A COMPOSITE TWO-STROKE CYCLE AS A PLATFORM FOR DESIGNING A NEW CLASS OF INTERNAL COMBUSTION ENGINES

I. INTRODUCTION

To date, engine designers have failed to develop an internal combustion engine capable of meeting the twin requirements of (i) clean burning and (ii) more fuel efficient. Moreover, while engine researchers have developed new combustion modes, those modes have yet to yield commercially viable engines. To meet the twin goals of cleaner burning and more fuel efficient operation, a new composite two-stroke cycle has been created as a platform for designing a new class of internal combustion engines. This Technical Note describes the basic composite two-stroke cycle platform and several combustion modes operating on the new platform.

II. THE COMPOSITE TWO-STROKE CYCLE

A combustion and expansion process start simultaneously at the beginning of the down stroke of the new two-stoke cycle, with intake and exhaust valves closed. The burning of fuel/air mixture during a combustion process may take place in three possible modes depending on the engine design objectives. These three modes are constant volume, constant pressure and constant temperature combustions. Each combustion mode can be obtained by coordinating the rate of heat release with the cylinder volume expansion.

A combustion process is designed first to achieve the goal of minimizing engine emissions and/or maximizing fuel efficiency. The expansion ratio is chosen as a compromise between the thermal efficiency and mechanical efficiency as well as to create an expansion stroke that is longer than the compression stroke. The difference in stroke lengths is utilized for replacing cylinder exhaust gas with fresh charge. At the end of a down stroke, the exhaust valve opens to begin a blow down process. An exhaust process begins when the piston begins its upward movement (toward TDC). After the piston has covered a portion of an upward stroke, the intake valve opens before exhaust valve closing for a diesel engine to admit compressed air to scavenge the cylinder helping the piston to expel exhaust gases out through the exhaust valve. In the case of a gasoline engine, the opening of intake valve and closing of the exhaust valve take place at the same time. A compression process begins after the intake valve is closed. When the piston reaches TDC, the processes of the new two-stroke cycle are repeated. The compression process takes place only during the later portion of the upstroke while the beginning portion is utilized for replacing cylinder exhaust gas with fresh charge.

Thus, the combination of the down stroke and upstroke thermodynamic processes creates a composite two-stroke cycle consisting of (i) a constant-volume, constant-pressure, or constant-temperature combustion process or a combination of them, (ii) an overexpanded expansion process of an overexpanded engine cycle, (iii) a shortened exhaust process of a four-stroke engine cycle, (iv) which shortened process is compensated for by a scavenging process of a two-stroke engine cycle. Because of more thermodynamic processes are involved in a composite cycle, it can meet far more performance requirements than existing cycles.

III. APPLICATION OF THE TWO-STROKE CYCLE TO SEVERAL COMBUSTION MODES

An engine designed to operate on overexpanded two-stroke cycle has the same cylinder-piston assembly of a four-stroke engine but with different intake and exhaust valve timings. Using the new two-stroke composite cycle as a platform, an overexpanded two-stroke diesel cycle is obtained by having the "Diesel Combustion" as the combustion process. Likewise an overexpanded two-stroke Otto cycle is obtained by having a constant volume combustion process. An overexpanded two-stroke HCCI cycle is obtained by having a constant volume autoignition combustion process. Lastly, a low combustion temperature overexpanded two-stroke cycle is obtained by having a combustion process with a limiting combustion temperature.

A. An Overexpanded Two-Stroke Diesel Cycle For High Fuel Efficiency

When a composite cycle has a "diesel combustion process" it becomes an overexpanded diesel cycle as shown in Figure 1. The P-V diagram of the original diesel cycle is indicated by solid line for comparison. A separate air compressor takes in the ambient air of V_1, P_1, and T_1 and delivers the partially compressed air to the engine manifold at point 1a. The air is further compressed from point 1a to point 2 to reach V_2, P_2 and T_2. From point 2 to point 3, a combustion process initiated by fuel injection, first at constant volume and then at constant pressure to reach point 3a. From point 3a to point 4', an expansion takes place. At point 4', the exhaust valve opens to begin a blow down process 4'-5. The ensuing exhaust process ends at point EC when the exhaust valve closes. The intake valve opens before the closing of the exhaust valve to begin a scavenging process. When the intake valve closes at point 1a, the in cylinder compression process 1a-2 starts and the sequence of processes repeats.

