A SHORT CUT FOR BRINGING HCCI ENGINES FROM LABORATORIES TO THE MARKET PLACE

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Introduction

HCCI engines hold the promise of providing cleaner burning and more fuel-efficient internal combustion engines. Characterized by the autoignition of a compressed lean homogenous charge, the entire compressed fuel/air mixture burns simultaneously avoiding the pressure rise (after burning) and further compression of already burned gases, which pressure rise is the primary cause for the high combustion temperatures that cause the formation of NOx. Researchers have yet to develop a viable means for controlling the timing of autoignition. Combustion is initiated by the compression of the homogenous charge to the required autoignition temperature. There is no commercially viable means, however, to precisely control the timing of autoignition because in a four-stroke HCCI cycle the chemical kinetics involved in the autoignition timing have thus far proved too complex to easily harness. With this background, an overexpanded two-stroke HCCI cycle has been created [1]. This Technical Note shows how to utilize this new two-stroke HCCI cycle as a short cut to bring HCCI engines from laboratories to the market place immediately without seeking any new technology.

An Overexpanded Two-Stroke HCCI Cycle

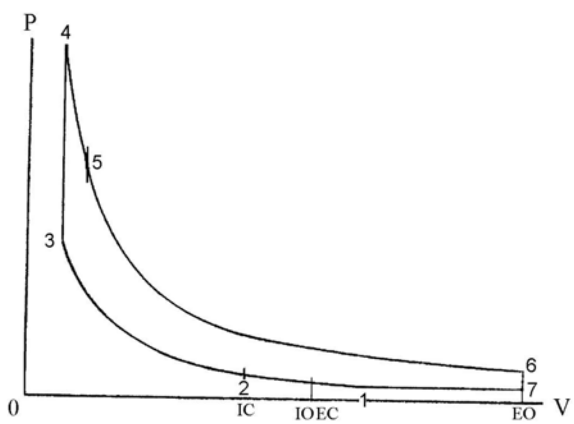

Figure 1: P-V diagram of an overexpanded two-stroke HCCI cycle.

Figure 1 presents a P-V diagram of an overexpanded two-stroke HCCI cycle that utilizes a three-stage fuel injection process (discussed in greater detail below). The compression process 1-2-3 has two parts. First, process 1-2 is performed in a separate air compressor with the entrance of the partially compressed lean homogenous mixture occurring at a point between points 1 and 2 indicated by IO. A Stage I fuel injection takes place in the separate air compressor to provide partly compressed lean homogeneous charge to the intake manifold.

In the second part of compression, a variable timing intake valve varies the closing timing at point 2 to control the engine compression ratio and thus the compression temperature at the end of the second part of compression process 2-3 to reach a temperature of 900°K (or other temperature just below the autoignition temperature). [2] Since the lean homogeneous charge enters the cylinder with a predictable temperature and because the very short duration of the compression process 2-3 (for pre-combustion chemical kinetic interaction), the required compression temperature T_3 at point 3 can be easily obtained regardless of engine rpm and load by controlling the timing of the intake valve closing.

Prior to the piston reaching TDC, a Stage II pilot fuel injection takes place, which pilot injection provides a "boost" in compression pressure and temperature (increasing temperature by approximately 100°K) of the existing compressed mixture sufficient to trigger autoignition of the lean homogenous charge at TDC. The equivalence ratio is selected to ensure that the autoignition combustion process 3-4 reaches (but does not exceed) a combustion temperature of 2000°K (or other temperature below the threshold temperature for the formation of NOx). Immediately after autoignition combustion of the lean homogenous charge, a Stage III fuel injection takes place to begin constant temperature combustion process 4-5.

The ensuing expansion process extends beyond V_1 to reach V_6 as shown in Figure 1. At point 6, the exhaust valve opens (indicated by EO) to begin a blow down process 6-7. Usually P_7 is less than one-half of P_6, proximately one-half of the exhaust gas escapes from the cylinder during the blow down process. An exhaust process begins when the piston reaches BDC and begins its upward motion and ends when the exhaust valve closes at EC. The intake valve opens simultaneously with the closing of the exhaust valve to admit partially compressed homogeneous charge from the separate compressor into the cylinder, to start the next cycle.

