FUEL SYSTEMS FOR MARINE GAS TURBINES

BY

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Introduction

Marine gas turbines may, for the purpose of this article, be classified as :—

(*a*) Emergency or stand-by sets

(*b*) Base-load sets.

In both cases they are used for power generation, not for main propulsion, and examples of each type, driving alternators or D.C. generators have been designed and constructed by W. H. Allen, Sons and Co. Ltd. for service on naval and mercantile vessels. To suit (*a*), the fuel system must operate in a fully automatic manner, so that on pressing a button the engine will start, accelerate to no-load speed, and accept load within one minute. To suit (*b*), a semi-automatic system is required, such that the machine is accelerated to no-load speed under the control of the operator and will then accept load automatically. Whatever the application, it is essential that a marine power-generating plant should be reliable, and this is particularly so of the fuel system. Reliability is an abstract quantity which cannot be measured other than by experience in service and the passage of time, and although it is possible to develop a satisfactory control system, either on a rig or during initial development of an engine, its behaviour in service is the means by which it must ultimately be judged.

Operating Principles

Control may be exercised in one of two ways :—

(*a*) Direct control of the fuel input by variation of the pump outlet. This is not possible where a gear pump is used, but is acceptable to a variable-stroke piston-type pump, for example.

(*b*) Constant excess output from the fuel pump, with by-pass of the surplus

The second type has been favoured by this Company, and fuel systems for both types of engine have been developed using it. The two basic components of the system are undoubtedly the fuel pump and the burner, which will be described. It is also necessary to provide by-pass control valves to ensure that the engine receives the right amount of fuel at the right time.

Two variables are available for operation of these valves :—

(i) Relay or servo oil pressure—this is controlled by the speed governor described in later paragraphs.

(ii) Air pressure—this is supplied from the compressor outlet and forms a useful additional method of control. It is used on both types of set but, as will be seen later, for a different purpose in each case. This does not constitute a power loss for the engine, as air flow is not required, merely pressure.

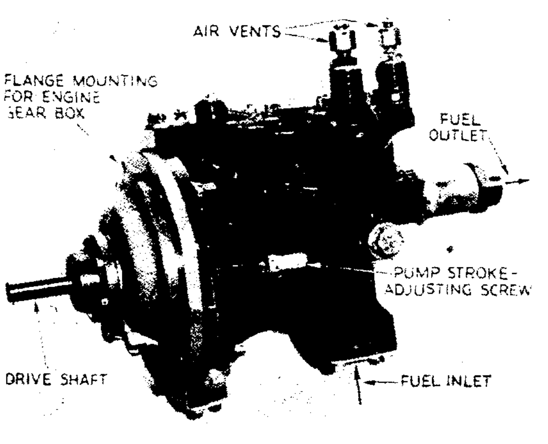

Fig. 1—Lucas Variable Stroke, Multi-Plunger Type Fuel Pump

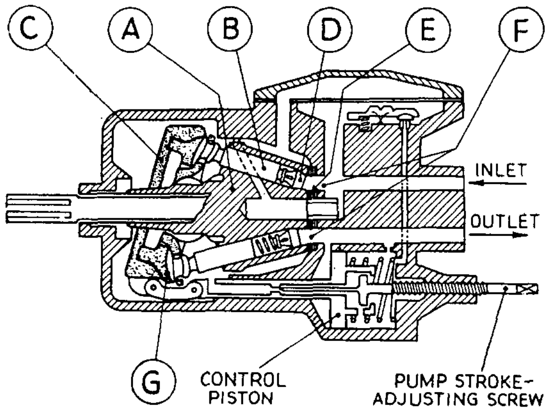

Fig. 2—Sectional Arrangement of Lucas Fuel Pump

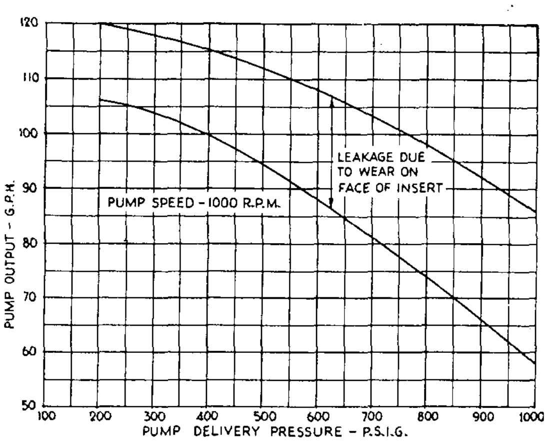

Fig. 3—Performance of Fuel Pump

The manner in which these variables are used will be described, but attention must first be focused on the pump and burner.

The Fuel Pump

The heart of the fuel system is the fuel pump. Its reliability, long life and maintenance of guaranteed output form the basis of success in any fuel system. The use of high pressures and fuels with little or no lubrication properties have made gear pumps generally unsuitable for marine sets. In the case of the Allen 350-kW set, marine Diesel fuel is used and this is particularly difficult for handling by a gear pump. There is hope, however, that a cheap, reliable gear pump will be used on Allen base-load sets where the fuel is gas-oil and the normal operating pressure does not exceed approximately 800 lb/sq in. The pump used so far is a positive displacement, variable - stroke, multi-plunger type manufactured by Joseph Lucas (Gas Turbine Equipment) Ltd., and illustrated in Fig. 1. The principle of operation of the pump is indicated in the sectional arrangement of a similar unit, Fig. 2. It is fitted with a rotor (A) having seven cylinders accommodating spring-loaded plungers (B) which are caused to reciprocate by rotation of the rotor against a swash plate (C), the angle of which can be varied between 10 and 15 degrees. The ends of the plunger bores terminate in holes of reduced diameter (D) which extend through to the rotor face (E) opposite the driving end. This face is ground flat and smooth and makes a pressure-tight seat with an insert in the pump housing. This insert has two kidney-shaped ports (F), one of which communicates with the pump inlet, the other with the outlet. Thus, each piston during one revolution of the rotor is connected, first with the suction port, then with the delivery port. The swash plate is arranged so that when the suction port is open the piston is moving out of its bore, and into its bore when the delivery port is open. The ends of the plungers at the driving end are ball-shaped and engage with a suitably shaped polished surface on the swash plate (G). Sealing between the rotor face and the pump insert carrying the kidney-

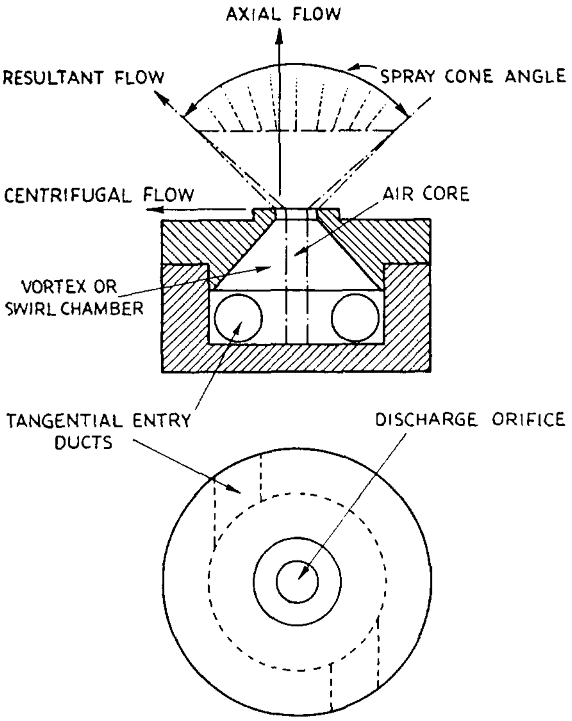

FIG. 4—SIMPLEX BURNER ORIFICE PLATE

shaped ports is vital, as any leakage will cause the pump output to fall as pressure is applied. A typical pump characteristic is shown in FIG. 3 and the effect of this leakage at 1,000 r.p.m. illustrated. When operating at its maximum setting the pump is capable of delivering 1,000 gallons per hour at 1,600 lb/sq in., at 4,000 r.p.m. Adequate filtration of the fuel is required to minimize wear of the sealing faces.