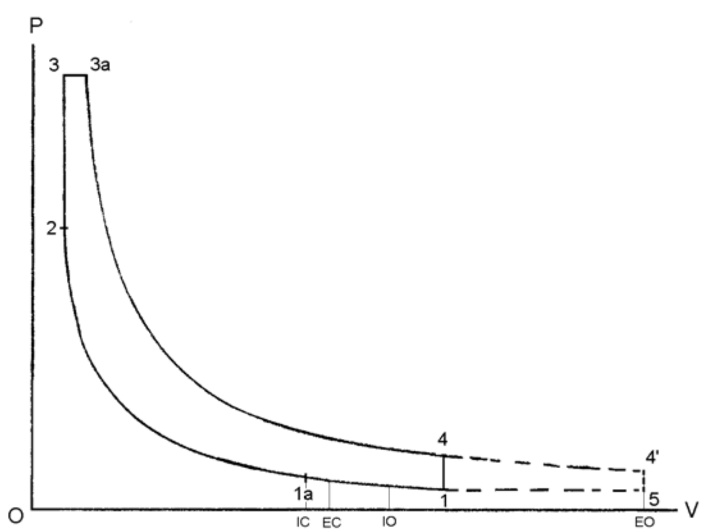

Figure 1: P-V diagram of an overexpanded two-stroke diesel cycle

Thermodynamic Analysis of an overexpanded two-stroke diesel cycle is carried out with $Q_{2\text{-}3\text{-}4}$=720 Btu/lbm and P_3=1000 psia. The air is considered as a perfect gas to represent the engine working fluid. The present thermodynamic analysis utilizes a formula based on heat energy balance (instead of mechanical work balance), in which thermo efficiency is computed in terms of heat addition Q^+ and heat removal Q^-, utilizing only the two basic equations; PV=RT and $T_2/T_1=(V_1/V_2)^{k-1}$. At point 1, assuming V_1=15.6 cubic feet, P_1=14.7 psia, and $T_1=311^o$ K. Let the overall- compression process 1-1a-2 has a compression ratio of 16 and the maximum combustion pressure equals 1000 and $Q_{2\text{-}3\text{-}3a}$=720 Btu/lbm. At point 2, V_2=15.6/16=0.975, $T_2=T_1(V_1/V_2)^{k-1}$=942.8, and P_2=713. At Point 3, $V_3=V_2$=0.975, P_3=1000, $T_3=T_2(P_3/P_2)$=1322, and $Q_{2\text{-}3}=C_v(T_3-T_2)$=116.8. $Q_{3\text{-}3a}$=720-116.8=603.2. At point 3a, $T_{3a}=T_3+Q_{3\text{-}3a}/C_p$=2718, $P_{3a}=P_3$=1000, and $V_{3a}=V_3(T_{3a}/T_3)$=2.0. Assume $V_{4'}=1.25V_4$ =19.5. Expansion process 3a-4' has an expansion ratio equal to $V_{4'}/V_{3a}$=9.75. At point 4', $P_{4'}$=41.2, $T_{4'}$=1093. At point 5, P_5=14.7 and $T_5=T_{4'}(T_5/T_{4'})$=390.0, $Q_{4'\text{-}5}=C_v$(390-1093)=-216.6, $Q_{5\text{-}1}=C_p$(311-390)=-34.1 and Q^-=250.7. The cycle efficiency=(720-250.7)/720=65%.

