VOL. XXIX SEPTEMBER/OCTOBER, 1942 No. 9/10

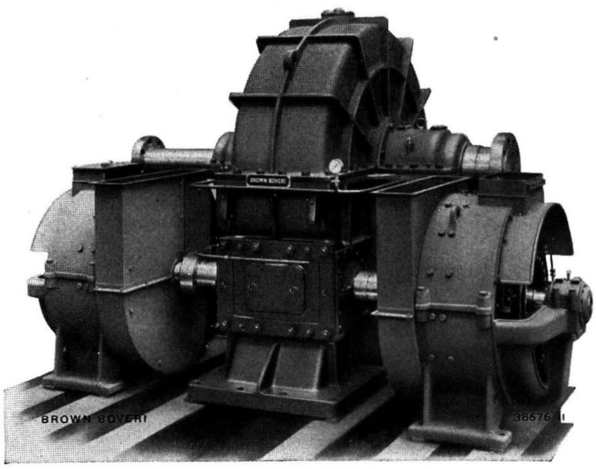
THE BROWN BOVERI REVIEW

Special Marine Number

BROWN BOVERI

BROWN BOVERI

MARINE EQUIPMENT



Diesel-electric d. c. propulsion equipment for a paddle boat.
Driving motors with gearing.

Propulsion Equipments for Merchant Vessels and Warships

Velox Steam Generators

Steam and Gas Turbines with Gearing

Oil-pressure Manœuvring Gear

Marine Condensers with Tubes expanded at both ends

Turbo- and Diesel-electric Propulsion Equipment

Electrical Propulsion Machinery for Submarines

Auxiliaries for Ships, Docks, and Harbour Installations

Turbo- and Diesel-driven Generators

Electrical Drives with Control Gear

Small Auxiliary Turbines

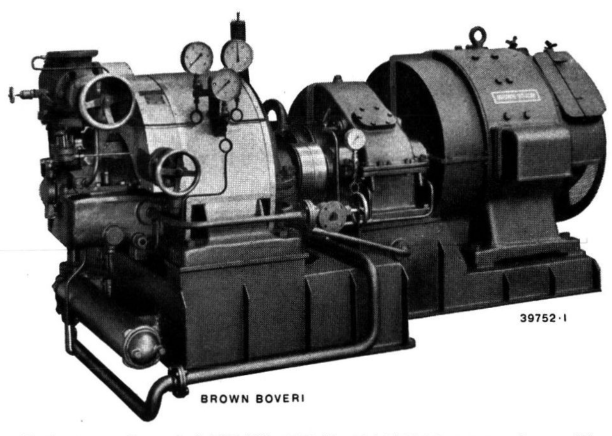
Lighting Turbo-sets and Turbo-fans

Quick-acting Regulators and Automatic Synchronisers

Exhaust Turbo-chargers and Scavenging Blowers for Diesel Engines

Air Conditioning Plants and Refrigerating Machines

Electrical Equipment for Floating Docks, Dredgers, Grain Elevators, etc.



Turbo-generator rated 420 kW, 220 V, 4200/1200 r. p. m., for auxiliary power station.

THE BROWN BOVERI REVIEW

THE HOUSE JOURNAL OF BROWN, BOVERI & COMPANY, LIMITED, BADEN (SWITZERLAND)

VOL. XXIX — SEPTEMBER/OCTOBER, 1942 — No. 9/10

CONTENTS:

40 YEARS' OF BROWN BOVERI MARINE MACHINERY DEVELOPMENT.

Decimal Index 629.12—8 (091)
629.12.06 (091)

ALTHOUGH this special marine number is dedicated more particularly to our friends in shipping, shipbuilding, and naval circles, and the articles presented — while laying no claim to completeness — have been chiefly selected with a view to giving this class of reader a general idea of the present state of design of our ship's propelling and auxiliary machinery, other readers will doubtless also be interested to learn a little of the part we have played in the development of the marine turbine and marine machinery in general, more especially so as the fiftieth anniversary of the foundation of our firm, celebrated last year, coincided with exactly forty years' activity in the marine turbine field.

They will perhaps be surprised to hear that:

Brown Boveri not only built (and that in the workshops at Baden!) the first *marine turbine on the Continent* in accordance with a project submitted in 1901, but just recently also marine turbines with, as far as can be ascertained, the highest rating per shaft in the world.

Brown Boveri not only took a leading part in all developments of the marine turbine on the Continent, e. g., in its application to torpedo-boats, light and battle-cruisers, and battle-ships, but were also the first firm to employ gearing in conjunction with main marine turbines.

Brown Boveri can point to pioneer achievements in the electrical propulsion of ships, generation of steam, and design of auxiliaries, which have found a permanent place in technical history.

How was this possible?

In the first place because our firm, in the persons of Walter Boveri and Charles Brown, was the first on the Continent to foresee the importance to which the steam turbine would ultimately attain not only as stationary prime mover, but above all for the propulsion of ships and in particular of warships. The torpedo-boat reciprocating steam-engine had at the time, i. e., at the end of the nineties, been developed to the limit of its capacity. Parsons had already applied the steam turbine to the propulsion of the "Turbinia" with success, thus enabling England to outstrip all other countries in the steam turbine field. By acquiring design and manufacturing rights in the Parsons turbine in 1900 Brown Boveri secured this technical advance for the Continent and thus laid the foundation of the subsequent important developments in this field. While it cannot be gainsaid that Parsons' patents were employed as basis for the Brown Boveri designs it is equally true that the Baden firm went its own way in many respects from the very beginning, so that the successes achieved can in all conscience be accredited to Brown Boveri.

At the instigation of Walter Boveri, who had a life-long predilection for the *marine turbine,* the "Turbinia Deutsche Parsons Marine A.-G." was founded on the 17th September, 1901. The partners comprised, in addition to our company, the Brown, Boveri & Cie. A.-G., Mannheim, and the Parsons Foreign Patent Co., Ltd., London. The "Turbinia" was a pure selling and patent holding company. Its activities were restricted to Germany, Russia, and a few northern lands, which explains why the majority of our marine turbines have been supplied to these countries. Up till 1914 the design of marine turbines was concentrated at Baden, where special departments had been created for marine turbines and auxiliaries. In that year, for national reasons, Brown, Boveri & Cie. A.-G., Mannheim, set up an independent marine turbine office, but for other countries design and development were still carried out at Baden. After the submission of the first marine turbine projects to the German Admiralty in the summer of 1901 the construction of marine turbines was begun at the Baden workshops in 1903 with the turbine plant for the German torpedo-boat "S 125", this being the first marine turbine plant to be built on the Continent. At practically the same time Brown, Boveri & Cie. A.-G., Mannheim, started work on a turbine plant for the light cruiser "Lübeck".

The development of the Brown Boveri-Parsons marine turbine in the ensuing years is of absorbing interest [1]. The lines it followed are best given by the following extract from the paper read by Walter Boveri before the "Schiffsbautechnische Gesellschaft" on the 23rd November, 1906:—

"If the German Navy were given the opportunity of trying out turbine-propelled ships on an absolutely equal footing with England it is thanks to my firm. None of the German shipbuilders was prepared at that time to risk such an experiment. After having been instrumental in introducing land turbines on the Continent, we therefore decided also to take up the design of marine turbines. We were well aware that from a business point of view the outlook was far from favourable, but financial considerations were not allowed to outweigh technical convictions and interests. The trials of the "S 125" and the "Lübeck" were undertaken entirely at our risk and we entered into a contract for the cruiser "Lübeck" the like of which has never since been concluded for a warship. As a matter of fact it was the first occasion on which a manufacturer [2] gave a guarantee for the maximum speed instead of for the usual indicated horse-power. This concession on our part not only met with no appreciation, but gave rise to adverse criticism and even enmity in many quarters. I will not speak of those who from conviction or for financial reasons are or were the opponents of the turbine: their opposition is only natural. But even those who desired to further the turbine have chiefly proved a bar to its progress."

[1] W. Boveri: Contribution to discussion on Prof. Riedler's paper: "Ueber Dampfturbinen", Jahrbuch der Schiffbautechnischen Gesellschaft, Berlin, 1904, p. 307.
W. Boveri: "Die Verwendung der Parsons-Turbine als Schiffsmaschine", with discussion, Jahrbuch der Schiffbautechnischen Gesellschaft, Berlin, 1907, p. 85.
Grauert: "Ueber Dampfturbinen." Reprint from "Marine-Rundschau", January, 1904.
H. Schmidt: "Die Dampfturbine im Schiffsbetrieb" and "Schiffshilfsmaschinen mit Turbinenantrieb", Deutscher Schiffbau 1908, issued on the occasion of the first German Shipbuilding Exhibition, Berlin, pp. 47 and 63.
J. Baasch: "Brown Boveri Marine Turbines", The Brown Boveri Review 1922, pp. 119/139.

[2] That is, the manufacturer of the machinery.

Be that as it may, in 1914, at the outbreak of the World War, a great part of the vessels in both the German and Russian Navies was equipped with Brown Boveri-Parsons marine turbines having an aggregate rating of 1·7 million S. H. P., while at the conclusion of hostilities this figure had reached approximately 2·8 million S. H. P.

Space will not allow of going into further details of the development of the Brown Boveri marine turbine, but we would refer the reader to the following historical retrospect "Milestones in the Forty Years' Activity of Brown Boveri in the Marine Machinery Field".

Another factor largely contributing to the furtherance of the Brown Boveri turbine was the cultivation of relations with various prominent shipbuilders, who acquired manufacturing licences (see above-mentioned historical retrospect). It is only natural that shipbuilders should seek the collaboration of a leading steam turbine firm, inasmuch as since the introduction of reduction gearing the marine turbine has become a high-speed machine, i. e., is of fundamentally the same design as the power-station turbine, so that the experience gained with the land machine in thousands of installations under very varied operating conditions can be applied to the marine turbine. From a business point of view, therefore, it is definitely of advantage for shipbuilders to be able to avail themselves of the improvements continually introduced by a prominent steam turbine manufacturing firm, while, vice versa, the latter benefits from the wider experience of the shipyards for the design of their turbines. As a matter of fact Brown Boveri also held shares in shipbuilding companies, e. g., from 1908 to 1924 in the Howaldtswerke, Kiel — again at the instigation of Walter Boveri — while during the existence of the American Brown Boveri Electric Corp. (1925 to 1931) the important New York Shipbuilding Corp. was also affiliated to it.

The fact that we have invariably been in the forefront of the steam turbine field and cultivated relations with prominent shipbuilders and shipowners, together with the circumstances cited at the beginning of these notes, explains the success achieved by our firm in the marine machinery field and how it has been able to maintain its activities right up till the present day.

In the *marine auxiliaries field* it was only natural that we should obtain our first successes with turbo-generators, which found application on warships practically at the same time as the marine turbine[1]. Here, too, the reciprocating steam engine had run its course. Thereupon, the development of turbo-fans for the forced draught of marine boilers[2] and condensing auxiliaries for the main turbine was taken up. In 1916, at the instigation of Sulzer Bros., Winterthur, we developed scavenging blowers for two-stroke-cycle marine Diesel engines. The success achieved in this field was chiefly due to the fact that we had already developed a special design of high-speed d. c. motor to a high degree of perfection. This technical advance gave us a world monopoly in the electrically-driven scavenging blower field for a long period. The turbo-charging blower with exhaust-gas turbine drive for increasing the power of four-stroke-cycle Diesel engines by the Büchi process was first applied to the marine Diesel engine and developed in Switzerland. To-day, there is hardly a fleet in the world in which Brown Boveri exhaust turbo-chargers are not employed.

Electricity found wide application on warships from the earliest times, but it was not until after the World War 1914—1918, when the Diesel engine was applied to the propulsion of merchant vessels, that it attained to great importance. Whereas Walter Boveri pushed the marine turbine, Charles Brown was from the beginning a strong advocate of the application of electricity to ships. As early as 1904 he was granted the German patent No. 169,559 covering an electrical drive for the propulsion of battle-ships at cruising speed, an arrangement, however, which became of significance only much later. In 1906 when a solution of the problem of the mode of pro-

[1] Deutscher Schiffbau 1908, p. 63: "Schiffshilfsmaschinen mit Turbinenantrieb".

[2] BBC Mitteilungen 1920, pp. 103 and 130: "BBC Marine-Turbolüfter".

pulsion for the new "Dreadnoughts" was being sought in England, John Brown & Co. of Clydebank got into touch with Charles Brown and requested him to study the possibilities of turbo-electric drive. The project submitted (under date of the 3rd December, 1906) evoked amazement through its completeness[1], embodying fundamentally as it did all of the features of the first turbo-electric propelling plants put into service in the U. S. A. Navy ten years later.

In 1929/30 we supplied from Baden the first pure Diesel-electric drives for warships, viz., for the Finnish coastal cruisers "Wäinämöinen" and "Ilmarinen". Moreover, we developed in collaboration with Brown, Boveri & Cie. A.-G., Mannheim, the Diesel-electric drive with alternating current fitted for the first time on the M. S. "Wupperthal", a vessel in the Hamburg-Australia service. This mode of drive has since found wide application in Germany. At the same time we began propagating, with considerable success, the introduction of alternating current on ships for the drive of the auxiliaries.

[1] The project is reproduced in Stodola's "Die Dampfturbine", 4th edition, p. 535, Fig. 608 and in Fig. 3 on page 226 of the present number of this journal.

Further developments in recent years of significance from a marine point of view are the *Velox steam generator* — the first supercharged marine boiler — which has been fitted on various ships, and the *combustion turbine*, the last link in the chain of development. The possibilities of the latter as ship's propelling machine are dealt with elsewhere in the present number of this journal.

"Navigare necesse est." Switzerland has also recognized the truth of this proverb, inasmuch as she now has her own maritime law and her own, if modest, merchant fleet, which it is intended to reinforce with new tonnage. As early as 1843 the national economist Friedr. List[2] wrote: "The sea is the testing place of the strength and spirit of enterprise of the nations and the cradle of their liberty. He who shows no interest in the sea is precluded from the good things and honours of the world." To us our more than forty years' activity in the marine turbine field has also proved a constant source of strength and spirit of enterprise.

(MS 909) *E. Klingelfuss. (E. G. W.)*

[2] From the preface to Boer's "Das Schiffbuch".

MILESTONES IN THE FORTY YEARS' ACTIVITY OF BROWN BOVERI IN THE MARINE MACHINERY FIELD.

A HISTORICAL RETROSPECT.

A. MARINE TURBINES AND VELOX BOILERS.

1900 (19th April). Foundation of the A. G. für Dampfturbinen, System Brown Boveri-Parsons at Baden (Switzerland), the interested parties being Brown, Boveri & Co., Ltd., Baden (Switzerland) and the Parsons Foreign Patent Co., Ltd., London. Brown Boveri acquired exclusive manufacturing rights in the Parsons turbine (steam, air, or gas, but excluding marine drives) for Switzerland, France (until 1903), Germany, Russia, Belgium (until 1904), and as from 1904 also for Norway, Sweden, Denmark, Spain, and Portugal. Manufacture of marine turbo-generators taken up (see "Auxiliaries").

1901 (21st June). Submission of first marine turbine project to the German Navy for the propulsion of a torpedo-boat (Fig. 1), steam reciprocating engines being retained for the cruising speed; in a second project, however, separate cruising turbines (Fig. 2) were put forward.
(17th September). Foundation of the "Turbinia", Deutsche Parsons Marine A. G. at Berlin, the associated parties being Brown, Boveri & Co., Ltd., Baden (Switzerland), Brown, Boveri & Cie. A.-G., Mannheim, and the Parsons Foreign Patent Co., Ltd., London. Acquisition of exclusive manufacturing rights in the Parsons marine turbine for Germany and later (1904) also for Russia.

1903 Brown, Boveri & Co., Ltd., Baden (Switzerland) entrusted with first contract on the Continent covering a marine turbine installation rated 7000 S.H.P. at 750—850 r. p. m. for the German torpedo-boat "S 125" on the basis of the second project with cruising turbines (Fig. 2) submitted in 1901[1].
Design of 14,000 S. H. P., 680 r. p. m. turbine installation for the German light cruiser "Lübeck"[2] built in the workshops of Brown, Boveri & Cie. A.-G., Mannheim.

1904 (19th September). Beginning of first trials with torpedo-boat "S 125".

[1] J. Baasch: "Brown Boveri Marine Turbines". Layout, Sections through Turbines, and Workshop View. The Brown Boveri Review 1922, pp. 120/121, Figs. 3—6.
[2] See same article pp. 126—128, Figs. 19—22.

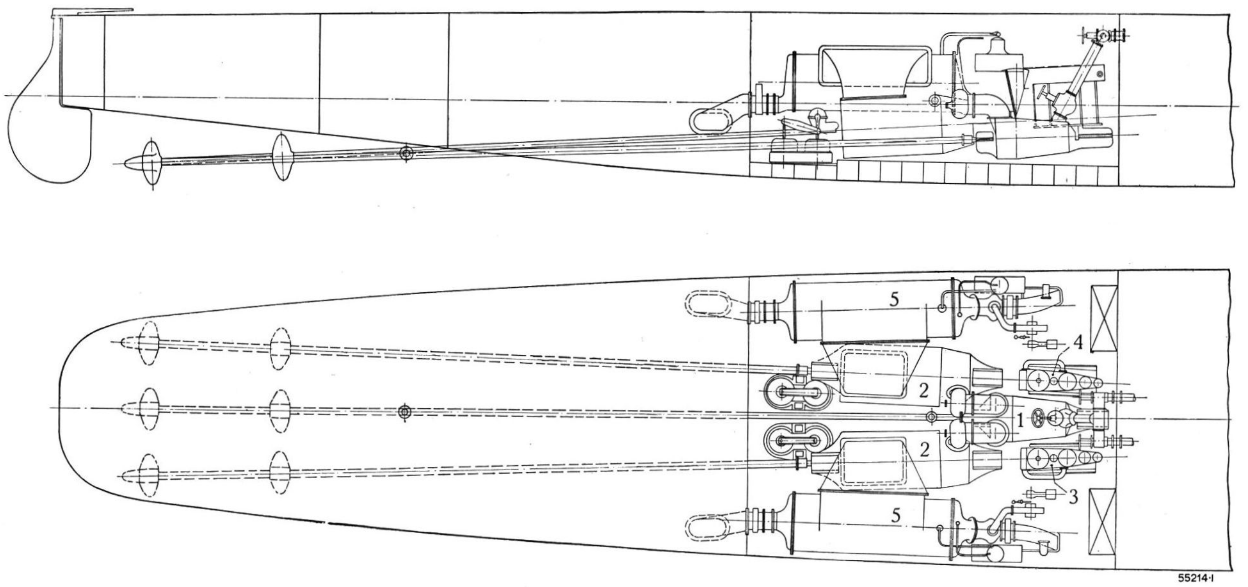


Fig. 1. — First project dated 21st June, 1901, covering a Brown Boveri-Parsons marine turbine installation for the propulsion of a German torpedo-boat. Rating 7000 S. H. P. at approx. 800 r. p. m.

1 = High-pressure turbine. 2 = Low-pressure turbines. 3, 4 = Steam reciprocating engines for cruising speed. 5 = Condensers.

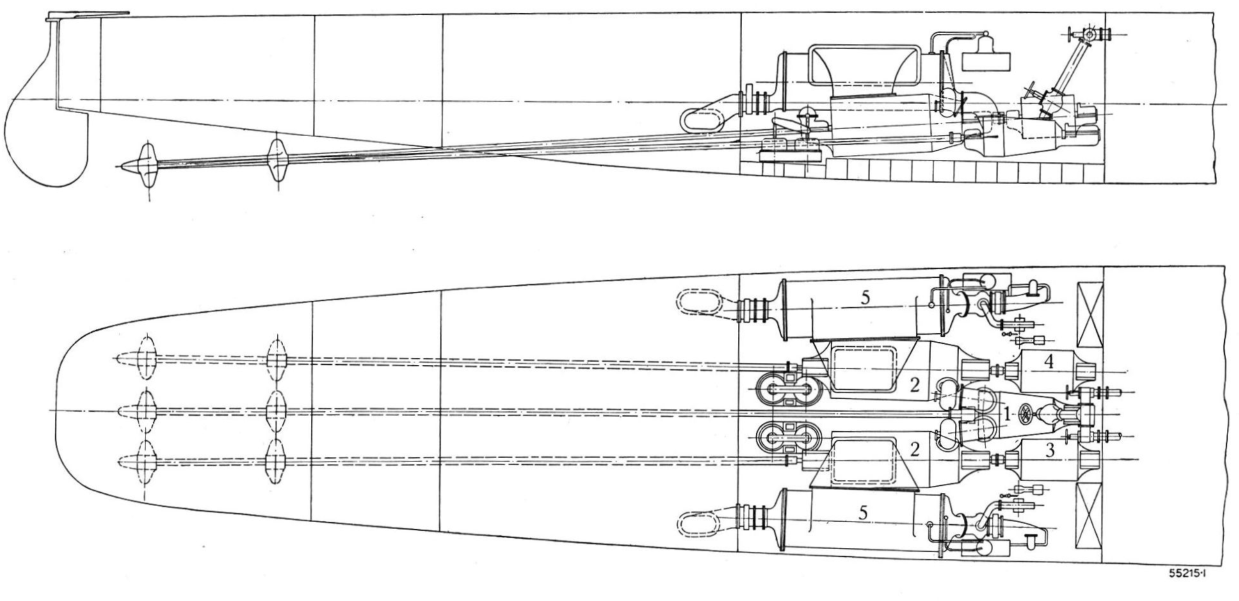


Fig. 2. — Second and final project for the same installation (Fig. 1) with turbines instead of reciprocating engines for the cruising speed.

1 = High-pressure turbine. 3 = High-pressure cruising turbine.
2 = Low-pressure turbines. 4 = Low-pressure cruising turbine.
5 = Condensers.

It was on the basis of this second project that the first marine turbine installation on the European Continent was constructed in the workshops of Brown, Boveri & Company, Limited, Baden (Switzerland) in 1903.

The drawings in both cases were worked out to the last detail, thus proving that well-conceived designs of the Brown Boveri marine turbine were already available at the beginning of 1901.

1905 (In spring). Beginning of trials with cruiser "Lübeck".
DRP 183,844 and 109,033, two Brown Boveri patents on which the present marine turbine design is based. These patents protect the division of the turbine into high-pressure and low-pressure cylinders with the h.p. turbine running faster than the l.p. machine. H.p. and l.p. turbines are coupled either through an electrical transmission or mechanically through gearing.
Licence agreement with the S. A. des Usines Franco-Russes (Anciens Etablissements Baird) at St. Petersburg.

1906 Design of first turbine installation rated 10,000 H.P. at 440 r.p.m. for a passenger vessel (mail steamer "Charles Roux" owned by the Cie. Générale Transatlantique for the Marseilles-Algiers route; trials July, 1908)[1], manufactured at the Le Bourget workshops of the Cie. Electro-Mécanique. Thereupon the turbines for the French torpedo-boats "Bouclier", "Chasseur", and "Francis Garnier" were also calculated and partly designed by Brown Boveri, Baden.
DRP 190,157 (bimetal lacing wire) and DRP 193,192 (sharpened blade edges), two patents which later became of great importance in turbine design.
Licence granted to Blohm & Voss, Shipbuilders, Hamburg.
Licence agreement with the Baltische Schiffbau- & mechanische Fabrik des Marineressorts, St. Petersburg. A series of other Russian shipbuilders followed some time later (Putilow, Bœcker, Nikolaijew).

1907 Design data evolved for turbine installation of German light cruiser "Dresden". First installation with superheated steam (Turbine manufacturer: Blohm & Voss).
Parsons pure reaction turbine abandoned for Brown Boveri combined impulse and reaction type.

1908 Licences granted to Naval Shipyards at Kiel, Wilhelmshaven, and Danzig.
Design data evolved for the 70,000 S.H.P., 320 r.p.m. turbine installation of the battle-cruiser "Von der Tann"[2] (Turbine manufacturer: Blohm & Voss) and for the 2×7000 H.P. turbine installation of the torpedo-boats G 169—172 (Shipbuilders: Germaniawerft, Kiel). Turbines manufactured at Mannheim.
Acquisition of Howaldt's Shipyard, Kiel, by the Turbinia.

1909 Licence granted to Burmeister & Wain, Copenhagen and design of the turbines for the Danish torpedo-boats "Stulven" and "Flyvefisken", built by Burmeister & Wain.
Design of first turbine installations each with a total rating of 40,000 S.H.P. at 280 r.p.m. for the battle-ships "Kaiser" and "Kaiserin" (Turbine manufacturers: Marinewerft Kiel and Brown, Boveri & Cie. A.-G., Mannheim)[3].
Design data evolved for the turbine installations of the torpedo-boats "G 174" and "G 175". First Brown Boveri single-shaft combined impulse and reaction marine turbine units[4].

1910 Six 15,000 S.H.P., 700 r.p.m. combined impulse and reaction turbine installations with one impulse stage, for further boats "G 7 – G 12"[5]. Of these, four vessels are still in uninterrupted service, i. e., more than thirty years afterwards.
Design data evolved for the turbines of four Russian battle-cruisers, each with a rating of 100,000 S.H.P.[6]
DRP 228,926 covering a condenser for continuous operation with twin water boxes permitting cooling tubes to be cleaned during service or leaky tubes to be cut out[7].

1911 Collaboration in design of 70,000 S.H.P., 180 r.p.m. turbine installations for the liners "Vaterland" and "Bismarck" of the Hamburg-America Line (Turbine manufacturer: Blohm & Voss). The "Vulcan-Werke", Hamburg, had acquired a special licence prior to this for the construction of the turbine installations of the sister ship "Imperator".
Design of exhaust-steam turbine (5000 S.H.P., 180 r.p.m.) for the S. S. "Joh. Heinr. Burckhardt" of the Hamburg-America Line (later "Reliance")[8]; this drives a separate propeller shaft.

1912 Contract for six 20,000 S.H.P., 560 r.p.m. turbine installations for Russian destroyers. Turbines made at Baden[9].

1913 DRP 271,482. Surface condenser with V-shaped steam chamber down to level of condensate (OV condenser), a design which has now been copied by many other firms[10].

1914 Design of first marine geared turbine installation (light cruiser "Karlsruhe")[11].

1916 Design of geared turbine installations with a rating of 20,000 S.H.P., 4200/3200/480 r.p.m., for destroyers. Power split up between four cylinders, two before and two after the gearing[12].
DRP 304,689. Double-flow l.p. turbine with astern stage incorporated in the exhaust-steam part.

1919 First project evolved for application of exhaust-steam turbine to triple-expansion reciprocating engines on common shaft, for the Hamburg-America Line.

1920 Design and construction at Baden of first Brown Boveri 3000 S.H.P., 3000/75 r.p.m. turbine installation with double reduction gearing for a merchant vessel[13], the gearing being of the single helical type throughout and with double mesh of the turbine

[1] The Brown Boveri Review 1922, p. 134, Figs. 38 and 39, and illustration on cover.
Schiffbau 1907, p. 142.
[2] The Brown Boveri Review 1922, p. 129, Fig. 26.
[3] The Brown Boveri Review 1922, pp. 130—132, Figs. 29—34.
[4] The Brown Boveri Review 1922, p. 124, Figs. 12 and 13.
[5] The Brown Boveri Review 1922, p. 125, Fig. 14.
[6] The Brown Boveri Review 1922, pp. 129 and 130, Figs. 27 and 28.
[7] BBC Mitteilungen 1916, p. 22, Figs. 1 and 2.
[8] The Brown Boveri Review 1922, pp. 134—135, Figs. 41—44.
[9] The Brown Boveri Review 1922, p. 125, Figs. 16 and 17.
[10] BBC Mitteilungen 1916, p. 105, Figs. 9—11.
[11] The Brown Boveri Review 1922, pp. 141 and 142, Figs. 48—50.
[12] The Brown Boveri Review 1922, p. 143, Figs. 51 and 52.
[13] The Brown Boveri Review 1922, pp. 137 and 144—146, Figs. 53—55.

pinions. (Fig. 3 on page 273 of the present number of this journal.)

Design of 5200 S. H. P., 3200/2300/85 r. p. m. turbine installation for the S. S. "Thuringia" and "Westphalia" of the Hamburg-America Line [1]. Turbine manufacturer: Brown, Boveri & Cie. A.-G., Mannheim.

1922 Design of 1500 S.H.P., 3600/38 r.p.m. geared turbine installations for the paddle-tug "Dordrecht" operating on the Rhine[2]. Ratio of double reduction gear 95:1. Manufacturer: Brown, Boveri & Cie. A.G., Mannheim.

1926 Oil-pressure manœuvring gear applied to marine turbines for the first time[3].

Construction of 2×1100 S. H. P., 5060/135 r. p. m. geared turbine installations at Baden for the Japanese railway ferry-boat "Seikan Maru I"[4]. Smallest marine turbine unit built by Brown Boveri.

1928 Construction of geared turbine installations at Baden for the Danish torpedo-boats "Dragen" and "Laxen". Total rating 6000 S.H.P., 5000/3850/550 r.p.m. each.

Design and construction of exhaust-steam turbines with gearing for application to existing steam reciprocating engines on a common shaft, for the Rotterdam Lloyd and Hamburg-America Lines[5]. (These installations were constructed exactly as proposed in the project submitted to the HAL in 1919.)

1929 Design data evolved for the first high-pressure primary marine turbine with a high degree of superheat (60/24 kg/cm² abs, 440° C) for the S. S. "Uckermark" of the Hamburg-America Line operating on the Far East route. Turbine manufacturer: Blohm & Voss, Hamburg. This installation was based on the design of, and experience with, the first super-high-pressure primary turbine in the world (50 kg/cm² abs, 450° C) built by us for Langerbrugge in 1925.

1931 First projects with Velox boilers for a 26,000 S.H.P. merchant vessel. Weight of orthodox water-tube boiler plant 700 t, of Velox boiler plant 185 t.

DRP 583,057. Surface condenser with pre-curved cooling tubes expanded at both ends, a design particularly important for marine applications.

1933 Order for two Velox boilers with an evaporation of 15 t/h and 45 t/h, respectively, for a navy[6].

1934 Contract for two 50 t/h Velox boilers for other navies.

1935 Condenser with pre-curved tubes expanded at both ends (DRP 583,057) employed for the first time in the marine field.

Design of first Brown Boveri geared turbine installations with auxiliaries driven from the main turbine (5300 H. P., 5200/3700/124 r. p. m. for the banana-carrying steamers "Katiola" and "Kita" of the Chargeurs Réunis)[7].

First Velox boiler with an evaporation of 35 t/h, 50 kg/cm² abs, 450° C, on a merchant vessel (S. S. "Athos II" of the Messageries Maritimes)[8]. Substituted for a cylindrical boiler the Velox unit, supplying a primary turbine, increased the output from 10,000 to about 16,000 S. H. P.

1936 Order for two large sets of gear-wheels for marine turbine installations manufactured by another firm (Fig. 4 on page 280).

1937 Welded turbine rotor (patented) applied to marine turbines for the first time.

First merchant vessel equipped exclusively with Velox boilers (S. S. "Bore II" of the Ångfartygs A. B. Bore, Åbo)[9].

1939 Construction of two 3300 S.H.P., 165 r.p.m., geared turbine installations and the appertaining Velox boiler for 46 kg/cm² abs at 435° C, for a navy.

Design and construction at Baden of marine turbine installations with (as far as can be ascertained) highest rating per shaft in the world. (Figs. 1, 5, 6, and 10 on pages 277—283 of the present number of this journal.)

B. ELECTRICAL PROPULSION OF SHIPS.

1904 DRP 169,559 covering electrical cruising speed drive of warships propelled by turbines.

1906 Projects for the electrical propulsion of the "Dreadnought", the first ship of this class, submitted to John Brown & Co., Ltd., Clydebank. 24,000 S. H. P., 1500 and 320 r. p. m., respectively (Fig. 3).

1918 DRP 341,597 covering the electrical propulsion of ships with synchronous propeller motor operating as induction motor during starting and manœuvres.

1919 Special commission, headed by Messrs. Aichele and Hunziker, set up at Baden to study the electrical propulsion of ships.

1929 Contract for complete Diesel-electric machinery for the propulsion of the two Finnish coastal cruisers "Wäinämöinen" and "Ilmarinen", the first warships exclusively with Diesel-electric drive[10].

1931/32 Development, in collaboration with Brown, Boveri & Cie. A.-G., Mannheim, and the MAN, Augsburg, of Diesel-electric three-phase drive for high-powered ships with Diesel-generator sets operating in parallel and switched in without the usual synchronizing[11].

[1] The Brown Boveri Review 1922, p. 148, Fig. 60.
The Brown Boveri Review 1924, p. 82, Fig. 2.

[2] The Brown Boveri Review 1922, p. 149, Fig. 62.
The Brown Boveri Review 1925, pp. 96 and 97, Figs. 2 and 3.

[3] The Brown Boveri Review 1930, p. 49, Fig. 82 and p. 262 of this number.

[4] The Brown Boveri Review 1927, pp. 215—217, Figs. 1—5.

[5] The Brown Boveri Review 1931, p. 62, Figs. 125—127.
Jahrbuch der Schiffbautechnischen Gesellschaft, Berlin, 1934, pp. 123—126.
The Shipbuilder & Marine Engine Builder 1928, pp. 644—647; 1932, pp. 151 and 475.
Schiffbau, Berlin, 1928, p. 555.

[6] Jahrbuch der Schiffbautechnischen Gesellschaft, Berlin, 1939, p. 149.

[7] The Brown Boveri Review 1936, p. 69, Fig. 147.

[8] The Brown Boveri Review 1939, p. 44, Fig. 110.
Jahrbuch der Schiffbautechnischen Gesellschaft, Berlin, 1939, pp. 147 and 148, Figs. 15 and 16.

[9] The Brown Boveri Review 1938, p. 37, Fig. 68.
Jahrbuch der Schiffbautechnischen Gesellschaft, Berlin, 1939, pp. 133 to 149.

[10] The Brown Boveri Review 1933, p. 7, Figs. 11 and 12.

[11] The Brown Boveri Review 1932, pp. 148—154, Figs. 1—8.

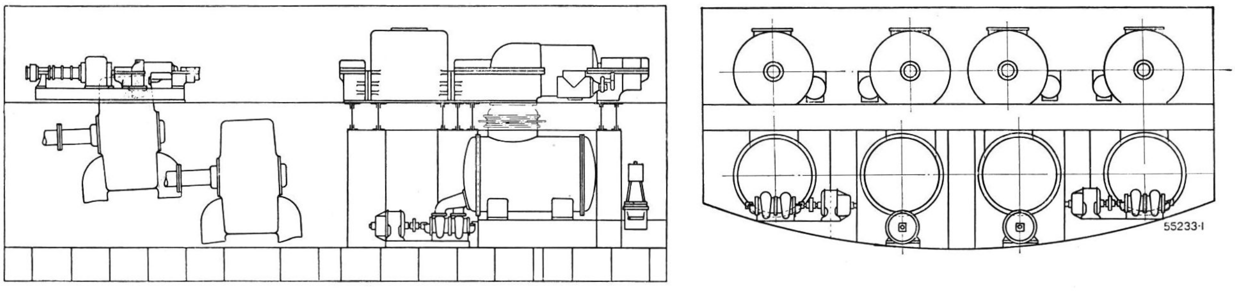


Fig. 3. — First project (in 1906) for the electrical transmission of the power from the turbines to the propeller shaft. Aggregate rating 4×5500 S. H. P.

1932 Design of Diesel-electric cruising-speed drive for the Norwegian mine-layer "Olav Tryggvason" [1]. Manufacturer: Norsk Elektrisk & Brown Boveri, Oslo.

1933 Construction of Diesel-electric drive for the paddle-vessel "Genève" [2].

1935/36 Construction of first Diesel-electric three-phase installation with generator sets operating in parallel for the 6800 S.H.P. cargo vessel M.S. "Wuppertal" by Brown, Boveri & Cie. A.-G., Mannheim.

1936 Design of Diesel-electric propelling plant for the two Norwegian scouting cruisers "Nordkape" and "Senja" with separately and self-excited Ward-Leonard generators[3]. Manufacturer: Norsk Elektrisk & Brown Boveri, Oslo.

1937 Design data evolved for the Diesel-electric propulsion of the paddle-vessel "Wien", the first vessel operating on the Danube to be equipped with Diesel-electric drive[4]. Manufacturers: Oesterreichische Brown Boveri Werke A.-G., Vienna.

1938 Construction of the first Diesel-electric tug drive with Brown Boveri current regulator for automatically adapting the propeller motor speed to the various towing duties[5].

C. MARINE AUXILIARIES.

1901 Construction of marine turbo-generators taken up. The first set was ordered in the same year for the French Navy (Etablissements d'Indret) and comprised a 280 kW turbo-generator in tandem (2×140 kW) 14 kg/cm² abs, 3000 r. p. m.

To date approximately 500 Brown Boveri turbo-generators or turbines for the drive of marine generators with an aggregate rating of over 70,000 kW have been supplied. The prime movers of the turbo-generators were pure reaction turbines until 1907.

1907 Introduction of combined impulse and reaction turbines as prime movers for turbo-generators.

1909 Construction of turbo-fans for marine boiler plants taken up[6]. To date over 400 Brown Boveri turbo-fans have been supplied.

1910 Construction of horizontal and vertical turbo-feed and other marine pumps begun.

1913 Geared turbines employed for the drive of turbo-generators (contract for twenty-four 360 kW, 3600/750 r. p. m. geared turbines for the drive of marine generators) [7].

1916 Construction of scavenging turbo-blowers for two-stroke-cycle marine Diesel engines taken up.

To date approximately 280 units have been supplied, these including the largest blower sets ever built with drive by d. c. turbo-motors [8]. The largest installation with our scavenging blowers is that of the M. S. "Saturnia" with a rating of 40,000 H. P.

1918 Series production of small impulse turbines for the drive of marine auxiliaries taken up. [9]

Development of combined condensing pump sets for high efficiency; first application 1920 to the turbine-propelled vessels "Thuringia" (later "General Artigas") and "Westphalia" (later "General St. Martin") of the Hamburg-America Line [10].

1923 Development of turbo-charging blowers with exhaust-gas turbine drive for increasing the power of Diesel-engines. The first tests were made on a Swiss Locomotive & Machine Works two-stroke-cycle engine, but were later continued in collaboration with Alfred Büchi on four-stroke-cycle engines. It was as a result of this collaboration that the Büchi Syndicate was formed in October, 1926. The first Brown Boveri turbo-charging sets with exhaust-gas turbine drive were ordered in 1927 by Messrs. Franco Tosi, Legnano, for Italian motor-ships. To-day, hundreds of four-stroke-cycle marine Diesel engines equipped with Brown Boveri exhaust turbo-chargers for increasing their power by the Büchi process are in operation all over the world [11]. The largest marine installation of this type is that on the M. S. "Reina del Pacifico" with a total rating of 22,000 H. P. (Fig. 1 on page 300).

(MS 909) *E. Klingelfuss and J. Baasch. (E. G. W.)*

[1] The Brown Boveri Review 1933, p. 7, Figs. 13 and 14.
The Brown Boveri Review 1935, pp. 211—213, Figs. 7—10.
[2] The Brown Boveri Review 1935, pp. 167—170, Figs. 1—6.
[3] The Brown Boveri Review 1938, p. 40, Fig. 72.
The Brown Boveri Review 1939, pp. 51 and 52.
[4] The Brown Boveri Review 1938, p. 39.
The Brown Boveri Review 1939, p. 52.
[5] The Brown Boveri Review 1940, p. 67, Figs. 151 – 154.
[6] BBC Mitteilungen 1920, pp. 103—107, Figs. 1—8.
[7] BBC Mitteilungen 1920, p. 250, Fig. 12.
[8] The Brown Boveri Review 1924, pp. 67—80.
[9] BBC Mitteilungen 1919, pp. 183—191 and 211—218.
[10] Werft, Reederei, Hafen, 1923, p. 495.
[11] The Brown Boveri Review 1937, pp. 175 – 190.

BROWN BOVERI TURBINES FOR THE PROPULSION OF MERCHANT VESSELS.

Decimal Index 621.125

The new merchant shipping to be built after the war will involve rapid delivery of suitable propelling machinery. Hereafter the Brown Boveri two-cylinder marine turbine with double-reduction gear for outputs up to 8000 S. H. P. per propeller shaft is described, which in virtue of its low steam consumption and reliable, robust construction is particularly suited to the propulsion of merchant steamers. Developments in the design of exhaust-steam turbine sets are also dealt with.

IT is doubtful whether any questions are meeting with more discussion in shipping circles at the present moment than the replacement of the lost merchant tonnage and the most suitable propelling machinery for the new vessels. The technical journals of maritime countries contain interesting and well-conceived contributions to these problems.

In view of the multiplicity of considerations which have to be taken into account in the construction of new vessels the problem of the most suitable propelling machinery can only be satisfactorily solved through close collaboration between the marine-engine designer and the shipbuilder, for the machinery with the highest thermal efficiency does not necessarily form the most fitting plant in every case. Apart from economic factors, various national considerations, such as employment of home fuels, often also play a decisive part in the selection of the mode of propulsion. The machinery for a cargo vessel of the tramp class will be different from that intended for a fast cargo liner. In the first case high efficiency is not so important as in the second. The plant should above all be as simple as possible and not require particularly skilled attendance. High speed is also of less significance than for cargo liners plying over a definite route where competition from other companies is continually encountered.

These fundamental questions of mode of propulsion and standardization of types of ships are beyond the scope of this article. It is only intended here briefly to outline the present state of development of the Brown Boveri low and medium power marine turbines.

Inasmuch as reciprocating steam and Diesel engines are not manufactured by Brown Boveri, while the design of steam turbines for the turbo-electric propulsion of ships does not essentially differ from that of land turbines and a separate article in the present number of this journal deals with gas turbines for marine applications, the notes hereafter will be confined to high-speed geared turbo-sets and exhaust-steam turbines for utilizing the residual energy in the exhaust steam of reciprocating engines.

I. HIGH-SPEED GEARED TURBINE SETS.

By far the greater proportion of the existing merchant shipping is still propelled by reciprocating steam engines[1], the chief reasons for this being that the reciprocating type of engine is not only extremely simple, but has attained to a high degree of reliability and, in consequence, to an extraordinarily long life. In contrast to the Diesel engine it permits any desired solid or liquid fuel to be used and for a long time was superior to the steam turbine in efficiency. The introduction of the high-speed geared turbine, however, not only offset this advantage of the reciprocating engine, but permitted the merits of turbine propulsion, e. g., possibility of employing high temperatures and pressures, oil-free condensate for feeding boilers, and maximum reliability, to be availed of and at the same time the space requirements and weight of the machinery to be reduced. In consequence, geared turbines are finding wider and wider application on ships, even for low powers.

About ten years ago[2] Brown Boveri evolved a turbine set which already incorporated the more important features of the present design, viz.—

1. Small high-speed turbines of simple and therefore reliable construction.
2. Two-stage impulse wheel as high-pressure astern turbine mounted on the pinion shaft of the h. p. turbine.
3. Double reduction gear.
4. Condensing auxiliaries and main oil pump combined into one set with double drive.

[1] Of the pre-war world merchant fleet of more than 30,000 vessels with about 70 million gross register tons, 4% of the vessels and 14% of the total tonnage were propelled by steam turbines, approximately 2/3 of all ships or 60% of the total tonnage by reciprocating steam engines, 19% of the vessels or 24% of the total tonnage by Diesel engines, while about 10% of the vessels or 2% of the total displacement, were sailing vessels, partly with auxiliary drives.

[2] See "The Marine Engineer and Motorship Builder" 1933, pp. 353—357.

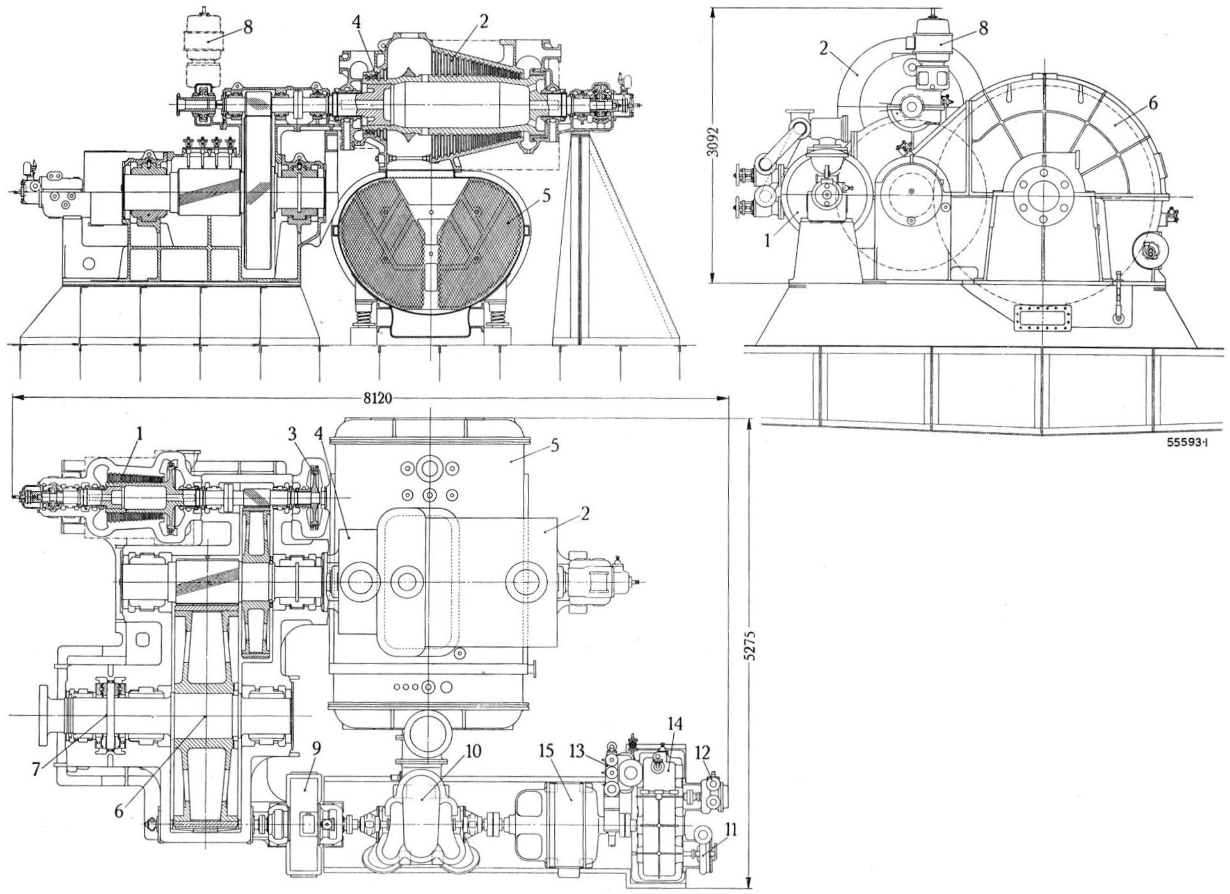


Fig. 1. — Geared turbine set, 1935 model. Output 4800 S. H. P. Propeller speed 120 r. p. m.

1. High-pressure turbine, speed 5165 r. p. m.
2. Low-pressure turbine, speed 3700 r. p. m.
3. High-pressure astern turbine.
4. Low-pressure astern turbine.
5. Condenser for cooling water at 28° C.
6. Double reduction gear.
7. Propeller thrust bearing.
8. Shaft turning gear.
9. Hydraulic coupling.
10. Circulating water pump.
11. Condensate pump.
12. Main oil pump — gear type.
13. Auxiliary turbine.
14. Reduction gear.
15. 80 kW d. c. generator.

The combination of the main turbines and auxiliaries into a single unit results in an efficient and space saving propelling set.

With the vessel under way the power for the auxiliaries is supplied by the highly efficient main machine through a hydraulic coupling, while at speeds of below approximately 60% of the normal, as well as during manœuvres, the auxiliary turbine is automatically put into service, the auxiliary machine set being simultaneously separated from the main drive through the draining of the hydraulic coupling.

Such sets were first supplied for two banana-carrying steamers in 1935 (Fig. 1). They had a normal rating of 4800 S.H.P. (maximum 5300 S.H.P.) at a propeller speed of 120 r.p.m. and when running astern delivered approximately 70% of the normal ahead rating. At sea the main turbines drive, in addition, an 80 kW d. c. generator combined with the auxiliary machine set, which supplies the ship's network. The control system will be clear from the diagram in Fig. 2.

On the basis of the operating experience made with these sets a series of turbines with ratings between 1600 and 8000 S. H. P. has been developed for the propulsion of merchant vessels. The chief characteristics of these are given in table I. Apart from the turbine and propeller speeds on which the design is based the table contains the weights of the main machine, including condenser designed for a 95% vacuum with cooling water at 15° C (North Atlantic service), auxiliary machine set, auxiliary oil pump, oil cooler, double oil filter, steam-jet air ejector, and condensate injector.

The table also gives the full-load steam consumption figures for the steam pressures and temperatures now usually corresponding to the ratings in question. In the case of the low outputs in particular the steam pressures have been so selected that the same boilers can be employed as in normal reciprocating steam engine practice. These small turbine sets can naturally also be operated with higher live-steam temperatures than given in the table, in which case the steam consumption will be correspondingly lower. For vessels

TABLE I.

Brown Boveri Marine Turbines for Merchant Steamers.

Outputs, Speeds, Steam Consumption, and Weights. Cooling Water Temperature 15 ° C, Vacuum 95 %.

Normal output S. H. P.	1600		2500		3200		4000		5000		6500	8000
R. P. M. Propeller shaft	90		90		100		110		110		125	125
H. p. turbine	7500		7200		6500		6000		5800		5500	4800
L. p. turbine	6000		5300		5000		4500		4000		3500	3250
Live steam before valve:												
Pressure kg/cm² abs	16	25	16	25	25	32	25	32	32	38	38	38
Temperature ° C	350	350	350	375	375	400	375	425	425	425	425	425
Steam consumption including auxiliary condensing machine set kg/S.H.P. h	4·05	3·85	3·95	3·65	3·60	3·40	3·55	3·25	3·22	3·20	3·13	3·10
Heat consumption including auxiliary machine set without preheating [1] kcal/S.H.P.h	2915	2750	2845	2660	2625	2510	2590	2450	2425	2405	2350	2330
Overall weight, including condenser, auxiliary machine set, and lubricating oil system, without water and oil kg	37,000		48,000		55,500		60,000		71,000		77,500	86,000
Or converted to kg/S. H. P.	23·1		19·1		17·3		15·0		14·2		11·9	10·75

[1] With single-stage preheating of the feed water to 80—100 ° C by steam extracted from the main turbine these figures are reduced by 3—4 %.

navigating in tropical waters the weight indicated in the table is increased by reason of the larger condenser entailed, while the steam consumption is also greater due to the impairment of the vacuum involved by the higher cooling water temperature. For instance, given cooling water at 30 ° C the weight is increased by about 6 % and the steam consumption by 4—5 %.

With the help of the section through a 6500 S. H. P. unit shown in Fig. 3 it is now intended to describe more exhaustively the fundamental construction of the new sets.

The *high-pressure turbine* is of the combined impulse and reaction type with a two-stage impulse wheel followed by a reaction drum. By reason of the high speed of 5500 r. p. m. few reaction rows are necessary despite the small diameter of the drum. The small blade diameter results in relatively long blades so that liberal radial clearances can be allowed without greatly impairing the efficiency. In consequence of this and the shortness of the rotor the high-pressure turbine can withstand rapid load changes with impunity and is thus particularly suitable for high steam temperatures. The thrust of the steam on the blades is virtually eliminated by the dummy piston, the small

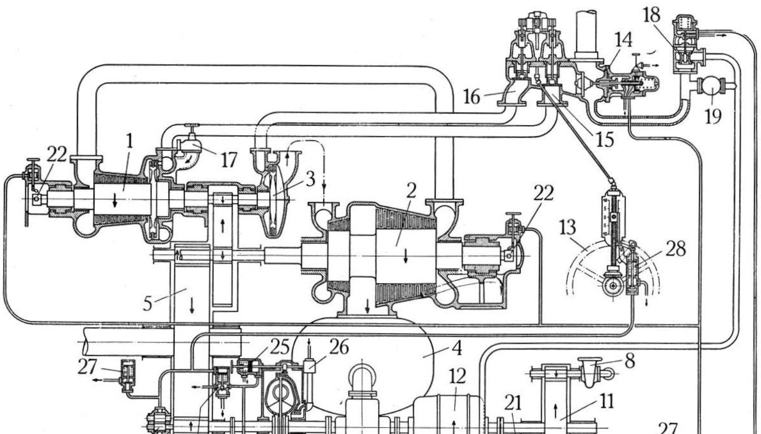


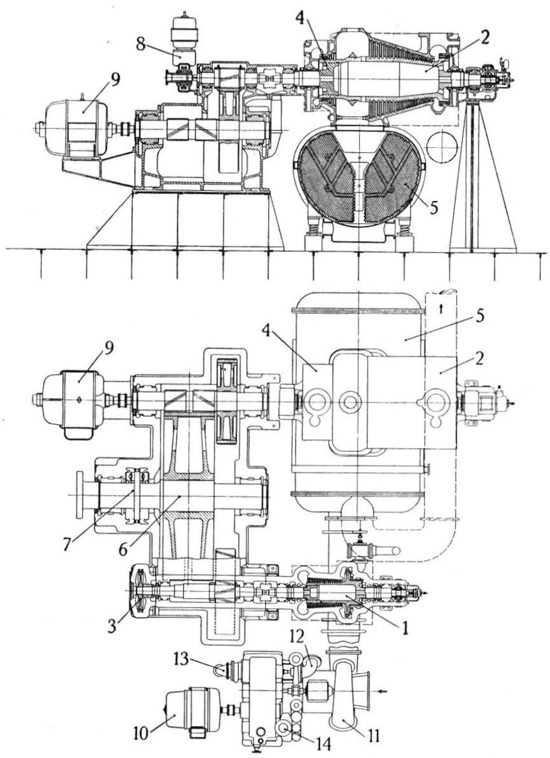
Fig. 2. — Governing diagram for 4800 S. H. P. set in Fig. 1.

1. High-pressure turbine.
2. Low-pressure turbine incorporating astern stage.
3. High-pressure astern turbine.
4. Condenser.
5. Double reduction gear.
6. Hydraulic coupling.
7. Main circulating water pump.
8. Condensate pump.
9. Main oil pump — gear type.
10. Auxiliary turbine.
11. Double reduction gear of auxiliary turbine set.
12. 80 kW d. c. generator.
13. Manœuvring wheel with direction indicator.
14. Main quick-acting stop valve.
15. Steam admission valve for ahead running.
16. Steam admission valve for astern running.
17. Manual overload valves.
18. Main stop valve of auxiliary turbine set.
19. By-pass valve.
20. Quick-acting stop valve of auxiliary turbine.
21. Steam admission valve of auxiliary set.
22. Safety governor of turbine.
23. Oil pump for hydraulic coupling.
24. Control valve for hydraulic coupling.
25. Operating piston for hydraulic coupling.
26. Oil pipe from gravity tank for filling the hydraulic coupling.
27. Oil safety valves.
28. Shut-down valve for hydraulic coupling (on operating pedestal).
29. Non-return valve with by-pass.
30. Connection to the auxiliary oil pump system.

The Brown Boveri pressure-oil governing gear enables the hydraulic coupling and auxiliary turbine to be reliably but simply cut in and out during manœuvres.

diameter of which enables the gland losses to be maintained within reasonable limits even for high steam pressures. Any residual thrust is taken up by the segmental thrust bearing of the turbine.

The *low-pressure turbine* turning at 3500 r. p. m. is of the pure reaction type, the low-pressure astern stage being located on the exhaust end of the shaft. A dummy piston and an axial thrust bearing (the latter to take up any small differences in thrust and to fix the position of the rotor) are also provided here. The l. p. cylinder is of high-class cylinder-quality cast iron, while the cylinders of the h. p. ahead and astern turbines are steel castings. Connections on the l. p. cylinder permit steam to be tapped off for feed-water preheating purposes. Live steam connections on the admission steam branches for ahead and astern running enable the voyage to be continued at reduced speed with the l. p. turbine alone in the event of the h. p. turbine being out of commission. Further connections, on the exhaust part, allow the exhaust steam of the h. p. turbines to be conducted to the condenser, so that it is possible for the vessel to be propelled by the h. p. stage alone when the l. p. turbine is not available.



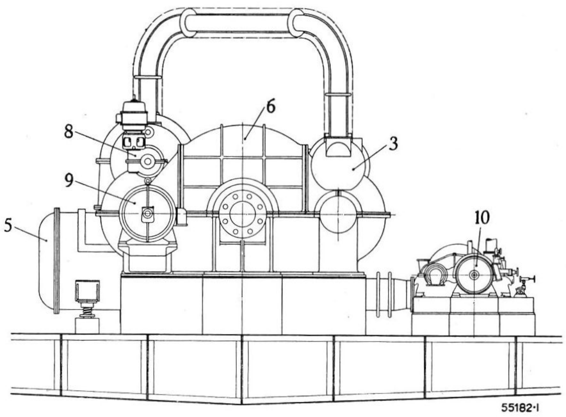


Fig. 3. — 6500 S. H. P. geared turbine set.
Propeller speed 125 r. p. m. Section and view from propeller end.

1. High-pressure turbine, speed 5500 r. p. m.
2. Low-pressure turbine, speed 3500 r. p. m.
3. High-pressure astern turbine.
4. Low-pressure astern turbine.
5. Condenser.
6. Double reduction gear.
7. Propeller thrust bearing.
8. Shaft turning gear.
9. Helux d. c. generator.
10. D. c. motor of auxiliary machine set.
11. Circulating water pump.
12. Condensate pump.
13. Main oil pump — gear type.
14. Auxiliary turbine.

Features of these new propelling machines are simplicity, economy, and reliability.

The *h. p. astern turbine* is of the impulse type with two pressure stages, the impulse wheel being located on the h. p. pinion shaft at the propeller end. This arrangement with a separate h. p. astern turbine instead of the more usual h. p. astern blading in one of the ahead turbines (in the case of three-cylinder plants usually the medium-pressure turbine) enables the gland between the ahead and astern blading to be dispensed with. This is a great advantage inasmuch as the gland in question would have to be constructed with a large clearance to take account of expansion and shaft deformation which would naturally involve additional steam losses. A source of frequent trouble is thus avoided. In our opinion this design gives maximum reliability, thus making it particularly suitable for steam with a high degree of superheat.

The turbine shafts are connected to the pinion shafts of the reduction gear through geared couplings of our own design. These allow of axial displacement, i. e., free expansion, of the turbine shafts under all operating conditions and serve to offset any slight inaccuracies in the alignment of the turbine and gear pinion shafts which may arise during erection or subsequently through distortion of the ship. Moreover, these couplings allow of displacement of the pinion shafts by the amount of the bearing play when changing from ahead to astern running or vice versa, whereby the pressure on the teeth changes in direction each time.

In contradistinction to the *gear* depicted in Fig. 1 that here has separate first-reduction wheels for the h. p. and l. p. turbines. The large pinion on the intermediate shaft, therefore, has only to transmit a little more than half of the power, so that it can be kept smaller in diameter and width. In consequence, the large wheel on the propeller shaft will also be smaller.

The first high-speed reduction-gear stage is of the single-helical type. The thrust of the teeth is compensated by the patented Brown Boveri arrangement with slightly tapered thrust collars on either side of the pinion, thus preventing any axial thrust being transmitted to the shafts. The slow-speed stage of the reduction gear is of the double-helical type, the correct axial position of the pinion of the second stage being maintained by the meshing of the teeth. The propeller thrust block combined with the reduction gear is designed for a propeller thrust corresponding to the speed of the vessel and fixes the axial position of the gear-wheel shaft.

