*Goering, Carroll E., Marvin L. Stone, David W. Smith, and Paul K. Turnquist. 2003 (rev. printing 2006). Thermodynamics and engine cycles. Chapter 4 in Off-Road Vehicle Engineering Principles, pp. 57-74. St. Joseph, Mich.: ASAE. Copyright American Society of Agricultural Engineers.*

# CHAPTER 4

# THERMODYNAMICS AND ENGINE CYCLES

## 4.1 Introduction

In this chapter, a brief engine history is presented to trace some of the thermodynamic ideas that are used in modern engines. The ideal gas law and polytropic compression/expansion law are reviewed as lead-ins to cycle analysis. Then the Otto cycle is presented as the ideal model for four-cycle SI engines. The dual cycle is presented as the ideal model for four-cycle CI engines. These models provide upper limits against which certain performance parameters of actual four-cycle engines can be compared. Finally, two-cycle SI and CI engines are discussed.

## 4.2 A Brief Engine History

Internal combustion (IC) engines, as the name implies, produce power through a combustion process occurring within the piston chambers. In contrast, combustion occurs outside the piston chambers in a steam engine–the combustion heats a boiler to produce steam and the steam is delivered to the piston chambers to produce power. One of the earliest experiments with IC engines occurred when Abbe Hautefeuille, a Frenchman, built a closed chamber in which he exploded gunpowder. The resulting high pressure was used to raise a water column that was in a connecting chamber. By 1680, a Dutch physicist named Huyghens replaced the water column with a piston, which would move when the gunpowder was exploded. Such engines of the exploding type were not very efficient, primarily because the gases in the combustion chamber were not compressed before ignition occurred.

More than 50 years later, in 1838, a scientist named Burnett pointed out the advantages of compressing the gases in the combustion chamber before ignition. Then, in 1862, *Beau de Rochas* set forth his famous four principles of operation for an efficient IC engine. They were:

1. The combustion chamber should have the smallest possible surface-to-volume ratio,
2. The expansion should take place as rapidly as possible,
3. The piston should have the longest possible stroke, and
4. The pressure should be as high as possible at the start of expansion.

The first two principles were aimed at reducing the heat loss to a minimum, while the second two were aimed at obtaining the maximum possible work from each power stroke.

*Nikolaus Otto* built on Beau de Rochas' principles, patenting his famous Otto-cycle engine in 1876. Two years later he built a successful IC engine. Otto was the first to use the four-stroke cycle, i.e., the intake, compression, power, and exhaust strokes that are still used in most IC engines today. With the expiration of the Otto patent in 1890, there was a spurt in development and commercialization of IC engines.

Otto's contemporaries thought that having only one power stroke in two crankshaft revolutions was a serious disadvantage and experimenters turned their attention back to two-stroke cycles of the explosive type–that is, without compression. In 1881, three years after Otto patented his engine, Sir Dugald Clerk built a two-stroke cycle engine with compression, but he abandoned it due to mechanical difficulties. Joseph Day simplified the design and, in 1891, he patented a two-stroke cycle engine that used a gas-tight crankcase as a pumping cylinder. Day's design is still used on modern two-stroke cycle engines of the spark-ignition type. By 1906, the Cushman Company was producing successful two-cycle, two-cylinder spark-ignition engines for use in farm tractors.

Dr. *Rudolph Diesel* patented his compression-ignition engine in 1892. The first diesel engine in the United States was built for the Busch brewery in St. Louis under license from Rudolph Diesel.

# 4.3 Four-Cycle Engine Analysis

As was mentioned in the previous section, the intake, compression, expansion and exhaust processes can be carried out in two or four piston strokes. Thus, there are two types of engines, those with *two-stroke cycles* and those with *four-stroke cycles*. For brevity, these are usually called two-cycle and four-cycle engines, respectively. *Four-cycle engines* have become much more popular than *two-cycle engines.* Thus, the reader can assume that all discussions in this textbook refer to four-cycle engines unless two-cycle engines are specifically mentioned.

## *4.3.1 Basic Thermodynamic Equations*

Two thermodynamic laws are used in cycle analysis, the ideal gas law and the polytropic compression law. The ideal gas law is

$$pV = MRT \tag{4.1}$$

where

$p$ = absolute gas pressure, kPa
$V$ = gas volume, $m^3$
$M$ = mass of trapped gas, kg
$T$ = absolute temperature of gas, K
$R$ = ideal gas constant = 8.314/mole wt of gas

In cycle analysis, changes in the condition of a gas are calculated rather than the absolute condition, and the gas constant drops out of the analysis.

The mass of gas in a combustion chamber remains essentially constant during the compression process. If the temperature also remained constant, Equation 4.1 indicates that the pressure would vary inversely with the volume. That is, the pressure would double if the volume was halved. Such a process is called an *isothermal* process. In an engine, the temperature increases considerably during the compression stroke; therefore, the pressure increases more than in an isothermal process. A *polytropic* process follows

$$pV^n = C_p \tag{4.2}$$

where $C_p$ = a constant and $n$ = an exponent between 1 and 1.4.

If the combustion chamber were perfectly insulated, so that there was no energy loss, then $n = k = 1.4$ and the process is called an *adiabatic* process. Thus, the use of k instead of n in Equation 4.2 indicates the process is an adiabatic process. For an isothermal process, $n = 1$. The value of n is always between 1 and 1.4 in an IC engine. It is usually a little above 1.3 on the compression stroke, but can fall to 1.26 or lower on the power stroke.

