

THE
INSTITUTION OF LOCOMOTIVE
ENGINEERS, LONDON.

The Application of the Diesel
Engine to Locomotives.

BY
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TWENTY-EIGHTH PAPER
(OF TRANSACTIONS).

SESSION 1914.

To be read on Saturday, September 26th, 1914

(Unless otherwise arranged),

at Caxton Hall, Victoria Street, Westminster.

Published by the Institution.

46, Malling Street, Lewes.

Price Two Shillings Net.

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The Institution of Locomotive Engineers

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INTRODUCTORY.

A few weeks after the offer of this paper had been made, the writer was interested in finding the following sentences towards the end of Mr. A. J. Hill's Presidential Address last January :—

“As to whether the steam locomotive will ever be entirely superseded by electricity or by some form of internal combustion engine is beyond my powers to forecast. I do think, however, that there are possibilities of such being the case, and I would strongly urge upon all young locomotive engineers to keep well up-to-date in these branches of the profession.”

As a paper on this subject has not previously been read before your Institution, it is hoped that what follows will be of interest, and cover a little ground which may not be as familiar to you as that of the steam engine.

The Diesel engine was invented twenty-two years ago, for in the year 1892 the late Doctor Diesel (Fig. 1) obtained a patent covering his specification of the type of engine which bears his name. But even at the present day the Diesel engine is regarded as a somewhat mysterious and awesome thing. It has been libelled in the Navy, having received the nickname of “Stinking Jimmy.” In a recent law case the atmosphere of the court came under consideration, and the following paragraph appeared in the “Daily News” on May 27, 1914. “‘Sitting there in that court hour

after hour,' said one of the crew of the now famous H.M.S. 'London,' 'was as bad as living day and night with "Stinking Jimmy."' 'Stinking Jimmy' is the pet name of that wonderful piece of mechanism known to science as the Diesel engine, and the Diesel engine, grim ornament of every battleship, breathes forth fumes which no one who has not breathed them can possibly imagine. The nether regions are Hampstead Heath compared to it."

A gross libel, Gentlemen. The fumes, often taken up a hollow mast entirely clear of the decks, are certainly poisonous, but if one had to die, it would be preferable to face that inevitable end amongst the fumes of a properly controlled Diesel engine, than in the smoke from a steam boat's funnel.

In the first alternative, one would at least die clean. The technical Press continually comments on the cleanliness of a Diesel engine-room as compared with that of a steam boat, to say nothing of that foretaste of Gehenna known as a stoke-hold, which is non-existent when the Diesel engine is installed.

Briefly the engine is just a glorified and scientific oil engine of higher power per cubic inch of cylinder volume than is obtained in other types of either steam or internal combustion engines.

Perhaps a few words on the theory of this prime-mover will not be out of place.

THE THEORY OF THE DIESEL ENGINE.

There are three distinct thermo-dynamic cycles, and all heat-engines operate on one or other, or a combination of two of these. Carnot's cycle (Fig 2), or the cycle of constant temperature, is the one, on a modification of which the steam engine works.

Fig. 3 shows the Otto cycle of constant volume, followed by the oil or gas engine. Fig. 4 illustrates the Brayton or constant pressure cycle; and Fig. 5 is the Diesel cycle, which is a combination of Figs. 3 and 4.

In each of these four figures horizontal distances represent volumes, pressures being shown by the vertical heights. Examining Fig. 2, compression of the working substance takes place from A to B. Heat is received at constant temperature from B to C, expansion occurs from C to D, and heat is rejected at constant temperature from D to A. This figure represents the ideal steam-engine heat cycle.

The highest possible thermal efficiency of a perfect steam engine using saturated steam at 180 lbs. per square inch, and condensing at 60 deg. F., is approximately 38 per cent., which figure, however, can never be approached in practice; from 10 to 15 per cent. efficiency can be actually obtained (Ewing).

In Fig. 3 compression occurs from A to B, heat enters at constant volume from B to C, *i.e.*, whilst the volume of gas remains unchanged, expansion takes place from C to D, and heat is rejected also at constant volume from D to A.

In the Otto cycle oil or gas engine, making allowance for the comparatively low compression, the ideal theoretical efficiency is about 50 per cent., and in practice from 18 to 23 per cent.

In Fig. 4 there is compression from A to B, heat received at constant pressure from B to C, expansion from C to D, and heat rejected at constant pressure from D to A.

Fig. 5, the actual Diesel heat cycle, shows the combination of Figs. 3 and 4. Compression from A to B, heat enters at constant pressure from B to C, expansion takes place from C to D, and heat is rejected at constant volume from D to A. This cycle gives a theoretical thermal efficiency of about 63 per cent., and under the conditions obtaining in practice of from 32 to 41 per cent.

Now the thermal efficiency of any heat engine depends on the difference in the quantity of heat received and the quantity rejected in the exhaust. And Professor Ewing, in his work on "Heat Engines," page 434, states that the highest possible thermal efficiency of any internal combustion engine exhausting at 60 deg. F, and reaching a temperature of, say, 3440 deg. F.—the temperature of combustion of coal gas under the conditions obtainable—is approximately 74 per cent., so it will be seen that though the Diesel engine does not reach this ideal figure, it approaches it more nearly than any other engine.

The high efficiency of the Diesel engine is not simply due to the heat cycle on which it works, for in the abstract one heat cycle is as efficient as another, it is due to the working conditions under which the cycle can be employed; in other words, the practical application. The huge and unavoidable waste of heat between the furnace and the engine, where steam is employed, is thoroughly understood, and need not be entered into in this paper, but the reason for the higher efficiency of the Diesel engine over that of the ordinary oil engine may be explained as follows: Repeating the statement above, if the initial temperature and

pressure of combustion are high, and the final temperature and pressure are low, then the result is a high thermal efficiency, and the maximum amount of work is got out of the fuel. Thus it depends on the range of temperature through which the heat passes. Therefore, without a large ratio of expansion and the resultant cooling, the thermodynamic advantage of a high initial temperature is lost.

In all internal combustion engines the burnt gases are discharged at atmospheric pressure and consequently a large ratio of expansion is only possible when there is a high initial compression. Compression is therefore an essential condition of efficiency. In the ordinary oil, gas or petrol motor, a mixture of fuel and air is compressed, and at a predetermined point in the compression stroke, is ignited by means of an electric spark or hot tube. The pressure that can be reached by this mixture is comparatively small, for when any substance is compressed, heat is generated, and when the pressure reaches a certain point, depending on the mixture, it ignites, possibly, long before the end of the compression stroke, causing back-fires and negative work on the piston.

Hence the pre-ignition pressure of these engines is limited.

Now in the Diesel engine, air is the substance compressed, and not an explosive mixture of air and fuel, so there can be no back-fire, and the compression pressure may therefore be greatly increased. In fact, the heat generated by this high compression is employed to fire the charge of fuel eventually admitted to the cylinder, no spark or hot tube being employed for this purpose.

It follows that the three reasons for the high efficiency of the engine are: First, efficiency is proportioned to the compression prior to ignition; secondly, the temperature reached due to high compression is greater than is required for ignition, so that combustion is very complete and a poorer quality of fuel can be employed; thirdly, the complete combustion absorbs the whole calorific value of the fuel.

Figs. 6 to 9 illustrate indicator cards from a steam engine (Fig. 6) working approximately on the Carnot cycle, an oil engine (Fig. 7) working on the Otto cycle, and two types of Diesel engine on the Diesel cycle (Figs. 8 and 9). For comparison the four cards are drawn to the same vertical pressure scale and on the same base, which accounts for the elongated appearance of the steam engine card

(Fig. 6). The cycle of operation in Fig. 7 is as follows, and refers to the mechanical cycle, and not the heat cycle mentioned previously: As the piston moves on its outward stroke from A to B, a mixture of fuel and air is drawn into the cylinder, the piston reaching the end of the first stroke at B. It commences to return, compressing the mixture and reaching the end of the second stroke at point C, the pressure then being about 140 lbs. per square inch. Ignition now takes place by means of an electric spark, or hot tube, and the pressure instantly rises, due to the combustion of the gas in a confined space to its maximum of about 300 lbs. at D; in other words, an explosion takes place.

The piston is driven forward and completes its third stroke at E. The exhaust valve is open at, and slightly before this point, and the piston returns, sweeping out the burnt gases from E to A, completing the fourth stroke of the cycle at A. It then repeats.