Should the propeller speed deviate from that assumed for design purposes the set can be adapted to the desired propeller speed by modifying the ratio of the double reduction gear.

The shaft turning gear, acting on the l. p. turbine pinion, is driven by a motor. In the event of failure of the supply, however, it can be operated by hand.

A Helux d. c. generator is driven from one of the two intermediate gear-shafts running at 1000 r. p. m. A regulator maintains the voltage constant at speeds down to about 60 % of the normal revolutions. Apart from the driving motor of the condensing pump set the ship's network is also supplied by this generator. When power requirements are particularly high, e. g., on vessels carrying refrigerated cargo, a Helux generator can be mounted on each of the two intermediate shafts.

The type OV *surface condenser* with a cooling surface of approximately 440 m^2 is spring-mounted, with its axis athwartships, underneath the l. p. cylinder. The slightly corrugated tubes of 16·5/19 mm diameter are expanded in the tube plates at both ends by the special Brown Boveri process. The lower portion of the condenser shell, which is fabricated by welding, forms a hot well and also takes the float regulating gear for the closed feed-water circuit. The air is evacuated from the condenser by two steam-jet air ejectors. Under normal service conditions only one of these is in operation, but for starting or when a large amount of air has found ingress into the condenser both are employed.

The circulating and condensate pumps together with the gear-type main oil pump are combined into an *auxiliary machine set* with double drive. Under normal operating conditions the set is driven by the d. c. motor, but at low propeller speeds and during manœuvres live steam is automatically supplied to the auxiliary turbine. The electrical drive permits the auxiliary machine set to be located in the most convenient part of the engine room, which would have been impossible if direct drive from the main reduction gear had been adopted. The 1500 r. p. m. motor drives each pump at its most suitable speed through the gearing of the auxiliary set.

The small turbine [1] with speeds ranging from 8000 to 10,000 r. p. m. can utilize pressure drops down to a back-pressure of approximately 0·8 kg/cm^2 abs, in which case its steam consumption is of the order of 850 kg/h. The exhaust steam passes into the main condenser.

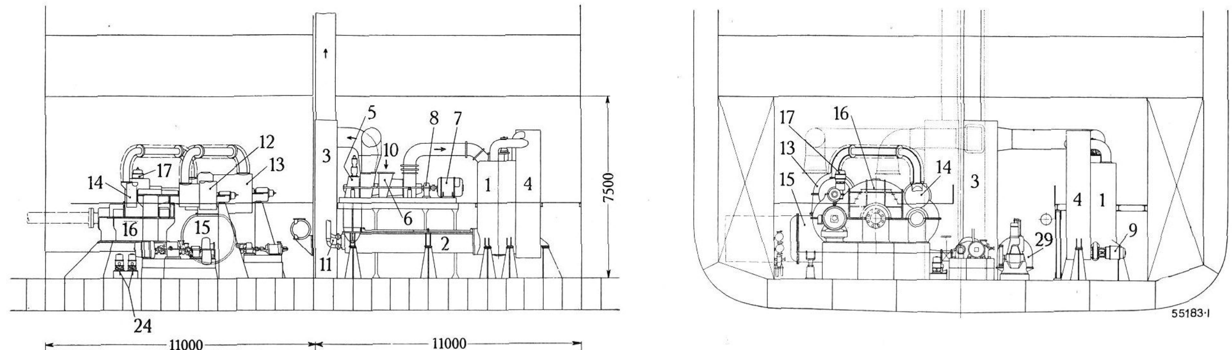
Governing Gear. — The machine set is governed by hand, similarly to the sets supplied earlier, as illustrated in Fig. 2. The Brown Boveri full-automatic marine-turbine governing gear utilizing oil under pressure[2], as supplied for larger machines, can, however, also be employed to advantage here.

The quick-acting closing valve provided before the steam admission valves for ahead and astern running is opened by oil under pressure. As long as the auxiliary machine set is out of commission and the oil pump not furnishing an adequate pressure no steam can be admitted to the main machine. At an overspeed of about 5 % a speed limiter comes into action and closes the main stop valve so far that a further rise in the turbine revolutions and, in consequence, shutting down of the plant through one of the safety governors set to operate at 10 % overspeed, are prevented. The safety governors only trip out in cases of real emergency and not when temporary speed increases occur, such as when the propeller rises clear of the water in very rough seas.

The new series of turbines of which the 6500 S. H. P. set is shown in section and described, cover the power range from 1600 to 8000 S. H. P. In the case of twin-

[1] E. Klingelfuss and R. Schmid: "Turbine-driven Marine Auxiliaries", page 290 of present number of this journal.

[2] E. Klingelfuss and V. Tödtli: "Control and Safety Features of Brown Boveri Marine Turbines", page 262 of present number of this journal.



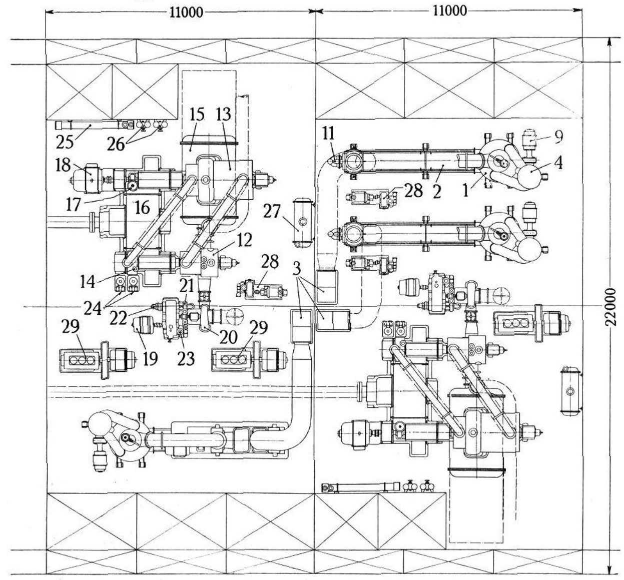


Fig. 4. — Velox-boiler and geared-turbine installation for a passenger steamer. Output 2×8000 S. H. P. Propeller speed 125 r. p. m. Three Velox-boiler sets each with an evaporation of 32 t/h, 40 kg/cm² abs, 415 °C.

1. Combustion chamber.
2. Evaporator and superheater shell.
3. Feed-water preheater.
4. Water separator.
5. Gas turbine.
6. Blower.
7. Starting motor.
8. Gearing.
9. Circulating pump.
10. Air inlet.
11. Live steam connection.
12. High-pressure turbine.
13. Low-pressure turbine incorporating l. p. astern stage.
14. High-pressure astern turbine.
15. Condenser.
16. Double reduction gear with built-on propeller thrust bearing.
17. Shaft turning gear.
18. Helux d. c. generator.
19. D. c. motor of auxiliary machine set.
20. Main circulating water pump.
21. Condensate pump.
22. Main oil pump — gear type.
23. Auxiliary turbine.
24. Motor-driven auxiliary oil pump.
25. Oil cooler.
26. Change-over double oil filter.
27. Feed-water preheater.
28. Turbo feed pumps.
29. Diesel-driven generator sets.

The compact construction of the Velox boiler enables it to be located in the engine room.

screw vessels (when the condenser is located with its axis running from stem to stern instead of athwartships under the l. p. turbine, according to space conditions) this range is extended to 16,000 S. H. P. These turbine sets can therefore be employed not only for cargo vessels and tankers, but also for small and medium-size passenger steamers.

The new turbines have been designed in special consideration of the present difficulties encountered in obtaining material, which it is to be assumed will continue for some considerable time. The relatively high turbine speeds and the double-reduction gear give a low overall weight and, moreover, small forgings. The small, simple turbine rotors can be supplied by the steelworks at short notice, while the slight stresses permit low-alloyed steels to be employed.

The chief advantages of this modern type of turbine, i. e., low overall weight, small space requirements, and high efficiency at full and fractional loads, are particularly pronounced in conjunction with a Velox boiler plant, for then not only the machine, but the whole plant will take up a minimum of space and have the lowest possible weight. Figs. 4 and 5 illustrate a twin-screw plant with Velox boilers which has an aggregate rating of 16,000 S. H. P.

With steam at 38 kg/cm² abs and 400 °C at the turbine admission valve (40 kg/cm² abs, 415 °C at the boiler) and cooling water at 15 °C the steam consumption at full-load, i. e., 2×8000 S. H. P., including auxiliary machine sets, steam-jet air ejectors, turbo-feed pumps, and preheating of feed water to 80 °C with steam from the l. p. turbine, is $3 \cdot 4$ kg/S. H. P. h,

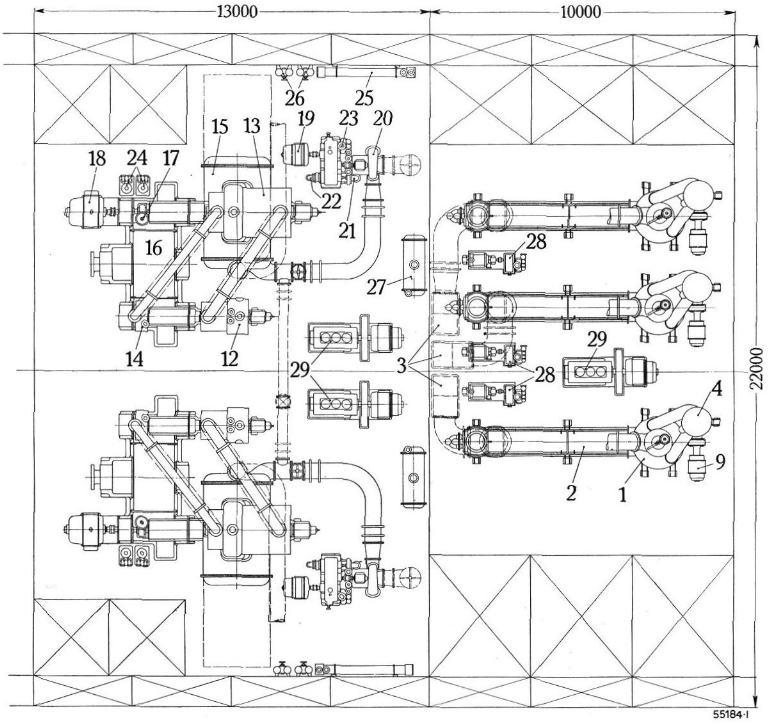


Fig. 5. — Same plant as in Fig. 4, but with separate engine and boiler rooms.
Denomination same as in Fig. 4.

New reciprocating engine plants will be provided from the beginning with exhaust-steam turbine sets to enhance the efficiency. It is for this reason that new models of exhaust-steam turbines have been developed which are characterized by particularly low weights.

About ten years ago the S. S. "Ammon" and S. S. "Amasis" of the Hamburg-America Line and the S. S. "Blitar" of the Rotterdam Lloyd Line were subsequently fitted with Brown Boveri exhaust-steam turbines to improve the efficiency. These sets were described in detail elsewhere at the time[1], while operating results were also subsequently published[2]. A special feature of these exhaust-steam turbine sets is the ring-spring coupling connect-

[1] The Shipbuilder and Marine Engine-Builder, 1932, page 151.

[2] The Shipbuilder and Marine Engine-Builder, 1932, page 473.

or 54,400 kg/h. With this load two of the three 32 t boilers are in service and attain an efficiency of 92·5 %. This corresponds to a full-load oil consumption of 256 g/S. H. P. h or a thermal efficiency of 24·7 %.

The Velox plant for the above efficiency, comprising three boilers and including the feed pumps, weighs 130 tons in working order. The turbine installation, including condensing plant (filled with water), lubricating oil system (filled with oil), feed water preheater, pipes between boilers, turbines, and Diesel-driven generators, but excluding the propeller shaftings and propellers, has a weight of 232 tons. The overall weight is thus 362 tons or 22·6 kg/S. H. P.

A similar turbine set already supplied on several occasions, but of lighter construction and for a higher propeller speed, is shown in Fig. 6.

II. APPLICATION OF EXHAUST-STEAM TURBINES TO RECIPROCATING ENGINES.

Notwithstanding its inferiority on the score of fuel consumption the reciprocating steam engine will continue to be used as propelling machine for new ships with ratings up to 3000—4000 H. P., alongside the geared turbine and Diesel engine.

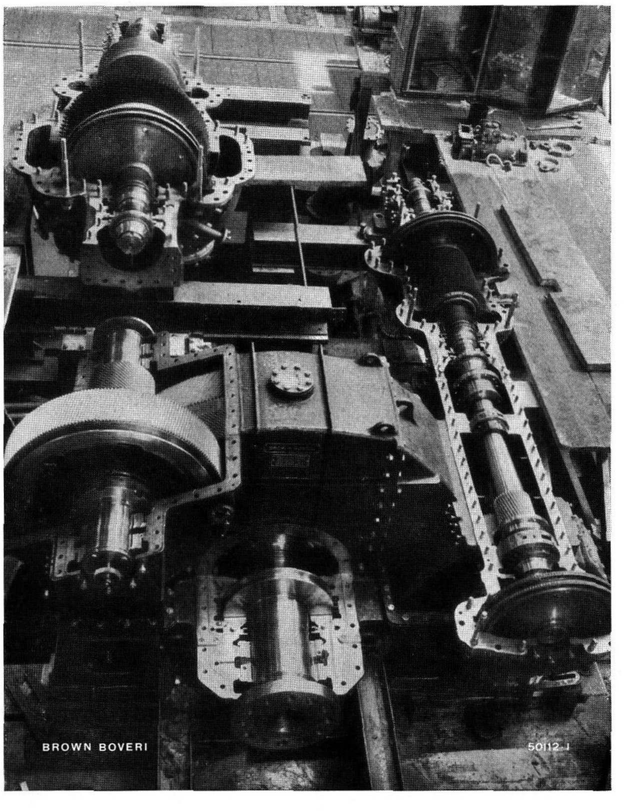


Fig. 6. — Geared-turbine set during assemby on test bed.

Normal output 3000 S. H. P., propeller speed 160 r. p. m., live-steam pressure 46 kg/cm² abs, temperature 435 °C. Lightweight design with gear casing fabricated by welding. Weight of entire set including complete condensing plant, feed-water preheater, spare parts, and tools 30,600 kg.

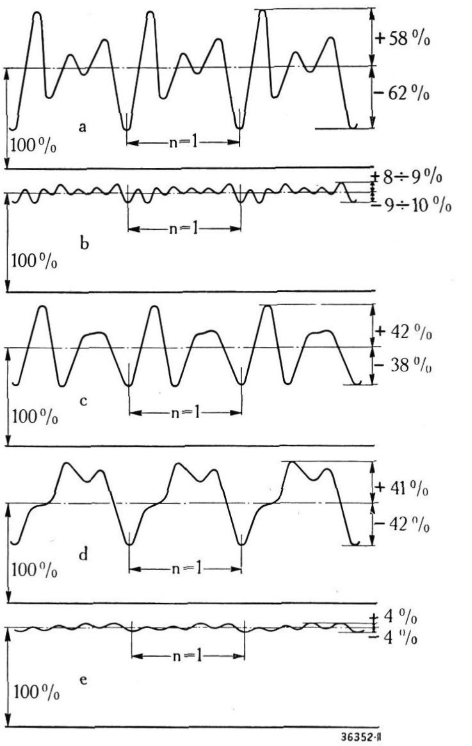


Fig. 7. — Torque curves of ships' propeller shafts.

a. S. S. "Cap Norte". Triple-expansion reciprocating steam engine before conversion, 2850 B. H. P., 82 r. p. m. (Werft, Reederei, Hafen, 1st September, 1931).
b. S. S. "Ammon". Triple-expansion reciprocating steam engine with Brown Boveri exhaust-steam turbine, 2800 B. H. P., 75 r. p. m.
c. S. S. "Cap Norte". Triple-expansion reciprocating steam engine with exhaust-steam turbine supplied by another firm, 2850 B. H. P., 82 r. p. m. (Werft, Reederei, Hafen, 1st September, 1931).
d. S. S. "Palembang". Triple-expansion reciprocating steam engine with exhaust-steam turbine supplied by another firm, 4150 B. H. P., 77 r. p. m. (Het Schip, 14th August, 1931).
e. S. S. "Wangoni". Blohm & Voss geared turbine 3000 S. H. P., 80 r. p. m.

Brown Boveri exhaust-steam turbines offset to an extremely high degree the torque fluctuations on the propeller shaft due to the reciprocating engine.

ing the reciprocating engine and the exhaust-steam turbine which remain mechanically coupled even during manœuvres and astern running. The periodical torque peaks of the reciprocating steam engine occurring during each revolution of the propeller shaft are taken up by the ring-spring coupling and are thus not transmitted to the reduction gear. The gyrating masses of the highly geared-down exhaust-steam turbine and of the gearwheels have a big compensating effect on the torque of the propeller shaft. As will be clear from diagram b in Fig. 7 the combination of reciprocating steam engines with our exhaust-steam turbine sets results in a practically uniform torque on the propeller shaft similar to that inherent in geared-turbine plants. This is in marked contrast to other systems. Fig. 8 is a further illustration of the set supplied for the S. S. "Blitar", which has proved so satisfactory in more than ten years' service.

Fig. 9 depicts an exhaust-steam turbine set of the new type for an aggregate rating of 4000 I. H. P. and a propeller speed of 100 r. p. m. Many of the components which have already proved their worth in the earlier plants, e. g., ring-spring coupling, steam change-over valve, etc., have been retained in the new designs. The chief features of the new model are:

1. Direct transmission of power developed by reciprocating steam engine through ring-spring coupling and gear-wheel shaft to the propeller shaft.
2. Steam change-over valve on admission branch of turbine. This either conducts the exhaust steam of reciprocating engines to the blading or bypasses it through the turbine exhaust branch to

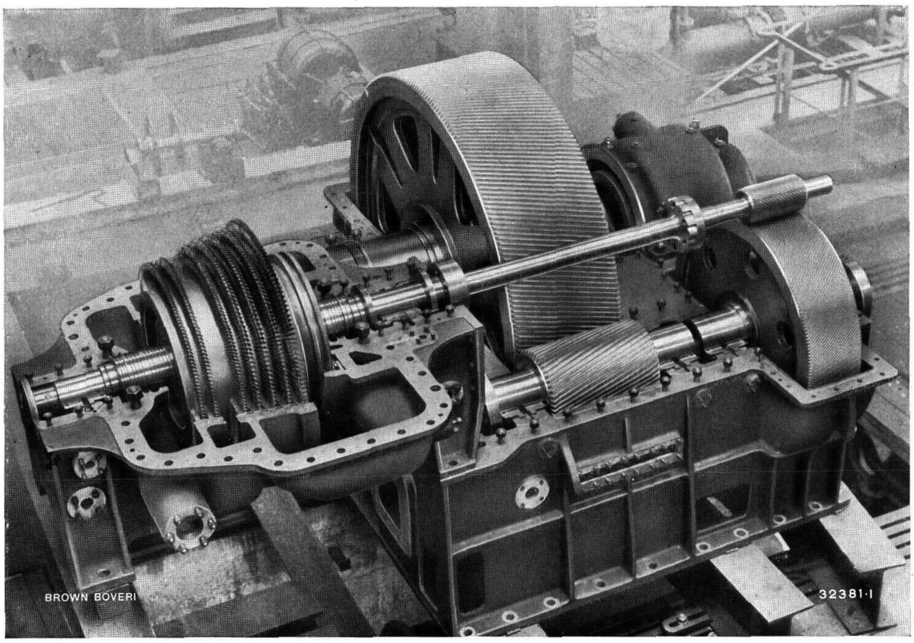
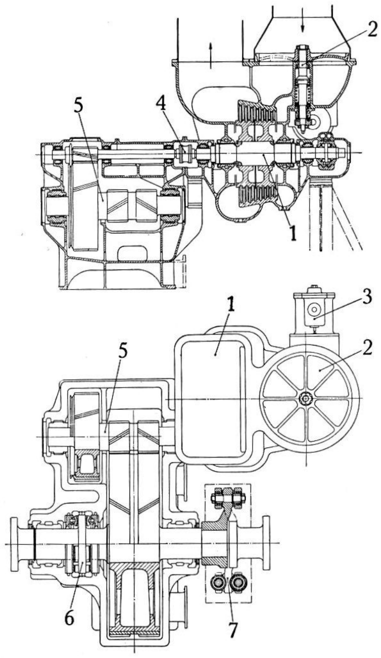


Fig. 8. — Brown Boveri exhaust-steam turbine for a reciprocating engine installation with an aggregate rating of 4700 I. H. P.

Output of turbine approximately 1200 H. P. Speed 3420/86 r. p. m. 1931 model.



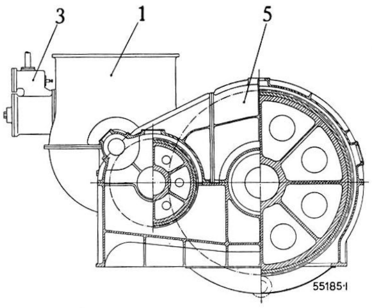


Fig. 9. — Brown Boveri exhaust-steam turbine for a reciprocating engine installation with an aggregate rating of 4000 I. H. P.

1. Exhaust-steam turbine.
2. Rotary valve for changing over exhaust steam of reciprocating engine to the turbine or condenser.
3. Driving unit of steam change-over valve.
4. Geared coupling.
5. Double reduction gear.
6. Propeller thrust bearing.
7. Ring-spring coupling for connecting reciprocating engine shaft to gear-wheel shaft.

Experience made with sets already supplied has enabled the new design to be made lighter and simpler.

the condenser, thus simplifying the piping. The valve, which is controlled by oil under pressure from the engine reversing gear, can be assembled and tried out with the engine in our workshops.

3. Geared coupling allowing of axial displacement between turbine and pinion shafts.

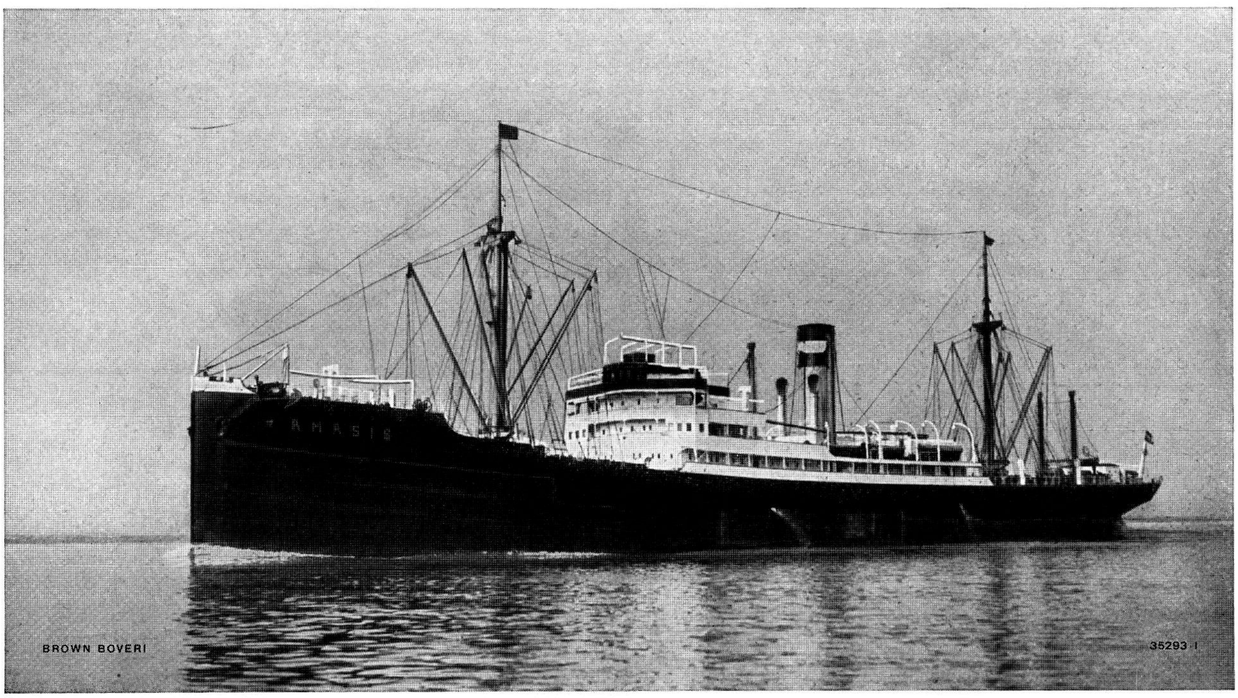
4. Reduction gear with single-helical first-reduction train and double-helical second-reduction train, a thrust collar taking up the tooth thrust in the first-reduction stage. The axial position of the intermediate shaft is maintained by the teeth of the large wheel, that of the latter by the propeller thrust block. The wheel of the first gear train, which is mounted on the intermediate shaft, fixes the axial position of the turbine pinion and the thrust collar. The intermediate shaft is of the solid type and runs in two bearings.

The live-steam manœuvring valves fitted on the first plants supplied are no longer provided, inasmuch as experience has proved sufficiently short reversing times to be possible without these.

Special design features have enabled the weight of the complete exhaust-steam turbine set, totalling 57—60 tons in the case of the first plants constructed, to be reduced by about 25 %, without affecting reliability. The set depicted in Fig. 9 weighs only about 45 tons.

(MS 894) *C. G. Wahl. (E. G. W.)*

S. S. "Amasis" of the Hamburg-America Line with Brown Boveri exhaust-steam turbine.



SOME REFLECTIONS ON THE PROPULSION OF SHIPS BY MEANS OF COMBUSTION TURBINES.

Decimal Index 629.12—843.8

The exhaust-gas and combustion turbine installations built by Brown Boveri have proved their reliability in continuous service. In planning new ships, this novel form of propulsion should, therefore, be given earnest consideration. In this article, it is shown for what type of service advantages are to be expected in the light of the present state of development of the combustion turbine.

DURING the last few decades, the reciprocating steam engine, the Diesel engine and the steam turbine with their advantages and disadvantages have between them shared the widely ramified domain of ship propulsion. Recently, attention has been focussed on the combustion turbine because of its simple design, not equalled by that of any other form of propelling machine. The principle upon which this prime mover operates, consists in burning fuel in air, in using the air as a heat carrier, thus avoiding any supplementary working medium such as water and steam, and in transforming the heat into mechanical energy in a turbine. Such a solution renders both steam generator and condensing plant superfluous, and eliminates the out-of-balance forces set up by reciprocating machine parts. The simplest form of combustion turbine consists of a turbo-compressor, of a combustion chamber and of a gas turbine (Fig. 1). A small fraction of

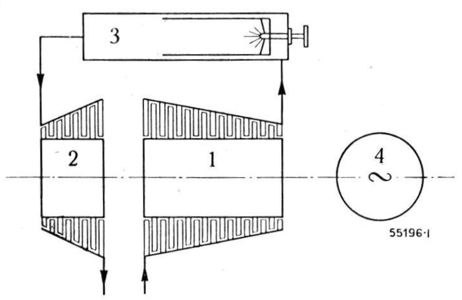


Fig. 1. — Diagram of a combustion turbine.

1. Compressor. 3. Combustion chamber.
2. Gas turbine. 4. Generator.

The air delivered by the compressor 1 is heated to 600° C by internal combustion in the combustion chamber 3. The power absorbed by the compressor, together with the useful power, is generated by the hot gases during their expansion down to atmospheric pressure in the gas turbine 2.

the compressed air serves for combustion of the fuel which is injected in the form of a fine spray. The remainder of the air is mixed with the hot gases to lower the temperature sufficiently to ensure absolute reliability of the turbine and in particular of the blades. The useful power is the difference between the power developed by the turbine and that absorbed by the compressor. Up to a few years ago, the economical realization of the principle outlined above was not possible. The attainable efficiencies of both compressor and turbine were too low and the creep strength of blade steel was too small, so that the admissible gas temperatures did not enable a sufficient power output to be obtained. The magnitude of the temperature at the turbine inlet, in particular, is of the foremost importance for the thermal efficiency and for the size of the combustion turbine. With given efficiencies for the compressor and for the turbine, the useful power output increases with the temperature. Fig. 2 shows how considerable is the increase in power output brought about by an increase in temperature.

Brown Boveri have been connected with the development of the combustion turbine over a period of thirty years and have done pioneer work in this field already at the time when results were not at all encouraging.[1] The development of the combustion turbine was considerably assisted by the favourable experiences ob-

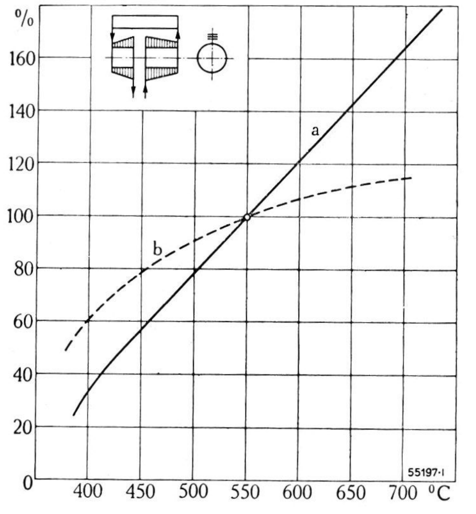


Fig. 2. — Combustion turbine without air preheater.

Useful output and thermal efficiency as a function of the temperature at the turbine inlet. The curves show clearly how the useful power and the thermal efficiency increase with increasing temperature in front of the gas turbine.

a. Useful output. b. Thermal efficiency.

tained in continuous service with exhaust-gas turbine driven supercharging units for Diesel engines, for Velox steam generators and for chemical processes. The fruits of this varied and lengthy development work were the 4000 kW combustion turbine for the Electric Supply Co. of Neuchâtel and the 2000 H.P. gas turbine locomotive for the Swiss Federal Railways. These two plants represent the first really serviceable com-

[1] W. G Noack: Pressure-charging, Velox Boiler and Gas Turbine. A Review of their Origin and Development by Brown Boveri. The Brown Boveri Review 1941, p. 183.

bustion turbine installations in the world. The first mentioned 4000 kW unit is installed in a stand-by power station of the Neuchâtel Electricity Works, where the question of fuel consumption is unimportant, as the unit operates only a few hours during the year. Attention was therefore given mainly to simplicity and cheapness of the plant, which, accordingly, is not provided with any arrangement for recuperating the heat in the exhaust gases of the turbine, in order to save the heat exchanger. The fuel oil consumption at full load is 350 g/H.P. h, corresponding to a thermal efficiency of 18·1 %.

In the case of the 2000 H.P. gas turbine locomotive, a small heat exchanger was incorporated. The limited space available made, however, a compromise necessary, with a consequently unfavourable effect on the efficiency. In spite of this, the fuel oil consumption at the best point, namely at $^3/_4$ load, is 355 g/H. P. h, corresponding to a thermal efficiency of 17·8 %.

If a turbine unit such as that of Neuchâtel, with which no compromise has had to be made, is provided with a heat exchanger, the fuel oil consumption may be reduced to 315—290 g/H. P. h, corresponding to efficiencies of 20—22 %. Such plants can already to-day bear comparison with good steam and Diesel installations.

Further improvements over and above these values are attainable by carrying out the compression in two stages with intermediate cooling and by effecting the expansion in two stages with intermediate reheating. In this manner the fuel oil consumption may be further reduced to 265 g/H. P. h and the efficiency raised to 24 %. This means, however, sacrificing something of the extreme simplicity of the first gas turbine installation.

Parallel to its further development from the thermodynamic point of view, efforts are also being made to open up new application fields for the combustion turbine. Ship propulsion appears to offer promising possibilities in this respect.

The opinion of an American naval expert [1] on this question is very illuminating. This expert believes that, because of its simplicity and because of the absence of the boiler, condenser and reciprocating parts, the combustion turbine will in due course become the most reliable form of propulsion for ships of the future. At the conclusion of his statement, he expressed the opinion that already in its present state of development, the combustion turbine is suitable for use in ships.

The application of the combustion turbine to ship propulsion brings up a number of interesting problems. These concern mainly the question of speed control and the necessity of providing means for reversing. In its simplest form, consisting of a compressor, a turbine, a combustion chamber and an air heater, the speed of the combustion turbine may be regulated down to 30—40 % of the normal value, but the propeller shaft cannot be reversed for going astern, without the use of special devices. A possibility of speed regulation and of reversing is, however, offered by the variable pitch propeller. By varying the angle incidence of the propeller blades, any speed between full speed ahead and full speed astern may be obtained, with constant speed of the propelling machine. This mechanical element has been taken over from the Kaplan turbine, which latter has been built in sizes up to 8 m diameter and for powers up to 80,000 H. P. and has proved its reliability under widely different operating conditions. So far, some fifty ships have been provided with this form of propeller in powers up to 3500 S. H. P. and diameters up to 4·5 m. After minor initial troubles, the adjustable pitch propeller has, even during the last three severe winters, given full satisfaction in Swedish ships.

If it is preferred not to use an adjustable pitch propeller, the main shaft must be driven by means of a separate "power" turbine with built-in reversing stage, that is to say, incorporating a feature well known in ordinary marine steam turbine practice. The gas turbine of the charging set has, with this arrangement, only to supply the power absorbed by the compressor. If a comparison is made with the steam turbine plant, the combustion chamber and the charging set must be regarded as taking the place of the steam boiler, because they supply the necessary motive medium for the propelling turbine.

When examining the combustion turbine in regard to its suitability for ship work, a fundamental distinction should be made between merchant and naval vessels. In naval vessels, the space required for the machinery, the weight per unit of power of the driving installation and simplicity of operation are of decisive importance. The fighting value of a ship is largely affected by this last factor, since with complicated installations, the effects of an engagement on individuals of the crew may lead to faulty operation. For this reason the manœuvres have to be so well rehearsed that they are carried out entirely automatically without the necessity of thinking. A simple plant facilitates this task and allows a scratch crew quickly to become familiar with its duties. The efficiency of the propelling machinery is of importance only during cruising, when it affects the radius of action of the ship.

[1] Lybrand-Smith, Captain, U.S.A. Navy: Contribution to a discussion "Transactions of the American Society of Mechanical Engineers", February, 1941.

In the case of freight and passenger vessels the profits, that is, the difference between the takings and the expenditure, represent the deciding factor. Low weight and low space requirements are advantageous as far as the takings are concerned, whilst the operating costs depend largely on the outlay for fuel. In addition, attendance and maintenance also appreciably affect the final balance. The following considerations will serve to show how far the combustion turbine is, already to-day, able to satisfy these varied requirements.

1. *Naval Vessels.*

It has been mentioned that in the simplest form of combustion turbine, the speed can be varied within limits which, with an adjustable pitch propeller, are sufficient for all practical purposes. Reversing also becomes possible by means of the variable pitch propeller. On the other hand, at reduced loads, a relatively high fuel consumption will have to be reckoned with. Unless the ideal simplicity of the unit is to be sacrificed, other means will have to be looked for to enable cruising to be effected without involving the use of the combustion turbine. As an economical solution of this problem, it has been suggested to employ a Diesel engine for cruising. According to the type of vessel and to the necessary propelling power, such Diesel engines may be connected to auxiliary shafts, or they may drive the propeller shafts through disengageable couplings. In this manner, a low fuel consumption is possible during cruising, while the combustion turbine serves exclusively as a low-weight peak-load machine for obtaining full speed with a favourable fuel consumption of about 400 g/S.H.P. h. The weight of a combustion turbine installation of 10,000 S.H.P. inclusive of the reduction gear and the propeller thrust bearing is 7—8 kg/S.H.P., so that a considerable saving of weight can be achieved over other forms of drive. For comparison purposes, it may be mentioned that the main driving machinery of modern destroyers weighs 12—15 kg/S.H.P., whilst for torpedo boats 8—11 kg/S.H.P. have to be reckoned with. For large ship units, combustion turbines are not likely to enter into consideration at present, because the highest power now obtainable in a single-stage combustion turbine unit is of the order of 10,000 H.P. The combustion turbine will, therefore, for the time being, serve for propelling small fast units, such as large speed boats, torpedo boats and destroyers.

2. *Merchant Vessels.*

The prospects of the combustion turbine for merchant vessels are very favourable, as the available space allows the use of air heaters and of two-stage compression and combustion. The normal service power requirements are of the same order as those for which the unit is designed, so that the latter operates in the neighbourhood of the point of best efficiency.

As already mentioned, economical operation is of the greatest importance for the application of the combustion turbine to merchant vessels. The different types of marine drive employed at present have fuel consumptions approximately as follows, whereby it should be mentioned that Diesel engines necessitate fuel oil costing approximately twice as much as that used for steam or gas turbine plants: [1]

Reciprocating steam engines	370—420	kg/S.H.P. h
Steam turbine plants	275—350	,,
Diesel engine plants	165—180	,,

These figures refer to the output at propeller shaft, inclusive of the auxiliaries used in normal service, and assuming the fuel to have a net calorific value of 10,000 kcal/kg. Compared with the above figures, the fuel consumption of combustion turbines with two-stage compression and combustion, is 280—320 g/S. H. P. h, so that competition with steam turbines becomes possible. The combustion turbine may even compare favourably with Diesel engines, where the above-mentioned difference in price between Diesel oil and bunker oil compensates the higher efficiency of the Diesel engine.

[1] Bleicken: "Hansa", Deutsche Schiffahrtszeitschrift, No. 17, 1941.

TABLE I.

Type of drive	Year of construction	Steam kg/cm² abs/°C	Speed of machinery and propeller r. p. m.	Weights Boiler t	Weights Main Machinery t	Total t	Specific weight kg/S.H.P.	Fuel consumption kg/S.H.P. h
Scotch type boilers, turbines .	1927	16/320	2200/110	430	160	590	95	0·4
La Mont boilers, turbines . . .	1938	36/450	7000/130	140	97	237	38·2	0·31
Diesel, double-acting, two-stroke compressorless	1938	—	105	—	530	530	85·5	0·177
Gas turbines	—	—	6500/110	—	90	90	14·5	0·290

Comparison between different forms of propulsion for a 10,000 t freight ship of 6200 S. H. P. with a speed of 16 knots.

The combustion turbine results in the lowest weight of machinery.

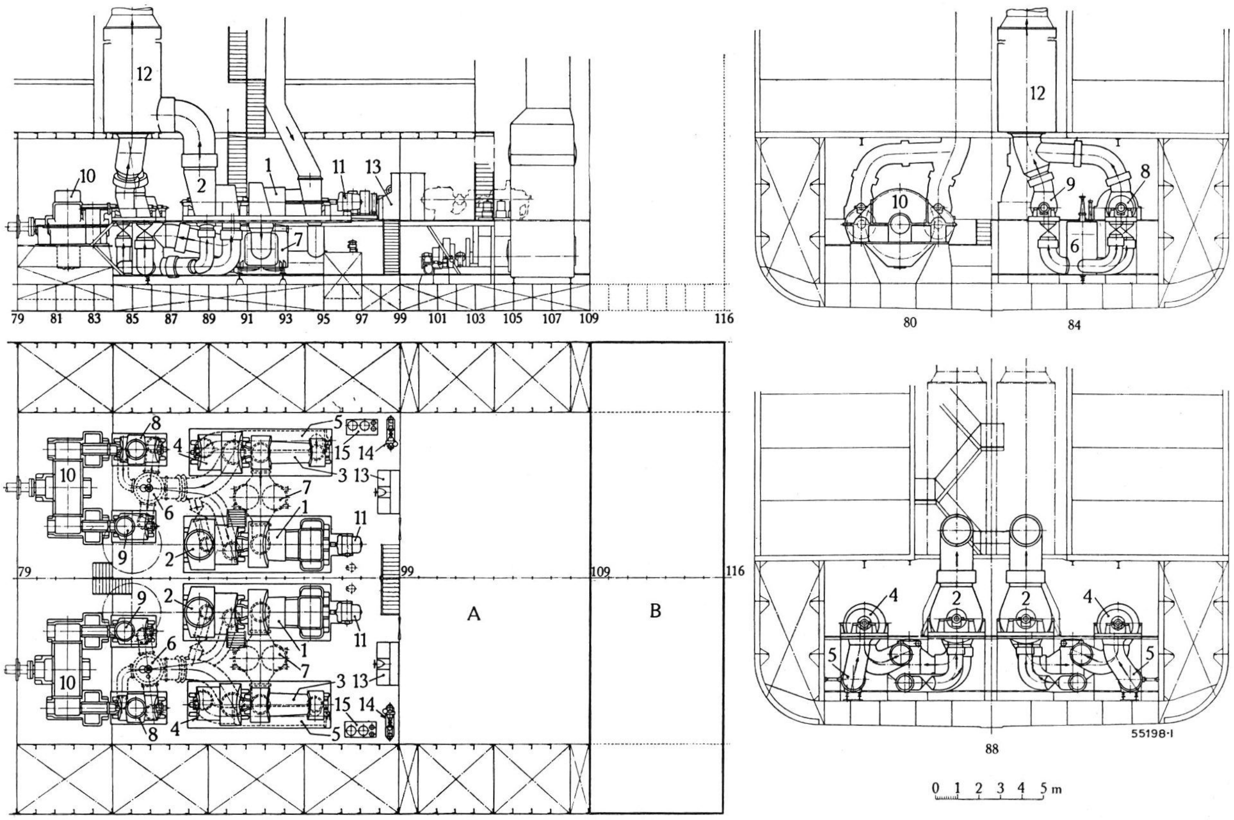


Fig. 3. — Fast liner with gas turbine propulsion, 2×7100 S. H. P., 125 r. p. m.

A. Auxiliaries. B. Space saved.

1. Low-pressure compressor.
2. Driving turbine of the low-pressure compressor.
3. High-pressure compressor.
4. Driving turbine of the high-pressure compressor.
5. High-pressure combustion chamber.
6. Low-pressure combustion chamber.
7. Air cooler.
8. Useful power turbine with built-in reversing stage.
9. Useful power turbine.
10. Reduction gear.
11. Starting motor.
12. Exhaust heat boiler.
13. Control desk.
14. Fuel pump.
15. Fuel heater.

Even when compared with a modern high-pressure water-tube boiler plant, there is a considerable saving in space, amounting to about 6 metres in length.

The useful power developed by the combustion turbine may be transmitted to the propeller shaft either mechanically or electrically. In the case of mechanical drive, it is advantageous to employ an adjustable pitch propeller. With fixed propeller blades, the propeller shaft is driven through a reduction gear by means of a separate turbine with built-in reversing stage. The electrical transmission requires at the reversing point corresponding to about 40 % speed, a torque which cannot be delivered by the simple type of combustion turbine. The electrical generator must, therefore, be driven by a separate "power-turbine." Since reversing is effected electrically, a reversing stage is superfluous. With both forms of drive the manœuvres can be effected quickly and easily.

The advantages offered in regard to weight and space requirements by the combustion turbine propulsion of ships, are shown in table I and Fig. 3.

Table I is an informative comparison between machine weights of different types of propulsion. The combustion turbine alternative is based on a plant with two-stage compression and combustion. It is seen that this solution enables a considerable saving in weight to be achieved over all other forms of drive, whilst retaining the conventional design of ship used for merchant vessels. In certain types of ship, space requirements are more important than a specially light propelling installation. The project studies illustrated in Fig. 3 show that in this respect also the combustion turbine offers advantages. The projects are based on the S. S. "Pretoria" of the German-Africa line. The gain in space is all the more remarkable, in that the comparison is made with a steam turbine plant operating with high-pressure steam at 80 kg/cm^2 g and 480^0 C, which already results in a compact plant. The fuel consumption of 290 g/S.H.P. h corresponds to a thermal efficiency of 21·8 % and bears comparison with any economically operating steam turbine plant.

It can be deduced from the above considerations that the propulsion of ships by means of combustion turbines, already to-day, is able to compete with regard to economy with other existing forms of propulsion. The neat layout, the simplicity and security of the propelling machinery will secure for it a wide application in the future.

(MS 888) *R. Schmid. (Hv.)*

THE DIESEL-ELECTRIC THREE-PHASE PROPULSION OF HIGH-POWERED SHIPS.

Decimal Index 629.12—833.6

Nowadays, in those cases where the choice falls on Diesel engines as the most fitting means of propelling a ship, the electrical method of transmitting the power to the propeller shaft must also be given due consideration, inasmuch as the last six years have seen great developments in the field of Diesel-electric ship's propulsion. The view that the usefulness of the Diesel-electric drive was confined to special low-powered ships has been abandoned and a series of large cargo and passenger vessels fitted out with Diesel-electric three-phase propelling equipment. This new method of propulsion has eminent advantages over the drive of the propeller shafts by direct-coupled or geared Diesel engines. In what these consist is described from the modern point of view in the following article.

UNTIL a few years ago Diesel-electric drive was employed exclusively for special vessels of relatively low power where for certain reasons electrical transmission afforded particular advantages. With these ships either extremely flexible operation and good manœuvrability from the bridge were essential (tugs, ferries, ice-breakers, etc.) or reliability and economical operation at various ship's speeds were specified (passenger vessels on rivers, small warships) or the auxiliary power demand was high in proportion to that for propulsion and could be readily met by the main Diesel-generator sets when the vessel was stopped or running at reduced speed (dredgers, special cargo vessels, fishing boats). Direct-current transmission was adopted in these cases on account of the simplicity with which the speed of the propeller motor can be regulated in either direction, for it involves neither variation of the revolutions and reversal of the Diesel generator sets, nor switching operations in the main circuit. For ships with high shaft-horse-power ratings, however, d. c. transmission is uneconomical, inasmuch as the voltage is limited, and heavy and expensive machines with large commutators and bulky cables are entailed. In the case of ships with over approximately 3000 S. H. P. per shaft, therefore, the three-phase system, using simpler machines without commutators and operating at higher voltages, has a clear field. Taking the example of a 6000 S. H. P. single-screw vessel, the Diesel-electric d. c. drive is about 45 % heavier and more expensive than three-phase transmission with machines operating at round about 3000 V. Moreover, the d. c. system has an approximately 4 % lower efficiency.

In order to keep the price, weight, and space requirements down to a minimum, the Diesel-electric three-phase drive — in contradistinction to turbo-electric propulsion where even for high powers only one high-speed generator set per propeller shaft is provided — must be divided into a number of Diesel-alternator sets running at as high a speed as possible and connected in parallel to a busbar system feeding the propeller motors. This arrangement permits individual Diesel-alternator sets to be brought in or dropped out according to the desired speed of the vessel. The numerous advantages accruing from the splitting up of the primary power are partially offset by the fact, however, that a change to a higher ship's speed involves synchronization of the alternator or alternators to be brought in before paralleling with the machines already in operation. The presumed complication of manœuvres entailed thereby chiefly explains the initial reluctance of shipowners to adopt the three-phase drive with alternators operating in parallel.

The merits of the Diesel-electric system of ship's propulsion with high-speed three-phase alternator sets operating in parallel were recognized by Brown Boveri at an early date, a system having been evolved more than ten years ago, whereby the individual alternator sets could be paralleled in a simple manner without synchronizing gear. This patented process is based essentially on the discovery that three-phase alternators with heavy damping windings and turning at approximately the same speed when thrown on to the busbars unexcited, fall into synchronism rapidly without undue current surges upon being simultaneously excited (see oscillogram, Fig. 1). The workability of the system was proved, as early as 1931, by exhaustive tests carried out with up to six parallel-operating Diesel-alternator sets and induction and synchronous propeller motors.[1] As a sequence to this pioneering work Brown, Boveri & Cie. A. G., Mannheim built the propulsion equipment for the 6800 S. H. P. cargo vessel "Wuppertal" of the Hamburg-America Line which after being put into commission at the end of 1936 was described in the technical press as being by far the most important development in the field of electric ship's propulsion.

[1] E. Klingelfuss: "Brown Boveri Diesel-electric Ship Propulsion with Alternating Current". The Brown Boveri Review, 1932, pp. 148—154.

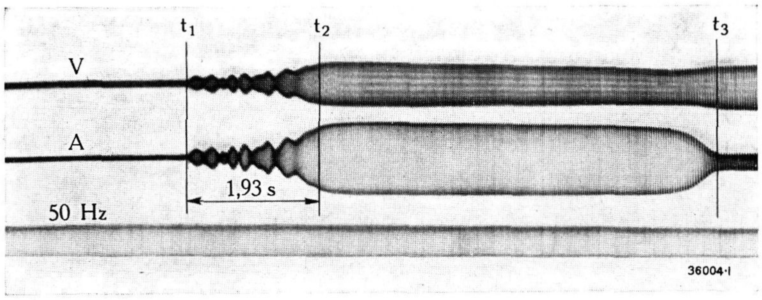


Fig. 1. — Oscillogram of starting test with Diesel-electric propulsion equipment incorporating six Diesel-alternator sets and one propeller motor.

t_1 . . . Alternator excitation switched in.
t_2 . . . Synchronizing process terminated.
t_3 . . . Propeller motor, accelerated up to full speed as induction motor and running with normal slip, can be easily synchronized by switching in d. c. excitation.

The synchronizing process is effected rapidly and without undue current surges notwithstanding the intentional discrepancy in the alternator speeds which differed from machine to machine by 30 r. p. m. from 770 to 920 r. p. m. This simplified synchronizing process renders *manœuvres extremely simple, despite the splitting up of the propulsion equipment.*

The orthodox arrangement was also departed from in the case of the auxiliaries for this vessel. In order to enhance the economy and simplify the machinery a three-phase supply furnished at sea exclusively by the Diesel-alternators of the propelling plant was employed for the first time on a cargo vessel (Figs. 2 and 3).

The experience made with the propulsion equipment of this ship was excellent. Particularly striking was the simplicity with which manœuvres could be effected, notwithstanding the parallel operation of the alternators (Fig. 4). The success attending this first plant led to a whole series of large passenger and cargo vessels being equipped with similar Diesel-electric three-phase drives by various firms between 1937 and 1939.[1]

The propulsion equipment of the E. S. "Wuppertal" has been dealt with fully in the marine press[2] and therefore need not be gone into here. On the other hand, a cursory recapitulation of the merits of this method of propulsion, based on present knowledge and experience, might not be out of place. In the following notes it is intended to lay special stress on those advantages which cannot be expressed in terms of money, weight, or space requirements, but which will probably prove a deciding factor in the choice of marine propelling machinery upon the return of normal conditions. Many shipowners will find themselves left with a very considerably reduced fleet and will therefore be in a position to plan the propulsion machinery of new ships much more systematically than when new vessels were only laid down at more or less lengthy intervals and fitted out with the propelling machinery which happened to be in vogue for the class of ship and the service conditions in question. It will be seen hereafter that when a fleet can be rebuilt systematically the Diesel-electric three-phase drive attains to even greater significance than under pre-war conditions, and that this mode of propulsion is fully justified not only for special ships, but also for high-powered vessels in general.

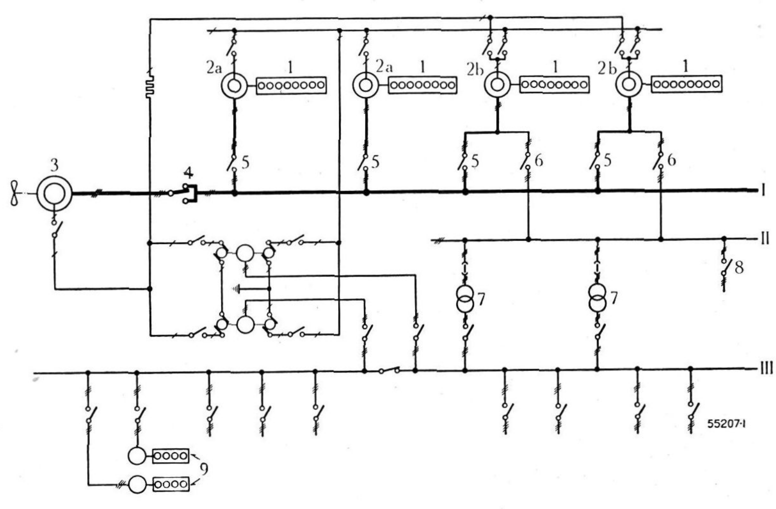


Fig. 2. — Diagram of connections of Diesel-electric three-phase propulsion equipment *with four Diesel-alternator sets.*

Alternators 2a and 2b are paralleled in a simple manner by a patented system. At sea one of the two alternators 2b can be simultaneously connected by switches 5 and 6 to the propulsion system I from which the propeller motor 3 is fed through reversing switch 4, and to the high-voltage auxiliary power system II coupled to the low-voltage auxiliary power system III through transformers 7. Large power-consuming auxiliaries 8, such as electric boilers, etc., can be connected directly to high-voltage system II in order to keep the transformers 7 small. For manœuvres the main alternator 2b supplying both systems I and II, is disconnected from system I. It then only meets the auxiliary power requirements at normal frequency and voltage, the other alternators being used for the manœuvres. Thereafter, the alternator 2b disconnected from system I is again synchronized with the other alternators. The auxiliary Diesel-alternators 9 are only required in harbour or when it is necessary to proceed for a long time so slowly that the main alternator voltage and frequency falls below the admissible values for the auxiliary system.

[1] Since this article was written a statement has been published to the effect that in the U. S. A. a large number of ships for various purposes and aggregating over 300,000 S. H. P. are to be Diesel-electric propelled with three-phase alternators operating in parallel. The rating of the individual ships lies between 4500 and 12,000 S. H. P., that of the Diesel engines being 1600 H. P. at 750 r. p. m.

[2] E. g., Werft, Reederei, Hafen, 1937, No. 8. Elektrotechnische Zeitschrift 1937, Nos. 40/41. VDI Zeitschrift 1937, No. 15. The Motor Ship, May, 1937.

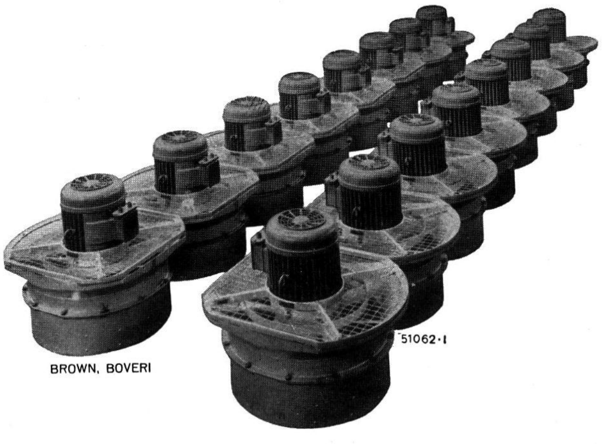


Fig. 3. — **Sixteen Sulzer vertical propeller-type fans on a modern motor-ship, driven by Brown Boveri three-phase pole-changing squirrel-cage motors.**

Three-phase motors can be employed to advantage for many auxiliary drives, due to their being smaller, lighter, less expensive, and more robust than the usual marine d. c. motors. The first merchant vessel to be built in Europe on which extensive use was made of a three-phase supply for the auxiliary motors, lighting, heating, etc., was the E. S. "Wuppertal" of the Hamburg-America Line. Experience with this installation has been good throughout and in the interim three-phase a. c. has found wider and wider application for the electrical auxiliaries of large ships, even where the propeller is driven mechanically.

1. *Weight — Space Requirements — Location of Rooms.*

When contrasting weights and space requirements of various methods of propulsion a common basis of comparison is imperative. In order to draw a parallel between Diesel-electric three-phase propulsion and direct or geared Diesel drive, the ship's auxiliary generating plant, for instance, must also be included, inasmuch as in the case of three-phase drive the entire auxiliary power system is fed by the main Diesel-alternators at sea. It is likewise necessary to take the propeller shaft into account, for, due to the propeller motor being placed right aft, this shaft is substantially shorter in the case of the electric drive than with the other modes of propulsion. On this basis even the first Diesel-electric three-phase propulsion equipments were at least the equal of direct and geared Diesel drives as regards weight and space requirements, although the ratio of Diesel-engine to propeller-shaft revolutions was only of the order of 2 : 1. In these installations four-stroke-cycle Diesel engines with Brown Boveri turbo-chargers or single-acting two-stroke Diesel engines, with ratings between 2000 and 3600 B. H. P. at speeds between 250 and 235 r. p. m. were employed to drive the main alternators. In the interim, however, leading Diesel-engine builders have evolved machines continuously developing up to approximately 1600 B. H. P. at 500—750 r. p. m. or 3000 B. H. P. at 360—500 r. p. m., and it is in this very field that they are at present making the greatest efforts. In consequence, upon the return of normal conditions medium-power Diesel engines running at still higher speeds, and suitable for installation on merchant ships, will doubtless be available.[1]

The diagram in Fig. 5 shows the effect of increased engine revolutions on the weight of Diesel-electric three-phase and geared Diesel propulsion equipment, respectively, for a single-screw cargo ship of 6500 S. H. P. at 125 r. p. m. For purposes of comparison, only standard Diesel engines designed for continuous operation on merchant ships have been considered. The unit weights of the engines naturally vary within certain limits according as four- or two-stroke-cycle or single- or double-acting engines are concerned; for the comparison mean values of the weights given by different firms have been inserted. In the case of the geared Diesel engine drive it is assumed that there are

[1] The Motor Ship, January, 1941, pp. 328/329: "Marine Oil Engine Progress in 1940."

Schweizerische Bauzeitung, 28th March/4th April, 1942: "Die Aufladung des Zweitakt-Dieselmotors."

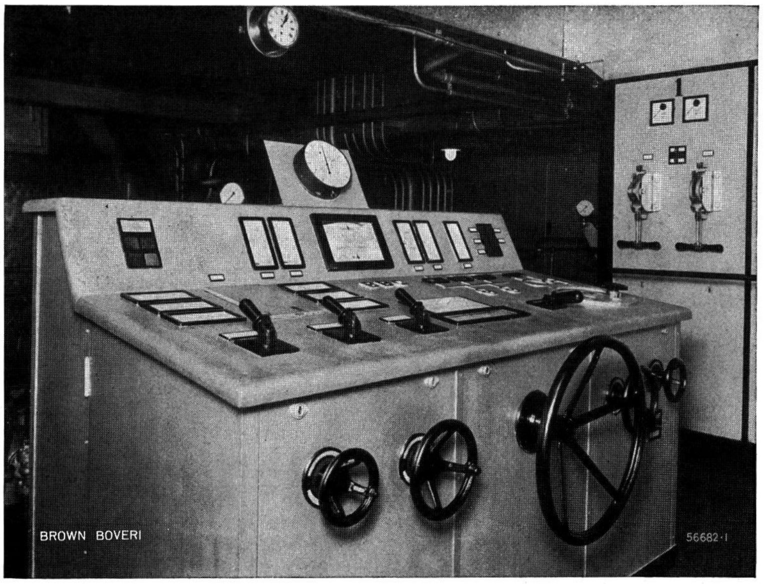


Fig. 4. — **Control desk of a Diesel-electric propulsion equipment with three Diesel-generator sets.**

(Manufactured by Brown, Boveri & Cie. A. G., Mannheim.)

This embodies all apparatus necessary for carrying out and supervising manœuvres and the requisite metering instruments and signalling apparatus for service checks. The carrying out and supervision of the manœuvring processes are extremely simple. All electrical operations are automatically effected in correct sequence when the large handwheel in the centre of the switch desk is turned, so that the getting under way, stopping, and reversal of the ship can be undertaken by any operator without lengthy preliminary instruction.

