

April 23, 1946.

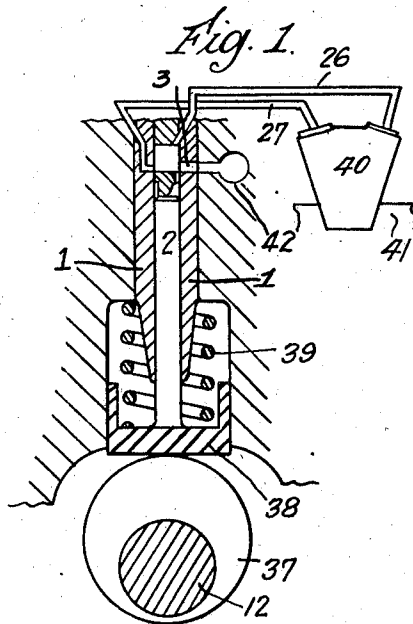
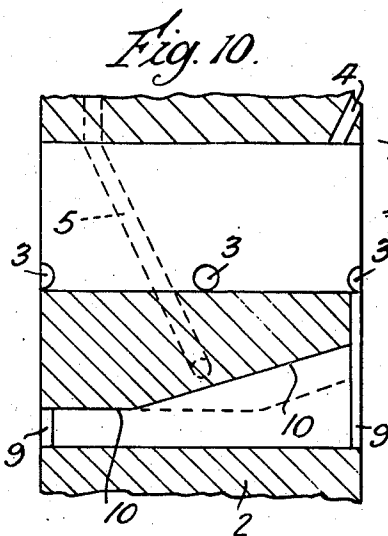
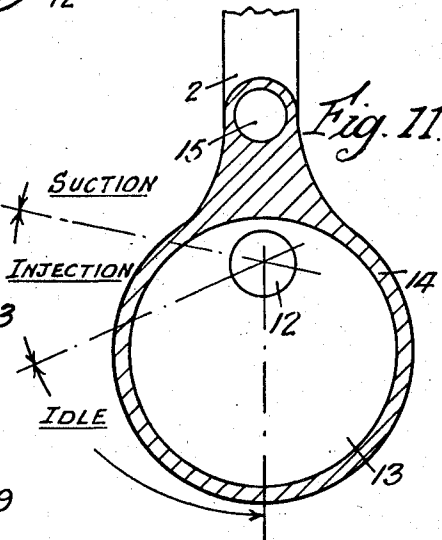
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2,398,834

