APPLIED THERMAL ENGINEERING (UMT303)

Air Standard Cycles

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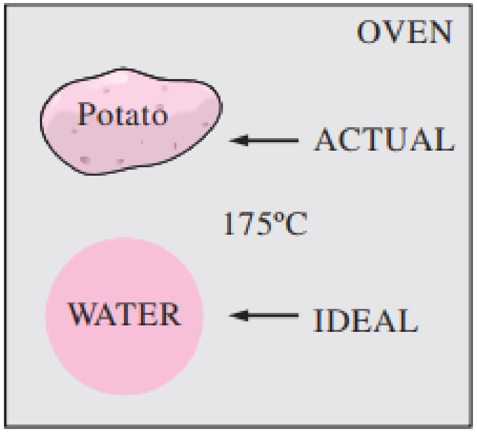
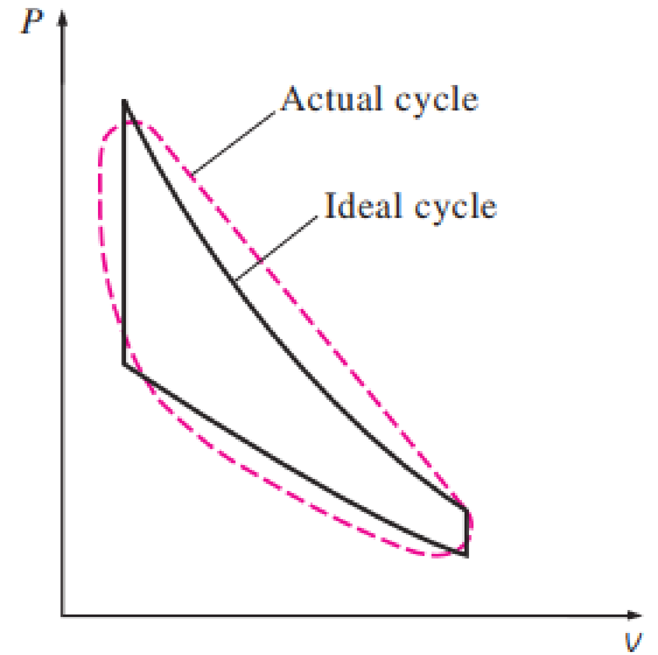
BASIC CONSIDERATIONS IN THE ANALYSIS OF POWER CYCLES



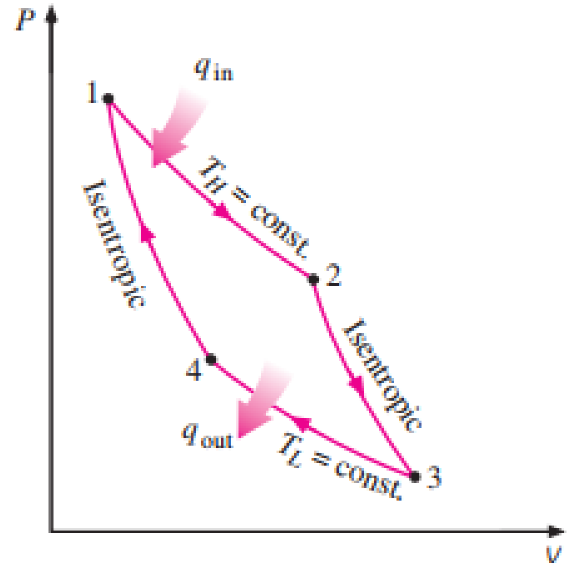


$$\eta_{th} = \frac{W_{net}}{Q_{in}} \quad \text{or} \quad \eta_{th} = \frac{w_{net}}{q_{in}}$$

The idealizations and simplifications commonly employed in the analysis of power cycles can be summarized as follows:

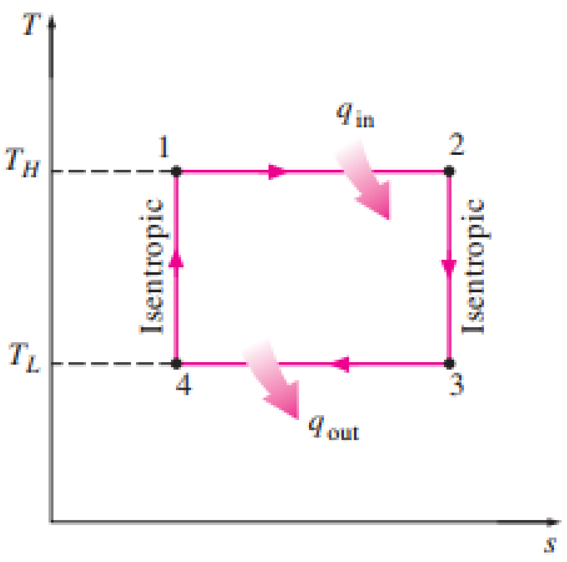
1. The cycle does not involve any *friction.* Therefore, the working fluid does not experience any pressure drop as it flows in pipes or devices such as heat exchangers.
2. All expansion and compression processes take place in a *quasiequilibrium* manner.
3. The pipes connecting the various components of a system are well insulated, and *heat transfer* through them is negligible

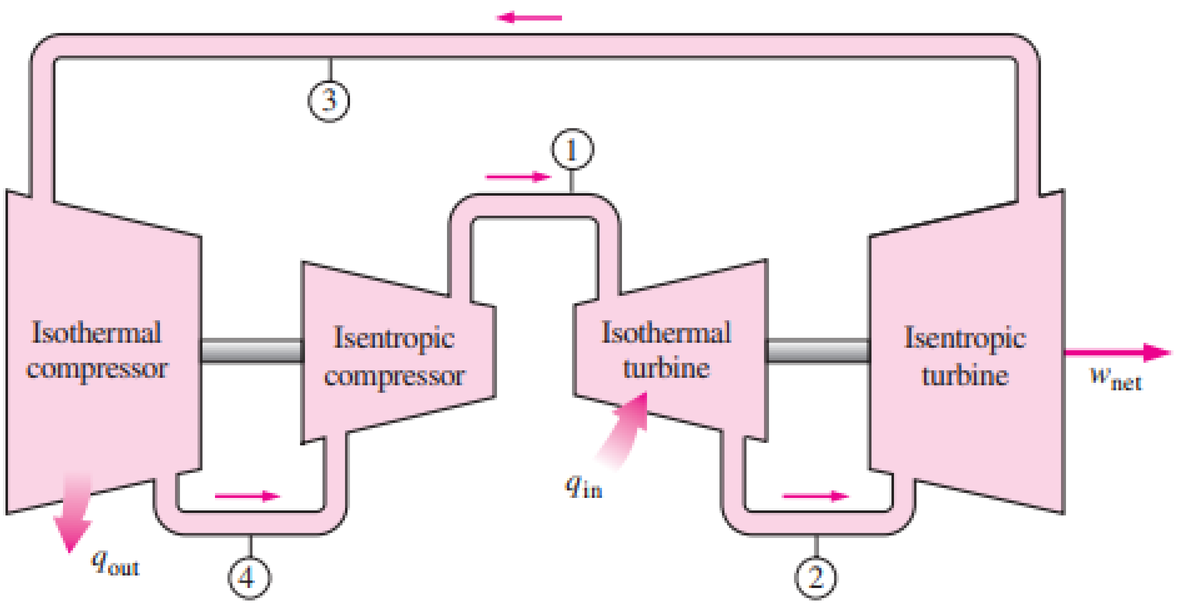
THE CARNOT CYCLE AND ITS VALUE IN ENGINEERING



