VOL. XXVI JUNE, 1939 No. 6

# THE BROWN BOVERI REVIEW

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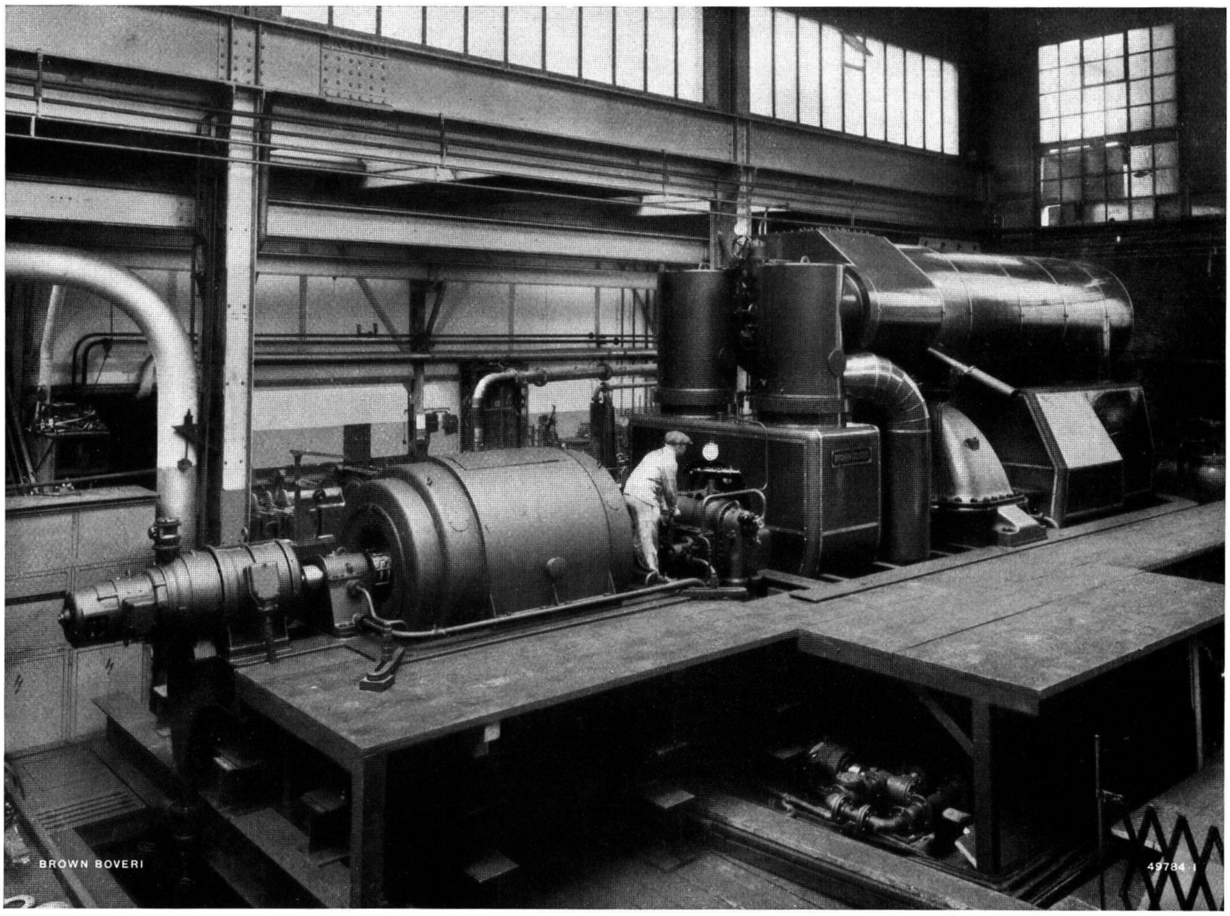


GAS-TURBINE SET FOR AN EFFECTIVE ELECTRICAL OUTPUT OF 4000 kW, 3000 r. p. m., 50 cycles, FOR THE NEUCHATEL ELECTRICITY WORKS. SET ON TEST BED IN BADEN.

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Printed in Switzerland

# HIGH-PRESSURE TURBINES
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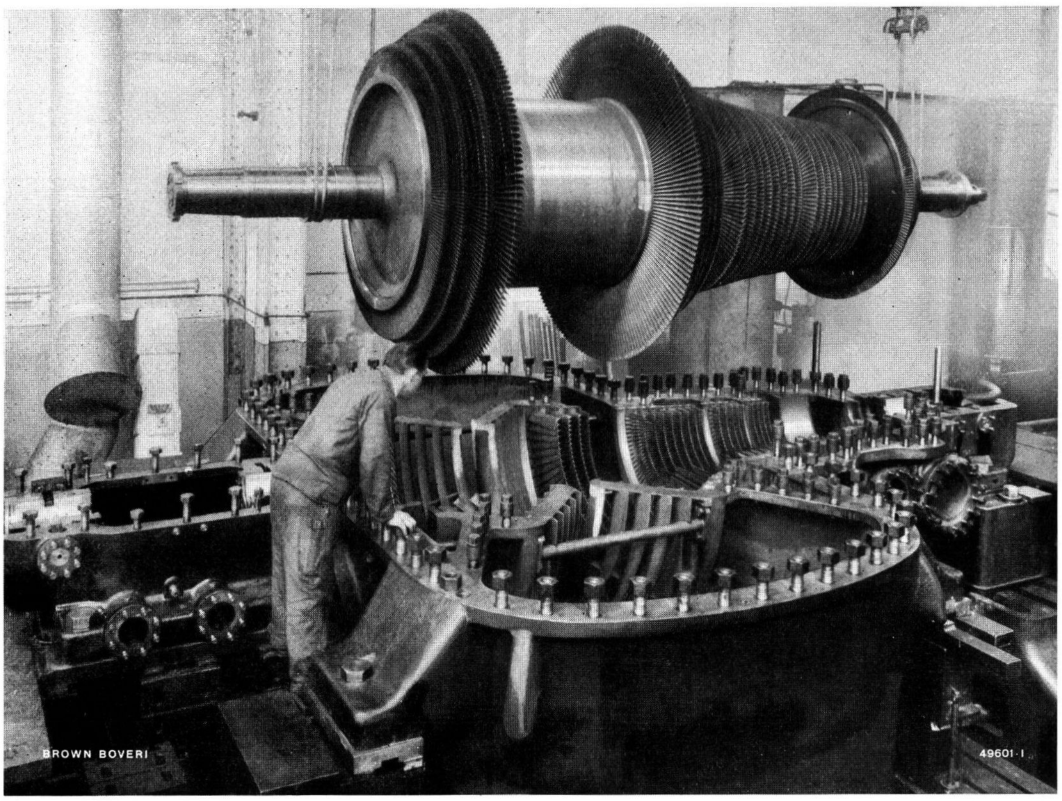
Single and multiple-extraction turbines and back-pressure turbines

for all the output ranges met with in practice




**Single-cylinder steam turbine for 32,000 kW at 3000 r. p. m. with rotor lifted out. This turbine is being erected, at present, in the Warsaw Electricity Works where a similar set is already running.**

## Surface-condensing plants
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# THE BROWN BOVERI REVIEW

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

VOL. XXVI JUNE, 1939 No. 6



## THE COMBUSTION GAS TURBINE: ITS HISTORY, DEVELOPMENT, AND PROSPECTS.[1]

Decimal index 621.438.1.

*After a brief historical summary of the development stages through which the gas turbine has passed, various fields of application are described in which the combustion turbine, in its simplest form, has found useful applications. The author then shows what can be done to improve the process and what the outlook is for gas turbine plants incorporating the said improvements.*

THE term "constant-pressure gas turbine" hitherto generally employed to denote a turbine actuated by the steady flow of the products of a continuous combustion under pressure in a combustion chamber, is inaccurate. The expression "constant-pressure turbine" was chosen to distinguish this machine from the constant-volume turbine, in which combustion takes place intermittently, by explosion in a closed chamber with a substantial rise in pressure. In the so-called "constant-pressure gas turbine" neither the combustion chamber pressure nor that before the turbine remains constant in service as, for instance, the steam pressure of a boiler; on the contrary, they depend on, and vary with, the load.

It is preferable, therefore, to call such a turbine a "continuous combustion gas turbine", or, briefly, "combustion turbine", in contradistinction to the explosion turbine.

### I. HISTORY.

The introduction of the gas turbine in the field of power generation is the realization of a long cherished dream of engineers. Hardly any other kind of machine has received more attention from inventors, skilled and otherwise. A great number of suggestions has been made in regard to the method of operation of gas turbines, whilst patents covering design details and component parts run into thousands.

From the patent office records of different countries, inventors appear to have started tackling this problem at a very early date, for the first patent was granted in 1791 in England to one John Barber (Fig 1). Since then, the number of patents has steadily increased.

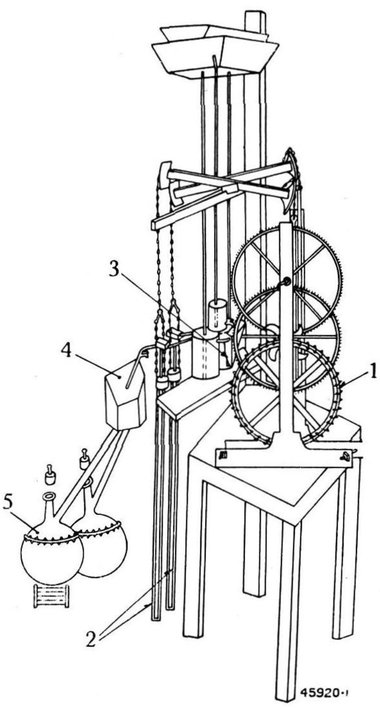



**Fig. 1. — Drawing of first patent for a gas turbine (John Barber 1791).**
1. Turbine.
2. Gas and air compressor.
3. Combustion chamber.
4. Receiver.
5. Gas producer.

What may have induced the first gas turbine inventors to want to produce a substitute for the reciprocating steam engine, at that time the only form of heat engine in use? The question of efficiency was probably given but scant consideration, for the theory of heat was then little advanced. Two reasons can be suggested. First, the natural desire to replace the steam plant consisting of steam boilers, steam engine, and condensing plant, by more simple equipment, and second, the wish to produce directly, by means of a turbine, the rotary movement (which, in the majority of cases, is necessary for the transmission of the power), and to do away with the crank and connecting rod mechanism of the steam engine. That the desired simplification has been achieved can be seen from Fig. 2, which shows the combustion turbine in its simplest form; b is the combustion chamber in which the gases are produced at a high temperature by the continuous combustion of some kind of fuel in an atmosphere of compressed air.

According to the fuel used, such turbines are referred to as gas, oil, natural gas, blast furnace gas, or pulverized-coal turbines. With all these fuels, the temperature of the combustion gases is high and

[1] Paper read at a meeting of the Institution of Mechanical Engineers, in London on 24th February, 1939.

must be reduced to a value compatible with the creep strength of the gas turbine blading. This can be done either by the addition of a large excess of air, or by the injection of water, or by a partial abstraction of heat by water-cooled surfaces, or other similar means. Of these methods, cooling by injection of water must be ruled out of consideration, because of the loss associated with the latent heat of evaporation. The plant shown *in Fig.* 2 is based on the use of excess air as a cooling medium, and of oil as fuel.

The total quantity of air (combustion plus cooling air) is drawn from the surrounding atmosphere by

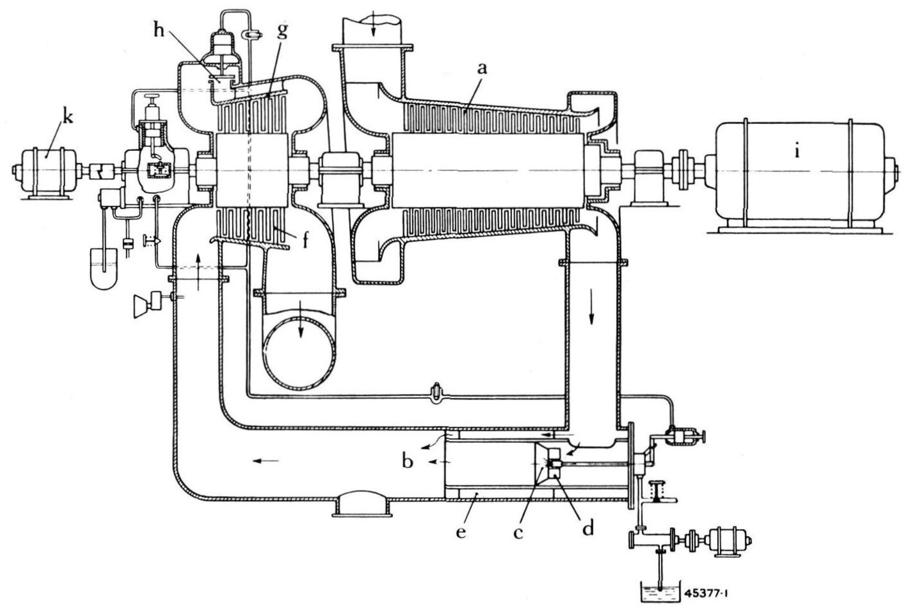



**Fig. 2. — Diagram of the simplest form of combustion turbine plant. With reaction type gas turbine and axial compressor for oil fuel, and with excess air cooling.**

a. Axial compressor.
b. Combustion chamber.
c. Combustion nozzle.
d. Burner.
e. Cooling-air jacket.
f. Gas-turbine blading.
g. Gas turbine.
h. Safety valve.
i. Generator.
k. Starting motor.

the axial compressor a, compressed to the combustion pressure of the order of 20—30 lb. per sq. in. gauge, and forced into the combustion chambre b. Part of the air serves as combustion air for the oil which enters the burner d at c, whilst the remainder is forced through the annular space between the wall of the combustion chamber and the burner jacket. Here it takes up heat from, and incidentally cools, the burner jacket, after which it mixes with the products of combustion, reducing their temperature to that admissible for the gas turbine blading f. In this simplest form of gas turbine, the gases go straight from the turbine g to the chimney. The speed of the gas turbine is governed by controlling the fuel oil supply, and by means of a bypass valve h, which acts as safety valve. In addition to the compressor referred to above, the gas turbine drives the generator i either directly or through gearing. The only auxiliary machines of the plant are the starting motor k, which is of a size to ensure the compressor supplying enough air to the combustion chamber for lighting up, and the lubricating and fuel oil pumps.

It will be seen that the early inventors' aim to simplify the steam plant has been attained, there being no boiler with auxiliary equipment such as feed water pumps, water treatment plant, etc., no condenser with circulating water, air extraction, and condensate pumps, no water supply system, cooling towers or such like. The boiler is replaced by a simple combustion chamber, although the compressor is now larger and more expensive than the forced and induced draught fans of a normal boiler installation.

Later inventors and scientific workers appear to have been striving for an improvement over the efficiency of the steam process, for, in about 1850, Redtenbacher wrote to Zeuner: "The fundamental principle of the generation and use of steam is wrong. It is to be hoped that steam engines will disappear in a not far distant future, as soon as we know more about the nature and effects of heat". The fact that Redtenbacher's prophecy has so far not come true, must be attributed mainly to the invention of the steam turbine which gave a new lease of life to the steam cycle, and for a long while caused the gas turbine problem to be relegated to the background. With the advent of the steam turbine, one of the principal aims of the gas turbine inventors was realized, namely the direct generation of power by a rotary movement without the help of pistons, cranks, etc., whereas this could not be achieved with the gas turbine, as long as the machine required for compressing the air was of the reciprocating type.

As far as is known, the first gas turbine to be built and tried was the hot air turbine developed by

Stolze, which is said to have been designed in 1872, although the trials were made between 1900 and 1904 (Figs. 3 and 4). This turbo set is of particular interest, because a multi-stage reaction gas turbine and a multi-stage axial compressor (most probably the first of its kind) were used. In view of the state of engineering development and limited knowledge of aerodynamics at that time, it is hardly surprising that this design was not a success. Even Sir Charles Parsons had to abandon the axial compressor at about the same time, after having done an enormous amount of work in connexion with it, this being one of the few cases in which this great inventor, in spite of his engineering genius, and of the resources at his disposal, was compelled to give up a question which he had embarked.[1]

It is a strange coincidence that the first centrifugal compressor ever built (a Brown Boveri three-cylinder turbo-compressor with 25 impellers, working all in series, to develop a pressure of 60 lb. per sq. in. abs.) was for a gas turbine designed by Armangaud Lemale which used paraffin oil as fuel. With a temperature of about 1030 deg. F. at the nozzles, obtained by injecting water into the combustion chamber, the turbine was just able to supply the power required for compressing the air. The net output was nil, and the efficiency was zero. The Société des Turbo-Moteurs, which built this turbine, did, however, reap some reward for their efforts, as the experience gained enabled them to improve considerably the performance of torpedoes by injecting and burning paraffin oil in the compressed air actuating the propelling engine. The conditions were here extremely favourable, as it was easily possible to avoid excessive temperatures, owing to the availability of an abundance of water for cooling purposes.

