INTERNAL-COMBUSTION ENGINES

In a steam power plant, the steam is an inert medium to which heat is transfenred from a burning fuel or from a nuclear reactor. It is therefore characterized by large heat-transfer surfaces: (1) for the absorption of heat by the steam at a high temperature in the boiler, and (2) for the rejection of heat from the steam at a relatively low temperature in the condenser. The disadvantage is that when heat must be transferred through walls (as through the metal walls of boiler tubes) the ability of the walls to withstand high temperatures and pressures imposes a limit on the temperature of heat absorption. In an internal-combustion engine, on the other hand, a fuel is burned within the engine itself, and the combustion products serve as the working medium, acting for example on a piston in a cylinder. High temperatures are internal, and do not involve heat-transfer surfaces.

Burning of fuel within the internal-combustion engine complicates thermodynamic analysis. Moreover, fuel and air flow steadily into an internal-combustion engine and combustion products flow steadily out of it; no working medium undergoes a cyclic process, as does steam in a steam power plant. However, for making simple analyses, one imagines cyclic engines with air as the working fluid that are equivalent in performance to actual internal-combustion engines. In addition, the combustion step is replaced by the addition to the air of an equivalent amount of heat. In what follows, each internal-combustion engine is introduced by a qualitative description. This is followed by a quantitative analysis of an ideal cycle in which air, treated as an ideal gas with constant heat capacities, is the working medium.

The Otto Engine

The most common internal-combustion engine, because of its use in automobiles, is the Otto engine. Its cycle consists of four strokes, and starts with an intake stroke at essentially constant pressure, during which a piston moving outward draws a fuellair mixture into a cylinder. This is represented by line $0 \rightarrow 1$ in Fig. 8.8. During the second stroke $(1 \rightarrow 2 \rightarrow 3)$, all valves are closed, and the fuellair mixture is compressed, approximately adiabatically along line segment $1 \rightarrow 2$; the mixture is then ignited, and combustion occurs so rapidly that the volume remains nearly constant while the pressure rises along line segment $2 \rightarrow 3$. It is during the third stroke $(3 \rightarrow 4 \rightarrow 1)$ that work is produced. The high-temperature, high-pressure products of combustion expand, approximately adiabatically along line segment $3 \rightarrow 4$; the exhaust valve then opens and the pressure falls rapidly at nearly constant volume along line segment $4 \rightarrow 1$. During the fourth or exhaust stroke $(line\ 1 \rightarrow 0)$, the piston pushes the remaining combustion gases (except for the contents of the clearance volume) from the cylinder. The volume plotted in Fig. 8.8 is the total volume of gas contained in the engine between the piston and the cylinder head.

