9-8. THE BRAYTON CYCLE

The *Brayton cycle* is a modified diesel or compression-ignition cycle which is secured by extending the adiabatic expansion process (3-4), Fig. 9-2, until the pressure at station 4 is attained. This cycle shown in

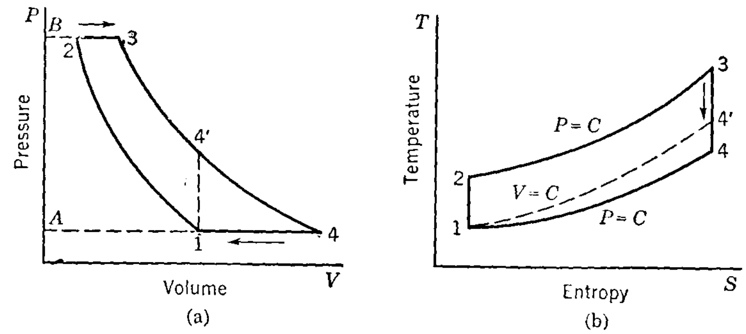

Fig. 9-7. Brayton air-standard cycle. (a) Pressure-volume diagram; (b) temperature-entropy diagram.

Fig. 9-7 is the basis for the operation of the *gas-turbine power plant.* From this illustration it is easy to see that the efficiency of the Brayton cycle is better than the corresponding diesel cycle shown as 1-2-3-4′-1. The increase in net work is a result of the reduction in the amount of heat rejected and is secured without any additional heat reception.

A *gas-turbine power plant* is an application of the steady-flow heat-engine cycle. Turbo-machinery is used by a plant operating on the Brayton cycle, since this type can handle large volumes more efficiently

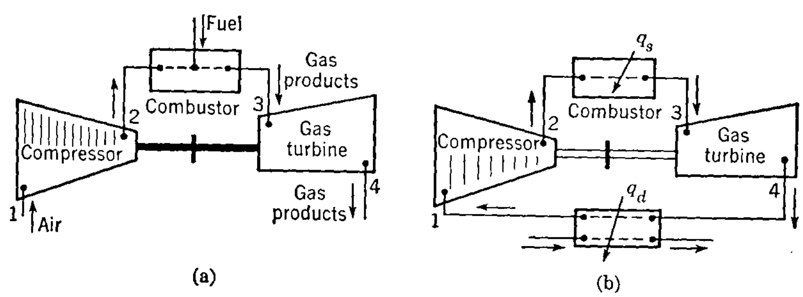

Fig. 9-8. Gas-turbine power plant. (a) Actual arrangement; (b) air-standard cycle.

than can the reciprocating types of the Otto and diesel power plants. A continuous-combustion or a combustion-gas turbine power plant consists basically of three main parts: (1) an *air compressor,* (2) a *combustion chamber* with a *fuel burner,* often called the *combustor,* and (3) a *gas* *turbine.* These three elements are shown in Fig. 9-8, and the symbols correspond to those of Fig. 9-7.

In the theoretical cycle of this power plant, Fig. 9-7(a), atmospheric air with a pressure P_1 and temperature T_1 is received. This air is isentropically compressed in the compressor to P_2 and T_2, according to the path 1-2, Fig. 9-7. Heat is received at constant pressure P_2 and the temperature rises to T_3, as the fuel is ignited and burned in the combustor after being mixed with air as it came from the compressor. The *products of combustion* leave the combustor at point 3 and enter the turbine, where they expand isentropically to the initial pressure so that $P_4 = P_1$. After the expansion process 3-4, just described, the gases are exhausted from the turbine.

The analysis of this cycle on the basis of the air-standard efficiency can be done by using a unit mass between the states 2 and 3 for the reception of heat and between states 4 and 1 for the rejection of heat. In steady-flow machinery it is easier to carry out heat transfer and combustion processes at constant pressure than at constant volume, and the turbine is not adapted to the high temperatures characteristic of the reciprocating engine. For these reasons, the Brayton cycle will serve as the air-standard cycle for the analysis of the gas-turbine power plant.

In this cycle, a part of the gross work delivered by the turbine is returned to the system to operate the compressor. The work to operate the compressor, Fig. 9-7(a), is proportional to the area 1-2-B-A-1, while the work delivered by the turbine is represented by the area B-3-4-A-B. The net work then is represented by the area bounded by the outline of the Brayton cycle.

By steady-flow analysis of each component of the cycle we arrive at the following results:

$$q_s = h_3 - h_2; \quad \mathrm{wk}_t = h_3 - h_4; \quad \mathrm{wk}_c = h_2 - h_1; \quad q_d = h_4 - h_1$$

Thus the net work (wk_n) is:

$$\mathrm{wk}_n = \mathrm{wk}_t - \mathrm{wk}_c = (h_3 - h_4) - (h_2 - h_1) \tag{9-12}$$

and
$$\eta_t = \frac{\mathrm{wk}_n}{q_s} = \frac{(h_3 - h_4) - (h_2 - h_1)}{h_3 - h_2} = \frac{(h_3 - h_2) - (h_4 - h_1)}{h_3 - h_2}$$

$$= 1 - \frac{h_4 - h_1}{h_3 - h_2} \quad \text{(gas turbine power plant)} \tag{9-13}$$

in which wk_t = specific turbine work, Btu per lb;
wk_c = specific compressor work, Btu per lb; and
η_t = air-standard (thermal) efficiency of the gas-turbine power plant.

