Engineering Cycles (Review)

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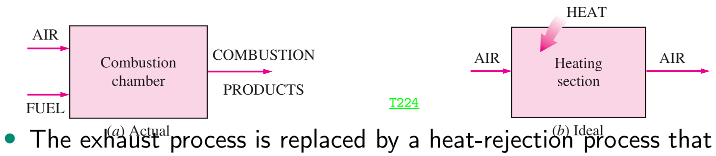
ME 6101: Classical Thermodynamics

`http://zahurul.buet.ac.bd/ME6101/`

Air-Standard Cycle 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.

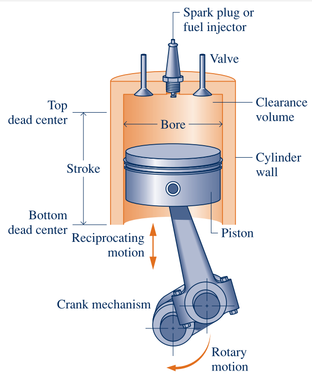
T223 T224



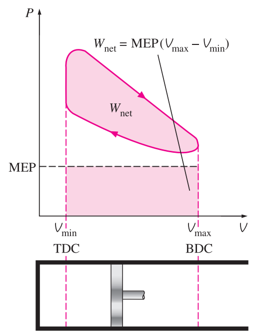
- The exhaust process is replaced by a heat-rejection process that restores the working fluid to its initial state.
- Air has constant specific heats determined at room temperature.

⇒ A cycle for which the air-standard assumptions are applicable is frequently referred to as an **air-standard cycle**.

Overview of Reciprocating (R/C) Engines



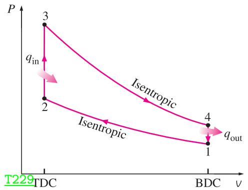
T625



T227

- TDC ≡ Top Dead Centre, BDC ≡ Bottom Dead Centre
- Compression Ratio $\equiv r = \frac{V_{max}}{V_{min}} = \frac{V_{BDC}}{V_{TDC}}$
- Displacement Volume $\equiv V_d = \frac{\pi}{4} \times S \times B^2 = V_{max} - V_{min}$
- Mean Effective Pressure $\equiv MEP = \frac{W_{net}}{V_d}$

The Otto Cycle: Ideal Cycle for SI Engines



T229

- $1 \rightarrow 2$: Isentropic compression
- $2 \rightarrow 3$: Reversible, Constant-volume heat addition
- $3 \rightarrow 4$: Isentropic expansion
- $4 \rightarrow 1$: Reversible, Constant-volume heat rejection

- Isentropic processes: $1 \rightarrow 2$ & $3 \rightarrow 4$. Also, $V_2 = V_3$ & $V_4 = V_1$

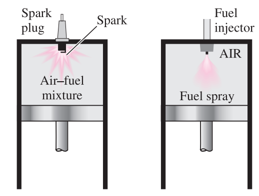
$\rightarrow \frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{k-1} = \left(\frac{V_3}{V_4}\right)^{k-1} = \frac{T_4}{T_3} \rightarrowtail \frac{T_1}{T_2} = \frac{1}{r^{k-1}}$ & $\frac{T_3}{T_2} = \frac{T_4}{T_1}$

- $q_{in} = u_3 - u_2 = c_v(T_3 - T_2)$: $q_{out} = u_4 - u_1 = c_v(T_4 - T_1)$
- $\eta_{th,Otto} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_2}{T_1}\frac{[T_3/T_2 - 1]}{[T_4/T_1 - 1]} = 1 - \frac{T_2}{T_1} = 1 - \frac{1}{r^{k-1}}$

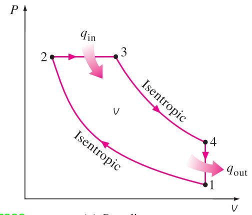
$$\eta_{th,Otto} = 1 - \frac{1}{r^{k-1}}$$

r = compression ratio,
k = ratio of specific heats, c_p/c_v

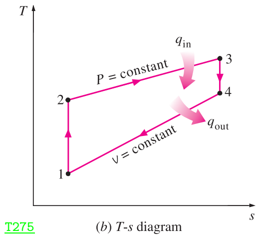
The Diesel Cycle: Ideal Cycle for CI Engines