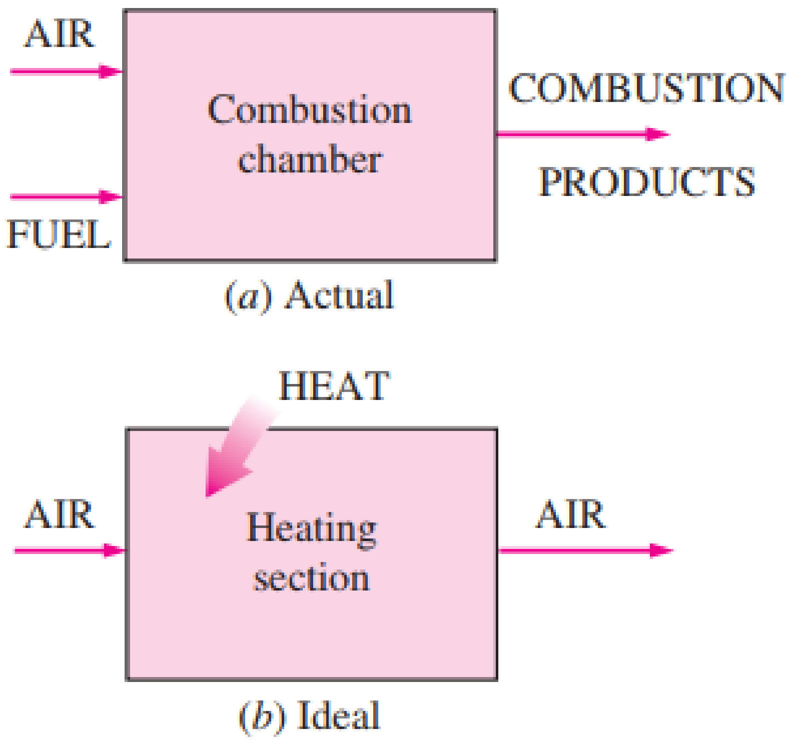
Nicolas Léonard Sadi Carnot
(1796–1832)

$$\boxed{\eta_{\text{th,Carnot}} = 1 - \frac{T_L}{T_H}}$$





AIR-STANDARD ASSUMPTIONS



The actual gas power cycles are rather complex. To reduce the analysis to a manageable level, we utilize the following approximations, commonly known as the **air-standard assumptions:**

1. The working fluid is air, which continuously circulates in a closed loop and always behaves as an ideal gas.
2. All the processes that make up the cycle are internally reversible.
3. The combustion process is replaced by a heat-addition process from an external source.
4. The exhaust process is replaced by a heat-rejection process that restores the working fluid to its initial state.

Another assumption that is often utilized to simplify the analysis even more is that air has constant specific heats whose values are determined at *room temperature* (25°C, or 77°F). When this assumption is utilized, the air-standard assumptions are called the **cold-air-standard assumptions**.
A cycle for which the air-standard assumptions are applicable is frequently referred to as an **air-standard cycle**

AN OVERVIEW OF RECIPROCATING ENGINES

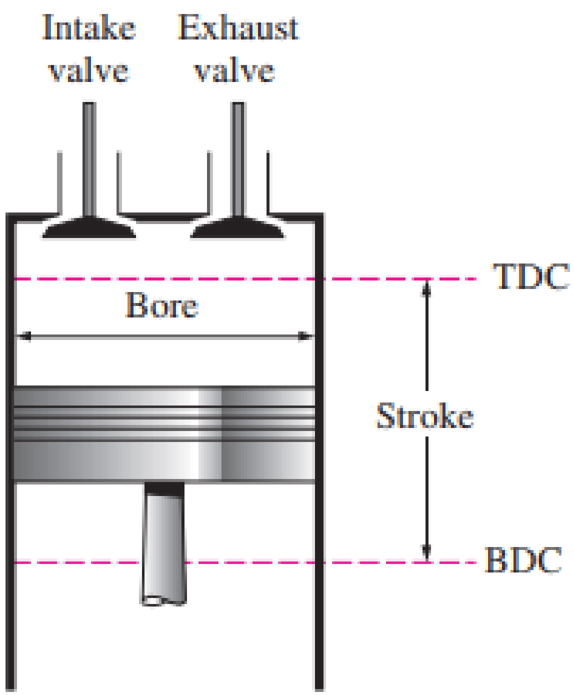
compression ratio

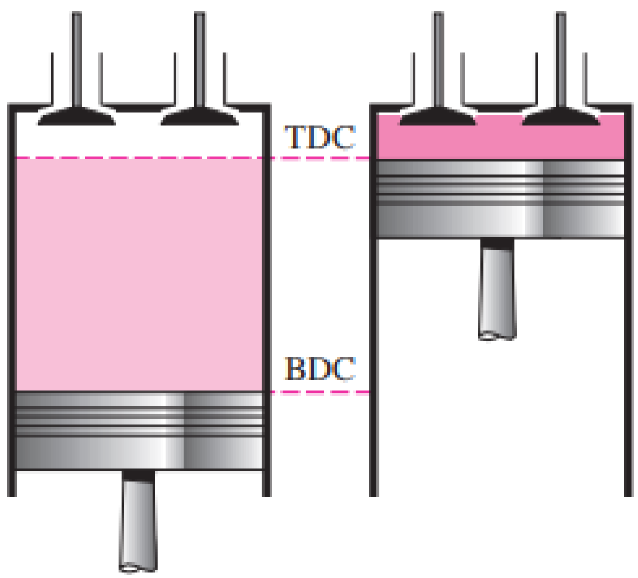
$$r = \frac{V_{max}}{V_{min}} = \frac{V_{BDC}}{V_{TDC}}$$

mean effective pressure (MEP)

$$W_{net} = \text{MEP} \times \text{Piston area} \times \text{Stroke} = \text{MEP} \times \text{Displacement volume}$$

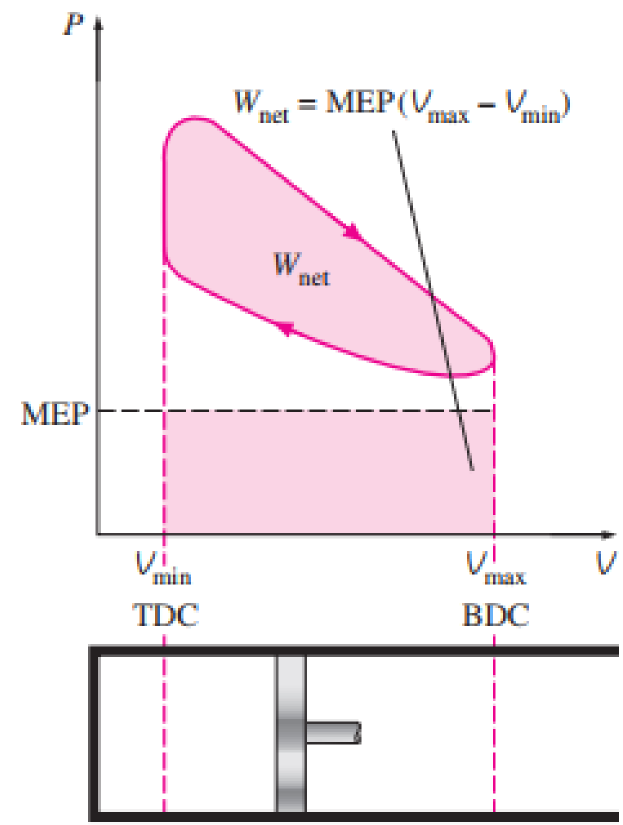
$$\text{MEP} = \frac{W_{net}}{V_{max} - V_{min}} = \frac{w_{net}}{v_{max} - v_{min}} \quad \text{(kPa)}$$