The change of specific enthalpy for a gas system is, by Eq. (7-11), equal to $c_p(T_2 - T_1)$, thus:

$$h_4 - h_1 = c_p(T_4 - T_1); \quad \text{also} \quad h_3 - h_2 = c_p(T_3 - T_2)$$

so
$$\eta_t = 1 - \frac{c_p(T_4 - T_1)}{c_p(T_3 - T_2)} = 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

It can be shown that $(T_4 - T_1)/(T_3 - T_2)$ is equal to T_1/T_2; thus:

$$\eta_t = 1 - \frac{T_1}{T_2} = 1 - \frac{T_4}{T_3} = 1 - \frac{1}{(r_c)^{k-1}} = 1 - \frac{1}{(r_p)^{(k-1)/k}} \tag{9-14}$$

in which r_p = pressure ratio, P_2/P_1.

Thus, the air-standard efficiency of a constant-pressure combustion cycle with complete expansion is identical with the efficiency of a constant-volume explosion cycle (Otto) with an incomplete expansion as used in spark-ignition engines. Therefore, the theoretical efficiency depends only on the pressure ratio (same effect as compression ratio for the Otto cycle) and is independent of the load just as it was for the Otto cycle. On the basis of pressure ratio, the efficiency of the Brayton cycle is the same as for the Carnot cycle, but under this condition the two cycles are not being compared on the same basis of equality of source and drain temperatures. Therefore, under the latter condition the Brayton cycle does not have as high an efficiency as the Carnot cycle. However, it does point out that the efficiency is a function of the pressure ratio.

Figure 9-9 shows the effects of irreversibilities which are encountered in the compressor and turbine of the Brayton cycle. The system has

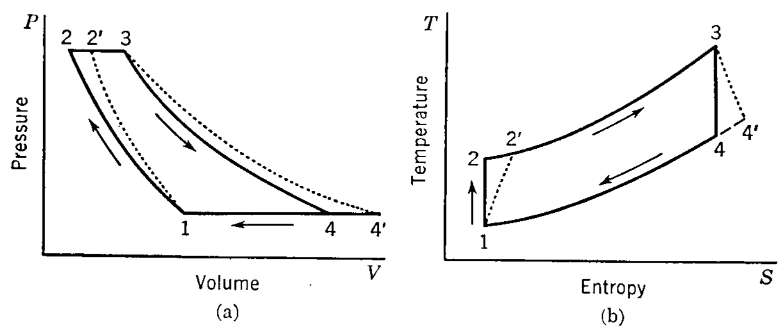

Fig. 9-9. Effects of irreversibilities. (a) Pressure-volume diagram; (b) temperature-entropy diagram.

greater entropy at the conclusion of these processes than it would have if the processes had been executed perfectly. Thus the enthalpies are also larger and the net work is $(h_3 - h_4') - (h_2' - h_1)$, which is a smaller amount than for the reversible cycle, both because the turbine work is less and the compressor work is greater. Therefore, if the turbine and/or the compressor has a low efficiency, the net output of the cycle may be zero or even less than zero (negative). The following example illustrates the performance of the Brayton cycle.

Example 9-4. An air-standard Brayton cycle operates with a pressure ratio of 5. The temperatures at the inlets to the compressor and turbine are 100F and 1300F, respectively. Find: (a) work of compression; (b) work of expansion; (c) net work; (d) work ratio; and (e) air-standard efficiency.

Solution (a): Figure 9-7 applies to this example.

$$T_2 = 560(5)^{(1.4-1)/1.4} = 560 \times 1.585 = 888\text{R}$$

$$\text{wk}_c = h_2 - h_1 = c_p(T_2 - T_1) = 0.24(888 - 560) = 78.7 \text{ Btu per lb} \qquad \text{Eq. (9-12)}$$

Solution (b):

$$T_4 = 1760\left(\frac{1}{5}\right)^{(1.4-1)/1.4} = \frac{1760}{1.585} = 1110\text{R}$$

$$\text{wk}_t = (h_3 - h_4) = c_p(T_3 - T_4) = 0.24(1760 - 1110)$$

$$= 156 \text{ Btu per lb} \qquad \text{Eq. (9-12)}$$

Solution (c):

$$\text{wk}_n = \text{wk}_t - \text{wk}_c = 156 - 78.7 = 77.3 \text{ Btu per lb} \quad \text{Eq. (9-12)}$$

Solution (d):

$$\text{WR} = \frac{\text{wk}_n}{\text{wk}_t} = \frac{77.3}{156} = 0.496 \quad \text{or } 49.6 \text{ per cent} \quad \text{Eq. (9-10)}$$

Solution (e):

$$\eta_t = 1 - \frac{T_1}{T_2} = 1 - \frac{560}{888} = 1 - 0.631 = 0.369 \quad \text{or } 36.9 \text{ per cent Eq. (9-14)}$$

9-9. METHODS TO IMPROVE GAS-TURBINE CYCLE EFFICIENCY

Increasing the pressure ratio and elevating the inlet temperature, Fig. 9-10, are effective means of improving the efficiency, but their application is limited by the decreasing resistance of the blades of the compressor and turbine to *corrosion* and *creep*. Another way to increase efficiency is by improving the performance of the compressor and turbine by applying aerodynamic principles to *improve the shape of the blades*. Other thermodynamic methods for improving the performance of the gas-turbine power plant will now be presented. These methods are not handicapped by the metallurgical limit.

It has been seen that the *principle of regeneration* has been used as a scheme for elevating the average temperature of the heat-reception