Chapter 8
Closed System Cycles

8.1 Thermodynamic Cycles

In previous chapters we discussed thermal efficiency of heat engines and coefficient of performance of refrigerators and heat pumps from a general viewpoint, without asking for the processes occurring within the engines. Now we will discuss the working principles of several closed system cycles.

All engines considered are steady state devices that do not accumulate energy or mass over time. To realize a steady state thermodynamic engine, a working fluid is subjected to a series of processes such that the process curve in a property diagram forms a closed loop. As the engine operates, the working fluid runs through the same cycle of processes again and again.

The closed loop integral of the energy vanishes, $\oint dE = 0$, since energy is a state property;[1] the engine does not accumulate energy. Thus, integration of the differential energy balance $dE = \delta Q - \delta W$ over the full cycle yields

$$W_\odot = \oint \delta W = \oint \delta Q = Q_\odot = Q_{in} - |Q_{out}| \ , \tag{8.1}$$

where $W_\odot$ and $Q_\odot$ are the total net work and net heat exchanged for the cycle. Moreover, $Q_{in} > 0$ is the total heat transferred into the cycle, and $Q_{out} < 0$ is the total heat transferred out.

Net work and net heat are positive for a clockwise process—a heat engine—and are negative for a counter-clockwise process—a refrigerator or a heat pump.

For a closed system, the engine contains the constant mass m of the working substance, and the net work and heat per unit mass are

$$w_\odot = \frac{W_\odot}{m} = \frac{Q_\odot}{m} = q_\odot = q_{in} - |q_{out}| \ . \tag{8.2}$$

[1] We have $\int_1^2 dE = E_2 - E_1$. For a closed loop inital and endpoint are the same, that is $E_2 = E_1$.

H. Struchtrup, *Thermodynamics and Energy Conversion*,
DOI: 10.1007/978-3-662-43715-5_8,

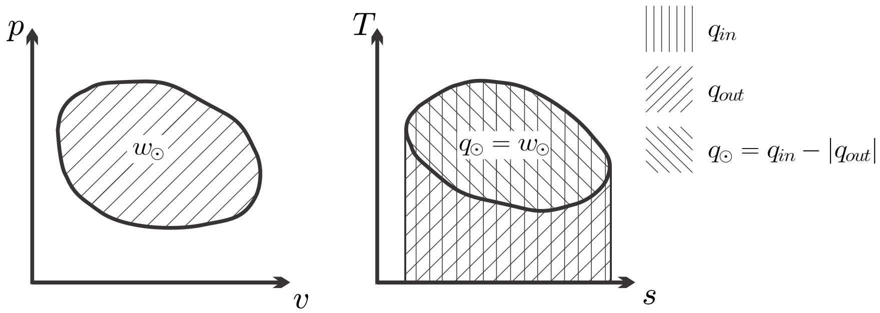


Fig. 8.1 Thermodynamic cycle in p-v- and T-s-diagram. Heat in, q_{in}, and heat out, q_{out}, are indicated for a heat engine, which runs clockwise.

For an engine that runs through the cycle with the frequency $\dot{n}$ (measured for instance in rounds per minute, rpm) the net power and the heat transfer rates are

$$\dot{W}_\odot = \dot{n}mw_\odot = \dot{n}mq_\odot = \dot{Q}_\odot \quad , \quad \dot{Q}_{in} = \dot{n}mq_{in} \quad , \quad \dot{Q}_{out} = \dot{n}mq_{out} \; . \qquad (8.3)$$

Figure 8.1 shows a reversible thermodynamic cycle in the p-v- and T-s-diagrams. For a reversible process in a closed system, the net work of the cycle per unit mass of working fluid is

$$w_\odot = \oint p dv \; . \qquad (8.4)$$

In the p-v-diagram, the net work is just the area enclosed by the process curve as indicated in the figure. Note that the integral is positive for a clockwise cycle, and negative for a counter-clockwise cycle.

Similarly, the net heat exchanged for a reversible cycle is the area enclosed by the cycle in the T-s-diagram,

$$q_\odot = \oint T ds \; , \qquad (8.5)$$

where the heat is positive for a clockwise cycle. Heat in and heat out,

$$q_{in} = \int_{ds>0} T ds \quad , \quad q_{out} = \int_{ds<0} T ds \; , \qquad (8.6)$$

can be read from the T-s-diagram as the areas below the respective process curves. This is indicated in the figure for a heat engine (clockwise cycle). With this, the net work ($w_\odot = q_\odot$), the heat in, and the heat out can all be read from the T-s-diagram. This implies that the thermal efficiency or the coefficient of performance for a reversible cycle can be completely determined from the T-s-diagram, by means of the relations

$$\begin{aligned} \eta_{th} &= \frac{w_\odot}{q_{in}} = \frac{\oint T ds}{\int_{ds>0} T ds} = 1 - \frac{\left|\int_{ds<0} T ds\right|}{\int_{ds>0} T ds} \,, \\ \mathrm{COP_R} &= \frac{q_{in}}{|w_\odot|} = \frac{\int_{ds>0} T ds}{\left|\oint T ds\right|} = \frac{1}{\frac{\left|\int_{ds<0} T ds\right|}{\int_{ds>0} T ds} - 1} \,, \\ \mathrm{COP_{HP}} &= \frac{q_{out}}{|w_\odot|} = \frac{\int_{ds<0} T ds}{\left|\oint T ds\right|} = \frac{1}{1 - \frac{\int_{ds>0} T ds}{\left|\int_{ds<0} T ds\right|}} \,. \end{aligned} \tag{8.7}$$

While in this chapter we discuss only reversible cycles, we note that the processes in real engines always suffer from irreversibilities, so that their thermal efficiencies or COP's will be smaller than those that will be computed below.

8.2 Carnot Cycle

As a first example for the evaluation of a thermodynamic cycle we consider the Carnot cycle. We introduced the Carnot engine as a fully reversible engine—no internal or external irreversibilities—operating between two reservoirs of temperatures T_H, T_L. The Carnot cycle is *one* possible realization of such an engine. For a heat engine, it consists of the following four processes

1-2 rev. isothermal compression at T_L
2-3 rev. adiabatic (isentropic) compression from T_L to T_H
3-4 rev. isothermal expansion at T_H
4-1 rev. adiabatic (isentropic) expansion from T_H to T_L

Heat is only exchanged with the reservoirs during the isothermal processes, at which the working substance is at the temperature of the reservoirs, and therefore there are no external irreversibilities associated with the cycle. Since there are no internal or external irreversibilities, the above cycle is a fully reversible cycle that exchanges heat only with the two reservoirs.

The T-s-diagram allows us to compute the thermal efficiency of the cycle: The area enclosed by the cycle is the net work,

$$w_\odot = q_\odot = (T_H - T_L)\Delta s \,. \tag{8.8}$$

The area below the curve 3-4 is the heat in,

$$q_{in} = q_{34} = T_H \Delta s \,. \tag{8.9}$$

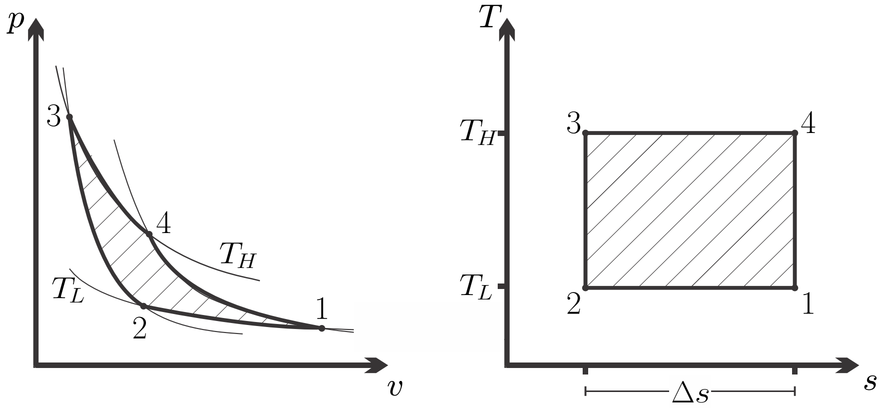


Fig. 8.2 Carnot cycle in the p-v- and T-s-diagrams

Accordingly, the thermal efficiency is, as expected, the Carnot efficiency

$$\eta_C = \frac{w_\odot}{q_{in}} = 1 - \frac{T_L}{T_H} \; . \tag{8.10}$$

We now compute the efficiency by considering the cycle for an ideal gas. With the work and heat of reversible processes computed in the last chapter, we find the values for heat and work of the individual processes as

