Sept. 21, 1937. H. TRIEBNIGG 2,093,592

INTERNAL COMBUSTION ENGINE

Original Filed Feb. 16, 1932 2 Sheets-Sheet 1

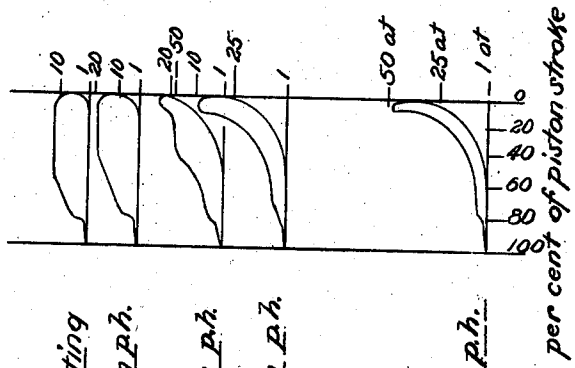


Fig. 2

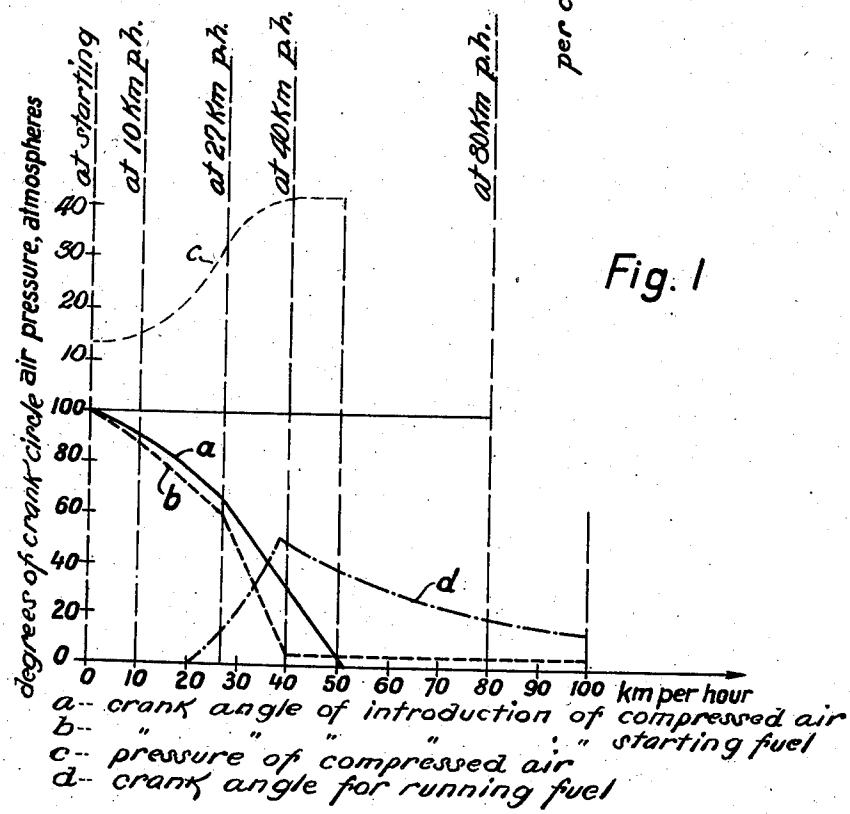


Fig. 1

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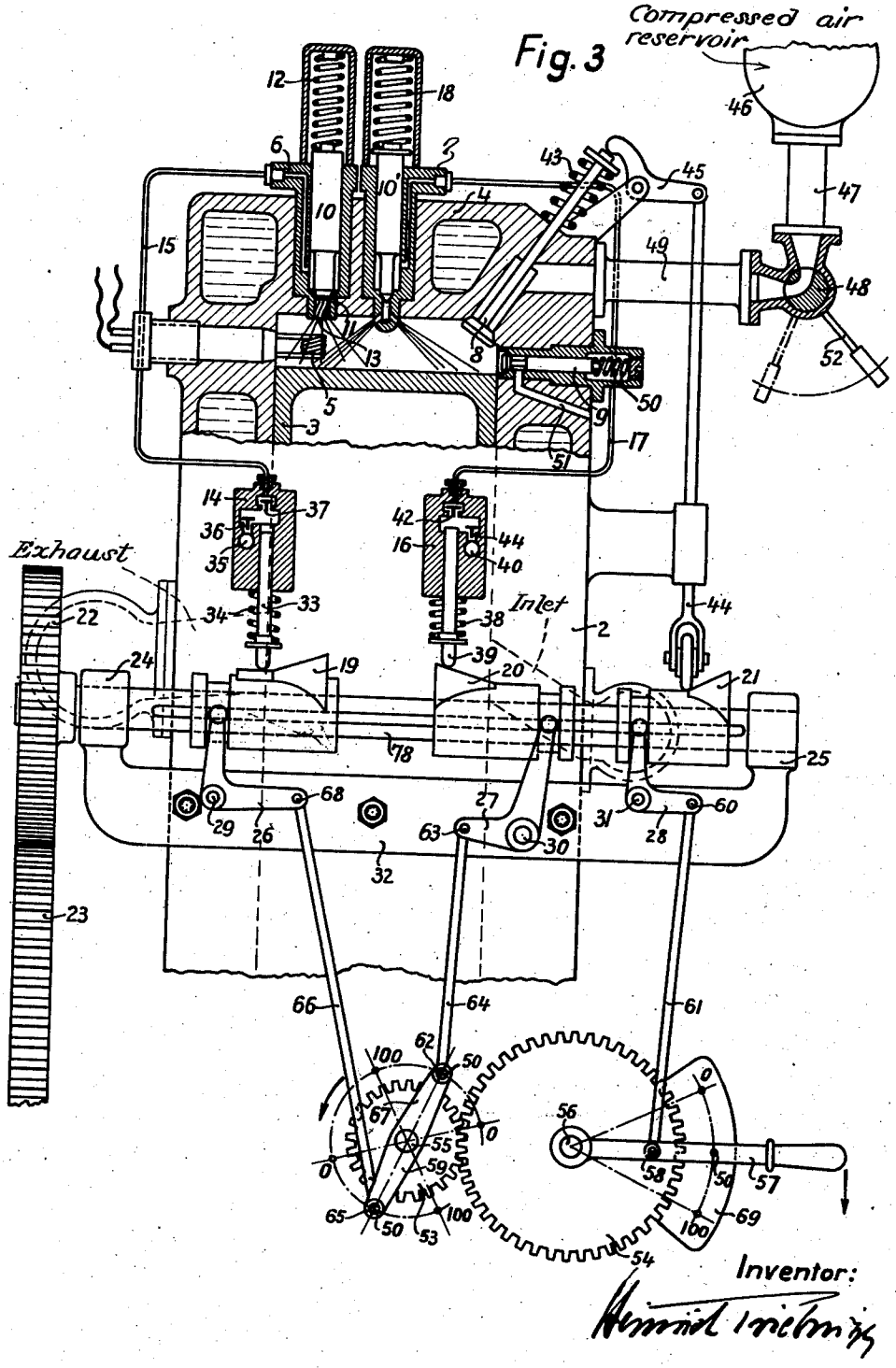
Heinrich Triebnigg

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Fig. 3



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2,093,592

INTERNAL COMBUSTION ENGINE

Heinrich Triebnigg, Cologne, Germany, assignor to Humboldt-Deutzmotoren A. G., Cologne, Deutz, Germany

Original application February 16, 1932, Serial No. 593,378. Divided and this application September 24, 1934, Serial No. 745,344. In Germany February 18, 1931

5 Claims. (Cl. 123—26)

This invention relates to locomotives driven by internal combustion engines, preferably Diesel engines, and apparatus for starting and accelerating such locomotives.

This application is a division of application Serial No. 593,378, filed February 16, 1932, now Patent No. 2,010,469, dated August 6, 1935.

The invention makes possible the starting of the driving engine of a locomotive from its state of rest without disengaging the driving connection to the axles and without the use of transmission gears to step up the driving ratio as the locomotive accelerates. According to the invention, the combustion of fuel and compressed air takes place beginning with the very first revolution, and the usual manner of starting the engine by introducing solely compressed air into the working cylinders is abandoned. The advantage gained hereby is that compressed air storage capacity of the engine need be only very small, since the pressure is raised in the engine cylinder by the combustion of fuel. Thus it is even possible to supply the compressed air by a relatively small auxiliary compressor. The compressed air and fuel without air are brought into the combustion chamber, each through a separate valve for the major part of the working stroke of the piston. Ignition is effected by a glowing body, for example an electrically heated element.

In the operation of the engine during starting, according to the invention, a low combustion pressure is used, as compared with the ordinary Diesel engine operation in which self-ignition or compression-ignition in consequence of the high degree of compression, takes place. The low combustion pressure may be secured for example by blowing out or releasing part of the air drawn in by the piston during the suction stroke, or by throttling the intake manifold. The pressure in this operation may even be reduced to atmospheric pressure in the case that a small tractive force is required. On the other hand, the compressed air and fuel are charged into the cylinder beginning about the inner dead center over a comparatively large part of the working stroke, for example 50%. By these two measures, i. e., lowering the combustion pressure and extending the time of charging the combustion chamber during the working stroke, an indicater diagram is obtained which is similar to that of a steam engine. By reason of this indicator diagram, the engine may act on the axles of a locomotive simply by means of connecting rods and cranks as is usual in steam locomotives, no other gears being required.