In Fig. 8 the same cycle of operations occurs, but in this case it will be noticed, first, that the compression is very much higher; second, that the pressure is reached before ignition takes place at C; third, that the period during which ignition occurs is more prolonged, hence the Diesel is sometimes called a slow combustion engine; fourth, that the pressure does not increase as a result of ignition, and the Diesel is therefore not an explosion engine.

In the two-stroke cycle (Fig. 9) assume the cylinder to be full of air at atmospheric pressure, and the piston at the end of the outward stroke. During the first stroke it moves from A to C, compressing the air to 500 lbs. per square inch at C., fuel is injected and burns from C to D, expansion takes place from D to E, exhaust from E to A. Scavenge air enters between E and A, helping to sweep out the exhaust, and continues to enter until point B is reached; then compression takes place and the cycle repeats.

Although the two-stroke cycle engine has fewer working parts than the four-stroke, and apparently many advantages, it has not, up to now, made the progress that at one time was thought probable, though, without question, it is the type of the immediate future for the larger powers.

Theoretically, twice the horse-power can be got from an engine working on this cycle, with its power stroke every revolution, as can be obtained from one of the same size working on the four-stroke, with one power stroke every second revolution. Hence, for the same power, the two-stroke engine is both lighter and smaller.

Against this advantage must be set the extra heat which has to be removed, for, as combustion, expansion and exhaust all take place during one stroke, it follows that in engines making the same number of revolutions per minute, only one-third of the time is available, whilst the high temperatures met with rather than the high pressures, offer the more serious problems in Diesel engine work ; thus, although the high temperatures employed give high thermal efficiency, they form the most difficult problem from a mechanical standpoint.

With regard to the power to which Diesel engines can be built. At the present time the maximum B.H.P. per cylinder of the four-cycle type is 300, and of the two-cycle type 2,000 B.H.P. per cylinder, the engine being made by Messrs. Sulzer, running at 150 R.P.M., with a cylinder diameter of 40in. Messrs. The M.A.N. Co., of Nurnberg, and Messrs. Krupp, of Kiel, are building similar engines of six cylinders, developing a total B.H.P. of 12,000. The mechanical efficiency of four-stroke-cycle engines varies from 75 to 84 per cent., and of two-stroke engines from 70 to 72 per cent. As to the fuel, almost any oil may be used, though, naturally, the better the oil, the better the result.

The ordinary large oil engine, when in actual work, requires about 0.6 lb. of fuel per B.H.P. per hour of, say, 20,000 British Thermal Units per lb. calorific value. Paraffin, a refined oil, is usually employed, but in the Diesel engine either crude oil, residuum oil, or even tar oil can be satisfactorily used with a guaranteed consumption of 0.4 lb. per B.H.P. hour.

Before the advent of the Diesel engine, residual oil was run to waste, whilst crude petroleum and tar oils were unsuited to the direct generation of power in internal combustion engines.

Coming to the question of fuel consumption, the following table shows average practice :—

Type of Engine.	Fuel Consumption per B.H.P. per hour.	Calorific Value in B.T.U.'s per lb.
Large Marine Steam Engine	1.5 lbs. of coal.	14,500
Modern Locomotive	2.0 " "	14,500
Large Oil Engine	0.6 " paraffin.	20,000
Diesel Engine... ..	0.4 " residuum.	19,000

These figures show the relative weights of fuel carried for equal distances and powers, and in passing it might be mentioned that small Diesel engines are nearly as efficient as large ones, and a Diesel at half-load almost as efficient as at full load.

Further comparing the fuel costs of locomotives using coal, with Diesel engines using residual oil :—

A 1,000 B.H.P. locomotive burns $2 \times 1,000$ lbs. of coal per hour, at 12s. 6d. per ton,

$$= 12.5 \times \frac{2000}{2240} = 11s. 2d.$$

A 1,000 B.H.P. Diesel engine burns $0.4 \times 1,000$ lbs. of oil per hour at 50s. per ton. This figure is obtained by averaging the various prices at ports throughout the world,

$$= 50 \times \frac{400}{2240} = 8s. 11d.$$

Residuum oil is the substance left after the lighter oils, such as petrol and paraffin, have been distilled off the crude natural oil. Before the Diesel engine was invented this residuum was practically a waste product of no commercial value. Its present price is inflated, but shows a tendency to cheapen.

In oil-producing countries crude oil costs about 15s. per ton. Freightage from America to England, until recently, was £3 per ton, but in May, 1914, it dropped to between 15s. and £1 per ton.

On the Continent tar oils costing 40s. a ton are extensively used for Diesel engines.

The following paragraph is taken from the "Motor Boat" of August 13, 1914 :—

"As showing the reduction that has taken place in the price of oil, a number of contracts have recently been made in America by the American Government, and also several of the railway companies, for a fairly large quantity of oil fuel at between 16s. and 20s. per ton.

"It was anticipated that the price in this country, in the course of a few months, would have dropped to 55s., or even less, had the war not arisen."

The following table gathers together some of the figures already given in the text :—

Heat Cycle.	Engine.	Thermal Efficiency of an Ideal Machine.	Actual Thermal Efficiency Obtained.
Carnot	Steam.	38%	{ Non-condensing, 6-8%.
Otto	Gas or Oil.	50%	{ Condensing, 10-15%
Diesel	Diesel.	63%	18-21% 32-41% _o
Prof. Ewing's Experiment...	—	74%.	—

THE CONSTRUCTION AND OPERATION OF THE DIESEL ENGINE.

The construction and operation of the Diesel engine may be followed by reference to the diagrammatic arrangement of a single cylinder engine shown in Fig. 10, which illustrates a vertical two-stroke engine of the port scavenge and exhaust type.

As already mentioned, the two-stroke engine has a firing or power stroke each revolution ; thus, for a given cylinder diameter and stroke twice the power of a corresponding four-stroke engine making the same number of revolutions per minute is obtained, also a more even turning moment is exerted on the crank shaft.

But it entails getting rid of twice the amount of heat, requiring extra precautions with the cooling water.

If the engine has to be capable of running in either direction, the two-stroke has the additional advantage of requiring fewer working parts for its valve gear than the four-stroke engine, which must be fitted with inlet and exhaust valves, levers and cams.

In the four-stroke engine port scavenge and exhaust cannot be employed, due to the sequence of the cycle, and though not adopted by all makers at the present time, this method is used with entire success by several manufacturers of two-stroke engines.

The scavenge port is shown at A and the exhaust port at B. No valve gear is employed for either, the ports are simply uncovered by the piston towards the end of the power stroke.

The scavenge air delivered at a pressure slightly above the atmosphere is supplied by a pump. This pump may be part of the high pressure air compressor, or a separate unit driven by links or by an extension of the crank shaft.

In the four-stroke engine, the whole of the first stroke is devoted to drawing air into the cylinder, and the fourth to expelling the products of combustion. These two strokes are dispensed with in the two-stroke engine; hence the employment of scavenge air at from three to five pounds pressure to push out the burnt gases.

When the piston has nearly reached the bottom of the stroke, it uncovers first the exhaust port B, the burnt gases begin to pass out, and the pressure falls very rapidly. Having travelled still further, the piston now uncovers the scavenge port A, air enters and is deflected by a projection on the piston, to the top of the cylinder, turning here and sweeping the remaining gases out before it, leaving the cylinder full of pure air. Having reached the bottom dead centre, the piston begins to rise, covering the ports and compressing the air, thus doing work on the air, and assuming adiabatic compression, considerably raising its temperature, until, at the top of the stroke, the pressure is about 500 lbs. per square inch, and the temperature nearly 1,000 deg. F., hot enough to ignite the charge of oil then injected through the fuel valve.

The actual pressure at the end of the compression stroke depends partly on the ratio of the cylinder volume to the clearance volume, usually 12 to 1, and also on the nature of the compression, viz., either adiabatic or isothermal. In the cylinder of a Diesel engine it is nearly, but not quite, adiabatic.

The formula employed is :—

$p_1 v_1^n = p_2 v_2^n$ where p_1 is the pressure at the beginning of the stroke = that of the atmosphere = 14.7 lbs. increased by 1.3 lbs. due to the pressure of the air delivered by the scavenge pump $\therefore p_1 = 16$ lbs. per square inch.

p_2 = pressure at end of stroke.

v_1 = volume at beginning of stroke.

v_2 = volume at end of stroke.

n = a constant = 1.4.