FUEL INJECTION IN INTERNAL-COMBUSTION ENGINES

Filed July 23, 1941

3 Sheets-Sheet 1



Inventor:
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By his attorneys,
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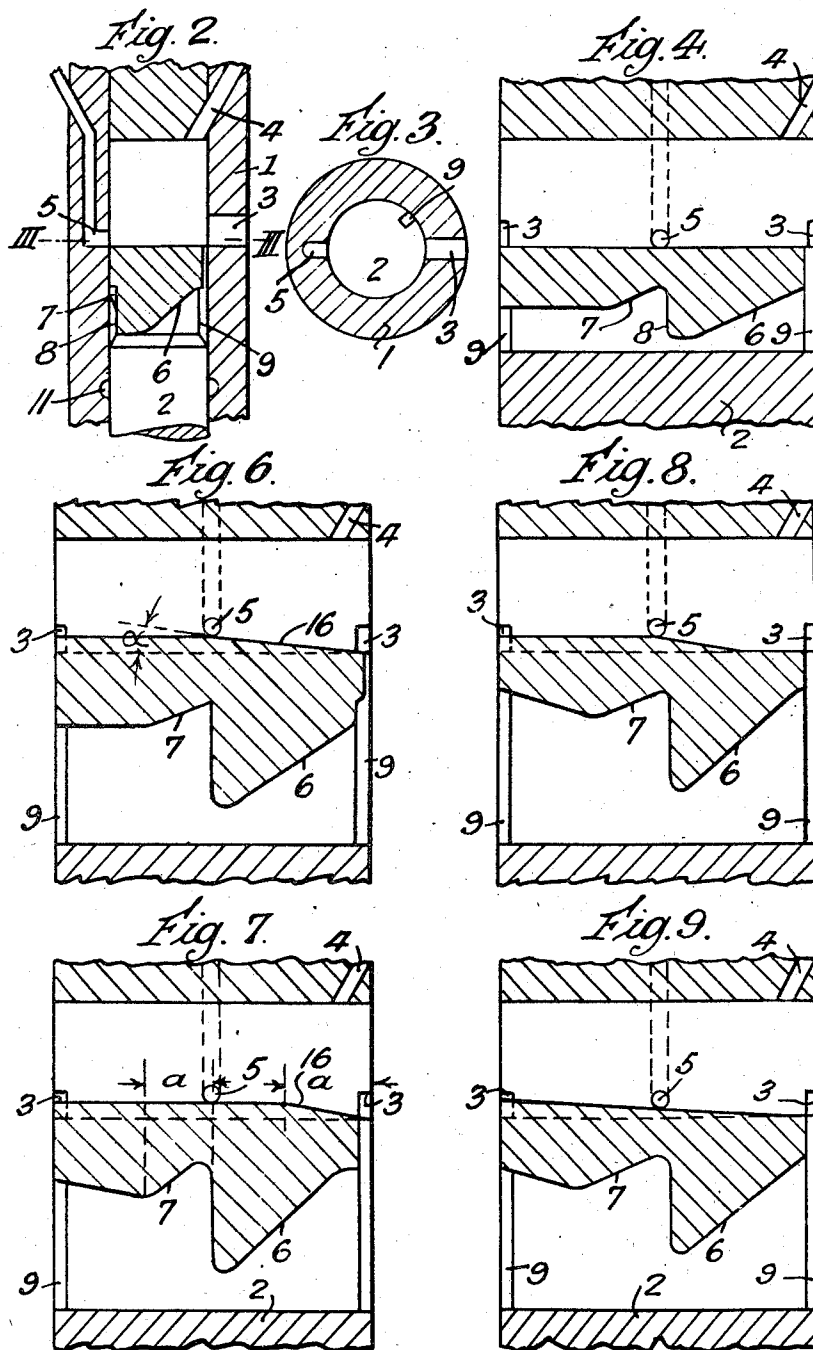
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3 Sheets-Sheet 2



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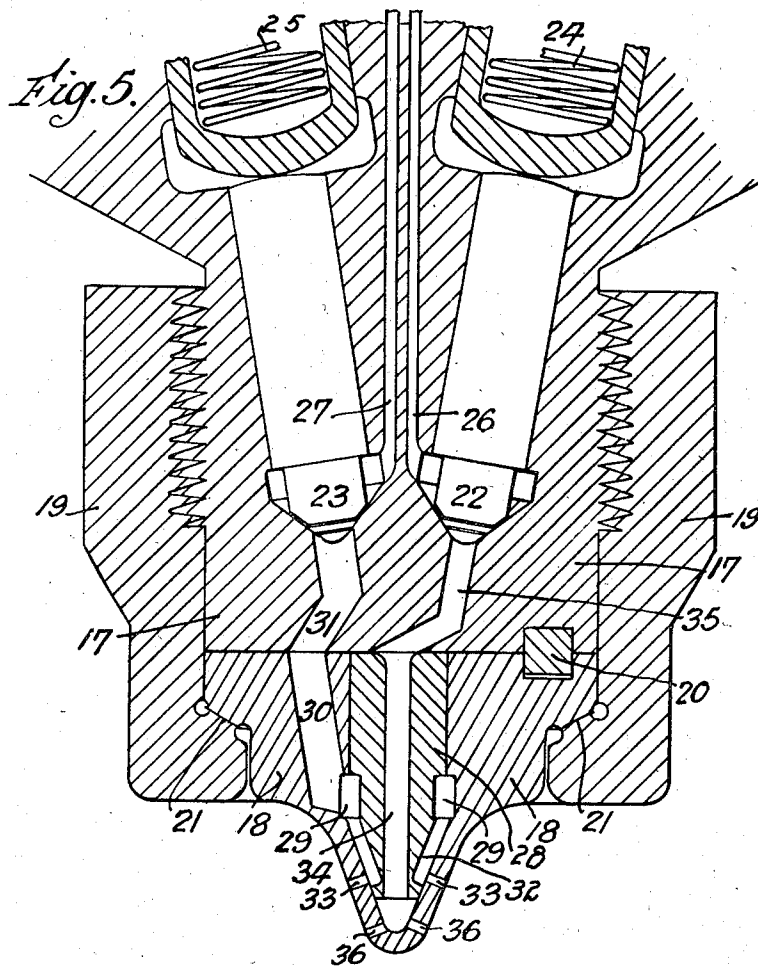
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FUEL INJECTION IN INTERNAL-COMBUSTION ENGINES

