KU LEUVEN

B-KUL-Z55337
Applied Thermodynamics

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Adapted from: Y.A. Cengel, M. A. Boles, Thermodynamics – An Engineering Approach (5th Edition)

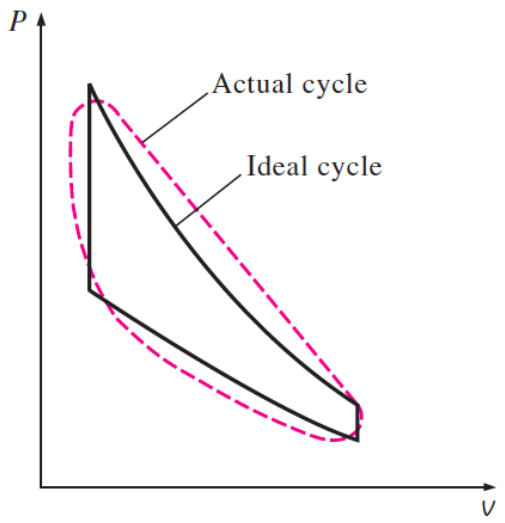
Chapter 8

Gas Power Cycles

(NB: this corresponds to the Chapter 9 of the reference book)

Basic considerations

- Power cycles (net generation of work) vs. refrigeration cycles (net consumption of work)
- **Ideal cycle**: simplified cycle which does not take into account internal irreversibilities. The ideal cycle allows to study the effects of the major parameters influencing the **Real Cycle** that it represents
- Internal reversitbility means: no pressure drops in the lines, no friction, no heat losses to the surroundings, reversible expansion and compression.



Example: The Otto cycle is the ideal cycle representing a spark-ignition automobile engine. The compression ratio (ratio between high and low pressure) has a similar effect (qualitatively but not quantitavely) on the thermal efficency of both the ideas and real cycles.

- **Thermal efficiency** of the cycle:

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}}$$

Basic considerations

- The most efficiency power cycle is the **Carnot cycle**.
- An ideal cycle is internally reversible, whereas the Carnot cycle is also externally reversible (it can be turned reversibely into a heat pump).
- Generally speaking, the thermal efficiency of an ideal cycle is lower or equal to that of the Carnot cycle.
- As a reminder, in Carnot cycle, there is no temperature difference beteen the heat source/since and the high/low temperatures of the Cycle. The efficiency is given by:

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

- **Gas cycle**: cycle in which the working fluids remains in gas state throughout the whole cycle (no condensation).

Hypothesis of air as working fluid

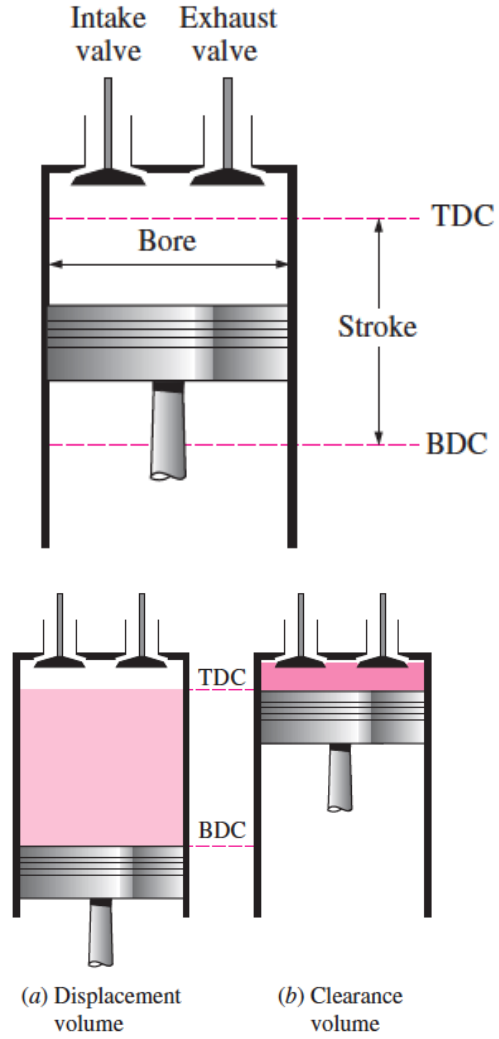
- In internal combustion engines (diesel, gasoline, gas turbine), the composition of the working fluid varies throughout the cycle (it is first air, then combustion gases after burning the fuel)
- Since air is mainly composed of nitrogen, which is inert during the combustion, we can, in good approximation, consider the combustion gases as air.
- The cycles describing internal combustion engines are complex, but can be idealized using the following hypotheses:
  1. The working fluid is air
  2. The cycle is internally reversible
  3. The energy from the combustion is replaced by a heat transfer from an external source
  4. The exhaust process is replaces by a heat-rejection process that restores the working fluid to its initial state

Reciprocating engines

- Reciprocating engines are a subset of the internal combustion engines « family »
- They can found in cars, motorcycles, small airplanes, boats, but also in stationary cogeneration (a.k.a combined heat and power, CHP) units.
- Main characteristics of the piston-cylinder:
  - *Stroke*
  - *Bore (= diameter of the piston)*
  - *top dead center, TDC*
  - *bottom dead center, BDC*
  - *clearance volume*
- The compression ratio is defined by:

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

- This is volume ratio and not a pressure ratio!



