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

(57) An improved method and apparatus for operating medium to high-speed four cycle compression ignition engines causes the amount of NO_x in the exhaust gases to be reduced substantially. The four cycle compression ignition engine is of the type wherein fresh working fluid (64) is introduced through an intake valve (28), the working fluid is compressed, fuel is injected and burns thereby expanding the working fluid, and the working fluid is scavenged through an exhaust valve (30). The improvement comprises timing the opening of the intake valve and the closing of the exhaust valve so that no fresh working fluid is permitted to pass out the exhaust valve.

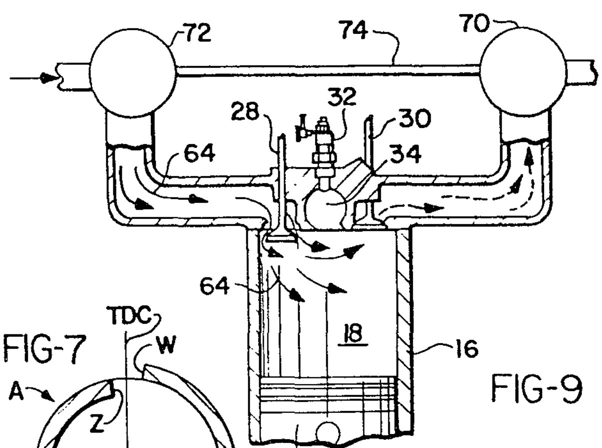
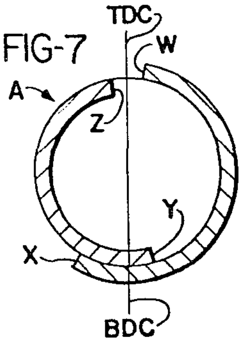


FIG-9

FIG-7


EP 0 036 261 A1

Croydon Printing Company Ltd.

A METHOD OF OPERATING A FOUR CYCLE COMPRESSION IGNITION ENGINE AND AN ENGINE OPERABLE ACCORDING TO THE METHOD

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 duty 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 emmissions in some respects than comparable spark ignition engines. While carbon monoxide emmissions 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 emmissions 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 exiting 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.

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 the exhaust valve is completely closed prior to the time the piston reaches top dead center, and the intake valve opens after the piston passes top dead center so that no fresh air is permitted to pass out the exhaust valve. 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 a significant reduction in the effective horsepower or mileage.

According to one aspect of the present invention, a method of operating a medium to high-speed four-cycle compression ignition engine of the type wherein fresh air is introduced through an intake port, the air is compressed, fuel is injected and burns to expand the air, and the air is scavenged through an exhaust port, is improved by so timing the opening of the intake valve and the closing of the exhaust valve that no fresh air is permitted to pass out through the exhaust port. The apparatus of the present invention includes 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. In some instances, 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 of the present invention will reduce significantly the presence of NO_x in the exhaust gases of all medium to high-speed compression ignition engines, the results are most noticeable in those compression ignition engines equipped with a turbocharger. 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 addition, it is believed that a turbocharged compression ignition engine is particularly suitable for the method of the present invention. With such an engine, the valves are timed in the manner of the prior art so that there is valve overlap at the end of the exhaust stroke and the beginning of the air intake stroke, when an even greater amount of fresh air passes out the exhaust port.

Accordingly, it is an object of this invention to provide an improved method of operating a medium to high-speed four stroke compression ignition engine 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.

Other objects and advantages of the invention will be apparent from the following description, the accompanying drawings, and the appended claims.

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 the invention during the intake stroke;

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

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

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

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

As shown in Figs. 1 through 4, the method and apparatus of 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.

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 the camshaft 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 of the present invention is as follows. As shown in Fig. 1, the crankshaft 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 drawn 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 decent 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 of 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 of 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 of the present invention, the prior art configuration depicted in Fig. 6 does not occur at any time during the operation of the Diesel engine of the present invention.

Fig. 7 is a valve timing diagram for the operation of a Diesel engine of the present invention. The circle generally designated A can be considered as the path traced by a point positioned on the crankshaft 14 of the present invention. The line segment TDC represents the postion 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 of 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 according to 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 of 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 is that a much greater amount of fresh air 64 is present in the combustion chamber 18 during the operation of the 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 emmission 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 gm/mi	NO_x gm/km	HC (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 turbocharged 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. An improved method of operating a medium to high-speed four-cycle compression ignition engine of the type wherein fresh air (64) is introduced through an intake port (22), the air is compressed, fuel (65) is injected and burns thereby expanding the air, and the air (68) is scavenged through an exhaust port (26), characterized by:

timing the opening of the intake valve (28) and the closing of the exhaust valve (30) such that no fresh air (64) is permitted to pass out through the exhaust port (22).

