(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) Publication number: 0 036 261 B1

(12) EUROPEAN PATENT SPECIFICATION

(45) Date of publication of patent specification: 28.11.84

(51) Int. Cl.^{3}: **F 02 B 1/06**, F 01 L 1/00

(21) Application number: 81300874.5

(22) Date of filing: 03.03.81

(54) A method of operating a four cycle supercharged compression ignition engine and an engine operable according to the method.

(30) Priority: 03.03.80 US 126769
20.02.81 US 234622

(43) Date of publication of application:
23.09.81 Bulletin 81/38

(45) Publication of the grant of the patent:
28.11.84 Bulletin 84/48

(84) Designated Contracting States:
AT BE CH DE FR GB IT LI LU NL SE

(56) References cited:
FR-A-1 529 537
FR-A-2 158 942
US-A-3 144 009

(73) Proprietor: McIntire, Ray Garrett
896 Black Foot Trail
Jamestown Ohio (US)

(73) Proprietor: Colvette, Douglas
912-4 LaCosta
Sarasota Florida (US)

(72) Inventor: McIntire, Ray Garrett
896 Black Foot Trail
Jamestown Ohio (US)
Inventor: Colvette, Douglas
912-4 LaCosta
Sarasota Florida (US)

(74) Representative: Warren, Keith Stanley et al
BARON & WARREN 18 South End Kensington
London W8 5BU (GB)

EP 0 036 261 B1

Courier Press, Leamington Spa, England.

Description

This invention relates to compression ignition engines and in particular to the operation of medium to high-speed compression ignition engines in such manner as to reduce the amounts of oxides of nitrogen in the exhaust gases.

As result of increasingly stringent federal standards with respect to emmissions from automobile and light dury truck exhausts, alternative power plants for automobiles and light duty trucks are being investigated. One popular alternative power plant is the compression ignition engine, commonly known as the Diesel engine.

The Diesel engine has several advantages over conventional spark ignition engines. In particular, Diesel engines burn heavier fuel which is cheaper than gasoline, they have a higher thermal efficiency than spark ignition engines, and they have significantly lower emissions in some respects than comparable spark ignition engines. While carbon monoxide emissions are low because the Diesel engine operates with excess air, and hydrocarbons are normally a small constituent of Diesel exhaust, Diesel engines characteristically produce unacceptably high amounts of oxides of nitrogen (NO_x) and therefore are presently unable to meet government standards with respect to NO_x emissions for automobiles and light duty trucks.

The standard Diesel engine used in some automobiles and most trucks today is a four-stroke or four cycle engine. In the first or intake stroke, the intake valve opens and the piston decends to draw fresh air into the cylinder. In the second or compression stroke, the intake valve closes and the piston rises to compress the air which becomes heated. Near the end of the compression stroke, fuel is injected into the cylinder and burns.

In the third or expansion stroke, the burning mixture expands and forces the piston down. At this time both the intake and the exhaust valves are closed.

In the fourth or exhaust stroke, the exhaust valve opens and the burned gases are forced out of the cylinder by the rising piston.

Since the working fluid, namely air, is a compressible gas that enters and leaves the cylinder in more than an instantaneous period of time, the closing of the exhaust valve at the end of the exhaust stroke typically occurs subsequent to the opening of the intake valve at the beginning of the air intake stroke. In other words, the exhaust valve remains open until after the piston reaches top dead center, and the intake valve opens before the piston reaches top center. The reason for this "valve overlap" is to effect a more thorough scavenging of the exhaust gases from the cylinder, which brings about an increase in power out of proportion to the amount of air involved.

When the exhaust stroke begins and the exhaust valve opens, the motion of the exhaust gases is started by the cylinder pressure existing when the exhaust valve is opened and is promoted by the piston motion during the exhaust stroke. The scavenging of exhaust gases tends to continue during and after the top center period. Therefore, the intake valve is opened to allow fresh air to enter the cylinder to displace the last traces of exhaust gases in the cylinder, and a necessary result of this procedure is that a certain amount of fresh air is drawn through the cylinder and out past the exhaust valve where it mixes with the exhaust gases.

It is believed that the occurrence of this valve overlap, during which fresh air is drawn in through the intake valve and out through the exhaust valve, is a major cause of the formation of unacceptable amounts of NO_x in the exhaust gas of a Diesel engine.

Some prior art internal combustion engines have been modified such that valve overlap is eliminated, with the alleged result that the levels of certain emissions in the exhaust gases are reduced. For example, in Bohnlein French patent No. 2,158,942, an engine is disclosed in which the exhaust valve is completely closed before the inlet valve is opened, and it is alleged that the levels of carbon monoxide (CO) in the exhaust gases are reduced as a result. However, since the engine is not supercharged, it would appear that this mode of operation would result in significant reductions in horsepower output and performance if employed in a modern, high-speed engine.

Another example is shown in Soubis French Patent No. 1,529,537. That patent discloses an internal combustion engine in which the valves and valve seats are designed to eliminate valve overlap conditions during engine operation.

In both cases a prevention of a back flow of burnt gases into the inlet system is envisaged.

Conversely, it is an object of this invention to provide an improved medium to high-speed four stroke supercharged compression ignition engine, and method of operating the same, in which the amount of NO_x present in the exhaust gases is at an acceptable level without an appreciable decrease in horsepower generated or fuel efficiency, and fresh air is prevented from flowing through the exhaust port.

The present invention provides an improved method and apparatus for operating a medium to high-speed, four cycle, compression ignition engine in which the valve timing is adjusted so that, for the entire range of engine speeds, the exhaust valve is completely closed before the intake valve opens so that no fresh air is permitted to pass out the exhaust valve. Furthermore, fresh air is forced into the combustion chamber so that the air pressure within the chamber is above atmospheric, so that there is sufficient air to burn substantially all of the fuel. Some exhaust gases may remain in the cylinder at the beginning of the next cycle. In this fashion, the conditions which create unacceptably high amounts of NO_x in the exhaust gases are reduced without effecting a significant reduction in the effective horsepower, mileage or performance, as with prior art engines.