Given that the four-stroke and two-stroke cycles have the same combustion processes, the only difference between the two cycles is the expansion ratio of 7.8 (four-stroke cycle) as compared with 9.75 (two-stroke cycle). At point 4, T_4=1195, and $Q_{4\text{-}1}$=272.3. The cycle efficiency=62%. The piston displacement ratio=$(V_{4'}-V_3)/(V_1-V_2)$=(19.5-0.975)/(15.6-0.975)=1.27. The thermal efficiency ratio=65/62=1.05. The power density ratio=twice the thermal efficiency ratio divided by the displacement ratio=2x105/1.27=1.65, which is also equal to the mechanical efficiency ratio. The brake efficiency ratio is equal to the thermal efficiency ratio times the mechanical efficiency ratio=1.05x1.65=1.73. Therefore a two-stroke diesel engine designed to operate on an overexpanded two-stroke diesel cycle could reduce the specific fuel consumption by 43%.

B. An Overexpanded Two-Stroke Otto Cycle

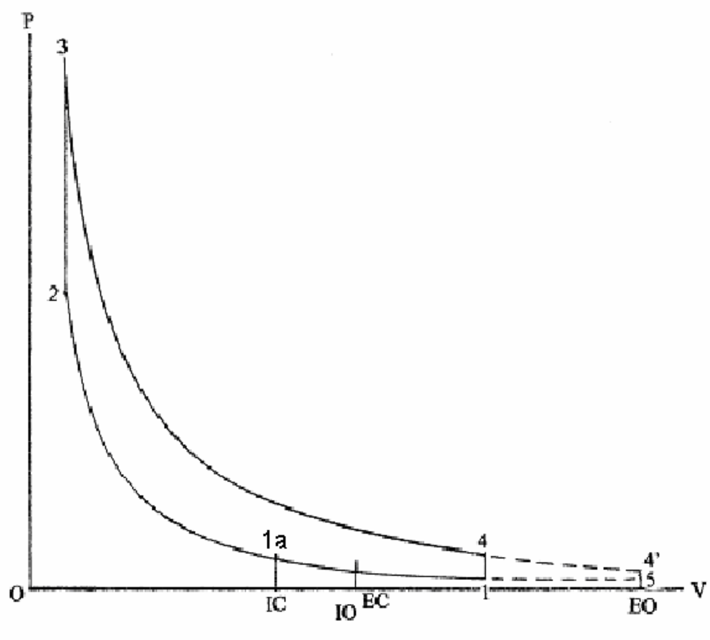

Figure 2: P-V diagram of an overexpanded two-stroke Otto cycle

Figure 2 is a P-V diagram of an overexpanded two-stroke Otto cycle. The original Otto cycle is shown by solid line for comparison. For spark ignition, the compression ratio of much smaller value of 8.5 is chosen. Process 2-3 is a constant volume combustion process. Assume $V_{4'}=1.25V_1$. The expansion process 3-4' has an expansion ratio of 8.5x1.25=10.63. The exhaust valve opens at EO and closes EC as shown. The intake valve opens at IO at the same point of EC without a scavenging process. A separate compressor provides a partial compressed fuel-air mixture to the engine manifold at a pressure equal to P_{1a}. The overall procession process 1-1a-2 has a compression ratio of 8.5.

At point 1, assuming V_1=15.6 cubic feet, P_1=14.7 psia, and T_1=311^o K. The overall- compression process 1-1a-2 is 8.5, and V_2=15.6/8.5=1.84. At point 2, $T_2=T_1(V_1/V_2)^{k-1}$=732 and P_2=294. At Point 3, $V_3=V_2$=1.84 For Q_{2-3}=1370 Btu/lbm $T_3=T_2$+1370/0.308=5180 and $P_3=P_2(T_3/T_2)$=2080. Expansion process 3-4' has an expansion ratio of 19.5/1.84=10.63. At point 4', $T_{4'}=T_3(V_{4'}/V_3)^{k-1}$=2015 and $P_{4'}$=76.2. At point 5, P_5=14.7 and $T_5=T_{4'}(P_5/P_{4'})$=388.7, $Q_{4'-5}=C_v$(388.7-2015)=-501, and $Q_{5-1}=C_p$(311-388.2)=-33.4. The cycle efficiency=(1370-501-33.4)/1370=61%.

