

2nd Law

The Second Law of Thermodynamics: Processes occur in a certain direction and energy has quality as well as quantity.

Thermal Energy Reservoirs: A hypothetical body with a relatively large thermal energy capacity (mass \times specific heat) that can supply or absorb finite amounts of heat without undergoing any change in temperature.

Heat Engines

- Devices that convert heat to work
- Receive heat from high-temp source
- Convert part of this heat to work
- Reject remaining waste heat to low-temperature sink - Kelvin-Planck Statement
- Operate on a cycle
- MUST waste some energy by transferring to low-temperature reservoir in order to complete cycle

Notation:

- $Q_{in} = Q_H$ = amount of heat supplied from a high-temp source
- $Q_{out} = Q_L$ = amount of heat rejected to a low temperature sink
- W_{out} = amount of work delivered out of system by working fluid
- W_{in} = amount of work input to system

$$W_{net,out} = W_{out} - W_{in} \text{ (kJ)}$$

$$W_{net,out} = Q_{in} - Q_{out} \text{ (kJ)}$$

Thermal Efficiency

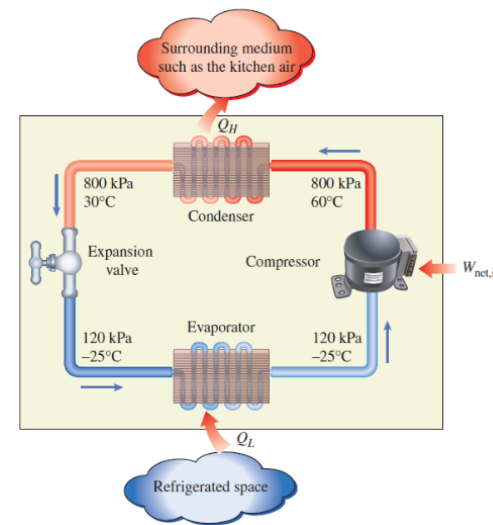
$$\eta = \frac{W_{net,out}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{Q_L}{Q_H}$$

Refrigerators and Heat Pumps

Coefficient of Performance: efficiency of a refrigerator.

$$COP_R = \frac{\text{Desired Output}}{\text{Required Input}} = \frac{Q_L}{W_{net,in}} = \frac{Q_L}{Q_H - Q_L} = \frac{1}{Q_H/Q_L - 1}$$

Clausius Statement: It is impossible to construct a device that operates in a cycle and produces no effect other than the transfer of heat from a lower-temperature body to a higher-temperature body.



Reversible and Irreversible Processes

Reversible Process: A process that can be reversed without leaving any trace on the surroundings - theoretical to find limits.

Irreversible Process: A process that is not reversible.

Irreversibilities:

- Friction
- Unrestrained expansion
- Mixing of two fluids
- Heat transfer across a finite temperature difference
- Electric resistance
- Inelastic deformation of solids
- Chemical reactions

The Carnot Cycle

The Carnot Cycle is composed of four reversible processes - two isothermal and two adiabatic - and it can be executed either in a closed or steady-flow system.

- Reversible Isothermal Expansion - process 1-2, $T_H = \text{constant}$
- Reversible Adiabatic Expansion - process 2-3, $T_H \rightarrow T_L$
- Reversible Isothermal Compression - process 3-4, $T_L = \text{constant}$
- Reversible Adiabatic Compression - process 4-1, $T_L \rightarrow T_H$

The Carnot Cycle is completely reversible - in which case it becomes the Carnot refrigeration cycle.

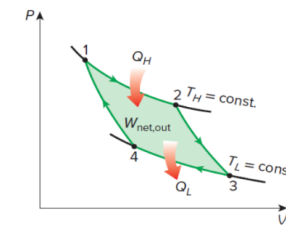


FIGURE 7-35

P-V diagram of the Carnot cycle.

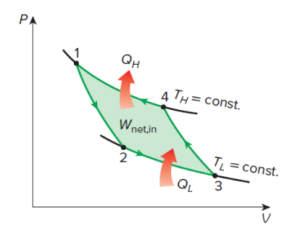


FIGURE 7-36

P-V diagram of the reversed Carnot cycle.

The Carnot Principles: The efficiency of an irreversible heat engine is always less than the efficiency of a reversible one operating between the same two reservoirs and the efficiencies of all reversible heat engines operating between the same two reservoirs are the same.

The Thermodynamic Temperature Scale: A temperature scale that is independent of the properties of the substances that are used to measure temperature.

$$T_H = T_L \frac{Q_H}{Q_L}$$

The Carnot Heat Engine

Any heat engine: $\eta_{th} = 1 - \frac{Q_L}{Q_H}$

Carnot heat engine: $\eta_{th,rev} = 1 - \frac{T_L}{T_H}$

$$\eta_{th} \begin{cases} < \eta_{th,rev} & \text{irreversible heat engine} \\ = \eta_{th,rev} & \text{reversible heat engine} \\ > \eta_{th,rev} & \text{impossible heat engine} \end{cases}$$

Amount of heat rejected per cycle: $Q_{L,rev} = \frac{T_L}{T_H} Q_{H,rev}$

Quality of Energy: The higher the temperature of the thermal energy, the higher its quality. Directly relates to face that you can use temperature to measure efficiency in $\eta_{th,rev}$.