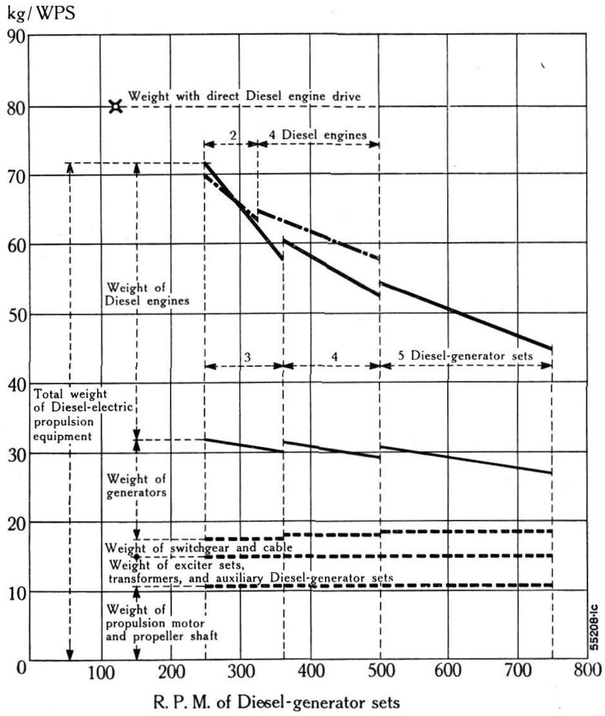


Fig. 5. — Comparison of weights of different propulsion equipments for a 6500 S. H. P. single-screw cargo vessel.

With direct Diesel drive.

With geared Diesel drive and hydraulic or electro-magnetic couplings and engines of different speeds.

With Diesel-electric three-phase drive and engines of different speeds.

Apart from the propulsion equipment proper, complete with all auxiliaries and appurtenances, the unit weights include the auxiliary Diesel-alternator installation and the propeller shaft.

Propeller revolutions at the full ship's speed 125 r. p. m.
Maximum auxiliary power demand at sea 400 kW; in harbour 180 kW.

The Diesel-electric three-phase drive with high-speed Diesel engines as at present available for merchant vessels is much lighter than Diesel-mechanical drive. The three-phase drive with, for instance, three 1850 kVA, 360 r. p. m. Diesel-alternator sets for generating the entire electrical power required at sea is about 36 t lighter than the geared Diesel drive and 140 t lighter than the direct Diesel drive.

hydraulic or electro-magnetic couplings between the Diesel engines and the gear pinions. Inasmuch as with this method of propulsion an even number of Diesel engines is invariably employed and the weight of the gearing becomes greater with increasing gear ratio, higher engine revolutions have not such a favourable effect as with the three-phase system. The weight for a corresponding direct Diesel drive will also be found on the same diagram.

From Fig. 5 it will be clear that even with the Diesel engines at present available the Diesel-electric three-phase drive is lighter than both the geared and direct Diesel drives if the propeller shaft and electrically-driven auxiliaries are taken into consideration. Taking the example of a 6500 S. H. P. cargo ship with three 1850 kVA, 360 r. p. m. Diesel-alternator sets for the generation of the entire electrical power required at sea a reduction in weight of approximately 140 t is achieved, compared to direct Diesel drive. The corresponding diminution in relation

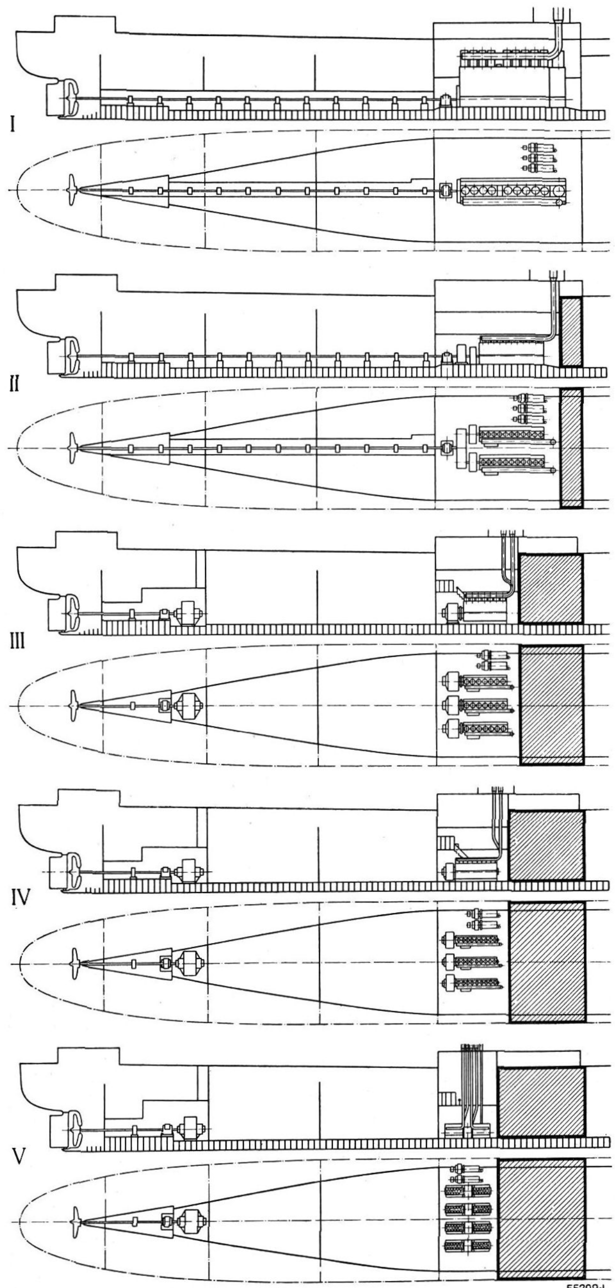


Fig. 6. — Comparison of space requirements of the propulsion equipment for a 6500 S. H. P. cargo vessel with a propeller speed of 125 r. p. m.

I. Direct Diesel drive with one Diesel engine.
II. Geared Diesel drive with two 320 r. p. m. Diesel engines.
III. Diesel-electric drive with three 250 r. p. m. Diesel-alternator sets.
IV. Diesel-electric drive with three 360 r. p. m. Diesel-alternator sets.
V. Diesel-electric drive with four 600 r. p. m. Diesel-alternator sets.

The engine room is substantially shorter and lower with Diesel-electric drive than with direct or geared Diesel drive, the more so the higher the speed of the Diesel-alternator set can be selected. Moreover, the shafting tunnel traversing the holds aft, involved by the other methods of propulsion, is eliminated. This saving in space and the reduction in weight compared to other Diesel engine propulsion systems considerably enhances the economy of the ship.

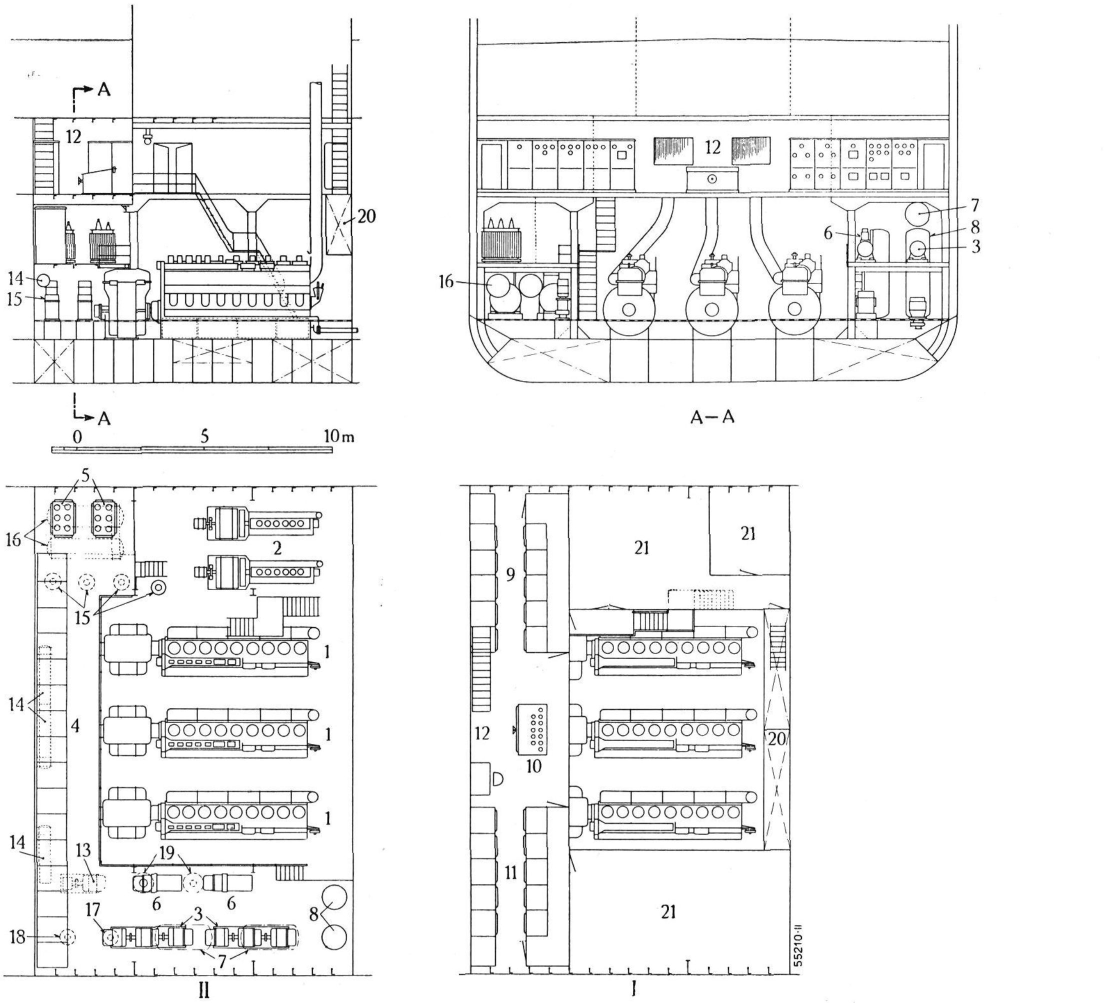


Fig. 7. — General layout of a modern 6500 S. H. P. Diesel-electric three-phase propulsion equipment.

I. Deck 1.
II. Engine room.
1. Main Diesel-alternator sets.
2. Auxiliary Diesel-alternator sets.
3. Exciter sets.
4. High-voltage switchgear for main generators, propeller motor, and transformers.
5. Transformers.
6. Compressors.
7. Compressed-air receivers for switchgear.
8. Compressed-air receivers for starting Diesel engines.
9. Switchboards for main alternators, propeller motor, and transformers.
10. Control desk.
11. Switchboards for auxiliary Diesel alternators, exciter sets, and low-voltage distribution system.
12. Control room.
13. Standby oil pump.
14. Oil cooler.
15. Fresh and sea-water pumps.
16. Fresh water cooler.
17. Ballast pump.
18. Bilge pump.
19. Sanitary and fire-fighting pump.
20. Fuel tanks.
21. Space for various auxiliaries, workshop, stores, etc.

The control room with control desk and switchboards for the main electrical machines is completely separated from the engine room. Owing to the freedom from noise and oil vapour, conditions in the control room are much more pleasant for the staff entrusted with manœuvres.

to geared Diesel drive with 320 r. p. m. engines is of the order of 36 t.

Fig. 6 shows that also on the score of space requirements the superiority of the Diesel-electric drive over the other methods of Diesel-engine propulsion becomes more and more pronounced the higher the main engine revolutions can be selected. The smaller and lower engine room releases valuable space amidships for the accommodation of passengers or cargo. Moreover, the shafting tunnel through the holds aft, involved by the other forms of propulsion, is eliminated, the propeller motor being placed right aft in a location having little value as cargo space. The low weight and small space requirements enhance the economy of the vessel to quite an appreciable extent. Even after the cessation of hostilities material-saving designs will probably prove of decisive importance, inasmuch as a shortage of raw materials is to be reckoned with for some time to come.

The electric drive gives greater freedom of choice in the location of the rooms due to the fact that the layout of the power plant is not tied to the position of the propeller shaft. For the sake of simplicity, the engine room in Fig. 6 is left in the same position as for direct or geared Diesel drive, although the Diesel-alternator sets could just as well have been located elsewhere, e. g., aft in the vicinity of, or above, the propeller motor, to liberate space amidships for other purposes. It is also possible to install the Diesel-alternator sets in separate water-tight engine rooms separated by bulkheads, so that should one engine room be put out of commission the vessel can still be propelled by the machinery in the others. Following modern power station practice, the control rooms with control desk and switchboards for the main electrical machines can be completely separated from the engine rooms proper (Fig. 7). When laying out electric ships a special point must be made of breaking away from the orthodox location of the rooms and fully utilizing the advantages accruing from electric propulsion so as to make the most of the available space. The naval architect is thus afforded an excellent opportunity of designing ships which are more economical and simpler to operate.

2. *Cost of Propulsion Equipment — Standardization of Propelling Machines — Stocks of Spare Parts — Charges for Engine-room Staff — Building Time.*

The initial cost of the first Diesel-electric three-phase drives, including auxiliary power plant, was only slightly greater than that of corresponding direct or geared Diesel installations. These drives were the first of their kind, however, and their further development with Diesel engines running at higher speeds will doubtless result in a much lower purchase price. The cost of the three 1850 kVA, 360 r. p. m. Diesel-alternator sets of a 6500 S. H. P. propulsion equipment is, for instance, only 93 % of that of 250 r. p. m. alternator sets of the same rating. In consequence, the price of the complete propulsion installation will already be about 5 % lower.

Since the main Diesel alternators also feed the auxiliary power system and thus deliver the auxiliary power economically, it is of advantage systematically to operate every possible duty electrically. This applies in particular to auxiliaries which, being employed for different purposes, are not in operation at the same time and therefore involve no increase in the auxiliary power demand. The cost of heating an electric ship electrically, for instance, is lower than with any other system, inasmuch as, firstly, it is cheaper to run cable than heating pipes and, secondly, the auxiliary power demand is practically not increased by the electric heating, because during the heating periods the power consumption for ventilating and refrigerating purposes is greatly diminished.

When rebuilding a fleet the economy of the propulsion equipment of one single ship must, moreover, not be taken as exclusive criterion for the mode of drive, but all of the new vessels to be laid down should be systematically considered with a view to equipping as many ships as possible with the same type of propelling machinery, notwithstanding differences in shaft rating. The fulfilment of this latter condition permits of:

(a) Reduction of initial cost of propulsion equipment.
(b) Diminution of number of spare parts required.
(c) Simplification of overhaul and repair of propelling machinery.
(d) Facilitation of instruction of operators and of exchange of crews.

With no other mode of propulsion can all of these advantages be better achieved than with the Diesel-electric three-phase drive, which permits one and the same type of Diesel engine and alternator to be employed for ships of widely differing shaft rating, inasmuch as the latter only governs the number and not the power of the parallel-operating primary sets. The question of the unit rating of the main Diesel-alternator sets will be chiefly decided by the high-speed marine Diesel engines available on the market, although the value selected should allow one or two of the sets to be employed for feeding the auxiliary power system economically in harbour or when the vessel is running at slow speeds. In this way auxiliary Diesel-alternator sets can be dispensed with, thus leaving only one or two small emergency Diesel-alternator sets to be provided. To ensure economical operation in harbour it may prove advisable in certain cases to select one of the main Diesel engines with a smaller number of cylinders, i. e., a lower rating.

Results of investigations undertaken for a shipowner in connection with the building of eleven ships with ratings between 6000 and 16,000 S. H. P. are shown in table I. Fifty-five similar Diesel-alternator sets can be utilized for the eleven vessels. From the point of view of service conditions or room location it may be more favourable in the case of the 12,000 and 16,000 S. H. P. ships to replace the seven and nine Diesel-alternator sets, respectively, by three and four Diesel alternators of double the rating, each driven by two 2250 B. H. P. Diesel engines, and only to retain the 1650 kW unit rating for one set in each case. In this way fifty-five similar Diesel engines and thirty-five 1650 or ten 3300 kW alternators can be used. The manufacturing costs of the propelling machines for the eleven ships would in this manner certainly be substantially lower than if various types of machine were to be employed for ships with different shaft ratings.

A saving not to be under-estimated in connection with the three-phase drive is that accruing from the possible reduction in the stock of spare parts to be maintained. In the case of direct drive the value of the necessary spare parts to be carried on board is tantamount to approximately 10 % of the purchase price of the Diesel engine. According to the rules of the classification societies only one set of spare parts need be carried in cases where the installation comprises Diesel engines of the same type. Taking the example of an installation with five Diesel-alternator sets the spare parts will only cost approximately 3 % of the initial outlay for the plant. Moreover, the present conflict has proved that the stocks of spares normally carried on board are in any case inadequate, and to prevent service interruptions in times of difficulty a store of important components must also be available on shore. Spare parts are more easily obtained and at less cost when a large number of ships belonging to the same shipowner has Diesel-electric three-phase drive with identical Diesel-alternator sets.

If all of these points are given due consideration and exploited correctly the cost of the propulsion equipment of electric ships will doubtless prove to be lower than for any other Diesel-engine propelling system.

Another factor to be taken into account is that, by reason of the circumstances detailed hereafter, the wages bill will be smaller, although the same engine-room staff is required as with Diesel-mechanical drive. Ships usually remain only a short time in the home

TABLE I.

Class of ship:	I	II	III	IV	V
Number of ships	4	2	2	2	1
Specified rating per ship . . . S.H.P.	6000	9000	2×4500	2×6000	2×8000
Revolutions of propeller shaft at full speed . . r. p. m.	120	100	120	120	100
Maximum auxiliary power demand at sea . . . kW	300	900	1200	2000	2300
Maximum auxiliary power demand in harbour . kW	180	400	1300	1500	1550
Maximum necessary aggregate rating of main alternators . . . approx. kW	4980	7930	8250	11,390	14,800
Unit rating of Diesel alternators . . . kW			1650[1]		
Unit rating of Diesel engines . . . B.H.P.			2250 at 360 r. p. m.		
Number of Diesel-alternator sets per ship . .	3	5	5	7	9
Total rating of main alternators . . . kW	4950	8250	8250	11,550	14,850
Necessary power required for auxiliary system at sea (including demand for excitation and ventilation of main electrical machines) . . . kW	480	1170	1470	2360	2770
Power available at terminals of alternators for propulsion purposes . . . kW	4470	7080	6780	9190	12,080
Actual rating per ship . . . S.H.P.	5950	9440	2×4520	2×6110	2×8040
Number and rating of auxiliary alternators . . kW	2×200	1×450 [2]	[3]	[3]	[3]
Weight of Diesel-electric propulsion equipment, including auxiliary alternator installation and propeller shaft . . . t	355	570	535	750	985
Reduction in weight per ship compared to direct Diesel drive . . . t	170	210	205	280	355
Total reduction in weight for the 11 vessels . t			2425		
Total number of main Diesel-alternator sets for each class of ship	12	10	10	14	9
for the 11 ships . .			55		

1 The rating of the main Diesel-alternators is selected to comply with the following requirements:—
(a) The same type of Diesel-alternator to be used on all ships.
(b) In the case of ships classes III—V it should be possible to feed the auxiliary power system economically from one of the main Diesel-alternator sets in harbour to avoid installing separate auxiliary Diesel-alternator sets.

2 Only one auxiliary Diesel-alternator set need be installed, inasmuch as one of the main sets can be employed in harbour should the auxiliary set be out of commission.

3 No auxiliary Diesel-alternator sets proper are installed, but only two 120—200 kW emergency Diesel-alternator sets which are also employed for exciting the first main Diesel-alternator set at starting.

port and if some of the home-coming men are to be given the leave to which they are entitled, substitutes must be held ready on shore. In the case of electric vessels, however, where machinery and operation are identical, men can be readily transferred from one ship to another, thus greatly simplifying the staffing problem for the shipowner. Moreover, operating experience can be exchanged between the men thus transferred and their new shipmates, and the incorporation of this in new vessels will rapidly lead to the perfectioning of this method of propulsion.

Medium-powered Diesel engines and alternators, which will be required in large numbers, are particularly suitable for manufacture on mass production lines and, like the propeller motors, can be built more rapidly than the large engines involved by direct drive. Since it is the machinery of a ship which takes the longest time to complete, electric ships will thus have a shorter overall building time than vessels with direct or geared Diesel drive, an advantage which should prove of great importance in the future.

3. *Fuel Consumption — Adaptability to Various Operating Conditions.*

The fuel consumption of high-speed Diesel engines per H. P. is somewhat higher than that of slow-speed machines. On the other hand, both the electric and the geared Diesel drives permit the most favourable propeller speed to be selected for the ship, whereas with direct Diesel drive the propeller speed is often chosen uneconomically high in order to avoid employing excessively heavy and expensive Diesel engines. The fact that the propeller speed is better adapted to the type of ship permits the propeller driving power to be kept low, thus usually more than offsetting the slightly higher fuel consumption of the high-speed Diesel engines. Another point to be considered is that electric ships can in many cases be built with single screws, whereas with direct drive, for reasons of safety, etc., twin-screw propulsion is usually resorted to, nothwithstanding the approximately 5 % lower propelling efficiency entailed.

The overall losses of 6—8 % of the shaft-horse-power rating involved by electrical transmission find in the geared Diesel system more or less a counterpart in the losses in the gearing and hydraulic or electro-magnetic couplings which aggregate at least 4—4 1/2 % of the shaft-horse-power rating. The shaft losses are lower with electrical drive than with the other modes of propulsion, due to the much shorter propeller shaft requiring — according to the size of the ship and the layout of the propelling machinery — six to twelve fewer lineshaft bearings. The losses per lineshaft bearing are of the order of 1/3 % of the shaft-horse-power rating. As will be seen from

TABLE II.

Method of propulsion	Losses in percentage of shaft-horse-power rating: in the propeller line-shaft bearings %	in gearing and couplings %	in electrical machines and cables %	Total %
Direct-coupled Diesel	2—4	—	—	2—4
Geared-Diesel . . .	2—4	4—4 1/2	—	6—8 1/2
Diesel-electric . . .	1/3—1	—	6—8	6 1/3—9

table II the losses between the Diesel-engine coupling and the stern gland are practically the same for Diesel-electric and geared Diesel drive and only 4—5 % higher than with direct drive.

On the score of fuel consumption, therefore, the difference between the three methods of propulsion is not nearly so marked as is usually supposed. In the case of a vessel running at below approximately 60—80 % of its full speed, fuel consumption is actually higher with direct drive than with electrical drive, inasmuch as with the latter as many economical ship's speeds can be obtained as Diesel-alternator sets are installed, as long as the latter are kept approximately fully loaded (Fig. 8). It is moreover possible to fulfil every schedule entirely at economical speeds, for, if necessary, the ship can be run partly above and partly below the average speed so that

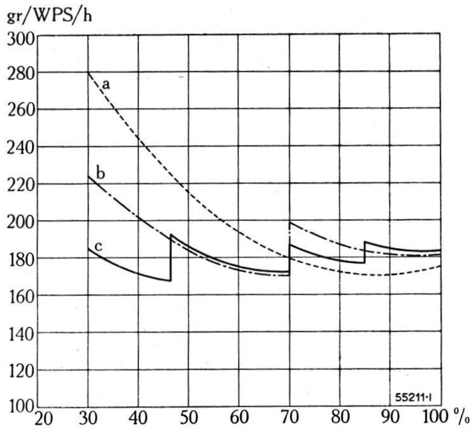


Fig. 8. — Comparison of specific fuel consumption of a 6500 S. H. P. single-screw cargo vessel with a propeller speed of 125 r. p. m.

a. Direct Diesel drive with one Diesel engine.
b. Geared Diesel drive with two 320 r. p. m. Diesel engines.
c. Diesel-electric drive with four 360 r. p. m. Diesel-alternator sets.

The same full-load specific fuel consumption at the coupling of the Diesel engines was assumed for the three types of drive, while the transmission losses between the Diesel engine couplings and the stern gland were taken to be 3 % in the case of a, 7 % in the case of b, and 7·5 % in the case of c.

Since the Diesel-electric drive holds a great advantage over direct drive at low speeds the fuel consumption of electric ships, measured over lengthy periods and including all runs at reduced speed, is not greater than in the case of motor-ships of the same size and operating on the same route.

the Diesel-alternator sets in service will constantly operate at full load. The Diesel-electric three-phase drive, therefore, holds the advantage on all routes demanding a reduced speed for a long period, e. g., during the passage of canals, when entering port, in fog, etc.

Due to the splitting up of the propelling plant into medium-power units the Diesel-electric three-phase drive not only enables the most varied schedules to be coped with economically, but also renders the auxiliary services more flexible on account of the ample power always available. For instance, instead of the hoisting and conveying appliances, such as winches, pumps, etc., already available on the ship, other appliances with industrial-type three-phase motors can temporarily be installed and supplied from the ship's power plant, to enable other cargoes than that for which the vessel was built to be loaded or unloaded.

4. *Reliability — Overhauls and Repairs — Manœuvring Properties.*

Reliability is the most important requirement of any ship's drive, and in the case of the new vessels to be built still more attention will be paid to this point, inasmuch as every ship will be urgently required for years to come. For the same reason and to keep the time in port and the attendant expenses low the engine-room staff should be able rapidly to effect machinery repairs and overhauls, if possible with the vessel under way. These requirements are fulfilled by the Diesel-electric three-phase drive to a high degree.

The breakdown of a Diesel-alternator set does not involve stoppage of the ship, but simply a slight reduction of its speed. The machine components are much smaller than with direct Diesel drive of the same power and can readily be removed and repaired without affecting the availability of the other sets. By employing a single type of Diesel engine for a number of vessels belonging to the same shipowner, the engine-room staff will become so familiar with the machines that they will be able to carry out repairs satisfactorily themselves. If the greater part of these can be effected with the vessel under way, the ship's engineers will stand a better chance of enjoying the whole of their shore leave.

There is less wear and tear on the Diesel engines with three-phase drive than with direct or geared Diesel drive, inasmuch as the Diesel engines can be run up light when getting under way and do not have to be reversed during manœuvres, the electrical machines taking over the starting, stopping, and reversal of the propeller. This reduces the compressed-air consumption to a minimum, so that compressors and compressed-air cylinders are smaller than with direct or geared Diesel drive. In particular, however, the life of the Diesel engines is lengthened and the time which can be allowed between overhauls increased, due to the fact that when reversing electrically the injection of cold compressed air into the hot cylinders, which causes high stresses to be set up, is no longer necessary.

As far as the electrical machines and apparatus are concerned experience has shown that these fully meet all requirements if their design and the materials from which they are built are selected to conform to the special conditions obtaining on board ship. Moreover, the specifications set up by classification societies for electrical machinery and apparatus in marine applications are generally much more stringent than those for land plants, thus ensuring installations of liberal dimensions and suitable construction. Only long-known designs, well tried out in land plants, are employed throughout, extra protection merely being provided to take account of the special conditions prevailing on board ship. The main machines are ventilated on the closed-circuit principle to prevent ingress of salt-laden air and oil vapour. Thermometers on the hot and cold air connections and on the cooling-water inlet and outlet enable a check to be kept on the efficiency of the cooler. The stator windings have remote indicating thermometers, thus permitting a reduction in the cooling effect, due to clogging of the cooling tubes, to be noticed in good time. No other method of drive allows of such extensive protection of the machines against excessive overload with simple apparatus. A differential system of protection nips electrical machine faults in the bud by disconnecting the faulty machines from the network and cutting off their excitation. Repairs can be readily effected by the crew, even winding replacements being possible on board due to the ease with which the electrical machines can be dismantled.

Synchronous machines are now generally not only employed for the alternators, but also for the propeller motors. The synchronous propeller motor is lighter and more efficient than the induction motor. Moreover, it can be built with a relatively large air-gap clearance (up to 6 mm) and is therefore extremely reliable even under bad weather conditions (shaft displacement involves no risk of damage). A "fog" motor, such as was fitted on the E. S. "Wuppertal" for running at reduced speed, as, for instance, in fog or when entering and leaving port, and particularly as standby for the main motor, was not considered necessary in subsequent three-phase plants. For starting and reversing purposes the propeller motors are provided with a robust induction winding. Manœuvres can therefore be carried out both simply and reliably.

Air-blast circuit-breakers are preferably employed as main switch for the alternators, special features of these being their small dimensions, low weight, and — due to their freedom from oil — high reliability. These circuit-breakers withstand inclined positions with impunity and lend themselves particularly well to remote control. Owing to the compressed-air plants of the circuit-breakers and Diesel engines being built for approximately the same pressure, they can be combined and the one employed as standby for the other. The operation of the entire control gear necessary for starting, stopping, and reversal of the propeller motor as well as for regulating the ship's speed is vested in a control desk where all of the metering instruments and signalling apparatus required for checking purposes are also assembled. The control operations are thus easy to carry out and supervise. The electrical instruments enable a continual check to be kept on the individual machines. Experience has also proved that electrical propelling equipment requires far less upkeep than other propulsion machinery.

Brown Boveri has evolved a patented system for the reversal of synchronous-motor-driven propellers, with which the greater part of the braking energy is dissipated in resistors instead of in the motor. It is based on the fact that a ship can be effectively braked when the propeller is turning only slowly forward or is at rest at the beginning of the reversing manœuvre. With the Brown Boveri system the propeller motor is first disconnected from the alternator and braked electrically — by switching its three-phase winding over on to braking resistors — and allowing it to operate as generator with approximately normal excitation. The motor will be braked more or less quickly according to the degree of excitation and size of the braking resistors. It is not until the ship has lost sufficient speed — which is indicated by the reduction in the motor current — that the propeller motor is again switched from the braking resistors over on to the alternators, to be reversed asynchronously. Since it is only in the quite short period during which the motor is picking up speed asynchronously in the astern direction that the current rises above the rated value, both the alternators and the propeller motors are subjected to far less thermal stressing than with pure asynchronous reversal. In consequence, a large number of manœuvres can be carried out in succession without fear of overloading

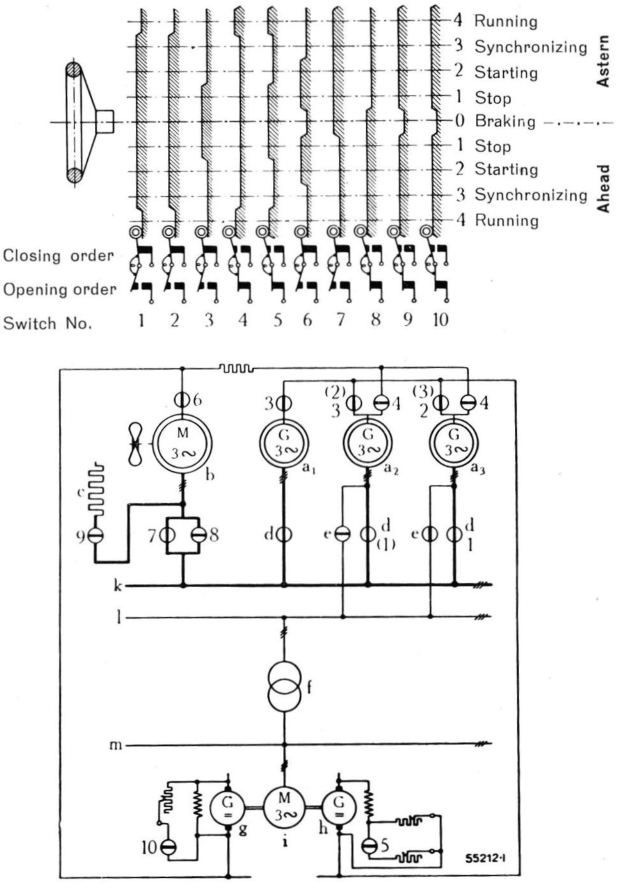


Fig. 9. — Diagram of connections of an electric ship's propulsion equipment with synchronous braking of propeller motor.

Manœuvring switch in position 4 "Ahead".

1. Main switch of Diesel-alternator set connected to both main and auxiliary power systems.
2–3. Switch for exciting from alternator exciter.
4. Switch for exciting from motor exciter.
5. Switch for over-exciting alternator exciter.
6. Exciter switch for propeller motor.
7. Main switch of propeller motor for running ahead.
8. Main switch of propeller motor for running astern.
9. Switch for braking resistor.
10. Switch for adjusting braking current.

$a_{1,2,3}$. Diesel-driven three-phase alternators.
b. Synchronous propeller motor.
c. Braking resistor.
d. Main switch for Diesel-alternators.
e. Switch for feeding auxiliary power system either by alternator a_2 or a_3, according to position of separate selector switch.
f. Transformer.
g. Motor exciter.
h. Alternator exciter.
i. Motor of exciter set.
k. Propulsion power system.
l. High-voltage auxiliary power system.
m. Low-voltage auxiliary power system.

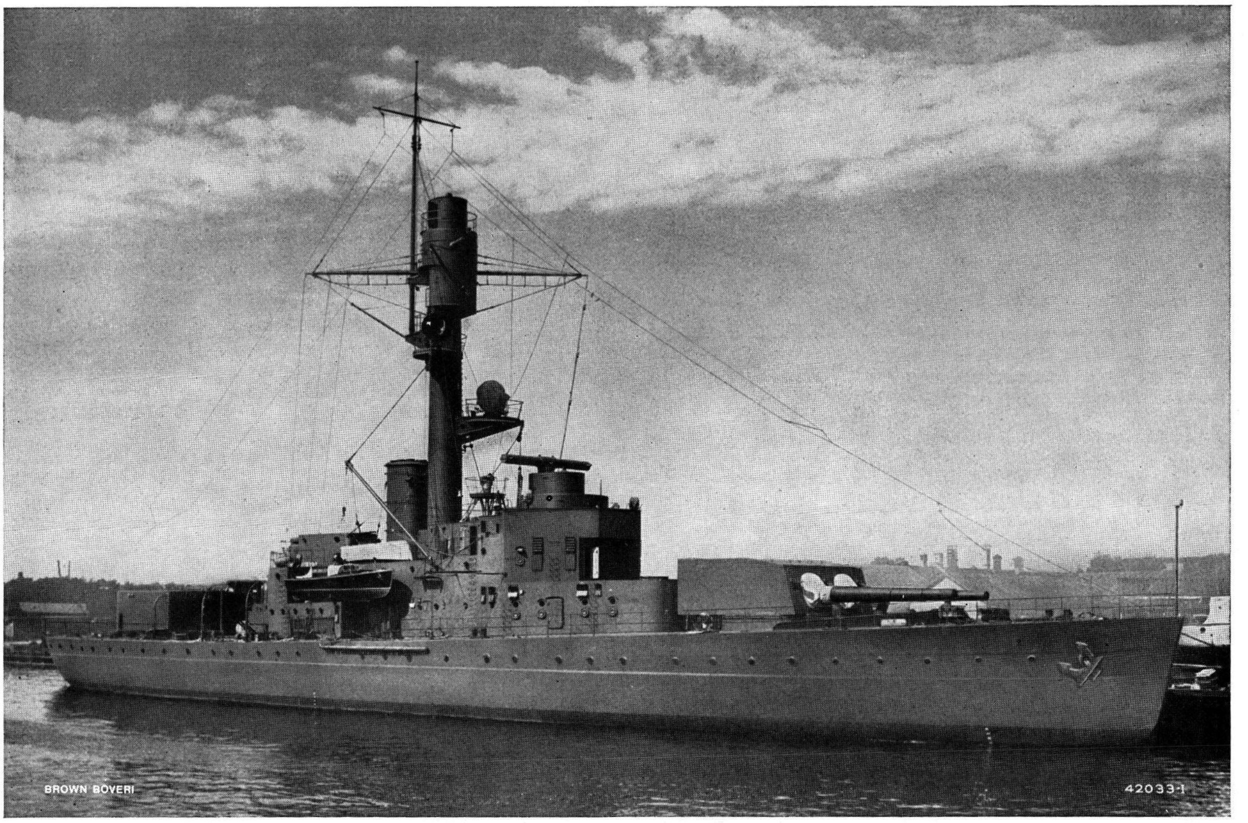
With this Brown Boveri patented reversing system the alternators and propeller motors are subjected to far less thermal stress than with pure asynchronous reversing. The reversing process is simple and permits a large number of manœuvres to be carried out in succession without risk of overloading the propulsion machinery.

the propulsion equipment. Transition from generative braking to asynchronous reversal can, moreover, be effected quickly or slowly, according as more importance is attached to stopping the ship within as short a distance as possible or to keeping the stressing of the propulsion equipment down to a minimum. Too rapid reversal of the propellers, with the consequent abrupt interruption of the flow induced by the propeller blades and the ensuing heavy vibration of the stern, is positively avoided. Manœuvring times are short and the ship can be brought to a standstill in a simpler and surer manner than with any other method of reversal. Fig. 9 shows the simple method of control of the propeller motor on this principle.

The foregoing notes prove that the reliability of the Diesel-electric drive is unsurpassed by any other method of propulsion. This can be still further enhanced by installing the Diesel-alternator sets together with the exciter sets, transformers, switchgear, etc., in independent engine rooms separated from one another by watertight, fire-proof bulkheads. Should one engine room be out of commission, due to ingress of water or outbreak of fire, propulsion and the supply of the auxiliary power system with electrical energy can be maintained, without any interruption whatsoever, by the engine rooms not involved in the breakdown. The Diesel-electric drive is thus also particularly suitable for use on warships where importance is attached to the splitting up of the propulsion equipment between several engine rooms and to economical operation at the various ship's speeds. It might also be mentioned that in the case of multiscrew vessels with three-phase synchronous motor drive the propellers in the same plane can be kept running synchronously in opposite directions in a simple manner without supplementary machines, thus substantially reducing the vibration of the ship caused by the propellers.

Although only ships with ratings up to 2×8000 S.H.P. have been considered it goes without saying that there is no limit to the power of the ships to which the three-phase Diesel-electric drive can be applied, inasmuch as the aggregate propelling power only determines the number and not the rating of the Diesel-generator sets to be operated in parallel.

(MS 890) *Th. Egg. (E. G. W.)*



Finnish coastal cruiser "Ilmarinen" with d. c. Diesel-electric propulsion equipment.

The entire propelling and auxiliary plant for both this and the sister ship "Wäinämöinen", the first warships to be exclusively equipped with Diesel-electric drive, were manufactured at Baden. The electrical equipment for submerged running, together with numerous auxiliaries, has also been supplied for various Finnish submarines.

A SPECIAL DESIGN OF VELOX BOILER FOR SHIPS.

Decimal Index 621.181.139

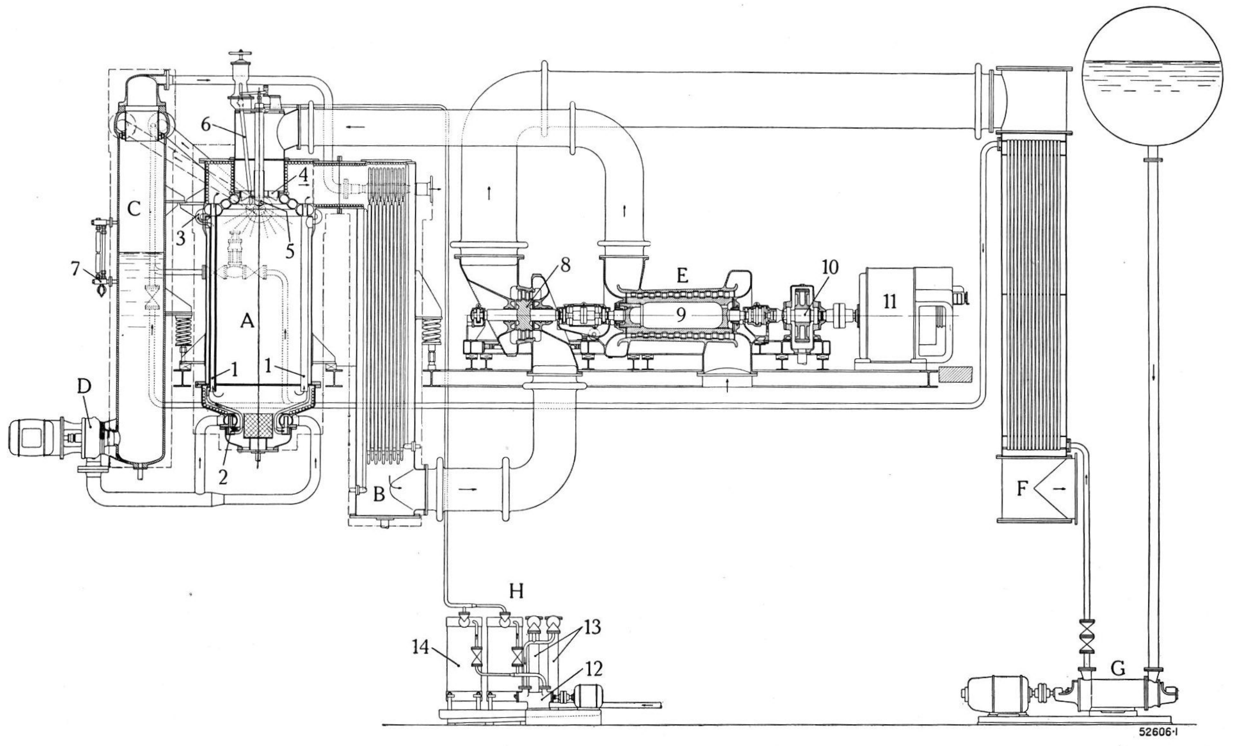


Fig. 1. — Standard design of Velox boiler for outputs from 10—150 t/h.

A. Combustion chamber.
  1. Evaporator tubes.
  2. Circulating water inlet.
  3. Steam-water mixture outlet.
  4. Swirl vanes.
  5. Burner.
  6. Ignition device.
B. Superheater.
C. Separator.
  7. Water level indicator.
D. Circulating pump.
E. Charging set.
  8. Gas turbine.
  9. Compressor.
  10. Reduction gear.
  11. Starting and regulating motor.
F. Economizer.
G. Feed pump.
H. Fuel supply.
  12. Fuel pump.
  13. Filter.
  14. Heater.

Velox steam generators are better than usual boilers in regard to efficiency. They have a considerably lower weight and take up less room. Their regulation takes place entirely automatically and they may be put into service in a few minutes. These characteristics are achieved by means of the pressure combustion and the use of high gas velocities resulting in high rates of heat transfer and further by forced circulation of the water and separation of the steam by centrifugal action. A special characteristic of the Velox steam generator is the gas turbine which is driven by the heating gases of the boiler itself, and which delivers the energy required for driving the charging compressor without the aid of external power.

The design of Velox boiler which has hitherto so successfully been employed for land purposes, can, except for certain special requirements of marine service, be used without change for ships.

Although the weight and volume of this type of Velox steam generator are already well below those of usual boilers, a new design has been developed, which — in addition to possessing the advantage of great simplicity of the component parts — achieves a further reduction of the weight and space requirements. This design accordingly merits special consideration where relatively large outputs have to be accommodated in small ship spaces.

The following article describes the new design after briefly recalling the special characteristics of the Velox steam generator design as used up to the present.

THE Velox steam generator has, so far, not been applied to ships to anything like the extent which might be expected in view of its undisputable suitability for this type of service. Its general introduction would probably have taken place much more rapidly if its manufacture had been entrusted to shipyards, and if shipowners had had the opportunity of forming an opinion from a greater number of ship installations than have been built up to the present. It would seem, therefore, that just as has been the case with the Diesel engine, so now with the Velox boiler, a certain time will have to lapse before its advantages are generally recognized and appreciated by shipbuilders and shipowners.

For land service, on the other hand, the introduction of the Velox has been relatively rapid. Not only are there to-day over seventy-five installations with an aggregate output totalling more than $2^1/_2$ million kg of steam per hour, but there are already a number of plants having more than 40,000 operating hours to their credit. This fact is mentioned here, because the design

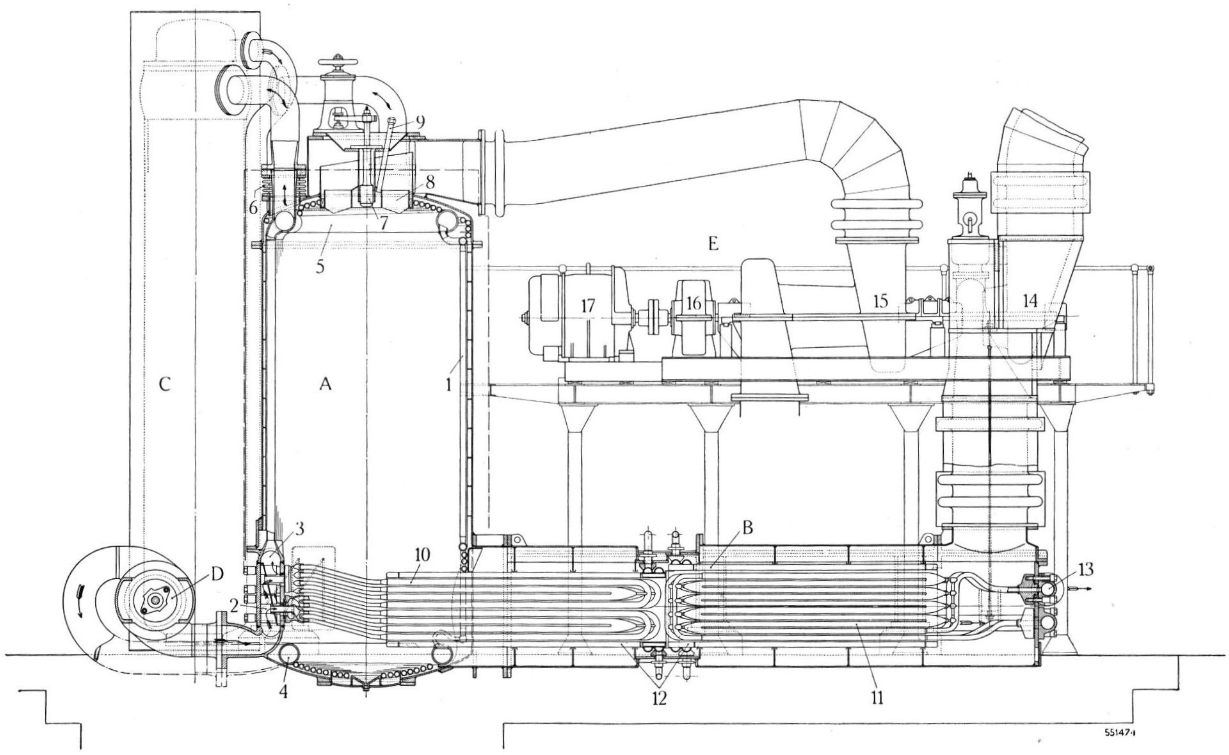


Fig. 2. — Special Velox design for plants with minimum weight and space requirements.

A. Combustion chamber.
1. Wall screen.
2. Circulating water supply.
3. Steam-water mixture outlet.
4. Circulating water distributor.
5. Steam-water mixture heater.
6. Bellows joint at the steam-water mixture outlets.
7. Burner.
8. Swirl vanes.
9. Ignition device.

B. Horizontal tube casing.
10. Evaporator tubes.
11. Superheater tubes.
12. Wall screens.
13. Live steam outlet.

C. Separator.

D. Circulating pump.

E. Charging set.
14. Gas turbine.
15. Compressor.
16. Reduction gear.
17. Starting and regulating motor.

This special design achieves a minimum in weight and space requirements. It is therefore applied wherever these two characteristics are required.

of Velox mainly entering into consideration for merchant ships and for large warships will be that which has successfully been employed for land service. This design has already been described in detail not long ago[1], and it is therefore sufficient to show again here only a section of the boiler (Fig. 1) and briefly to mention the most recent constructional improvements. These concern mainly the arrangement of the evaporator and superheater elements. Instead of building them one into the other and mounting them into the combustion chamber, they have — as was the case of the first Velox designs — been separated, the superheater being accommodated in a special casing outside the combustion chamber. This results in an increase in length of the gas paths, so that the cross sections of the gas passages may be increased while maintaining the same velocities, thus avoiding the tendency to obstruction by deposits of slag coming from the fuel. Further, the superheater is now located in a part of the gas path where the temperature is relatively low, thus excluding any danger for the tubes. Finally, the refractory material, which formerly filled the space between the evaporator elements and the wall of the combustion chamber, and which served as an additional heat insulation, has been replaced by a double wall of steel plate, the intermediate space being cooled by means of the combustion air.

These modifications have been due almost exclusively to experience with poor qualities of fuel or unsuitable feed water, that is to say, they were caused by factors which have occurred in every type of boiler during its first period of service. Difficulties due to the unusual design and operating principle of the Velox boiler have been practically inexistent. We succeeded right from the beginning in finding for the novel conditions, such as the pressure combustion, the large rates of heat transmission, the forced circulation and the centrifugal separation, suitable design methods which are still employed unchanged. This applies particularly to the charging set which consists of a gas turbine and of an axial compressor, and which to-day is used not only for Velox boilers, but also extensively in exactly the same form for a number of other applications.

As already mentioned, the usual land boiler design of Velox is also suitable for marine purposes. Where, however, large outputs have to be accommodated in

[1] The Brown Boveri Review, 1941, page 221.

very restricted spaces, and where low weight is essential, a special design has been evolved which deserves careful consideration (Fig. 2). In this design, to keep down the height, everything has been removed from the combustion chamber which does not directly belong to the furnace or is not immediately concerned with the production of the heating gases. The combustion chamber consists only of a pressure-resisting boiler plate enclosure, lined with evaporator elements, with the burner and the air distributor mounted in the cover. The greater part of the evaporating surface is, however, accommodated in a horizontally disposed casing, joining at right angles the lower part of the combustion chamber. A similar casing contains the superheater. Because these two casings are of limited height, they may be located underneath the operating floor on which the charging set is mounted. The length of the charging set, therefore, determines the length of the tube casings.

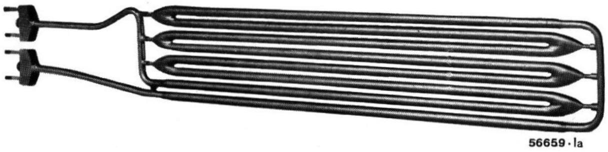


Fig. 5. — Superheater element.

Each superheater element consists of several tube banks which are kept in position by means of pins sliding in sockets.

Special attention was given to simplifying the individual components as much as possible. The horizontally arranged evaporator elements consist of simple tubes, swept externally by the hot gases and with water inlet and steam-water mixture outlet, and are both located in a single header (Fig. 3). The headers of all elements are connected in the usual Velox manner to a common collector. In order to employ for each element only a single bolt fixing, the water is fed into a tubular inlet which is concentrically located in the steam-water mixture outlet. Approximately 60—70 % of the total steam quantity is produced in the evaporator elements, the remainder being generated by the screening walls, that is to say, in tubular screens, which replace the refractory lining used in conventional boilers.

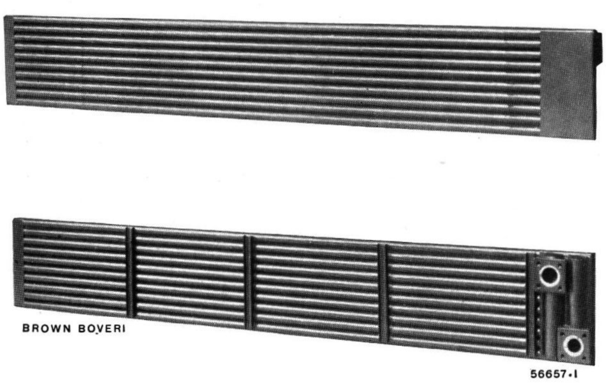


Fig. 4. — Wall screening tubes of the Velox design shown in Fig. 2.

The wall screens of combustion chamber and of the tube casing are constituted by tube walls which are built up of tubes connecting closely together.

The wall screens also (Fig. 4) are built up in the form of interchangeable elements, i. e., there is always a number of closely spaced tubes connected together by common inlet and outlet headers. Moreover, the screening walls of the superheater casing are generally also used as evaporator surfaces. In this way the heat quantity converted inside the superheater casing is increased, so that the temperature of the gases entering the casing may be made higher. This results in a small superheater and in reducing the drop of superheat temperature with decreasing load.

The construction of the superheater is exactly the same as in the case of land boilers (Fig. 5). A special feature of the superheater elements is the flexible mounting of the tubes, and the method of supporting them by means of pins free to slide in sockets. The latter are welded to the tubes, so that they are protected against excessive heating.

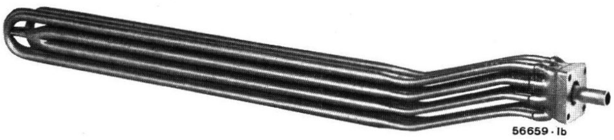


Fig. 3. — Evaporator element of the Velox design shown in Fig. 2.

Each element consists of six hairpin-shaped tubes, the inlet and outlet orifices of which are contained in a common head piece which is bolted to the collector in the combustion chamber by means of four substantial bolts. All heating surfaces of the Velox are replaceable and can be kept in the form of spares ready for immediate insertion.

It was also possible to simplify considerably the construction of the economizer elements (Fig. 6). By increasing the length of the gas path, either by allowing the economizer to extend deeply into the chimney, or by making it double-pass, the "hydraulic radius" or, what is the same thing, the spacing of the tubes could be increased. This means that the spacing between tubes may be made so large that, instead of employing as hitherto, flat connecting pieces from tube to tube, single common cross connecting pipes may be used for all upper and lower tube ends of an element. The slightly contracted tube ends are welded to these cross connecting pipes. Whereas previously the greater part of the economizer tubes were connected in parallel and accordingly the full rise in temperature of the feed water took place during its flow through a single element, the individual elements are now connected in series, the tubes of an element being connected mainly in parallel. This enables the total rise of temperature of the water to be distributed over several elements.

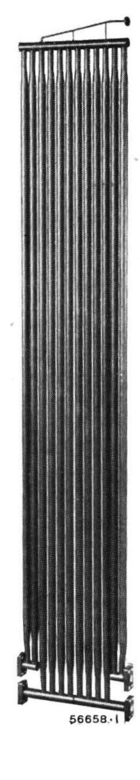


Fig. 6. — Economizer element.

Water inlet and outlet are at the same end of the element, so that the tubes are free to expand in one direction. Part of the tubes of one element may be traversed in the same direction. Such an arrangement is the most advantageous when in the greater part of the tubes the water flows in the opposite direction to the hot gas stream. The individual elements of an economizer are connected in series.

In ships it is particularly important to reduce the number of external pipe connections to a minimum. In this design of Velox boiler, the external pipes are limited to the connections between the combustion chamber and the separator, and to those between the circulating pump and the combustion chamber. Some of the connecting pipes can be made to form heating surfaces inside the boiler, whilst others can be accommodated under the shrouding. Special measures enable those connections which have to be brought out to be kept extremely short. Formerly, the differential thermal expansions of the combustion chamber casing, of the separator, pump, and piping had to be taken up in long connecting pipes and by expansion joints. In the new Velox construction, all pipes traversed by the circulating water, the steam and water mixture or the saturated steam, and which, therefore, are at approximately the same temperature, are rigidly joined together, their free expansion relative to the combustion chamber being made possible by means of bellows pieces at the points where the pipes pass through the walls of the casing. faces are located in two adjacent casings connected by a cross channel. If the charging set is mounted above these casings, care has to be taken to allow sufficient room for lifting their covers when dismantling the superheater and evaporator elements. As the individual weights of the covers and of the evaporator and superheater elements even in the largest units do not exceed 200 kg, and the dimensions of these parts are also quite small, the space which has to be left free above the casings is fixed more or less entirely by that required to provide ready accessibility. Table I gives data in regard to the floor space requirements and the overall height of such units. The space given in this table does not include that needed for the economizer, as the size and the method of mounting depends on the conditions of each particular case. The figures, therefore, hold good for both lightweight and warship boilers and for heavier merchant-ship boilers of high efficiency.

The small breadth enables even in the smallest of ships at least two, and in the wider ships more,

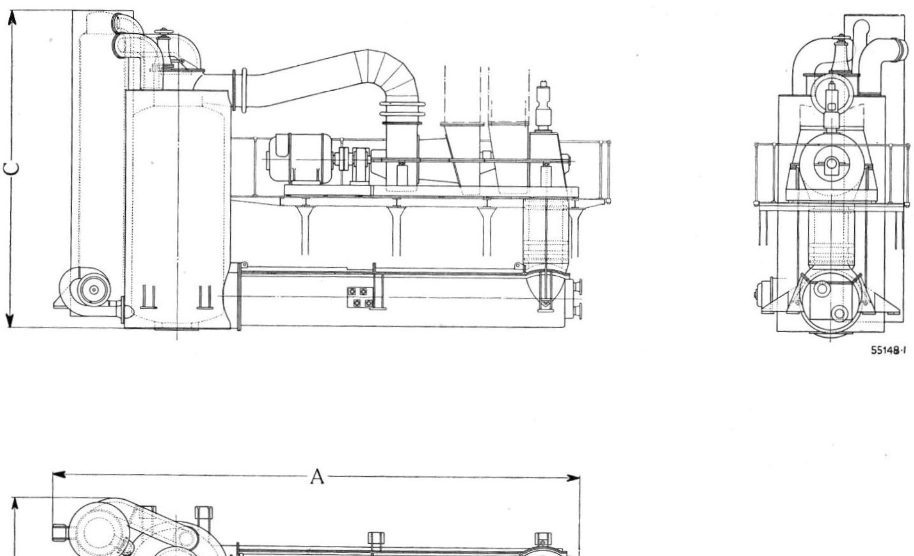


Fig. 7. — External view of a Velox steam generator with separator and superheater in a common casing.
The simplest arrangement is that in which the charging set is installed directly above the tube casing. For the main dimensions see table I.

The components of the Velox steam generator not specifically mentioned above are the same as in the case of the normal boiler design illustrated in Fig. 1. Their operating conditions are also the same. The new arrangement has not, therefore, brought about any fundamental change, so that the experience obtained with the construction hitherto employed can also be applied to the new design. In its most simple form the evaporator and the superheater are arranged one behind the other in the same casing (Figs. 2 and 7). Where the length of the space available does not allow this, the evaporator and superheater sur-

TABLE I

Space requirements of Velox boilers with evaporator and superheater in a common casing.

(In accordance with Fig. 7.)

Max. Output	Dimensions in mm		
	Length A	Breadth B	Height C
20 t/h	7800	2200	5000
40 t/h	8800	2400	5500
60 t/h	9500	2500	6000
100 t/h	10,500	2750	6500

boilers to be installed next to one another, whilst still leaving adequate room available for lateral fuel tanks. The most suitable arrangement is that in which steam generator and engines are accommodated in the same compartment. Examples of this type of layout are shown in the article on Brown Boveri turbines for the propulsion of merchant vessels on pages 232/233 of the present special number, to which the reader is referred.

As already mentioned, where the length has to be kept short, evaporator and superheater may be arranged side by side instead of behind one another. The two casings necessitated by this construction, are then connected together and the charging set is mounted above the superheater casing next to the combustion chamber.

Table II gives for different classes of service, the efficiencies and steam pressures and superheats and the approximate individual weights of Velox plants with evaporator and superheater in common casing expressed in kg per kg of steam generated per hour at maximum rating. These figures show clearly the extraordinarily low weights which can be based on for Velox plants. They are well below those of usual boiler designs, even where the Velox is compared with highly forced boilers, operating at considerably reduced efficiency at a forced rating.

