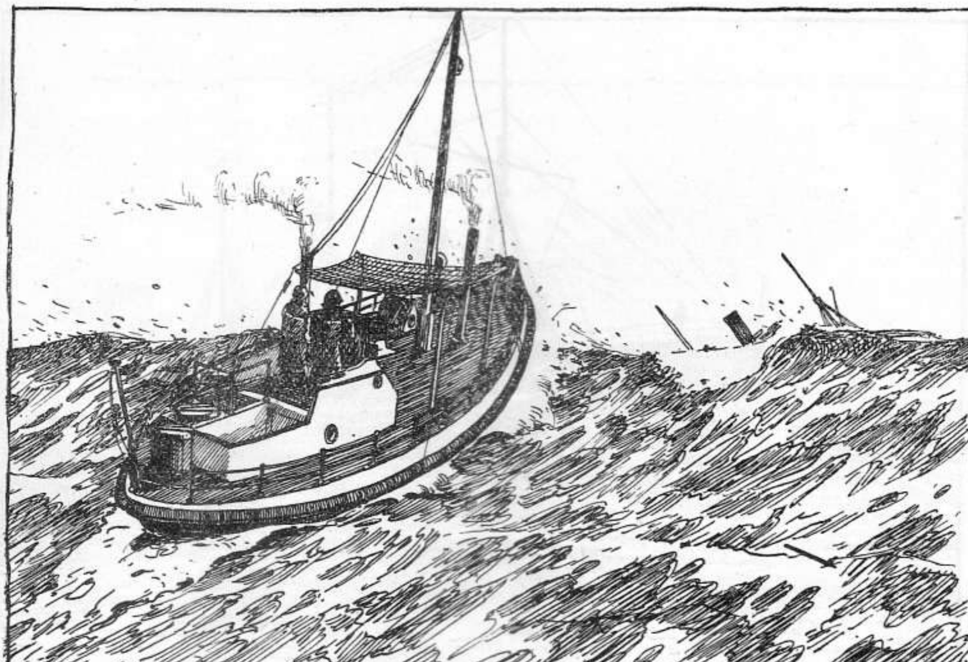


Fig. 2. Large motor life-boat "Jnsulinde" (length 60 ft), Holland.

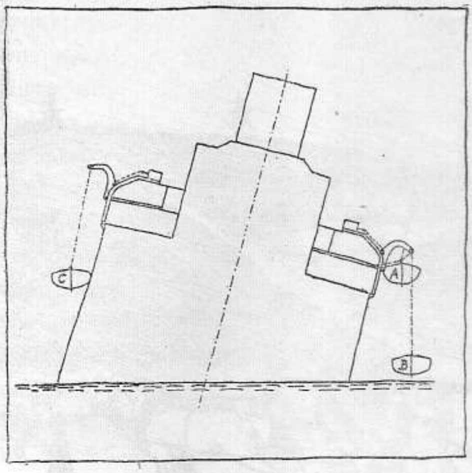


Fig. 9. Maclachlan boat davits in activity if the ship has a heavy list,

A. lowered for passengers stepping in,
B. lowered at the weather side,
C. lowered at the leeside.

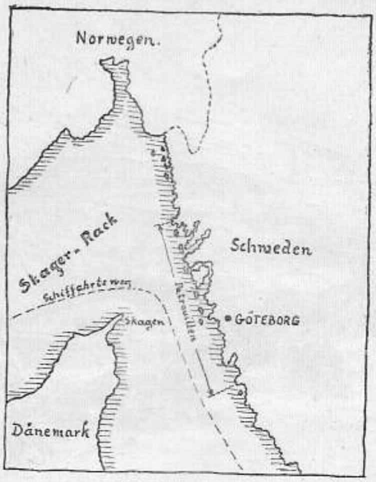


Fig. 3. Stripe of the Swedish western coast, where a steady service of two patrol motor-boats is entertained.

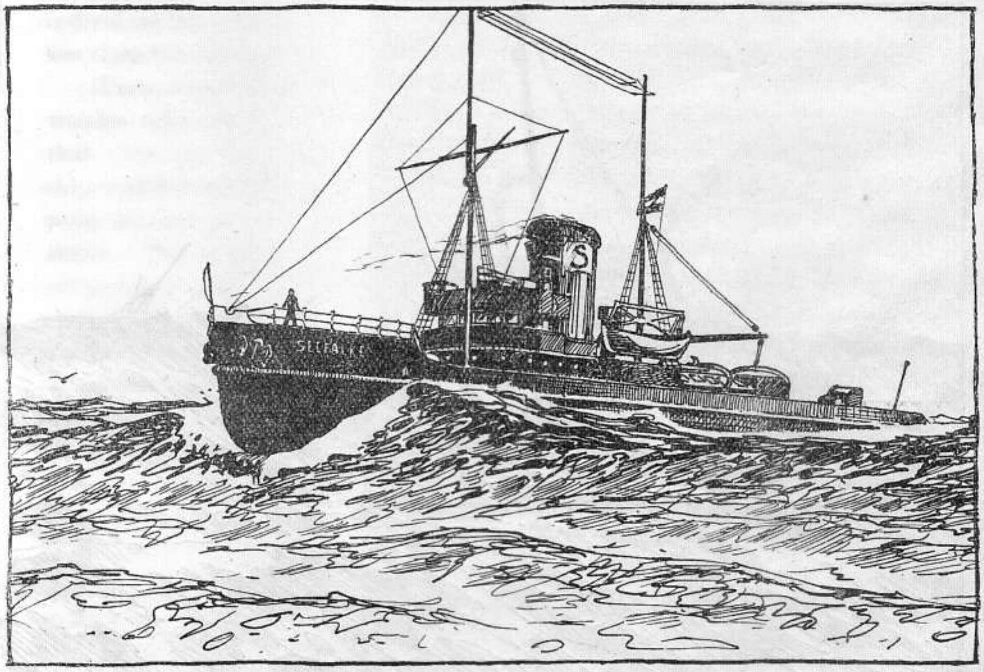


Fig. 4. 570 ton motor salvage tug " Seefalke " (Germany).

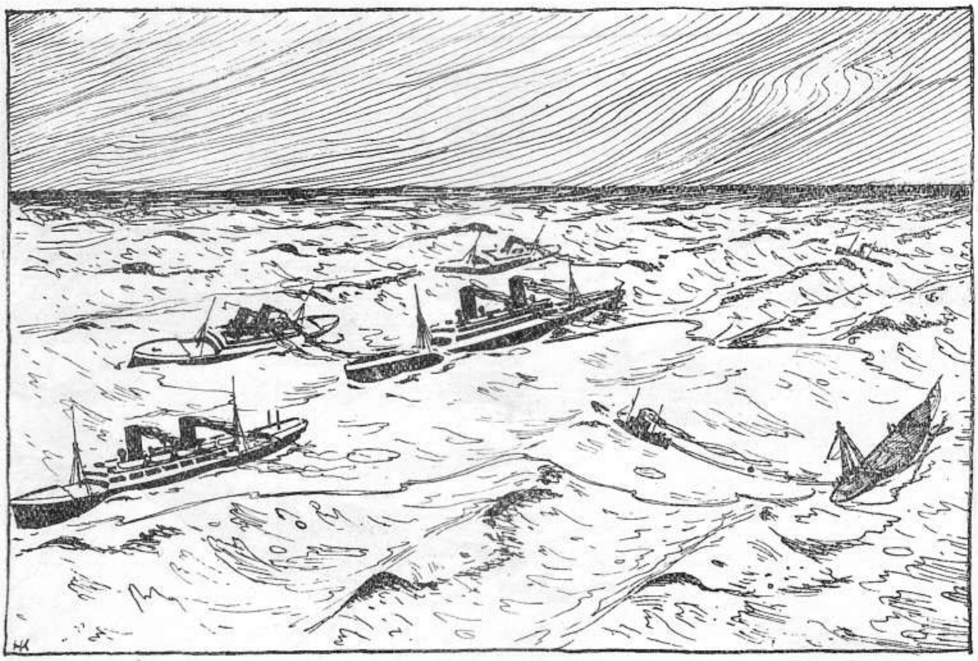


Fig. 5. Salvage of the crew of the "Pommern" by the salvage tug "Heros."

Fig. 6. Salvage of the crew of the "Laristan" by the passenger steamer "Bremen."

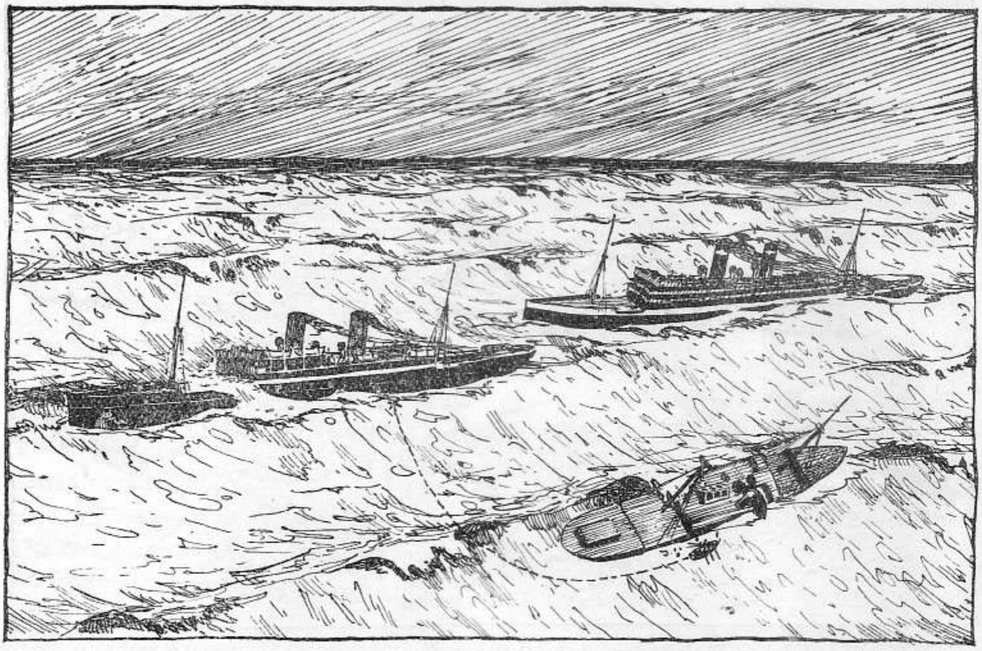


Fig. 7. Salvage of the crew of the "Herrenwijk" by the passenger steamer "Estonia."

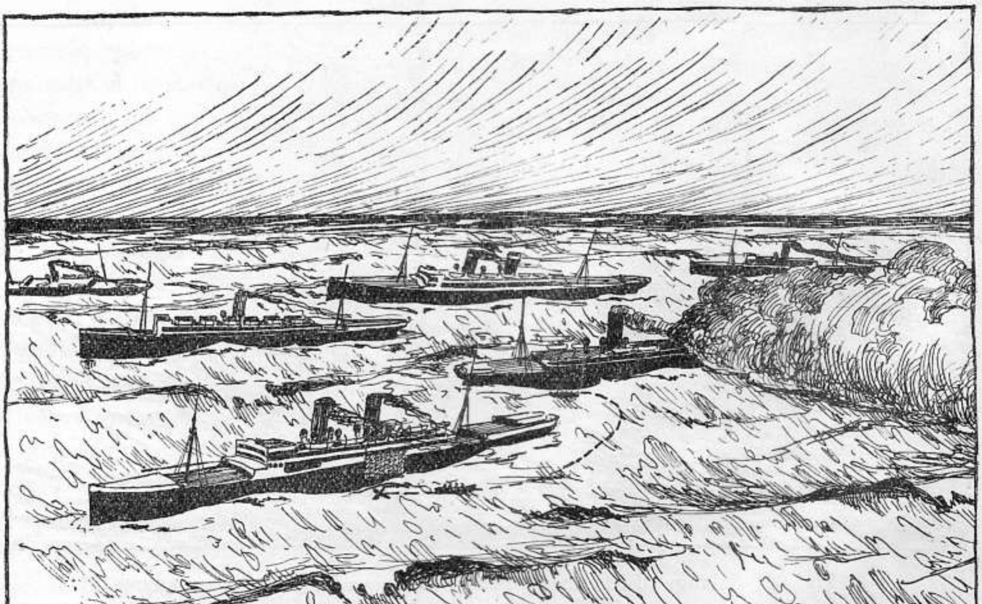
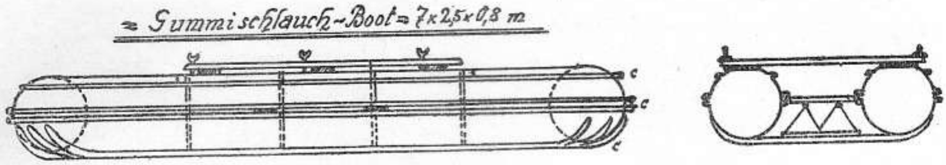


Fig. 8. Saving the shipwrecked passengers and crew of the "Volturno" by the boats of rescue ships.



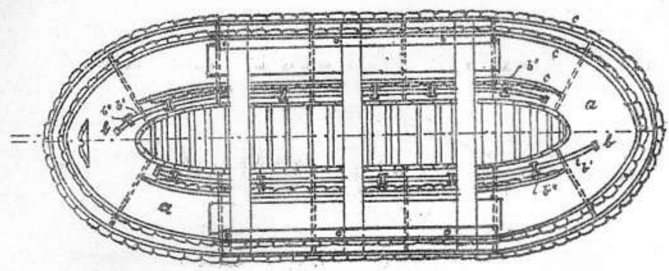


Fig. 10. Inflatable india-rubber boat. Capacity 40 persons at a draught of one ft, weight 270 lbs, cubic measure or buoyancy tube 250 cbft; port if boat is filled with water 14400 lbs.—a buoyancy tube, b ai inlet, b′ hose for distribute the air in the compartments, b″ air valve, c grasp lines.

