The Free-Piston Gas-Generator Turbine as a Power Plant for Ship Propulsion

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Free-piston engines have been used for many years as air compressors in single- and multi-stage form in both France and Germany. Their development for use as gas generators to supply "power gas" for use in turbines has, however, only recently reached a stage justifying production for marine, electric generating, and locomotive purposes.

In the paper the authors describe the inherent characteristics of the free-piston gas-generator/turbine engine system and the advantages of the inward compressing types of gas generator, which have been developed jointly by their two companies.

The advantages of the free-piston gas-generator engine system, including its inherent low cost of manufacture, ease of installation, and good maintenance features, are outlined, likewise its requirements in respect of turbines and reverse-reduction gears. Reference is also made to future possibilities, in particular advantages that are likely to accrue when a free-piston gas-generator/turbine is combined with a steam turbine.

I.—INTRODUCTION

(a) *Gas Generators*

A gas-generator power plant may be defined as one in which the power is first generated in the form of hot gas under pressure that is then converted into mechanical power at the shaft of a turbine or reciprocating engine. A steam engine would clearly come under this general definition, the boiler in this case being the gas generator, but the term "gas generator" or "gasifier" is actually used to mean a combination of internal combustion engine and air compressor in which the exhaust gases are added to the output from the compressor, the whole power output being delivered in the form of hot gas under pressure.

For instance, in a compound internal combustion turbine plant of the conventional type, the compressor, combustion chamber, and high pressure turbine driving the compressor may be said together to constitute a gas generator, since they produce gas whose energy is converted into work in the low pressure turbine.

Crankshaft gas generators have been built in Sweden by Götaverken. They consisted of a Diesel engine driving a reciprocating air compressor through the usual cranks, crankshafts, and connecting rods. The compressor is used to scavenge and supercharge the Diesel engine, which exhausts against a pressure about equal to the supercharge pressure. The combined exhaust products and excess scavenge air are then passed to the propelling machinery, which in these cases are reciprocating engines.

(b) *The Free-piston Gas Generator*

A free-piston gas generator is, therefore, a particular type of gas generator using the free-piston principle.

A free-piston engine consists essentially of a single cylinder two-stroke Diesel engine operating a piston type of air compressor. Now, in the conventional design of such a combination the reciprocating motion of the engine piston is transformed into a reciprocating motion of the compressor piston. If the stroke and the stroke/connecting rod ratio of the compressor were the same as that of the engine, and the two were direct coupled, the two pistons would execute motions that would be absolutely identical. Therefore, from the kinematic standpoint, it is quite unnecessary to use this double transformation from a reciprocating motion to a rotary one, and back again from rotary to reciprocating.

The cylinders of the engine and of the compressor can be put opposite each other, and the two pistons rigidly connected by means of a piston rod. The expansion of the gas in the engine cylinder moves the piston, which compresses the air in the compressor cylinder. The piston is, therefore, absolutely free to move in either direction, according to the difference between the pressures acting on opposite sides.

In a gas generator, the whole of the air from the compressor is delivered to the engine cylinder, which is therefore highly supercharged, the large excess of air passing out to the exhaust as scavenge air. The hot exhaust products join with the heated scavenge air and the whole constitutes the "power gas" that is delivered to the turbine. The power gas consists of at least 75 per cent of unburnt air, the remainder being the products of combustion of the fuel.

The overall compression and expansion ratio of the cycle is very high, which is well known to be one of the essentials for high thermal efficiency. The first part of the compression takes place in the compressor cylinder, and the high pressure part in the Diesel cylinder. Similarly, the high pressure part of the expansion takes place in the Diesel cylinder and the low pressure part in the turbine.

(c) *Working Cycle of Free-piston Gas Generator*

A diagrammatic sketch of a free-piston gas generator and turbine plant is shown in Fig. 1. It will be seen that there are two sets of moving parts (1), this arrangement being adopted on all machines in order to provide perfect balancing of the reciprocating masses. The gas generator illustrated here is of the "inward compressing" type, which means that the air in the compressor cylinders (4) is compressed on the inward stroke, as will be described presently.

Sectional drawings of a type GS-34 inward compressing

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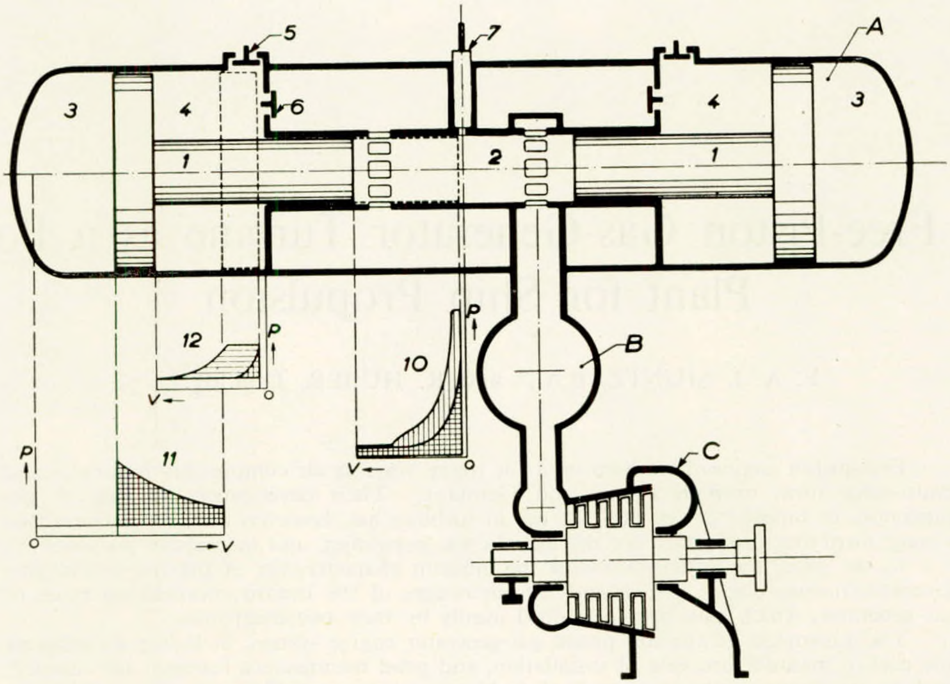


FIG. 1—*Diagrammatic sketch of a free-piston gas generator and gas turbine, showing pressure-volume diagrams of the engine, compressor and cushion cylinders*

A.—Gas generator; B.—Gas collector; C.—Gas turbine
1.—Piston; 2.—Engine cylinder; 3.—Cushion cylinder; 4.—Compressor cylinder; 5.—Suction valves; 6.—Delivery valves; 7.—Fuel injector; 10.—*p-V* diagram of engine cylinder; 11.—*p-V* diagram of cushion cylinder: 12.—*p-V* diagram of compressor cylinder

free-piston gas generator are given in Fig. 2, and a photograph of the machine in Fig. 3 (Plate 1). An 1,800/2,000 s.h.p. marine set, consisting of two GS-34 gas generators each driving a turbine, and a reduction gear, is shown in Fig. 4.

The operation of such a set will now be described with reference to Fig. 1.

The gas generator (A) contains two opposed piston assemblies (1) of equal and symmetrical stroke. The engine cylinder (2) is in the centre; it operates as a two-stroke Diesel engine supercharged to several atmospheres. The two single-acting compressor cylinders (4) are located at each end of the central housing. The end spaces (3) constitute the cushion or "bounce" cylinders which store the energy for the return stroke. Fresh air is drawn in through the suction valves (5) and is delivered through the delivery valves (6) into the "engine case", which surrounds the engine cylinder. The fuel is injected by several injectors (7) mounted in the central plane of the combustion chamber.

Consider now the cycle of operation. Beginning at the inner dead point (I.D.P.), the pressure of the gases in the combustion space of the engine cylinder added to the pressure of the air remaining in the clearance space of the compressor cylinders drives the pistons outwards, compressing the air in the cushion cylinder. The work done during this stroke is represented by the vertically cross-hatched areas in the pressure volume diagrams (10) and (12) for the engine and compressor cylinders. This work is subsequently recovered, except for friction losses, as shown on the pressure-volume diagram (11) for the cushion cylinder. As soon as the pressure in the compressor cylinders (4) has dropped to about atmospheric pressure, the automatic suction valves (5) open and a fresh charge of air is drawn into the cylinders. Meanwhile the engine cylinders have uncovered first the exhaust ports, shown on the right in Fig. 1, and soon afterwards the scavenge ports, shown on the left. This allows the air in the engine case to scavenge the engine cylinder from end to end; the excess scavenge air and exhaust products are collected in the gas collector (B), which acts as a smoothing chamber, and they subsequently flow to the turbine (C).

During the inward stroke, the air compressed in the cushion cylinders (3) drives the pistons back towards the centre. The charge in the compressor cylinders (4) is compressed and delivered to the engine case through the automatic delivery valves (6). As soon as the engine pistons cover the ports in the engine cylinder, the scavenging process comes to an end. The trapped air is then at the pressure of the gas collector because the exhaust ports must necessarily close last on the in-stroke as they must obviously open before the scavenge ports on the out-stroke.

This combustion air is then compressed by the pistons as they continue to move inwards, the injection being timed to take place towards the end of this stroke. The pistons come to rest at the inner dead point when the energy stored in the cushion cylinder, which provides the work done in the compressor cylinders and the work of compression in the engine cylinder, as well as the friction losses, is all expended. The work of this stroke is illustrated by the horizontally cross-hatched areas on the diagrams (10), (11) and (12) of Fig. 1.

(d) Different Types of Free-piston Gas Generator

The first type of gas generator developed in France was an asymmetric type with a double-acting compressor piston at one end. This layout, although it seemed to offer some advantages, involved the use of heavy link gear for the transmission of power from the Diesel piston not directly coupled to the compressor piston, thus sacrificing one of the great advantages of the free-piston engine system, namely absence of highly loaded bearings. Nevertheless, this unit gave quite encouraging results and in fact achieved an efficiency of 43 per cent, fuel into gas horsepower. (For a definition of gas horsepower see section III(b).)

The alternative single-acting outward compressing and inward compressing types of gas generator were then considered; these are illustrated diagrammatically in Fig. 5.

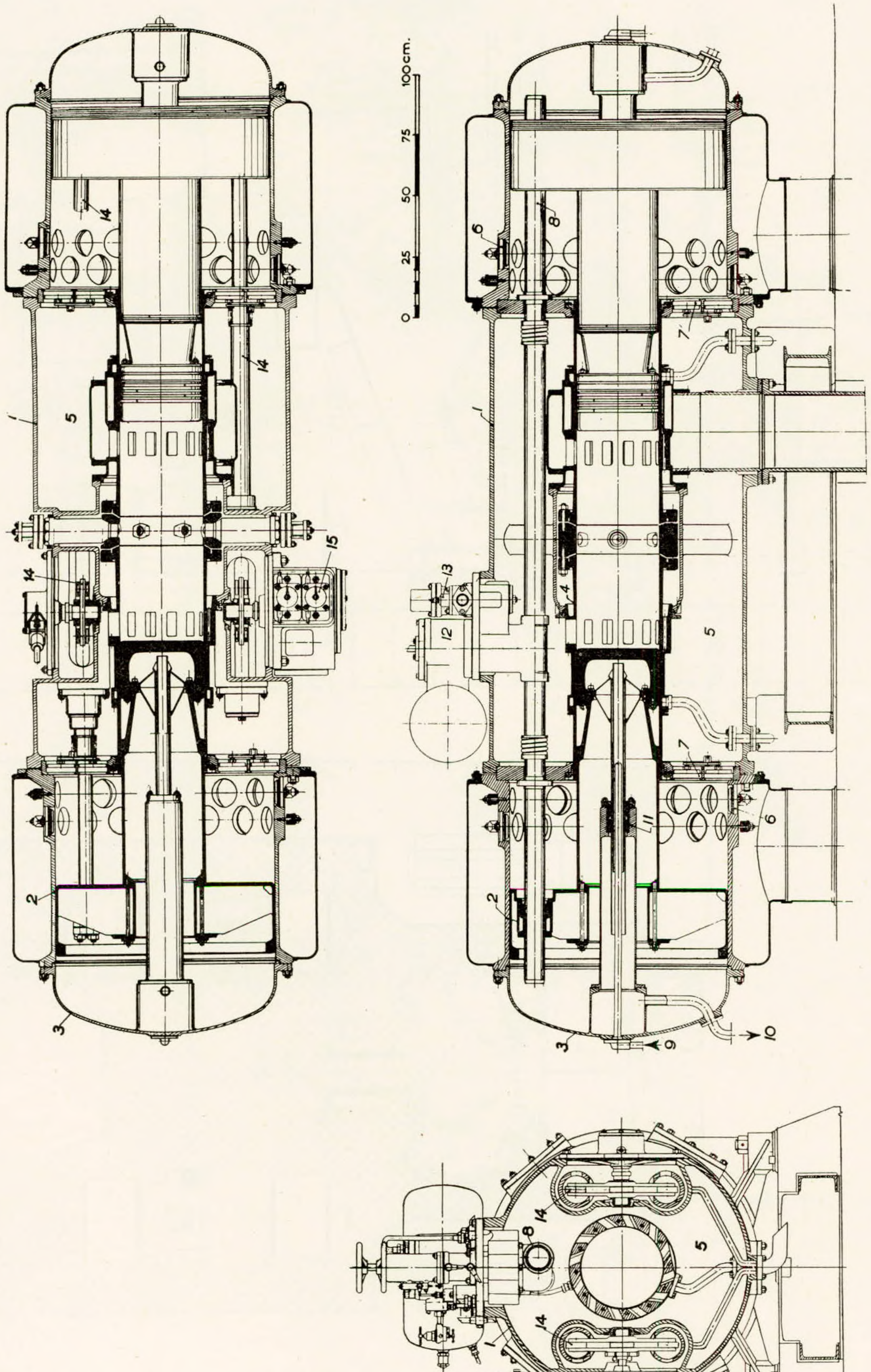


FIG. 2—*Sectional drawings of type GS-34 free-piston gas generator*

1.—Engine case; 2.—Compressor cylinder; 3.—Cushion head; 4.—Engine cylinder; 5.—Scavenge air receiver, inside of engine case; 6.—Suction valve; 7.—Delivery valve; 8.—Balance pipe; 9.—Piston cooling oil inlet; 10.—Piston cooling oil outlet; 11.—Gland; 12.—Starting valve; 13.—Cushion control or stabilizer; 14.—Synchronizing mechanism; 15.—Fuel injection pump.

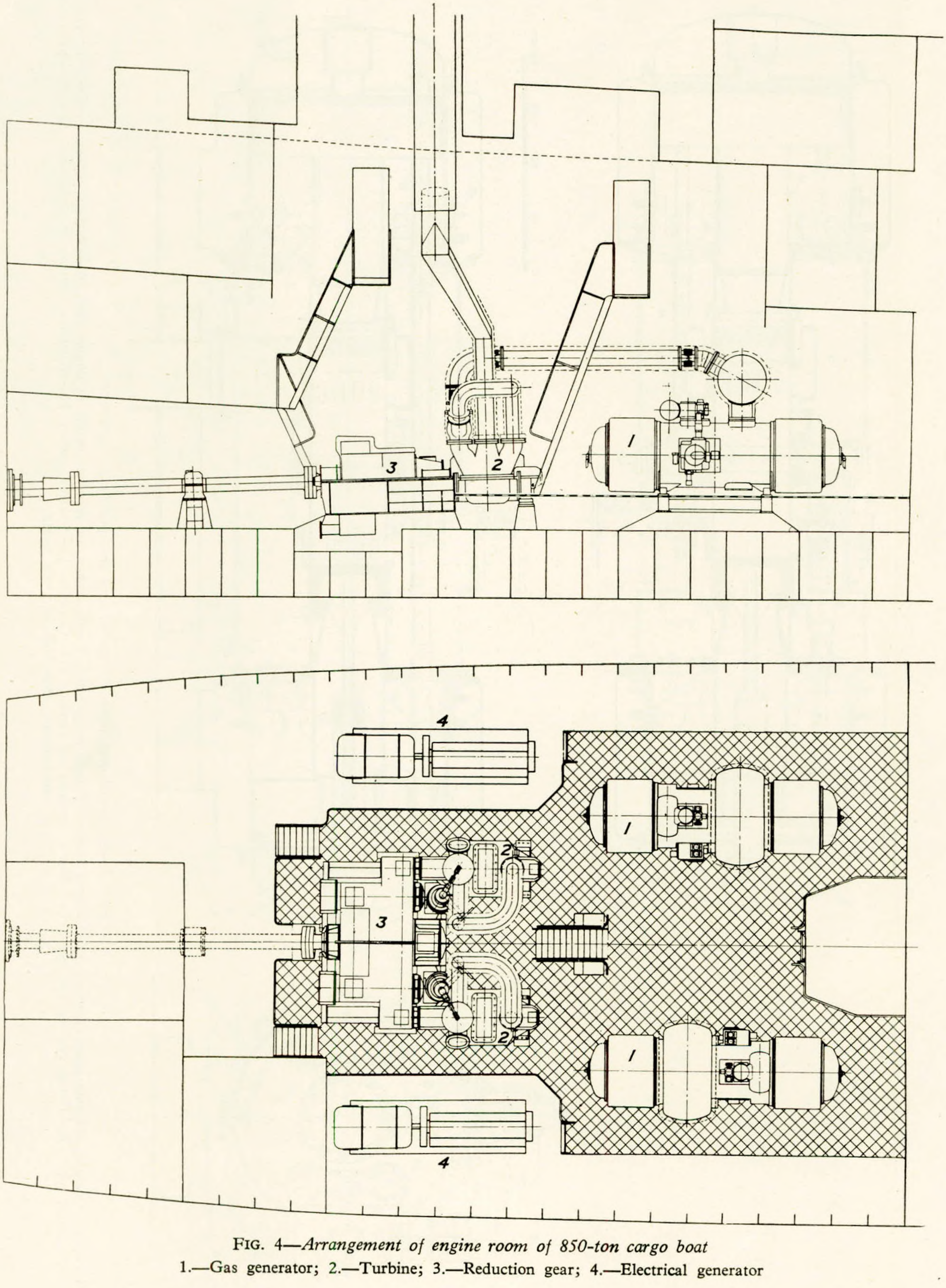


FIG. 4—*Arrangement of engine room of 850-ton cargo boat*

1.—Gas generator; 2.—Turbine; 3.—Reduction gear; 4.—Electrical generator

Although an outward compressing gas generator was fully designed with a view to manufacture of a prototype, it was finally decided to concentrate on the inward compressing type of gas generator for the following reasons:—

(*i*) With the outward compressing type of gas generator it is necessary to employ stepped piston assemblies having additional pistons of smaller diameter mounted on the outsides of the compressor pistons to provide a cushion. This means that not only are the overall dimensions and weights appreciably greater, but the manufacture is almost certainly more expensive.

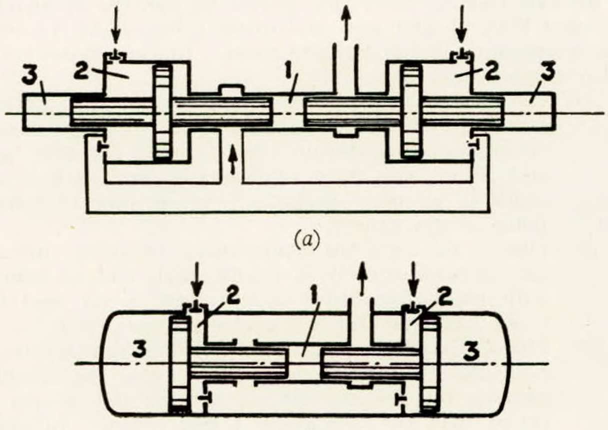


FIG. 5—*Diagrams of outward and inward compressing types of free-piston gas generators*
(*a*) *Outward compressing*
(*b*) *Inward compressing*

1.—Engine cylinder; 2.—Compressor cylinder; 3.—Cushion cylinder.

(*ii*) The maximum output obtainable from an outward compressing type of gas generator must be less than that obtainable from an inward compressing type, because the inherent greater weight of the moving parts will reduce the speed of oscillation.

(*iii*) Piston oil-cooling, which can be arranged extremely easily in the case of the inward compressing type of gas generator, becomes very much more difficult and complicated in the outward compressing type.

(*iv*) The inward compressing type is inherently simpler, particularly with regard to maintenance; for instance, the pistons can be much more easily withdrawn (see Fig. 11 (Plate 2)). These are about the only parts that have to be withdrawn during an overhaul.