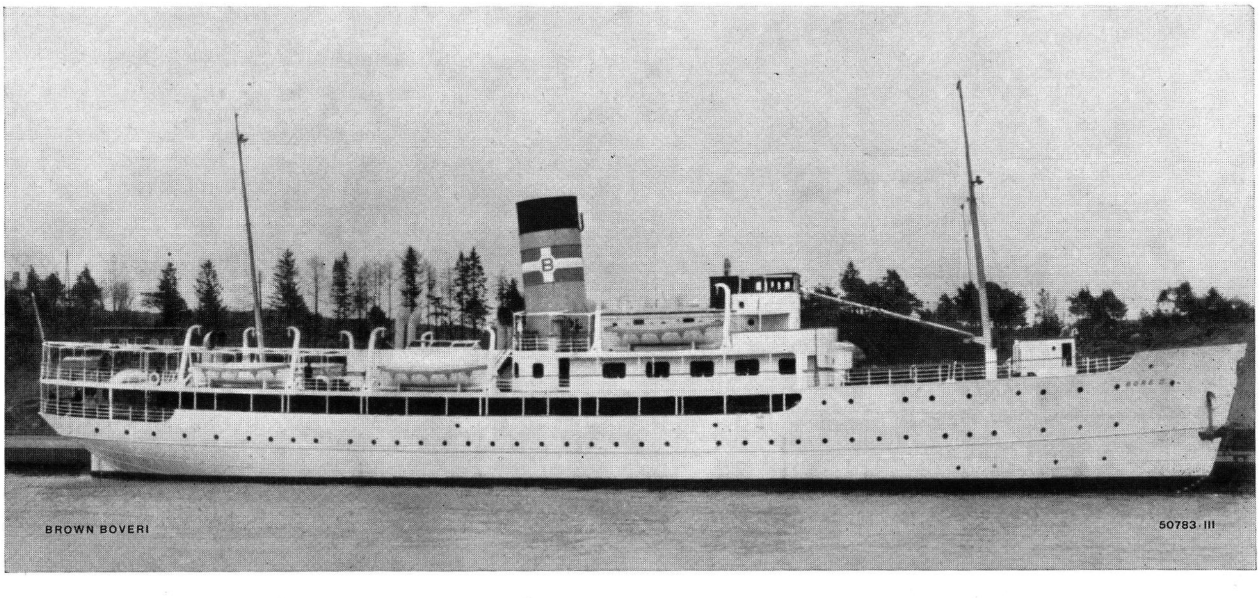
It might be expected that the extraordinarily low weights and small dimensions would lead to high material stresses. Such a conclusion is, however, incorrect. Both the heat and the mechanical stresses are no higher in the Velox boiler than in boilers of usual design, and they are considerably lower than in marine boilers, where maximum output is obtainable only at forced rating. The high specific outputs in the case of the Velox are due to the pressure combustion and to the high gas velocities. These result in small dimensions, but have no effect on the temperature conditions. The life of a Velox steam generator is, therefore, at least as great as that of the usual type of boiler. Its maintenance costs are, however, considerably lower. In regard to efficiency, ease of control, and speed of putting into service, no other type of boiler can be compared with it.

TABLE II.

Specific weights of Velox steam generators with evaporator and superheater in a common casing, ready for service, i. e., including water and oil contents as well as auxiliary machines, but excluding the feed pump.

Type of boiler	Light weight with small economizer		Normal construction (heavier design) with large economizer	
Efficiency of boiler plant % . . .	86—87		91·5—92·5	
Steam pressure kg/cm^2	40	75	40	75
Steam superheat °C	400	420	400	420
Specific weight in kg per kg of steam produced per hour				
for max. continuous rating of 20 t/h	1·12	1·25	1·50	1·70
40 t/h	0·94	1·05	1·25	1·40
60 t/h	0·80	0·90	1·10	1·25
100 t/h	0·68	0·76	0·90	1·00

(MS 887) *Dr. W. G. Noack. (Hv.)*



S. S. "Bore II" owned by the Ångfartygsaktiebolaget Bore, Åbo (Finland). This mail and passenger steamer was the first vessel to be exclusively equipped with Velox boilers.

The boiler plant comprises two Velox boilers each with an evaporation of 8 t/h at 16 kg/cm^2 abs and 320° C. The vessel operates a ferry service and is tied up during the day. Orthodox steam boilers would have had to be kept under steam during the whole of this time, whereas the Velox boilers are started up just before the departure of the vessel.

THE DISTRIBUTION OF PRESSURE AND THE REPARTITION OF STEAM FLOW IN MARINE TURBINES ON OVERLOAD.

Decimal Index 625.125.013

The most favourable points for the introduction of the overload steam can be determined by working out a number of alternatives. The numerical computation of the pressure distribution and of the repartition of the steam in the different portions of the turbine is both long and tedious. New graphic methods have, therefore, been devised, which enable all problems connected with the introduction of overload steam to be solved in a simple and elegant manner.

INTRODUCTION.

THE marine turbine, which has to operate over a large speed and load range, is always planned as a multiple-flow turbine. There are a number of ways of admitting the steam to marine turbines at heavy loads. They consist, in principle, of by-passing steam to the chamber of the impulse wheel, which latter receives steam from a number of nozzle groups, or, to one or more points beyond the wheel chamber. Every introduction of the steam at an intermediate point causes a change in the steam conditions at the beginning and at the end of the expansion over the blade groups, and thereby greatly affects the steam flow. Considerable changes then take place in the distribution of the heat drop and in the power generated in the individual turbine groups, at the same time affecting the operating economy.

In order to avoid throttling losses, the point of introduction of the steam should be as near as possible to the wheel chamber of the governing stage, but on the other hand, in order to achieve a high overload capacity and to avoid impairing the efficiency of preceding stages, it should be located as far away as possible from the first blade groups. The most economical arrangement for the path of the overload steam can be decided upon only after the distribution of the pressure and the repartition of the steam flow has been determined for the various loads coming into consideration, and after the corresponding steam consumptions have been calculated.

It seemed, therefore, desirable to devise a simple graphical method which would enable the changes in the steam quantities and pressures resulting from the introduction of additional steam to be followed with sufficient accuracy for all practical purposes, such method being based on the well-known steam cone law.[1]

[1] A. Stodola, Dampf- und Gasturbinen, 6th Edition, p. 262.

List of symbols:

G = The steam quantity flowing through the turbine at partial load.

G_0 = The full-load steam quantity (the overload valves are closed).

G_a = The quantity of steam flowing through the nozzles of the regulating stage.

G_1 = The additional steam quantity entering the wheel chamber.

G' = The steam quantity entering the intermediate stage.

G_t = The total amount of steam admitted to the turbine.

p_a = The steam pressure in front of the nozzles of the regulating stage with regulating valves fully open (initial pressure).

p_e = Final pressure at the end of the expansion of the steam in the turbine (condenser pressure).

p_1 = Wheel chamber pressure, i. e., the steam pressure in front of the first group of blades after the regulating stage.

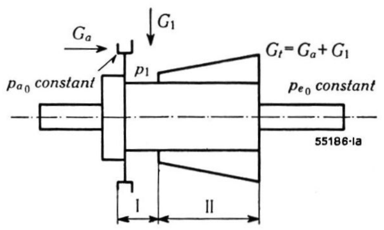
p_z = The steam pressure in the intermediate stage (interstage pressure).

The steam quantities are expressed in kg/s or kg/h, and the pressure in kg/cm^2 abs. The index $_0$ indicates that a corresponding magnitude refers to the normal condition for which the blading has been designed. Unless otherwise mentioned, it is assumed in the following that the initial pressure $p_a = p_{a_0}$ and the final pressure $p_e = p_{e_0}$ remain constant.

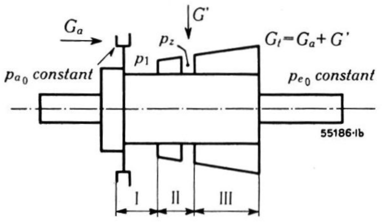
The principal cases of introduction of additional steam occurring in marine steam turbine practice are indicated diagrammatically in Figs. 1a and 1b, and are dealt with successively below. In all these cases the governing valves are assumed to be fully open, the overload steam being introduced either into the wheel chamber of the governing stage (case 1a) or into an intermediate stage after the wheel chamber (case 1b).

I. INTRODUCTION OF THE OVERLOAD STEAM INTO THE WHEEL CHAMBER (Fig. 1a).

In this case the governing stage must be regarded as the up-stream part, and the remaining blading as the down-stream part, which parts are denoted for brevity by I and II respectively. For part I, the wheel chamber pressure p_1 represents the final pressure. For part II, it represents the initial pressure.



a. Case of live steam supplied to the wheel chamber.



b. Case of live steam supplied to any intermediate stage.

Fig. 1. — Diagrammatic representation of the introduction of the overload steam with steam quantity and steam pressure notations.

I, II, III Turbine sections.
G_a The steam quantity supplied to the nozzles of the governing stage.
G_1 The steam quantity supplied to the wheel chamber.
G' The steam quantity supplied to the intermediate stage.
G_t The total steam quantity supplied to the turbine (total steam quantity).
p_a Steam pressure in front of the nozzles of the governing stage with nozzle valves fully open (initial pressure).
p_e Pressure at the end of the expansion in the turbine (condenser pressure).
p_1 Wheel chamber pressure, i. e., the steam pressure in front of the first blade group after the governing stage.
p_z Steam pressure in the intermediate stage (interstage steam pressure).

The index $_0$ indicates that the corresponding magnitude is referred to the normal operating condition.

(a) Let the turbine be designed for a steam quantity G_0, for an initial pressure in front of the nozzles p_{a_0}, a wheel chamber pressure p_{1_0}, where $p_{1_0} > 0 \cdot 546\, p_{a_0}$ and for an end pressure p_{e_0}.* It is required to find the new wheel chamber pressure p_1 and the steam fractions G_a and G_1 for a total steam quantity G_t. Let us consider first the general case, and let us assume that both the initial and the final pressure alter slightly, the alteration being such that the pressure in front of the nozzles $p_a < p_{a_0}$ and that the final pressure $p_e > p_{e_0}$.

Construction (Fig. 2).

Draw along the ordinate axis to any but to the same scale: $OP = p_{a_0}$; $ON = p_a$; $OB = p_{1_0}$; $OE = p_e$ and $OD = p_{e_0}$. From the centre O, strike the arc of radius $r = OB = p_{1_0}$ up to the point of intersection C with the horizontal DC ($y = p_{e_0}$). Through the point E ($y = p_e$) draw a second horizontal and make $EF = \alpha DC$, where $\alpha = G_t/G_0$; the radius OF then represents the required wheel chamber pressure p_1. In order to determine graphically the steam fraction G_a, strike about the origin

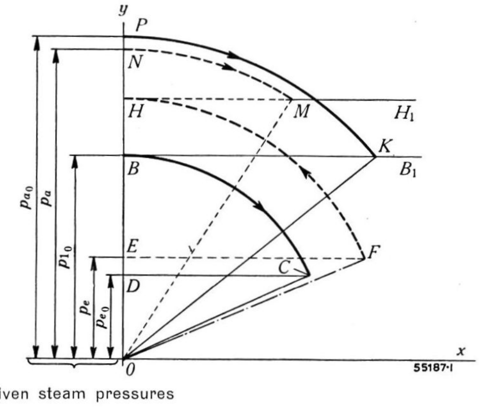


Fig. 2. — Live steam supplied to the wheel chamber $p_{1_0} > 0 \cdot 546\, p_{a_0}$; $p_a \neq p_{a_0}$ and $p_e \neq p_{e_0}$.

Determination of the wheel chamber pressure $p_1 = OF = OH$ and of the steam fraction $G_a \dfrac{HM}{BK} G_0$.

$$EF = \frac{G_t}{G_0} DC.$$

O a circular arc of radius $r = OF$ up to the point of intersection H with the ordinate axis. Through the point B ($y = p_{1_0}$) and the point H just found, draw two horizontal lines BB_1 and HH_1, and strike about the centre O two arcs of radius $r_1 = OP = p_{a_0}$ and $r_2 = ON = p_a$ up to the points of intersection K and M with the above horizontal lines, thus obtaining the steam fraction G_a, which is given by the relation $G_a = \dfrac{HM}{BK} G_0$. The remaining steam fraction G_1 which has to be supplied to the wheel chamber by means of the overload valve, is evidently $G_1 = G_t - G_a$.

Mathematical Proof.

According to the steam cone law, the following pair of equations has to be satisfied:—

For part I:
$$\frac{G_a^2}{G_0^2} = \frac{p_a^2 - p_1^2}{p_{a_0}^2 - p_{1_0}^2} \tag{1}$$

For part II:
$$\frac{G_t^2}{G_0^2} = \frac{p_1^2 - p_e^2}{p_{1_0}^2 - p_{e_0}^2} \tag{2}$$

In the equations (1) and (2) both the full-load steam quantity G_0 and the overload steam quantity G_t are assumed to be known; the unknown quantities are the steam fraction G_a and the wheel pressure p_1.

The graphical construction shown in Fig. 2 gives for the triangles OEF and ODC the following relations:

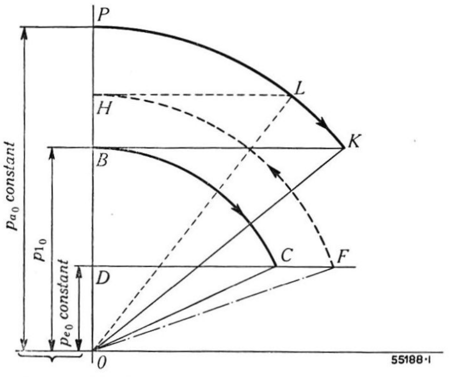
$$\frac{EF^2}{DC^2} = \frac{OF^2 - OE^2}{OC^2 - OD^2} \tag{3}$$

or

$$\frac{G_t^2}{G_0^2} = \frac{OF^2 - p_e^2}{p_{1_0}^2 - p_{e_0}^2} \tag{3'}$$

Comparison of (3′) with equation (2) shows that the graphically determined length OF does indeed represent the wheel chamber pressure p_1.

* The case of a governing stage operating with a heat drop greater than that corresponding to the critical pressure ratio, is dealt with under d.



Given steam pressures

Fig. 3. — Steam supplied to the wheel chamber.
$p_{1_0} > 0{\cdot}546\ p_{a_0}$; $p_{a_0} =$ const. $p_{e_0} =$ const.

Determination of the wheel chamber pressure $p_1 = OF = OH$, and of the steam fraction $G_a = \frac{HL}{BK} G_0$.

$$DF = \frac{G_t}{G_0} DC.$$

The triangles OHM and OBK (Fig. 2) give further

$$\frac{HM^2}{BK^2} = \frac{OM^2 - OH^2}{OK^2 - OB^2} \tag{4}$$

or

$$\frac{HM^2}{BK^2} = \frac{p_a{}^2 - p_1{}^2}{p_{a_0}{}^2 - p_{1_0}{}^2} \tag{4'}$$

Comparison of (4′) with (1) shows that the ratio of the graphically determined lengths is

$$\frac{HM}{BK} = \frac{G_a}{G_0}, \text{ whence } G_a = \frac{HM}{BK} G_0.$$

(b) If the steam pressures $p_{a_0} =$ const. and $p_{e_0} =$ const., as almost always is the case, the construction becomes simplified (see Fig. 3). Here the distance $DF = \frac{G_t}{G_0} DC$ is set off, the radius OF gives, as before, the new wheel chamber pressure p_1, and the steam fraction $G_a = \frac{HL}{BK} G_0$.

The process shown in Figs. 2 and 3 may be replaced by the following illustrative construction, which for some applications is preferable.

(c) Let the turbine be designed for a steam quantity $G_0 = OK_0$ (Fig. 4), the initial pressure in front of the nozzles $p_{a_0} = OL =$ const., for the wheel chamber pressure $p_{1_0} = K_0 D_0$, where $p_{1_0} > 0{\cdot}546\ p_{a_0}$, and for a constant condenser pressure p_{e_0}. It is required to find the wheel chamber pressure p_1, and the steam fractions G_a and G_1, corresponding to the total overload steam quantity G_t.

Construction:

Strike the arc of radius $r = OL = p_{a_0}$ about the centre O and through the given point D_0 (coordinates $x = OK_0 = G_0$ and $y = K_0 D_0 = p_{1_0}$) draw a horizontal line $D_0 B$ and denote the point of intersection with the above arc by A_0. Now draw the elliptical arc LD_0, the abscissæ of which are increased over those of the arc LA_0 in the ratio of BD_0 to BA_0.

The standard graphical method of construction for the ellipse may also be employed, since the minor semi-axis $= OL = p_{a_0}$ and the major semi-axis $= ON = p_{a_0} \frac{BD_0}{BA_0}$. The length ON can be read off graphically, since N is the point of intersection of the extension of radius OA_0 with the perpendicular $K_0 D_0$.

The condenser pressure may be assumed to be practically equal to 0. For the down-stream part II, the wheel chamber pressure represents the initial pressure, which according to the well-known proportionality law is represented by the straight line OF'' connecting the origin O with the given point D_0.

By means of the diagram Fig. 4, it is possible to read off directly the wheel chamber pressure p_1 and the steam fractions corresponding to any overload steam quantity G_t. For example, for a total steam quantity, $G_t = OK'$ we obtain, on following the line $K'MDF$, the wheel pressure $p_1 = K'M$ and the steam fractions $G_a = LF$ and $G_1 = FF'$.

The abscissæ of the point of intersection F'' gives the steam quantity $G_t = OK''$ for which the wheel chamber pressure $p_1 = p_{a_0}$ and for which the steam quantities $G_a = 0$ and $G_1 = G_t = OK''$.

It is immaterial for this process and for all these here described, what scale has been chosen for the base length $OK_0 = G_0$. All that is necessary is that the scale of the pressures shall be the same. The steam quantities and pressures can then be read off directly to the chosen scales.

Mathematical Proof:

The steam cone law gives the following two equations:—

For part I:

$$G_a{}^2 (p_{a_0}{}^2 - p_{1_0}{}^2) - G_0{}^2 (p_{a_0}{}^2 - p_1{}^2) = 0 \tag{5}$$

For part II:

$$(G_a + G_1)^2 (p_{1_0}{}^2 - p_{e_0}{}^2) - G_0{}^2 (p_1{}^2 - p_{e_0}{}^2) = 0 \tag{6}$$

where $G_a + G_1 = G_t$.

If $p_{e_0} \approx 0$, equation (6) gives the well-known proportionality law between steam quantity and steam pressure, which is evidently represented graphically by the radius OF'' in Fig. 4. From equation (5) we obtain:

$$p_1 = \sqrt{p_{a_0}{}^2 - \frac{G_a{}^2}{G_0{}^2} (p_{a_0}{}^2 - p_{1_0}{}^2)} \tag{7}$$

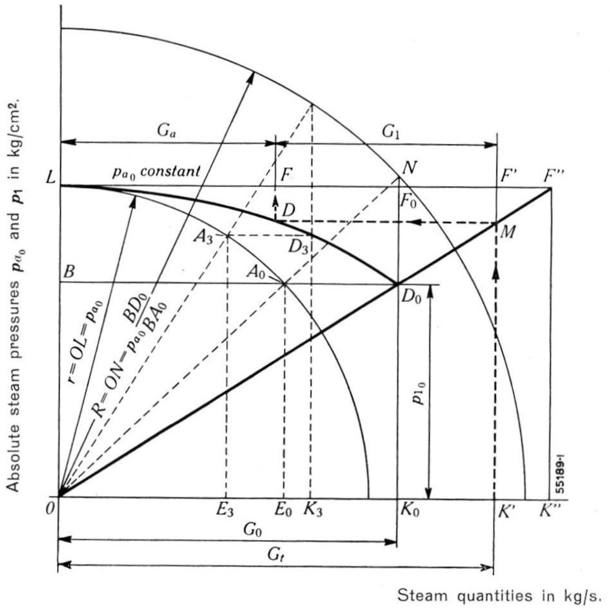


Fig. 4. — Steam supplied to the wheel chamber $p_{1_0} > 0{\cdot}546\ p_{a_0}$ and $p_{e_0} \eqsim 0$.

The elliptic arc LD_3D_0 represents the wheel chamber pressure as the final pressure $p_1 = f_1\ (G)$ for the preceding part I with a constant initial pressure p_{a_0}.

The straight line OF'' represents the wheel chamber pressure as the initial pressure $p_1 = f_2\ (G)$ for the subsequent part II with a constant final pressure $p_{e_0} \eqsim 0$.

With a given total steam quantity $G_t = OK'$ the corresponding wheel chamber pressure $p_1 = K'M$ and the steam fractions $G_a = LF$ and $G_1 = FF'$ are determined by the broken line $K'MDF$.

The graphically determined pressure curve LD_3D_0 gives for every value of G_a the corresponding value of the wheel chamber pressure p_1; for example, to an assumed steam quantity $G_a = OK_3$ corresponds a definite steam pressure $p_1 = K_3D_3$, whereby equation (7) always remains satisfied. In order to prove this, let us consider the auxiliary triangles OA_3E_3 and OA_0E_0 (Fig. 4). We then obtain for D_3

$$p_1^2 = K_3D_3^2 = E_3A_3^2 = OA_3^2 - OE_3^2 = p_{a_0}^2 - OE_3^2 \qquad (8)$$

and from the law of the ellipse there follows the relation

$$\frac{OE_3^2}{OK_3^2} = \frac{OE_0^2}{OK_0^2} = \frac{p_{a_0}^2 - p_{1_0}^2}{G_0^2} \qquad (9)$$

whence

$$OE_3^2 = \frac{G_a^2}{G_0^2}(p_{a_0}^2 - p_{1_0}^2) \qquad (10)$$

Substituting from (10) in (8) we obtain

$$p_1^2 = p_{a_0}^2 - \frac{G_a^2}{G_0^2}(p_{a_0}^2 - p_{1_0}^2) \qquad (11)$$

that is, as given by equation (7).

(d) If in normal operation the wheel chamber pressure $p_1 \leq 0{\cdot}546\ p_{a_0}$, i.e., the regulating stage, operates with more than the critical heat drop, and the pressure remains as before $p_{e_0} \eqsim 0$, it is possible to proceed as follows:—

The variable flow through the nozzles is determined by the well-known ellipse law $G_a = \chi\, G_{max} = \chi\, G_0$ where the coefficient χ, using the above notation for the steam pressures, is given by

$$\chi = \frac{p_{a_0} - p_1}{p_{a_0} - 0{\cdot}546\, p_{a_0}} \sqrt{2\,\frac{p_{a_0} - 0{\cdot}546\, p_{a_0}}{p_{a_0} - p_1} - 1} \qquad (12)$$

or

$$\chi = \frac{1 - \dfrac{p_1}{p_{a_0}}}{0{\cdot}454} \sqrt{\frac{0{\cdot}908}{1 - \dfrac{p_1}{p_{a_0}}} - 1} \qquad (12')$$

i. e., for given values of G_a/G_0, we obtain definite values of the pressure ratio p_1/p_{a_0} which are tabulated in table I. If the lengths $OK_0 = G_0$ and $OL = p_{a_0}$ are looked upon as unity, and if the values of p_1 according to table I are set off against the corresponding abscissæ G, the curve line LD_0K_0 represents the capacity of the nozzles.

Fig. 5 shows that the flow through the nozzles does not change up to the total steam quantity $G_t = OK'$, as the expansion remains above the sound velocity limit.

TABLE I.

Nozzle flow capacity as a function of the pressure ratio according to equation (12′).

$\chi = \frac{G}{G_{max}} = \frac{G_a}{G_0}$	Pressure ratio $\frac{p_1}{p_{a_0}}$ from (12′)	$\chi = \frac{G}{G_{max}} = \frac{G_a}{G_0}$	Pressure ratio $\frac{p_1}{p_{a_0}}$ from (12′)
0·00	1·000	0·55	0·926
0·05	0·999	0·60	0·908
0·10	0·998	0·65	0·890
0·15	0·995	0·70	0·869
0·20	0·991	0·75	0·845
0·25	0·986	0·80	0·819
0·30	0·979	0·85	0·785
0·35	0·971	0·90	0·745
0·40	0·962	0·95	0·690
0·45	0·951	1·00	0·546
0·50	0·940		

With larger steam quantities, the wheel chamber pressure is greater than $p_1 = 0{\cdot}546\ p_{a_0}$, so that $G_a < G_0$.

The wheel chamber pressure $p_1 = K''M$ and the steam fractions $G_a = LF$ and $G_1 = FF''$ corresponding to any particular overload steam quantity $G_t = OK''$

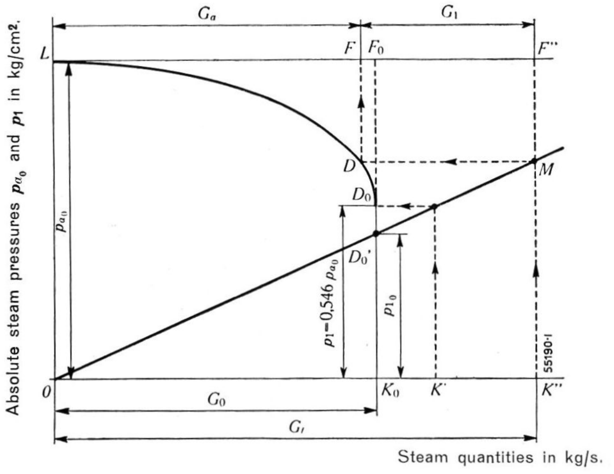


Fig. 5. — Live steam supply to the wheel chamber $p_{1_0} \lesseqgtr 0{\cdot}546\ p_{a_0}$ and $p_{e_0} \approx 0$.

The curve LD_0K_0 represents the wheel chamber pressure $p_1 = f_1\,(G)$ as the back pressure for the preceding part I with constant initial pressure p_{a_0}. [The capacity of the nozzles $G_a = \%\ G_{max}$, where % from equation (12')]. The straight line OM represents the wheel chamber pressure $p_1 = f_2\,(G)$ as the initial pressure for the subsequent part II with a constant final pressure $p_e \approx 0$.
With a given total steam quantity $G_t = OK''$ the corresponding wheel chamber pressure $p_1 = K''M$ and the steam fractions $G_a = LF$ and $G_1 = FF''$ are determined by the broken line $K''MDF$.

can be read off directly by the following broken line $K''MDF$. It is also seen that the amount of steam flowing through the nozzles has become less and that this reduction of the amount FF_0 has to be made good by the overload valve.

II. ADMISSION OF THE ADDITIONAL STEAM INTO THE INTERMEDIATE STAGE BEYOND THE WHEEL CHAMBER (Fig. 1b).

(a) Let the turbine be designed for a normal full load steam quantity $G_0 = OK_0$ (Fig. 6), for the initial pressure in front of the nozzles $p_{a_0} = OL = $ const., for the wheel chamber pressure $p_{1_0} = K_0\,D_0$, where $p_{1_0} > 0{\cdot}546\ p_{a_0}$, for an intermediate stage pressure $p_{z_0} = K_0E_0$ and for a condenser pressure $p_{e_0} = $ const. Let the pressure drop between the wheel chamber and the additional steam inlet point $p_{1_0} - p_{z_0} = D_0\,E_0$ be distributed over several stages, every stage operating below sound velocity. It is required to find the inter-stage pressure p_z, the wheel chamber pressure p_1 and the steam fractions G_a and G', corresponding to a total given steam quantity G_t.

Construction:

Draw the back-pressure curve $p_1 = f_1\,(G)$ for the wheel chamber in exactly the same manner as illustrated in Fig. 4. This curve is denoted in Fig. 6 by LD_0. Since it may be assumed that the final pressure $p_{e_0} \approx 0$, the interstage pressure varies in a linear manner. In Fig. 6 this pressure is indicated by the line OE_0, which is to

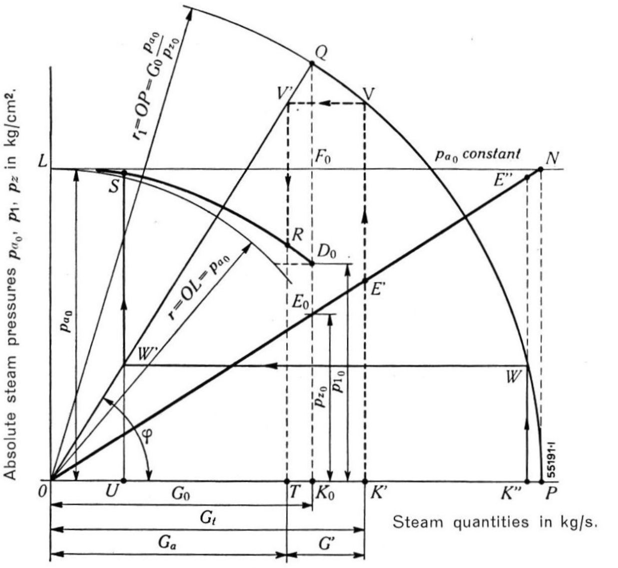


Fig. 6. — Live steam supply to a subsequent intermediate stage $p_{1_0} > 0{\cdot}546\ p_{a_0}$ and $p_{e_0} \approx 0$.

Determination of the steam pressures and steam quantities for a given total steam quantity $G_t = OK'$ by means of the broken line $K'\,VV'\,R$:

Wheel chamber pressure p_1	$= TR$
Interstage pressure p_z .	$= K'\,E'$
Steam fraction I G_a . .	$= OT$
Steam fraction II G' . .	$= TK'$.

be extended to its point of intersection N with the horizontal line $p_{a_0} = $ const. Project the point of intersection N on to the abscissæ axis to P; with the radius $r_1 = OP$ strike the arc about the origin O up to the point of intersection Q with the straight line K_0Q ($x = G_0 = OK_0$). (In case the projection point P falls outside the drawing, determine the radius $r_1 = OP = G_0\ \frac{p_{a_0}}{p_{z_0}}$ by calculation.)

Connect the point of intersection Q with the origin O by means of the radius OQ, which together with the arc PQ enables the pressures p_z, p_1 and the steam quantities G_a, G' corresponding to any given value of the total steam quantity G_t to be determined immediately. For example, for a given total steam quantity $G_t = OK'$, the broken line $K'VV'R$ gives the values: $p_z = K'E'$; $p_1 = TR$; $G_a = OT$ and $G' = TK'$. For any other total steam quantity $G_t = OK''$, a similar broken line $K''WW'S$ gives the values: $p_z = K''E''$; $p_1 = US$; $G_a = OU$ and $G' = UK''$.

The abscissæ of the point of intersection N, i. e., $G_t = OP$, gives the maximum steam quantity, for which the steam pressures are $p_1 = p_z = p_{a_0}$ and the steam quantities are $G_a = 0$ and $G' = G_t$, i. e., when the turbine parts I and II cease to develop power and act only as a brake.

Mathematical Proof:

The analytical determination of the four unknowns p_z, p_1, G_a and G' is effected by means of the following four equations:—

For part I:

$$G_a^{\,2}\,(p_{a_0}^{\,2} - p_{1_0}^{\,2}) - G_0^{\,2}\,(p_{a_0}^{\,2} - p_1^{\,2}) = 0 \quad (13)$$

For part II:

$$G_a^2 (p_{1_0}^2 - p_{z_0}^2) - G_0^2 (p_1^2 - p_z^2) = 0 \quad (14)$$

For part III:

$$(G_a + G')^2 (p_{z_0}^2 - p_{e_0}^2) - G_0^2 (p_z^2 - p_{e_0}^2) = 0 \quad (15)$$

and

$$G_a + G' = G_t \quad (16)$$

If it may be assumed that $p_{e_0} \lesssim 0$, equation (15) takes the form of the well-known proportionality law.

$$p_z = \frac{G_t}{G_0} p_{z_0} \quad (15')$$

which is represented by the radius ON.

Addition of equations (13) and (14) gives

$$G_a^2 (p_{a_0}^2 - p_{z_0}^2) - G_0^2 (p_{a_0}^2 - p_z^2) = 0 \quad (17)$$

This equation gives for the steam fraction G_a upon inserting the value for p_z given by (15'):

$$G_a = \sqrt{\frac{G_0^2 \frac{p_{a_0}^2}{p_{z_0}^2} - G_t^2}{\frac{p_{a_0}^2}{p_{z_0}^2} - 1}} \quad (18)$$

We shall now proceed to prove that the steam fraction G_a corresponding to any overload steam quantity $G_t = OK'$ really is equal to OT according to equation (18).

The angle φ formed by the radius OQ and the abscissæ axis is defined by the relation

$$\tan\varphi = \frac{K_0 Q}{OK_0} = \frac{\sqrt{r_1^2 - OK_0^2}}{OK_0} = \frac{\sqrt{G_0^2 \frac{p_{a_0}^2}{p_{z_0}^2} - G_0^2}}{G_0} = \sqrt{\frac{p_{a_0}^2}{p_{z_0}^2} - 1} \quad (19)$$

but from the triangle OTV' we have

$$OT = \frac{TV'}{\tan\varphi} = \frac{K'V'}{\tan\varphi} = \frac{\sqrt{r_1^2 - OK'^2}}{\tan\varphi} = \frac{\sqrt{G_0^2 \frac{p_{a_0}^2}{p_{z_0}^2} = G_t^2}}{\sqrt{\frac{p_{a_0}^2}{p_{z_0}^2} - 1}} \quad (20)$$

which is exactly the same as the analytically derived equation (18) for the steam fraction G_a.

Equation (13) is identical to equation (5) which, as already shown, represents the pressure curve LD_0 for the wheel chamber. Since this curve fixes graphically the relation between the steam pressures p_1 corresponding to every value of the steam quantity G_a, to any value of $G_a = OT$ there can correspond only the wheel chamber pressure $p_1 = TR$. It follows from this that the process developed in Fig. 6 satisfies the system of equations (13)—(16).

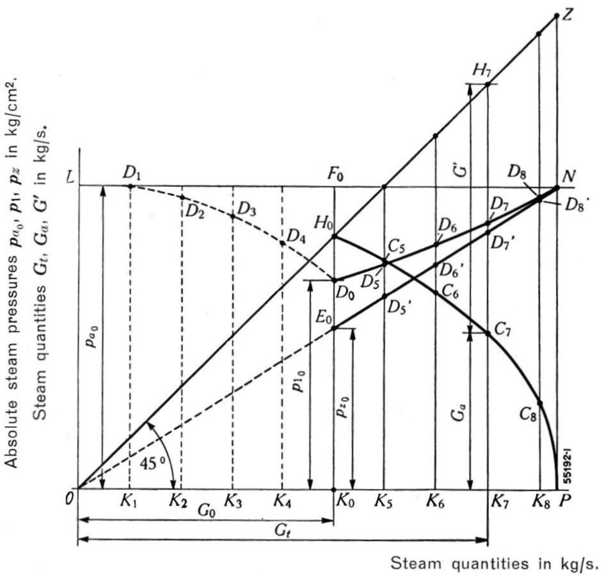


Fig. 7. — Live steam supply to a subsequent intermediate stage $p_{1_0} > 0{\cdot}546\ p_{a_0}$ and $p_{e_0} \lesssim 0$.

Steam pressure and steam fractions as a function of G_t.
Curve $D_0 D_5 D_6 \ldots\ N$ = Wheel chamber pressure p_1,
Curve $H_0 C_5 C_6 \ldots\ P$ = Steam fraction G_a.
OZ = Auxiliary radius.
ON = Steam pressure in front of part III.

In order to give a clear idea of the way in which the quantities p_1, G_a and G' vary with the total steam quantity G_t, they may be plotted as a function of G_t. Fig. 7 shows such a diagram.

The curve $D_0 D_5 D_6 \ldots\ldots N$ shows the relation $p_1 = f_1(G_t)$. The ordinate intercepts between this curve and the radius ON give the variable pressure drop $p_1 - p_z = f_2(G_t)$, which does work in turbine part II. The ordinate intercepts between the curve $D_0 D_5 D_6 \ldots N$ and the horizontal $F_0 N$ shows the pressure drop doing work in part I. The curve $H_0 C_5 C_6 \ldots P$ represents the steam quantity $G_a = f_3(G_t)$. The ordinate intercepts between this curve and the auxiliary radius OZ which cuts the abscissae axis at the origin O at the angle of 45° give the value of the steam quantity $G' = f_4(G_t)$.

The graphical relations between the total steam quantity and the values p_1, p_z, G_a and G' shown in Fig. 7 form the basis of calculations for the determination of the steam consumption, the efficiency, etc., which is then carried out in the usual manner.

(b) If the turbine is designed for the same conditions as under IIa, but with the difference that $p_{1_0} < 0{\cdot}546\ p_{a_0}$, the values of the table I are to be used as they were for the case of steam admission into the wheel chamber. In other respects, the procedure is the same developed in Fig. 6.

(MS 857) *W. Rohrbach. (Hv.)*

CONTROL AND SAFETY FEATURES OF BROWN BOVERI MARINE TURBINES.

Decimal Index 621.125—5

Marine turbines for small outputs are usually hand-controlled by means of a common hand-wheel operating both forward and reverse valves. For large outputs, especially for warships, with greatly varying loads, and hence with a relatively large number of valves, hydraulic control offers considerable advantages: Every running point can be set and all manœuvres can be carried out with speed, safety, and without effort by means of a hand-wheel at the control stand. The hydraulic control enables, moreover, the machinery to be controlled from any point of the ship. It also enables all safety features to be combined in the simplest possible manner. The individual valves may be located most conveniently on the turbine cylinders, thereby avoiding awkward high-pressure piping and linkages.

CONTRARY to the land turbine, the speed, and hence the power, as well as the direction of rotation of marine turbines must be adjusted by hand according to the orders of the telegraph. A particularly important point is the reversing. This must be effected quickly and surely, without overstressing the machinery (turbine blading and gear teeth), as the safety of the ship frequently depends thereon.

I. HAND CONTROL OF MARINE TURBINES.

In the case of turbine plants of small outputs, hand control (Figs. 1 and 2) is sufficient for all practical purposes. The live steam supply to the turbines flows from the main steam pipe through the emergency stop valve A (Fig. 1). During a long stop (when in port), this valve is screwed down on to its seat by means of a spindle provided with a hand-wheel. Before putting the plant into service again, the spindle is first lifted, but the valve does not open until sufficient oil pressure is available, and it remains open only as long as an adequate pressure is maintained in the oil system.

Steam then flows to the forward or reverse turbines through the manœuvring valves B or C, only one of which can open at a time. This is ensured by the reversing gear D which is operated by only one hand-wheel, so that faulty manœuvres are excluded.

The emergency stop valve A is under the control of the safety governors F and G, as well as of the speed limiting device K. The latter prevents a certain maximum value from being exceeded when the propeller comes out of the water during heavy seas, thus obviating unnecessary tripping of the emergency governors and shutting down of the turbines. If the speed exceeds the value which can be adjusted by the hand-wheel L, oil is allowed to escape from the pipe system O, the oil pressure under this piston M falls, the valve A closes somewhat and throttles the supply of steam to the turbine until the speed falls again. The emergency stop valve A accordingly operates temporarily as a regulating valve. The hand-wheel A enables any particular propeller speed, and hence ship speed, to be adjusted, and maintained constant.

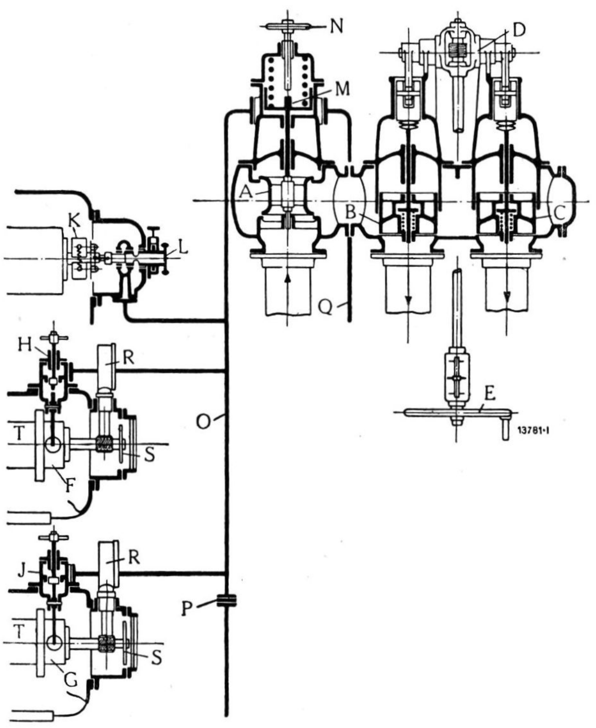


Fig. 1. — Hand-controlled marine turbine plant.

A. Emergency and main stop valve.
B. Ahead control valve.
C. Astern control valve.
D. Reversing gear.
E. Hand-wheel with indicator device.
F. Emergency governor of the high-pressure turbine.
G. Emergency governor of the low-pressure turbine.
H. Hand trip device for the high-pressure turbine.
I. Hand trip device for the low-pressure turbine.
K. Automatic speed limiter and speed regulator on an intermediate shaft of the main gear.
L. Speed adjusting device.
M. Servo-motor.
N. Hand-wheel for screwing the main stop valve down in the closed position.
O. Oil pressure pipe.
P. Orifice plate.
Q. Oil return pipe.
R. Tachometer.
S. Direction indicator.
T. Shaft end of the high-pressure and of the low-pressure turbines.

Even this simple, mechanical hand-control system guards the turbine against excessive speed, inadequate lubricating oil pressure and against sudden rise of the speed when the propeller comes out of the water in heavy seas.

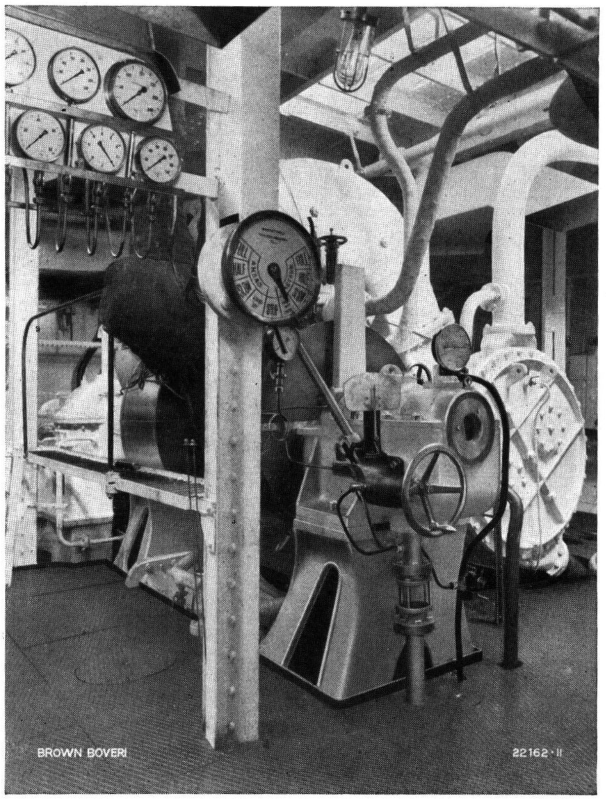


Fig. 2. — Brown Boveri marine turbine plant with hand control.

For plants of small output simple hand control of the turbine is sufficient for all requirements. The control hand wheel and the indicating panel above it for the different speeds are fixed to the turbine bearing pedestal. A ship's telegraph (left top) as well as the necessary measuring instruments for the steam and oil pressures, together with a tachometer, driven from the end of the turbine shaft, complete the control stand.

The main stop valve A therefore serves three purposes and accordingly operates:

1. As an emergency stop valve upon tripping of the emergency governor or of the quick-stop devices H or J.
2. As an automatic regulating device for maintaining constant or for limiting the speed of the propeller shaft.
3. As a guarantee that the turbine plant can be put into service, and kept in service only as long as sufficient lubricating oil pressure is present.

II. HYDRAULIC CONTROL OF MARINE TURBINES.

The number of nozzle valves on large marine turbines, especially in warship plants, is relatively great; in addition, there are overload and change-over valves. Briefly, the operation of all these valves by hand during quick manœuvring takes up time and requires considerable effort. There is also the danger of faulty manœuvres which may be prevented by means of interlocking devices; which, however, lead to complications.

We have, therefore, already for many years[1], applied to marine turbines the principle of our rodless oil pressure governing, as used on our land turbines since 1905. The arrangement and the method of operation of a hydraulic marine turbine governing system are shown schematically in Fig. 3.

The operation of the emergency stop valve (4) as well as of the overspeed governors (12) and (13) is exactly the same as that described in Fig. 1; here again a speed limiting device is provided such as that shown under K (Fig. 1), although for the sake of simplicity, it has been omitted from Fig. 3.

The novelty lies in the fact that the nozzle valves of the forward and reverse turbines are operated from

[1] For the first time in 1926, simultaneously in a number of plants, see Brown Boveri Review 1930, p. 49. Also Ingeniere Naval (Madrid) 1934, p. 138: E. Klingelfuss "Perfeccionamientos introducidos en las turbinas del tipo Marino Brown Boveri".

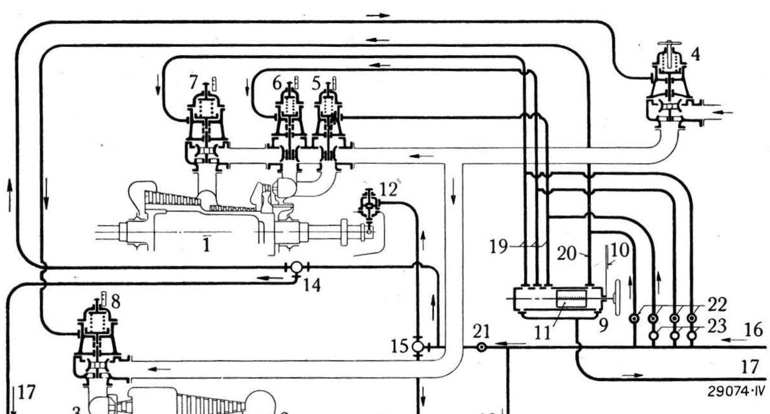
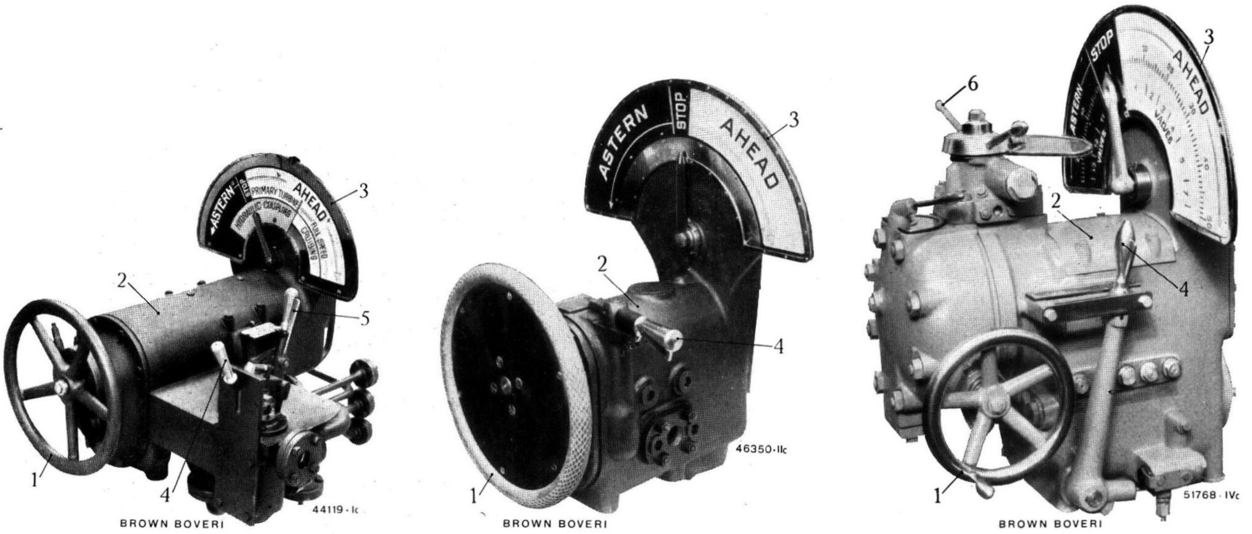


Fig. 3. — Diagram of a hydraulic marine turbine control system.

1. High-pressure ahead turbine.
2. Low-pressure ahead turbine.
3. Astern turbine.
4. Emergency and main stop valve.
5–7. Nozzle valves of the ahead turbine.
8. Nozzle valve of the astern turbine.
9. Manœuvring apparatus.
10. Hand-wheel for 9 with indicator plate.
11. Piston valve of the manœuvring apparatus.
12–13. Emergency governor and hand-trip device (quick-stopping device).
14. Additional trips at different points of the engine room.
15. Three-way cock.
16. Pressure oil from the main oil pump.
17. Oil return to the oil tank.
18. Lubricating oil to the bearings and gear.
19. Servo-motor oil pipes to the ahead valves.
20. Servo-motor oil pipes to the astern valves.
21–23. Orifice plates.

By rotating the single handwheel (10) into the ahead or astern directions, all manœuvres on the largest marine turbine plants can be effected without effort and securely by a single man. At the same time all safety devices can be combined in a simple way with the hydraulic governing system. All valves may also be operated by hand in an emergency.



Figs. 4a—4c. — Different designs of manœuvring apparatus for hydraulically controlled marine turbines.

1. Control hand wheel.
2. Control device with control valve.
3. Indicator.
4. Quick-trip device (hand trip for emergency stop valve).
5. Device for cutting in and out of the cruising turbine, which is connected by means of a hydraulic coupling with the main turbine.
6. Lever operated manœuvring device which in this case may also be remotely controlled. The hand-wheel 1 (Fig. 4c) is here only for emergency operation.

Hydraulic control enables any additional duties to be solved in a simple manner. The apparatus Fig. 4a enables for instance a cruising turbine to be cut in and out by means of the lever 5, which is interlocked in such a way that only when the cruising turbine is out of service can the machinery be run at higher speeds or put into reverse. The apparatus 4c is for remote control from any point of the ship. The indicating plates are painted with luminous paint, so that safe manœuvring is also possible in the dark.

the manœuvring apparatus by fluid under pressure. In the arrangement shown in Fig. 3 pressure oil is used which is supplied by the main oil pump (16). Part of the pressure oil is diverted at (18) for the lubrication of the bearings and of the gear, a further part flows through the orifice plate (21) to the safety system consisting of the overspeed governors (12) and (13), through the quick stop trip (14) and the emergency stop valve (4), and finally a part of the pressure oil flows through the orifice plates (22) as power oil to the underside of the servo-motor pistons operating the nozzle valves (5 and 8).

The method of working of the control station (9) is exceedingly simple. By means of the hand-wheel (10) the control valve (11) is displaced to the right or to the left. With the control valve in the position shown in the drawing, the pipes (19) and (20) are not under pressure, the oil supply returning from the manœuvring apparatus to the oil tank by the pipe (17). All valves are closed and the turbine is at rest. If now, in response to the order "Slow speed ahead", the control valve is shifted to the left by means of the hand-wheel (10), the pressure in the power oil supply pipe to the forward valve (5) gradually increases due to the control valve (11) slowly closing the escape port of the control valve. The pressure in the pipe increases according to the extent of the closure, and the valve (5) opens gradually until the oil escape port is completely covered up, whereupon the full oil pressure comes into play and the valve (5) is full open. *Every position of the control valve therefore corresponds to a definite oil pressure and hence also to a definite opening of the steam inlet valve and therefore to a definite speed of the propeller shaft.* This speed can be read off on a scale marked on the hand-wheel (10). Further rotation of the hand-wheel (10) causes the remaining valves (6) and (7) to open in the same manner. In order to reverse, i. e., for going astern, the hand-wheel (10) is rotated in the opposite direction until pressure builds up in the pipe (20) causing the reversing valve (8) to open. The reversal from full speed ahead to full speed astern can be effected in a few seconds. As no ship can respond to a reversal in such a short time, from 30 to 60 seconds will normally be allowed for going from the ahead to the astern positions.

The manœuvring apparatus — various forms of which are shown in Figs. 4a to 4c — can, together with the central control apparatus, be mounted in any convenient part of the engine room (Figs. 5 and 6). Moreover, remote operation of the manœuvring apparatus is possible from any part of the ship, for instance from the bridge.

Instead of operating the nozzle valves separately by means of servo-motors, they can be grouped together and operated by the cam-shaft actuated by a common rotary piston type of servo-motor (Fig. 7). In place of oil, an incombustible fluid, such as condensate to which rust-preventing oil has been added, may be used.

III. SAFETY FEATURES OF MARINE TURBINES.

The protection of the machinery installation of a ship against possible break-downs is of so great importance — more important even than for land turbine plants, where in most cases an emergency supply of current from another source is available — that no means of increasing the reliability of the installation may be overlooked. To appreciate this, it is only necessary to consider the result of a failure of the supply of lubricating oil to the bearings and to the gears. Of course, the safety devices must themselves contain no possible causes of disturbance. They should first draw the attention of the staff at an early stage to any abnormal operating condition and, if necessary, automatically reduce the load or even shut down the plant.

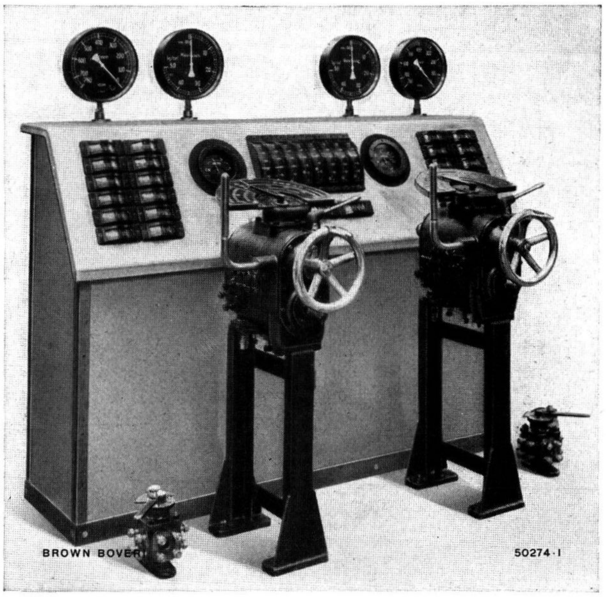


Fig. 5. — Common control stand of a two-shaft marine turbine plant with hydraulic control.

By means of the two control devices it is possible by simple rotation of a hand-wheel, to control both port and starboard turbine plants. The main instruments for supervising the operation of the machinery are mounted in the switchboard panel. The turbines may be controlled from any point of the ship, for instance from the bridge. In the engine room it is possible at any time to switch over to hand operation of all valves.

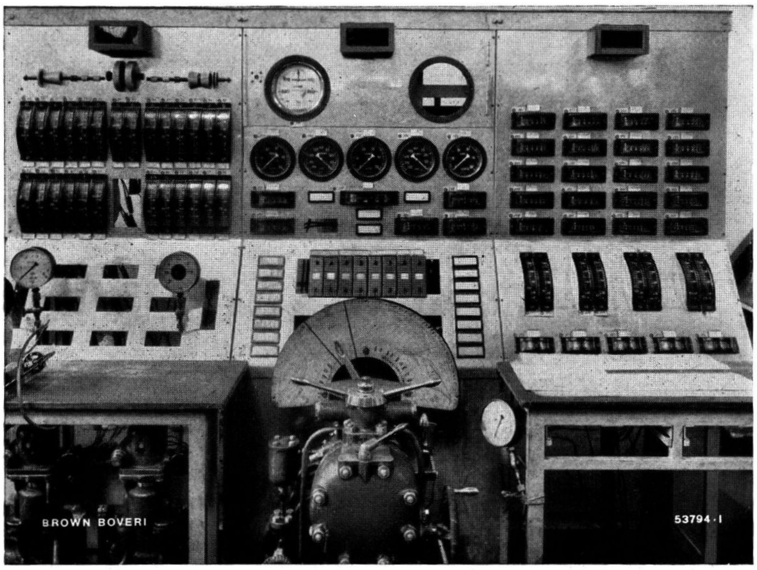


Fig. 6. — Control stand with manœuvring apparatus for a high power marine turbine on the test bed.

Marine turbines of high power can, by means of the Brown Boveri hydraulic control system, be controlled without effort and securely by a single man.

Fig. 8 shows diagrammatically how in a marine turbine installation the principal safety devices may be combined with the hydraulic control system. The

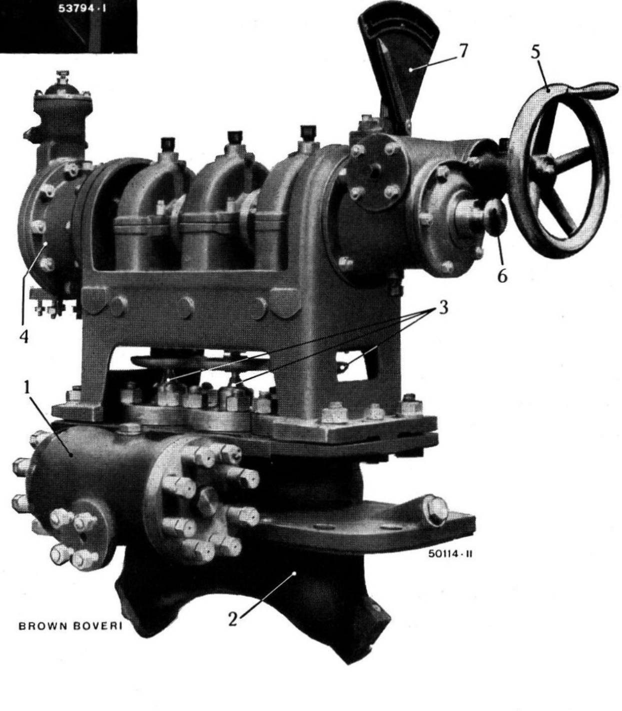


Fig. 7. — Ahead valve group of a hydraulically-controlled marine turbine operated by means of a common rotary servo-motor.

1. Steam inlet with built-in steam strainer.
2. Nozzle valve chest.
3. Three nozzle valves operated by cams.
4. Rotary servo-motor.
5. Hand operation (reserve).
6. Change-over from hydraulic to hand operation.
7. Position indicator for the nozzle valves.

Change-over from hydraulic to hand control can be effected in any position and damaged valves also uncoupled, thereby providing for maximum reliability.

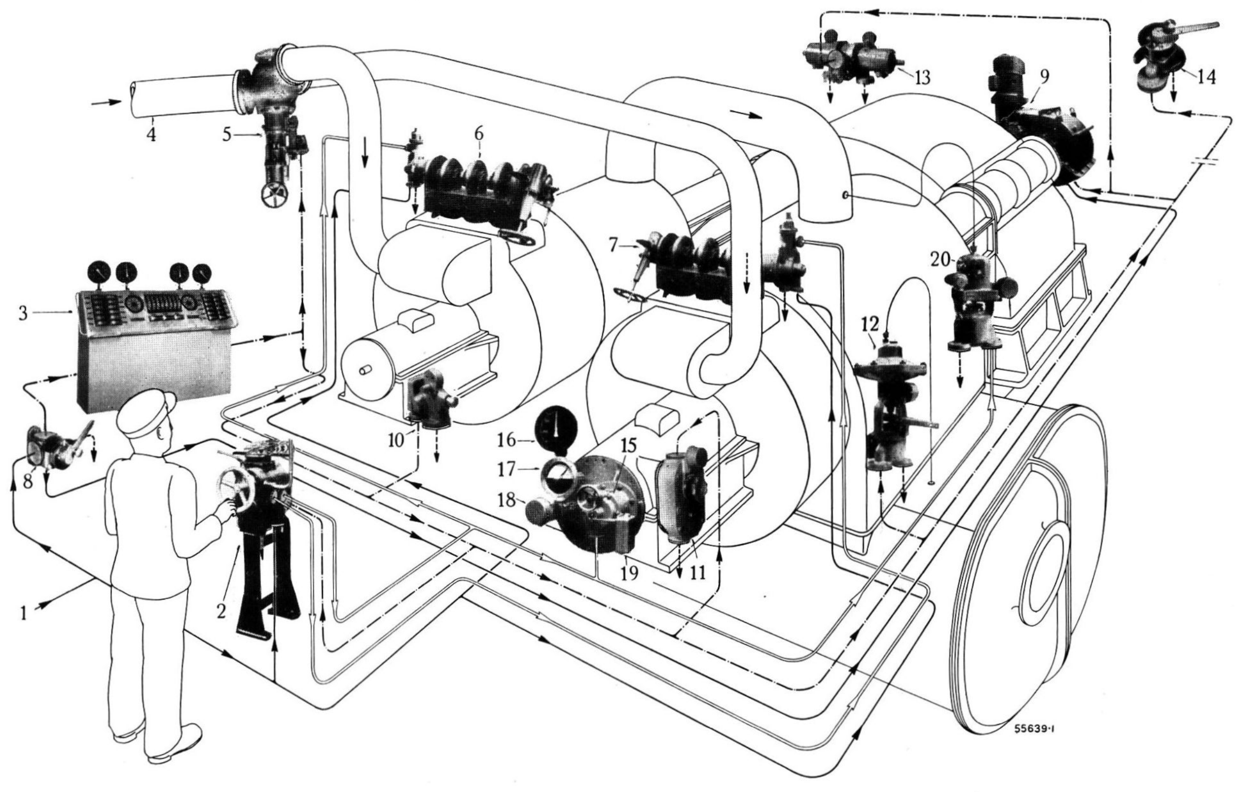


Fig. 8. — Diagrammatic representation of the hydraulic governing system of a marine turbine plant with all safety devices.