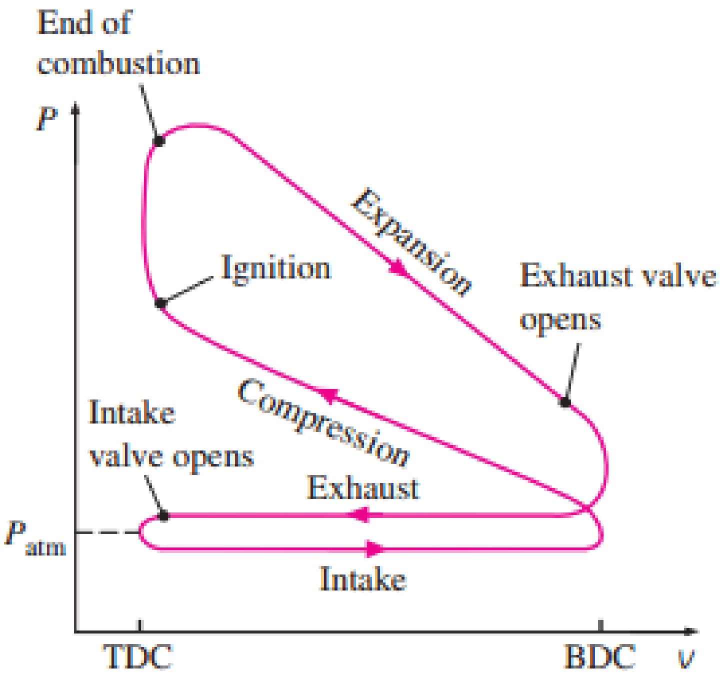
(a) Displacement volume

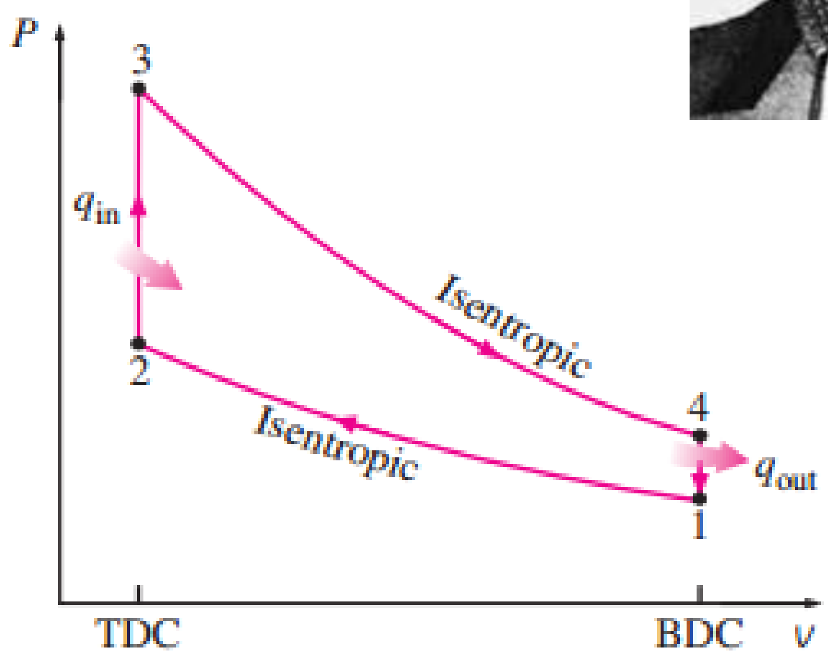
(b) Clearance volume



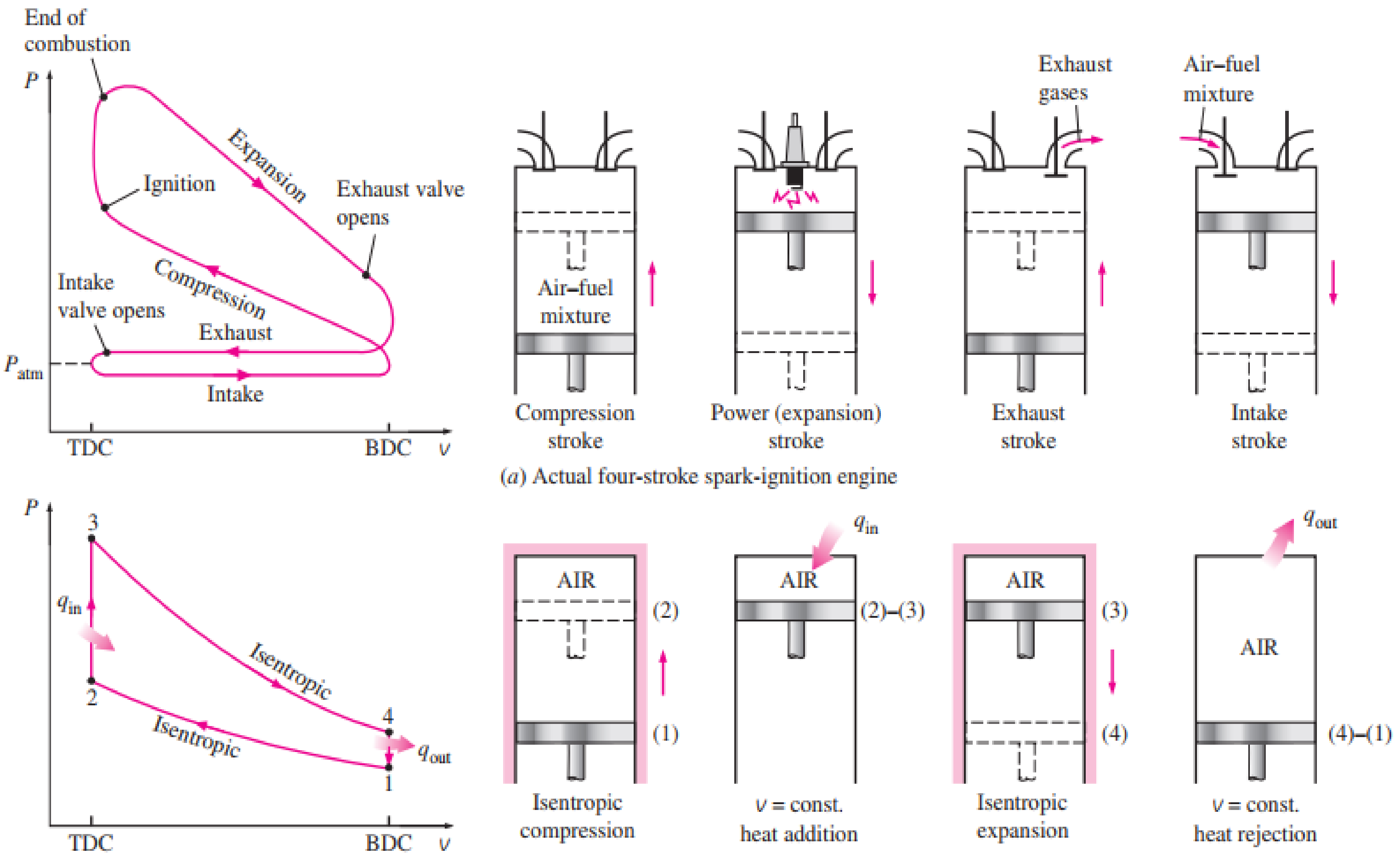
OTTO CYCLE – THE IDEAL CYCLE FOR SPARK-IGNITION (Petrol) ENGINES

Nicolaus August Otto
(1832 -1891)





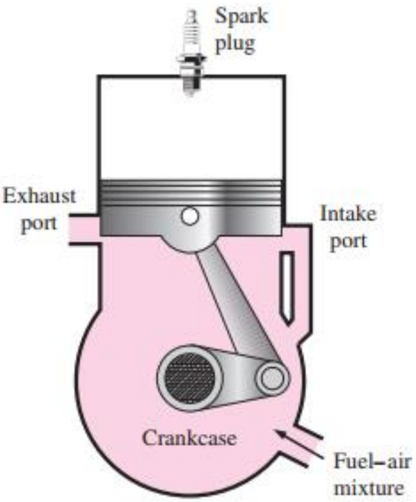
OTTO CYCLE – THE IDEAL CYCLE FOR SPARK-IGNITION (Petrol) ENGINES



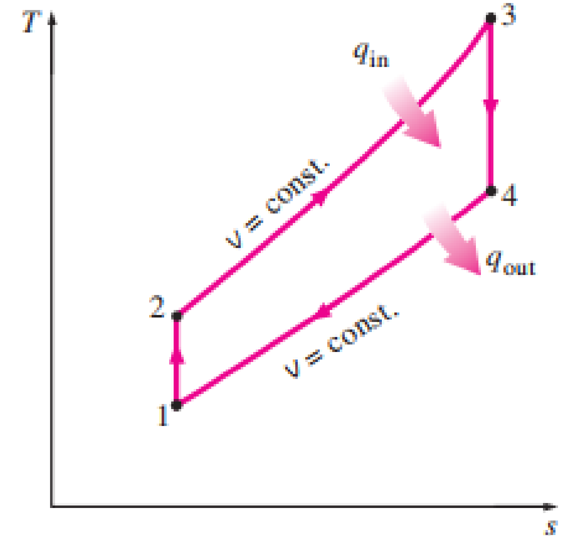
(*a*) Actual four-stroke spark-ignition engine

(*b*) Ideal Otto cycle

OTTO CYCLE – THE IDEAL CYCLE FOR SPARK-IGNITION (Petrol) ENGINES



1-2 Isentropic compression
2-3 Constant-volume heat addition
3-4 Isentropic expansion
4-1 Constant-volume heat rejection



Two-stroke engines are commonly used in motorcycles and lawn mowers.