To summarize, the overexpanded two-stroke HCCI cycle is designed with key compression and combustion temperature parameters in mind. First, the autoignition temperature of hydrocarbon fuel is between 900° – 1000°K. Second, the threshold temperature at which NOx forms is generally believed to be between 1800° – 2100°K. [2] The three-stage fuel injection process that has been designed to accommodate these key temperature parameters as follows:

- Stage I: Fuel is injected under low-pressure into a separate air compressor to provide a partially compressed lean homogeneous charge upon intake to the engine cylinder. Once admitted to the cylinder, the partially compressed lean homogeneous charge is further compressed by the upward movement of the piston to reach a compression temperature of approximately 900°K (or other appropriate temperature selected to be just below the temperature at which autoignition would be triggered).

- Stage II: A small amount of fuel is injected as a pilot injection just prior to the piston reaching top dead center ('TDC') in order to trigger autoignition when the piston reaches TDC. The purpose of this pilot injection is to increase the compressed mixture temperature by approximately 100°K, which increase should be sufficient to trigger autoignition.

- Stage III: Following autoignition combustion of the lean homogeneous charge, a third injection of fuel is made in an amount necessary to achieve additional combustion at a constant temperature 2000°K (or other selected temperature) to generate additional power without producing NOx.

The key to controlling autoignition is the predictability of the in-cylinder temperature of the compressed homogenous charge made possible by the characteristics of an overexpanded two-stroke engine cycle. More specifically, for an engine operating on the new cycle, the crank angle from when the intake valve is closed to TDC is less than 90°, i.e., the compression process is extremely short in duration. In addition, the homogeneous charge entering the cylinder at intake has a predictable temperature. Lastly, the rate of chemical kinetics at temperatures below 900°K is negligible. Accordingly, it is reasonable to expect that obtaining the required compression temperature of 900°K can be accomplished solely by controlling the compression ratio, which is controlled by varying the timing of the closing of the intake valve.

For the same amount of fuel injection per expansion stroke as for a four-stroke diesel engine:

$Q_{3\text{-}4\text{-}5}$ = 720 Btu/lbm.

Starting at point 1 of Figure 1:

V_1 = 15.6 cubic feet

P_1 = 14.7 psia, and

T_1 = 311ºK.

The first injection of fuel takes place in the separate air compressor with:

$Q_{1\text{-}1a}$ = 308 Btu/lbm.

An overall compression ratio of 14.0 is chosen to achieve the required condition of T_3 equal to 893.7ºK (slightly below 900ºK or other autoignition temperature). The second pilot fuel injection of 30.9 Btu/lbm takes place just before the piston reaches TDC. At point 3:

V_3 = 1.11 cubic feet

P_3 = 591 psia,

autoignition combustion process 3-4 takes place. At point 4:

$T_4 = T_3 + (308 + 30.9)/C_v = 2000$ºK,

V_4 = 1.11 cubic feet

$P_4 = P_3T_4/T_3 = 1323$ psia.

From point 4 to point 5, the third stage of fuel injection begins at point 4 under a constant limiting temperature of 2000ºK with $Q_{4\text{-}5}$ equal to 381.1 (720 - 338.9) Btu/lbm,

Procedure For Obtaining A Constant-Temperature Combustion Process 4-5

The key to achieving constant-temperature combustion is to coordinate the rate of heat addition with volume expansion. A simple procedure for calculating the timing and rate of heat addition (in the form of multiple fuel injection pulses) necessary to achieve a constant-temperature combustion process of an air cycle has been developed. The computational steps for achieving a constant-temperature process 4-5 (beginning at V_4) are set forth in the following table:

Table 1

1	C_i	1	2	3	4	5	6	7	8	9
2	dQ_i		47.64	47.64	47.64	47.64	47.64	47.64	47.64	47.64
3	$T_{4,\,i}$	2000	2000	2000	2000	2000	2000	2000	2000	2000
4	$T'_{4,\,i}$		1845	1845	1845	1845	1845	1845	1845	1845
5	$V_{4,\,i}$	1.11	1.358	1.660	2.030	2.483	3.037	3.714	4.543	5.556
6	$P_{4,\,i}$	1323	1081	884.7	723.4	591.4	483.5	395.4	323.4	264.3
7	Re_j	17.56	14.36	11.75	9.61	7.85	6.42	5.25	4.29	3.51
8	$T_{6,\,j}$	635.7	688.9	746.5	809.0	877.2	950.6	1030.3	1117.0	1210.3
9	$P_{6,\,j}$	23.94	25.93	28.10	30.45	33.04	35.80	38.80	42.10	45.57
10	$T_{7,\,j}$	390.3	390.5	390.5	390.6	390.3	390.3	390.4	390.0	390.4
11	$Q_{7\text{-}6,\,j}$	75.58	91.9	109.4	128.9	150.0	172.6	197.1	223.9	252.5
12	$Q_{1\text{-}7,\,j}$	34.3	34.3	34.3	34.4	34.3	34.3	34.3	34.1	34.3
13	$Q^-_{,\,j}$	109.9	126.2	143.7	163.3	184.3	206.9	231.4	258.0	286.8
14	Q^+_j	338.9	386.7	434.5	482.3	530.2	578.0	625.9	673.7	721.6
15	ϕ_j	0.282	0.332	0.362	0.402	0.442	0.482	0.521	0.561	0.601
16	$Eff_{t,\,j}$	68%	67%	67%	66%	65%	64%	63%	62%	61%