T232 Gasoline engine   Diesel engine



T233 (a) P-v diagram

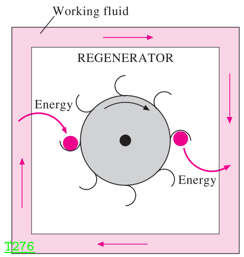


T275 (b) T-s diagram

- Isentropic processes: $1 \longrightarrow 2$ & $3 \longrightarrow 4$.
- Cut-off ratio, $r_c = \frac{V_3}{V_2}$, and $V_4 = V_1$.
- $q_{in} = h_3 - h_2 = c_p(T_3 - T_2) \quad : \quad q_{out} = u_4 - u_1 = c_v(T_4 - T_1)$
- $\eta_{th,Diesel} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_2}{kT_1}\frac{[T_3/T_2 - 1]}{[T_4/T_1 - 1]} = 1 - \frac{1}{r^{k-1}}\left[\frac{r_c^k - 1}{k(r_c - 1)}\right]$

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

Stirling & Ericsson Cycles



T276

- **Stirling** and **Ericsson** cycles involve an isothermal heat-addition at T_H and an isothermal heat-rejection at T_L. They differ from the Carnot cycle in that the two isentropic processes are replaced by two constant-volume regeneration processes in the *Stirling cycle* and by two constant-pressure regeneration processes in the *Ericsson cycle*.
- Both cycles utilize regeneration, a process during which heat is transferred to a thermal energy storage device (called a regenerator) during one part of the cycle and is transferred back to the working fluid during another part of the cycle.

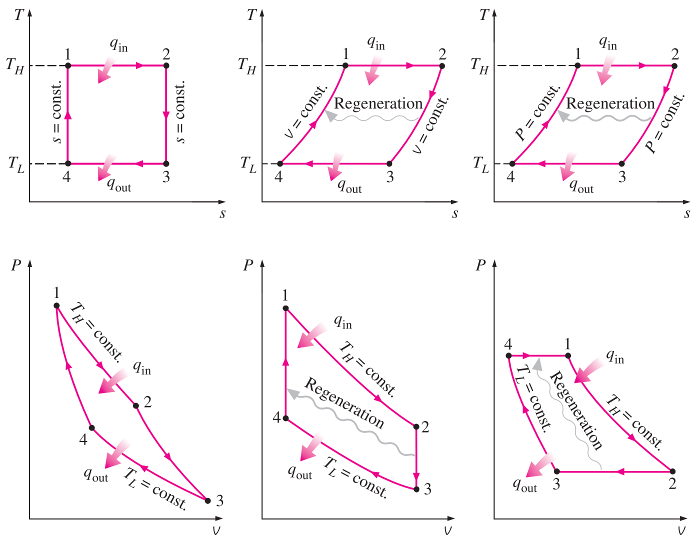
Stirling cycle is made up of four totally reversible processes:

$1 \longrightarrow 2$: Isothermal expansion (heat addition from the external source)

$2 \longrightarrow 3$: Isochoric regeneration (internal heat transfer from the working fluid to the regenerator)

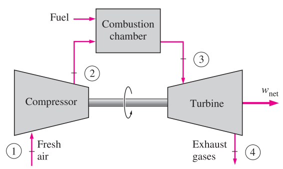
$3 \longrightarrow 4$: Isothermal compression (heat rejection to the external sink)

$4 \longrightarrow 1$: Isochoric regeneration (internal HT from regenerator back to fluid)

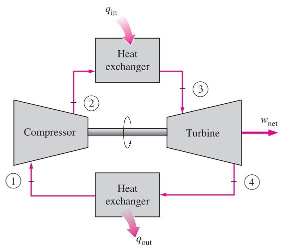


T236 (a) Carnot cycle (b) Stirling cycle (c) Ericsson cycle

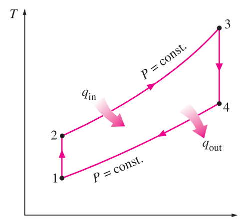
Brayton Cycle: Ideal Cycle for Gas Turbines



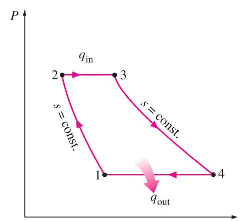
T237 *An open-cycle gas-turbine*



T238 *A closed-cycle gas turbine*



(a) T-s diagram



T239 (b) P-v diagram

- $q_{in} = h_3 - h_2 = c_p(T_3 - T_2)$
- $q_{out} = h_4 - h_1 = c_p(T_4 - T_1)$
- $r_p \equiv \frac{P_2}{P_1} \Rightarrow \frac{T_2}{T_1} = r_p^{(k-1)/k} = \frac{T_3}{T_4}$
- $\eta_{th,Brayton} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)}$

