July 22, 1941. M. G. FIEDLER 2,250,364

ENGINE AND METHOD OF FUELING THE SAME

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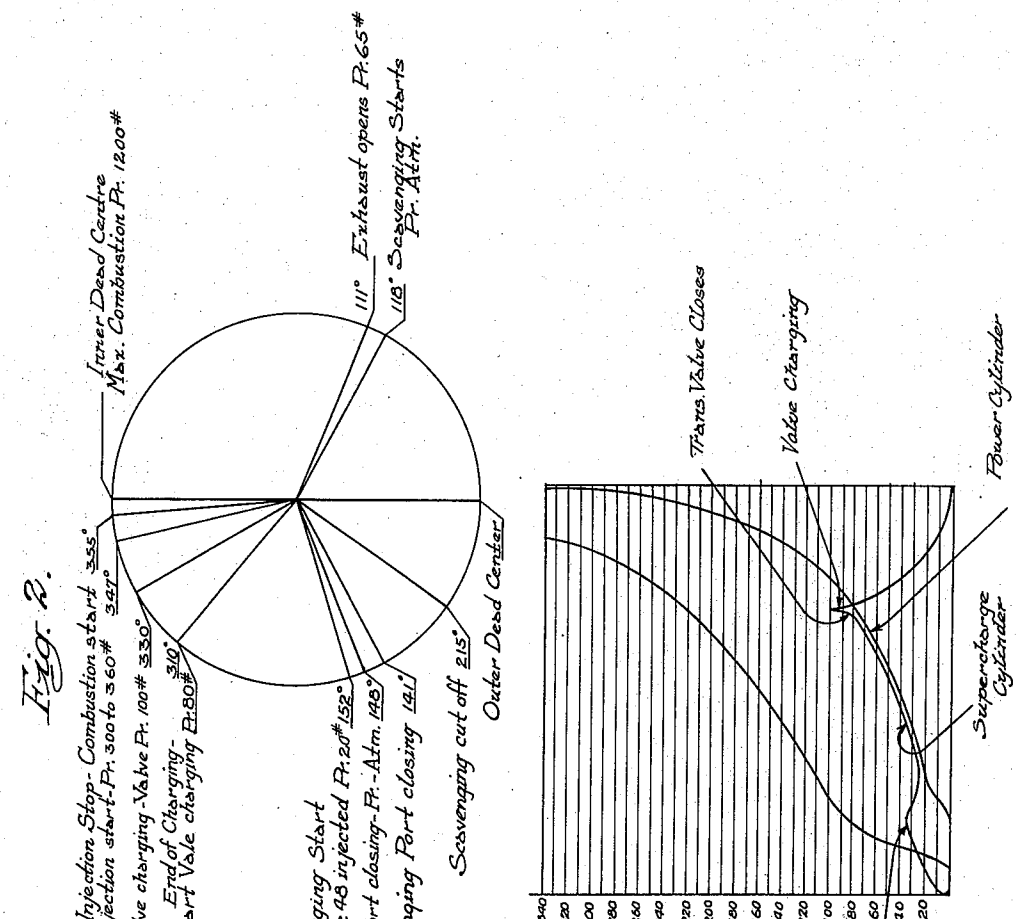


Fig. 2.

Fig. 3.

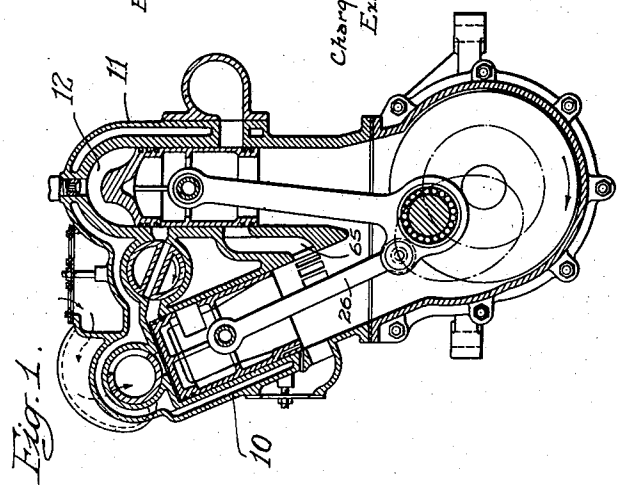


Fig. 1.