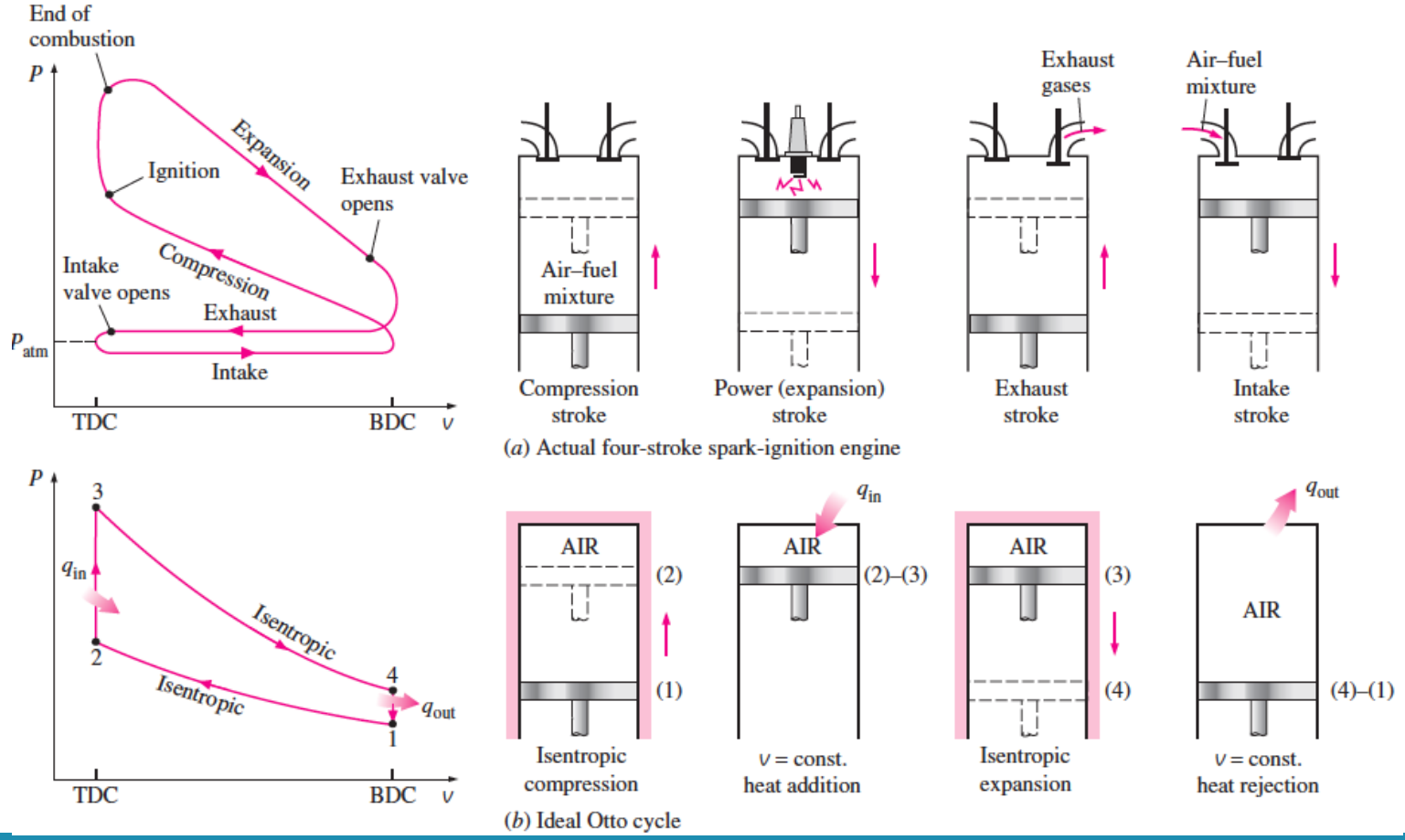
Reciprocating engines

- Two main types:
  - *spark ignition engines*, idealized by the **Otto cycle**
  - *compression ignition engines*, idealized by the **Diesel cycle**

Reciprocating engines

The spark-ignition engine

- The Otto cycle is the ideal cycle for this kind of engine
- Spark-ignition engines (=gasoline engines) are usually four-stroke engines:



Real cycle

Ideal cycle

Reciprocating engines

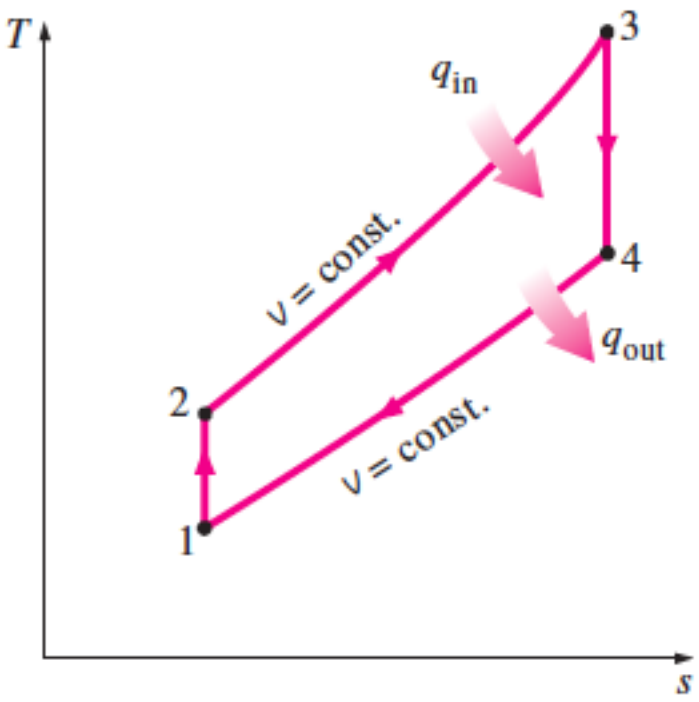
The spark-ignition engine

- The Otto cycle comprises four internally reversible processess:
  - 1-2: isentropic compression
  - 2-3: constant-volume heat addition
  - 3-4: isentropic expansion
  - 4-1: constant volume heat rejection
- It is a closed cycle (contrary to the real proces) and therefore:

  $q_{in} - q_{out} + w_{in} - w_{out} = 0$

- No work is exchanged during the heat transfers. Therefore:

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



Reciprocating engines

The spark-ignition engine

- Therefore, the thermal efficiency is given by:

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

- Processes 1-2 and 3-4 are isentropic and $v_2=v_3$ and $v_4=v_1$. Therefore,

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

- The cycle efficiency is therefore equal to:

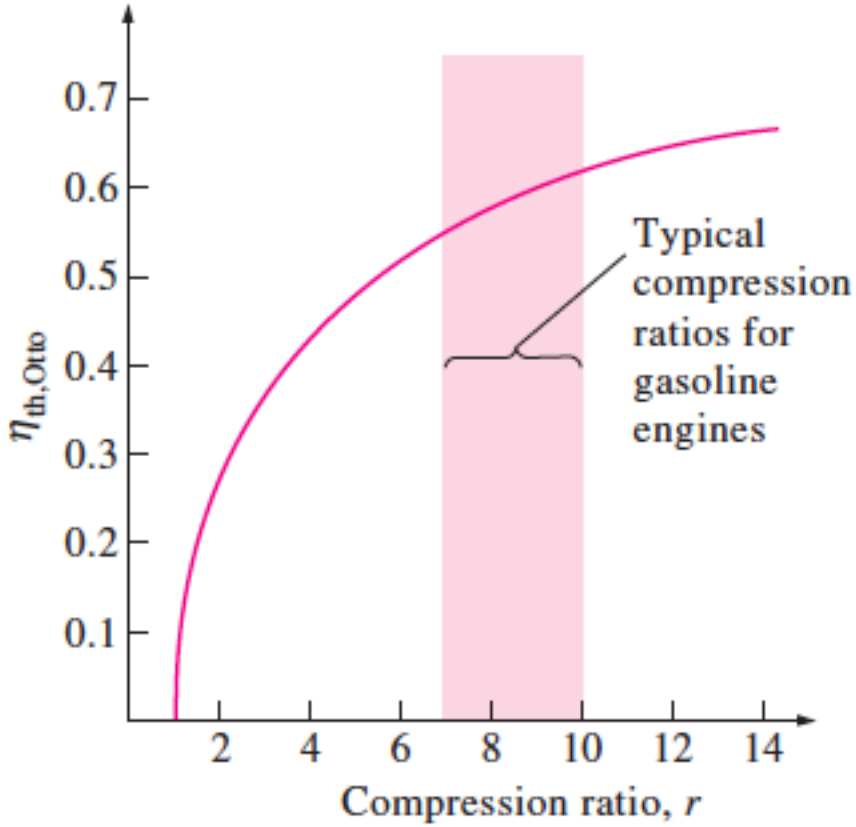
$$\eta_{th,Otto} = 1 - \frac{1}{r^{k-1}}$$

Where the compression ratio is given by: $r = \frac{V_{max}}{V_{min}} = \frac{V_1}{V_2} = \frac{v_1}{v_2}$

Reciprocating engines

The spark-ignition engine

- The thermal efficiency of the ideal Otto cycle (cfr equation from the previous slide) is provided hereunder.