(*e*) *1,250 Gas H.P. Gas Generator Type GS-34*

Fig. 2 shows sectional drawings of the GS-34 gas generator, which has been developed in France and works on the principle just described. About sixty of these machines have been built and forty are in operation for electrical generation and rail traction as well as in marine applications.

The leading particulars of the GS-34 are as follows:—

Engine cylinder bore ...	340 mm.	13·4 inches
Compressor cylinder bore ...	900 mm.	35·4 inches
Maximum stroke possible ...	2 × 550 mm.	2 × 21·6 inches
Stroke at continuous maximum rating (C.M.R.)...	2 × 443 mm.	2 × 17·4 inches
Number of oscillations per minute at C.M.R. ...	570	
Mean piston speed	8·4 m./sec.	1,650 ft./min.
Gas pressure at C.M.R. ...	3·0 kg./cm.^{2}	42·7lb. per sq. in. gauge
Gas temperature at C.M.R. with ambient temperature	437 deg. C. 20 deg. C.	819 deg. F. 68 deg. F.
Gas horsepower at C.M.R.*	1,250 gas h.p.	
Specific fuel consumption*	144 gr./gas h.p. hr.	0·320 lb./gas h.p. hr.
Thermal efficiency into gas*	43 per cent (on lower calorific value)	

* Based on adiabatic expansion, as defined in section III(b).

The one hour rating is 1,400 gas h.p. at 50lb. per sq. in. gauge.

In addition to the basic components already discussed, the following may be seen in Fig. 2:—

(*i*) The synchronizing gear (14), which in this case takes the form of a kinematic linkage, connecting the two sets of moving parts. This gear does not in any way

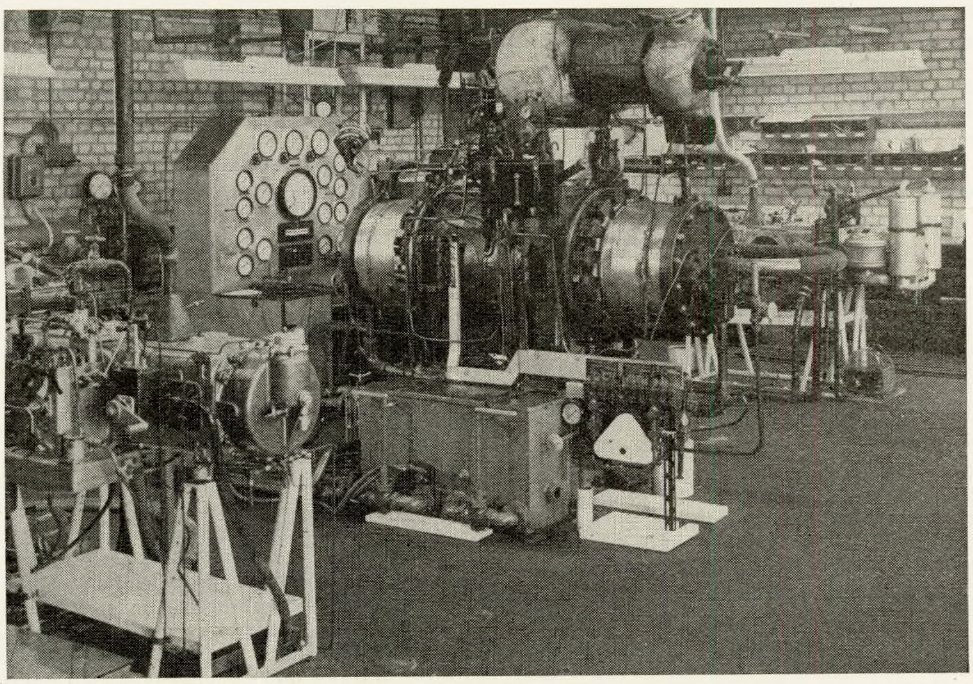


FIG. 10—*CS-75 prototype on test bed*

limit the stroke of the pistons, and theoretically it does not carry any load. However, it is obvious that there will be in practice small inequalities in the forces on the two ends, due to friction, wear, intake pressure, etc.; it is the function of the synchronizing gear to distribute these equally, keep the pistons in step and so maintain a perfect balance of the reciprocating masses.

(*ii*) The fuel injection pumps (15), which are driven from the synchronizing gear.

(*iii*) The cushion balance pipe (8), running from end to end, which ensures that the pressures in the two cushion cylinders are always equal. To the balance pipe are connected the device controlling the cushion pressure, known as the cushion control or "stabilizer" (13), and the starting valve (12), by means of which high pressure air is introduced to the cushion cylinders to start the engine.

(*iv*) The piston cooling oil system (9) and (10) can also be seen.

(f) 420 Gas H.P. Gas-Generator Type CS-75

A smaller unit known as the CS-75 is at present under development in this country. It is designed on the same general lines as the GS-34, as may be seen from the sectional drawing in Fig. 6. Photographs of a full scale model of the CS-75 are shown in Fig. 7 (Plate 1), and photographs of the moving parts and engine cylinder liner in Figs. 8 and 9 (Plate 1).

One difference that may be noted between the GS-34 and the CS-75 is that the synchronizing mechanism of the latter consists of a rack and pinion gear. There is only one fuel injection pump, and it is driven from the pinion shaft.

The prototypes have so far completed about 1,700 hours' running. One of them is shown on the test bed in Fig. 10. Its leading particulars are as follows:—

Engine cylinder bore ...	7·5 inches
Compressor cylinder bore ...	20·75 inches
Maximum stroke possible ...	2 × 12·75 inches
Stroke at continuous maximum rating (C.M.R.)...	2 × 9·5 inches
Number of oscillations per minute at C.M.R. ...	1,000
Mean piston speed	1,580 ft. per min.
Gas pressure at C.M.R. ...	44½lb. per sq. in. gauge
Gas temperature at C.M.R. with ambient temperature	465 deg. C. (869 deg. F.) 20 deg. C. (68 deg. F.)
Gas horsepower at C.M.R.*	420 h.p.
Specific fuel consumption*	0·325lb. per gas h.p. hr.
Thermal efficiency into gas*	42 per cent (on lower calorific value)

* Based on adiabatic expansion, as defined in section III(b).

The one hour rating of this unit is 500 gas h.p. at 50lb. per sq. in. gauge working pressure.

II.—FREE-PISTON GAS GENERATOR VERSUS DIESEL AND GAS TURBINE ENGINE SYSTEMS

The free-piston gas generator is in effect simply an oscillating gas boiler, which produces gas at a temperature of the same order as a modern steam boiler produces steam for use in turbines, but with an efficiency that has so far only been achieved by the compression ignition engine. It is, in fact, a compromise between the Diesel crankshaft engine and the gas turbine. It is, however, one which avoids their respective main disadvantages, namely the bulk, weight, heavily loaded bearings and torque reactions associated with marine crankshaft engines, and the high temperatures and bulk associated with gas turbines and heat exchangers.

There is also the very important point that neither the free-piston gas generator nor the complementary turbine, which transforms its "power gas" into shaft horsepower, require any better metals than have been in use in Diesel engines and steam turbines for many years past.

It possesses the following additional advantages:—

(a) Low First Cost and Ease of Installation

A number of gas generators of moderate size can be piped together to supply a turbine of any required power. In this way a very wide range of power can be obtained with one size of gas generator, and this means that it will become possible for the first time to obtain for marine engines the advantages in respect both of first cost and maintenance which go with series production of standardized units. In this connexion it is also worth noting that:—

(*i*) Gas generators being so much smaller than corresponding marine Diesel engines, and having no crankshafts, their manufacture can be carried out with tools and lifting equipment of moderate size, such as are available in most machine shops—a very important point in time of war.

(*ii*) Due to the moderate temperatures involved, turbines can be made entirely in ferritic steel; and, of course, with the gas-generator system there is no need for bulky and expensive heat exchanger equipment.

(*iii*) Due to their light weight and freedom from vibration, both the free-piston gas generators and the complementary turbines are very much easier and cheaper to install than the comparable Diesel engine. In addition, gas generators can, due to their small overall height when installed horizontally, be positioned between decks and, moreover, in any convenient position, since there is no need for them to be in line with the propeller shaft.

(b) Good Power/Weight Ratio

Due to the fact that high pressure loads are transmitted directly from the engine pistons to the compressor pistons and that very high supercharging is an inherent feature, the weight of gas-generator machinery is inherently low. In point of fact, the weight of GS-34 and CS-75 gas generator/turbine cum reduction gear plants works out approximately as follows:—

1. GS-34 gas generators cum reversing turbine and normal reduction gear 38lb. per s.h.p.
2. CS-75 gas generators cum epicyclic reverse-reduction gear ... 22lb. per s.h.p.

The weight of CS-75 sets is substantially lower than that of the GS-34 sets, not only because the weight per horsepower of the CS-75 gas generator is lower than that of the GS-34, but also because it has been assumed that much lighter reverse-reduction gearing (epicyclic) and auxiliary gear will be used in conjunction with the CS-75 gas generator.

(c) Ease of Maintenance

Maintenance cannot, in fact, fail to be both easier and cheaper than for a Diesel engine, for the following reasons:—

(*i*) Any one gas generator can be stopped without affecting the rest, which means that the ship can continue on its course while the individual gas generator is being dealt with.

(*ii*) It takes only fifteen minutes to withdraw the two piston assemblies, for instance to change a piston ring, and this very soon after running at full load; and another forty minutes to reassemble and get the engine running again. Ease of dismantling is illustrated in the photographs in Fig. 11 (Plate 2) which show successive stages in the removal of the moving parts of the GS-34, the cushion head weighing 300 kg. (660lb.), the compressor piston 240 kg., and the engine piston and trunk 290 kg. For the CS-75 the weights are, of course, proportionately less, the cushion head and the compressor piston each weighing about 150lb.

(*iii*) The gas generators, being comparatively light (GS-34 8½ tons, CS-75 2 tons), their removal for overhaul is a quick and easy business. In point of fact, if a spare gas generator were to be carried on the ship, all overhauls could be carried out while at sea. In any case, it would be possible to reduce very considerably current overhaul periods by carrying a stock of reconditioned gas-generator units at the main ports of call.

(d) Fuel Consumption

It seems probable that the straight free-piston gas-generator/turbine engine system will have about the same fuel consumption at full load as the corresponding Diesel engine. The specific fuel consumption curve is fairly flat down to quite low loads: at 50 per cent of full load the specific consumption on gas h.p. would be about 4 per cent higher than at full load, and at 25 per cent of full load it would be about 12 per cent higher. These figures would apply to a plant where several gas generators supply a single turbine. Where there is only one gas generator it is necessary to blow off some of the "power gas" at the powers below about 30 per cent of full load, for the reasons given in section III(c). In such a case, the specific fuel consumption below 30 per cent of full load would be increased.

In the final section of this paper, mention is made of a compound power plant in which the gas generator plant is combined with a steam turbine plant, offering the possibility of appreciably better fuel economy than the simple set both in respect of the grades of fuel usable and consumption per horse-power-hour.

(e) Ability to Use Low Cost Fuel

The inherent high supercharge and large amount of excess air make for good combustion of residual fuels and, in this connexion, it has the following additional points in its favour:—

(*i*) Auxiliary power can be provided via auxiliary turbines taking "power gas" from the main battery of free-piston gas generators.

(*ii*) There is no fear of combustion products contaminating crankcase lubricating oil for the reason that fresh oil alone is supplied by a metering pump to the Diesel and compressor pistons and, in any case, combustion products are blown out with the scavenge air.

(*iii*) There is no risk of vanadium trouble as far as turbine blades are concerned, because the maximum temperature of the free-piston "power gas" is normally less than 500 deg. C.

(f) Starting

Free-piston gas generators start very readily owing to the possibility of increasing the engine compression ratio for the first few strokes simply by increasing the starting air pressure, and in this connexion the quantity of high pressure air used is very much less than with a comparable Diesel engine.

III.—BASIC PRINCIPLES AND OPERATING CHARACTERISTICS

(a) Motion of Pistons

The pistons of a free-piston engine are free only in the sense that they are not constrained mechanically (the synchronizing mechanism does not restrict the motion, but merely ensures that it is the same for both sets of moving parts). However, their motion is quite precisely controlled by the gas pressures acting on them and by their own inertia.

Each piston assembly may be thought of as a spring-loaded mass, whose frequency of oscillation depends on its inertia and on the stiffness of the pneumatic springs, and whose stroke may be altered by altering the spring forces.

Beginning with the inner dead point, the outward accelerating force, ignoring friction, is the difference between the forces due to the engine cylinder pressure and compressor pressure on the one hand and the cushion pressure on the other. The work done by this accelerating force is converted into kinetic energy, the maximum piston velocity being attained around mid-stroke. As the pistons compress the air in the cushion cylinder they gradually give up their kinetic energy and come to rest at the outer dead point (O.D.P.). In order to increase the outer dead point so as to increase the stroke at any constant gas delivery pressure, all that is required is to increase the kinetic energy of the pistons. This is done, of course, by increasing the work done on the pistons in the engine cylinder, or in other words, by increasing the amount of fuel injected.

The process is similar on the return stroke. The inward accelerating force is the difference between the forces due to the cushion pressure on the one hand and those due to the pressures in the compressor and engine cylinders on the other. The pistons come to rest at inner dead point when all their kinetic energy has been expended. In order to reduce the inner dead point so as to increase the engine compression pressure, the work done on the pistons by the cushion must be increased; this requires an increase in cushion pressure, which can be achieved by adding air to the cushion cylinder.

So in a free-piston engine, both the stroke and the engine compression pressure are variable. The outward piston acceleration at inner dead point is considerably greater than its inward acceleration at outer dead point. This means that the pistons spend relatively less time near inner dead point, where the engine cylinder temperature and pressure are high, and the rate of heat loss is large, and relatively more time near outer dead point, while scavenging is going on, than if their motion were constrained to approximately a simple harmonic motion as in a crankshaft engine.

It follows from the analogy of the spring-loaded mass that the frequency of oscillation, or speed, of a free-piston engine cannot be independently controlled as in a crankshaft engine, but is determined by the stiffness of the pneumatic springs and the inertia of the moving parts. The spring stiffnesses increase with pressure, so that the speed increases as the gas delivery pressure rises.

However, the whole speed range is comparatively small. Roughly speaking, the speed at no load, when the delivery pressure is very low, is about two-thirds of the speed at full load, when the pressure is high.

For a given delivery pressure, an increase of stroke obtained by increasing the outer dead point results in a comparatively small decrease in speed—about 5 per cent for the whole working range, while an increase in the engine compression pressure produced by a comparatively small change in inner dead point results in an increase of speed—about 10 to 15 per cent for the whole practical range of compression pressures.

(b) Output of Gas Generator

The output of a gas generator is usually expressed as a "gas horsepower". This is defined as the power available from an adiabatic expansion of the gas down to atmospheric pressure; in other words it is the power that would be obtained if the gas were used in a perfectly efficient turbine exhausting to atmosphere. The gas horsepower is, therefore, the product of the adiabatic heat drop and the mass flow of gas.

The adiabatic heat drop is given by the well-known expression:—

$$i = C_p\, T_1 \left(1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right)$$

where i = adiabatic heat drop in heat units per pound of gas
C_p = specific heat at constant pressure
T_1 = gas delivery temperature, absolute
P_1 = gas delivery pressure, absolute
P_2 = atmospheric pressure, absolute
γ = ratio of specific heats.

In fact, the adiabatic heat drop depends principally upon T_1 and P_1, and the variations in C_p and γ are secondary effects only. The gas delivery temperature obviously depends on the inlet air temperature, to which is added the temperature rise through the gas generator. This temperature rise consists of two elements:—

(*i*) The temperature rise due to the compression of the air; ideally this is merely the adiabatic temperature rise, but in practice there are always pumping and other losses that heat the air above its adiabatic temperature.

(*ii*) The temperature rise due to the mixing of the air with the exhaust gases; this rise comes about because the air in effect picks up the heat rejected from the engine cylinder (excluding the heat to the coolant), and it varies inversely as the indicated efficiency of the engine cylinder.

It follows from this that the delivery temperature T_1 rises with the delivery pressure P_1, but that apart from this an increase of T_1 is, with one exception, indicative of an increase in gas generator losses. The exception is a reduction in heat loss to the coolant, which admittedly would improve the efficiency of the engine but which might have a harmful effect on the life of the liner, pistons and piston rings.

It must be emphasized that the rule, familiar in internal combustion turbine practice, that the higher the turbine inlet temperature the higher the efficiency, does not apply to a gas generator/turbine set. The reason is that in this case the peak temperature in the thermodynamic cycle occurs during combustion in the engine cylinder. This temperature can be raised by increasing the engine compression ratio, which would improve the engine indicated efficiency and, assuming the coolant loss to be unchanged, would actually lower the gas delivery temperature T_1.

For a given engine running at or near its optimum combustion conditions, the gas delivery temperature may be taken, to a first approximation, to be a function of delivery pressure alone. This is not strictly correct, as there is a tendency for it to rise when the stroke is increased, owing to a drop in the engine cylinder indicated efficiency and increased pumping losses generally.

The mass flow of gas depends on a number of factors, and a distinction must be made between the gas flow per cycle and the gas flow per unit time.

At constant gas delivery pressure, the gas flow per cycle depends both on the inner dead point and on the outer dead point, which will be considered separately. An increase in inner dead point gives an increase in compressor cylinder clearance volume, thereby increasing the fraction of suction stroke occupied by the re-expansion. Thus the first atmospheric point, where the suction valves open, is moved outwards, so reducing the effective suction stroke (this is, in effect, the fraction of the suction stroke during which air is actually being drawn into the cylinder). This reduces the gas flow per cycle. At the same time, the speed will fall because of the lower compression pressure, so that there will be a two-fold reduction in the gas flow per unit time. The second atmospheric point, where the suction valves close, is practically at outer dead point, so an increase of outer dead point gives an equal increase in effective suction stroke; if the inner dead point is left unchanged, there will be only a small drop in speed, and the gas flow per unit time will be increased nearly in proportion. Although a given change of inner dead point has more effect on the gas flow per unit time than the same change of outer dead point, a much greater variation of outer dead point than of inner dead point is permissible in practice, and the gas flow at constant delivery pressure is usually altered by adjusting the outer dead point, that is to say, by adjusting the quantity of fuel injected.

When the gas delivery pressure is raised, the first atmospheric point moves outwards because the air in the clearance space of the compressor cylinder is now re-expanded from a higher pressure. Consequently, at constant outer dead point, the gas flow per cycle is reduced. There is, however, a simultaneous increase of speed, and the combined effect is to increase the gas flow per unit time over the lower part of the working pressure range and to reduce it over the higher part.

This effect of reduced gas flow becomes so marked at very high gas delivery pressures, say, 100lb. per sq. in. gauge, that a stage would be reached where an increase of delivery pressure at constant outer dead point, in spite of the higher adiabatic heat drop, would cause an actual reduction in gas horsepower.

The gas-generator efficiency, defined as the ratio of gas horsepower to power input in the form of fuel, increases with delivery pressure, but above a certain turbine pressure ratio this increase becomes very small, and there is no great inducement to try to raise the rated full load pressure much above about 3½ to 4 atmospheres.

(*c*) *Limited Range of Stroke: Characteristic When Driving a Turbine*

In practice there are both upper and lower limits of stroke for any given free-piston gas generator.

The lower limit is set by the necessity to uncover the ports in the engine cylinder sufficiently to give time for scavenging. The minimum stroke, therefore, depends on the position of the scavenge ports.

The upper limit depends primarily on the dimensions of the machine, as it would obviously be unsuitable to allow it to operate with an outer dead point too close to the mechanical limit. It may be mentioned here that all gas generators are fitted with an overstroke trip that automatically stops the fuel injection if the outer dead point exceeds a given value, so that a mechanical contact at the end of the stroke will not occur. Furthermore, at the present stage of development, it is found that at high delivery pressures and large outer dead points the combustion efficiency tends to fall off somewhat, as evidenced, for instance, by a dirty exhaust, and this condition is treated as the practical limit of stroke at high pressures.

In order to ensure that the stroke should not accidentally stray outside these limits, the stroke of the fuel rack is limited by minimum and maximum stops to limit the range of fuel quantity injected in accordance with the instantaneous delivery pressure. At each extreme of the allowable fuel rack stroke, a pin comes on to a cam whose position is controlled by the delivery pressure.