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Fig. 4.

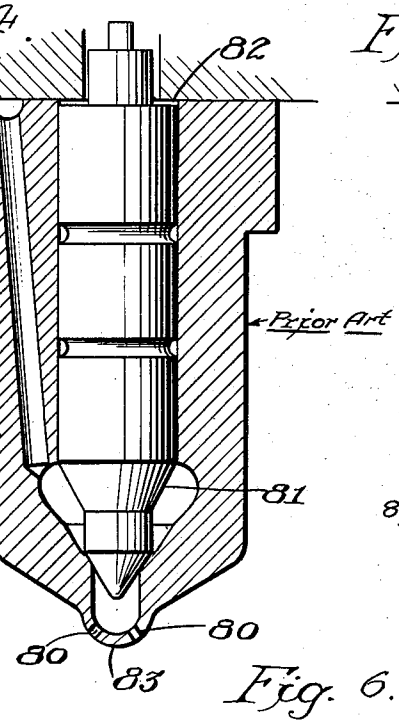


Fig. 5.

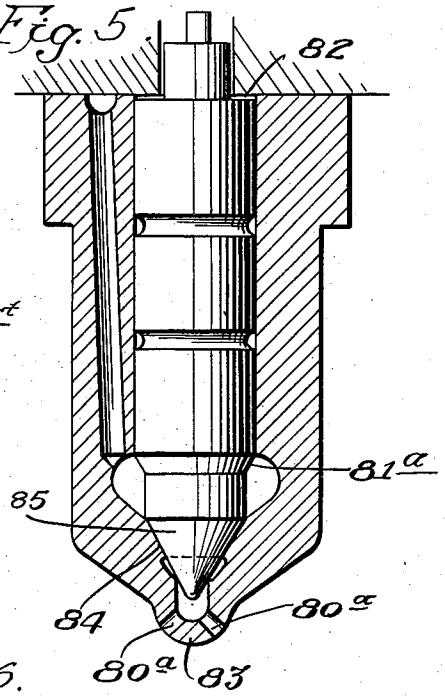


Fig. 6.

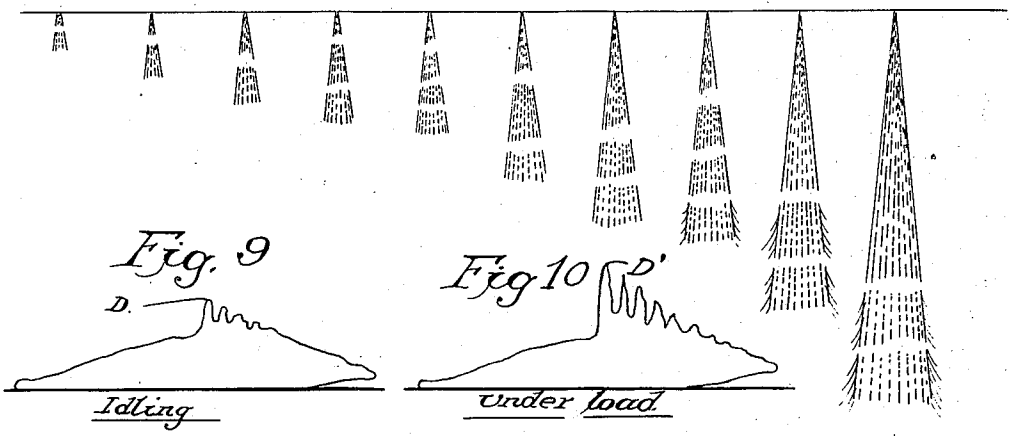


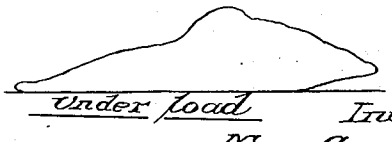
Fig. 9

Fig 10

Fig. 11.