Table A.

Number and kind of accidents at sea in 1927 and 28.

1927

Br.R.Ts.	Number of ships stranded	Ships foundered through storm	Ships destroyed by fire	Distress unknown.	Loss of ships, total.
100	13	33	10	3	71
200	24	39	6	11	80
300	18	23	4	5	51
400	10	10	2	1	23
500	9	8	3	4	24
600	4	1	2	1	8
700	10	6	2	1	19
800	4	6			10
900	1	4	1	4	10
1000	12	19	2	8	41
2000	23	20	7	9	59
3000	14	9	2	1	26
4000	13	3	3	1	20
5000	10	2	1		13
6000	2	1	1	1	5
7000	3	2		1	8
8000	1				1
9000	1	1			2
10000					
12000		1			1
Totals	180	194	46	52	472

Number of persons shipwrecked 10172

1928

Br.R.Ts.	Number of ships stranded.	Ships foundered through storm.	Ships destroyed by fire.	Distress unknown.	Total loss of ships.
100	30	29	5	11	75
200	22	30	4	11	75
300	19	23	6	4	52
400	11	10	2	2	25
500	9	13	3	3	28
600	8	7			15
700	8	9	1	1	19
800	6	4	2	2	14
900	7	5	1	3	16
1000	15	23	4	6	52
2000	17	14	5	8	44
3000	15	10	3	3	31
4000	11	2		1	19
5000	10	3	2		15
6000	2	5			7
7000	1	3		1	5
8000		1			1
9000		1			1
10000		1	1		2
20000	1				1
Totals	202	193	39	59	493

Number of persons shipwrecked 11050

Are Fast Dependable Sea Schedules Possible?

(Paper No. 607)

By Elmer A. Sperry, Hon. M., Soc. of Naval Architects in Japan.

Recent studies of the performance of ships at sea have virtually established the fact that the difficulties universally encountered, where great loss in speed accompanied by large increase in power demand is experienced in heavy weather, are due to the violent motions of the ship itself rather than to the direct effect of weather. From present knowledge it is no longer necessary to accept as inevitable this loss of speed and increased power demand, failure to make schedules, and the universal discomfiture of voyagers, since it is now perfectly feasible to obtain measurably fair-weather performance under the most severe weather conditions.

From the earliest times when brave men ventured on the high seas in ships, their toils and tribulations have been uniformly charged to storms at sea. There seems to be quite definite historic evidence that the early Japanese seamen did two remarkable things: first, they built sampans that were sufficiently seaworthy to carry them far afield; and, second, they understood navigation, and had so perfected their astronomical observations as to be able to navigate these ships for many days completely out of sight of land and succeeded in reaching the mainland, the Korean coast, and returning in safety. This seems to have transpired many centuries before the Phœnicians succeeded in reaching the coasts of England. This latter performance did not partake of the hardihood of that of the early Japanese, because the Phœnicians were simply shore-huggers and made the entire trip practically at all times within sight of land.

The difficulties encountered in operating ships were always so closely connected to the disturbed conditions of the sea that the two were looked upon as synonymous. Nothing better illustrates how this actually forms a part of contemporary thought regarding ships' behavior than the observations of J. L. Kent of England, to which he has devoted much time and which have become classic in this field. In this work he has used up-to-date scientific methods and equipment which have contributed immensely to both the accuracy and value of the observations themselves, regardless of the fact that he may have followed an age-long error of attributing them to wrong causes. It was only last year that Mr. Kent read his latest paper, entitled "The Propulsion of Ships Under Different Weather Conditions."

Certain weather conditions are generally understood to superimpose well-known problems upon the propulsion and headway of ships, and within the writer's knowledge the question has never been raised that this situation is not solely chargeable to sea conditions. Mr. Kent has apparently driven the last rivet in this structure by a series of observations in which he directly measured the tail-shaft thrust under conditions where the variations were charged to weather conditions. He shows by curves (see Plate II, Trans. British Inst. Naval Architects, Vol. LXIX, 1927, reproduced as Fig. 1) how, as the weather increases in intensity, the thrust increases and the speed of the ship drops off materially. For instance, a thrust of 10 tons is necessary to propel a ship of 13,500 tons displacement at 8 knots, and a thrust of 31½ tons is necessary at 13.4 knots under conditions that he denominates fine weather. But as the weather changes the speed drops off more and more, accompanied by a very marked change in thrust, so that when the " weather " has forced the ship back to 12 knots the thrust has risen to 40 tons while at 10 knots it is 48 tons, and when the ship is retarded to 8 knots there is no less than a 56-ton thrust, although the propeller revolutions have reduced only a trifling percentage.

From observations on a ship of 23,000 tons displacement Mr. Kent gives another set of curves equally interesting. Here the thrust required to drive the ship at 10 knots is 24 tons. At 12.7 knots it rises to 48 tons, practically following the law of the cube in fine weather, but when the weather changes and gradually grows rougher, the ship falls off in speed while the thrust keeps on increasing. At 11 knots there is a thrust of more than 58 tons and at the original figure of 10 knots the thrust has risen above 60 tons. In the first case the thrust has increased 5.7 times, in the latter case it has increased 2.5 times, all ascribed to increasing intensities of rough sea as compared with fair-weather performance. Mr. Kent states that accompanying these various weather conditions, as was to be expected, the ship itself was obsessed with certain increasing amounts of roll, pitch, and yaw, requiring also, of course, increased helm angles for keeping the ship even on its sinuous course. This observation of the more or less violent motions on the part of the ship itself may be far more significant than heretofore realized.

Now suppose we turn to a perfectly calm sea and take a ship or its model—we have used both—and artificially impart to her roll, pitch, and yaw conforming in point of proper angles measurably to those observed by Mr. Kent, and also apply about the same angles of correcting helm. Quite a startling result is reached. The behavior here follows quite completely in all particulars the curves given in Mr. Kent's papers, which of course are charged to weather. Certainly these results cannot be charged to the weather, for here the sea is calm, and we are therefore forced to the conclusion that the ship's own motions certainly play a paramount part and may constitute the sole cause of the increase in power demand with diminished speed.

These results are easily confirmed by trials that may be performed on any completely stabilized ship, there being upwards of twenty such ships plying the seas at the present time.

Here we turn again to heavy weather conditions, where there has been ample opportunity to observe the following with stabilized ships: first, a positive increase in speed; second, a definite drop in the power demand; and third, the yawing and the use of large helm angles cease and the course is so straightened that the quartermaster practically loses his job the moment any ship rolling and wallowing in a heavy storm is thrown into the stabilized condition. This applies to all sizes of ships with which we have experience.

Another significant fact is that at the time the ship is stabilized in a storm, the greatest gain in speed and saving in power come at that heading with reference to the wind and waves that ordinarily would cause the greatest disturbance of the ship, the greatest angle of roll and consequent yawing, together with pitch.

Fig. 2 shows the relative size of the stabilizer and a naval airplane carrier. This ship, with its large G. M., has been assumed to have about the same characteristics as a merchant marine ship of 16,000 tons displacement. The gyro is about $\frac{1}{8}$ larger than necessary to completely stabilize this ship, the standard performance being at $\frac{7}{8}$ speed of the gyro wheel.

Fig. 3 shows the gyro of Fig. 2 in the ship while undergoing its acceptance tests, showing its two precessional positions while in the act of stabilizing the ship. A little pilot gyro in the background is controlling this artificial precession.

Fig. 4 shows the "gyro gallery" on the Savarona, with the gyro compass at the right and the top of the gyro stabilizer at the left. On her trials this ship possessed no less than four gyros performing different major functions.

Five gyro stabilizer equipments are under order and being constructed at this writing.

There is abundant evidence that a ship is only and truly natural when she is upright. If she is made to roll, pitch and yaw from any cause as we have seen in fair weather or foul, she makes us pay dearly for this maltreatment. The disintegrating strains which she suffers is only one of the prices we are paying. The more immediately observable price is the greater demand for power accompanied by less result in speed and headway. We have always thought of this as being due to the weather and sea conditions alone, but it is not the sea. It is the ship, demonstrated completely, we maintain, from the fact that when we impart to her artificially fair-weather attitudes and freedom from motion in a stormy sea, she delivers practically perfect fair-weather performance in all details. And per contra, when we artificially give her bad weather action and motions in perfectly calm weather, she faithfully hands us bad weather performance with all the extra power demand and lessening of speed. This has received abundant proof and can be demonstrated by anyone who is possessed of the necessary simple equipment. What other conclusion are we to draw, therefore, but that the movements of the ship itself in the various planes are the true source of her difficulty in power and speed, and that all we have to do to remove this serious handicap to ocean schedules is to remove the cause?

The retardation of a ship seems to be proportional to the sum total of all the motions that obsess her about the horizontal axis, i.e., roll and pitch, but varies somewhat faster than the square of the angles about the vertical axis because of the twofold factors brought in by yawing; viz., not only "the bad angle of attack," i.e., the broadside component during headway of the enormous mass of the ship, but the large helm angles constantly applied in the almost vain effort at its correction.

Abaft and Beam Winds.

It is fortunate that pitch does not take place about the ship's center, but about a point much farther aft. Some 21-ft. models tested in waves pitch about an axis three-fourths of the ship's length back of the bow. This is significant because all abaft winds are found to have but little effect in producing pitch. Their points of attack are so near the axis of pitch that the forces have only slight effect on the ship owing to the slight leverage present. These abaft seas when quartering do, however, produce the most serious rolling of the ship of any seas encountered. This also invariably produces heavy yawing and there is always more or less pitching accompanying the heavy rolling. But all of the rolling, much of the pitch, and all the yaw are found to yield completely to the stabilizer. There is then no impedence to the full and even slightly super-speed performance of the ship, and under these conditions she requires only normal power. The observed tendency of a stabilized ship to make speed in heavy weather under these conditions is probably explained by the aft component of the wind.

Forward wind components cause the greatest pitching because of the great leverage present, due to the remotely located pitch axis, but there is one favorable factor, i.e., the larger mass moments involved and the tendency toward lower pitch periods owing to this remote axis. Now we know that the most favorable condition for producing pitch obtains when the seas synchronize with this period and, conversely, when the wave period becomes so short or frequent that the great mass can no longer follow these quick impulses. It will be seen at once that so great a mass oscillating as a long horizontal pendulum, cannot partake of quick motion, so if the wave increments reach it at too high frequency it simply refuses to oscillate or to be much disturbed, merely picking up a displacement, the surface of which lies at the mean wave depth and does little pitching.

Now let us consider characteristic storm waves, ocean waves produced by the wind. Authorities agree that the important ones vary in length from 300 ft. to 500 ft., the latter occurring, say, only once or twice a year. Before going into a discussion of this phase—speed of progression and its relation to the speed of the ship itself, let us look into the character of the disturbance that we know as waves and find the depth from valley to crest or, what is more to the point, the energy crest. Also, what is more important still, the actual depth to which the disturbance extends and the character and amount of this disturbance at different depths. Ocean waves have been the subject of much study and out of the accumulation of observations the facts have now been quite well established. We need only go into the aspects that affect the problem before us. First we know that storm wave crests always have concave sides and are more or less sharp and narrow. The top part of the wave, seen in dotted line to the right in Fig. 5, does not represent any tonnage of water and possesses very little energy. It may therefore be neglected from all practical consideration. The really effective waves, neglecting these frothy protuberances, may best be represented by trochoidal diagrams. This figure gives such a half diagram for a 400-ft. ocean wave produced by a wind that has risen to the proportions of a gale. We do not often meet with waves of this length. We at once observe that where the theoretical cycloidal depth below sea level is about 10 ft. and the peak of the crest is 10 ft. above the true trochoidal line is only 6 ft. below and 4 ft. above. This brings the mean area much more near the mean sea level than the more academic cycloidal diagram.

We have also learned the shallowness of the disturbance of waves. The effect of the wave motion dies out in geometrical proportion to the depth below the surface. Thus, at a depth of 15 ft. below sea level in the case of this 400-ft. wave, the wave disturbance has died out to less than 1 per cent of that at the surface. The area of the vertical line of circles to the right illustrates the amount of this motion and how rapidly it dies out with depth. Thus the broad expanse of the bottom of a large ship even in the heaviest weather finds itself in practically calm water. The ship's largest single area is free from any disturbed wave zone. Not only is there no motion imparted, but the quiet conditions constitute a definite aid in restraining the ship against roll and pitch.

It is thus seen that fair weather conditions actually exist at astonishingly shallow depths in any of the storm waves encountered practically in any season of the year and shallow surface disturbances are the only impulses left that can affect the ship. This explains fully why it is so easy to bring a large ship into a completely stabilized condition and to hold her continuously in this condition.

Let us now consider the pitch-producing power of head winds. Suppose the ship is standing still and the average storm wave period is assumed to be, say, 7½ seconds. This may be considered as about synchronous with the natural pitching period of a ship having about 16 to 18 seconds as its period of roll. Here we have an ideal condition for producing pitch, and heavy pitching ensues forthwith, rising and falling in amplitude as the nodes of true synchronism pass and recur. But as we have seen from the discussion above, the great masses of the ship, acting about a remote center and constituting so long a horizontal pendulum, cannot respond to wave impulses when they arrive too often or become too high in frequency.

As we run head on into waves their mean virtual period referred to the ship begins to shorten, growing less and less as the speed of the ship increases. Referring to the large storm wave with its, say, 7½-second period, suppose the speed of these waves to be taken at, say, 25 knots and we should run into them head on at the same period. Their period, or the rate at which they successively reach the ship, would be exactly halved. When this speed reaches 32 knots this virtual period has risen to about 2.75 seconds, too rapid for the ship to respond, and pitching virtually ceases. This is especially true with large ships, and was observed with a converted battle cruiser on her recent trials.