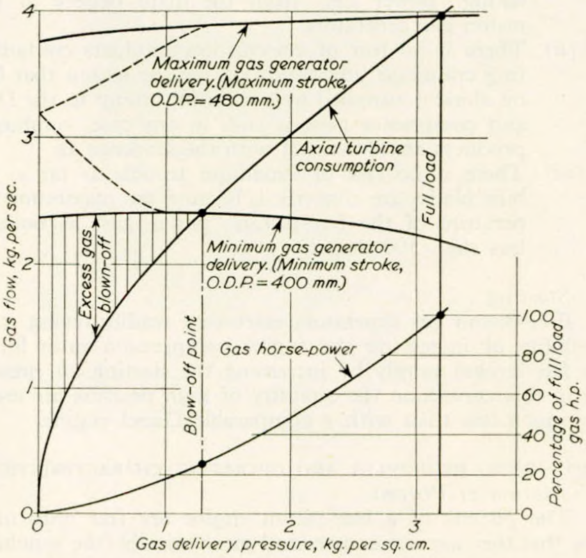


FIG. 12—*Flow characteristics of type GS-34 gas generator supplying an axial turbine*

The effect of this limited range of piston stroke on the characteristic when driving a turbine is illustrated in Fig. 12. There is a corresponding limited range of gas flow at each delivery pressure, and in particular it is not possible to reduce the gas flow below the amount corresponding to the minimum stroke. This means that even at very low delivery pressures, when the gas generator is idling the gas flow is still a substantial fraction of the full load gas flow.

If the gas generator is delivering to an axial turbine, its delivery of gas will exceed the turbine requirements at low delivery pressures. This follows from the fact that an axial turbine behaves like a fixed orifice, in that the gas flow through it falls progressively as the supply pressure is reduced, the turbine having, of course, been designed so that its gas flow requirement matches the gas-generator delivery at full load.

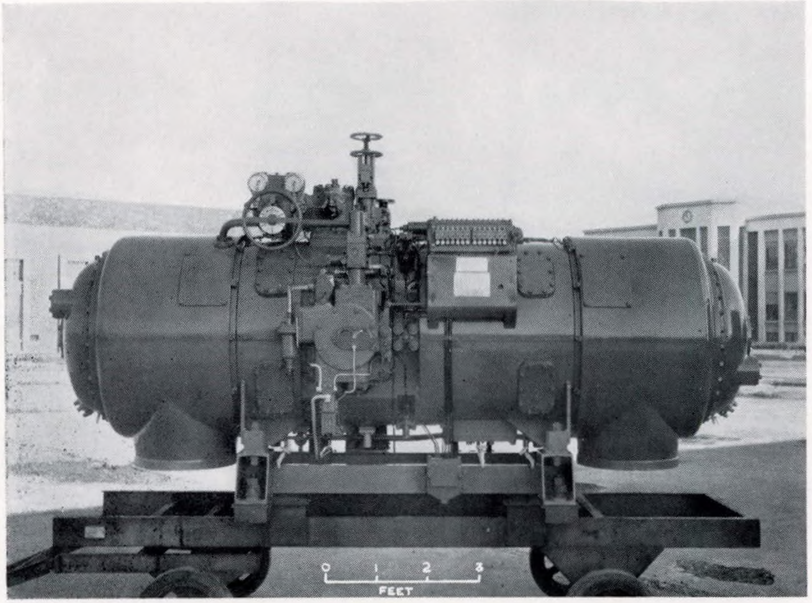


FIG. 3—*GS-34 free-piston gas generator ready for lifting aboard*

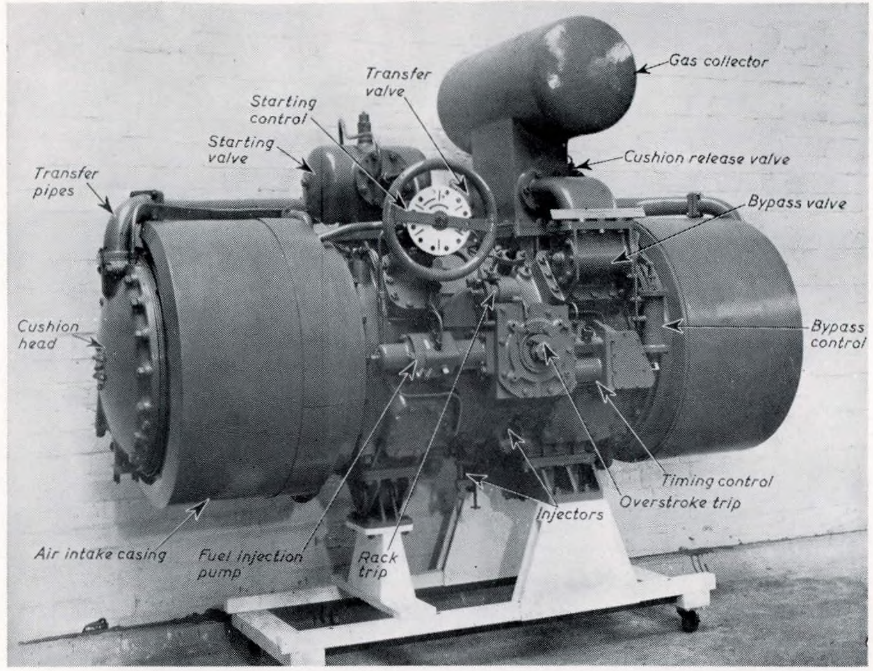


FIG. 7—*Full-scale model of CS-75 free-piston gas generator*

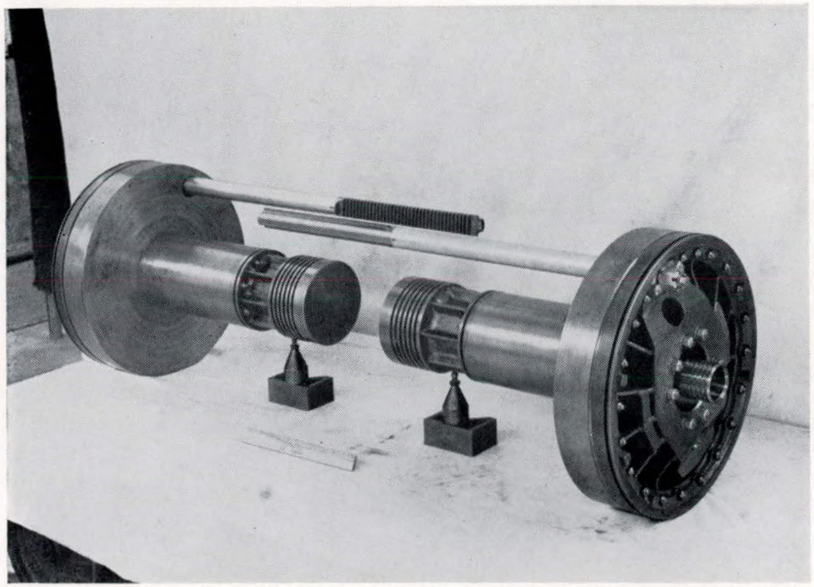


FIG. 8—*CS-75 prototype—moving parts*

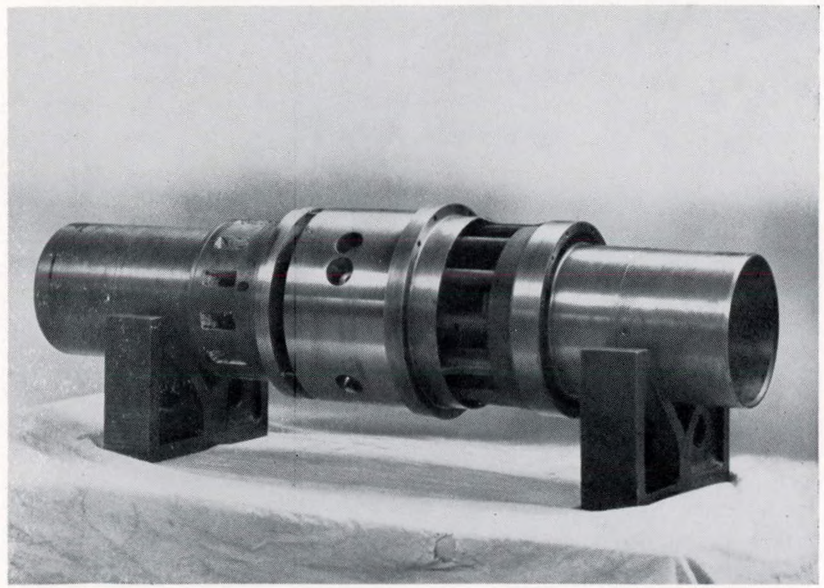
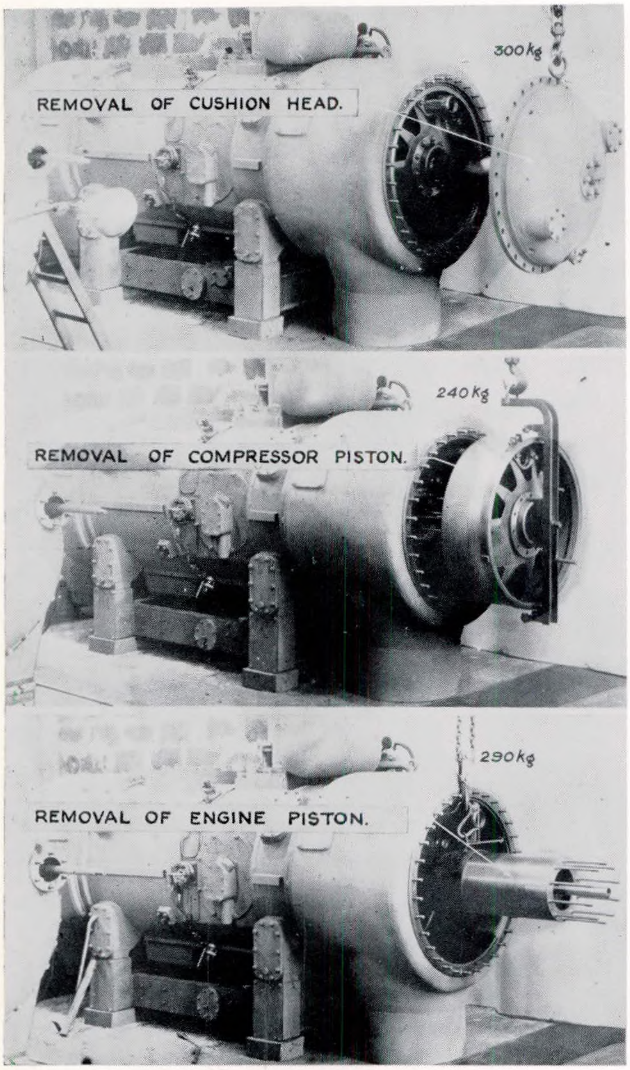


FIG. 9—*CS-75 prototype—7½-in.-bore cast-iron cylinder liner with aluminium bronze reinforcing ring fitted*

Plate 2



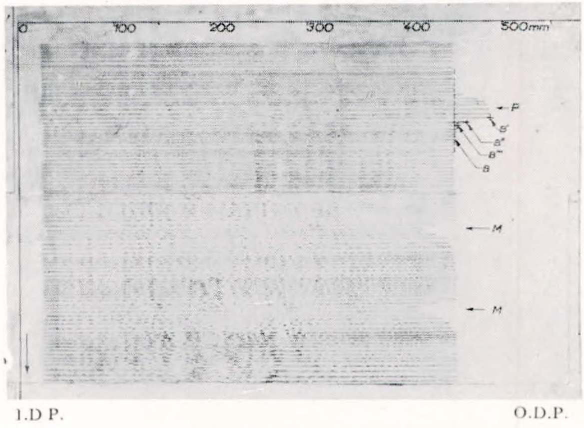


FIG. 18 *(above)—GS-34 stroke records during momentary disturbance*

FIG. 11 *(left)—Successive stages in removal of moving parts of GS-34 free-piston gas generator*

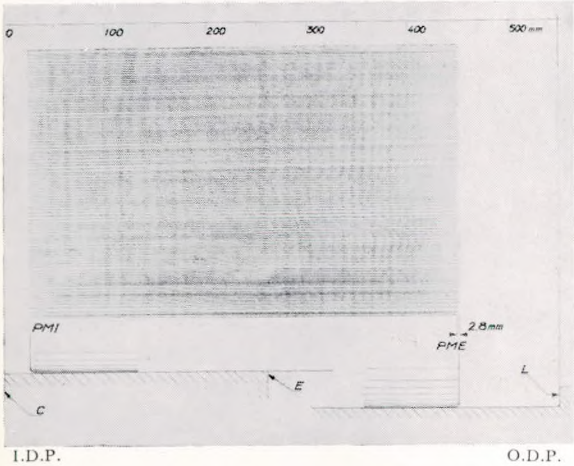


FIG. 17—*GS-34 stroke records during steady running*

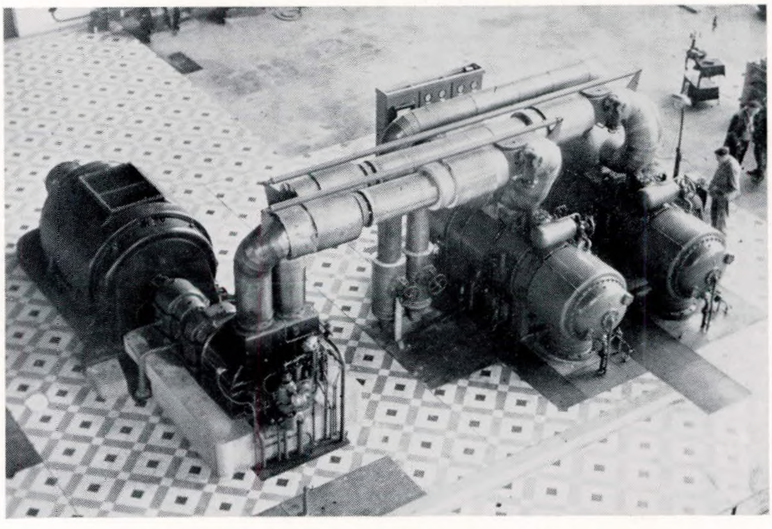


FIG. 21—*1,250 kW. free-piston gas generator/turbine set installed at the Electricité de France, Rheims*

In fact there comes a point, as the load is reduced, where the gas-generator delivery can no longer be reduced to match the turbine requirement. This point is known as the blow-off point, and occurs normally at about 30 per cent of full load power. Below this point the excess gas delivered by the gas generator must be blown off to atmosphere. It should be noted that the gas-generator delivery at minimum stroke is made as small as possible by running at a large inner dead point, i.e. a low compression pressure. This remedy is, of course, limited by the need to provide an adequate engine compression.

A more promising way of overcoming this difficulty, by exploiting the gas generator's property of very easy starting, is to provide a battery of gas generators for a single turbine, one or more of them being shut down for light load running.

(d) Reasons for Good Efficiency

The reasons for the good efficiency of gas generators, some of which have already been touched upon, may be briefly summarized as follows: —

- *(i)* The maximum engine cylinder pressure is not limited by the allowable thrust on big end and crankshaft bearings: the load on the engine piston is transmitted as a direct thrust through the piston rod to the compressor piston. As a result the overall engine compression ratio can be made much higher than in a crankshaft engine.
- *(ii)* Owing to the high outward piston acceleration at inner dead point, the peak engine temperature and pressure endure for a relatively shorter time than in a crankshaft engine, with a consequent reduction in thermal loading on the engine pistons and liner and in coolant loss.
- *(iii)* In view of the end-to-end scavenging and of the large scavenge air ratio, scavenging is probably very thorough.
- *(iv)* Since the gas-generator delivery can be reduced progressively with the delivery pressure, at any rate down to the blow-off point, no throttle valve is needed at the turbine inlet so that there are no throttling losses at part load.
- *(v)* Special streamline automatic valves, both suction and delivery, have been developed for use on free-piston engines. The good performance of these valves leads to low compressor cylinder pumping losses.
- *(vi)* Some reduction in the pumping losses can be obtained by grouping the gas generators in pairs and connecting their engine cases together, and, by means of a device known as a dephaser, arranging the two gas generators of a pair to run exactly 180 degrees out of phase. The advantage of this arrangement is that the compressor delivery period of one gas generator coincides with the scavenging period of the other, so that the compressor delivery pressure is reduced.

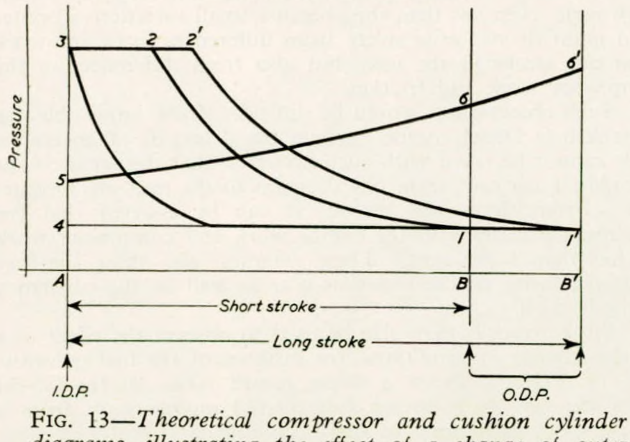


FIG. 13—*Theoretical compressor and cushion cylinder diagrams, illustrating the effect of a change of outer dead point*

IV.—STABILITY OF OPERATION AND CONTROL GEAR

(a) Control of Cushion Pressure

The operating stability is one of the chief problems of the free-piston engine. By operating stability is meant the characteristic of a free-piston engine that enables it to maintain the correct return of the pistons towards inner dead point at all working pressures and at all strokes; in other words, to maintain sufficient but not excessive compression of the combustion air at all operating conditions.

The operating stability of a gas generator depends essentially on the dimensions of the cushion cylinder and of the device controlling the cushion pressure, called the cushion control or stabilizer.

An important condition for stable operation is illustrated in Fig. 13, which shows on a stroke base the pressures in the compressor cylinder and cushion cylinder for two running conditions, one with a long stroke and the other with a short stroke. Point *A* represents the inner dead point at a given working pressure. With a short stroke the pistons reach the outer dead point at point *B*, and with a long stroke at point B^1. In the first case the indicated work in the compressor cylinder during the return stroke is represented by the area 1-2-3-4-1; in the second case by the area 1^1-2^1-3-4-1^1. The difference between these two areas, namely the area 1-1^1-2^1-2, represents the increase in the indicated compressor work during the return stroke owing to the increase in stroke *B*-B^1. This excess work must be provided by an increase in the energy accumulated in the cushion when the outer dead point increases from *B* to B^1.

The pressure in the cushion cylinder varies along the curve 5-6 with a short stroke and 5-6^1 with a long stroke. In the first case the cushion energy is, therefore, represented by the area 4-1-6-5-4 and in the second case by the area 4-1^1-6^1-5-4. Furthermore, the difference between these two quantities of energy, namely 1-1^1-6^1-6, represents the increase in the energy stored in the cushion when the stroke increases from *B* to B^1.

It can be seen from this figure that in order to reach the same inner dead point at *A* from *B* or B^1, the area 1-1^1-6^1-6 must be equal to the area 1-1^1-2^1-2. This condition determines the slope of the curve 5-6-6^1; in other words, the volume of air required in the cushion cylinder. In practice, the position of the inner dead point should not be exactly constant. It is usually an advantage to obtain a low engine compression at short strokes, that is to say at low powers, and a high engine compression at long strokes. In this case the area 1-1^1-6^1-6 should be greater than 1-1^1-2^1-2.

The above discussion relates to a constant working pressure. However, a generator must be able to run between very wide

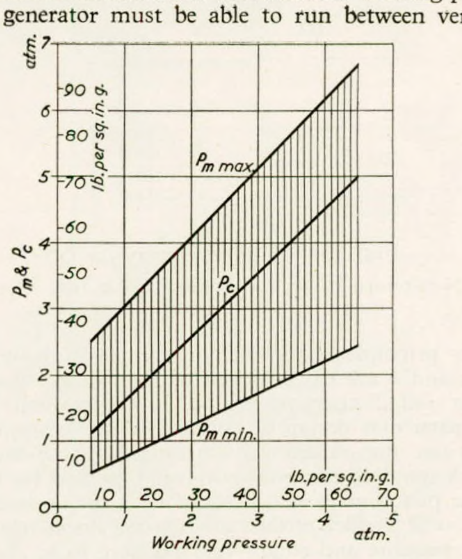


FIG. 14—*Maximum and minimum cushion pressures and engine-case pressure as functions of the working pressure*

p_m.—Cushion pressure; p_c.—Engine-case pressure limits of working pressure. When the working pressure varies from 1 to 5 atmospheres absolute, the mean pressure in the cushions must be varied to correspond, in order to maintain suitable compression pressures in the Diesel cylinder. The adjustment of the pressure in the cushions to suit the actual working pressure must be automatic. This operation is carried out by the cushion control or "stabilizer". It adjusts the mean pressure in the cushions to correspond with the pressure of the scavenge air in the engine case corresponding to the turbine load. The engine case pressure may be regarded as constant for a given load, whereas the air in the cushions is compressed by several atmospheres and expanded again at each stroke. Thus, for one part of the stroke the pressure in the cushion is higher than the engine case pressure and for the other part it is below it. Fig. 14 shows that this is the case under all working conditions. Thus, it is always possible to transfer air from the engine case into the cushions during one part of the stroke, or conversely to discharge air from the cushions to the engine case during the other part of the stroke. For this purpose a piston valve, with non-return air valves, is fitted between the engine case and the balance pipe to the cushions, as shown in Fig. 15.