The fuel pump is engine-driven and operates over the normal operating range at virtually constant speed apart from the permanent governor droop as load is changed.

The Burner

The burner is the controlling component of any gas turbine fuel system, for the design of the system is entirely dependent upon its characteristic. Not only must it supply a given amount of fuel at a given pressure, but it must atomize the fuel to a fine degree over the entire engine operating range. It must also be made of suitably hard material to prevent wear by very high friction of the swirling fuel, and must have the largest possible orifice to prevent blockage by small particles of dirt which may pass through the filter.

The fuel requirement is obviously dependent on the particular engine, but the extremes are set by the minimum flow required to light-up and accelerate the engine, and the maximum flow required to limit the temporary speed droop when the largest required increments of load are put on the engine.

The simplex burner, as its name suggests, is the simplest type normally encountered, and it is used on Allen stand-by sets. It is basically an orifice preceded by a conical swirl chamber, into which fuel is admitted by means of two or more tangential holes or slots. A high degree of swirl is produced, but the chamber does not run full, as the fuel forms an annulus surrounding a central air core, as shown in FIG. 4. The fuel is discharged from the orifice into the combustion chamber as a hollow conical sheet which soon disintegrates into drops followed by a fine mist. The injection pressure is proportional to the square of the quantity injected ; thus, for a range of maximum/minimum fuel flow of 5/1, a pressure range of 25/1 would be required. The minimum pressure is restricted to not less than 50 lb/sq in. to ensure reasonable atomization at light-up ; consequently, a maximum pressure of 1,250 lb/sq in. would be required under maximum conditions, in the above example.

The burner is usually classified according to its flow number defined as :—

$$\text{Flow Number} = \frac{\text{Quantity of fuel injected (g.p.h.)}}{\sqrt{\text{Injection pressure (lb/sq in.)}}}$$

The quantity has no units and is dimensionally incorrect, but it is widely used to specify an orifice plate. This square law relationship becomes an embarrassment as the flow range increases, because of the large pressure range required. This problem has been overcome by two alternative designs, both of which have been used on Allen marine gas turbines.

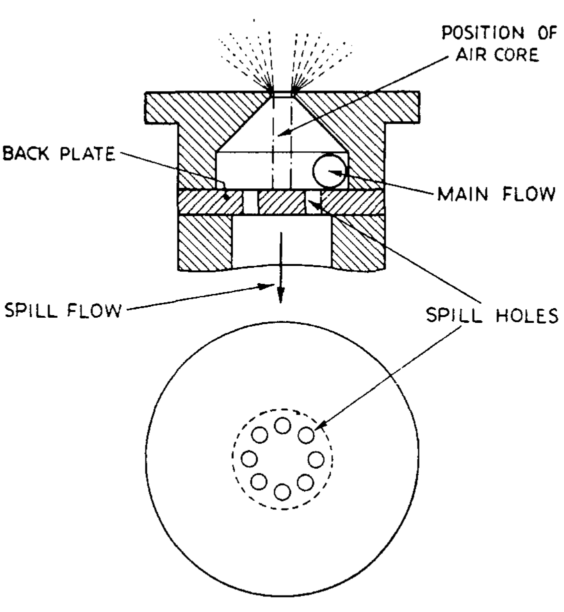

FIG. 5—SPILL BURNER ORIFICE PLATE

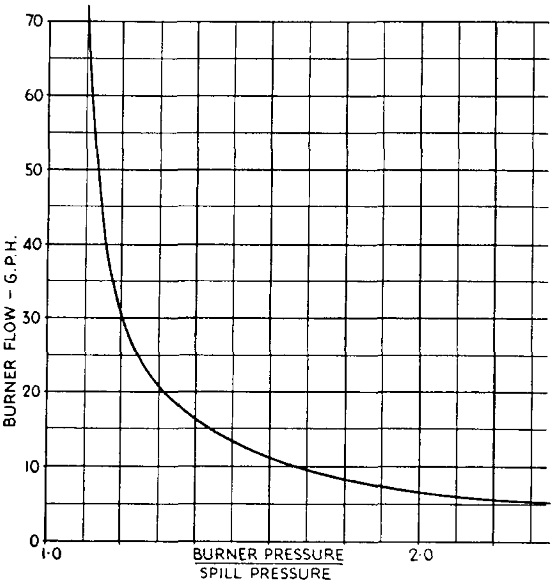

FIG. 6—CALIBRATION OF SPILL BURNER

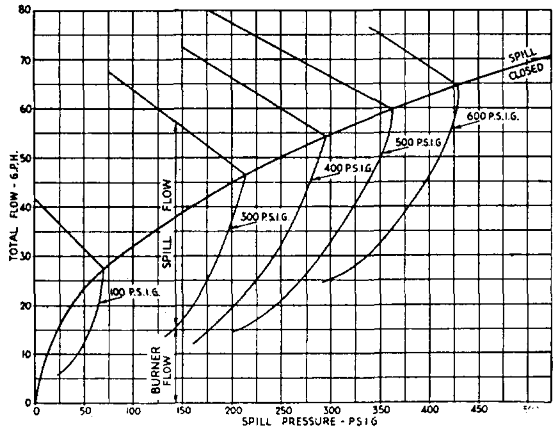

FIG. 7—CALIBRATION OF SPILL BURNER

(*a*) *Spill burner*

With this type the swirl chamber is modified as shown in FIG. 5, so that the back plate, instead of being solid, is provided with holes. As the restriction across these holes is less than that of the orifice, the fuel, after being admitted in the usual way, would pass out of these holes instead of through the orifice. If, however, a variable resistance in the form of a valve or a second pump, were placed in this spill line (as it is called), some fuel would pass through the orifice, depending on the magnitude of the resistance. With this burner the degree of atomization may be maintained over a large flow range with a moderate pressure range, since a high and constant swirl velocity is set up in the swirl chamber and the output is controlled by the amount of fuel taken out along the spill line. The spill holes must be so arranged that the central air core previously mentioned does not pass along the spill line and aerate the fuel, which has a disastrous effect on the functioning of the valve controlling the spill flow. Another useful advantage of the spill burner is that the discharge orifice may be relatively large in diameter, which decreases the risk of blockage by dirt in the fuel. The surface finish of the orifice is important and it has been found that by lapping, the discharge flow will be increased. The ratio burner pressure/spill pressure controls the flow and from the calibration shown in FIG. 6, it will be observed that a large increase in fuel flow will be produced by a small change in the burner/spills presure ratio. This is a bad feature as it imposes strict limits on the performance of the spill control valve, which must hold the burner exactly on its calibration to produce the required flow. This situation is improved if the burner can be arranged to work at a constant burner pressure, as is done on the Allen 125-kW set. The