The Carnot Refrigerator and Heat Pump

Any refrigerator or heat pump:

$$COP_R = \frac{1}{Q_H/Q_L - 1} \text{ and } COP_{HP} = \frac{1}{1 - Q_L/Q_H}$$

Carnot refrigerator or heat pump:

$$COP_{R,rev} = \frac{1}{T_H/T_L - 1} \text{ and } COP_{HP,rev} = \frac{1}{1 - T_L/T_H}$$

$$COP_R \begin{cases} < COP_{R,rev} & \text{irreversible refrigerator} \\ = COP_{R,rev} & \text{reversible refrigerator} \\ > COP_{R,rev} & \text{impossible refrigerator} \end{cases}$$

Entropy

- Processes can only occur in a certain direction and that direction must comply with the increase of entropy principle: $S_{\text{gen}} \geq 0$.
- Entropy is a nonconserved property, and there is no such thing as the conservation of entropy principle. Entropy is conserved during the idealized reversible processes only and increases during all actual processes.
- Entropy generation is the measure of the magnitudes of the irreversibilities during the performance of engineering systems. It is also used to establish criteria for the performance of engineering devices.

Clausius Inequality: $\oint \frac{\delta Q}{T} \leq 0$

$$\delta W_C = \delta Q_R - dE_C$$

Reversible cyclic device:

$$\frac{\delta Q_R}{T_R} = \frac{\delta Q}{T} \rightarrow \delta W_C = T_R \frac{\delta Q}{T} - dE_C$$

For a system undergoing a cycle: $W_C = T_R \oint \frac{\delta Q}{T}$

$$\oint \left(\frac{\delta Q}{T} \right)_{\text{int,rev}} = 0 \text{ and } dS = \left(\frac{\delta Q}{T} \right)_{\text{int,rev}} \text{ (kJ/K)}$$

Entropy: $\Delta S = S_2 - S_1 = \int_1^2 \left(\frac{\delta Q}{T} \right)_{\text{int,rev}}$

Special Case:

Internally Reversible Isothermal Heat Transfer Process $\Delta S = \frac{Q}{T_0}$ (kJ/K) where T_0 is the constant temperature of the system and Q is the heat transfer for the internally reversible process.

Generated Entropy:

$$S_{\text{gen}} = \Delta S_{\text{total}} = \Delta S_{\text{sys}} + \Delta S_{\text{surr}} \geq 0 \rightarrow \Delta S_{\text{sys}} = S_2 - S_1 = \int_1^2 \frac{\delta Q}{T} + S_{\text{gen}}$$

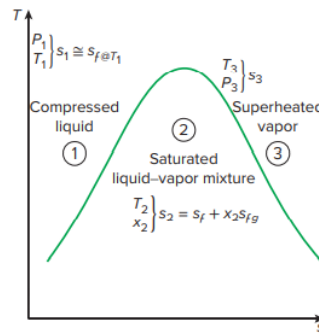
The Increase of Entropy Principle:

$$S_{\text{gen}} \begin{cases} < 0 \text{ impossible process} \\ = 0 \text{ reversible process} \\ > 0 \text{ irreversible process} \end{cases}$$

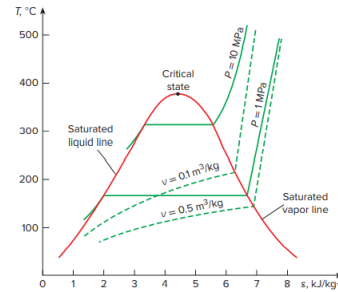
Entropy Change of Pure Substances

Once the state of the system is fixed the value of entropy is also fixed.

$$\Delta S = m \Delta s = m(s_2 - s_1) \text{ (kJ/K)}$$



T-s Diagram for water ↓



Isentropic Processes

$$\Delta S = 0 \text{ or } S_2 = S_1 \text{ (kJ/kgK)}$$

Property Diagrams involving Entropy

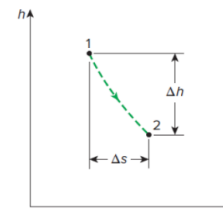
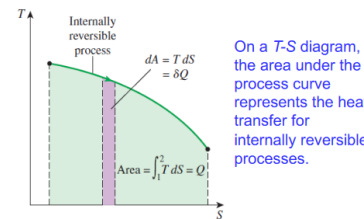


FIGURE 8-18
For adiabatic steady-flow devices, the vertical distance Δh on an h - s diagram is a measure of work, and the horizontal distance Δs is a measure of irreversibilities.