Admiral Taylor stated not long ago that the largest ships are not troubled much with pitching. There is usually superimposed on any series of storm waves a low-period swell or following effect. This may cause some show of pitching, but this is moderate and can be completely controlled by the automatic feathering of small bow rudders which fold in flush when not in use. At 30 or more knots these bow rudders are less in area than one-half of one of the sixteen blades of the four propellers and offer a head resistance at full 16-deg. feathering angle of 0.4 of 1 per cent of that of the ship, when not feathering, of $\frac{1}{16}$ of 1 per cent, and, when folded in, practically no head resistance whatever. The feathering is automatically controlled by a little pilot gyro of high precision. (See Fig. 6) As the bow tends to rise it is automatically pulled down by the reaction, and vice versa as it is in the act of going down, accomplished by a reversal of the feathering. There is no power demand except when in use. When feathering to the full extent this demand is insignificant, taking about the same power as one searchlight.

With ships operating at high speed, the question inevitably arises, what lies out ahead? In daylight with high visibility the problem differs in no way from current practice with ships of even higher speed than is here proposed. In cases of necessity at night it is possible to use searchlights with completely light-tight shutters so that the dark-accustomed eye of the observer is not blinded while the intense arc is being made ready for operation. Then with the searchlight so placed in the extreme forward position that no part of the ship is illuminated by the intense beam, the beam is utilized with extremely satisfactory results. For instance, a letter "A" four feet high on a white target has been repeatedly seen and read at 8000 yards. There are two requisites in effective use of intense light beams of this kind: (1) their orientation in all planes should be controlled from a distance and, (2) the observer should be stationed at as great a divergent angle as possible from the beam alignment. For instance, it is found that an observer standing close to the searchlight is unable to see an airplane five miles away, whereas if he is removed at a distance of only about 30 feet his ability to observer anything coming into the beam is found to be sufficient for all practical purposes.

The beams that are here referred to are those of the super high-intensity variety emanating from no less than a 60 in. reflector. (See Fig. 7, giving an idea of size by the workmen alongside.) An ordinary newspaper has been carried in an airplane up into the beam and has been read at a distance of 50 miles from the source of light. This is a matter of Government record.

For the assistance of the forward lookout in low visibility, supersonic methods may be resorted to. These have now been brought to such perfection that a 40-in. sphere has been detected at about 1000 yards. It will be observed that this subtends an angle of only 0.063 deg., but a derelict floating just at the surface would naturally present a much greater surface than this and would therefore be detected at a correspondingly greater distance. Very definite reflections have been obtained from ships' masses standing at an angle of 45 deg. at a distance of several miles. This device, when further perfected, owing to the fact that the ship's noises or vibrations cannot in any way interfere with it, is destined to become a lookout of value to extremely fast ships.

A third method developed in Russia is the infra-red-ray method by which minute temperature differences may be detected perfectly through fog, striking the receiving station with the exponential factor of four. Thus a minute temperature difference is initially received at sixteen times its actual variant and, furthermore, is easily amplified to any desired point, giving most useful indications, records, and alarms of even comparatively small objects at astonishing distances. The small receiving element operates in the focus of a large accurate reflector which is stabilized and oriented the same as searchlight reflectors shown in Fig. 7. In fact, the lamp, at times of low visibility, can be replaced in a few minutes by this infra-red receptive cell which thus detects unbelievable differences in temperature of a distant object. This method was first proposed for detection of icebergs at great distances, but the sensitivity and dependability have more recently been so increased that it gives evidence of great usefulness as a lookout or detector of distant objects, as they are all found to have slight temperature differences from both the surrounding air and sea. For instance, the exact azimuth of distant cities is easily picked up. Thus practical lookouts, eliminating the human element, acting as safeguards for making fast time at sea, are at hand.

Ship-to-Shore Service.

Transportation by air and sea are closely related and modern enterprise has sought to combine them. The shore-to-ship service at the one end and the ship-to-shore service at the other utilize the newer art in connection with fast water transportation, the object being to clip off no less than one day or more at each end of the voyage.

Such a proposed service is in all probability an interim one as undoubtedly within a reasonable lapse of years, airship service will be running on most of the present-day liner routes.

The able and constructive criticism by Mr. A. C. F. Henderson of a fine paper by Sir Eustace d'Eyncourt, K.C.B., D.Sc., F.R.S., etc., on "A Proposed Aircraft Carrying Mail Steamer," presented before the Sixty-fourth Session of the British Institution of Naval Architects, indicates many limitations which necessarily surround such a proposition. He states: The present design of the North Atlantic liner is the result of such gradual improvement from time to time as has been found possible with increasing facilities and additional experience. Those who have been responsible for the building and running of steamers in this trade find it a difficult economic proposition under the most favorable conditions, and it cannot be expected that the revolutionary design of the type put forward should command the approbation of those who by long training are best capable of recognizing its commercial possibilities."

All this emphasizes the point that only the most insignificant departure from the liners of today should for a moment be considered.

The proposition of carrying a large number of airplanes on board, and also providing an extensive top-side landing deck even larger than the ship itself seems entirely untenable and unnecessary. The presence of this overshadowing upper deck would disarrange practically all of the present-day practice as to accommodations, standardized through years of experience. The space required for stowing airplanes and the extra personnel would encroach on the ship's accommodations to an extent that would not be tolerated either by the owners or the traveling public.

Let us consider a ship-to-shore service of a character that will save a calendar day or a little more for both de luxe passengers and mail. Chamberlin established the practicability of launching an airplane from a large ship by using a very long runway without a catapult. A method using a catapult is described in this paper.

The Catapult.

When reviewed in detail, the 25–, 50–, and even 100-passenger planes are surprisingly small and may be easily handled with simple equipment,—especially small when compared with the liner as well as the importance of their contribution. They would naturally be fast amphibians, otherwise not special in any way. An important point is that they would be delivered overboard after a run of only 35 ft. by a simple mechanical catapult from a perfectly plain platform, no track or truck being necessary.

The writer many years ago was impressed by the enormous, though perfectly safe, quantity of stored energy in a rotor of even a small gyro stabilizer. The extremely small rotor employed in the well-known flywheel engine starter for airplane engines illustrates this point. The flywheel catapult was built twelve years ago and after being perfected, has never failed to deliver airplanes into the air at full flying speed at the end of a short runway. Only a very small (2 to 4) horsepower is required, which quickly brings the wheel up to the speed selected for this work, which is only about one-third normal in current gyro practice.

The frustrum drum shown in Fig. 8 is idle on the shaft of the flywheel and coupled to it by a liberally designed asbestos-lined cone clutch located at the large end of the reel, the flywheel being shown inside. The cable clamp is seen at the small end of the reel, which passes out horizontally around a forward roller on the platform and back to the airplane, where it is divided in the form of a halter and attached to two widely separated downwardly and rearwardly protruding pins firmly attached to the airplane structure. The wheel being at speed, the lever A throws the cone to the left and the clutch, starting to engage, starts the airplane very gently. The reel quickly comes up to the flywheel speed, whereupon slipping ceases at the clutch and the airplane is accelerated as the cable climbs the frustrum. The gradient of this frustrum is based on the algebraic product of the deceleration of the flywheel and the acceleration of the plane. At its larger end the airplane is delivered at flying speed at the moment the mass has been decelerated to the extent of only one-fourth its speed, thus giving up to the airplane half its total stored energy. To give some idea of the small power requirement, the little pulley may be observed to the left, connected with the flywheel shaft. With this apparatus the full size J-N Curtiss planes were repeatedly delivered at flying speed. Over 100 successful take-offs were made with this machine prior to the armistice. From the above it is evident that the catapult to handle even the largest planes is but a diminutive affair, weighing only a fraction as much and far cheaper to build than the catapults heretofore employed on our naval vessels. No truck is employed.

The launching platform itself is comparatively narrow, broadened forward. The forward part is folded back and the narrow rearward projection supporting the tail of the plane is folded forward. When the platform is not in use, the whole is hardly observable in its low, contracted forward position. The acceleration on this short take-off is somewhat less than the Navy standard of 2.3 the acceleration due to gravity, and has no ill effect on either the plane or passengers, whose seats are provided with cushioned backs. The start is almost imperceptible and the acceleration uniform. The platform is placed low over the bow and the forward end is swung up into the vertical with suitable canvas cover, effectually shielding the plane during the first days of the trip. The plane is mounted intact, but leashed firmly to the platform and ship. The fact that the ship is stabilized simplifies the matter of properly securing the plane. When ready for the take-off, the shield is removed and the forward end of the platform is lowered into place, the ship is brought into the wind, and all leashes are removed except the automatic stays and chocks which instantly disappear, but only at the will of the pilot. The plane is therefore only freed when the engines are all running to the pilot's perfect satisfaction and the wind, as indicated by a wind vane, is exactly parallel with the fuselage.

The catapult is located out of the way under the platform (see diagram, Fig. 9), and also constitutes a part of the automatic operation under the control of the pilot who, by the simple act of throwing a switch or pushing a button in his cockpit when all elements on his plane are functioning to his entire satisfaction, clears the plane and simultaneously energizes the catapult reel or drum which delivers the plane at flying speed at the end of the short runway. The accelerating cable and halters let go of the plane at the end of the runway automatically by a simple loop slipping off from a straight rearward and downward projecting pin on each rear corner of the plane (see insert, Fig. 9). The cables are caught, retarded and brought to rest before the ends reach the drum by grips located under the platform which seize and retard both it and the drum before the halter reaches the chocks. This is all performed automatically as the cables release from the plane. Both forward and rear parts of the platform are then folded.

After the take-off the wireless gives constant record of the progress of the plane to shore. The plane then searches for the radio beam, on which it flies to the nearby landing.

Take-off from Stabilized Ship.

It will be the work of comparatively few minutes to hoist and install the second plane from the forward deck up onto the launching platform, so several planes may be launched. These are stowed in "temporary" protecting housing above the standard forward deck fittings. This operation of hoisting and installing succeeding planes on the platform goes forward in a space temporarily shielded from the wind. The planes are put aboard and secured on their racks the last thing before sailing. Their presence does not interfere with the forward deck fittings.

The freedom from motion of the stabilized ship is of the greatest importance in securing safety and complete smoothness in the take-off, as well as in manipulating the machines and installing them on the platform. This has reached practical demonstration as one airplane carrier is now completely stabilized, securing the extremely valuable contributions originally expected.

Immunity from Fire Hazard.

The greatest flight hazard that still remains is the fire hazard. This will be eliminated by using Diesel fuel in place of the present volatile and inflammable aviation gasoline. The elimination of gasoline on board will insure safety on the ship. The Diesel fuel may be taken from the ship's fuel tanks and costs only about one-tenth as much as gasoline. Changing over to Diesel fuel secures four important advantages :

1) Complete immunity from the fire hazard.
2) Much lower first cost.
3) Only about two-thirds of the fuel is employed by weight.
4) Volume for volume, the fuel is found to have in excess of 30 per cent greater heating value.

The Diesel engine (see Fig. 10) will eventually be even lighter than the present gasolinee ngine, especially when considered with its fuel for a reasonably long flight. The mean effective pressure may be brought to a higher point and the weight of fuel required to be carried for a given mileage will be only about two-thirds what is now required with aviation gasoline. It may be noted, in passing, that the underlying principle of supercharging in this special Diesel engine is the final step in attaining this extremely light weight and high duty performance, thus adapting the Diesel to air transportation. These engines will be 1000-hp. units and the larger p anes will be multi-engined.

Automatic Flight.

These planes will be flown automatically with great smoothness and comfort, eliminating very largely the uncertainty of the human element, much as is rapidly becoming common practice in the use of such devices as "Metal Mike" on board ships.

Fig. 11 shows a recent form of the automatic flight mechanism, experience with which has been very successful. In 1914 the forerunner of this apparatus was called the "airplane stabilizer" and work has gone forward continuously with it ever since, until it has now reached a very high degree of reliability. As the apparatus has been perfected, one very interesting feature has been noted and that is that the quality of flight is very greatly improved whenever the automatic is thrown in. For instance, in bumpy weather when passengers are likely to be attacked by air sickness, the flight is so smoothed out and improved by the use of the automatic that this condition is greatly relieved.

By the time this service can be inaugurated, wireless beams will be so perfected that they can be used not only from shore, but from the ship itself, both of these beams being capable of proper orientation, directing, and setting. The one on the ship is given great accuracy by using the gyro compass as a base, and also eliminating the serious factor of ship's yaw on the beam, by employing the gyro pilot, or "Metal Mike," especially when utilizing the beam at great radius. The shore beams will do double service: (1) as an aid to navigation of the ship itself through fog and at night; and (2) as a most valuable guide to the planes under the same conditions. The wireless beam emanating from the ship will be of the utmost value on the last leg of the shore-to-ship journey in saving time by guiding the plane directly to the ship. The converse of this is true with all ship-to-shore operations.

Where the ship's course lies adjacent to the shore for quite an appreciable percentage of the time, it is evident that some combinations with rail transportation may be effected, so that the airplane flights will have two advantages: (1) later take-off; and (2) shorter duration of flight to reach the ship, the converse of this being true at the far end.

Shore-to-Ship Mail Service.

Let us now consider fast mail service. The mail of the day following the ship's departure may reach the ship between 500 and 700 miles out, and at the other end, the take-off with the passenger planes will get it to its destination one day ahead. The dating up of the mail on the two ends—date of transmission and date of receipt—would thus be shortened by two out of, say, a four-day service on the Atlantic, giving in effect a dependable two-day mail service between Europe and North America. This may make unnecessary trans-oceanic air service This is such a tremendous gain over anything that now exists as to justify the employment of an extra stamp and extra return to the transportation companies on all such fast trans-oceanic mail.