$$\begin{array}{lll} \text{1-2 isothermal:} & w_{12} = -RT_L \ln \frac{p_2}{p_1} & , \; q_{12} = -RT_L \ln \frac{p_2}{p_1} \; , \\ \text{2-3 isentropic:} & w_{23} = u\left(T_L\right) - u\left(T_H\right) & , \; q_{23} = 0 \; , \\ \text{3-4 isothermal:} & w_{34} = -RT_H \ln \frac{p_4}{p3} & , \; q_{34} = -RT_H \ln \frac{p_4}{p3} \; , \\ \text{4-1 isentropic:} & w_{41} = u\left(T_H\right) - u\left(T_L\right) & , \; q_{41} = 0 \; , \end{array} \tag{8.11}$$

Since the processes (2-3) and (4-1) are isentropic, we have

$$\begin{aligned} 0 &= s_3 - s_2 = s^0\left(T_H\right) - s^0\left(T_L\right) - R \ln \frac{p_3}{p_2} \; , \\ 0 &= s_4 - s_1 = s^0\left(T_H\right) - s^0\left(T_L\right) - R \ln \frac{p_4}{p_1} \; . \end{aligned} \tag{8.12}$$

From comparison of the two equations we find that the pressures at the corner points of the process are related as $\frac{p_3}{p_2} = \frac{p_4}{p_1}$, or, alternatively, $\frac{p_3}{p_4} = \frac{p_2}{p_1}$.

The thermal efficiency of the cycle is

$$\eta_C = \frac{w_\odot}{q_{in}} = \frac{w_{12} + w_{23} + w_{34} + w_{41}}{q_{34}} \; . \tag{8.13}$$

With the above results for work and heat, and the relation between the pressures, we find, once more, the Carnot efficiency,

$$\eta_C = \frac{-T_L \ln \frac{p_2}{p_1} - T_H \ln \frac{p_4}{p_3}}{-T_H \ln \frac{p_4}{p_3}} = 1 - \frac{T_L}{T_H} \,. \tag{8.14}$$

That we found the well-known result again from detailed calculations for an ideal gas proves that the ideal gas temperature scale (Sec. 2.13) is identical to the thermodynamic temperature scale (Sec. 5.6).

The net work of the ideal gas Carnot cycle,

$$w_\odot = R\,(T_H - T_L) \ln \frac{p_2}{p_1} \tag{8.15}$$

grows with the temperature difference $(T_H - T_L)$ and the pressure ratio $\frac{p_2}{p_1}$. Thus, for large efficiency and large work output the Carnot cycle should operate at large temperature difference $T_H - T_L$ and at large pressure ratios $\frac{p_2}{p_1}$. Then, the volume ratio between smallest and largest volume,[2] $\frac{v_1}{v_3} = \frac{p_2}{p_1}\left(\frac{T_H}{T_L}\right)^{\frac{1}{k-1}}$, becomes large, which is quite unpractical for designing a compact engine. Moreover, the overall pressure ratio $\frac{p_3}{p_1} = \frac{v_1}{v_3}\frac{T_H}{T_L} = \frac{p_2}{p_1}\left(\frac{T_H}{T_L}\right)^{\frac{k}{k-1}}$ becomes large, which makes effective sealing difficult. For example, an engine with air as working gas operating at $T_L = 300\,\mathrm{K}$, $T_H = 750\,\mathrm{K}$, and a pressure ratio of $\frac{p_2}{p_1} = 5$, has the thermal efficiency $\eta_C = 1 - \frac{T_L}{T_H} = 0.6$, the overall volume ratio $\frac{v_1}{v_3} = 49.5$, and the overall pressure ratio $\frac{p_3}{p_1} = 124$; it produces the net work $w_\odot = 208 \frac{\mathrm{kJ}}{\mathrm{kg}}$. Internal combustion engines with comparable net work operate with significantly smaller volume and pressure ratios, and thus are more compact and lighter, and suffer less from sealing problems.

Another problem for the Carnot cycle is that the isothermal heat exchange processes (1-2, 3-4) require slow processes, so that the cycle frequency $\dot{n}$ must be low. To produce significant amounts of power $\dot{W} = \dot{n} m w_\odot$ the engine would have to contain a large mass m, that is it must be large. Engines that operate on higher frequencies $\dot{n}$ can be more compact.

The Carnot engine operates between two reservoirs of constant temperature. If the engine is to be heated by burning of a fuel, one does not have a constant high temperature reservoir, but a flow of hot combustion product at flame temperature T_F which is gradually cooled to T_H in the heat exchange. This implies that there will be a temperature difference between engine and combustion gas, and thus an external irreversibility. If one uses such a hot flow to heat the engine, one will have warm exhaust which has still work potential. If the exhaust is expelled into the environment, the equilibration of temperature between the warm exhaust (at T_H or higher) and the environment (at T_L or lower) is an external irreversibility. To eliminate, or at least

[2] For the isothermal process 1-2 we have $p_1 v_1 = p_2 v_2$ which gives $\frac{v_1}{v_3} = \frac{p_2}{p_1}\frac{v_2}{v_3}$; the adiabatic relation for the process 2-3 gives $\frac{v_2}{v_3} = \left(\frac{T_H}{T_L}\right)^{\frac{1}{k-1}}$.

reduce this loss, the exhaust must be used for preheating of the combustion air. We shall come back to this point in Secs. 11.7 and 12.1.

In principle, one could make an effort to design an engine that follows the Carnot cycle. Obviously, due to irreversibilities, the real engine would have an efficiency below the Carnot efficiency. Moreover, for the reasons listed above, such an engine would be relatively large and heavy in relation to the amount of power it could deliver. Thus, in fact, one does not try to build engines that follow the Carnot cycle for practical applications.

The Stirling cycle with an ideal gas, which will be discussed in Sec. 13.1, is an alternative realization of a Carnot engine, but compared to the Carnot cycle it has a significantly smaller volume ratio. Stirling engines, i.e., engines designed to follow the Stirling cycle, are commercially available. Naturally, these have efficiencies below the Carnot efficiency due to unavoidable internal and external irreversibilities.

8.3 Carnot Refrigeration Cycle

Inversion of the Carnot cycle gives the Carnot refrigeration (or heat pump) cycle with the following processes:

1-2 rev. adiabatic (isentropic) compression from T_L to T_H
2-3 rev. isothermal compression at T_H
3-4 rev. adiabatic (isentropic) expansion from T_H to T_L
4-1 rev. isothermal expansion at T_L

The process curves in the p-v- and T-s-diagrams are depicted in Fig. 8.3. As before, the curve in the T-s-diagram is independent of the working fluid, while the p-v-diagram is sketched for an ideal gas as working fluid.