Suppose that a gas changed from state 1 at the beginning of a compression to state 2 at the end of the compression. Then, the ideal gas law can be used to show that

$$\frac{p_1 V_1}{T_1} = \frac{p_2 V_2}{T_2} \tag{4.3}$$

The polytropic law can be used to show that

$$\frac{p_2}{p_1} = \left(\frac{V_1}{V_2}\right)^n \tag{4.4A}$$

The pressure rises from $p_1$ to $p_2$ as the gas is compressed from volume $V_1$ to smaller volume, $V_2$. The ratio, $V_1/V_2$, is thus referred to as the *compression ratio*, r, i.e.,

$$r = \frac{V_1}{V_2} \tag{4.5}$$

Then Equation 4.4A can also be rewritten as

$$\frac{p_2}{p_1} = r^n \tag{4.4B}$$

By combining the ideal gas law and the polytropic law, it can be shown that

$$\frac{T_2}{T_1} = r^{n-1} \tag{4.6}$$

### 4.3.2 The Otto Cycle

The theoretical Otto cycle is shown in Figure 4.1. It includes intake from Point 0 to Point 1, adiabatic compression from Point 1 to 2, constant-volume heat input from Point 2 to 3, adiabatic expansion from Point 3 to 4, blow down from Point 4 to 1, and exhaust from Point 1 to 0. The volume, $V_2$, is called the *clearance volume*; it is the cylinder volume when the piston is at HDC. The cylinder *displacement* is the change in volume from Point 1 to 2, i.e.,

$$D_c = V_1 - V_2 \tag{4.7}$$

where $D_c$ = displacement of a single cylinder, L.

Although Equation 4.1 shows volumes measured in $m^3$, use of liters is permitted since we are working with ratios of volumes and engine displacements are generally measured in liters.

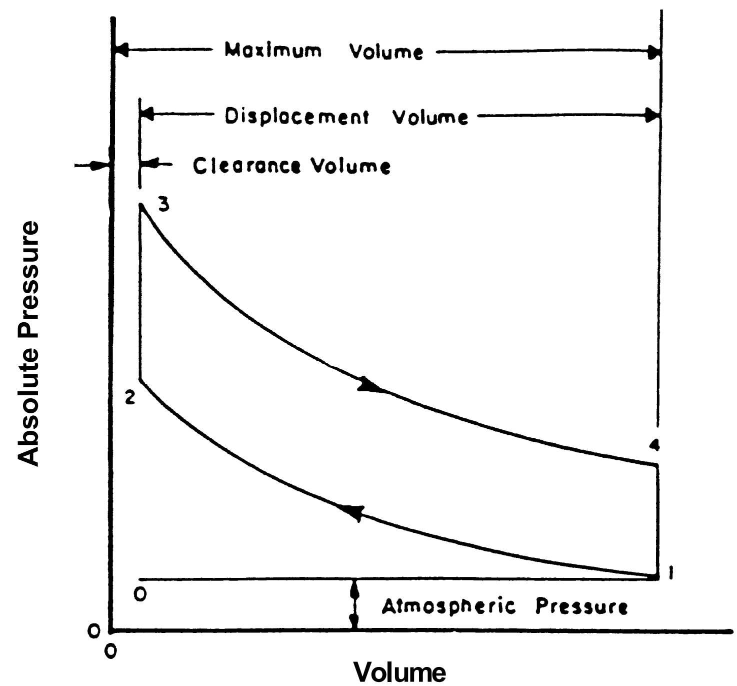


**Figure 4.1. The theoretical Otto cycle.**

In analyzing the Otto cycle, it is assumed that the conditions at Point 1 are atmospheric and thus $p_1$ and $T_1$ are given. It is also assumed that cylinder displacement, $D_c$, and compression ratio, r, are given. It can then be shown that

$$V_2 = \frac{D_c}{r-1} \tag{4.8}$$

and that

$$V_1 = \frac{r\,D_c}{r-1} \tag{4.9}$$

Finally, it can be shown that $T_3$ is the maximum temperature occurring in the cycle. Thus, $T_3$ is also specified as one of the givens for the cycle. The *adiabatic flame temperature*, $T_3$ = 2700 K, is usually specified as the given value for $T_3$. It is then possible to calculate the conditions at all Points in the Otto cycle, as shown in Table 4.1.

The work accomplished in the Otto cycle can be calculated by integrating the quantity p dv through the cycle. The results can be simplified to

$$W = p_1 D_c \left[ \frac{(r - r^k) + \frac{T_3}{T_1}(r - r^{2-k})}{(k-1)(r-1)} \right] \tag{4.10}$$

where
- W = work, J/cycle
- $p_1$ = initial pressure, kPa
- $D_c$ = displacement volume, L
- k = 1.4

By definition, the *cycle mean effective pressure*, $p_{cme}$, is

$$p_{cme} = \frac{W}{D_c} \tag{4.11}$$

where $p_{cme}$ = cycle mean effective pressure, kPa.

The cycle power is

$$P_c = \frac{p_{cme} D_c N_e}{2(60{,}000)} \tag{4.12}$$

where
- $P_c$ = cycle power, kW
- $N_e$ = crankshaft speed of the engine in which the cycle is implemented, rpm

**Table 4.1 Otto cycle summary.**

| Quantity | Point 1 | Point 2 | Point 3 | Point 4 |
|---|---|---|---|---|
| P, kPa | $p_1$ | $p_1\, r^k$ | $(T_3 / T_1)\, p_1\, r$ | $(T_3 / T_1)\, p_1\, r^{1-k}$ |
| V, L | $r\, D_c / (r-1)$ | $D_c / (r-1)$ | $D_c/(r-1)$ | $r\, D_c / (r-1)$ |
| T, °K | $T_1$ | $T_1\, r^{k-1}$ | $T_3$ | $T_3\, r^{1-k}$ |

The 2 in the denominator would become a 1 for a two-cycle engine. Finally, it can be shown that the cycle efficiency is

$$e_{Otto} = 1 - r^{1-k} \tag{4.13}$$

where $e_{Otto}$ = Otto-cycle efficiency, decimal.