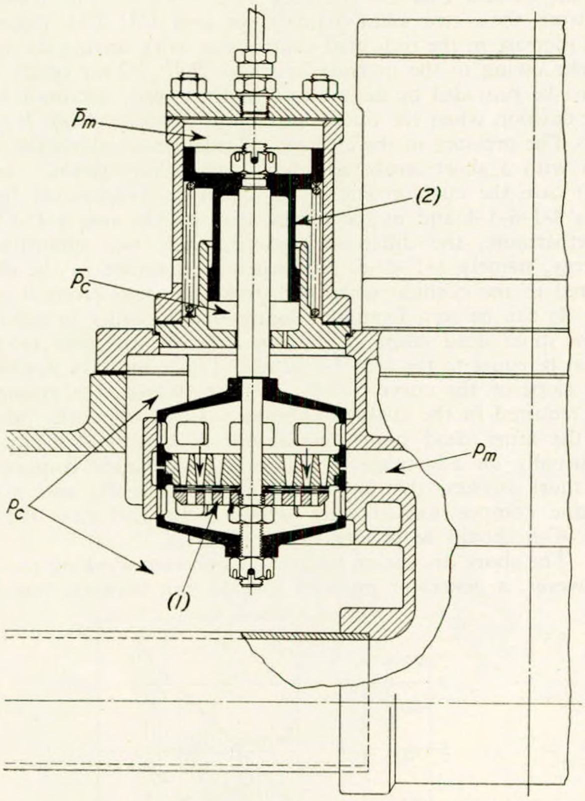


FIG. 15—*Cushion control for GS-34*

1.—Non-return valve; 2.—Differential piston; p_m.—Cushion pressure; p_c.—Engine case pressure.

The principle of the cushion control is shown in Fig. 16, where *a* and *b* are the positions of the piston valve during the charging and discharging periods of the cushions respectively. In the particular design of cushion control shown in Fig. 15, there is one non-return valve arranged in the moving piston valve. A standard compressor valve (1) is used for this purpose.

The piston valve is controlled by a spring-loaded differential piston (2), which enables any desired linear relation between cushion pressure and engine case pressure to be obtained. If a non-linear relation is desired, this can be obtained by using a cam in the control mechanism, or by interposing a special type of pressure-reducing valve, called a "modulator", between the engine case and the cushion control.

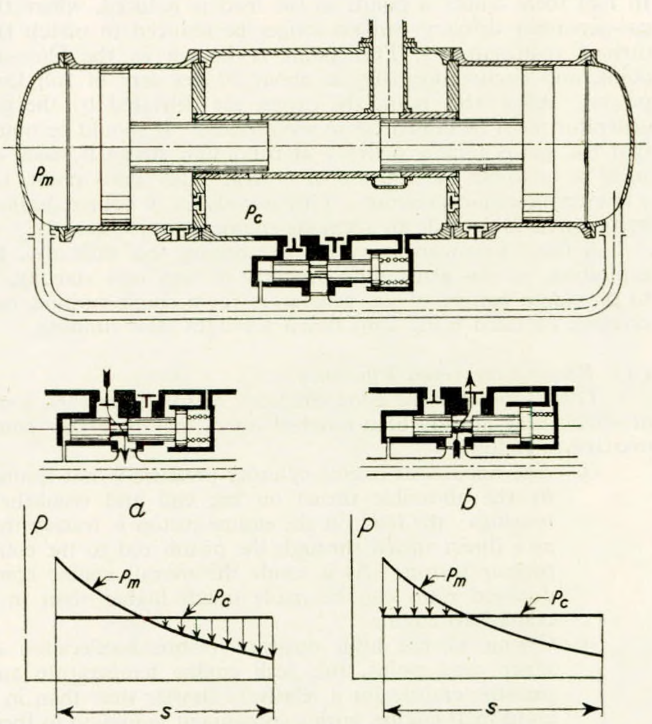


FIG. 16—*Diagrams illustrating operation of cushion control*

a.—Charging the cushion; *b*.—Discharging the cushion; p_m—Cushion pressure; p_c—Engine case pressure; *s*.—Stroke.

(b) *Stroke Records*

The observation and recording of successive strokes of the piston constitutes a very precise means, without equivalent on crankshaft engines, of keeping a watch on the functioning of a gas generator. Such readings make it possible to tell with great accuracy the positions of the inner dead point and outer dead point with respect to the centre line and to deduce from them the compression ratio of the combustion air, the instantaneous output, and the regularity of successive cycles.

A diagram taken on the GS-34 at constant fuel pump rack setting is shown in Fig. 17 (Plate 2). This diagram shows a great regularity in the length of the stroke. The maximum variation of the outer dead point with respect to its mean position does not exceed $\pm$ 1·4 mm. For this outer dead point the mean pressure in the cushion cylinder is equal to 4·6 kg./cm.^{2} gauge (61lb. per sq. in. gauge). The variation of energy due to the irregularities is $\pm$ 41 kg. metres, and as the indicated engine work is 3,430 kg. metres, the calculated fluctuations of the engine power will be $\pm$ 1·2 per cent. In fact, the engine work varies even less than this, because small variations of outer dead point do not arise solely from differences in engine work from one stroke to the next, but also from differences in the compressor work and friction.

Such observations would be difficult, if not impossible, on a crankshaft Diesel engine, because the diagrams of an engine cycle cannot be taken with such accuracy that discrepancies of less than 1 per cent from one diagram to the next are measurable. From the stroke records it can be asserted that the maximum variations in the engine work and compressor work are less than 1 per cent. These diagrams also show the very great regularity of the injection gear as well as the constancy of the friction.

Stroke records may also be used to observe the effect of a sudden change of conditions, for instance of the fuel quantity. Fig. 18 (Plate 2) shows a stroke record taken on the GS-34 when the fuel rack setting was altered momentarily from a steady state, an increase of fuel occurring at *P* and a decrease at *M*. In both cases the change of outer dead point is about 40 mm., and the normal dead point is reached within a few strokes of the end of the disturbance without any sign of instability such as hunting of the outer dead point around the mean position. These diagrams show that the gas generator has a very strong tendency to return to its normal steady state after each disturbance.

(c) Load Control

The variation of the power of the turbine fed by one or several gas generators is obtained by varying the quantity of fuel to each gas generator and thus the mass flow and the pressure of the gas delivered. In this way any loss of efficiency through throttling the gases upstream of the turbine is avoided.

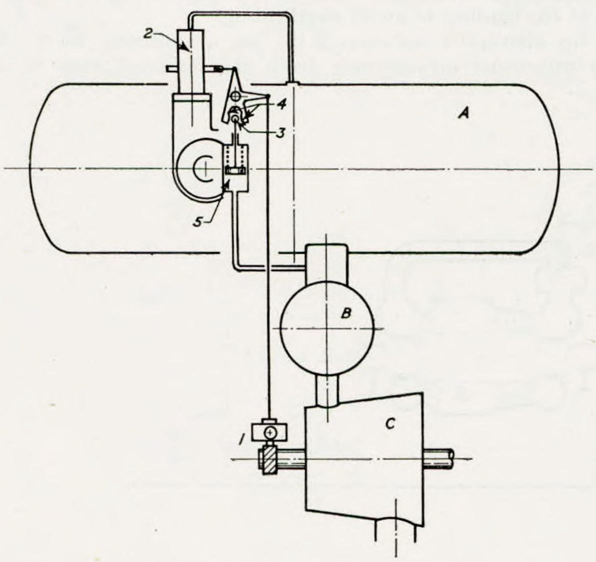


FIG. 19—*Diagram of fuel control gear*

A.—Gas generator; B.—Gas collector; C.—Turbine.
1.—Turbine speed governor
2.—Injection pump
3.—Fuel quantity stop
4.—Minimum and maximum fuel cams
5.—Control cylinder.

As shown in Fig. 19, the speed governor (1) of the turbine *C* acts directly upon the racks of the injection pumps (2) of the gas generator or gas generators *A*. A stop (3) placed between two cams (4) and controlled by the gas pressure in the cylinder (5) fixes the limits of the stroke of the fuel rack, and so adjusts the maximum and minimum quantity of fuel to the various loads defined by the working pressure. However, as already explained, the minimum delivery of a gas generator at reduced loads is more than the turbine can absorb. At such loads the excess gas is discharged to atmosphere. This blow-off takes place through a valve placed on the gas pipe between the gas generators and the turbine.

V.—PERFORMANCE

The power delivered by a gas generator is expressed as gas horsepower, by assuming an adiabatic expansion of the gases from the gas generator delivery pressure to atmospheric pressure. This gas horsepower must be multiplied by the efficiency of the turbine/reduction gear in order to obtain the useful shaft horsepower at the reduction gear outshaft.

The available gas horsepower increases with increasing working pressure, as is shown by Fig. 20, on which are plotted, on a base of working pressure, the following: the power, the gas temperature, the number of strokes per minute, the efficiency into gas, and the quantity λ, which is the ratio of the air trapped in the cylinder to the air theoretically required for complete combustion.

At the present state of development of free-piston gas generators of the GS-34 type, the gas pressure at continuous

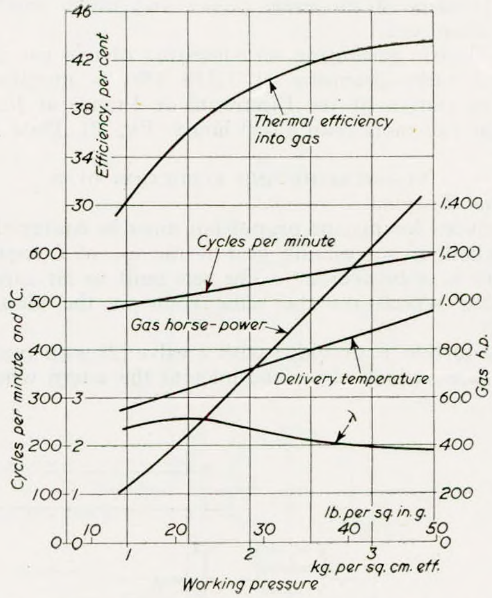


FIG. 20—*Curves of GS-34 characteristics plotted against working pressure*

maximum rating is fixed at about 3 kg./cm.^{2} (43lb. per sq in. gauge).

At a gas pressure of 3 kg./cm.^{2}, the value of λ is 2. This value is relatively low, but nevertheless, owing to the great excess of scavenge air, perfect combustion with an invisible exhaust is obtained. Overload tests at a gas pressure of 3·6 kg./cm.^{2} (51lb. per sq. in. gauge) at 1,530 gas horsepower have been carried out, the exhaust being practically invisible.

It is beyond question that the rated power for continuous duty, fixed at the moment at 1,250 gas horsepower for the GS-34, could be increased in the near future. Tests are at present being carried out with the object of increasing the power output.

In due course actual performance figures will be provided by two 850-ton 1,250 s.h.p. coastal vessels, which are now nearing completion in the yard of Augustin Normand at Le Havre. In the absence of these figures, the following table gives the component and overall efficiencies of two electrical generating sets. The figures are actual in the case of the 700 kW. set, several of which are now in service, and estimated in the case of a 6,000 kW. set incorporating eight gas generators, which has been ordered by the Electricité de France.

Power at the alternator terminals, kW.	700	6,000
Efficiency into gas of the gas generators (measured), per cent ...	43	43
Turbine efficiency (measured on a six-stage 1,000 h.p. impulse turbine), per cent	86	86*
Reduction gear efficiency (estimated), per cent	97	—†
Alternator efficiency, per cent... ...	96	97
Losses in the gas pipes, as a percentage of the adiabatic work, per cent ...	1·3	1·3
Losses in the auxiliaries as a percentage of the work, per cent ...	1·2	1·0
Overall efficiency at the alternator terminals, per cent	33·5	35

* Estimated efficiency for turbine at 3,000 r.p.m.
† Direct coupled.

These efficiencies are comparable to those of a good Diesel set, but the fuel costs will be in many cases below those of a Diesel engine because the gas generator, with its single highly supercharged combustion chamber of 1,000 h.p., can much more easily be run on heavy fuel than the Diesel engine with many cylinders of moderate power and much smaller combustion chambers.

An electric generating set consisting of two gas generators and of a turbo-alternator of 1,250 kW. is installed in the generating station of the Electricité de France at Reims, and has so far run more than 4,000 hours (Fig. 21, Plate 2).

VI.—TURBINE AND REDUCTION GEAR

(a) Axial Turbines

Turbines for marine propulsion must be equipped with an astern wheel or a reversing gear if the use of reversible pitch propellers is to be avoided. The sets built so far carry one or two astern wheels on the same rotor as the ahead wheels (Fig. 22).

The turbine is provided with a valve (2) with two opposed seating faces, which closes the inlet to the astern wheels in its upper position, and in its lower position the inlet to the ahead wheels. This valve is operated by a servo-motor (3) controlled from the operating station by a hydraulic relay. Another relay controls the fuel injection pump.

Before starting the gas generators, the valve (2) is placed at its mid-position. After starting the gas generators, the gas pressure is about 0·4 kg./cm.^{2} (3·7lb. per sq. in. gauge); the turbine remains stationary. When the controlling handwheel is turned in the ahead direction, the valve moves upwards and the gas flowing to the astern wheel is throttled, so that the torque of the ahead turbine predominates and the turbine begins to rotate in the ahead direction. The progressive decrease of the passage area towards the astern turbine causes an increase in the gas pressure. During this period of regulation the gas generator is running at minimum delivery.

As soon as the gas passage towards the astern turbine is completely closed the relay between the operating station and the gas generator comes into action and increases the quantity of fuel injected. When the handwheel is rotated further in the ahead direction, the delivery and the gas pressure are increased up to full load.

To reduce power these operations are reversed. The gas pressure and the delivery are first brought back to the minimum values compatible with the turbine flow area by reducing the quantity of fuel; then the passage towards the astern turbine is gradually opened.

The astern power is controlled by the same handwheel, turned in the opposite direction.

The turbine wheels that are not supplied with gas rotate in the direction opposite to their normal direction of rotation; this might lead to important windage losses. Tests on a bladed wheel have shown that such losses can, by means of a special design, be limited to about 2½ per cent of the power delivered. These tests have also shown that the ahead turbine, when motored in the astern direction, does not require special ventilation of the blading to avoid overheating.

An alternative solution is to use a reversing gear. The most promising arrangement from the point of view of efficiency, bulk, and cost would be an epicyclic reverse-reduction gear, incorporating oil-operated brakes. For a double reduction from about 10,000 r.p.m. to 120 r.p.m. at the propeller shaft, such a gear would have an efficiency of about 94-95 per cent. There would, of course, be no turbine reverse wheel windage loss, and no other transmission losses would have to be allowed for.

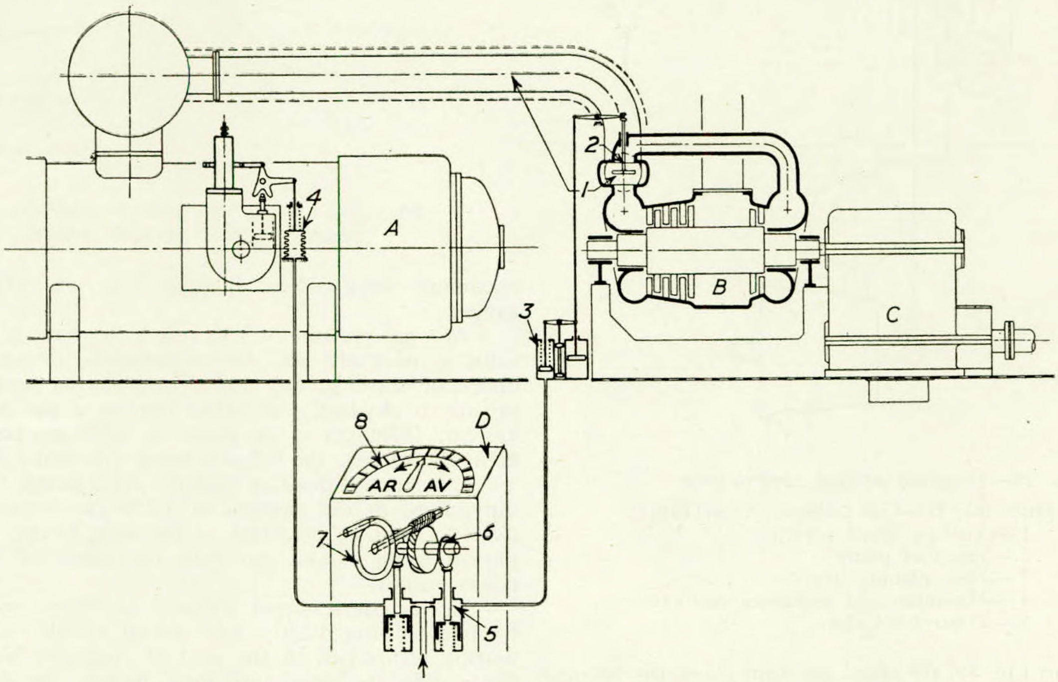


FIG. 22—*Diagram of control scheme for marine set with reversing turbine*

A.—Gas generator; B.—Reversing turbine; C.—Reduction gear; D.—Control block.
1.—Gas pipe to turbine; 2.—Turbine inlet control valve; 3.—Oil servo motor; 4.—Gas-generator fuel quantity control cylinder; 5.—Oil pressure control valve; 6.—Operating cam; 7.—Operating handwheel; 8.—Indicating quadrant.

(b) Radial Flow Turbines

Another possibility is to use a radial inward flow turbine instead of an axial flow turbine, with a single stage rotor. For an application where the power follows the cube law, such as marine propulsion, a good relation between the inlet gas velocity and the rotor tip speed could be maintained down to fairly low loads.

The radial turbine, which is still in the project stage, would be fitted with adjustable nozzle vanes at inlet and with an outlet diffuser to recover some of the leaving loss. At inlet to the rotor, the gas stream would have no axial component of velocity, but nozzle vanes could be rotated through nearly 180 degrees so as to reverse the direction of rotation of the turbine. Thus, no separate reverse turbine would be required and the attendant windage losses would be eliminated.

By adjusting the nozzle vanes from nearly circumferential at full load to nearly radial at very light load, the nozzle throat area could be increased as the load falls, with a corresponding increase in the turbine's swallowing capacity. This would be particularly suitable for use with sets consisting of only one or two gas generators, owing to the impossibility of reducing their gas delivery below a certain figure, as explained in a previous section. Preliminary calculations indicate that the blow-off point could be lowered very considerably as a result, if not eliminated altogether.

The delivery conditions of a gas generator at full load, owing to the moderate pressure ratio, would be suitable for a single stage turbine, with a rotor tip speed of about 1,500ft. per sec. The combined thermal and centrifugal hoop stresses at the moderate mean rotor temperature of about 300 deg. C. would not be excessive at this tip speed, and a ferritic steel could be used. The rotor could be made from a forging, and the turbine as a whole should be a good deal cheaper than an axial one.

VII.—TYPICAL MARINE INSTALLATIONS

A large range of powers can be obtained with the same type of gas generator, as is shown by Figs. 23 and 24, which give examples of installations of a single gas generator set of 900 s.h.p. and a ten gas generator set of 9,000 s.h.p. The comparison of the weights of these two installations brings out one of the most interesting points of free-piston gas-generator installations, namely, that the weight per horsepower is practically independent of the total power of the installation.