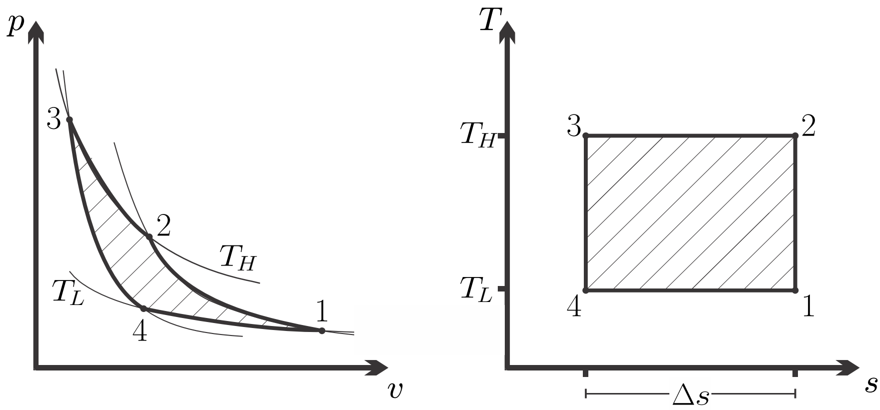


Fig. 8.3 Carnot refrigeration cycle (inverse Carnot cycle)

The analysis of the inverse cycle is just analog to the analysis in the preceding section, and we do not go through the details. The well-known coefficients of performance for a Carnot refrigerator and a Carnot heat pump can be easily read from the T-s-diagram as

$$\mathrm{COP}_{\mathrm{R},C} = \frac{|q_{in}|}{|w_{\odot}|} = \frac{T_L \Delta s}{(T_H - T_L)\,\Delta s} = \frac{1}{\frac{T_H}{T_L} - 1}\,, \tag{8.16}$$

$$\mathrm{COP}_{\mathrm{HP},C} = \frac{|q_{out}|}{|w_{\odot}|} = \frac{T_H \Delta s}{(T_H - T_L)\,\Delta s} = \frac{1}{1 - \frac{T_L}{T_H}}\,. \tag{8.17}$$

8.4 Internal Combustion Engines

In an internal combustion engine, heat is provided to the system by burning an air-fuel mixture *inside* the system. Engine operation requires the exchange of the working fluid after one cycle is completed, to bring in new fuel and oxygen. Accordingly, internal combustion engines exchange mass with their surroundings. Nevertheless, during the working cycle the system is closed, and thus internal combustion engines can be analyzed as closed systems.

Internal combustion engines are the dominant power source for cars, trucks, ships and non-electric trains. In cars one usually finds Otto engines, while in trucks, ships and trains Diesel engines are used.

Figure 8.4 shows a sketch of a single piston-cylinder assembly of an internal combustion engine. The piston is connected through rods to the crankshaft. The crankshaft is driven through the expansion process, which pushes the piston down, and it provides the work for the compression processes. Most engines have several cylinders that run through the cycle with a phase shift to ensure even load on the crankshaft; in single cylinder engines a fly wheel might be mounted to the crankshaft.

As the crank shaft turns, the piston moves between *bottom dead center* and *top dead center*. The volume the piston moves through is known as the *swept volume* V_s, the remaining volume at top dead center is the *clearance volume* V_c. Valves allow the exchange of working fluid with the surroundings.

The main difference between Otto and Diesel engine is that an Otto engine draws in air-fuel-mix while in a Diesel engine the fuel is injected into the cylinder later in the process. The combustion process in the Otto engine is triggered by a spark plug, while in the Diesel engine the fuel begins to burn as soon as it is injected. Accordingly, the figure shows spark plug (for Otto engine) and injector (for Diesel engine).

We now follow through the processes in a four-stroke engine, starting at top dead center:

Stroke I: The first stroke is the intake stroke. The valves are open, and as the piston moves towards bottom dead center air-fuel-mix (Otto) or air (Diesel)

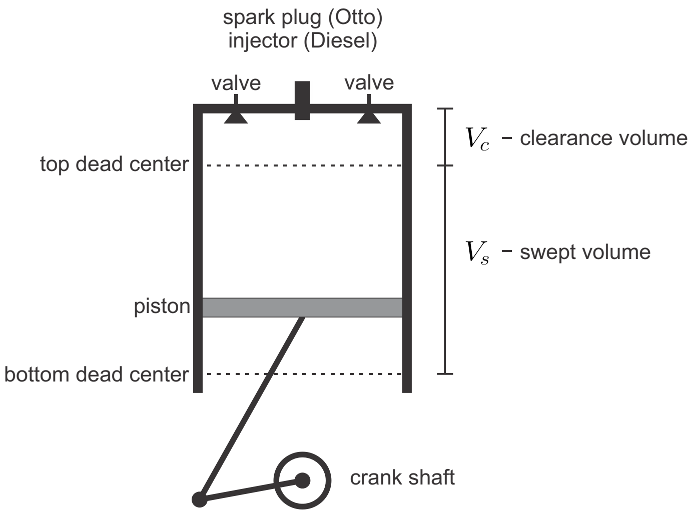


Fig. 8.4 Cylinder and piston of an internal combustion engine

enters the cylinder. Since the valves are open, the pressure in the cylinder is nearly constant. Whence bottom dead center is reached, the valves close.

Stroke II: In the second stroke the piston returns to top dead center. Since the valves are closed, the working fluid (air-fuel-mix or air) is compressed. The process is fast, and nearly adiabatic, pressure and temperature increase. The compression work is provided by the crankshaft. Shortly before the piston reaches top dead center, the combustion is triggered, either by firing the spark plug (Otto), or by injecting fuel which begins to burn in the hot air (Diesel).

Stroke III: As the fuel burns, the temperature of the working fluid is further increased. The hot combustion gas expands in the third stroke, as the piston returns to bottom dead center. The expansion is fast, nearly adiabatic, pressure and temperature decrease. The expansion work is transferred to the crankshaft.

Stroke IV: For the last of the four strokes the valves open, which leads to a sudden pressure drop. The piston returns once more to top dead center and pushes the expanded combustion products out. This is the exhaust stroke.

We summarize the processes in the four strokes in the following list:

Stroke I	:	intake	valve open
Stroke II	:	compression, combustion starts	valve closed
Stroke III	:	combustion continues, expansion	valve closed
Stroke IV	:	exhaust	valve open

A rough schematic p-V-diagram for all four strokes is shown in Fig. 8.5.

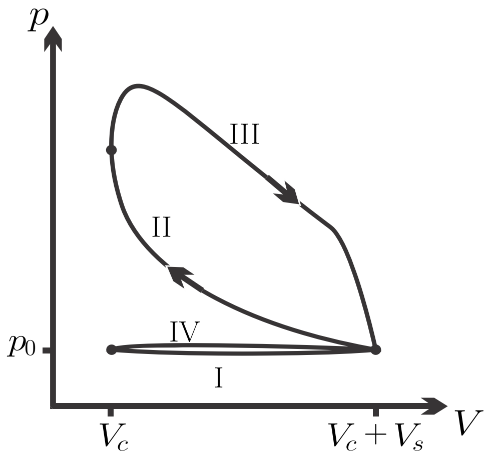


Fig. 8.5 Schematic p-V-diagram for an internal combustion engine

During strokes I and IV the valves are open, and the mass in the cylinder changes. As indicated in the p-V-diagram, in real engines there is a small depression during intake, and a small compression during exhaust, which we ignore. Thus, assuming that during intake and exhaust the pressure equals the exterior pressure p_0, the piston work for the two processes is $W_I = p_0 V_s$ and $W_{IV} = -p_0 V_s$. The work for these two strokes just cancels, $W_I + W_{IV} = 0$, and does not need to be considered further.

During strokes II and III the valves are closed, and the working fluid goes through a closed system cycle. Due to the chemical processes occurring in the combustion processes, the full analysis of this closed cycle is difficult. However, the ratio between the amounts of air and fuel is rather large. Therefore, the amount of fuel can be ignored for a basic analysis, and the working fluid can be considered as air for the complete cycle. This leads to the following modelling assumptions:

(a) The working fluid is air.

(b) The energy that is supplied through the combustion of fuel can be described as a heat transfer into the system.

(c) The exchange of the working fluid in exhaust and intake during which hot expanded combustion product is exchanged against cold precombustion working fluid can be considered as a heat exchange with the surroundings.

This *air standard analysis* allows to study internal combustion engines in a simplified yet significant manner. Further simplifications will appear in the following sections which deal with the Otto and Diesel cycles, and some variants.