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

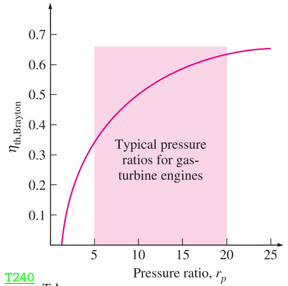
- $w_{net} = c_p\left[(T_3 - T_4) + (T_1 - T_2)\right]$

$\rightarrowtail \frac{w_{net}}{c_p} = T_3\left[1 - \frac{1}{r_p^{(k-1)/k}}\right] + T_1\left[1 - r_p^{(k-1)/k}\right]$

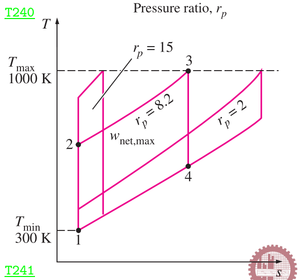
- For max. work: $\frac{\partial w_{net}}{\partial r_p} = 0$:

$\Rightarrow$ $$r_p = \left(\frac{T_3}{T_1}\right)^{k/2(k-1)} : \text{for max. } w_{net}$$

- If $T_3 = 1000\ K,\ T_1 = 300\ K \Rightarrow r_p = 8.2$.

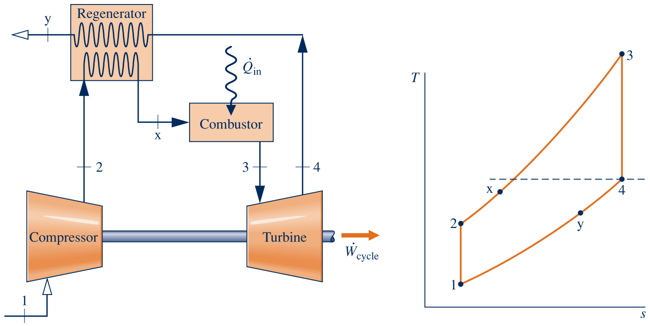


T240



T241

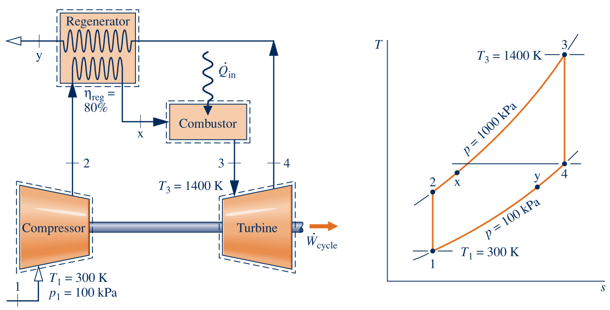
Brayton Cycle with Regeneration



T628

- $q_{regen,act} = h_x - h_2$: $q_{regen,max} = h_4 - h_2$
- $$\text{Effectiveness, } \epsilon \equiv \frac{q_{regen,act}}{q_{regen,max}} = \frac{h_x - h_2}{h_4 - h_2} \simeq \frac{T_x - T_2}{T_4 - T_2}$$
- $$\eta_{th,regen} = 1 - \left(\frac{T_1}{T_3}\right) r_p^{(k-1)/k}$$

Moran 9.7: ▷



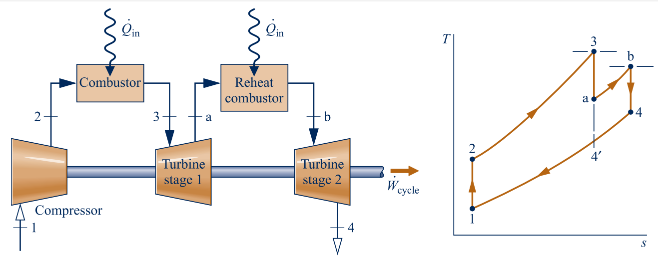
T629

- $\epsilon \equiv \frac{q_{regen,act}}{q_{regen,max}} \simeq \frac{T_x - T_2}{T_4 - T_2} \rightsquigarrow T_x = \epsilon(T_4 - T_2) + T_2 = 745.5$ K
- $w_{net} = w_t + w_c = (h_3 - h_4) + (h_1 - h_2) = C_p[(T_3 - T_4) + (T_1 - T_2)]$