Power at propeller shaft, h.p.... ...	900	9,000
Weight of gas generators, tons ...	8·5	85
Weight of turbine and reduction gear, tons	4·0	65
Weight of gas piping, tons	0·5	7
Weight of auxiliaries of propelling machinery, tons	2·5	15
Total weight, tons	15·5	172
Specific weight, lb. per h.p.	38	42

It is self-evident that, according to the available space, the gas generators may be located very differently with respect to the turbine and reduction gear. For instance, in an installation which must be as short as possible, the gas generators can be placed above the turbine and reduction gear. It is also possible to use part of the delivery gases to feed electrical generating sets, in which the Diesel engine is replaced by a gas turbine. The weight and the bulk of such sets is, of course, much less than those of equivalent crankshaft Diesel engine sets. An installation including a group of free-piston gas generators feeding on the one hand the main turbine and on the other hand the turbine of the electrical generating set would have in addition the advantage of using only one type of fuel and of having a great reliability in service. The number of spares would be reduced, because the whole installation would be based on a single engine bore.

Periodical overhauls of a gas generator installation can be carried out in rotation during the voyage. By installing one reserve gas generator, the whole power will remain available, even during the overhaul of a gas generator. Turbine overhauls are much less frequent. Endurance tests have shown that the turbine does not become fouled. Thus, the electrical generating set of the oil tanker *Bethsabee,* consisting of a gas turbine fed by a free-piston gas generator, has run for more than 4,500 hours without the turbine ever having had to be opened for cleaning.

Ships equipped with gas generator sets have shown great flexibility of the controls when the turbine is provided with an astern wheel. The manœuvrability is the same as with a steam turbine.

VIII.—COMPOUND GAS GENERATOR/STEAM PLANT

It seems likely that several advantages could be obtained from compounding a gas generator/turbine plant with a steam plant. The basic principle of such a plant would be to use in a steam boiler the heat rejected from the gas side, consisting of the heat in the gas turbine exhaust and the coolant loss of the gas generator. Such a plant could also include a combustion chamber between the gas generator and the gas turbine, in

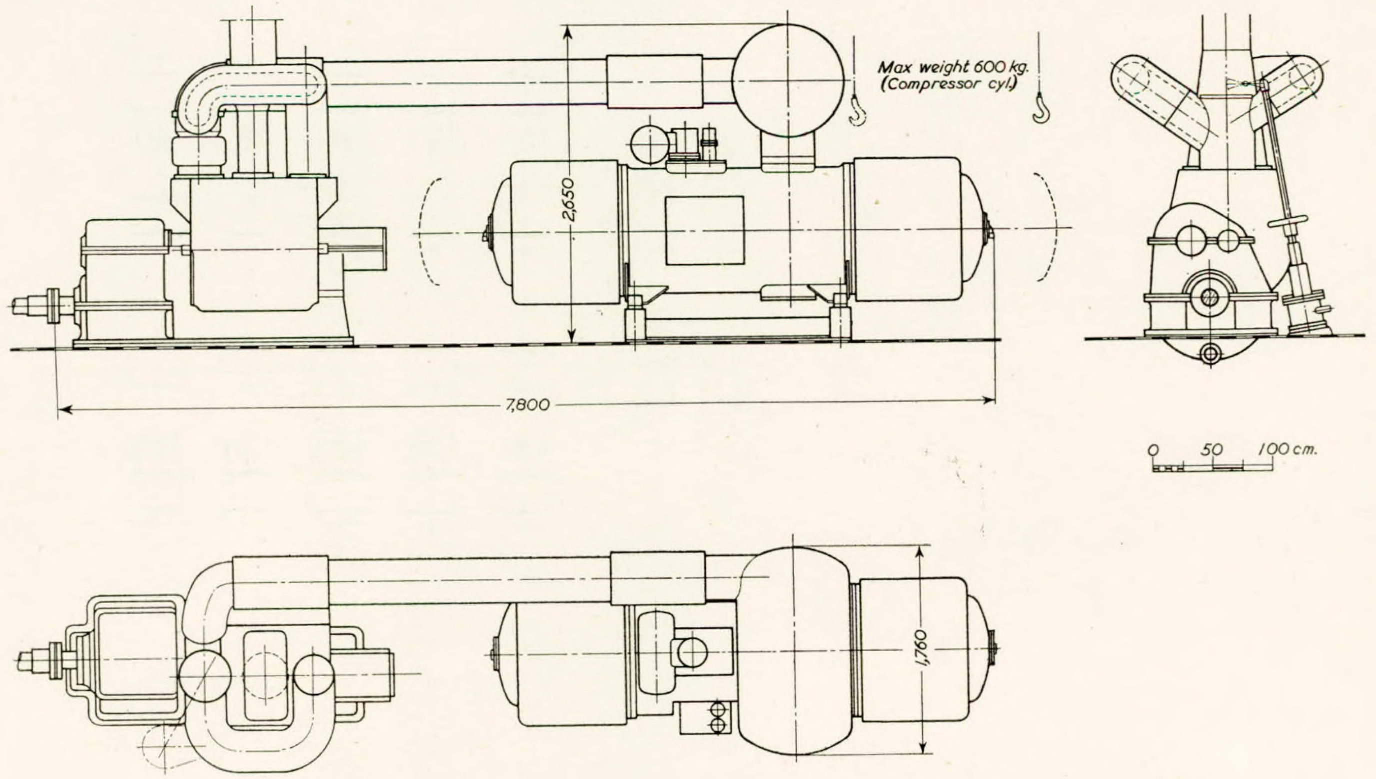


FIG. 23—*900-s.h.p. marine set with one reversing gas turbine fed by one GS-34 gas generator*

which additional fuel would be burnt at very high efficiency, but in that case a high temperature gas turbine would be required; an overall fuel consumption at the shaft of the order of 0·35lb. per shaft horsepower hour could be achieved.

If a combustion chamber is not provided, no special materials are needed for the gas turbine. A number of arrangements are possible, two of which are illustrated diagrammatically in Fig. 25.

In the first of these, Case A1 shown by the full lines, the shaft powers of the steam and gas turbine are equal. The combustion air for the steam boiler is provided entirely by the gas turbine exhaust, so that an air preheater cannot be used. This means that the feed water alone is available for cooling the stack gases, so that the provision of feed heaters would, whatever their advantage from other points of view, increase the stack losses. It will be seen that with the gas and steam conditions shown, the overall specific fuel consumption would be 0·40lb. per shaft horsepower hour.

In the other example, Case A2, a second steam circuit, shown dotted, has been added. In this circuit the steam at a

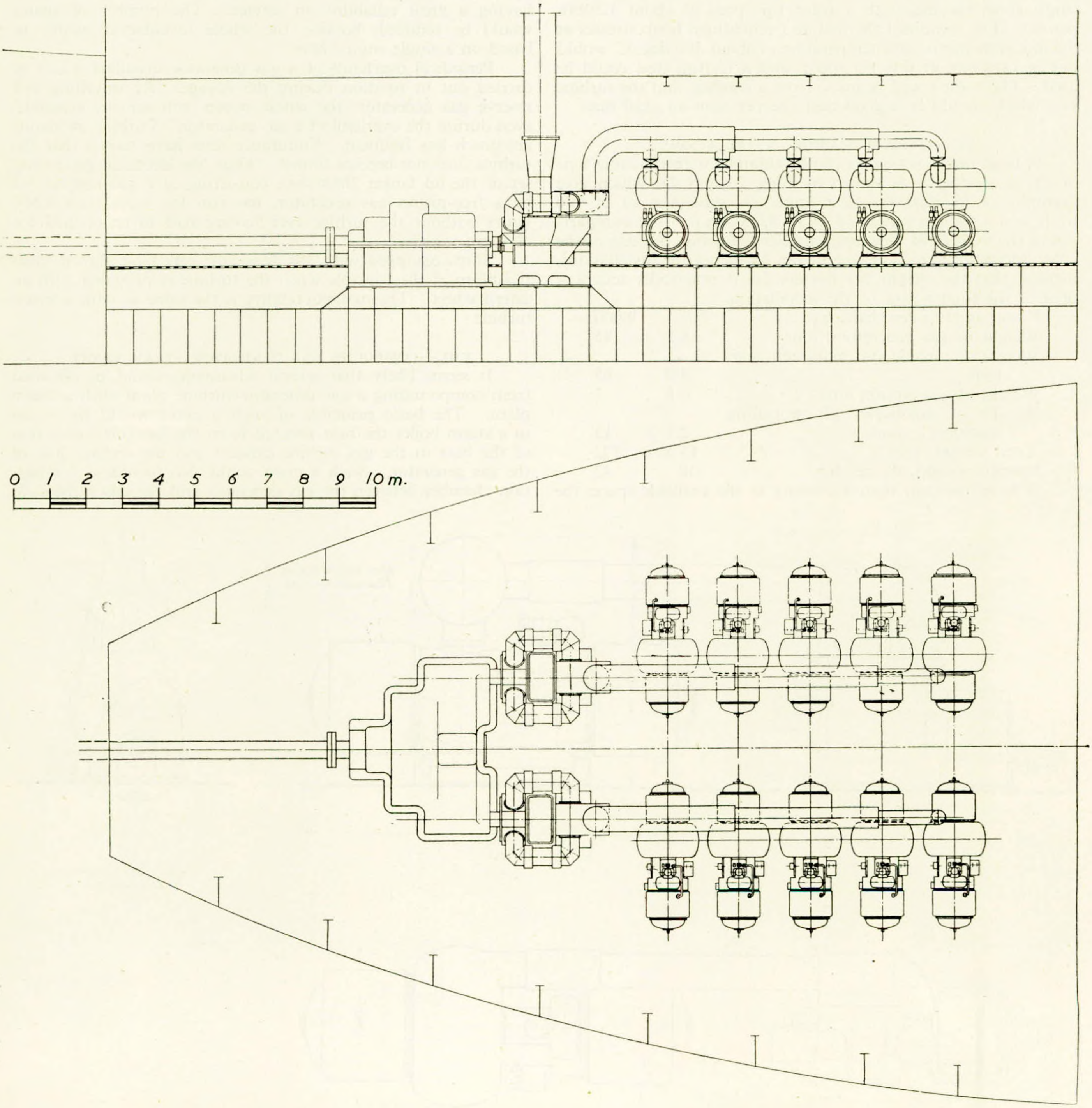


FIG. 24—*9,000-s.h.p. marine set with two reversing gas turbines, each fed by five GS-34 gas generators*

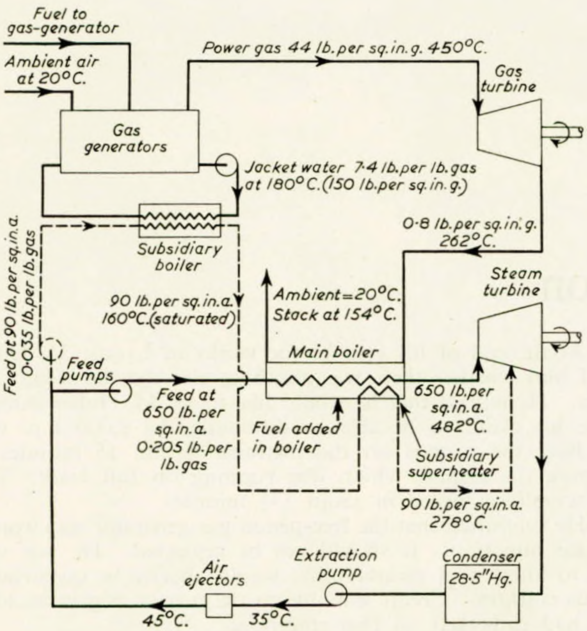


FIG. 25—*Compound gas generator/steam plant*

Case A1: No combustion chamber; equal power outputs (full lines)

Case A2: With subsidiary boiler working on coolant (dotted lines)

Case	A1	A2
Water flow lb./lb gas ...	0·205	0·240
Gas power, per cent ...	50	47·4 (of total)
Steam power, per cent ...	50	52·6 (of total)
Efficiency, per cent ...	34·4	36·1
Special construction, lb./s.h.p. hr.	0·400	0·381

lower pressure, arbitrarily taken to be 90lb. per sq. in. absolute, picks up as much as possible of the heat in the gas generator coolant. If this heat is converted into shaft work, the overall specific fuel consumption would be reduced to about 0·38lb. per shaft horsepower hour.

It should be noted that in such a plant, the heat added in the boiler fuel is utilized at a substantially lower efficiency than that in the gas generator fuel, and that an improvement in overall efficiency could be achieved by reducing the boiler fuel and the main steam flow, and so reducing the proportion of the total power provided by the steam side.

IX.—CONCLUSION

In the light of the foregoing, it is, in the authors' opinion, reasonable to assume that the following conclusions will prove correct:—

1. That the free-piston gas generator/turbine engine system can now be considered as a practical alternative to the steam, Diesel, and gas turbine engine systems.
2. That, while the efficiencies obtained so far are a little lower than those of the best Diesel engines, it seems likely that an evolution similar to that of Diesel engines will lead in due course to efficiencies at least equal to those of Diesel engines.
3. That, for a wide range of applications, the free-piston gas generator/turbine engine system will compare favourably with established engine systems, not only in respect of cost of production and of installation but also in regard to overall operating costs.

Finally, the authors would take this opportunity to pay tribute on the one hand to the Marquis de Pescara, who first succeeded in making a free-piston engine work, and on the other to the two leaders of French industry who had the foresight and courage to back the development of the Pescara free-piston engine system, M. Arthur Bommelaer, President of the Société Alsacienne de Constructions Mécaniques, and at a later stage, M. Ernest Mercier, President of the Union de l'Electricité up to the time when it was nationalized and became part of the Electricité de France.

Discussion

MR. H. N. PEMBERTON (Member of Council), in opening the discussion, said that all present would no doubt appreciate the significance of the Pescara free-piston engine and its application to marine propulsion.

The question naturally came to mind as to why they should only now be hearing of the details of these interesting developments.

About three years previously he had had the privilege of visiting France and seeing a prototype gas generator on test. Since that date it seemed to have been difficult to get any information at all about how the development was going, and he suggested that perhaps their French friends might have been a little more forthcoming at an earlier stage. Indeed, prospective licencees were still very much in the dark regarding manufacturing and operational problems.

It was probably fair to suggest that this country had in recent years concentrated very much on the gas turbine itself, and it had been in that direction that money had been spent. Progress in marine application was slow but the study so absorbing that little interest had hitherto been shown in the free-piston gas-generator turbine as a serious proposition for ship propulsion.

The gas turbine difficulties remained, though there was confidence that their ultimate solution would be achieved. Here, however, was an economic unit ready for immediate application and already tried out in service in small coaster-type ships.

One of the main difficulties in respect of the gas turbine was the limitation on gas temperature due to the inadequacy of materials and, of course, the problem of blade cooling, on which some progress was now being made.

In the gas-generator turbine unit the inlet gas temperature was no higher than in modern steam turbine practice. There were also all the advantages of a supercharged two-stroke Diesel engine without the mechanical contrivances which were necessary to convert reciprocating motion into rotary motion. From the practical point of view, they were presented with the means of getting rid of a number of problems when they eliminated connecting rods and crankshafts. They also had a reasonable economy in weight. It would be noted that in the smaller CS-75 unit a specific weight of 22lb. per shaft horse power was possible when epicyclic reverse reduction gear was used.

A further point which was of considerable interest was that in their reciprocating unit the authors had achieved a balanced engine in which there were no unbalanced forces to induce vibration in the hull structure.

Another possible trouble had been eliminated in that there was no crankcase in which lubricating oil might be contaminated and in which explosions might occur.

Further, according to the authors in section viii of the paper, by combining the free-piston gas generator and gas turbine with a boiler and steam turbine unit, it was possible to achieve the best of two worlds, that of the internal combustion engine and that of the steam turbine. From the figures given in the paper, however, it was questionable whether sufficient economy of fuel constumption could be obtained which would justify this somewhat complicated arrangement.

At the end of his visit to the works in Lyons, M. Huber asked him whether there was anything else that he would like to see. He replied that he would like to see M. Huber demonstrate his claim to be able to dismantle the 2,000 h.p. unit and have the pistons on the platform within 15 minutes of stopping the engine, which was running on full load. This was actually achieved in about 13½ minutes.

He submitted that the free-piston gas-generator was worthy of their attention. It should not be neglected. He was very glad to know that several units would shortly be constructed in this country. Credit was due to the marine engine builders who had embarked on that enterprise.

MR. J. CALDERWOOD, M.Sc. (Vice-President) congratulated the authors not only upon a most interesting and valuable paper but also upon the way in which it had been presented. He said that they had put it over in a manner which made it very easy to follow for those who had perhaps not had time to read it before coming to the meeting, as must have happened to many.

He was sure that he could spend the whole evening discussing points on the subject with the authors, but he would try to keep his remarks reasonably short.

There was one thing in relation to the development which interested him very much, and it applied to the other people who had worked on it. That was where the loss in efficiency was. If one studied the cycle academically, as nearly everybody did when they first began to be concerned with the development, one would expect a fuel consumption very substantially better than that of a Diesel engine.

Working back from the efficiencies which were being obtained—in that respect, the figures in the paper were equal to or better than any other results that had been published—there seemed to be a loss in mechanical or pumping losses of about 25-30 per cent of the probable indicated power. Could the authors state whether that was represented largely by pressure losses? Little detailed information about tests relating to pressures at the compressor delivery, scavenging pressure losses, losses through valves, etc., had been published. It seemed to him that that fact alone justified the claim of the authors towards the end of the paper that that type of cycle, so far as efficiency in performance was concerned, was really in the position of the Diesel engine of 20 to 25 years ago, and that when fully developed the consumption of the gas-generator turbine should at least equal that of the Diesel engine.

The authors referred to pairing cylinders. That would appear to him immediately to give an improvement of about 5-8 per cent in consumption due to the smaller pressure fluctuations in the scavenge air receiver. With regard to the paired cylinders, he would be interested to know whether the authors had yet carried out any comparative tests with separately working cylinders. He would also be interested if the authors could tell the meeting how they controlled the synchronizing of the two cylinders. He had known for some time that they were synchronizing pairs of cylinders but he had been greatly puzzled as to how the synchronizing was done.

The reasons stated in the paper in favour of the inward type of compressor were very valid. There were, however, arguments on the other side. He had never been able to make up his mind on the pros and cons of the subject. It was only within the last few days that it had occurred to him that there was a reason for the inward type which the authors had not stated. The type seemed to him to be much more self-balancing with the requirements of the turbine in that if the compressor was delivering more air than the turbine could take, it would automatically reduce its delivery on the inward-compressing type without alteration of any of the other functions, whereas in the case of the outward-compressing type there appeared to be an increase in delivery. The outward-compressive type seemed to him to be in a state of unstable balance, and he would like the views of the authors on the subject.

He was a little disappointed about the weights. He had expected them to be lower. Actually, they were not very much lower, particularly those for the CS-75 size, than those of a highly pressure-charged Diesel engine running at the same revolution speed. He had expected that they would have been substantially lower.

He would like some information from the authors about the piston rings. He knew that in the early stages of all free-piston work there had been considerable trouble with excessive piston ring wear. He knew the special type of rings which had been adopted on certain free-piston air compressors, and which seemed to have got over the trouble. Could the authors tell the meeting whether the trouble had been overcome in the case of the gas generators, and if so, how?

There was one point on which he wished to disagree with the authors on behalf of all their sea-going friends. The authors suggested that the sets would be overhauled one at a time at sea. For the sake of the sea-going engineer, he would say that it was better that they should wait until they got into port because the sets were so light that they could be lifted out for a major overhaul and replaced by spare sets.

He wished to raise one point about cooling losses. The authors remarked that the losses were low and, therefore, the heat stresses were low. However, so far as he could see, if one tried to make a heat balance from the figures in the paper, they must be percentage-wise higher than in a straight Diesel engine. He wondered whether the authors could give any information about any heat balances.

On the question of low cost fuel, the authors made rather a sweeping claim, and he considered that they ought to modify it. "Low cost fuel" covered an enormous range of types of oil, and he did not think that the authors meant that the engine could run safely on anything which could be burnt in a boiler.

The authors also said that there was no vanadium trouble occurring. Obviously, there would not be at the temperatures quoted, but with the worst of the low cost fuels it was impossible to get perfect combustion. In those circumstances, would there not be blade deposits of carbon?

One point in the paper had interested him considerably. The authors referred very briefly to the use of a radial flow turbine instead of an axial flow turbine. That seemed to him personally to be the ideal solution for that type of turbine. The pressure ratios were such that a single-stage machine might be possible. The efficiency should be reasonable, though perhaps not quite so good as with an axial flow turbine, but loss of effiicency would be compensated by reduction in capital cost. Further, it might be possible to save the cost of the reversing gear. He did not know how far the authors had gone with that method, but he would be extremely interested if they could expand the point in replying to the discussion.

MR. B. WOOD, M.A., said that he was not a marine engineer—he was present merely through the courtesy of the Institute—and all he could hope to do was to give the land view of the development. He did not propose to call it a new development because to some people it was not new; it was just about as new as the gas turbine which, of course, was very old.