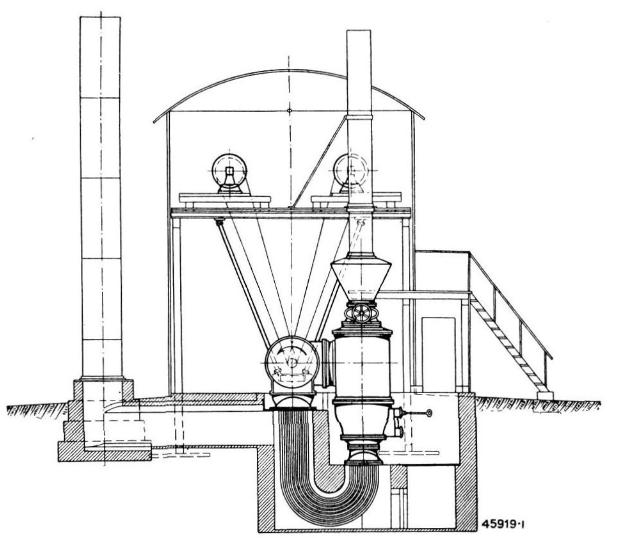



**Fig. 3. — Stolze's hot air turbine (1900—4).**

The reason why the first turbine did not prove a success can be seen in Fig. 5, which shows the thermal efficiency of a gas turbine plant of the type illustrated in Fig. 2, as a function of the overall turbine and blower efficiency, and for different turbine admission temperatures. It will be seen that even with an overall efficiency of 53 per cent, corresponding to a turbine efficiency of 78 per cent and

[1] Stoney, Dr. G., Engineering 1937, vol. 144, p. 695, "Scientific Activities of the Late Hon. Sir Charles A. Parsons, O. M., F. R. S."

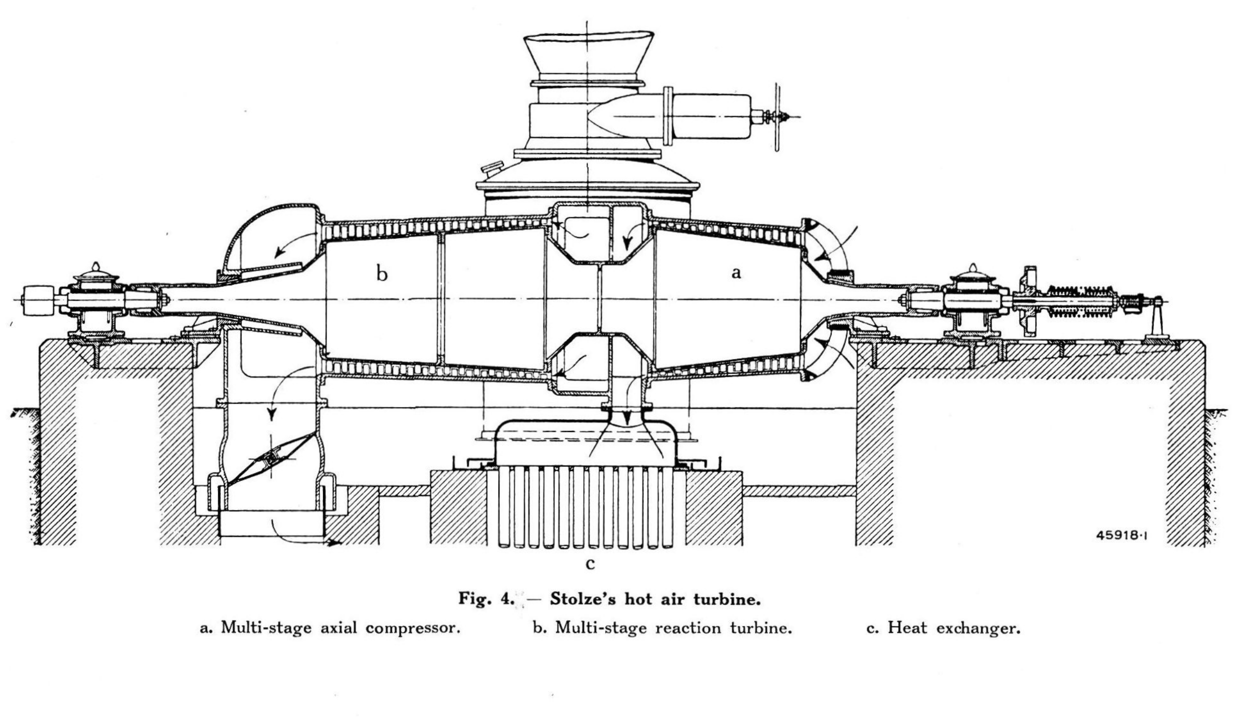



**Fig. 4. — Stolze's hot air turbine.**

a. Multi-stage axial compressor. b. Multi-stage reaction turbine. c. Heat exchanger.

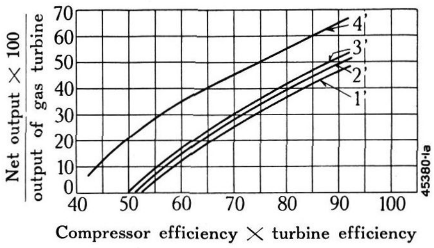



**Fig. 5. — Thermal efficiency at coupling of gas turbine as a function of the overall efficiency of compressor and turbine. For different temperatures at the turbine inlet without recuperation of exhaust heat.**

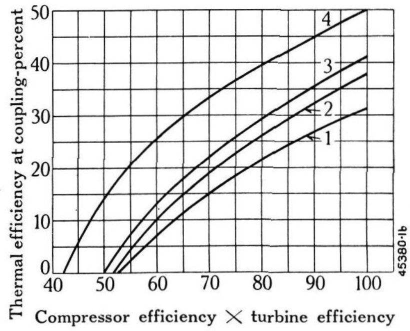
Thermal efficiency. $\frac{\text{Heat equivalent of output at coupling}}{\text{heat supplied by fuel}}$

1′ $\frac{\text{Net output} \times 100}{\text{Output of gas turbine}}$ at 1000 deg. F. before the turbine.

2′ $\frac{\text{Net output} \times 100}{\text{Output of gas turbine}}$ at 1200 deg. F. before the turbine.

3′ $\frac{\text{Net output} \times 100}{\text{Output of gas turbine}}$ at 1500 deg. F. before the turbine.

4′ $\frac{\text{Net output} \times 100}{\text{Output of gas turbine}}$ at 2200 deg. F. before the turbine.




1. Thermal efficiency at coupling of turbine at 1000 deg. F.
2. Thermal efficiency at coupling of turbine at 1200 deg. F.
3. Thermal efficiency at coupling of turbine at 1500 deg. F.
4. Thermal efficiency at coupling of turbine at 2200 deg. F.

Assumed pressure ratio in compressor:—

$P_1/P_2$. 4·2 at 1000 deg. F.
$P_1/P_2$. 6 0 at 1200 deg. F.
$P_1/P_2$. 7·2 at 1500 deg. F.
$P_1/P_2$. 12·0 at 2200 deg. F.

a compressor efficiency of 68 per cent (values unlikely to be attained at that time), the efficiency of the gas turbine cycle with a gas admission temperature of 1000 deg. F. is zero, and would not have exceeded 5 per cent with a temperature of 1500 deg. F., a value which even now is still inadmissible. The cause of this unhappy result must be sought in the great volume of compressed air required for mixing with the combustion gases to reduce the combustion temperature of approximately 3300—3600 deg. F. to the value admissible for the gas turbine blading.

To get over this difficulty, which was thought insurmountable at the time, Holzwarth, in 1905, resorted to the explosion or constant-volume turbine. In this turbine, the fuel (oil, blast furnace gas, and pulverized coal have so far been used) is fed to a closed combustion chamber filled with compressed air, and the mixture exploded, thus causing the pressure to rise to several times (approximately 4½ times) its original value. The combustion chambers, nozzles, impeller, and blades are water-cooled. The output of the compressor is only a fraction of that required for the combustion turbine, and a poor efficiency of the compressor has no longer such a disastrous effect, because (1) only a small excess of air is necessary for combustion, since water is used for cooling; and (2) the air need only be compressed to approximately one-quarter of the final explosion pressure.

These advantages are, however, accompanied by a considerable complication and increase in cost of the plant. First, a number of automatically operated valves is necessary for the combustion chamber; and second, to recover the heat given up to the cooling water, the latter must be evaporated and the steam used to actuate a turbine for driving the compressor.

In order that this turbine may be able to develop the power required by the compressor, it must run condensing, for which purpose a condenser with all its auxiliaries, and a cooling-water plant are needed. The first gas turbine of this type was built between 1906 and 1908 by Messrs. Körting of Hanover to the design of Dr. Holzwarth. On the basis of the results obtained with this experimental turbine, Messrs. Brown Boveri built and tested between 1909 and 1913, to the order of Dr. Holzwarth, a second gas turbine with a nominal rating of 1000 H.P., which, however, gave a net output of only about 200 H.P. Several further Holzwarth gas turbines were built by Messrs. Thyssen during the period 1914 to 1927, but none of them was ever put into continuous operation. In 1928, Messrs. Brown Boveri again took up the manufacture of a Holzwarth gas turbine and proposed for it a cycle which might be called the two-chamber, two-stroke cycle. The diagrammatic arrangement of the plant is shown in Fig. 6, whilst Fig. 7 is a view of the turbine on site. This unit is installed in a German steel plant, where it has been in operation with blast furnace gas since 1933. The results obtained have led to an order being placed for a larger turbine of 5000 H.P. now being built by Messrs. Brown Boveri at Mannheim.

The explosion turbine is really outside the scope of the present paper, and it is accordingly impossible to deal here more fully with the Holzwarth turbine, but it would be unfair not to acknowledge the enormous amount of work contributed during thirty years of unrelenting effort by Dr. Holzwarth and his collaborators, to the development of the gas turbine, and not to say a word of praise for their tenacity and the material sacrifices made by them in following up this idea.

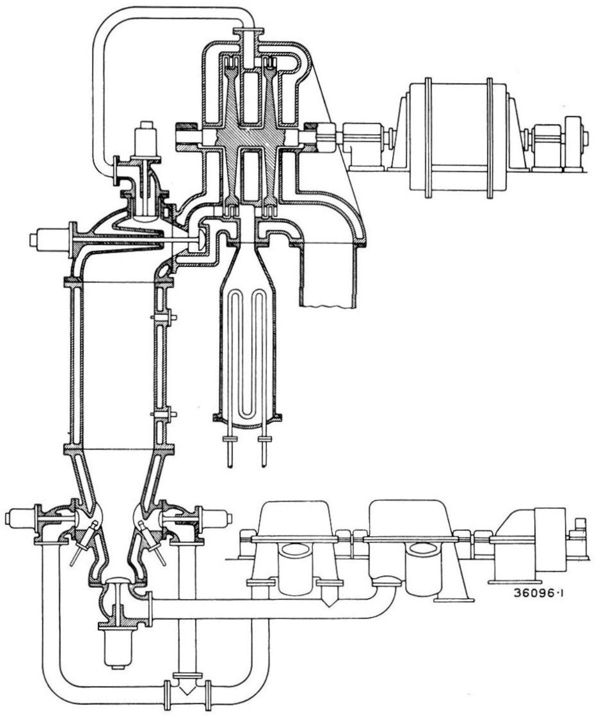



**Fig. 6. — Diagram of the Holzwarth explosion turbine. Two-chamber two-stroke design.**

The work done by Messrs. Brown Boveri in connexion with the Holzwarth gas turbine resulted in the development of the Velox boiler, the principle of which is now well known to most engineers, but which nevertheless is referred to again here (Fig. 8), because the experience gained with it led back to the combustion turbine. The Velox steam generator is a boiler fired under pressure, the pressure being produced by a compressor driven by a gas turbine, actuated by the exhaust gases of the boiler. Part of the pressure produced in the compressor is used to maintain high gas velocities in the heat-transmitting parts of the boiler, thus ensuring high rates of heat transfer. The remainder of the pressure head is used to drive the gas turbine. This application of the gas turbine as an auxiliary rendered essential the creation of a compressor-set having a high efficiency, otherwise the exhaust gas turbine would be unable to develop the power required for driving the compressor, and the deficiency would have to be supplied by another source, which would have reflected seriously on the efficiency of the boiler.

The problem was solved by the development of a four- or five-stage reaction turbine and a ten- to twelve-stage axial compressor, the design taking into account the results of the latest research in the field of aerodynamics.

## II. PRESENT STAGE OF DEVELOPMENT OF THE COMBUSTION TURBINE.

The attainment of efficiencies of 70 per cent and over in large sets of this kind suggested that the possibilities of the combustion turbine cycle for a prime mover without the use of steam be once more investigated. Fig. 5, which has already been used to explain why former designs failed to deliver any useful output, shows that with overall efficiencies of 70—75 per cent it is possible to obtain the following net output efficiencies of the gas turbine cycle:—

| Gas temperature at turbine inlet, deg. F. | Cycle efficiency, per cent |
|---|---|
| 1000 | 15—18 |
| 1200 | 19—23 |
| 1500 | 22—26 |