Since the four-stroke cycle and two-stroke cycle have the same combustion processes, the only difference between the two cycles is the expansion ratio of 8.5 (four-stroke cycle) instead of 10.63 (two-stroke cycle). At point 4, $T_4=T_3(V_3/V_4)^{k-1}$=2200. $Q_{4-1}=C_v$(2200-311)=581.8. The cycle efficiency=(1370-581.8)/1370=58%. Piston displacement ratio=(V4'-V3)/(V1-V2)=(19.5-1.84)/(15.6-1.84)=1.28. The thermal efficiency ratio=61/58=1.05. The power density ratio=twice the thermal efficiency ratio divided by the displacement ratio=2x1.05/1.28=1.64, which is also equal to the mechanical efficiency ratio. The brake efficiency ratio equal to the thermal efficiency ratio times the mechanical efficiency ratio=1.05x1.64=1.72. For the same engine rpm, the torque ratio is also equal to 1.72. The exhaust valve timing controls the amount of the air/fuel mixture entering the cylinder and thus the engine output. A two-stroke gasoline engine designed to operate on an overexpanded two-stroke Otto cycle could be develop to reduce the specific fuel consumption by 42%.

C. An Overexpanded Two-Stroke HCCI Cycle

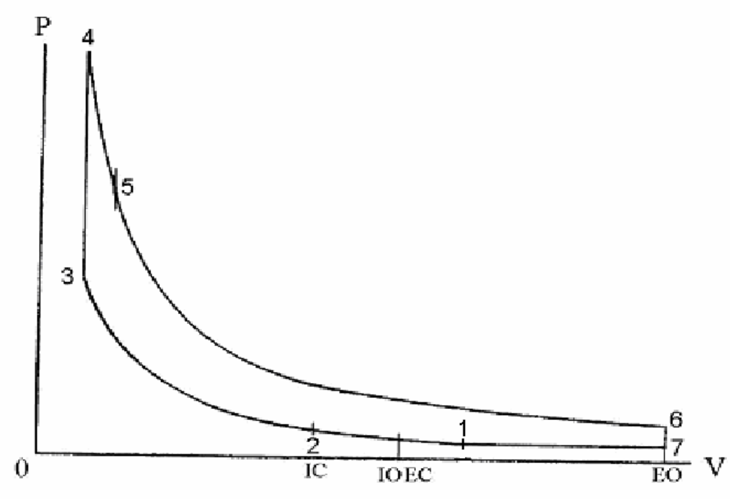

Figure 3: P-V diagram of an overexpanded two-stroke HCCI cycle

Figure 3: P-V diagram of an overexpanded two-stroke HCCI cycle

Figure 3 presents a P-V diagram of an overexpanded two-stroke HCCI cycle utilizing a three stages 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. The second part of the compression process 2-3 takes place in the engine cylinder (by the upward movement of the piston) to reach a compression temperature T_3 of approximately 900 K (selected to be just below the autoignition temperature).

A variable timing intake valve varies the closing timing at point 2 to control engine compression ratio and thus the compression temperature at the end of the second part of the compression process 2-3 to reach a temperature of 900 K (or other temperature just below the autoignition temperature). 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.

Before the piston reaches TDC, the 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 such that the autoignition combustion process 3-4 reaches (but does not exceed) a combustion temperature of 2000^o K (or other temperature below the threshold temperature for the formation of NOx). Immediately after autoignition and combustion of the lean homogenous charge, the 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 3. At point 6, the exhaust valve opens (indicated by EO) to begin a blow down process 6-7. Approximately 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.

