How the CCI Mechanism Enables Higher Efficiency

Limitations inherent in the standard crank-piston mechanism inhibit the adoption of higher efficiency cycles

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Abstract

A review of the classic idealized engine cycles leads to consideration of the limitations inherent in the ubiquitous slider-crank piston actuation mechanism. The compromises inherent in using the same cylinder for induction, compression, combustion, expansion and exhaust are discussed. The Compact Compression Ignition (CCI) engine avoids these compromises by distributing the processes between three closely coupled optimized cylinders. The losses inherent in gas transfer between cylinders are discussed and simulations, including those losses and finite combustion rates, indicate that the CCI engine design can adopt an efficient high pressure high temperature Brayton cycle.
Review of classic idealized thermodynamic cycles

The fundamental basis of high efficiency in heat engines is using high temperature for heat addition together with low temperature for heat rejection. To allow appropriate numerical comparison with similar structural and thermal/emission limits the following cycles all work within the same maximum pressure and temperature. Specifically the peak temperature is 10 times ambient ($T=10$) and the peak pressure is 200 times ambient ($P=200$). The working fluid is a perfect gas with a specific heat ratio of 7/5.

![Logarithmic Pressure vs. Volume](image)

Figure 1. Logarithmic Pressure vs. Volume

Figure (1) shows these ratios using logarithmic coordinates. In these coordinates distances correspond to ratios not differences. Lines of constant pressure, volume, temperature and entropy are straight with slopes of 0, $\infty$, -1 and -7/5 respectively.

The Otto Cycle (Nicolaus August Otto 1832-1891)

The practical application of this cycle is the automobile engine even though, as we shall demonstrate, there are better choices.
Figure 2a shows that this cycle comprises an isentropic compression, instant (constant volume) heat addition, isentropic expansion and gas discharge involving instant (noisy) release of pressure to atmosphere. The efficiency is

\[
\eta_o = 1 - \left( \frac{T}{p} \right)^{(2/5)}
\]  

(1)

The implied compression ratio would be \( P/T = 20 \) and the efficiency 70%. But premature spontaneous ignition of gasoline fuel in air reduces useful Otto engine compression ratios to about 12 so that the efficiency would be 63%. (In Otto’s time available fuels limited compression ratio to much lower values - early tests were at CR=2.5 [1].) Even so, the efficiency exceeded that of the contemporary Brayton engines. There are many phenomena which further reduce the 63% value: heat loss to the walls, friction in bearings and sealing rings, seal leakage and the combustion being far from instantaneous, involving chemical reactions and flame propagation. Together these effects reduce the gasoline engine’s actual efficiency to 35% or less.

The Diesel Cycle (Rudolf Christian Karl Diesel 1858-1913)

Figure 2b shows that this cycle comprises isentropic compression, constant pressure combustion, isentropic expansion and constant volume exhaust. With equal expansion and compression ratios it fits the regular piston movement and makes best use of the pressure limit. The efficiency is

\[
\eta_d = 1 - \frac{5}{7T} \frac{A^{7/5} - 1}{1 - A^{-1}}
\]  

(2)

where \( A = \frac{T}{p_0} \).

Using the same pressure and temperature limits the efficiency is 74%. However, the implied compression ratio is 44 which, for reasons to be discussed below, is beyond current practice.

The Atkinson Cycle (James Atkinson 1846-1914)

Figure 2c shows that this cycle [1], [2], [3] comprises isentropic compression, constant volume combustion (as for the Otto cycle) followed by isentropic expansion beyond the initial volume down to atmospheric pressure. The efficiency is
\eta_b = 1 - \frac{7}{5T} \frac{A - 1}{1 - A^{-3/5}}

Using the same peak pressure and temperature limits its efficiency is 75%. Again, the expansion ratio is greater than the compression but, unlike Brayton, Atkinson developed mechanisms which changed the piston motion so that the induction-compression stroke length is less than the expansion-exhaust stroke. Tests in 1888 by the Society of Arts in London [1] confirmed that an Atkinson gas engine could achieve higher efficiency than the Otto and it was awarded the gold medal. However, it did not prevail commercially because its extra size and complexity made it heavier and more expensive than the Otto engine.