According to one aspect of the present invention, there is provided a method of operating a medium to high-speed four-cycle supercharged compression ignition engine of the type wherein an intake valve is moved to open an inlet port to allow fresh air to enter a combustion chamber, said air is compressed, fuel is injected into said combustion chamber and burns thereby expanding said air, and an exhaust valve is moved to open an exhaust port to allow burned gases to be scavenged through said exhaust port, characterized by:

timing said opening of said inlet port and said closing of said exhaust port such that said exhaust port is completely closed before said inlet port is opened at all engine speeds.

According to another aspect of the present invention, there is provided a supercharged, medium to high-speed, four-cycle automotive compression ignition engine of the type having at least one cylinder defining a combustion chamber and having a reciprocating piston therein linked to rotate a crankshaft; an intake manifold having intake valve means communicating with said combustion chamber, exhaust valve means communicating with said combustion chamber, and camshaft and linkage means for intermittently opening and closing said intake valve means and said exhaust valve means such that fresh air may be drawn into said combustion chamber from said intake manifold through said intake valve means, and exhaust gases may be exhausted from said combustion chamber through said exhaust valve means, characterized by:

means associated with said camshaft, and linkage means for timing said opening of said intake valve means and said closing of said exhaust valve means such that said exhaust valve means is fully closed before said inlet valve means is opened throughout the entire range of operating speeds of said engine to prevent fresh air from flowing directly from said intake valve means through said exhaust valve means.

The apparatus embodying the present invention may include a camshaft having cams so shaped and positioned that during operation of the engine, the exhaust valve of each cylinder is fully closed before its respective intake valve is opened.

The aforementioned timing of the valves is achieved by adjusting the relative positions of the cams actuating the intake and exhaust valves relative to one another as well as the contour of the flank and nose portions of the cam. Although there is a virtually infinite number of possible combinations of cam contours and relative cam combinations, the desired effect is to time the closing of the exhaust valve at the end of the exhaust stroke so that the air entering the cylinder does not pass through the exhaust port without being burned. This requires that the closing of the exhaust valve occur before the opening of the inlet valve, thus eliminating valve overlap. Since the method of the invention can be performed using a standard compression ignition engine on which only relatively minor adjustments have been made, the invention is ideally suited for retrofit applications. By substituting a camshaft ground in the manner of the invention for the standard camshaft of a conventional compression ignition engine in a vehicle, that vehicle will have significantly reduced emissions, regardless of its vintage.

Although the method will reduce significantly the presence of NO_x in the exhaust gases of all medium to high-speed compression ignition engines, the results are, according to the invention, most noticeable in those compression ignition engines equipped with a turbocharger or other supercharging device. If an engine is turbocharged, a greater differential exists between the pressure of the fresh or unburned air flowing into the combustion chamber and the pressure of the exhaust gas or burned air in the combustion chamber than is the case with a non-turbocharged engine. As a result, air enters the combustion chamber during the intake stroke at a faster rate than with a non-turbocharged engine, and a greater amount of air enters the combustion chamber, even though the intake valve is opened for a shorter period of time.

Similarly, with the turbocharged engine, there exists a greater differential in pressure between the exhaust gases or burned air in the combustion chamber and those in the exhaust manifold than exists with a non-turbocharged engine. This increased pressure differential causes the exhaust gases within the combustion chamber to scavenge more rapidly than would a non-turbocharged cylinder.

The overall result is that a sufficient volume of fresh air enters the cylinder to impart a powerful thrust to the piston upon burning, and subsequently the cylinder is scavenged without the "blow by" that occurs in prior art compression ignition engines and causes excessive NO_x in the exhaust gases.

In order that the invention may be more readily understood, reference will now be made to the accompanying drawings in which:

Fig. 1 is a side elevation in section of an engine embodying the invention during the intake stroke;

Fig. 2 is a side elevation in section of an engine embodying the invention during the compression stroke;

Fig. 3 is a side elevation in section of an engine embodying the invention during the combustion or expansion stroke;

Fig. 4 is a side elevation in section of an engine embodying the invention during the scavenging or exhaust stroke;

Fig. 5 is a partial side elevation in section of the cam and valve assembly of an engine embodying the invention;

Fig. 6 is a side elevation in section of a prior art compression ignition engine at the end of the exhaust stroke and the beginning of the intake stroke;

Fig. 7 is a valve timing diagram of an engine embodying the present invention;

Fig. 8 is a valve timing diagram of a prior art compression ignition engine;

Fig. 9 is a side elevation in section showing a turbocharger schematically; and

Fig. 10 is a partial side elevation in section of a compression ignition engine of the open chamber type also showing a cam and valve assembly embodying the invention.

As shown in Figs. 1 through 4, the method and apparatus embodying the present invention can be integrated into a standard, high-speed, four stroke, compression ignition engine. The power generating portion of such engines typically consists of a piston 10 which is pivotally connected to a piston rod 12 mounted on a crankshaft 14 which transmits the piston movement to a drive train (not shown). The piston 10 reciprocates within a cylinder 16 that defines a combustion chamber 18 which communicates with an intake manifold 20 by means of an inlet port 22 and with an exhaust manifold 24 through an exhaust port 26. The inlet and exhaust ports 22, 26 are shaped to receive intake and exhaust valves 28, 30 respectively, which can be moved to open and close passages in the inlet and exhaust ports. Means (not shown) are provided for forcing fresh air into the intake manifold 20 such that, when the intake valve 28 is open, the air introduced into the combustion chamber 18 will have a pressure above ambient pressure.

A fuel injection nozzle 32, which is connected to a fuel source (not shown), communicates with a pre-combustion chamber 34. The pre-combustion chamber 34 in turn communicates with the combustion chamber 18.