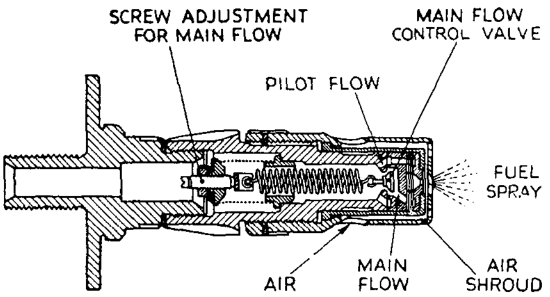

Fig. 8—Section of Duplex Burner

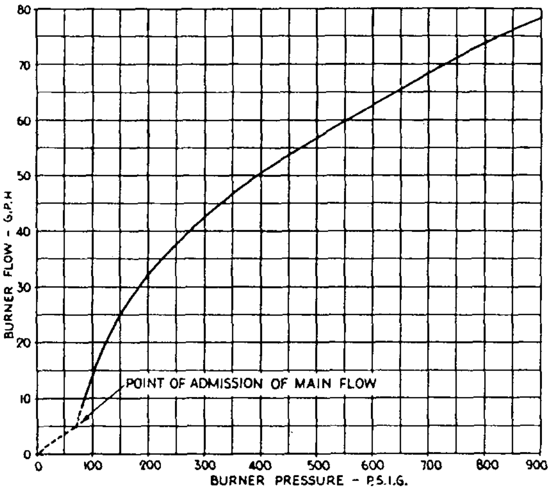

Fig. 9—Calibration of Duplex Burner

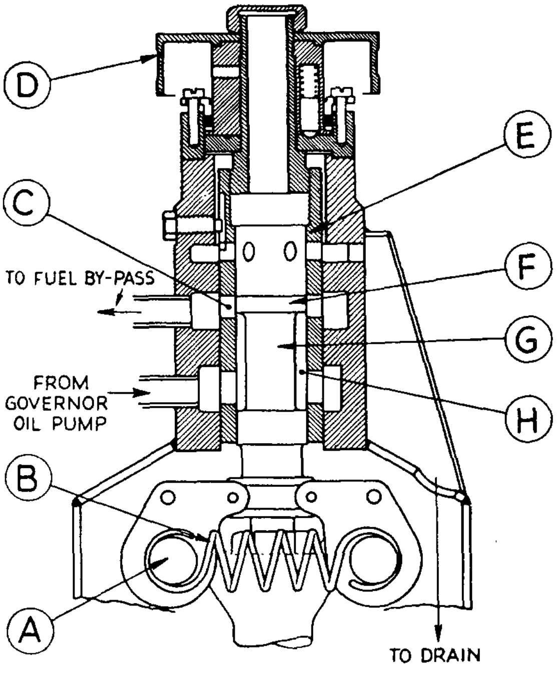

Fig. 10—Speed Governor

control is then entirely on the spill side and, although the characteristic is still steep, control becomes more simple. The calibration is shown in another form in Fig. 7.

Spill burners are generally difficult to work with, and have been found unsuitable for Allen gas turbines. They have also never become popular in aircraft engines for similar reasons. The best form of control has been obtained with a two-pump system, the second pump being placed in the spill flow-line.

(*b*) *Duplex Burner*

This uses two sets of holes or slots to supply the fuel to a swirl chamber, illustrated in Fig. 8. One set, of relatively small area, supplies pilot flow to cover starting, and the other set gradually comes into operation as the flow builds up. This second or main flow is controlled inside the burner by a small valve which can be set by means of a spring to open at a certain injection pressure. This type will cover a wide flow range with good atomization, but the cone angle will vary slightly over the operating range due to the change in area of flow path as the valve is opened. A typical calibration is shown in Fig. 9, admission of the main flow being shown by the change in slope of the characteristic.

The Speed Governor

This governor, shown in Fig. 10, is driven from the engine gearbox by means of a bevel gear. The plunger (G) moves as the radius of rotation of the balls (A), controlled by springs (B), changes with engine speed. Relay oil is supplied to the governor input at a constant pressure of 60 lb/sq in. controlled by a relief valve. This oil passes by way of an annular space (H) through ports (C), which are progressively opened as the engine speed decreases, and closed as the engine speed increases, by section (F) of the plunger. Adjustment or setting of the governor is achieved by turning the hand wheel (D), which moves the sleeve (E) and the position of ports (C), or by using a remotely-controlled speeder motor instead of the hand

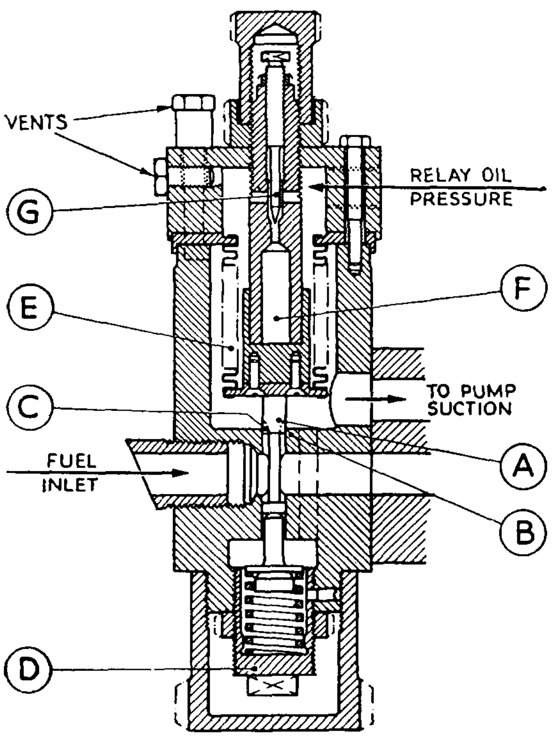

Fig. 11—Section of Automatic By-Pass Valve

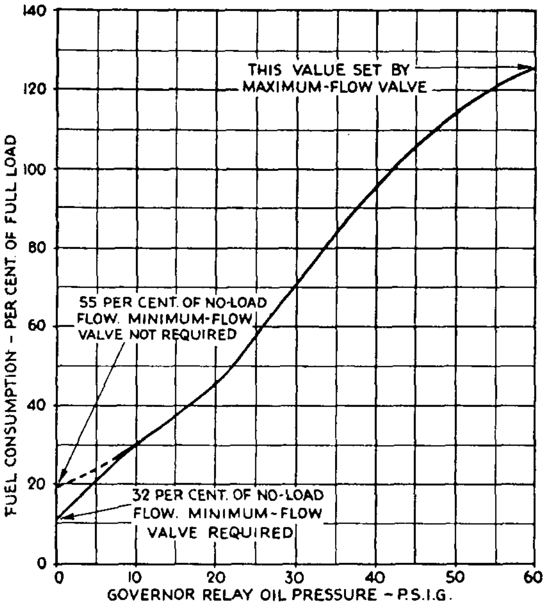

Fig. 12—Calibration of Automatic By-Pass Valve

wheel. Moving the sleeve downwards will produce a reduction in engine speed, and vice versa. The hand wheel is usually locked in position on an emergency set, but may be adjustable on a base-load set, where speed adjustment is required to synchronize with the grid or other machines running in parallel.