The invention may be best understood by the following description thereof taken in conjunction with the accompanying drawings, wherein

Fig. 1 shows curves of the regulation of a Diesel engine according to the instant invention for different speeds attained by the locomotive,

Fig. 2 represents the indicator diagrams of the engine at corresponding speeds of the locomotive indicated on the axis of abscissas in Fig. 1, and

Fig. 3 is an elevational view, partly in section, of the engine with the regulating means therefor.

The means for executing the above described mode of operation, such as the fuel valves for airless injection, the compressed air valve and scavenging air inlet and exhaust ports or valves, respectively, are of the usual kind. The actuating means for the valves are designed accordingly. An ignition device, as for example, an electrically heated wire spiral of a heat resisting material, as nichrotherm or a ceramic substance as carborundum or silicon, is always necessary for the starting operation. The heating element must possess a high caloric capacity which renders it indifferent to the cooling effect of fuel or compressed air thrown against it. Finally the valve for diminishing the compression in the engine is new, as compared with the usual type.

The mode of operation and means according to this invention in the transition from starting the engine with combustion of fuel and compressed air to the normal Diesel-cycle with compression-ignition will now be described. During acceleration of the engine a gradual transition takes place to the normal working cycle, for example the so-called Diesel-cycle, in which self-ignition of the fuel occurs due to the high compression temperature. As the engine is accelerated, the charge of compressed air is gradually decreased to zero by reduction of the charging time relative to the working stroke. Simultaneously the time of fuel injection is reduced, which time is generally proportional to the charge of fuel and which is much longer during starting than in normal working of the engine. The normal compression pressure is also restored during the same time. In accordance with the increase in compression pressure, the pressure of the charging air must be raised to overcome the increasing pressure in the cylinder.

According to the invention, two different fuel valves or sets of valves and also two fuel pumps are applied, one of which operates during starting, and the other one during normal running of the engine. As shown in Fig. 3, the former discharges fuel into the chamber with a large dispersion of low force in the vicinity of a glowing

ignition element and the latter in a spray of lesser dispersion of higher force. Each pump delivers fuel to one of the two different fuel valves in the cylinder head. One of these valves is designed especially for starting requirements, i. e. so that the fuel is dispersed at low force very finely without impinging on the cylinder walls. This valve is loaded, for example, to about 50 to 150 atmospheres. The nozzle holes accordingly are dimensioned small, or another kind of resistance is provided in the valve. The foggy spray of fuel will ignite surely upon touching the glowing ignition body. The pump pressure for this nozzle is comparatively low, as on account of the proximity of the valve to the ignition body, the fuel particles need not traverse the whole combustion chamber. It would indeed be disadvantageous if the fuel would reach the relatively cool cylinder wall without igniting. The second fuel valve must be dimensioned for the purpose of injection in the normal Diesel-cycle. This valve is loaded, for example to about 300 atmospheres by a spring, against the compression force of which the valve opens. The operating mechanism of the compressed air valve and of the fuel pumps are regulated together, so that when first starting, only compressed air and the first fuel pump operate and give their maximum charge. With growing speed, and in dependency thereon, both air and the first fuel pump are cut off gradually, and simultaneously the second pump begins to act through the other fuel valve. For a certain time during acceleration, the delivery of both pumps may overlap.

A more detailed manner of regulation is diagrammatically illustrated in Fig. 1. The abscissas represent the travelling speed of the locomotive in kilometers per hour. The vertical distances (ordinates) from the base line to the curve *a* are the degrees of angle of the crank circle in the working stroke counted from the inner dead center, during which compressed air is admitted to the cylinder. At 50 km. per hour the air is cut off altogether. The curve *b*, representing the angular duration of fuel injection through the first (starting) fuel valve, is very similar. At 40 kms. per hour, the fuel delivery of this valve is nearly interrupted. For higher velocities always a very small amount is discharged to avoid choking of the nozzles. The injection pressure *c* of starting compressed air rises from nearly 15 atmospheres at the beginning of operation to about 40 atmospheres at 50 kms. per hour for the reason that the increasing combustion pressure in the cylinder must be overcome. The second fuel pump begins to operate, for example at 20 kms. per hour, as shown by the curve *d*, and at almost 40 kms. per hour, where the actual Diesel-cycle begins, fuel is injected by the second fuel pump continuously while the crank is traveling about 50° from top dead center. As the speed increases above 40 kms. per hour the period of delivery drops continuously in accordance with the decrease of the tractive force as required by the demand of a constant output of the engine.