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3 Sheets-Sheet 3



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UNITED STATES PATENT OFFICE

2,398,834

FUEL INJECTION IN INTERNAL-COMBUSTION ENGINES

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Application July 23, 1941, Serial No. 403,722
In Great Britain July 25, 1940

1 Claim. (Cl. 123—139)

The invention relates to the injection of fuel in internal combustion engines and represents a further development of the invention disclosed and claimed in my Patent No. 2,265,692.

It is an object of the present invention to provide a cheaper system than the constant pressure system of the said patent, more particularly for smaller engines, by using a jerk-type pump.

According to the invention a fuel pump for a system as described and claimed in my said prior patent is provided with outlets from the pressure space to different needle valves at the nozzle, at least one outlet being controlled as to opening and closing by the pump plunger and at least one other outlet being in continuous communication with the pressure space irrespective of the position of the pump plunger.

In many cases it will prove advantageous to make use of the compressibility of the oil for providing the high pressure during the first part of the injection.

Thus, if the cam actuating the pump plunger is not fully adapted, for any reason, to the required delivery for the slow rate, a long lead or pipe from the pumping space to the nozzle may prove useful, or any other means for increasing the volume. This applies especially in cases in which the nozzle orifices of the slow rate are of such section that the pressure caused by the pump delivery exceeds the opening pressure necessary for the respective needle valve, and in which owing to resistance a further pressure is then built up in front of the nozzle orifices, compressing the fuel and limiting the delivery.

It is obvious that by this method of operation a much higher rate of main fuel feed can be allowed than in methods using only one single rate of feed, for with the latter, if economical operation is to be achieved, injection must start at its sole rate before the inner dead centre of the piston, and if this sole rate is high, the usual detonation and smoky exhaust will be the result, with poor overall performance. Thus, with a single rate of feed, the rate is determined not by the amount of fuel which the engine will "take" after the inner dead centre is passed or when combustion is in full swing, but by the maximum rate to prevent detonation and like defects.

Heretofore it has not been possible to vary the rate of fuel feed into the cylinder by means of a single jerk-type plunger per cylinder, even if more than one nozzle were fitted. The single rate of feed could be slightly affected by the shape of cams used to operate the plungers, but, as the pressure at the nozzle depends on very many

factors, a satisfactory control over a wide speed range, such as is required for transport engines, was not attainable by these means, and the less so when the delivery pressure was controlled at the same time.

It is a further object of the invention to provide means for feeding fuel at a variable rate into the combustion space or cylinder at all required speeds of the engine.

Still a further object of the invention is to provide a pump to effect injection in this manner up to very high speeds.

It is among the objects of the invention to reduce ignition lag and so to time the different rates of fuel feed, that high velocity of flame propagation may be maintained over a longer period than heretofore usual in medium and small size engines.

The invention makes it possible to create a local combustion in one part or in a recess of the combustion space by means of the high pressure-slow rate feed and, while this combustion is still going on, to inject the fuel at high rate and lower pressure into the strata where this local combustion is proceeding. By these means it will sometimes be possible to dispense with controlled air movement in the engine cylinder, as the resultant progressive flame front in a suitably shaped combustion space will in this case provide a more advantageous effect than that striven after by controlled motion of the combustion air.

As with the increase of compression ratio the economy of the cycle is usually improved, and as engine cylinder pressures about and even above 1000 lb./sq. in. can well be dealt with in the present state of engineering knowledge, it will be desirable to increase the compression ratio so as to reach these figures. The advantages would be, firstly that the starting of the engine would be safe and quick, and secondly that the ignition lag would be greatly reduced, and thus the benefits of the variable rate injection method would be obtained to a greater extent.

It would be possible to inject fuel only a very few degrees before the inner dead centre and not to lose diagram area on the compression side and to start burning first on a low rate of feed at high flame propagation speed and then to inject the rest of the fuel at a very high rate after the inner dead centre has been passed. With the very high compression ratios mentioned, the combustion space becomes very small and the outward moving piston multiplies the volume of the cylinder available for combustion very rapidly. Thus the rapid rate of feed cannot cause an unde-

sired pressure rise, but it must be kept high enough to maintain the peak pressure, which as above stated need not or only slightly exceed the compression pressure. If this condition is to be fulfilled the resulting total injection time will not exceed about 10° of crank angle for the high rate at full load, that is, about one half of that usual in known engines.