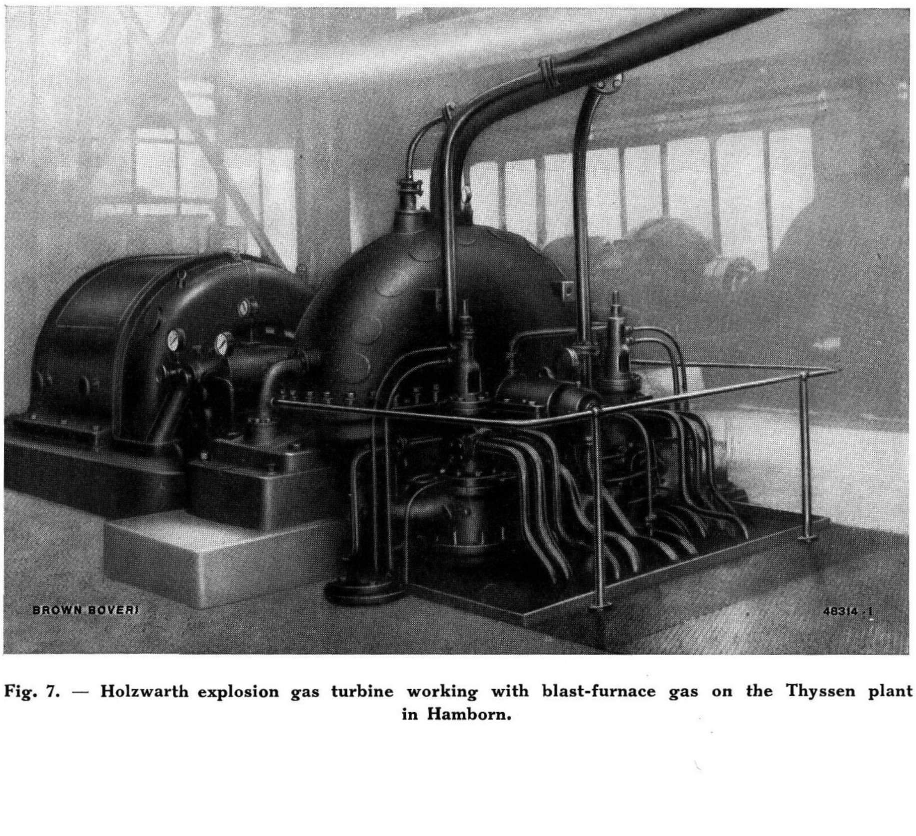



**Fig. 7. — Holzwarth explosion gas turbine working with blast-furnace gas on the Thyssen plant in Hamborn.**

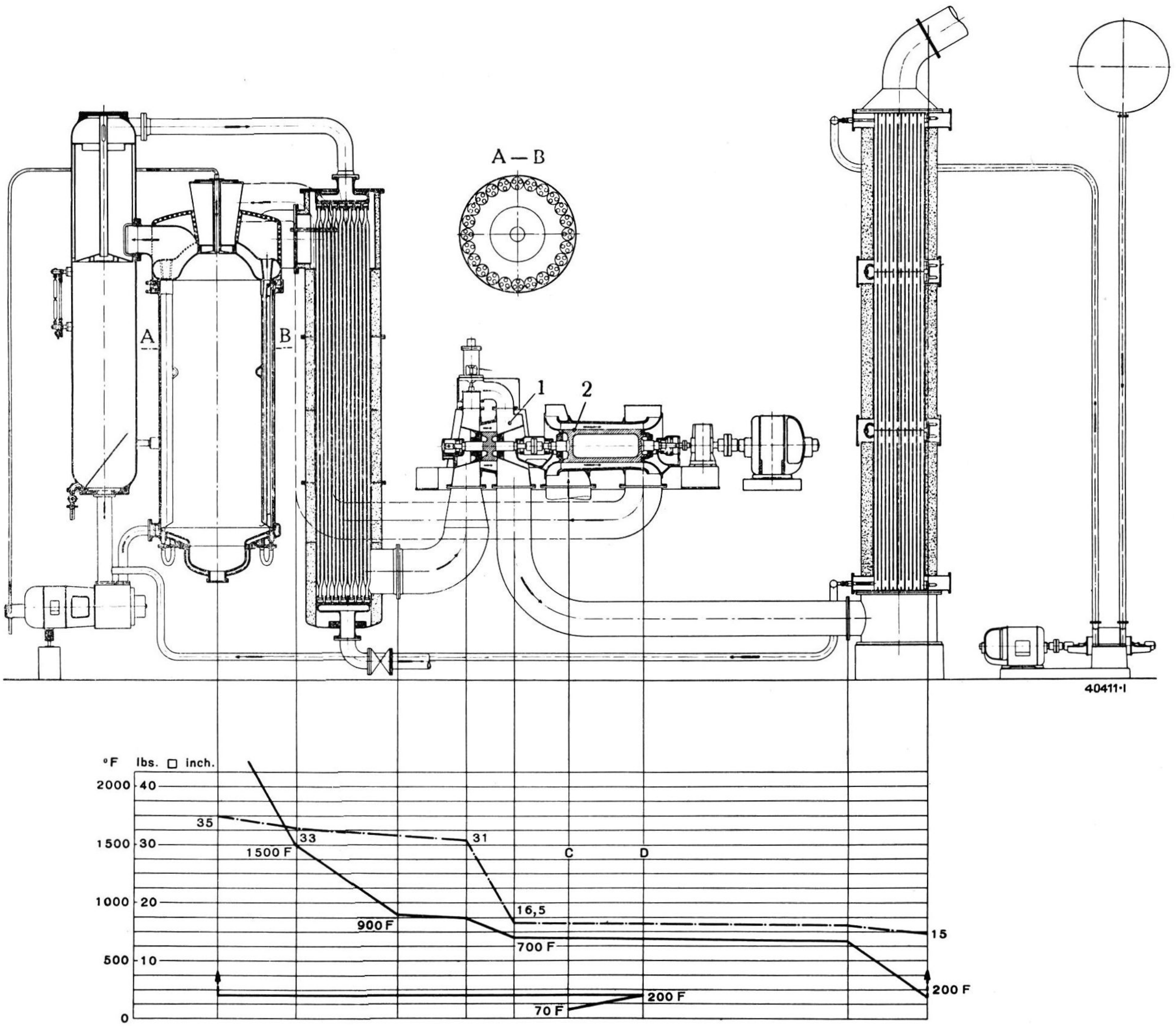



**Fig. 8. — Velox steam generator.**

A highly efficient gas-turbine-driven compressor was developed as an auxiliary.
1. Gas turbine. 2. Compressor.

The question was: What temperature could safely be employed without harm to the gas turbine blading? On the basis of experience obtained with a large number of Velox boilers, and from the operating records of hundreds of exhaust-gas turbines for Diesel engine supercharging units, a temperature of 1000 deg. F. was considered as absolutely safe for uncooled blades made of the available heat-resisting steel, due allowance being made for inevitable temporary fluctuations of temperature during governing operations.

The efficiency corresponding to this temperature, referred to the output at the coupling and the net calorific value of the fuel, and assuming an overall compressor and turbine efficiency of 73—75 per cent, readily obtainable with net outputs of 2000—8000 kW could, according to the curves in Fig. 5, be expected to attain 17—18 per cent.

## III. APPLICATIONS.

The next question was: What use could there be for a turbine of this efficiency? In raising this question, it must be remembered that before high pressures were introduced, 18 per cent was considered quite a satisfactory efficiency for a steam turbine of this size, but with high pressures and temperatures, regenerative heating of the condensate, air-preheating, and all the dodges in use in modern power stations, it is now possible to attain for such outputs coupling efficiencies approximating to 25 per cent. To attempt to compete with the modern base-load steam turbine would, therefore, be useless.

## IV. SUPERCHARGING IN CHEMICAL PROCESSES.

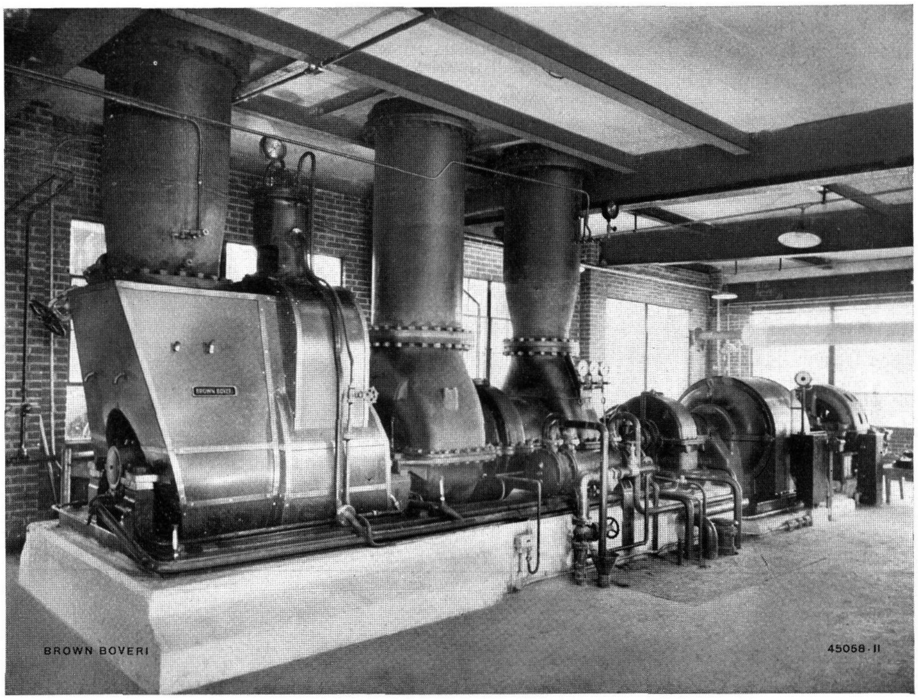



**Fig. 9. — Gas turbine supercharging plant for the Houdry oil refinery process.**
(The set has been in constant service for 2½ years in the plant of the Sun Oil Company, Marcus Hook, near Philadelphia (U. S. A.).

The demand for hot compressed air in connexion with the Houdry cracking process[1] created the first interesting field of application for the gas turbine. Many chemical processes are favourably affected by an increase in pressure, either by intensifying the action of catalysts or otherwise assisting the reaction, and resulting in an improvement in quality or an increase in quantity of the product, or by enabling smaller and therefore cheaper apparatus to be used. In many such processes, supercharging is impossible in practice, owing to the cost of compressing the air, but if the gases resulting from the process can be expanded in a gas turbine, and the power thus produced is sufficient to drive the compressor, supercharging is in many cases both feasible and economical.

This applies to the Houdry cracking process, where the power generated by the exhaust gases in the gas turbine more than suffices for supplying that taken by the compressor, the excess being used for driving a generator coupled to the set through gearing.

[1] See The Oil and Gas Journal, 1938, vol. 37, p. 40.

Although the pressure drop in the apparatus used for the cracking process results in the electrical energy obtained being considerably less than that which would be delivered by a pure gas turbine, the energy produced is sufficient to cover the power requirements of the cracking process, for which a Velox boiler supplies the necessary steam. The gas turbine has an output of 5300 kW, and the input to the compressor is 4400 kW, the excess of 900 kW being converted into electrical energy. The unit in question, shown in Fig. 9 has been in operation day and night for more than two years in the Marcus Hook Refinery of the Sun Oil Company, near Philadelphia, U. S. A. Fig. 10 shows the unit opened up, the gas turbine being on the left, and the multi-stage axial compressor on the right. The experience gained with this gas turbine resulted in orders for twelve additional units, of which two are already running, one in Italy and the other in Texas.

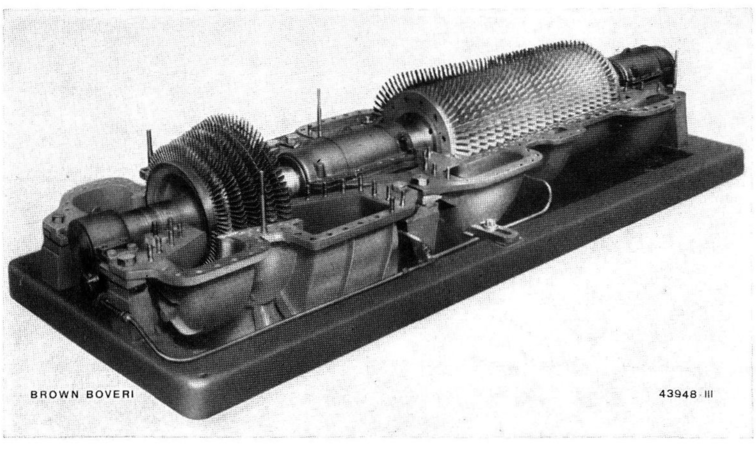



**Fig. 10. — Gas turbine compressor unit shown in Fig. 9, open.**

## V. STANDBY AND PEAK LOAD PLANTS.

Although the combustion turbine cannot yet compete with the modern steam plant for base-load purposes, it deserves serious consideration as a standby and peak load unit, on account of its special advantages. It is of simple design, cheap, light, requires little space, and, last but not least, is not dependent on any water supply. This means that the foundations and buildings are cheap, and that the plant can be installed right at the point of consumption.

These advantages are of particular importance in the case of so-called bomb-proof emergency stations, such as are now frequently built to meet the power requirements of key industries in time of war, and for the maintenance of the essential services of towns (waterworks, etc.), which normally receive current over transmission lines, in the event of the power supply being interrupted. An example of this type of plant is the 4000-kW set ordered for the city of Neuchâtel some eight months ago. Fig. 11 illustrates the extraordinary simplicity of such a power station. The set operates on the principle of the diagram shown in Fig. 2. The only auxiliary plant is the separate small Diesel-driven alternator, which in the event of a complete failure of power serves to supply the starting motor of the gas turbine.

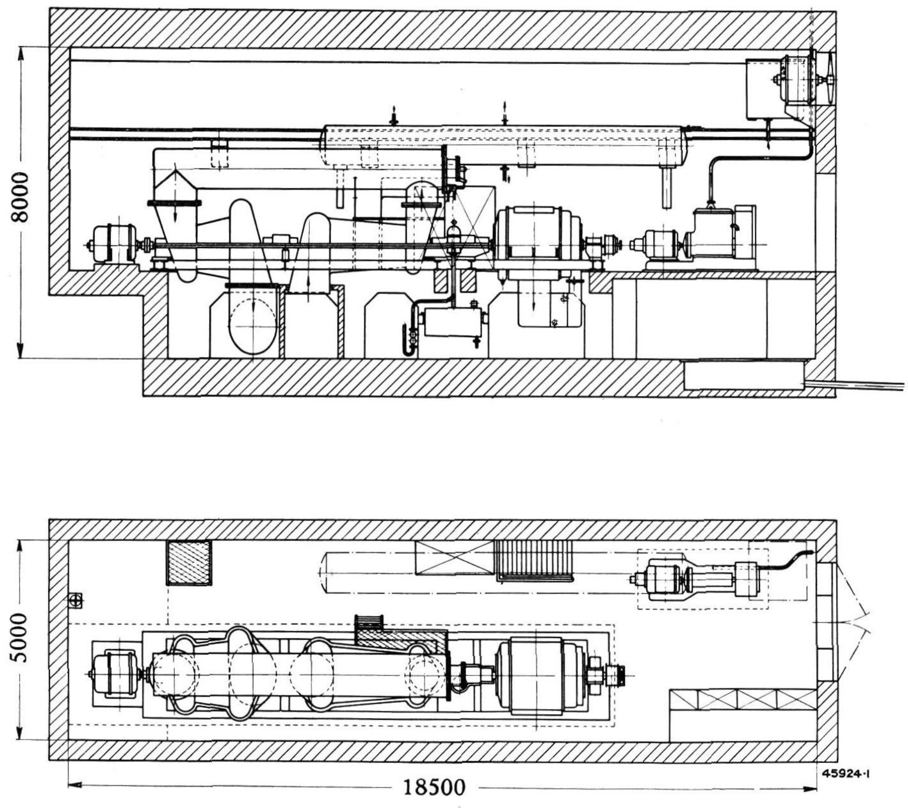



Fig. 11. — Bomb-proof gas-turbine-driven 4000 kW standby plant. City of Neuchâtel (Switzerland).

## VI. IMPROVEMENTS OF THE CYCLE.

The applications referred to above correspond to the simplest design shown in Fig. 2. An important improvement is attained by the utilization of the heat of the exhaust gases of the turbine to preheat the compressed air, as shown in Fig. 12. Here the coupling efficiencies of the gas turbine set obtainable with heat exchangers of different sizes for an output of 2000 kW, and a turbine admission temperature of 1000 deg. F. are plotted against the pressure ratio of the compressor. It will be seen from these curves that the recuperation of the heat of the exhaust gases in a heat exchanger of 5000 sq. ft. raises the cycle efficiency from 16·5 to 21 per cent, i. e. brings about an improvement of 27 per cent.

A further considerable improvement can be achieved by dividing the gas turbine into several cylinders and reheating the gas between those cylinders to the initial temperature, as shown in Fig. 13, which is drawn for a 2000-kW unit operating with an admission temperature of 1000 deg. F. and one intermediate reheat stage. As can be seen from this diagram, it is thus possible to obtain an efficiency of 19 per cent without preheating the combustion air, and $22^1/_2$ per cent with the above-mentioned preheater of 5000 sq. ft. The number of reheating stages is limited by the increased cost of the plant, as well as by the pressure drop in the piping and combustion chambers.

A further innovation which improves the efficiency at fractional loads, but not at full load, can be obtained by using two turbines, one driving the compressor only without giving any power, and the other supplying only power.

The efficiencies of the single-shaft arrangement as in Fig. 2, and the two-shaft arrangement as in Fig. 11, for

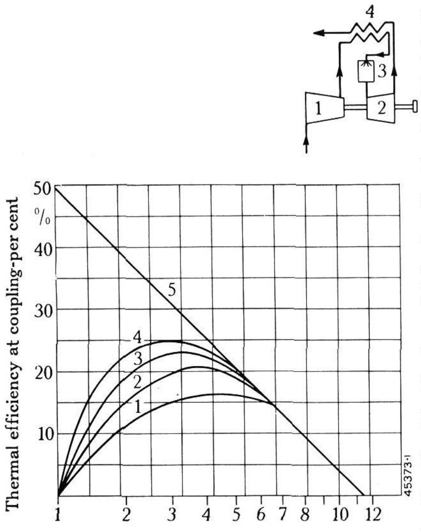



**Fig. 12. — Efficiency at coupling of 2000 kW combustion turbine.**

With temperature of 1000 deg. F. at the turbine inlet, and air preheating by exhaust gases with different sizes of preheater. Surface of heat exchanger for 2000 kW:

| | | |
|---|---|---|
| Curve 1. | 0 sq. ft. | 1. Compressor. |
| Curve 2. | 5,000 sq. ft. | 2. Gas turbine. |
| Curve 3. | 15,000 sq. ft. | 3. Combustion chamber. |
| Curve 4. | 30,000 sq. ft. | |
| Curve 5. | ∞ sq. ft. (theoretical). | 4. Heat exchanger. |

Temperature before turbine, 1000 deg. F.; temperature of intake air, 68 deg. F.; compression without cooling.

a unit of 2000 kW with air preheater, and operating with an admission temperature of 1000 deg. F., are compared in Fig. 14. The improvement at fractional loads is brought about by the fact that the turbine driving the compressor can always be operated at the most suitable speed for the compressor, independently of the speed of the turbine supplying the useful power.

These improvements can advantageously be incorporated in the following proposed applications of the combustion turbine.

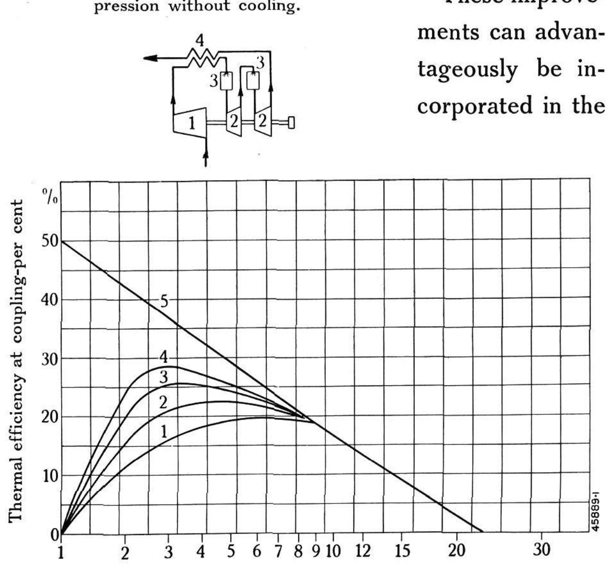



**Fig. 13. — Efficiency at coupling of a 2000 kW combustion turbine.**

With a temperature of 1000 deg. F. at the turbine inlet and air preheating by exhaust gases with different sizes of preheater, and reheating of the gases to the initial temperature between two stages. Surface of heat exchanger for 2000 kW:

| | | |
|---|---|---|
| Curve 1. | 0 sq. ft. | 1. Compressor. |
| Curve 2. | 5,000 sq. ft. | 2. Gas turbine. |
| Curve 3. | 15,000 sq. ft. | 3. Combustion chambers. |
| Curve 4. | 30,000 sq. ft. | 4. Heat exchanger. |
| Curve 5. | ∞ sq. ft. | |

Temperature before turbine 1000 deg. F.; temperature of intake air, 68 deg. F.; compression without cooling. Turbine efficiency, 86 per cent; compressor efficiency, 83 per cent.