In 1939 he went to Switzerland to see the first commercial gas turbine on load at Baden, and on the way back he stopped at Belfort by arrangement with Mr. Muntz to see one of the first free-piston gas generators undergoing tests. Therefore, it had appeared to him to be a very long time before the paper was presented. If there was any criticism of the paper, it was perhaps that the reasons for the delay were not sufficiently explained. The consequence of the mystery which surrounded certain aspects might be that some people would take an agnostic attitude and would only believe the plant were a success if they were shown it. They had, therefore, been waiting for some years to have their convictions overridden by being able to see the machine actually doing the job.

Unlike the gas turbine, which was so simple that anyone became an expert after a first reading of it, the free-piston engine was extraordinarily complicated, and he was sure that there were very few people outside the organizations of the authors who really understood it at all. It had to be borne in mind that there was no fixed stroke, no fixed inner and outer dead centre and no fixed speed. They had to cope not only with thermodynamics but also with mechanical dynamics. He thought the designer did not know until a machine was built at what speed it would care to run. It seemed to him that the development was a very awkward one to pursue. They started off by building an engine, and when they had got it to work they could then build identical engines, but he believed that there might be difficulty in expanding up to a bigger size. That was also the experience in the case of boilers. No boiler manufacturer really knew what his new boiler would do; he discovered it only by trial. He believed that it was the same with the free-piston engine. Until theory and practice were more canalized, they would more or less have to wait and see what happened.

Going back to the times before the war, the attraction of the free-piston engine was that it offered possibilities of providing efficient compression in a piston compressor with, at that time, a better efficiency than was obtainable with a rotary compressor. It also avoided the difficulty of high temperature, which at one time held back the development of the gas turbine. But unfortunately the position deteriorated from the point of view of the free-piston generator during the war because of the enormous amount of money which was spent on the development of refractory alloys and axial compressors. Therefore, after the war the gas turbine seemed to have sidetracked the free-piston engine, rendering it as an unnecessary step on the way.

Just after the war, Sulzer Brothers in Switzerland built a 7,000 h.p. free-piston generator complete and ran it on test, but apparently they completely abandoned interest in it and just wrote it off. Some other people had done some similar things in a smaller way. Therefore, it seemed as if there must have been some very severe difficulties, and it would be interesting to learn more about them. Lubrication seemed to have been one of the difficulties, as was shown by the piston ring troubles mentioned by Mr. Calderwood, but he could not believe that those were the only troubles.

It was the fashion with any new prime mover to claim that it could do everything and beat everything, but they all knew that that was not true and that it was sometimes necessary to overstate a case to encourage people. In the light of what had happened with the gas turbine—it had to be admitted that some of the gas turbines were terrible flops—it would be far better not to overstate the case but to try to keep near the truth so that people would not be greatly disappointed.

Reference had already been made to the weight issue. That was an example of excessive optimism. The engines were not light. They were lighter than the marine Diesels, but they were not light compared with what was available. The specific weights quoted on page 213 of 38lb. per h.p. at 900 h.p. and 42lb. per h.p. at 9,000 were not as good as those obtainable by existing gas turbines of commercial types. The Ruston machine of 900 kW weighed $11\frac{1}{2}$ tons without regenator and 5 tons without alternator, which was 9·31b. per h.p., so that it was much lighter than the free-piston generator. At the other end of the scale, a 22·5 MW Brown Boveri set weighed 320 tons, or 200 tons without alternator, which was 18·5lb. per h.p.

If they were right in thinking that the free-piston machines would have roughly the same specific weight irrespective of the size, it was clear that they were a long way behind what was regarded by the aircraft people as preposterously clumsy and idiotically heavy land gas turbines.

He thought that weight did not matter to Merchant Marine engineers, although he was only guessing. Weight *per se* meant nothing to him. For land purposes, weight was not objectionable. What was wanted was efficiency, reliability and moderate first cost. Probably those considerations applied in ships other than naval boats.

What he would like to know was what the machines were going to cost. He assumed that a lot of the work was in simple machining of cast iron and that the cost per ton should not be high. He suggested that it might come out at about £500 to £700 per ton as in the case of Diesel engines, in which case the machine should be able to compete very easily in many duties with gas turbines because its efficiency would be very much better and its cost would be either the same or less than that of the other.

Interesting figures might be quoted for comparison. The Ruston gas turbine cost £3,000 per ton and the Brown Boveri machine £1,200 per ton. Therefore, even though the weights of the gas turbines were low, it was possible that the cost per kilowatt would be higher than that of the new form of machine with the efficiency of the Diesel engine.

ING. E. R. GROSCHEL said that the paper was an excellent one dealing with a fascinating subject, and the authors were to be congratulated upon the way they had handled the problems involved.

There were, however, a few questions that he wished to ask, and one of them was connected with the specific fuel consumption relative to medium- or slow-speed Diesel engines such as the Doxford.

On page 205 of the paper the authors mentioned a specific fuel consumption of 144 gr. per gas h.p. hr. (0·320lb. per gas h.p. hr.) and a thermal efficiency of 43 per cent. A shipowner was, however, primarily interested in the overall fuel consumption of the whole installation. What would be the fuel consumption of an installation of, say, 5,000 s.h.p. in the case of a free-piston gas generator cum turbine of present-day design? He ventured to say that from that point of view the free-piston gas generator cum turbine installation could hardly compete with, say, a Doxford direct-drive installation, the Doxford engine, as far as he could recollect, having a consumption of 0·325lb. per b.h.p. per hr. for a supercharged experimental engine. He supposed that the free-piston gas generator was quite capable of running on heavy fuels such as Ord oil of 1,500 seconds Redwood I and over, as most of the large marine piston engines did nowadays.

While the free-piston gas generator cum turbine installation undoubtedly possessed a higher efficiency than the ordinary combustion chamber gas turbine installation—due, as the authors rightly pointed out, to the high compression/expansion ratio—the combustion chamber gas turbine was capable of further development in using solid fuel (coal dust), the cheapest fuel available. He doubted if the coal dust-fed reciprocating piston engine would ever arrive.

On page 214 of the paper an installation was shown of a 9,000-s.h.p. marine set with two reversing gas turbines, each fed by five GS-34 gas generators. A battery of free-piston gas generators must present quite a silencing problem. At the Skoda Works before the war he had some experience in experimenting with a Junkers free-piston air compressor. The operational noise was deafening, and it was advisable to wear ear plugs when the compressor was operating. Perhaps that problem had now been overcome with free-piston gas generators? What had the naval authorities to say with regard to the noise of operation? During the war, the German battleships could be heard some miles away when their Diesel machinery was working; on a quiet night, this was not very pleasant for them for it drew attention to their position.

The authors stressed the case for easy piston maintenance. This was generally the case with opposed piston engines (except double crankshaft engines) where no high-pressure cylinder head joints had to be broken. It was, therefore, applicable to large opposed-piston marine engines. He wondered if any experience in maintenance costs over several years for free-piston gas generator units had been accumulated and, if so, how these compared with, say, a Doxford installation. The Doxford engine was a very economical one, and bearing in mind that it was a slow-speed engine, it ran almost indefinitely, whereas the free-piston gas generator was a high-speed unit.

It was stated that at loads below 30 per cent of full load the free-piston gas generator had to blow-off gas. This uneconomical feature would seem to exclude it from trawler installations where a comparatively small unit had to work for many hours at small fractional loads, as was the case when trawling. Of course, a trawler would not warrant the installation of several free-piston gas generators. However, the Doxford engine ran almost indefinitely at 20-25 revolutions per minute and ticked over quite happily.

He thanked the authors for a most enjoyable lecture. The free-piston gas generator was undoubtedly a very fascinating machine and should have many applications.

COMMANDER(E) N. H. MALIM, R.N., said that he wished to make a few remarks from the point of view of the Royal Navy.

The paramount interest was the saving of weight and space in H.M. ships. That was probably the over-riding requirement, but at the same time they must not sacrifice any of the other essentials such as reliability and ease of production (under which heading he also included fitting out), operation and maintenance.

The "weight and space" was weight and space of machinery plus fuel for a given endurance, which was the factor that was really important to the Royal Navy. It was not just machinery weight. Economy in operation was not of such importance to the Royal Navy as it was to the merchant service except insofar as the saving of fuel weight was concerned.

From what they had heard, it would seem that the reliability of the free-piston gas generator, particularly the GS-34, was well established. They had been told something of the relatively easy methods of production and of the simplification of maintenance compared with the Diesel engine.

From certain preliminary studies which had been carried out by the Royal Navy, the savings in weight and space—again, of machinery plus fuel—appeared to be substantial compared with the existing modern machinery in certain applications. In general, they covered the range of 5,000-8,000 h.p., which necessitated the use of a multiplicity of gas generators. It was on that side of it that he would like to ask the authors if they could give the answers to one or two points.

He wondered why, in the illustrations which had been given of the higher-powered merchant units, the authors had split the banks of generators. Each bank was supplying its own single turbine, and the turbines were coupled together into a single gearbox. It would seem that they were doubling up on the losses in the gear case, the bearing losses, and it would not seem to be very economical from the turbine point of view.

It was normal practice in the Royal Navy to operate for long periods under wartime conditions at cruising powers which varied from 10 to 30-40 per cent of the full installed power. It was under those conditions that the Royal Navy was most interested in economy in order to keep the fuel weight down.

It was also necessary, while steaming under those reduced power conditions, to be able to come up to full power very rapidly. He was wondering whether, when one was steaming—or whatever the word was now—on a bank of just a few of the generators, it was possible to bring the extra generators into operation quickly so that one could increase power rapidly.

In relation to the multiplicity of generators, he also wondered what system had been devised for controlling the banks of generators so that one obtained equal output and distribution of gas load between them in the installation.

MR. E. W. CRANSTON, Wh.Sc. (Member of Council) said that he must first congratulate the authors on producing such an interesting and detailed paper on a subject which was little known—or was little known before the paper was published—in this country.

In Switzerland, from about 1940 onwards—a considerable amount of experimental work was carried out with free-piston gas generators with the object of supplying power gas for gas turbine experiments.

Materials suitable for high temperatures experienced in combustion turbines were not available at that time, and the gas generators allowed very useful development work to proceed using ordinary blading material in the gas turbines. A description of those free-piston gas generators was given in Mr. Calderwood's paper* read before the North East Coast Institution of Engineers and Shipbuilders in 1946.

The first unit constructed was with a cylinder 250 mm. bore and a combined stroke of 1,100 mm., giving an output of 475 b.h.p. at the turbine. The unit ran at 280 cycles per minute, and power gas was supplied at 4 atmospheres gauge and 500 deg. C. Allowing a turbine efficiency of 85 per cent, the fuel consumption was equivalent to 0·38lb. per b.h.p. per hr., which agreed fairly well with the author's figures.

Afterwards, a 2,000-h.p. unit was made with a cylinder 400 mm. bore and 1,220 mm. combined stroke running at 350 cycles per minute. In that unit, the air supply first of all passed through an axial compressor driven through a small gas turbine fed from the power gas supply. Afterwards the air was cooled and entered the free-piston unit at a pressure of 1·8 atmospheres absolute. That modification to the usual arrangement had two advantages: (1) the size and weight of the free-piston unit were both reduced because of the reduction in diameter of the compressor piston; (2) the pressure of the power-gas supply could be increased.

Turning to the application of the free-piston gas generator to marine use, he considered that it would be limited to special installations for the following reasons:—

(*a*) The maximum size of gas generator would appear to be about 2,000 h.p. For large powers a number of generators would be required, and no doubt five working and, perhaps, one spare would be the limit.

(*b*) A shipowner running cargo ships requiring an installation up to 10,000 s.h.p. had the alternative of using, say, the well-known direct-coupled Diesel engine with a fuel consumption of about 0·35lb. per b.h.p. per hr. without any complications of using step-down gearing or reversing gearing, or the free-piston gas generator with a turbine which could be reversible with reduction gear or with electric transmission. In any case, the fuel consumption of the latter would be something over 0·38lb. per s.h.p. on the basis of 0·32lb. per h.p. of the gas producer, a turbine efficiency of 85 per cent and an allowance for either windage losses in the astern turbine or electric transmission losses. Further, Diesel engines were available to run reliably on heavy fuels up to 3,000 seconds Redwood I at 100 deg. F. or over (see W. Kilchenmann's paper, April 1953), whereas the capabilities of the gas generator to burn heavy fuels was not generally known. He would like to ask the authors what provision was made for the collection and removal of sludge which must find its way past the rings on the power pistons.

(*c*) It was true that the free-piston gas generator did away with the connecting rods, bearings and crankshaft of the Diesel engine, but usually those parts caused very little trouble on a modern engine. On the other hand, pressure charging of engines made the conditions for piston rings more onerous, and particulars of piston ring life and liner wear for the gas generators would be welcome.

(*d*) For installations over about 10,000 s.h.p., gas generators were hardly practicable because of the large number of units required, although they could be considered for special ships where headroom was limited. The most obvious development for larger powers was the use of the pressure-charged two-stroke engine with exhaust-gas driven turbo-blowers. Those single-acting engines could be built in single units up to 15,000 b.h.p., could burn heavy fuel, and had a fuel consumption of about 0·35lb. per b.h.p.

COMMANDER(E) F. ROBERTS, O.B.E., D.S.C., R.N.(ret.) (Member) said that he was grateful to the authors for their paper, which was on a subject in which he was very interested. He wished to ask a few questions about details which did not appear in the paper.

He wished to know what was the maximum firing pressure of the cycle and what was the estimated or approximate maximum temperature.

Secondly, he wished to know what the losses were in the cushion cylinder owing to heat generated during compression, some of which must pass through the sides of the cylinder and must be counted against the theoretical efficiency of the process. He wondered whether that loss could be reduced by a suitable lagging of the cushion cylinder.

Thirdly, considering the power/weight ratio, an increase in oscillation speed would obviously bring down the weight for a given power. What were the limitations on the oscillation speed or the cycle speed of the free-piston engine, and at the increased speed, what was the incidence of Diesel knock? Could the power/weight ratio be improved by the use of light alloy pistons or pistons made partly of light alloys and the use of light alloys in various parts of the structure of the engine?

Finally, what was the lubricating oil consumption per horsepower generated, so that one could make a comparison with the Diesel, the gas turbine and the steam turbine plant in that respect?

MR. A. F. EVANS (Member) said that when Gothawerken started the development in about 1938, everyone laughed at it. Gothawerken used a supercharged two-cycle engine which propelled by its exhaust gases an ordinary steam engine, and excellent results were obtained.

He would be glad if the authors would say whether they had investigated the use of the gas produced by the engine for an air engine of, say, 100 h.p. with cut-off regulation. That would be very interesting.

He wished to make one very small criticism of the paper. The authors had shown quite a complicated lot of gear in respect of cooling the pistons, thus throwing heat away. Why did they not insulate their piston crowns?

MR. A. L. WACHAL said that the development of the free-piston power unit first attracted his attention some twenty-five years ago, but surprisingly not in the countries—Spain, France and Great Britain—which are now associated with the development of the idea, but in Poland.

A. Wicinski, a Polish engineer, designed and developed an opposed-piston unit very similar to that now described. His idea evolved from a single-piston unit in which the piston oscillated from one end of the cylinder to the other, discharging exhaust and compressed air through the consecutive sets of ports. It would be interesting to know if, and to what extent, the work done in Poland had contributed to the stage now

* Calderwood, J. 1946. "Some Researches on Internal Combustion Prime Movers". Trans.N.E.C.I.E. and S., Vol. 62, p. 283.

reached in the development of the opposed-piston power plant.

Much had been said in praise of the unit and it was hard to conceive that it possessed so few drawbacks. In particular, one wondered about the combustion efficiency at high power output conditions, as there seemed to be little to facilitate an air swirl in a flat cylindrical combustion chamber. Under these conditions, one would expect premature Diesel smoke.

He would also like to know whether any attempt had ever been made to combine two or more opposed piston units in a common Diesel combustion chamber, thus forming a star-shape arrangement in which it would appear that there would be more scope for creation of beneficial air movement. One felt that in such a design, further reduction in the power/weight ratio would also be conceivable.

Correspondence

H. J. ALEXANDER (Member) thought the paper was full of interest and promise and the authors were to be congratulated. It would be appreciated if one point were made clear. In bad weather some vessels were susceptible to propeller racing; as yet no governor could give a near positive efficiency to cut down power when the ship's propeller was part out of the water and to increase power when the propeller was well covered. This would, in the case of the free-piston engine with gas turbine, produce a situation when the slowing of turbine wheel, due to total sudden immersion, created a build-up in the pipe line and gas collector to the point of blow-off. The speed governor gear (Fig. 19) would also cut down fuel while the propeller regained normal revolutions. Would the back pressure cause bad scavenging and cut-down of fuel due to the speed governor cause an engine, or multi-engine set, to stall?

MR. J. F. R. ELLISON (Member) wrote that the free-piston gas-generator appeared to be an important stepping stone in the development of the pure gas turbine, as the latter was still a problem from the metallurgist's point of view, owing to the extremely high gas temperature which must be allowed if an efficiency comparable with the Diesel engine and reasonable life from the turbine blades was to be obtained. He used the term "stepping stone" advisedly, as the most perfect form of prime mover was surely that which employed only rotary motion. Admittedly the free-piston gas generator had made it possible to eliminate crankshafts, heavy connecting rods and the like, with their attendant balancing and torsional problems, but the parts of Diesel machinery which involved heavy upkeep costs and caused most trouble, namely pistons, piston rings and liners, particularly when using residual fuel, formed an inherent part of the free-piston design.

In this connexion, he noted that the piston speed and, presumably the mean effective pressure, as the unit was highly supercharged, were both very much higher than in normal Diesel practice. Would the authors say how the pistons and rings stood up to these rigorous conditions, as it would appear that the latter must be kept in good order, so as to maintain the required high compression pressure.

It was understood that the disc type compressor valves shown in the drawing of the GS-34 unit were now being replaced by the well-known reed or "mouth organ" type. The writer had had experience of this type of valve in medium- and high-speed Diesel engines. The life of the thin steel valve plates was very short, especially if carbon build-up on the faces prevented them seating properly. It would be interesting to know how many hours the free-piston unit could run on, say, 1,500 seconds Redwood I at 100 deg. F. residual fuel, before the compressor delivery valves had to be withdrawn for cleaning, an operation which did not appear to be possible without taking off the end covers and compressor pistons. It was all very well to say, as the authors did on page 206 under the heading "Ease of Maintenance", that the two piston assemblies could be withdrawn in fifteen minutes and assembled ready for running in forty minutes. This might be possible in the builder's engine works, but with the various lines of demarcation and restrictive practices applied to repair work in this country today, it was a very different story.

It would add to the general usefulness of the paper if the overall performance of the compressor, turbine and gearing were expressed in terms of lb. of fuel per s.h.p., instead of quoting gas horsepower at the compressor outlet, but as far as could be seen from the figures on page 205, this figure worked out at about 0·38lb. per s.h.p. per hr., assuming a turbine efficiency of 86 per cent and gearing efficiency of 98 per cent. This result was about equal to but certainly not less than a slow-speed Diesel engine operating on residual fuel.

It was surprising to learn that the windage losses in the ahead blading of the turbine when rotating astern were only 2½ per cent and that the heat developed was not excessive. If steam turbine designers could achieve this result it would mean the elimination of costly desuperheaters or other methods of reducing the steam temperature when going astern.

The weight figures in terms of shaft horsepower compared with slow- and medium-speed (geared) Diesel machinery appeared attractive at first sight, but did this include the large lubricating oil pumps and coolers which would be needed to deal with the piston cooling oil? What quantity of lubricating oil would be required to circulate the twenty pistons of the 9,000 s.h.p. installation illustrated on page 214?

The compounding of the free-piston compressor turbine with a steam turbine might show an improved performance in theory, but it was complicated and would require considerable skill in operation to achieve the maximum efficiency. There was also the attendant risk of oil ingress to boilers, which was difficult to avoid in such combinations.

The free-piston compressor turbine was a most interesting development, and the authors were to be congratulated on the success which had already been achieved with this novel form of prime mover.

MR. K. MADDOCKS, B.Sc.Tech. (Associate Member) thought the authors were to be congratulated on making available this timely information on a plant which promised a solution to some of the major difficulties so far encountered in the application of the gas turbine for marine propulsion.