Fig. 12.



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Fig. 7.

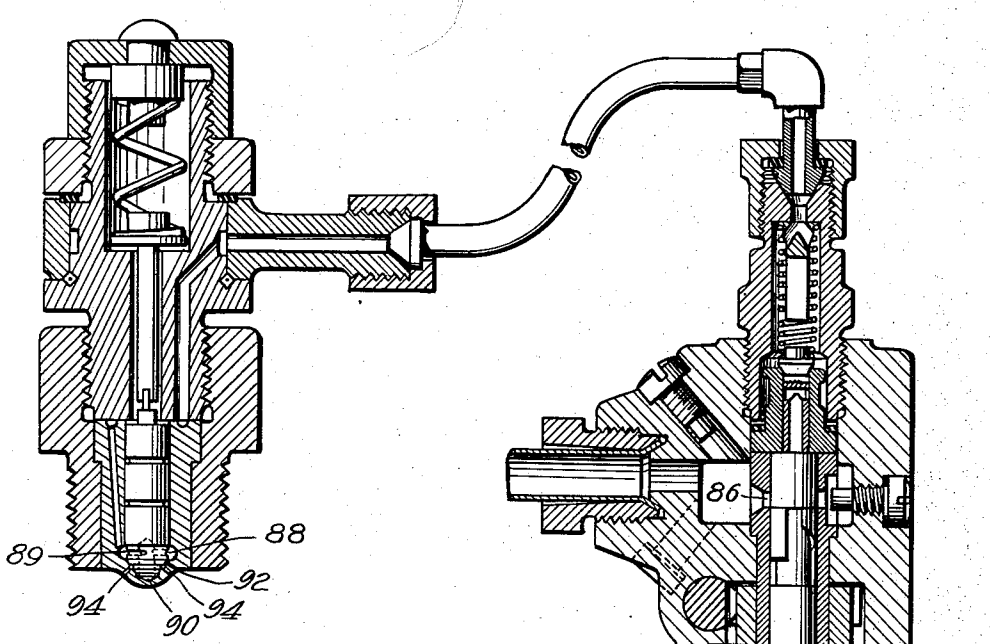
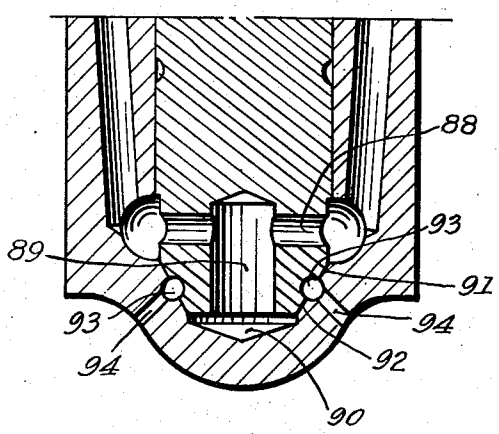


Fig. 8.



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UNITED STATES PATENT OFFICE

2,250,364

ENGINE AND METHOD OF FUELING THE SAME

Max George Fiedler, Chester, Pa., assignor to Fiedler-Sellers Corporation, Philadelphia, Pa., a corporation of Pennsylvania

Continuation of application Serial No. 34,134, July 31, 1935. This application January 31, 1938, Serial No. 188,017

3 Claims. (Cl. 123—32)

This invention relates to a method of operating compression ignition engines of the oil burning type, or that type usually referred to as Diesel engines, and is a continuation of my prior application Serial No. 34,134, filed July 31, 1935. An important object of the invention is to provide a method of operation of such engines whereby they may be made to operate substantially in accordance with what is known as the Otto cycle, and to operate at variable speeds comparable to those of the ordinary gasoline engine employed in automobiles without detonation throughout such speed ranges.