The remaining factor to be treated now is the detail of shore-to-ship service in the delivery of the next day's mail to the ship. The upper decks and works, masts, antennae, smoke stacks, and bridges of these great ships are too completely occupied already with most highly organized and necessary construction. Moreover, certain places are reserved for outdoor sports. It is impossible, therefore, to think of dropping mail onto the decks or even any points dangerously close. An alternative method must be sought.

Method of Getting Mail Aboard.

Imagine a fleet of mail-carrying amphibians approaching the ship. The wireless beacon has guided them to the fast moving ship and wireless communication has given the accurate time of arrival of this fleet. A most noteworthy contribution of the stabilized ship is now brought into play, namely that of casting a most impressively calm lee.

A standard test of the effectiveness of many installations of the gyro stabilizer for ships was to wait for heavy weather, go to sea, and make great circles a mile or two in radius, so as to take the sea, wind, and waves at every possible angle. During this complete circle, which in some instances is repeated two or three times, a position is reached twice in the course of each circle where the wind and waves are exactly abeam with the ship in the trough of the sea. Here we have always witnessed a phenomenon that is very impressive, that is the wonderful lee-making capacity of a stabilized ship, where the surface of the sea for a considerable area on the two sides of the ship stands out in the greatest possible contrast. On two occasions the ship was slowed down to mere steering headway at these points so as to observe more closely this unusual sight, a perfect lee.

With a stabilized ship an extensive lee of calm water almost instantly becomes available for the landing and protection of the air fleet. Fig. 12 is an illustration of the lee as it would appear from an airplane. It will be noticed that to approach the ship directly broadside into this lee the amphibians will be coming up in the most favorable position possible, namely, into the wind. Reversible propellers will allow them to make a very short landing, so they need not actually alight upon the surface until they are quite close to the ship and therefore well within the smooth area of the extensive lee of the great ship. The planes once down, they are taxied to various points alongside, where their mail is hoisted and swung into side doors by cranes. Or, in extreme instances, the amphibians themselves may be hoisted aboard. Those not remaining will at once make their take-off in the smooth water of the great lee area, being guided landward by the radio beams, and the ship will at once proceed upon her course. The orientation of the land beam will be initially directed from the ship's radio and the fleet of amphibians will rise into it and be directed by the shortest course to their first landing field. The amphibians will find complete ease of maneuverability within this calm lee area.

Alternative Method of Getting Mail Aboard.

Alternative to this it is of course possible to use other methods. The following has been suggested.

It is proposed that the mail be stored in comparatively small water-tight containers, probably made of reinforced rubber and treated with highly actinic aluminum compounds on the exterior and with striping of radium paint for use at night. These containers are inflated so they float and they are leashed together with sturdy, flexible and somewhat elastic coupling of suitable length, so that the function of the shore-to-ship service will be to fly forward of the ship, very close to the water, whereupon these mail container chains are released, as is now standard practice with torpedoes. They would then be fished up in the lee thrown up by the ship as she slows down as is the current practice in taking on and discharging pilots. This greatly facilitates the quick handling of the mail containers on board.

The containers are hoisted aboard without the exposure to danger of any of the personnel. They will be lifted only a comparatively few feet to the proper water-tight side doors in the hull, which are opened on the lee side to receive them. A number of sets of these connected containers would thus come aboard very quickly. Powerful searchlights and modern flood lights will aid this operation when night arrival of the mail is necessary.

It is hoped that the above presentation puts into concrete form some of the elements, functions, and operations which will give the highest degree of dependability in fast combined sea and air service, reducing the time for mails between the continents to a point which will hold the field for many years, and justifying the establishment of the sea and air cycle outlined.

Land transport has for many years felt the pressure of an ever growing public demand for strict maintenance of schedules. In consequence, it has tightened up the specifications for performance on both equipment and personnel, and a much more reliable and satisfactory service has resulted. With the very great increase in volume of travel, this has advanced to such a point that any failure to maintain schedules receives instant and vigorous public disapproval.

The vast number of tourists the world over have made demands that have brought transport on both land and sea to its present high state of reliability, comfort, and luxury of appointments. The traveler has been more tolerant of ocean travel than at the present time is justified, sharing in the commonly accepted tradition that ships must respond to the dictates of Neptune. But let the clever class of traveler once catch the idea that shipping is using this only as an excuse behind which to hide bad or indifferent performance, and he will be heard from forthwith. As a matter of fact, this patience has about reached its logical end. Ocean travel is increasing so rapidly that there is already more than an emphatic suggestion that shipping break away from the ancient tradition and meet an insistent demand not only for fast ocean travel, but absolute dependability as to schedule.

The rapid onward march of science and engineering has achieved marvelous things. Kipling exemplified its achievement when he said:

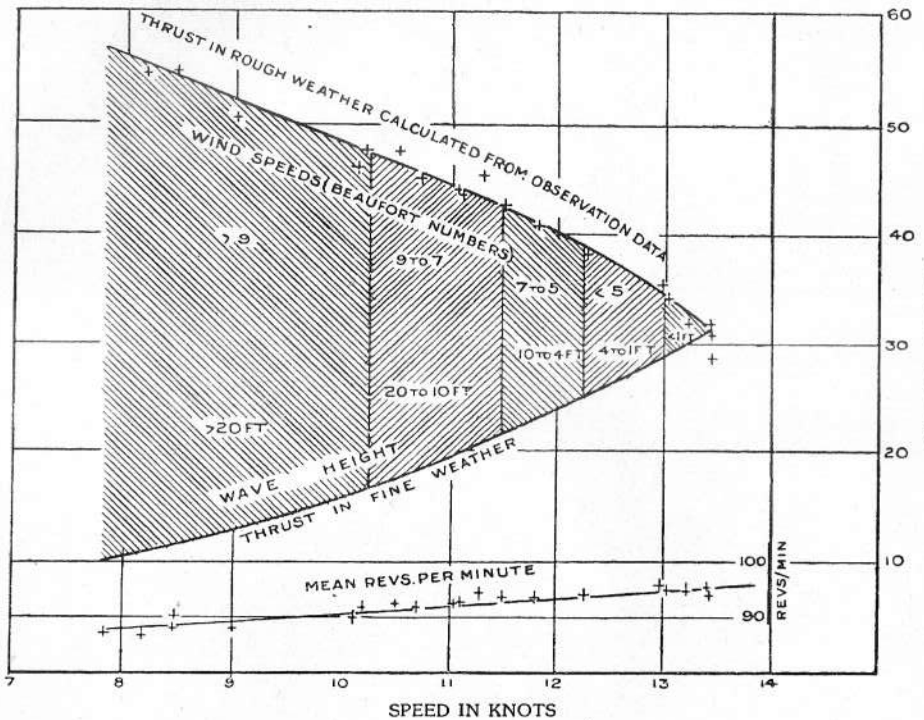
"I am the prophet of the utterly absurd,
The patently impossible and vain,
And when the thing that couldn't has occurred,
Give me time to . . . ."

Within our lifetime we have all witnessed things that seemed so impossible as to border on the absurd, so completely conquered by science and engineering as to become the commonplaces of to-day.

It is a fact that in the art of shipping, science and engineering have advanced to the point where, first, they can offer complete solutions to all the problems involved and can prove that they are by no means as difficult as supposed. Careful observations from aircraft over very stormy seas have corroborated scientific conclusions as to the superficial character of such seas. Second, the problems involved in maintenance of schedules are not at all insuperable, but really very simple. Third, all the equipment involved for guaranteed performance is ready and dependable. Fourth, this equipment also imparts and guarantees for the first time practically perfect comfort to the voyager. Fifth, all of this dovetails into and in a measure is only made completely possible with high speed ships of, say, 30 to 35 knots.

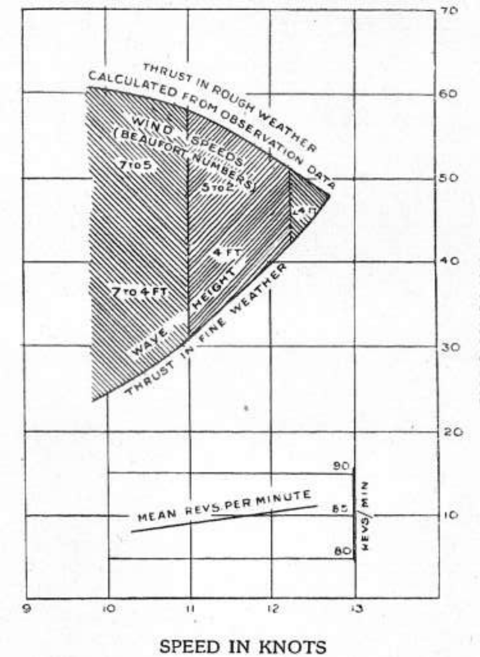
When the public, as stated, are aroused to this situation, which sooner or later they are bound to be, rigorous pressure will promptly be brought to bear upon shipping interests and, as invariably happens, those of greatest vision among the operators will not be slow to seize upon this opportunity and reap the rich rewards that are bound to follow.

Fig. 1—Curves from Mr. Kent's paper.



Increased Thrusts due to Weather on S.S. *Oroya*, based on Observations taken January and February 1926.

Approximate S.H.P. = 5,130; Displacement = 13,500 tons; + = Observations.



Increased Thrusts due to Weather on S.S. *Oropesa*, based on Observations taken March 1926.

Approximate S.H.P. = 6,460; Displacement = 23,000 tons.

(Paper No. 607)

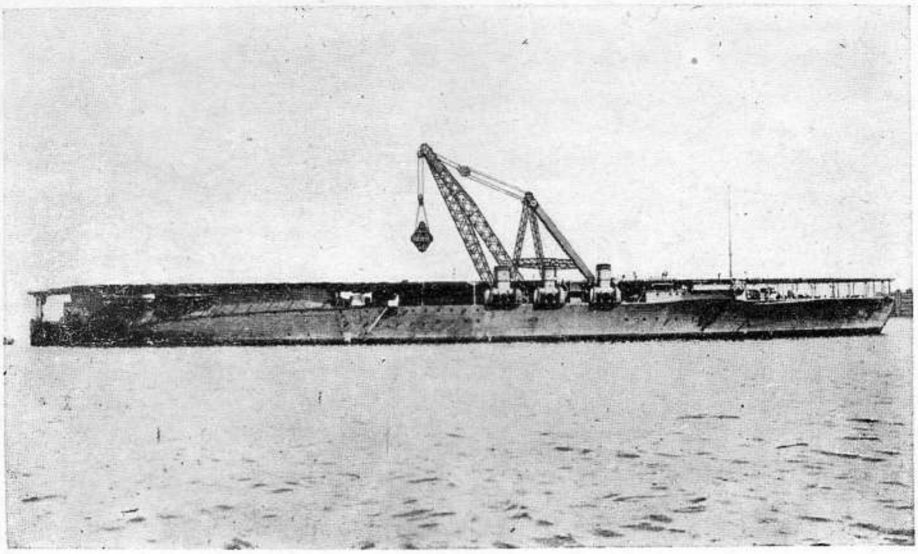


Fig. 2—Showing comparative size of gyro stabilizer of Fig. 3 and the ship. Even this small equipment is too large, as this ship is completely stabilized when the gyro is running at 7/8 normal speed.

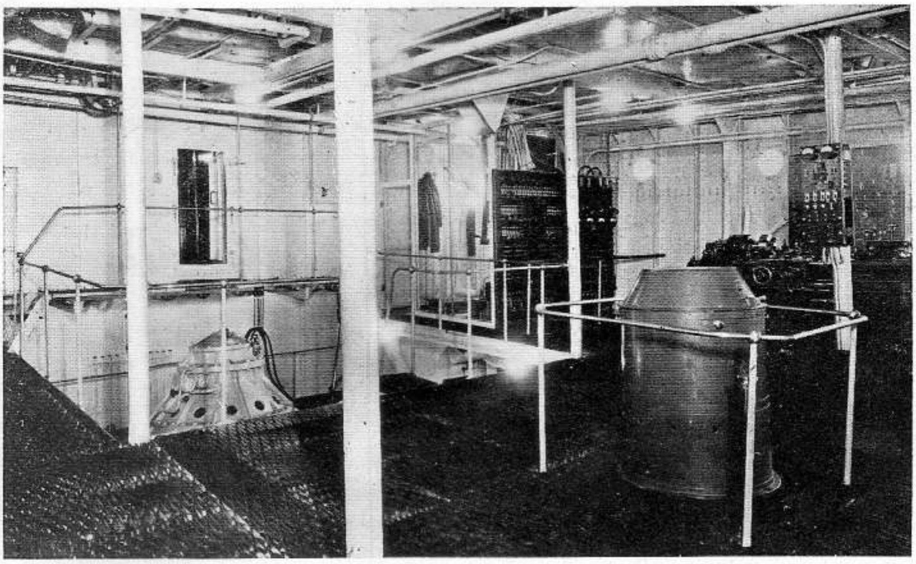


Fig. 4—Gyro gallery on the SAVARONA, showing gyro compass to right and top of gyro stabilizer to left. On her trial trip this yacht had four gyros on board, all performind different major functions.