The energy flows into and out of the cycle are

$$Q_{in} = \frac{W}{e_{Otto}} \tag{4.14}$$

and

$$Q_{out} = Q_{in} - W \tag{4.15}$$

where

$Q_{in}$ = energy into cycle, J/cycle
$Q_{out}$ = energy rejected from cycle, J/cycle

The Otto-cycle efficiency serves as a goal against which to measure indicated thermal efficiencies attained in actual spark-ignition (SI) engines running at full load. Also, $p_{cme}$ serves as a goal against which to measure achieved values of indicated mean effective pressure in SI engines running at full load. There are a number of simplifying assumptions that underlie the theoretical Otto cycle. These include zero friction, air as the only working fluid, zero heat transfer, constant-volume heat addition and constant-volume heat rejection. Also, as shown in Figure 4.2, an actual pV diagram includes pumping losses during the intake and exhaust stroke that are not included in the theoretical Otto cycle. All of these assumptions are violated to some extent in a running engine and thus the $e_{Otto}$ and $p_{cme}$ values calculated for the Otto cycle can never be achieved in practice.

### *4.3.3 The Classic Diesel Cycle*

Rudolph Diesel's original idea was to develop a compression-ignition engine that would burn the coal dust that was a waste product of that time. He soon found the coal dust to be unsuitable, but continued to develop his engine to burn liquid fuel. At the time that Diesel was developing his engine, he was well aware of the Otto cycle. The Diesel cycle differs from the Otto cycle only in that the process from Point 2 to 3 is a constant-pressure process rather than a constant-volume process. Let's try to reconstruct Diesel's rationale for substituting the constant-pressure process. Diesel realized his engine needed a compression ratio high enough to self-ignite the fuel at the end of the compression stroke. From Equation 4.6, by setting $T_2$ = 750 K (the self-ignition temperature of diesel fuel), taking $T_1$ = 300 K (a reasonable ambient temperature) and assuming n = 1.33, the compression ratio would have to be at least 16:1 to self-ignite the fuel. An even higher r would be required for lower values of $T_1$. From Equation 4.4B, assuming typical barometric pressure of 100 kPa, the pressure, $p_2$, at the end of the compression stroke would be 3994 kPa or higher. Then, if energy was released into the combustion chamber in a constant-volume process, the pressure

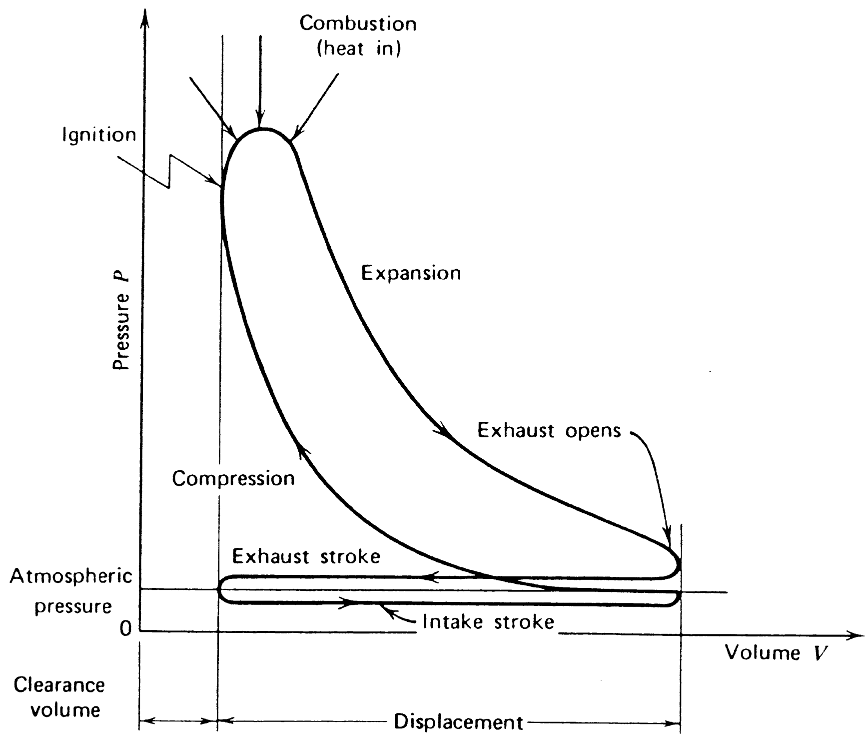


**Figure 4.2. Actual pV diagram of a spark-ignition engine.**

would rise much higher than 3994 kPa, i.e., higher than the engine could withstand. Diesel's solution was to release the energy in such a way that the pressure would rise no higher than the pressure $p_2$.

We will not analyze the theoretical diesel cycle because it is not a good model of the modern diesel engine. The dual cycle is a better model, because modern diesel engines have sufficient strength to allow some energy input at nearly constant volume. The theoretical dual cycle provides for part of the energy input at constant volume and the remainder at constant pressure. The dual cycle is a more general cycle, i.e., it includes the Otto cycle and the theoretical diesel cycle as special cases.