Thus, as the engine speed changes, the output or relay pressure from the governor will change, and this provides a suitable variable for the operation of the fuel-control valves, referred to earlier under ' Operating Principles '.

Automatic By-Pass Valve

This is the main fuel controller under steady running conditions at all loads, and a typical valve is shown in Fig. 11. The surlpus fuel is by-passed automatically by variation of relay pressure as the engine speed changes. This pressure varies the position of a profiled valve (A) in a sleeve (B) which closes as the relay oil pressure rises. Thus, the fuel by-pass is reduced and more fuel is supplied to the engine to increase the power output. A typical calibration is shown in Fig. 12 as a relationship between relay oil pressure and fuel consumption. The total amount of fuel being delivered would be controlled by the setting of the pump stroke, the difference between this and the fuel consumption being the amount by-passed. At a relay of 60 lb/sq in., the profiled valve blanks-off the by-pass opening (C) completely (except for slight leakage through the clearance space) and the entire pump output is being delivered to the engine. At 0 lb/sq in., the by-pass is completely open and the minimum quantity of fuel to maintain combustion is being passed to the engine, the rest being returned to pump suction. The shape of the characteristic is controlled by the shape of the profile, and the line may be moved up or down vertically by means of a screw adjustment at the bottom of the valve (D). The principle of operation is that the relay pressure will expand or contract the top bellows (E) which, in turn, will move the profiled valve up or down. The flow path is shown, the fuel being returned to pump suction after leaving the valve. The profile consists of three symmetrical flats, tapered as shown at (A) in Fig. 11.

The gradient was initially developed by filing on a mild-steel blank, followed by grinding on a stainless-steel valve for engine service.

Fluctuations and general instability of fuel pressure were also experienced and a variable damping device was fitted inside the bellows. This consisted of an oil dashpot arrangement (F) with a fine adjusting screw (G). Prevention of fluctuations is essential, as once they have started, there is no cure, and they magnify, spreading to other valves in the system. It is likewise essential to vent all traces of air from the system.

Hand By-Pass Valve

As stated in the introduction, hand control of the fuel flow is sometimes required, and is provided on the base-load type of set. A conventional needle valve can be used for this purpose and has proved quite satisfactory. It is placed in parallel with the automatic by-pass valve, and by-pass is controlled by gradually closing the valve to accelerate the engine at the desired rate. On reaching no-load speed, the valve should be completely closed and the engine is then under the control of the automatic by-pass valve. The effect of opening the hand throttle on a typical set of valves is shown in FIG. 13.

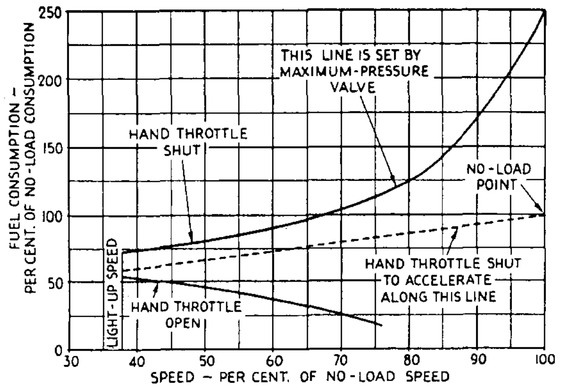

FIG. 13—EFFECT OF HAND THROTTLE ON ACCELERATION OF BASE LOAD SET

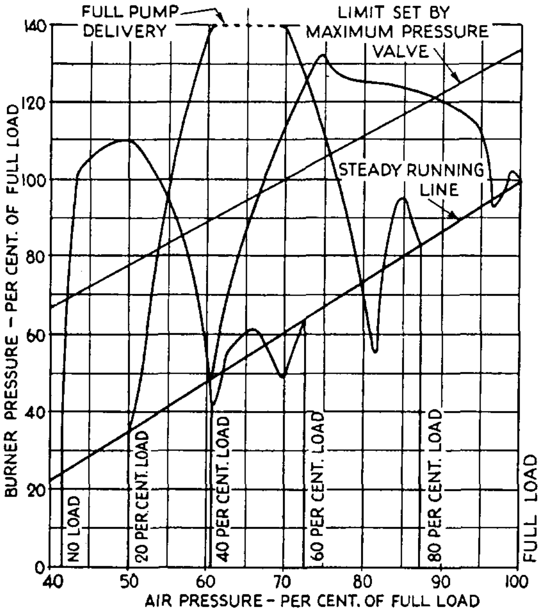

FIG. 14—BURNER PRESSURE RESPONSE TO SUDDEN ADDITIONS OF 60 PER CENT LOAD SHOWING EFFECT OF MAXIMUM-PRESSURE VALVE

Maximum-Pressure or Air/Fuel Ratio Control Valve

This valve has two applications, depending on the type of engine.

(*a*) *Base-Load Set*

The function of the valve in this case is to restrict the amount of fuel supplied to the burner, when load is being suddenly added to the set. The state of affairs is illustrated in FIG. 14, which shows response of the fuel system to additions of 60 per cent load. The information was obtained by use of an oscilloscope fitted with a cine-camera. It will be observed that on adding the load, the fuel pressure rises immediately to a high level, then falls, and, after a small amount of oscillation, settles on the steady running value for the new load. The peak is reached in one to two seconds, and the new steady running condition in four seconds. In the case of the 20 to 80 per cent load increment, the peak was never reached, as the full fuel-pump output was reached and this tended to act as a valve, as shown by the dotted line. The object of the maximum-pressure valve is to chop off these peaks and control the fuel consumption under these circumstances. In the case illustrated, without a control valve, the gas temperature produced would be approximately 1,100/1,200 degrees C., which is excessive even for such short periods when load application will be frequent. If, however, the fuel consumption is over-restricted, the engine speed will fall excessively, and there will be danger of compressor surge, which must be avoided. A compromise must, therefore, be achieved and the valve set to work accordingly. Fig. 15 illustrates the type of valve developed for this purpose. Air pressure is used as the controlling variable which regulates the fuel by-pass by means of a ball (A) seating on an orifice (B). The size of ball and orifice may be changed, also adjustment to the vertical position of the orifice to regulate the setting of the valve by a screw (C). Damping was not needed, as the valve operated in a stable, repeatable manner and produced no fluctuations. Air pressure is transmitted by the bellows (D) to balance fuel pressure, and control the fuel flow through the valve.