As an example assume a cylinder dia. of 13 inches, and a stroke of 18 inches, then $v_1 = 2,400$ cubic inches, and $v_2 = 2,400 \div 12 = 200$ cubic inches.

$$\text{then } p_1 v_1^n = p_2 v_2^n.$$

$$\therefore p_2 = p_1 \left(\frac{v_1}{v_2} \right)^n$$

$$= 16 \left(\frac{2400}{220} \right)^{1.4}$$

$$= 16 (12)^{1.4}$$

$$= 16 (\log 12 \times 1.4) = 16 \times 32.43 = 519 \text{ lbs. abs.}$$

$$= 504 \text{ lbs. sq. in. gauge pressure.}$$

The temperature at the end of compression is found by the formula

$$T_1 P_2^{\frac{n-1}{n}} = T_2 P_1^{\frac{n-1}{n}}$$

Where T_1 is the initial temperature absolute in degrees F. = atmospheric temperature $60^\circ\text{F.} + 20^\circ\text{F.}$, as the air is heated 20°F. by the hot cylinder as it enters, + 460° (the absolute zero) = 540°F.

T_2 = final absolute temperature F.°

P_1 = initial pressure in lbs. abs. = 16.

P_2 = final pressure, = 519.

$$n = 1.4 \therefore \frac{n-1}{n} = \frac{1.4-1}{1.4} = \frac{.4}{1.4} = \frac{4}{14} = \frac{2}{7}$$

$$\text{then } T_1 P_2^{\frac{n-1}{n}} = T_2 P_1^{\frac{n-1}{n}}$$

$$\therefore T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}}$$

$$= 540 \left(\frac{519}{16} \right)^{\frac{2}{7}}$$

$$= 540 (32.44)^{\frac{2}{7}} = 540 (\log. 32.44 \times \frac{2}{7}).$$

$$= 540 \times 2.702 = 1459 - 460 = 999^\circ\text{F.}$$

$$= \text{say } 1000^\circ\text{F.}$$

The fuel valve is shown at C. The oil is delivered to the valve by the fuel pump D. This pump is driven by an eccentric on the cam shaft, and the quantity of oil delivered is regulated by means of the lever R. The oil from the pump fills an annular space round the valve needle. Above it high pressure or blast air from the compressor enters. This air is delivered to the valve at a pressure greater than that in the cylinder, so that when the valve is

lifted just before (1 to 3 deg. before) the piston reaches the top of its stroke, the fuel is forced in in the form of a fine spray, and is immediately ignited by the heat in the cylinder. The charge of gas continues to burn for an appreciable period of time, representing about one-tenth of the next downward stroke of the piston, maintaining, during this time, a pressure equal to that already in the cylinder. As the piston continues to descend, the gases expand doing work, and the pressure falls until the ports are uncovered, and the cycle of operations is repeated. The maximum lift of the valve is not more than $\frac{1}{4}$ in., obtained by means of a lever E mounted on the fulcrum shaft F. This shaft is stationary, serving as an axle, about which the lever oscillates.

On the other end of the lever is a roller in contact with a cam G, keyed to a sleeve, which is also keyed to the cam shaft, and, therefore, constrained to revolve with it, but free to move axially along the shaft.

As the cam revolves, a peak on it comes in contact with the roller, causing the lever to open the valve, admitting a charge of fuel to the cylinder.

The fulcrum shaft is mounted eccentrically in bearings, and to it is attached the handle H. If this handle is pulled over it causes the fulcrum shaft to move away from the cam shaft, and thus takes the fuel valve lever off its cam, so that the valve remains closed. The same operation brings the starting valve lever K, also carried by the fulcrum shaft, into contact with the starting air cam L. This arrangement prevents the two valves from being in operation at the same time.

The cam shaft is driven by the crank shaft through two pairs of spiral gears and an intermediate vertical shaft M, and in the two-stroke engine revolves at the same rate as the crank.

Assuming that the normal direction of rotation is clockwise, the piston is shown travelling towards the bottom centre; it passes and begins to ascend. Just before the top of the cylinder is reached, the fuel cam turns into a position where its peak is about to depress the fuel lever, but should the engine be running in the opposite direction it is clear that this cam would not be in the same position relative to the piston. Some arrangement must therefore be made to alter the "timing" of the valve. This is accomplished by a duplicate cam on the same sleeve, adjacent to the first one, timed for reverse running.

When going ahead, the latter cam revolves without touching the valve lever, but when the engine is reversed the sleeve is moved along the cam shaft until the reverse cam comes under the fuel valve lever. This is performed by the forked lever N. The starting-air valve P is timed and operated in an exactly similar manner, lever H swinging it clear of its cam when not in operation, and N sliding the second or reverse cam into working position.

The sequence of operations in reversing is as follows: Assume the engine to be running ahead as indicated by the arrow. Handle H is pulled over into the neutral or stop position, in which the fuel and starting valve levers E and K are clear of all the cams. The valves remaining closed, no fuel enters the cylinder, and the engine slows down. Handle N is then pulled over, causing the cam sleeve to slide along its shaft until the reverse cams for both fuel and air valves are in line with their respective levers. F is pulled still further over, bringing the starting-air lever K into contact with its cam, and allowing the fuel valve lever to travel still further away from the fuel cam. The peak on the starting-air cam is so timed in relation to the piston that it causes the air valve to open as the piston ascends, admitting high-pressure air from the compressor to the cylinder, driving the ascending piston back, and thus starting the engine in the reverse direction.

For one or more revolutions the engine runs as a compressed air motor; as it gains speed handle H is pushed over again into the normal vertical position, shutting off the starting air, and the engine again runs on fuel, but in the opposite direction.

The fuel valve, lever, and cam are shown in greater detail in Fig. 11.

The valve needle is held on its seat by a spring in the valve casing. The revolving cam raises this needle, allowing oil in the annular space round the needle to be forced into the cylinder by the blast air. The three positions of the lever are A, ahead or astern; B, stop; C, starting air.

Figs. 12, 13 and 14 give further details of the fuel valve, showing the pulverising rings employed to break up the oil, and finally deliver it in the form of a very fine spray. This valve is shown opening into the cylinder, but the more usual form is that in which the needle opens outwards. The efficiency of the engine is greatly affected by the design of the fuel valve, for any oil which gets past it in a solid form is not completely consumed, and in addition to the loss of

the available heat units, the imperfectly consumed oil leaves a carbon deposit in the cylinder. The amount of fuel oil injected per stroke is proportional to the volume of the cylinder, as the latter contains the air in which the fuel is burnt. Hence, for any cylinder, the volume of which is known, the amount of fuel to be injected may be approximately calculated by reference to the composition of the oil. Assume that crude petroleum, 1 lb. of which contains .87 lbs. of carbon and .13 lbs. of hydrogen, is used. Then, as $2\frac{2}{3}$ lbs. of oxygen are required to consume 1 lb. of carbon, and 8 lbs. of oxygen are required to consume 1 lb. of hydrogen, therefore 1 lb. of crude oil requires $.87 \times 2\frac{2}{3} + .13 \times 8$, = 3.36 lbs. of oxygen. Air is composed of oxygen and nitrogen in the proportions of 1 to $3\frac{1}{2}$; therefore $4\frac{1}{2}$ lbs. of air contain 1 lb. of oxygen, or 3.36 lbs. of oxygen are contained in $3.36 \times 4\frac{1}{2} = 14.56$ lbs. of air. Hence, theoretically, about 15 lbs. of air are required to consume 1 lb. of crude oil, but in practice at least twice this amount is employed, as the time for complete combustion is extremely short, as shown by the following example: Assume an engine to make 200 R.P.M., combustion takes place during $1/10$ of the stroke, then time available for combustion is:—

$$1/10 \times \frac{1}{200 \times 2} \times 60 = 1/66 \text{ of a second.}$$

The starting-air valve is shown on the left in Fig. 15, which is a sectional elevation of one of Messrs. Mirrlees, Bickerton and Day's high-speed Diesel engines. The valve is only in operation when starting up the engine, either from rest or at the time of reversal. The engine shown works on the four-stroke cycle, employing valves for both air inlet and exhaust.

The employment of these valves ensures a better scavenging action than is usually obtained with ports, and saves head room, as a shorter cylinder is required, for in the ported engine the length of the cylinder is increased by the length of the ports. But there is the disadvantage associated with an increased number of valves, cams and levers, together with the additional number of pockets in the cylinder cover, the latter becoming complicated in design, expensive to mould, and difficult to cool effectively when working.

The fuel pump is an interesting and important detail of the engine, and Fig. 16 is a diagrammatic view, from which the action of the pump can be followed.

The fuel pump eccentric mounted on the cam shaft is seen in the centre of the figure at A ; directly below is a crosshead B, to which the pump plunger C is attached.

Oil flows by gravity from a tank into the casing D, which is thus kept full of fuel.