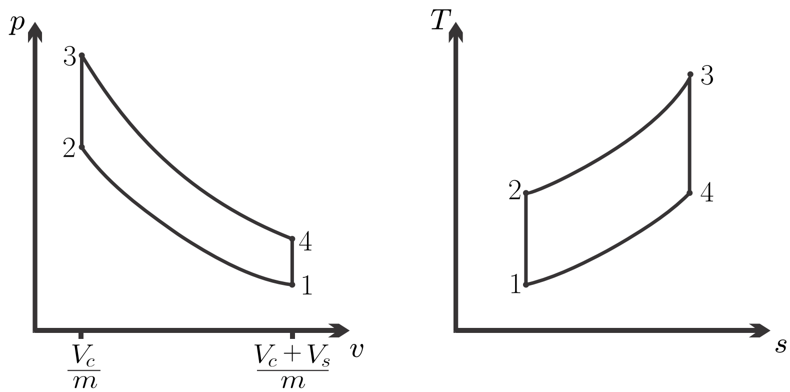


Fig. 8.6 Air-standard Otto cycle

8.5 Otto Cycle

The Otto cycle is named after Nicolaus.A. Otto (1832-1891), who build the first working internal combustion engine. In the Otto cycle, the fuel is mixed into the air outside the cylinder by injection, so that air-fuel mix enters the engine. The compression and expansion processes are assumed to be adiabatic—the processes are fast, and there is hardly time for effective heat exchange. When the spark plug ignites the compressed air-fuel-mix, the mixture explodes. The reaction is so fast, that the piston does not move much, and for modelling we describe this as an isochoric heat transfer into the working fluid. Thus, the compression and expansion strokes of the Otto engine can be modelled as

Stroke II:	1-2	adiabatic compression
	2-3	isochoric heating (air-fuel mix explodes)
Stroke III:	3-4	adiabatic expansion
	4-1	isochoric cooling (exchange of working fluid)

The respective process curves are shown in Fig. 8.6. For further analysis we assume reversible processes for which we find heat and work as

$$\begin{aligned}
&\text{1-2 isentropic: } w_{12} = u_1 - u_2 < 0 \;\;, \;\; q_{12} = 0 \;, \\
&\text{2-3 isochoric: } w_{23} = 0 \;\;, \;\; q_{23} = u_3 - u_2 > 0 \;, \\
&\text{3-4 isentropic: } w_{34} = u_3 - u_4 > 0 \;\;, \;\; q_{34} = 0 \;, \\
&\text{4-1 isochoric: } w_{41} = 0 \;\;, \;\; q_{41} = u_1 - u_4 < 0 \;.
\end{aligned} \tag{8.18}$$

The thermal efficiency of the Otto cycle is

$$\eta_{Otto} = \frac{w_\odot}{q_{in}} = \frac{w_{12} + w_{23} + w_{34} + w_{41}}{q_{23}} = 1 - \frac{u_4 - u_1}{u_3 - u_2} \;. \tag{8.19}$$

Under cold-air assumptions the internal energy is $u(T) = c_v(T - T_0) + u_0$ so that

$$\eta_{Otto} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{T_1}{T_2}\frac{\frac{T_4}{T_1} - 1}{\frac{T_3}{T_2} - 1} . \tag{8.20}$$

For the two isentropic processes we have, still using the cold-air assumption,

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1} = \left(\frac{V_c + V_s}{V_c}\right)^{k-1} = \left(\frac{v_4}{v_3}\right)^{k-1} = \frac{T_3}{T_4} . \tag{8.21}$$

We conclude that $\frac{T_4}{T_1} = \frac{T_3}{T_2}$, and write the thermal efficiency of the cold-air-standard Otto cycle as

$$\eta_{Otto} = 1 - \frac{1}{r^{k-1}} , \tag{8.22}$$

where we have introduced the compression ratio

$$r = \frac{V_c + V_s}{V_c} . \tag{8.23}$$

The simplified analysis shows that the compression ratio r is the most important parameter for the evaluation of the Otto cycle. Under the cold air assumption the thermal efficiency depends solely on the compression ratio, and it grows with increasing compression ratio. Obviously, one will aim for large compression ratios to have efficient engines.

The compression ratio cannot be arbitrarily large. As the air-fuel-mix is compressed, its temperature increases, $T_2 = T_1 r^{k-1}$. When the temperature of the air-fuel-mix reaches its auto-ignition temperature, it will start to react before the spark plug induces the explosion at the appropriate point in the process. Premature combustion, known as engine knocking, reduces the power output since it leads to increased pressure during compression, and, even more importantly, it damages the engine. To prevent knocking, the compression ratio must be limited to values which guarantee that auto-ignition cannot occur. High grade fuel, i.e., gasoline with larger octane numbers, has a higher auto-ignition temperature, and must be used in engines with larger compression ratios.

Typical values for the compression ratio in Otto engines are between 8 and 12. These values yield efficiencies for the idealized cycle discussed above between 0.565 and 0.63. Real engines have about 30% loss to irreversible processes (friction, heat transfer) within the engine, and another 30% loss to friction in the drive train, so that their actual efficiency is $\eta = 0.7 \times 0.7 \times \eta_{Otto} = 0.28$ (for $r = 8$).

Since the working cycle of the engine happens during only two of the four strokes (recall that inlet and exhaust work cancel), the power delivered by an engine which runs at a frequency $\dot{n}$ is

$$\dot{W} = \frac{\dot{n}}{2} m w_{\odot} \,, \tag{8.24}$$

where $m = p_1 (V_c + V_s) / RT_1$ is the mass of air in the cylinder.

8.6 Example: Otto Cycle

As an example we consider an engine operating on the ideal Otto cycle with a compression ratio $r = 9$. The total swept volume of all cylinders is $V_s = 3\,\text{litres}$, and the engine runs at $\dot{n} = 3000\,\text{rpm}$. The intake is at $T_1 = 300\,\text{K}$, $p_1 = 0.98\,\text{bar}$, and the heat transfer through the combustion process is $q_{23} = 717 \frac{\text{kJ}}{\text{kg}}$. We determine temperatures, pressures and specific volume for the four corner points of the process, and compute heat and work for all subprocesses. For the computation we treat the working fluid as air with constant specific heats (cold-air approximation), $R = 0.287 \frac{\text{kJ}}{\text{kg K}}$, $c_v = 0.717 \frac{\text{kJ}}{\text{kg K}}$, $k = 1.4$.

We begin with the computation of the measurable properties, their numerical values are found in the table below this paragraph. From the ideal gas law we have $v_1 = \frac{RT_1}{p_1}$. Since the process 4-1 is isochoric, we have $v_4 = v_1$, and the other two volumes follow from the compression ratio as $v_2 = v_3 = \frac{v_1}{r}$. The compression process 1-2 is isentropic, so that $p_2 = p_1 r^k$; the ideal gas law gives $T_2 = \frac{p_2 v_2}{R}$. The heat for the isochoric heating process (the explosion) is $q_{23} = u_3 - u_2 = c_v (T_3 - T_2)$, so that $T_3 = T_2 + \frac{q_{23}}{c_v}$; from the ideal gas law $p_3 = \frac{RT_3}{v_3}$. Since the process 3-4 is isentropic, the pressure at the end of the expansion is $p_4 = p_1 r^{-k}$; the temperature follows again from the ideal gas law, $T_4 = \frac{p_4 v_4}{R}$. The numerical values are

	$v/\frac{\text{m}^3}{\text{kg}}$	$T/\,\text{K}$	$p/\,\text{bar}$
1	0.879	300.0	0.98
2	0.098	725.3	21.2
3	0.098	1725.3	50.5
4	0.879	714.0	2.33

The clearance volume follows from the compression ratio $r = \frac{V_c + V_s}{V_c}$ as $V_c = \frac{V_s}{r-1} = 0.375\,\text{litres}$. The mass in the cylinders is $m = \frac{V_c}{v_2} = 3.83\,\text{g}$.