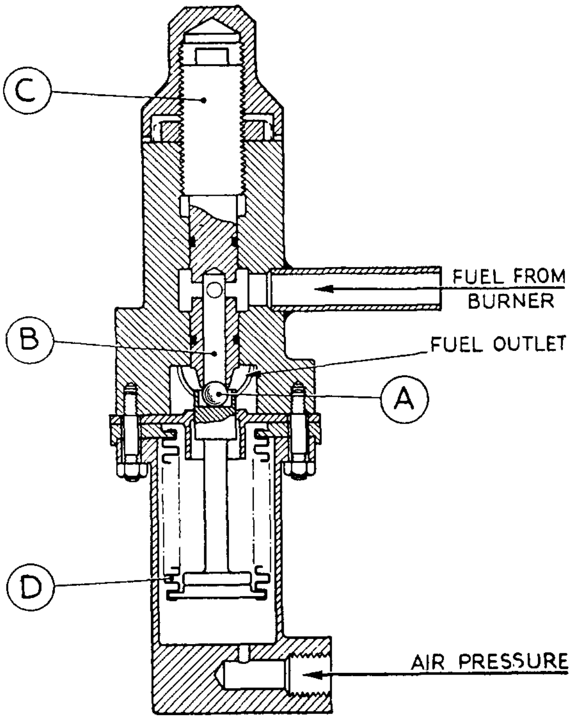

Fig. 15—Maximum-Pressure Valve (Base Load Set)

(*b*) *Emergency Set*

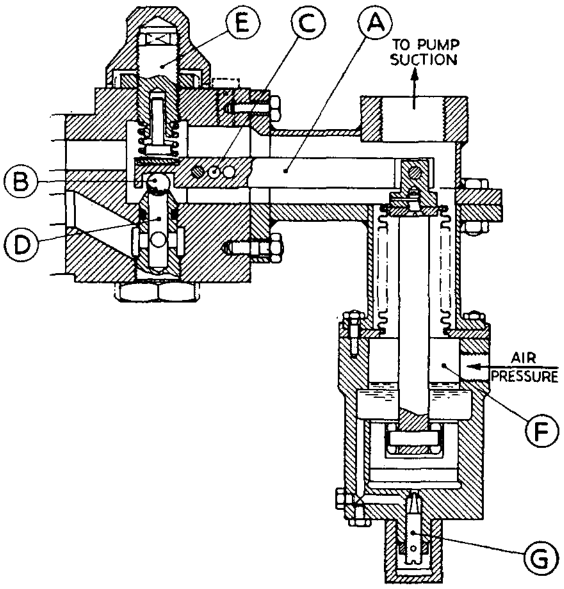

Fig. 16—Maximum-Pressure Valve (Emergency Set)

On the emergency set, starting must be an automatic event and some form of control is essential to provide sufficient fuel for acceleration, yet insufficient to produce a dangerously high temperature. A maximum-pressure valve attached to the automatic by-pass valve has been developed for this purpose, and its method of operation is shown in Fig. 16. During acceleration the governor is not responsive and the relay pressure is at its maximum value of 60 lb/sq in. The automatic by-pass valve is, therefore, closed and, apart from a small amount of leakage, there is no flow through it. Without the maximum-pressure valve the entire pump output would pass to the engine. Air pressure is used to actuate a lever arrangement (A) which balances the fuel pressure acting on a ball (B). This lever is fitted with an adjustable fulcrum (C) for experimental purposes and the balance is so arranged that fuel is by-passed to the pump suction until no-load speed is nearly reached. At this stage the relay pressure comes into operation, and the pressure falls, bringing the profiled valve into operation as the by-pass controller. Also at this stage, air pressure, with the assistance of the lever, overcomes the fuel pressure and the ball valve closes, shutting off this by-pass (D) completely. This remains closed during steady running, although it may be called into operation, as on the base-load set, to prevent over-fuelling on sudden load changes. Adjustment of the valve is possible by means of the fulcrum position, also a spring on top of the lever at the ball-valve end may be adjusted by means of the screw provided (E). Damping of the air side was found to be necessary, and is provided by means of an oil dashpot (F) and an adjustable fine screw arrangement (G).

Minimum-Pressure Valve

A valve for controlling the maximum fuel pressure has been described and, conversely, it may be necessary to limit the minimum fuel pressure. In cases where load is being removed from the set, the fuel pressure may momentarily fall to such a low value that combustion cannot be maintained. The object of a minimum-pressure valve is to prevent this state of affairs by restricting the sudden excessive by-pass of fuel and always to ensure that sufficient is being delivered to the burner to keep the flame alight. Air pressure is the controlling variable, as with the maximum-pressure valve, and two types of valve were developed for use on the base-load set. The valve used on the emergency set is a type of relief valve set to maintain flow at light-up only, and is not required during load changes. It is not sensitive to air-pressure changes.

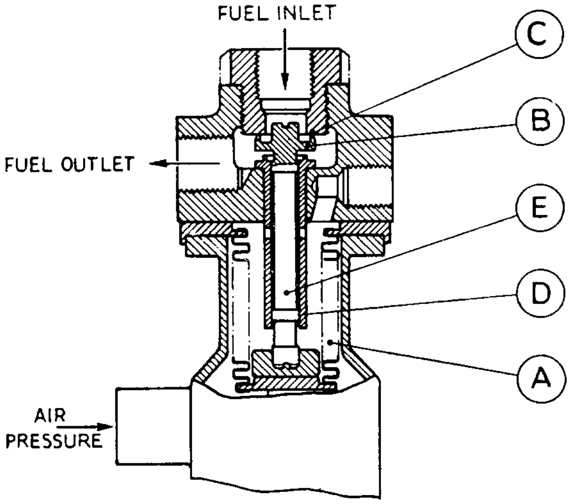

Fig. 17—First Minimum-Pressure Valve (Base Load Set)

The first type (Fig. 17) was similar to the maximum-pressure valve, in that it balanced fuel pressure against air pressure, utilizing a bellows (A) and an annular valve (B) locating on a flat seating (C), instead of a ball and orifice. It was decided to restrict the minimum flow to approximately half the no-load flow for all load changes, but unrepeatable results were obtained, and the minimum flow would change for no apparent reason from time to time. Eventually, it was thought that the flow of fuel through the valve was causing it to tilt, and a guide cylinder (D) was placed around the stem (E). This improved matters but it then developed excessive chattering, especially at high air pressures, which was found to be difficult to cure.

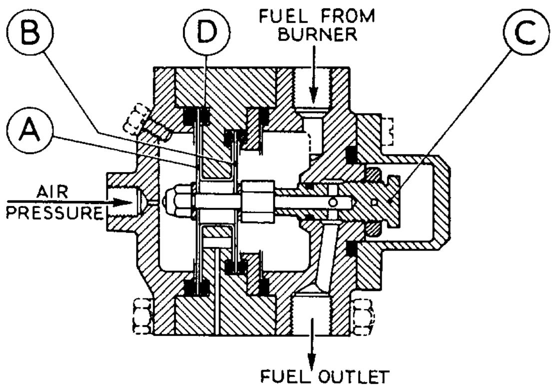

Fig. 18—Diaphragm Type Minimum-Pressure Valve (Base Load Set)

A complete change of design to a diaphragm type, illustrated in

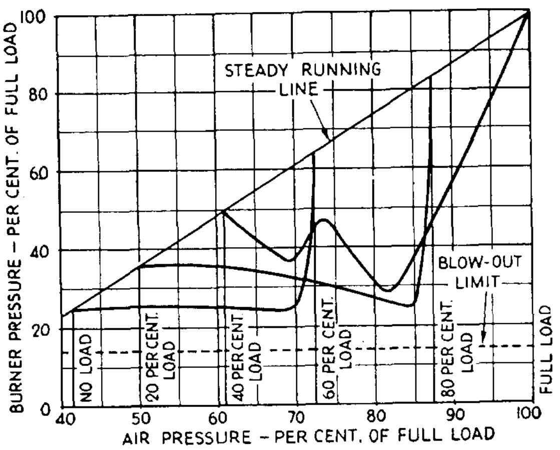

FIG. 19—BURNER PRESSURE RESPONSE TO SUDDEN REMOVAL OF 60 PER CENT LOAD (BASE LOAD SET)