In recent years some automobile engines have been said to use an Atkinson cycle. These engines do not modify piston motion but use intake valve timing to reduce the effective compression ratio - making it less than the expansion ratio. But the Atkinson engine only achieves an efficiency benefit over the Otto when its expansion is greater than the compression of the Otto. This valve timing device actually reduces the compression ratio and leaves the expansion ratio unchanged so the efficiency is reduced. The advantage of the modified timing has more to do with making better use of turbocharger boost than with the extra expansion promoted by Atkinson.

The Brayton Cycle (George Bailey Brayton 1839-1892)

Figure 2d shows that this cycle comprises isentropic compression, constant pressure combustion, isentropic expansion and constant pressure exhaust. The Brayton cycle makes the best use of a pressure limit. The efficiency is

\eta_b = 1 - \frac{1}{p^{2/7}}

Against the same peak pressure and temperature its ideal efficiency is 78%. However, the expansion ratio required is 97, a value beyond practicality in one chamber. Brayton recognized that a greater expansion ratio than compression ratio was needed and he therefore flowed the gas from a compression cylinder to a larger expansion cylinder. His engines [1] phased the piston motions 180 degrees apart. In hindsight one can state that this arrangement suffers either from extreme pressure loss in the flow between cylinders or, if timed so that a similar pressure exists during transfer, the useful expansion ratio is reduced to about 2. His ideal cycle efficiency was therefore limited to 24% and actual efficiency to less than 5% [1].

The Extreme Expansion Cycle

It is of at least academic interest to consider the extension of the Brayton cycle expansion to the point where the expanded gas reaches ambient temperature, as shown in figure 3a.
This extra expansion does extra work but to complete the cycle it is then necessary to use isothermal compression. Practical isothermal compression can be approached using a series of compressors with intercoolers. The expansion ratio then becomes the ‘even more unattainable’ 696.

However, as observed in [4], and figure 3b, the use of a recuperator to feed exhaust heat back into the compressed gas avoids the extreme expansion ratio. The efficiency of this ideal cycle, recuperated or not, is 85.5%. This extreme expansion cycle has no well-established name. Of all the above cycles it is the only one which utilizes all the exhaust energy. It therefore competes with the use of ‘add-on’ heat engines (sometimes called bottoming cycles or combined cycles) for exhaust heat energy recovery.

**Extreme Expansion Cycle Efficiency - first law method**

Considering the first law of thermodynamics the work done is the difference between the energy supplied and the energy rejected. So, for this cycle, the work done by unit mass of air per cycle is:

\[
\text{Work} = c_p (T_3 - T_2) - R T_1 \ln \left( \frac{p_1}{p_4} \right) \tag{5}
\]

Where \( T \) stands for absolute temperature and \( p \) stands for absolute pressure, \( c_p \) is the gas specific heat at constant pressure and \( R \) is the gas constant for air. The first term in the right hand side of equation (5) is the heat supplied and the second is the heat rejected by the isothermal compression. The thermal efficiency is therefore:

\[
\eta_e = \frac{\text{Work}}{\text{Heat}} = 1 - \frac{R T_1 \ln \left( \frac{p_1}{p_4} \right)}{c_p(T_3 - T_2)} \tag{6}
\]

But \( p_1 / p_4 = (T_3 / T_2)^{c_p/R} \) so that:

\[
\eta_e = 1 - \frac{T_1 \ln \left( \frac{T_3}{T_2} \right)}{(T_3 - T_2)} \tag{7}
\]
Extreme Expansion Cycle Efficiency - second law method