As shown in Fig. 5, a typical valve 38 in a compression ignition engine pivots against a rocker arm 40 in which is pivotally journaled a push rod 42. The push rod 42 terminates in a cam follower 44 which rolls against a cam 46 fixedly journaled to a camsahft 48. The camshaft 48 is turned by the crankshaft 14 by means of a linkage (not shown) well-known in the art. As the camshaft 48 rotates, the eccentricity of the cam shape causes the cam follower 44 to rise and fall thereby causing the valve 38 to engage and disengage a typical port 50 defining a port. The valve 38 is urged against its valve seat by means of a spring 52 which operates between the cylinder head 54 and the retainer portion 56 of the valve 38.

The timing of the opening and closing of the intake and exhaust valves 28, 30 is a function not only of the positions of their respective cams 46 in relation to one another on the camshaft 48 but also of the cam contour. The cam contour is comprised of a base circle portion 58, a nose 60, and two flanks 62. The shapes of the flanks 62 and the nose 60 of a cam 46 determine the rate at which each valve is opened and the duration that it remains open.

The method of operating the Diesel engine embodying the present invention is as follows. As shown in Fig. 1, the cranksahft 14 may turn in a clockwise direction, drawing the piston 10 downward within the cylinder 16, and at the same time, the intake valve 28 is moved away from the inlet port 22, thus allowing fresh air 64 from the intake manifold 20 to be forced into the cylinder. This process begins when the piston is approximately 1° to 3° past top dead center, that is, when the crankshaft 14 has turned 1° to 3° beyond the position it was in at the time the piston 10 reached its maximum ascent within the cylinder 16. The intake valve 28 remains open until the piston 10 has reached approximately 30° past bottom dead center, that is, the crankshaft 14 has turned 30° beyond the position it was in at the time the piston 10 reached its furthest descent within the cylinder 16.

As shown in Fig. 2, the compression stroke begins with the closing of the intake valve 28 and the travel of the piston 10 upward within the cylinder 16. As the air 64 is compressed within the cylinder 16, it becomes hotter.

When the piston 10 is near top dead center a charge of fuel 65 is injected through the nozzle 32 as a fine spray into the hot air 64, and ignition takes place. As shown in Fig. 3, the expanding gases 66 force the piston 10 downward on the third stroke of the cycle, and the movement of the piston is transmitted to the crankshaft 14 by the piston rod 12.

As shown in Fig. 4, the exhaust valve 30 opens when the piston 10 is approximately 30° before bottom dead center, and the scavenging or exhaust stroke begins. The piston 10 reaches bottom dead center and begins its ascent up the cylinder 16 to force the exhaust gases 68 out through the exhaust port 26 and the exhaust manifold 24. When the piston 10 is near top dead center, the exhaust valve 30 closes the exhaust port 26 completely, thereby cutting off the flow of exhaust gases 68 through the port and trapping a small amount of exhaust gas within the cylinder 16. As the piston 10 passes top dead center and begins the first or intake stroke, the intake valve 28 opens the inlet port 22, and fresh air 64 is admitted. Thus, in the method embodying the present invention, a small amount of exhaust gas 68 may remain in the cylinder, and no fresh air 64 is permitted to "blow by" and mix with the exhaust gases in the exhaust manifold 24.

The foregoing explanation of the method and apparatus embodying the present invention is contrasted with the operation of a conventional Diesel engine of the prior art as shown in Fig. 6. Fig. 6 depicts the position of the piston 10, intake valve 28 and exhaust valve 30 at the end of the exhaust stroke and the beginning of the intake stroke.

In the operation of Diesel engines of the prior art, both valves 28, 30 are open at this time to allow fresh air 64 to enter the combustion chamber 18, thereby completely scavenging the exhaust gases 68 from the combustion chamber. However, a certain amount of "blow by" occurs wherein fresh air 64 passes into the combustion chamber 18 and out the exhaust port 26 without supporting the combustion of the fuel. In order to reduce significantly the presence of unacceptable levels of NO_x in the exhaust gases of the engine embodying the present invention, the prior art configuration depicted in Fig. 6 does not occur at any time during the operation of the Diesel engine embodying the present invention.

Fig. 7 is a valve timing diagram for the operation of a Diesel engine embodying the present invention. The circle generally designated A can be considered as the path traced by a point positioned on the crankshaft 14. The line segment TDC represents the position of the crankshaft 14 — and hence the piston 10 — at top dead center, that is, when the piston has risen to its highest point in the cylinder 16. The line segment BDC represents the position of the crankshaft 14 and piston 10 at bottom dead center, that is, the point at which the piston has reached its furthest descent within the cylinder 16.

Thus to depict the valve sequence for a Diesel engine embodying the present invention, the piston begins at a point TDC on the valve diagram and begins to descend as the crankshaft turns in a clockwise manner. The inlet valve opens at line segment W, which represents a cylinder position approximately 3° after top dead center, and remains open to line segment X approximately 30° after bottom dead center. The area bounded by lines W and X represents the period of time during the first cycle when the intake valve 28 is open.

Line X also designates the beginning of the second or compression stroke. This stroke continues to a point near top dead center at which time the fuel is sprayed into the combustion chamber 18 through the nozzle 32 and the expansion stroke begins. During the expansion stroke, the crankshaft 14 is turning from line TDC to line Y, located within circle A. Line Y denotes the opening of the exhaust valve 30 and the beginning of the exhaust stroke shown in Fig. 4.

The exhaust stroke begins at approximately 30° before bottom dead center and continues to a point denoted by line Z which is approximately 3° before top dead center. Line segment Z denotes the point at which the exhaust valve is completely closed. The segment of the timing cycle between lines Z and W represents a period of crankshaft rotation during which both the intake valve 28 and the exhaust valve 30 are closed. It is crucial to the operation of a Diesel engine embodying the present invention that this segment appear on the valve timing sequence.