Work and heat for the four processes are, with $u_i - u_j = c_v (T_i - T_j)$,

$$\begin{aligned}
w_{12} &= c_v (T_1 - T_2) = -304.9 \frac{\text{kJ}}{\text{kg}} \,, && q_{12} = 0 \,, \\
w_{23} &= 0 \,, && q_{23} = c_v (T_3 - T_2) = 717 \frac{\text{kJ}}{\text{kg}} \,, \\
w_{34} &= c_v (T_3 - T_4) = 725.1 \frac{\text{kJ}}{\text{kg}} \,, && q_{34} = 0 \,, \\
w_{41} &= 0 \,, && q_{41} = c_v (T_1 - T_4) = -296.8 \frac{\text{kJ}}{\text{kg}} \,.
\end{aligned}$$

The net work for the cycle is

$$w_{\odot} = w_{12} + w_{23} + w_{34} + w_{41} = 420.2 \frac{\text{kJ}}{\text{kg}} ,$$

the power produced is

$$\dot{W} = \frac{\dot{n}}{2} m w_{\odot} = 40.3\,\text{kW} ,$$

and the thermal efficiency of the cycle is

$$\eta = \frac{w_{\odot}}{q_{23}} = 1 - \frac{1}{r^{k-1}} = 0.585 .$$

8.7 Diesel Cycle

In the Diesel cycle, named after its inventor Rudolf Diesel (1858-1913), only air is drawn in and compressed, and the fuel is injected after the compression. Here, one utilizes the temperature increase of the air in compression: as the fuel is injected it starts to burn in the hot compressed air. The compression ratios of Diesel engines can be substantially higher than those of Otto engines, with values of up to $r = 30$.

However, injection of the fuel is slow, and the injected fuel disperses into droplets which are not as well mixed with the air as the gasified fuel in the Otto engine. Therefore combustion is slower than in the Otto engine, the piston moves as the fuel burns. This process can be modelled as an isobaric heat transfer process, so that the Diesel cycle is modelled as follows:

Stroke II:	1-2	adiabatic compression
Stroke III:	2-3	isobaric heating (slow combustion during fuel injection)
	3-4	adiabatic expansion
	4-1	isochoric cooling (exchange of working fluid)

The respective process curves are shown in Fig. 8.7, which also indicates clearance volume, swept volume, and the injection volume V_i, which is the volume at the end of the isobaric heating process. For further analysis we assume reversible processes for which we find heat and work as

$$\begin{array}{lll} \text{1-2 isentropic: } w_{12} = u_1 - u_2 < 0 & , & q_{12} = 0 , \\ \text{2-3 isobaric: } w_{23} = p_3 (v_3 - v_2) > 0 & , & q_{23} = h_3 - h_2 > 0 , \\ \text{3-4 isentropic: } w_{34} = u_3 - u_4 > 0 & , & q_{34} = 0 , \\ \text{4-1 isochoric: } w_{41} = 0 & , & q_{41} = u_1 - u_4 < 0 . \end{array} \quad (8.25)$$

The thermal efficiency of the Diesel cycle is

$$\eta_{Diesel} = \frac{w_{\odot}}{q_{in}} = \frac{w_{12} + w_{23} + w_{34} + w_{41}}{q_{23}} = 1 - \frac{u_4 - u_1}{h_3 - h_2} . \quad (8.26)$$

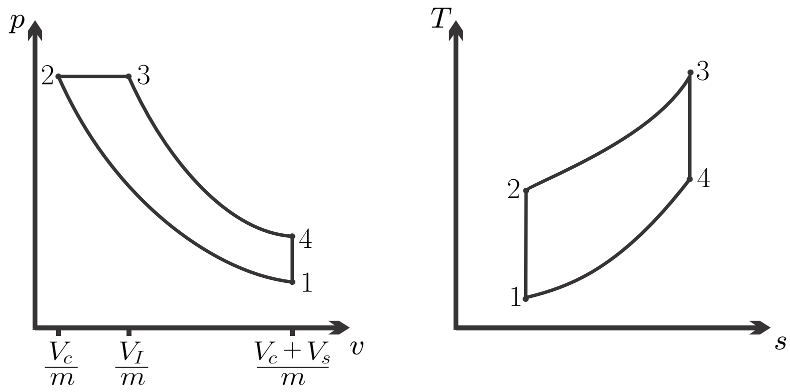


Fig. 8.7 Air-standard Diesel cycle. Note that in the T-s-diagram the isochoric line has a larger slope than the isobaric line.

Under cold-air assumptions energy and enthalpy are $u(T) = c_v(T - T_0) + u_0$, $h(T) = c_p(T - T_0) + h_0$, so that

$$\eta_{Diesel} = 1 - \frac{c_v}{c_p}\frac{T_1}{T_2}\frac{\frac{T_4}{T_1} - 1}{\frac{T_3}{T_2} - 1} . \tag{8.27}$$

For the two isentropic processes we have, still using the cold-air assumption,

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1} = r^{k-1} \quad , \quad \frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{k-1} = \left(\frac{V_4}{V_2}\frac{V_2}{V_3}\right)^{k-1} = \frac{r^{k-1}}{r_c^{k-1}} , \tag{8.28}$$

where $r_c = V_3/V_2 = V_i/V_c$ is the cut-off ratio. For the isobaric process we have (ideal gas law!) $\frac{T_3}{T_2} = \frac{V_3}{V_2} = r_c$, so that the thermal efficiency of the Diesel process becomes[3]

$$\eta_{Diesel} = 1 - \frac{1}{r^{k-1}}\frac{1}{k}\frac{r_c^k - 1}{r_c - 1} . \tag{8.29}$$

For the *same* compression ratio r the Diesel efficiency is below the Otto efficiency. However, the Diesel cycle allows significantly *higher* compression ratios than the Otto cycle, and that is the reason why it has a larger efficiency. For a cycle with cut-off ratio $r_c = 2$ and compression ratio $r = 20$, we find the thermal efficiency as 0.65. Irreversible losses to friction in the engine and drive train reduce the thermal efficiency, actual engines have efficiencies around 37%.

From the p-v-diagram we can see that in a Diesel cycle the maximum pressure is sustained for a longer period, in contrast to the Otto engine, where

[3] With $\frac{T_4}{T_1} = \frac{T_4}{T_3}\frac{T_3}{T_2}\frac{T_2}{T_1} = \frac{r_c^{k-1}}{r^{k-1}} r_c r^{k-1} = r_c^k$.

the maximum pressure is just a short spike. Due to this, Diesel engines must be build more sturdily, which is the main reason why they are more expensive. On the other hand, they are more efficient, and thus require less fuel. The cost of gasoline and Diesel fuel, respectively, also depends on taxation, and thus it might be more cost effective to drive a Diesel powered car in some countries, while an Otto powered car might be more cost effective in other countries.

8.8 Example: Diesel Cycle

An air standard four-stroke Diesel cycle has a clearance volume $V_c = 0.5\,\text{litre}$, and a compression ratio $r = 16.16$. Intake temperature and pressure are $17\,°\text{C}$ and $1\,\text{bar}$, respectively, and the temperature at the end of the expansion is $707\,°\text{C}$. The engine runs at $4400\,\text{rpm}$. We determine temperatures, pressures and specific volume for the four corner points of the process, and compute heat and work for all subprocesses. For the computation we treat the working fluid as air with variable specific heats.