The autoignition temperature of hydrocarbon fuel is between 900° – 1000° K. The value of the compression ratio should be such that the compression must reach a compression temperature of slightly less than 900 K and autoignition combustion at TDC is triggered by a small pilot fuel injection. On the other hand, the threshold temperature at which NOx forms is generally believed to be between 1800° – 2100° K. This example has a combustion process consists of a lean fuel/air mixture autoignition combustion to limit the combustion temperature 2000^o K. These key temperature parameters provide the framework for the following three-stage fuel injection process:

- 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') so as 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 is sufficient to trigger autoignition. The appropriate equivalence ratio is selected so that the combustion temperature reached after autoignition of the homogenous charge is approximately 2000° K (or other selected temperature that is sufficiently low to avoid the formation of NOx).
- Stage III: Following autoignition combustion of the lean homogeneous charge (which combustion temperature is below the threshold temperature for the formation of NOx), 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) in order to generate additional power without producing NOx.

The key to controlling autoignition is the predictability of in-cylinder temperature of the compressed homogenous charge made possible by the high performance two-stroke engine. Specifically, for the new engine, the crank angle from when the intake valve is closed to TDC is less than 90-degrees. The homogeneous charge enters the cylinder upon intake at a predictable temperature. Because of the extremely short duration of the compression process together with the negligible rate for chemical kinetics at temperatures below 900^o K, obtaining the required compression temperature value of 900^o K can be accomplished solely by controlling the compression ratio by varying the timing of the closing of the intake valve.

A small pilot injection sufficient to increase the compressed mixture temperature by approximately 100^o K is made just prior to the piston reaching TDC timed to trigger autoignition combustion at TDC. Lastly, for additional power output, the third stage of fuel injection occurs on the heels of the autoignition combustion (of the lean homogenous charge) to achieve additional combustion at a constant limiting temperature of 2000^o K. Achieving constant temperature combustion requires a simultaneous large volume increase, which increase reduces thermal efficiency because of a reduced effective expansion ratio. This reduction, however, is offset by the increased engine output, which also increases engine power density and thus results in increased mechanical efficiency.

For the same amount of fuel injection per expansion stroke as for a four-stroke diesel engine, Q_{3-4-5}=720 Btu/lbm. Starting at point 1 of Figure 2, V_1=15.6, P_1=14.7, and T_1=311º K. The first fuel injection takes place in the separate air compressor with Q_{1-1a}=308Btu/lbm. An overall compression ratio of 14.0 is chosen to achieve the required condition of T_3=893.7º K (slightly below 900º K). 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, P_3=591, autoignition combustion process 3-4 takes place. At point 4, $T_4=T_3+(308+30.9)/C_v$ =2000º K, V_4=1.11, $P_4=P_3T_4/T_3$=1323. From point 4 to point 5, the third stage fuel injection begins at point 4 under constant limiting temperature of 2000º K with Q_{4-5}=720-338.9=381.1 Btu/lbm approximated by five equal parts, Q_{4-4a}, Q_{4a-4b}, Q_{4b-4c}, Q_{4c-4d}, and Q_{4d-5} each part with a heat addition of 76.22 Btu/lbm. At point 4a, the adiabatic temperature T'=2000-76.22/0.308=1753, $V_{4a}=V_4(2000/1753)^{2.5}=1.39V_4=1.53$. At point 4b, $V_{4b}=1.39V_{4a}=2.13$ At point 4c, $V_{4c}=1.39V_{4b}=2.96$. At point 4d, $V_{4d}=1.39V_{4c}=4.11$. At point 5, $V_5=1.39V_{4d}=5.72$. $P_5=V_4(P_4/V_5)=254.4$. Expansion ratio of expansion process 5-6 is equal to 19.5/5.72=3.41 At point 6, V_6=19.5, P_6=45.8, T_6=1224.4º K. At point 7, $T_7=T_6(14.7/45.8)$=393º K, Q_{6-7}=0.308(393-1265.5)=-256.1, Q_{7-1}=0.432(311-393)=-35.4 Q^-=256.1+35.4=291.5 and Q^+=720. The cycle efficiency=(720-291.5)/720=59.5%.