In contrast, a valve timing diagram of a Diesel engine operated according to the method of prior art is shown as circle A′ in Fig. 8. The start of the first or intake stroke is shown by line segment W′ which occurs before top dead center. The intake valve 28 remains open until line segment X′, typically about 25° past bottom dead center. The compression stroke begins at line X′ with the closing of the intake valve 28 and continues through to a point near top dead center, at which time the fuel is sprayed into the combustion chamber 18 from the nozzle 32 and the third or expansion stroke begins.

The expansion stroke continues through to line segment W′, located within the circle A′. Line Y′ denotes the opening of the exhaust valve 30 and the beginning of the exhaust stroke. The exhaust stroke continues through to a point Z′, typically after top dead center.

Thus, the segment of the valve timing diagram of Fig. 8 denoted by the double cross-hatching represents the time during the four-stroke cycle of the prior art in which both the intake and the exhaust valves 28, 30 are open, as shown in Fig. 6. It is at this time that fresh air 64 enters the combustion chamber 18 as the exhaust gases 68 are leaving the combustion chamber 18, and some fraction of the fresh air 64 leaves the cylinder along with the exhaust gases 68. By eliminating the time during which both the intake valve 28 and the exhaust valve 30 are open, "blow by" of fresh air 64 entering the combustion chamber 18 is prevented, and the amount of NO_x formed in the exhaust gases 68 is reduced.

The method and apparatus embodying the present invention are particularly effective when used in conjunction with a turbocharged Diesel engine as shown in Fig. 9. An exhaust turbine 70 located in the exhaust manifold 24 is driven by the exhaust gases 68 leaving the combustion chamber 18 during the exhaust stroke. The exhaust turbine 70 is coupled to an inlet turbine 72 by a drive shaft 74, and the inlet turbine is rotated by the exhaust turbine 70 to force fresh air 64 into the combustion chamber 18 during the air intake stroke.

The result of forcing air into the combustion chamber, for example by turbocharging, is that a much greater amount of fresh air 64 is present in the combustion chamber 18 during the operation of the engine than in the case of a normally aspirated engine, and consequently more fuel can be injected and a greater horsepower generated for a given cylinder.

Since higher pressures are involved, there is a greater amount of blow by of fresh air 64 in the operation of a prior art Diesel. The elimination of valve overlap eliminates all blow by and thereby reduces significantly the amount of NO_x in the exhaust gases 68.

Although the invention has been discussed previously as used in connection with a compression ignition engine which includes a precombustion chamber, the invention has been successfully tested in combination with an engine of the open chamber type, as shown in Fig. 10. In an open chamber type engine, the cylinder head 54′ is designed so that the fuel injection nozzle 32′ injects fuel directly into the combustion chamber 18′.

The piston 10′ has an upper surface 76 which defines a recess 78 to receive a charge 65′ of fuel. However, the configuration and operation of the cam and lifter assembly 79 are the same as that shown in Fig. 5. A typical valve 38′ in a compression ignition engine pivots against a rocker arm 40′ in which is pivotally journalled push rod 42′. Push rod 42′ terminates in a cam follower 44′ which rolls against a cam 46′ fixedly journalled to camshaft 48′.

As discussed previously, rotation of the camshaft 48′ causes cam follower 44′ to rise and fall in response to the eccentricities of the shape and contours of cam 46′. This cam is ground to the proper contour to time the opening and closing of valve 38′ to eliminate blow by of unburned air 64′.

The open chamber engine shown in Fig. 10 may be turbocharged, and is shown schematically with turbocharging apparatus. As was discussed in connection with Fig. 9, the turbocharger 80 of Fig. 10 is preferably of the exhaust gas type, and includes an exhaust turbine 70 which is rotated by the force of escaping exhaust gases 68′, an inlet turbine 72′, and a drive shaft 74′ which joins the inlet turbine to the exhaust turbine. The rotation of the exhaust turbine 70′ causes the drive shaft 74′, and hence the inlet turbine 72′, to rotate, thereby compressing the fresh air 64′ entering the combustion chamber 18′. This compressed fresh air 64′ permits a greater amount of fuel to be injected into and burned in the combustion chamber 18′, resulting in greater horsepower for that engine configuration than without turbocharging.

In accordance with the above discussion, Tables 1 and 2 show the effect of variations in valve overlap on the amount of NO_x present in the exhaust gases of a medium speed turbocharged Diesel engine of the open chamber type. By "medium speed" is meant a Diesel engine which is designed for a maximum operating speed of from 2400 to 2600 rpm. at full load, and as compared with high-speed engines which operate in a speed range in excess of 2600 rpm. The testing equipment and procedures used in generating this data were capable of duplicating the City and Highway Modes of the Federal Test Procedures as outlined in Part 86 of Chapter 1, Title 40 of the Code of Federal Regulations as applicable to light-duty vehicles. The testing facility at which the tests were performed was one of ten such facilities in the country listed by the U.S. Environmental Protection Agency as being equipped to perform emission tests in accordance with the aforementioned federal procedures.

Three different cam designs yielding three different amounts of valve overlap were tested in a standard turbocharged Diesel engine mounted in one of two light-duty vehicles. All tests were run in accordance with the 1975 Federal Test Procedure. In this Federal Test Procedure, the vehicle to be tested was placed on a dynamometer set at predetermined resistance to simulate wind and rolling friction, and its exhaust gases were sampled while the vehicle was put through a series of accelerations, decelerations and idle periods in a way designed to simulate actual driving conditions. The results for the entire test were reported in terms of grams of a particular pollutant per mile or per kilometre of vehicle operation on the dynamometer.