Mollier diagram: The h - s diagram

$$\delta Q_{\text{int rev}} = T ds \quad Q_{\text{int rev}} = \int_1^2 T ds$$

$$\delta q_{\text{int rev}} = T ds \quad q_{\text{int rev}} = \int_1^2 T ds$$

$$Q_{\text{int rev}} = T_0 \Delta S \quad q_{\text{int rev}} = T_0 \Delta s$$

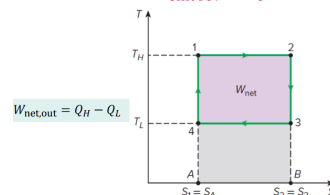


FIGURE 8-19 The T-s diagram of a Carnot cycle

What is Entropy?

Boltzmann Relation: $S = k \ln W$

$$k = 1.3806 \times 10^{-23} \text{ J/K}$$

Gibb's Formulation: $S = -k \sum P_i \log P_i$

Pure Crystal: $T = 0 \text{ K}$ then Entropy = 0

The entropy of a pure crystalline substance at absolute zero temperature is zero (Third Law of Thermodynamics)

The Tds Relations

The First Tds or Gibbs Equation:

$$\delta Q_{\text{int,rev}} - \delta W_{\text{int,rev,out}} = dU$$

$$\delta Q_{\text{int,rev}} = Tds \text{ and } \delta W_{\text{int,rev,out}} = PdV$$

$$Tds = dU + PdV \text{ (kJ)}$$

$$Tds = du + Pdv$$

The Second Tds Equation:

$$h = u + Pv$$

$$dh = du + Pdv + vdP \text{ and } Tds = du + Pdv$$

$$Tds = dh - vdP$$

Finally:

$$ds = \frac{du}{T} + \frac{Pdv}{T} = \frac{dh}{T} - \frac{vdP}{T}$$

Entropy Change of Liquids and Solids

Liquids and solids can be approximated as incompressible substances $\rightarrow dv = 0$.

$$ds = \frac{du}{T} = \frac{cdT}{T} \rightarrow c_P = c_v = c \text{ and } du = cdT$$

$$S_2 - S_1 = \int_1^2 d(T) \frac{dT}{T} = c_{\text{avg}} \ln \frac{T_2}{T_1} \text{ (kJ/kgK)}$$

Isentropic: $s_2 - s_1 = c_{\text{avg}} \ln \frac{T_2}{T_1} = 0 \rightarrow T_2 = T_1$

Entropy Change of Ideal Gases

$$Pv = RT$$

$$du = c_v dT$$

$$dh = c_p dT$$

$$ds = c_v \frac{dT}{T} + R \frac{dv}{v} \rightarrow s_2 - s_1 = \int_1^2 c_v(T) \frac{dT}{T} + R \ln \frac{v_2}{v_1}$$

$$s_2 - s_1 = \int_1^2 c_p(T) \frac{dT}{T} - R \ln \frac{P_2}{P_1}$$

Constant Specific Heats (Approximate Analysis)

$$s_2 - s_1 = c_{v, \text{avg}} \ln \frac{T_2}{T_1} + R \ln \frac{v_2}{v_1}$$

$$s_2 - s_1 = c_{p, \text{avg}} \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1}$$

Variable Specific Heats (Exact Analysis)

$$s_2 - s_1 = \int_1^2 c_v(T) \frac{dT}{T} + R \ln \frac{v_2}{v_1}$$

Choose absolute zero as reference temperature and define a function s° as:

$$s^\circ = \int_0^T c_p(T) \frac{dT}{T}$$

$$s_2^\circ - s_1^\circ = \int_1^2 c_p(T) \frac{dT}{T}$$

On a unit mass basis:

$$s_2 - s_1 = s_2^\circ - s_1^\circ - R \ln \frac{P_2}{P_1} \quad (\text{kJ/kgK})$$

Isentropic Processes of Ideal Gases

Constant Specific Heats

$$\left(\frac{T_2}{T_1} \right)_{s=\text{const}} = \left(\frac{V_1}{V_2} \right)^{k-1}$$

$$\left(\frac{T_2}{T_1} \right)_{s=\text{const}} = \left(\frac{P_2}{P_1} \right)^{(k-1)/k}$$

$$\left(\frac{V_1}{V_2} \right)^{k-1} = \left(\frac{P_2}{P_1} \right)^{(k-1)/k}$$

Variable Specific Heats

Relative Pressure, $P_r = \exp(s^\circ/R)$

Relative Specific Volume, T/P_r

$$\left(\frac{P_2}{P_1} \right)_{s=\text{const}} = \frac{P_{r2}}{P_{r1}}$$

$$\left(\frac{v_2}{v_1} \right)_{s=\text{const}} = \frac{v_{r2}}{v_{r1}}$$

Reverse Steady-Flow Work

$$w_{\text{rev}} = - \int_1^2 v dP - \Delta ke - \Delta pe \rightarrow w_{\text{rev}} = -v(P_2 - P_1) - \Delta ke - \Delta pe$$