Row 1 is the column number "C_i" with i equal to 2 to 9 for eight heat addition steps or fuel injection pulses to add 381.1 Btu/lbm at constant temperature of 2000° K. Row 2 "dQ_i" are eight equal heat addition steps or fuel injection pulses (381.1/8.0). Row 3 "$T_{4,\,i}$" are the constant limiting temperature of 2000° K. Rows 4, 5, and 6 are computed by using following equations (1) to (3) respectively:

$$T'_{4,\,i} = 2000 - dQ_i/C_v \qquad (1)$$

$$V_{4,\,i} = V_{4,\,i-1}(2000/T'_{3,\,i})^{2.5} \qquad (2)$$

$$P_{4,\,i+1} = P_{4,\,i}(V_{4,\,i}/V_{4,\,i+1}) \qquad (3)$$

Row 7 "Re_j" is the expansion ratios equal to $V_6/V_{4,\,j}$. with V_6 equal to 1.25 times V_1. Row 8 "$T_{6,\,j}$" is equal to $2000(Re_j)^{0.4}$ and Row 9 "$P_{6,\,j}$" is equal to $P_5(Re_j)^{1.4}$. Row 10 "$T_{7,\,j}$" is equal to $T_{6,\,j}(14.7/P_{6,\,j})$. Row 11 is the heat energy difference between states 7 and 6. Row 12 is the heat energy difference between states 7 and 1. Row 13 "Q^-_j" is the heat energy difference between states 6 and 1 or heat rejected with the exhaust gas. Row 14 "Q^+_j" is the total heat addition 338.9 $\pm$ dQ_j with j equals 2 to i. When j equals i, $V_{4,\,i}$ equals V_5. Each j value represents a heat addition process 3-4-5. For J equals 1, there is no constant temperature combustion portion and the new cycle becomes a two-stroke version of Atkinson cycle. Row 15 is the fuel equivalence ratio ϕ_j, obtained by dividing Row 14 by 1200 (Btu/lbm value for ϕ equal to 1). Row 16 $Eff_{t,j}$ is the cycle thermal efficiency equal to $(Q^+_j - Q^-_j)/ Q^+_j$.

The quantities of injection pulses are computed from dQ_i of Row 7. For the injection pulse timings, crank angles corresponding to $V_{4,\ i}$ of Row 5 are computed first based on the ratio between connecting rod length and crank radius. Then injection pulse timings are computed from crank angles and engine rpm.

For comparison, thermodynamic analysis of a four-stroke diesel cycle 1-2-3-3a-4-1 is carried out with the following parameters:

$Q_{2\text{-}3\text{-}3a} = 720$ Btu/lbm, and

$P_3 = 1000$ psia.

At point 1, assume:

$V_1 = 15.6$ cubic feet

$P_1 = 14.7$ psia, and

$T_1 = 311°K$.

At point 2 with a compression ratio of 16:

$V_2 = 15.6/16 = 0.975$ cubic feet,

$T_2 = T_1(V_1/V_2)^{k-1} = 942.8°K$, and

$P_2 = 713$ psia.

At Point 3:

$V_3 = V_2 = 0.975$ cubic feet,

$P_3 = 1000$ psia,

$T_3 = T_2(P_3/P_2) = 1322°K$, and

$Q_{2\text{-}3} = Cv(T_3 - T_2) = 116.8$ Btu/lbm

$Q_{3\text{-}3a} = 720 - 116.8 = 603.2$ Btu/lbm.

At point 3a:

$T_{3a} = T_3 + Q_{3\text{-}3a}/C_p = 2718°K$

$P_{3a} = P_3 = 1000$ psia, and

$V_{3a} = V_3(T_{3a}/T_3) = 2.0$ cubic feet.

Expansion ratio equals 15.6/2 = 7.8. At point 4:

T_4 = 1195°K, and

Q_{4-1} = 272.3 Btu/lbm.

The cycle efficiency is equal to (720 – 272.3)/720 = 62%.