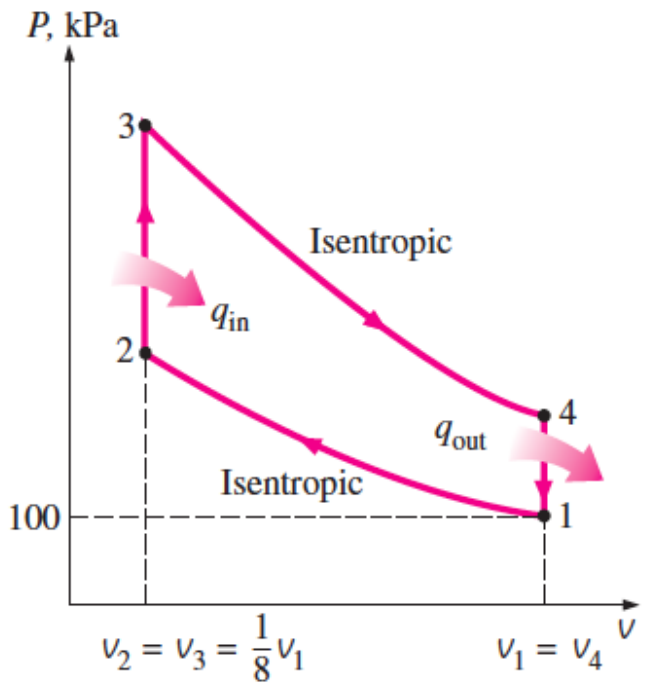
- The compression ratio is limited by the auto-ignition of the fuel. The resistance to auto-ignition is given by the octane rating.
- Autoignition leads to engine knock, which cannot be tolerated.

Reciprocating engines

The spark-ignition engine

Example 9-2

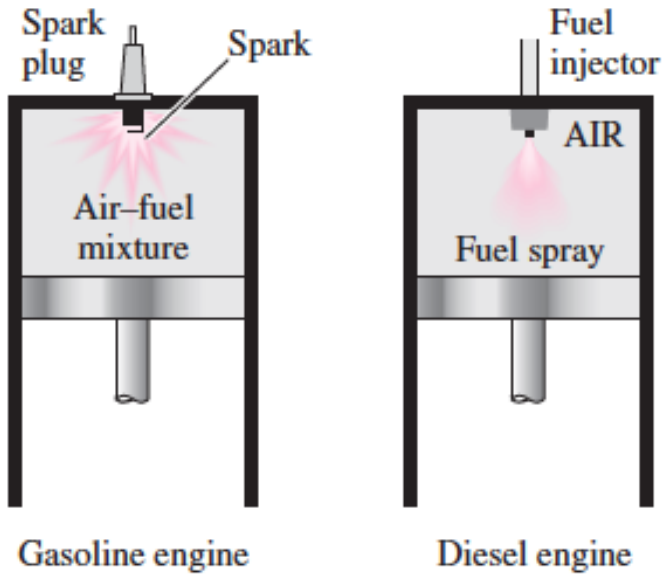
An ideal Otto cycle has a compression ratio of 8. At the beginning of the compression process, air is at 100 kPa and 17°C, and 800 kJ/kg of heat is transferred to air during the constant-volume heat-addition process. Accounting for the variation of specific heats of air with temperature, determine (*a*) the maximum temperature and pressure that occur during the cycle, (*b*) the net work output, (*c*) the thermal efficiency, and (*d*) the mean effective pressure for the cycle.



Reciprocating engines

The compressions-ignition engine

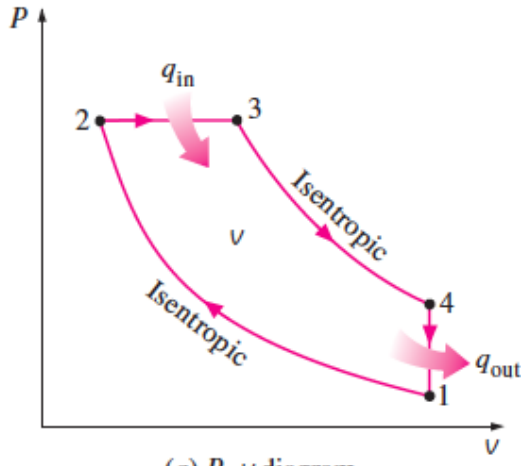
- The diesel cycle is the ideal cycle representing a compression ignition engine (or diesel engine)
- This type of engine was invented in 1890 by Rudolph Diesel.
- The main difference with the gasoline engine is the fuel ignition mechanism:
  - Gasoline engine: the air-fuel mixture is compressed up to the a temperature lower than autoignition temperature of the fuel. The combustion is started by firing a spark plug.
  - Diesel engine: Air is compressed up to a temperature higher than the autoignition temperature of the fuel. Combustion is started by injecting the fuel in the cylinder.



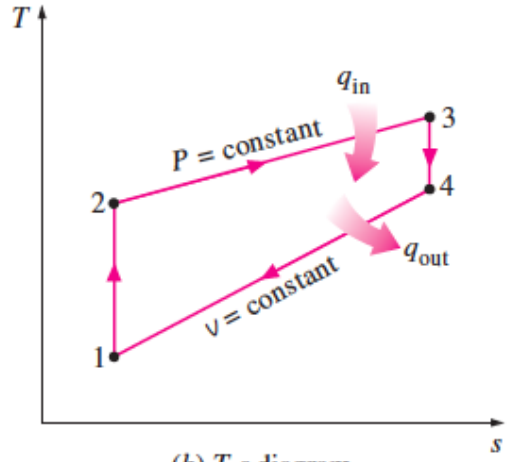
Reciprocating engines

The compressions-ignition engine

- Since the diesel engine is not subject to autoignition, it can be operated with **high compression ratios** (from 12 to 24)
- In a diesel engine, injection occurs just before the to top dead center and continues during the first stroke. The combustion therefore occurs during a relatively long time. As a consequence it can be idealized by a **heat addition at constant pressure**



(a) P- v diagram



(b) T-s diagram

Reciprocating engines

The compressions-ignition engine

- Thermal efficiency:
  - During the heat addition phase 2-3, we can write:

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

  - During the heat rejection phase, we can write:

$$-q_{out} = u_1 - u_4 \rightarrow q_{out} = u_4 - u_1 = c_v(T_4 - T_1)$$

  - If the gas is considered as perfect, the thermal efficiency can therefore by computed by:

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

Reciprocating engines

The compressions-ignition engine

- The comrpession ratio r_c is the ratio between the volumes after and before the combustion phase.