When KE and PE are negligible: $w_{\text{rev}} = - \int_1^2 v dP$

For steady flow of a liquid through a device that involves no work, like a pipe, the work term is zero:

$$v(P_2 - P_1) + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1) = 0$$

Isentropic Efficiencies of Steady-Flow Devices

Turbine: $\eta_T = \frac{w_a}{w_s}$ where w_a is actual turbine work and w_s is isentropic turbine work.

$$\eta_T = \frac{h_1 - h_{2a}}{h_1 - h_{2s}}$$

Compressors and Pumps:

$\eta_C = \frac{w_s}{w_a} = \frac{h_{2s} - h_1}{h_{2a} - h_1}$ when kinetic and potential energies are negligible.

$$\eta_P = \frac{w_s}{w_a} = \frac{v(P_2 - P_1)}{h_{2a} - h_1}$$

Nozzles: $\eta_N = \frac{V_{2a}^2}{V_{2s}^2} = \frac{h_1 - h_{2a}}{h_1 - h_{2s}}$ where V is velocity and

$$h_1 = h_{2a} + \frac{V_{2a}^2}{2}$$

Entropy Balance

Total entropy entering - total entropy leaving + total entropy generated = change in total entropy of the system.

$$S_{\text{in}} - S_{\text{out}} + S_{\text{gen}} = \Delta S_{\text{system}}$$

$$\Delta S_{\text{system}} = S_{\text{final}} - S_{\text{initial}} = S_2 - S_1$$

Refer to Lecture 7 Page 21 for more information.

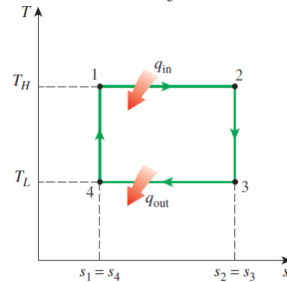
Power Cycles

Ideal Cycle: A cycle that resembles the actual cycle closely but is made up of totally internally reversible processes. These cycles are not necessarily externally reversible.

Carnot Cycle

The Carnot Cycle is composed of 4 totally reversible processes: isothermal heat addition, isentropic expansion, isothermal heat rejection, and isentropic compression.

For both ideal and actual cycles the thermal efficiency increases with an increase in the average temperature at which the heat is supplied to the system or with a decrease in the average temperature at which the heat is rejected from the system.



Derivation of the Efficiency of the Carnot Cycle

$$q_{\text{in}} = T_H(s_2 - s_1) \quad q_{\text{out}} = T_L(s_3 - s_4) = T_L(s_2 - s_1) \quad s_2 = s_3 \text{ and } s_4 = s_1$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}}$$

Air-Standard Assumptions

- The working fluid is air which continuously circulates in a closed loop and always behaves as an ideal gas.
- All the processes that make up the cycle are internally reversible.
- The combustion process is replaced by a heat-addition process from an external source.

- The exhaust process is replaced by a heat-rejection process that restores the working fluid to its initial state.

Cold Air Standard Assumptions: When the working fluid is considered to be air with constant specific heats at room temperature (25°C)

Air Standard Cycle: A cycle for which the air-standard assumptions are applicable.

Reciprocating Engines

$$\text{Compression Ratio: } r = \frac{V_{\text{max}}}{V_{\text{min}}} = \frac{V_{\text{BDC}}}{V_{\text{TDC}}}$$

$$W_{\text{net}} = \text{MEP} \times \text{Piston Area} \times \text{Stroke} = \text{MEP} \times \text{Displacement Volume}$$

Mean Effective Pressure (MEP):

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

The engine with a larger MEP delivers more work per cycle and thus performs better.

The Otto Cycle

The ideal cycle for spark-ignition engines.

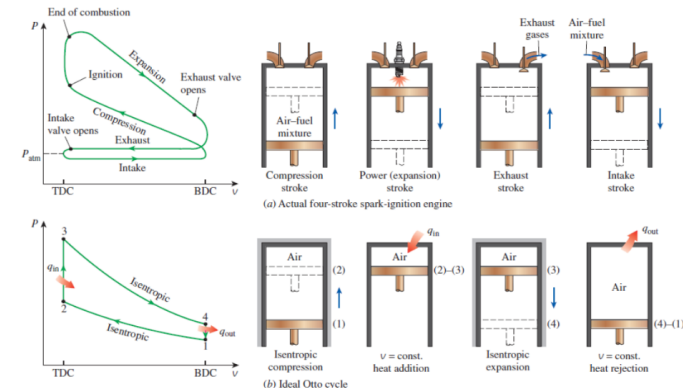


FIGURE 9-12

Actual and ideal cycles in spark-ignition engines and their P-v diagrams.