FIG. 18, was decided upon. The bellows on the air side was replaced by two diaphragms (A) and (B), which produced a more compact valve. Initial setting or adjustment was possible by means of the screw (C) on the fuel side of the valve. This was the first experience with thin diaphragms, using their deflection under pressure to open or close an orifice. The material used was a cast steel known as ‘ gauge plate ’, which was hardened by quenching from 780 degrees C., and tempered at 300 degrees C. A main diaphragm (A) and a smaller supporting diaphragm (B) were used, both being supported in the casing by ‘ O ’ rings (D). It was possible to vary the flow a limited amount at a particular air pressure by altering the thickness of the diaphragm. Two alternatives were tried, but after a certain amount of testing, the thicker diaphragm was found to be cracked, presumably due to fatigue with repeated deflection. Three cracks were noted, starting at the hole in the centre of the diaphragm, radiating to the outer edge and, when removed from the valve, the diaphragm fell apart into three distinct pieces. The thinner diaphragm was therefore used for the rest of the work. Other diaphragm materials such as beryllium copper may be more suitable than steel for this purpose. The aforementioned combination of diaphragms produced the calibration shown in FIG. 26. It was found impossible to hold the flow down to the required level over the complete air-pressure range, but the small increase at higher air pressures was acceptable.

This is again a by-pass valve, arranged so that under steady running conditions, all the by-pass fuel passes through it to the automatic by-pass valve. When load is removed, the engine speed rises and relay oil pressure falls almost to zero. The automatic by-pass is fully open and it is in these circumstances that the minimum-flow valve begins to operate, restricting the by-pass so that combustion will continue. FIG. 19 shows an extract from the film taken during removal of 60 per cent load from the base-load set. In this particular case the characteristic of the complete fuel system was such that a minimum-pressure valve was not required. However, the immediate fall in fuel consumption is clearly illustrated as the load is removed. If this drop were sufficient to cause blow-out, the minimum-pressure valve would restrict the troughs as did the maximum-pressure valve to the peaks (FIG. 14).

Spill Control Valve

In describing the spill burner, the need for a restriction in the spill flow line was mentioned, and during work on spill burners, interesting experience was gained with a spill control valve to provide this restriction. It was also mentioned that the ratio burner/spill pressure was of great importance in keeping the burner flow under control. This demanded meticulous performance from the valve which, in practice, it was found difficult to provide. A suitable valve was developed, however, but this was affected by air from the core in the burner swirl chamber passing down the spill line and, in spite of careful venting on starting, aeration was never overcome. Further investigation was abandoned as the spill burner concerned was replaced by a duplex burner which

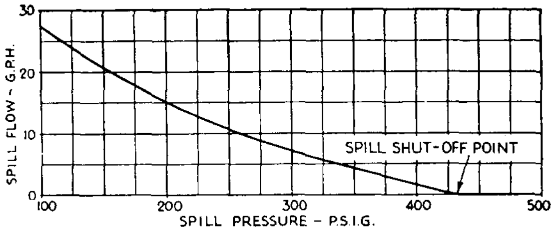

FIG. 20—CALIBRATION OF SPILL CONTROL VALVE

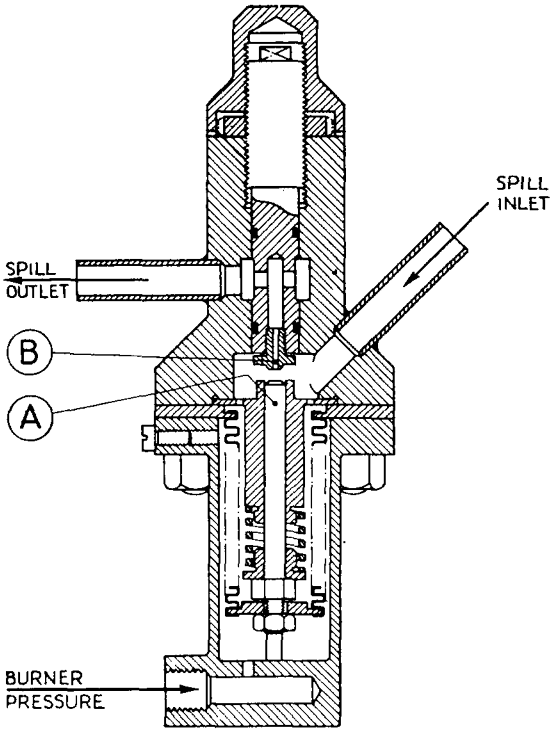

FIG. 21—FIRST SPILL CONTROL VALVE (BASE LOAD SET)

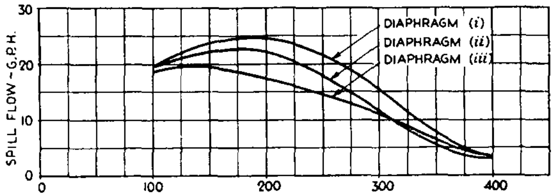

FIG. 22—CALIBRATION OF DIAPHRAGM TYPE SPILL CONTROL VALVE

eliminated the need for spill flow and a spill valve.

The first design of valve was similar to the other valves, using a balance between two pressures by means of bellows. The valve was similar to the minimum-pressure valve, except that the air pressure was replaced by burner pressure. The design of the system was such that as fuel-injection pressure increased, less spill flow was required until, at a pressure of approximately 450 lb/sq in., spill flow was no longer required and the burner became a simplex type injecting all the fuel into the combustion chamber. This fact was used to provide the calibration shown in FIG. 20, where, as the spill pressure increases, the spill flow decreases until the shut-off point is reached. A needle-type valve was first used, actuated by a bellows and the burner pressure, but this was found to have a short working range and closed the spill line too early. It was modified to the design shown in FIG. 21, by using a flat-ended valve (A) closing a small orifice (B). Its range was however, still restricted, which meant that if the valve were set to give correct light-up flow, the spill line closed too early, and if set to give the correct shut-off point, the light-up spill flow was too high. A temporary solution was provided by placing a sensitive needle valve in the line between the burner and the spill valve. This enabled the light-up flow to be reduced to the required value, and as a fixed orifice, it did not affect the shut-off point determined by the variable opening of the spill valve. This worked quite well but it was obviously undesirable to have two valves, and a complete change was made, as with the minimum-flow valve, to a diaphragm type. Three thicknesses of diaphragm were used, as indicated by the calibration (i), (ii), (iii), FIG. 22, the two together (A) forming the main controlling deflector, and a third (B) used as a guide or support, as shown in FIG. 23. Small differences in calibration were found with each diaphragm. The effect of thickening the main diaphragms wa to restrain the closing process and allow more spill flow over the range of spil pressures, while the supporting diaphragm also has a restraining influence