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = \Delta u \qquad \text{(kJ/kg)}$$

$$q_{in} = u_3 - u_2 = c_v(T_3 - T_2)$$
$$q_{out} = u_4 - u_1 = c_v(T_4 - T_1)$$

$$\eta_{th,Otto} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)}$$

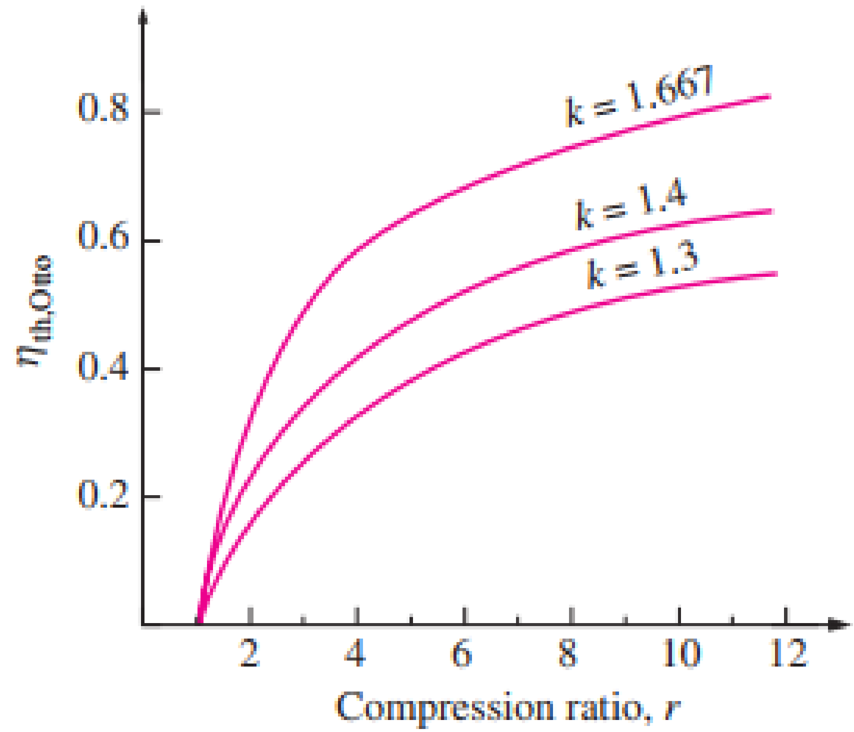
$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{k-1} = \left(\frac{v_3}{v_4}\right)^{k-1} = \frac{T_4}{T_3}$$

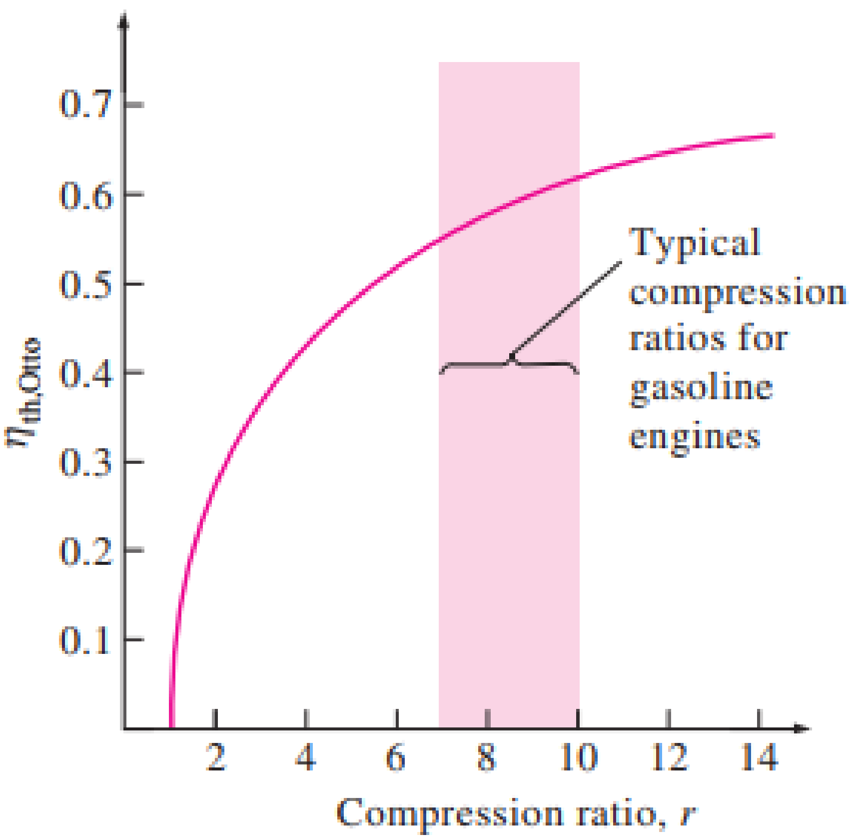
$$\boxed{\eta_{th,Otto} = 1 - \frac{1}{r^{k-1}}}$$

$$r = \frac{V_{max}}{V_{min}} = \frac{V_1}{V_2} = \frac{v_1}{v_2}$$

OTTO CYCLE – THE IDEAL CYCLE FOR SPARK-IGNITION (Petrol) ENGINES

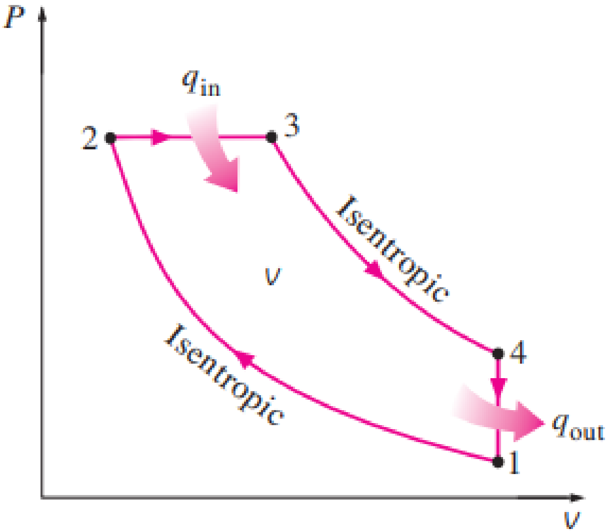
The thermal efficiency of the Otto cycle increases with the specific heat ratio *k* of the working fluid.





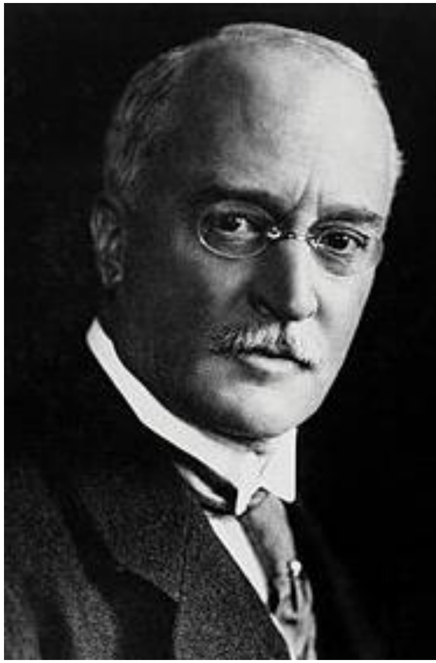
Thermal efficiency of the ideal Otto cycle as a function of compression ratio ($k = 1.4$)

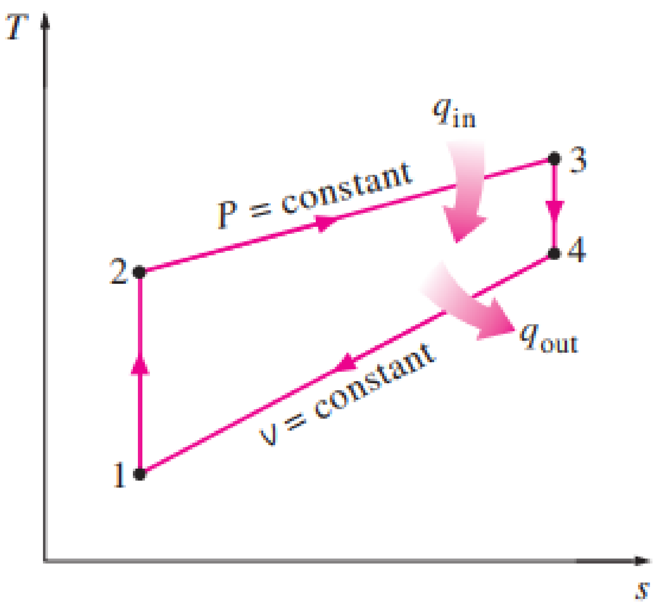
DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES



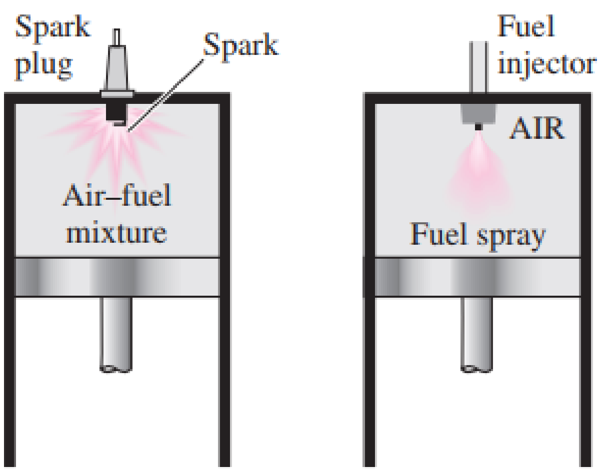
(a) P- v diagram

Rudolf Diesel
(1858-1913)