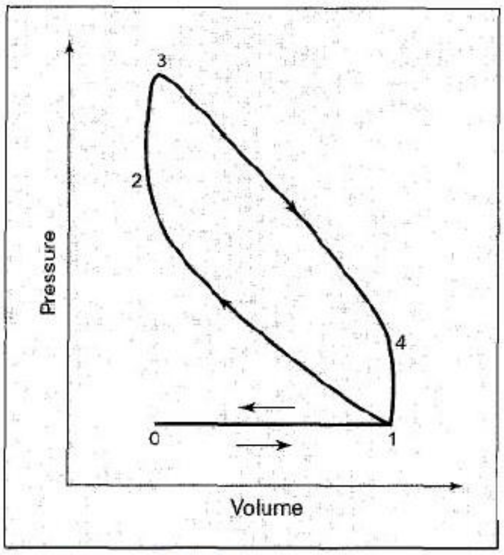

Figure 8.8 Otto engine cycle

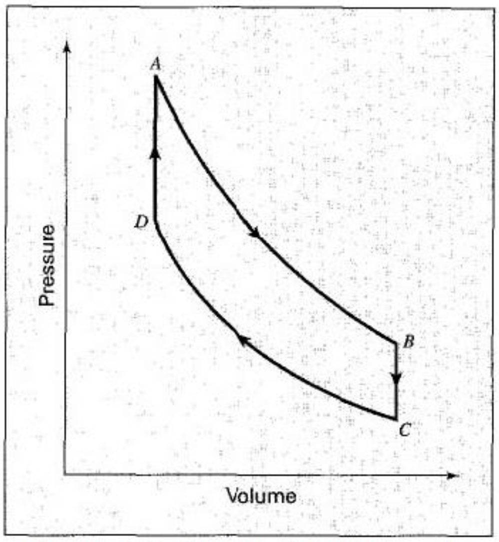

Figure 8.9 Air-standard Otto cycle

The effect of increasing the compression ratio, defined as the ratio of the volumes at the beginning and end of the compression stroke, is to increase the efficiency of the engine, i.e., to increase the work produced per unit quantity of fuel. We demonstrate this for an idealized cycle, called the air-standard Otto cycle, shown in Fig. 8.9. It consists of two adiabatic and two constant-volume steps, which comprise a heat-engine cycle for which air is the working fluid. In step DA, sufficient heat is absorbed by the air at constant volume to raise its temperature and pressure to the values resulting from combustion in an actual Otto engine. Then the air is expanded adiabatically and reversibly (step AB), cooled at constant volume (step BC), and finally compressed adiabatically and reversibly to the initial state at D.

The thermal efficiency Q of the air-standard cycle shown in Fig. 8.9 is simply:

$$\eta = \frac{|W(\text{net})|}{Q_{DA}} = \frac{Q_{DA} + Q_{BC}}{Q_{DA}} \tag{8.3}$$

For 1 mol of air with constant heat capacities,

$$Q_{DA} = C_V(T_A - T_D) \qquad \text{and} \qquad Q_{BC} = C_V(T_C - T_B)$$

Substituting these expressions in Eq. (8.3) gives:

$$\eta = \frac{C_V(T_A - T_D) + C_V(T_C - T_B)}{C_V(T_A - T_D)}$$

$$\eta = 1 - \frac{T_B - T_C}{T_A - T_D}$$

The thermal efficiency is related in a simple way to the compression ratio, ***r* = *Vc / VD.***

Each temperature in Eq. (8.4) is replaced by an appropriate group *PV/R, in accord with the* ideal-gas equation. Thus,

$$T_B = \frac{P_B V_B}{R} = \frac{P_B V_C}{R} \qquad T_C = \frac{P_C V_C}{R}$$

$$T_A = \frac{P_A V_A}{R} = \frac{P_A V_D}{R} \qquad T_D = \frac{P_D V_D}{R}$$

Substituting into Eq. (8.4) leads to:

$$\eta = 1 - \frac{V_C}{V_D}\left(\frac{P_B - P_C}{P_A - P_D}\right) = 1 - r\left(\frac{P_B - P_C}{P_A - P_D}\right)$$

For the two adiabatic, reversible steps, $P\,V^{\gamma}$ = *const. Hence:*

$$P_A V_D^{\gamma} = P_B V_C^{\gamma} \qquad \text{(because } V_D = V_A \text{ and } V_C = V_B\text{)}$$

$$P_C V_C^{\gamma} = P_D V_D^{\gamma}$$

These expressions are combined to eliminate the volumes:

$$\frac{P_B}{P_C} = \frac{P_A}{P_D}$$

This equation transforms Eq. (8.5):

$$\eta = 1 - r\frac{(P_B/P_C - 1)P_C}{(P_A/P_D - 1)P_D} = 1 - r\frac{P_C}{P_D}$$

Since
$$\frac{P_C}{P_D} = \left(\frac{V_D}{V_C}\right)^{\gamma} = \left(\frac{1}{r}\right)^{\gamma}$$

$$\eta = 1 - r\left(\frac{1}{r}\right)^{\gamma} = 1 - \left(\frac{1}{r}\right)^{\gamma - 1}$$

This equation shows that the thermal efficiency increases rapidly with the compression ratio r at low values of γ, but more slowly at high compression ratios. This agrees with the results of actual tests on Otto engines.

The Diesel Engine

The Diesel engine differs from the Otto engine primarily in that the temperature at the end of compression is sufficiently high that combustion is initiated spontaneously. This higher temperature results because of a higher compression ratio that carries the compression step to a higher pressure. The fuel is not injected until the end of the compression step, and then is added slowly enough that the combustion process occurs at approximately constant pressure.

For the same compression ratio, the Otto engine has a higher efficiency than the Diesel engine. However, preignition limits the compression ratio attainable in the Otto engine. The Diesel engine therefore operates at higher compression ratios, and consequently at higher efficiencies.

Example 8.3

Sketch the air-standard Diesel cycle on a PV diagram, and derive an equation giving the thermal efficiency of this cycle in relation to the compression ratio r (ratio of volumes at the beginning and end of the compression step) and the expansion ratio r_e (ratio of volumes at the end and beginning of the adiabatic expansion step).

Solution 8.3

The air-standard Diesel cycle is the same as the air-standard Otto cycle, except that the heat-absorption step (corresponding to the combustion process in the actual engine) is at constant pressure, as indicated by line DA in Fig. 8.10.