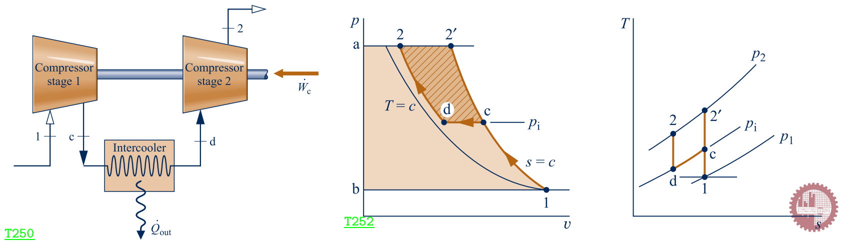
$\Rightarrow$ w_{net}= 427.4 kJ/kg

- $q_{in} = (h_3 - h_x) = C_p(T_3 - T_x) = 753.2$ kJ/kg
- $\eta = \frac{w_{net}}{q_{in}} = 56.7\%$

Turbines with Reheat, Compressors with Intercooling

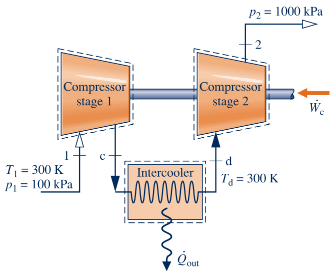


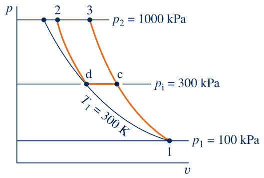
T246



T250 T252

Moran 9.9: ▷

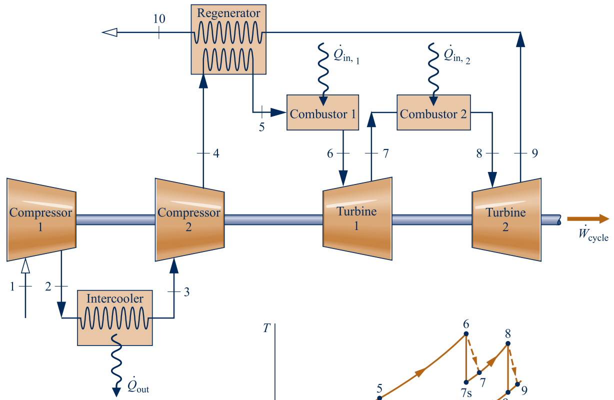


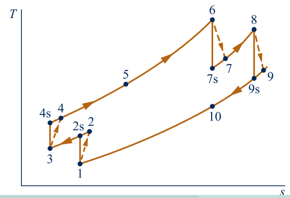


T631

- $h_1 = h(100\ kPa, 300\ K),\ s_1 = s(100\ kPa, 300\ K)$
- $h_c = h(300\ kPa, s_1),\ h_d = h(300\ kPa, 300\ K)$
- $w_c = w_{c1} + w_{c2} = (h_1 - h_c) + (h_d - h_2)$ = -234.9 kJ/kg◀
  If single stage compression is done:
- $h_2 = h(1000\ kPa, s_1)$
- $w_c = (h_1 - h_2)$ = -280.1 kJ/kg◀

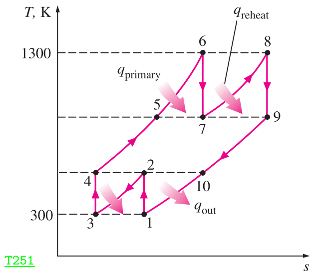
Brayton Cycle with Intercooling, Reheating & Regeneration





T249

Cengel 9.8: ▷ GT with reheating & intercooling: $r_p = 8$, $\eta_{isen} = 1.0$, $\epsilon = 1.0$



T251

- In case of staging, for min. compressor work or for max. turbine work:

$$P_i = \sqrt{P_{min} P_{max}}$$

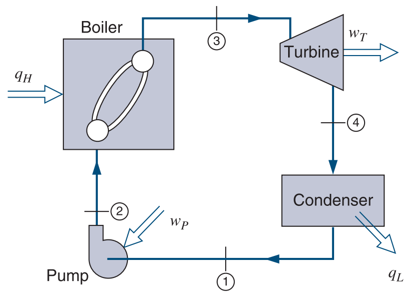
- $w_{turb} = (h_6 - h_7) + (h_8 - h_9)$
- $w_{comp} = (h_2 - h_1) + (h_4 - h_3)$
- $w_{net} = w_{turb} - w_{comp},\ bwr = \frac{w_{comp}}{w_{turb}}$
- **Without regen:** w_{turb} = 685.28 kJ/kg ◀ w_{comp} = 208.29 kJ/kg ◀
- $q_{in} = (h_6 - h_4) + (h_8 - h_7)$ = 1334.30 kJ/kg ◀