On the upward stroke of the plunger oil enters the barrel through a non-return valve E. The next downward stroke forces this oil up through the delivery valve F, into the fuel valve in the cylinder cover, against the pressure of the injection air waiting there to force it into the cylinder when the valve opens.

This, then, is quite a simple single-acting force pump. But unless there is some method of control the same quantity of fuel would be delivered each stroke, even when the engine was working at only a fraction of its full load, causing it to run away.

To regulate the amount of fuel delivered a trip gear is employed, by means of which the suction valve E is kept off its seat, allowing a portion of the oil to flow back into the casing. The action of the trip gear is as follows: In stationary and marine Diesel engines, a governor G is usually employed. The rising of the weights, due to an increase of speed, lifts link H, lever K, and link L, connected to another lever M outside the pump casing, and mounted on a spindle N, which passes through a gland into the casing. On this spindle is keyed the trip-lever O, which raises the suction valve. The spindle, trip-lever and suction valve are shown in the enlarged view. Now as the crosshead is continually moving, it follows that the lever K attached to it, and finally the suction valve E, have a fixed and definite travel, but the locus of this travel may be changed by the governor through link H, which raises the left-hand end of lever K, so that though the suction valve opens as before, it cannot now return to its seat until later in the stroke, allowing part of the oil which otherwise would have gone past the delivery valve to flow back into the casing.

In Fig. 10 no governor is shown, but the pump cross-head lever is extended to form a handle, by means of which the fuel is regulated. When starting up the engine it is necessary to flood the fuel valve with oil; and this is done by a small hand-operated plunger—not shown—forming part of the main pump. A separate pump is generally fitted for each cylinder, all the pumps being arranged in one casing, and the trip gears operated by one rod from the governor or control handle, and timed to their respective cylinders.

Fig. 17 shows a two-stage vertical compressor driven by an extension of the crank shaft, the high pressure stage being above the low pressure. Automatic suction and delivery valves are fitted to both stages, and the air drawn in from the atmosphere is first compressed by the lower piston, passing afterwards through a length of piping surrounded by cooling water to the suction valve of the high pressure stage. As the piston descends this air enters the pump, and on the return stroke is further compressed, and finally delivered to a reservoir composed of one or more drawn-steel air bottles, from which the engine receives a supply of high pressure air for starting up, and for the injection of fuel.

The theory of the compressor is simple ; a bicycle pump is an inefficient example, but to obtain the best thermal, volumetric, and mechanical efficiency great care is necessary in the design.

It has been found in practice that for injecting the fuel $6\frac{1}{2}$ to 7 litres, say 400 cubic inches of free air—air at atmospheric pressure—are required per B.H.P. per minute, equal to 15 cubic feet per hour.

The piston speed of the engine may be taken at 800 feet per minute, and the stroke-bore ratio at 1.5 to 1. From these figures the required volume of the low pressure stage is found.

As previously mentioned, the pressure in the engine cylinder, when the fuel valve opens, is at least 500 lbs. per square inch, so that the injection air must be delivered very much in excess of this, certainly not less than 750 lbs. per square inch, some makers going as high as 1,000 to 1,500 lbs.

It follows that the compressor has to raise the pressure of atmospheric air, say, 15 lbs. abs. to 750 lbs. Assuming this to be done in two stages in geometrical progression, then pressure ratio = $R = \frac{750}{15} = 50$.

Initial Pressure in Stage I. = $P = 15$ lbs. per sq. in.
 Final " " " = $P_1 = P/R$.
 = $15/50 = 106$ lbs. per sq. in.
 Final " " " II. = 750 lbs. per sq. in.

The final pressure in each cylinder of a multi-stage compressor can be found from the table below.

Number of Stages employed.	FINAL PRESSURE IN CYLINDER OF STAGE.				
	I.=P ₁	II.=P ₂	III.=P ₃	IV.=P ₄	x=P _x
1	PR	—	—	—	—
2	P $\sqrt{R}$	P ₁ $\sqrt{R}$	—	—	—
3	P $\sqrt[3]{R}$	P ₁ $\sqrt[3]{R}$	P ₂ $\sqrt[3]{R}$	—	—
4	P $\sqrt[4]{R}$	P ₁ $\sqrt[4]{R}$	P ₂ $\sqrt[4]{R}$	P ₃ $\sqrt[4]{R}$	—
x	P $\sqrt[x]{R}$	P ₁ $\sqrt[x]{R}$	P ₂ $\sqrt[x]{R}$	P ₃ $\sqrt[x]{R}$	P _{x-1} $\sqrt[x]{R}$

Where P = initial pressure in Stage I.

P_{1 2...x} = final pressure in Stages I, II....x.

R = $\frac{\text{Final Delivery Pressure}}{\text{Initial Pressure.}}$

Then having found the pressures in both stages and knowing the volume of the first stage, the volume of the second stage is given by formula $P_1 V_1^n = P_2 V_2^n$, where $n = 1.25$. The ideal compressor would work on the isothermal law of $PV = C$, i.e., without increase of temperature. It is not, however, possible to obtain this result, as the quantity of cooling water would be so great. It follows that the numerical value of "n" lies somewhere between 1.406 and 1, the values for perfect adiabatic and isothermal compression respectively.

In a series of tests of an air compressor, Professor Goodman, of Leeds University, obtained the values 1.27 and 1.28, but 1.25 is generally accepted.

To approach this figure the whole of the pump and the valves must be water-jacketed, and the air between each stage carried through a length of piping surrounded by water. The valves should be large in diameter in proportion to their lift, the latter not exceeding $\frac{1}{8}$ in. There should be the least possible clearance when the piston is at the top of the stroke. The quantity of cooling water employed varies greatly with different makers and designs, and should be as much as is reasonably possible. When the delivery pressure is high an increase in the number of stages makes for efficiency. For pressures above 500 lbs: three stages are advisable, and from 1,000 to 1,500 lbs. per square inch, four stages.

But as pressures, temperatures, stages, and cooling water are mutually dependent, a reasonable working compromise is easily obtained.

The scavenge pump is sometimes formed by the underside of the low pressure stage of the compressor, but more usually a separate piston is employed, both sides of which deliver air, resulting in a double-acting pump, which need not be water-cooled.

It is a single-stage low pressure air pump, forcing air at from 3 to 5 lbs. pressure through automatic or other valves into a reservoir in direct connection with the air valves or ports in the engine cylinder. The capacity of the pump varies from $1\frac{1}{4}$ to $1\frac{3}{4}$ times that of the engine cylinder.

Before passing to the application of this engine to locomotives there are one or two other little points to mention.

The engine cylinders and pistons are cooled either by water or oil. The usual allowance of circulating water is $4\frac{1}{2}$ gallons per B.H.P. per hour, at an initial temperature of 60 deg. F. This quantity is increased if the final temperature rises much above 120 deg. F.

Cast steel has in the past been extensively used for cylinder covers, but cast iron is now finding more favour. A really reliable cast steel has yet to be discovered; even the best that can be obtained often gives cause for great disappointment when employed in the manufacture of Diesel engines.

A small percentage of nickel is an improvement which, incidentally, adds very considerably to the cost.

Careful design, high-class materials and workmanship, and intelligent handling of the finished machine are the essentials of a successful engine. Given these, the Diesel engine will work with entire satisfaction.

THE APPLICATION OF THE DIESEL ENGINE TO THE LOCOMOTIVE.

There are three ways in which the Diesel engine can be used for driving locomotives, each possessing certain advantages.

First, the engine may be installed in power stations, and employed to drive dynamos generating current transmitted to motors carried on the locomotive by means of the overhead wire and trolley pole.

This is an indirect application, and is beyond the scope of this paper, coming rather under the heading of the electrification of railways, but the greater thermal efficiency of Diesel engines when compared with any other type of prime-mover, may be taken full advantage of in this direction.

A number of generating stations in England and throughout the world now employ Diesel engines for this purpose.

The second and more direct method of application is that of a Diesel engine coupled to a dynamo, and carried on the locomotive.

An example of this type is shown in Fig. 18. These engines are a considerable departure from the usual type, and work on a system invented by Ochelhaeüser, and improved by Professor Junkers, of Aachen.

A number of these engines have recently been put on the market by Messrs. The A.E.G. Electric Co., Ltd., of London and Berlin, who have kindly furnished photographs and the following description.

The most noticeable feature is the employment of two pistons in each cylinder, thus avoiding cylinder covers.

The lower piston is coupled to the central crank, and the top piston to a cross-beam, from which two long connecting-rods descend to side cranks. These engines work on the two-cycle principle, with port scavenge and exhaust.