The second law of thermodynamics focuses on the rise in entropy caused by all actual processes (think friction!) and, equivalently, on the decrease in available energy - the potential for doing work in a given environment. All the cycles considered above are internally reversible. This means that no internal processes lose available energy (also called exergy) and the efficiency can therefore be calculated entirely based on the loss due to ‘failure’ to add the heat at infinite temperature and to reject it at environment temperature. This extreme expansion cycle rejects its heat at environment temperature - which incurs no available energy loss. The only loss that needs consideration is that due to the finite temperature of heat addition. The work obtained is therefore equal to the available energy supplied.

$$\eta_e = \frac{\text{Work}}{\text{Heat}} = \frac{\text{Available energy supplied}}{\text{Heat}} = \frac{c_p(T_3 - T_2 - T_1 \ln\left(\frac{T_3}{T_2}\right))}{c_p(T_3 - T_2)} = 1 - \frac{T_1 \ln\left(\frac{T_3}{T_2}\right)}{T_3 - T_2}$$ (8)

Carnot Cycle

Using air for a Carnot cycle requires both isothermal compression for heat rejection and isothermal expansion for heat addition. These processes are approachable in principle with multiple intercooling and reheat stages. There is an open cycle with the same peak pressure and temperature as in the above examples and with Carnot efficiency. It is similar to the extreme expansion cycle with recuperation. But there seems to be no corresponding Carnot cycle without a recuperating heat exchanger.

$$\eta_e = 1 - \frac{1}{T}$$ (9)

Idealized Cycles Summary

<table>
<thead>
<tr>
<th>Head</th>
<th>Efficiency</th>
<th>Compression Ratio</th>
<th>Expansion Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Otto</td>
<td>0.70</td>
<td>20</td>
<td>20</td>
</tr>
<tr>
<td>Diesel</td>
<td>0.74</td>
<td>44</td>
<td>44</td>
</tr>
<tr>
<td>Atkinson</td>
<td>0.75</td>
<td>20</td>
<td>44</td>
</tr>
<tr>
<td>Brayton</td>
<td>0.78</td>
<td>44</td>
<td>97</td>
</tr>
<tr>
<td>Extreme Expansion</td>
<td>0.85</td>
<td>44</td>
<td>696</td>
</tr>
<tr>
<td>Extreme Expansion - recuperated</td>
<td>0.85</td>
<td>6</td>
<td>97</td>
</tr>
<tr>
<td>Carnot</td>
<td>0.90</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Comments

1. The two cycles which ‘fit’ the standard piston-crank mechanism with its equal compression and expansion ratios have the lowest efficiencies. The prevalence of this mechanism reflects simplicity and small size rather than a thermodynamic optimum.
2. The ideal efficiencies are so much greater than achieved in practice that it brings into question the value of this analysis. One reason is that the compression and expansion ratios are higher than can in practice be achieved in a single cylinder. The reasons for this are discussed in the next section.
3. The higher the efficiency the greater the difference between the expansion and compression ratios.
4. Although not obvious from the above, all the cycles except the Otto and Brayton lose efficiency at lower heat addition (power) values. They all tend to the efficiency controlled by their compression ratio.
Idealized Turbocharged Engine

All larger diesel engines are now turbocharged. Adding the turbocharger dramatically reduces size and cost for a given power. There are aerodynamic design compromises involved in matching the compressor and turbine of a turbocharger and then more in linking it to a reciprocating engine required to operate across a wide range of speed and power. Figure (4) is a composite pressure volume diagram of a cylinder cycle and associated compression and expansion processes.

![Figure 4. Ideal Composite Pressure - Volume for Turbocharged Engine](image)

Pressure at 5 is the cylinder exhaust pressure and at 9 is the cylinder boost pressure. The 4-stroke cylinder operation is 9-2-3-4-5-8-9. The indicated work is represented by the areas marked A and C. The corresponding expander operates in the cycle 5-6-7-8-5. Its indicated work is represented by the areas B and D. The compressor operates in the cycle 1-2-9-7-1 with a work requirement indicated by the areas D and C. Because the expander and compressor work balance, the cycle has the efficiency of the Brayton but with much lower cylinder compression ratio and expansion ratio requirements. For a specific example with the same peak pressure and temperature and an arbitrary halving of the in-cylinder expansion ratio the pressure ratio across the turbine would be 2.639 and across the compressor would be 7.793. The required expansion ratio is now 48.5 and the compression ratio 10.2. In this ideal case the efficiency remains at the Brayton value of 0.78. So, turbocharging at high efficiency levels emphasises the need for the expansion ratio to be high relative to the compression ratio.