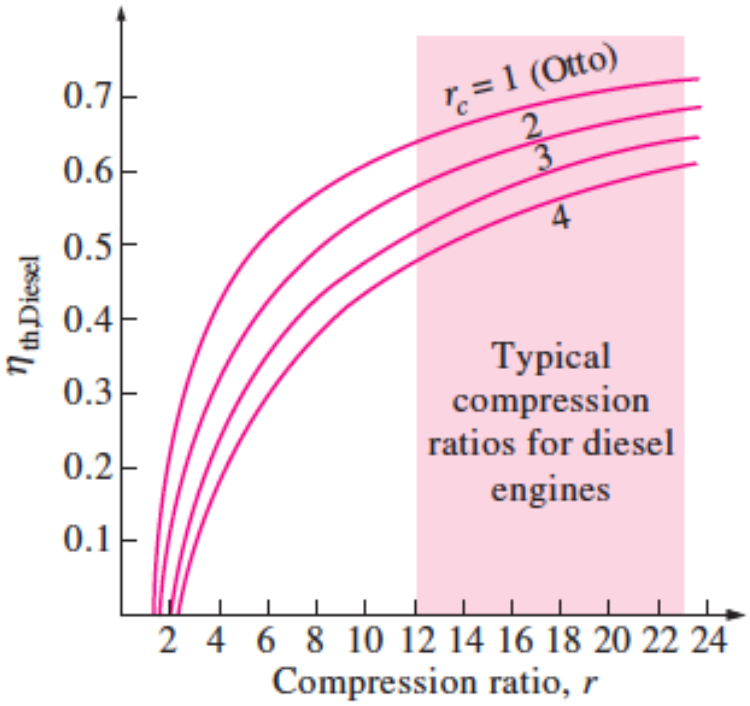
$$r_c = \frac{V_3}{V_2} = \frac{v_3}{v_2}$$

- The cycle efficiency is thus given by:

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

The expression between brackets is always higher than 1. The efficiency of the Otto cycle is therefore higher than that of the diesel cycle for the same compression ratio.

However, diesel engines operate with higher compression ratios, which ultimately makes them more efficient than gasoline engines



Large diesel engines: 35-45%

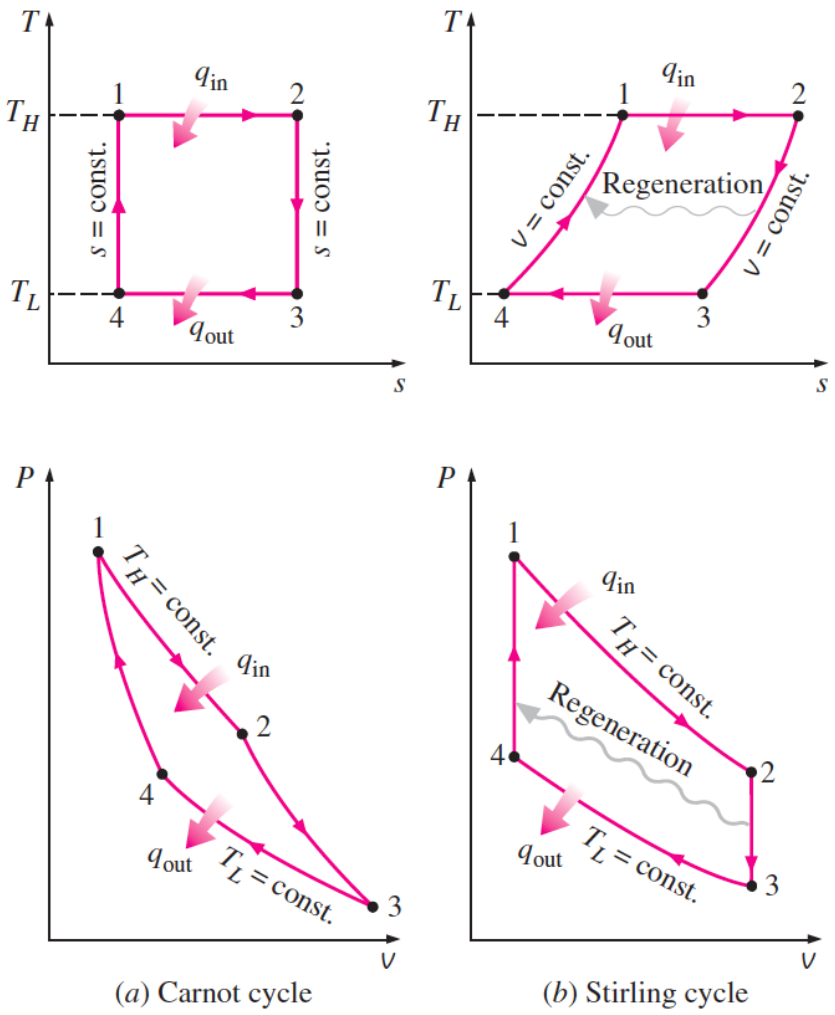
Stirling engines

- Otto and Diesel cycles comprise internally reversible processes
- However, they are not externally reversible because of finite-temperature heat transfer between the source/since and the working fluid. This gradients is explained by the fact that heat addition and rejection do not occur at a constant temperature.
- Their efficiency is therefore always lower than the efficiency of the Carnot cycle, which is externally reversible.
- There are however two other ideal cycles that are externally reversible: the Stirling and the Ericsson cycle.

Stirling engines

- The Stirling engine comprises the following processes:
  - 1-2: isothermal expansion (with an external heat addition)
  - 2-3: isochoric regeneration (heat transfer from the fluid to the regenerator)
  - 3-4: isothermal compression (with an external heat rejection)
  - 4-1: isotchoric regeneration (heat transfer from the regenerator to the working fluid)