TABLE 1

Test No.	Engine Type	Valve Overlap (Degrees)	NO_x(gm/mi) & (gm/km)	
1.	Turbocharged Diesel	+28 to +30	9.65	6.00
2.	Turbocharged Diesel	+1 to +3	2.35	1.46
3.	Turbocharged Diesel	−1 to −3	1.85	1.15

TABLE 2

Test No.	Engine Type	Valve Overlap (Degrees)	NO_x(gm/mi) & (gm/km)	
4.	Turbocharged Diesel	+28 to +30	6.35	3.95
5.	Turbocharged Diesel	+1 to +3	1.94	1.20
6.	Turbocharged Diesel	−1 to −3	1.86	1.16

TABLE 3

Test No.	Engine Type	Valve Overlap	NO_x		HC (gm/mi) & (gm/km)	
			gm/mi	gm/km		
7.	Turbocharged Diesel	−2	1.35	0.84	1.01	0.63
8.	Turbocharged Diesel	−2	1.68	1.04	.52	0.32
9.	Turbocharged Diesel	+28 to +30	3.19	1.98	.456	0.283
10.	Turbocharged Diesel	+28 to +30	4.07	2.53	.347	0.216

Table 1 shows the data generated by the vehicles which were put through a total of three Federal City Mode tests, each time with a cam design yielding a different degree of valve overlap.

In Test 1, a van having a standard, unmodified, turbocharged Diesel of a type exemplifying a prior art engine was tested. The engine displacement was 3.7 liters (226 $in.^3$) and the dynamometer was set to simulate resistance for a 1818.2 kg (4000 lbs.) vehicle. The amount of valve overlap, that is, the range of crankshaft angles during which both the inlet valve and the outlet valve were open (see Figs. 6 and 8), was approximately 30°. The amount of NO_x generated for the entire Federal City Mode was 6.00 gm/km (9.65 gm/mi).

In Test 2, a pick-up truck having a trubocharged Diesel engine whose cams had been modified so that the valve overlap was reduced to approximately 1° to 3° was tested. The amount of NO_x present in the exhaust gases for the City Mode was 1.46 gm/km (2.35 gm/mi).

In Test 3, a pick-up truck having the same type of turbocharged Diesel engine whose cam had been modified in accordance with the present invention was tested. The amount of valve overlap in this test was approximately −1° to −3°. The amount of NO_x generated was approximately 1.15 gm/km (1.85 gm/mi). Clearly, a turbocharged Diesel engine whose cam has been modified in accordance with the present invention displays a significant decrease in the amount of NO_x generated in the exhaust gas during normal use.

Similarly, Table 2 depicts the same three vehicle and engine combinations subjected to the Federal Highway Mode on the same test facilities described above. The data from tests 4, 5 and 6 show that a modification of the engine to effect a negative valve overlap results in a significant decrease in the amount of NO_x in the exhaust gases.

Table 3 shows the data generated by the testing of a light duty truck having a four-cylinder turbocharged compression ignition engine of the open chamber type at the aforementioned facilities and under the same types of tests. The engine had a displacement of 3.7 liters (226 $in.^3$) and a compression ratio of 18:1. The track underwent the test on a dynomometer set at 1818.2 kg (4000 pounds).

In tests 7 and 8, the subject was the aforementioned vehicle whose engine included a cam shaft modified in the manner of the invention to eliminate valve overlap and fresh air blow by. The amount of negative overlap was approximately 2°. In test 7, the vehicle was driven according to the Federal City Mode and generated 1.84 gm/km (1.35 gm/mi) of NO_x and 0.63 gm/km (1.01 gm/mi) of hydrocarbons. In test 8, the same vehicle was driven according to the Federal Highway Mode. The vehicle generated 1.04 gm/km (1.68 gm/mi) of NO_x and 0.32 gm/km (0.52 gm/mi) of hydrocarbons.

In tests 9 and 10, the same vehicle was retested according to the Federal City and Highway Modes respectively, but this time the engine was fitted with a standard cam shaft which allows approximately 30° of overlap. The results showed that significantly higher amounts of NO_x were generated. In particular, in test 9, the vehicle generated 1.98 gm/km (3.19 gm/mi) of NO_x while driven according to the City Mode and in test 10 generated 2.53 gm/km (4.07 gm/mi) of NO_x when driven in the Highway Mode.

It should be noted that the amounts of hydrocarbons (HC) that were generated by the vehicle and measured during the tests were greater when the engine was modified according to the invention. However, this increase is believed to be relatively insignificant when compared to the relatively large reduction of oxides of nitrogen.

Claims

1. A method of operating a medium to high-speed, four-cycle supercharged compression ignition engine of the type wherein an intake valve (28) is moved to open an inlet port (22) to allow fresh air (64) to enter a combustion chamber (18), said air is compressed, fuel (65) is injected into said combustion chamber and burns thereby expanding said air, and an exhaust valve (30) is moved to open an exhaust port (26) to allow burned gases (68) to be scavenged through said exhaust port, characterized by:

timing said opening of said inlet port and said closing of said exhaust port such that said exhaust port is completely closed before said inlet port is opened at all engine speeds.

2. A method as claimed in claim 1, wherein the supercharging includes operating a turbocharger (80) communicating with said inlet port (22) to force fresh air (64) into said combustion chamber (18).

3. A method as claimed in claim 1 or 2, wherein said engine is of the open chamber type.

4. A method as claimed in claim 1, 2 or 3, wherein said engine has a maximum operating speed of from 2400 to 2600 revolutions per minute.

5. A supercharged medium to high-speed, four-cycle automotive compression ignition engine of the type having at least one cylinder (16) defining a combustion chamber (18) and having a reciprocating piston (10) therein linked to rotate a crankshaft (14); an intake manifold (20) having intake valve means (22, 28) communicating with said combustion chamber, exhaust valve means (26, 30) communicating with said combustion chamber, and camshaft (48) and linkage means (40, 42, 52, 56) for intermittently opening and closing said intake valve means and said exhaust valve means such that fresh air (64) may be drawn into said combustion chamber from said intake manifold through said intake valve means, and exhaust gases (68) may be exhausted from said combustion chamber through said exhaust valve means, characterized by means (58, 60, 62) associated with said camshaft, and linkage means for timing said opening of said intake valve means and said closing of said exhaust valve means such that said exhaust valve means is fully closed before said inlet valve means is opened throughout the entire range of operating speeds of said engine to prevent fresh air from flowing directly from said intake valve means through said exhaust valve means.