Consequently the injection will be terminated earlier in the cycle and more fuel will be burnt with less excess air. The effects on the overall efficiency of the engine of these means are obvious.

The accompanying drawings illustrate the invention by means of different embodiments, and therein

Figure 1 is a diagrammatic showing of a jerk-type pump and an injector with their connections,

Figure 2 is a part sectional view of the jerk type pump of Figure 1,

Figure 3 is a cross-section of the same on the line III—III,

Figure 4 is a developed view of the piston groove and cylinder ports,

Figure 5 is a sectional view of the nozzle of Figure 1,

Figures 6 to 10 are views corresponding to Figure 4 showing modifications, and

Figure 11 is a diagrammatic view of an alternative pump drive.

Referring to Figures 1, 2 and 3, the pump comprises a cylinder 1 and a plunger 2. The plunger is operated by a cam 37 mounted on a shaft 12 positively driven from the engine crank shaft and at the same speed. A cup 38 at the lower end of the plunger 2 is pressed against the cam 37 by a spring 39 so as to follow the contour of the cam at all times.

In Figure 2 the plunger is shown in its outermost position. Fuel enters from the supply passage 42 by an inlet port 3 extending from the edge of the plunger at the position shown towards the pump cylinder head. An outlet port 4 is located in the cylinder head, and a second outlet port 5 is located opposite the inlet port 3 but extending a shorter distance in the direction of the cylinder head.

The plunger has a groove below its head of a particular shape which is shown developed in Figure 4. The lower edge of the groove is circular, and the upper edge has firstly a portion 6 mainly helical with a short circular part to cooperate with the inlet port 3. The upper edge has a second portion 7, of which a small part is helical and a larger part circular to cooperate with the delivery port 5. An axial portion 8 joins the two portions adjacent to the port 5 at the position of minimum feed. Diametrically opposite the portion 8 a notch 9 provides communication between the pressure space and the groove. A seepage groove 11 is provided in the pump cylinder below the lowest position occupied by the groove.

In the position shown in Figures 2 and 4 no feed will take place, since the fuel displaced by the plunger in its rising movement will return to the port 3 through the notch 9. If the plunger is turned on its axis to the right in Figures 2 and 4, the notch 9 as shown in Figure 3 will no longer register with the port 3, and, when the plunger has risen far enough to cover port 3, delivery will commence. Since port 5 has also been covered, delivery will only take place through port 4. After a certain time the edge 7 will uncover the port 5, and then delivery will take place through port 5.

The nozzle 40 is fixed in the engine cylinder

head, a part of which is shown at 41 to deliver fuel to the combustion chamber within the cylinder. The construction of the nozzle is shown in Figure 5 as comprising an injector body 17 and a nozzle body 18 clamped thereto by a cap nut 19. The bodies 17 and 18 are maintained in a precisely defined location by a dowel 20, and a conical seating 21 between the nozzle body and cap nut substantially prevents fuel leakage.

The injector body 17 contains two needle valves 22 and 23 normally held closed by springs 24 and 25, which act on piston shaped members formed at the heads of the needles. Fuel is fed from the ports 4 and 5 of the injection pump by leads 26 and 27 respectively to the spaces around the needle points. The spring 24 is preferably considerably stronger than the spring 25, so that when both ports are under pressure the needle valve 22 closes, and fuel is fed only through port 5, lead 27 and needle valve 23. The feed pressure is then lower than when fuel is being fed through needle valve 22 because of the smaller spring pressure.

The nozzle body 18 has a central hollow of partly cylindrical form filled by a plug 28. The plug has a conical end to fit a conical projection of the body 18. An annular chamber 29 is formed by recesses in the plug 28 and body 18 and is in communication by ducts 30 and 31 with the needle valve 23. A groove 32 on the conical part of the plug completes the communication with a row of nozzle orifices 33.

The plug 28 has a central duct 34 communicating by a duct 35 with the needle valve 22. At its lower end the duct 34 is in communication by the space in the tip of the nozzle body with a second row of orifices 36. The conical end of the plug 28 beds into the interior of the conical projection of the nozzle body 18 to form a seal between the sets of orifices 33 and 36 and their respective supply ducts. Packing may be inserted for the same purpose between the contact faces of the bodies 17 and 18.