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

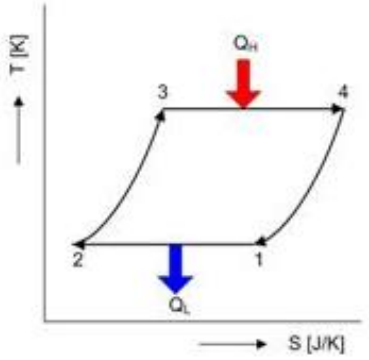


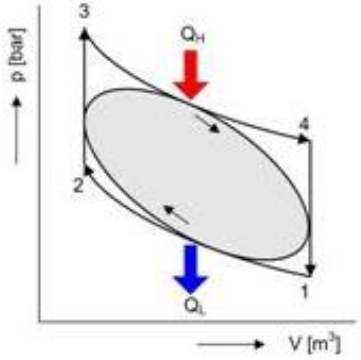
(a) Carnot cycle (b) Stirling cycle

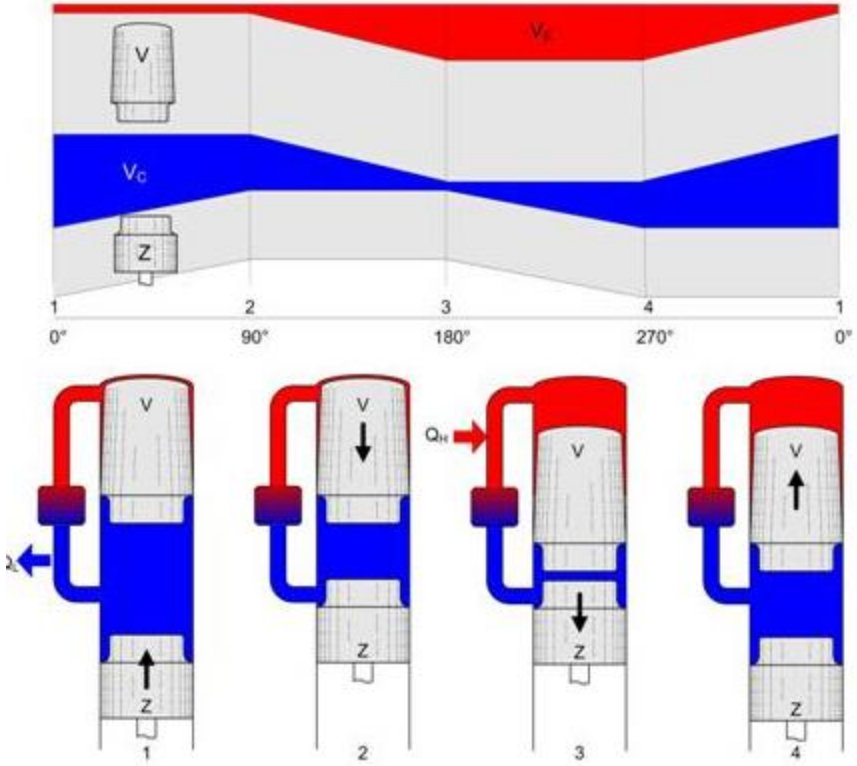
Stirling engines

Stirling cycle

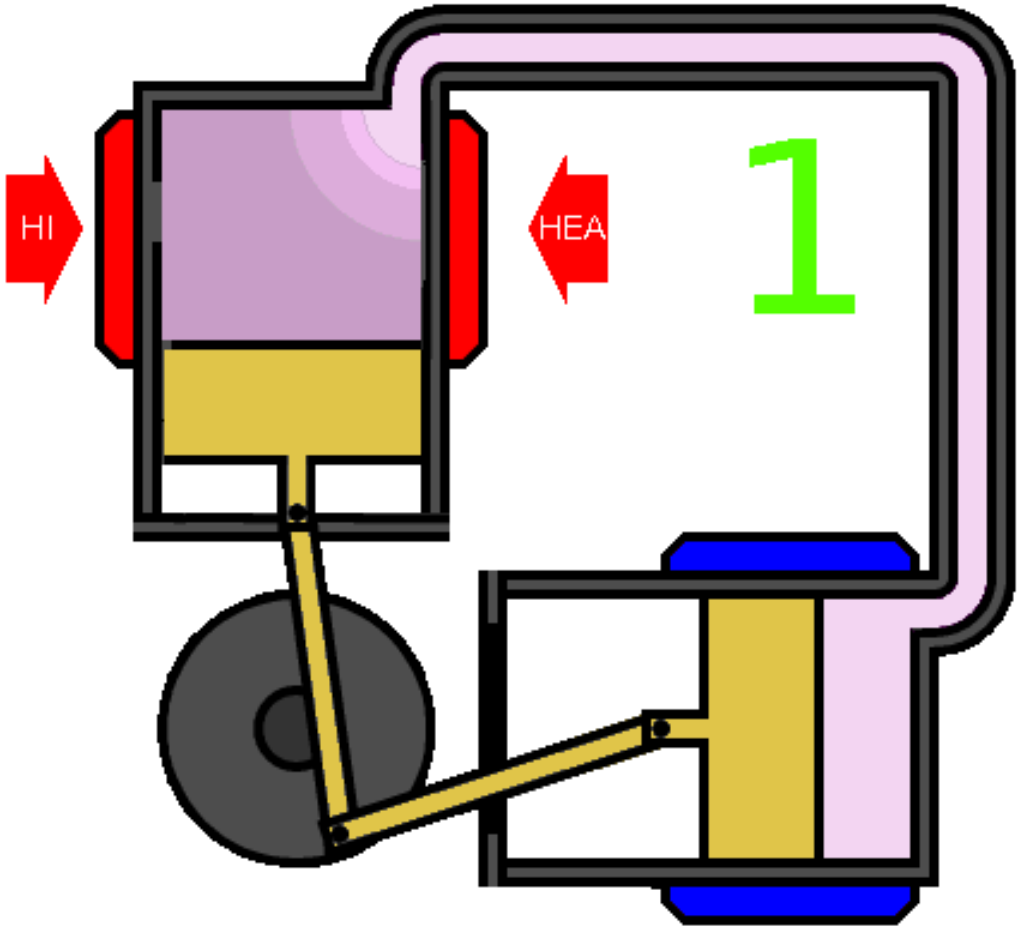
Stirling engine



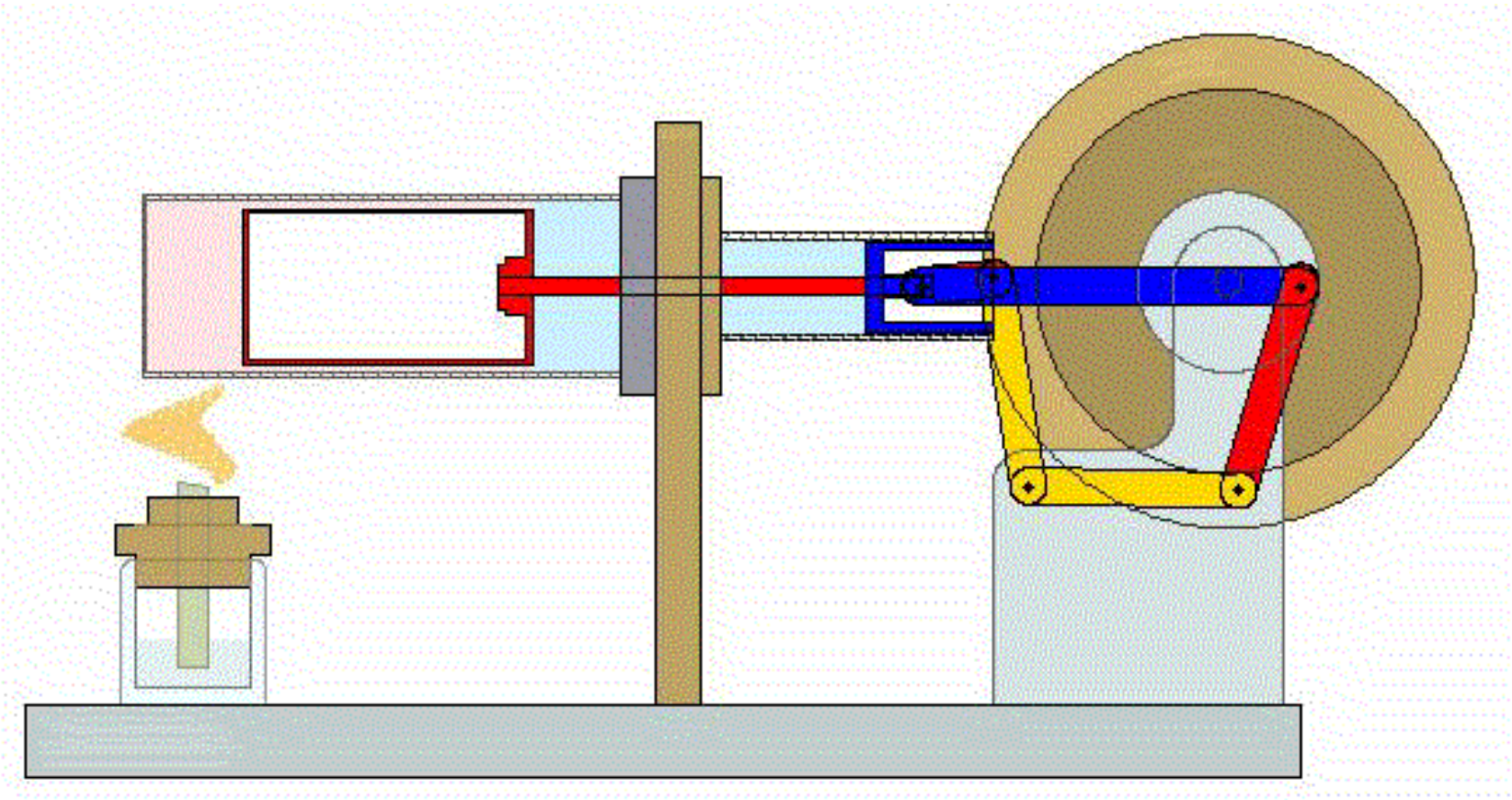




Stirling engines



Stirling engines

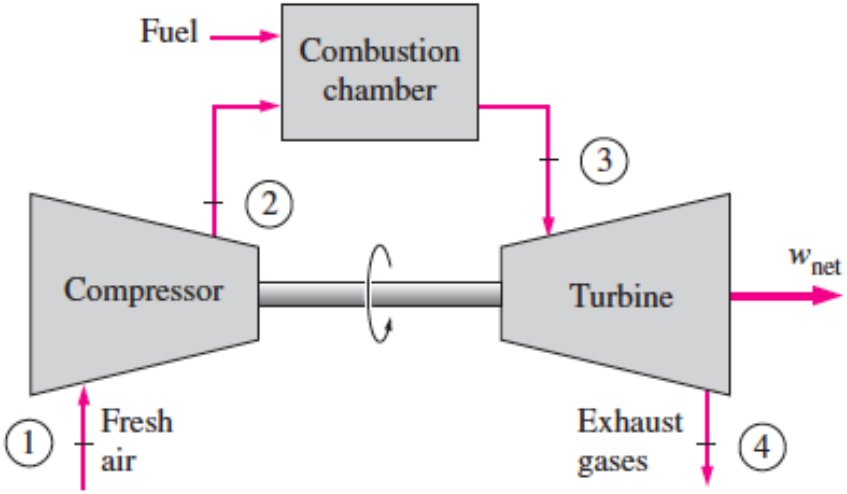


Stirling Engines

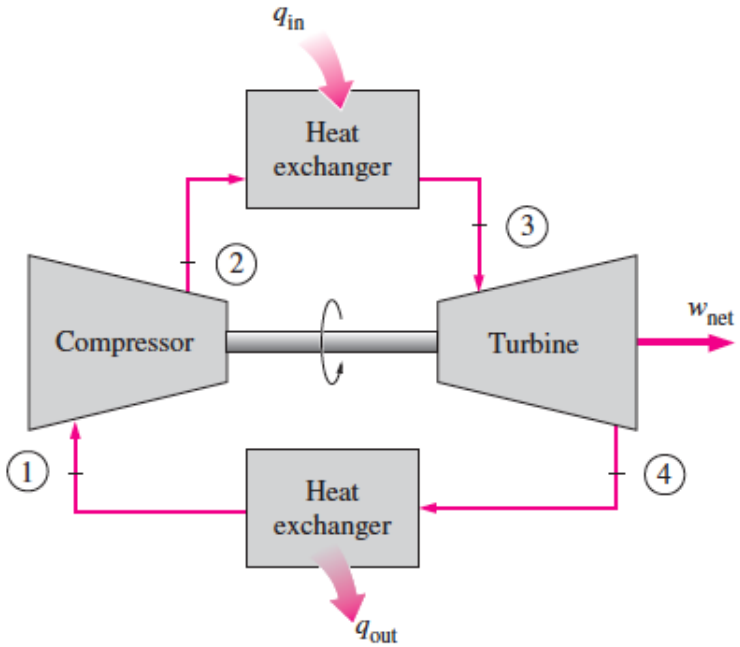
- Stirling engines are external combustion engines: the fuel does not need to be burned inside the engine. These engines are therefore more versatile in terms of heat source.
- Typical characteristics of small Stirling engines
  - Electrical power: 1-50 kWe
  - Heat source: gas, wood pellets
  - Thermal efficiency ~15%
  - Heat source temperature ~400-525°C