The effective compression ratio can be lowered by advancing or retarding the intake valve closing relative to the maximum volume event. This means that the valve is closing while the piston is moving and significant pressure drops occur across the valve. If boost levels increase, turbocharger efficiency increases or the extreme expansion cycle is adopted this requirement for lower compression ratio than expansion ratio becomes more significant, the loss using the valve closing approach increases, and so alternating stroke piston mechanisms or the CCI become appropriate for highest efficiencytaking advantage of turbocharging.

Limitations of the Conventional Piston Crank Mechanism

As shown above the only cycles which ‘fit’ the equal expansion and compression ratio occurring in the conventional cylinder are the Otto and Diesel. The geometry simply makes the others impossible. The reasons why the compression ratios used are much lower than these ideal values suggest are more subtle.

1. When the fuel is gasoline mixed with the ingoing air the mixture ignites early if the compression ratio is too high. For normal (regular) gasoline this limit is close to 10. To avoid this problem, and hence reach higher efficiency, the diesel engine injects the fuel when it is needed. Diesel fuel does not itself limit compression ratio (except that it must not be too low) but as compression increases the time within the cycle for the injection and mixing process to occur at maximum efficiency becomes shorter and the effectiveness of further compression is reduced. Specifically, suppose the
Combustion time (fuel quantity does not change) requires 30 degrees of crank angle and combustion starts at the minimum volume time. The expansion ratio remaining at the end of combustion is then low as shown in the table.

<table>
<thead>
<tr>
<th>Compression Ratio</th>
<th>Expansion Ratio at End of Combustion</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>6.25</td>
</tr>
<tr>
<td>16</td>
<td>8.00</td>
</tr>
<tr>
<td>44</td>
<td>11.34</td>
</tr>
<tr>
<td>97</td>
<td>13.00</td>
</tr>
</tbody>
</table>

2. The combustion chamber volume becomes smaller inversely with the increasing compression ratio. But perhaps more important than the reduction of volume is the change of shape. To keep the sides of the fuel sprays from direct injection away from the walls the combustion space must be deeper than a flat-topped piston allows. It is customary to use a ‘crater’ in the top of the piston. For illustration we will assume its depth is 1/3 rd of its diameter and that the stroke is 1.3 times the bore. The portion of the piston top outside this crater is known as the ‘squish’ region. As compression ratio increases the squish region starts to dominate.

<table>
<thead>
<tr>
<th>Compression Ratio</th>
<th>Crater Diameter/ Piston Diameter</th>
<th>Squish Surface Area/ Total Surface Area</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>0.76</td>
<td>0.19</td>
</tr>
<tr>
<td>16</td>
<td>0.64</td>
<td>0.32</td>
</tr>
<tr>
<td>44</td>
<td>0.45</td>
<td>0.56</td>
</tr>
<tr>
<td>97</td>
<td>0.34</td>
<td>0.71</td>
</tr>
</tbody>
</table>

The squish surface area is a source of heat loss. Near the time of minimum volume air is rapidly expelled from the squish region and then air - possibly with hot products - is returned rapidly over the lip of the crater. The velocities and areas add to the heat loss from the chamber.

3. To avoid the connecting rod impacting the side of the cylinder bore the maximum stroke/bore is about 1.3 and this value is assumed for the above calculations. Higher stroke/bore ratios require more complex and larger mechanisms but do reduce the need for ‘squish’. In fact, at a compression ratio of 16 and a stroke/bore ratio of 5 there is no ‘squish’ as the flat top piston has a clearance 1/3 of the bore. The ‘most efficient engines in the world’ are the long stroke marine diesels. They use a stroke/bore ratio of up to 4 which is possible because they use the bulky cross-head mechanism.