1. Pressure fluid from the pump.
2. Control apparatus.
3. Control stand with measuring instruments.
4. Main steam pipe.
5. Emergency and main stop valve.
6. Nozzle valves for ahead.
7. Nozzle valves for astern.
8. Four-way cock.
9. Shaft turning device with safety interlock preventing the starting of the main turbines when the turning device is engaged.
10. Protection device against excessive speed provided by the emergency governor and the hand trip device.
11. Safety device against excessive axial thrust of the turbine rotor.
12. Safety device against excessive loss of condenser vacuum (protection against overheating of the idling turbine blading in the low pressure cylinder).
13. Protection against inadequate oil pressure.
14. Hand trip devices arranged at different points of the engine room.
15. Protection against undesired operation of the emergency governor, as for instance when the speed rises momentarily due to the propeller coming out of the water in high seas (speed limiter, acting also as regulator of the constant propeller speed).
16. Tachometer.
17. Direction indicator.
18. Generator for the remote tachometer.
19. Contact socket for the electrical signalling devices.
20. Protection against excessive steam pressure at particular points, for instance in the interconnecting pipes between the turbines.

—— Pressure fluid system at full pressure.
═══ Relay system with variable pressure.
-·-·- Safety device.

The rodless hydraulic governing system enables all protective features to be solved in a simple and clear manner.

diagram is with the legend sufficiently clear in itself, so that only a few complementary explanations are necessary.

All safety devices are connected to the safety pipe system operating on the quick-closing main stop valve. If any safety feature comes into action, the pressure oil in the safety system is allowed to escape, the emergency stop valve closes, and the turbines come to a standstill. Some safety devices operate so as to throttle the steam supply temporarily, thereby limiting the load just as long as the abnormal condition persists.

Control of the Shaft Position and of the Axial Thrust.

Inadmissible displacements of the turbine rotors from their normal position are shown by axial indicators (Fig. 11). The pressure limiting device (20) must also be considered as a protection for the axial position of the rotors, because if excessive pressure develops at any point of the steam expansion, the unbalanced part of the axial thrust may assume inadmissible values. This pressure-gradient safety feature protects also at the same time the corresponding turbine parts from excessive pressure, for which they are not designed.

Protection against Overspeed.

This is ensured first by the emergency governor (10) incorporated in every turbine, and secondly, by the speed limiter (15) which is usually fitted to an intermediate shaft of the gear. The speed limiter (15) is to prevent the overspeed governor from operating, except in an emergency.

Protection against inadmissible Fall of Vacuum in the Condenser.

With excessively poor vacuum, the light running low-pressure blading (e. g. of the reverse turbine going ahead) may become overheated. The vacuum control device (12) eliminates this danger.

Protection of the Shaft Turning Gear.

The shaft turning gear is employed, for instance, during warming up of the main turbines. As long as the shaft turning gear is in engagement, the main turbines (including those on the other side of the ship in multi-shaft arrangements) may not be started, as this would cause damage to the turning gear. This protection can easily be realized with the hydraulic governing system.

Special attention has been given to the design of the *quick-closing main stop valve,* which constitutes the principal organ of the entire safety system. In order that it shall be independent of any secondary power supply, its servo-motor can be arranged for operation by steam instead of by oil pressure. If the main quick-closing stop valve is shut, then all other steam valves also close automatically, i. e., they must be closed before it is possible to restart. The quick-closing valve serves also under certain conditions — as already explained under I — temporarily as a regulating valve. A special device, protected by patent, enables the proper functioning of the quick-closing stop valve to be tested out in service.

Hand-operated Quick-stopping Devices.

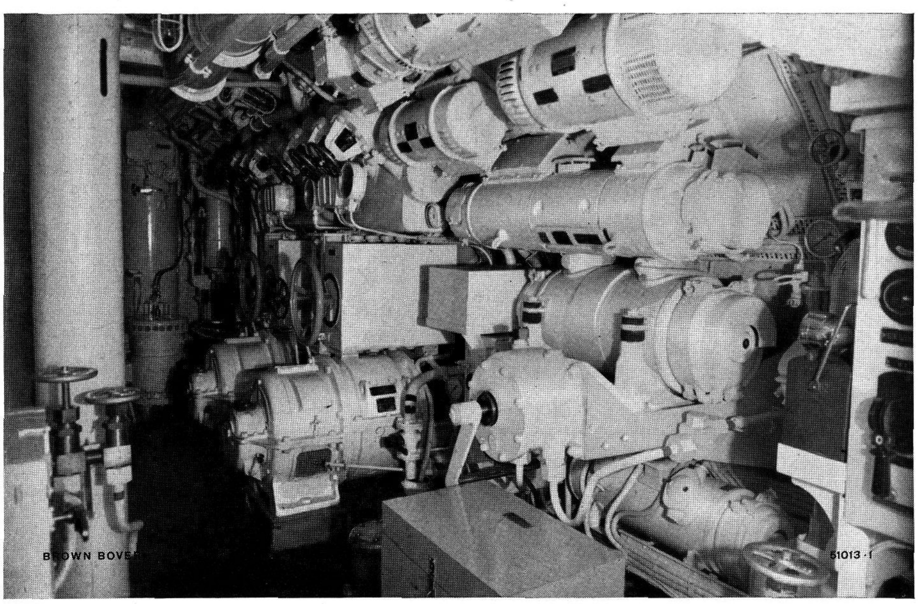
In case of trouble it is necessary to be able to shut down the plant from different points of the engine room. This is achieved by means of the "quick-stopping devices". These are located near to every emergency governor (10), on every manœuvring apparatus, on the control apparatus (item 4, Fig. 4), as well as at different points of the engine room (item 14, Fig. 8).

Supervision.

All the necessary devices for supervision and operation are combined together in the central control stand (Figs. 5 and 6). These consist of pressure and temperature gauges which enable a continuous check to be kept on the operating conditions in different parts of the plant. The optical signalling devices, which light up immediately any abnormal condition occurs at any important place, are also mounted here.

This short description is itself only a hint of the care and attention which has been expended in order to obtain the greatest reliability of the marine turbines built by us. These remarks will also have shown that our marine turbines may well be considered as outstanding achievements in the field of marine engineering, and that they are the outcome of our own research and design technique.

(MS 856) *E. Klingelfuss and V. Tödtli. (Hv.)*



View of auxiliary engine room on former Polish submarine "Orzel" the electrical equipment of which (main and numerous auxiliary motors) was supplied from Baden.

In the centre foreground two diving tank blowers with starters mounted directly on the driving motors. The extremely limited space conditions together with the severe weight restrictions make the designer's task no easy one.

THE DISTRIBUTION OF THE PRESSURE IN MARINE TURBINES WITH VARYING LOAD.

Decimal Index 621.125.013

In the following study, a simplified relation, namely "The separation pressure equation" is derived, enabling the pressure distribution to be determined in turbine stages operating below the velocity of sound, and by means of which the pressure gradient upon change of load may be easily deduced by a simple graphical process. [1]

A marine turbine operates in general at a variable load which is approximately proportionately to the third power of the ship's speed. In planning such a turbine, it is important to establish exactly the variable pressure distribution which fixes the dimensions of the blading, and hence also determines the cost and the weight of the installation. There are a number of laborious methods of calculation which enable the pressure gradient for other rates of steam flow to be determined for a turbine which has been designed for a certain load. In the case of the reaction turbine, which generally operates with relatively small heat drops, the relation between quantity of flow and the pressure ratio is unfortunately not so simple as in the case of a turbine with stages operating above the speed of sound. Notwithstanding this, a very simple graphical process can also be devised in this case, enabling the pressure gradient over the stages to be determined with sufficient accuracy for all practical purposes. The method is applicable to bladings, the flow sections of which remain unchanged with load variations, that is to say, it excludes governing stages.

It is known [2] that the relationship between the steam quantity G, and the initial and final pressures p_1 and p_2 is determined by the equation of an elliptical cone which takes the following form:—

$$\frac{G}{G_0} = \sqrt{\frac{p_1^{\,2} - p_2^{\,2}}{p_{1_0}^{\,2} - p_{2_0}^{\,2}}} \qquad (1)$$

where the index $_0$ indicates that the magnitude is referred to the normal operating conditions for which the blade sections have been designed. The "steam cone law" defined by equation (1) is, strictly speaking, only approximate; it is, however, sufficient for most calculations involved in the design of marine turbines.

Only when there are relatively large differences in the initial temperatures of a group of blades, is it necessary to take a temperature correction into account. For this purpose the "quantity pressure equation" developed by Flügel can be used.

$$\frac{G}{G_0} = \sqrt{\frac{T_{1_0}}{T_1}}\ \sqrt{\frac{p_1^{\,2} - p_2^{\,2}}{p_{1_0}^{\,2} - p_{2_0}^{\,2}}} \qquad (2)$$

In this equation T_{1_0} and T_1 are the absolute steam temperatures in front of the blading.

Taking as starting point a given live steam temperature, the ratio of the initial temperatures of a group of blading for different pressure gradients can be taken as

$$\frac{T_{1_0}}{T_1} = \left(\frac{p_{1_0}}{p_1}\right)^{\frac{m-1}{m}} \qquad (3)$$

Equation (2) then becomes

$$\frac{G^2}{G_0^{\,2}} = \left(\frac{p_{1_0}}{p_1}\right)^{\frac{m-1}{m}} \frac{p_1^{\,2} - p_2^{\,2}}{p_{1_0}^{\,2} - p_{2_0}^{\,2}} \qquad (4)$$

The exponent m of the expansion curve (polytrope) can easily be obtained from the formula

$$m = \frac{\log_e p_1 - \log_e p_2}{\log_e v_2 - \log_e v_1} \qquad (5)$$

wherein v_1 and v_2 are the specific volumes. Experience shows that the exponent m is about 1·2 in the superheat region and 1·1 for wet steam.

In order to obtain an easily calculated expression for the temperature correction, it is sufficient to insert in equation (4) for the pressure ratio p_{1_0}/p_1 the approximate value G_0/G (in accordance with the well-known approximate proportionality law).

Equation (4) then takes the form

$$p_1^{\,2} = \left(\frac{G}{G_0}\right)^{\frac{3m-1}{m}} \left(p_{1_0}^{\,2} - p_{2_0}^{\,2}\right) + p_2^{\,2} \qquad (6)$$

This formula is very suitable for working out the "interstage pressures" p_1 between the individual turbine cylinders with varying steam quantities G and different steam connections of the turbine groups, provided the basic values p_{1_0} and p_{2_0} and

[1] The graphical method developed in the article entitled "The Distribution of Pressure and the Repartition of Steam Flow in Marine Turbines on Overload" at page 256 of the present number of this journal is here made use of.

[2] A. Stodola, Dampf- und Gasturbinen 6th Edition, p. 262.

the new final pressure p_2 are known. The correction factor $\left(\frac{G}{G_0}\right)^{\frac{3m-1}{m}}$ can be obtained as a function of the steam quantity ratio and of the exponent m from the curves Fig. 1, where it is assumed that G/G_0 is less than 1. In case m equals 1, that is to say, when the initial point of the load change remains on the isothermal, equation (6) reduces to the "steam cone law" (1). As shown by elaborate calculations, there is in practice a good agreement between the values given by equations (2) and (6).

If, instead of having to determine for particular stages the "interstage pressures" with variable load, it is required to find the steady pressure distribution, it is

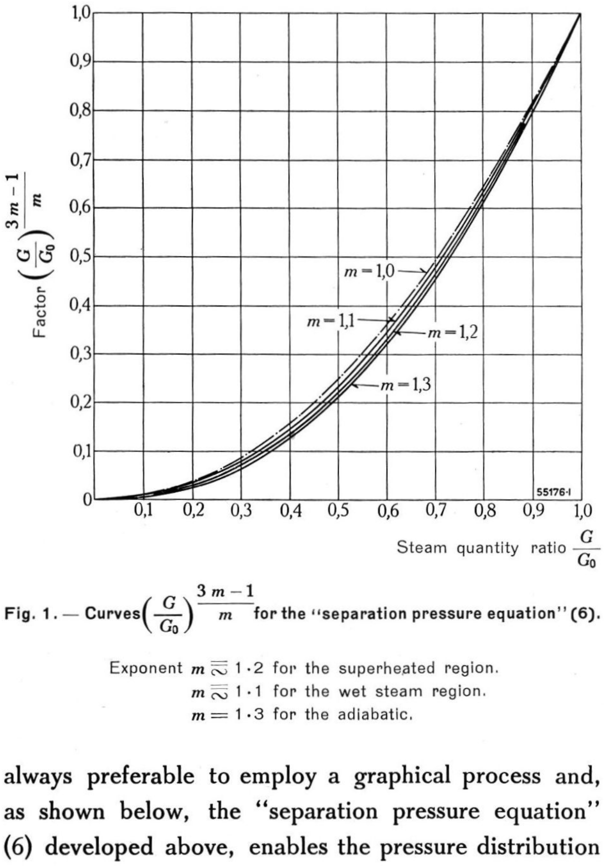


Fig. 1. — **Curves $\left(\frac{G}{G_0}\right)^{\frac{3m-1}{m}}$ for the "separation pressure equation" (6).**

Exponent $m \approx 1{\cdot}2$ for the superheated region.
$m \approx 1{\cdot}1$ for the wet steam region.
$m = 1{\cdot}3$ for the adiabatic.

always preferable to employ a graphical process and, as shown below, the "separation pressure equation" (6) developed above, enables the pressure distribution in the blading with change of load to be determined graphically in the easiest possible manner.

Let, for instance, the pressure gradient at "normal load" (design load) corresponding to a quantity of steam G_0 in the stages $A_1, A_2 \ldots A_5$ be represented by the curve $B_1\ B_2 \ldots B_5$ (Fig. 2). Let it be required to find the new pressure distribution for a fractional load corresponding to a steam quantity $G = \alpha\, G_0$, where α is less than 1, the new final pressure $p_2 = A_5\, H_5$ being known. All stages operate in the region below the speed of sound, and the expansion takes place in a superheated region.

Through the given final points B_5 and H_5, draw two horizontal lines $B_5\, D_4$ and $H_5\, E_4$. The pressure scale in Fig. 2 is arbitrary; it must, however, be the same for all pressures. Strike about the point A_4 a circular arc with radius $A_4 B_4 = p_{1_0}$ up to the point of intersection C_4 with the horizontal line $B_5\, D_4$. On the second horizontal $H_5\, E_4$ mark off E_4 a distance $E_4\, F_4 = \alpha^{\frac{3m-1}{2m}}\, D_4\, C_4$, where $\alpha = G/G_0$ is given, and $\alpha^{\frac{3m-1}{2m}}$ is obtained directly from Fig. 3 ($m = 1{\cdot}2$). About the point A_4 draw a new circular arc with

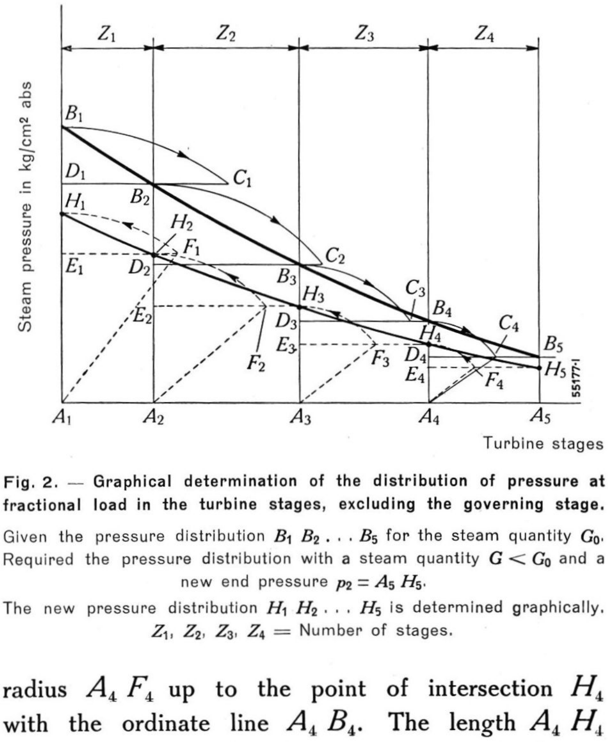


Fig. 2. — **Graphical determination of the distribution of pressure at fractional load in the turbine stages, excluding the governing stage.**

Given the pressure distribution $B_1\ B_2 \ldots B_5$ for the steam quantity G_0.
Required the pressure distribution with a steam quantity $G < G_0$ and a new end pressure $p_2 = A_5\, H_5$.
The new pressure distribution $H_1\ H_2 \ldots H_5$ is determined graphically.
Z_1, Z_2, Z_3, Z_4 = Number of stages.

radius $A_4\, F_4$ up to the point of intersection H_4 with the ordinate line $A_4\, B_4$. The length $A_4\, H_4$ then represents to the chosen pressure scale, the required initial pressure p_1 for the blade group $A_4\, A_5$.

In order to follow the pressure gradient further, the process is repeated, considering the steam pressure just found $A_4\, H_4$ as the given final pressure of the blade group, in place of the pressure $p_2 = A_5\, H_5$.

In this manner the curve $H_1\ H_2 \ldots H_5$ can be obtained, giving the pressure gradient over the stages with the new steam quantity $G = \alpha\, G_0$. In the process here described we have calculated "backward", because the end pressure $p_2 = A_5\, H_5$ for the new steam quantity G was given. This graphical process can, however, also be used for "forward" working if the new initial pressure p_1 is known.

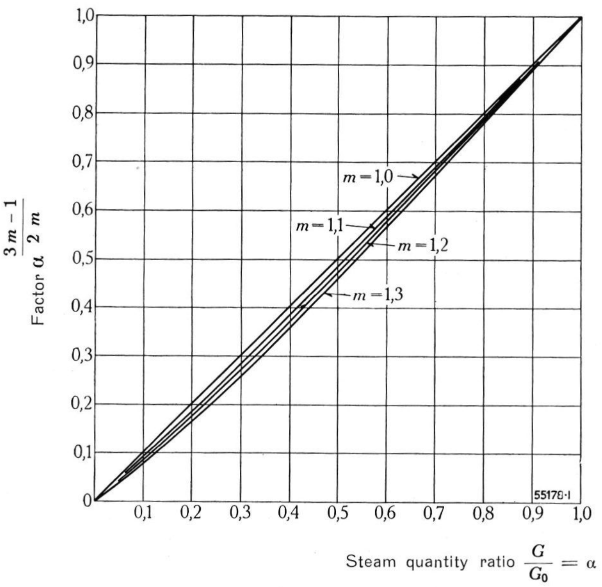


Fig. 3. — Curves $\alpha^{\frac{3m-1}{2m}}$ for fractional load $(G < G_0)$.

Exponent $m \approx 1 \cdot 2$ for the superheated region.
$m \approx 1 \cdot 1$ for the wet steam region.
$m = 1 \cdot 3$ for the adiabatic.

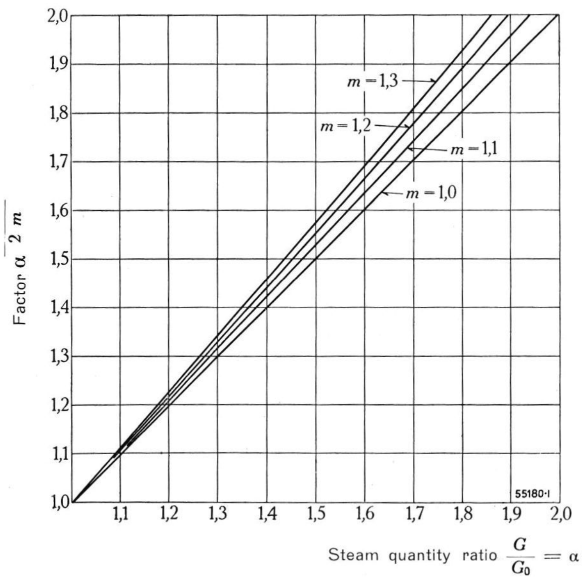


Fig. 5. — Curves $\alpha^{\frac{3m-1}{2m}}$ for overload $(G > G_0)$.

Exponent $m \approx 1 \cdot 2$ for the superheated region.
$m \approx 1 \cdot 1$ for the wet steam region.
$m = 1 \cdot 3$ for the adiabatic.

Fig. 4 shows the application of the graphical method to the determination of the pressure gradient on overload, when $\alpha = G/G_0$ is greater than 1. Here also it is assumed that the end pressure $p_2 = p_{2_0} = A_5 B_5$ remains constant. The length $D_4 F_4 = \alpha^{\frac{3m-1}{2m}} D_4 C_4$ must here be marked off along the horizontal $B_5\, D_4$.

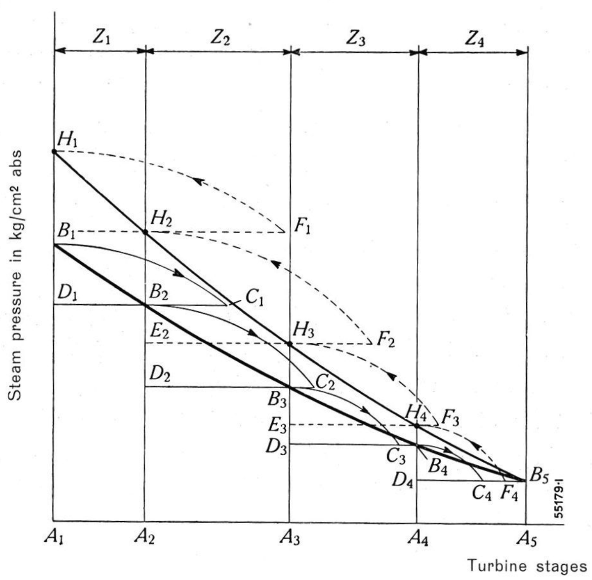


Fig. 4. — Graphical determination of the distribution of pressure at overload in the turbine stages, excluding the governing stage.

Given the pressure distribution from $B_1\, B_2 \ldots B_5$ for a steam quantity G_0. Required the pressure distribution with a steam quantity $G > G_0$ and a constant end pressure $p_2 = p_{2_0} = A_5\, B_5$.

The new pressure distribution $H_1\, H_2 \ldots B_5$ is determined graphically. Z_1, Z_2, Z_3, Z_4 = Number of stages.

The values of $\alpha^{\frac{3m-1}{2m}}$ in the case of overload can be obtained from Fig. 5.

In conclusion, we give the mathematical proof of the graphical process developed in Fig. 2. From the triangles $A_4\, E_4\, F_4$ and $A_4\, D_4\, C_4$ follows the relation

$$\frac{E_4\, F_4^{\,2}}{D_4\, C_4^{\,2}} = \frac{A_4\, F_4^{\,2} - A_4\, E_4^{\,2}}{A_4\, C_4^{\,2} - A_4\, D_4^{\,2}} \tag{7}$$

But, by construction

$$E_4\, F_4 = \alpha^{\frac{3m-1}{2m}}\, D_4\, C_4;\; A_4\, E_4 = A_5\, H_5 = p_2;$$
$$A_4\, C_4 = A_4\, B_4 = p_{1_0} \text{ and } A_4\, D_4 = A_5\, B_5 = p_{2_0},$$

hence it follows that

$$\alpha^{\frac{3m-1}{2m}} = \frac{A_4\, F_4^{\,2} - p_2^{\,2}}{p_{1_0}^{\,2} - p_{2_0}^{\,2}} \tag{8}$$

whence

$$A_4\, F_4^{\,2} = \left(\frac{G}{G_0}\right)^{\frac{3m-1}{m}} (p_{1_0}^{\,2} - p_{2_0}^{\,2}) + p_2^{\,2} \tag{9}$$

On comparing equations (6) and (9) it is seen that the length $A_4\, H_4$ which is equal to $A_4\, F_4$ really does represent the required steam pressure p_1, thereby proving the correctness of the graphical method here developed.

If for the sake of simplicity m is taken = 1, the calculation is simplified in that in Fig. 2 the length $E_4\, F_4 = \alpha\, D_4\, C_4$ and in Fig. 4 the length $D_4\, F_4$ becomes equal to $\alpha\, D_4\, C_4$, and therefore the curves Figs. 3 and 5 can be dispensed with.

(MS 858) *W. Rohrbach. (Hv.)*

MARINE CONDENSING PLANTS.

Decimal Index 621.175.182

The Brown Boveri OV-condenser with rolled-in tubes offers a number of special advantages for marine service. Equipped with a new type of condensate flow-regulator (closed feed system), the OV marine condenser may be looked upon as one of the best designs of the present day, as well in regard to thermal performance and constructional details, as in regard to the use of up-to-date manufacturing methods (welding technique).

THE fundamental ideas which Brown Boveri, correctly appreciating the nature of the process of condensation, already in 1916 successfully incorporated in their OV condenser, have now become common property. [1]

The fundamental principle according to which a large flow section, affording ready access to the tubes, must be provided for the steam immediately it enters the condenser, whilst at the same time, the amount of cooling surface with which it comes into contact is limited to that strictly necessary for its condensation, was realized for the first time in the OV condenser. In the V-shaped steam space, which extends right down to the surface of the condensate, the exhaust steam distributes itself along the whole length and depth of the condenser, and can flow freely into the wedge-shaped lateral steam lanes (Figs. 1 and 6). This uniform and economical utilization of the inner cooling surface enables the space and material requirements to be kept down to a minimum.

The depth of the tube nest in the direction of flow is small, and hence the pressure drop on the steam side necessary to overcome the resistance to flow offered by the tube nest, is also small. This means that one of the main causes leading to *undercooling of the condensate is eliminated.* Concerning this much discussed question of undercooling of the condensate, it is well to keep the following facts in mind:—

When in practical service, discrepancies are observed between actual and theoretical conditions (e. g., condensate temperature lower than that corresponding to the condenser pressure, or low heat transfer coefficient), the cause is likely to lie not so much in the design (e. g., arrangement of the tubes), as in the leakage at the turbine glands, and in the equipment connected to the condensing system. If much air gets into the condenser, the partial pressure of the steam is reduced, and hence the effective condensate temperature is lowered.

Now a far greater quantity of air has to be reckoned with in marine installations than in land ones. This is to be attributed, less to leakage into the main turbines than to the fact that more opportunities are provided for the ingress of air by the extensive connected pipe system for the auxiliary machinery, and by the condensate returns. Our efforts are, therefore, directed in the first place towards assuring freedom from leakage into the condenser system, not only of air, but also of sea water.

One way of obviating leakage into the condensing system lies in the elimination of flange joints by the extensive use of *welding.* At the same time, a saving in weight is achieved, which is very desirable for marine installations. Thus in the condenser, the shell as well as all connecting branches, are welded; in addition flange connections are avoided in the pipe system wherever possible. A feature of paramount importance for freedom from leakage of the condenser,

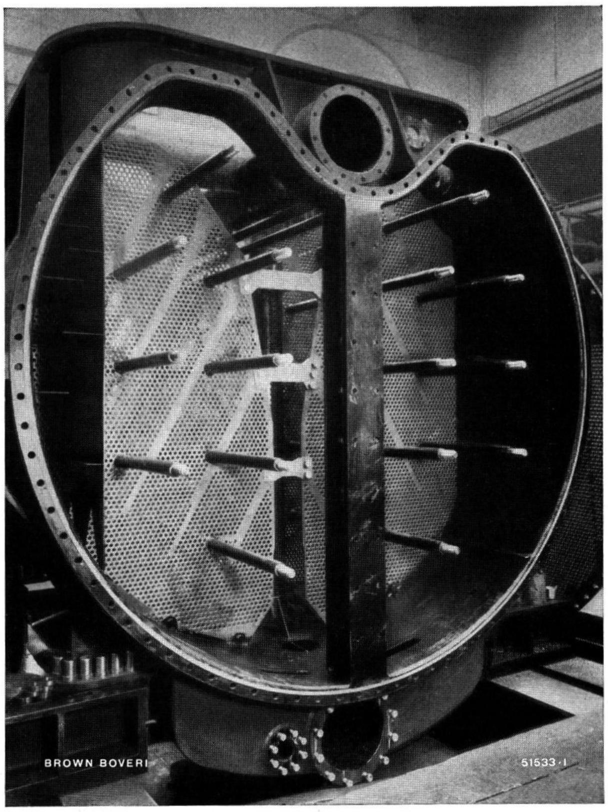


Fig. 1. — OV marine condenser of 1050 m^2 cooling surface under erection.

The V-shaped steam space extending right down to the bottom of the condenser can be clearly seen between the bored tube plates. Lateral wedge-shaped steam lanes open up the tube bundles. Air coolers are arranged on each side of the main condensation space and are separated from it by vertical partitions (see also Fig. 6). The condenser tubes are rolled-in at both ends.

The OV-condenser with rolled-in tubes takes up little space and delivers a practically air-free condensate with no sensible undercooling. Marine condensers with ferruled tubes must to-day be regarded as obsolete.

[1] Ad. Meyer: "Über Oberflächenkondensatoren / Der OV-Kondensator". BBC Mitteilungen 1916, p. 105.

is the rolling-in of the tubes into the tube plates at both ends.

Condensers with tubes sealed by means of grummets and ferrules, give considerable trouble in service. The stresses to which the tubes are subjected are much more severe in marine condensers than in land installations; during manœuvring with the main turbines, and during the occasional introduction of excess steam delivered by the boilers, they come into contact alternatively with hot and cold steam, and are thereby caused to undergo considerable expansions and contractions. This relative motion has to be taken up by the glands, which suffer accordingly. Moreover, the tubes are subjected to additional stresses due to temporary deformations of the ship's hull, especially in heavy seas.

easily and regularly between the fixed tube plates in following thermal expansion and contraction, without subjecting the rolled joints to any undue stresses. At the same time every tube can expand individually, by deflecting more or less in the direction of the initial curvature. As this direction is the same for all tubes, there is no possibility of the tubes fouling one another.

To ensure proper drainage of the tubes during shut-downs, the entire bundle is made slightly convex upward.

Whereas the material of tubes for use in condensers with grummets and ferrules, must have a hard structure, in order that it shall not suffer damage when the packings are tightened, soft annealed material, free from all residual strains is used for rolled-in tubes.

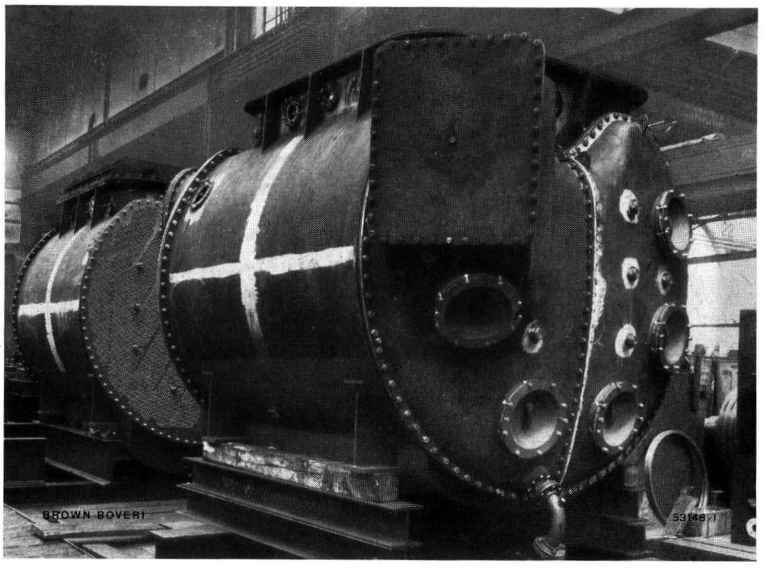


Fig. 2. — Two OV marine condensers under hydraulic pressure test.

The condenser in the background is being tested on the steam side, that in the foreground on the water side. The numerous manholes in the waterbox covers facilitate inspection of the condenser tubes. The zinc anti-corrosion plates are also fixed to the manhole covers. Our OV condenser, invented already in 1911, represents a masterpiece of thermal engineering design and of workshop technique.

The result is that, unless constant attention is given to them, the glands begin to leak, and sea water and air find their way into the condensate. Since the number of glands runs into thousands, the total effect of even minute leakages may lead to serious trouble.

Rolling the tubes in at both ends completely eliminates the source of this trouble. Moreover, the absence of the glands enables the tubes to be placed closer together and hence allows a given cooling surface to be accommodated in a smaller volume.

In order that the tubes may be able to expand and contract freely with changes of temperature, they are given a slight initial curvature, in accordance with a protected design. Depending on the number of supporting plates, the tubes are given two or more waves. The waves are in the horizontal plane, and are such that the supporting plates are located at the points of inflection. The lightly bent tubes yield

Such tubes have a more uniform crystal structure and are less liable to corrosion than hard drawn ones. They enable in particular, the much feared tube failures to be avoided. Exhaustive tests have shown namely that the alloy plays only a subordinate part in the matter of resistance to corrosion, and that uniform crystal structure and absolute freedom from residual strains are the primary requisites. [1]

For the marine engineer, it should be mentioned that bending of the tubes can be effected on site with very simple means. The replacement and rolling-in of tubes into the tube plates is equally simple. To ensure against any alteration of the material structure during rolling-in, a special expanding tool with automatic

[1] H Stäger and J. Biert: "The Corrosion of Copper-Zinc Alloys", The Brown Boveri Review 1934, p. 180; and the Brown Boveri Review 1935, p. 62/63.

limitation of the travel and of the pressure is supplied with the condenser.[1]

Condensers with rolled-in tubes have been used by us for land plants since 1931, and for marine installations since 1935. The service results have been so satisfactory that we would not now, of our own initiative, use any other method of fixation; moreover, a number of shipyards and builders of marine condensers have acquired the manufacturing rights for our protected design of condenser.

When planning the condenser, the object in view is to obtain the most economical vacuum for a given cooling water temperature, or, in other words, the lowest temperature t_d corresponding to the steam pressure (vacuum) in the condenser. In the ideal case the exhaust steam temperature would be equal to the cooling water temperature t_w. This is not possible in practice, as it would mean infinitely large cooling water quantities and cooling surfaces. The economical relation between the attainable steam temperature, the cooling water quantity and the cooling water temperature as well as the condenser cooling surface are best shown by the following formula:—

$$t_d = t_w + \frac{W}{2\,Q} + \frac{W}{k\,O}$$

W = heat quantity to be carried away (kcal/h)
Q = cooling water quantity (kg/h)
O = surface of the condenser (m^2)
t_d = exhaust steam temperature corresponding to the condenser vacuum (^{0}C)
t_w = cooling water temperature (^{0}C)
k = heat transfer coefficient (kcal/m^2 h ^{0}C).

[1] J. Lalive: "Rolled-in Bent Tubes employed in Brown Boveri Steam Condensers", The Brown Boveri Review 1939, p. 241.

The deciding factor for the attainable vacuum, i. e., for the lowest attainable steam temperature, is the cooling water temperature t_w. The two other members of the equation represent the departure from the ideal case ($t_d = t_w$) due to the limitation of the cooling water quantity Q, and of the cooling surface O, of the condenser. The larger the cooling water quantity Q and the cooling surface O, the closer the approximation to the ideal case. Economical considerations fix limits to both these factors.

In marine plants plenty of cooling water can be obtained without having to expend much pumping power, but the cooling surface is limited by space and weight considerations. Ships' condensers are, therefore, usually designed for a cooling water quantity of some 60 to 80 times the steam quantity. In order to reduce the amount of power required for pumping, it is advantageous to use single-pass condensers, and in many cases these can be conveniently supplied with water by means of scoops built into the hull of the ship.

A noteworthy development also of interest for marine installations is the continuous service condenser introduced to the engineering world by us.[1]

In this arrangement, the front and back water chambers are divided into two parts, each being provided with its own inlet and outlet connections for the cooling water. There is no subdivision on the steam side. By shutting off the cooling water in one set of connections, the corresponding half of the water chambers can be opened whilst the other half remains in service. This enables the condenser to be inspected for leaky tubes and, if necessary, the joints to be tightened up, without having to shut down the

[1] DRP 228,926, June, 1910.

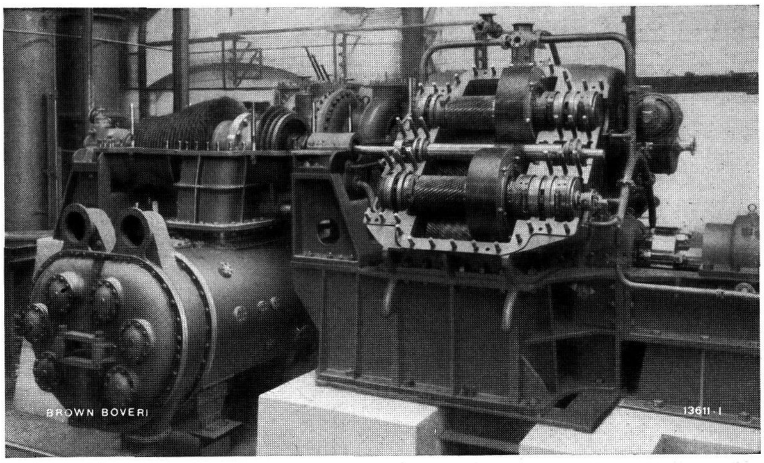


Fig. 3. — Marine turbine plant of 3000 S. H. P., 75 r. p. m. with OV continuous service condenser.

In the continuous service condenser the water boxes are divided into two parts. Each water box part is provided with its own cooling water inlet and outlet. Upon closing the same, the corresponding water box half can be taken out of service and the cover swung open like a door. This enables the condenser to be inspected for leaking tubes, and if necessary to tighten up defective ones without taking the condenser out of service, thereby enabling salt water leakage into the feed water to be stopped at the earliest moment. Moreover, it is also possible to clean the condenser whilst in service.

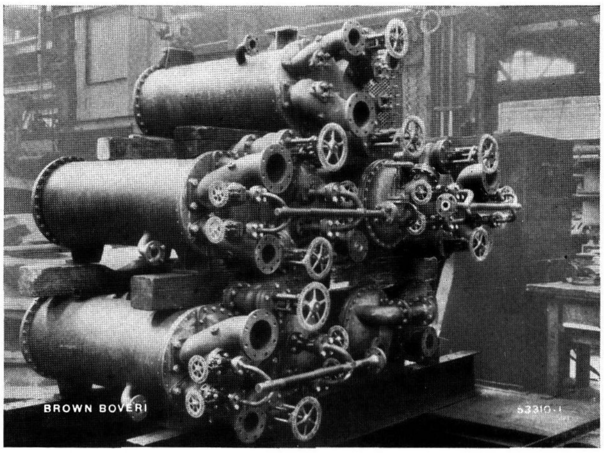


Fig. 4. — Two-stage, high-power, steam-jet air ejectors for marine turbine plants.

For marine service the steam-jet air ejector represents the simplest and most reliable form of air extraction device.

plant. By means of our patented "Columbus" tube cleaner[1] it is possible to clean the tubes with the condenser in service, which is occasionally desirable after long period navigation through dirty river waters. Fig. 3 shows a marine turbine plant of the year 1920 with an OV continuous service condenser.

Air Extraction.

This is effected by means of a steam-jet air ejector through special air coolers, which — as can be seen from Figs. 1 and 6 — are located on either side in the coldest parts of the condenser. For this purpose part of the cooling tubes are separated from the active cooling surface by means of vertical partitions. Thus the tubes of the air cooler are not in the main steam path; they form a narrow second nest of tubes of considerable depth, beyond which practically only saturated air is extracted.

The air ejector (Fig. 4) is of the two-stage type with intercooler. Condensate and not sea water is used as a cooling medium, thus enabling the heat of the ejector steam to be used for feed heating, whilst at the same time avoiding any danger of sea water entering the system.[2]

Condenser Pumps.

Condensing pumps are usually of the centrifugal type. Cooling water pumps are either of the centrifugal or of the propeller type. They can be supplied with horizontal or with vertical shaft and with electrical or with steam turbine drive.[3] Generally, however, the condenser pumps as well as the main oil pump are connected together and are provided with both steam turbine driving through a reduction gear, and with electric motor drive, the latter being fed by a Helux generator driven from the propeller shaft.[4]

These combined condenser pump sets represent the best solution from the point of view of economical operation.

Condensate Regulation
(Closed feed system).

The importance of the closed feed system (Fig. 5), in which the condensate extraction pump and the feed pump are in series in a closed circuit, so that the condensate does not come in contact with air, is well known to every steam plant engineer. Because the amount of condensate returning and the amount of feed water required by the boiler are not always equal, a storage tank has to be interposed between the condensate pump and the feed pump. We use the condenser itself as a storage tank, or, a condensate tank mounted directly on the underside of the condenser and to which make-up water may be added from an equalizing tank when necessary. The first marine plant with closed feed system was supplied by us already in 1920.[5] A float-controlled valve maintains the level of the condensate above the extraction pump constant within given limits. If the boilers re-

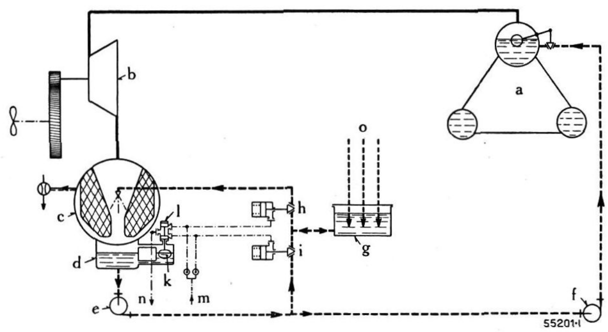


Fig. 5. — Diagram of a closed feed system with automatic condensate regulation.

a. Boiler with feed regulating valve.
b. Main turbine.
c. OV condenser with air extractor.
d. Condensate tank.
e. Condensate pump.
f. Boiler feed pump.
g. Condensate reserve and equalizing tank.
h. Condensate adding valve.
i. Condensate overflow valve.
k. Float.
l. Control valve.
m. Pressure oil (supply).
n. Pressure oil (return).
o. Condensate returns.

The condenser serves as a de-aerator for the entire water quantity fed to the boilers. The arrangement is simple, accessible and reliable.

[1] The Brown Boveri Review 1935, p. 53/54.

[2] O. Frey: "On Steam-jet Air Ejectors", The Brown Boveri Review 1941, p. 382.

[3] The Brown Boveri Review 1937, p. 70.

[4] E. Klingelfuss & R. Schmid: "Turbine-driven Marine Auxiliaries", p. 290 of present number of this journal.

[5] A. Gorgel: "Die Maschinenanlage der Passagier- und Frachtdampfer 'Thuringia' und 'Westphalia' der Hamburg-Amerika Linie". Werft, Reederei, Hafen 1923, Heft 19.

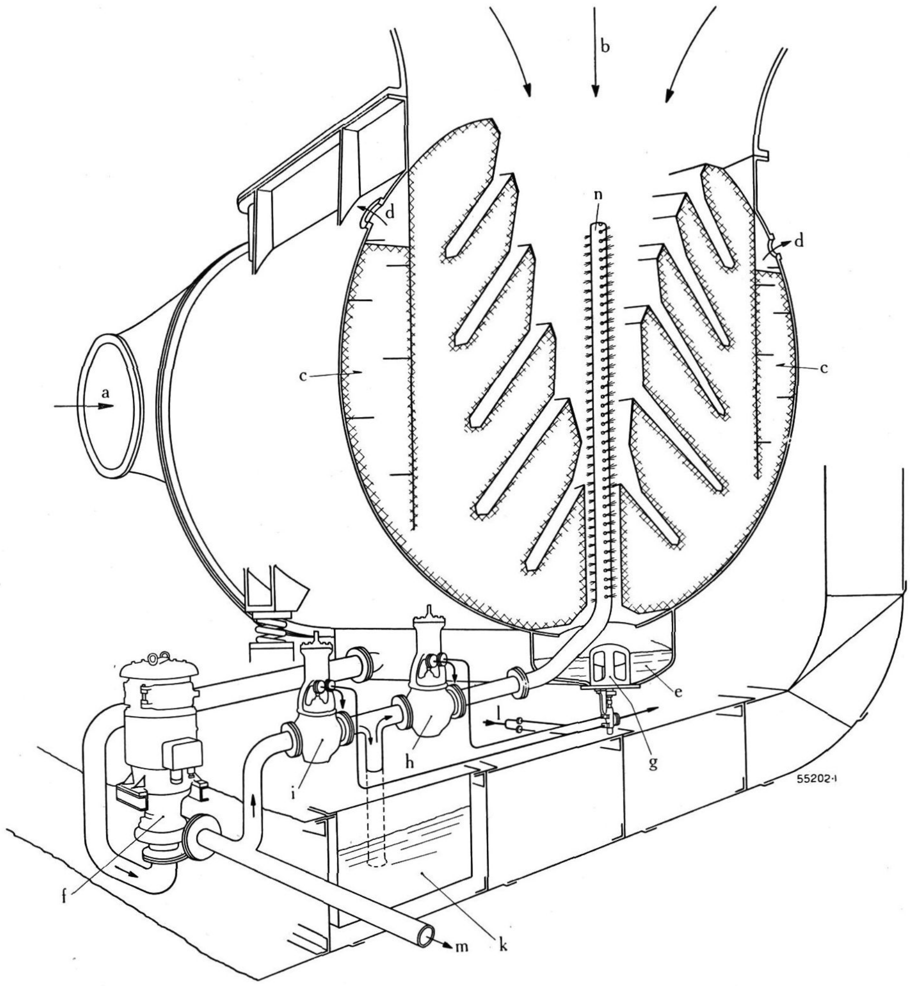


Fig. 6. — Fundamental arrangement of an OV marine condenser with condensate regulation.

a. Cooling water regulation.
b. Exhaust steam from the main turbine.
c. Air cooler.
d. Air extraction.
e. Condensate tank.
f. Condensate pump.
g. Float and control valve.
h. Condensate adding valve.
i. Condensate overflow valve.
k. Feed water equalizing tank.
l. Pressure liquid.
m. To steam pump.
n. Spray tube for make-up water.

The float g has only to provide the necessary impulse, whilst the flow-regulating valves (h and i) are operated by the fluid under pressure. They can also be operated by hand and may be installed independently of the condenser at the most convenient spot.
The new power-operated condensate regulation increases the reliability.

quire more water than there is condensate returning, the float valve admits water from the equalizing tank in the double bottom of the ship; in the converse case the float valve allows water to flow from the delivery branch of the extraction pump into the equalizing tank, which is provided with an overflow to the feed water storage compartments. The regulation of the water level in the boiler takes place in the ordinary way by means of the boiler feed regulating valve.

Whereas formerly, the valves regulating the water flow used to be controlled directly by the float, we have recently introduced hydraulic control for these valves (Fig. 6). The float is located on the condenser in such a position as to be affected as little as possible by the rolling of the ship. It controls a small relay valve, which in turn controls the supply of the fluid under pressure to the operating gear of the valves regulating the water flow. The latter can be installed independently of the float gear in the most convenient place. The float gear is relieved of all work. It has only to transmit the governing impulse. On the other hand, plenty of power is available for the operation of the water-flow regulating valves, so that reliable functioning is assured.

As can be seen from the diagram Fig. 6, all make-up water is effectively de-aerated by being injected into the condenser, because the latter is and remains the best and simplest de-aerating device.

(MS 880) *V. Tödtli. (Hv.)*

MARINE GEARS.

Decimal Index 621.831 : 629.12

The article shows by means of some examples of manufactured gears, the present state of development of our gear technique and the achievements of our workshops in this field.

THE sight of a large gear such as that illustrated in Fig. 1, may, by its dimensions, awaken a certain amount of admiration in the layman, but the latter does not, however, appreciate what a marvel of workshop technique stands in front of him, and how many years of laborious research have gone into the making of such a gear, the general design of which seems simple enough to him.

The requirements which a gear has to satisfy may indeed be formulated quite briefly: High efficiency, smooth running with little noise, and long life are in fact all that is needed. Yet these apparently simple requirements conceal a number of difficult problems. The solution of these problems lies mainly with the workshops. In addition, there are the hydrodynamic lubrication problem and, of course, certain problems of design, but in the main, everything depends on the accuracy with which the workshops are able to manufacture such a gear. This is so, because high efficiency, smooth running and quietness, as well as long life, depend essentially on the precision with which the gear is made.

Our success in the manufacture of turbo-gears is due to:

(a) Our long service and manufacturing experience extending over many years.
(b) Our research work in the field of reduction gears.
(c) Our original and progressive design technique.
(d) Our first-class workshop equipment for cutting large gear wheels and our specially trained personnel for attending gear cutting machines.

Our Experience.

Our experience is the result of a thirty year activity in the manufacture of turbine gears ranging in output from 4 to 80,000 H. P., for the most varied purposes. We refer the reader to earlier publications[1] and would mention here only that our experience covers some 2500 gears with a total output of 3·6 million H. P. Among these are the following extremes:—

Greatest power per pinion	approx. 26,000 H. P.
Largest diameter of wheel	approx. 4000 mm
Greatest tooth width . .	1×750 single helical 2×660 double helical
Heaviest wheel	35 t
Highest pinion speed . .	48,000 r. p. m.
Greatest peripheral speed .	97 m/s 181 m/s in tests
Largest reduction ratio . .	1 : 18 as single reduction gear 1 : 95 as double reduction helical gear 1 : 266 as double reduction epicyclic gear.

In regard to application to ships, it is interesting to note that on the Continent, Brown Boveri was not only the first firm to employ the marine turbine for direct drive, but was also the first firm to employ gearing for marine propulsion turbines. Noteworthy is that the incentive for the development of geared marine turbines came in connection with the manufacture of propelling machinery for ships and especially for warships, thus the previous compromise between the speeds of the slow running propeller and of the fast running turbine was eliminated at one stroke by the introduction of gearing.

Our Research.

Whilst during the early stages the design of gearing was based mainly on experience, the problems of lubrication, tooth wear, the formation of "pitting" as well as the cause of noise later became the subject of systematic research. Some of the results of this research have already been made public.[2]

[1] Klingelfuss: "Brown Boveri Dampfturbinen mit Übersetzungsgetrieben", BBC Mitteilungen 1920, pp. 243, 278.

Baasch: "Brown Boveri Marine Turbines, III. Geared Turbines", The Brown Boveri Review 1922, p. 139.

Baasch: "Brown Boveri Gears", The Brown Boveri Review 1926, pp. 47/153.

[2] A. Meldahl:

"Testing Gear-Wheel Material." The Brown Boveri Review 1939, p. 235. See also "Engineering" 21st July, 1939, p. 63.

"Theory of Gear Pumps." The Brown Boveri Review 1939, p. 259.

"Regarding the most favourable Shape of Helical Hobs for cutting Involute Gears." The Brown Boveri Review 1940, p. 142.

"Contribution to the Theory of the Lubrication of Gears and of the Stressing of the lubricated Flanks of Gear Teeth." The Brown Boveri Review 1941, p. 374.

"Why does a Gear sing? How can the Singing be avoided?", p. 284 of the present number of this journal.

F. Modugno: Teoria e costruzione degli ingranaggi ad assi paralleli con applicazione ai riduttori marini, p. 237 (Publishers: Spoleto 1940).

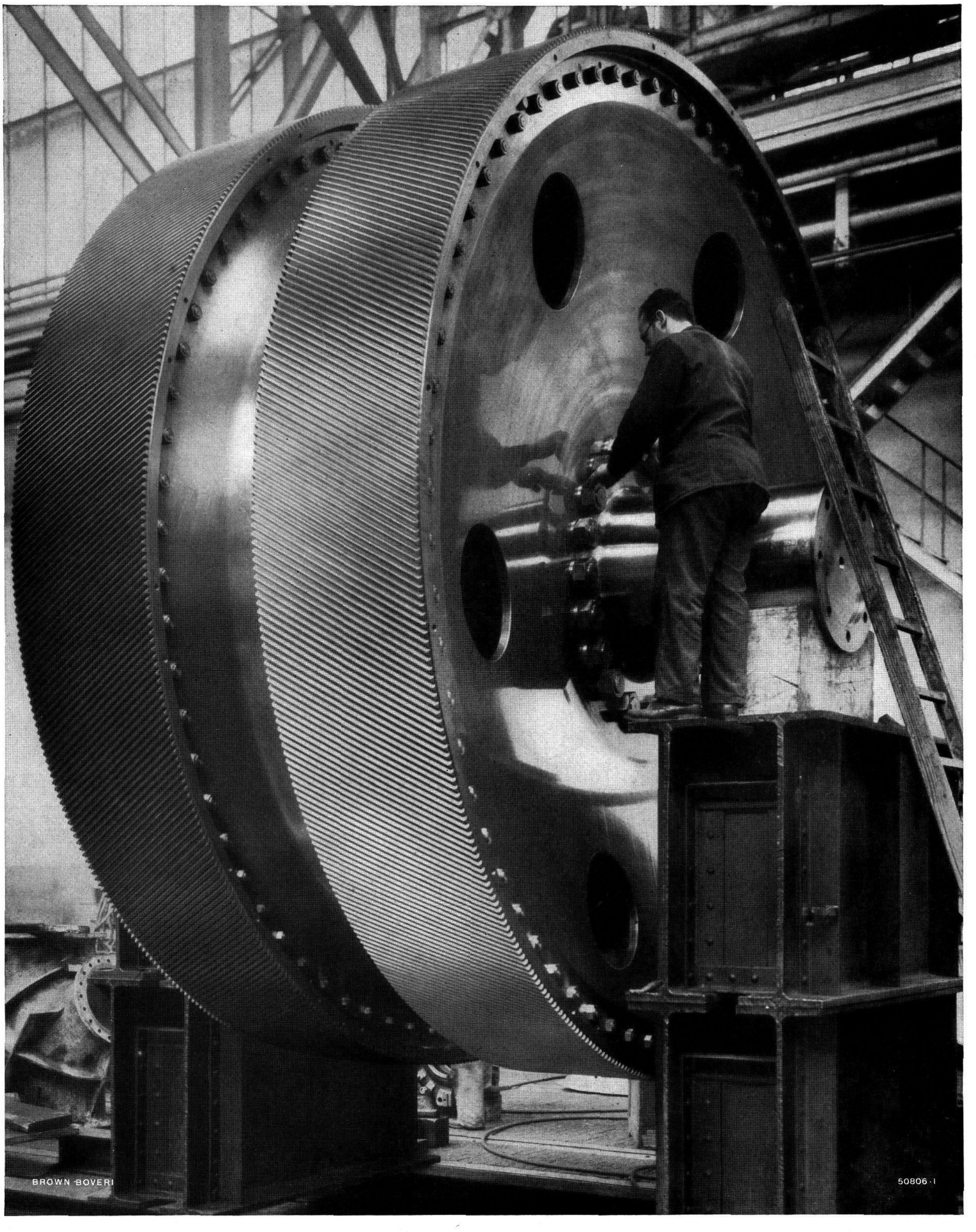


Fig. 1. — Large marine gear wheel.

A masterpiece of modern engineering workshop technique and precision. The greatest pitch error on a diameter of nearly 4000 mm amounted to only $^{3}/_{1000}$ mm, and the average eccentricity to only $^{1}/_{100}$ mm.

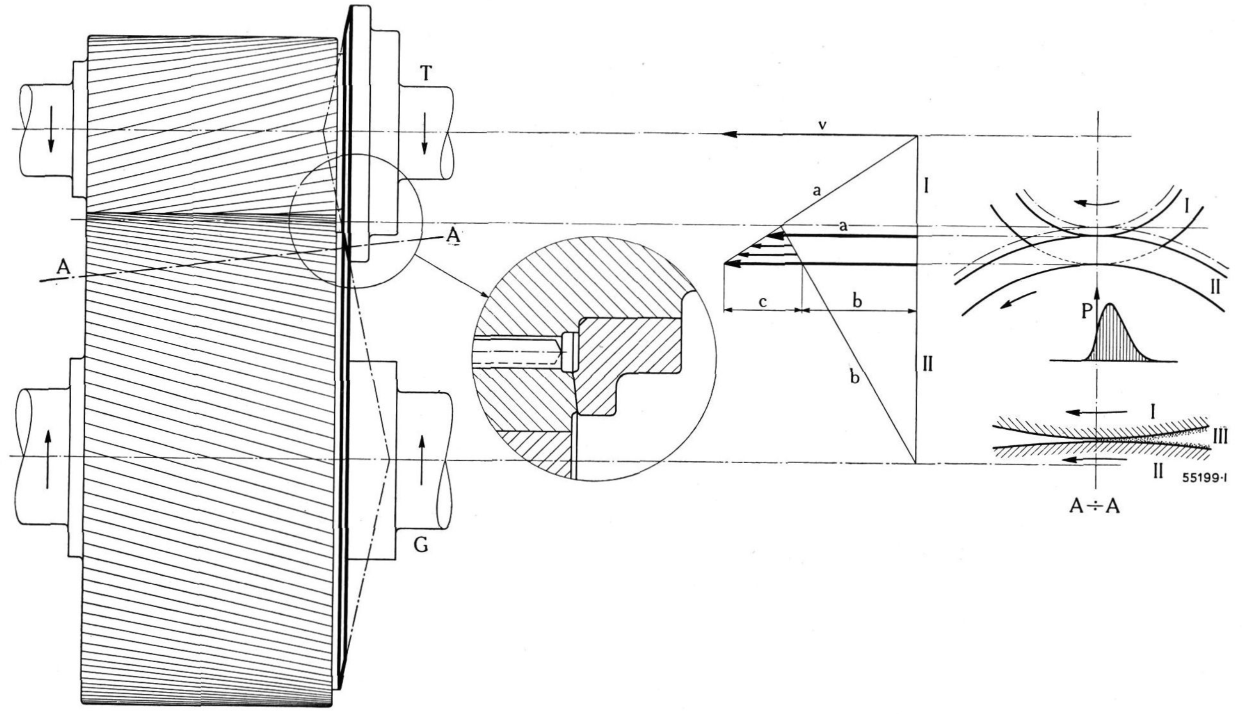


Fig. 2. — The "Thrust collar", a noteworthy design element for single helical gears.

I. Pinion. T. Driving. P. Pressure in the oil film. a. Peripheral velocity of pinion.
II. Wheel. G. Driven. v. Velocity. b. Peripheral velocity of wheel.
III. Oil. c. Slip velocity at the thrust collar.

The thrust collar permits the use of single helical gears with all their advantages. It takes up the axial thrust at the place where it originates, so that no thrust is transmitted to the pinion and wheel shafts and to the casing. The slightly conical bearing surface of the thrust collar rests against a corresponding surface on the gear wheel rim. These conical surfaces make contact along a generatrix. A strong oil film forms in the wedge-shaped zone in front of this line. The thrust collar losses are smaller than those of a thrust bearing, due to the fact that the thrust surfaces are near the pitch circle and hence the difference in velocities is small.

They have received considerable attention from engineering circles. The best evidence in this respect, is provided by the fact that we have been consulted, and asked to render assistance, in a number of cases where troubles were being experienced with gears of other make.

Our Design Technique.

The fundamental object of the gear is to couple together two rotating machines, each of which is operating under most favourable *normal* speed conditions. This explains why there are limits to the standardization of gears and why in many cases a special design is necessary.