## VII. PROPOSED NEW FIELDS OF APPLICATION.

An interesting application of the gas turbine is the gas turbine locomotive, as an alternative to the steam locomotive. It is well known that the efficiency of a steam locomotive is only of the order of 8—12 per cent, the latter figure allowing for all improvements made in the course of the last few years. The gas turbine with its coupling efficiency of 17—20 per cent finds here a promising field of application, since it should be possible to obtain an efficiency of approximately 15 per cent with mechanical transmission, or approximately 14 per cent with electrical transmission. These efficiencies do not naturally allow of competition with the Diesel engine, but the difference between the fuel consumptions of the two machines is in many cases fully compensated by the difference in the price of Diesel oil and fuel oil. Moreover, in many cases the problem of getting more power from a given size of locomotive is of greater importance than the efficiency. A noteworthy advantage compared with steam locomotives is the fact that the gas turbine locomotive needs no water,

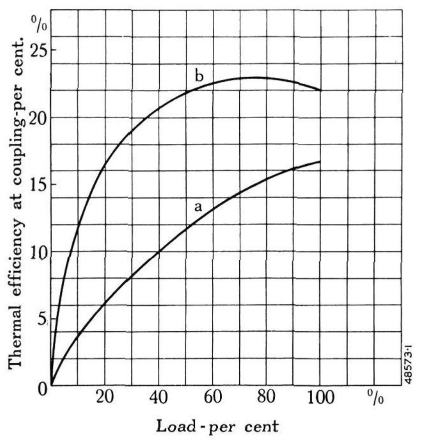



**Fig. 14. — Thermal efficiency at coupling as a function of the load.**

a. Thermal efficiency without recuperation; single-shaft arrangement with one turbine.
b. Thermal efficiency with recuperation and two-shaft arrangement with two turbines, one driving the compressor, the other driving the generator.

Temperature before turbine, 1000 deg. F.; surface of heat exchanger, 7500 sq. ft. for 2000 kW.

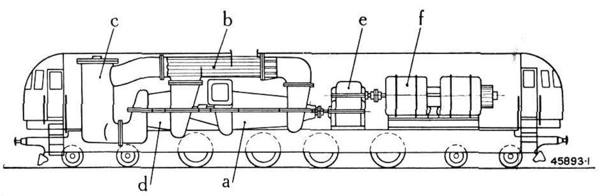
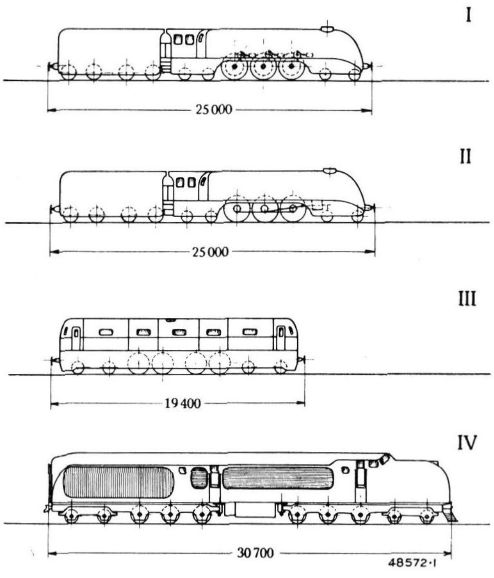



**Fig. 15. — Diagram of a gas-turbine-electric locomotive.**

| | |
|---|---|
| a. Compressor. | d. Gas turbine. |
| b. Heat exchanger. | e. Gearbox. |
| c. Combustion chamber. | f. Generator. |




| Technical data | High-pressure locomotive with reciprocating engine | Standard locomotive with reciprocating engine | Gas-turbine locomotive with electrical transmission | High-pressure locomotive with condensing plant and electrical transmission |
|---|---|---|---|---|
| Locomotive | I | II | III | IV |
| Railway system | North | North | Project | Union Pacific Railroad |
| Type | 2 $C_0$ 2 | 2 C 2 | 2 $D_0$ 2 | 2 $C_0$—$C_0$ 2 |
| Builder | Swiss Locomotive and Machine Works, Winterthur, and Alsacienne | Alsacienne | Brown Boveri | General Electric Company |
| Power at rim of wheel, H. P. | 3,000 | 3,000 | 3,000 | (2) × 2,500 |
| Thermal efficiency at rim of wheel, per cent | 12 | 10 | 17 | 14·5 |
| Service weight with tender, tons | 194 | 180 | 135 | (2) × 234 |
| Speed, m. p. h. | 75 | 75 | 85 | 100 |
| Fuel | Coal | Coal | Fuel oil | Fuel oil |
| Length including tender, feet | 81 | 81 | 63 | (2) × 100 |

**Fig. 16. — Brown Boveri gas turbine locomotive compared with steam and steam-electric locomotives.**

thus doing away with cleaning of boilers and interruptions of service resulting therefrom.

In the case of electrical transmission, only the gas turbine part is new, the generators, motors, and switchgear being adapted from the Diesel-electric locomotive without modification. The gas turbine which drives the generator through gearing can always be operated at the most suitable speed for the compressor, so that the engine gives a high efficiency at all speeds and loads. Such a locomotive is, of course, heavier and more expensive than one with mechanical transmission, but difficulties may have to be overcome with the latter, especially in case of high outputs, since it will be necessary to provide in addition to the forward turbines, reverse turbines or corresponding reversing gears.

Fig. 15 shows a gas turbine locomotive with electrical transmission; a is the blower, b an air preheater, c the combustion chamber, d the gas turbine driving the generator f through the gearing e, whilst Fig. 16 compares different designs and sizes of locomotives.

Fig. 17 shows a project for a gas-turbine-driven locomotive for an output of 3000 H. P. at the generator coupling, and an output of approximately 2500 H. P. at the wheel tread. The design of the locomotive and the main and auxiliary machines is clear from the references to the drawing.[1]

## VIII. SHIP PROPULSION.

A further promising field of application for the combustion turbine is to be found in the propulsion of ships. Although competing with modern geared turbine plants cannot yet be contemplated, the question of placing 2000—4000 H. P. combustion turbines instead of reciprocating engines on oil-fired ships already deserves full investigation by the interested parties, since efficiencies over 20 per cent can be obtained with a combustion turbine plant through utilization of the heat of the exhaust gases for preheating the air, which means an increase of approximately 20 per cent on those attained with reciprocating engines.

The prospects for merchant ships will depend upon the conditions applying, and each case must be examined individually. A very different state of things prevails, however, in the case of warships, and in particular torpedo boats or destroyers, which are usually driven by steam turbines. Here the drive is a compromise, since the top speed entails outputs ten or even twenty times that required at normal cruising speed. Consequently, the steam and oil

[1] The Swiss Federal Railways ordered a 2000 H. P. locomotive of this type from Brown, Boveri & Co., Ltd, just recently.

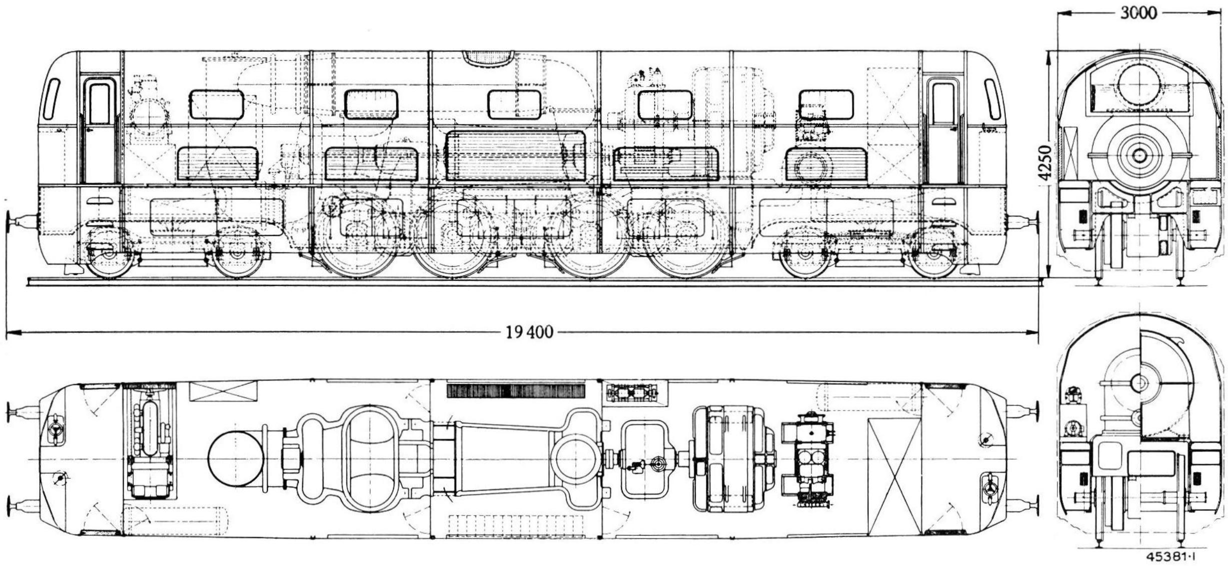



**Fig. 17. — Gas-turbine-electric-driven locomotive 3000 H. P. at generator coupling or 2500 H. P. at wheel tread.**

consumptions at both the top and cruising speeds differ considerably from those customary in the case of stationary plants. The efficiency of such a marine turbine plant at full speed is approximately 14—18 per cent, i. e. of the order of that which can be obtained with a gas turbine plant without preheating the combustion air. At cruising speed the efficiency may be even as low as 11—14 per cent. There is, therefore, no reason why a simple form of gas turbine, which in regard to the average oil consumption over the whole speed range of the ship would be equal to the average of a steam drive, should not be considered. If, however, the many advantages afforded by the gas turbine drive over the steam turbine drive are taken into account, even a slightly higher fuel oil consumption could be accepted.

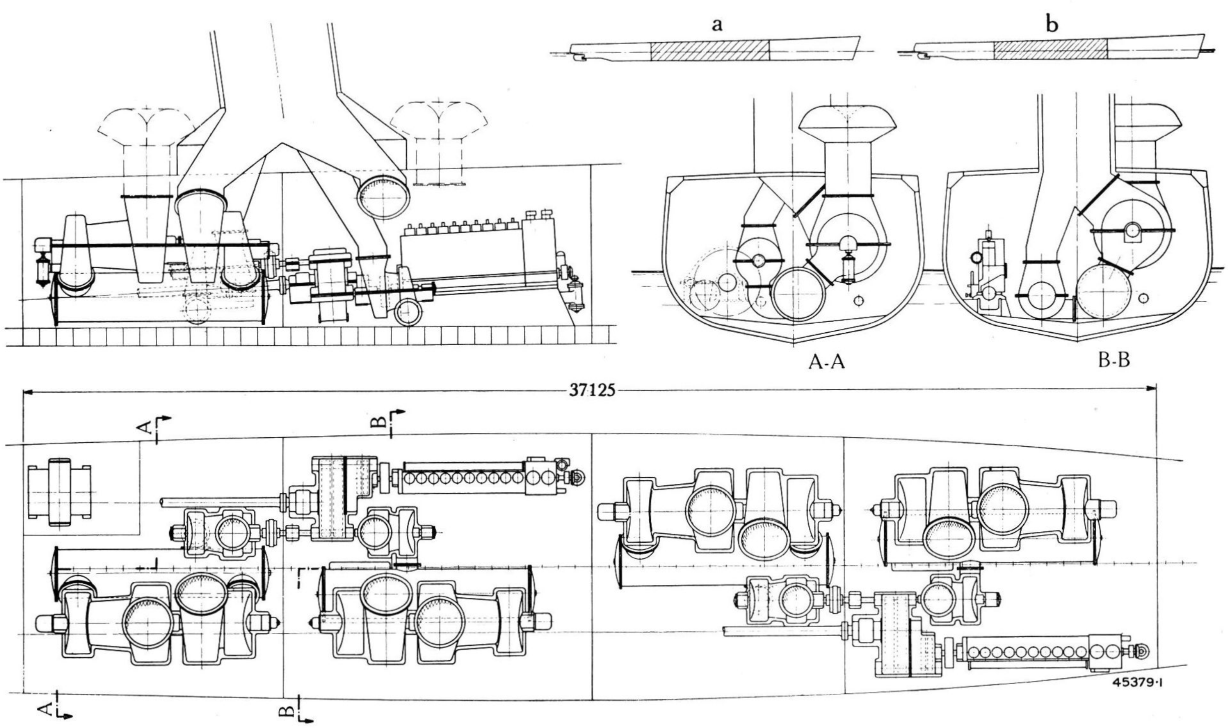



**Fig. 18. — Comparison between a (2) × 16,000 S. H. P. steam turbine-driven destroyer and a (2) × 18,000 S. H. P. combined combustion turbine- and Diesel engine-driven destroyer.**

a. (2) × 16,000 S. H. P. steam turbine: weight of machine, 30 lb. per S. H. P.; fuel consumption at cruising speed, 1·12 lb. per S. H. P.; at maximum output, 0·86 lb. per S. H. P.

b. (2) × 18,000 S. H. P. gas turbine: weight of machine, 20 lb. per S. H. P.; fuel consumption at cruising speed, 0·4 lb. per S. H. P.; at maximum output, 0·84 lb. per S. H. P.

These advantages are:—

1. The great simplification of the plant, brought about by the omission of the boiler with its feed water evaporating plant.
2. Reduced vulnerability when in action, due to the absence of large volumes of water and steam stored in boilers and steam pipes.
3. The simplification due to the omission of the condensing plant with all its auxiliaries, corrosion problems, and other troubles.
4. The rapidity with which the gas turbine plant can be brought into service, only as many seconds being required as minutes are needed for a boiler installation.
5. The fact that the plant stops and becomes cool within a few seconds of cutting off the fuel supply, because the stored energy is quickly absorbed by the compressor in pumping cold air through, and thus cooling, the combustion chamber and gas turbine.

A further appreciable improvement is obtained if Diesel engine drive is resorted to for the cruising speed. This can be done because the gas turbine is much lighter and requires less space, so that the complete plant—turbine plus Diesel engine—is still more compact and lighter than a steam turbine plant. This is shown by Fig. 18 which compares a plant comprising two 16,000 S. H. P. steam turbines with a 2×18,000 S.H.P. combustion turbine and Diesel engine plant.

## IX. WIND TUNNELS.

A comparatively small but interesting field of application is the drive of wind tunnel blowers for testing aeroplane forms. Such drives, which require a considerable amount of power (up to 5000 H.P.) for testing purposes, and, therefore for short periods only, are very unfavoured consumers of current, and are charged for energy at high rates: moreover, the periods during which power can be taken are frequently restricted. Here also the gas turbine drive can be resorted to, as, on account of its relatively low cost and the small masses which have to be heated up, it is more suitable than, for instance, a steam plant. As will be seen from Fig. 19, the turbine is located outside the tunnel and drives the compressor, whilst a second turbine, arranged in the tunnel, drives the fan through gearing.

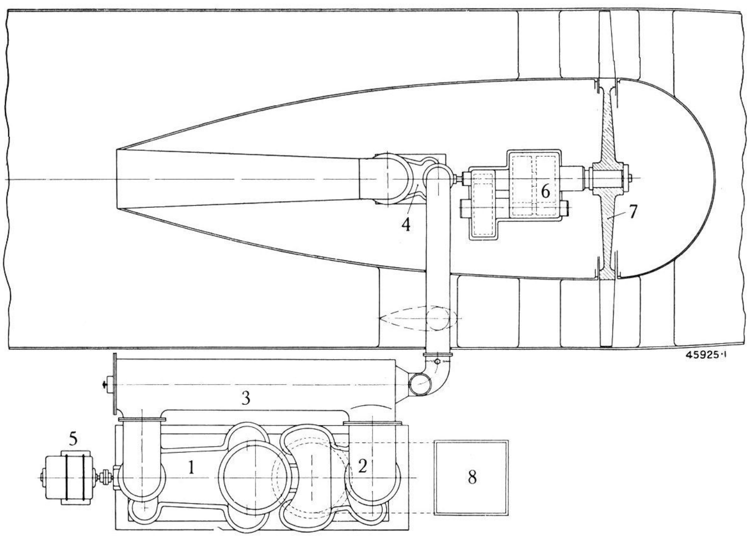



**Fig. 19. — Gas turbine plant as a prime mover for a wind tunnel blower.**

1. Axial-flow blower.
2. Gas turbine for driving item 1.
3. Combustion chamber.
4. Gas turbine for driving item 7.
5. Pony motor.
6. Gear.
7. Fan.
8. Chimney.