2. A method as claimed in claim 1 wherein the exhaust valve (30) fully closes before the intake valve (28) opens at all engine speeds.

3. A method as claimed in claim 1 or 2 wherein the engine is a turbocharged compression ignition engine.

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

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

6. An improved medium to high-speed compression ignition engine of the type having a plurality of cylinders (16), each having an intake valve (28) in an intake port (22) and an exhaust valve (30) in an exhaust port (26), characterized by:

means (46, 48) for timing the opening of the intake valve and the closing of the exhaust valve of each cylinder such that no fresh air (64) is permitted to pass out through the exhaust port.

7. An engine as claimed in claim 6 wherein the means for timing the opening of the intake valve (28) and the closing of the exhaust valve (30) includes a camshaft (48) having cams (46) shaped and positioned thereon such that during operation of the engine, the exhaust valve (30) of each cylinder is fully closed before the respective intake valve is opened.

8. An engine as claimed in claim 6 or 7 wherein the engine is a turbocharged compression ignition engine.

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

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

FIG-1

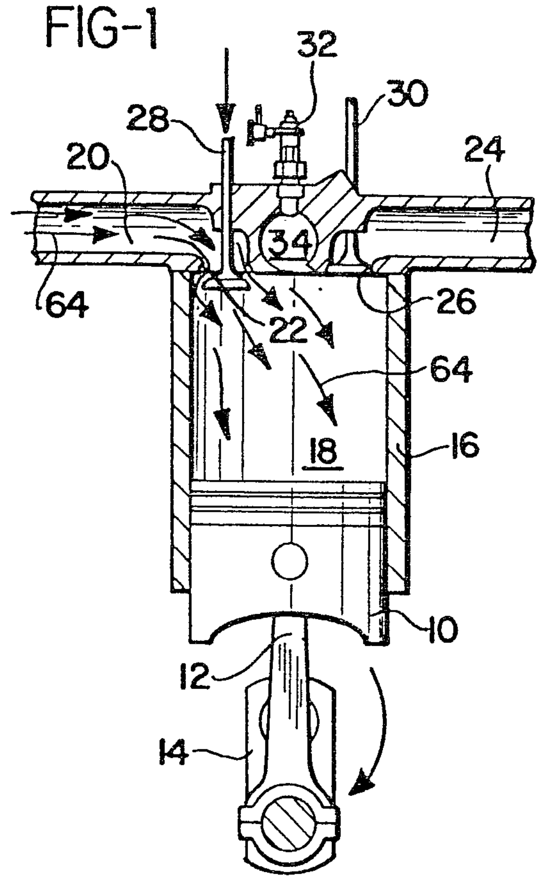


FIG-2

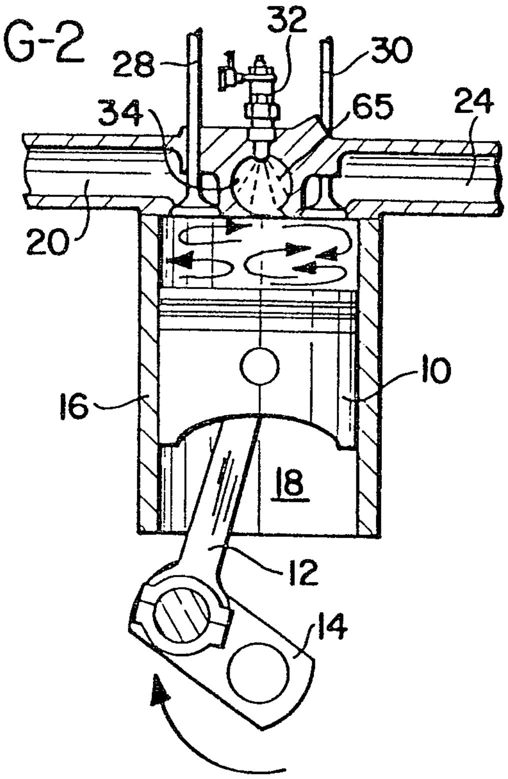


FIG-3

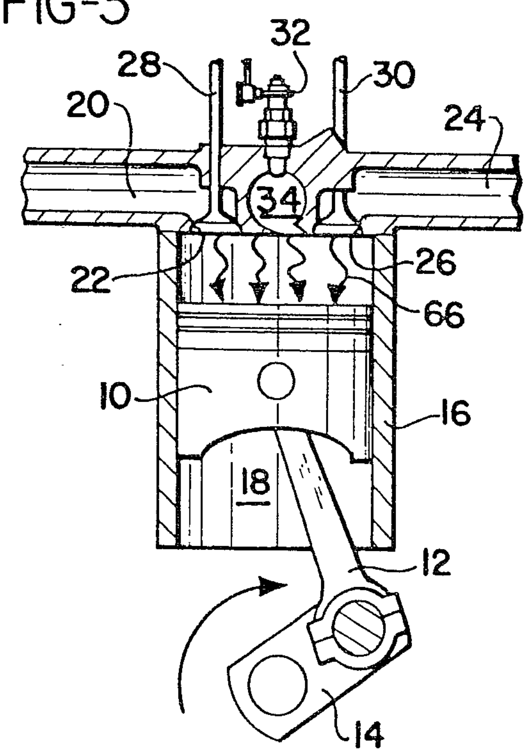


FIG-4

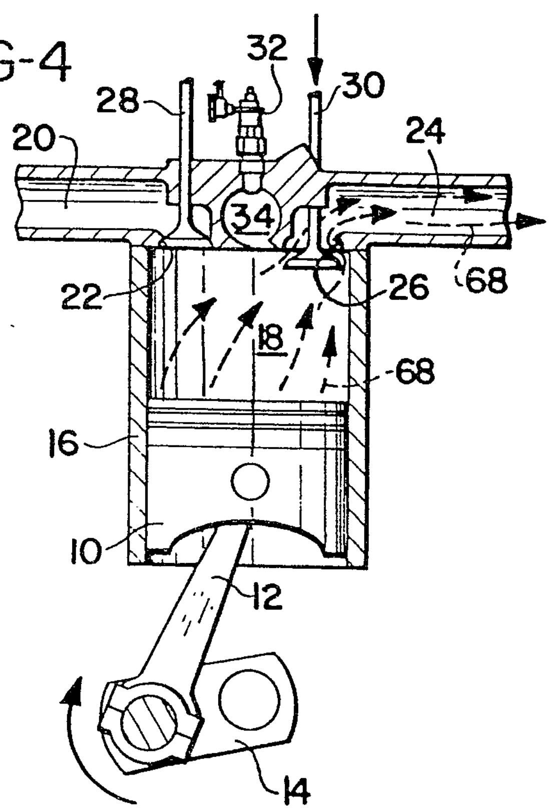


FIG-5

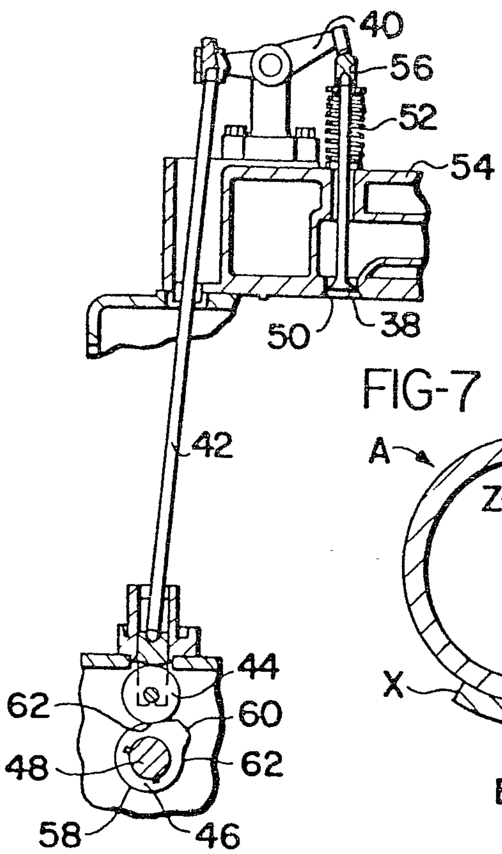
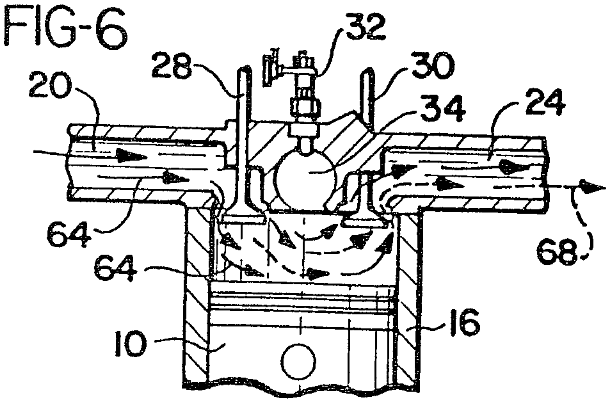