In general, all our gears are provided with helical involute shaped teeth. Helical teeth have the advantage that there are always a number of teeth meshing at the time, so that load is gradually applied to the full width and the stresses remain moderate. Experience shows that they run much more quietly than straight tooth spur gears. We employ, wherever possible, single helical gears.

Since it has become possible by the application of the Mitchell principle to deal safely with axial thrusts of any magnitude, there is fundamentally no longer any necessity to use herring-bone wheels. Single helical wheels have the great advantage that they are lighter, that they can be more accurately machined and more easily erected, and accordingly run more quietly than herring-bone wheels. Since the space between the two tooth flanks such as is necessary in herring-bone gears for the end of the travel of the hob can be done away with, gears with single helical teeth can be built both smaller and lighter.

The axial thrust, inevitable with single helical gears, is taken care of by means of an element introduced by us to engineering, namely, the thrust collar on the pinion shaft. The operating principle is shown in Fig. 2. Because the axial thrust is not transmitted to the shafts, but is taken up directly at the spot where it is produced, couplings with axial clearance may be employed to connect the machines together.

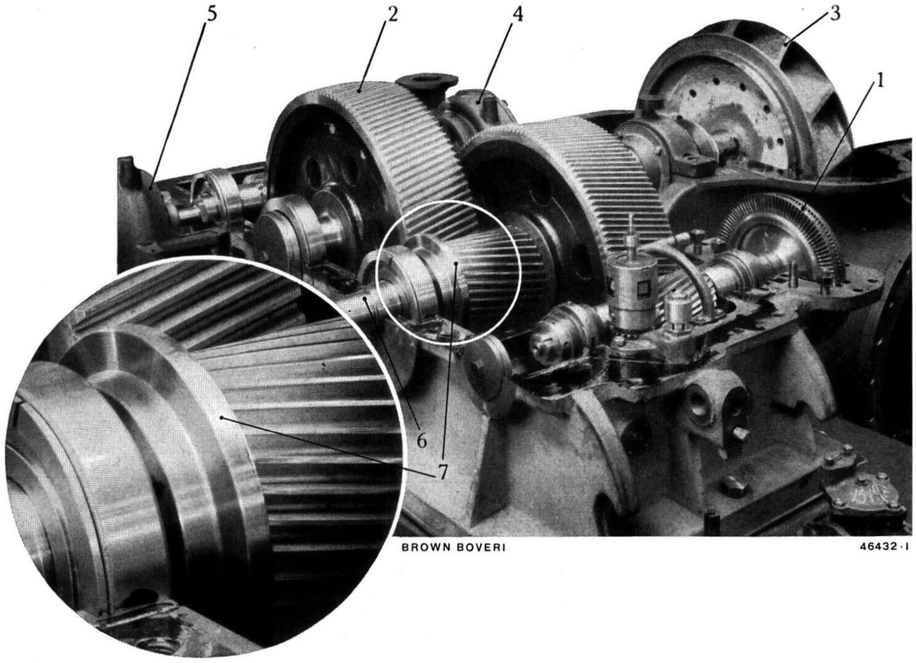


Fig. 3. — Auxiliary unit for a condensing plant with helical gearing and dual drive.

1. Driving turbine n = 9500 r. p. m.
2. Helical gear unit with 3 pinions on 2 wheels.
3. Cooling water pump n = 357 r. p. m.
4. Condensate pump n = 1800 r. p. m.
5. Oil pumps n = 1800 r. p. m.
6. Shaft extension for coupling with a direct current motor.
7. Thrust collar.

The auxiliaries of the main propulsion machinery are combined by gearing into a single unit. By the use of a number of pinions and gear wheels the most suitable speeds can be given to both the driving and the driven machines. On the high seas, the set is driven electrically, the power being supplied from the main turbine or from the propeller shaft. When in port, the set is driven by the auxiliary turbine 1.
The illustration is a typical example of the use of helical gearing with thrust collar (see also Fig. 2).

We were the first to make extensive use of single helical wheels, mainly for auxiliary machinery (Fig. 3)[1], but also for propelling turbines.[2]

For large outputs, especially when — as in the case of marine turbines — it is necessary to be able to run in both directions, only double helical gears as are shown in Figs. 4 and 5, can come into consideration. For large outputs, single helical wheels would become too wide; a uniform distribution of the pressure over the entire tooth width could not be ensured, because of the deflection of the pinion shaft.

In the case of high power plants, it is advantageous to subdivide the power among three or four pinions (Figs. 4 and 6); the distribution of the tangential force over a number of points of the wheel periphery enables a better utilization of the latter; it can be made both smaller and lighter. The subdivision of the power also results in shorter, more compact and more reliable turbines. The latter, especially in the case of four pinions, are disposed on both sides of the gear.[3] Fig. 6 shows one side of a geared marine turbine plant of large output, employing four pinions. To facilitate erection, the pinion shaft bearings are made adjustable in all directions.

Where weight is of importance, the large *gear wheel* is built up of separate forged and cast steel parts (Figs. 1 and 4). If weight is not important, the wheel body may consist of a special cast iron.

The same applies to the gear casing, which may accordingly be of cast iron or of welded construction (Figs. 6 and 7).

A factor of the first importance for the life of the gear is the lubrication. Especially in this field we have, as already mentioned in the beginning of this

[1] Auxiliary machinery of this type is described in detail in the article by Klingelfuss/Schmid: "Turbine-driven Marine Auxiliaries", on p. 290 of the present number of this journal.

[2] See The Brown Boveri Review 1922, pp. 137, 138, 145, also the article by C. G. Wahl: "Brown Boveri Turbines for the Propulsion of Merchant Vessels" on p. 227 of the present number of this journal.

[3] An arrangement of this sort for torpedo boat units of 20,000 H. P. at 4200/3200/480 r. p. m., of which a number of examples have been supplied, is illustrated in section and in plan in the Brown Boveri Review 1922, p. 143.

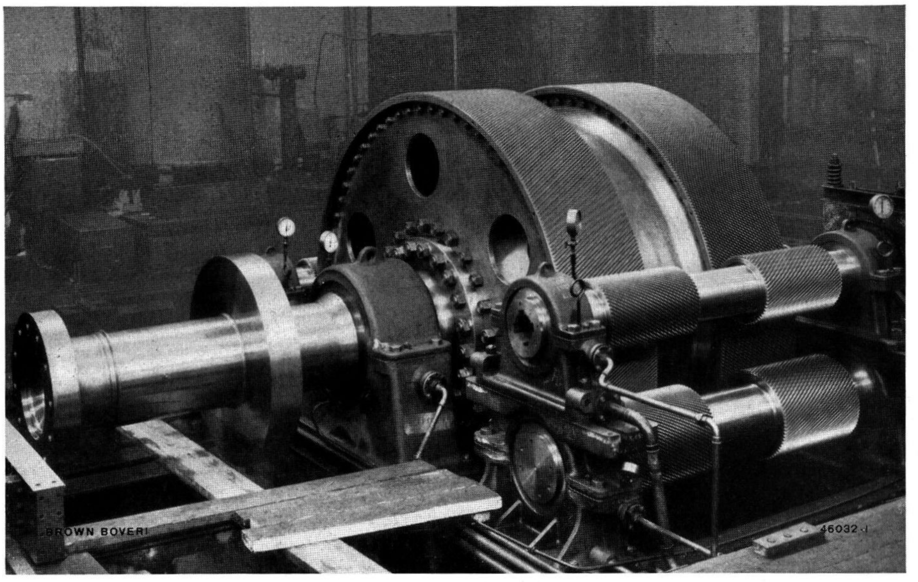


Fig. 4. — High-power, three-pinion marine gear during works tests.

Wheel diameter 3200, tooth width 2 × 650 mm. The finished gear sets are carefully tested for accuracy.

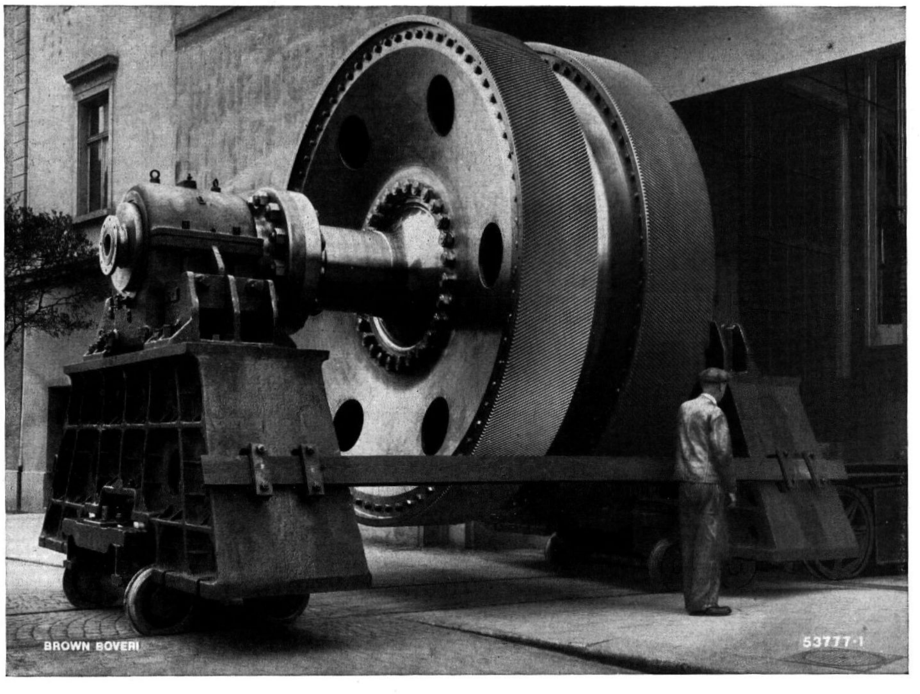


Fig. 5. — Transport of a high-power marine gear into the overspeed and balancing pit.

The finished gear wheels are thoroughly tested dynamically before being mounted in the casing.

article, done a considerable amount of research and have given the most careful attention to the design of the lubricating arrangements. The individual oil spray nozzles (Fig. 8) can be shut off and dismantled for cleaning during operation. It is in principle correct to employ as heavy an oil as possible for the lubrication of the teeth, but for the bearings, on the other hand, a thin oil is more suitable. As it is not practical to have two oil systems, an oil of average viscosity is employed, and that supplied to the teeth is cooled to a lower temperature than that going to the bearings.

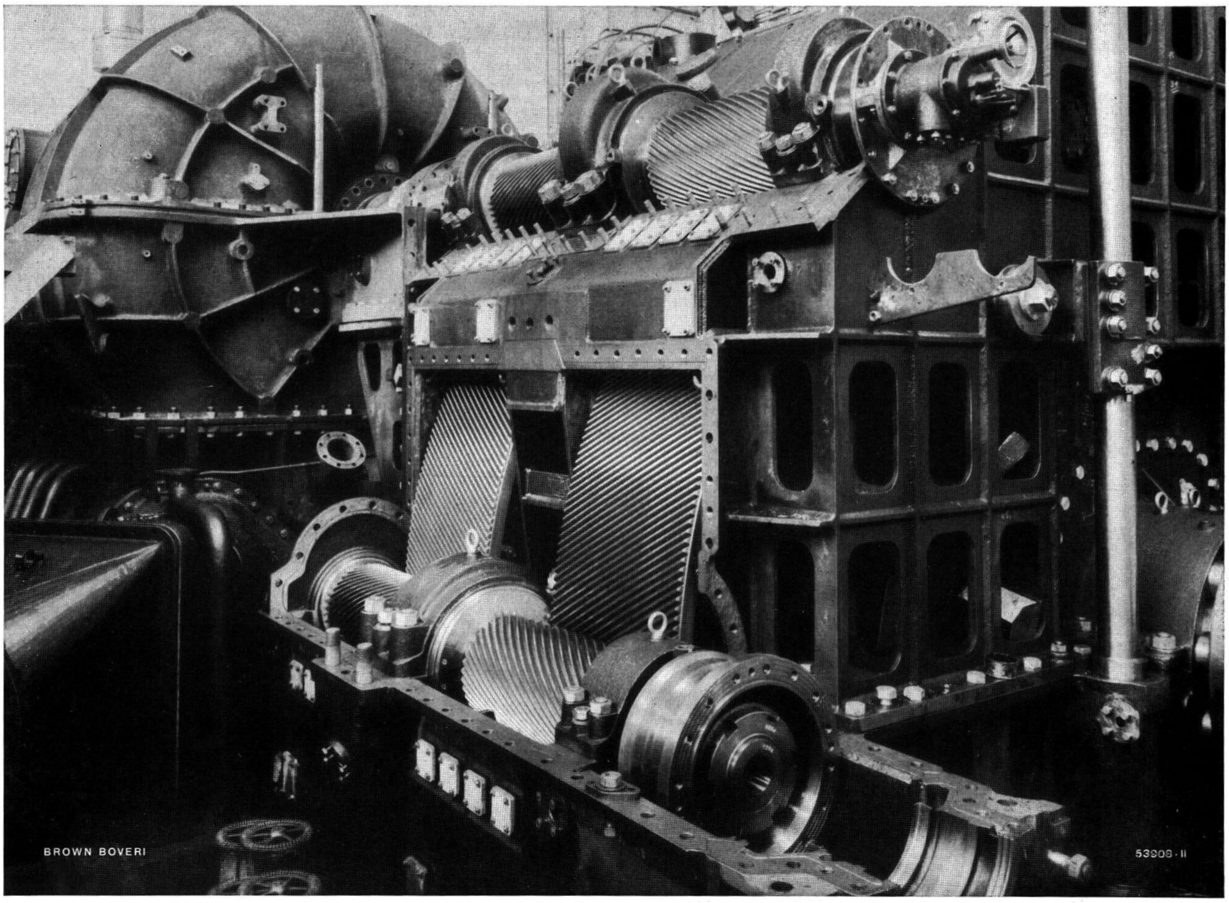


Fig. 6. — Partial view of a high-power marine turbine plant during erection at the works.

The total power is distributed among four pinions. The four turbine cylinders are disposed in front and behind the gear, the low-pressure cylinders visible in the background being on a high level, so as to enable the condensers to be placed immediately below them.

An extensive subdivision of the gear casing provides for easy erection and overhaul of the pinions. The longitudinally movable couplings allow any pinion to be removed without dismantling the corresponding turbine rotor, and vice versa.

Our Workshop Equipment for Gear Manufacture.

As already mentioned in the beginning of this article, the fundamental conditions for a long life and for a quiet running of the gear is accurate manufacture. This in turn requires a hobbing machine free from errors, accurate hobs and a carefully trained personnel.

Systematic investigations conducted on our gear-cutting machines, and untiring efforts given to the improvement of the table drive, have after decades of expensive and laborious work, enabled a unique degree of precision of the machines to be obtained. The results achieved are described elsewhere in this issue [1]. Fig. 9 shows as an addition to that article, the results of the measurements of the tooth pitch error on a high power gear wheel of approximately 4000 mm diameter, of which a number of examples have been built.

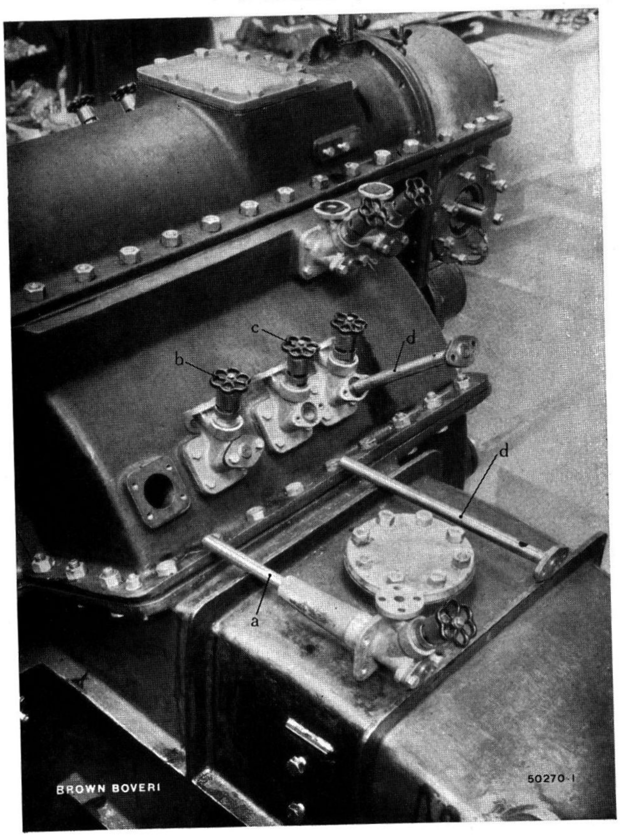
Our gear-cutting machines are installed in a special room, the temperature of which is maintained constant within 1 ^{0} C throughout the year by means of an air conditioning plant (Fig. 10). How sensitive these machines are to the changes of temperature between night and day was previously discovered by us when the first gear hobbing machine stood unprotected in the workshops exposed to daylight. It was possible from the markings on the tooth flanks, to state exactly which parts had been machined in daylight, and which had been machined at night.

The machining of the large gear such as that illustrated in Fig. 10 requires, including setting up of the machines, a total time of about three months. During the last finishing out the machine is in service uninterruptedly day and night for ten days. This is

[1] A. Meldahl: "Why does a Gear sing? How can the Singing be avoided?", p. 284.

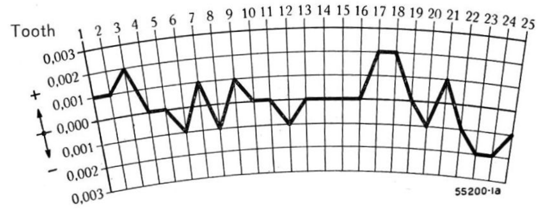
Fig. 7. — Lower casing half of a unit similar to that shown in Fig. 6 in the process of welding.

Experienced specialized workers weld the entire casing, consisting of steel plate and steel castings, together. The finished casings are then annealed. The manufacture of these parts also demands a high standard of workmanship.

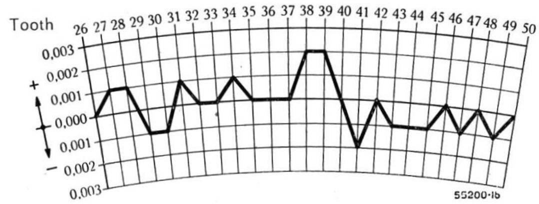


Left-hand wheel section.

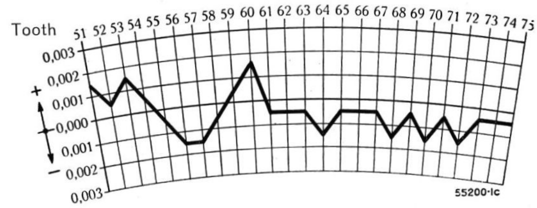
Measurements in millimetres.
Largest pitch error from tooth to tooth = **0·002 mm.**



Measurements in millimetres.
Largest pitch error from tooth to tooth = **0·002 mm.**

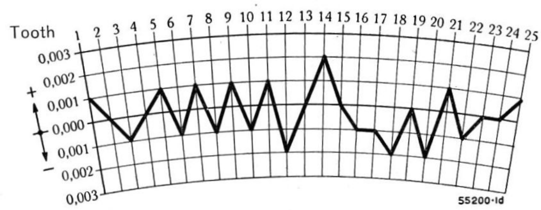


Measurements in millimetres.
Largest pitch error from tooth to tooth = **0·002 mm.**

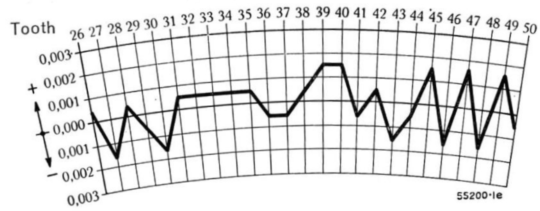


Right-hand wheel section.

Measurements in millimetres.
Largest pitch error from tooth to tooth = **0·003 mm.**



Measurements in millimetres.
Largest pitch error from tooth to tooth = **0·003 mm.**



Measurements in millimetres.
Largest pitch error from tooth to tooth = **0·002 mm.**

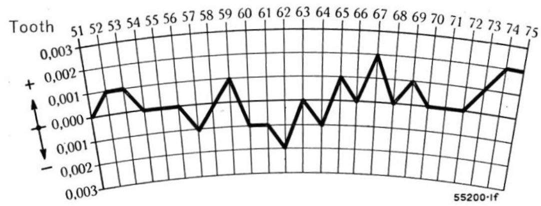


Fig. 9. — Measured pitch errors from tooth to tooth of a gear wheel as illustrated in Fig. 1 of about 4000 mm diameter.

The largest deviation measured in three tooth groups 120° apart of 25 teeth each amounted only to $^3/_{1000}$ mm.

Our gear cutting machines enable masterpieces of precision to be produced.

Fig. 8. — Lubrication arrangements of a marine turbine.

A reliable lubricating system is essential for every gear. In large gears, the lubricating oil is sprayed by nozzles between the meshing surfaces of wheel and pinion.

a. Oil spray nozzle ready for mounting.
b. Oil spray nozzle in service position.
c. d. The individual nozzles may be shut off and removed for cleaning during service.

Even minor components of our gears such as these spray nozzles are carefully designed and facilitate the work of the attendants in supervising the plant.

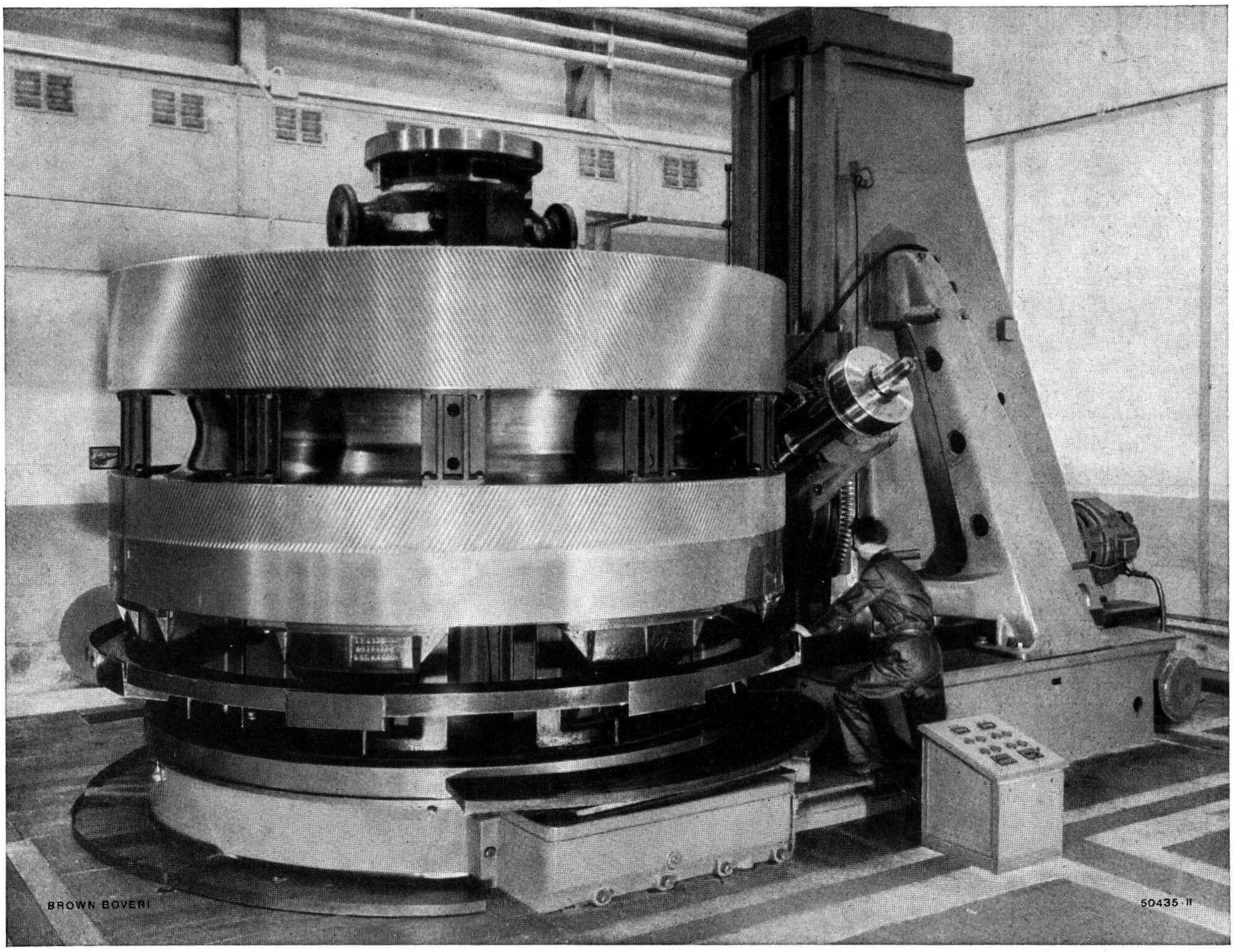


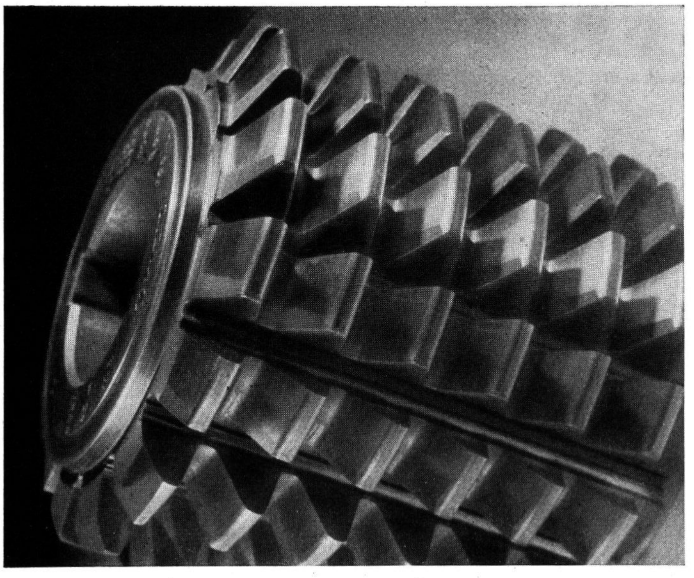
Fig. 10. — **Our largest gear-cutting machine** can cut gear wheels up to 5000 mm diameter and 50 tons in weight. In order to assure the highest precision of cutting, the machines are accommodated in a closed bay, the temperature of which is maintained constant throughout the year within 1° C by means of an air conditioning plant. The check gauges, accessories and hobs are kept in the same room.

the most critical period of manufacture, as any interruption during this cut would render the wheel unusable. Every precaution is accordingly taken to prevent any disturbance, such as the failure of the electric supply causing the machine to come to a standstill.

All improvements of the gear-cutting machine would be useless if the form of the tool, that is, of the hob, generating the teeth were not perfectly accurate. During the above-mentioned final cut of the large wheel lasting ten days, every tooth of the hob travels a distance of 120,000 m, during which time the accuracy of its profile must absolutely be maintained. The hobs are made in our own workshops.

All gear cutting machines, whether in service or not, are under the constant supervision of the same personnel. A similar systematic control is extended to the products of these machines. The analysis of the results of this control continually shows us the way to further improvements of our arrangements for the manufacture of precision gears.

(MS 881) *E. Klingelfuss & V. Tödtli. (Hv.)*



WHY DOES A GEAR SING?

HOW CAN THE SINGING BE AVOIDED?

One of the high-class tools employed to cut our noiseless gears.

Decimal Index 534.837:621.831

During the last ten years numerous measurements have been carried out on large helical gears. The results of these measurements are published in the present article. They show how the untiring efforts of the workshop staff in putting into practice the deductions of theoretical investigations have enabled the precision of manufacture to be increased during these years.

A gear, the speed ratio of which is absolutely constant, makes fundamentally no noise. It is, however, extremely difficult to get sufficiently closely to this ideal condition. With straight tooth gears it is hardly possible to avoid periodic variations of speed due to the deformation of the teeth, even if this deformation is allowed for when designing the tooth profile, because perfect compensation is only achievable for one particular load. In the case of helical gears, however, it is in theory possible to obtain an absolutely uniform speed of the driven wheel, even if the tooth form is not entirely perfect.

The first designers of fast running gears for large powers, therefore, invariably chose helical teeth, and in order to neutralize the axial thrust of single helical gears, they made them double helical. Notwithstanding this, they all made the painful experience that certain gears would begin to sing when the load was applied. Depending on the conditions, the singing might take the form of a deep organ note or of a shrill scream, and whilst in other respects the gear would appear to be quite satisfactory, the singing might assume such an intensity as to be quite unbearable in the immediate vicinity of the machine.

The cause of this singing was explained by Sir Charles A. Parsons in a lecture delivered to the Institution of Naval Architects.[1] Parsons also described an ingenious process, namely the "creep process", for avoiding it. This lecture seems to have been generally overlooked and even firms specializing in the manufacture of gear cutting machines appear to have remained ignorant of it, for only so is it possible to explain why again and again, helical gears cut on machines built by first-class machine-tool makers would sing when on load, sometimes to such an extent as to be quite unusable.

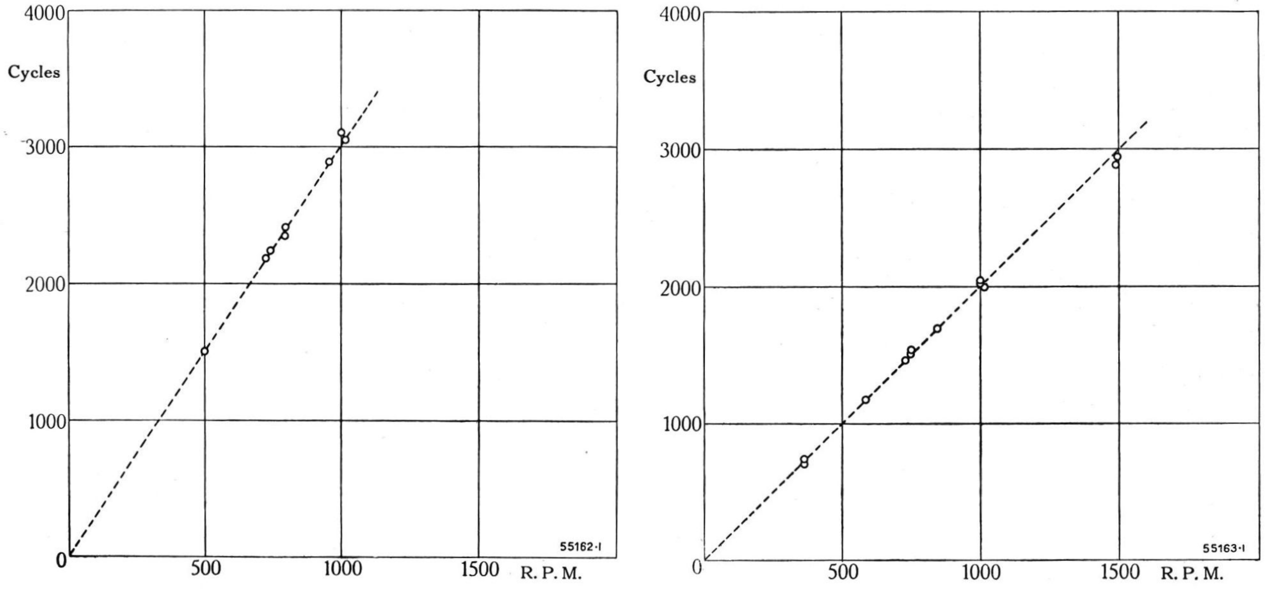
Since such gears — even when they make an unbearable noise — usually show no signs of undue wear, provided they are in other respects properly designed, only the pitch of the note which they give out can furnish a clue to the cause of the singing. If the note frequency is found to coincide with the number of teeth meshing per second, then it is fairly certain that the setting of the pinion is incorrect. Such an error can also be recognized from the contact marks and eliminated accordingly. Often, however, no such relation can be traced between the number of teeth and the note frequency.

About ten years ago the author had occasion to measure the tone frequency of the notes given out by a large number of gears. The gears had been cut mainly on two machines of different origin; power, speed and reduction ratio varied within wide limits. The results of these measurements are plotted in Figs. 1 and 2 as a function of the speed.

It was found that the frequency of the objectionable note was definitely independent of the design of the gear. The only determining factor was the machine on which the gear was cut. If a wheel was cut first on one machine and then given a finishing cut on the other, the note changed also. As shown by the two curves, the frequencies of all the notes measured lie on a straight line passing through the origin; the slope of this line corresponds in the one case to 180 and in the other to 120 vibrations per revolution.

Since the table of each machine had a worm drive and the number of teeth of the dividing wheel was 180 in one case and 120 in the other

[1] Engineering, March 14th, 1913.



Figs. 1 and 2. — Measured audible frequencies plotted as function of the speed.

All gears here considered were cut on a hobbing machine with a 180-tooth dividing wheel.
---- Corresponds to 180 disturbances per revolution.

All gears here considered were cut on a hobbing machine with a 120-tooth dividing wheel.
---- Corresponds to 120 disturbances per revolution.

The main disturbing tone always has a frequency corresponding to the number of teeth of the dividing wheel.

the measurements showed therefore conclusively that the worm drive was mainly responsible for the singing.

An entirely different means of determining the disturbing frequency is afforded by the so-called shadow-lines. If the gear has fine teeth, pressure marks are sometimes visible, these marks taking the form of lines parallel to the axis of the wheel. It is then possible to count accurately on the periphery of the wheel the number of teeth of the dividing wheel of the gear cutting machine. With coarser teeth and smaller obliquities, the individual marks are spread out further apart, forming a diamond pattern, which appears to the eye as a number of helical lines around the wheel. In every case, it is readily possible to determine the disturbing frequency from the number and position of these lines.

With some practice, and given a suitable illumination, it is even possible to see these shadow lines directly after making the finishing cut. In the case of large wheels, which are cut at constant temperature, this visual test is often of uncanny precision. To mention an example, shadow lines could still be discerned on a cleanly cut gear, although the error as determined by the most careful measurements amounted in this case to only a fraction of a second of arc.

That a worm drive can easily cause a periodic disturbance is evident from the following reasoning: In order that a worm drive shall operate properly, it is necessary first that the worm shall have exactly the same form as the hob used for cutting the worm wheel, and secondly, that the worm shall be in exactly the same position relative to the wheel as was occupied by the hub during the cutting operation. The necessary information in regard to these points can be furnished only by the maker of the worm wheel; no subsequent check is possible. If, for instance, the worm runs eccentrically, or if it wobbles, or if it moves backward and forward in its thrust bearings, a careful examination of the machine may finally enable the fault to be discovered. If, however, the same fault was present during the cutting process, the effect on the table of the gear cutting machine is, as can readily be seen, exactly the same. *There is, however, in practice absolutely no possibility of determining this by any later examination of the dividing wheel.*

Every maker who has seriously taken up the manufacture of large gears must, therefore, sooner or later give his attention to the subject of the table drive of the gear cutting machines. It is no mere chance that a large proportion of the gear making firms make their own dividing wheels, and in some cases even build the gear hobbing machine themselves, because only in this manner have they a complete check on all the essential elements for the manufacture of a faultless gear.

The question is, what possibilities are there of testing the table drive, and, if possible, of measuring

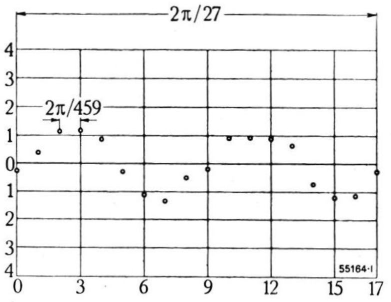


Fig. 3. — Variation of pitch measured on a large gear wheel.

Every point represents the average of 27 independent measurements.

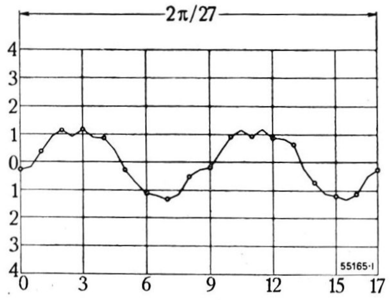


Fig. 4. — "Apparent error" of frequency 54 per revolution.

This frequency results from a first attempt at connecting together the measured points by a periodic curve. No such periodic error exists in reality.

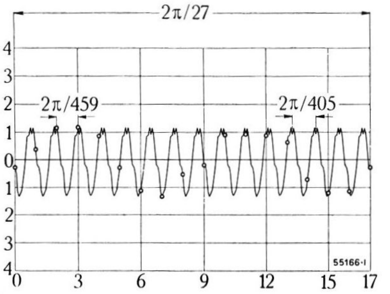


Fig. 5. — Real periodic error of frequency 405.

This curve also connects all the measured points. Although at first sight it appears unnecessarily complicated, it really does represent the true error. The saw teeth are due to accidental errors.

The interpretation of the pitch measurement results is not easy and requires much experience and skill.

the drive errors? There are indeed different possibilities, which may be classed as follows:—

(a) Direct measurement on the machine.

(b) Measurements on cut wheels, namely:

1. On special check wheels.
2. On ordinary gear wheels.

Direct measurements on the machine can, for instance, be carried out by rotating the worm through a definite fraction of a revolution; at the same time accurately measuring the corresponding movement of the table. If the worm drive is correct, the table must each time move through exactly the same angle; generally, the motion of the table will fluctuate periodically. From the individual measurements it is possible to construct the summation curve and from the latter to derive the total variation of the table drive.

Already the first measurements of this kind which were effected about ten years ago at the suggestion of the author, furnished extremely valuable information as to the manner in which the table drive functioned, and immediately led to considerable improvements. The measurement is, however, rendered difficult by the fact that the table must be brought to rest after every movement. If during the period of rest the oil film in the bearings is squeezed out, it is difficult to obtain a uniform motion of the table which is essential for accurate measurements. Methods and arrangements were, therefore, devised to enable measurements to be carried out on the machine while in motion; it would, however, take up too much space to describe these methods in detail here.

If special test gears are cut for the purpose of testing the worm drive, it would at first sight appear advisable to give the test wheel at least three times the number of teeth of the dividing wheel, in order that several measurements might be effected within one pitch of the worm wheel. Both the cutting operation and the measurements then require a considerable time.

If measurements are to be made on ordinary gear wheels, the number of teeth of which is fixed by other considerations, many interesting points arise which it is proposed to deal with in detail below. In the first place, only whole pitches can be measured with the accuracy which is here required, intermediate points are out of the question. Hence, only isolated points of the required error curve are available, whereby the interval between the points may often be greater than the pitch of the error looked for. Notwithstanding this, such a series of pitch measurements may provide very useful information.

Fig. 3 shows the result of such a pitch measurement. Every point represents the average of 27 separate measurements, so that observation errors are practically excluded. At first sight an error appears to be present, which repeats itself 54 times in the circumference (Fig. 4). This, however, is an illusion. *In reality, there is a periodic error of frequency 405 per revolution.* Fig. 5 shows that a wave curve with 405 waves also passes

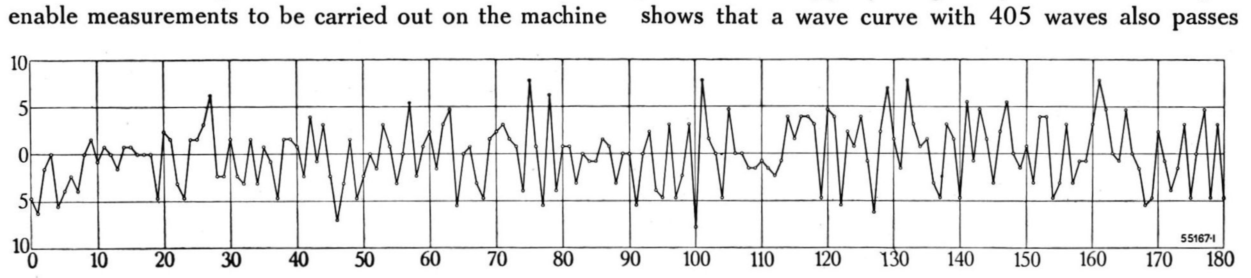


Fig. 6. — Pitch variations measured on an old dividing gear wheel with 180 teeth.

The "apparent error" of frequency 60 per revolution is clearly visible. The real error has a frequency $180 + 60 = 240$ per revolution.

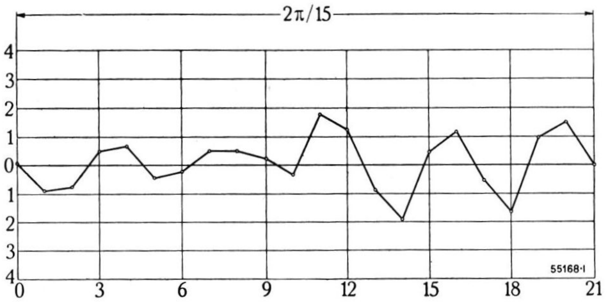


Fig. 7. — Pitch variations measured on a dividing wheel of 315 teeth.

Each point is the average of 15 independent measurements. The "apparent error" has a frequency of $5 \times 15 = 75$ per revolution. The real error has a frequency of $315 - 75 = 240$ per revolution (measurements of the maker).

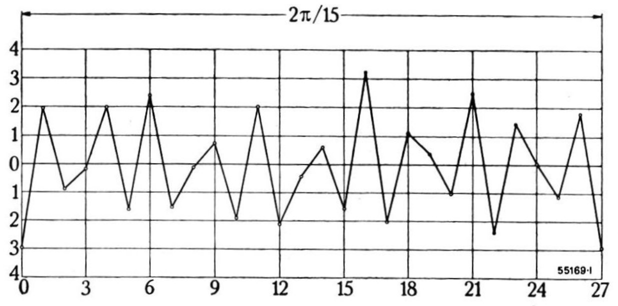


Fig. 8. — Pitch variations measured on a dividing wheel of 405 teeth.

Each point is the average of 15 independent measurements. The "apparent error" has a frequency of $11 \times 15 = 165$ per revolution. The real error has a frequency of $405 - 165 = 240$ per revolution.

All these dividing wheels show an error of frequency 240 per revolution corresponding to the number of teeth of the dividing wheel of the parent machine of the gear cutting machine maker.

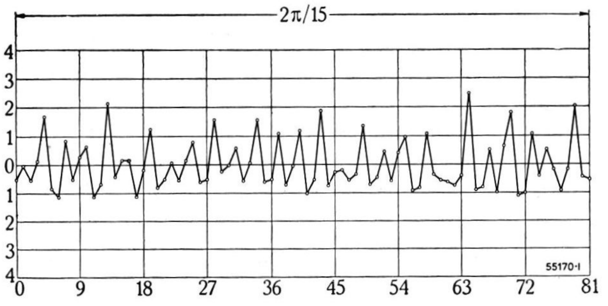


Fig. 9. — Pitch variations measured on a check wheel.

Each point is the average of 15 independent measurements. A periodic error of frequency $27 \times 15 = 405$ per revolution is clearly visible.

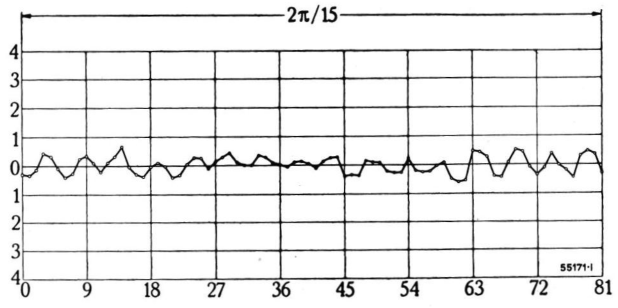


Fig. 10. — Pitch variations measured on a check wheel.

Each point is the sliding average of three neighbouring points of Fig. 9, that is of a total of 45 measurements. The sliding average suppresses the error of frequency 405 per revolution. A second error of frequency 240 per revolution is thereby caused to appear.

This check wheel was cut on a hobbing machine with the dividing wheel of Fig. 8. In addition to the worm error, the error of frequency 240 per revolution which according to Fig. 8 is present in the dividing wheel also shows up.

through all the measured points; *moreover, the dividing wheel of the hobbing machine has 405 teeth.* There is in this case no cause for periodicity 54 per revolution; it is the apparent frequency resulting from the interference pattern of the 459 test points, i. e., the teeth of the wheel cut and of the 405 waves.

The author had occasion to test a large number of dividing wheels in this manner. The pitch measurements were supplied partly by Brown Boveri and partly by allied firms. Measurements were available for:

3	dividing	wheels	with	180 teeth
1	"	wheel	"	320 "
1	"	"	"	315 "
1	"	"	"	405 "

All dividing wheels had been supplied by the same specialized maker of gear cutting machines, and had been cut on the firm's original parent hobbing machine. The results of these measurements were as follows:—

The dividing wheels with 180 teeth, showed an error with a frequency of 60 per revolution (Fig. 6).

The dividing wheel with 320 teeth showed an error with a frequency of 80 per revolution.

The dividing wheel with 315 teeth, showed an error with a frequency of 75 per revolution (Fig. 7).

The dividing wheel with 405 teeth, showed an error with a frequency of 165 per revolution (Fig. 8).

At first sight no relation could be seen between these errors.

On regarding them, however, as the apparent errors resulting from the interference pattern of the number of test points and of the unknown real error frequency, the following results are obtained:—

$$\begin{aligned} 180 + 60 &= 240 \\ 320 - 80 &= 240 \\ 315 - 75 &= 240 \\ 405 - 165 &= 240. \end{aligned}$$

The real error had, therefore, in each case a frequency of 240 per revolution, which corresponded exactly to the number of teeth of the dividing wheel of the parent hobbing machine.

After having thus once again confirmed the original experience of Parsons, every effort was made to eliminate this error. Given the accuracy required of the modern gear, this meant a severe trial of the ability

The step by step improvement of the large hobbing machine at our Baden works.

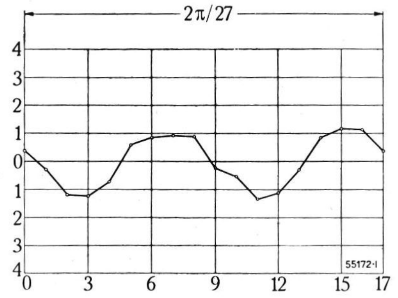


Fig. 11. — Dividing wheel recut. The error of frequency 240 per revolution of Figs. 8 and 10 has disappeared.

The "apparent error" with a frequency $2 \times 27 = 54$ per revolution is still present. The real error has a frequency of 405 per revolution and comes from the worm drive.

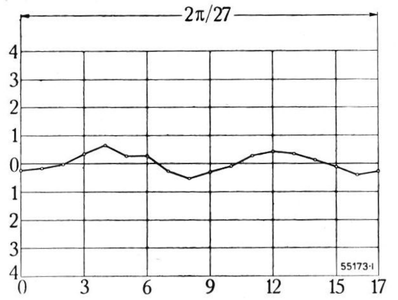


Fig. 12. — Worm drive improved; the "apparent error" of frequency $2 \times 27 = 54$ per revolution is considerably reduced.

An error of this magnitude is of no consequence for the quiet running of the gear.

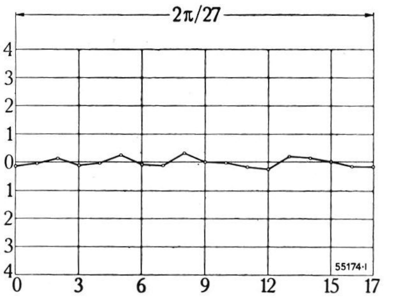


Fig. 13. — Worm drive again improved and readjusted; no periodic error can be traced.

The smooth shape of the curves and the agreement of the measurements show with what accuracy and skill the measurements were made. Both manufacture and measurements represent outstanding achievements of precision.

On this machine, we cut wheels up to 5 m diameter and for shaft powers of up to 80,000 H. P.

and patience of all concerned. The result was, however, most satisfactory and undoubtedly represents an outstanding achievement in modern heavy machinery engineering.

First a check wheel was cut; the result of which is shown in Fig. 9. The worm error of frequency 405 per revolution can be clearly seen. If this error be eliminated by taking a sliding average over every three test points, i. e., the arithmetic mean of three neighbouring points (Fig. 10), the error with a frequency of 240 per revolution appears, which error had already been shown up by the pitch measurements effected on the dividing wheel and which originated from the parent machine of the maker. In this condition, the machine could not give satisfaction.

First, the dividing wheel had to be recut, in order to eliminate the error of frequency 240 per revolution coming from the parent machine. A first gear

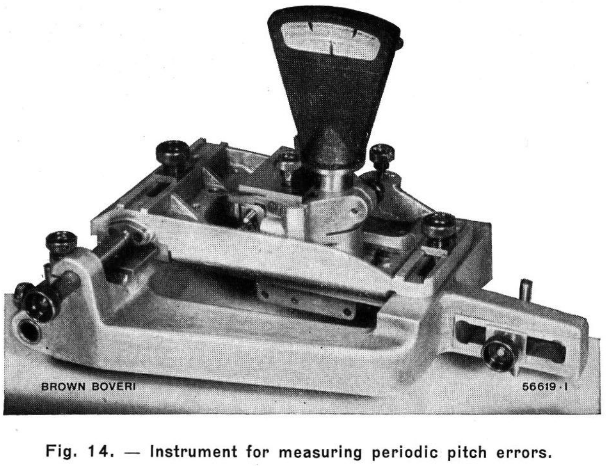


Fig. 14. — Instrument for measuring periodic pitch errors.

With this instrument periodic errors may be measured directly. This measuring device was developed by us on a new principle and built in our Baden works. It represents the latest advance in this field.

wheel cut then gave the pitch measurements reproduced in Fig. 11; the error of frequency 240 has disappeared, but the error of frequency 405 is still present. By means of improvements to the worm drive and careful adjustments, the result given by Fig. 12 was obtained upon cutting a second wheel; the periodic error of frequency, 405 per revolution has considerably diminished. *A third trial was finally so far successful that the drive error could be considered to be excluded.* Fig. 13 shows no periodic error of frequency 405 per revolution. As the average error, resulting from the fact that the measuring device was read to the nearest unit, was about $^1/_7$ of the scale division; the remaining errors can definitely be considered as "accidental". The measurement was itself an achievement of precision.

If is evident that, because of the expense, such precision measurements can only be carried out exceptionally. We have, therefore, recently developed an instrument for measuring *directly* the periodic error on large helical gear wheels. Measurements made by this new method have confirmed exactly the results obtained by the much more complicated pitch measurements; this procedure, therefore, represents an important advance in this field, and enables us to control the accuracy of our reduction gears much better than was hitherto possible (Fig. 14).

A fundamentally different way of avoiding the transmission of periodic errors of the table drive to the work-piece, if not entirely, at least to a considerable extent, is afforded by the "creep process" described in the above mentioned Parsons lecture. It consists essentially in providing a planetary drive for the table. With the direct table drive, the errors of the dividing wheel and of the worm drive repeat themselves at every revolution so that the cutter

Fig. 15. — Gear wheel cut 30 years ago by the "creep process".
Thus looked the first wheels of this kind. At that time, as can be seen, errors of several tenths of a millimetre had to be distributed. Notwithstanding this, the gears ran quietly.

Fig. 16. — Gear wheel cut with a modern planetary drive.
The chequered surface of the tooth flanks is characteristic of the automatic compensation of the error. Wheels of this type cut on a good machine run practically without noise.

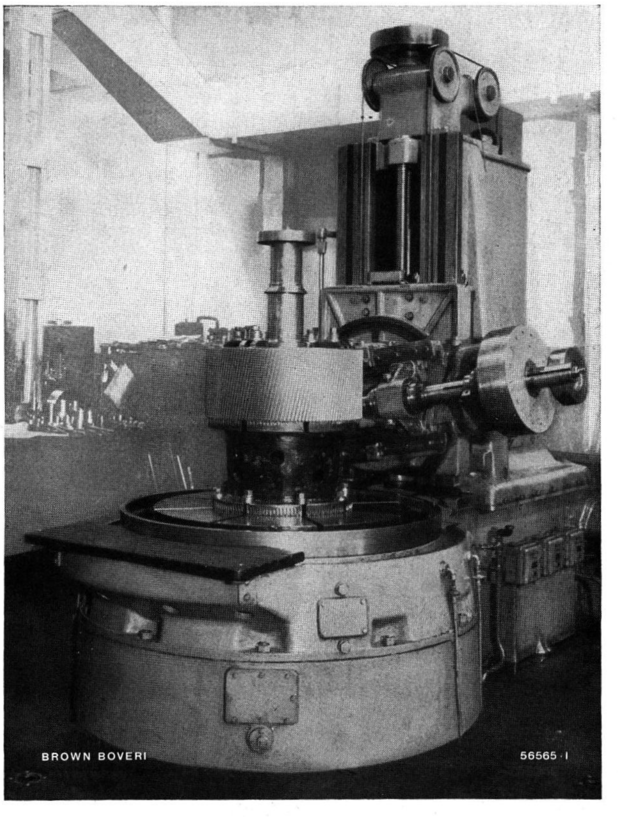


Fig. 17. — Gear wheel cutting machine with planetary table drive.
This table drive was built in our works and to our own design. The drive assures a substantial compensation of errors; remaining errors are distributed by the planetary drive and thus rendered innocuous.
The only machine of its kind, it serves mainly for the continuous improvement of our gears.

therefore copies the same error on the same straight line parallel to the axis of the gear wheel. Every error of the table drive corresponds to a complete generatrix line of the cut gear. In the case of a planetary drive, on the other hand, the errors of the individual wheels do not repeat themselves after a complete revolution of the table. The consequence is that the hob does not repeat every error along a generatrix but along a helix. The errors do not, therefore, manifest themselves at the same instant on the finished gear; instead, only the high spots on the tooth flanks bear the load.

Experience shows that such gears run very quietly. The distribution of the error gives the tooth flanks a chequered appearance, by which the expert may recognize immediately that the gear has been cut on a planetary machine (Figs. 15 and 16). Laymen sometimes make the unjustified objection that such chequered tooth flanks are "inaccurate". If, however, a good planetary machine has been employed, the objection is quite unjustified.

The errors still remaining in the planetary machine are, as mentioned above, rendered innocuous by even distribution. Pitch measurements on wheels, cut on a planetary machine, can therefore have no real meaning. At every point along the width of the wheel a series of different values would be obtained. None of these values would agree with the real behaviour of the wheel.

The special advantage of the hobbing machine with planetary drive lies in the fact that under certain conditions, it achieves an automatic improvement of the gear produced as opposed to the hobbing machine with direct drive, which furnishes at the best a copy of the dividing wheel. We have, therefore, thoroughly investigated the theory of transfer of error and of the error elimination in hobbing machines with planetary drive, and basing on these studies we have fitted our own design of planetary drive to a hobbing machine of medium size (Fig. 17). The conversion was a complete success, and the gears cut on this machine run very quietly indeed. The machine is unique of its type; it is one of the contributions which will ensure that our firm will continue its high achievements in the field of gear manufacture.

(MS 893) *A. Meldahl. (Hv.)*

TURBINE-DRIVEN MARINE AUXILIARIES.

Decimal Index 629.12.063

The steam turbine has proved extremely suitable and found wide application for the drive of marine auxiliaries of all descriptions. The following notes give an insight into the part Brown Boveri has played in the development of this field. The factors which have to be taken into account in design are touched upon, while it is also shown that high turbine speeds make for enhanced reliability. A number of illustrations give an idea of the present state of development of turbine-driven auxiliaries.

MARINE machinery may be divided into two classes, i. e., the main machinery, for the drive of the propeller, and the auxiliaries. The term "auxiliaries", however, must not be allowed to detract from the significance of this category of marine equipment, inasmuch as it constitutes a bigger problem for both the designer and the ship's engineer than the main machinery itself. The auxiliaries are many and varied, and it is their reliability which ultimately determines the seaworthiness of the vessel. It is therefore not surprising that the classification societies should specify ample standby equipment for, e. g., lighting sets and pumps. At an early date[1] turbine-driven auxiliaries were recognized to be capable of fully meeting the most stringent requirements of marine operation, and forty years of untiring development work separate the original auxiliary turbines with their relatively slow speeds and correspondingly big spread from the present compact high-speed machines.

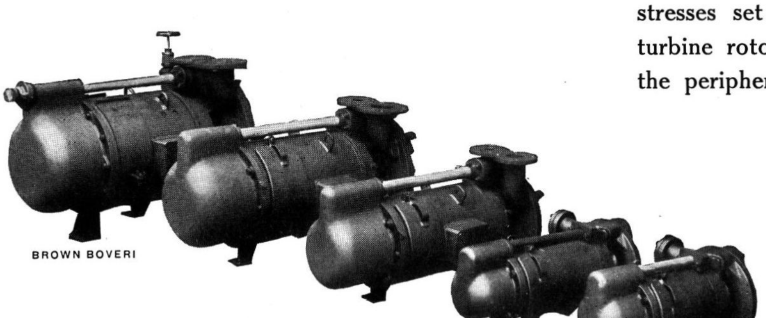


Fig. 1. — A series of lighting turbo-sets for installation on ships, dredgers, floating cranes and the like.

Standard sizes rated 0·5, 2, 5, 10 and 20 kW, live-steam conditions up to 30 kg/cm² abs, 350 ° C, 2 kg/cm² abs back-pressure. Speed 4500 to 3600 r. p. m. according to output. Compact, robust, and rainproof construction. Mass production methods of manufacture, together with permanent stocks of standard machines, ensure rapid delivery at best possible prices.

Basic factors for the design of marine auxiliaries were and still are:

Reliability.
Economy.
Minimum weight and space requirements.
Simple attendance and good overhaul facilities.

I. RELIABILITY.

The reliability of a machine depends not only on the suitability of the design, but above all on the stresses set up in the material. In the case of the turbine rotor these are determined fundamentally by the peripherial speeds of the impellers which are approximately the same at all revolutions, inasmuch as, for reasons of efficiency, they are fixed by the steam conditions. A further factor is the thermal expansion of both the turbine rotor and the casing, which, however, becomes less pronounced the smaller the absolute dimensions of the parts in question. The stresses involved by rapid starting or sudden changes in operating conditions, for instance, are thus smaller the higher the revolutions. This fact is corroborated by experience and, in view of the present unmistakable tendency to increase steam pressures and superheat temperatures, will become of greater and greater signi-

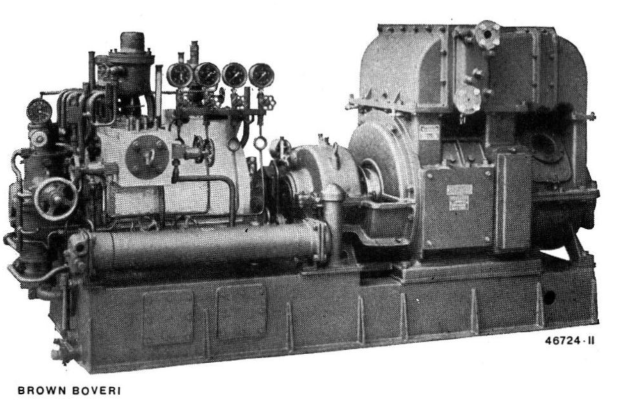


Fig. 2. — Example of a modern marine-type turbo-generator with forced-air cooling.

The turbo-generator is combined with gearing and has a water-jacketed forced-air cooling system, thus ensuring permanent reliability even at high ambient temperatures. In condensing operation the rotor expands the steam from 25 kg/cm² abs, 340 ° C to 0·1 kg/cm² abs in two impulse stages with two-row impeller wheels. Rating 120/156 kW, speed 10,500/3000 r. p. m. The features of this set are low weight and minimum space requirements.

[1] Grauert: "Ueber Dampfturbinen", Marine Rundschau 1904, where the first Brown Boveri marine turbo-generators of 65-kW rating are described. The prime movers of these sets had pure reaction blading.