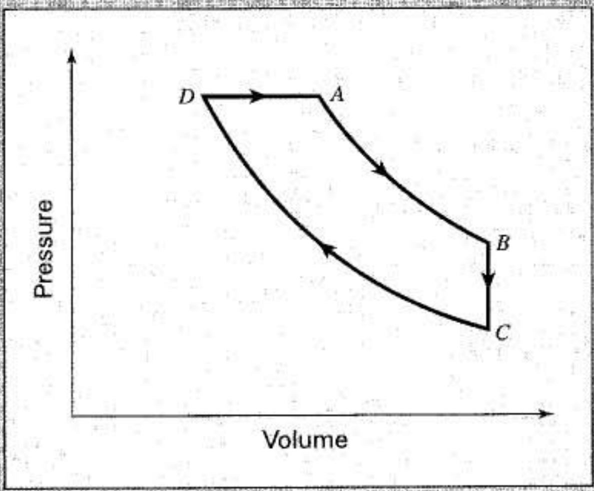

Figure 8.10 Air-standard Diesel cycle

On the basis of 1 mol of air, considered to be an ideal gas with constant heat capacities, the heat quantities absorbed in step DA and rejected in step BC are:

$$Q_{DA} = C_P(T_A - T_D) \qquad \text{and} \qquad Q_{BC} = C_V(T_C - T_B)$$

The thermal efficiency, Eq. (8.3), is:

$$\eta = 1 + \frac{Q_{BC}}{Q_{DA}} = 1 + \frac{C_V(T_C - T_B)}{C_P(T_A - T_D)} = 1 - \frac{1}{\gamma}\left(\frac{T_B - T_C}{T_A - T_D}\right) \qquad (A)$$

For reversible, adiabatic expansion (step AB) and reversible, adiabatic compression (step CD), Eq. (3.29a) applies:

$$T_A V_A^{\gamma-1} = T_B V_B^{\gamma-1} \qquad \text{and} \qquad T_D V_D^{\gamma-1} = T_C V_C^{\gamma-1}$$

By definition, the compression ratio is $r \equiv V_C / V_D$; in addition the expansion ratio is defined as $r_e \equiv V_B / V_A$. Thus,

$$T_B = T_A\left(\frac{1}{r_e}\right)^{\gamma-1} \qquad T_C = T_D\left(\frac{1}{r}\right)^{\gamma-1}$$

Substituting these equations into Eq. (A) gives:

$$\eta = 1 - \frac{1}{\gamma}\left[\frac{T_A(1/r_e)^{\gamma-1} - T_D(1/r)^{\gamma-1}}{T_A - T_D}\right] \qquad (B)$$

Also $P_A = P_D$, and from the ideal-gas equation,

$$P_D V_D = RT_D \qquad \text{and} \qquad P_A V_A = RT_A$$

Moreover, $V_C = V_B$, and therefore:

$$\frac{T_D}{T_A} = \frac{V_D}{V_A} = \frac{V_D/V_C}{V_A/V_B} = \frac{r_e}{r}$$

This relation combines with Eq. (B):

$$\eta = 1 - \frac{1}{\gamma}\left[\frac{(1/r_e)^{\gamma-1} - (r_e/r)(1/r)^{\gamma-1}}{1 - r_e/r}\right]$$

or

$$\eta = 1 - \frac{1}{\gamma}\left[\frac{(1/r_e)^{\gamma} - (1/r)^{\gamma}}{1/r_e - 1/r}\right] \qquad (8.7)$$

<u>The Gas-Turbine Engine</u>

The Otto and Diesel engines exemplify direct use of the energy of high-temperature, highpressure gases acting on a piston within a cylinder; no heat transfer with an external source is required. However, turbines are more efficient than reciprocating engines, and the advantages of internal combustion are combined with those of the turbine in the gas-turbine engine.

The gas turbine is driven by high-temperature gases from a combustion chamber, as indicated in Fig. 8.11. The entering air is compressed (supercharged) to a pressure of several bars before combustion. The centrifugal compressor operates on the same shaft as the turbine, and part of the work of the turbine serves to drive the compressor. The higher the temperature of the combustion gases entering the turbine, the higher the efficiency of the unit, i.e., the greater the work produced per unit of fuel burned. The limiting temperature is determined by the strength of the metal turbine blades, and is much lower than the theoretical flame temperature of the fuel. Sufficient excess air must be supplied to keep the combustion temperature at a safe level.