## X. BLAST FURNACE PLANTS.

Interesting possibilities for the application of the gas-turbine drive are offered by blast furnace plants, especially when the air supply for the blast furnace and its heating are combined with the operation of the combustion turbine by using the same compressor, together with a common combustion chamber for both purposes. This combination becomes particularly interesting if the air heaters are fired under pressure, in which case their dimensions can be considerably reduced, as shown in Fig. 20. In the top left-hand corner of Fig. 20 is shown a comparison between the dimensions of ordinary air heaters and supercharged ones; on the right-hand side are shown the air heaters with the accessory machines.

## XI. COMBINED GAS TURBINE AND STEAM PLANT.

The more important fields of application for which gas turbine plants have already been supplied or proposed have been enumerated. The potentialities of this type of prime mover are, however, far from

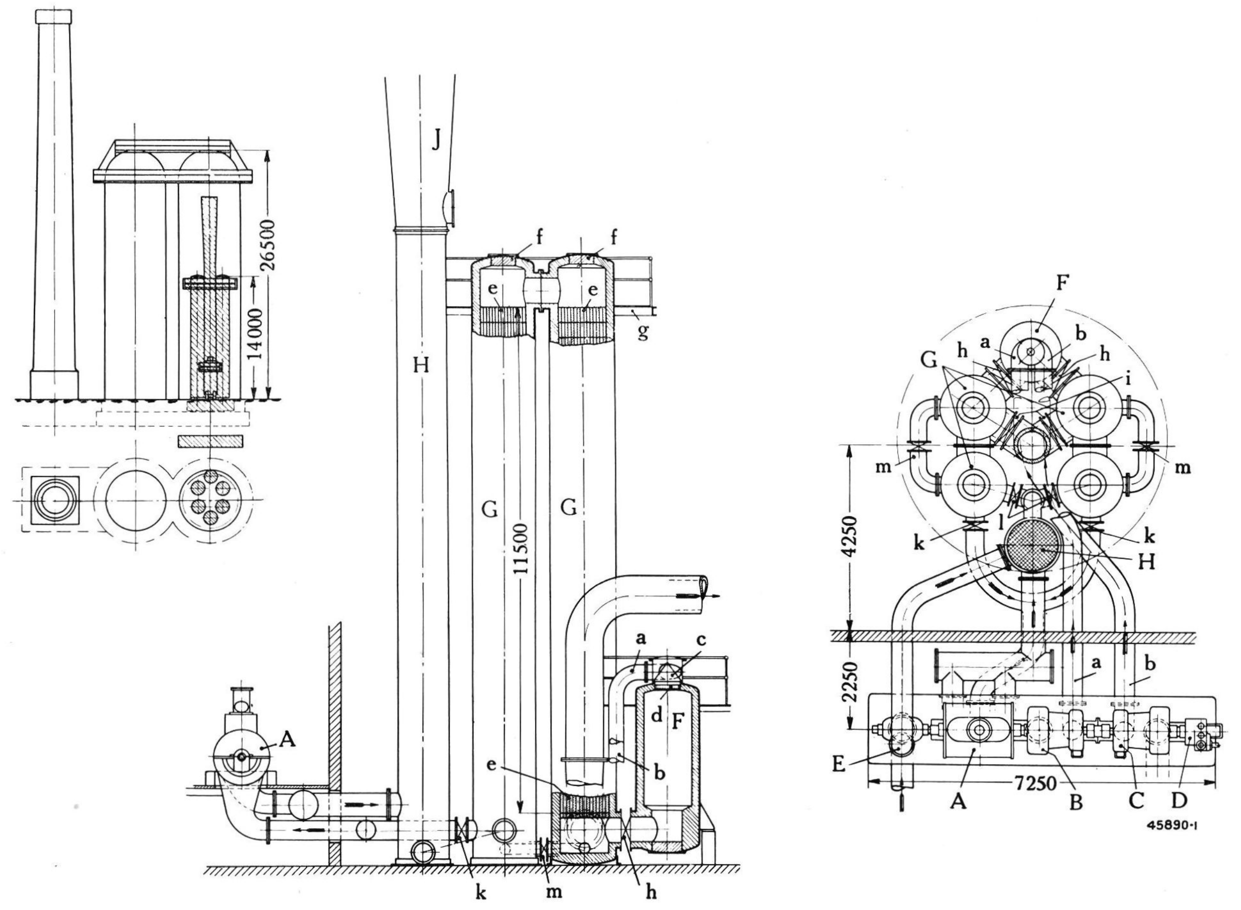



**Fig. 20. — Velox blast heater, with combustion under pressure supercharged by gas-turbine-driven air and gas compressors.**
Output 25,000 cu. ft. per min.; blast temperature 1650 deg. F.; combustion pressure 22 lb. per sq. in. gauge; duration of cycle 10 minutes.

Supercharging set:
A. Gas turbine.
B. Combustion air compressor.
C. Gas compressor.
D. Auxiliary motor.
E. Booster blower for the blast air.

Blast heater:
F. Combustion chamber.
a. Combustion air supply.
b. Gas supply.
c. Mixing chamber.
d. Burner.
G. Regenerators.
e. Refractory.
H. Air preheater for the blast (recuperation).
J. Stack.

Change-over valve gear:
h. Hot gases.
j. Hot blast.
k. Cold gases.
l. Cold blast.
m. Bypass for temperature regulator.

exhausted by this account. Mention may be made of at least one additional field, namely the combination of the combustion turbine with the utilization of the heat of the exhaust gases for producing steam in exhaust gas boilers, together with, or instead of, preheating the combustion air. In many cases where there is a demand for steam in addition to energy, the combustion turbine with exhaust gas boiler is more economical and cheaper than a high-pressure steam plant with back-pressure or extraction turbines.

## XII. GLIMPSE INTO THE FUTURE.

Fig. 5 shows that an increase in the overall efficiency of compressor and turbine from 70 to 75 per cent raises the cycle efficiency from 15 to 18 per cent, representing an improvement of 20 per cent. Every 1 per cent increase in the efficiency of the compressor or the turbine means, therefore, an improvement of 4 per cent in the cycle efficiency. Here, then, is a field from which much progress may be expected.

Fig. 5 also shows the improvement accruing from the use of higher temperatures at the gas turbine inlet. Such temperatures will be admissible either when materials of a correspondingly high creep strength are available, or when some means is devised for protecting the blading of the materials now used against the effects of such high temperatures. It can be seen from the curves that with an overall compressor and turbine efficiency of 76 per cent,

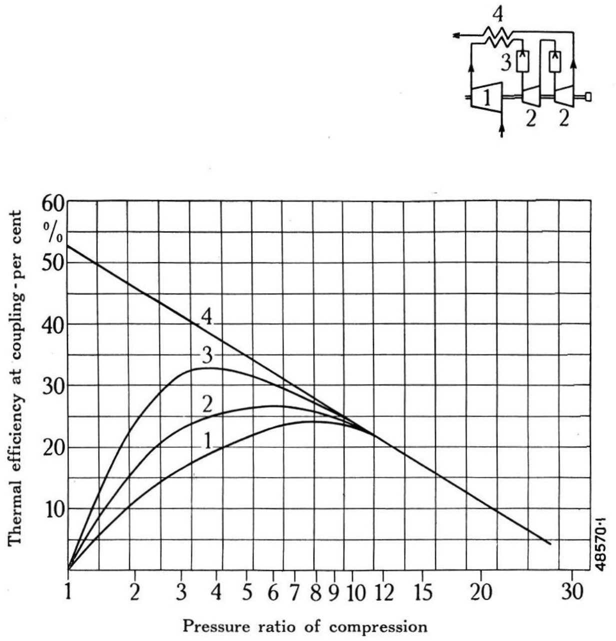



**Fig. 21. — Efficiency at coupling of a 2000 kV combustion turbine.** Temperature, 1200 deg. F. at turbine inlet; preheating of the air by exhaust gases with different sizes of preheaters; reheating of the gases to the initial temperature between two stages.

Surface of heat exchanger for 2000 kW:

Curve 1. 0 sq. ft.
Curve 2. 5,000 sq. ft.
Curve 3. 30,000 sq. ft.
Curve 4. ∞ sq. ft. (theoretical).

1. Compressor.
2. Gas turbine.
3. Combustion chambers.
4. Heat exchanger.

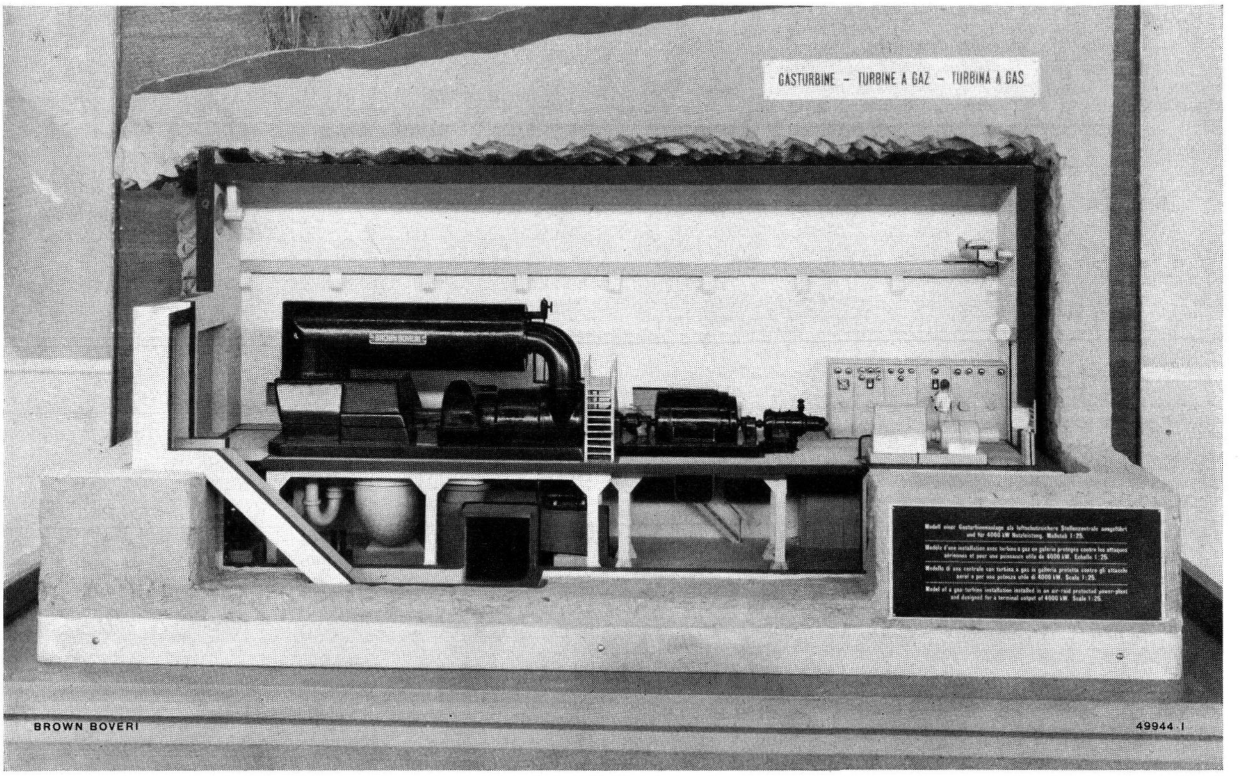
Temperature before turbine, 1200 deg. F.; temperature of intake air, 68 deg. F.; turbine efficiency, 86 per cent; compressor efficiency, 83 per cent.

increasing the temperature at the turbine inlet from 1000 to 1200 deg. F. raises the cycle efficiency from 18 to 23 per cent, representing an improvement of 28 per cent. Fig. 21, which gives the cycle efficiency for a two-cylinder gas turbine of 2000 kW with intermediate reheating and a temperature of 1200 deg. F., shows that the efficiency attains 24 per cent without air preheating, 26·5 with a preheater of 5000 sq. ft. heating surface, and 33 per cent with a preheater of 30,000 sq. ft. heating surface.

The prospects of the gas turbine, if it is possible—as the author believes it is—to raise the temperature in the near future to 1200 deg. F., can be readily appreciated. This belief is based upon the experience obtained with a number of Diesel engine supercharging units which have been in operation for a considerable time at temperatures approaching this value.

The picture which this paper has attempted to convey, of the history, present state, and future of the combustion turbine, is necessarily incomplete, but it will serve to show that the subject is a promising one, full of interesting possibilities.

*(MS 688)* *Dr. Ad. Meyer.*




**Model of a gas-turbine plant. Underground bomb-proof power station of 4000 kW output.**

# A NEW ELECTRIC GOVERNOR FOR PRIME MOVERS.

Decimal index 621.316.720.

*This article describes a new electric governor for prime movers which meets satisfactorily the most modern requirements as regards frequency and output regulation. Its design, which differs fundamentally from that of the mechanical governors used up till to-day, allows of it being supplied direct from the generator voltage, the sources of trouble inherent to other driving methods being eliminated.*

## I. THE MECHANICAL DRIVE OF THE GOVERNOR PENDULUM.

IN many turbine plants, difficulties are encountered in installing a mechanical drive for the governor pendulum. These difficulties arise from the restricted space available and from the layout, which differ from case to case. Belt drive, with its inherent unreliability, has been replaced, since some years, by mechanical transmission, wherever it was found possible so to do. Some firms took into account the danger of the governor belt breaking by taking special constructive measures to render impossible the, otherwise, inevitable running away of the turbine when the governor belt is removed.

In certain cases, belt drive had to be abandoned, because of its jerky running and the unfavourable influence thereof on the governor. Considerable improvements were introduced, however, by utilizing jointless silk belts, etc. The putting on of these latter belts and the replacement thereof, nevertheless, meant having overhung belt pulleys at both ends.

The replacement of the belt drive by a rigid mechanical one was not always easy. According to the speed and layout of the set, this kind of drive may also incorporate weak points.

It is true that a rigid mechanical drive results in an absolutely faithful transmission of the rotor movement to the speed governor pendulum and, for this reason, would seem to be the only correct kind of drive. However, this rigid connection of governor pendulum and main shaft of set frequently transmits oscillations of the set to the speed organ. With the degree of governor precision attained and, indeed, necessary to-day, it may very easily happen that these continuous angular oscillations of the governor get amplified, this notably, when the governor is not purely a speed governor but is called on to respond to acceleration, which may be advantageous for other reasons.

## II. THE ELECTRIC DRIVE OF THE GOVERNOR PENDULUM.

It will be quite obvious why turbine designers should have tried to eliminate all those defects by adopting an electrical drive of the governor pendulum. The simplest solution is to replace the belt by an electric motor fed at generator voltage, respectively frequency. The type of motors used for the purpose are rotating field units designed as synchronous, induction or reaction motors. Synchronous motors have not been found advantageous, partly for those reasons already given when discussing mechanical drives, namely the danger of the motor falling out of step. Ordinary induction motors, on the other hand, have too great a slip, so that reaction-type motors are the best for the purpose as they run with, practically, no slip and are of simple design, requiring no D.C. excitation of the rotor. The reaction motor is synchronous in character although the character of the torque curve in function of angular deviation is not exactly the same as in synchronous motors with salient poles excited by D. C. current, for which reason the danger of an amplification of pole-wheel oscillations is somewhat lower than with purely synchronous motors. In many cases, however, a sufficiently exact speed transmission to the governor pendulum can be attained with the ordinary induction motor on condition that it be of ample size so that the slip is very small, as, for example, about 0·1 to 0·2 %.

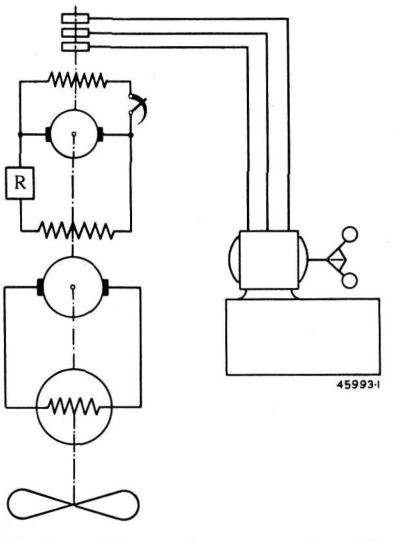



**Fig. 1. — Diagram for a generator with main and auxiliary exciters, for independent supply of the motor of the governor pendulum.**

It is usual to use an auxiliary transformer for feeding the three-phase governor pendulum motor from the generator busbars. However, it was soon found that this solution had big draw-backs for the following reasons:—

When short circuits or similar disturbances occur, resulting in considerable voltage drops across the generator terminals, the governor motor speed also drops considerably, suddenly. This means that the centrifugal turbine-governor pendulum will regulate as though the speed of the main set had dropped too low and the turbine will be regulated to full open position. If now the short circuit lasts for a certain time, there will be a danger that the increase in speed of the rotor of the main set will cause a falling out of step. This occurs when the short circuit is a three-pole one because, then, the electric power delivery is brought down to, practically, zero and the turbine drives the generator rotor with the maximum torque owing to the action of the turbine governor, as explained. As long as the short circuit persists the only braking action exercised on the set is that of the heat losses due to the current flowing in the generator and due to the short circuit set up, to which must be added, of course, the losses proper of the generator, due to the increased speed. The minimum voltage across the terminals of the generator due to the short-circuit voltage of the transformer in series with it and which may be of the order of about 10 % of the rated voltage is obviously insufficient to keep the governor-pendulum motor running. On the other hand, the damping winding of the pole wheel cannot develop any torque as long as the short circuit persists. Thus, the pole wheel may be accelerated to considerably greater speed up till the moment when the system voltage builds up again thanks to the defective section of the system being cut out or to the generator itself being cut off from the system, after which the generator voltage reappears. This last process is generally accompanied by a voltage rise across generator terminals which may be of dangerous magnitude, because, besides the much increased speed of the generator, there is full excitation by the voltage regulator, when no over-current regulator is installed.