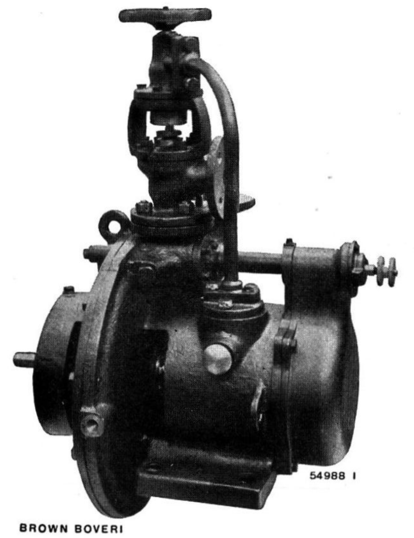


Fig. 3. — Small turbine of simple design for the drive of pumps, fans, and other marine auxiliaries.

This turbine is available in two sizes, 25 and 60 H. P. It can be operated with steam up to 30 kg/cm² abs, 425° C, against back-pressures of up to 2 kg/cm² abs. For speeds between 3000 and 5500 r. p. m. drive is direct, from 600 to 2500 r. p. m. through built-on gearing. This turbine is simple in construction, of excellent design, reliable, and inexpensive. It is the product of forty years' experience in the building of marine auxiliaries.

ficance as time goes on. It is therefore a fallacy to suppose that high-speed turbines are less reliable than low-speed units.

Reliability can also be enhanced by adopting certain protective measures, e. g., totally-enclosed construction (Fig. 1), or water-jacketed forced-air cooling of the generators (Fig. 2). Further improvements can likewise be achieved by providing special safety devices as, for instance, against excessively high back-pressures.

In the case of auxiliaries for small ships, where the crew is not particularly skilled in the handling of machinery, simple construction and attendance are virtually indispensable. Grease-lubricated ball or roller bearings greatly simplify attendance and improve reliability.

Ships with electrically-driven auxiliaries must be provided with standby steam-driven sets for all of the more important duties. The new small turbine illustrated in Fig. 3 is particularly suitable for this purpose.

II. ECONOMY.

The total power required by the auxiliaries necessary for the operation of the main turbine may be up to 10 %. This does not include, however, the generators for feeding the ship's power system to which the lighting installation and driving motors of the various auxiliaries both on and below deck are connected. In the case of warships the steam consumption of the auxiliaries attains even 15—25 % of that of the main machines according to the speed of the vessel. From these figures it will be clear that the economy of the individual auxiliaries must be given careful attention, especially in those cases where the exhaust steam can only partially be utilized for preheating the feed water. Drive by small independent condensing turbines does not constitute an economical proposition. The introduction of the geared marine turbine and high-pressure superheated steam further aggravated the driving problem by reason of the lower steam consumption of the main turbines and the greater power required by the auxiliaries.

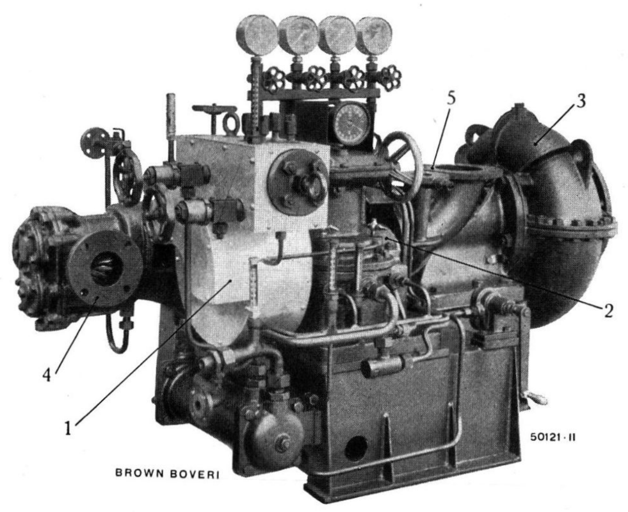


Fig. 4. — Pump set for a 3500 S. H. P. marine turbine installation.

1. Driving turbine, speed 13,500 r. p. m.
2. Reduction gear with various wheel-shafts.
3. Circulating pump, 1200 r. p. m.
4. Lubricating-oil pump, 1200 r. p. m.
5. Condensate pump, 4000 r. p. m.

The arrangement of the various pumps to form a set driven by a single auxiliary turbine brings about a substantial reduction in the steam required for the auxiliaries. The reduction gear enables both the prime mover and the various pumps to be run at their most efficient speed.

A substantial reduction in steam consumption can be achieved by arranging the pumps of the main turbine and condensing plant to form a single set and with the application of geared turbines to merchant vessels we directed all our effects to the realization of this arrangement[1]. In lieu of a large number of small independent turbines one single high-speed machine is provided which drives the various pumps through reduction gear. In this way it is possible to assign not only to the prime mover, but also to each pump, the revolutions conducive to maximum efficiency in each case. Fig. 4 depicts one of the latest pump sets.

[1] Dipl.-Ing. A. Gorgel: "Die Maschinenanlage der Passagier- und Frachtdampfer 'Thuringia' und 'Westphalia' der Hamburg-Amerika Linie", Werft, Reederei, Hafen 1923, p. 495.

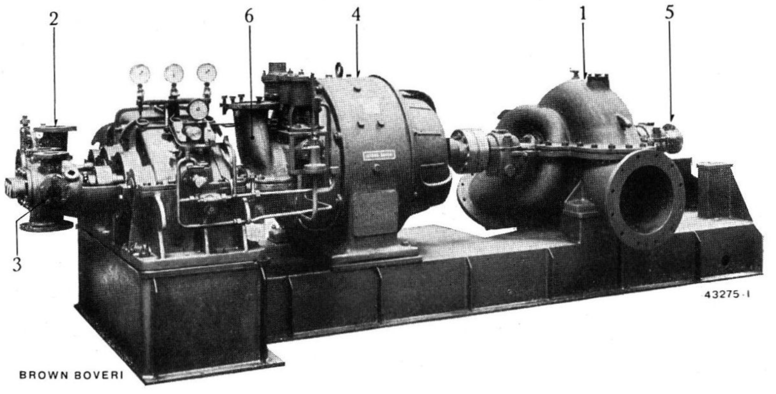


Fig. 5. — Pump set with double drive for a marine turbine installation.

The set comprises the circulating (1), condensate (2) and lubricating-oil (3) pumps, and a generator (4). It is coupled at (5) to an intermediate shaft of the main gearing through a hydraulic coupling and when the vessel is at sea is driven by the main turbine. When entering or leaving port the set is uncoupled and the auxiliary turbine (6) takes over the drive.

The drive of the auxiliaries from the main turbine results in minimum steam consumption.

From a thermal efficiency point of view a better arrangement is to drive all of the condensation pumps, together with the lubricating-oil pump and the generator, from the main turbine[1] when at sea, the various machines being, for instance, hydraulically coupled to one of the intermediate shafts of the main gearing. Two French banana-carrying vessels equipped with Brown Boveri turbine drive in 1935 have the pumps and the auxiliary generator driven in the foregoing manner (Fig. 5).

The electrical transmission of the necessary power for the auxiliaries from the main turbine, however, affords still greater advantages. To this end, a so-termed "Helux" generator is driven from the main propelling machinery or from the propeller shaft and supplies the driving motor of the pump set. The generator is preferably designed large enough also to cover the requirements of the ship's auxiliary power system with the vessel under way. The advantage of electrical transmission over the hydraulic system is that the pump set is not tied to the main turbine and can be located in a position giving the most favourable layout as regards piping and space requirements[2].

Where the pumps are driven from the main turbine — and this applies to both hydraulic and electrical transmission — a further, auxiliary turbine must be provided for their drive during manœuvres and when leaving or entering port. Upon the speed of the main drive falling below approximately 60 % of the normal revolutions steam is admitted to the auxiliary turbine which takes over the drive of the pump set. (Up to this point the auxiliary turbine revolves with the main drive, but without delivering power.) At the same time the hydraulic coupling is disconnected or, as the case may be, the electrical connection to the Helux generator interrupted. The reverse cycle of operations takes place when the pump set is changed over for drive by the main turbine at sea.

In order further to enhance the economy of modern turbine-driven ships feed-water preheating with steam bled from the main turbine has been resorted to. This measure, however, is only feasible and economically justified when no exhaust steam is available from the auxiliaries, i. e., in the case of pure electrical drive. With such installations the electrical energy is generated by a Helux machine at sea and by a Diesel- or turbine-driven generator in harbour and when entering or leaving port. In the case of passenger vessels the total power required by the auxiliaries is so high that separate generating sets with condensing turbines prove an economical proposition. Where the auxiliaries are electrically driven, steam-driven standby sets must also be provided for particularly important duties, e.g., for feed-water, bilge, and fire-fighting pumps.

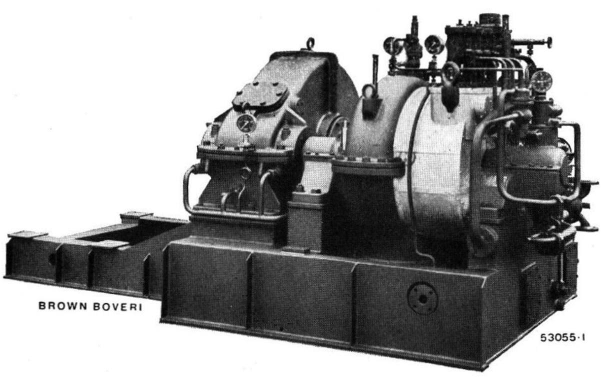


Fig. 6. — 500 kW geared turbine for utilizing the steam generated in an exhaust-gas boiler on a motor-ship.

This turbine, operated with steam generated in an auxiliary boiler by the heat inherent in the exhaust gases of the main Diesel engine, drives a d. c. generator supplying the ship's auxiliary power system. The pressure of the dry-saturated steam is 8 kg/cm² and the exhaust-steam pressure 0·1 kg/cm² abs; speed 10,200/1200 r. p. m.

The application to the main Diesel engine of exhaust-gas boilers supplying turbo-generators results in maximum economy.

[1] The first step in this direction was taken by Brown Boveri in 1922, on the paddle-tug "Dordrecht". The circulating pump on this turbine-driven ship was coupled to one of the intermediate gear-shafts. The radially-arranged pump blades also maintained the water circulation at a sufficiently high rate during astern running.

[2] C. G. Wahl: "Brown Boveri Turbines for the Propulsion of Merchant Vessels", page 227 of the present number of this journal and E. Klingelfuss: "Modern Steam Turbine Drives for Ships of Low and Medium Power", The Brown Boveri Review 1936, page 271.

On motor-ships the heat inherent in the exhaust gases is frequently utilized in special steam boilers, the resultant low-pressure steam being employed not only for the preparation of fresh water and for culinary and heating purposes, but even also for the generation of

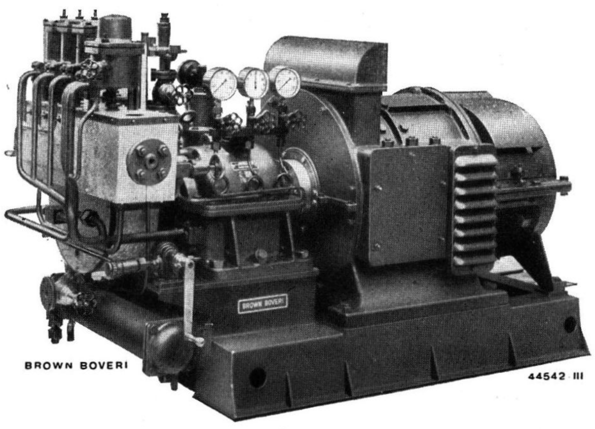


Fig. 7. — High-pressure turbo-generator for a marine installation.

Rating 100 kW, 12,000/4000 r. p. m.

Live-steam conditions 70 kg/cm² abs, 450° C, back-pressure 1·2 kg/cm² abs.

The generator has only one bearing and is driven through gearing by the turbine. The impulse wheel of the latter is overhung on the pinion shaft. The small dimensions of the high-speed turbine make it especially suitable for high-pressure steam with a high degree of superheat. In order also to achieve low steam consumption figures on low loads, such turbines are equipped with a large number of automatic nozzle valves to regulate the steam admission.

Example of a particularly space-saving arrangement.

electrical energy. The turbo-set illustrated in Fig. 6 is intended for the latter application. In the case of two-stroke Diesel engines, up to 25 kW per 1000 S.H.P. can be recuperated from the exhaust gases and delivered to the auxiliary power system. For four-stroke engines the maximum recuperation is 35 kW per 1000 S.H.P. This represents an improvement of up to 5% in fuel consumption.

III. MINIMUM WEIGHT AND SPACE REQUIREMENTS.

The dimensions and weights of the auxiliaries can be substantially reduced by increasing the revolutions. As early as 1913 Brown Boveri employed gearing to reduce the turbine revolutions from 3600 to the speed of 750 r.p.m. suitable for 360 kW marine generators[1]. Inasmuch as we already disposed of gear designs of our own, this was a natural development and finally led to the arrangement of the gearing and steam turbine in a single block (Fig. 7). The turbine impeller is overhung on the gear pinion shaft and rotates in a casing freely supported on the lower part of the gear-case. This arrangement gives an extremely compact unit having the additional advantage of only requiring one single gland. The gear-case also serves as oil sump.

Vertical-shaft geared turbines (Fig. 8) can also be employed for the drive of the auxiliaries. This model permits a large amount of space to be economized.

IV. SIMPLE ATTENDANCE AND GOOD OVERHAUL FACILITIES.

Simple and clear construction, good accessibility, i. e., appropriate subdivision of casing, large assembly apertures, and hinged casing components (Fig. 8) facilitate the attendance of the auxiliaries. Ease of dismantling, suitable tools, and detailed working instructions permit men having had no particular training in the care of machinery to maintain their reliability.

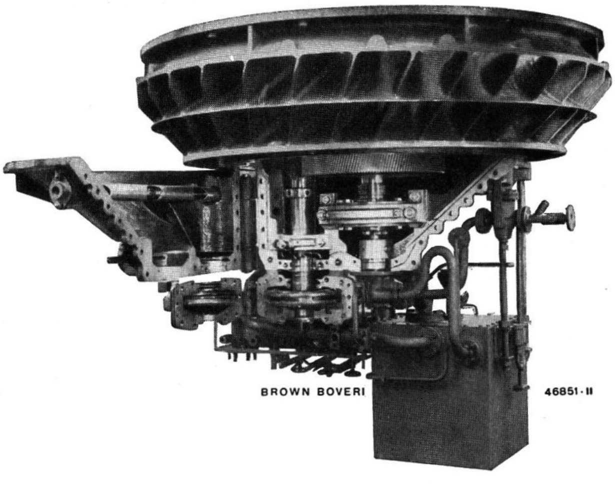


Fig. 8. — Vertical-shaft boiler-room fans for marine installations.

Rate of air 150,000 m³/h, total static head 240 mm w. g., speed 10,300/1880 r. p. m. The vertical arrangement with high-speed axial-flow fan and geared turbine gives a lightweight set with small space requirements and excellent air inlet and outlet conditions.

The great variety of marine auxiliaries built by Brown Boveri will be evident from an earlier number of this journal[2]. In all these developments the firm has always striven to achieve the arrangement affording the highest degree of reliability and economy commensurate with minimum space requirements and weight.

(MS 889) *E. Klingelfuss. / R. Schmid. (E. G. W.)*

[1] In the case referred to twenty-four such geared turbines were supplied for turbo-generators on Russian "Borodino" class cruisers. See BBC Mitteilungen 1920, p. 250, Fig. 12.

[2] See The Brown Boveri Review 1937, p. 69—71.

ELECTRICAL AUXILIARY MACHINES AND APPARATUS FOR SHIPS.

Decimal Index 629.12.066

Due to the excellent experience made with the electrical drive of marine auxiliaries, wide use is now made of this form of drive not only on motor-ships, but also on steam-propelled vessels. Brown Boveri have been building electrical machines and apparatus to satisfy the special requirements of marine operation for more than forty years. Hereafter a number of these are briefly described.

I. D.C. OR A.C. FOR MARINE AUXILIARIES?

WHEREAS until a few years ago ship's auxiliary power systems were chiefly operated with direct current, increasing use has recently also been made of alternating current, particularly for ships with Diesel or turbo-electric three-phase propulsion where the main alternators supply both the propeller motors and the auxiliary power system. A. C. installations have undeniable advantages over d. c. plants, the most important being decreased weight, smaller space requirements and lower cost of electrical equipment, more robust machines, simpler control gear, less attendance, fewer spare parts, and — due to absence of commutators — diminished maintenance costs. The question of the suitability of a. c. for the drive of marine auxiliaries has been exhaustively discussed in the technical press.[1]

In its application to the drive of marine auxiliaries, however, alternating current has certain drawbacks. For instance, a. c. motors are less adaptable to the speed of the machines to be driven and their revolutions cannot be so readily varied as with direct current. There are a number of auxiliaries, however, which do not absolutely require a variable speed, while in other cases the delivery or pressure of the auxiliaries (fans and pumps) to be driven can be adapted to the working conditions by means of throttle valves. For drives where a certain amount of speed variation is indispensable pole-changing squirrel-cage motors or, in certain cases, high-speed commutator motors can be employed.

The design of a. c. installations involves close study of the mutual operation of alternators and motors. Inasmuch as the total power available on ships is small in comparison to the individual power requirements, it is imperative, when selecting motors and their method of starting, to consider their influence on the supply voltage and, vice versa, the effect of the latter on the starting process. The alternator, therefore, cannot be designed simply for the maximum loads to be expected, as in the case of machines for large land networks, but must be built with low inherent regulation and an ample margin for power factor. The excitation is preferably controlled by an automatic quick-acting regulator designed so as to rapidly smooth out fluctuations in voltage caused by sudden load alterations. By reason of its momentary over-regulating characteristic the Brown Boveri quick-acting regulator is eminently suitable for this application. An essential condition for the correct functioning of the regulator is liberal design of the alternator field winding and the exciter.

Alternating current is definitely finding wider and wider application for auxiliaries on large ships, but small vessels will probably continue to be provided with direct current for auxiliary purposes. Brown Boveri supply approved machines and apparatus for both current systems.

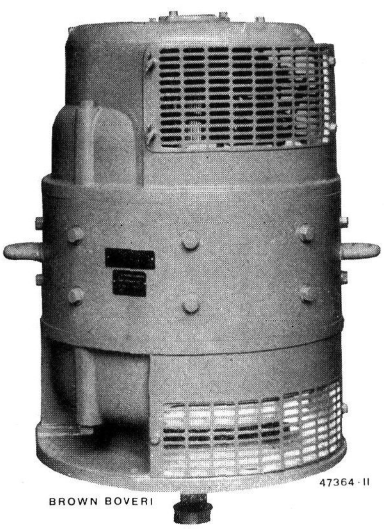
II. GENERAL CONSTRUCTION OF GENERATORS AND MOTORS.

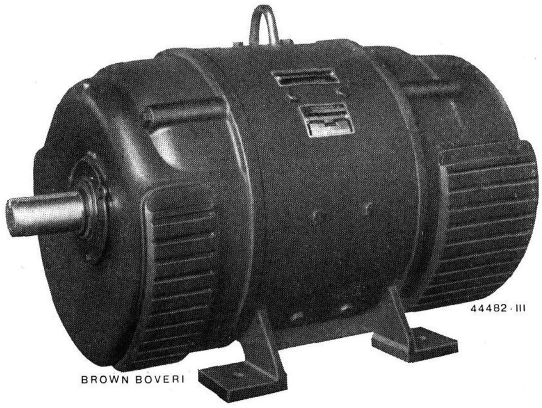
Varying degrees of protection are necessary for safeguarding marine machines according to their location on the ship. Drip-proof protection suffices for the majority of applications. Machines which are liable to be splashed, however, should be of the splash- or hose-proof type. Large machines for operation in rooms at a particularly high temperature or in locations where the air is saturated with oil vapour are either connected to fresh air ducts or cooled on the closed-circuit principle. Where conditions entail total enclosure of small machines the surface-cooled model is preferable. Installation on deck, e. g., the drive of deck winches, involves watertight machines and apparatus to meet the danger of waves breaking over the deck. Such watertight deck drives are usually combined with the machine to be driven to form a self-contained unit.

Small and medium-size machines have end-shield bearings throughout, these normally being of the greased-lubricated ball type, only needing inspection

[1] Forschungsheft 12 (April 1942) der schiffbautechnischen Gesellschaft, Berlin.
Electrical Engineering (U. S. A.), July, 1940.
The Motor Ship, April, 1942.

Various patterns of d. c. marine machines.





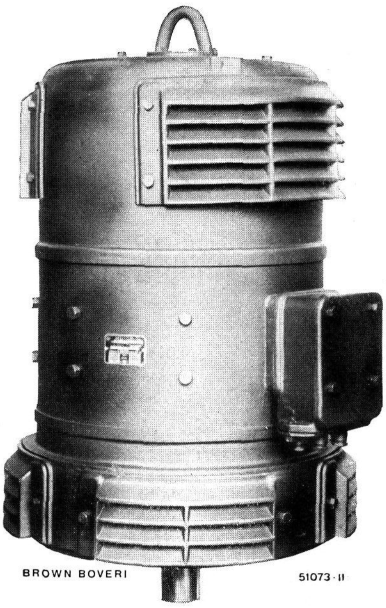


Fig. 1. — Drip-proof.

Fig. 3. — Hose-proof.

Fig. 2. — Splash-proof.

The robust and simple design of our machines and apparatus also ensures full reliability in inclined positions and under conditions of hull vibration. Sturdy non-hygroscopic insulation safeguards against the noxious effects of damp, salt-laden sea air and oily vapours. Important components subject to wear can be readily supervised and replaced.

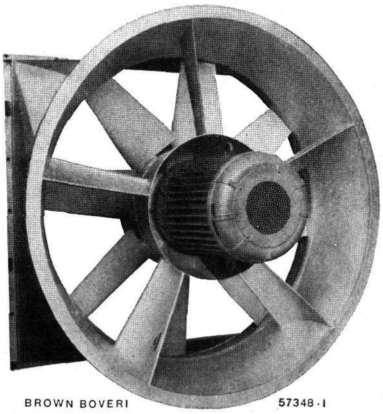


Fig. 4. — Three-phase surface-cooled totally-enclosed squirrel-cage motor driving Sulzer propeller-type fan.

The motor fan, mounted on the shaft extension at the non-driving end under a cover, forces air over the outside of the stator which is ribbed to increase the cooling surface.

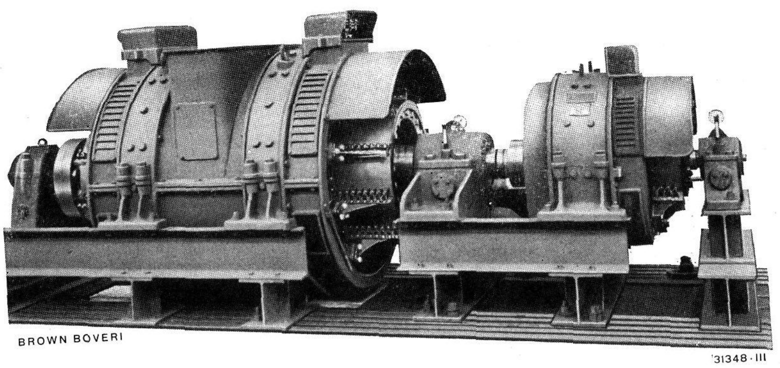


Fig. 5. — High-powered Diesel-driven d. c. generator coupled to auxiliary generator.

Design with split casing, canopy over the commutator, and forced oil-lubricated plain bearings.

and re-packing with grease once annually. Motors out of commission at sea and therefore continually subject to vibration while at rest, e. g., motors driving Diesel-engine barring gear, generally have plain bearings.

As will be seen from Fig. 1 drip-proof machines have all ventilating apertures covered with perforated sheet metal or wire screens. In the case of the splash-proof pattern shown in Fig. 2 these screens are replaced by louvred covers which prevent ingress of jets of water falling on the machine at any angle from the perpendicular to the horizontal. The hose-proof model depicted in Fig. 3 is provided with baffle-type covers over the ventilating apertures which protect the interior of the motor against powerful jets of water from any direction. With the totally-enclosed machine illustrated in Fig. 4 the interior of the motor is completely shut off from the atmosphere. A fan mounted on the shaft extension of the motor and enclosed by a cast-iron cover blows air over the motor casing and thus accelerates the rate of heat dissipation. Large machines have pedestal bearings and, to facilitate mounting and demounting, split casings (Fig. 5).

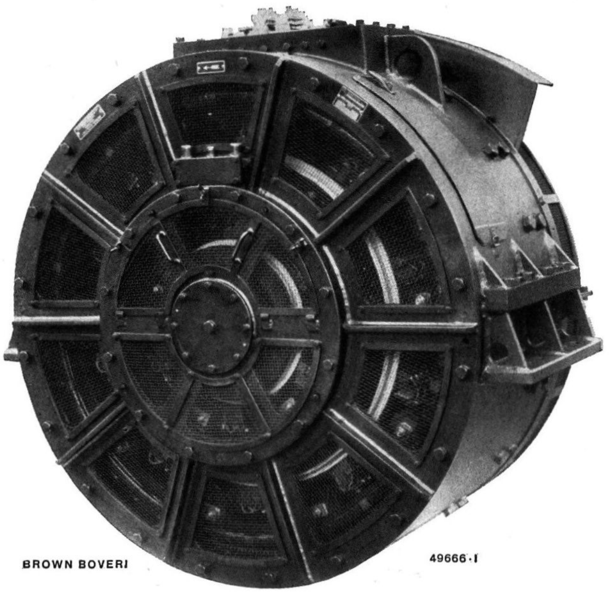


Fig. 6. — Compact, lightweight Diesel-driven d. c. generator with auxiliary generator embodied in main commutator.
Notwithstanding the compact design, commutators and brush gear are readily accessible after removal of the cover.
The Diesel-generator set has a much shorter overall length than separate main and auxiliary generators.

III. MACHINES OF SPECIAL DESIGN.

Warship installations involve lightweight, compact machines. Fig. 6 depicts a d. c. generator with built-in auxiliary generator for Diesel drive. This arrangement enables generating sets with a very short overall length to be obtained. The stator of the auxiliary generator is welded to the end shield of the main generator and embodied in the commutator of the latter. The armatures of the two machines are mounted on a common cast-steel hollow shaft coupled rigidly to the Diesel engine at one end and running in a grease-lubricated roller bearing at the other. In the case of alternators the exciter and main machine are combined to obtain a short overall length.

A further special design worthy of mention is the d. c. turbo-machine employed as generator for drive by steam turbine and as motor chiefly for the drive of scavenging and diving-tank blowers (Figs. 7 and 8). Brown Boveri took up the manufacture of these machines more than forty years ago and in the interim they have been perfected by taking account of operating experience and of the progress made in the metallurgical field. Turbo-machines are now available with ratings of 60—750 kW at speeds of approximately 6000—2500 r. p. m. The strikingly robust and compact design ensures quiet, vibrationless running at the high speeds. The armature winding is specially arranged to give uniform current distribution and excellent heat dissipation properties. Since, for mechanical reasons, the commutator of turbo-machines must be kept as short as possible, and this again entails optimum heat dissipation, very effective ventilation of the commutator surface is provided for. At the armature end the commutator segments are rigidly shrunk on to the shaft or commutator bush, while at the opposite end they are supported by a flexible membrane which permits of their axial expansion and thus prevents the commutator running out of truth when the temperature of the commutator segments rises.

Among the special machines developed are also small monobloc convertor sets which are frequently used for transmitting stations (Fig. 9). Characteristic features of these sets are the bolting together of the stators of the two machines to form a single frame and the common shaft supported in two end shields. Bed-plates are unnecessary, inasmuch as the self-contained sets can be fixed directly on the framework of the ship.

IV. SWITCHGEAR.

Standard marine switch cubicles and switchboards are drip-proof and enclosed on all sides to give maximum protection against accidental contact (Fig. 10). On account of the vibration to which it is subjected particular care is paid to the wiring.

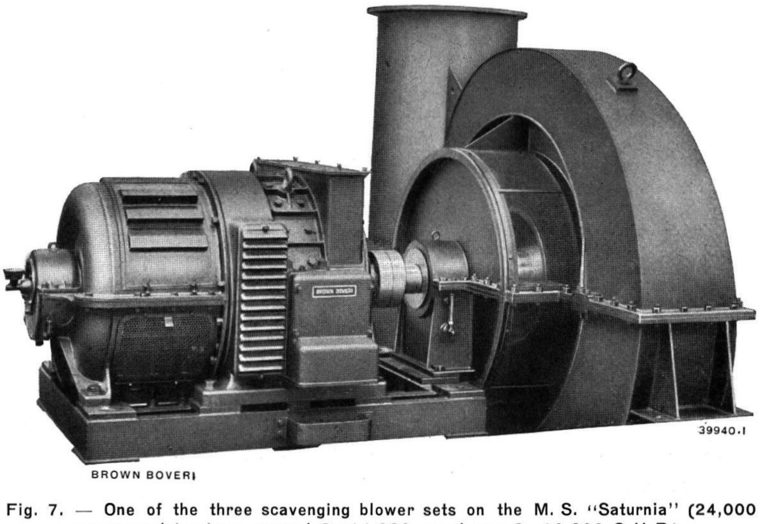


Fig. 7. — One of the three scavenging blower sets on the M. S. "Saturnia" (24,000 gross register tons, normal 2×14,000, maximum 2×20,800 S. H. P.)
The driving motor of this set is the largest d. c. turbo-motor ever built (770 kW at 220 V and 2000—2470 r.p.m.). The motors of scavenging blower sets are connected to fresh-air ducts when installed in Diesel engine rooms.

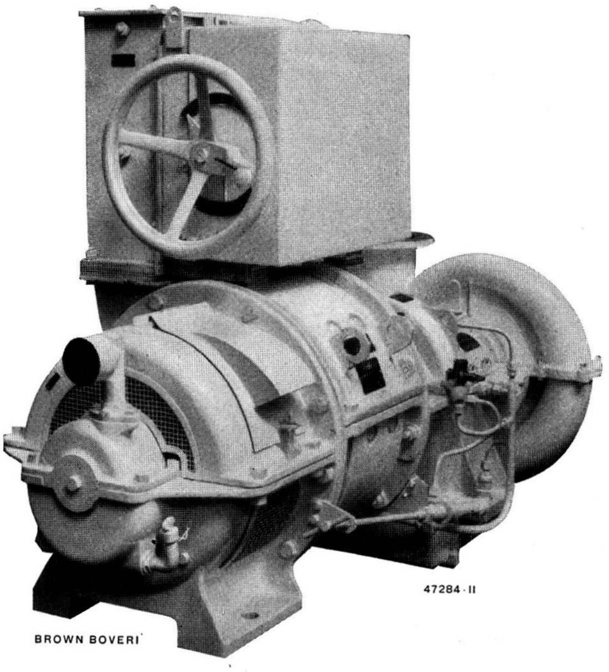


Fig. 8. — Diving-tank blower set.

The turbo-motor drives a high-speed blower through gearing, the impeller being overhung on the pinion shaft. Motor, gearing, and blower are combined into a compact set. To save space the starter is mounted directly on the motor. These sets are supplied for blower speeds up to 35,000 r. p. m. and delivery volumes up to 60 m³/min. They are 30—50 % lighter in weight than direct-driven sets and also require less space.

In the case of a. c. installations the apparatus we have developed for land applications can also be employed if suitably adapted to the conditions obtaining on board ship.

For d. c. drives we have evolved a special marine type of starter. This is hose-proof and designed for bulkhead mounting. Switching components and tripping gear are clearly arranged in the front of the case so that upon opening the door all of the more important parts can be readily inspected and replaced if necessary. The motor is disconnected on both poles by a contactor with thermal over-current release fitted in the starter casing. Special design features prevent the starter being left on the starting steps. Should the motor be inadvertently cut out on the starting steps the contactor takes over disconnection, thus completely obviating excessive burning of the starting contacts.

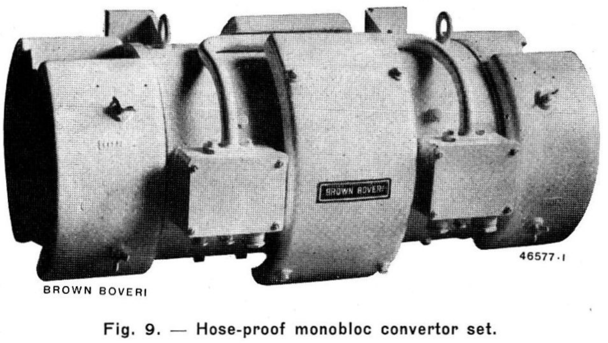


Fig. 9. — Hose-proof monobloc convertor set.

Compact, lightweight pattern without bed-plate for mounting directly on the ship's framework.

For small d. c. motors face-plate starters (Figs. 11 and 12) are employed, whereas in the case of large motors the starters are of the cam-operated type (Fig. 13). Should speed variation through field reduction be required the starters are provided with a number of field regulating steps and the necessary regulating resistors.

Apart from these starting apparatus a series of marine-type reversing starters with series regulation

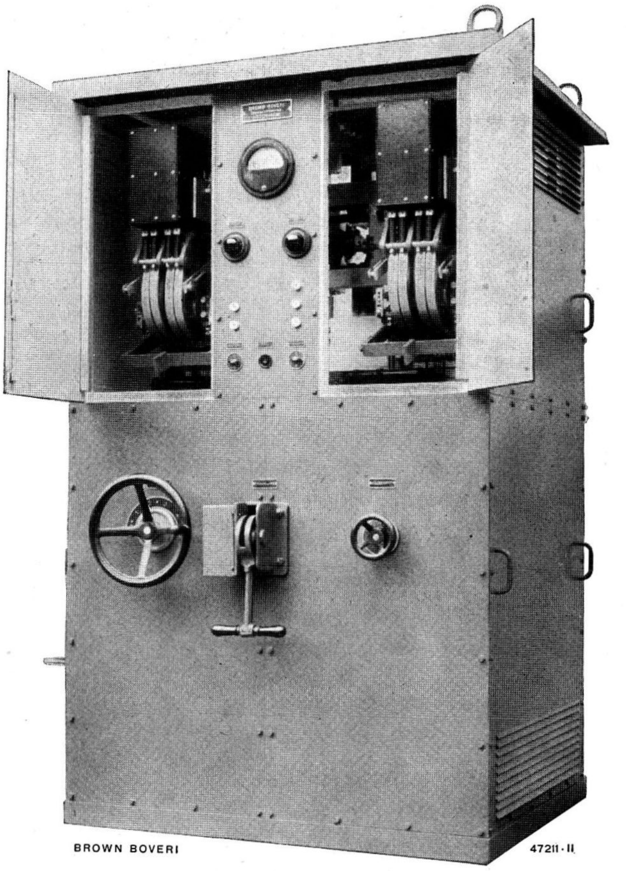


Fig. 10. — Drip-proof switch cubicle for the d. c. motor of a scavenging blower set, enclosed on all sides.

All apparatus and instruments required for the starting, speed regulation, protection, and supervision of the motor are incorporated in this cubicle.

have been developed particularly for Diesel-engine barring-gear drives. By reason of the high frequency of operation involved with such drives these starters are of the cam-operated type.

Certain conditions entail automatic or lightweight and compact starters. In the first case pure contactor-type starters are employed. A compact, lightweight starter, used in particular for submarine auxiliaries,

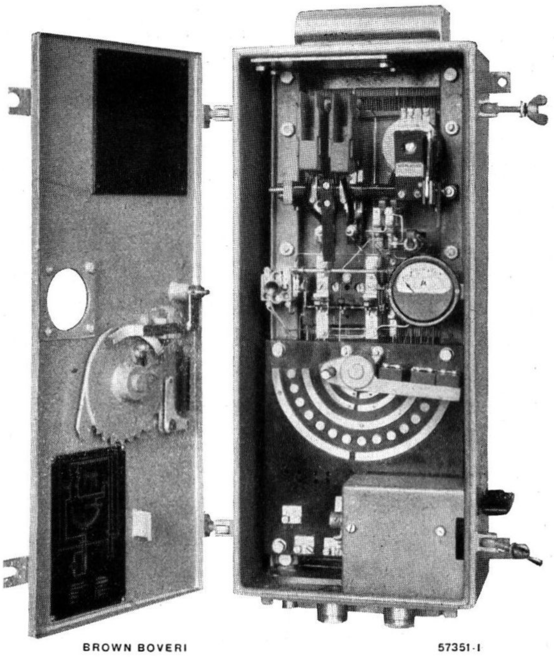


Fig. 11.

Fig. 12.

Marine-type face-plate starter, hose-proof pattern, for current ratings up to 75 A d. c., with isolating switch in separate casing in bottom right-hand corner, and over-current and low-voltage releases.

is shown in Fig. 14, this being provided solely with off and on positions. Upon the handwheel being rotated into the on position, the motor, which is provided with a heavy series starting winding, is switched on to the supply through a resistor, the starting winding and resistor being automatically short-circuited after a certain elapse of time. The special starting winding and the limiting of the starting time to an absolute minimum enable a very small resistor to be employed. Somewhat larger switching current surges occur, however, than with the standard multi-step starters.

In order to reduce the number of starters on ships and thus simplify the installation, there is at present a tendency to provide a single starter for a whole series

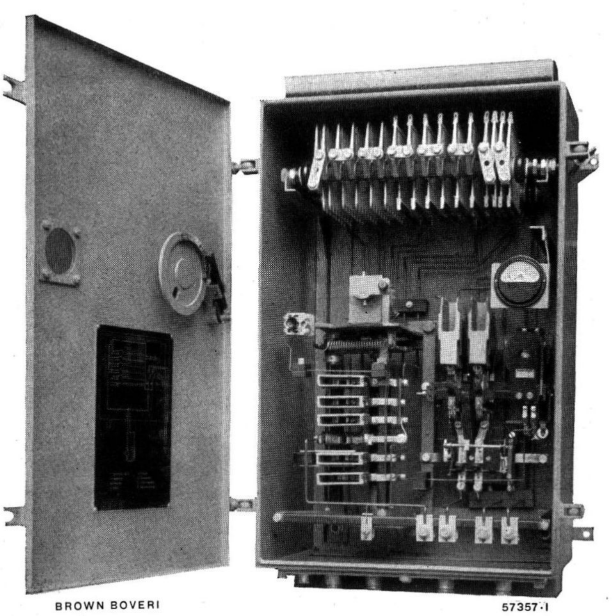


Fig. 13. — Marine-type d. c. starter with cam-operated switch units and over-current and low-voltage releases; current ratings up to 400 A.

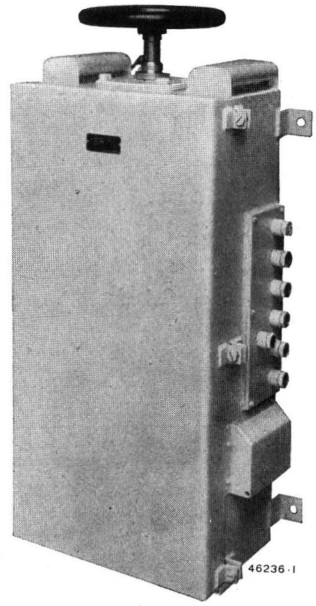


Fig. 14. — Compact, lightweight d. c. starter with automatic starting step for submarine auxiliary drives. The motor is started simply by rotating the handwheel from the off to the on position.

All d. c. marine starters can be supplied with or without speed regulating steps (field regulation) and with or without isolating switch. They fully protect both the operator and the motor, and render faulty operation impossible.

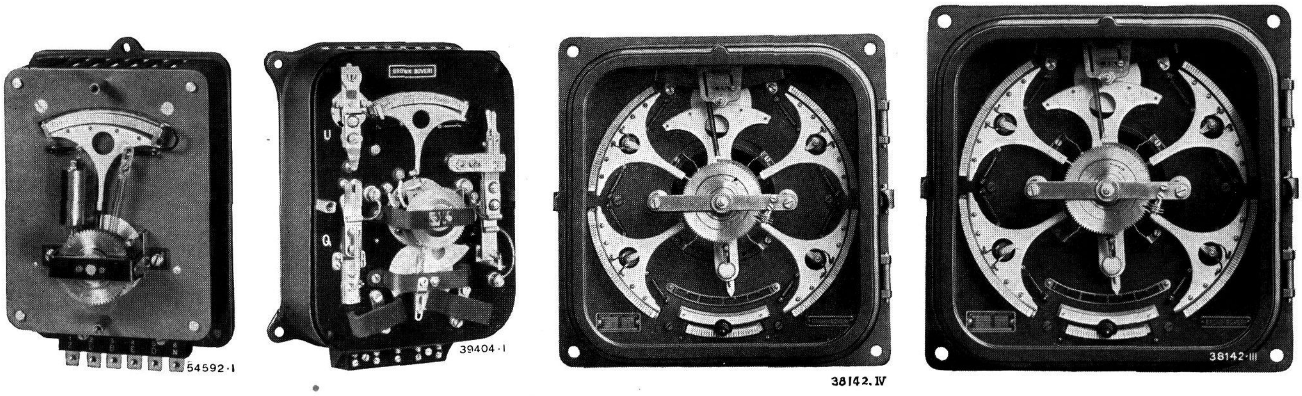


Fig. 15. — Automatic voltage regulators for d. c. and a. c.

A suitable quick-acting regulator is available for all generator outputs encountered in marine practice. To date over 800 such voltage regulators have been supplied for ships of all categories.

of motors, this being switched on to the motor to be started when required.

V. QUICK-ACTING REGULATORS AND AUTOMATIC SYNCHRONIZERS.

The well-known Brown Boveri quick-acting regulators have also become an important auxiliary apparatus on board ship for the automatic regulation of the voltage of d. c. and a. c generators and the correct distribution of the load between the individual generating sets. Compared to the compound-wound generators hitherto generally provided for d. c. marine installations shunt-wound generators with automatic voltage regulators have the advantage that the equalizing conductor between the individual generators can be dispensed with and two-pole automatic circuit-breakers employed for their protection. Details of an interesting application of our quick-acting regulators in conjunction with parallel-operating Helux- and turbo-generators on a motor-tanker with exhaust-gas boiler, were given on page 50 of the Brown Boveri Review for 1939. Quick-acting regulators are also particularly suitable for generators driven from the propeller shaft at variable speed. In the case of three-phase alternators, however, the voltage must be varied approximately in proportion to the frequency to avoid overloading the auxiliary motors. This condition can be readily met with specially connected Brown Boveri regulators. Such regulators have already been supplied for three-phase shaft-driven alternators with ratings up to 650 kVA and a frequency range of 53—40 cycles. By reason of their reliability and high degree of accuracy under conditions of vibration and in inclined positions quick-acting regulators with rolling contacts are finding wider and wider application for ships of all classes.

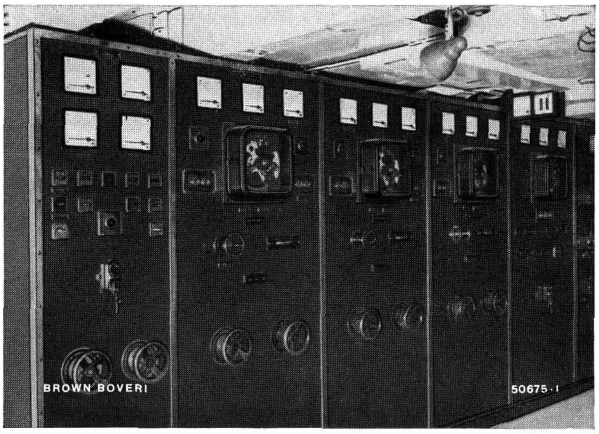


Fig. 16. — Switchboard in three-phase auxiliary power station on M. S. "Oranje" with three Brown Boveri automatic voltage regulators and one Brown Boveri synchronizer.

The automatic synchronizer renders paralleling of three-phase alternators simple.

Of the regulators illustrated in Fig. 15 the smaller apparatus is chiefly employed for the Brown Boveri Helux ship's lighting system[1]. Installations of this type are suitable for small vessels where the electrical energy is chiefly only required for lighting purposes and where, on account of the low demand, a separate lighting set would prove too costly and uneconomical. Such equipments comprise a d. c. generator which, being driven from the propeller shaft, i. e., by the propelling machine, operates at variable speed, a battery to provide the supply when the vessel is stopped or running at very slow speeds, and a Helux regulator taking care of the automatic cutting in and out of the generator, regulation of the voltage, and charging of the battery.

Apart from the above-described automatic voltage regulators, *automatic current-regulators* and *synchronizers* (Fig. 16) are also available. The regulators limit or regulate the current of special drives, and in particular of certain ship's drives, while the synchronizers render paralleling of generators just as easy in a. c. plants as in d. c.

(MS 907) *Th. Geiger. (E. G. W.)*

[1] The Brown Boveri Review 1933, pp. 84—86.

THE CHARGING OF FOUR-STROKE-CYCLE MARINE DIESEL ENGINES.

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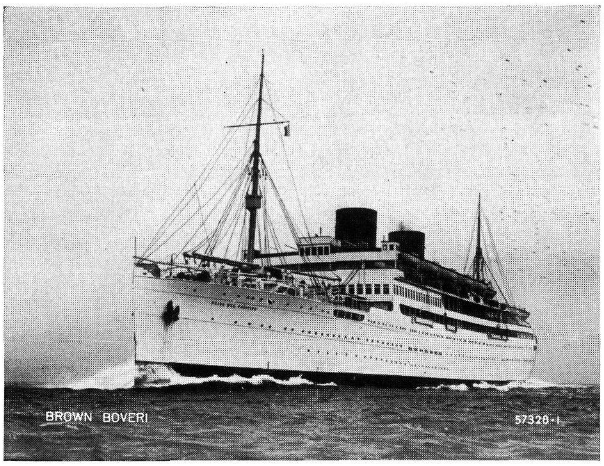

Fig. 1. — M. S. "Reina del Pacifico" owned by the Pacific Steam Navigation Company, Liverpool.

Built by Messrs. Harland & Wolff, Belfast, and propelled by four Burmeister & Wain marine Diesel engines exhaust turbo-charged to 5500 B. H. P. each. The aggregate shaft rating of 22,000 B. H. P. gives the vessel, which operates between England and the West Coast of South America, a mean speed of 20 knots.

The employment of exhaust turbo-charging enables a practically 50 % higher power to be lodged in the same engine room.

Exhaust turbo-chargers permit the power of four-stroke-cycle engines to be increased by 50 % or more. The power-weight ratio of exhaust turbo-charged engines is up to 30 % greater and space requirements correspondingly less than with ordinary engines of the same power. Exhaust turbo-charging gives engines a degree of flexibility otherwise unattainable. By reason of their greater simplicity, reliability, and economy, exhaust turbo-chargers have a brighter future than mechanically-driven chargers. Our present models are the fruit of nearly twenty years' experience.

EXHAUST turbo-chargers for increasing the power of four-stroke-cycle marine Diesel engines on the Büchi principle have found wide and rapid application. As early as 1928, i. e., immediately after conclusion of the fundamental tests on stationary engines[1], a whole series of English and Italian passenger and cargo vessels with propulsion engines exhaust turbo-charged by the Büchi process was put into commission. By 1930 more than fifty exhaust turbo-charged four-stroke-cycle marine engines of all sizes with an aggregate power of over 140,000 B. H. P. were already in service. Fig. 1 shows the M. S. "Reina del Pacifico" of the Pacific Steam Navigation Co., Liverpool, launched in 1930. This vessel with its four 5500 B. H. P. exhaust turbo-charged Diesel engines was one of the most imposing and fastest motor-ships of the period and is proof of the trust placed in the then relatively new process by the shipowners.

As experience has shown, this confidence on the part of the company in question was fully justified. Not only are all of the original installations still in service — which as manufacturers of the appertaining exhaust turbo-chargers is particularly gratifying to us — but the number of our customers has also steadily grown until to-day more than 800,000 B. H. P. of exhaust turbo-charged engines are in operation on sea-going vessels alone, the mean increase in power over the ordinary engine being of the order of 45—50 %. Fig. 2 shows the development of sales of Brown Boveri exhaust turbo-chargers up till October, 1942.

Sundry advantages accruing from exhaust turbo-charging have contributed to this success. The fitting of an exhaust turbo-charger involves an increase of

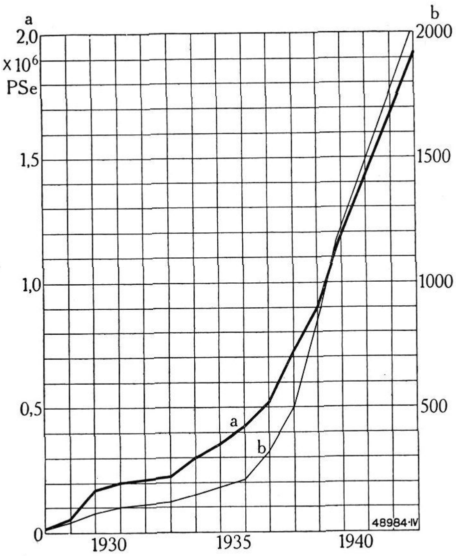

Fig. 2. — The development of exhaust turbo-chargers for Diesel engines (Büchi process).

a = Total power of charged Diesel engines in B. H. P.
b = Total number of charged Diesel engines.

The total power of exhaust turbo-charged engines on ships of all kinds considerably exceeds the aggregate rating of such machines for other applications.

[1] Stodola, "Leistungsversuche an einem Dieselmotor mit Büchi'scher Aufladung", Z. VDI 1928, p. 421.

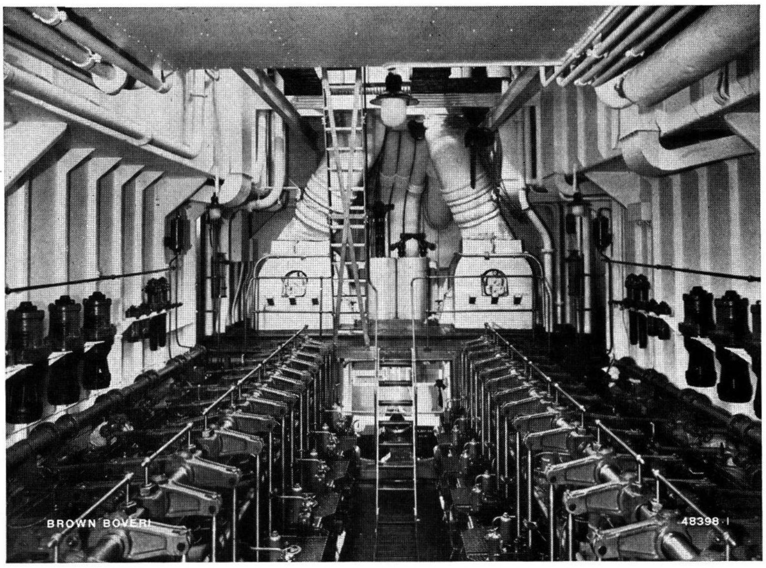

Fig. 3. — **Engine room of the M. S. "Don Isidro" owned by the Hijos de J. De La Rama Steam Ship Co., Iloilo (Philippine Islands).**

The two nine-cylinder four-stroke-cycle Diesel engines built by the Friedr. Krupp Germaniawerft A.-G., Kiel, are each charged to 3500 B. H. P. at 235 r. p. m. on the Büchi principle with Brown Boveri exhaust turbo-chargers.
In the background will be seen the exhaust turbo-chargers separately mounted on a platform.

The space requirements of exhaust turbo-charged Diesel engines are only about 70 % of ordinary engines of the same power.

only 2—3 % in the overall weight of the engine, whereas the augmentation in power is of the order of 50 %, thus making the power-weight ratio about 30 % greater than for a similar ordinary engine. Exhaust turbo-chargers, therefore, also enable a considerable amount of material to be saved.

By exhaust turbo-charging the engine, space requirements are also reduced. With the charging set fitted at one end an exhaust turbo-charged engine is 20—30 % shorter than an ordinary engine of the same power, height, and width. In the case of slow-speed marine engines the charger can be installed separately in space not otherwise utilized (Fig. 3).

The exhaust turbo-charger gives the engine a degree of flexibility otherwise unattainable. Being regulated solely by the quantity and temperature of the exhaust gases its revolutions are greater the higher the load on the engine. In consequence, the quantity and pressure of the charging air are correctly proportioned at all loads and the engine can be overloaded to a high degree. Fig. 4 shows the characteristic operating curves of a marine Diesel engine exhaust turbo-charged on the Büchi principle.[1]

[1] For description of Büchi process see E. Klingelfuss: "Increasing the Power of Diesel Engines and Aeroplane Engines by the Büchi Process." The Brown Boveri Review 1937, p. 175.

Fuel consumption is in general lower than with ordinary engines. This is explained by the negligible increase in mechanical losses (notwithstanding the higher power), the favourable effect of the scavenging of the combustion chamber on the combustion process, and the fact that the charger requires no engine power to drive it. Another important point is that, even at low load, the fuel consumption is much less than in ordinary installations by reason of the high degree to which the quantity and pressure of the charging air is adapted to actual load conditions. This will be clear from the typical diagram in Fig. 4.

The completely flexible drive of the charger solely through the exhaust gases of the engine constitutes the simplest, most reliable, and most economical method conceivable. Mechanical driving gear, which in the case of centrifugal chargers in particular presents such difficulties, is unnecessary, and the power required for the drive of the exhaust turbo-charger (which may attain 12 % of the Diesel engine rating) is furnished without detracting from the power of the engine.

It was maintained from the very beginning that exhaust turbo-charged and scavenged engines would be subject to no greater thermal and mechanical stress than ordinary engines, and this theory has been fully corroborated by practical experience. Customers

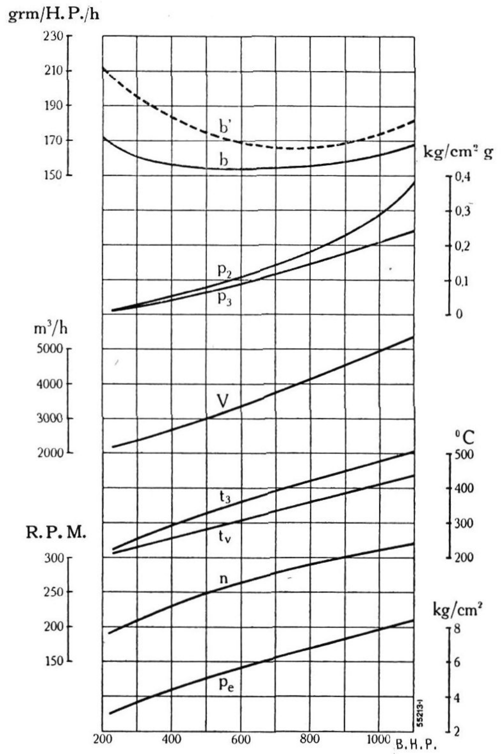

Fig. 4. — Official acceptance tests on a Swiss Locomotive and Machine Works marine Diesel engine by Lloyd's Register of Shipping.

Abscissa: Power in B.H.P.
Ordinates:
b = Fuel consumption with exhaust turbo-charging.
b_1 = Mean fuel consumption of various ordinary Diesel engines.
n = Engine speed.
p_1 = Pressure in suction pipe of blower = 0·98 kg/cm².
p_2 = Charging pressure.
p_3 = Pressure in exhaust piping to turbine.
p_e = Mean effective working pressure in Diesel engine cylinder in kg/cm².
t_3 = Temperature in exhaust piping to turbine in °C.
t_v = Exhaust temperature after valves in °C.
V = Quantity of air aspired.

The full-load engine speed is 300 r.p.m. The small fuel consumption at full and low load is particularly striking.

The dotted curve shows the mean fuel consumption of various ordinary Diesel engines of different make, but of the same power.

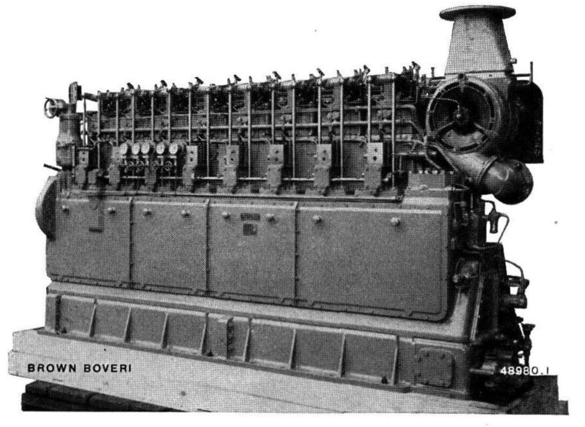

Fig. 5. — Sulzer eight-cylinder four-stroke-cycle marine Diesel engine with Brown Boveri exhaust turbo-charger.

Six such engines were supplied for the Danube ships "Stadt Wien", "Stadt Passau", and "Traun" of the Erste Donau-Dampfschiffahrtsgesellschaft. The exhaust turbo-charged engine power is 460 B.H.P. at 500 r.p.m. The same engine would only develop 320 B.H.P. non-supercharged.

The exhaust turbo-charger is fitted on the engine over the auxiliaries, the width and height of the engine thereby remaining unaltered, while the overall length is only slightly increased.

report from lengthy observation that exhaust turbo-charged engines require not only no more, but even less inspection and upkeep than ordinary engines. In many installations it has been found possible to increase the intervals between overhauls with a consequent reduction in maintenance costs. The exhaust turbo-charger itself needs very little attention, inasmuch as it does not become dirty and, apart from the bearings which have to be replaced only after long periods of service, has no parts subject to wear.

It is only natural that the exhaust turbo-charger should have met with a certain amount of competition as the years went by. For instance, plants with electrically-driven chargers have been introduced, but, due to the large amount of power required to drive the charger, far from attain the efficiency of turbo-charged engines. In point of fact up to 15 % more fuel is required at full load and considerably more at low load.

Then there are engines which drive their chargers mechanically, the chief drawbacks of these being as follows:—

The power for the compression of the charging air has to be furnished by the Diesel engine itself, while the charging pressure remains practically constant throughout the entire range of loads. For these reasons the fuel consumption is considerably higher at both full and low load than with exhaust turbo-charged engines.

The system with which the underside of the engine piston is employed as air pump (Werkspoor) is restricted to cross-head, i. e., large, engines.

In the case of eccentric vane blowers and centrifugal fans, etc., the drive is generally difficult, inasmuch as on the one hand the blowers become large and heavy if a small transmission ratio is selected, while on the other a large transmission ratio involves particularly great care in the design of the drive. The use of such chargers is still further complicated when the main engines are required to be reversed.

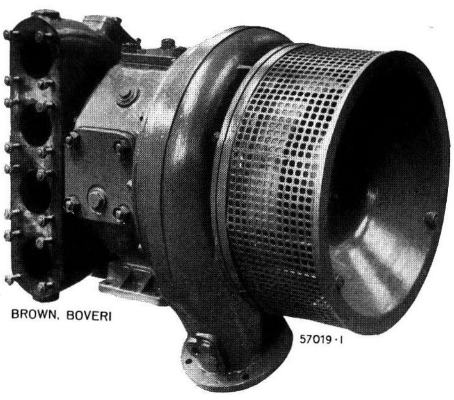

Fig. 6. — Brown Boveri exhaust turbo-charger, standard model.

The charger is intended for an eight-cylinder engine and has thus four separate turbine inlets which, together with the gas-inlet and turbine casings, are water-jacketed. The charger has a separate lubricating system.

Important new factors augur well for the future of exhaust turbo-charging.

So-termed high-capacity chargers permit the power of four-stroke-cycle Diesel engines to be increased by 80 % or more. This, however, entails close collaboration between the engine and charger builders, for such results can only be obtained when the charger is fitted on the engine in the most workmanlike manner possible and all gas- and air-conducting components are thoroughly adapted to one another.