4. All mechanisms have inherent cyclic frequency limitations beyond which bearing or metal stresses cause failure. Interestingly, exact scale models of mechanisms built with the same materials have exactly the same stresses if they run at the same linear speed. So, for instance, if the dimensions are all halved, the operating frequency can be doubled with the same stress level. It follows that geometrically similar engines have torque proportional to linear dimension cubed and power proportional to linear dimension squared. So smaller engines rotate more rapidly. The ignition chemistry of fuels is not ‘aware’ of the engine size. Faster rotation helps the gasoline engine combustion by allowing more compression before auto-ignition. Faster rotation of the diesel engine hurts because the ignition delay between start of injection and combustion becomes a larger portion of the time available for combustion. This is the reason that the largest engines are compression ignited diesel and the smallest are spark ignited gasoline. Automotive engines are in a size range where both methods work - but they do not have the same efficiency!

The following section shows that a major re-arrangement of the engine offers a way to avoid all the above limitations.

Summary

The efficiency limitations of a 4-stroke cycle in a single cylinder can be summarized by:

1. Expansion Ratio is limited to Compression Ratio
2. Expansion and Compression Ratio are both limited by
   (a) Combustion chamber becoming a small crater with a dominant ‘squish’ region leading to extra air motion and excessive surface area
(b) Optimum time for combustion, near minimum volume, becomes too small relative to the cycle time.

3. Stroke to Bore ratio is a compromise between size and efficiency. Large ratios favor combustion chamber shape and high efficiency, but large ratios lead to less power from a given engine size or weight. This is increasingly the case when the ratio exceeds about 1.3, which is the limit to a simple compact piston crank arrangement. Small ratios lead to a more compact engine for the same power but less efficiency.
4. The higher rotation frequencies of smaller engines makes use of the (more efficient) diesel engines in the smaller sizes more difficult because the ignition delay of the fuel becomes large relative to the combustion time and makes combustion ‘harsh’ and noisy. Development of multi-injection per cycle fuel injection systems has helped make smaller diesel engines acceptably smooth and quiet.

**The High Efficiency Sequential Pistons Engine**

This new engine concept, first described in reference (5), takes the working gas in sequence through three cooperating cylinders. In figure 5, for each diagram, the induction cylinder is at the bottom, the combustion cylinder with opposed pistons is in the middle and the exhaust cylinder is at the top. It is suggested and confirmed by numerical simulation that this arrangement allows a practical approach to the Brayton cycle.

The Brayton (sometimes called the Joule) cycle is also the underlying ideal of the gas turbine. The gas turbine also features flow sequential from compressor to combustor and then expander.
Figure 5. Schematic of the Compact Compression Ignition (CCI) Engine Operation
**Cycle Description**

The following cycle description uses the position of the crankshafts to identify the sequence of processes. The crank angle is customarily measured from a zero at the time the combustion chamber (the middle cylinder) is at its minimum volume. This is the top left diagram of the figure 5 sequence. Following the gas through the engine starts at the opening of the intake valves at -90 degrees (or 270 degrees) crank angle. From -90 to +90 is an induction process with the induction cylinder going from minimum to maximum volume. The intake valves then close and compression takes place for 90 degrees. At that time (180 degrees crank angle) the transfer passages between the induction and combustion cylinders open. They stay open for 90 degrees while compressed gas transfers during further compression from the induction to the combustion cylinders. At this time the induction cylinder starts its next cycle while the combustion cylinder, for another 90 degrees, completes the compression with combustion at and around 0 crank angle. The expansion process is similar but in reverse. Until 90 degrees the combustion cylinder expands the gas. Then the transfer passage to the exhaust cylinder opens (this is also the time the inlet transfer closes - there is no overlap) and the passage remains open for 90 degrees while transfer of exhaust occurs during continuing expansion. Transfer stops with the expansion space at about half its volume. Depending on the fuel/air ratio and the relative cylinder sizes, some exhaust is retained in the combustion cylinder. From 180 degrees to 270 degrees exhaust expansion continues in the exhaust cylinder. At 270 degrees the exhaust cylinder is at maximum volume. For the next 180 degrees the exhaust is expelled. Although the whole process from start of intake to end of exhaust takes 2.5 revolutions each cylinder repeats its cycle every revolution.