### *4.3.4 The Dual Cycle*

The dual cycle is illustrated in Figure 4.3. It is similar to the Otto cycle, except the process between Points 2 and 2a is constant-volume heat input, while 2a to 3 is constant-pressure heat input. Equations 4.4A through 4.9, 4.11, and 4.12 are all valid for the dual cycle. In analyzing the dual cycle, it is assumed that conditions at Point 1 are atmospheric and thus $p_1$ and $T_1$ are given. As with the Otto cycle, r, $D_c$ and $T_3$ are also given. In addition, it is necessary to specify the fraction of energy input that occurs at constant pressure, i.e.,

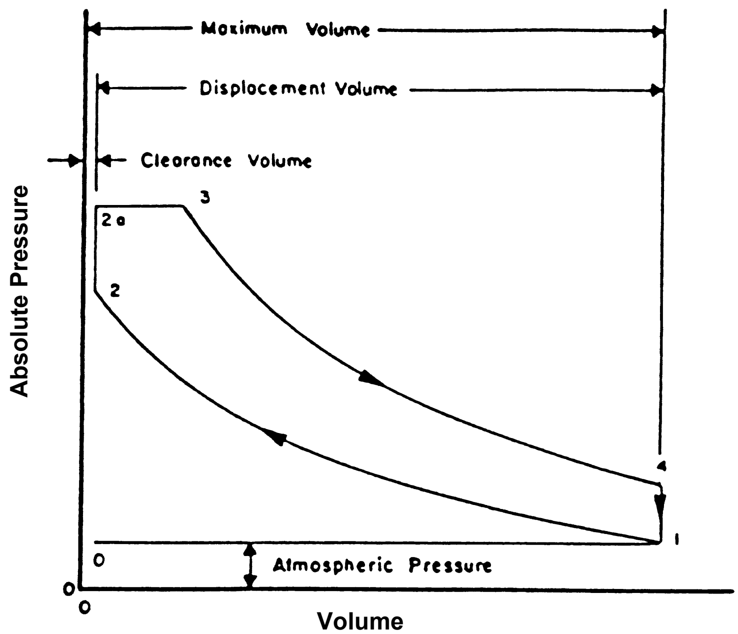


**Figure 4.3. The theoretical dual cycle.**

$$\beta_{dc} = \frac{Q_{inp}}{Q_{inp} + Q_{inv}} \tag{4.16}$$

where

$\beta_{dc}$ = fraction of energy input at constant pressure
$Q_{inp}$ = heat input at constant pressure, J/cycle
$Q_{inv}$ = heat input at constant volume, J/cycle

The *fuel cutoff ratio*, $r_{co}$, defines the volume at which energy input ceases and can be calculated as

$$r_{co} = \frac{u+1}{u + \frac{T_1}{T_3} r^{k-1}} \tag{4.17}$$

where $$u = k\left(\frac{1}{\beta_{dc}} - 1\right) \qquad \text{for } \beta_{dc} > 0 \tag{4.18}$$

In calculating conditions at all points in the dual cycle, note that the pressures, volumes and temperatures at Points 1 and 2 are identical to those in the Otto cycle. The other dual-cycle pressures are

$$\frac{p_{2a}}{p_1} = \frac{p_3}{p_1} = \left(\frac{r}{r_{co}}\right)\left(\frac{T_3}{T_1}\right) \tag{4.19}$$

and

$$\frac{p_4}{p_1} = \left(\frac{r_{co}}{r}\right)^{k-1}\left(\frac{T_3}{T_1}\right) \tag{4.20}$$

The volumes at Points 1, 2 and 4 are identical to those in the Otto cycle. The remaining volumes are

$$V_{2a} = V_2 \tag{4.21}$$

and

$$V_3 = r_{co}\, V_2 \tag{4.22}$$

The temperatures at Points 1, 2 and 3 are identical to those in the Otto cycle. The temperatures at the remaining Points are:

$$T_{2a} = T_3 r_{co}^{-1} \tag{4.23}$$

and

$$T_4 = T_3 \left(\frac{r_{co}}{r}\right)^{k-1} \tag{4.24}$$

The cycle mean effective pressure is

$$\frac{p_{cme}}{p_1} = \frac{r - r^k + T_r (r - r^{2-k} r_{co}^{k-1} + r\, r_{co}^{-1}(k-1)(r_{co}-1))}{(k-1)(r-1)} \tag{4.25}$$

where

$$T_r = \frac{T_3}{T_1}$$

The dual-cycle efficiency is

$$e_{dual} = 1 - r^{1-k}\left[\frac{\beta_{dc}\,(r_{co}^k - 1) + k(r_{co}-1)(1-\beta_{dc})}{k(r_{co}-1)}\right] \tag{4.26}$$

The dual-cycle efficiency increases from that of the theoretical Diesel cycle at $\beta_{dc} = 1$ to that of the theoretical Otto cycle at $\beta_{dc} = 0$, as shown in Figure 4.4. The fourth principle of Beau de Rochas is consistent with the efficiency being highest when $\beta_{dc} = 0$, i.e., when all of the energy is input at constant volume. The smaller the volume into which a given amount of energy is released, the higher the pressure resulting from that energy input, and thus, according to Beau de Rochas, the higher the efficiency. This change in cycle efficiency has practical implications for diesel engines, as will be explained in the next section.