Certainly they had here an elimination of the headaches caused by: (*a*) the design of turbine inlet blades to withstand elevated temperature; (*b*) a satisfactory combustion chamber to produce that temperature; and (*c*) the heat transfer problem, which presumably led to the construction of the regenerator, a most prominent feature of the *Auris* installation. There remained, however, another basic problem; viz., machine efficiency which also prevented development of a plant capable of providing output power in a usable quantity. By the adoption of the concept of "gas horsepower" in quoting results for the GS-34 type, the efficiency of the turbine was neglected. It was suggested that a more realistic figure for comparison of specific fuel consumption and thermal efficiency with other installations would be provided if the adiabatic efficiency of an existing turbine were incorporated. This was assumed to be of the order of 85 per cent.

Synchronizing gear was described as a kinematic linkage for the GS-34 type and presumably replaced the rack and pinion gear used for the CS-75 type. Fig. 2 did not clarify the details of this gear, and the applicable reference number (14) in the vertical sectional elevation was shown located in the exhaust passage. Further explanation would be of assistance in appreciating the complete function of this gear in maintaining balance between the two pistons; e.g. the main variable forces acting on the pistons were due to the cushioning air on one side and the products of combustion on the other. The balance pipe partly took care of the former. Thus it would appear that the synchronizing gear was mainly used to control the injection system also a secondary adjustment of the cushioning pressure.

Regarding manœuvring, the authors' comments on a few reflexions on "stand-by" conditions of the 9,000-s.h.p. installation would be appreciated. For the direct reversing arrangement it was suggested that the flow of gases be directed to either the ahead or astern turbine by the operation of a single inlet control valve. With full power flow of gas this procedure might be justified for an emergency reversal, but a temporary gas bypass would seem to be a modification producing safer conditions. This could possibly be effected by use of the low power "blow-off" arrangements. Had any estimate yet been made of "Full ahead" to "Full astern" times?

Considering an extended "stop" where full power must be available, such as awaiting the arrival of a pilot offshore or when on hurricane watch at an anchorage, was the speed of starting up all the gas generators simultaneously sufficiently small to warrant their shut-down under these conditions? Also, in view of the use of the larger cushioning cylinders for the admission of starting air, did not this require a large reservoir capacity? A comparison of this quantity with the quantity required for twelve starts of the equivalent Diesel engine would be of interest.

No doubt an indication of some of the latter points would be forthcoming when the service performance of the smaller French installations was published. He, and no doubt many other marine engineers, hoped that this would not be long delayed. The keen interest shown in this type of machinery was evidenced by the fact that reputable British, American and other foreign companies were now developing these units.

MR. ALBERT STUBBS, B.Sc., noted that GS-34 was provided with a fuel injection ring which was bolted to two cylinders through conical joints, whereas GS-75 had not got the separate fuel injection ring. Was the injection ring of the same cast material as the cylinders? Were the conical joints expected to give improved tightness as compared with plain flat surfaces? Why was this separate injection ring not repeated in the GS-75 design?

Page 211 stated GS-34 gas pressure fixed at 3 kg. per cm.^{2} at C.M.R. Was this the optimum pressure condition when considered with a low gas turbine inlet temperature of 819 deg. F.? If gas turbine inlet temperature were required to be 1,200 deg. F., could this be obtained from a gas generator of the free-piston type, and what would be the effect on the design of the generator?

Was the best application of the free-piston gas generator to be found where the power units were of medium size only and where gas temperature was limited to approximately 900 deg. F. Medium-sized plant of the form shown in Fig. 25 suffered from over-complications which increased capital costs, maintenance and operational difficulties.

MR. D. D. WILLIAMSON (Member) thought the authors were to be congratulated on introducing a new engine system for the propulsion of ships which offered the shipowner an increase of dead weight, also an increase of cubic capacity, or smaller ships to do the same service. At a time when freights were low the introduction was most opportune; the system offered economies which would appear to justify taking whatever technical risks there were. The extra cargo carried was of the order of 5 per cent by weight or 8 per cent by measurement.

In respect of reliability, this system might be regarded in the light of a multi-boilered turbine installation; any one of the gas boilers would get the ship home, unlike the Diesel engine, where the failure of any one of the cylinders might bring about a complete stoppage. There was reason to believe that the gas turbine would prove to be as equally reliable a unit as the steam turbine. The comparative pressure was 50lb. per sq. in. gauge for the gas turbine against 600lb. per sq. in. gauge for the steam turbine, the temperature drop in the gas turbine being a modest 850 deg. F.-480 deg. F., whereas the temperature drop in a steam turbine at the same initial temperature was 850 deg. F.-92 deg. F.

Until more running experience was available, it was impossible to judge what the endurance was likely to be, and on this depended the cost of maintenance. The life of the liners could not be expected to equal those of a Diesel engine of 1,000 h.p. per cylinder, i.e. equal output. The diameters were about 1 to 2 and the wear could reasonably be expected to be about double. However, these liners were of relatively simple construction.

Authors' Reply

In reply to the discussion, the authors thanked Mr. Pemberton for his encouraging comments. They appreciated the point that it might prove complicated to use both steam and gas power on the propeller shaft; actually the arrangement shown in Fig. 25 should be regarded simply as an example illustrating the principle of using the heat in the exhaust to raise steam, and did not necessarily represent the best way of employing the steam.

The authors agreed with Mr. Calderwood that the efficiency obtained was somewhat less than the theoretical efficiency, and there was certainly scope for development in this direction. There were pumping losses in the compressor cylinder and engine case, and the mean pressure in the compressor cylinder during the delivery period was 10-15lb. per sq. in. higher than the mean gas pressure delivered by the gas generator. This loss was partly due to the limited volume of the engine case in which the air delivered by the compressors was stored till the scavenge ports opened, and this could be reduced, as Mr. Calderwood suggested, by "twinning", i.e. connecting two gas generators together with, in effect, a common engine case and synchronizing them exactly out of step with each other so that one engine scavenged the other.

In order to make two gas generators run 180 degrees out of phase, a pneumatic device was fitted in a pipe connecting the cushions of the two engines. When the two engines were not correctly out of phase, this device caused air to flow from one cushion to the other, i.e. from the leading gas generator to the lagging one. As a result the first gas generator slowed down and the second one was accelerated. This device worked on the principle that the sum of the two instantaneous cushion pressures was practically constant when the two gas generators were dephased by 180 degrees, and that this sum oscillated with increasing amplitude as they came more nearly into phase.

Other sources of loss were piston ring friction and a small loss in the cushion cylinder. These could no doubt be reduced by improved design. Any reduction in the length of stroke for a given output would reduce friction losses and any reduction in the clearance volume in the compressor cylinder would help in this direction. Reduction of direct heat loss to the engine pistons should also improve the efficiency. Hot piston crowns had been tried a long time ago but they had not been satisfactory mechanically.

Mr. Calderwood had discussed what would happen if the gas generator were delivering more gas than the turbine could take. In such a case the pressure at the turbine inlet, i.e. the gas generator delivery pressure, would rise until it was sufficient to force all the gas through the turbine. The set would then be operating at a higher power. To reduce the power the fuel quantity could be reduced, so giving a smaller stroke and less mass flow, and there would be a corresponding drop of pressure.

The outward compressing type of gas generator was not at a disadvantage in this particular respect because with either type the gas generator delivery was simply controlled by the quantity of fuel injected into the engine cylinder. In fact the outward compressing type had the advantage of being able to run at a lower percentage of full power without blowing off excess gas, as described in the paper.

When considering the weights quoted in the paper, it should be remembered that no serious effort had yet been made to produce a lightweight design of gas generator, and there was plenty of scope for improvements in this direction. It was also possible to increase the specific weight by having lighter moving parts and so raising the speed of oscillation.

The wear of the top engine piston ring with an untreated surface was less than 1 mm. per 1,000 hours. Thus, on a GS-34 running on residual fuel, a wear of 0·3 mm. per 1,000 hours had been measured at the scavenge end and of 0·6 mm. per 1,000 hours at the exhaust end. Tests were now being carried out with chromium-plated piston rings.

The difficulties encountered at first with the piston rings had been overcome by effective cooling of the pistons, which on the GS-34 were fitted with a heat barrier between crown and piston ring groove.

With regard to cooling losses as derived from a heat balance, the heat supplied to the gas generator in the form of fuel reappeared partly as heat in the coolant, including both water and oil, and partly as heat in the gas; there would be in addition a small radiation loss. The heat in the gas was represented by the temperature rise from ambient temperature to the gas delivery temperature. Taking the figures given in the paper for the GS-34, this was a temperature rise from 20 deg. C. to 437 deg. C., i.e. 417 deg. C., and it could be calculated that it amounted to 81·3 per cent of the heat in the fuel. Thus, by difference, the cooling and other losses were less than 20 per cent of the heat in the fuel, and this was confirmed by direct measurement. The useful heat, giving the gas horsepower, was only part of the heat in the gas, being the work that could be obtained from the gas in a perfect turbine by an adiabatic expansion to atmospheric pressure. In this case the temperature at the end of adiabatic expansion from 42·7lb. per sq. in. gauge would be 221 deg. C., and the work obtained was 43 per cent of the heat in the fuel. The drop from this value to the ambient temperature of 20 deg. C., i.e. 201 deg. C., represented heat rejected in the exhaust of the perfect turbine; it amounted to 38·3 per cent for the case considered.

GS-34 gas generators had operated in industrial applications for several thousand hours on fuel of 500 seconds Redwood I at 100 deg. F. containing 3·1 per cent sulphur. Tests with fuel oil No. 2, which had a viscosity of 1,500 seconds Redwood I at 100 deg. F. had been carried out on the test bed for several hundred hours and had given satisfactory results.

The authors were not able to give performance figures for a specially designed radial turbine of high power, as this was still in the project stage. They had, however, made some tests driving a centrifugal supercharger in reverse, which gave very promising results.

Mr. Wood was quite correct in stating that both thermodynamics and mechanical dynamics had to be considered together when designing a free-piston engine, but the authors could not agree with him when he went on to conclude that such characteristics as the speed could not, therefore, be predicted. In fact the motion of the pistons was governed by quite definite laws, admittedly different from those obtaining in other types of engine, but which were nevertheless amenable to analysis and calculation. Nor was there any particular difficulty in scaling an engine to a different size; in fact this had been done successfully with compressors.

However, having said this, it must be admitted that a new size of engine undoubtedly required a considerable amount of practical development work to get satisfactory performance.

It was never very easy to say on what basis different engine systems should be compared, but the authors thought Mr. Wood's figures for the weights of simple gas turbines were hardly significant in view of the substantial difference in fuel consumption. There was no doubt that the gas turbines by themselves were lighter, but their weights would be greatly increased if the heat exchangers necessary to reduce the fuel consumption to a comparable value were taken into account. Another point was that the figures quoted by Mr. Wood referred to engines driving an alternator, and did not appear to include the weights of the reduction gear and reversing means that would be necessary for marine applications. The authors' view was that for marine propulsion, provided the weight was reasonably low, there was little incentive to reduce it.

In reply to Mr. Groschel, the specific consumption of a 5,000 h.p. set with reverse reduction gear was 0·38lb. per shaft horsepower at the present stage of development. Several improvements now being carried out would reduce this figure.

The noise of a gas generator was greater than that of a slow-speed Diesel engine, but noticeably less than that of a medium-speed or high-speed Diesel engine. The GS-34 gas generators installed on board ship had shown that the noise was quite acceptable.

It was too soon to be able to give comparative figures for the cost of maintenance of the piston rings, pistons, etc., but it was thought that the ease with which the pistons could be withdrawn should make this relatively small. As regards the advantage of having eliminated the crankshafts, connecting rods, etc., it was admitted that in a good engine expenditure on these items might be small over a considerable period, but it seemed to be the case that the heaviest bills for repairs came when work inside the crankcase of a conventional Diesel engine was involved.

The authors did not agree with Mr. Groschel that a trawler would not warrant the installation of several gas generators. On the contrary, one of the advantages of the free-piston gas generator system was the possibility of using more than one gas generator. Quite apart from the question of blow-off, this made for cheaper maintenance, greater reliability, and flexibility of layout, as discussed in the paper. Furthermore, the authors understood that, while it could not be said that a trawler did not operate at 30 per cent or less of its normal maximum continuous power, it did so only very little more frequently than in any other vessel, i.e. when manœuvring or when "laid to" in very heavy weather on the fishing grounds; but that it would certainly be unusual for a trawler to operate for prolonged periods at 30 per cent of its normal maximum power.

In reply to Commander Malim, greater reliability was obtained by using two turbines instead of one. The loss of efficiency was very slight, because turbines of 4,500 h.p. and 9,000 h.p. had practically the same efficiency.

Satisfactory performance at cruising speed could be achieved on warships by shutting down some of the gas generators in the set. The cruising gas generators would supply a special sector of the turbine or possibly a cruising turbine.

A gas generator could be started in 15 to 20 seconds. If it had been kept warm by circulating the coolant from the other gas generators, it could take on load very rapidly, and less than a minute would be needed from the beginning of the starting operation to full load.

The load was controlled by means of an oil circuit, the oil pressure being varied by the operating handwheel of the set. This oil pressure acted on the fuel racks of the gas generators through a spring-loaded bellows unit. Since the same oil pressure was supplied to every gas generator, they shared the load equally.

There was no difficulty in grouping ten or more gas generators to supply a single turbine and it was not clear why Mr. Cranston should regard five gas generators per turbine as the maximum number.

The specific consumption of a gas generator set with a reversing turbine was 0·39lb. per shaft h.p. This was slightly more than that of slow-speed Diesel engines but in installations of more than 10,000 h.p. there were generally two engines on two lines of shafting. A gas generator set could easily develop powers greater than 10,000 h.p. on a single line of shafting. The difference in power of a ship with one line of shafting compared to one with two was about 11 per cent and, therefore, the fuel consumption per mile for a gas generator set would not be greater than that of a Diesel engine. Nor should it be forgotten that gas generators were at the beginning of their development when comparing them with the highly developed Diesel engines.

With regard to the disposal of sludge, the contaminated oil which made its way out through the scavenge ports was drained automatically from the bottom of the engine case.

The authors were not sure that it was right to minimize the advantage of doing away with the crankshaft. They had heard of a good many cases of trouble with these and when there was trouble it could be extremely serious. Apart from trouble in service, the cost and special equipment required for the manufacture of large crankshafts must not be lost sight of, especially in time of war.

In reply to Commander Roberts, the maximum Diesel cylinder pressure in the CS-75 was about 1,800lb. per sq. in., in the GS-34 it was slightly lower. The estimated maximum cycle temperature for the CS-75 was about 1,800 deg. C., corresponding to a fuel-air ratio only about half the theoretical, but this temperature in itself was not important, as it only occurred momentarily soon after inner dead point. The important thing was the rate of heat flow to the surface of the combustion chamber and the Diesel pistons, and this must be kept low enough to be dealt with by adequate cooling. The indicated loss in the two cushion cylinders of the CS-75 had been measured to be about 2 per cent of the heat in the fuel. This loss appeared in the form of heat in the cushions. A fraction of it would be radiated away from the cushion head, and the remainder would heat up the walls of the compressor cylinders and the compressor pistons, and in the absence of cooling this heat must be carried away by the air passing through the compressor cylinders. The net effect would be to raise the gas delivery temperature, and a substantial fraction, say, one-quarter, of the heat carried forward in this way would be recovered in the turbine as useful power.

The speed of a free-piston engine was inversely proportional to the square root of the weight of the moving parts, so that a reduction in weight to one-half would be required to get an increase in speed of about 40 per cent. This would probably be somewhere near the practical limit with either of the two present engines in a fully developed design. This would give a mean piston speed at full load of 2,250ft. per min. which would probably be regarded as too high for most marine applications.

It was probable that there would be more difficulty in maintaining control of the rate of pressure rise in the cylinder at the higher speed, and some increase in combustion noise was likely. However, this should not be serious, because of the relatively small increase in speed that was possible for the reason given.

The GS-34 oil consumption was 0·0036lb. per shaft horsepower/hour.

In reply to Mr. Evans, the authors considered that free-piston gas generators were not really suitable for powers as low as 100 h.p., although a reciprocating engine with cut-off regulation could be used to drive auxiliaries for a main engine of much larger power. The chief disadvantage of a reciprocating engine in this application was that it would be rather bulky for a given power, because of the relatively low gas delivery pressure.

The heat removed from the pistons by the cooling oil was certainly a loss to be regretted, and insulated piston crowns had been tried during the early stages of development to avoid this, but had been abandoned because they would not stand up to the duty and cracked. However, recent developments made it seem possibly worth while trying again. It was probable that it should not be attempted to insulate the crowns by any means completely, because they would then get extremely hot—getting on for 1,000 deg. C. at full load. Thus the full theoretical saving would not be obtained. Again, radiation from the hot crown to the cylinder walls would transfer some of the heat to the water jackets.

In reply to Mr. Wachal, the first free-piston compressors were built in France in 1925. Some years later the authors heard of the Polish machines, but the work of the Polish engineers did not in any way contribute to the development of the authors' engines.

Separate combustion chambers had been fitted to the GS-34 to increase the turbulence; 5 per cent of the fuel was injected into these post-combustion chambers after the injection by the direct injectors. The effect of these additional chambers was to increase the gas velocities at the end of the combustion period. In the CS-75 the direction and penetration of the sprays were chosen so as to distribute the fuel as uniformly as possible throughout the combustion chamber, to avoid depending too much on the swirls bringing the air to find the fuel.

A multicylinder engine with a single combustion chamber had never been considered practically because such an arrangement seemed more complicated and would increase the combustion chamber surface area and hence the heat losses.

Mr. Alexander had asked what would happen when the turbine slowed down owing to total sudden immersion of the propeller. There would be very little effect on the gas pressure, because the swallowing capacity of the turbine was substantially independent of its speed. In practice speed governors were not fitted to marine sets, and the load was controlled manually as shown diagrammatically in Fig. 22; there was, however, no objection in principle to controlling the load indirectly by adjusting the setting of a speed governor. In any case a sudden reduction in load would not cause the gas generator to stall because the stroke was prevented from becoming too short for proper scavenging by the minimum fuel cam.

The authors agreed with Mr. Ellison that the engine cylinder pressures and the piston speeds were higher than in a Diesel engine of the same power per cylinder, but the acceleration of the pistons at inner dead point was also substantially higher. As a result the pistons moved away very rapidly from their inner dead point and the piston rings were subjected to the high pressure and temperature gases for a relatively short time.

With regard to Mr. Ellison's points about valves, a special design of streamline valve, with the blades parallel to the direction of flow, had been developed for free-piston gas generators. This was the type now used for the GS-34 suction valves, and used from the beginning on the higher speed CS-75 prototypes. Fig. 26 showed the latest design of CS-75 suction valve, the type C3 Oxnam streamline valve, and illustrated the general construction; the delivery valves were similar. Fig. 27 showed how the hole area and clearance volume of streamline suction valves compared with those of three plate valves together giving about the same performance.

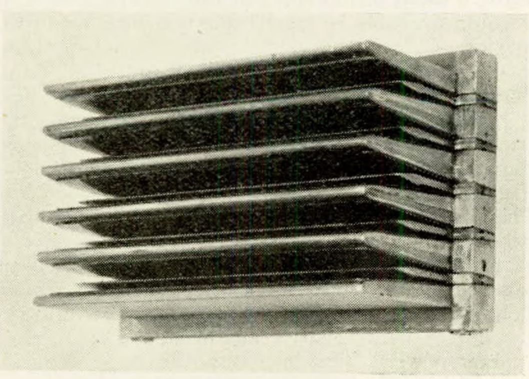


FIG. 26—*Type C3 Oxnam streamline suction valve viewed from outlet side*

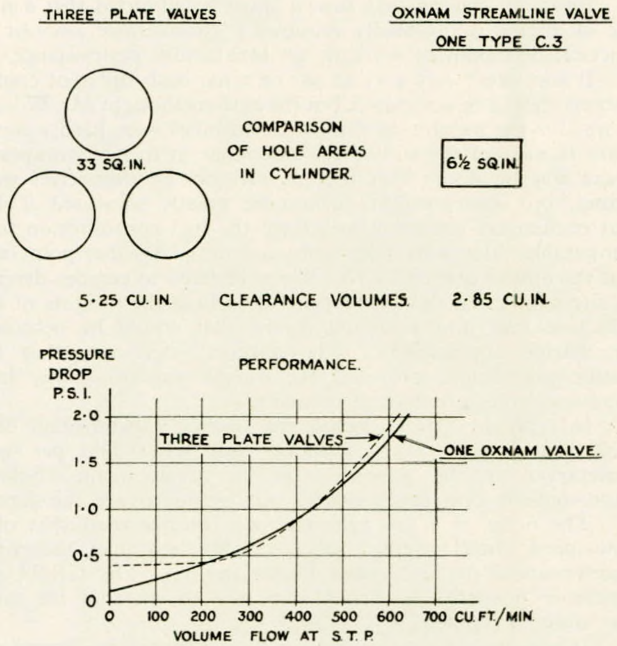


FIG. 27—*Comparison of compression suction valves*

These automatic valves were located at inlet and outlet from the compressor cylinders and consequently at a point where contamination from the exhaust gases did not occur. The combustion products went through the exhaust ports to the gas collector, and any blow-back through the scavenge ports into the engine case did not reach the delivery valves, so that the type of fuel used had no effect on the delivery valves.