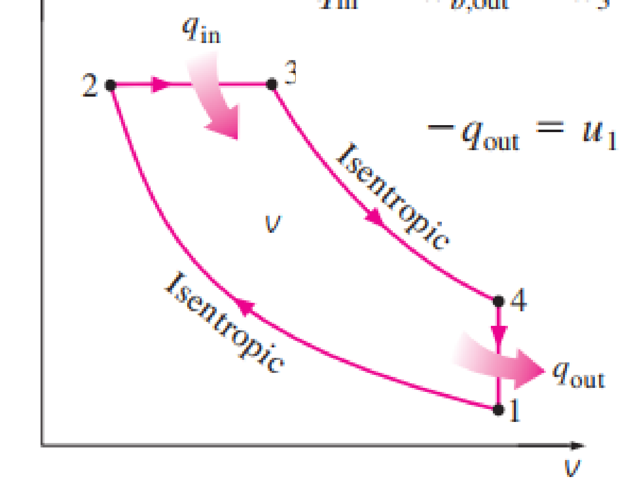


(b) T-s diagram



DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES

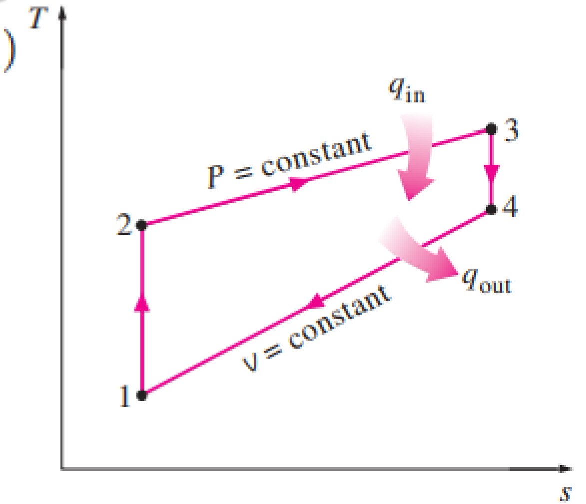
$$q_{in} - w_{b,out} = u_3 - u_2 \rightarrow q_{in} = P_2(v_3 - v_2) + (u_3 - u_2)$$
$$= h_3 - h_2 = c_p(T_3 - T_2)$$
$$-q_{out} = u_1 - u_4 \rightarrow q_{out} = u_4 - u_1 = c_v(T_4 - T_1)$$



(a) P- v diagrai

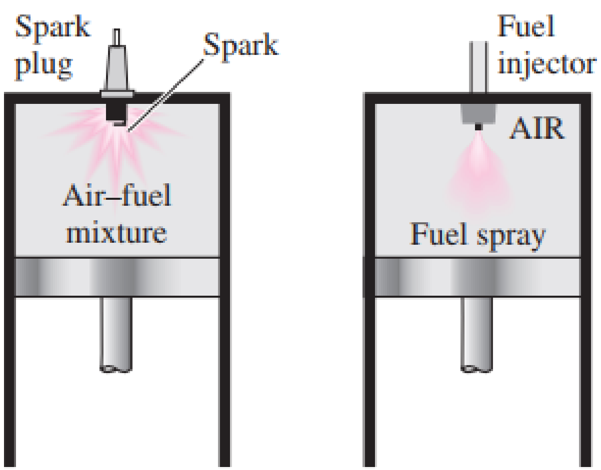
cutoff ratio

$$r_c = \frac{V_3}{V_2} = \frac{v_3}{v_2}$$



(b) T-s diagram

$$\eta_{th,Diesel} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_4 - T_1}{k(T_3 - T_2)} = 1 - \frac{T_1(T_4/T_1 - 1)}{kT_2(T_3/T_2 - 1)}$$



Gasoline engine

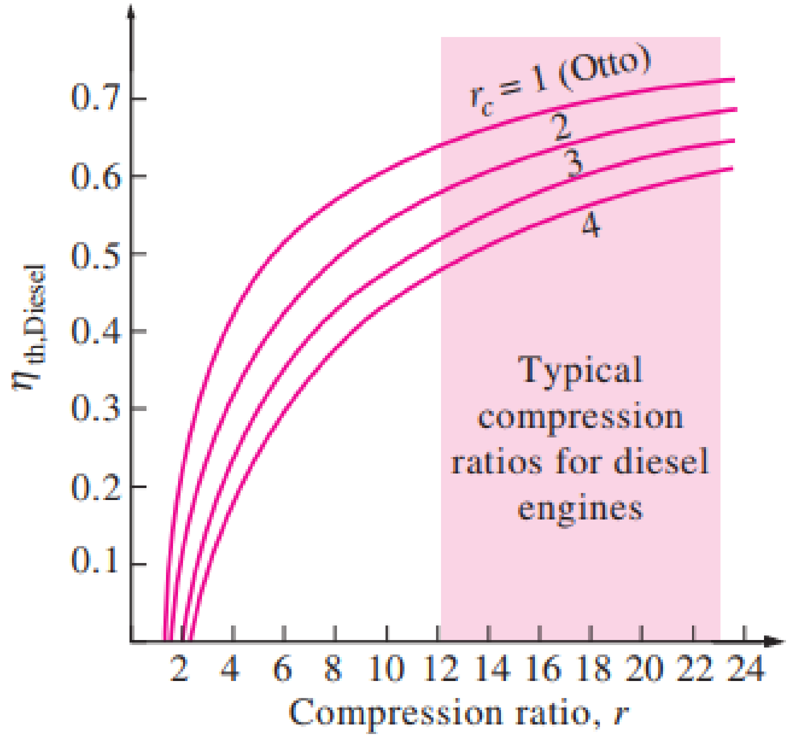
Diesel engine

$$\boxed{\eta_{th,Diesel} = 1 - \frac{1}{r^{k-1}}\left[\frac{r_c^k - 1}{k(r_c - 1)}\right]}$$

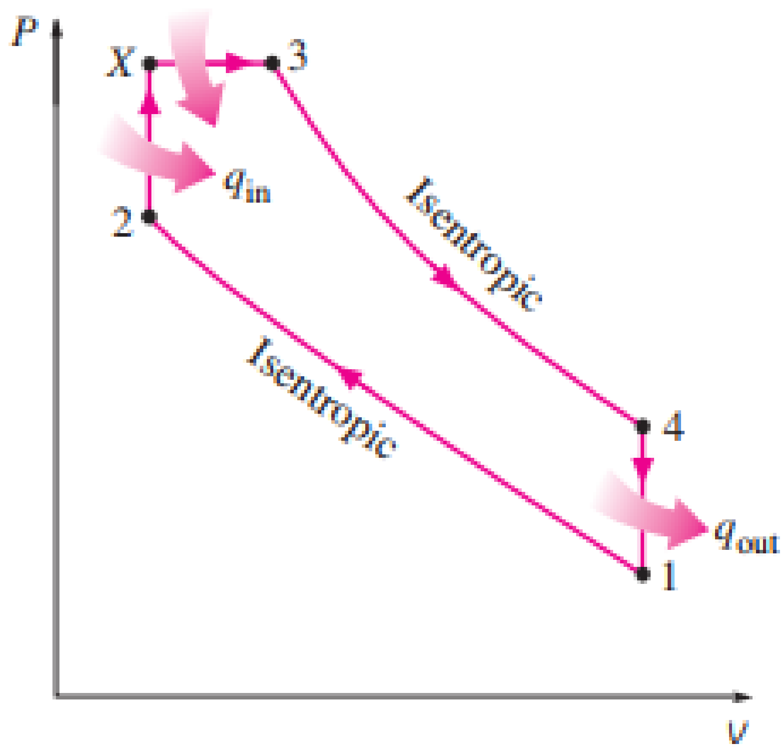
In diesel engines, the spark plug is replaced by a fuel injector, and only air is compressed during the compression process.

DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES

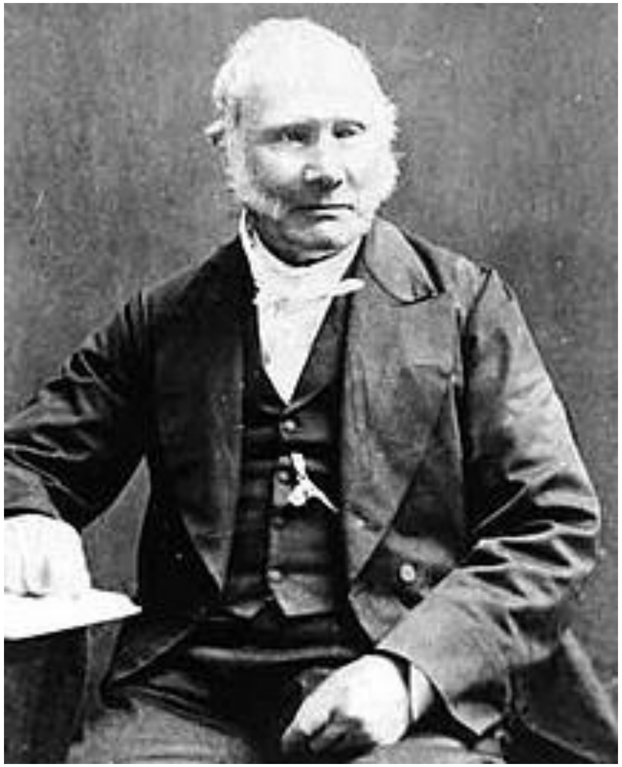
Thermal efficiency of the ideal Diesel cycle as a function of compression and cutoff ratios ($k = 1.4$)



Dual Cycle



STIRLING AND ERICSSON



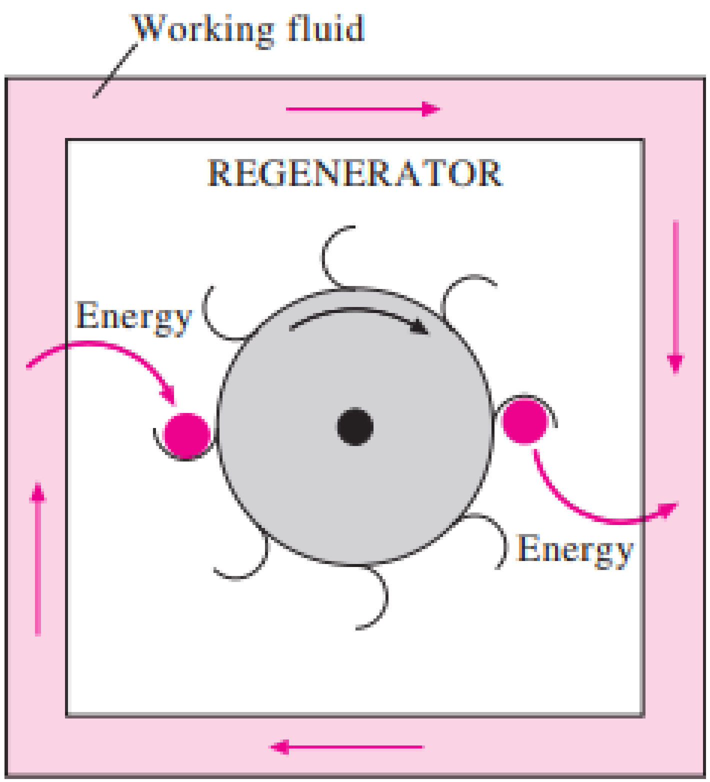
Robert Stirling
(1790-1878)

John Ericsson
(1803-1889)

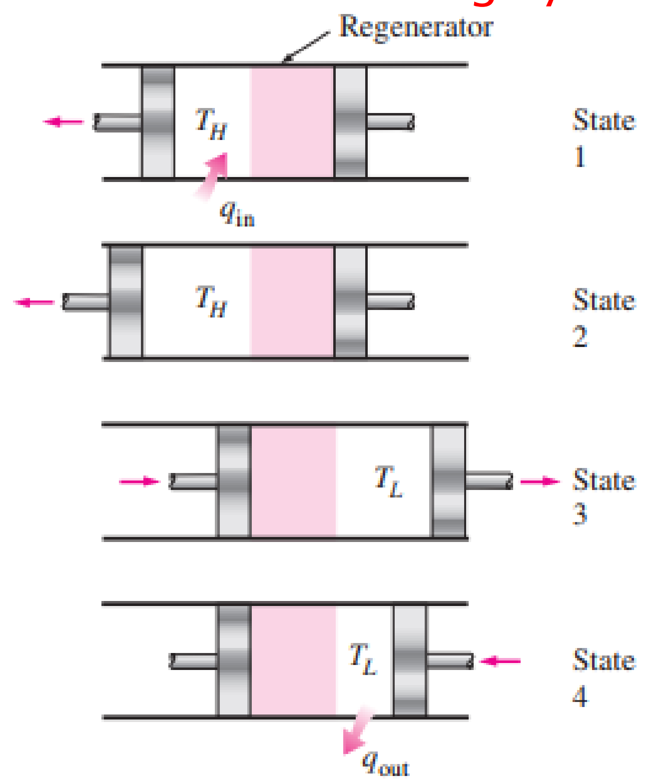
STIRLING AND ERICSSON CYCLES

$$\eta_{\text{th,Stirling}} = \eta_{\text{th,Ericsson}} = \eta_{\text{th,Carnot}} = 1 - \frac{T_L}{T_H}$$

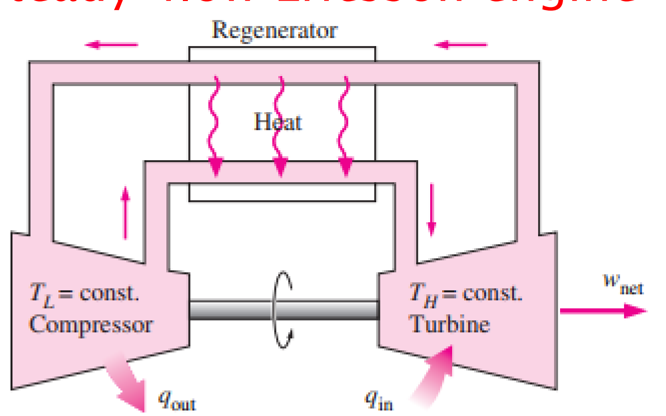
A regenerator is a device that borrows energy from the working fluid during one part of the cycle and pays it back (without interest) during another part.