6. An engine as claimed in claim 5, wherein said timing means (58, 60, 62) comprises said camshaft (48) having cams (46) shaped and positioned thereon such that said exhaust valve means (26, 30) is fully closed before said intake valve means (22, 28) is opened.

7. An engine as claimed in claim 5 or 6, wherein the means for supercharging includes a turbocharger (80) associated with said intake manifold (20).

8. An engine as claimed in claim 5, 6, or 7, wherein said engine is of the open chamber type.

9. An engine as claimed in claim 5, 6, 7, or 8, wherein said engine has a maximum operating speed of from 2400 to 2600 revolutions per minute.

Revendications

1. Procédé de conduite d'un moteur à allumage par compression, à quatre temps, à surcompresseur, de vitesse moyenne à élevée, du type dans lequel une soupape d'admission (28) est déplacée pour ouvrir une lumière d'admission (22) pour permettre à l'air frais (64) d'entrer dans une chambre de combustion (18), ledit air est comprimé, du carburant (65) est injecté dans ladite chambre de combustion et brûle en dilatant ledit air, et une soupape d'échappement (30) est déplacée pour ouvrir une lumière d'échappement (26) pour permettre aux gaz brûlés (68) d'être évacués par ladite lumière d'échappement, caractérisé par:

— on détermine l'instant de l'ouverture de ladite lumière d'admission et l'instant de la fermeture de ladite lumière d'échappement de façon telle que ladite lumière d'échappement soit complètement fermée avant que la lumière d'admission soit ouverte, pour toutes les vitesses du moteur.

2. Procédé selon la revendication 1, dans lequel la surcompression comprend le fonctionnement d'un turbocompresseur (80), communiquant avec la lumière d'admission (22) pour forcer l'air frais (64) dans la chambre de combustion (18).

3. Procédé selon l'une des revendications 1 ou 2, dans lequel le moteur est du type à chambre ouverte.

4. Procédé selon l'une des revendications 1, 2 ou 3, dans lequel le moteur a une vitesse de fonctionnement maximum de 2400 à 2600 tours/minute.

5. Moteur à allumage automatique par compression à quatre temps, à surcompression, de vitesse moyenne à élevée, du type ayant au moins un cylindre (16) définissant une chambre de combustion (18) et ayant dedans un piston (10) à mouvement alternatif, embiellé pour faire tourner un vilebrequin (14), un collecteur d'admission (20) ayant des moyens de soupape d'admission (22, 28) communiquant avec ladite chambre de combustion, des moyens de soupape d'échappement (26, 90) communiquant avec ladite chambre de combustion, et des moyens d'arbre à came (48) et d'embiellage (40, 42, 52, 56) pour ouvrir et fermer par intermittence lesdits moyens de soupape d'admission et lesdits moyens de soupape d'échappement, de façon telle que de l'air frais (64) puisse être aspiré dans ladite chambre de combustion à partir du collecteur d'admission, à travers ledit moyen de soupape, et que

des gaz d'échappement (68) puissent être évacués hors de la chambre de combustion par ledit moyen de soupape d'échappement, caractérisé par:

— des moyens (58, 60, 62) associés aux moyens d'arbre à came et d'embiellage pour déterminer l'instant d'ouverture dudit moyen de soupape d'admission et l'instant de fermeture dudit moyen de soupape d'échappement, de façon que le moyen de soupape d'échappement soit complètement fermé avant que le moyen de soupape d'admission soit ouvert, dans tout le domaine de vitesses de fonctionnement du moteur, pour empêcher l'air frais de passer directement du moyen de soupape d'admission à travers le moyen de soupape d'échappement.

6. Moteur selon la revendication 5, caractérisé en ce que les moyens de détermination d'instants (58, 60, 62) comprennent ledit arbre à cames (48) portant des cames (46) de formes et de positions telles que lesdits moyens de soupape d'échappement (26, 30) soient complètement fermés avant que lesdits moyens de soupape d'admission (22, 28) soient ouverts.

7. Moteur selon l'une des revendications 5 ou 6, caractérisé en ce que les moyens pour surcomprimer comprennent un turbocompresseur (80) associé audit collecteur d'admission (20).

8. Moteur selon l'une des revendications 5, 6 ou 7, caractérisé en ce que ledit moteur est du type à chambre ouverte.

9. Moteur selon l'une des revendications 5, 6, 7 ou 8, caractérisé en ce que ledit moteur a une vitesse de fonctionnement maximum de 2400 à 2600 tours/minute.

Patentansprüche

1. Verfahren zum Betrieb einer im mittleren und hohen Drehzahlbereich arbeitenden, aufgeladenen, Viertakt-Diesel-Brennkraftmaschine der Type, in der ein Einlaßventil (28) zur Öffnung eines Einlasses (22) für den Eintritt von Frischluft (64) in die Verbrennungskammer (18) bewegt, und die Luft komprimiert, Kraftstoff (65) in die Verbrennungskammer eingespritzt und verbrennt, und dabei die Luft expandiert, und ein Auslaßventil (30) zur Öffnung einer Auspufföffnung (26) für den Austritt der durch die Auspufföffnung auszustoßenden Verbrennungsgase (68) bewegt wird, gekennzeichnet durch die Zeitpunkteinstellung des Öffnens des Einlasses (22) und des Schließens des Auslasses (26) derart, daß der Auslaß (26) vollständig geschlossen ist, bevor der Einlaß (22) in allen Drehzahlbereichen der Maschine geöffnet wird.

2. Verfahren nach Anspruch 1, dadurch gekennzeichnet, daß die Aufladeeinrichtung einen mit dem Einlaß (22) in kommunizierender Verbindung stehenden Turbolader (80) zum verstärkten Einbringen von Frischluft (64) in die Verbrennungskammer (18) einschließt.

3. Verfahren nach Anspruch 1 oder 2, dadurch gekennzeichnet, daß die Maschine eine Type mit offener Kammer ist.

4. Verfahren nach Anspruch 1, 2 oder 3, dadurch gekennzeichnet, daß die Maschine eine maximale Betriebsdrehzahl von 2400 bis 2600 U/min hat.