The overexpanded two-stroke HCCI cycle is compared with the four-stroke diesel cycle of Figure 1, the piston displacement ratio=$(V_6-V_3)/(V_1-V_2)$=(19.5-0.975)/(15.6-0.975)=1.27 and the thermal efficiency ratio is 59.5/62=0.96. The power density ratio= twice of the thermal efficiency ratio divided by the displacement ratio=1.92/1.27=1.51, which is also equal to the mechanical efficiency ratio. The brake efficiency ratio is equal to the thermal efficiency ratio times the mechanical efficiency ratio=0.96x1.51=1.45. For the same engine rpm, the torque ratio is also equal to 1.45. The exhaust valve timing controls the amount of the homogeneous charge entering the cylinder and thus the engine output. A two-stroke diesel engine designed to operate on an overexpanded two-stroke HCCI cycle would reduce the specific fuel consumption by 31%.

For an HCCI engine, fuel and air is premixed as in a S.I. engine and combustion is initiated by compression as in a C.I. engine. However, the combustion process of an HCCI engine has a unique feature. Because the whole mixture is burned simultaneously, there is no post combustion temperature gradient to produce hot temperature zones for NOx formation. Therefore, a combustion process with combustion pressure equal to or less than the compression pressure will not produce post combustion temperature gradient because there is no combustion pressure rise. As a result the same benefits of an HCCI engine can be achieved.

D. An Overexpanded Two-Stroke Low-Temperature Cycle

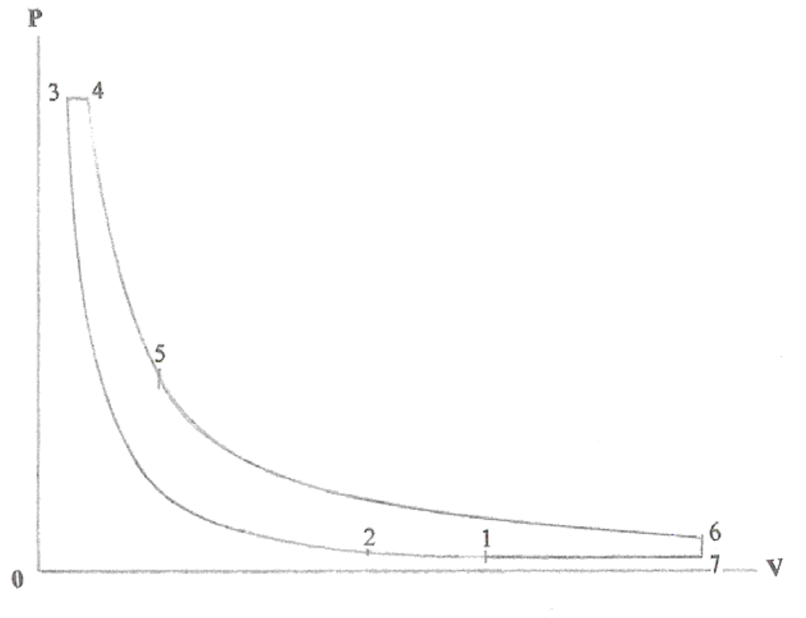

Figure 4: P-V diagram of an overexpanded limited temperature two-stroke diesel cycle

Figure 4 shows a P-V diagram of an overexpanded limited temperature two-stroke diesel cycle. At point 1, V_1=15.6, P_1=14.7, and T_1=311° K. Process 1-2 is a compression process (by air compressor C with a pressure ratio of 1.5 for discussion purpose). P_2=22.05, T_2=349.3° K and V_2=11.7. At point 3, assuming V_3=0.975 (for an overall compression ratio of 16), then P_3=713, T_3=943.2° K. At point 4, P_4=P_3, T_4=2000° K. The temperature increase from point 3 to point 4 is divided into say five equal steps of (2000-943.2)/5=211.4. At point 3a, T3a=T3+211.4=1154.6, V_{3a}=V3(T_{3a}/T_3)=1.194, and $Q_{3\text{-}3a}$=C_p(1156.6-943.2)=92.19. At point 3b, T_{3b}=T_{3a}+211.4=1366.0, V_{3b}=1.413. $Q_{3a\text{-}3b}$=91.3. At point 3c, T_{3c}=1577.4, V_{3c}=1.632, and $Q_{3b\text{-}3c}$=91.3. At point 3d, T_{3d}=1788.8, V_{3d}=1.851, and $Q_{3c\text{-}3d}$=91.3. At point 4, T_4=2000, V_4=2.07, and $Q_{3d\text{-}4}$=91.2. Total heat addition between point 3 and point 4, $Q_{3\text{-}4}$=456.5 Btu/lbm.