In Fig. 2 are shown several indicator diagrams taken during starting and acceleration of a ported-valve two stroke cycle engine, each diagram being placed substantially above the abscissa of Fig. 1 corresponding to the speed at which it is taken. The diagram shown at the furthest left shows the change of pressure in starting. Compressed air and fuel are forced into the cylinder during about 60% of the work stroke. Compression does not take place at this stage and the combustion pressure is relatively low. The second diagram corresponds to a speed of 10 kilometers per hour. Here there is only a slight compression, on account of the pressure and heat losses due to the low piston speed, or on account of the holding open of the compression relief valve for a certain portion of the compression stroke. The combustion pressure is somewhat higher than in the first diagram and the time during which the fuel and compressed air are fed in can therefore be slightly reduced, substantially to one-half of the work stroke. The third diagram was taken at a speed of 26 kilometers per hour and shows a further increase of the compression pressure, and a further reduction of the period during which compressed air and fuel are introduced. Furthermore the influence of the second fuel pump, which begins to operate at a speed of about 20 kilometers per hour, can be recognized by the pressure peak in the first part of the work stroke. The fourth diagram, which corresponds to a traveling speed of about 40–50 kilometers per hour, is an ordinary Diesel diagram, which shows still a little supercharging by compressed air, and the fifth diagram, which corresponds to the traveling speed of 80 kilometers, is a normal Diesel diagram without super-charging.

The regulating mechanism and part of the engine adapted to carry out the cycle described are illustrated in Fig. 3 of the drawings, in which **2** is the working cylinder, and **3** is the piston in its upper dead center position. The cylinder **2** has an inlet port **200** and an exhaust port **201**, both controlled by the piston **3**. Through the cylinder head **4** there extends an electrically heated ignition element **5**, the fuel valve **6** for starting, the fuel valve **7** for the ordinary Diesel-cycle, the compressed-air starting valve **8** and the compression relief valve **9**. The needle **10** of valve **6** is pressed on its seat **11** by spring **12**, and it terminates in a spiral-grooved point **13** which causes large dispersion of the fuel. The valve opens in the known manner by fuel pressure transmitted by the fuel pump **14** through the pipe line **15**. The valve **7** is fed by the fuel pump **16** through the pipe **17**. The spring **18** on the needle **10'** of valve **7** is stronger than spring **12** of valve **6**. Both fuel pumps and also the compressed air valve **8** are operated by sloped cams **19, 20, 21** slidably mounted on the common shaft **78** which is driven by gears **22** and **23** from the main crank shaft not shown. The shaft **78** is supported in bearings **24** and **25**. There are three bell crank levers **26, 27** and **28** fulcrumed at **29, 30** and **31** in the bracket **32**. The stroke of the plunger **33** of the fuel pump is varied with the lift of the cam **19**. The cam is formed so that the plunger always starts to pump through the pressure valve **37** while the piston **3** of the engine is about its inner dead center, and finishes its delivering stroke at a later period in direct proportion to the height of the lift. The spring **34** effects the suction from the suction channel **35** through the valve **36**. The pump **16** acts on the same principle. The plunger **39** is actuated by the cam **20** in the delivering stroke and by the spring **38** in the suction stroke. The suction channel is designated **40**, the suction valve **44**, and the pressure valve **42**. The cam **20** is formed so as to pump only about the inner dead center of the engine piston. By shifting the cam, mainly the amount of fuel is varied, but only slightly is the time of injection changed. The cam **21** opens the compressed air valve **8** against the

compression force of the spring 43 by means of the push rod 44 and the bell crank lever 45. This cam is of similar cross section to the cam 19, i. e. by engaging with the push rod 44, it opens the valve always about the inner dead center and changes the time or duration of communication of the combustion chamber with the source of compressed air. The compressed air is admitted to the valve 8 from the vessel 46 through pipe 47, rotary valve 48 and pipe 49. This valve 48 serves to throttle the air pressure for the initial operation and to render free the full pressure during acceleration of the engine in the manner as shown in the diagram of Fig. 1 by the curve c. The rotary valve is adjusted by the hand lever 52. The compression relieving valve 9 is held open by the spring 50. The combustion space then communicates with the atmosphere by way of the hole 51. If the piston 3 moves upward slowly during the compression stroke, as in the very first movement during starting, the current of the outgoing air will not close the valve 9. But as the engine gains speed the force of the air blown out through the valve 9 will close the valve against the tension of spring 50, and as the piston always starts its upward movement from the lower dead center with the velocity equal to zero, the closing of the valve occurs earlier in the piston stroke as the speed of the engine is higher. Thus the full compression is automatically reinstated in dependency of the engine speed. Of course instead of the automatic valve, another cam on the shaft could actuate the relief valve.