  - Working fluid: helium

Gas turbines & Brayton cycles

- The Brayton cycle was proposed by George Brayton in 1870.
- It is used by rotative compression/expansion machines (turbomarchines), whereas originally it was composed of reciprocating compressors and expanders.
- A part of the work (about half) produced by the turbine is consumed by the compressor. The ratio is called the back work ratio



Gas turbine (open Brayton cycle)



Closed Brayton cycle

Gas turbines

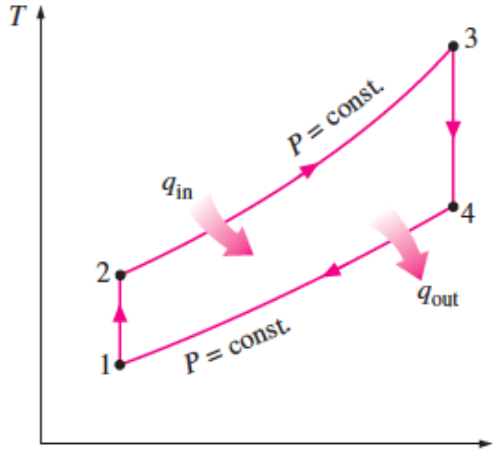
- A gas turbine can be modeled by a closed Brayton cycle
- The combustion process is replaced by an isobaric heat addition from an external source
- The discharge/suction process (renewal of the working fluid) is replaced by an isobaric heat rejection.
- The Brayton cycle is composed of 4 internally reversible processes:

  1-2: isentropic processes

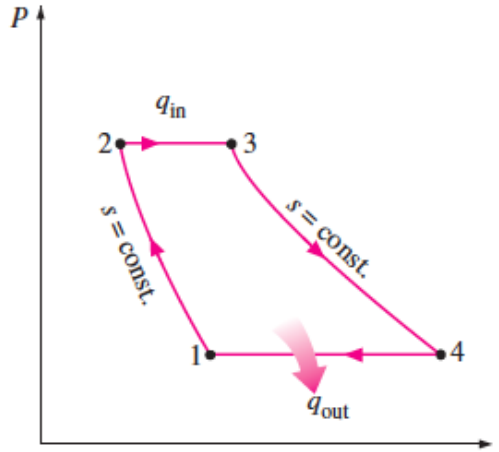
  2-3: isobaric heat addition

  3-4: isentropic expansion

  4-1: isobaric heat rejection



(a) T-s diagram



(b) P-v diagram

Gas turbines

- Thermal efficiency:
  - Each component of the cycle (compressor, heat exchangers, expander) is open system operating in steady-flow conditions
  - Heat transfers are expressed by:

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

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

  - The efficiency is thus given by:

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

  - If the working fluid is perfect gas, this becomes:

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

Gas turbines

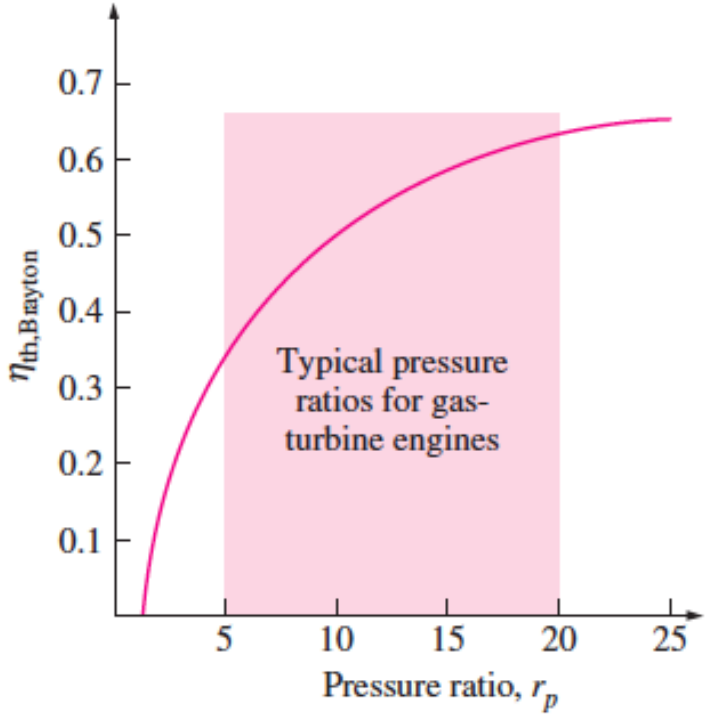
- Therefore, the efficiency of the ideal Brayton cycle is given by:

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

With:

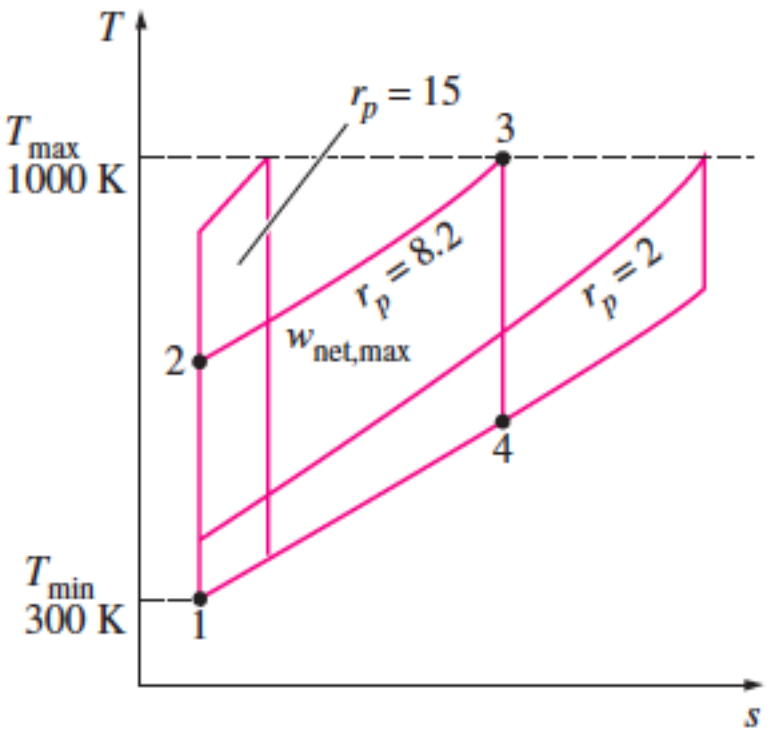
$$r_p = \frac{P_2}{P_1}$$

- In the case of air: k=1.4



Gas turbines

- The highest temperature in the cycle is T_3, it is the *Turbine Inlet Temperature - TIT*.
  - For a given maximum temperature, the is a pressure ratio that maximizes the power of the cycle (given by the area within the T-s diagram)
  - There is therefore a tradeoff betwee efficiency and power (=> compacity and cost)



Gas turbines

- The first gas turbines were developped in the years 1930-40, with efficiencies around 17%. The low efficiency was due to the low isentropic efficiency of compressors and turbines and the low TIT (540°C in 1940 vs. 1425°C today)
- Today, gas turbines are mainly used in the following applications:
  - Turboreactors for airplanes and helicopters
  - Electricity generation:
    - Example: turbine General Electric: TIT: 1425°C, 282 MWe, 39.5%
    - Gas turbines can also be used in conjunction with steam cycles
  - Micro gas turbines are currently in the development phase, e.g. for cogeneration in buildings

Gas turbines

Adapted from: Y.A. Cengel, M. A. Boles, Thermodynamics – An Engineering Approach (5th Edition)