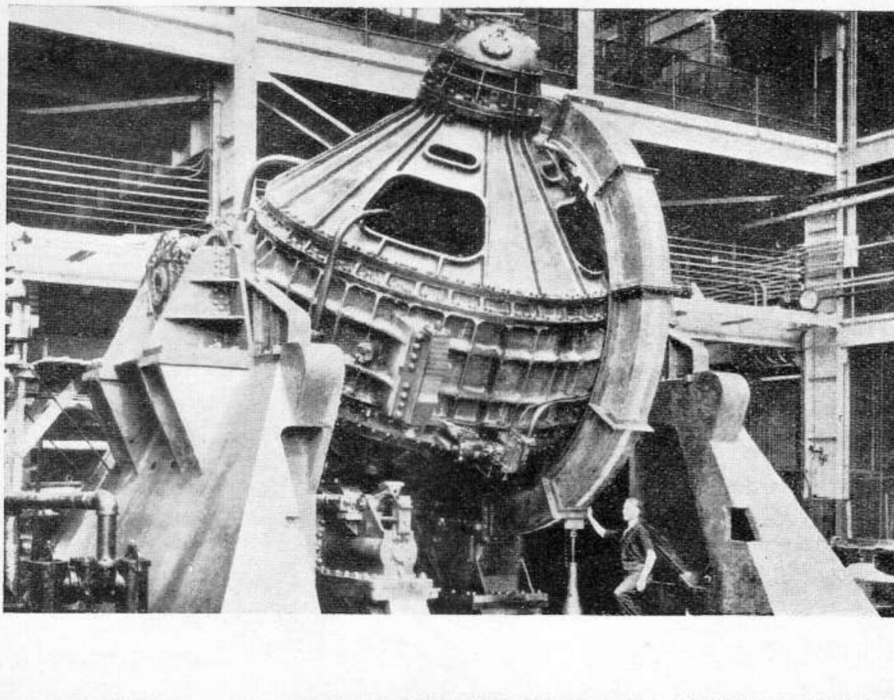


Fig. 3—Largest stabilizer yet installed, in shop test, showing the two extreme positions of precession.

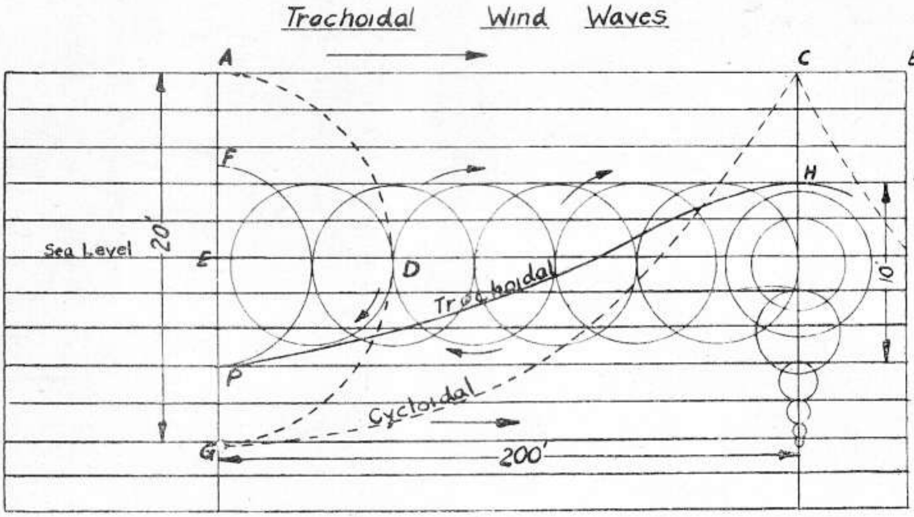


Fig. 5—Note at what shallow depths the sea disturbance indicated by the decreasing size of the circles falls to practically zero, even with this great 400-ft. wave.

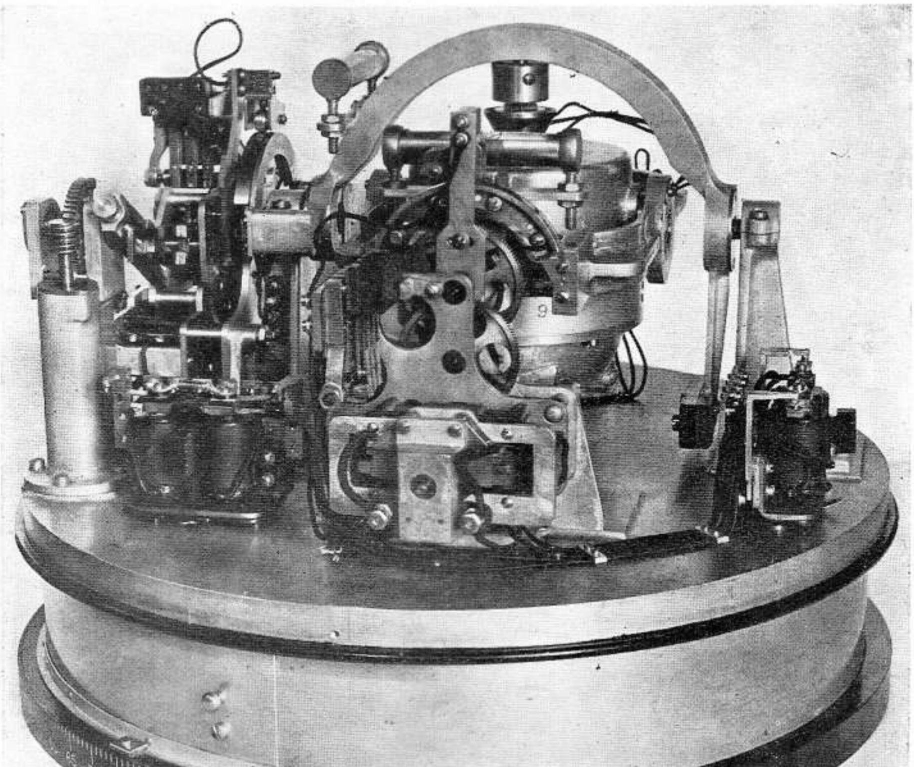


Fig. 6—The pilot gyro which controls feathering. This gyro equipment gives a base line with an accuracy of 2 to 4 minutes of arc.

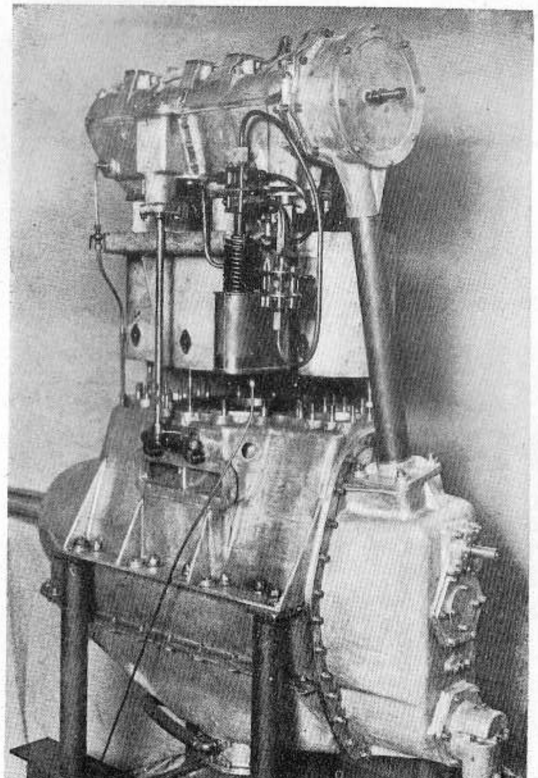


Fig. 10—One form of early Diesel type aviation engine.

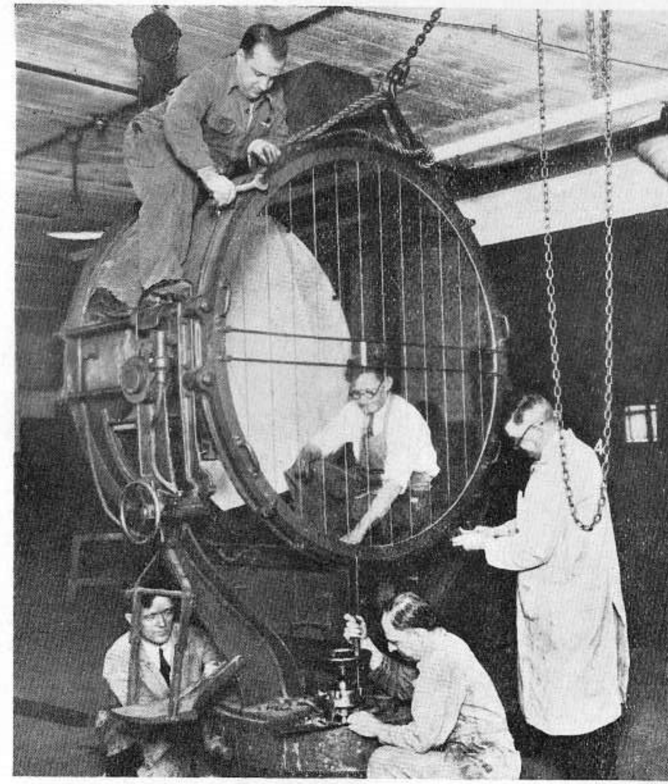


Fig. 7—60-in. searchlight for high intensity beam work or infra-red reception.

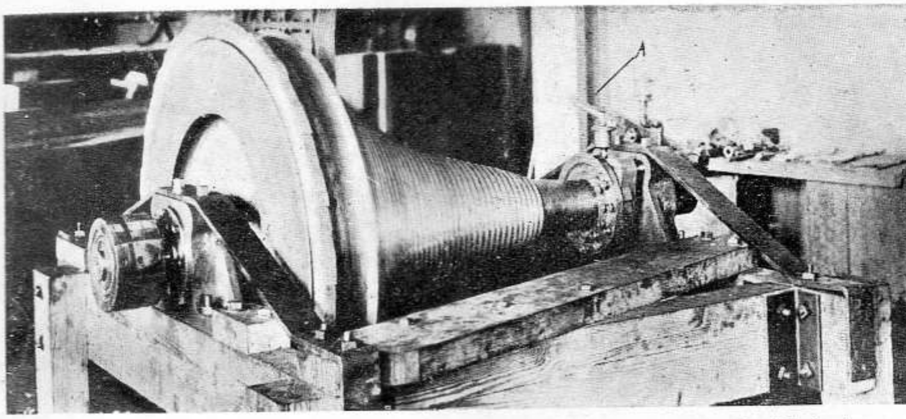


Fig. 8—One of the early forms of hyperbolic frustrum catapult reel with its large cone clutch enclosing a 30-in. steel flywheel with which it can be engaged or disengaged. The stored energy in the wheel at comparatively low speed was found sufficient to deliver Curtiss J-N planes into the air at flying speed at the end of a short runway.

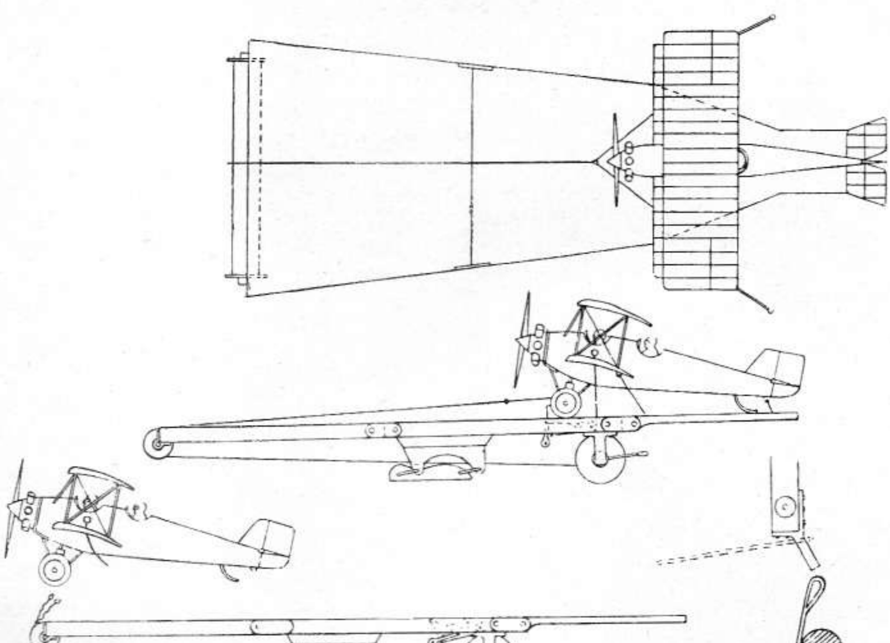


Fig. 9—Diagram showing plan and elevation of simple form of folding take-off platform, with airplane and catapult in place. The sketch to the left shows the automatic release of the halters as the plane takes the air. The trigger, under the control of the pilot, is also seen.