Four-stroke cycle

1 cycle=4 stroke=2 revolution

Two-stroke cycle

1 cycle=2 stroke=1 revolution

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

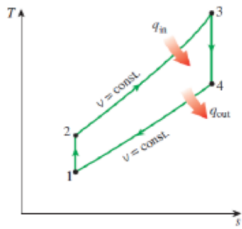
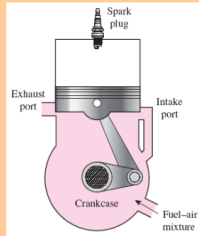


FIGURE 9-15
T-s diagram of the ideal Otto

The two-stroke engines are generally less efficient than their four-stroke counterparts but they are relatively simple and inexpensive, and they have high power-to-weight and power-to-volume ratios.



Schematic of a two-stroke reciprocating engine.

For air $k=1.4$ and the thermal efficiency of the Otto Cycle increases with the specific heat ratio, k , of the working fluid.

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

$$q_{in} = u_3 - u_2 = c_v(T_3 - T_2)$$

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

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

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

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

$$\text{Cold-Air Standard Assumption: } \eta_{th,Otto} = 1 - \frac{1}{r^{k-1}}$$

Diesel Cycle

The ideal cycle for the compression-ignition engines.

In diesel engines, only air is compressed during the compression stroke, eliminating the possibility of autoignition (engine knock). Therefore, diesel engines can be designed to operate at much higher compression ratios than SI engines, typically between 12 and 24.

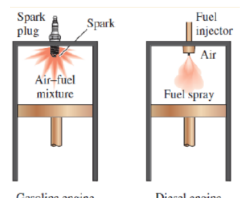


FIGURE 9-20 In diesel engines, the spark plug is replaced by a fuel injector, and only air is compressed during the compression process.

- 1-2 isentropic compression
- 2-3 constant-pressure heat addition
- 3-4 isentropic expansion
- 4-1 constant-volume heat rejection.

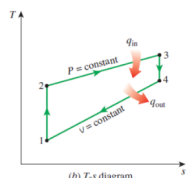
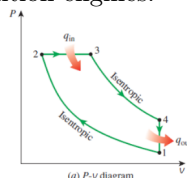


FIGURE 9-21 T-s and P-v diagrams for the ideal Diesel cycle.

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

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

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

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

$\eta_{th,Diesel} = 1 - \frac{1}{r^{k-1}} \left[\frac{r_c^k - 1}{k(r_c - 1)} \right]$ The thermal efficiency of the Diesel Cycle decreases as the cutoff ratio increases and increases as the compression ratio increases.

Brayton Cycle

The ideal cycle for gas-turbine engines.

The combustion process is replaced by a constant-pressure heat-addition process from an external source, and the exhaust process is replaced by a constant-pressure heat-rejection process to the ambient air.

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

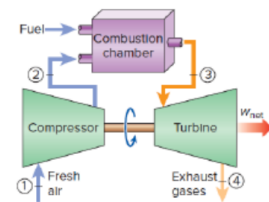


FIGURE 9-25 An open-cycle gas-turbine engine.

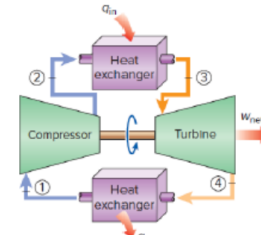
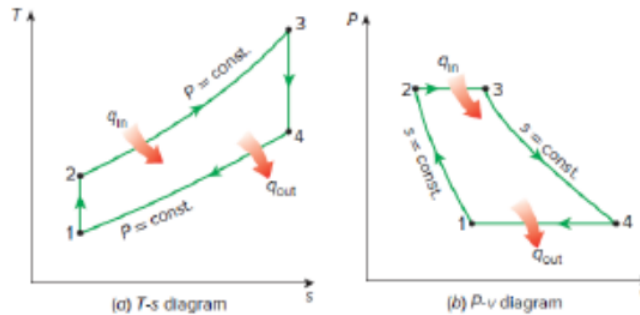


FIGURE 9-26 A closed-cycle gas-turbine engine.



The thermal efficiency of the ideal Brayton Cycle increases as the pressure ratio increases.

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

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

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

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

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

$$r_p = \frac{P_2}{P_1} \text{ Pressure Ratio}$$

Only for ideal cycle and cold-air standard assumptions:

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

Development of Gas Turbines

- Increasing the turbine inlet (or firing) temperatures
- Increasing the efficiencies of turbomachinery components (turbines, compressors)
- Adding modification to the basic cycle (intercooling, regeneration or recuperation, and reheating)

Deviation of Actual Gas-Turbine Cycles from Idealized Ones:

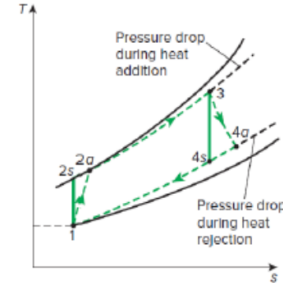


FIGURE 9-32

The deviation of an actual gas-turbine cycle from the ideal Brayton cycle as a result of irreversibilities.