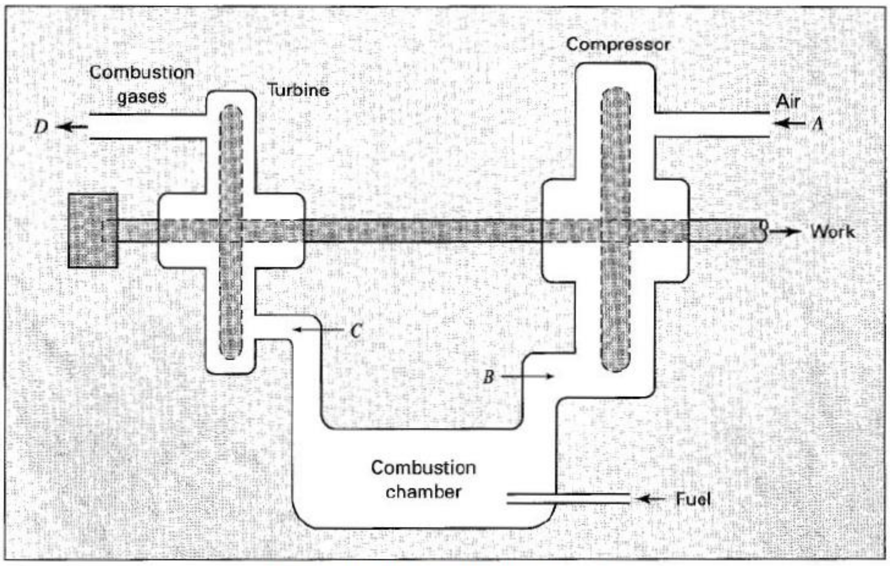

Figure 8.11 Gas-turbine engine

The idealization of the gas-turbine engine (based on air, and called the Brayton cycle) is shown on a PV diagram in Fig. 8.12. Step AB is the reversible adiabatic compression of air from P_A (atmospheric pressure) to P_B. In step BC heat Q_{Bc}, replacing combustion, is added at constant pressure, raising the air temperature prior to the work-producing isentropic expansion of the air from pressure P_C to pressure P_D (atmospheric pressure). Step DA is a constant-pressure cooling process that merely completes the cycle. The thermal efficiency of the cycle is:

$$\eta = \frac{|W(\text{net})|}{Q_{BC}} = \frac{|W_{CD}| - W_{AB}}{Q_{BC}} \tag{8.8}$$

where each energy quantity is based on 1 mol of air.

The work done as the air passes through the compressor is given by Eq. (7.14), and for air as an ideal gas with constant heat capacities:

$$W_{AB} = H_B - H_A = C_P(T_B - T_A)$$

Similarly, for the heat-addition and turbine processes,

$$Q_{BC} = C_P(T_C - T_B) \qquad \text{and} \qquad |W_{CD}| = C_P(T_C - T_D)$$

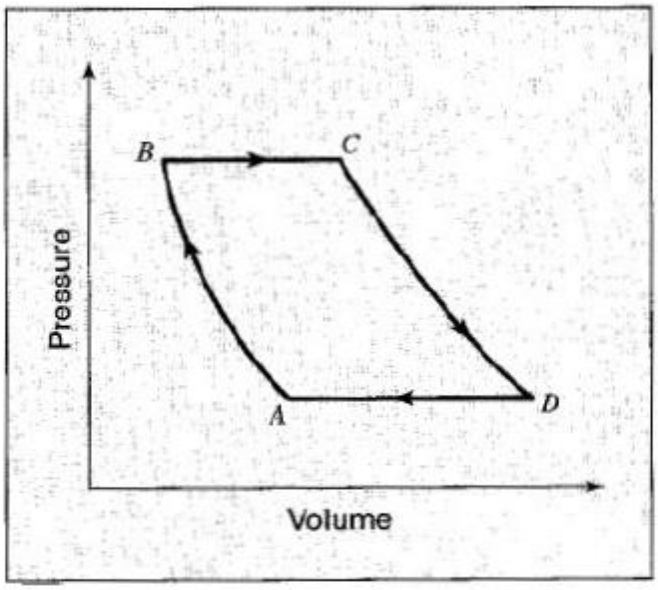

Figure 8.12 Ideal cycle for gas-turbine engine

Substituting these equations into Eq. (8.8) and simplifying leads to:

$$\eta = 1 - \frac{T_D - T_A}{T_C - T_B} \tag{8.9}$$

Since processes AB and CD are isentropic, the temperatures and pressures are related by Eq. (3.29b):

$$\frac{T_B}{T_A} = \left(\frac{P_B}{P_A}\right)^{(\gamma-1)/\gamma} \tag{8.10}$$