Continual improvements in material and design have also enabled the specific load and, in consequence, the exhaust temperatures of two-stroke-cycle engines to be substantially increased. This re-introduces the problem of an exhaust-driven scavenging and charging pump for two-stroke-cycle Diesel engines. Promising tests recently carried out have proved the exhaust turbo-charger also to be the most natural and efficient charger for this type of engine. As a result, it may be expected soon to find regular application in this field, both as scavenging and charging pump proper and as booster to meet temporary load peaks.

The loss of power ensuing when Diesel engines are run on gas is common knowledge. By employing exhaust turbo-charging, however, it is not only possible to offset this inherent drop in output, but the power developed with liquid fuels can even be improved upon. Tests on small vehicle engines extending over a period of several years have resulted in power increases up to 60 % over non-supercharged gas operation being recorded.

The chargers are now mostly mounted directly on the engine, usually across one end (Fig. 5). Where this is not feasible they can be separately installed (Fig. 3). Our present models are standardized and form self-contained units requiring little attention. Ten different sizes, weighing from 40 to 2800 kg net per unit, are available. These are designed to give charged engine ratings of approximately 70 to 5000 B. H. P. A comprehensive stock of finished components ensures rapid delivery. Fig. 6 shows one of our standard turbo-charging sets. All sizes comprise three main components, viz., the water-cooled gas-inlet casing, the likewise water-cooled turbine casing, and the uncooled blower casing with suction branch or silencer. All types can be supplied with two, three, or four separate turbine inlets, as desired, so that the Büchi process with scavenging of the combustion chamber can be applied to engines with any of the usual number of cylinders.

Brown Boveri can look back on nearly twenty years' experience in the construction of exhaust turbo-charging sets. This has not been confined to the turbo-chargers alone, but also covers the study of the mutual operation and assembly, etc., of the chargers and Diesel engines.

(MS 908) *R. Stahel. (E. G. W.)*

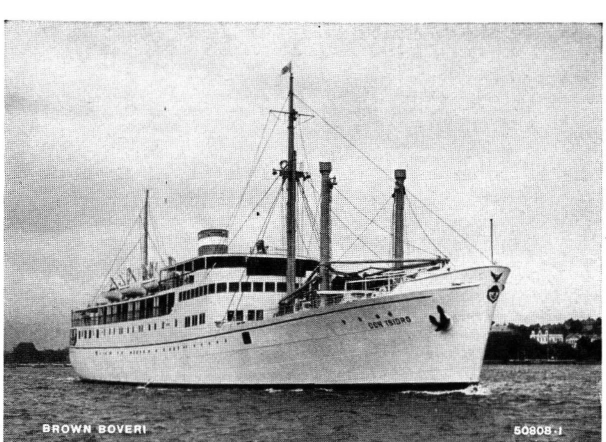

Fig. 7. — M. S. "Don Isidro" owned by the Hijos de J. De La Rama Steam Ship Co., Iloilo (Philippine Islands).

Two marine Diesel engines constructed by the Friedr. Krupp Germaniawerft A.-G., Kiel, and equipped with Brown Boveri exhaust turbo-chargers drive this vessel which was built for service between the various islands in the Philippine group.

With the approximately 40 % increase attained by exhaust turbo-charging the total rating is 7000 B. H. P.

Brown Boveri exhaust turbo-chargers are to be found *all over the world*.

WORKSHOP TESTS ON SHIP MACHINERY.

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In order to be able to test ship machinery already in the manufacturers' works under conditions corresponding as closely as possible to those specified by survey authorities, a whole series of checks, tests and manipulations are necessary, which are briefly described in the present article. For this purpose special equipment and facilities have to be available at the manufacturers' works. These are enumerated with special reference to the Velox boiler, which is particularly suitable for workshop tests on marine turbines.

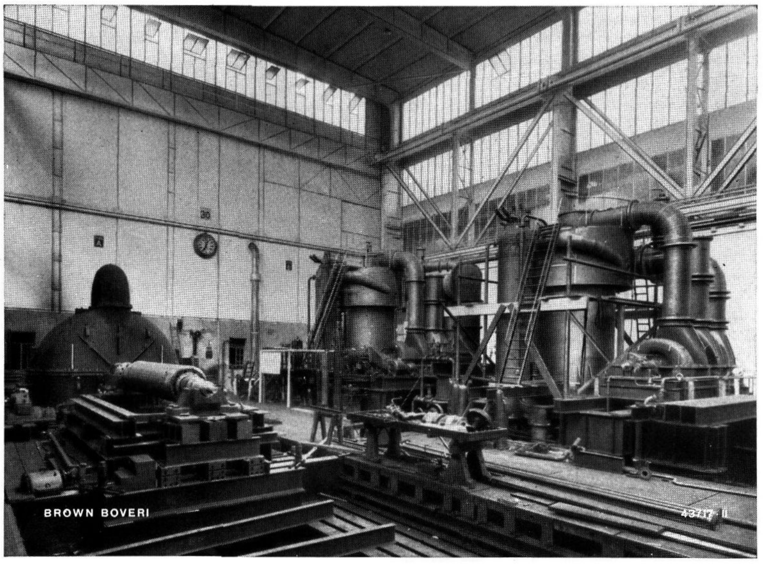


Fig. 1. — Large erection bay.

Well lighted testing and erection bays, built with much foresight, are available for carrying out tests on large objects. Three travelling cranes each capable of lifting a load of 75 tons and five cantilever cranes for loads of 8—15 tons facilitate the erection of the test articles in the roomy 16 m high bay.

Two Velox steam generators for the Oslo Electricity Works, each for an output of 75 t/h on the test beds. On the left, the 31,500 kW single-cylinder turbo-set for the same installation which was tested with the Velox boilers.

WHEREAS in the case of the works tests on steam turbines for land purposes it is generally considered sufficient, because of the more simple erection conditions of such plants, to set up only the principle parts, in the case of marine plants the greatest possible number of components are assembled together and tested, involving considerable expense. It is evident that every machine intended for marine purposes should undergo extensive and thorough testing under conditions approximating as closely as possible to those on board, before leaving the makers' works, because the correction of defects later on the ship is connected with considerable difficulties. Such complete test programmes can only be carried out in workshops where testing facilities have been planned with far-sighted vision as to size and variety of the objects to be dealt with. Already at the time of signing the contract, consideration should be given to the available testing facilities, and steps should be taken to acquire any auxiliary equipment necessary for the tests. The limiting factors for the testing of large marine turbines are:

1. Capacity and flexibility of the boiler plant.
2. Speed range and load capacity of the available water brakes.
3. Supply of cooling water available for the condenser and for the oil coolers.
4. Space and crane facilities on the test beds.

As shown by Figs. 1, 2, 3, we have constantly been extending our test plants

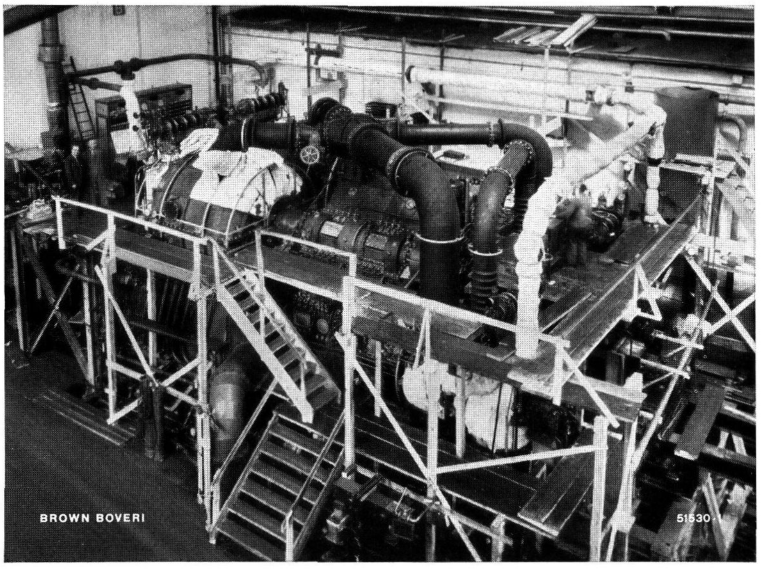


Fig. 2. — Four-cylinder high-power geared marine turbine set up for works tests and acceptance trials.

The 25 m long test beds enable large marine turbine units to be set up with their accessories as on board ship for carrying out works tests.

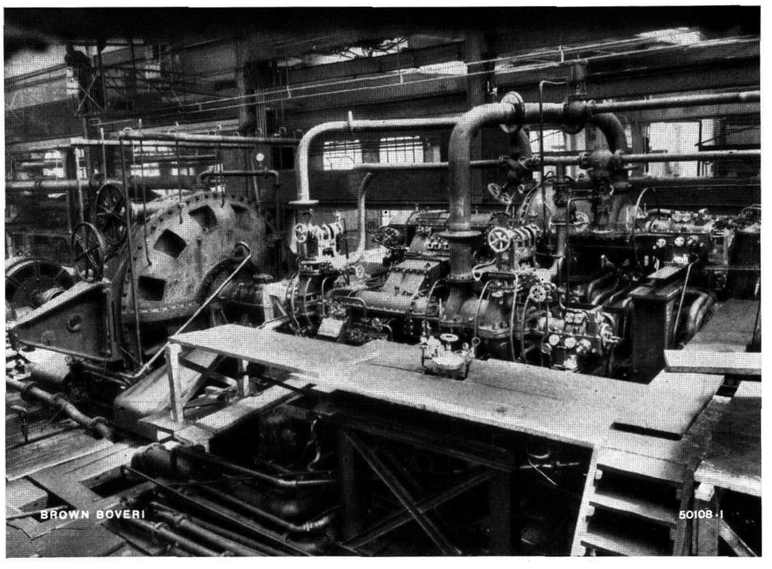


Fig. 3. — Multi-cylinder geared marine turbine on the test bed.

3000 S. H. P. marine turbine set up on the test bed with its condenser for carrying out trials. Full load can be applied by means of a water brake. Medium and small size units can be tried out in several well-equipped test bays.

by the construction of large assembly bays and have thus kept in a position to meet all exigencies for the thorough testing of large machines (for land plants as well as for marine purposes). The installation of a Velox steam generator having an output of 20 t/h, occupying but a very small space in the boiler house of the test department, enabled the flexibility, the service availability and the economy of the steam requirements for test purposes to be considerably increased. Only a few minutes are required to bring this boiler from cold state to normal pressure, and the fully automatic pressure and feed water regulation enables the output to be increased from no load to full load in a fraction of a minute. The rise in steam pressure when the load is suddenly dropped remains within the range of control of the regulating devices, so that even extreme load fluctuations do not cause the safety valves to blow.

The Velox boiler has proved itself very suitable for the testing of large marine turbines, especially for quick reversing trials. Numerous tests on marine turbines have shown conclusively, that even during rapidly effected manœuvres and reversing, no special arrangements are necessary for disposing of excess steam. The curves illustrated in Figs. 4 and 5, taken during the tests on three Velox boilers, each of 75 t/h output, show how well this type of steam generator behaves during unusually sudden load changes. Fig. 6 shows the compact construction of the Velox, enabling large outputs to be accommodated in relatively small spaces, without sacrificing anything of its accessibility.

Only a few special problems connected with the many interesting tests carried out on marine turbines are dealt with here.

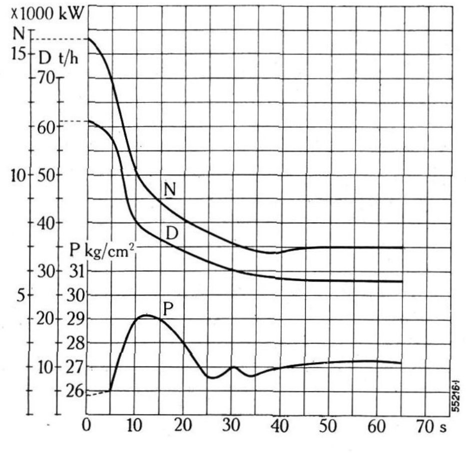


Fig. 4. — Regulation characteristics of a Velox boiler on sudden removal of the load.

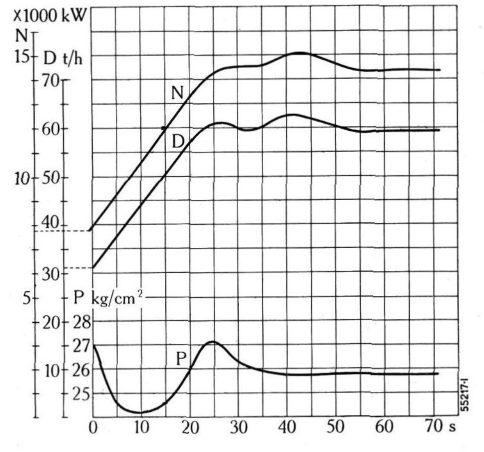


Fig. 5. — Regulation characteristics of a Velox boiler on sudden application of the load.

N. Terminal output of the turbo-set in kW. D. Steam output of Velox boiler in t/h. P. Absolute live steam pressure of the Velox boiler in kg/cm².

During these tests, the Velox boiler operated alone supplying a turbo-set the load of which was reduced from 16,000 to 8000 kW and increased again to 16,000 kW. Because of their great flexibility, Velox steam generators are particularly suitable for widely varying loads.

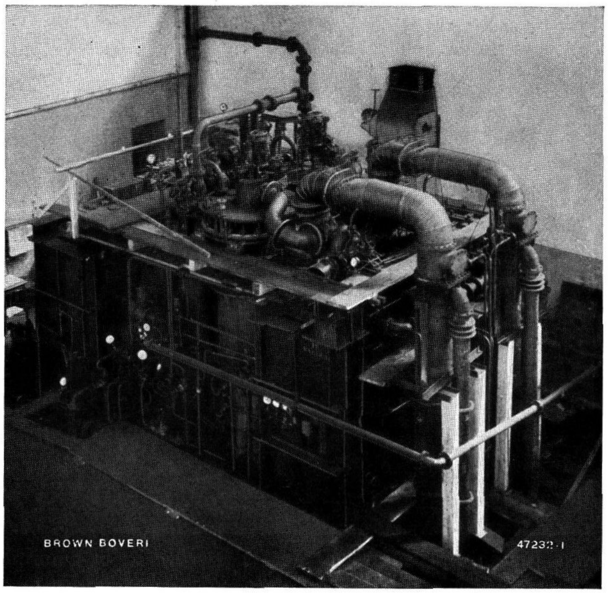


Fig. 6. — Marine Velox boiler plant for the mail and passenger steam ship "Bore II" of the Ångfartygs A. B. Bore, Åbo, on the test bed.

Output	16 t/h.
Steam pressure	16 kg/cm² abs.
Steam temperature	320° C.

The compact yet accessible design of Velox boilers make them particularly well suited for mounting in ships.

I. AIR LEAKAGE TESTS.

The widely ramified network of pipes which in ship installations, together with the condenser, is under vacuum, provides countless sources of air leakage. Locating and removing them often causes much difficulty. The question of the amount of air leakage to be expected must be solved already when designing the air removal equipment. If official steam consumption tests at the works have been agreed upon, care has to be taken already on the test bed to reduce these leakages to a minimum. If it is desired to obtain an idea as to the amount of leakage, reliable results can be obtained by direct measurement of the air quantity evacuated by the ejector. It is then important that the ejected air shall first be well cooled before being measured, in order that the greater part of the water vapour which may have been withdrawn from the air shall be condensed. The quantity of leakage air to be expected in large plants, is of the order of 30—90 kg/h, the lower value applying to well sealed installations, and the higher one being considered as the admissible limit. In general, it is advisable to dimension the ejecting apparatus liberally.

II. TESTS IN AN INCLINED POSITION.

In order to allow for extreme conditions, surveying bodies have specified definite angles of slope under which the machines must still operate satisfactorily. Whereas with electrically driven machines the angle of slope can readily be altered during the tests (Fig. 7), it is generally considered sufficient in the case of the turbine-driven machines to carry out a test in one extreme position (Fig. 8).

III. ACCESSIBILITY AND EASY DISMANTLING.

A fundamental requirement of all ship machinery is that it shall be readily accessible and easily dismantled, in order to facilitate overhauls and repairs. As the erection of ship machinery on board is generally done by the shipyard, that is, without the assistance of erectors from the manufacturers' works, this requirement must be strictly observed. A thorough testing, taking into account the conditions on board, is possible at the makers' works. The turbo-fan [1] with vertical turbine and gear is a typical example

[1] E. Klingelfuss & R. Schmid, "Turbine-driven Marine Auxiliaries" see Fig. 8, page 293 of the present number of this journal.

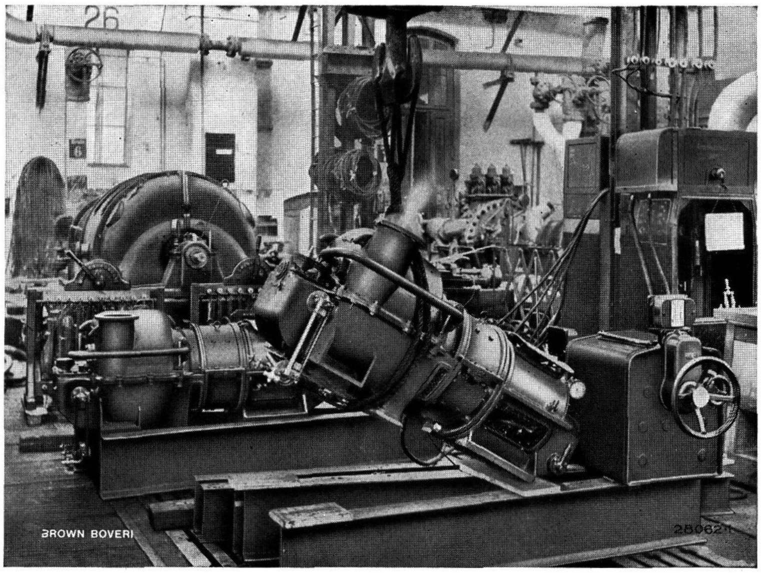


Fig. 7. — Inclined position test on a tank blower.

Electrically driven machines can be brought to a maximum angle of tilt whilst in full operation by slinging them from a crane.

of such an auxiliary. By removing only a few bolts, and by swinging open the hinged casing half, complete accessibility is afforded to all parts, without having to use any crane or any other lifting device.

IV. MEASUREMENT OF THE BLADING TEMPERATURE WHEN GOING ASTERN.

The temperature of the blading of the ahead turbines increases more or less rapidly with increasing speed of the reversing turbines. The values then attained generally exceed those normally prevailing. This is due mainly to the high ventilation losses of the idling forward turbines, and is particularly noticeable in the long low-pressure blades. The rise in temperature depends, moreover, to a considerable extent on the vacuum, which in fact has accordingly to be taken into account during the tests. The rapid, and to a large extent local variations of temperature in the rotors and casings occurring when steaming astern for long periods can be followed during the works tests by means of continuous measurement of the expansions of the shaft and casing, of the blading clearances and of the blading temperature (Fig. 9). In this manner it is possible thoroughly to investigate the behaviour of these parts under various operating conditions. Frequently, the hottest spot for the locating of the service instruments is not known. It can, however, then be determined during these tests by the application of a number of thermometers in different places.

V. HYDRAULIC PRESSURE TESTS.

The perfect sealing of the jointing flanges of high-pressure turbines constitutes an important problem of marine turbine manufacture. The hydraulic pressure tests, which surveying authorities require to be made prior to the assembly of the complete plant, give a good picture of the strength and freedom from leakage of the casing, as such tests are made at pressures well above operating pressure.

VI. OVERSPEED TESTS.

Because of the enormous powers required for overspeeding large bladed rotors at atmospheric pressure, such parts are generally tested at the makers' overspeed

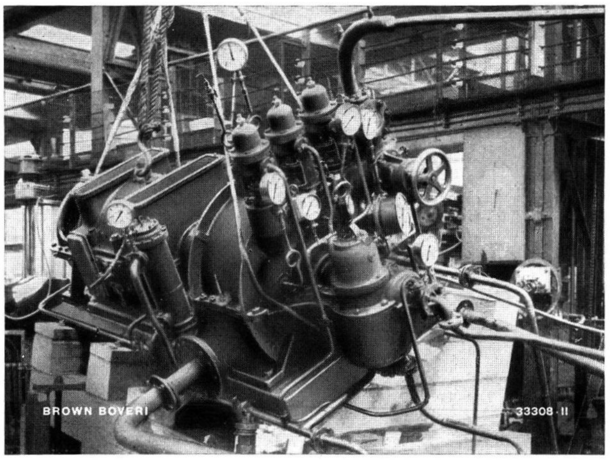


Fig. 8. — Inclined position test on a turbine driven ship's lighting set.
After concluding the ordinary tests, turbine driven auxiliaries are re-erected for further tests at the specified angle.

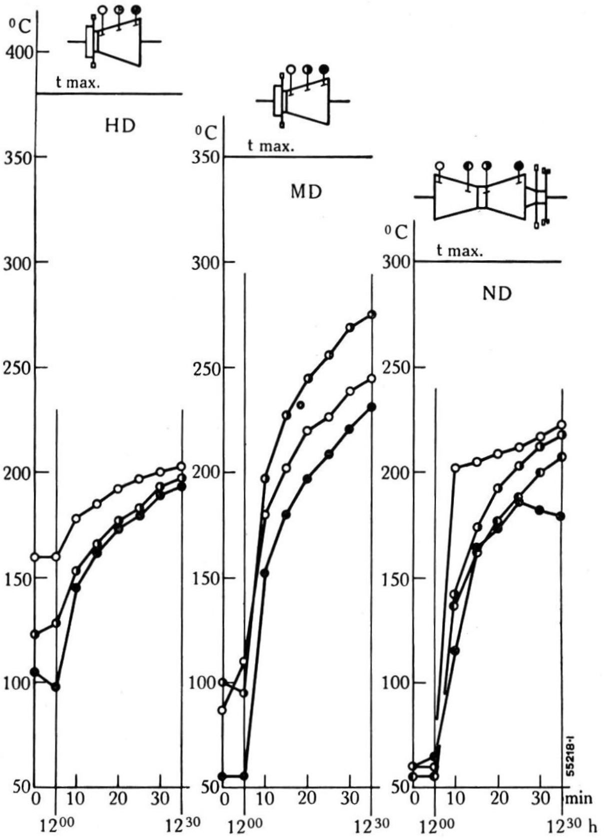


Fig. 9. — Temperature measurements on ahead turbines during astern running.

HD. High-pressure cylinder.
MD. Intermediate-pressure cylinder.
ND. Low-pressure cylinder.
t_{max}. Maximum admissible temperature.

During prolonged steaming astern, the blading and the cylinders of the forward turbines running in reverse, especially those of the low-pressure stages, attain temperatures which are in excess of the normal operating temperatures. The diagrams show the rise in temperature measured in the three cylinders of a high-powered multi-cylinder geared marine turbine during a 30-minute reverse running trial on load at the works. To determine the variation exactly, measurements were made at a number of points.

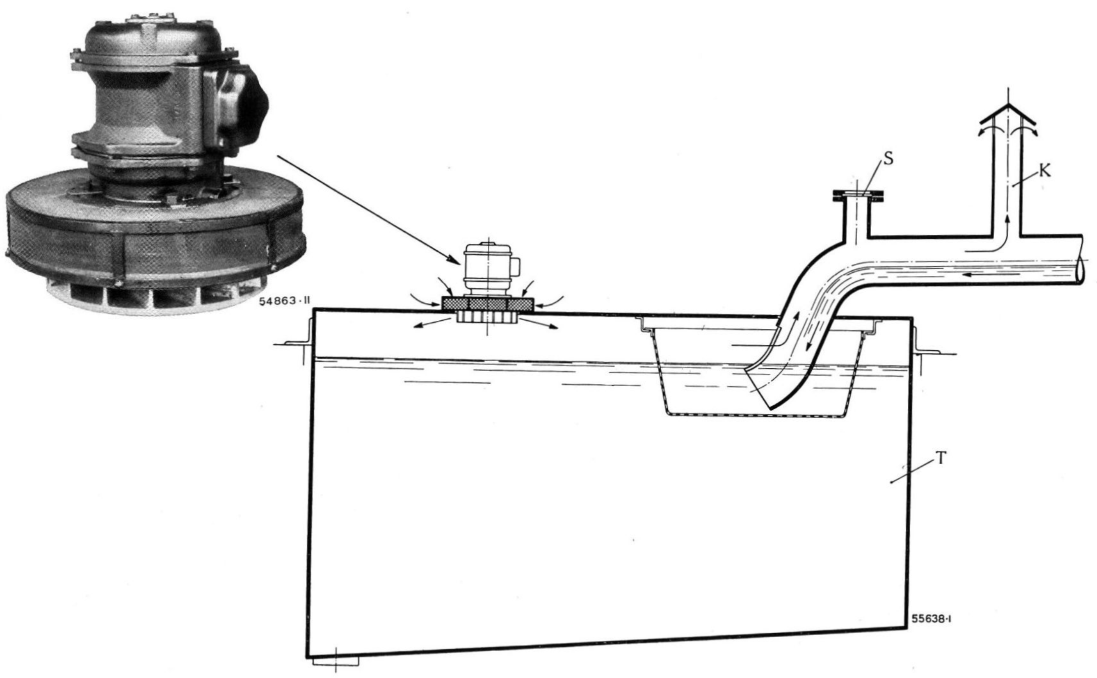


Fig. 10. — Rust-prevention by ventilation.

Diagrammatic representation of a ventilating installation with the oil tank T, the vent pipe K and the humidity check glass S. On the left, the simple design of motor combined with the fan, which is built in a number of sizes. Both during shut-down and during service, rusting of the oil-lubricated parts of gears and turbines, which may attain dangerous proportions, may occur due to the circulation of oil containing water. The addition of a ventilating fan enables their rusting to be avoided. The water is then continually extracted from the circuit in form of fog.

pits before being bladed. When the turbine is completely assembled on the test bed, it is then run for a few minutes at a speed close to that at which the overspeed governor has been set to trip, whilst a careful watch is kept on the bearing temperatures and on the amplitude of the vibrations.

VII. WARMING-UP AND STARTING TESTS FROM THE COLD STATE.

By coupling up with a water brake and adjusting the latter to a definite load, it is possible to carry out starting and load tests under conditions approximating very closely to the real ones. The time required for bringing the turbine up to a definite load may be specified. During such tests the ahead and the astern turbines are forcibly warmed up while being rotated by the turning device, by means of permanently located heating nozzles. Specially provided drainage arrangements of large section which can be opened from a central control point enable the water condensing in the cylinders during warming-up to be evacuated directly to the condenser. It is interesting to record the thermal expansions of the cylinders and shafts during such tests, and thus to follow the progress of the warming-up process, and the attainment of the steady state.

VIII. VENTILATING DEVICE FOR PREVENTING THE RUSTING OF GEAR PARTS AND OF OIL PIPES.

A difficult problem is the prevention of rust formation on the oil-lubricated parts during long periods of rest and during service. Some time ago we developed an arrangement also employed with success in land installations for drying out the walls of oil tanks, oil pipings and the casing of oil-lubricated gears. Fig. 10 shows the ventilating arrangement diagrammatically. As can be seen from the drawing, the air drawn in by a fan flows over the surface of the oil and thence through the free section of the oil pipes to the bearing pedestals and gear casings. The air finally escapes through a number of vent pipes situated at high level points of the oil system. Glass inspection windows may be provided at different places on which the water vapour condenses when the ventilation is inadequate, thus furnishing a means of observing the effectiveness of the ventilating device. Tests in which the air thus delivered to the atmosphere was passed through a cooler, showed that depending on the output of the fan, from 0·8 to 1·5 kg of water vapour could be removed per hour. In this manner it is possible to avoid rusting of gear parts or of oil passages, or the accumulation of water when the plant is either stationary or in service.

(MS 855) *W. Hiltpold. (Hv.)*

BRIEF BUT INTERESTING

Modern Short-wave Transmitter for Ship to Shore Traffic.

Decimal Index 621.396.932
621.396.61.029.58

IN recent years there has been a general tendency to substitute valve transmitters for the spark transmitters introduced at the beginning of the century. Since the significance of short waves for communication over great distances was recognized, ships are now also being increasingly equipped with short-wave transmitters which enable the vessel to keep in touch with the home port from distant parts with an extremely low transmitting power. Our short-wave dual transmitter type SO 15/300 is particularly suitable for coastal wireless stations where rapid selection of method of operation and change of wave-length are of vital importance. It is designed for modulated and unmodulated telegraphy as well as for telephony over a wave-range of 2·7—15 megacycles. Simultaneous operation on any two desired ship wavebands is possible with a high-frequency output of 300 W in telegraphy service. Ingenious remote control gear permits full-automatic operation of the station from a distant point over a single telephone line. The transmitter can be set to work on any of the specified wave-lengths within 20 seconds, while changes in wave-length or method of operation can also be effected within a few seconds.

Apart from the transmitters referred to we are at present designing various other modern types for coastal wireless stations and ships.

(MS 862) *Dr. R. Stuber. (E. G. W.)*

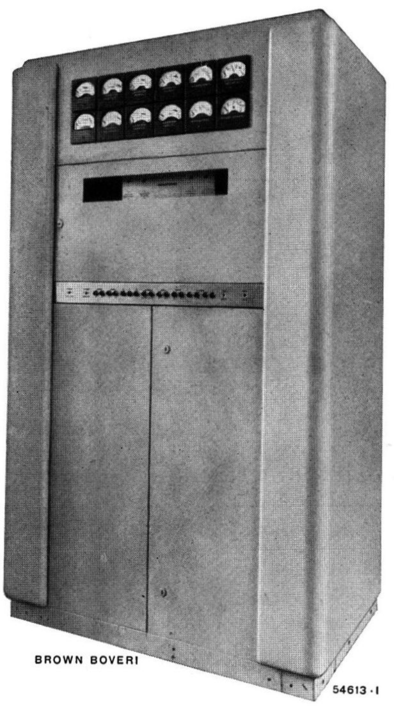


Fig. 1. — Short-wave dual transmitter for coastal wireless stations.
Increased safety through simultaneous transmission on two ship wavebands. Rapid service through full-automatic change of wave-length and method of operation.

The Brown Boveri Frigibloc as High-capacity Refrigerating Unit on board Ship.

Decimal Index 621.576.3

THE frigibloc is a complete refrigerating unit suitable for outputs exceeding 250,000 kcal/h (about 80 tons of refrigeration). Capacities of this magnitude are now common for refrigeration and air conditioning on board large modern ships.

The frigibloc comprises exclusively rotating machines which ensure a compact design and smoothly-running plant. The erection is thereby simplified, inasmuch as only light and cheap foundations are necessary.

The turbo-compressor consists of a well-tried-out centrifugal machine utilizing the most up-to-date refrigerants, such as methylene chloride or freon. The properties of these substances are very much more favourable than those of ammonia, sulphur dioxide, methyl chloride, etc. A particular advantage of these new refrigerants consists in their moderate pressures throughout the refrigerating cycle, which are in part below that of the atmosphere.

For ships' installations, preference has been given hitherto to carbon dioxide (CO_2) as a refrigerant. Its chief drawbacks reside in its high operating pressures (about

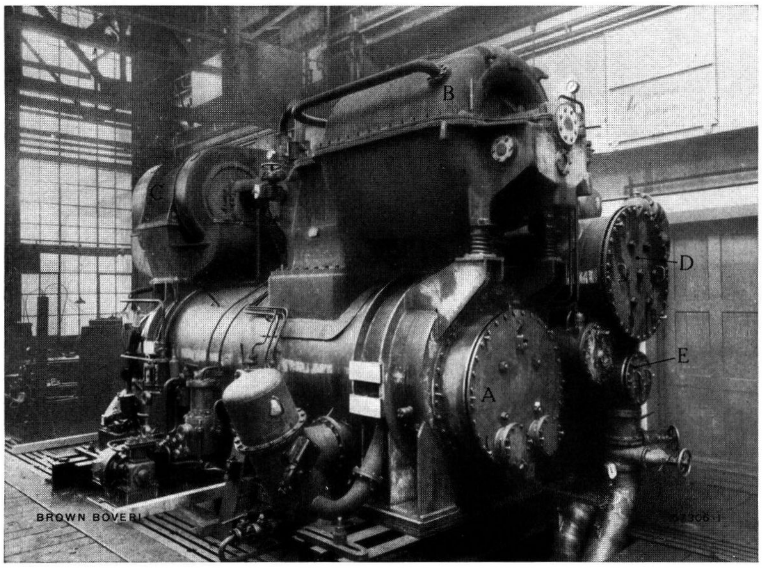


Fig. 1. — Frigibloc for 1·7 million kcal/h on the test bed.

A. Evaporator.
B. Turbo-compressor.
C. Driving motor.
D. Condenser.
E. Under-cooler.

The frigibloc is a complete refrigerating unit. The compact design combined with easy access to all parts are factors of outstanding importance for ships' plants.

This design is generally adopted by Brown Boveri for refrigerating outputs down to about 200,000—250,000 kcal/h.

76 kg/cm² at 30° C) and low critical point. For these reasons, ammonia has been employed of late to an increasing extent for marine refrigerating plants. It is, however, far from ideal, as it is both inflammable and toxic; furthermore its operating pressures are still considerably above those of more recent refrigerants. In presence of water, ammonia attacks copper and its alloys, which are used extensively on board ship on account of their ability to resist sea water. The refrigerants used for frigiblocs have just as favourable properties in this respect as carbon dioxide.

The frigibloc has been designed with particular care to ensure its tightness. This, together with the moderate operating pressures throughout the entire cycle, ensure that practically no refrigerant escapes. The compressor shaft traverses the casing only once, so that there is a single gland to the atmosphere. Thanks to an effective oil seal, no leakage can occur here.

Formerly direct expansion was extensively adopted for ships' refrigerating plants, the refrigerant itself going to the places which need cooling. Long pipelines having a number of joints are required for the refrigerant, which are a potential source of trouble and danger. It requires, for instance, no great imagination to visualize the serious consequences due to the failure of an ammonia pipeline on board ship. For these reasons, indirect transmission with brine or chilled water for conveying cold to the places where it is required has met with increasing favour of late. The frigibloc is particularly well adapted for this form of transmission.

Standard a. c. or d. c. motors are mostly employed for driving the turbo-compressor of the frigibloc, but steam turbines are also suitable.

The design of the frigibloc has been examined and approved by marine underwriters for use on board ship.

(MS 913) *D. Marples.*

Brown Boveri Geared Coupling.

Decimal Index 621.825.2

FOR the connection of high-speed shafts Parsons proposed and developed at a very early date a claw-type coupling, which has been used right up till the present day. The claws of this coupling (Fig. 1), however, require a well-conceived lubricating system and very accurate and costly grinding-in or scraping of the rubbing surfaces in order to avoid wear of the latter should the axes of the connected shafts become slightly out of line or out of level.

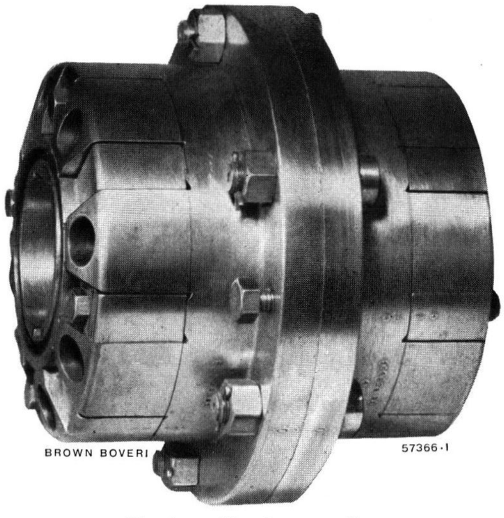


Fig. 1. — Claw-type coupling.

This type of coupling has been frequently employed for steam turbines and turbo-compressors. Limited compensation for misalignment, due to the long claws, and lubrication difficulties at high speeds restrict their application.

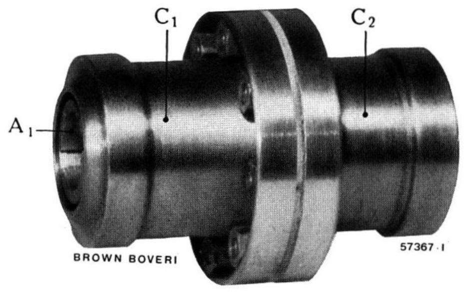


Fig. 2. — Geared coupling hitherto constructed, with shell in halves joined by bolted flanges.

A_1. Hub of coupling.
C_1, C_2. Halves of shell with coupling flange.

The torque is transmitted by flanges having a large external diameter. The centrifugal force on the bolt-heads and nuts limits the speed.

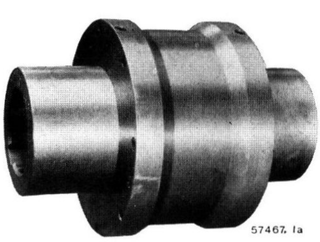


Fig. 3. — Brown Boveri geared coupling with one-piece shell.

For this reason the so-termed geared coupling with which the engaging parts are machine-cut gear teeth requiring no additional operations has found wider and wider application in recent years. This coupling (Fig. 2) comprises two flanged hubs A_1 and A_2 with gear teeth cut in the periphery of the flanges and a shell in halves C_1, C_2 having two rows of internally cut teeth which mesh in the external teeth of the hubs around their whole periphery. The two-part shell was intended to facilitate dismantling, but the necessary bolted flanges, which transmit the peripherial forces, gave the coupling a large overall diameter and this was found to be a great drawback when it was required to be located in the body of a bearing, as is usually the case with turbo-generators and compressors, for instance.

This type of coupling has been further developed by Brown Boveri as shown in Figs. 3—5, 8, and 9 and is now available for the transmission of powers up to 50,000 kW and speeds up to 25,000 r. p. m. As with the earlier design there are hubs A_1 and A_2 (Fig. 4) on each of the shafts to be connected, these having narrow flanges in the periphery of which gear teeth are cut. The shell B over the two hubs, which has two narrow rows of internal teeth corresponding to those on the hubs, is now in one piece, however, and transmits the torque directly. The shell B is supported and centred on the hubs A_1 and A_2 at both ends by screwed-on plates C_1 and C_2 with round-lipped flanges D_1 and D_2 fitting in recesses under the centre of the rows of teeth on the hubs. Inasmuch as the necessary screws have to transmit no peripherial forces they can be kept small.

As shown in Fig. 5 the rows of teeth on the hubs A_1 and A_2 and the length of the shell B are so arranged that when dismantling the coupling the shell B need only be axially displaced to the end of the opposite hub A_1 after the end plate C_2 has been taken off, whereupon the two shafts can be lifted out. With this design the stresses on the teeth remain the same, but the diameter

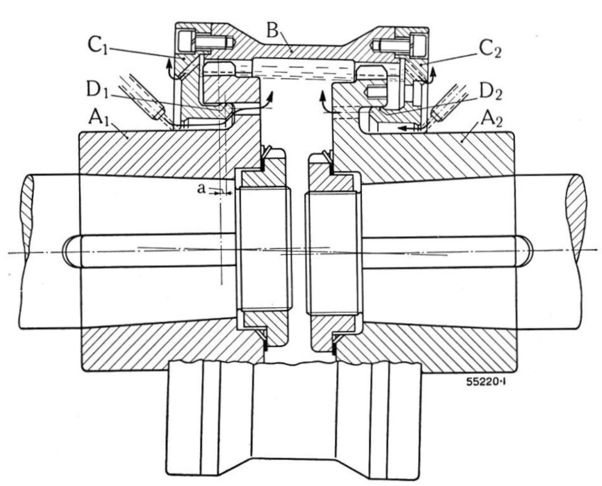


Fig. 4. — Brown Boveri geared coupling with spray lubrication.

A_1, A_2. Hubs on shafts to be coupled. C_1, C_2. End plates.
B. Single-piece shell. D_1, D_2. Round-lipped flanges.
a. Maximum deflection of teeth with coupling out of alignment.

In the drawing the shafts are shown skewed to the same degree.

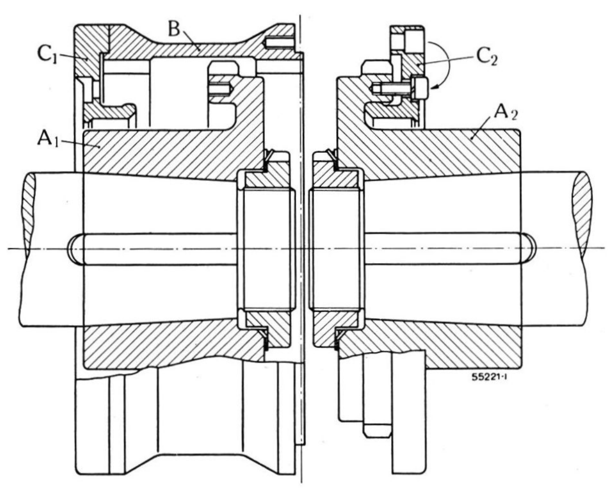


Fig. 5. — Brown Boveri geared coupling ready for inserting or lifting out the shafts.

For this purpose end plate C_2 is bolted to hub A_2 and shell B pushed over hub A_1.

The teeth are immersed in a ring of oil and the torques are transmitted by a boltless one-piece shell B from one hub to the other. A reduced number of component parts and higher admissible speeds are features of this new design.

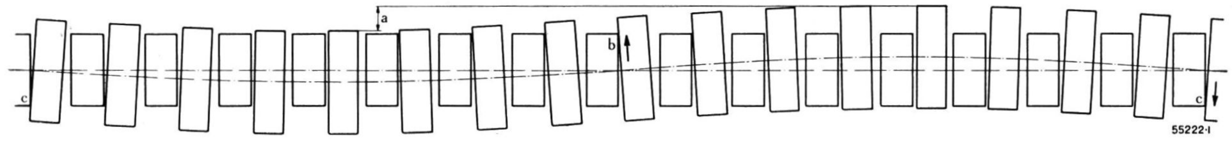


Fig. 6. — Development showing various positions of shell teeth between hub teeth.
The hubs and shell are mutually skewed by the amount "a". The teeth continually move to and fro axially by the amount "a" in service.

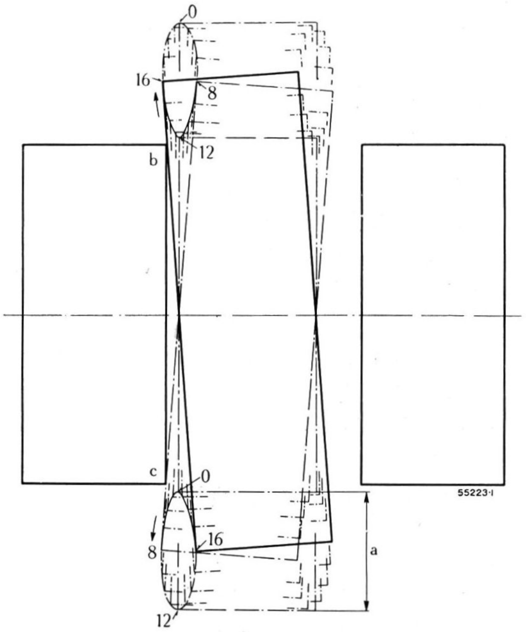


Fig. 7. — Position of a shell tooth between the corresponding hub teeth during one revolution of the coupling shell skewed by the amount "a".
From positions 8—16 the film of oil is compressed at b, thus producing a pressure zone at this point. During each revolution of the coupling, pressure zones are produced alternately at b and c. The resulting oil film prevents metallic contact of the teeth and, in consequence, wear.

of the coupling is smaller, its weight lower, and manufacturing costs reduced.

For the lubrication of the teeth of the coupling, which is closed by the end plates C_1 and C_2, a charge of oil is poured into its interior. In service the centrifugal force causes the oil to spread to the inside of shell B and form a ring of oil immersing all of the teeth. In the case of high-speed couplings spray lubrication is employed as shown in Fig. 4.

When the shafts are out of line or out of level the teeth of the shell are skewed as compared to the teeth of the hub. In this case the teeth of the shell make cyclic rolling movements between the teeth of the hubs (Figs. 6 and 7).

Fig. 6 is a development showing the various positions of the shell teeth between the teeth on the hubs, while Fig. 7 depicts the different positions of one shell tooth between the corresponding hub teeth during one revolution of the coupling. The path of a shell tooth between two hub teeth can be compared to the rudder of a paddle boat rising and sinking in the water and moving in the longitudinal direction. During this motion pressures are set up in the film of lubricating oil between the teeth, which are greater the thinner the layer of oil. These pressures, however, are favourably influenced by the wedge-shaped character of the layer of lubricating agent. If the viscosity of the oil and the load on the teeth are correctly selected pressure transmission without metallic contact — such as has proved to be so satisfactory in the case of thrust and journal bearings — is achieved, i. e., bearing parts are not subject to wear.

A geared coupling supplied for the transmission of 1400 kW at 1500 r. p. m. is shown in Figs. 8 and 9.

(MS 919) *J. Niederhauser. (E. G. W.)*

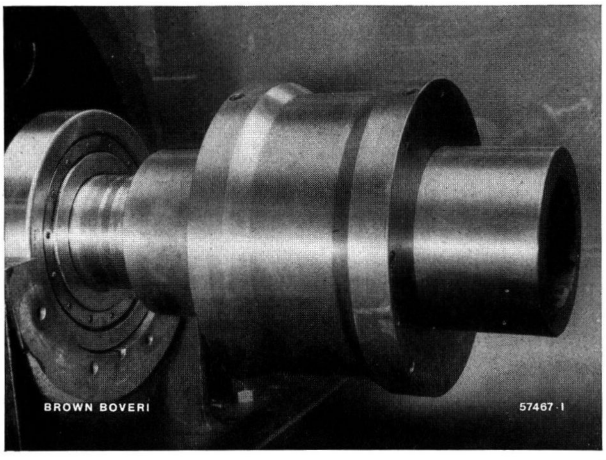


Fig. 8. — Brown Boveri geared coupling for transmitting 1400 kW at 1500 r. p. m.

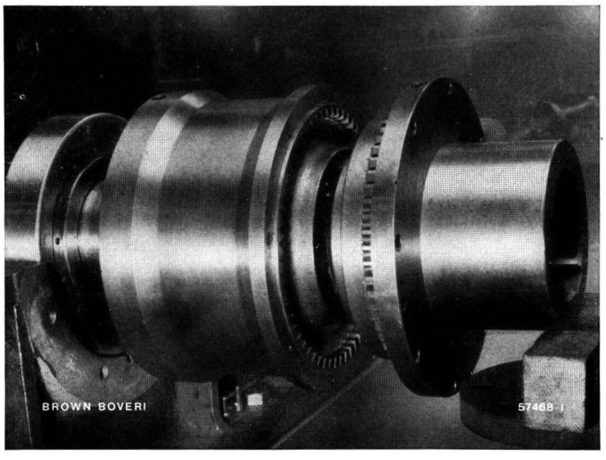


Fig. 9. — Geared coupling shown in Fig. 8, ready for inserting or lifting out the shafts.

Oil-filled model for low and medium speeds fitted between gearing and a d. c. generator.

Electric Welding Machines for Shipyards and Shipowners.

Decimal Index 621.791.7 : 629.12

IN the course of the last few years electric welding has found wider and wider application in shipbuilding. Articles and notes on the subject are constantly appearing in the shipbuilding press and lay stress on the shorter building time and saving in material to be achieved by electrically welding instead of riveting ships' hulls. The urgency of replacing lost tonnage rapidly and with as little material as possible has naturally accentuated this tendency, but already before the outbreak of hostilities the advantages accruing from electric welding in shipbuilding, i.e., reduced weight, shorter building time, and lower cost, were recognized in many quarters. As early as 1930 Brown Boveri supplied one of the first large welding installations for shipbuilding purposes. This was a multi-operator welding installation with an aggregate rating of 36,000 A, 40 V d. c., for the French Shipyard Penhoët, St. Nazaire (Fig. 1). This installation is noteworthy in that a constant voltage of only 40 V is employed instead of the more usual 60—65 V and that, notwithstanding this low voltage, high-class welds are obtainable through the use of special patented welding current regulators (Fig. 2) which permit the welding current to be regulated progressively. The losses in the regulators are about 50 % less and in consequence the welding machines approximately 30 % smaller than if designed for operation at the more usual voltage of 60—65 V. By reason of this low voltage and the use of our welding current regulators, working conditions are much more pleasant for the welders and production is increased.

In shipyards the load is frequently liable to be concentrated at certain points and to relieve the main welding station we have evolved multi-operator monobloc welding sets (Fig. 5) with built-on regulating apparatus which can be taken to the points of greatest demand by means of trucks or cranes and put into service without delay.

Fig. 1. — One of three welding stations in the French Shipyard Penhoët, St. Nazaire, with four convertor sets each for a welding current of 2000 A.

The multi-operator welding installation with an aggregate rating of 36,000 A at 40 V d. c. supplied by Brown Boveri to this shipyard in 1930 was one of the first large welding plants to be built for shipbuilding purposes.

The material-saving and weight-reducing potentialities of electric welding induced Brown Boveri thoroughly to investigate this manufacturing process more than forty years ago. A number of electric welding machines were developed both for d. c. and a. c. and through extensive research an indispensable adjunct for the manufacture of

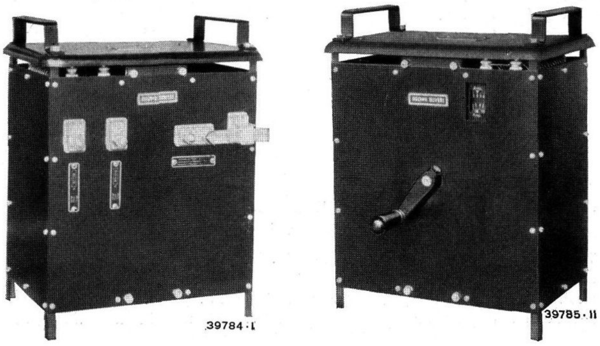


Fig. 2. — Welding current regulators.

One regulator improves the efficiency of the installation through its inductance and the other permits the welder to select the best current setting for the work in hand.

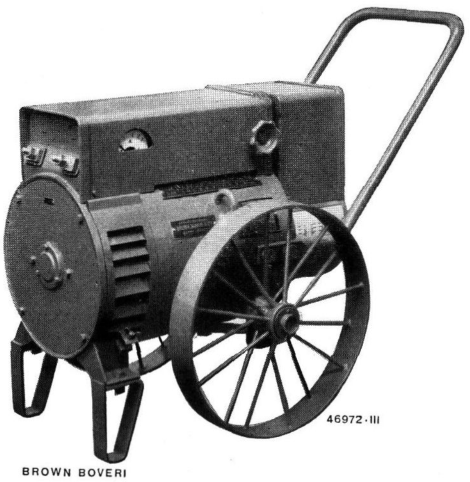


Fig. 3. — The Brown Boveri instantaneous-response welding sets for d. c. arc welding are characterized by excellent welding properties, wide range of regulation of welding current, high efficiency, and low weight.

They are mounted on sprung trucks and the appertaining apparatus is readily accessible after removal of the casing.

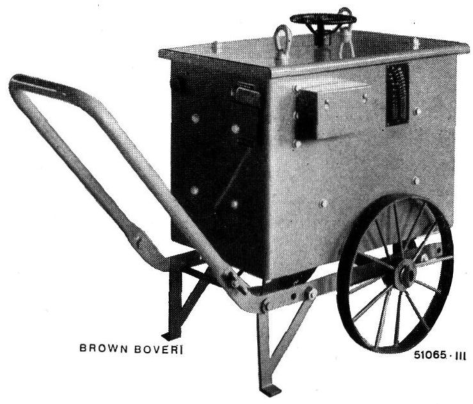


Fig. 4. — Arc welding transformer, with special characteristic to give *a steady arc. The welding current is progressively regulated.*

Eminently suitable for horizontal work on iron.

machines and apparatus created.[1] In this way the well-known Brown Boveri instantaneous-response arc welding generators[2] for welding-currents of 200—750 A (Fig. 3) and the welding transformers for 175—500 A (Fig. 4) were developed. The welding current is progressively adjustable, and ease of arc establishment and reliable welding operation are inherent features of all of the welding units of which more than 10,000 are in use all over the world.

The question of choice of d. c. welding sets or the less expensive welding transformers depends on the national specifications applicable and the work to be welded. Alternating current is employed chiefly for horizontal welds on iron and sheet-iron, whereas direct current can be used

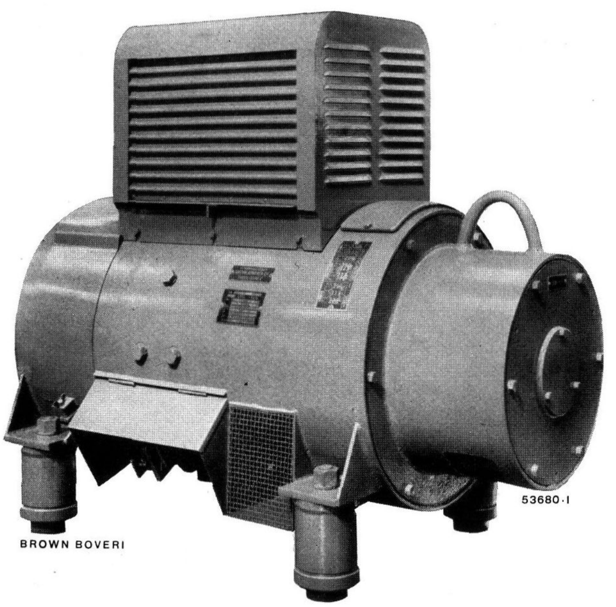


Fig. 5. — Mobile monobloc multi-operator welding set for currents of 1200—1500 A at no-load voltage of 40—50—60 V.

The motor and generator are contained in the same casing which also serves as mounting for the complete control gear. Due to their mobility such sets facilitate the planning of work in that they can be used to supplement an existing welding current distribution system and thus enable special yard conditions to be coped with.

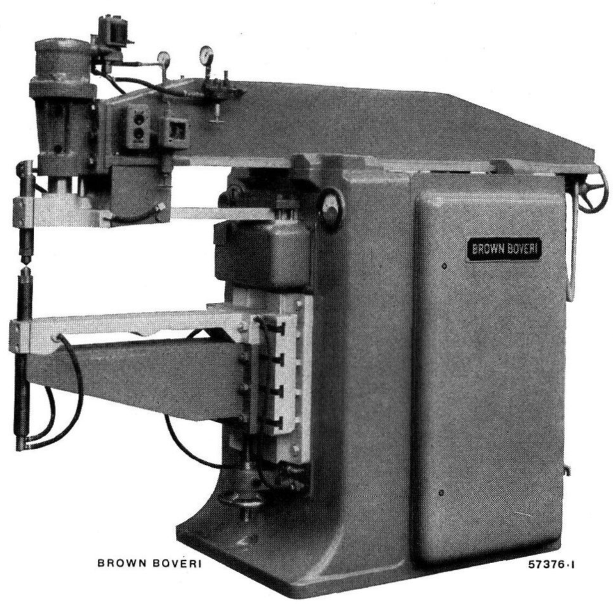


Fig. 6. — Spot and seam welding machine rated 240 kVA, with automatic control of welding current, welding time, and electrode pressure.

Welding now replaces riveting for many purposes, thus saving material and reducing labour costs.

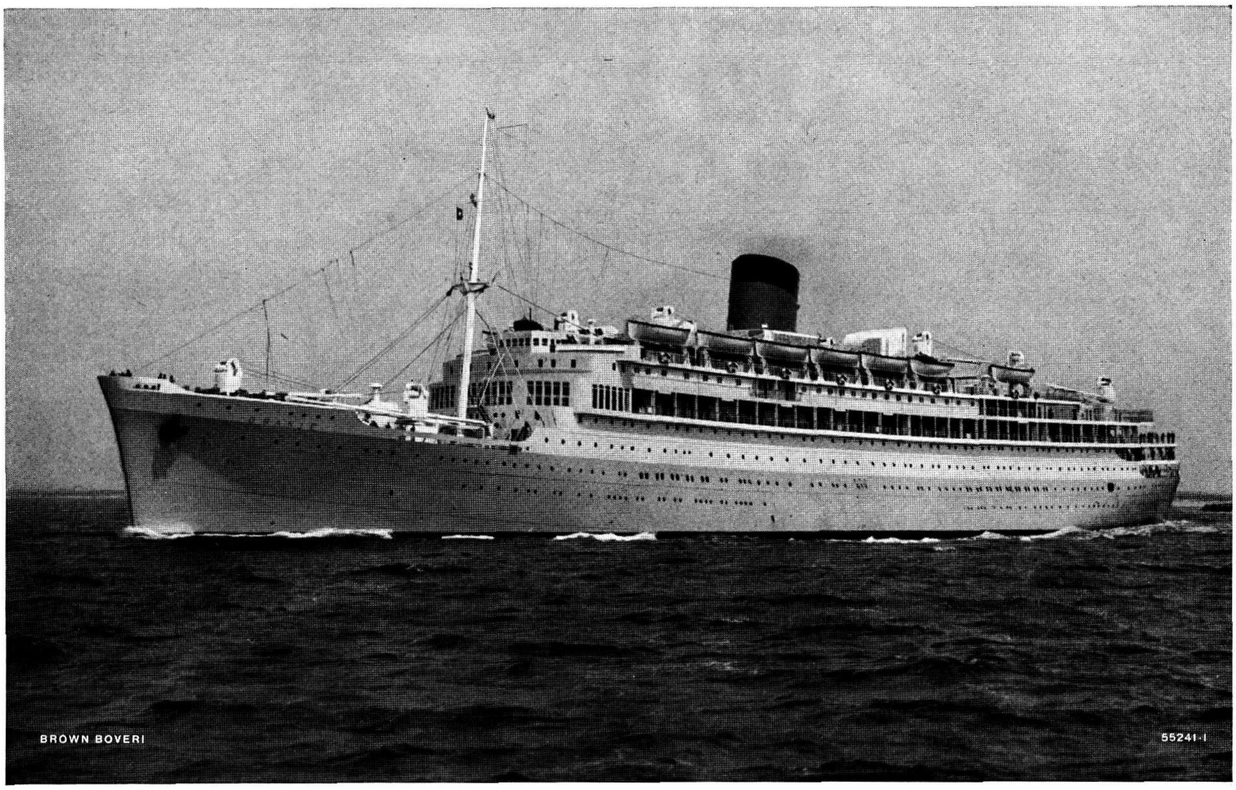
[1] H. Zschokke: "The Technology of Welding and the Examination of Welding Work. A Survey of Test Results." The Brown Boveri Review, 1937, p. 311.

[2] H. Kocher: "New Features incorporated in the Brown Boveri Welding Sets of Instantaneous-reaction Type for Direct-current Arc Welding." The Brown Boveri Review, 1942, p. 100.

for any welding work and will be found of particular advantage for welding thin sheets, non-ferrous metals of all kinds, and for casting repairs. Both systems are, inter alia, eminently suitable for rapid repair work on ships or in shipyards. Marine models can be delivered from stock.

A recent addition to the Brown Boveri range of welding equipment are resistance welding machines of the spot and seam welding type (Fig. 6). These have completely automatic control gear, i. e., the welding time, welding current, and pressure with which the parts to be joined must be pressed together are continuously regulated to adjusted values. A synchronous switch prevents current surges being transmitted to the supply system and also ensures sparkless disconnection of the welding current, thus substantially lengthening the life of the switch contacts.

(MS 923) *Th. Egg. (E. G. W.)*



M. S. "Oranje" of the N. V. Stoomvaart Mij. Nederland. (20,000 gross registered tons, 3 × 12,500 S. H. P.)

Brown Boveri equipment on this ship comprises scavenging blowers for the Sulzer main Diesel engines, 105 d. c. and a. c. auxiliary motors aggregating over 4900 H. P., and the voltage regulators and an automatic synchronizer for the three-phase auxiliary generators.

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