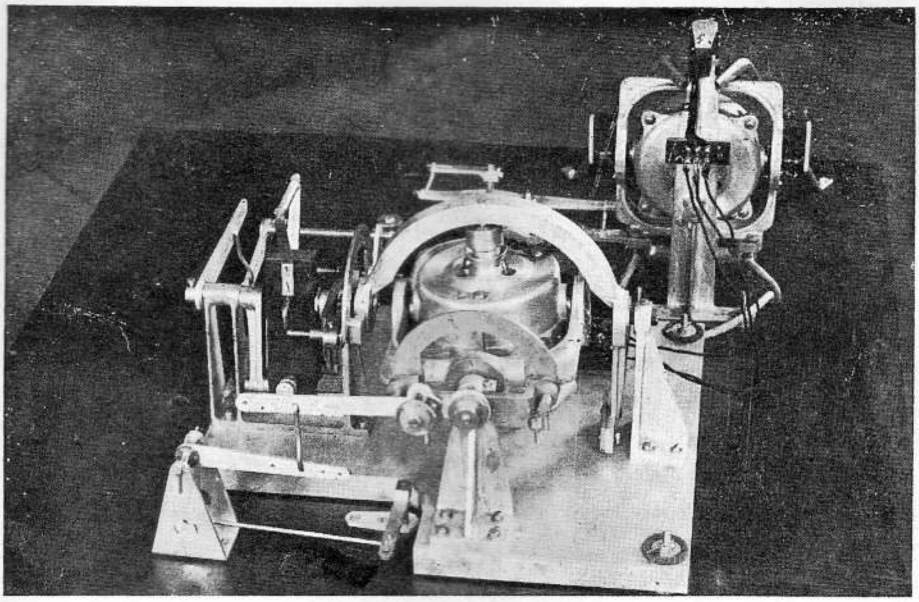
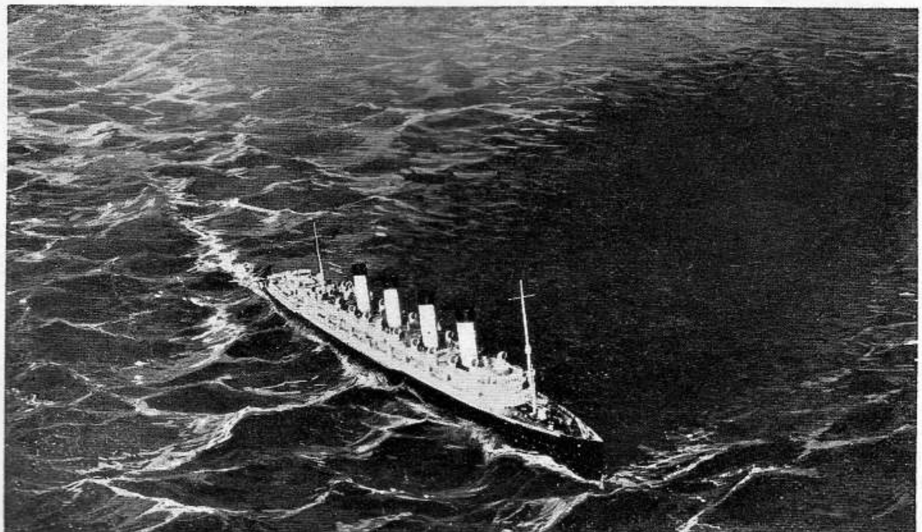


Fig. 11—Gyro pilot for automatic flight of heavy planes.



Airplane view.

Fig. 12—Illustrating the wonderful lee-making capacity of a great stabilized ship where absolute freedom from rolling prevents it from setting up any new series of waves on the lee side. The complete calmness of the water in the lee is not only impressive but extremely useful for landing and protection of the air fleet.

雜　　　　　錄

淺間丸の概要

本船は桑港、「ホノルヽ」、横濱、神戸、上海、香港間の定期航路に從事する日本郵船株式會社の優秀旅客船３隻中の第１船にして、第２船龍田丸と共に三菱造船株式會社長崎造船所に於て建造せられ、第３船秩父丸は横濱船渠株式會社に於て建造中である。

淺間丸は昭和２年９月10日に起工され、昭和３年10月30日に進水し、昭和４年９月引渡を了し、10月初旬處女航海の途に上り、太平洋上に其雄姿を浮べる事となつた。

本船の要目及び構造設備は雜纂第86號（昭和４年５月號）82頁掲載の龍田丸の記事と殆んど全部同様なるを以て、之を參照せらるゝ事として玆には之を省略し、以下本船の機關に就て略述する事とした。

機　　　關

本船の主機關は「ズルツァー」單働２衝式「ディーゼル、エンジン」４臺を裝備し、合計16,000軸馬力を發生する計畫にて、４軸推進式である。

主機關裝備上の特長とも稱すべきは、主機の高さ低くして第２甲板以下に納まつた事であつて、其の爲めに第２甲板上に前後を通ずる大通路を一直線に通す事が出來た計りでなく、同甲板機關室上を船員室炊事室等に利用する事を得、延いて甲板部全體の設計を樂にする事が出來た。

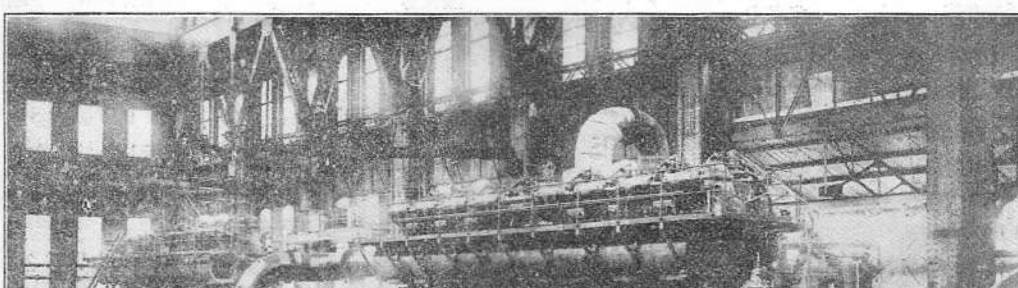
船內の諸補機は全部電動化せられ、蒸氣は只加熱用として小規模に使用せられるのみである。

今機關室內諸機械の主なるものを列擧すれば次の如くである。

後機室諸機械

（1）主　機　關

機關數	４基
型式	「ズルツァー」單働２衝式８ST 68型「ディーゼル、エンジン」
軸馬力（４基合計）	16,000
每分回轉數	120
重要寸法	
筩數（１基に付）	８
筩徑	680粍



淺　間　丸　の　主　機　關

行程	1,000 粍
曲肱軸徑	450 〃
勢車徑	2,200 〃
全長	16,180 〃
臺盤の幅	3,080 〃
最高部の高さ（臺盤の底より）	7,000 〃

主機關直結喞筒

空氣壓縮喞筒	3 段式にして各主機關の前部に 2 臺直結せらる
燃料喞筒	「プランヂヤー」式にして空氣壓縮喞筒の滑頭より作動せらる

(2) 主機附屬器

起動用氣蓄器	高壓14 箇、容量各 800立、壓力 70氣壓 低壓4箇、容量各 8 立方米、壓力32氣壓	
消音器	3 箇	
火の粉除器	1 箇	
淸水冷却器	2 箇	總冷却面積 980 平方米
潤滑油冷却器	2 箇	總冷却面積 80 平方米

(3) 主機用獨立卿筒（全部電動）

名稱	臺數	型式	容量（1 臺に付）
「ターボブロアー」	3	「ピー・ビー・シー」式	363 K.W.
冷却淸水喞筒	3	渦卷式	毎時 440 噸
冷却海水喞筒	4	同上	高壓毎時 80 噸 低壓 〃 500 噸
潤滑油喞筒	4	齒車式	高壓 〃 8立方米 低壓 〃 50立方米

(4) 共 他

名稱	臺數	型式	容量（1 臺に付）	
「ビルヂ」喞筒	4	渦卷式	毎時	150 噸
燃料油「サービス」喞筒	3	齒車式	〃	20 噸
燃料油移動喞筒	2	同上	〃	150 噸
燃料油淸淨機	3	「デラバル」式	〃	900「ガロン」
潤滑油淸淨機	2	「シヤープレス」式	〃	350「ガロン」
「バラスト」喞筒	1	「ピストン」式	〃	250 噸
消火喞筒	1	竪型渦卷式	〃	80-120 噸
「エマージエンシービルヂ」喞筒	1	同上	〃	140 噸
機關室通風機(大)	1	三菱 FOL 120 型	毎分	1,200 立方米
同 (小)	2	同 110 型	〃	850 立方米

前機室諸機械

名稱	臺數	型式	容量（1 臺に付）	
主「ディーゼル」發電機	4	「アレン」4 衝式		408 K. W. 225 V.
補助「ディーゼル」發電機	1	新潟 4 衝式		100 K. W. 225 V.
補助空氣壓縮喞筒（電動）	1	「ズルツアー」4C 34型		380 馬力
同 （電動）	1	「ズルツアー」2 C34型		200 馬力
「エマージエンシー」空氣壓縮喞筒	1	「ズルツアー」MC 12		12 馬力
補助「ディーゼル」冷却水喞筒（電動）	2	渦卷式	毎時	120 噸
潤滑油移動喞筒（電動）	1	齒車式	〃	$6\frac{1}{2}$ 噸
潤滑油淸淨機（電動）	2	「シヤープレス」式	〃	350「ガロン」
雜役喞筒（電動）	1	竪形渦卷式	〃	80-120 噸
衞生喞筒（電動）	3	同 上	〃	120 噸
淸水喞筒（電動）	3	同 上	〃	60 噸
「ビルヂ」喞筒（電動）	1	同 上	〃	30 噸
機關室通風機（電動）	1	三菱 FOL 110 型	毎分	850 立方米
汽罐	2	「スコツチ」型	徑 長	10 呎 10 呎 6 吋
蒸化器	1	「モリソン」式	毎日	50 噸

公 試 運 轉

第1回公試運轉は昭和4年8月1日午前8時、長崎を出港して三重沖に於て施行し、次の如き好成績を收めた。

速力（3 浬標柱間 3 往復）

最高 21.013 節　　平均 20.713 節

馬力（4 軸合計平均）

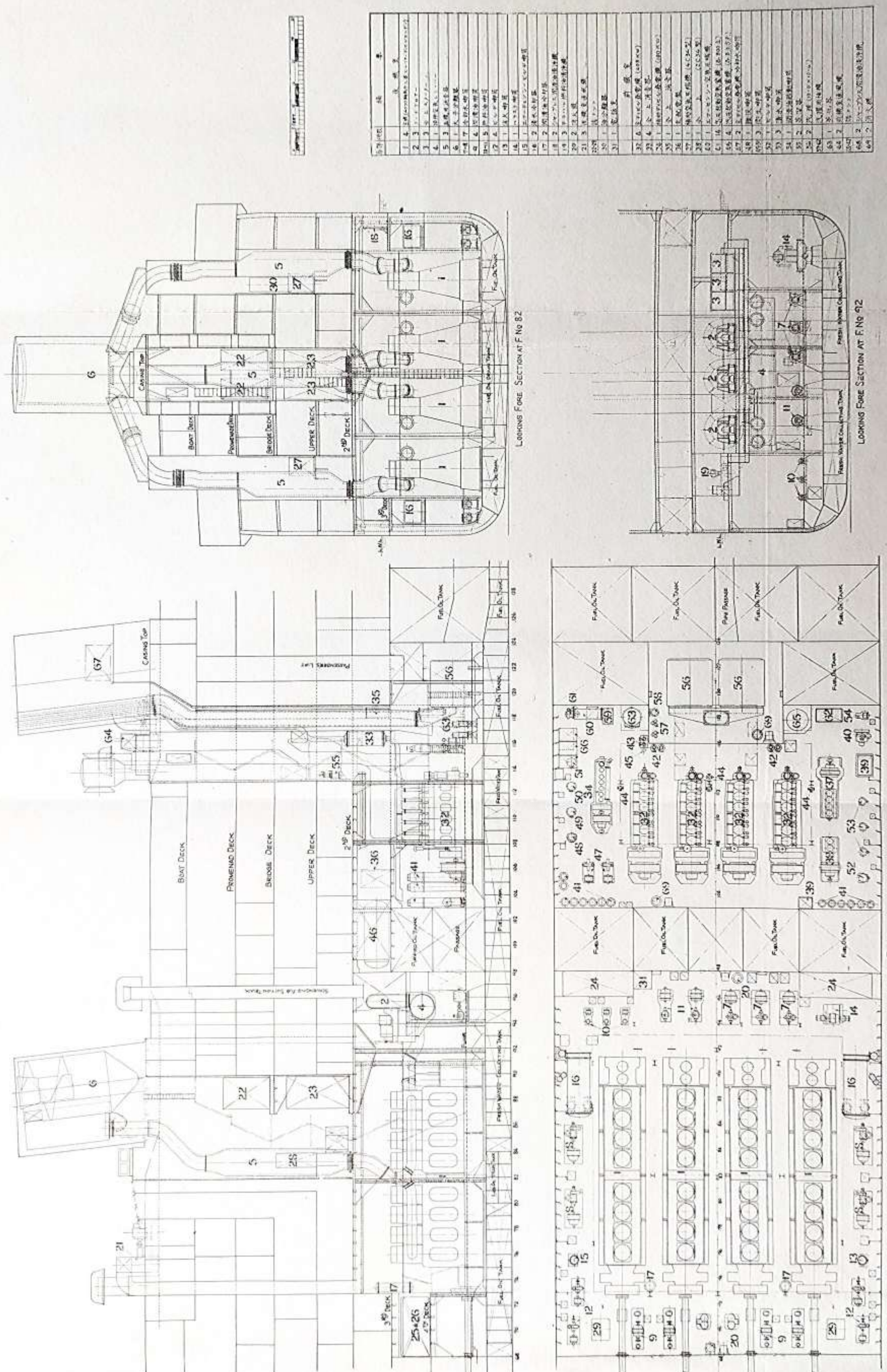
指示馬力 23,206　　軸馬力 18,946

運轉中各部に異狀を生ぜず、殊に此種「モーター」船に、とかく有勝ちの震動が起らなかつたのは、特筆に値する。

其後引續き各種運轉を施行し、全部無事に結了を告げたが、就中燃料消費試驗の成績は次の通りである。

回轉數	軸馬力	消費量 1 時間 1 軸馬力に付
113	13,756	172 gramme

淺間丸機關室全體裝置圖



時　　報

遠洋航路補助法施行細則中改正

昭和四年十月五日遞信省令第四十一號を以て、加速通風機を備ふる船舶は速力試驗に該通風機を使用するを得ること、並に受命航路の起點、終點及び寄港地に支店又は代理店を設置すべき件に就て、明治四十二年十二月の遞信省令第五十六號の該施行細則を改正した。（昭和四年十月五日官報）