And $$\frac{T_D}{T_C} = \left(\frac{P_D}{P_C}\right)^{(\gamma-1)/\gamma} = \left(\frac{P_A}{P_B}\right)^{(\gamma-1)/\gamma} \tag{8.11}$$

With these equations T_A and T_D may be eliminated to give:

$$\eta = 1 - \left(\frac{P_A}{P_B}\right)^{(\gamma-1)/\gamma} \tag{8.12}$$

Example 8.4

A gas-turbine engine with a compression ratio $P_B/P_A = 6$ operates with air entering the compressor at 298.15 K (25°C). If the maximum permissible temperature in the turbine is 1033.15 K (760°C), determine:

(*a*) The efficiency η of the ideal air cycle for these conditions if $\gamma = 1.4$.

(*b*) The thermal efficiency of an air cycle for the given conditions if the compressor and turbine operate adiabatically but irreversibly with efficiencies $\eta_c = 0.83$ and $\eta_t = 0.86$.

Solution 8.4

(*a*) Direct substitution in Eq. (8.12) gives the ideal-cycle efficiency:

$$\eta = 1 - (1/6)^{(1.4-1)/1.4} = 1 - 0.60 = 0.40$$

(*b*) Irreversibilities in the compressor and turbine reduce the thermal efficiency of the engine, because the net work is the difference between the work required by the compressor and the work produced by the turbine. The temperature of the air entering the compressor T_A and the temperature of the air entering the turbine, the specified maximum for T_C, are the same as for the ideal cycle. However, the temperature after irreversible compression in the compressor T_B is higher than the temperature after *isentropic* compression T'_B, and the temperature after irreversible expansion in the turbine T_D is higher than the temperature after *isentropic* expansion T'_D.

The thermal efficiency of the engine is given by:

$$\eta = \frac{|W(\text{turb})| - W(\text{comp})}{Q}$$

The two work terms are found from the expressions for isentropic work:

$$|W(\text{turb})| = \eta_t C_P(T_C - T'_D)$$

$$W(\text{comp}) = \frac{C_P(T'_B - T_A)}{\eta_c} \qquad (A)$$

The heat absorbed to simulate combustion is:

$$Q = C_P(T_C - T_B)$$

These equations combine to yield:

$$\eta = \frac{\eta_t(T_C - T'_D) - (1/\eta_c)(T'_B - T_A)}{T_C - T_B}$$

An alternative expression for the compression work is:

$$W(\text{comp}) = C_P(T_B - T_A) \qquad (B)$$

Combining Eqs. (A) and (B) and using the result to eliminate T_B from the equation for η gives after simplification:

$$\eta = \frac{\eta_t \eta_c(T_C/T_A - T'_D/T_A) - (T'_B/T_A - 1)}{\eta_c(T_C/T_A - 1) - (T'_B/T_A - 1)} \qquad (C)$$

The ratio T_C/T_A depends on given conditions. The ratio T'_B/T_A is related to the pressure ratio by Eq. (8.10). In view of Eq. (8.11), the ratio T'_D/T_A can be expressed as:

$$\frac{T'_D}{T_A} = \frac{T_C T'_D}{T_A T_C} = \frac{T_C}{T_A}\left(\frac{P_A}{P_B}\right)^{(\gamma-1)/\gamma}$$

Substituting these expressions in Eq. (C) yields:

$$\eta = \frac{\eta_t \eta_c(T_C/T_A)(1 - 1/\alpha) - (\alpha - 1)}{\eta_c(T_C/T_A - 1) - (\alpha - 1)} \qquad (8.13)$$

where $$\alpha = \left(\frac{P_B}{P_A}\right)^{(\gamma-1)/\gamma}$$

One can show by Eq. (8.13) that the thermal efficiency of the gas-turbine engine increases as the temperature of the air entering the turbine (T_C) increases, and as the compressor and turbine efficiencies η_c and η_t increase.

The given efficiency values are here:

$$\eta_t = 0.86 \qquad \text{and} \qquad \eta_c = 0.83$$

Other given data provide:

$$\frac{T_C}{T_A} = \frac{1033.15}{298.15} = 3.47$$

and $$\alpha = (6)^{(1.4-1)/1.4} = 1.67$$

Substituting these quantities in Eq. (8.13) gives:

$$\eta = \frac{(0.86)(0.83)(3.47)(1 - 1/1.67) - (1.67 - 1)}{(0.83)(3.47 - 1) - (1.67 - 1)} = 0.235$$

This analysis shows that, even with a compressor and turbine of rather high efficiencies, the thermal efficiency (23.5%) is considerably reduced from the ideal-cycle value of 40%.