Scavenge air enters at the lower row of ports, sweeping out the burnt gases through similar ports near the top of the cylinder. On the left is a two-stage air compressor, below it a double-acting scavenge pump, which delivers into a reservoir in communication with the cylinder scavenge ports. On the right is the generator, the stator of which is bolted to a flange on the crank case, giving a very rigid construction. This engine is capable of developing from 10 to 40 kilowatts—13 to 54 H.P. The injection and starting air bottles are carried in the crank case. The whole arrangement is extremely neat and compact, and entirely self-contained.

Fig. 19 shows a similar engine installed as a light shunting locomotive for a train weighing 150 tons.

This type of Diesel-driven electric locomotive has the advantage over the generating station method that it is inde-

pendent of all external current conductors. Great economy is obtained, as the voltage is regulated to the load by the speed of the engine, and not absorbed in resistances.

The A.E.G. Co. supply these engines up to 200 K.W. (280 H.P.).

This method, however, does not lend itself very well to large powers, for at the best it is indirect, the power being first in one form of energy—mechanical; then it is changed to electrical in the dynamo, and transmitted in that form to motors on the driving axles back into mechanical energy. Admittedly, very little is lost in the process, due to the high efficiency of electric generators, but there is the additional weight of both dynamo and motor to be carried, which, for large powers, becomes prohibitive; there is also the question of the additional cost of the electric transmission gear.

Mr. Durntall, in his paper on "The Evolution and Development of the Internal Combustion Locomotive," has put the mechanical-electrical locomotives before you and has covered the ground in such a capable manner that there is no need in this paper to more than mention the connection of the Diesel engine with it as one form of prime-mover likely to come into prominence in the near future.

The third, and possibly most interesting method of employing the Diesel engine for locomotive driving is the direct application of the engine to the driving wheels. Only one locomotive has been built up to the present time in which the Diesel engine is directly connected in this way.

It was completed early last year by Messrs. Sulzer Bros., Ltd., of Winterthur, and has been described in "Engineering." Messrs. Sulzer have kindly supplied, for the purpose of this paper, a number of drawings of their engine, together with a full description of the various parts.

A photograph of the locomotive is shown in Fig. 20. It will be seen that the engine is entirely enclosed, and being alike at both ends, can be worked from either end. It is built on two $\frac{7}{8}$ in. stayed plate frames extending the whole length, supported by two bogies.

The Westinghouse brake is fitted to all wheels, in addition to a hand brake on the driving wheels. The four-coupled driving wheels are seen in the middle, and are driven through side rods attached to a disc crank, the axle of which forms the engine crank shaft.

The overall length is 54ft. 5½in. ; the driving wheel base, 11ft. 9½in. ; the bogie wheel base, 7ft. 3in. ; the bogie centres are 34ft. 6in. apart ; the diameter of the driving wheels, 5ft. 8¾in. ; and of the bogie wheels, 3ft. 3½in. The driving axles are 7⅞in. dia., and the journals 8¼in. dia. by 10¼in. ; the coupling pins 4in. dia. by 4¾in.

The engine will take curves of 590ft. radius, and in running order weighs 95 tons, developing a maximum of 1,600 B.H.P. at 300 R.P.M., equal to a speed of 63 M.P.H., which, however, it could not develop over a long period, as the mean effective pressure at this power is about 170 lbs. per sq. in. ; this is abnormally high ; from 90 to 120 lbs. per sq. in. is the maximum usually reached in Diesel engines. The makers give the nominal brake horse-power as 1,000.

A side elevation and plan, with the casing removed, is shown in Figs. 21 and 22 ; also two cross-sectional views in Figs. 23 and 24. The main engine occupies the centre of the locomotive. It is a four-cylinder, single-acting, direct reversible two-cycle V engine. The cylinders are in pairs, inclined at an angle of 45 deg. to the vertical centre line. The connecting rods of opposed cylinders are attached to the same crank pin, the big end of one being forked, bearing on the pin on each side of the other rod as shown in Fig. 23.

There are, therefore, only two cranks, set at 180 deg., for the four cylinders. The disc cranks, to which the side rods are attached, are also shown in this view. The crank shaft is carried in three bearings set in a cast steel frame, to which the cylinders are bolted. In the space above the crank shaft, and between the two pairs of cylinders, is a small three-stage air compressor, and two double-acting scavenge pumps, all driven by means of links and levers from the main connecting rods.

The exhaust pipes are taken up to a long horizontal silencer, the outlet of which passes through the roof, the burnt gases escaping here almost without noise or colour, having been broken up by a series of deflecting plates arranged at the joints of the silencer. Forward of the main engine is a horizontal auxiliary compressor driven by a vertical two-cylinder Diesel engine of 250 B.H.P.

Behind are eight vertical starting-air bottles and two injection bottles.

This large reserve of air is necessary because the locomotive may be called upon to make frequent stops and reversals, and having stopped, becomes dependent on the air

supply for getting away again ; hence, one reason for the auxiliary compressor, which may be kept working whilst the main engine and small compressor are standing.

The driver manipulates the engine from either end of the locomotive, each cab containing a reversing hand-wheel connected by levers and rods to the engine, levers operating the fuel valve, starting valve, fuel pump, brakes, whistle, sand gear, and the air bottle pressure gauges.

Fig. 25 is an outline drawing of the engine, showing the arrangement of the valve gear. Full lines indicate the position of the gear for running ahead, and chain dotted lines for reverse.

The eccentric shown in full is on the near side, whilst that shown dotted is on the far side of the engine.

These two loose eccentrics control all the valves, both for ahead and reverse running, and are operated by means of the link A, which terminates in the control gear at both ends of the locomotive. The eccentric D is moved into the reverse position by means of link A, lever B, and the short link C, Eccentric E at the same time taking up a corresponding position.

The eccentric, by means of a pin joint at the top, actuates link F, and lever G, which in turn gives a reciprocating motion to cams H. The peaks on the cams acting on rollers and bell-crank levers, open the various valves.

It will be noticed that this gear varies considerably from that described earlier in the paper. It is, however, simply another means of arriving at the same end, the eccentrics taking the place of the revolving vertical shaft and sliding cam sleeve, whilst the reciprocating cams act like the revolving cams. It would be hard to say which was the better method, though it is certain that Messrs. Sulzer would consider all the available methods before deciding on the type most suitable to their design of engine when applied to locomotive work.

On the left-hand cylinder the two scavenge valves are shown, these being employed instead of ports. They open inwards into the cylinder, admitting air normally at a pressure of 5 lbs. per sq. in., which can be increased to 20 lbs. by the help of a second scavenge pump on the auxiliary compressor, when at starting or for any other reason the engine is required to exert for a short time its maximum power.

This additional air and initial pressure increases the final pressure, supplying the larger volume of air required to consume additional fuel, and increasing the B.H.P. from the normal of 1,000 to 1,600. Both valves open together, being operated by one cam, the profile of which is clearly shown.

The starting-air valve is shown near the centre, partly behind the scavenge valves.

On the right-hand side the fuel and one of the scavenge valves are shown ; others are left out for clearness.

The method of varying the opening of the fuel valve, and consequently the amount of fuel admitted, is shown separately in Fig. 26. By means of the vertical rod K and curved lever, the push rod J can be moved nearer or further from the fulcrum of the curved lever. When nearer, the travel of the push rod obviously becomes less ; hence, also, that of the valve.

The starting valve is worked by another cam shown in detail in Fig. 27. A rod L pushes an eccentrically mounted bell-crank lever clear of its cam when starting air is not required. The two rods K and L are interlocked and coupled up to the operating gear in the cabs, so that both valves cannot be in operation at the same time. When the eccentrics are set over for reverse running, only the timing of the valves is altered.

Fig. 28 is an enlarged view of the cylinder and cover, showing in greater detail the fuel and scavenge valves.

The cylinder liner is made of hard cast iron. It is spigoted into the cover, and secured by bolts passing right through to the cast steel engine frame.

The nickel steel scavenge valves open inwards into the compression clearance space above the spigot. A number of grooves are turned in the spindle to retain lubricating oil and prevent the escape of air. A guide containing a roller, on which the operating lever rests, forms the end of the spindle, a coil spring ensuring the return of the valve to its seat. The fuel valve is central to facilitate the spraying and burning of the oil. The nickel steel needle opens outwards. It is lifted by a forked lever, which is keyed to a shaft passing through a gland in the valve casing.