It is well known that the combustion of hydrocarbons may be either a direct oxidation or a decomposition followed by oxidation of the destruction products. In practice, there is a race between the two processes, the conditions being more favorable to hydroxylation when a fuel has been properly divided and mixed with air before it is burned, at which time the flame is blue and has no tendency to soot. The conditions are more favorable to destructive combustion when the fuel is exposed very suddenly and in a highly vaporized condition to flame temperatures, the fuel particles decomposing rapidly before they can find oxygen, and under these conditions there is a yellow radiation caused by the burning carbon and a tendency to form soot.

In the ordinary Diesel engine, air is compressed to the greatest possible extent to prevent ignition lag, and a readily ignitable oil is employed for the same reason. I have found that in high speed Diesel engines having a high compression ratio the first fuel entering the combustion space does not ignite but meets with the turbulent compressed air. Part of the entering fuel will mix with more or less turbulent air until, in some part of the chamber a suitable mixture for self-ignition has been established and combustion will start. If the duration of injection is continued after combustion has occurred, this fuel will meet air which is not only strongly turbulent but mixed with combustion residue. As injection continues, the air becomes saturated with the products of combustion until the interference with further combustion is so serious that free carbon will be generated. In the ordinary Diesel this occurs at a point where approximately 50% of the available air has been consumed. The time necessary to establish the proper mixture for the first auto-ignition is equal to the ignition lag and is influenced to a great extent by turbulence during injection, but also by the shape of the combustion chamber, the spray characteristics, duration of injection, and, to some extent, by the compression pressure. To a large extent the ignition lag in the engine is directly dependent upon the degree of turbulence, increasing with such turbulence due to the fact that the air usually rotates at high speed around the cylinder axis throwing the entering fuel into the coldest zone adjacent the cylinder wall. This can only be counteracted through high compression and it follows that previous engines having high turbulence must work under very high compression pressures, often as high as 40 to 50 atmospheres, and even under these circumstances detonation frequently results.

Analyzing the standard spray characteristics as produced today in the usual solid fuel injection engines, it is found that the fuel, through being subjected to high injection pressures (3,000 to 20,000 lbs.) and through being forced through very small orifices is substantially in a vapor stage and thus readily subject to "cracking." In addition to that, as has been photographically demonstrated, the spray itself is very compact and cannot be broken up even by the most violent turbulence. This spray is, therefore, exposed to high compression temperatures without the ability to mix with air and the result is, inevitably, "cracking." The combustion will, therefore, follow the second type mentioned above and since part of the combustion will be a hydrogen-oxygen or oxygen-methane reaction at the high temperatures and pressures existing in Diesel engines, will undoubtedly, be extremely violent and destructive. Furthermore, since the injection is through such fine orifices, the injection period necessarily, and of course purposely under the Diesel system, continues after combustion has actually begun and, obviously, the later injected fuel will crack producing further hydrogen-oxygen reactions.

I have discovered that the proper operation may be provided by observing the following precepts:

1. In small bore, short stroke engines the use of a relatively large combustion space and the maintenance of the air in this space in a high state of turbulence in order that the mixture may be as complete and rapid as possible while maintaining the turbulence of such character that there is no localizing disturbance or deformation of the jet.

2. The instantaneous, or substantially instantaneous injection of the fuel into the combustion chamber, and the use of low compression pres-

sures (approximately 240 pounds at the time of starting injection) thereby causing a combustion lag, in order that all of the fuel may be delivered thereto before combustion begins without regard to the speed or load conditions under which the engine is operating.

3. The introduction of the fuel spray into the combustion chamber in the form of a loosely bonded liquid foam and under relatively low injection pressures (preferably less than 1200 lbs.) thereby enabling the relatively cool fuel to readily combine with the air in the combustion chamber and establish an auto-ignition mixture as rapidly as possible, and, furthermore, promoting the combustion lag during which establishment of the mixture is assured.