FIG-6



PRIOR ART

FIG-7

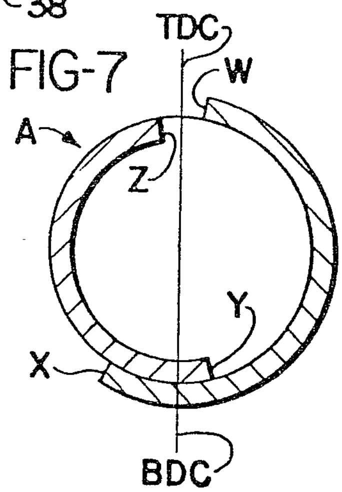
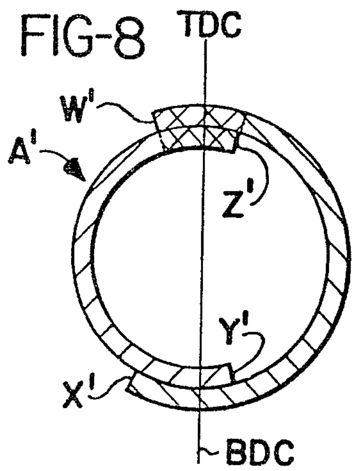


FIG-8



PRIOR ART

FIG-9

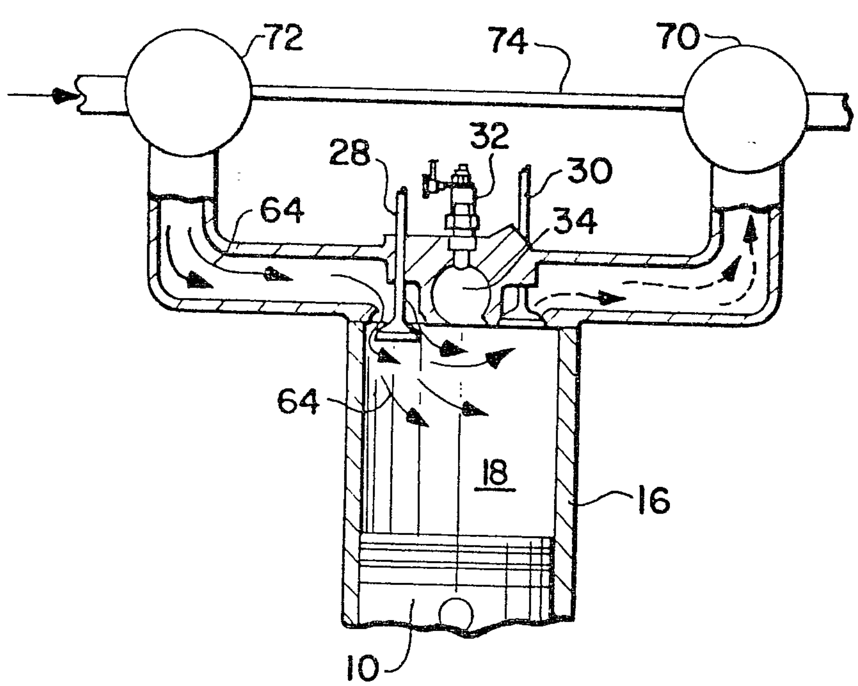
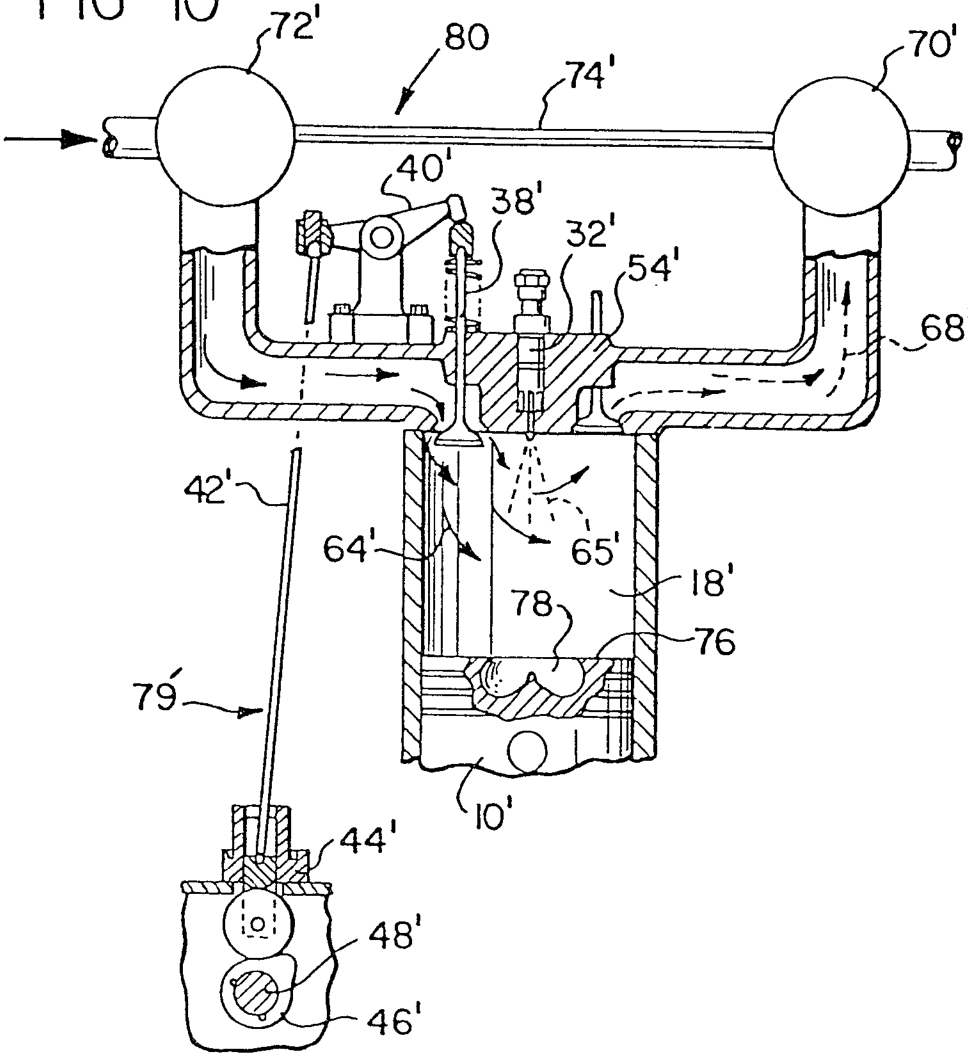