We outline how the various properties are determined, and collect their numerical values in the table below this paragraph. The values for T_1, p_1 and T_4 are given above. From the ideal gas law we have $v_1 = v_4 = \frac{RT_1}{p_1}$; from the compression ratio $v_2 = \frac{v_1}{r}$. The compression process 1-2 is isentropic, so that for the relative volume $v_r(T_2) = \frac{v_r(T_1)}{r}$. With tabulated data for $v_r(T)$ follows T_2, while the ideal gas law gives the pressures $p_2 = p_3 = \frac{RT_2}{v_2}$. With T_4 given, the pressure at 4 follows from the ideal gas law, $p_4 = \frac{RT_4}{v_4}$. The expansion 3-4 is isentropic, so that $p_r(T_3) = p_r(T_4)\frac{p_3}{p_4}$. With tabulated data follows T_3, and from the ideal gas law $v_3 = \frac{RT_3}{p_3}$. With this, all points are identified, and energies and enthalpies can be found from the property table. Altogether we have

	$v/\frac{\text{m}^3}{\text{kg}}$	T/K	p/bar	v_r	p_r	$u/\frac{\text{kJ}}{\text{kg}}$	$h/\frac{\text{kJ}}{\text{kg}}$
1	0.832	290	1.0	319.5		208	
2	0.0515	840	46.81	19.77		626	867
3	0.113	1845	46.81		1075	1531	2060
4	0.832	980	3.38		77.61	743	

The swept volume follows from the compression ratio $r = \frac{V_c + V_s}{V_c}$ as $V_s = V_c(r-1) = 7.58\,\text{litre}$, the injection volume is $V_i = \frac{v_3}{v_2}V_c = 1.1\,\text{litre}$, and the cut-off ratio is $r_c = \frac{V_i}{V_c} = 2.2$. The mass in the cylinders during the working cycle is $m = \frac{V_c}{v_2} = 9.71\,\text{g}$.

Work and heat for the four processes are

$$
\begin{aligned}
w_{12} &= u_1 - u_2 = -418\tfrac{\text{kJ}}{\text{kg}} &, \; q_{12} &= 0\,, \\
w_{23} &= p\,(v_3 - v_2) = 288\tfrac{\text{kJ}}{\text{kg}} &, \; q_{23} &= h_3 - h_2 = 1193\tfrac{\text{kJ}}{\text{kg}}\,, \\
w_{34} &= u_3 - u_4 = 788\tfrac{\text{kJ}}{\text{kg}} &, \; q_{34} &= 0\,, \\
w_{41} &= 0 &, \; q_{41} &= u_1 - u_4 = -535\tfrac{\text{kJ}}{\text{kg}}\,.
\end{aligned}
$$

The net work for the cycle is

$$w_{\odot} = w_{12} + w_{23} + w_{34} + w_{41} = 658\frac{\text{kJ}}{\text{kg}}\,,$$

the power produced is

$$\dot{W} = \frac{\dot{n}}{2} m w_{\odot} = 234.3\,\text{kW}\,,$$

and the thermal efficiency of the cycle is

$$\eta = \frac{w_{\odot}}{q_{23}} = 0.552\,.$$

This is somewhat lower as the efficiency under the cold-air approximation, which is $\eta = 1 - \frac{1}{r^{k-1}}\frac{1}{k}\frac{r_c^k - 1}{r_c - 1} = 0.606$. The difference is due to the effect of temperature dependent specific heats.

8.9 Dual Cycle

In real engines the combustion process is neither just an instantaneous, i.e., isochoric, explosion as assumed above for the Otto cycle, nor is it a slow isobaric combustion as assumed above for the Diesel cycle. To have a somewhat better model, one can modify the description of the fuel combustion process as a combination of isochoric and isobaric heating. The resulting cycle is the *dual cycle*:

Stroke II:	1-2	adiabatic compression
	2-3'	isochoric heating
Stroke III:	3'-3	isobaric heating
	3-4	adiabatic expansion
	4-1	isochoric cooling

Figure 8.8 shows the corresponding diagrams. Depending on the choice of parameters, one can model Otto and Diesel engines as a dual cycle. The dual step would have a more pronounced isochoric and smaller isobaric step for Otto engines, and a smaller isochoric but more pronounced isobaric step for Diesel engines. Moreover, Otto engines have compression ratios below 12, while Diesel engines have higher compression ratios.

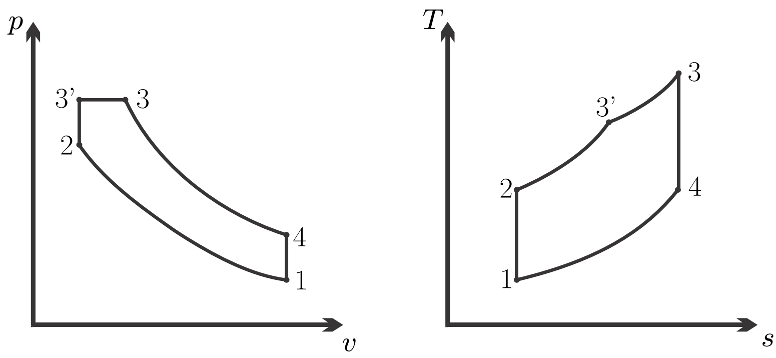


Fig. 8.8 Dual cycle

8.10 Atkinson Cycle

At the end of the expansion process in the Otto and Diesel cycles, the working fluid is at elevated pressure and temperature, and more work could be extracted. One method to use this work is to drive a turbo charger, which essentially is a compressor that is used to fill the cylinders with air-fuel mix (Otto) or air (Diesel) at elevated pressures. With this, the mass of air in the engine is higher than when it only draws air at environmental pressure p_0. Since the power output of the engine is proportional to mass, $\dot{W} = \frac{\dot{n}}{2} m w_{\odot}$, turbo charging increases the power output, or gives the same power from a smaller engine.

An alternative method to make use of the work potential is used in Atkinson engines. The Atkinson cycle is a modification of the Otto cycle where the compression and expansion strokes have different lengths. Specifically, the expansion stroke is longer than the compression stroke, so that—in the best case—the pressure at the end of expansion is equal to the intake pressure. The early engines build by James Atkinson (1846–1914) relied on mechanical valve control. Since valves can be controlled electronically, Atkinson's ideas become more prominent, and engines based on the Atkinson cycle are routinely used in modern hybrid cars.

To achieve the Atkinson cycle in an actual engine one uses clever control of the valves. In the Otto engine the valves close at the end of the intake stroke (Stroke I), and as the piston reverses its direction, compression starts immediately (Stroke II). In an Atkinson engine the valve remains open during the beginning of the second stroke, so that some of the intake is pushed back into the intake manifold. The valve closes a bit later (at volume V_1) and compression commences. Thus, compression occurs only on part of Stroke II. The valves remain closed for the full expansion (Stroke III), so that the expansion stroke is longer than the compression stroke. At the end of expansion, the valves open and the exhaust stroke (Stroke IV) begins. Computerized control

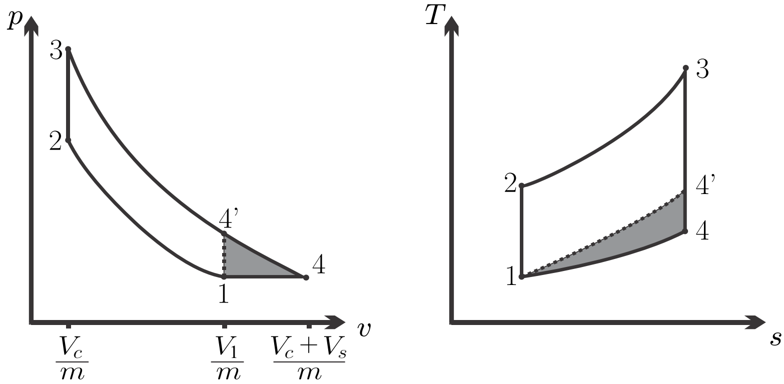


Fig. 8.9 Atkinson cycle 1-2-3-4-1 compared to Otto cycle 1-2-3-4'-1. The shaded area is the extra work delivered by the Atkinson cycle for the same heat input.

of the valve timing allows to vary the length of the compression stroke to optimize engine performance for the current driving conditions.

The processes in the ideal air-standard Atkinson cycle are:

1-2 adiabatic compression
2-3 isochoric heating (air-fuel mix explodes)
3-4 adiabatic expansion
4-1 isobaric cooling (exchange of working fluid)

The corresponding process diagrams are depicted in Fig. 8.9. The figure also indicates the point $4'$ at which the expansion would finish in the Otto cycle. The shaded areas in the two diagrams indicate the additional work generated in the Atkinson cycle as compared to the Otto cycle. The compression ratio is $r = \frac{V_1}{V_c}$ and the expansion ratio is $r_e = \frac{V_c+V_s}{V_c}$.