The total heat into the dual cycle is

$$Q_{inp} + Q_{inv} = \frac{p_{cme} D_c}{e_{dual}} \tag{4.27}$$

and

$$Q_{out} = (Q_{inp} + Q_{inv}) - (p_{cme}\, D_c) \tag{4.28}$$

The Otto cycle and original Diesel cycle are special cases of the dual cycle. If $\beta_{dc}$ is nearly zero (Equation 4.18 fails if $\beta_{dc} = 0$), then Point 3 of Figure 4.3 moves to the left to Point 2a, $r_{co} = 1$, and $T_r = T_3/T_1$. Then the cycle mean effective pressure and the cycle efficiency become those of the Otto cycle. If $\beta = 1$, then Point 2 of Figure 4.3 moves upward to Point 2a and the cycle mean effective pressure and cycle efficiency become those of the original diesel cycle.

The theoretical dual cycle is based on many of the same simplifying assumptions that applied to the Otto cycle. All of the simplifying assumptions are violated to some extent in a running compression-ignition engine and thus the pressure and temperature values and the cycle efficiency can never be achieved in practice. However, these values serve as goals against which achieved values can be compared.

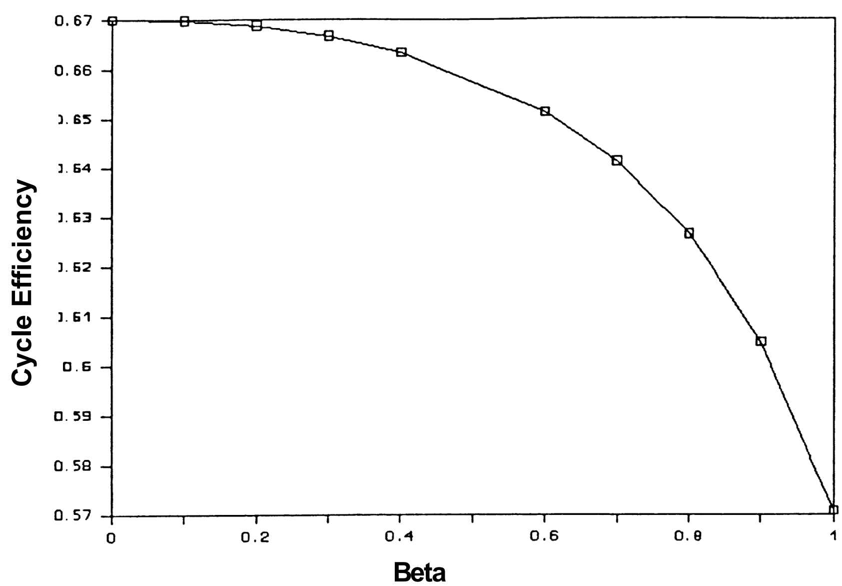


**Figure 4.4. Dual-cycle efficiency versus fraction of energy input at constant pressure ($r = 16$, $T_3/T_1 = 9$).**

# 4.4 Part-Load Efficiency of SI and CI Engines

To help provide a theoretical basis for the comparison of SI and CI engines, let us consider the brake thermal efficiency of engines, i.e., the fraction of the fuel equivalent power that is converted into power at the flywheel. By manipulation of equations in Chapter 2, the following equation for brake thermal efficiency can be derived:

$$e_{bt} = e_{it}(1 + \frac{p_{fme}}{p_{bme}})^{-1} \quad (4.29)$$

Let us consider how the two engines respond to changes in load when the speed of both is held constant. With constant speed, note that the $p_{fme}$ is constant for both engines, and that the $p_{bme}$ varies in direct proportion to the torque load on each engine. As $p_{bme}$ increases with increasing torque load on either an SI or a CI engine, the second term in Equation 4.29 increases, thus improving the brake thermal efficiency because of rising mechanical efficiency. Both engines have higher mechanical efficiency at full load and lower mechanical efficiency at part load. How does the indicated thermal efficiency vary in each of the engines as the load changes? To answer that question, we study the dual-cycle efficiency.

The best theoretical model for the indicated thermal efficiency is the cycle efficiency. The dual-cycle efficiency is given by Equation 4.26 and is plotted in Figure 4.4. As previously noted, $\beta_{dc}$ is the fraction of the fuel energy that enters the cycle at constant pressure while fraction $(1-\beta_{dc})$ enters at constant volume. The design of CI fuel-injection systems is such that injection begins at a fixed Point before HDC and the *rate* of injection cannot be changed when the engine load changes. Instead, the *duration* of injections must change to suit the load on the engine; the heavier the load, the later in the cycle the injections must terminate and the larger the value of $\beta_{dc}$. Conversely, smaller values of $\beta_{dc}$ correspond to higher values of cycle efficiency and, at $\beta_{dc} = 0$, the cycle efficiency rises to that of the Otto cycle. The dual-cycle efficiency equation thus predicts that CI engines increase their indicated thermal efficiencies as torque load declines. On the other hand, the theoretical Otto cycle predicts no change in indicated thermal efficiency when the load on an SI engine changes. Another prediction of Figure 4.4 is that the cycle efficiency of an SI engine would be higher than that of a CI engine if both were operated at the same compression ratio. As discussed in Chapter 5, however, engine knock considerations limit the compression ratios of SI engines to much lower values than those for CI engines. Thus, CI engines operate more efficiently than SI engines, both at full load and at part load. Part-load efficiencies are discussed in more detail in Chapter 7.

# 4.5 Two-Cycle Engines

As with four-cycle engines, two-cycle engines can be of either the SI or the CI type. Although there are many more SI than CI engines of the two-cycle type, both types will be discussed in the following sections.