As the whole of this valve is enclosed, the pressure inside the casing is equal to that of the cylinder, so that tight glands round the needle are avoided, reducing friction and

minimising the liability of the needle sticking ; hence, lighter springs are employed than would be the case if a valve of the open type were used. The cover, as well as the cylinder, is water jacketed ; the passages between the cover and jacket are shown. One of the exhaust ports, a number of which form a band round the liner, is shown ; they are uncovered by the piston in its descent. The diameter of each of the four cylinders is 15 in., and the stroke $21\frac{3}{4}$ in. Fuel is injected by blast air at a pressure of 1,000 lbs., and starting air is supplied from the bottles at 700 lbs. per sq. in.

Figs. 29, 30 and 31 show the two double-acting scavenge air pumps, and the three-stage compressor already referred to as occupying a position between the working cylinders of the engine.

Their capacity is only sufficient to keep the engine going, and is not large enough to keep the bottles charged as well if, due to frequent stopping, a great demand has been made on them. Fig. 31 is a section through the compressor. It is mounted on the bracket which carries the engine cylinders; the through bolts from the cylinder covers, and one of the connecting rods A, are shown. Attached to this rod near the big end is a double link B, transmitting through a lever C, and shaft D, a reciprocating motion to the lever E, which, in turn, drives the compressor pistons, of which there are three, the low pressure or first-stage on the left, and on the right the second and third stages.

Air at atmospheric pressure is drawn into stage one at F, and discharged by the rising piston at G ; it then enters stage two through the underside of the compound valve H, is further compressed and discharged through the top of valve H and the pipe K into an intercooling chamber. On the next down stroke of the piston it enters the third stage, is there again compressed to the working pressure of 1,000 lbs. per sq. in., and delivered through pipe L to the injection bottles. Notice the very small clearances between the pistons and their covers when at the top of the stroke, and between the valves and the cylinders.

All cylinders and valves are water-jacketed. The cooling water inlets and outlets are indicated.

On the far side of the compressor are two double-acting scavenge pumps, driven in the same way by the connecting rod of the adjacent cylinder, making it necessary for the shaft D to be in two separate lengths, as one connecting rod is up when the other is down, the cranks being at 180 deg.

These pumps are clearly shown in Figs. 29 and 30. Air to the top side of the piston is drawn in at M and N, and discharged at O and P, the underside receiving air through R and S, and delivering it at T and U to the engine scavenge valves. The scavenge pumps are not water-cooled, due to the comparatively light load at which they work, as the air is compressed to only a few pounds above the atmosphere.

The auxiliary compressor is shown in Fig. 32. On one bed plate is carried a two-cylinder vertical Diesel engine, and a horizontal compressor and scavenge pump. The connecting rods of both engine and pump act on the same crank shaft. Dealing first with the engine. It is a two-cylinder, single-acting, two-stroke vertical engine of 250 B.H.P. at 350 R.P.M., the cylinders 12in. dia. by 15in. stroke, cranks at 180 deg., with two scavenge valves, one fuel and one starting valve to each cylinder. It exhausts through ports in the cylinder, and is not reversible. The valves are operated by a cam shaft in the usual way.

The cylinder, cover and piston are water-cooled. A governor is fitted to regulate the output to the amount of air required. The valves are similar to those already described. Through bolts from cylinder covers to bed-plate, and forced lubrication are again employed. By the side of the engine connecting rods, and working on the same pins, are the connecting rods of the three-stage compressor.

The scavenge air for the auxiliary and main engines is compressed on the left-hand side of the large pistons; the right-hand side of both forms the low pressure or first stage of the compressor.

The smaller piston shown is that of stage two, whilst stage three is obtained by a similar but still smaller piston occupying a corresponding position opposite the second crank. Water-jackets and intercoolers are fitted. The air is delivered to distributing valves controlled in the cabs. From these valves it passes to the bottles, or direct to either engine as required. Primary balancing is effected by weights in the two fly-wheels.

Assume the locomotive to be standing in the shed, with oil tanks, water tanks, and starting bottles full, *i.e.*, in running condition ready to be taken out, but with the main and auxiliary engines stopped. The valve connecting the starting bottles to the main engine is opened, and the locomotive at once begins to move on compressed air. In less

than half-a-minute a speed of between five and six miles an hour is attained, at which speed the engine is able to work on fuel, so the starting-valve lever is pushed back, releasing the fuel lever, and the engine gets away, showing a considerable saving of time compared with the steam engine, which has to wait until steam is formed.

The auxiliary is now started by air from the bottles, to charge them again, and assist the smaller compressor. The latter, of course, only works when the engine is running. When the engine is returned to its shed at night, the auxiliary is kept working for a few minutes to fill up the bottles ready for the next start. Should all the bottles by any chance be empty—a possible though not probable contingency—one at least of them must be filled by means of a small portable compressor of 3 or 4 H.P., kept in the shed. When this bottle is charged the auxiliary is set going, and finally the main engines are started. To stop, the fuel is first cut off and then the brakes applied, the Westinghouse brakes taking the required air from the starting bottles.

This locomotive has been in work on the Prussian State Railways for the last eighteen months, and is the only direct-coupled Diesel engine locomotive made up to the present time. It has, however, given entire satisfaction, both to the makers and to the railway. In a letter from Messrs. Sulzer it is stated that "it is their first attempt to utilise internal combustion engines for a direct drive on a locomotive of any high power, and that they are so satisfied with the results that they are ready to offer locomotives up to 2,200 effective horse-power at a total weight of 136 tons." They add that they are "also studying the question of Diesel electric locomotives, and have supplied a number on this principle to the Saxon State Railways; the engines in this case are of only 200 B.H.P., but the results are quite satisfactory."

Apparently Messrs. Sulzer have great confidence in this type of locomotive, and it says something for the quality of their work that the first serious attempt should have answered so well.

Messrs. Sulzer have a reputation as engineers which stands very high, and they may be relied upon not to stop at their first success. Probably any future Diesel locomotive of theirs will be an advance on the one described. Engineers do not expect to reach perfection at one stride, and there are several points about this engine which will probably come in for adverse criticism.

The weight and length of the engine compare quite favourably with that of the modern British steam locomotive of similar power, as shown in the following table :—

Locomotive.	Type.	Weight in working order of Engine and Tender.	Overall Length of Engine and Tender.
Sulzer	4—4—4	95 tons	54ft. 6in.
N.E.R.	{ Class R 4—4—0	105 tons	58ft. 3½in.
N.B.R.	4—4—2	119 tons 16 cwts.	63ft. 0½in.
L. & N.W.R. ...	{ Sir G. Claughton 4—6—0	117 tons	63ft. 4¾in.

The figures for the three British locomotives are taken from Mr. Dearberg's paper on "The Standardisation of Large Passenger Locomotives."

Until one becomes accustomed to the Diesel engine, it appears very much more complicated than the steam engine, but on better acquaintance the prejudice against these new parts becomes at least considerably modified. The auxiliary engine may be at first regarded unkindly, but when one considers that it takes the place of the boiler and furnace, and is ready for work in infinitely less time, it can be regarded with less disfavour. A larger direct-driven compressor and scavenge pump, enabling a smaller auxiliary to be used, would probably have been an advantage.

This small auxiliary could then have been of the so-called semi-Diesel or hot bulb type, as the Diesel engine is essentially for large rather than small powers. The hot bulb engine, with its lower compression, can be started by hand, and though not quite as efficient, the cost of the additional fuel consumed is negligible, a difference of not more than 0.1 to 0.2 lbs. of fuel per B.H.P. per hour.

A point has already been reached in steam locomotives beyond which the demand on the stoker cannot be greatly increased, at any rate, with coal as the fuel, though if the adoption of liquid fuel in the furnace became general, one stoker could attend to engines of greater power. But burning oil in the furnace is more costly than burning coal, except in oil-producing countries, so if used at all it should be employed to the greatest advantage by direct burning in internal combustion engines.

In the Diesel locomotive, the stoker is replaced by an engine attendant, whose duty is to see that the engine is running freely. The control is with the driver, who attends to the signals and keeps a watchful eye on the running of the engine as indicated by gauges in his cab.

The comfort of both men is greater, as they are not exposed in wet and cold weather, or half melted by heat from a firebox. The attendant is not exhausted by the constant shovelling of coal; he is subordinate to the driver, and works his way up to that position as in present practice, passing successively from cleaner and attendant to driver.

An engine of 2,000 H.P. makes no greater demand on either man's strength than one of 1,000 H.P.