**Table 2. CCI engine cycle sequence**

<table>
<thead>
<tr>
<th>Event</th>
<th>Crank Angle</th>
<th>Figure 6 Notation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Start of Induction</td>
<td>-90</td>
<td></td>
</tr>
<tr>
<td>End of Induction, Start of Compression</td>
<td>90</td>
<td>1</td>
</tr>
<tr>
<td>Start of Inlet Transfer</td>
<td>180</td>
<td>2,3</td>
</tr>
<tr>
<td>End of Inlet Transfer</td>
<td>270</td>
<td>4</td>
</tr>
<tr>
<td>Start of Combustion</td>
<td>360,0</td>
<td>5</td>
</tr>
<tr>
<td>End of Combustion</td>
<td></td>
<td>6</td>
</tr>
<tr>
<td>Start of Exhaust Transfer</td>
<td>90</td>
<td>7</td>
</tr>
<tr>
<td>End of Exhaust Transfer</td>
<td>180</td>
<td>8,9</td>
</tr>
<tr>
<td>End of Expansion, Start of Exhaust</td>
<td>270</td>
<td>10</td>
</tr>
<tr>
<td>End of Exhaust</td>
<td>90</td>
<td></td>
</tr>
</tbody>
</table>
In figure 6 processes 1 to 2 take place in the induction space but at 2-3 the relevant volume become the sum of the induction space volume and the combustion space volume. Ideally the transfer into the combustion space ends with a ‘zero’ volume in the induction space so that there no discontinuity in the compression process 3 to 4. The same is true for the expansion 5 to 6.

**Thermodynamic Advantages**

Consider in turn the efficiency limitations summarized above:

1. Because the expansion and compression use different cylinders it becomes possible to make the expansion cylinder larger than the compression cylinder so that the expansion advantage of the Atkinson or Brayton over the Otto and Diesel can then be realized.

2. By moving the compressed gas to a smaller diameter combustion cylinder with a geometric compression ratio of approximately 8, both the chamber shape and time available at optimum combustion conditions are improved.

3. The compromise in the bore/stroke ratio between size and efficiency is removed when the compression and expansion cylinders can use small stroke to bore ratios for small engine size while the combustion cylinder achieves a good shape for efficiency.

4. Modern small diesels, like their bigger brothers, are turbocharged. This requires the use of 3 or 4 cylinders operating in parallel. If instead, as in the CCI, the cylinders are operated in series the combustion can be in a larger cylinder with
a lower geometric compression ratio and hence a lower surface/volume ratio. The higher temperature at the start of combustion (but not necessarily at the end) and the longer period for combustion serve to make it easier to achieve smooth, relatively quiet, combustion in a smaller engine size. Also turbocharger performance deteriorates (Reynold’s Number effects) at small sizes while the corresponding effects are not felt in reciprocating components until they are much smaller.

**Structural Advantages**

1. In high pressure conventional engines the crankshaft bearing diameters approach the bore size to keep the bearing and crankshaft stresses acceptable. This is because the high pressure during the combustion is exposed to the same piston area as is used for induction. When, as in the CCI engine, the combustion occurs in a smaller diameter cylinder the required crank diameter is reduce accordingly. The maximum load magnitude is reduced by the square of the combustion to induction bore ratios.

2. In the CCI engine there are no connecting rods. This avoids entirely the troubles often encountered in 2-stroke engine small end connecting rod bearings due to absence of load reversal allowing a squeeze film mode of lubrication.