$Q_{4\text{-}5}$=720-456.5=263.5 Btu/lbm approximated by five injection pulses, $Q_{4\text{-}4a}$=$Q_{4a\text{-}4b}$=$Q_{4b\text{-}4c,}$ =$Q_{4c\text{-}4d}$=$Q_{4d\text{-}5}$=263.5/5=52.7 Btu.lbm. At point 4a, T'=2000-52.7/0.308=1829, V_{4a}=V_4(2000/1829)$^{2.5}$=1.25V_4=2.59. At point 4b, V_{4b}=1.25V_{4a}=3.24. At point 4c, V_{4c}=1.25V_{4b}=4.05. At point 4d, V_{4d}=1.25V_{4c}=5.06. At point 5, V_5=1.25V_{4d}=6.33 P_5=V_4(P_4/V_5)=233.2. At point 6, V_6=19.5 Expansion ratio of expansion process 5-6=19.5/6.33=3.08. P_6=48.3, T_6=1275. At point 7, P_7=14.7, T_7=T_6(14.7/48.3)=388.0. $Q_{6\text{-}7}$=0.308(388.0-1275)=-273.2. $Q_{7\text{-}1}$=0.432(311-388.0)=-33.3. Q^-=277.8+33.7=306.5. Efficiency=(720-306.5)/720=57%

By dividing fuel injection into parcels, better fuel/air mixing is achieved. With less temperature difference between burning each parcel and the surrounding temperature, the entropy increase is less. Furthermore, with the combustion temperature kept at maximum during the later part of a combustion process, whatever combustible substance in the cylinder is burned Out. Having combustion pressure equal to or less than the compression pressure, no fuel element is forced into cylinder crevices for escaping burning. Therefore NHC is also minimized.

A Suggested Test Program

For engine experiments, crank angles corresponding to cylinder volumes are computed. Each fuel injection pulse takes place within each corresponding crank angle section. An indicator diagram is taken for the running testing engine. A horizontal line is drawn through point 2 (the end point of compression process 1-2). In each crank angle section, if there is area above the horizontal line, this section of crank angle is widened to eliminate that area. If there is an area under the horizontal line, that crank angle section is narrowed. Based on these modifications, a second test is run with the modified timing of the fuel injection pulses. The goal would be to repeat this trial and error procedure until an approximation of a constant pressure combustion process is obtained and the corresponding fuel injection pulse timings represent the fuel injection rate necessary to produce the constant pressure combustion. After this is done, the procedure for constant pressure combustion is repeated for combustion at constant temperature. A hyperbola is drawn from point 4. If there is area above the hyperbola, this section of crank angle is widened to eliminate that area. If there is an area under the hyperbola, that crank angle section is narrowed. Based on these modifications, another test is run with the modified timing of the fuel injection pulses until it converges.

IV. SUMMARY

The present Technical Note has described a new composite two-stroke cycle that serves as a platform for the creation of a group of new cycles for developing a new class of reciprocating engines. The overexpanded two-stroke diesel and Otto cycle engines can be obtained by modifying the intake and exhaust valve timings of the respective four-stroke cycle engines without altering their fuel systems. The significant reduction in the calculated sfc for both cases demonstrates the potential of these two new cycles to achieve significant increase in fuel efficiency. The discussion of the overexpanded two-stroke HCCI cycle and overexpanded low-temperature combustion diesel cycle shows the potential for minimizing engine emissions while simultaneously achieving a significant sfc reduction.