In order to counteract these defects, certain safety measures are adopted to prevent the turbine being regulated up to a higher speed when this kind of trouble occurs. Generally, the protective apparatus consists, chiefly, of a voltage relay which is also connected to the generator terminals and which regulates the turbine to the closed or no-load position when the voltage drops to less than about 35 %.

In many cases, however, this safety measure has not been found absolutely satisfactory:—

(1) The safety apparatus did not act for certain cases of trouble, either because it was incorrectly set or because it worked defectively.

(2) The apparatus in question is generally of one-pole design which means that it cannot give complete protection at all. On the other hand, single-pole protection can be the cause of unnecessary interventions of the said apparatus, because although a single-pole short circuit may have occurred, synchronous parallel running and power exchange is always possible on the single-phase which is still sound.

(3) The protective apparatus is set for instantaneous action, because when a serious case of trouble occurs, it is certainly disadvantageous that there should be a time lag in the closing of the turbine. This instantaneous action can, however, be the cause of superfluous regulation processes which are disadvantageous to the stability of the system.

(4) The arbitrary and sudden closing down of the turbine when the voltage drop is such that the pendulum motor would be likely to trip is not really always desirable. It is true that the exchange of power between the set under consideration and the system supplied drops as the system voltage falls, but it persists, nevertheless, at a reduced figure. The arbitrary closing down of the turbine just at the critical moment of trouble signifies entire renunciation of all speed regulation of the set, so that the picking up again of those generators which may have fallen out of step is very questionable. The main factors of the pick up of the said generators and bringing them into step again is the excitation, respectively the strength of the main field in conjunction with the deviation of the speed or slip frequency. The greater the difference in speed the stronger must be the excitation supplied to bring the pole wheel into step again (the reinforcement of the excitation to this end has a limit, however, which is set by the admissible strength of the compensating currents which then flow). For example, tests have shown that a generator excitation corresponding to about 125% of the no-load voltage excitation was sufficient to bring a rotor back into step when it had fallen out of step, this when the speed of the said rotor had been first regulated to a value which deviated less than 3 % from the system frequency. This example confirms the surprisingly good resynchronization noticed in practical system operation, but also shows how important is the maintenance of speed regulation at moments of trouble. It should be remembered that speed differences are unavoidable after the cutting out

of defective sections even with perfect turbine regulation. The biggest differences between two sets after a defect is cut out is seen when one set runs under full load and the other under no load (phase advancer, for example). If the governors are set for a static of 3 %, the speed difference will also be 3 %, in this case.

These defects inherent to direct supply of the pendulum motor led to connecting the organ of the speed governor to some source of power which was not affected by what may take place on the system. This source of power is a three-phase auxiliary generator mounted on the shaft of the main generator which, when feasible, is built to give the standard industrial frequency of 50 cycles, for example, at the rated speed of the turbine shaft. With steam, turbo or hydraulic high head sets running at about 500—3000 r. p. m., there is no special difficulty encountered in using an auxiliary generator, as this corresponds to a 12- to 2-pole type of machine. When there are exceptional conditions in the plant, as regards voltage regulation and, where a separate auxiliary exciter is necessary, the latter can be utilized to form the source of three-phase current wanted. The auxiliary generator is generally necessary when a wide voltage regulation range has to be covered and where the lowest excitation current which is specified is very small, or, even, negative. This last case generally occurs for generators which are used to charge or feed extensive high-voltage transmission lines. As this auxiliary exciter is most advantageously designed for constant voltage in function of load on account of the voltage regulating conditions, it is very suitable for supplying the pendulum motor if three slip rings are put in which are connected to the armature winding.

If trouble develops on the main generator, the auxiliary exciter continues being excited, the suppression of the main field of the generator, respectively of the main exciter, is carried out on its own excitation circuit and does not affect the auxiliary exciter.

This solution which is, doubtless, superior to all other electric or mechanical drives as regards safety, meets with serious difficulties at low generator speeds.

For example, on the low-head generating sets of the power stations on the Rhine, the speed of 75 r. p. m. of the generating sets would call for auxiliary generators with 80 poles. Even if an exceptional frequency is chosen for the pendulum motor, such as 25 cycles, this would mean 40 poles, which, with regard to the low output of the machine of some hundreds of watts would mean an expensive and uneconomical design.

On the other hand, with regard to the running up of the pendulum motor in time, when the turbine is being put into service, the auxiliary exciter has to be amply dimensioned, especially when it has its own excitation, which is generally the case for the sake of safety. Fig. 2 shows the curve of the generator speed rise in conjunction with the starting curve of the pendulum motor, which, under these conditions, begins to rotate when the speed of the turbine has attained about 70 % of the rated speed. As the turbine is not set to full opening for starting up, the pendulum motor has ample time — about 1 minute — to take over the speed regulation.

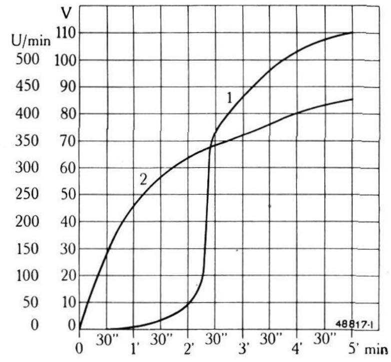



**Fig. 2. — Curve of voltage and speed of a governor pendulum drive, in function of the starting time (minutes).**

1. Voltage, V.
2. Speed, r. p. m.

If the speed rise or starting-time constant

$$T_a = \frac{GD^2}{4} \cdot \left(\frac{\pi \cdot n}{30}\right)^2 \frac{1}{P_n}$$

were smaller, the time available for the acting of the governor pendulum would be correspondingly shorter, which would result in a considerable over-speed at starting. In automatic plants, this process is especially important because in these stations, starting generally takes place with the full torque.

Apart from the constructive difficulties for the pendulum generator already described, when the speed of the main set is unfavourable, there are, often, difficulties in mounting the generator on the main shaft. In vertical shaft sets this means a prolongation of the shaft which is alway undesirable. Especially in existing plants, the putting in of the auxiliary generator is a matter of great difficulty and not possible without detracting from the accessibility of other important parts and from the dismantling facilities thereof.

## III. THE FREQUENCY REGULATOR.

Apart from the defects of the speed governors used up till now, especially as regards their drives, the knowledge which has been acquired to-day as regards the requirements of modern transmission-system regulation places new demands on governors which it is impossible for ordinary mechanical speed governors to meet. It would be outside the scope of this article to go into the details of this problem, which were exhaustively discussed at the SEV meeting in Berne on May the 1st, 1937 (see SEV Bulletin No. 22 of 1937). In order to make possible faultless collaboration of power stations working in parallel on a big network, it is necessary that frequency-output regulators, working to given laws, should govern the prime movers. The combination of a speed governor and output regulator which is, thus, required, makes it necessary to replace the mechanical speed governor by an electric frequency regulator, because the output regulator must be a purely electrical apparatus. By using a purely electrical speed, respectively frequency regulator, it becomes possible to impart to the apparatus a function dependent on the output up to a certain degree. It is only necessary to reproduce, for example, the influencing by wattless load a character which has already been imparted to the Brown Boveri quick-acting voltage regulator. However, perfect and entire solution of all the regulating duties desired can usually only be attained by separate installation of the frequency and output regulators. But here as well, the combination of two electric instruments, both being static measurement instruments, offers advantages as compared to the combination of an electric output regulator and a constantly rotating mechanical speed governor.

Apart from the requirements of system regulation, a possible uneven — and undesirable — curve of static characteristic of a set can be eliminated, as regards the external network it works on, by making use of a combined frequency and output regulator.

In full recognition of these conditions, a new electric regulator for prime movers was developed by Brown Boveri. According to what has been said, the duty to be fulfilled by this apparatus was the elimination of the defects of the drive of mechanical governors while doing away with a separate auxiliary generator and, on the other hand, the solving of the most varied regulating duties in conjunction with an influencing of the output. The fundamental design of this frequency regulator is shown in Fig. 3.

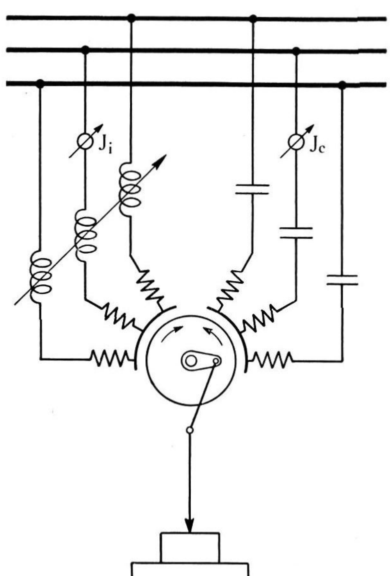



**Fig. 3. — Fundamental diagram of the three-phase frequency regulator.**

The governor has two windings which are opposed as regards sense of rotating fields. When the same current flows in both windings the torques equalize each other. A reactance is inserted in series with one winding and a capacity with the other, which demand a multiple of the voltage taken by the regulator winding proper, so that, practically, the current circuit of one winding can be considered as purely inductive and the other as purely capacitive.

In the state of equilibrium of the rotating system the following equations are valid:—

$$I_i = I_c \text{ (see Fig. 3)}$$

$$I_i = \frac{U}{L\,\omega_0};\; I_c = C U\,\omega_0$$

$$\frac{1}{L\,\omega_0} = C\,\omega_0$$

$$\omega_0 = \frac{1}{\sqrt{C L}}$$

This is the same result as for a resonance connection. Now, as the torque generated by each winding varies as the square of the current, the characteristic of the resultant torque on the spindle of the regulator is very advantageous, indeed. Fig. 4 reproduces the said torque. It will be seen that the slightest divergence of the frequency from the rated value set causes the resultant torque to increase rapidly. Further, the continuous character of the torque is apparent.

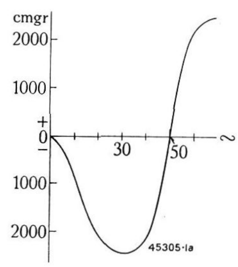



**Fig. 4. — Curve of torque in function of the frequency.**

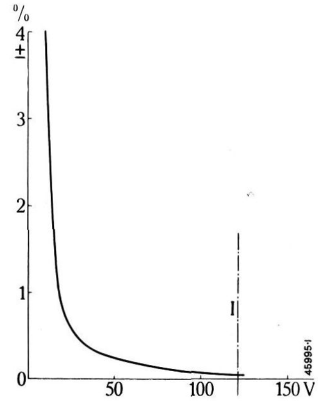



**Fig. 5. — Degree of insensitiveness in function of the voltage.**
I. Normal voltage.

The mathematical investigation of this characteristic leads to the following results:—

If the impedances of both regulator windings are disregarded, the effective torques within the range of this investigation, are as follows:—

For each system

$$M = k \cdot I^2$$

The currents in both systems are

$$I_i = \frac{U}{\omega L};\; I_c = U \cdot C \cdot \omega$$

$$M_i = k \cdot \left[\frac{U}{\omega L}\right]^2;\; M_c = k \cdot U^2 (\omega C)^2$$

The resulting torque is

$$M_{res} = M_c - M_i = k \cdot U^2 \left[ (\omega\, C)^2 - \frac{1}{(\omega\, L)^2} \right]$$

As the capacity and inductivity of the two main circuits are unchanged the constants can be enlarged to:—

$$k \cdot \frac{1}{L^2} = B;\ k \cdot C^2 = A$$

$$M_{res} = \left( A \cdot \omega^2 - B \frac{1}{\omega^2} \right) U^2.$$

In the state of equilibrium of both the effective torques we have:—

$$M_{res} = 0;\ \omega = \omega_0$$

$$M_{res} = U^2 \left( A \cdot \omega_0^2 - B \frac{1}{\omega_0^2} \right) = 0.$$

This condition is fulfilled, when

$$A\, \omega_0^2 = B \frac{1}{\omega_0^2}$$

$$B = A \cdot \omega_0^4;\ \omega_0^4 = \frac{B}{A} = \frac{1}{C^2\, L^2}$$

$$\omega_0 = \frac{1}{\sqrt[4]{C^2 \cdot L^2}} = \frac{1}{\sqrt{C \cdot L}}$$

By elimination of constants B we get:—

$$M_{res} = \left( A\, \omega^2 - B \frac{1}{\omega^2} \right) U^2 = \left( A\, \omega^2 - A \frac{\omega_0^4}{\omega^2} \right) U^2$$

$$\underline{M_{res} = A\, U^2 \left( \omega^2 - \frac{\omega_0^4}{\omega^2} \right)}$$

It must be remembered, however, that the voltage is not constant but is a function of $\omega$. The dependence of the voltage on the speed differs, according to the connection of the excitation cascade. In the most unfavourable case, that is to say with a big time constant of the main excitation circuit, this excitation current is constant and, therefore, the generator voltage is

$U = b \cdot \omega$ with $b =$ constant of the machine.

The above formulae can thus be given the following final forms:—

$$M_{res} = A \cdot b^2 \cdot \omega^2 \left( \omega^2 - \frac{\omega_0^4}{\omega^2} \right)$$

$$= A \cdot b^2 \cdot (\omega^4 - \omega_0^4)$$

$$\underline{\frac{d\,M}{d\,\omega} = 4 \cdot A \cdot b^2 \cdot \omega^3}$$

To this we may compare the analogous function of the mechanical pendulum taken from the equation

$$K = m \cdot r \cdot \omega^2$$

$$\underline{\frac{d\,K}{d\,\omega} = 2 \cdot m \cdot r \cdot \omega.}$$

The gradient of the acting precision of the frequency or mechanical regulator in function of the speed is much more marked in the case of the electric regulator.

The setting of the frequency is made by altering the inductivity. A choke organ is used for this. Every position of the mobile interior armature of the choke corresponds to a determined frequency value. The choke allows of a close and smooth setting and has no contacts at all. The utilization of the three-phase connection shown has the advantage that the regulator continues to govern satisfactorily under a single-phase short circuit, because a very small component of the voltage of the defective phase suffices to maintain the regulating capacity of the regulator. In the case of a three-phase short circuit the speed regulation is maintained down to about 10 % of the rated voltage, with, of course, lower precision, which, nevertheless, is still about ± 3 % (Fig. 5). Thus, in the case of short circuits on the high-voltage side of the generator-transformer sets, the command over the speed is, practically, assured in all cases and the undesirable arbitrary closing down of the turbine when trouble arises on the system is eliminated.

By means of an exceptionally simple measure, the *frequency regulator itself* is made to close the turbine down in the case of a complete three-phase voltage drop. Contrary to the electric safety devices with voltage-drop relays, closing coils, etc. used on mechanical governors with motor drive of the governor pendulum, the closing of the turbine is effected with the new regulator, *without any additional element.* As the regulator has neither counter-weights nor main spring, it is of fundamentally astatic type. It suffices to generate an additional torque remaining constant over the whole range and acting in the closing sense, which torque acts on the rotating system of the regulator. This torque consists, for example, of a weight that need only be so heavy that it exercises the necessary force to close the servo-governor valve. There is, also, an adjustable damping device to allow of setting the closing time when there is a complete breakdown of the three-phase voltage.