The Sulzer locomotive has one serious disadvantage when compared with the steam locomotive, in that it starts from rest as a compressed air engine, and for the first few moments is therefore terribly inefficient, whilst the steam engine starts with the full power of the boiler behind its pistons. Hence, it does not get up speed so quickly, and shows to greater advantage on long runs than it would on suburban traffic, or shunting work, for which it is obviously not suited, though, at the same time, when it is standing, no fuel is being consumed. A possible solution of this difficulty is the adoption of clutches, and perhaps gearing, at both of which one has been taught to look askance. The clutch suggests sudden jerks and personal discomfort, but here again it is just a question of getting the right design.

Messrs. The Coil Clutch Co., Ltd., of Johnstone, write stating that their clutches will take up a load without shock, can be positively locked when full power is being transmitted, and can be made to work up to 8,000 H.P.

They add that they would have no difficulty in supplying special clutches capable of starting the full load from rest in either direction.

There is, however, no room for these clutches as well as the engine cranks between the frames of a locomotive, so that it seems a lay shaft and gearing will be necessary.

When geared turbines for ship propulsion were first proposed, they came in for much adverse criticism, but they have outlived it, and are giving entire satisfaction in an increasing number of ships. Probably something on the same lines could be employed with equal success. With

their adoption, starting troubles would be avoided, as the full power of the engine would be immediately available.

As time goes on the Diesel engine will be considerably simplified, and the evolution of the perfect motor will occupy very much less time than that taken by the steam engine. But experimenters must be prepared for only partial success at first, and a large expenditure of money and brains before perfection is reached.

On land and in marine installations success is rapidly approaching, and the locomotive is probably the most difficult proposition of all. That success will come sooner or later is certain, but how soon depends on the enthusiasm and capital pioneers are prepared to spend in attaining it.

The writer wishes to express his indebtedness to his friend Mr. W. Hamilton Purdie for many suggestions in connection with this paper, and also to the firms who have provided him with information and illustrations, namely :—

Messrs. Mirrlees, Bickerton and Day, Ltd., Stockport.

Messrs. The A.E.G. Electric Co., Ltd., London.

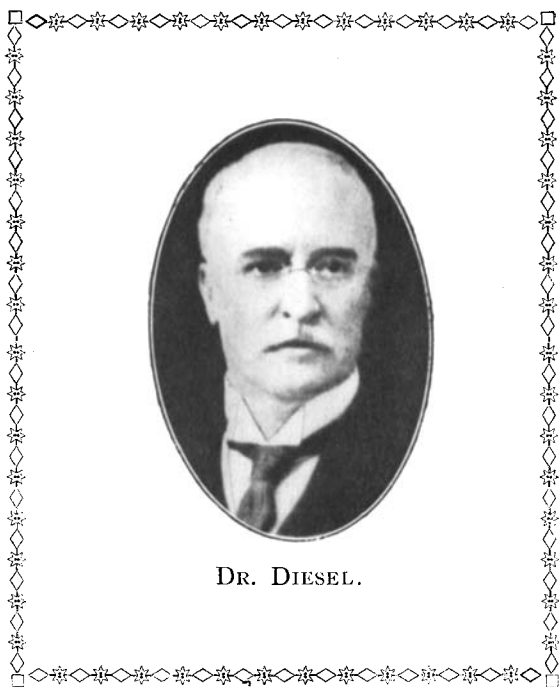
Messrs. Sulzer Bros., Ltd., London and Winterthur.

Messrs. Alley and MacLellan, Ltd., Glasgow.

Messrs. The Coil Clutch Co., Ltd., Johnstone.

He is also indebted to the following works of reference, and further information on the various subjects dealt with will be found in :—

Prof. Ewing's work on "Heat Engines"; Mr. A. P. Chalkley's "Diesel Engines"; Messrs. Watts and Taylor's "The Diesel Engine"; Prof. V. B. Lewes's "Oil Fuel"; and to the Technical Press in general, including "Engineering" and "Internal Combustion Engineering."



DR. DIESEL.

Fig. 1.

— HEAT CYCLE DIAGRAMS —

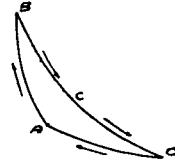


FIG. 2. — CARNOT —
or CONSTANT TEMPERATURE CYCLE.

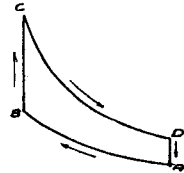


FIG. 3. — OTTO —
or CONSTANT VOLUME CYCLE.

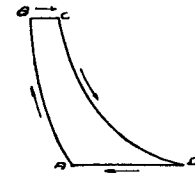


FIG. 4. — BRAYTON —
or CONSTANT PRESSURE CYCLE.

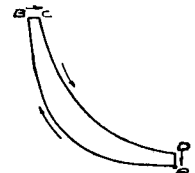


FIG. 5. — DIESEL —
A SIMILARITY OF FIG. 3 AND 4.

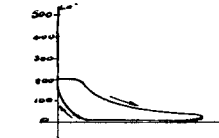


FIG. 6. — STEAM ENGINE —

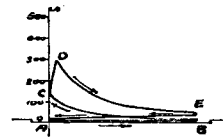


FIG. 7. — CUMMINS ENGINE —

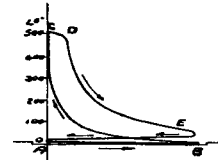


FIG. 8. — FIVE STROKE DIESEL ENGINE —

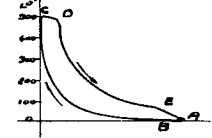


FIG. 9. — TWO STROKE DIESEL ENGINE —

— INDICATOR CARDS —

Figs. 2, 3, 4, 5, 6, 7, 8, and 9.

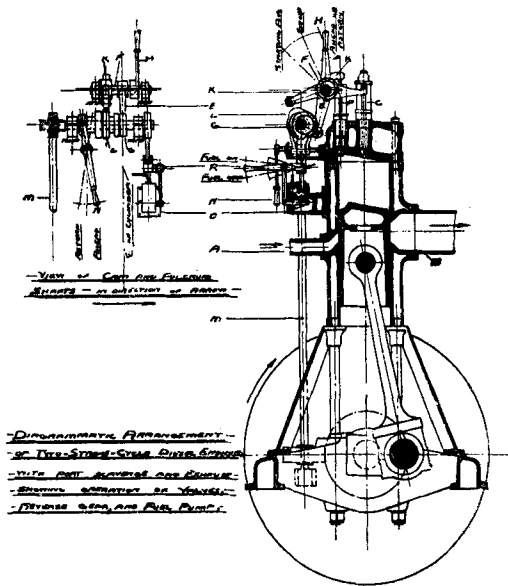


Fig. 10.

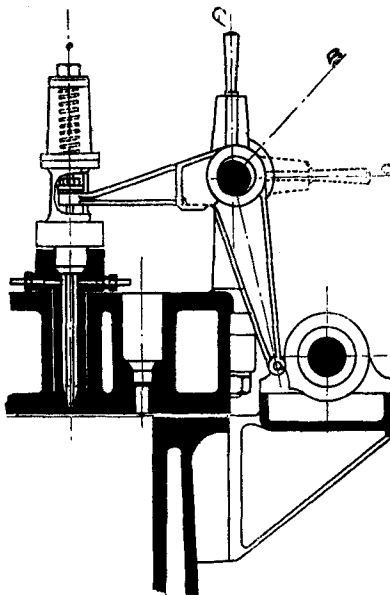
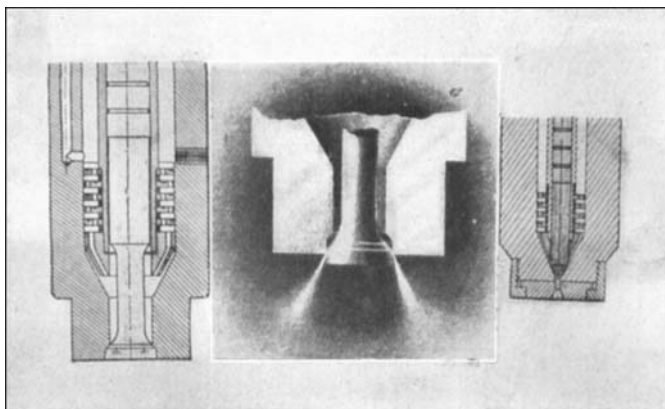


Fig. 11.



Figs. 12, 13 and 14.