## *4.5.1 Two-Cycle SI Engines*

The two-cycle engine shown in Figure 4.5 uses the crankcase as an air pump. Upward piston movement creates a vacuum in the crankcase, causing the spring-loaded check valve to open to admit air-fuel mixture into the crankcase. The check valve closes and the mixture in the crankcase is compressed as the engine moves on its down stroke. The piston uncovers the intake and exhaust ports in the cylinder walls during the down stroke. When the ports are uncovered, the compressed mixture in the crankcase rushes into the combustion chamber. A deflector on the piston top directs the incoming mixture towards the top of the cylinder. The incoming mixture helps to push the spent exhaust gases from the previous cycle towards the exhaust port. These flow processes continue while the piston moves to CDC (Crank Dead Center, also called Bottom Dead Center or BDC), reverses direction and starts upward. After the rising piston covers the intake and exhaust ports, the trapped mixture is compressed as the piston continues to rise. Near HDC, a spark plug fires to ignite the mixture and the resulting pressure increase forces the piston downward. The cycle is completed when the intake and exhaust ports are again uncovered. Use of the crankcase as an air pump interferes with its use as a sump for lubricating oil. Lubrication of the moving parts in this two-cycle SI engine is accomplished by mixing oil with the fuel.

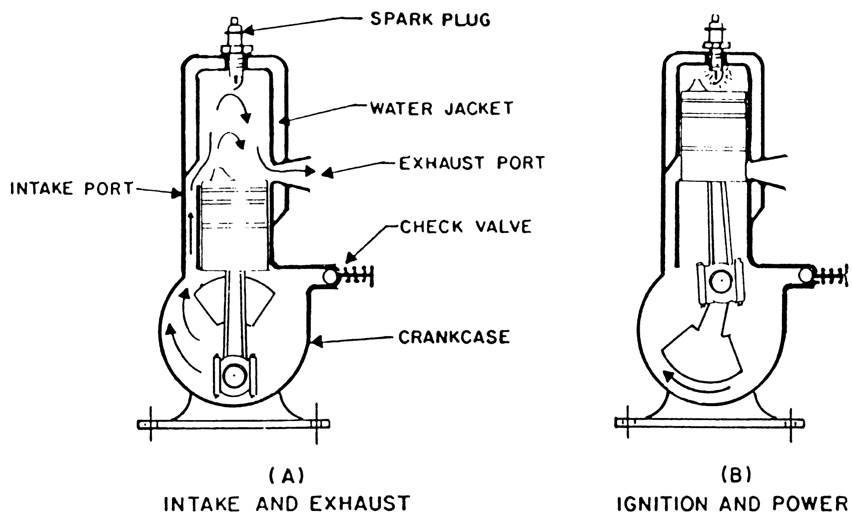


**Figure 4.5. Two-cycle spark-ignition engine strokes.**

The Otto cycle is not a good theoretical model for the two-cycle SI engine. The intake and exhaust ports restrict a portion of the piston travel to flow processes, rather than to the compression and expansion of a trapped fluid. The concepts of compression ratio and displacement involve the issue of what volume to use for $V_1$. One approach is to set $V_1$ equal to the trapped volume at the instant that the intake and exhaust ports are both covered by the rising piston. The definition of clearance volume is the same for two-cycle engines as for four-cycle engines. The ideal gas law and polytropic law are valid for any type of engine.

Two-cycle SI engines have a high power-to-size ratio. By producing a power stroke every crankshaft revolution, a two-cycle engine can produce more power than a four-cycle engine of the same physical size. On the other hand, two-cycle engines idle more erratically, have poorer fuel economy and can be more difficult to start than their four-cycle counterparts. Thus, two-cycle engines have been used primarily in applications in which their power-to-size ratio is an advantage, e.g., in chain saws, string trimmers, some lawn mowers, and outboard engines for boats. Some automobile manufacturers have developed prototype, two-cycle SI engines in which a mechanically-driven blower delivers air to the combustion chambers through conventional, cam-driven intake valves and exhaust is eliminated through conventional exhaust valves. Fuel is injected directly into the combustion chamber. These engines can use a conventional lubrication system, since the crankcase is available for use as an oil sump. The camshaft runs at crankshaft speed, while each spark plug fires once per crankshaft revolution. The reduced weight of these two-cycle SI engines is intended to help improve the fuel economy of automobiles. To date, however, these two-cycle automotive engines have not entered mass production.

### *4.5.2 Two-Cycle CI Engines*

A two-cycle CI engine is shown in Figure 4.6. A mechanically driven blower delivers air through an intake port in the cylinder wall. A conventional, cam-driven valve is used to expel exhaust gases. The crankcase is available as an oil sump and thus the engine can have a conventional lubrication system. Typical timing events are shown in Figure 4.7. When the piston is about halfway through its down stroke, the exhaust valve opens to release the exhaust gases and then the intake port is uncovered. Compressed air from the blower enters the combustion chamber and helps to sweep the residual exhaust gases from the chamber. The exhaust valve begins closing after CDC and, while it is closing, the rising piston covers the intake port. The trapped air is further compressed by the continued upward movement of the piston and fuel injection begins just before HDC. The injected fuel is ignited by compression and the resulting pressure increase forces the piston downward to complete the cycle.

Like its SI counterpart, the two-cycle CI engine has a high power-to-size ratio. The reduced weight is less of an advantage in the heavy-duty vehicles in which two-cycle CI engines are used. These engines have been popular as bus engines, where their small size permits them to fit in a compact engine compartment at the rear of the bus. They have also been used in some farm tractors and heavy-duty trucks. The added cost of the required air blower is a disadvantage of two-cycle CI engines, and they have been available in far fewer numbers than their four-cycle counterparts.