3. All oscillatory motions in the CCI engine are sinusoidal. Among other advantages is the fact that, unlike a conventional engine, complete balance can be achieved by weights on the counter-rotating crankshafts.

4. A further advantage of the counter-rotating shafts is that even though shaft speeds will fluctuated cyclically, as in all engines, the resultant change in moment of inertia is zero and even an idling single module unit will not rock on its mountings in the manner of an idling single cylinder conventional engine.

**Size Advantages**

1. The principal size advantage comes from the opportunity to use a very short stroke without the normally damaging effect on combustion chamber shape, surface area and piston stability. Rotation frequency can be higher so that induced air volume rate becomes larger than obtainable from longer stroke units.

2. Removal of the combustion (high pressure) chamber from the induction and exhaust cylinders allows a smaller crankshaft diameter and weight.

3. Conventionally, small size for a given power is achieved using multiple cylinders working in parallel and turbocharging. The CCI arrangement does not preclude using those methods. Parallel and turbocharged CCI modules engines present a dramatic opportunity for compact power.

**Efficient Gas Transfer Between Cylinders**

Of course all the above benefits assume that the gas exchange between the cylinders is achieved without incurring losses large enough to nullify the other benefits. Cylinders have been linked in series before. Double and triple expansion cylinders were used to enhance the efficiency of reciprocating steam engines in marine applications before the steam turbine. Brayton experimented with and sold reciprocating units with larger expansion than compression cylinder volumes. It is clear in hindsight that his arrangements, which all phased their pistons 180 degrees apart (like the steam expansion units) did not meet the requirements for efficient operation described below. The requirements can be stated as follows:

1. The duration of the transfer process must be long enough and the flow area large enough that the pressure loss arising from the flow should be small relative to the useful pressure differences developed by the engine.

2. Starting the gas exchange between two significant volumes at different pressure, leading to a ‘blow-down’ event, must be avoided.

3. Stopping the gas exchange should occur at a time when the flow is minimal.

4. The timing of the transfer process itself should not reduce the useful compression and expansion ratios inherent in the engine.
**Gas transfer in Brayton's engines**

In the historic Brayton engines the two cylinders were phased 180 degrees apart. With that phasing the only way to avoid excessive ‘blow-down’ is to make the transfer during the period corresponding to roughly half the stroke. The resulting useful expansion ratio is about 2 and efficiency cannot be high. So his engine fails on item 4.

**Gas transfer in the Scuderi engine**

An engine known as the Scuderi is currently being promoted. It, too, uses a pair of cylinders like the Brayton engines but their phasing is much closer together. Efficient transfer can occur starting when the combustion/expansion cylinder is at minimum volume and ending when the compression cylinder reaches its minimum volume. A compromise is needed between allowing sufficient time for the transfer to occur between the two minimum volume events and losing, as Brayton did, the useful expansion ratio needed for high efficiency.

**Gas transfer in the CCI engine**

Transfer from the induction to the combustion cylinder starts when the pressures are almost equal. Transfer occurs for 90 degrees of crank angle during which compression continues with both chambers reducing volume. The end of this transfer occurs when the induction cylinder reaches its minimum volume and the flow stops, so the closing corresponds to a no flow situation.

Transfer from the combustion cylinder to the exhaust cylinder takes place when the exhaust cylinder volume is minimal. So, if there is any pressure difference, and this can be adjusted by exhaust valve closing time; the flow needed to equalize pressures is very small and the losses accordingly are small. Transfer occurs during 90 degrees and during this time both cylinders are expanding but because the exhaust cylinder expansion rate is higher most of the exhaust gas leaves during this process. The end of the transfer occurs when the combustion cylinder is at maximum volume so that flow rate is again small and the loss due to closing the transfer port is minimal.

During both the compression and expansion processes the cylinders are cooperating to increase the compression or expansion ratio. Thus, these transfers serve to provide the benefits of two-stage processes and make high overall compression and expansion ratios practical.

It will be shown by the following simulations that the geometry permits port sizes to allow transfer without significant loss of efficiency.