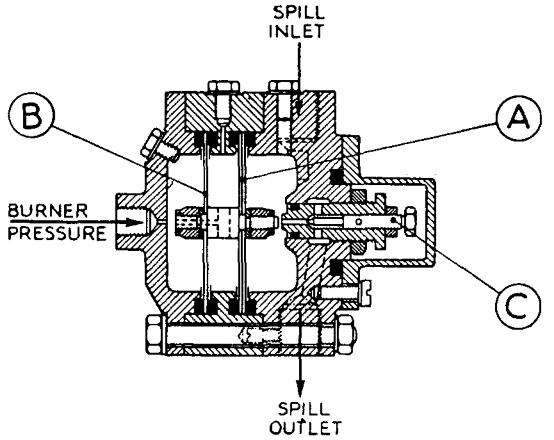

FIG. 23—DIAPHRAGM TYPE SPILL CONTROL VALVE

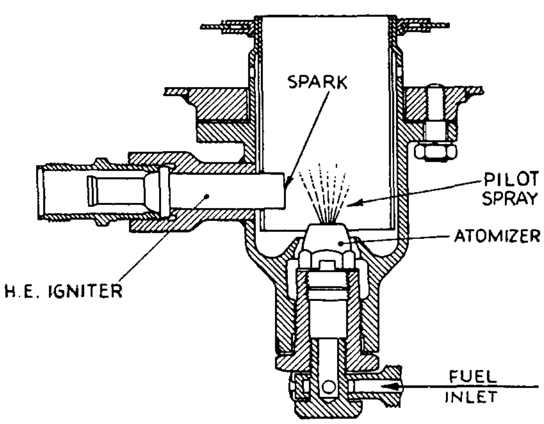

FIG. 24—SECTION OF TORCH IGNITER

After experiments with the thinnest diaphragms (iii), FIG. 22, buckling to the extent of 0·005 in. was found to have occurred, and they were replaced by a thicker variety. The fine adjustment previously provided by the external needle valve was built into the unit as shown in FIG. 23, by means of the small screw (C) which could be set and locked in position.

Emergency Shut-Off Valve

This valve is common to both types of engine, and is the means of safeguarding the engine in an emergency. The valve is placed in series with the burner and will allow fuel to pass so long as the governor input pressure is maintained at approximately 60 lb/sq in. If, for any reason, this should fall, the shut-off valve will close, due to release of the pressure on a bellows. The valve can be caused to operate and shut off the fuel supply by either of the following safety devices :—

(*a*) Bearings oil trip—this will operate if the bearings-oil pressure falls below a set value

(*b*) Overspeed trip—this device is responsive to speed and can be set to destroy the governor-input pressure at a set overspeed.

Conversely, in addition to being a shut-down device, this valve is a starting device. As the engine begins to rotate by means of the starter motor, governor-input pressure will build up quickly to 60 lb/sq in., and the valve will open, allowing passage of the fuel, followed by ignition and combustion. In addition to the shut-off valve, it is usual to fit a non-return valve at the burner inlet to prevent leakage of fuel into the combustion chamber during stationary periods. The spring would allow fuel to pass on opening the shut-off valve.

Igniters

A vital component of any system is the igniter, of which two types have been used, as follows :—

(*a*) *Torch Igniter*

This incorporates a sparking plug working from a 24-volt supply, and a pilot spray of fuel. Ignition of the pilot spray is thus the first stage, followed by ignition of the main fuel spray, and some idea of the fuel involved is shown in FIG. 26 and is referred to subsequently, under ‘ Performance of Complete Fuel Systems ’. A section of the igniter is shown in FIG. 24.

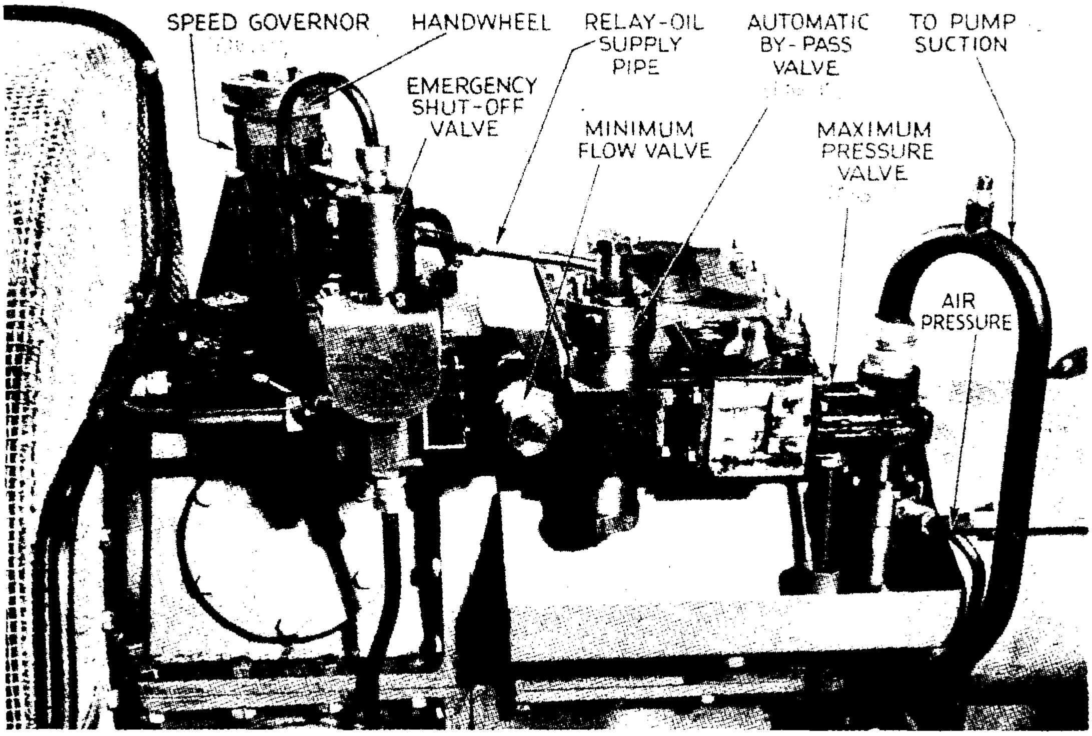

FIG. 25—ARRANGEMENT OF FUEL CONTROL VALVES IN A 350 kW GAS TURBINE GENERATING SET

(*b*) *Surface Discharge Plug*

This requires no pilot spray but ignites the main fuel spray directly. I requires an ignition box supplied from a 24-volt battery which will produce spark of considerable energy amounting to twelve joules. The position of thi type of igniter in the combustion chamber is of importance to provide efficien ignition, but it must be so placed that its surface does not become overheate as combustion proceeds.

The ignition of gas-oil as used on the Allen base-load set by means of th surface discharge plug, should present no difficulty, but the assistance of a pilc spray when burning marine Diesel fuel as used on the Allen emergency set, i of great value.