An important object of the invention is the provision of a means for super-charging the working cylinder so constructed and arranged that losses from interposition of receivers or recompression of any appreciable volume of air handled by the compressor unit is eliminated.

A further object of the invention is the provision of a solid fuel injection cycle in which the injection period is in all phases of operation of the engine of such duration that all the fuel has entered the combustion chamber before combustion actually starts.

Obviously, sudden injection of fuel in this manner is impractical in a quiescent chamber and, accordingly, a further object of the invention is to provide a structure wherein high turbulence is assured, the turbulence being non-directional in character, and which provides this turbulence without the use of any artificial or additive means for producing the same.

Another object of the invention is the injection of fuel in the engine in the manner outlined in the second and third precepts above.

Other objects of my invention are to provide an engine having two cylinders and pistons operatively mounted therein, one of which cylinders is the main working cylinder and the other cylinder is the supercharging cylinder and utilizing both of the cylinders as compression chambers; to provide a passageway between the working ends of the cylinders controlled by a valve through which the air is forced and retained in the main cylinder during the firing of the fuel mixture; to provide a novel form of valve for transferring the air from one cylinder to the other, in which valve a small volume of air under pressure is retained when the valve is closed and again released during the next succeeding operation of the pistons, thus providing air under pressure which will assist in building up the pressure in the cylinders when the next compression stroke takes place; to provide a novel form of rotary inlet valve for controlling the admission of air to the compression cylinder; to provide a novel means for scavenging the main cylinder at the end of the working stroke of the piston; to provide an engine in which the scavenging air is compressed by both of the pistons in the crank casing and controlled by the movements of the piston for transferring the air for scavenging purposes in a novel manner; to provide a novel construction and manner of operating the inlet and transfer valves; and to construct the engine so that any number of cylinders may be combined for operating upon the same crank shaft and to arrange the casing, of which the cylinders form a part, with novel inlet and transfer passages for controlling the inlet air, combustion mixture, and the scavenging air.

These together with various novel features of construction and arrangement of the parts, which will be more fully hereinafter described and claimed, constitute my invention.

Referring to the accompanying drawings,

Fig. 1 is a sectional view through an engine of a type suitable for use in carrying out my novel method;

Fig. 2 is a cycle diagram of an engine constructed in accordance with my invention;

Fig. 3 is an indicator card showing the pressure cycle of the engine;

Fig. 4 is a conventional injection nozzle of the type at present in use;

Fig. 5 is a sectional view through a nozzle modified for use in accordance with my invention;

Fig. 6 is a diagrammatic view illustrating the type of spray produced by the nozzle of Fig. 5;

Fig. 7 is a sectional view showing a preferred type of nozzle and the pump connection thereto;

Fig. 8 is a highly enlarged sectional view through a portion of the nozzle of Fig. 7;

Figs. 9 and 10 are card diagrams showing operation of the engine with the ordinary type of injection; and

Figs. 11 and 12 are card diagrams taken under identical conditions with those in Figs. 9 and 10, but with use of the new type of injection and operation.

The first precept of my invention may be conveniently carried out by the engine forming the subject matter of my copending application Serial No. 247,508, filed December 23, 1938, for "Internal combustion engine." The engine therein illustrated is of the type shown in Fig. 1, having main and supercharging cylinders **10** and **11**, the main cylinder being so constructed that a relatively large combustion space **12** is provided. The supercharging cylinder **11** delivers a sufficient amount of air beyond that obtained from compression in base **13** and introduced for scavenging the main cylinder to provide auto-ignition pressures and temperatures in the combustion chamber considerably prior to complete compression of the charge. The scavenging air is delivered from the base to cylinder **12** in a manner such that a high degree of non-directional turbulence is obtained in the cylinder. At the same time the maximum pressure attained in the cylinder should, preferably, not exceed 400 to 450 pounds, and under many conditions considerably lower pressures are feasible and desirable.