Irreversibilities in turbine and compressors, pressure drops, heat losses.

Isentropic efficiencies of the compressor and turbine:

$$\eta_C = \frac{w_s}{w_a} = \frac{h_{2s} - h_1}{h_{2a} - h_1} \text{ and } \eta_T = \frac{w_a}{w_s} = \frac{h_3 - h_{4s}}{h_3 - h_{4a}}$$

Brayton Cycle with Regeneration

In gas-turbine engines, the temperature of the exhaust gas leaving the turbine is often considerably higher than the temperature of the air leaving the compressor.

Therefore, the high-pressure air leaving the compressor can be heated by the hot exhaust gases in a counter-flow heat exchanger (a *regenerator* or a *recuperator*).

The thermal efficiency of the Brayton cycle increases as a result of regeneration since less fuel is used for the same work output.

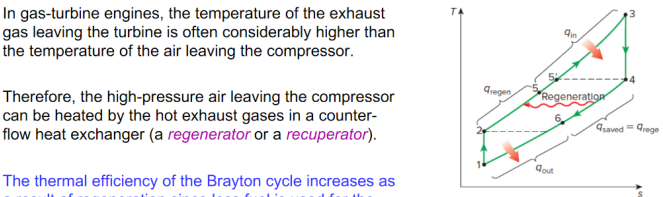
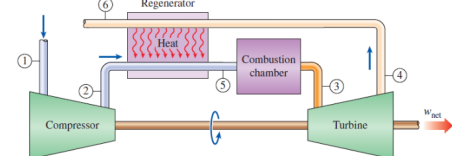


FIGURE 9-35
T-s diagram of a Brayton cycle with regeneration.



A gas-turbine engine with regenerator.

$$q_{regen,act} = h_5 - h_2$$

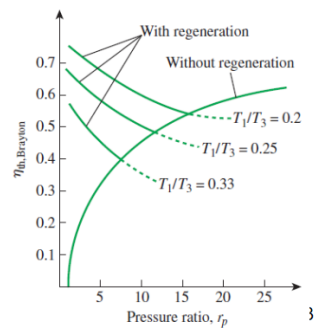
$$q_{regen,max} = h_{5'} - h_2 = h_4 - h_2$$

$$\text{Effectiveness of regenerator: } \epsilon = \frac{q_{regen,act}}{q_{regen,max}} = \frac{h_5 - h_2}{h_4 - h_2}$$

$$\text{Effectiveness under cold-air assumptions: } \epsilon = \frac{T_5 - T_2}{T_4 - T_2}$$

Efficiency under cold-air assumptions:

$$\eta_{th,regen} = 1 - \left(\frac{T_1}{T_3}\right) (r_p)^{(k-1)/k}$$



Carnot Vapor Cycle

The Carnot cycle is the most efficient cycle operating between two specified temperature limits but it is not a suitable model for power cycles. Because:

- Process 1-2** Limiting the heat transfer processes to two-phase systems severely limits the maximum temperature that can be used in the cycle (374°C for water)
- Process 2-3** The turbine cannot handle steam with a high moisture content because of the impingement of liquid droplets on the turbine blades causing erosion and wear.
- Process 4-1** It is not practical to design a compressor that handles two phases.
- The cycle in (b) is not suitable since it requires isentropic compression to extremely high pressures and isothermal heat transfer at variable pressures.

- 1-2 isothermal heat addition in a boiler
- 2-3 isentropic expansion in a turbine
- 3-4 isothermal heat rejection in a condenser
- 4-1 isentropic compression in a compressor

Rankine Cycle

The ideal cycle for vapor power cycles.

Many of the impracticalities associated with the Carnot cycle can be eliminated by superheating the steam in the boiler and condensing it completely in the condenser.

The cycle that results is the **Rankine cycle**, which is the ideal cycle for vapor power plants. The ideal Rankine cycle does not involve any internal irreversibilities.

- 1-2 Isentropic compression in a pump
- 2-3 Constant pressure heat addition in a boiler
- 3-4 Isentropic expansion in a turbine
- 4-1 Constant pressure heat rejection in a condenser

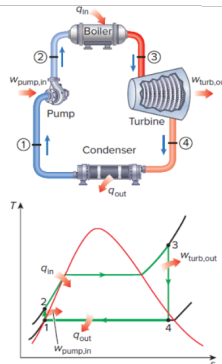


FIGURE 9-39 The simple ideal Rankine cycle.