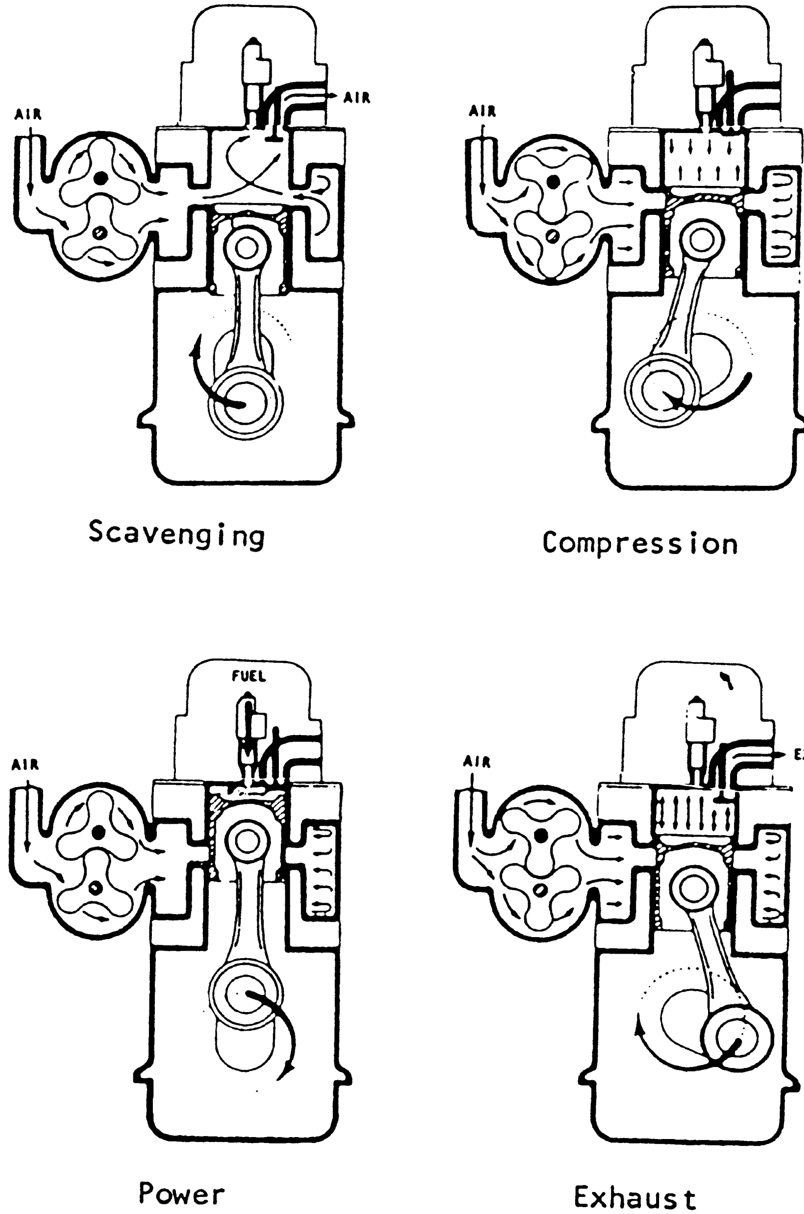


**Figure 4.6. Two-cycle compression-ignition engine strokes.**

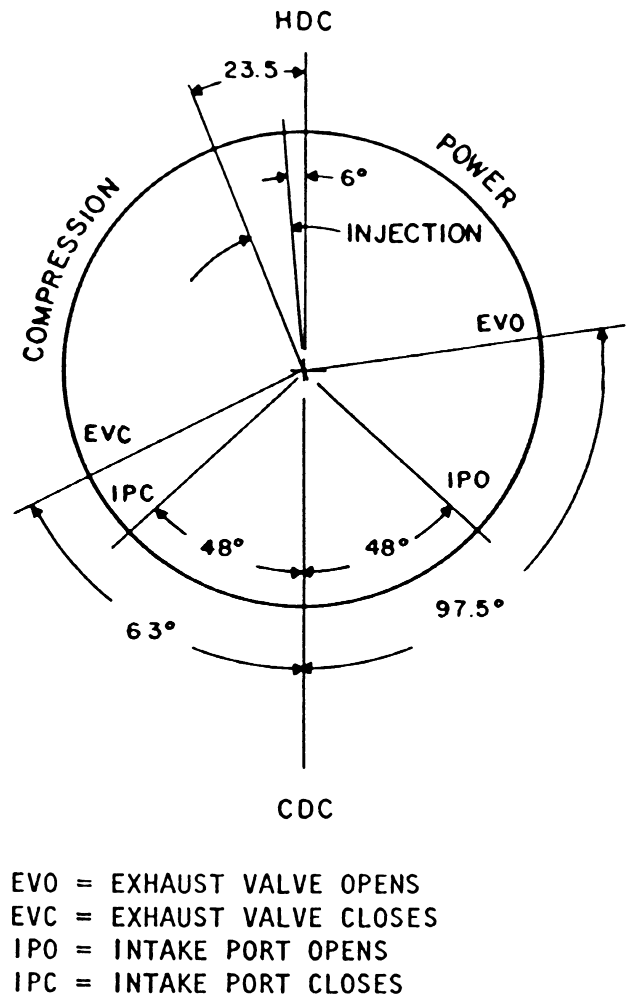


**Figure 4.7. Typical timing of a two-cycle compression-ignition engine.**

The dual cycle is not a good theoretical model for two-cycle CI engines. Also, as with their SI counterparts, there is some uncertainty as to what volume to choose for $V_1$ when calculating compression ratio and displacement. As shown in Figure 4.7, the piston can began compressing the trapped air when the exhaust valve fully closes, so $V_1$ could be taken as the volume at exhaust valve closure. Because of the blower, however, the trapped air at that Point is already partially pressurized.