The second precept may, obviously, be obtained in a variety of fashions as, for example, by multiplying the number of nozzles employed and thereby increasing the effective injection area so that the period necessary to injection of a predetermined amount of fuel may be reduced to the desired point. I have found that the injection period should not exceed 10° of crank travel, and preferably should be confined to approximately 7° thereof. Since the engine is now to operate as a constant volume engine, all of the fuel being injected before ignition, it is, of course, desirable that this fuel be entered prior to the arrival of the working piston **18** at top dead center, the arrangement being preferably that illustrated in the cycle diagram forming Fig. 2, in which the injection is illustrated as occurring in advance of the top dead center, a distance corresponding to the average combustion lag. While the injection period might be delayed beyond the point illustrated, this will, obviously, result in a loss of efficiency, the operation of the engine under such circumstances

being substantially that of gasoline engines operating with a retarded spark.

The compression pressure of the engine must be kept sufficiently low to insure a combustion lag enabling complete injection of the charge prior to initial combustion, and must be sufficiently rapid to insure against vapor formation. It will be noted that this is directly contrary to Diesel practice in which the production of highly vaporized sprays is sought and in which the compression pressures are carried to the highest possible point in order to avoid ignition lag.

Such an arrangement as that already suggested will result in a highly improved operation of the engine, but to insure complete elimination of detonation, a smooth operation of the engine over a wide speed range, and an economical fuel injection system, the construction should be restricted to a single injection nozzle of peculiar characteristics. As is well known to those familiar with Diesel construction as hitherto practiced, the usual nozzle comprises a valve lifted by the injection pressure and thus permitting the escape of the fuel to the cylinder through orifices **80** (see Fig. 4). Due to the relatively large area of the lifting surface **81** provided on such valves, with the attainment of injection pressure, the valve is violently thrown upwardly against a stop **82** and there remains throughout the injection period, the minute discharge orifices providing sufficient back pressure to maintain it in this position.

I have found that by materially reducing the lifting area of this valve, as at **81***a* in Fig. 5, and increasing the diameter of the discharge openings **80***a* to an extent such that the pressure beneath the valve is constantly relieved and at the same time providing the valve with a relatively heavy spring of high frequency, the valve will chatter against the seat, or rather upon the liquid passing over this seat, reducing the fuel to a loose foam of liquid particles which is discharged through the openings **80***a* at a much reduced pressure and in relatively large particles. The increase of the size of the openings **80***a* requires a thickening of the wall **83** through which these openings are formed in order that they may properly guide the fuel and insure correct distribution thereof. The fuel then has a spray characteristic typified in Fig. 6, and is in the form of well spaced, loose liquid particles as compared with the usual injection practice, which particles may be readily taken up and surrounded by the air in the combustion chamber, with the result that a thorough comingling is obtained insuring proper proportioning of air to fuel at the time the mixture attains the flash point. As indicated above, the openings should be much larger than those ordinarily employed, and in an injector found to give excellent results 6 openings were employed of a diameter of .9 millimeter as compared to 6 openings having .3 millimeter utilized in the usual injection nozzle; in other words, an increase of approximately 900 per cent in injection area.

In general, it may be stated that the area of the openings must exceed the maximum area afforded by the valve seat **84** and the valve **85** during the injection period. It will be obvious that the valve, during the injection period, acts not only as a valve but likewise as a flexible orifice, through which the fuel may enter, exercising upon the fuel because of its flexibility an agitating action producing a relatively loose foam. The valve seat itself should be of relatively large diameter in order to obtain large impact relief areas with a minimum lift and the valve spring should be strong enough so that at minimum pressure it will counteract the inertia effect of the valve stem under impact and continually attempt to reseat the valve and thus produce the chattering action on the fuel passing over the seat. To this end, as previously stated, a heavy valve spring having a high frequency is preferable.