5. Aufgeladene, im mittleren bis hohen Drehzahlbereich arbeitende Viertak-Diesel-Brennkraftmaschine der Type mit mindestens einen eine Verbrennungskammer (18) bildenden Zylinder, und einen darin hin- und herbeweglichen, zur Drehung einer Kurbelwelle (14) angelenkten Kolben (10); einen Ansaugkrümmer (20) mit einer mit der Verbrennungskammer in kommunizierender Verbindung stehenden Ansaugventil-Anordnung (22, 28), einer Abgasventil-Anordnung (26, 30) in kommunizierender Verbindung mit der Verbrennungskammer, und einer Nockenwelle (48) mit Gestängeelementen (40, 42, 52, 56) für das intermittierende Öffnen und Schließen der Ansaugventil-Anordnung und der Abgasventil-Anordnung derart, daß Frischluft (64) vom Ansaugkrümmer durch die Ansaugventil-Anordnung in die Verbrennungskammer eingesaugt und Abgas (68) auf der Verbrennungskammer durch die Abgasventil-Anordnung abgeführt werden kann, gekennzeichnet durch eine der Nockenwelle zugeordnete Steueranordnung (58, 60, 62) und Gestangeelemente für das zeitlich abgestimmte Öffnen der Ansaugventil-Anordnung und das Schließen der Abgasventil-Anordnung derart, daß die Abgasventil-Anordnung vollständig geschlossen ist, bevor die Ansaugventil-Anordnung geöffnet wird, um den direkten Durchfluß von Frischluft von der Ansaugventil-Anordnung durch die Abgasventil-Anordnung zu verhindern.

6. Diesel-Brennkraftmaschine nach Anspruch 5, dadurch gekennzeichnet, daß die die zeitliche Abstimmung bewirkende Anordnung (58, 60, 62) die Nockenwelle (48) umfaßt, die Nocken aufweist, die in ihrer Form und Position auf der Nockenwelle derart aus ausgelegt sind, daß die Abgasventil-Anordnung (26, 30) vollständig geschlossen ist, bevor die Ansaugventil-Anordnung (22, 28) geöffnet wird.

7. Diesel-Brennkraftmaschine nach Anspruch 5 oder 6, dadurch gekennzeichnet, daß die Aufladeeinrichtung einen dem Ansaugkrümmer (20) zugeordneten Turbolader (80) einschließt.

8. Diesel-Brennkraftmaschine nach Anspruch 5, 6 oder 7, dadurch gekennzeichnet, daß die Maschine eine Type mit offener Kammer ist.

9. Diesel-Brennkraftmaschine nach Anspruch 5, 6 oder 7, dadurch gekennzeichnet, daß die Maschine eine maximale Betriebsdrehzahl von 2400 bis 2600 U/min hat.