## IV. THE ELECTRIC ISOCHRONOUS REGULATOR.

Fig. 6 shows the way the regulation acts. The capacity is a fixed quantity, while the choke can be regulated, by displacement of its armature with relation to the fixed stator. The armature is flexibly connected to the regulating spindle that is to say to the guide blade (or nozzle) governing organ of the turbine. In the diagram, the piston of the damper is connected to the choke and the cylinder of the damper to the rod. The damper piston of the choke armature

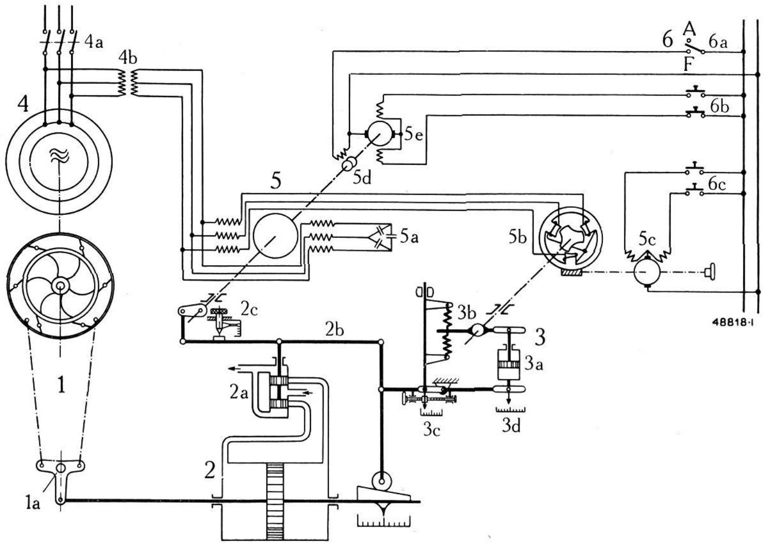



**Fig. 6. — Fundamental diagram of the isochronous regulator.**

1. Turbine.
   1a. Regulator spindle.
2. Servo-motor.
   2a. Control valve.
   2b. Valve recall.
   2c. Travel (output) limitation.
3. Isochronous recall.
   3a. Oil damping.
   3b. Recall.
   3c. Setting of continuous static.
   3d. Setting of momentary static.
4. Generator.
   4a. Main circuit breaker.
   4b. Voltage transformer.
5. Primary regulator.
   5a. Condenser.
   5b. Rotating choke.
   5c. Speed (output) displacement motor.
   5d. Electro-magnetic coupling.
   5e. Remote control motor.
6. Remote control.
   6a. Change-over switch.
   A. Automatic.
   F. Remote control.
   6b. Direct remote control.
   6c. Speed changing.

is also coupled through recall springs to a support point dependent on the position of the governing organ of the turbine. The travel of the supporting point and that of the damper cylinder are in a fixed, although adjustable, relationship to the travel of the governing organ of the turbine. The travel of the cylinder determines the momentary static of the regulator while the travel of the supporting point of the recall device determines the constant static of the regulator. Either can be set as desired and independently. A regulating process, as, for example, a cutting out process takes place in the following manner. The position indicated on the diagram for the control organ and of the choke corresponds to about half-rated load. The frequency rise which occurs when the cut out takes place causes the immediate intervention of the frequency regulator. The servo-motor displaces the governing organ of the turbine in the closing sense. The rapid movement of the rod gives the damping device (3a) no time to allow the passage of oil and the said device, therefore, acts as a rigid coupling; for that reason the armature of the choke will be considerably displaced and in the sense

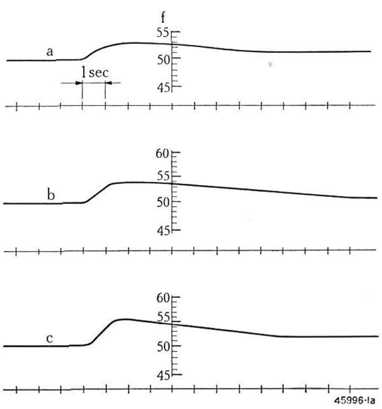



**Fig. 7. — Cutting-out diagram.**

a. About 30 % rated output, static coefficient 3 %.
b. About 45% rated output, static coefficient 3 %.
c. About 60 % rated output, static coefficient 3 %.

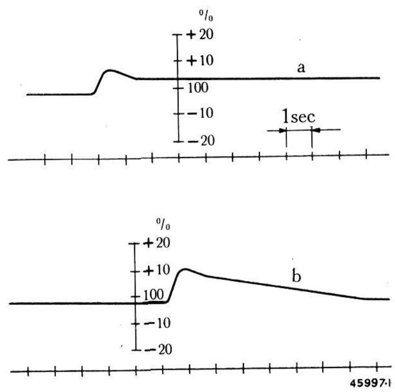



**Fig. 8. — Cutting out with simultaneous cutting off of the excitation.**

a. About 45 % rated output, static coefficient 8 %.
b. About 70 % rated output, static coefficient 8 %.
100 %. Rated speed ($\triangleq$ 50 cycles).

of higher frequency. This process causes a drop in the speed and, therefore, in the frequency. As soon as the latter has dropped below the value corresponding to the new position of the choke armature, the frequency regulator intervenes in counter sense, in other words it opens the turbine again. The flexible damping action acting together with the recall 3 b causes the recall in proper time of the armature of the choke and a regulation adjustment, without hunting, to the new state of complete equilibrium. The displacement of the no-load point, respectively of the taking over of load, in parallel service is effected by a rotation of the housing of the choke. Thus, an immediate reaction of the turbine is attained when the push-button is actuated, or by manual displacement of the housing.

The diagram of Fig. 7 shows the curve of generator frequency when there is a sudden cutting off of the set from the system, under various loads. Here, the regulator was set to constant static coefficient of about 3 %. The diagram shows the character of the speed variation, devoid of oscillations, and which is also that of the frequency. When the load is thrown off with simultaneous interruption of generator excitation, that is to say when the voltage across the terminals is brought down, the character of the speed curve is a similar one; but, according to the setting of the closing time proper of the frequency regulator the cutting-out speed peak is correspondingly somewhat higher (Fig. 8); in practice this is of no importance at all. The main thing is that in this case, as well, the turbine closes in time and reliably.

To the moving system of the frequency regulator there is coupled, electro-magnetically, a remote control device. The latter permits of leaving the turbine continually under automatic regulation and of starting it up by push-buttons from the control switchboard desk. A possibility is offered here of starting up without the changing over of the turbine governor to manual operation which is sometimes a rather lengthy process. At the same time, remote control allows of regulating the set to any opening desired, with the generator under no voltage. The switching in and out of this remote control can, as explained, take place at any time and without regard for any particular running condition.

## V. CHARACTERISTIC DATA OF THE FREQUENCY REGULATOR.

(1) Frequency and speed, respectively constant in function of the motive system, therefore astatic regulator.
(2) Degree of insensitivity $\pm$ 0·015 cycles measured on the main piston of the servo-motor, respectively on the governing organ of a turbine. The rated frequency at these values is 50 cycles, and, therefore, the degree of insensitiveness is $\pm$ 0·3 ‰ at rated voltage.
(3) Temperature, or heating up, error completely eliminated.
(4) The regulator functions, fundamentally, quite independently of the voltage.
(5) Adjustability: Constant static characteristic 0 to about 8 %, momentary statics up to about $\pm$ 20 %.
(6) Power input about 300 VA three-phase supply.
(7) Influence of the shape of the curve:— as Ferraris systems are used as measuring organs, the regulator reacts to the r. m. s. values of the A. C. currents of both excitation systems. In exceptional cases, when higher harmonics with exceptionally big amplitude with regard to the basic wave appear, a slight increase of torque in the capacitive measurement system may occur. Now as the deformation of the basic wave, that is the amplitude of the higher harmonics, generally increase with the load (for example in mutator operation) the only result is that the statics set for on the regulator is slightly influenced. This behaviour, however, is favourable to parallel operation in the case of sets loaded in this manner. The supply of the regulator by a deformed voltage curve corresponding to the oscillogram of Fig. 9, does not affect the regulator in any way at all; further, the frequency value set to does not suffer any displacement as compared to supply with a purely sine-shaped voltage curve.
(8) Speed:— is determined by the closing and opening time of the servo-motor.

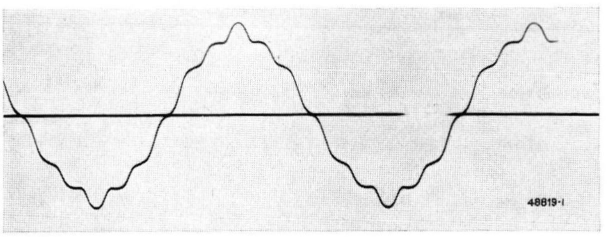



**Fig. 9. — Oscillographic record of a deformed voltage curve.**
No-load voltage curve at 120 V

| Higher harmonics | 1. | 3. | 5. | 7. | 9. | 11. |
|---|---|---|---|---|---|---|
| % | 1·00 | 1·35 | 3·97 | 1·50 | 0·73 | 6·5 |

## VI. BEHAVIOUR OF THE REGULATOR WHEN TROUBLE OCCURS ON THE SYSTEM.

### 1. *Three-phase short circuit.*

When a three-phase breakdown of the voltage occurs below about 8 % the frequency regulator initiates the closing down of the turbine without any auxiliary apparatus. This closing-down time can be set between about 30 to 3 seconds. Voltage breakdowns to values above 10 % cause regulation to a somewhat lower frequency, respectively the setting to a lower output. This property increases the stability of the network in cases of trouble, because when the network voltage drops the amount of power which can be exchanged in the network also drops.

### 2. *Single-phase short circuit.*

The closing down of the turbine only takes place, here, for phase voltage breakdowns below about 5 %. For superior values, the regulator takes over frequency control again and this with increasing exactitude as the voltage increases. As there is usually a transformer connected in front of the generator, the short-circuit voltage of which is above about 8 %, the automatic closing down of the turbine will only take place in exceptional cases when there is a voltage breakdown due to troubles on the network.

### 3. *Behaviour of the regulator with generators which have fallen out of step.*

In the case of the rotors of two generators $M_1$ and $M_2$, which have fallen out of step (see Fig. 10) and which are connected together through a line impedance $Z_L$, the frequencies $f_1$ and $f_2$ corresponding to the speed of the rotors make themselves apparent at each point of the connecting line. A frequency dominating at any point is determined solely by the relative value of the effective voltage amplitude of both frequencies at the said point. The frequency is all the more marked the higher the voltage. This will show how the frequency is represented on the line section. Across the generator

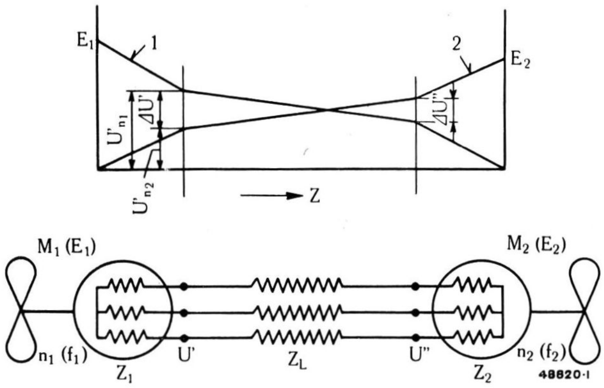



**Fig. 10. — Diagrammatic representation of the behaviour of the voltage in the case of generators falling out of step.**
1. Characteristic of voltage of frequency $f_1$.
2. Characteristic of voltage of frequency $f_2$.

terminals it corresponds all the more clearly to the value determined by the speed of the rotor, the higher the voltage (excitation) and line impedance are. The theory shows that the superimposition method is the simplest one of estimating quantitatively. In Fig. 10, $Z_1$ and $Z_2$ are the impedances of both generators, $Z_L$ is that of the line, $E_1$ is the no-load terminal voltage $M_1$, $E_2$ no-load terminal voltage $M_2$. Assuming supply from one end of the whole circuit by $M_2$ the terminal voltage $U'_{n2}$ is:

$$U'_{n2} = E_2 \frac{Z_1}{Z_1 + Z_2 + Z_L}$$

Assuming supply from $M_1$:

$$U'_{n1} = E_1 \frac{Z_2 + Z_L}{Z_1 + Z_2 + Z_L}$$

The effective voltage amplitude of the frequency $f_1$ across terminals of $M_1$ is

$$\Delta U' = U'_{n1} - U'_{n2}$$

As long as $\Delta U'$ shows positive values the frequency appearing at the generator terminals corresponds to the speed of the rotor. It is only when the generator is completely unexcited that the frequency of the network appears clearly at its terminals. In this case, however, the generator is cut off from the network, in any case, through the action of over-current or other protective devices. After the unit has been cut off, the turbine is closed down by the action of the regulator.

The above observations will make it clear how extremely important is the behaviour of the frequency actuating organ with relation to the available voltage amplitude of the basic frequency of the generator to be regulated.

The property, already described, of the electric regulator of doing duty down to about 10 % of the rated voltage is the deciding factor in allowing the regulator to be supplied straight from the main generator voltage or through auxiliary transformers.

The other property of the regulator to regulate to lower frequency when a drop in voltage is taking place, also tends to stabilize regulation when generators fall out of step. For this special case a more detailed description will be published later.

As the new regulator works, practically, without wattless current, its utilization, also in conjunction with a separate auxiliary generator, offers advantages, because the latter can be smaller than would be necessary if it had to supply a governor pendulum motor.

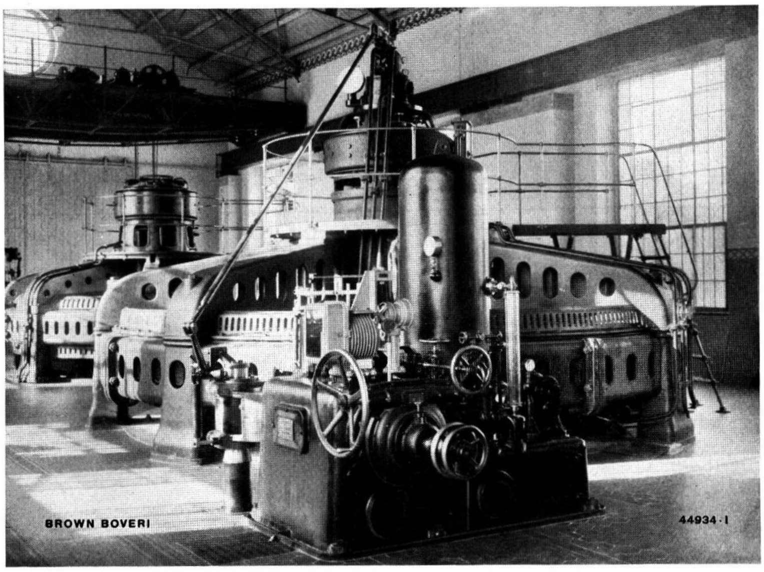



**Fig. 11. — Turbine governor of Messrs. Bell & Co., Kriens (Switzerland) with built on frequency regulator of Brown Boveri type, for the control of a Kaplan turbine belonging to the Electricity Works of the town of Aarau (Switzerland).**

APPENDIX.

- $GD^2$. Flywheel effect expressed in t $m^2$.
- n. r. p. m.
- $P_n$. Rated output in kW.
- $I_i$. Inductive current.
- $I_c$. Capacitive current.
- U. Terminal voltage.
- L. Inductivity of choke.
- C. Capacity.
- $\omega_0$. Frequency set.
- M. Torque.
- k. Regulator constant.
- $M_i$. Torque of inductivity.
- $M_c$. Torque of capacity.
- $M_{res}$. Resultant torque.
- $\omega$. Circuit frequency.
- m. Mass.
- K. Force.
- r. Radius.

*(MS 653)* *R. Keller. (Mo.)*

# NEW UNIFIED SERIES OF MACHINES OF MEDIUM POWER TYPES Mi, Gi, Wi WITH ROLLER BEARINGS.

Decimal index 621.313.333.

*The three machine series, Mi, Gi and Wi have all got the same fundamental characteristics of design and construction and they correspond very closely one to another as regards external shape and overall dimensions. The chief advantages of these series are:— the fulfilment of various special requirements from the mechanical and electrical point of view and also the simplification and unification of individual parts along with the possibility of quick delivery.*

## I. GENERAL NOTES.

THE new medium-power induction motors of Type Mi (Figs. 1 and 1 a) have been described in The Brown Boveri Review of November 1938. The present article gives some information on the methodical extension of this work of adaptation to other classes of machines of about the same power range, especially to D.C. machines of Type Gi and to A. C. generators of Type Wi; some characteristic features of the said machines are, also, gone into.

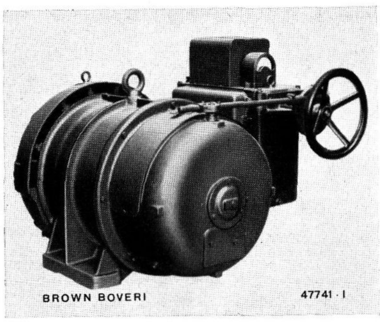



**Fig. 1. — Standard three-phase induction motor with centrifugal starter, 170 kW, 3000 V, 1000 r. p. m., 50 cycles.**

The three series Mi, Gi, Wi form a whole. Not only have they all the same cooling system and the same fundamental characteristics of design and construction but they correspond approximately as regards external shape and overall dimensions. A great number of their individual parts are interchangeable while identical additional parts can be used on them in order to give them certain protective qualities or to impart to them special features (compare Figs. 1, 4 and 11, for example).