Work and heat for the processes are

$$\begin{array}{lll} \text{1-2 isentropic: } w_{12} = u_1 - u_2 < 0 & , & q_{12} = 0 \, , \\ \text{2-3 isochoric: } w_{23} = 0 & , & q_{23} = u_3 - u_2 > 0 \, , \\ \text{3-4 isentropic: } w_{34} = u_3 - u_4 > 0 & , & q_{34} = 0 \, , \\ \text{4-1 isobaric: } w_{41} = p_1 \left(v_1 - v_4 \right) < 0 & , & q_{41} = h_1 - h_4 < 0 \, . \end{array} \tag{8.30}$$

Accordingly, the net work and the thermal efficiency of the engine are

$$w_{\odot} = w_{12} + w_{23} + w_{34} + w_{41} = h_1 - u_2 + u_3 - h_4 \, , \tag{8.31}$$

and

$$\eta_{Atkinson} = \frac{w_{\odot}}{q_{23}} = 1 - \frac{h_4 - h_1}{u_3 - u_2} \, . \tag{8.32}$$

As before, we evaluate the process under the cold-air approximation, where

$$h_4 - h_1 = c_p (T_4 - T_1) \quad , \quad u_3 - u_2 = c_v (T_3 - T_2) \ , \tag{8.33}$$

so that

$$\eta_{Atkinson} = 1 - k\frac{T_4 - T_1}{T_3 - T_2} \ . \tag{8.34}$$

For the temperatures at the corner points of the process we find

$$T_2 = T_1 r^{k-1} \quad , \quad T_4 = T_1 \frac{v_4}{v_1} = T_1 \frac{r_e}{r} \quad , \quad T_3 = T_4 r_e^{k-1} = T_1 \frac{r_e^k}{r} \ , \tag{8.35}$$

so that net work and thermal efficiency become

$$w_{\odot} = c_v T_1 \left[k \left(1 - \frac{r_e}{r} \right) + \frac{r_e^k - r^k}{r} \right] \ , \tag{8.36}$$

$$\eta_{Atkinson} = 1 - \frac{1}{r^{k-1}} k \frac{\frac{r_e}{r} - 1}{\left(\frac{r_e}{r}\right)^k - 1} \ . \tag{8.37}$$

It is easy to see that for $r_e > r$ the factor $\left[k \frac{\frac{r_e}{r} - 1}{\left(\frac{r_e}{r}\right)^k - 1} \right]$ is less than unity. Therefore, the thermal efficiency of the Atkinson cycle is larger than that of the Otto cycle with the same compression ratio.

In order to have the ideal Atkinson cycle performed, the heat addition q_{23} must be such that $T_3 = T_1 \frac{r_e^k}{r}$. If more heat is added, so that the temperature T_3 lies above this value, the pressure p_4 at the end of expansion will be above the inlet pressure p_1. In this case, the cooling process (which models exhaust and intake) would include an isochoric process:

1-2	adiabatic compression
2-3	isochoric heating
3-4'	adiabatic expansion
4'-4	isochoric cooling
4-1	isobaric cooling

The corresponding process diagrams are shown in Fig. 8.10. Work and heat for the processes are

$$\begin{array}{llll} \text{1-2} & \text{isentropic:} & w_{12} = u_1 - u_2 < 0 & , q_{12} = 0 \\ \text{2-3} & \text{isochoric:} & w_{23} = 0 & , q_{23} = u_3 - u_2 > 0 \\ \text{3-4'} & \text{isentropic:} & w_{34'} = u_3 - u_{4'} > 0 & , q_{34'} = 0 \\ \text{4'-4} & \text{isochoric} & w_{4'4} = 0 & q_{4'4} = u_4 - u_{4'} < 0 \\ \text{4-1} & \text{isobaric:} & w_{41} = p_1 (v_1 - v_4) < 0 & , q_{41} = h_1 - h_4 < 0 \end{array} \tag{8.38}$$

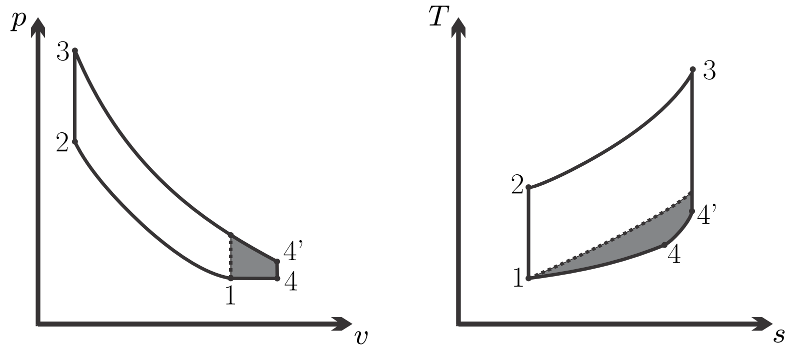


Fig. 8.10 Overheated Atkinson cycle with excess pressure at the end of expansion

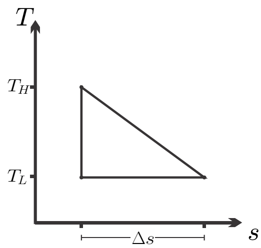
The thermal efficiency of the overheated Atkinson cycle is below that of the ideal Atkinson cycle. In the cold-air approximation one finds

$$\eta = 1 - \frac{1}{r^{k-1}} \left[k \frac{T_3 r_e^{1-k} - T_1}{T_3 r^{1-k} - T_1} + (k-1) \frac{T_3 r_e^{1-k} - T_1 \frac{r_e}{r}}{T_3 r^{1-k} - T_1} \right] , \tag{8.39}$$

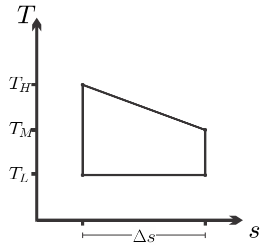
which reduces to (8.37) for $T_3 = T_1 \frac{r_e^k}{r}$.

Problems

8.1. Power Cycle in the T-s-Diagram
Compute the efficiency of the cycle in the sketch, and compare with the efficiency of the Carnot cycle.



8.2. Refrigeration Cycle in the T-s-Diagram
A closed system runs on the process indicated in the sketch as a refrigeration cycle. Is the cycle running clockwise or counterclockwise? Use the sketch to determine the COP and compare with the Carnot cycle. Draw the p-v-diagram of the cycle for the case that the working fluid is an ideal gas. Assume $T_M = (T_H + T_L)/2$.



8.3. Inventors

Two inventors have developed heat engines that operate between the temperatures of 900 K and 300 K. One inventor claims an efficiency of 50% for his engine, the other claims an efficiency of 66%. If you had some money to invest, which inventors start-up would you invest in? Explain!

8.4. Carnot Heat Engine

A heat engine operates on a fully reversible Carnot cycle between two reservoirs at 25 °C and 527 °C. The engine contains 4 g of air, and the largest volume is 2 litres. The pressure ratio for isothermal compression is equal to the pressure ratio for isentropic compression.

1. Draw the process diagrams, and make a table with temperature, pressure, specific volume, and specific internal energy at the corner points of the process.
2. Determine specific heat and work for the four processes in the cycle, the net work for the cycle, and the thermal efficiency.
3. When the engine runs at 350 rpm, determine the power produced.

8.5. Carnot Cycle

A Carnot power cycle using hydrogen as working fluid operates between the temperatures 320 K and 1000 K with maximum and minimum pressures given by 40 bar and 0.2 bar, respectively. The maximum volume of the hydrogen in the engine is 6.65 litres and the engine runs at 700 rpm. Draw the process in a T-s-diagram and in a p-v-diagram, then make a table with pressures, specific volumes, internal energies, and entropies at all corner points of the process. Compute the thermal efficiency, the mass of hydrogen in the engine, the maximum volume ratio, and the power produced. Based on the data obtained, discuss the feasibility of the process (apart from the fact that it would be impossible to build a fully reversible engine).