# 4.6 Chapter Summary

In a brief history of engine development, the origin of some engine thermodynamic concepts was traced. The ideal gas law and polytropic compression/expansion law were reviewed and applied to the analysis of the Otto cycle and the dual cycle. The Otto cycle provides upper limit targets for certain performance parameters of actual four-cycle SI engines, while the dual cycle provides upper limit targets for four-cycle CI engine performance parameters. The dual cycle provided a basis for comparison of the indicated thermal efficiencies of SI and CI engines. Finally, two-cycle SI and CI engines were discussed.

### *Homework Problems*

4.1 Starting with Equation 4.1, derive an equation for the mass density of air. Then use the equation to calculate the density of air when the ambient temperature is 30°C and the barometric pressure is 97 kPa.

4.2 Starting with Equation 4.1, derive an equation for the mass density of air. Then use the equation to calculate the density of air when the ambient temperature is 15°C and the barometric pressure is 99 kPa.

4.3 Derive a new equation by solving Equation 4.4A for n. Then use the new equation to calculate the value of n when the pressure increases from 100 kPa to 4000 kPa as a gas is compressed to one-sixteenth its initial volume. By inserting a pressure transducer in the combustion chamber and using an encoder to measure crank angle, researchers can obtain pressure volume data and use it to determine the value of n in actual engines. For a given r, the value of n is normally larger for compression and smaller for expansion processes.

4.4 Rework Problem 4.3, but calculate the value of n when the pressure decreases from 1266 kPa to 100 kPa as a gas is expanded to 7.5 times its initial volume.

4.5 Verify all the entries at Point 2 in Table 4.1, i.e., if the quantity is not a given, derive the equation for calculating the quantity.

4.6 Verify all the entries at Point 3 in Table 4.1, i.e., if the quantity is not a given, derive the equation for calculating the quantity.

4.7 Verify all the entries at Point 4 in Table 4.1, i.e., if the quantity is not a given, derive the equation for calculating the quantity.

4.8 For the Otto cycle, plot $p_{cme}/p_1$ versus r for $2 < r < 12$ if $T_3 = 9\ T_1$. Note that increased compression ratios put more stress on the engine. What value of r would you recommend to minimize such stress while gaining most of the improvement in $p_{cme}$?

4.9 Rework Problem 4.8, but let $T_3 = 10\ T_1$.

4.10 Plot $e_{Otto}$ versus r for $2 < r < 12$ if $T_3 = 9\ T_1$. Note that simplifying assumptions underlying the Otto cycle are all violated in practice; thus, an actual spark-ignition engine cannot achieve the plotted efficiencies, but they can serve as an upper limit.

4.11 The following data were taken from a test of a four-cylinder SI engine running on gasoline: $N_e = 2250$ rpm, $\dot{m}_f = 10.8$ kg/hr, $H_g = 47,600$ kJ/kg, $D_e = 2.38$ L, r = 7.4 and $p_{ime} = 915$ kPa. Assume $p_1 = 100$ kPa.

(a) Calculate indicated power.

(b) Calculate fuel equivalent power.

(c) Calculate indicated thermal efficiency.

Now, assuming $T_3 = 9\ T_1$,

(d) Calculate theoretical Otto cycle efficiency.

(e) Calculate theoretical Otto cycle cycle mean effective pressure.

(f) Calculate theoretical Otto cycle cycle power.

(g) Compare the latter three quantities as upper limits to their corresponding measured quantities from the engine test.

4.12 The following data were taken from a test of a four-cylinder SI engine running on gasoline: $N_e = 2000$ rpm, $\dot{m}_f = 7.8$ kg/hr, $H_g = 47,600$ kJ/kg, $D_e = 2.59$ L, r = 7.5 and $p_{ime} = 705$ kPa. Assume $p_1 = 100$ kPa.

(a) Calculate indicated power.

(b) Calculate fuel equivalent power.

(c) Calculate indicated thermal efficiency.

Now, assuming $T_3 = 9\ T_1$,

(d) Calculate theoretical Otto cycle efficiency.

(e) Calculate theoretical Otto cycle cycle mean effective pressure.

(f) Calculate theoretical Otto cycle cycle power.

(g) Compare the latter three quantities as upper limits to their corresponding measured quantities from the engine test.

4.13 Plot $p_{cme}/p_1$ versus $\beta_{dc}$ for $0.1 < \beta_{dc} < 0.7$ assuming $T_3 = 9\ T_1$. Plot two curves on the same graph, one for r = 16 and one for r = 20. Note that the simplifying assumptions underlying the dual cycle are violated in an actual engine and thus a compression-ignition engine cannot achieve the $p_{cme}$ values plotted on your graph. The plots serve as an upper limit to performance.

4.14 Rework Problem 4.13, except plot curves for r = 14 and for r = 18.

4.15 Plot $e_{dual}$ versus $\beta_{dc}$ for $0.1 < \beta_{dc} < 0.7$ assuming $T_3 = 9\ T_1$. Plot two curves on the same graph, one for r = 16 and one for r = 20. Do the curves verify that increasing departures from constant-volume burning result in decreasing cycle efficiencies?

4.16 Rework Problem 4.15, but plot curves for r = 14 and r = 18.

## *References and Suggested Readings*

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