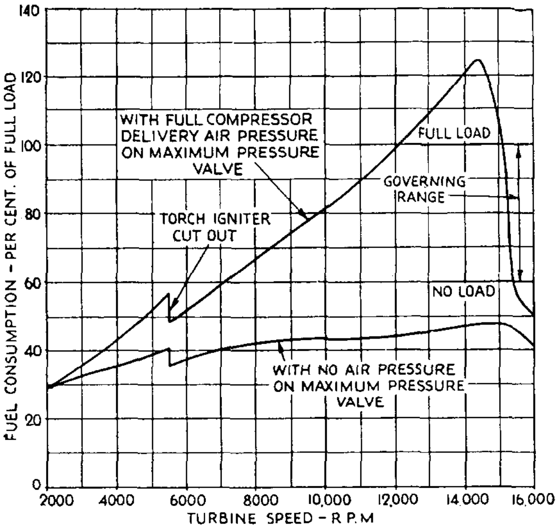

FIG. 26—CALIBRATION OF EMERGENCY SET COVERING FULL SPEED RANGE

Performance of Complete Fuel Systems

The following are typical c results required and obtained fror the fuel systems employed on Alle gas-turbine generating sets. A stated in the introduction, th results have been obtained from ri settings, confirmed by substanti periods of engine running. It wil however, be both necessary an interesting to observe the change if any, to the calibrations as engin running hours are accumulated also the effects of wear will be c particular importance in order t assess the life and reliability of vit control valves. The two types c engine will be taken separately.

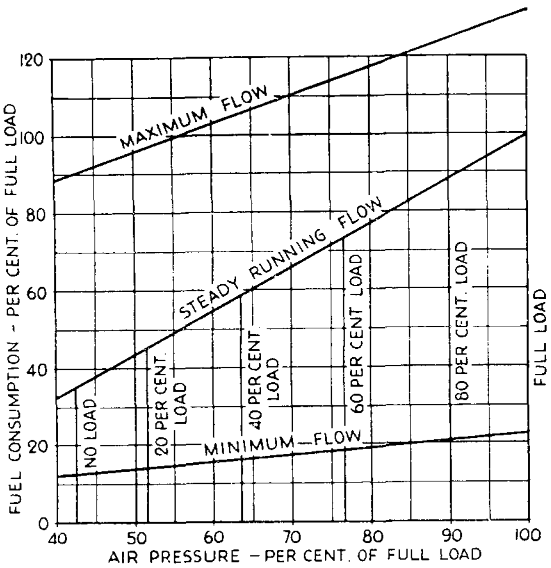

FIG. 27—CALIBRATION OF BASE LOAD SET FUEL SYSTEM WITH FULL SET OF VALVES

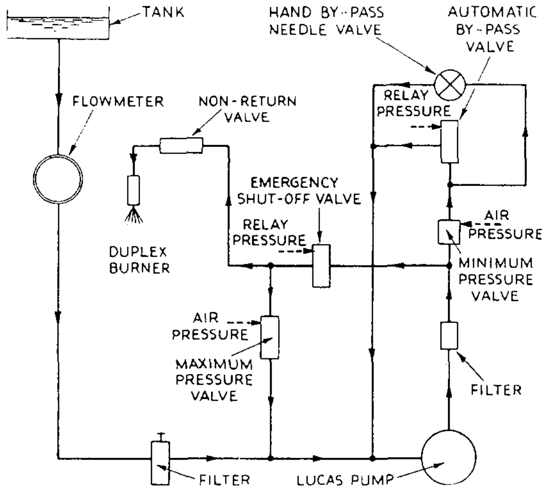

FIG. 28—DIAGRAMMATIC ARRANGEMENT OF FUEL SYSTEM WITH FULL SET OF VALVES FOR BASE LOAD SET

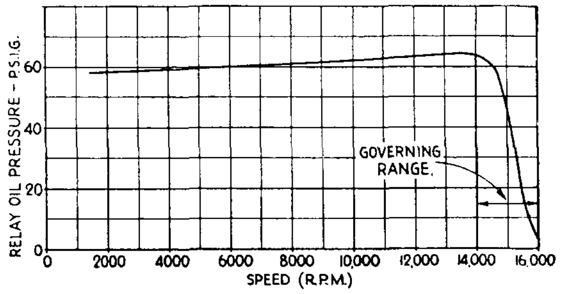

FIG. 29—GOVERNOR RELAY PRESSURE/SPEED RELATIONSHIP FOR EMERGENCY SET

(*a*) *Emergency or Stand-by Set*

The valve assembly here consists of an automatic by-pass valve, a minimum-pressure valve of the pre-set type, previously mentioned, and a maximum-pressure valve balancing air and fuel pressure during automatic starting, as already described. The valves are arranged together as one block, a typical case being the 350-kW set shown in FIG. 25. The calibration in FIG. 26 shows the increase in fuel consumption as the engine accelerates up to 14,000 r.p.m. At this stage the governor becomes operative and the relay pressure falls from 60 lb/sq in. to approximately 14 lb/sq in., as acceleration continues from 14,300 to the no-load speed of approximately 15,300 r.p.m. The automatic by-pass valve is thus opened progressively and more fuel is by-passed until settled values of relay pressure, fuel consumption and speed are reached at no-load. The sudden drop in fuel consumption at 5,500 r.p.m. is due to cut-out of the torch igniter, whose assistance is no longer required. The effect of air pressure on the maximum-pressure valve is clearly seen in FIG. 26 by reference to the lower line. Here air pressure to the bellows was blanked off, which produced no action on the 'air end' of the lever (FIG. 16). Thus, there was no opposition to fuel pressure which was able to push the ball off its seat and by-pass large quantities of fuel, as shown. This offers a means of adjusting the acceleration line within the limits, no air pressure to full compressor delivery pressure. If a controlled amount of restriction were placed in the air-pressure line the accelerated fuel consumption could be reduced according to the amount of restriction. A second method of adjustment is provided by the screw on top of the ball valve at the fuel side, and a third by adjusting the fulcrum of the lever. The range of operation, no load to full load, is shown in FIG. 26, the speed droop being between $2\frac{1}{2}$ and $3\frac{1}{2}$ per cent.

(*b*) *Base-Load Set*

The state of affairs on this engine is illustrated in FIG. 27 for a full set of valves, arranged as in FIG. 28, acceleration being controlled by the hand by-pass valve. Clearly, the fuel required for acceleration can be provided by manipulation of the valve until no-load speed is reached. The flow with the hand by-pass closed would be controlled by the maximum-pressure valve (FIG. 15). Over this period, relay pressure is at its maximum value of 60 lb/sq in. and falls in a similar matter to that on the emergency set, to its no-load value of approximately 15 lb/sq in., as illustrated in FIG. 29. Control of the engine is now the responsibility of the automatic by-pass, minimum-pressure and maximum-pressure valves, and will be as shown in FIG. 27. Fuel consumption is plotted against air pressure, as the controlling variable rather than speed. Speed droop of 3½ to 4 per cent will be controlled by the shape of the profiled by-pass valve, which dictates the difference between relay oil pressure (and consequently speed) from no load to full load. The by-pass valve profile used to produce the results indicated in FIG. 12 would require a minimum-flow valve, assuming the blow-out fuel flow to be 50 per cent of no-load flow, since at zero relay pressure the flow has fallen to 32 per cent. If the shape of the profile were such as to produce the dotted line falling only to 55 per cent of no-load flow, for example, then a minimum-pressure valve could be dispensed with. This is further dependent on the inherent temporary speed droop of the particular engine as, in fact, the relay pressure may never fall to zero, but only to, say 5 lb/sq in., in which case still more fuel would be available. It is in any case desirable that the relay pressure should not fall to zero, as the engine is momentarily ' off the governor ' at this position. Investigation of these points of detail is essentially confined to engine development and cannot be foreseen until engine experience has been gained. Nevertheless, the availability of a test rig proved of inestimable value for the considerable detailed setting and nvestigation of pumps, burners and valves, found to be required.