In the lower part of Figure 3, are shown two gear wheels 53 and 54 meshing together. They are fulcrumed at 55 and 56. Fixed to the wheel 54 is the hand lever 57 and the pin 58. The latter is connected to the pin 60 of the bell crank lever 28 by the rod 61. The wheel 53 carries a two-armed lever 59, 67 fixed thereon. The pin 62 of the upper arm 67 is connected to the pin 63 of bell crank lever 27 by rod 64, and the pin 65 of the lower arm 59 is connected to the pin 68 of the bell crank lever 26 by rod 66. There are indicated in dot-and-dash lines three positions of the levers 57, 59 and 67, the corresponding positions each marked by the same numerals 0, 50 and 100. These numerals signify approximately the travelling speeds of the locomotive. If the lever 57, beginning at zero while starting is lowered slowly in harmony with the actual speed, and the speed as indicated on the dial 69, both fuel pumps 14, 16 and the compressed air valve 8 are actuated in a similar manner as required by the diagrams of Fig. 1. The regulating mechanism is shown in the position for 50 km. per hour travelling speed. The push rod 44 is just out of engagement with the cam 21. The fuel pump 14 delivers very little to prohibit choking of the valve 6, and the cam 20 actuates the pump 16 for nearly the highest amount of delivery. Lowering the lever 57 will only shorten the duration of the delivery of fuel by the pump 16 on account of the increased speed of operation of cam 20 at higher speeds. Naturally the regulating mechanism may also be set to any other position than to a conformity of the actual speed and that marked on the dial 69, which only indicates a favorable regulation for starting and accelerating under normal conditions. E. g. while driving up a slope after having gained full speed before, the lever 59 will be set back to compressed air fuel combustion to enhance the tractive force of the engine.

Having described my invention, what I claim is:

1. In an oil injection engine, a cylinder, a piston reciprocating in said cylinder and cooperating therewith to form a combustion chamber, an ignition device in said chamber, a starting fuel valve in said chamber, a second fuel valve for injecting fuel into said chamber, a fuel pump for delivering fuel to said first valve, a fuel pump for delivering fuel to said second valve, a valve for compressed air, actuating mechanisms for said fuel pumps and said compressed air valve, regulating means for said actuating mechanisms, said last-mentioned means being adapted to control the duration of fuel supply to said first fuel valve, the duration of opening of said compressed air valve, and the quantity of fuel delivered to said second fuel valve.

2. In an oil injection engine, a cylinder, a piston reciprocating in said cylinder and cooperating therewith to form a combustion chamber, an ignition device in said chamber, a starting fuel valve emitting a spray of low force and great dispersion in said chamber, a second fuel valve emitting a spray of higher force and lesser dispersion for injecting fuel into said chamber, a fuel pump for delivering fuel to said first valve, a fuel pump for delivering fuel to said second valve, a valve for compressed air, actuating mechanisms for said fuel pumps and said compressed air valve, regulating means for said actuating mechanisms, said last-mentioned means being adapted to control the duration of the fuel supply to said first valve in accordance with the duration of opening of said compressed air valve and to gradually render operative said second fuel pump upon the gradual lessening of the duration of opening of said compressed air valve and the decrease in effect of said first fuel valve.

3. In an oil injection engine, a cylinder, a piston reciprocating in said cylinder and cooperating therewith to form a combustion chamber, an ignition device in said chamber, a starting fuel valve in said chamber, a second fuel valve for injecting fuel into said chamber, a fuel pump for delivering fuel to said first valve, a fuel pump for delivering fuel to said second valve, a valve for compressed air, actuating mechanisms for said fuel pumps and said compressed air valve, and a unitary regulating means for said actuating means for adjusting the period of feed of fuel from said first pump and compressed air from said valve, in reference to piston stroke of the engine, and for adjusting simultaneously in inverse ratio the quantity of fuel fed from said second pump per engine stroke.

4. An engine as described in claim 1, wherein said first mentioned fuel valve is adjusted for comparatively low injection pressure and said second fuel valve is adjusted for a higher injection pressure.

5. An engine as described in claim 3, wherein said regulating means is adapted to stop the feed of fuel from said second fuel pump at one limit position and to adjust said first fuel pump to feed a minimum charge of fuel at its other limit position.

HEINRICH TRIEBNIGG.