FIG-10



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Application number
EP 81 30 0874.5

DOCUMENTS CONSIDERED TO BE RELEVANT			CLASSIFICATION OF THE APPLICATION (Int. Cl 3)
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	
	FR - A - 2 158 942 (H. BÖHNLEIN) * claims 1 to 4; page 1, lines 1 to 4; page 2, lines 13 to 40; page 3, lines 1 to 2; page 4, lines 24 to 34; fig. 1 to 3 * --	1,2,3,6,7,8	F 02 B 1/06 F 01 L 1/00
A	US - A - 3 144 009 (H.S. GOODFELLOW et al.) * column 4, lines 43 to 55; fig. 9 * --		TECHNICAL FIELDS SEARCHED (Int. Cl.3) F 01 L 1/00 F 01 L 3/00 F 01 L 31/22 F 02 B 1/00 F 02 B 3/00 F 02 B 41/00 F 02 B 43/00 F 02 B 61/00 F 02 B 69/00 F 02 D 1/00
A	FR - A - 1 529 537 (INSTITUT FRANCAIS DU PETROLE, DES CARBURANTS ET LUBRIFIANTS) * claim * ---- ./..		CATEGORY OF CITED DOCUMENTS X: particularly relevant A: technological background O: non-written disclosure P: intermediate document T: theory or principle underlying the invention E: conflicting application D: document cited in the application L: citation for other reasons
X	The present search report has been drawn up for all claims		&: member of the same patent family, corresponding document

Place of search	Date of completion of the search	Examiner
Berlin	26-05-1981	CANNICI

0036261

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Application number
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DOCUMENTS CONSIDERED TO BE RELEVANT			CLASSIFICATION OF THE APPLICATION (Int. Cl.³)
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	
			TECHNICAL FIELDS SEARCHED (Int. Cl.³)
			F 02 D 13/00 F 02 D 21/08 F 02 D 23/00