FIG-1

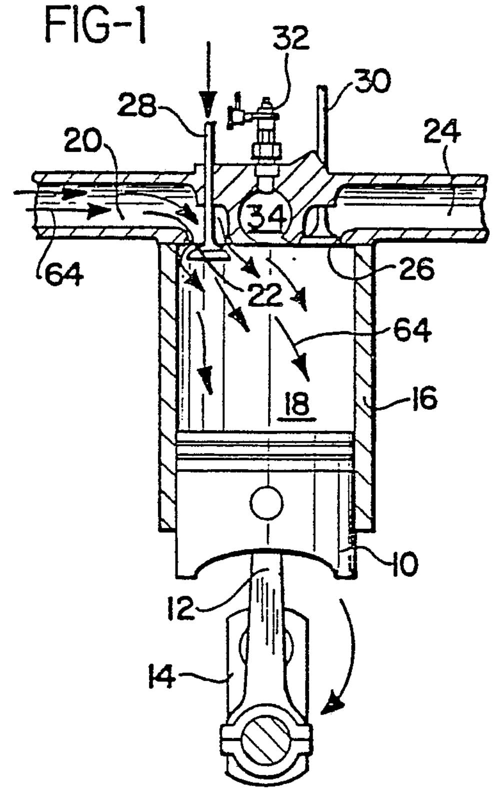


FIG-2

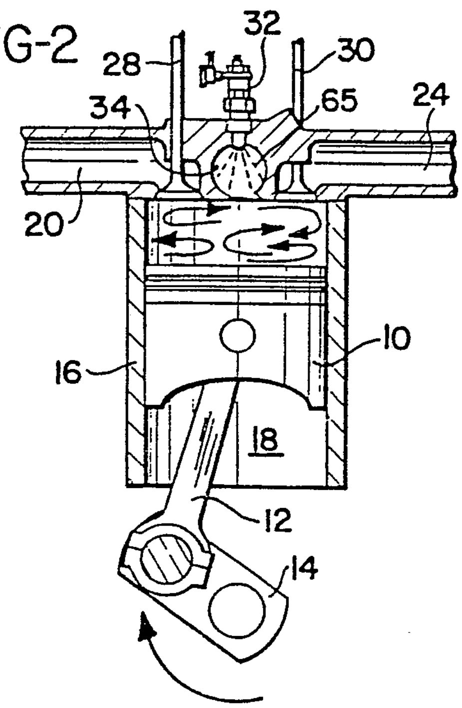


FIG-3

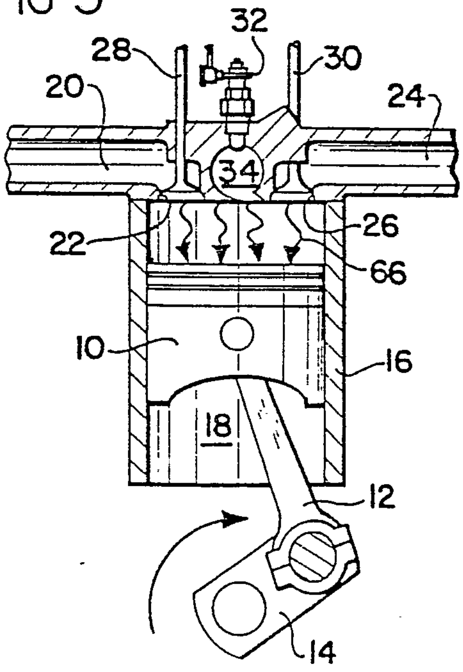


FIG-4

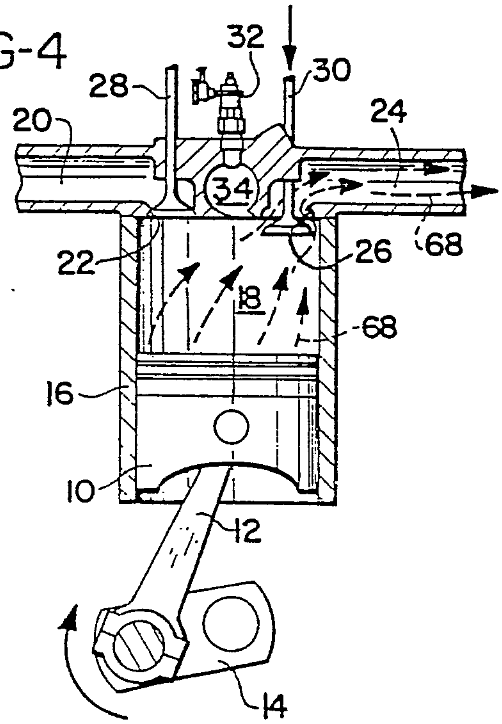


FIG-5

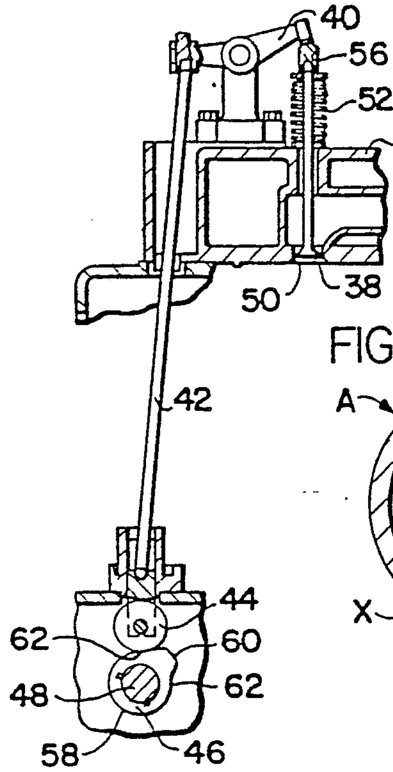
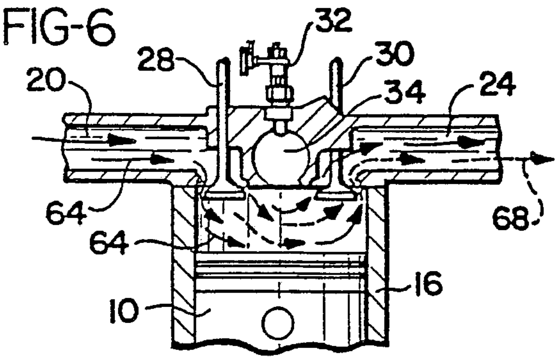


FIG-6



PRIOR ART

FIG-7

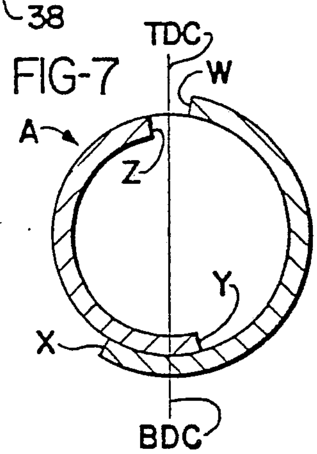
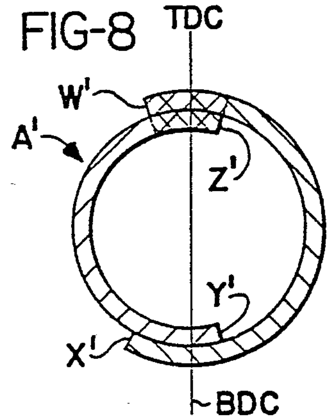


FIG-8



PRIOR ART

FIG-9

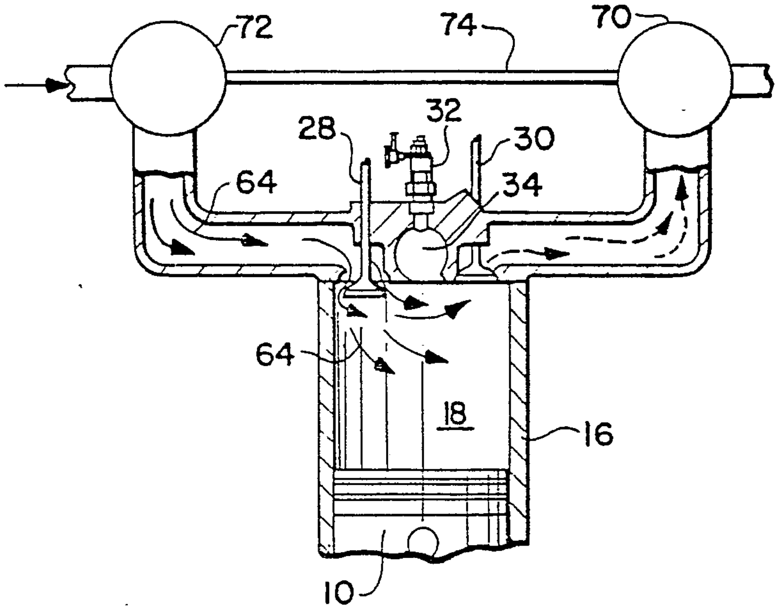


FIG-10