Assume that the mechanical efficiency of the four-stroke diesel engine is equal 90% at 720 Btu/lbm of Q^+, the brake efficiency equals 0.9x0.62 = 0.56. Indicated power is proportional to 446.4 (the product of 720 and 0.62). Total friction loss is proportional to 0.10 times 446.4 = 44.64. The friction loss per unit indicated power is 44.64 divided by the product of Q^+ and the thermal efficiency. The mechanical efficiency of overexpanded two-stroke HCCI engine at various Q^+ values are tabulated in Table 2 below.

Table 2

1	Q^+_j	338.9	386.7	434.5	482.3	530.2	578.0	625.9	673.7	721.6
2	$Eff_{t,\,j}$	68%	67%	67%	66%	65%	64%	63%	62%	61%
3	$Eff_{m,\,j}$	81%	83%	85%	86%	87%	88%	88%	89%	90%
4	$Eff_{b,\,j}$	55%	56%	57%	57%	57%	56%	55%	55%	55%

The displacement ratio between the two-stroke and four-stroke engines equals (19.5-1.11)/(15.6-0.975) = 1.26. The brake power density ratio is equal to twice the brake efficiency ratio divided by the displacement ratio equals 2(55/56)/1.26 = 1.96. For the same displacement, the two-stroke engine produces 96% more power without additional mechanical losses. Therefore it has an equivalent brake efficiency of (0.55 + 0.96)/1.96 = 77% corresponding to a fuel saving of 36%.

For automotive truck application, an overexpanded two-stroke HCCI engine can be designed with Q^+ value of 338.9 Btu/lbm for homogeneous charge autoignition combustion to provide enough power to maintain a speed of 70 mph on highway. Because of homogeneous charge autoignition combustion, no NOx or PM is produced. For Q^+ values equal to 338.9 to 731.6 Btu/lbm, the combustion temperature has a constant limiting value of 2000^o K and no NOx is produced. The long combustion duration of constant temperature combustion will not have any combustible substance unburned. For quick acceleration and step slope climbing, the last dQ_i value can be increased to make Q^+ more than three times 338.9 to have the equivalence ratio approaching 1.0. Because of its very short duration at full power, the fuel efficiency and amount of NOx formation need not be a concern. Due to the fact that high brake efficiency is achieved throughout the whole range of power output, there is no need for a hybrid two-stroke HCCI engine.

Constant-Temperature Combustion - Engine Testing Program

There are plenty of lean HCCI autoignition engine experimental results showing that both NOx and PM can be minimized. Therefore it is only necessary to conduct engine experiments to verify that by coordinating fuel injection with cylinder volume expansion, constant temperature combustion can be achieved in a real engine. Existing engine experiment setup (either two or four-stroke engine) can be utilized for constant temperature combustion experiments. A testing engine must have a compression ratio high enough to reach a high compression temperature such that the ignition delay can be minimized. To begin with, a number of equal injection pulses calculated from 47.4 Btu/lbm and pound of air mass per cycle. The timings of injection pulses are computed from the crank angles and engine rpm. The crank angles are computed from the cylinder volumes and the ratio between the connecting rod length and crank radius. With this initial injection quantities and timings, the testing engine is run and indicator diagram is taken. A hypo bola is drawn through the beginning point of the indicator pressure curve. The crank angle division is increase where the indicator pressure is above the hypo bola and reduced where the indicator pressure is below. With the new injection pulse timings, the engine experiment is repeated. If the indicator pressure curve and hypo bola do converge, constant temperature combustion has been obtained in a real engine.

Conclusion

This Technical Note has described a new overexpanded two-stroke HCCI cycle engine. By operating on an overexpanded two-stroke HCCI cycle, the new engine will greatly reduce engine out NOx and PM. Overexpansion compensates for what would otherwise be a great reduction in thermal efficiency resulting from the prolonged combustion process. A unique two-stroke cycle utilizes the portion of the upward piston stroke corresponding to the extended expansion stroke for a shortened exhaust process. For a piston displacement volume ratio of 1.26, between the two-stroke and four-stroke engines, the power density of the two-stroke engine is increases by 58% (2/1.26) at full load. For the same displacement volume and thus the same total friction losses, the extra 58% power output without additional friction losses greatly increases the brake efficiency. As a result the fuel saving of 36% can be obtained throughout the whole range of power output. Since such drastic fuel savings depend on achieving constant temperature combustion in real engine, an engine experimental program has been proposed to verify whether constant temperature can be achieved in a real overexpanded two-stroke cycle HCCI engine.

REFERENCES

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