**Simulation**

In 1995 thermodynamic simulations funded by DARPA through Caterpillar using comprehensive engine simulation technology at Lotus Engineering established in theory that the CCI concept could not only be more compact but also more efficient that conventional engines of similar power.

The author no longer has access to those comprehensive simulation tools. But the most important processes can be modelled using the ideal cycles as a guide and incorporating ‘reality’ using a quasi-steady one-dimensional compressible flow model for poppet valves and transfer ports together with heat release (combustion) rates that mimic direct fuel injection and diesel combustion rates. The resulting simulations are realistic enough to illustrate how the CCI overcomes the single cylinder limitations.
Comparison a naturally aspirated single cylinder (SC) conventional engine with a naturally aspirated single module CCI

The SC engine has equal compression and expansion ratios and can only approach the Otto or Diesel cycles. The more efficient Diesel requires very high compression ratio and combustion starting at minimum volume. Attempting that in practice, or by simulation, is limited by a conflict between the time required for injection and combustion and the very short time available near minimum volume. The result is that the highest efficiency is obtained at lower compression ratio with combustion starting before minimum volume.

Figure 7. Pressure vs. Crank Angle for a Simulated High Efficiency 4-Stroke Diesel Engine
Using the same simulation assumptions applied to the CCI the efficiency is higher and the engine size smaller for the reasons discussed above.
Figure 10. Logarithmic Pressure vs. Volume for 3 cylinders of a Simulated High Efficiency CCI Engine
When engines are scaled (i.e. all their linear dimensions are multiplied by the same factor) their power changes proportionally with the square of that linear scaling factor. So, for instance, a doubling of all dimensions leads to a four-fold power increase. This is a result of many factors including equal component stresses and mixing controlled combustion rates. The ratio between forcing frequencies from combustion and elastic resonant frequencies is also preserved. With this in mind the following comparison table between the SC and CCI simulation results for naturally aspirated engines includes only size-independent parameters. The simulations use the same simplifying assumptions, so the values of both efficiency and power are higher than can be obtained in practice, but the differences between the values are meaningful. The engines’ major dimensions are quoted relative to their respective induction piston bores.

Table 3. Comparison of size-independent parameters
Conclusions

1. Attempts to increase reciprocating engine efficiency using the conventional slider-crank mechanism face inherent barriers:
   (a) the more efficient cycles working within the same pressure and temperature limit require the expansion ratio to exceed the compression ratio and with this mechanism they are the same
   (b) high efficiency requires high temperature and hence pressure ratio, but at high pressure ratios the short time available within the cycle for injection and combustion prevents the best use of the pressure limit.
2. Although large expansion ratios with compact combustion chambers can be achieved with very large stroke/bore ratios the resulting engine becomes very large for its power. The “Compact Compression Ignition” concept offers a much more compact arrangement, enabling a high expansion ratio even with a higher power density.
3. A simple thermodynamic model suggests that the use of sequential cylinders in the manner of the CCI engine overcomes these barriers because:
   (a) separation of the induction and exhaust spaces allows the exhaust space to be larger and expansion more complete
   (b) separation of the combustion space from both induction and expansion spaces and the use of a geometric compression ratio of order 7 increases the combustion time within the cycle and makes an approach to constant pressure combustion feasible with normal or even longer injection duration
   (c) the surface to volume ratio of the combustion space is much more favorable than can be achieved when all the processes occupy the same cylinder.
4. The model includes one-dimensional compressible flow through the valves and transfer ports and shows that the losses due to the transfer processes do not negate the benefits of being able to optimize the chambers separately. Indeed the model performance approaches the thermal efficiency of the ideal Brayton cycle which in the case examined is 78%
The CCI engine shares with the gas turbine the sequential processing through separate compression, combustion and expansion components. It also shares with the gas turbine components exposed to hot flowing gas before expansion has fully cooled it. It seems likely that the speed and hence the eventual power of the CCI engine will depend on the extent to which the exposed components can be effectively cooled.

References