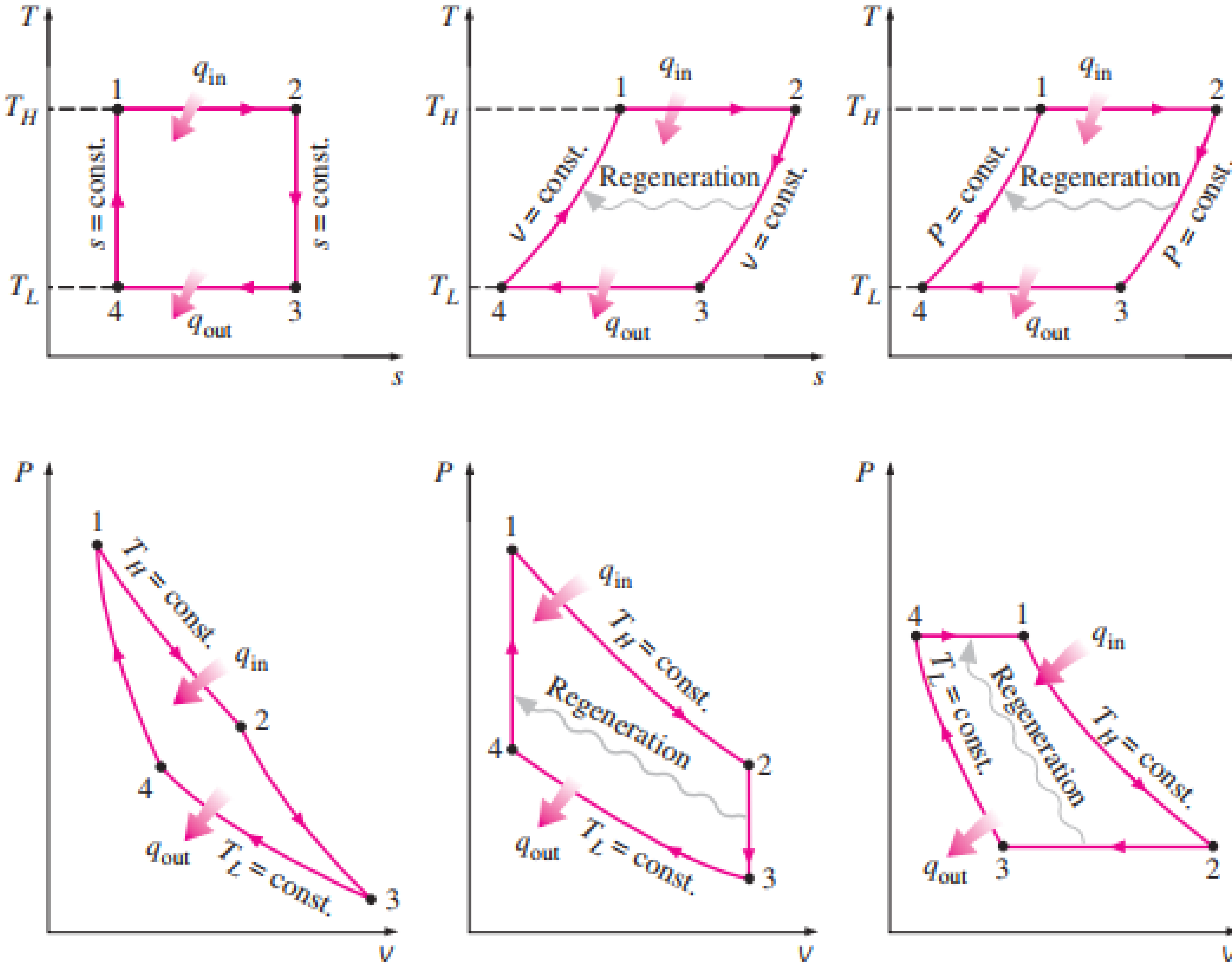
The execution of the Stirling cycle



A steady-flow Ericsson engine



STIRLING AND ERICSSON CYCLES

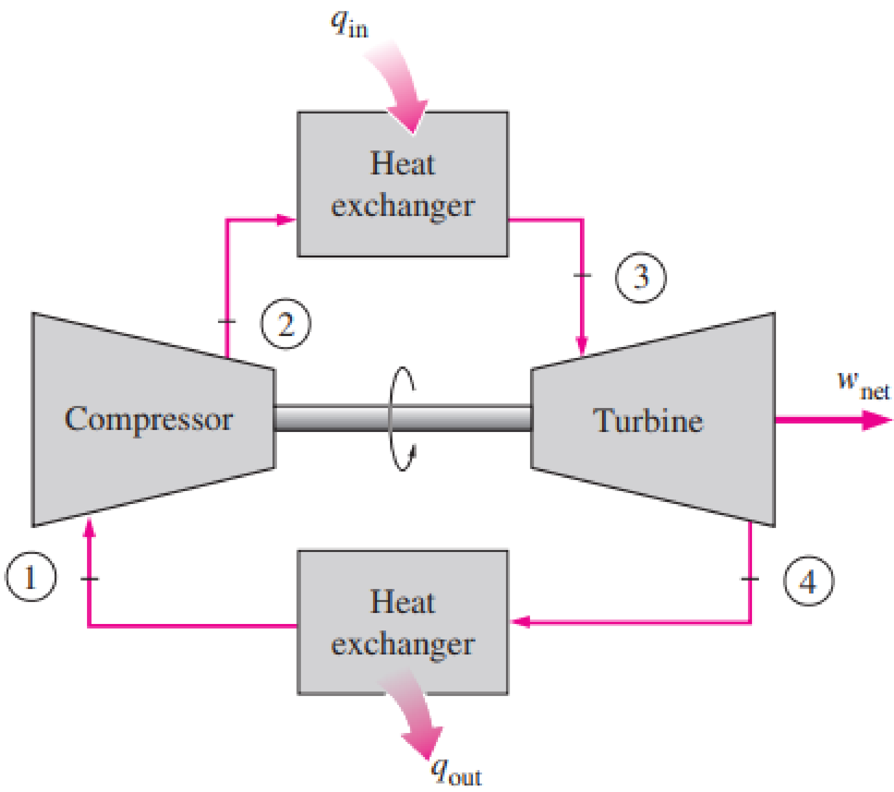


(*a*) Carnot cycle (*b*) Stirling cycle (*c*) Ericsson cycle

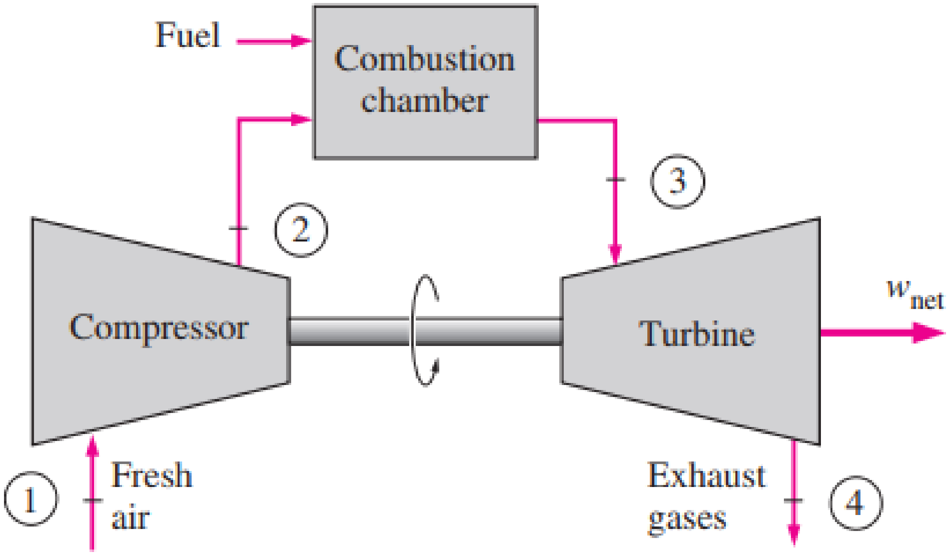
BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

George Bailey Brayton
(1830–1892)

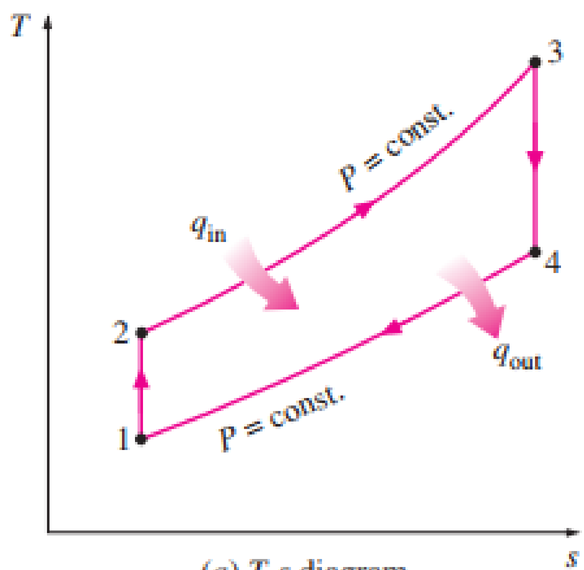
A closed-cycle gas-turbine engine



An open-cycle gas-turbine engine



Ideal BraytonCycle



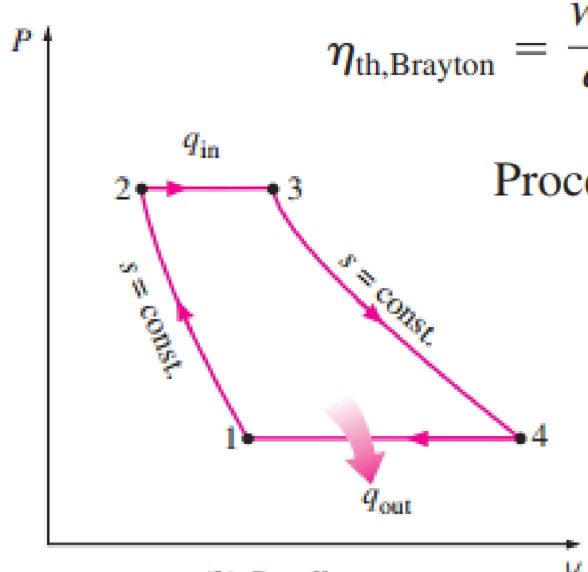
(a) T-s diagram

1-2 Isentropic compression (in a compressor)
2-3 Constant-pressure heat addition
3-4 Isentropic expansion (in a turbine)
4-1 Constant-pressure heat rejection

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_{exit} - h_{inlet}$$

$$q_{in} = h_3 - h_2 = c_p(T_3 - T_2)$$

$$q_{out} = h_4 - h_1 = c_p(T_4 - T_1)$$



(b) P-v diagram

$$\eta_{th,Brayton} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{c_p(T_4 - T_1)}{c_p(T_3 - T_2)} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)}$$