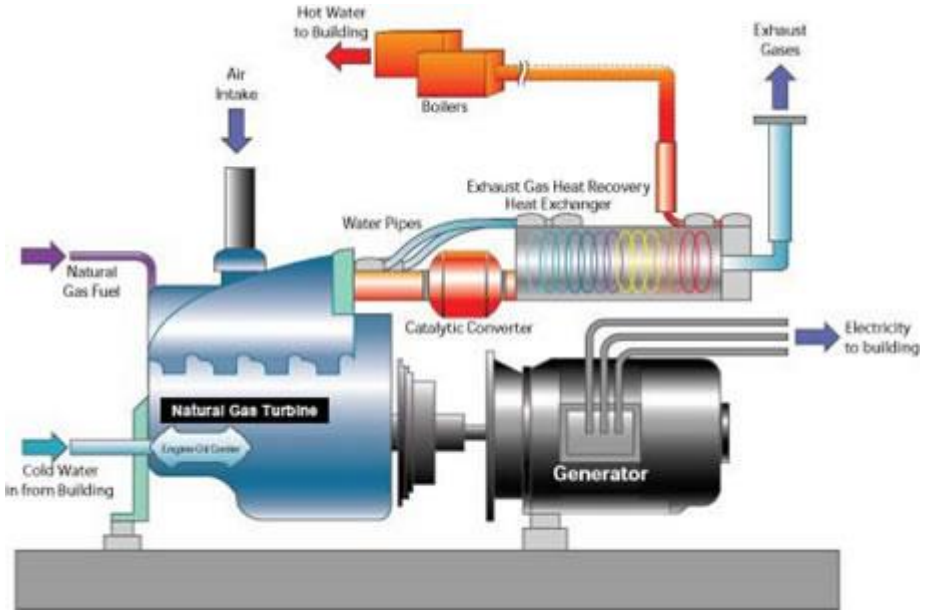
Combined heat and power (= cogeneration) using gas cycles

- **Cogeneration = Combined heat and power = CHP** = Combined generation of heat (hot water, steam, hot air, …) and electricity from a single primary energy source and a single piece of equipment.
- The primary energy source can be renewable (e.g. biomass, goethermal, solar)
- Decentralized CHP: used to cover local generation and consumption

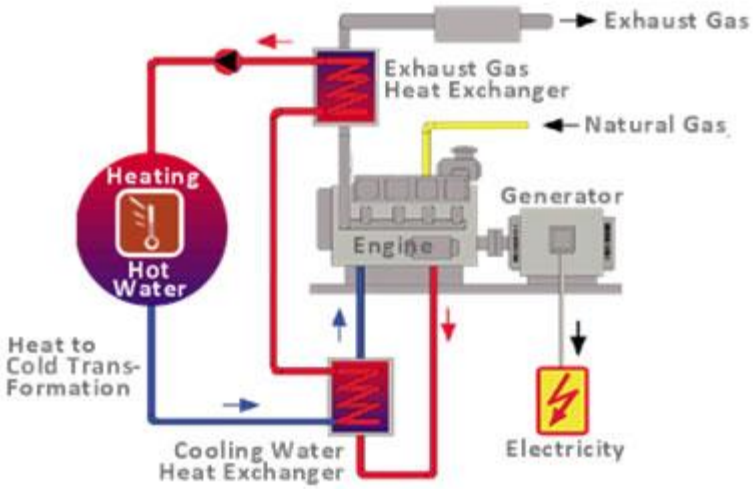
  ⇒ Small-scale cogeneration (>50 kWe, < 1MWe) or micro-cogeneration(<50 kWe according to the EU directive 2004/8/CE)

Combined heat and power (= cogeneration) using gas cycles

With a gas turbine:

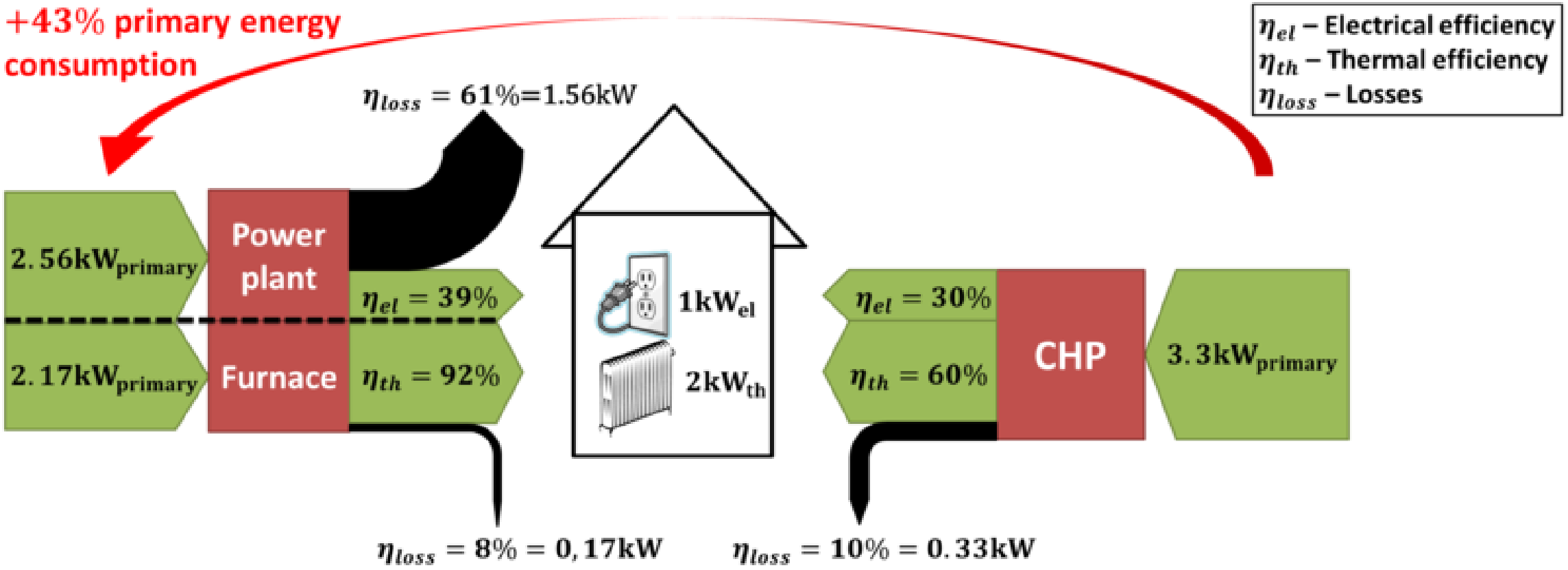


With an internal combustion engine:



Source: NextGrid

Interest of cogeneration: primary energy savings



Source: https://energyeducation.ca/encyclopedia/Cogeneration

Exercises in EES

9–37 The compression ratio of an air-standard Otto cycle is 9.5. Prior to the isentropic compression process, the air is at 100 kPa, 35°C, and 600 cm^3. The temperature at the end of the isentropic expansion process is 800 K. Using specific heat values at room temperature, determine (*a*) the highest temperature and pressure in the cycle; (*b*) the amount of heat transferred in, in kJ; (*c*) the thermal efficiency; and (*d*) the mean effective pressure. *Answers:* (*a*) 1969 K, 6072 kPa, (*b*) 0.59 kJ, (c) 59.4 percent, (*d*) 652 kPa

Exercises in EES

9–76 Air is used as the working fluid in a simple ideal Brayton cycle that has a pressure ratio of 12, a compressor inlet temperature of 300 K, and a turbine inlet temperature of 1000 K. Determine the required mass flow rate of air for a net power output of 70 MW, assuming both the compressor and the turbine have an isentropic efficiency of (*a*) 100 percent and (*b*) 85 percent. Assume constant specific heats at room temperature. *Answers:* (*a*) 352 kg/s, (*b*) 1037 kg/s