While the injection nozzle of Fig. 5 provides a highly improved operation, it was found in practice that this nozzle, after heating to approximately 700° caused detonation. It was finally determined that the foam formed in advance of the nozzle tip by the chattering of the valve upon its seat tended to vaporize at the nozzle tip with resulting detonation in operation. For this reason, the valve of Fig. 7 was developed. In this valve the lifting area is transferred to the center of the valve, ports **88** communicating with a passage **89** opening through the bottom of the valve and into a chamber **90** at the nozzle tip. The valve seat is divided into two sections **91** and **92** by a groove **93** aligned with the discharge ports **94**. By this construction all foaming fuel is discharged from the valve and the tip of the nozzle is kept cool by maintenance of a solid body of the fuel thereagainst.

Tests with an injection nozzle of this type have proven conclusively not only its value in improving the explosion characteristics of the engine but likewise that for maximum efficiency certain definite characteristics and proportions should be employed in the injection system exterior to the nozzle. I have found, for example, that the injection at the nozzle is responsive not so much to the injection to the system by a measuring pump of a predetermined amount of fluid as to the impact resulting on the fluid line from the initial opening of the fuel line to receive the fuel displaced by the pump. This is evidenced by the fact that the injection period does not appreciably vary through a considerable range of injection as regards the amount of fuel which passes through the nozzle. With the same nozzle it is possible to inject 10 to 100 cubic millimeters of fuel to the cylinder in the same interval. This is apparently due to the fact that with a greater fuel injection a greater residual pressure exists in the line connecting the pump and nozzle and, accordingly, the impact blow resulting on initial injection is transmitted with greater force to the valve. The injection apparently immediately follows the closing of the intake openings **86** of the pump by the piston **87** thereof. The speed of operation of the pump piston apparently has little effect on the injection although it is found that a greater fuel injection can be obtained through use of a relatively slow operation of the piston. In actual use, sharp, medium and eccentric cams have been utilized, and it is found that the eccentric cam gives much the best results, the sharp cam tending to upset equilibrium of the system. The indications are, accordingly, to the use of a large diameter pump operated by an eccentric cam through a short plunger travel.

I have, further, found that there are definite characteristics necessary to the discharge line connecting the pump and nozzle if proper injection characteristics are to be obtained. For example, a discharge nozzle having openings of the size above mentioned employed in an engine of the character described must have a discharge line of definite length; in the construction under

test, .533 meter. It is found that a shorter line causes double injection; that is to say, there are two definitely separated injection periods for each operation of the pump, apparently caused by the fact that the shorter line does not provide the dampening effect necessary and the impact blow becomes too great for the system. On the other hand, a longer line causes double injection, for the apparent reason that there is too great a volume of liquid in the system causing a lag. It might appear that the presence of too much fluid in the system occasioned by the use of a long line might be overcome by a reduction of the diameter of the discharge line, but this is not the case, for it is found that unless the diameter of the line is kept quite large the impact blow creates such high speed in the column that it cannot be controlled by the valve or flexible orifice. As a matter of fact, I have found that a relatively large line as compared to the standard line of 0 to 2 millimeters in diameter is essential and in practice employ a line at least 3, and preferably 4 millimeters in diameter. In any case, the line must be large enough to prevent too great an impact and, in general, the larger the amount of fluid to be injected, the larger the line should be. It has also been found that the amount of fuel injected through a given period can be varied within given limits by variation of the size of the pump employed. For example, a pump having a piston 10 millimeters in diameter provides an injection range between 10 and 100 cubic millimeters, while with the same line and conditions a pump having a piston of 13 millimeters in diameter provides an injection range of between 10 and 160 cubic millimeters in the same injection period.

I have determined that the temperature range at which injection should take place in order to permit the proper ignition lag enabling all of the fuel to be injected before combustion starts is provided by a compression pressure having a minimum of 120 pounds per square inch and a maximum of 400 pounds per square inch, the most efficient range being between 330 pounds and 360 pounds to the square inch. In this range the temperatures are sufficiently high to cause rapid heating of the material and are at the same time below the decomposition temperature when the fuel is delivered to the cylinder in the form of a coarse spray.