8.6. Carnot Heat Pump

A food drying unit requires a heating power of 200 kW which must be supplied at $T_H = 75\,°\mathrm{C}$, while the exterior temperature is $T_L = 15\,°\mathrm{C}$. For this purpose consider a closed system Carnot heat pump using xenon (ideal gas, monatomic, $M = 131.3\frac{\mathrm{kg}}{\mathrm{kmol}}$) as working fluid. The ratio between maximum and minimum volume of the unit is 40, and the largest pressure that can occur is 80 bar.

1. Draw the process in a T-s-diagram and in a p-v-diagram.
2. Make a table with pressures, temperatures and specific volumes, at all corner points.
3. Compute heat and work for all processes. Determine the net work and the COP.
4. When the engine operates at 500 rpm, determine the mass of xenon in the engine and the maximum volume (which could be distributed over several cylinders). Based on the data obtained, discuss the feasibility of the process.

8.7. Otto Cycle

The air entering a 3 litre Otto engine (ideal air-standard cycle, air as ideal gas with variable specific heats) with compression ratio $r = 9.2$ is at environmental conditions (98 kPa, 300 K). After the heat supply, the pressure has doubled. Determine heat added, net work output, thermal efficiency, and power output when the engine runs at 2000 rpm. Draw diagrams.

8.8. Otto Engine

An air standard four-stroke Otto engine has a swept volume of 2.5 litres, and a clearance volume of 0.4 litres. Temperature and pressure at intake are given by $T_1 = 290\,\text{K}$, $p_1 = 0.7\,\text{bar}$. The temperature at the end of combustion is $T_3 = 1400\,\text{K}$. Consider the working fluid to be air, as ideal gas with $R = 0.287\frac{\text{kJ}}{\text{kg K}}$, and with variable specific heats.

1. Draw the p-V- and T-s-diagrams for the process.
2. Determine the values of temperature, pressure, specific volume, and internal energy at the corners of the cycle. Make a table with these values.
3. Determine the net work per unit mass and the thermal efficiency.
4. Determine the mass of air in the cylinders and net power output of the engine when it runs at 4500 rpm.

8.9. Otto Engine

An air standard four-stroke Otto engine has a compression ratio of 9.4 and a clearance volume of 0.3 litre. Temperature and pressure at intake are $T_1 = 280\,\text{K}$, $p_1 = 1.1\,\text{bar}$, and the pressure after expansion is $p_4 = 2.984\,\text{bar}$.

Consider the working fluid to be air, as ideal gas with variable specific heats.

1. Draw the p-V- and T-s-diagrams for the process.
2. Determine the values of temperature, pressure, specific volume, and internal energy at the corners of the cycle. Make a table with these values.
3. Determine the net work per unit mass and the thermal efficiency.
4. Determine the mass of air in the cylinders and net power output of the engine when it runs at 1200 rpm.

8.10. Diesel Engine

An air standard four-stroke Diesel engine has a swept volume of 7 litres, and a clearance volume of 0.5 litre. The volume at the end of the fuel injection (isobaric heat supply) is 1 litre.

Temperature and pressure at intake are given by $T_1 = 300\,\mathrm{K}$, $p_1 = 1\,\mathrm{bar}$. Consider the working fluid to be air, as ideal gas with constant specific heats, $R = 287\frac{\mathrm{J}}{\mathrm{kg\,K}}$, $c_v = \frac{5}{2}R$, $k = \frac{c_p}{c_v} = 1.4$.

1. Draw a p-V-diagram of the process and then determine:
2. The values of all temperatures, pressures and specific volumes at the corners of the cycle. Make a table with these values.
3. The net work per unit mass and the thermal efficiency.
4. The mass of air in the cylinder and net power output of the engine when it runs at 2500 rpm.

8.11. Diesel Cycle

A 16 cylinder 170-litre Diesel engine operating on the ideal air-standard Diesel cycle has a compression ratio of 17, and a cut-off ratio of 2.2. Determine the amount of power delivered when the engine runs at 900 rpm based on the air-standard cycle, under cold-air assumption (that is: constant specific heats). Consider the following two cases for outside temperature and pressure: (a) $T_0 = 280\,\mathrm{K}$, $p = 1\,\mathrm{bar}$. (b) $T = 305\,\mathrm{K}$, $p = 0.9\,\mathrm{bar}$. Draw diagrams.

8.12. Diesel and Otto cycle

Draw a schematic, and the process curves in a p-V-diagram and a T-s-diagram for a Diesel and an Otto cycle. Mark swept volume, clearance volume etc. These are four-stroke engines: what are the four strokes? Indicate them in the diagram. Discuss the difference between Diesel and Otto cycles. Why can the Diesel have a higher compression?

8.13. Dual Cycle

The processes in a 4-stroke Diesel cycle with compression ratio of 14 are modeled as dual cycle with the following data: The engine draws air at 1 bar, 27 °C, the maximum temperature reached in the cycle is 2200 K, and the total amount of heat added is $q_{23} = 1520.4\frac{\mathrm{kJ}}{\mathrm{kg}}$. The working fluid can be considered as air (ideal gas, variable specific heats).

1. Draw the process in a p-V-diagram, and a T-s-diagram. Include intake and exhaust, and indicate the 4 strokes in the p-V-diagram.
2. Determine temperatures, pressures, and specific volumes in the points 1,2,3',3,4.
3. Determine the thermal efficiency of the cycle.

8.14. Atkinson Engine

A four-stroke-engine operating on the Atkinson cycle draws air at 0.9 bar, 17 °C. The working cycle consists of the following reversible processes:

1-2: Adiabatic compression of air with compression ratio 8.6
2-3: Isochoric heating to $T_3 = 1500\,\mathrm{K}$
3-4: Adiabatic expansion
4-1: Isobaric cooling

Consider the working fluid to be air, as ideal gas with $R = 0.287\frac{\mathrm{kJ}}{\mathrm{kg\,K}}$, and variable specific heats.

1. Draw p-v- and T-s-diagram for the cycle.
2. Make a list with temperature, pressure, specific volume at the four corner points of the cycle.
3. Determine the expansion ratio of the engine.
4. Determine net work and thermal efficiency of the cycle.
5. The engine runs at 2000 rpm and the engine delivers 28.75 kW. Determine the gas volume at bottom dead center.

8.15. Atkinson Engine

A four-stroke-engine operating on the ideal Atkinson cycle draws air at 0.9 bar, 17 °C. The working cycle consists of the following reversible processes:

1-2: Adiabatic compression of air with compression ratio 10.08.
2-3: Isochoric heating.
3-4: Adiabatic expansion with expansion ratio 16.
4-1: Isobaric cooling.

Consider the working fluid to be air, as ideal gas with $R = 0.287 \frac{\mathrm{kJ}}{\mathrm{kg\,K}}$, and variable specific heats.

1. Draw p-V- and T-s-diagram for the cycle.
2. Make a list with specific volume, temperature, pressure, at the four corner points of the cycle.
3. Determine net work and thermal efficiency of the cycle.
4. The engine runs at 2000 rpm and the engine delivers 18 kW. Determine the mass in the cylinders, and the swept volume.

8.16. Overheated Atkinson Engine

Show that the thermal efficiency of the overheated Atkinson cycle under the cold-air approximation is given by Eq. (8.39).

8.17. Overheated Atkinson Engine

An overheated 4-stroke Atkinson cycle has a compression ratio of 10, and an expansion ratio of 14.09; its swept volume is 1.2 litres. The cycle draws air at $p = 0.9\,\mathrm{bar}$, $T_0 = 0\,°\mathrm{C}$, and the total heat rejected into the environment is $397.38 \frac{\mathrm{kJ}}{\mathrm{kg}}$. Assume air-standard conditions, with air as ideal gas with variable specific heat, and reversible processes.

1. Draw the process into p-V- and T-s-diagrams.
2. Determine pressure, temperature, specific volume, specific internal energy and specific enthalpy at all relevant process points.
3. Determine the net work, the heat addition, and the thermal efficiency of the cycle.
4. The engine runs at 1750 rpm, determine the power output.