Processes 1-2 and 3-4 are isentropic, and $P_2 = P_3$ and $P_4 = P_1$. Thus,

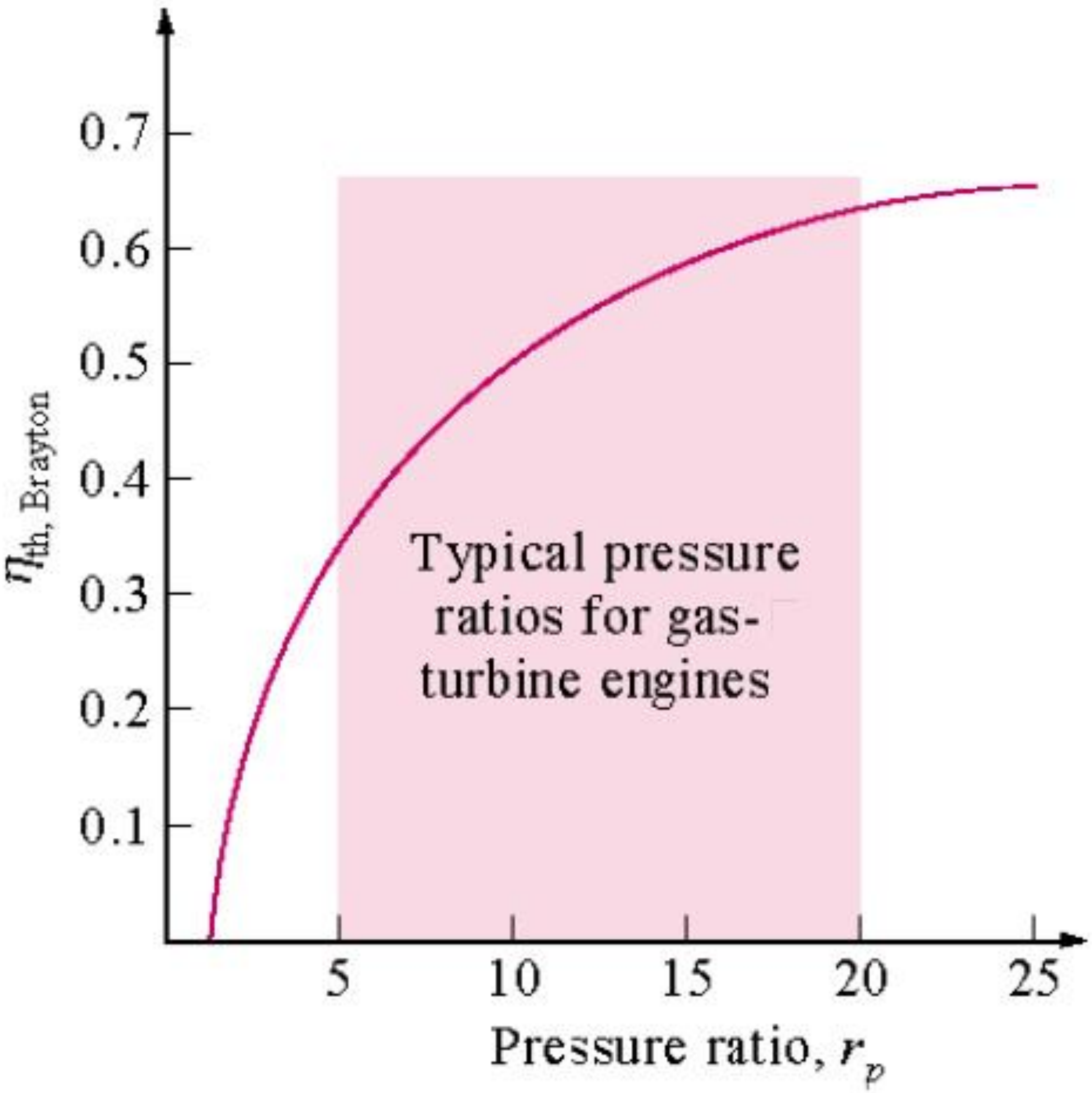
$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} = \left(\frac{P_3}{P_4}\right)^{(k-1)/k} = \frac{T_3}{T_4}$$

$$\boxed{\eta_{th,Brayton} = 1 - \frac{1}{r_p^{(k-1)/k}}}$$

pressure ratio

$$r_p = \frac{P_2}{P_1}$$

Thermal Efficiency of Ideal Brayton Cycle

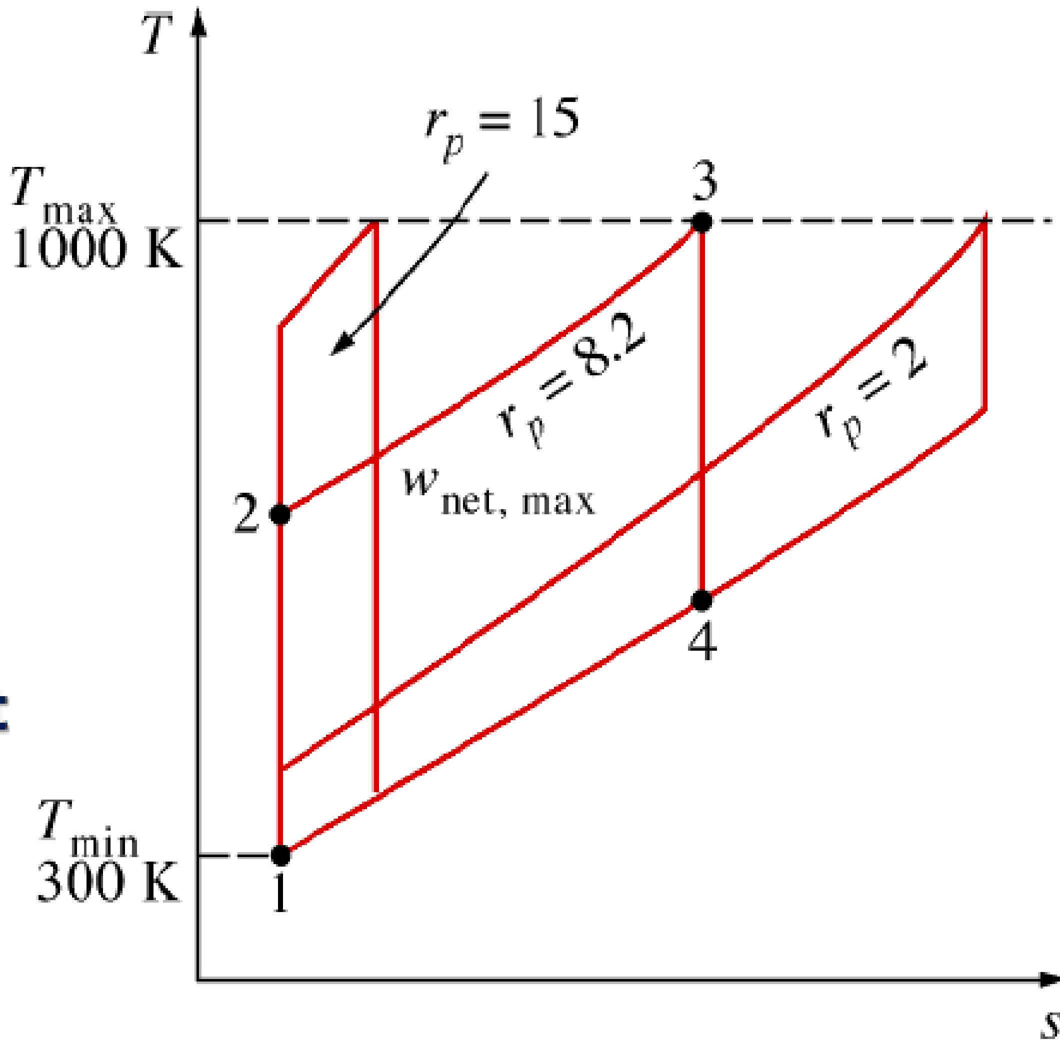


Net Work of Ideal Brayton Cycle

The highest temperature in the cycle is limited by the maximum temperature that the turbine blades can withstand. This also limits the pressure ratios that can be used in the cycle.

The air in gas turbines supplies the necessary oxidant for the combustion of the fuel, and it serves as a coolant to keep the temperature of various components within safe limits. An air–fuel ratio of 50 or above is not uncommon.

- ✓ Fixed T_{min} and T_{max}
- ✓ W_{net} first increases w/ r_p
- ✓ Reaches a maximum at $r_p = (T_{max}/T_{min})^{k/[2(k-1)]}$
- ✓ Finally decreases



Reference

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