It will be noted that there is again a sharp deviation from the ordinary Diesel practice and this deviation assists in providing the necessary lag enabling the establishment of the auto-ignition mixture with complete injection of the fuel prior to auto-ignition. Due to the fact that low pressures are employed in injecting the fuel, the fuel itself enters the cylinder at a much lower temperature than is possible when injection takes place under the usual Diesel system. Since the fuel injected is substantially incompressible, all work done in injecting the fuel in the cylinder must be converted to heat in the fuel itself. With the light injection pressure the fuel may enter without in any way approximating decomposition temperatures and may, accordingly, combine with the air to provide a complete auto-ignition mixture before such temperatures are reached. The resultant explosion is, therefore, a soft-burning as differentiated from the detonating characteristics of fuel injected under the usual circumstances.

The vast improvement in operation of an engine operating with fuel injection under the above-outlined circumstances as against the normal injection methods may be visualized by comparing the indicator cards forming Figs. 9 to 12. The indicator cards of Figs. 9 and 10 are cards taken from an engine utilizing the ordinary nozzle and developing a perfect highly atomized compact spray in accordance with the principles of present-day Diesel practice.

It will be noted that a succession of detonation peaks D appear even when the engine is idling, as in Fig. 9. These peaks are much exaggerated when the engine is operating under load, as indicated at D' in Fig. 10. On the contrary when the type of injection just described is employed in the same engine and under the same conditions, these peaks immediately disappear and when idling or under load the engine shows no tendency to the detonations responsible for these peaks. (See Figs. 11 and 12.)

Through the use of this method of fueling the engine and the nozzle construction hereinbefore described, I am able to produce an engine which is particularly adapted for use in automotive fields. In an engine from which the indicator cards forming Figs. 11 and 12 were taken when using the new fueling system, such engine having 4 cylinders and a bore and stroke of $3\frac{3}{32}$" and 4" respectively excluding the exhaust port area, when operating at 1800 revolutions produced 150 H. P., at 1,000 revolutions produced 80 H. P. It is, however, operable in the higher automotive ranges; that is to say, 3,000 to 4,000 revolutions.

It will be noted that an engine operated as above described operates neither on the Diesel nor the Otto cycle, differing from the former in that the fuel injection occurs through an extremely short period and entirely prior to combustion; that the injection employed is a spray of loose particles rather than a highly vaporized injection; that an ignition lag is deliberately sought for, to enable the injection to be made prior to combustion, and that volumetric capacity of the engine is greatly increased from the ordinary Diesel, with the result that much higher engine speed can be obtained. It differs from the Otto cycle both in the fact that the volume of air introduced is materially increased, and that the fuel is separately injected.

Since both the method illustrated and the construction described are capable of considerable modification without departing from the spirit of the invention, I do not wish to be understood as limiting myself thereto except as hereinafter claimed.

I claim:

1. The method of operating variable speed solid fuel auto-ignition type internal combustion engines which consists in compressing air in the engine cylinder to an extent affording auto-ignition temperatures and maintaining said air in a state of turbulence, and instantaneously injecting a complete charge of fuel into the cylinder in the form of a loosely bound mass of liquid particles readily mixable with the turbulent air to produce an auto-ignition mixture but at a temperature sufficiently low to preclude initiation of combustion prior to completion of said injection and prior to the establishment of all of said fuel in auto-ignition mixture.

2. The method of operating a variable speed solid fuel injection pressure ignition engine comprising substantially instantaneously injecting a loose foamy charge of fuel into the combustion chamber after the attainment of auto-ignition

temperature and delivering the fuel to the chamber in a wet state while maintaining the pressure in the chamber sufficiently high to cause combustion and sufficiently low to insure an ignition lag preventing combustion of the wet fuel prior to the complete injection of the charge.

3. The method of claim 2 wherein the fuel is introduced to the cylinder through relatively large openings and prior to injection is subjected to the action of a flexible orifice causing the fuel to foam and be divided into loose particles.

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