本協會の諸會合

役員會

昭和四年九月十二日（木曜日）午後五時三十分より本會事務所に於いて開催次の諸件を審議す。

（一）入會者承認の件、團體員株式會社新潟鐵工所代表者取締役社長笹村吉郎君、正員小副川要作君外十名。

（二）會費滯納者除名の件。

（三）特別會員推選の件。特別會員は成るべく現在數に止め此際は新規推選を見合すこと。

（四）本會雜誌寄贈の件。東京商工會議所よりの照會に對しては團體員に加入を勸誘し大阪毎日新聞社へは寄贈承認の旨回答すること。

（五）萬國工業會議造船、舶用機關論文印刷の件。造船、舶用機關部門に屬する論文數は約30件に及べり。是等の論文中會議に參加する外國人の論文約5件は全文を印刷し其他は總べて「アブストラクト」に止め會議當日迄には印刷出來上る由なり。而して工業會議にて全部の論文を印刷し工業會議會員へ配本する迄には今後少くとも一兩年を要する事と思はる（汎太平洋會議の例に徴し）。殊に本會々員の大多數は工業會議の會員にあらざる故、是等の論文を閲讀するする事能はざる憾みあり。依て此際本會に於て造船、舶用機關の部門に屬する論文全部を印刷し一般會員に配付することにしては如何との提案あり。協議の結果次の範圍内に於て實行する事に決す。但し萬國工業會議の承認を得る事。

（イ）昭和四年秋季講演會は萬國工業會議と時期を同ふする爲め中止す。從て**昭和五年四月發行の會報は刊行せざることゝなる。**

（ロ）前記論文は凡そ3箇月間位に亘り印刷を完了し、毎月1册づゝ會員へ配付する見込なるも、工業會議當日迄に印刷出來上る分丈はなるべく多くの論文を掲載する樣に努力する事。尚該論文集の表題は「**造船協會雜纂**」と稱し號數を追つて刊行するものとす。

（六）工業品規格統一に關し商工省より諮問の件。當協會船用品規格統一調査委員會に附議したる上にて回答する事。

（七）日本工學會定款改正の件。修正案通りにて異議なきこと。

（八）本年度總會、講演會、工場見學並晩餐會の件。萬國工業會議開催の爲め本年の秋季講演會及工場見學は中止する事。總會は昭和4年11月5日學士會館にて開會、閉會後同所に於て晩餐會を催し萬國工業會議の爲め來朝の外國人（造船、舶用機關の專門家）を招待する事。

（九）懸賞應募論文審査委囑の件。

（十）造船協會々報第43號及第44號掲載の論文擬賞の件。

（十一）世界動力會議へ參列の本會代表者の件に付照會の件。末廣會長參列の旨回答する事。

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	理事	濱田彪君
監事	今岡純一郎郎	監事	山本幸男君
評議員	近藤基樹君	評議員	斯波孝四郎君
評議員	山本武藏君	評議員	河上邦彥君
評議員	永村清君	評議員	湊一磨君
會務委員	陰山金四郎君		

編輯委員會

昭和四年九月十六日（月曜日）午後五時三十分より本會事務所に於て開催、板部成雄君、出淵巽君、片山有樹君、加藤熙彥君、菊植鐵三君、小室鉦君、大瀨進君、岡本方行君、牛尾平之助君、山縣昌夫君、横山要三君の各委員より提出の雜纂第九十二號（昭和四年十一月號）掲載豫定記事標題は平賀編輯主任より別項（本號役員會記事中（五）項參照）の通り役員會に於て決議せられし旨説明の後出席各委員の諒解を求め、次の通り申合

の後午後七時三十分散會す。

（一）昭和四年九月十二日開會の役員會に於て本年十月二十九日より開會の萬國工業會議につき造船、舶用機關の部會に於ける「ペーパー」約三十二、三種あり是等の「ペーパー」を萬國工業會議の同意を得て同會議に於て發表後當協會々員一般に配付するを緊要と認め之が具體的成案は編輯主任に一任せられたる旨報告あり。

（二）前記「ペーパー」三十二、三種の内十七種は萬國工業會議にて既に著手せるものにして此種の分は當協會々員全部に相當する部數の交付を受け未著手の分十五、六種を當協會に於て引受著手し前刷を作成し猶ほ前者と合して之を數回に亘り「雜纂」誌上に掲載する事、爲に當分「雜纂」の内容を前記「ペーパー」を以て充す事。但し「時事」及「會員動靜」其の他緊要の記事はなるべく掲載する事。

（三）通常の「雜纂」を停止する回數は出來得るだけ少數に止め度き事。

（四）以上の事由により十月中旬開會の編輯委員例會は特別の通知なき節は休會となす事。

（五）上記雜纂第九十二號（昭和四年十一月號）掲載豫定記事標題として各委員より提出の分は雜纂第九十四號（昭和五年一月號）に繰下げの豫定。

當日出席者

平賀讓君	板部成雄君	出淵巽君
片山有樹君	加藤熈彥君	菊植鐵三君
小室鉅君	横山一君	金井寛三君
鈴木增次郎君		

造船協會試驗水槽成績表現法調査委員會

昭和四年九月二十六日（木曜日）午後五時三十分より本會事務所に於て近藤委員長司會の下に第十七回の會合をなし、次の議案につき審議の上午後九時散會。

出淵委員及び山縣委員より提出の試驗成績表現法の成案を審議し可決其の他雜件の審議。

當日出席委員（順序不同）

委員長	近藤基樹君	常任委員	重光蔵君
	平賀讓君		川原五郎君
	八代準君		出淵巽君
	山縣昌夫君		

船用品規格統一調査會

昭和四年九月二十七日（金曜日）午後五時三十分より本協會事務所に於て越智委員長司會の下に第二十六回委員會の會合をなし次の原案を審議の上午後九時三十分散會。

「シヤツクル」通風筒兩標準案並に商工省工業品統一調査會よりの諮問に係る、小「ネヂ」、麻索、鋼索、梯形「ネヂ」、管「フランヂ」諸規格案に就て討議の結果「シヤツクル」にありては當日提案されたるD型3種の標準原案に對し、通風筒にありてはMIB型4種の標準原案に對し夫々多少の修正を加へ大體原案通り可決。又商工省諮問事項に就ては決定したる協議に基づき意見回答する事。

當日出席者（順序不同）

越智誠二君	川原五郎君	井上要君
鷲見周保君	横山要三君	山本武君
小野暢三君	新堀重太郎君	久保田芳雄君
武田毅介君	板部成雄君	

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

昭和四年八月末 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
〃	〃	未定	〃	11,000		4. 9	甲板、甲板室取付中	〃
〃	〃	〃	〃	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	龍骨据附終り	〃
浦賀船渠會社	〃	〃	〃	7,500		未定	23%	山下汽船會社
原田造船所	汽	鯛浦丸	〃	800	4. 7		艤裝中	九州炭礦汽船會社
〃	〃	未定	〃	17		4. 10	15%	大阪商船會社
大阪鐵工所	發	平洋丸	〃	9,500		4. 10	70%	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	40%	藤岡船舶部
野口造船所	帆	北平丸	〃	400		4. 9	99%	手塚半治郎
〃	〃	和平丸	〃	400	4. 8		艤裝中	〃
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
〃	發	第一宇高丸	〃	300		4. 9	65%	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	〃	〃	105		未定	60%	笹岡鐵男
〃	〃	〃	〃	110		〃	60%	和歌山縣
〃	〃	〃	〃	110		〃	50%	門司宗太郎
〃	汽	〃	〃	5.000		〃	15%	朽木商事會社
三井玉工場	發	嵜山丸	〃	2,750	4. 8		艤裝中	大連汽船會社
〃	汽	未定	〃	2,000		未定	30%	山九運輸會社
三菱彦島造船所	發	雄基丸	〃	320	4. 8		艤裝中	共同漁業會社
〃	〃	妙義丸	〃	320		4. 9	60%	〃
三菱長崎造船所	〃	淺間丸	〃	16,000	3. 10		艤裝中	日本郵船會社
〃	〃	龍田丸	〃	16,000	4. 4		〃	〃
〃	〃	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	38%	〃
〃	〃	照國丸	〃	11,800		4. 12	20%	日本郵船會社
〃	〃	靖國丸	〃	11,800		5. 3	11%	〃
堀常吉	帆	新寶丸	木	120		4. 9	80%	岩野英吉

登簿船調

昭和四年八月末現在

汽船

積量		內地	朝鮮	臺灣	關東州	合計
20噸以上 100噸	隻 噸	1,648 68,722	138 5,702	21 790	25 1,190	1,832 76,404
100 〃 300〃	隻 噸	414 74,504	12 2,274	7 1,048	15 2,342	448 80,168
300 〃 500〃	隻 噸	141 56,053	6 2,301	1 499	7 3,014	155 61,867
500 〃 1,000〃	隻 噸	229 173,467	8 6,209	— —	7 5,531	244 185,207
1,000 〃 2,000〃	隻 噸	237 347,042	14 17,497	— —	12 17,433	263 381,972
2,000 〃 3,000〃	隻 噸	198 481,414	6 13,107	— —	13 33,630	217 528,151
3,000 〃 4,000〃	隻 噸	147 492,542	— —	— —	14 50,659	161 543,201
4,000 〃 5,000〃	隻 噸	77 343,650	— —	— —	24 108,390	101 452,040
5,000 〃 6,000〃	隻 噸	136 765,294	— —	— —	16 87,190	152 852,484
6,000 〃 7,000〃	隻 噸	53 345,497	— —	— —	5 31,276	58 376,773
7,000 〃 8,000〃	隻 噸	41 300,682	— —	— —	2 14,307	43 314,989
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	11 127,723	— —	— —	— —	11 127,723
計	隻 噸	3,354 3,777,103	184 47,090	29 2,337	141 363,192	3,708 4,189,722
內 100噸以上	隻 噸	1,706 3,708,381	46 41,388	8 1,547	116 362,002	1,876 4,113,318
內 1,000噸以上	隻 噸	922 3,404,357	20 30,604	— —	87 351,115	1,029 3,786,076

帆船

船種	積量		內地	朝鮮	臺灣	關東州	合計
噸數船	20噸以上 100噸	隻 噸	13,017 579,981	677 21,283	122 6,817	46 1,854	13,862 609,935
	100 〃 300 〃	隻 噸	2,095 293,389	2 384	3 360	— —	2,100 294,133
	300 〃 500 〃	隻 噸	30 11,615	3 983	— —	— —	33 12,598
	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	15,146 891,119	632 22,650	125 7,177	46 1,854	15,999 922,800
石數船	200石以上 300石	隻 石	259 64,923	— —	10 2,399	25 6,284	294 73,606
	300 〃 400 〃	隻 石	133 45,651	— —	2 759	4 1,474	139 47,884
	400 〃 500 〃	隻 石	54 23,810	— —	1 416	2 874	57 25,100
	500 〃 1,000 〃	隻 石	13 7,880	— —	2 1,402	— —	15 9,282
	1,000 〃		— —	— —	— —	— —	— —
	計	隻 石	459 142,264	—	15 4,976	31 8,632	505 155,872
合計		隻 噸	15,605 905,345	682 22,650	140 7,675	77 2,717	16,504 938,387

10 石を 1 噸に換算し合計に算入す

總計		內地	朝鮮	臺灣	關東州	合計
總計	隻 噸	18,959 4,682,448	866 69,740	169 10,012	218 365,909	20,212 5,128,109

最近本邦海上運賃及傭船料

			8 月 中	9 月 中 旬
運賃	石炭（單位噸）			
	九州	横濱間	円 円 1.00-1.30	円 円 1.10-1.20
		伊勢灣間	1.00-1.30	1.10-1.20
		上海間	1.20-1.50	1.30-1.40
		香港間	2.30-2.50	2.20-2.30
		新嘉坡間	3.10-3.40	3.00
	豆粕（單位擔）			
	大連	横濱間	.10- .14	.9- .10
		伊勢灣間	.10- .14	.9- .10
		神戸間	.9- .11	.7- .9
	小麥（單位噸）			
		北米（太平洋岸）—日本間	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 75.00-95.00	円 85.00
		北米（太平洋岸）—日本間	弗 弗 7.00-7.50	弗 7.00
	鐵（單位噸）			
		北米（太平洋岸）—日本間	——	——
		紐育——日本間	弗 12.00	弗 12.00
傭船料		大型	円 円 1.30-1.80	円 円 1.30
		中型	1.50-2.50	1.50
		小型	一區 4.00 二區 2.80-3.00	4.00 2.80

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	7 月 中	8 月 中
亞歷山	英本國	棉實	志片 志片 10.6	志片 志片 10.6
濠洲	英本國 歐大陸	小麥	26.9-30.0	30.0-30.6
ビルバオ	カーディフ	鑛石	6.9-7.0	$6.10\frac{1}{2}$- 7.0
孟買	英本國 歐大陸	雜貨	17.0-19.0	22.3-22.6
ビルマ	〃	米	——	——
ダニューブ河	〃	穀	——	——
リヴアプレート	〃	〃	24.0-25.0	18.9-24.0
北米大西洋岸	〃	〃	——	——
メキシコ灣	歐大陸	〃	× $3.4\frac{1}{2}$	× $3.4\frac{1}{2}$

備考　×印は標準な480 封度とす

(2) 英國發（1噸當）

カーディフ	坡西土	石炭	志片 志片 11.0-11.6	志片 志片 11.0-11.6
同	リヴアプレート	〃	13.0-14.0	13.6-15.9
同	セントヴインセント	〃	10.0-11.6	10.0-10.6