In an alternative construction (not shown) two entirely separate nozzles are used and each is fed through a spring closed needle valve similar to the valves 22 and 23 just described. Here again it is preferred in most cases to make the closing springs 24 and 25 different in strength for the purpose indicated above.

The nozzle orifices 36 fed from port 4 are adapted to give a small feed at high pressure and the orifices 33 fed from port 5 to give a larger feed at a lower pressure. It may in some cases be desirable to provide both needle valves with springs of equal strength. Then the feed through port 4 will continue after port 5 comes into action.

Figures 6 to 10 show different shapes of pump plunger as alternatives to Figure 4.

In Figure 6 the plunger groove is of somewhat similar shape to Figure 4, but the plunger head is so shaped that above the portion 6 of the groove edge, the plunger edge is helical in the opposite direction at 16. In development this portion of the plunger edge makes an angle α with the horizontal, which latter would represent the edge of a right cylindrical plunger. As a result pre-injection at high pressure is increased in time as the total fuel injection rises due to the rotation of the plunger.

In Figure 7 the helical portion 16 of the plunger edge does not extend over the half circumference but only over a developed length a . The helical part of the portion 7 is of the same length a . In this case with increasing fuel feed the time of

pre-injection rises and the high rate of injection at low pressure starts earlier, though the actual commencement of injection is constant in time once the high rate of injection has come into operation.

Figure 8 shows a form of groove in which the helical portion 16 of the plunger edge is in a different position from that of Figure 7. Commencement of high pressure injection is constant in time, and low pressure high rate injection starts earlier and finishes later with the total injection increasing up to a certain point, after which only the cessation of injection becomes later.

With the arrangement of Figure 9 the top edge of the plunger is in the form of a single helix. There is a gradual steady increase in the absolute time of pre-injection, and the commencement of the high rate becomes earlier with increasing total injection.

Figure 10 shows a different arrangement of plunger groove and cylinder ports. Two inlet ports 3 are placed diametrically opposite each other. The high pressure low rate port 4 is in the cylinder head as before, but the low pressure high rate port 5 is lower in position than the inlet ports 3. The plunger has a groove of which the edge 16 comprises only one helical part and one circular part.

In the position shown injection is stopped due to the register of notch 9 with port 3. By turning the plunger a little to the right both high and low pressure feeds are provided over a very short time. Both times are increased by further rotation in the same direction, until at the position shown dotted the high pressure feed reaches its maximum time. Further rotation will increase the low pressure feed time, while the high pressure feed time remains constant.

The seepage groove 11 shown in Figure 2 will usually be necessary with the arrangement of Figure 10, where there is a pressure port at a lower level than the inlet ports 3.

An alternative drive for the plunger is shown in Figure 11. The shaft 12, positively driven from the engine crank shaft as before, carries an eccentric sheave 13. An eccentric 14 driven by the sheave is linked at 15 to the plunger 2. The angle diagram added to the figure shows the approximate timing of injection for one particular setting of the plunger in its rotary adjustment about its axis. Only the middle part of the plunger stroke is utilised for the actual fuel feed.

What I claim is:

A liquid fuel injection system for an internal combustion engine comprising means providing two sets of discharge orifices to discharge the fuel into the engine cylinder, one set having a considerably smaller cross-sectional area of discharge than the other, a jerk type pump with cylinder and plunger, means for actuating the plunger at a steady rate of lift, a first outlet in the pump cylinder wall at a part covered and uncovered by the plunger, a passage connecting the said outlet to the set of discharge orifices having the larger cross-sectional area, a second outlet in the pressure space of the pump cylinder out of reach of the plunger, and means including a passage entirely separate from the first-named passage for storing fuel and for connecting the said second outlet to the set of discharge orifices having the smaller cross-sectional area, the flow area provided by said set of discharge orifices of smaller cross-sectional area being insufficient to permit discharge of all of the fuel delivered by the pump when said plunger is performing a working stroke and said first passage is covered by said plunger whereby the pump will then deliver excess fuel, said storing means being adapted to store said excess fuel under pressure and then to discharge the said excess fuel through the said second outlet when that outlet is uncovered by the plunger.

GEORGE STEPHEN KAMMER.