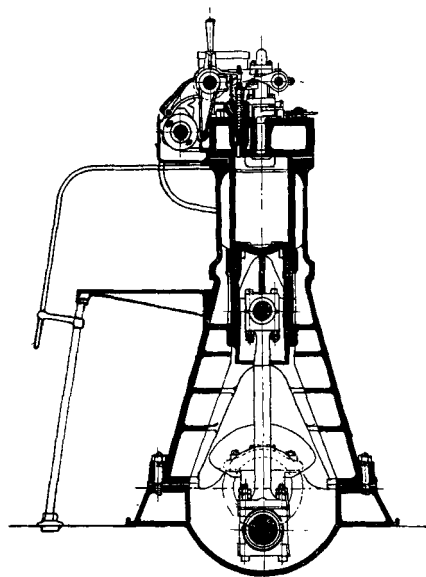


Fig. 15.

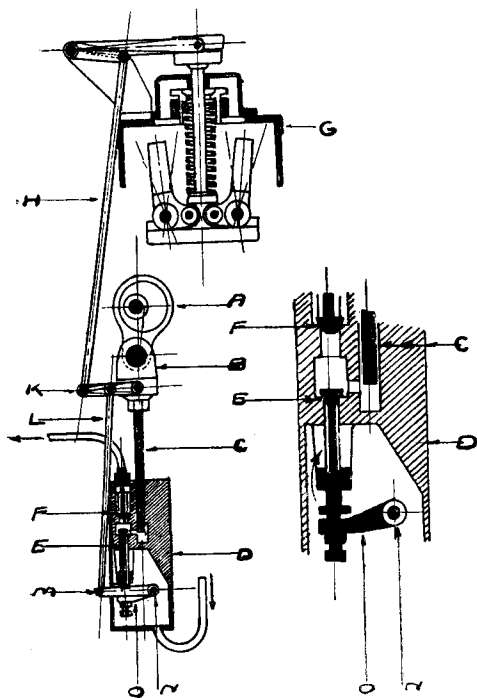


Fig. 16.

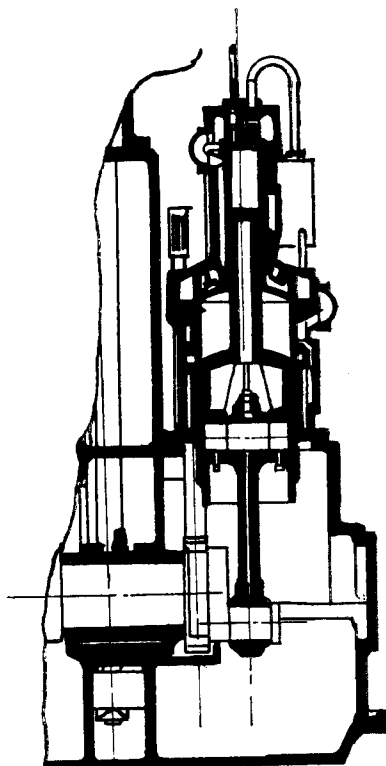


Fig. 17.

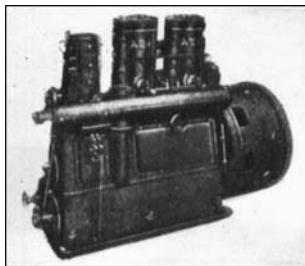


Fig. 18.

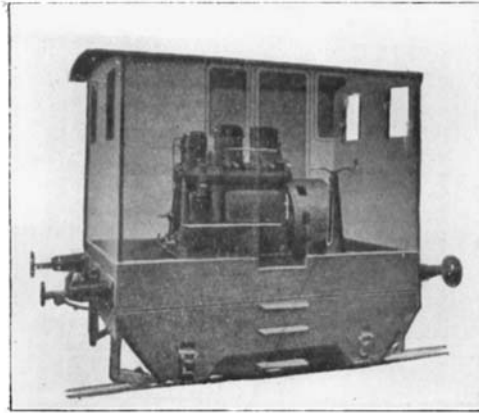


Fig. 19.

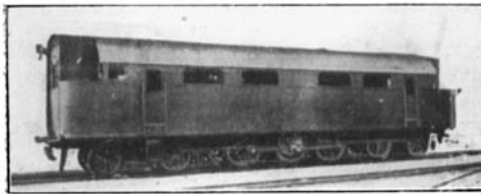
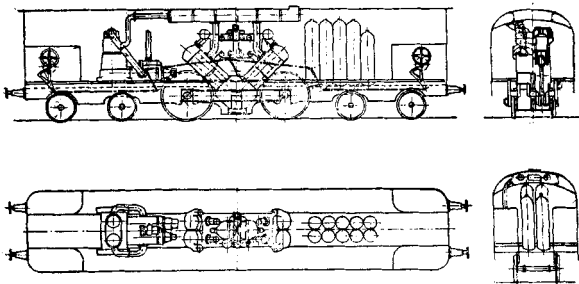
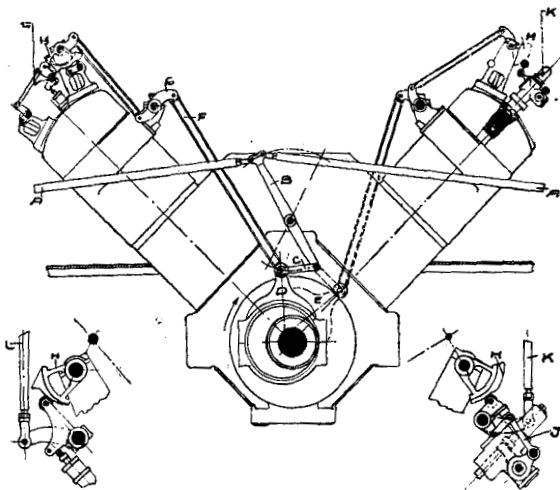


Fig. 20.



Figs. 21, 22, 23, and 24.



Figs. 27, 25 and 26.

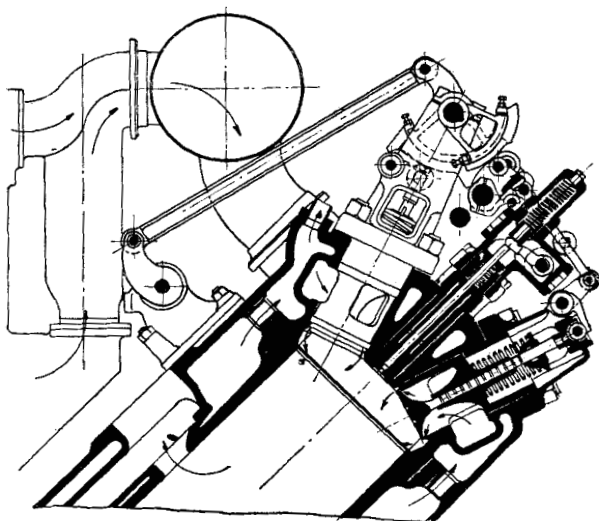
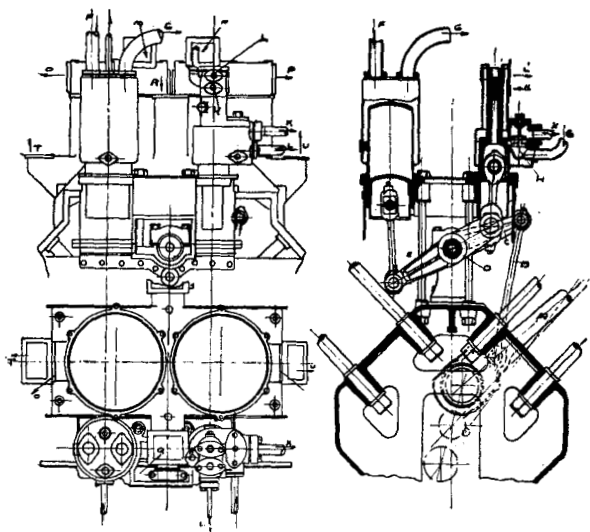


Fig. 28.



Figs. 29, 30 and 31.

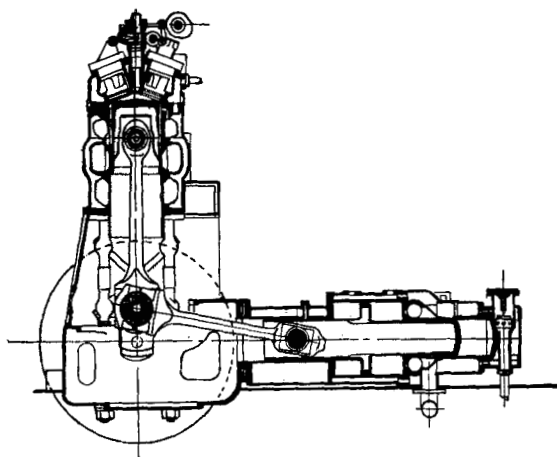


Fig. 32.