The difficult task of combining machines of fundamentally different classes, differing in electric construction and method of operation was only possible of fulfilment by making the design a common one to all, from the beginning. This, in turn, demanded the sacrifice of former models, machine tools, gauges and manufacturing devices and demanded long study and development work. To take one case in point, only, the adaptation for all machines of *one, single* cooling system which had been found to offer a maximum of advantages and to make it suitable to the different internal shapes of the machines (salient poles on stator only in Gi type, in rotor only in Wi type, neither in stator nor rotor in Mi type) demanded careful investigation and many tests, before quite satisfactory results could be attained, while holding rigidly to the aforesaid uniform cooling system.

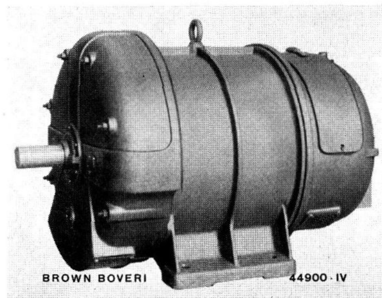



**Fig. 1 a. — Three-phase induction motor with built-on rotor starter, 75 kW, 380 V, 750 r. p. m., 50 cycles.**

Results have entirely justified the sacrifices made, by the great simplification attained in manufacture and by the facilities offered as regards stock holding, this for a very extensive field of manufacture. As regards the client, this means quick deliveries and rapid adaptation to a variety of special needs. The client also gets the benefit of unified replacement parts and spares. Further, for groups of motors the flow of cooling air is such that the hot air expelled from one motor does not affect its neighbour and

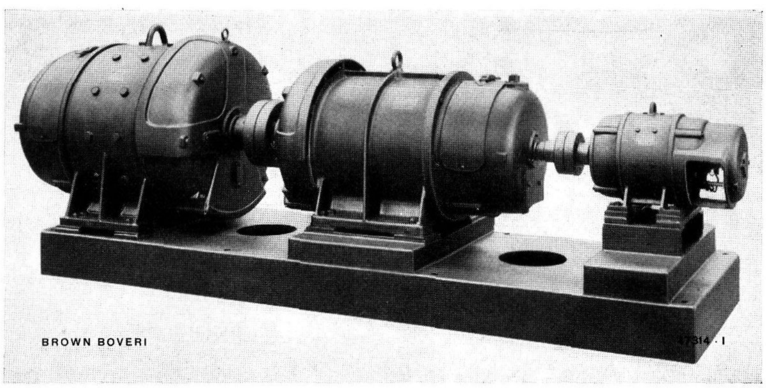



**Fig. 2. — Converter set, 1460 r. p. m., composed of :—**
On left :— D. C. generator 65 kW, 370 V.
In centre :— Three-phase induction motor 80 kW, 500 V.
On right :— Exciter 5·5 kW, 320 V.

the suitable laying of the connection leads is made easier (compare Figs. 2 and 3).

By their rigid adherence to ISA standards, which are making such rapid progress internationally, the machines of the i class are a guarantee that no difficulty will be countered when combining them to machines to be driven or to parts coming from some other source.

After these general remarks, we give some closer details on the Gi and Wi series.

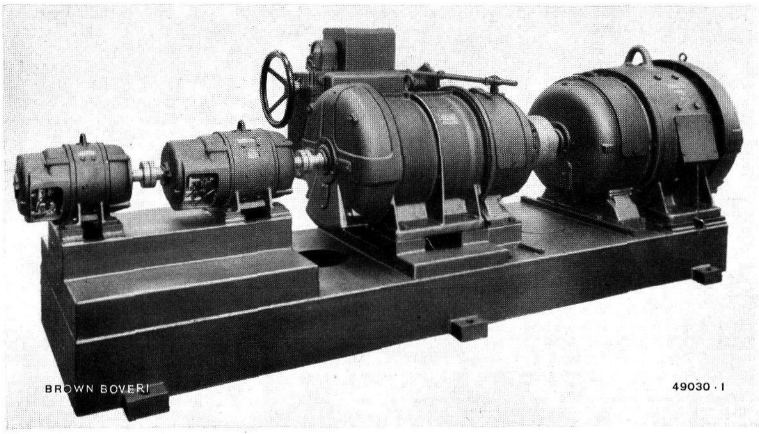



**Fig. 3. — Converter set, 1450 r.p.m. composed of:—**

On right:— D. C. generator 145 kW, 460 V, with semi-pipe ventilation, coupled to a three-phase induction motor 80 kW, 500 V, with semi-pipe ventilation.

On left :— Main exciter 3·5 kW, 75 V, coupled to an auxiliary exciter 12 V, 20 A.

## II. MEDIUM-POWER D. C. MACHINES OF TYPE Gi.

This series covers a power range of about 40 to 180 kW, referred to generators running at 1000 r.p.m. Externally, the series only differs from that of the Mi motors by the stator housing, being of cast steel in the Gi machines (Figs. 4 and 1). All these machines are suitable for operating as motors or as generators.

Fig. 4 shows a new D. C. standard machine, while Fig. 5 shows the rotor belonging thereto. In its standard design, this machine is well protected against drip water, foreign bodies and contact with the interior parts. The big terminal box, located somewhat inclined over the middle of the machine, makes connections easy to carry out, it is also easy to change connections on the terminals. If required, the leads can be brought in through sealing glands. According to needs — for example drive by a Diesel engine or, sometimes, when the machine is to form part of a converter set, it can be delivered with one bearing and a flanged shaft (ISA), see Figs. 6 and 7.

Fig. 8 shows a new D.C. generator with external slip-rings for voltage divider, as is sometimes called for in the upper range of machine voltages for connecting to lighting circuits, etc.

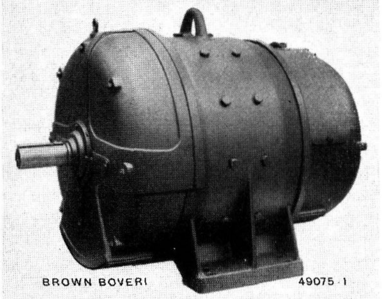



**Fig. 4. — Standard D. C. motor 180 kW, 230 V, 1000 r. p. m.**

Apart from this, essentially the same derivations of the standard design of the Gi machines can be formed, with the help of the same additional parts, as are used on the Mi machines. Thus, Fig. 9 shows a D. C. splash-proof motor and Fig. 10 a generator of semi pipe-ventilated and splash-proof type.

From the electrical point of view, as well, the new D. C. machines are very adaptable to the most varied kinds of service, as for example, varying voltage, speed variation, momentary increase in speed, load surges, etc.

Simply by changing over a contact piece on the terminals the sense of rotation can be reversed, at any time.

Further, the Gi machines can be built to run in different inclined positions, such as occur on board ship and can, indeed, be made suitable to all conditions met with on vessels. When the machine has to be vertically mounted, the standard bearing flange is generally utilized for drive below. But other kinds of vertical mounting can also be arranged for (drive above or suspended mounting with drive below or above) or else flanged mounting with a horizontal or inclined position of the shaft, according to needs.

The internal design takes account of wireless-trouble protection and condensers to this end can be lodged in the machine.

In short, nearly every possible requirement, electrically or mechanically, can be met by the new Gi machines.

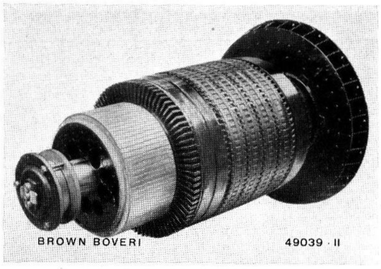



**Fig. 5. — Rotor of standard D. C. motor, 180 kW, 230 V, 1000 r. p. m.**

## III. MEDIUM-POWER A. C. GENERATORS OF TYPE Wi.

The Wi generators differ externally from the Mi motors with squirrel-cage rotor, essentially, only in the former having a built-on exciter. When the exciter is separately mounted, the external shape is the same (compare Fig. 12 with 1 a); stator and bearing flanges are the same for both classes of machine.

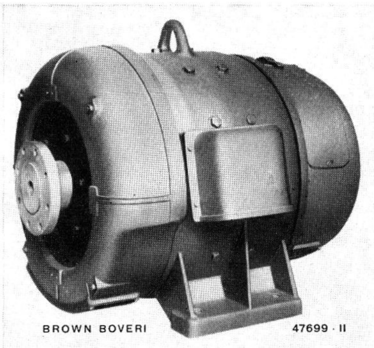



**Fig. 6. — D. C. generator with one bearing and shaft flange, 85 kW, 460 V, 500 r. p. m.**

Fig. 11 shows a standard 4-pole three-phase A. C. generator, Fig. 13 the 4-pole rotor of a somewhat bigger generator. The medium-power Wi generators cover a power range of about 30 to 525 kVA and a voltage range of about 110 to 6600 V; the normal speeds are between 500 and 1500 r. p. m.

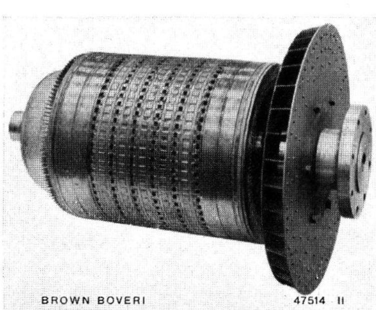



**Fig. 7. — Rotor of D. C. generator with one bearing and shaft flange, 85 kW, 460 V, 500 r. p. m.**

From the constructive point of view, the remarks made on the subject of the Mi and Gi machines are valid. These machines, as well, are drip-water proof in standard design (Fig. 11). Although, as a result of their being in enclosed rooms, it rarely happens that special protection is required, there is nothing to prevent special protection being provided with the help of the standard additional parts mentioned under chapter II.

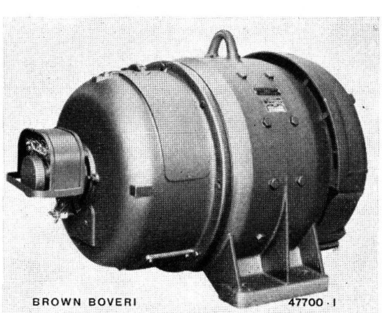



**Fig. 8. — D. C. generator with external slip-rings for connecting to a voltage divider, 85 kW, 460 V, 500 r. p. m.**

The reinforced design with one bearing for connection through a shaft flange, which is frequently called for in the case of Diesel-engine drive, is also built as a standard design (Figs. 14 and 15). The shield on the driving side is made in two parts.

The vertical design, also, being used increasingly, for belt drive as well, is built as a standard type with special bearing-shield foot, allowing of that displacement requisite to belt tightening. Fig. 16 shows a machine of this kind.

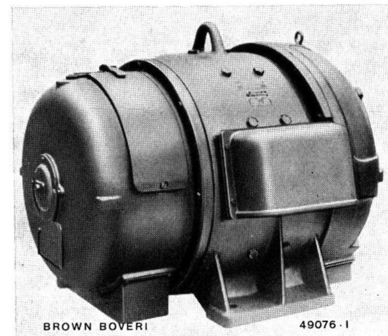



**Fig. 9. — D. C. motor in splash-proof design, 110 kW, 450 V, 1450 r. p. m.**

A further special feature, in this field, is the demand for very high over-speeds or run-away speeds. Account is fully taken of this requirement, among other measures, by a new way of supporting the pole windings, see the rotor of Fig. 13, which allows of the highest over-speeds encountered without any considerable drop in standard output of the machine.

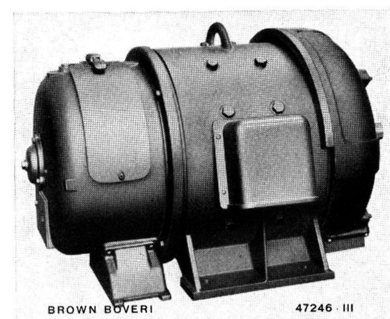



**Fig. 10. — D. C. generator, with semi-pipe ventilation, in splash-proof design, 55 kW, 230 V, 1000 r. p. m.**

To make this enumeration complete, mention must be made of two i machine classes which can be considered as extreme limits of the Wi series and

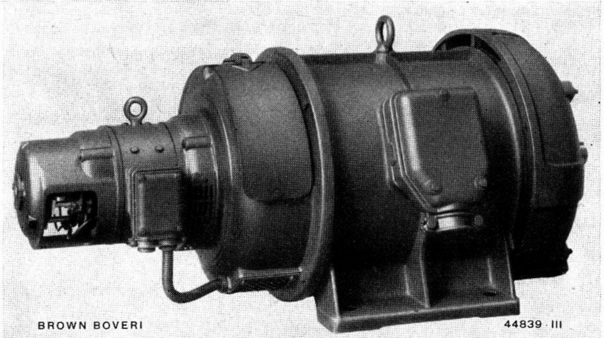



**Fig. 11. — Standard three-phase generator 125 kVA, 400/231 V, 1500 r. p. m., 50 cycles, with built-on exciter.**

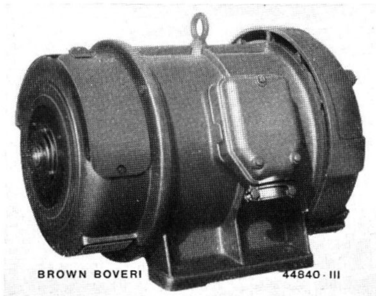



Fig. 12. — Standard three-phase generator 125 kVA, 400/231 V, 1500 r. p. m., 50 cycles.


Fig. 13. — Rotor with pole-winding supporting device, for an A. C. generator, 150 kVA, 500 V, 1500 r. p. m., 50 cycles.

which have been included in the work of standardization. In one case, these are the A.C. synchronous motors which, practically, correspond in construction to the Wi generators and, in the other, the synchronous induction motors of Type Si having the outer form of the Wi machines but the inner design more on the lines of the Mi machines.

## SUMMARY.

Brown Boveri have completed the total revision work on the medium-power i machines and imparted a uniform

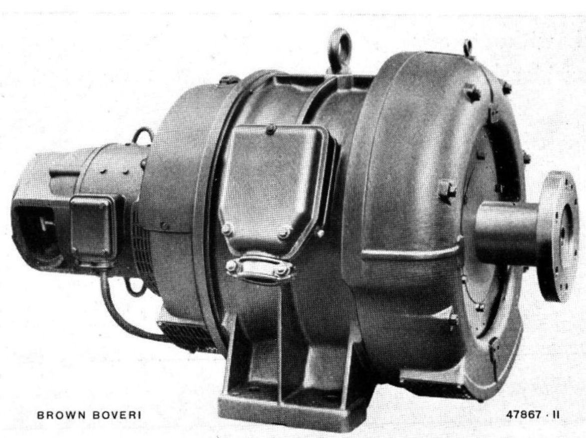



Fig. 14. — Three-phase generator with one bearing, 140 kVA, 225/130 V, 500 r. p. m., 50 cycles, with built-on exciter.

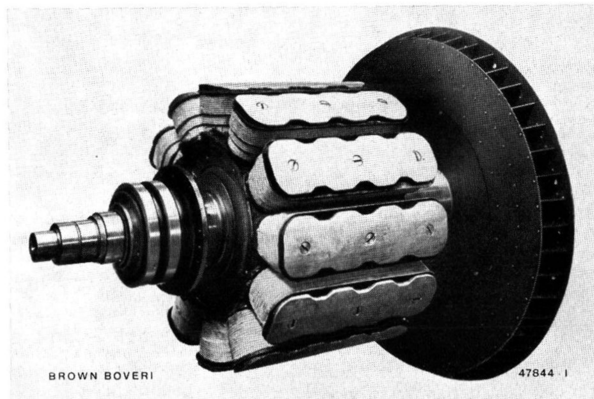



Fig. 15. — Rotor with shaft flange, for a three-phase generator, 140 kVA, 225/130 V, 500 r. p. m., 50 cycles.

fundamental character to a large number of electrical machine classes covering a considerable power range, so that these machines now form a uniform whole. This has considerable advantages, such as:— Quicker delivery, even when special electrical and mechanical conditions have to be fulfilled, this applies especially to protective measures; great simplification and uniformity of individual parts, advantages when parts for replacement and spares have to be acquired.

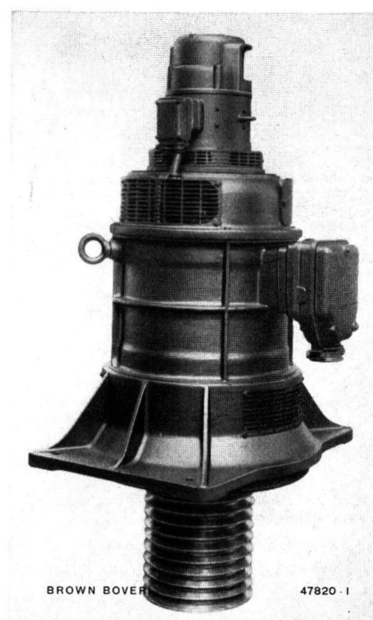



Fig. 16. — Three-phase A. C. generator of vertical type, with V-belt pulley, 135 kVA, 400/231 V, 1000 r. p. m., 50 cycles.

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