Steady-flow energy equation:

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_e - h_i \text{ (kJ/kg)}$$

Pump(q=0):

$$w_{pump,in} = h_2 - h_1 = V(P_2 - P_1)$$

$$h_1 = h_{f@P_1} \text{ and } v = v_1 = v_{f@P_1}$$

Boiler(w=0):

$$q_{in} = h_3 - h_2$$

Turbine(q=0):

$$w_{turb,out} = h_3 - h_4$$

Condenser(w=0):

$$q_{out} = h_4 - h_1$$

Finally:

$$w_{net} = q_{in} - q_{out} = w_{turb,out} - w_{pump,in}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}}$$

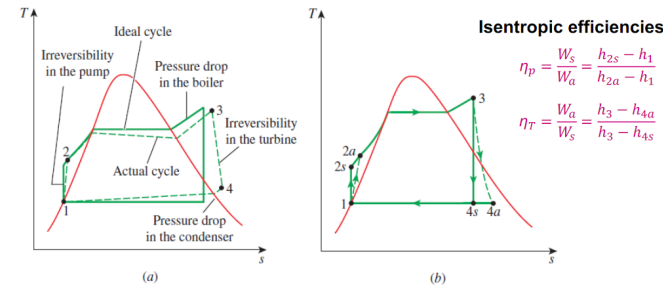
The efficiency of power plants in the US is often expressed in terms of **heat rate** which is the amount of heat supplied, in Btu's, to generate 1 kWh of electricity:

$$\eta_{th} = \frac{3412 \text{ (Btu/kWh)}}{\text{Heat rate (Btu/kWh)}}$$

Deviation of Actual Vapor Power Cycle from Idealized Ones

The actual vapor power cycle differs from the ideal Rankine cycle as a result of irreversibilities in various components.

Fluid friction and **heat loss to the surroundings** are the two common sources of irreversibilities.



(a) Deviation of actual vapor power cycle from the ideal Rankine cycle.
(b) The effect of pump and turbine irreversibilities on the ideal Rankine cycle.

How Increase Efficiency of Rankine Cycle?

Look at Lecture 12 starting Page 7 for diagrams.

Lowering the Condenser: Lowers $T_{low,avg}$, however, it increases the moisture content of the steam at the final stages of the turbine.

Superheating the Steam to High Temperatures: increases $T_{high,avg}$ and decreases the moisture content at the turbine exit (desirable), however it is limited by metallurgical considerations. Highest temp at inlet is 620°.

Increasing the Boiler Pressure: increases $T_{high,avg}$ however the moisture content of the steam at the turbine exit increases - can be corrected by reheating the steam.

The Ideal Reheat Rankine Cycle

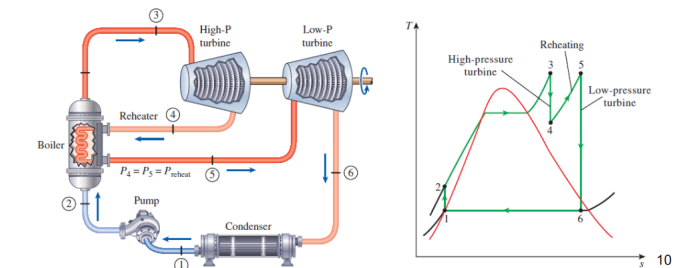
How can we take advantage of the increased efficiencies at higher boiler pressures without facing the problem of excessive moisture at the final stages of the turbine?

1. Superheat the steam to very high temperatures. It is limited metallurgically.
2. Expand the steam in the turbine in two stages, and reheat it in between (**reheat**)

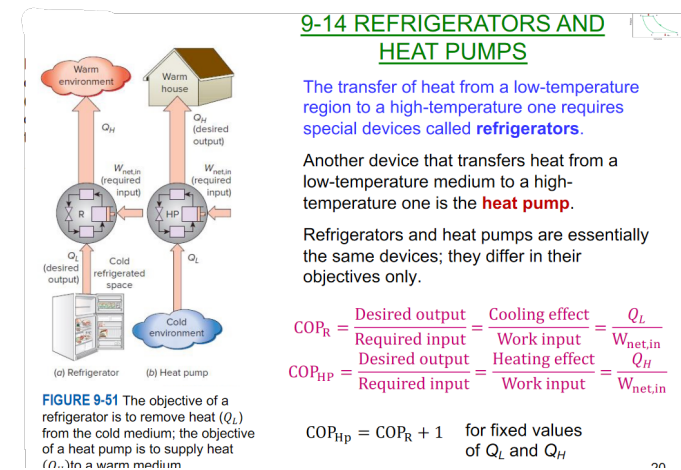
$$q_{in} = q_{primary} + q_{reheat} = (h_3 - h_2) + (h_5 - h_4)$$

$$w_{turb,out} = w_{turb,I} + w_{turb,II} = (h_3 - h_4) + (h_5 - h_6)$$

FIGURE 9-48 The ideal reheat Rankine cycle.