In the CS-75, the speed at maximum power reached about 1,100 cycles per minute, and the maximum compressor cylinder pressure would be about 60lb. per sq. in. gauge. The endurance of the valves under these conditions was not known, but the latest type of delivery valve had been tested in a test compressor at 2,500 cycles per minute and 120lb. per sq. in. gauge; during this test the conditions were such that considerable coking occurred, and the valve was removed from time to time for cleaning. The valve was inspected after 1,000 hours and, apart from the coking, was found to be in excellent condition.

With regard to Mr. Ellison's remarks about restrictive practices, the authors did not think that these would apply to the comparatively simple operation of removing pistons, since for this work only fitters and their labourers would be employed, and the problem of demarcation would not arise.

The windage losses in the astern wheel of a reversing turbine, which at the beginning had been 6 per cent, had now been reduced to 2 per cent by certain modifications, which consisted in fitting a screen near the astern wheel to prevent any flow of air radially outwards under the centrifugal action of the astern wheel rotating in the forward direction.

The figures for the weights given in the paper did not include the circulating pumps for oil and for water. For the 9,000 shaft horsepower installation illustrated in Fig. 24, the flow of water and oil required was about 200 $m.^3$/hr. each, and the weight of the pumps would be about 3 per cent of the weight of the installation. The coolers were smaller than on Diesel engines because less heat was lost to the water and oil.

In reply to Mr. Maddocks, the principal function of the synchronizing gear was to transmit from one set of moving parts to the other any slight differences in friction or air pressure in the two ends of the engine. The GS-34 synchronizing gear was illustrated in Fig. 28. It consisted of a parallelogram linkage similar to a lazy tongs. As the piston moved in and out, the tongs closed and opened, and the linkage rotated about the central pivot. The authors apologized for reference number 14 appearing in the wrong place on the longitudinal vertical section in the preprint of the paper. This had now been corrected.

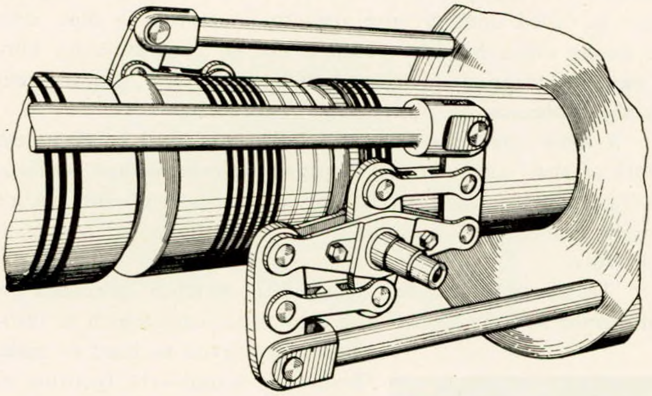


FIG. 28—*Sketch of GS-34 synchronizing gear*

Trials on board ship had shown that there was no difficulty in reversing quickly from full ahead to full astern. The time from full ahead to full astern depended on the speed of action of the relay operating the reversing valve. On existing installations this time was 5 to 7 seconds but it could be easily reduced to 2 seconds. As a guide to the sort of performance obtainable in practice, recent tests on the 850-ton coastal vessel *Merignac* showed that it took 13 seconds to reduce the propeller shaft speed from full ahead of 220 r.p.m. to zero, and a further 17 seconds to reach 100 r.p.m. in reverse.

The starting air required for the GS-34 was about 2 kg. at a pressure of 32 kg./cm.^{2}. Thus the air quantity was practically the same as on a Diesel engine of the same power, but the storage capacity could be less than for a Diesel engine because reversals from ahead to astern were carried out without stopping the gas generator. Thus, as mentioned in the paper, much less starting air was used in the long run.

In reply to Mr. Stubbs, there was a central steel ring in the GS-34 liner, the outboard portions being of cast iron, whereas the CS-75 liner consisted of a single cast iron piece with a reinforcing ring (Fig. 9, Plate 1). These were two methods of achieving the desired strength, and the divergence was due to the fact that the two gas generators had been independently designed. In fact the joints between the central steel ring and the outboard parts of the liner in the GS-34 were not conical, but stepped; it was regretted that the small scale print in Fig. 2 was misleading in this respect.

As discussed in section III(b) of the paper, it was a characteristic of the free-piston gas-generator turbine system that the efficiency did not suffer from using a turbine inlet temperature that would be considered low by conventional gas turbine standards. Consequently the concept of a desired turbine inlet temperature did not arise, and there would be no question of aiming at a higher temperature, such as 1,200 deg. F. for its own sake

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at the Institute on Tuesday, 11th May 1954

An Ordinary Meeting was held at the Institute on Tuesday, 11th May 1954, at 5.30 p.m., when a paper entitled "The Free-piston Gas-generator Turbine as a Power Plant for Ship Propulsion", by F. A. I. Muntz and Robert Huber, was presented and discussed. Mr. J. P. Campbell (Chairman of Council) was in the Chair. Members and visitors present totalled 180 and nine speakers took part in the discussion.

A vote of thanks to the authors, proposed by the Chairman, was accorded by acclamation. The meeting ended at 7.30 p.m.

Correspondence on "New Metals in Engineering"

Following the publication in the July TRANSACTIONS of the paper by Dr. L. B. Pfeil on "New Metals in Engineering", and the discussion thereon, the letter reproduced below has been received from Mr. S. A. E. Wells:—

"I have read with great interest the paper by Dr. Pfeil appearing in the July issue of the *Transactions of the Institute of Marine Engineers*, and the discussion which followed its presentation. As one of the principals concerned with the development of the new propeller alloy 'Novoston', I would like to draw attention to a discrepancy in the remarks made by Mr. G. S. Jackson in the discussion, when he says that 'Novoston' and 'Nikalium' are of similar composition.

"This is not true, for whereas 'Nikalium' is an alloy of the conventional 80/10/5/5 high tensile aluminium bronze type, in which the main addition elements, apart from aluminium, are iron and nickel, 'Novoston', on the other hand, although containing aluminium, iron and nickel, has a high manganese content in excess of 10 per cent. It is, therefore, quite a different type of material to 'Nikalium'.

"The difficulties associated with the casting of aluminium bronzes, particularly when required to produce large castings, are stressed by Dr. Pfeil in his paper and endorsed by Mr. R. N. Richardson in the discussion. 'Novoston' is substantially free from these difficulties and the necessity for a high strength, corrosion-resisting alloy suitable for propellers with good casting properties, was one of the main reasons for its development".

OBITUARY

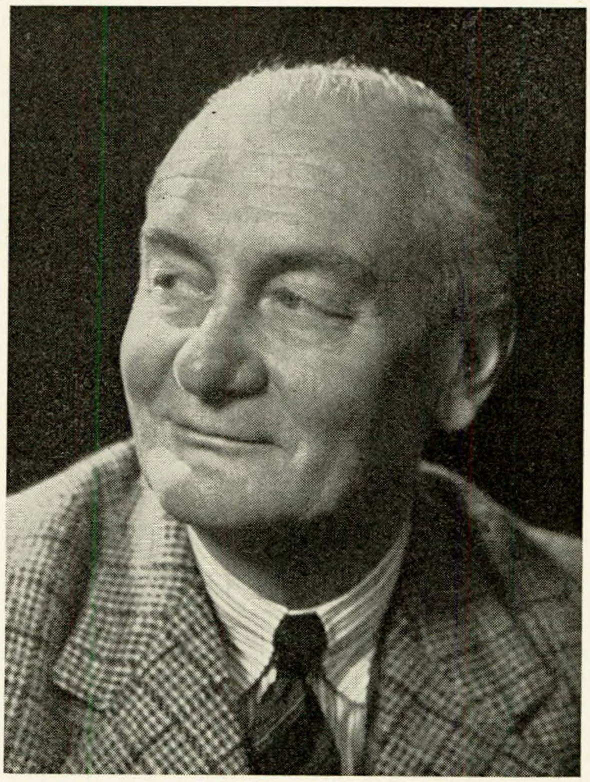
Arthur Edward Crighton

An appreciation by Robert Rainie, M.C. (Member)

It was an opportunity I willingly grasped when the Chairman, certain Members of Council and the Honorary Treasurer asked me to undertake the duty of writing an appreciation of my friend and most sincere comrade over a period of fifty-three years.

We met in Glasgow in 1901 where Arthur Crighton was serving his apprenticeship at John Brown's of Clydebank, and although there are those whom circumstances—during the intervening years—permitted to have more frequent contact with him than me, none, I believe, could have understood him better and admired him more.

Educated at Antwerp and Saint Lawrence College, Ramsgate, and after completing his apprenticeship at Clydebank, Arthur Crighton spent a year at Wallsend, and then joined the Royal Mail Steam Packet Company as a sea-going engineer in August 1908, leaving the sea with a First Class Certificate of Competency in September 1911.



Two years were spent at Belfast as an Assistant Outside Manager with Harland and Wolff, Ltd., followed by his appointment as Assistant Superintendent Engineer to the Royal Mail Company, and then, in 1914, he was appointed Superintendent Engineer of that company. The additional duties of Naval Architect were assumed in 1942. Arthur Crighton relinquished active duties in 1946, but remained as Consultant to the company until 1948, when he finally retired.

It was as Superintendent that he was best known in the shipping and shipbuilding world, and to the Council, Officers and Members of the Institute.

It was not within the power of *all* those who came in contact with him to put the X-rays on a man like Arthur Crighton, and to discover the qualities of mind, heart and character which brought him to the position he occupied in the shipping industry, and in the esteem of those who were privileged to call him friend.

The motives which actuated him in relation to the "Brethren of the Cloth", were indeed worthy motives. To enlarge on this I would quote Richard Hughes: —

"As a profession sea-going seems somewhat of an anomaly. The *raison d'être* of it is economic, and yet the practice of it is judged by standards which are not economic at all, which can only be called moral, and which are peculiar to it. For the working of a ship calls for certain qualities—virtues if you like—which do not seem to be appropriate today to the relations of employers and employed on shore. The shore worker's liability is limited: the seafarer's unlimited. The seafarer may be called upon to give the utmost that he is able, even to laying down his life. This is not an imposition on him, a piece of chicanery on the part of his employers, it is inherent in the profession he practices".

I know that it was because of a profound belief in the truth of that statement that Arthur Crighton worked continuously, though unobtrusively, for the welfare of the marine engineer in general, and his own staff, ashore and afloat, in particular.

To that belief can be attributed the practical provision for all members of the "cloth" and their dependents which he initiated and worked so hard to make financially sound—the Institute of Marine Engineers' Guild of Benevolence.

Arthur Crighton served the Institute in many capacities since first becoming a member in 1916. He was elected a Member of Council for four years, was a member of the Social Events Committee, and representative of the Institute on three Committees of the British Standards Institution. He became a Vice-President in 1927, and in recognition of his many services was made an Honorary Vice-President in 1947.

No one did more to enhance the position of superintendent engineers in the shipping world than Arthur Crighton, or to emphasize the increasing importance of technical departments of shipping companies to the success of their operations than he.

Eminently able to look at both sides of a question, but seldom in doubt as to where the right lay—with a gift of leading those who differed from him to his own point of view—Arthur Crighton fitted easily into his position and carried out his duties adroitly and wisely, developing cordial relationships with those with whom he had to deal.

All that should become a man Arthur Crighton stood possessed of: —

A devoted wife and family.

The high regard of his peers and colleagues.

The respect and loyalty of his subordinates.

The esteem, and indeed affection of those privileged to know him well, of whom I—perhaps the oldest of these—forever conscious of help in adversity, consideration and counsel always; with my pen "dipped in tears" do write—Good-bye, my Friend.

William Veysey Lang (Member 2041) was born in 1864. He was educated at Taunton, in Somerset, and apprenticed in 1884 to the Uskside Engineering Company, Newport, Monmouthshire, as a fitter and turner. From 1885-88 he was a seagoing engineer and at the end of this period he obtained a First Class Board of Trade Certificate. He then went to work at the Eastern Dry Dock in Newport as assistant to his father, but six years later he went into partnership with William Alsop at Ribbleside Works, Preston, Lancashire, where they built small craft such as tugs. After a year or so as assistant manager at Smith's Dock Company of South Shields he was appointed superintendent engineer to Watts, Watts and Co., Ltd., a position he held until 1928, when he went into partial retirement, being retained in a consultative capacity, however, for several years.

Mr. Lang had been a Member of the Institute since 1908 and retained his interest in its activities until his death, in his ninetieth year, on 18th July 1954.

Alexander Lightbody (Member 11227) was born at Carluke in Lanarkshire in 1882; he was educated at the Coatbridge Technical College and the Royal Technical College, Glasgow, and served an apprenticeship with Hugh Martin and Sons, Carluke, from 1902-07. From 1907-09 he was employed as a fitter with Babcock and Wilcox, Ltd., of Renfrew, and with the New Arrol Johnstone Gas Company, Paisley. He moved to Aberdeen in 1911 to join C. F. Wilson and Company, thus beginning his long career with this firm. For many years he was responsible for the design, production, and development of Wilson Diesel engines; in 1914 he was appointed works manager, he became responsible for research in 1919, and in 1924 he was chairman and director of the company. From 1932 until his death on 9th August 1954 he was managing director of the company; from the same date he was managing director also of Tullos Limited, agricultural and general engineers of Aberdeen, but resigned this appointment in 1947 in order to devote all his time to extending the work of the other company. During the last war Mr. Lightbody was selected by the Ministry of Aircraft Production to build and manage the Tullos factory in Aberdeen for the manufacture of sparking plugs for aero engines.

Mr. Lightbody was a member of the Institution of Mechanical Engineers and a former governor of Robert Gordon's College, Aberdeen. He had been a Member of the Institute since 1947.

Robert McWilliam (Member 9014) died in hospital on 24th August 1954, aged sixty-five. He served an apprenticeship with J. T. Chapman and Company, Hampton Wick, and went to sea in ships of the China Navigation Company and the Donaldson Line between 1912-16; from 1916-19 he served as engineer lieutenant with the Royal Naval Reserve. He obtained a First Class Board of Trade Certificate and for the next twenty years held various appointments—he was engineer-in-charge for the Western Canada Pulp and Paper Company in 1920-21, second engineer with the Port of London Authority from 1922-26, assistant to Mr. S. C. Ross, consulting engineer, during 1926-27, and from then until 1941 he was assistant to Mr. L. S. Polychroniadis, consulting engineer. In 1941 he was appointed engineer superintendent to the Stanhope Steamship Co., Ltd., and remained with that company until, in 1951, he went into partnership with Meikle, Tolliday, McWilliam and Co., Ltd., consulting engineers and marine surveyors in London.

Mr. McWilliam was elected to membership of the Institute in 1939.

George Mosley (Member 7016) was born in 1892. He served an apprenticeship with Armstrong, Whitworth and Co., Ltd., and then joined the Eagle Oil and Shipping Co., Ltd.; he sailed continuously in their tankers until 1947, when he retired after twenty-seven years as chief engineer. After his retirement he sailed for short periods as chief engineer of the steamships *Regent Hawk, Fort Eric* and *Margay* but owing to ill health he was obliged to retire completely in February 1953 and died suddenly in April 1954.

Mr. Mosley was elected a Member of the Institute in 1932.

Ewart Alfred Palliser (Member 8203) died suddenly on 15th August 1954. He was born in 1901 and from 1916-22 served an engineering apprenticeship with Richardsons, Westgarth and Co., Ltd., following this with a further year at Smith's Dock Co., Ltd. In 1923 he joined the British India Steam Navigation Co., Ltd., and remained in employment with the company until his death. He was promoted chief engineer in 1943.

Mr. Palliser had been a Member of the Institute since 1936.

Antonio Ravelo (Member 9983) was born in 1894. He served an apprenticeship with Blandy Brothers and Company, Las Palmas, from 1906-12, and then went to sea in ships of Cia. de Vapores Interinsulares Canarios, remaining with them until 1930 and serving for the last ten years as chief engineer; the next year the company was sold to Cia. Transmediterranea, for whom Mr. Ravelo worked until his death, which was sudden and unexpected, in 1951. Until 1938 he served as chief engineer at sea, finally of the motor ship *Ciudad De Mahon,* from 1938 until 1940 he was Second Shore Superintendent for the company in Las Palmas, when he was appointed Shore Superintendent. For two six-month periods between 1924 and 1927 he was a member of the Spanish Government's Board of Examiners of applicants for engineer's certificates, and from March 1929 until April 1931 he was a member of the consultative committee on shipping to the Spanish Government.

Mr. Ravelo was elected to membership of the Institute in 1944.

William George Riddell (Member 2770), formerly chairman and managing director of John Hastie and Co., Ltd., Greenock, died on 22nd July 1954 in Durban, South Africa. He was born near Jedburgh in 1871, served the greater part of his apprenticeship with James Howden and Company from 1888-1890, completing the last eighteen months with the Fairfield Shipbuilding and Engineering Co., Ltd. Afterwards he went to sea and obtained a First Class Board of Trade Certificate in 1896. He returned to the Fairfield Company as a draughtsman and in 1898 became under-manager with David Rowan and Company, Glasgow. In 1901 he went to Finland as manager of the shipyard and engineering works of Crichton (now Crichton-Vulcan) at Abo. Two years later he returned to Greenock as joint-managing director of John Hastie and Co., Ltd., with the late Mr. Walter Graham. On the latter's retirement in 1918 Mr. Riddell became managing director and later chairman, positions which he held until his retirement in 1934. For work done for the Admiralty in London in 1918 he was awarded the O.B.E.

In his later years Mr. Riddell wrote several books about his experiences, two of the best known being "Adventures of an Obscure Victorian" and "The Thankless Years"; these portray the Clyde engineer at work in the yards and at sea. Of his two sons, the younger, Mr. T. C. Riddell (Member), is now chairman and managing director of Hastie and Co., Ltd.

Mr. Riddell had been a Member of the Institute since 1913.

Llewelyn Owen Roberts (Associate 11204) was born in 1893. His apprenticeship was served from 1907-12 with the Graigddw Quarry Company, Blaenau Festiniog, and he continued in their employment until 1914. He was with the Army in the Middle East and Gallipoli from 1914-17 and for the next five years was fitter at Pembroke Dockyard, and second engineer in steamships; in 1920 he obtained a Second Class Board of Trade Certificate. From 1924-31 he was maintenance engineer in the transport department of William Evans and Co., Ltd., Cardiff, and went into business as a garage proprietor for the next five years. In 1936-7 he returned to sea as third engineer in a motorship and obtained a motor endorsement to his Second Class Certificate. From July 1937 until February 1939 he was first district engineer in charge of transports, Welsh area, for Russian Oil Products, and then chief engineer in charge of a motor ship. In 1939 he was engaged by the Mount Stuart Dry Dock Co., Ltd., first as charge hand fitter, then (from 1946) as safety officer in the Channel Dry Dock and Pontoon Co., Ltd., Cardiff. In 1948 he was appointed manager in Cardiff to Ferguson and Timpson, Ltd., a position he held until his sudden death in August 1954.

MARK LEE WATSON (Member 8163) was born in 1893. He served an apprenticeship with the Harton Coal Company, South Shields, from 1907-14, and sailed for the next two years as seventh to third engineer with the Eagle Oil Co., Ltd.; he was a first class engine room artificer in the Royal Navy from 1916-19 and sailed with the Rome Shipping Co., Ltd., for a year after that. He obtained a First Class Board of Trade Certificate. In 1920-21 Mr. Watson was a fitter with Palmers Shipbuilding and Engineering Co., Ltd., but returned to sea in 1922 in the service of the British Tanker Co., Ltd., with whom he served until his retirement due to ill health in 1949. For the first two years he was a second engineer and from 1924, except for some months during 1936 and 1937, when he qualified for and obtained a Motor Endorsement to his First Class Certificate, he was a chief engineer, being appointed senior post engineer in 1946. Mr. Watson died on 16th July 1954.

He had been a Member of the Institute since 1936.