會員動靜

○入　會

氏名	會員種別	職名、勤務先	住所
谷川正次	協同員	三井物産株式會社船舶部機關士	東京市淺草區七軒町三澁谷方
山崎文夫	同	日本郵船株式會社運轉士	香川縣仲多度郡琴平町吉田清吾方
木下辰三郎	准員	三菱造船株式會社彥島造船所技手	下關市外、彥島町江ノ浦
武内博	同	大連汽船株式會社二等機關士	大連市山縣通リ大連汽船株式會社內
吉川善勝	同	東京帝國大學工學部船舶工學科學生	東京府下、荏原町中延六六七
島本浩一	同	同上	東京市牛込區市ヶ谷、谷町七四和泉方
吉識雅夫	同	同上	東京府下、和田堀町和泉二五〇
古賀正巳	同	同上	東京市外、澁谷町豐澤三七小宮山カヨ方
原田進一郎	同	同上	東京府下、田園調布四四八號伊東方
有吉金太	同	海軍技手海軍技術研究所造船研究部	東京府下、池上町市野倉三〇三
高橋高藏	同	同上	東京市麻布區笄町七一
株式會社新潟鐵工所		代表者　取締役社長　笹村吉郎 團體員（第四級）	東京市麴町區丸ノ內三ノ二
滿洲船渠株式會社		代表者　取締役社長　和田敬三 同　上（同　上）	大連市濱町三番地

○准員より正員に會員種格變更者

正員	太田友彌	正員	石崎隣之助	正員	大久保敬三
正員	平田國太郎	正員	酒井温	正員	岩神武夫
正員	赤木猪三郎				

○轉居、轉任

櫻井久雄	東京府下、荏原町戶越八二一
佐木々誠	橫濱市神奈川區青木町上反町四九七
後藤彰	東京府下、井荻町上荻窪五六八
水智幸雄	大阪市住吉區昭和町中一丁目四〇山內振一郎方
古川强	神戶市兵庫、兵庫三番町一丁目一二
水田守道	滋賀縣八日市、飛行第三聯隊第一中隊幹部候補生、
霜鳥芳三	大阪市西淀川區佃町四二六
岩崎誠一	東京府立川飛行第五聯隊第四中隊へ入營中、來る十二月末除隊の上は三菱長崎造船所へ復歸の筈
岡村耕次	昭和四年九月十八日、日本郵船、六甲丸に乘船、(住所東京市外澁谷町神泉二九土屋正男方)
肥田倭夫	神戶市楠町七丁目一七ノ二二
松田竹太郎	米國出張海軍造船監督官、昭和四年十月二十七日歸朝の筈
吉田忠雄	東京府下、世田ヶ谷町代田一〇八五
佐々木誠	日本郵船株式會社橫濱支店へ轉勤(住所、橫濱市神奈川區青木町上反町四九七
森洋二	若松市山手通六丁目一五二、若松高等小學校運動場前（電話 644 番）
酒井温	兵庫縣武庫郡本山村岡本字裏ノ山二八六
吉岡好治	神戶市東須磨松風町六ノ二七
伊藤信雄	東京府下、蒲田町蒲田新宿六一八
吉川彌三郎	當分通信先、東京帝國大學工學部船舶工學科教室內
小原啓以智	大分縣中津市金谷、屋形治作方
宮本吉太郎	東京市外、巢鴨町一二五〇
川井芳一	東京市外、澁谷町櫻丘四四
耶葆琛	山東省青島特別市港務局工務科船機股股長
須原坦	大連市山城町二
林成昭	姬路市北條口一二一
篠田一郎	廣島縣御調郡土生町株式會社大阪鐵工場因島工場
淸水利治	大阪府豐能郡豐津村垂水九二九
大和茂幸	神奈川縣浦賀町大津一一一一
松原潔	神戶市東須磨下中島町一丁目五一
稻葉晃	三菱神戶造船所辭職、(住所、三重縣四日市市午起、稻葉勝吉方)

松平季雄	横濱市神奈川區青木町臺町一八一一	有井澄	神戸市上筒井通七丁目二五ノ三
岡敬藏	横濱市中區山王町六五六	齋藤郁之助	横濱市鶴見區平安町一丁目一〇二
落合惣三郎	大阪市東成區北生野町一ノ二三	江田太郎	東京市本郷區駒込追分町六〇同學舍內
平井進	長崎市伊良林町一ノ一四	土屋宗任	横濱市神奈川區青木町澤渡谷一六四四
松尾鶴松	東京市芝公園地二十一號地九號麻尾方	松尾玄次	神戸市東須磨濱山一一七
久埜茂	東京市麹町區永田町二ノ六八遞信省官舍	齋信佐吉	東京府下、荏原町小山五〇七
綾目喜一（舊姓丸谷）	大阪市港區新池田町二丁目四五	中西松右衞門	神戸市平野都由乃町一丁目五六
松岡松太郎	横濱市神奈川區青木町東輕井澤一九一七	村上勇次郎	東京府北多摩郡三鷹村上連雀六三一
笠井元一	大阪市住吉區相生通二丁目一八	竹之上文雄	神戸市大同町五丁目一三ノ八
三島忠雄	横濱市鶴見區豊岡町二ノ六一九	島本熊一郎	大阪市住吉區住吉町帝塚山三七四（電話、住吉、2807番）
東常任	（住所稱呼變更）神戸市大手町三丁目二十一番屋敷	福島延	勤務先、大阪遞信局海事部（住所、大阪市西淀川區大仁本町一ノ五二戸田丈次方）
松岡實郎	（社內勤務所屬變更）株式會社大阪鐵工所造船部造船工作課	大南德之亟	上海施高塔路東照里八〇號
		大賀悳二	札幌市南五條西八丁目

○會員改姓

（新）	（舊）
綾目喜一	丸谷喜一

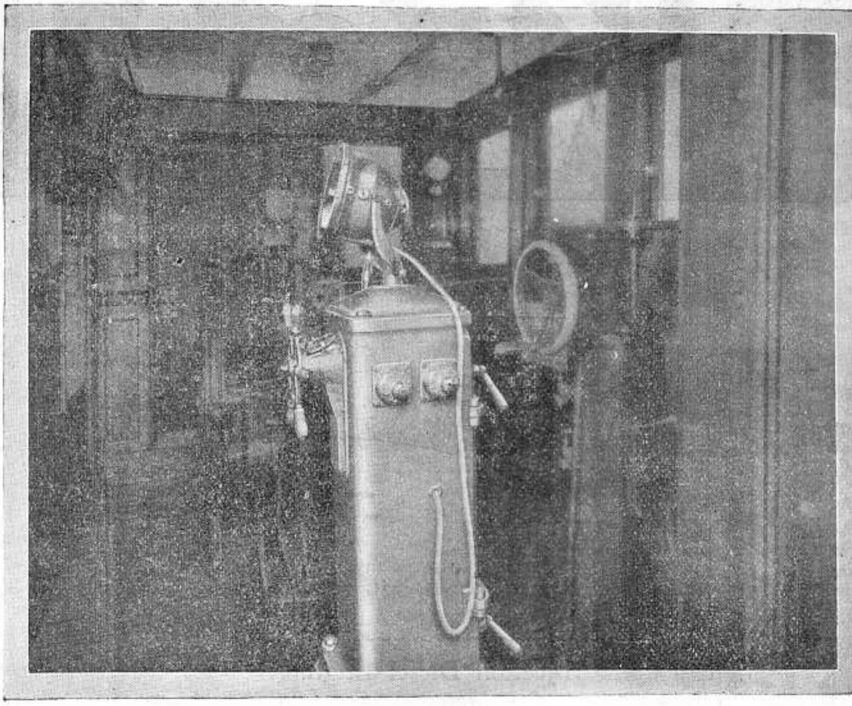
○死亡會員

協同員	山口半兵衞君	昭和四年九月三日兵庫縣武庫郡蘆屋自宅に於て死去
准員	石崎芳兵衞君	昭和四年九月五日郷里愛媛縣溫泉郡三津濱町須先町二一實家に於て死去

本會は此の訃音に接し謹みて哀悼の意を表す

造船協會役員

理事（會長）	末廣恭二		
理事（主事）	越智誠二		
理事（主計）	濱田彪		
理事（編輯主任）	平賀讓		
理事	藤島範平		
監事	今岡純一郎		
監事	山本幸男		
評議員	山本武藏		堤正義
男爵	徳大寺則麿		元良信太郎
男爵	斯波忠三郎		斯波孝四郎
	野中季雄		山本開藏
	淺井虎之助		永村清
	湊一麿		島谷敏郎
	河上邦彦		太田丙子郎
	近藤基樹		鵜飼宗平
	目良恒		田原得三
會務委員	陰山金四郎		湊一麿
編輯委員	西村眞次		高橋良之助
	横山要三		萩與可
	板部成雄		小室鉦
	野島休五		山縣昌夫
	加藤熙彦		菊植鐵三
	加藤弘		大瀬進
	牛尾平之助		片山有樹
	出淵巽		岡本方行
	家永文彦		
地方委員（神奈川）	小野暢三	（横濱）	波多野友次郎
（横須賀）	山本幹之助	（大阪）	高木清吉
（神戸）	目良恒	（呉）	渡邊隆吉
（佐世保）	久保綱彦	（長崎）	元良信太郎
（舞鶴）	河東卓四郎	（函館）	大塚巖
（浦賀）	藤田益三	（因島）	須田勝雄
（播磨）	神保敏男	（宇野）	鵜飼宗平
（福岡）	廣瀬瀧次	（關門）	加藤清一
內外通信委員（神戸）	中島喜代治	（大阪）	小田末治郎
（大阪）	荒木賢保	（長崎）	元良信太郎
（長崎）	岩井祐文	（香港）	武正敏男
（上海）	荒木重義		



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

三つ星ハガネ

各種特殊鋼

高速度鋼
普通工具鋼
專賣特許 耐蝕鋼
專賣特許 タービン翼用
專賣特許 耐蝕性合金鐵
專賣特許 磁石鋼
專賣特許 ゲージ鋼

製釘製線用ダイス鋼
發條用鋼
飛行機用鋼
自動車用鋼
打刃物用鋼
鑛用鋼

日本特殊鋼合資會社

東京市外大森
電話 高輪特二六〇八
大森六一二

各種高級工具

ミーリングカツター
ギヤーカツター
ギヤーホツブ
リーマー
ツイストドリル

タツプ及ダイス
各種ゲージ
各種成形磁石
各種鍛工品
ドロツプホージ品

東京營業所

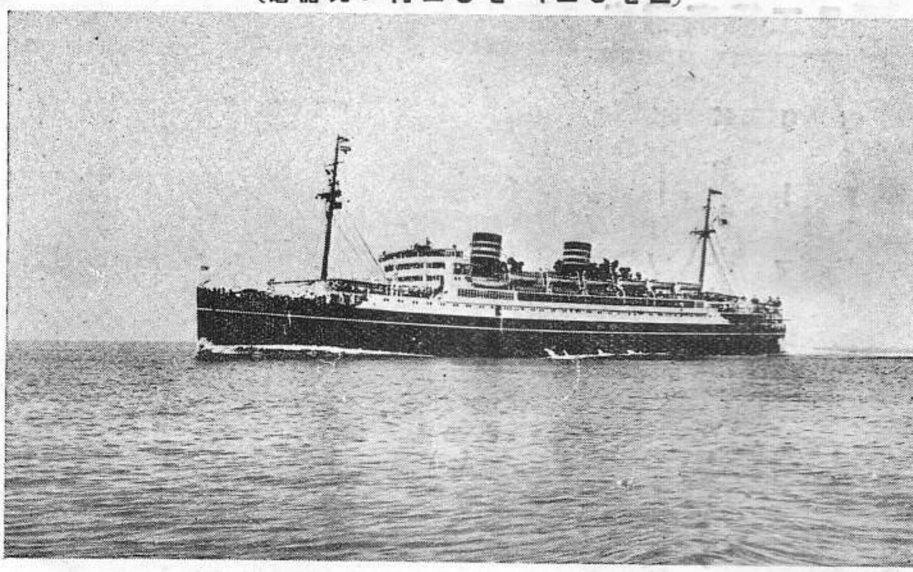
東京市芝區三島町十番地
電話 高輪 特六二七五
三九五六
七五九四

名古屋出張所

名古屋市中區南大津町一丁目
電話東二一九四

三菱造船株式會社

東京市麴町區丸ノ內二丁目四番地
(電話丸ノ內二〇一七、二〇二七)



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)

營業科目

○船舶、艦艇ノ建造竝修理
○火力發電所設備一式
○水力發電所設備一式
○各種汽罐
○各種喞筒類
○ターボブローア、ロードローラー、電車用電氣機、蒸氣機關車、電氣機關車、エヤーブレーキ其ノ他各種機械
○一般鐵構工事
○水タンク、油タンク、瓦斯タンク
○鋼板製管類(水道、下水、排水用其ノ他)
○鋼製客貨車々體及鋼製電車々體
○耐火アートメタル製事務用机、書類棚、椅子其ノ他家具類一式
○各種鑄物及打物
○特種合金　飯高メタル其ノ他

尚各種御計畫設計ニ關シテハ夫々專門ノ技術者參上御相談ニ應シ可申上候

工場

神戸造船所　神戸市兵庫和田崎町
長崎兵器製作所　長崎市茂里町
長崎造船所　長崎市飽ノ浦
彥島造船所　下關市外彥島

研究所

東京市本郷駒込

昭和四年十月十三日印刷
昭和四年十月十五日發行

編輯兼發行者　東京市下谷區谷中眞島町一番地　川尻政吾
印刷者　東京市[illegible]美土代町二丁目一番地　島連太郎
印刷所　東京市神田區美土代町二丁目一番地　三秀舍

發行所　東京市麴町區丸ノ內三丁目八番地(丸ノ內・仲・六號館三號)　造船協會
電話丸ノ內(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[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