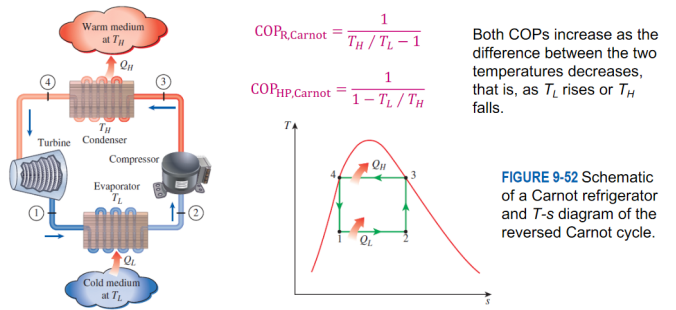


Refrigerators and Heat Pumps

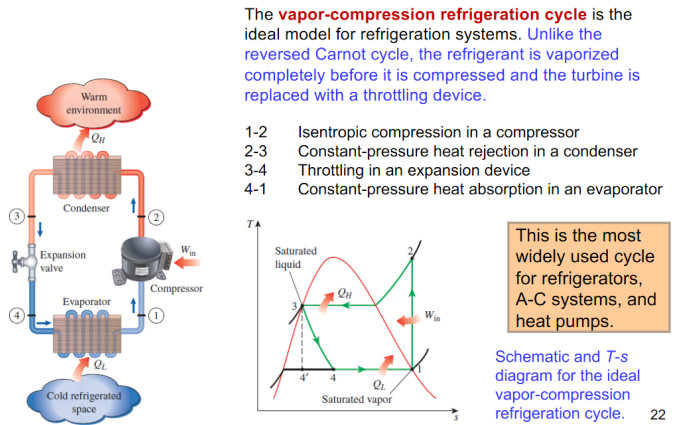


Reversed Carnot Cycle

The reversed Carnot cycle is the *most efficient* refig. cycle operating between T_L and T_H . It is *not a suitable model for refrigeration cycles* since processes 2-3 and 4-1 are not practical *because* process 2-3 involves the compression of a liquid–vapor mixture, which requires a compressor that will handle two phases, and process 4-1 involves the expansion of high-moisture-content refrigerant in a turbine.



Ideal Vapor Compression Refrigeration Cycle



Mechanisms of Heat Transfer

Conduction

The transfer of energy from the more energetic particles of a substance to the adjacent less energetic ones as a result of interactions between the particles.

Rate of heat conduction $\propto \frac{(\text{Area})(\text{Temperature Difference})}{\text{Temperature Difference}}$

$$\dot{Q}_{\text{cond}} = kA \frac{T_1 - T_2}{\Delta x} = -kA \frac{\Delta T}{\Delta x} \text{ (W)}$$

When $x \rightarrow 0$, $\dot{Q}_{\text{cond}} = -kA \frac{dT}{dx}$

Thermal Conductivity: k , a measure of the ability of a material to conduct heat.

Temperature Gradient: dT/dx , the slope of the temperature curve on a T-x diagram.

Thermal Diffusivity

Specific Heat: c_p (J/kgC), heat capacity per unit mass.

Heat Capacity: ρc_p (J/m³), heat capacity per unit volume.

Thermal Diffusivity: α (m²/s), represents how fast heat diffuses through a material.

$$\alpha = \frac{\text{Heat Conduction}}{\text{Heat Storage}} = \frac{k}{\rho c_p}$$

The larger the thermal diffusivity the faster the propagation of heat into the medium.

Convection

The mode of energy transfer between a solid surface and the adjacent liquid or gas that is in motion, and it involves the combined effects of conduction and fluid motion.

$$\dot{Q}_{\text{conv}} = hA_s(T_2 - T_\infty) \text{ where } h \text{ is the convection heat transfer coefficient in W/m}^2\text{K}$$

Radiation

The energy emitted by matter in the form of electromagnetic waves as a result of the changes in the electronic configurations of the atoms or molecules.

$$\dot{Q}_{\text{emit,max}} = \sigma A_s T_s^4 \text{ where } \sigma = 5.670 \times 10^{-8} \text{ W/m}^2\text{K}^4 \text{ or } 0.1714 \times 10^{-8} \text{ Btu/hft}^2\text{R}^4$$

The idealized surface that emits radiation at this maximum rate is called a blackbody. The radiation for all real surfaces is as:

$$\dot{Q}_{\text{emit}} = \varepsilon \sigma A_s T_s^4 \text{ (W) where } \varepsilon \text{ is the emissivity of the surface. } \alpha \text{ is the absorptivity of the surface.}$$

$$\dot{Q}_{\text{absorbed}} = \alpha \dot{Q}_{\text{incident}}$$

$$\dot{Q}_{\text{rad}} = \varepsilon \sigma A_s (T_s^4 - T_{\text{surr}}^4) \text{ (W)}$$

Convection and Radiation often take place in parallel:

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = h_{\text{combined}} A_s (T_s - T_\infty)$$

$$h_{\text{combined}} = h_{\text{conv}} + h_{\text{rad}} = h_{\text{conv}} + \varepsilon \sigma (T_s + T_{\text{surr}})(T_s^2 + T_{\text{surr}}^2)$$

General Knowledge

Isentropic: Constant entropy

Isothermal: Constant temperature

Adiabatic: No Q

Adiabatic and Reversible: Isentropic