⇒ η_{th} = 0.358 ◀ bwr = 0.304 ◀

- **With regen:** turbine and compressor works remains unchanged.
- $q_{in} = (h_6 - h_5) + (h_8 - h_7)$ = 685.28 kJ/kg ◀

⇒ η_{th} = 0.696 ◀ bwr = 0.304 ◀

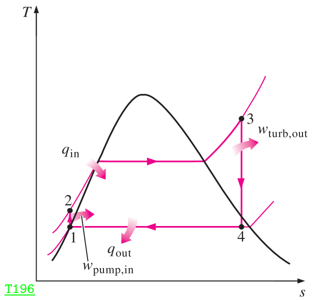
Rankine Cycle for Vapour Power Plant



T195

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

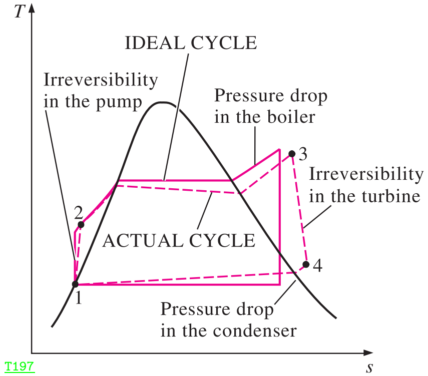
Cengel 10.1: ▷ Ideal Rankine Cycle: Steam enters the turbine at 3 MPa and 350°C, and condenser is at 75 kPa.

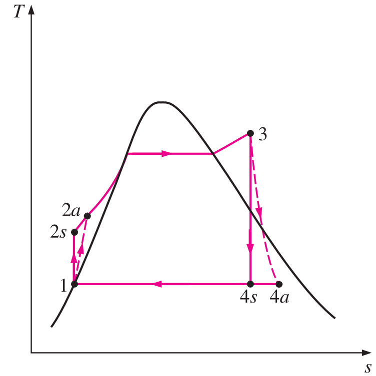


T196

- ⇒ Steady state ⇒ $dE_{cv}/dt = 0$
- ⇒ $Z_2 = Z_1$ & $\mathbb{V}_2 = \mathbb{V}_1$, $\dot{m}_i = \dot{m}_e = \dot{m}$
- ⇒ $\boxed{0 = q - w + (h_i - h_e)}$
- Turbine: $q = 0$, $w = w_T = h_3 - h_4$
- Pump: $q = 0$, $w = w_P = h_1 - h_2$
- Boiler: $w = 0$, $q = q_B = h_3 - h_2$
- Condenser: $w = 0$, $q = q_C$
- h_2 is in compressed state & difficult to estimate. Hence, $w_P = -\int_1^2 v dP$.
- ⇒ $w_{net} = w_T + w_P = (713.0 - 3.03)$ kJ/kg $= 710.0$ kJ/kg. ▶ $w_T \gg w_P$.
- ⇒ $q_{in} = q_B = h_3 - h_2 = 2728.6$ kJ/kg
- ⇒ Thermal efficiency, $\eta_{th} = \frac{w_{net}}{q_{in}} = 0.260 = 26.0\%$ ◀
- ⇒ Back work ratio, $bwr = |\frac{w_P}{w_T}| = 0.0043$ ◀

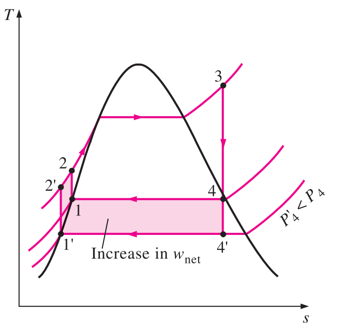
Deviation of Actual Vapour Power Cycle



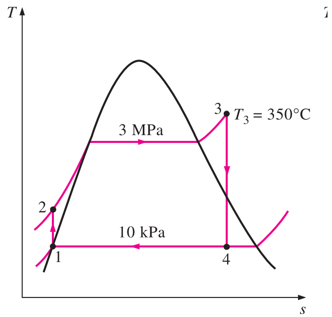


T197

Effect of Lowering Condenser Pressure



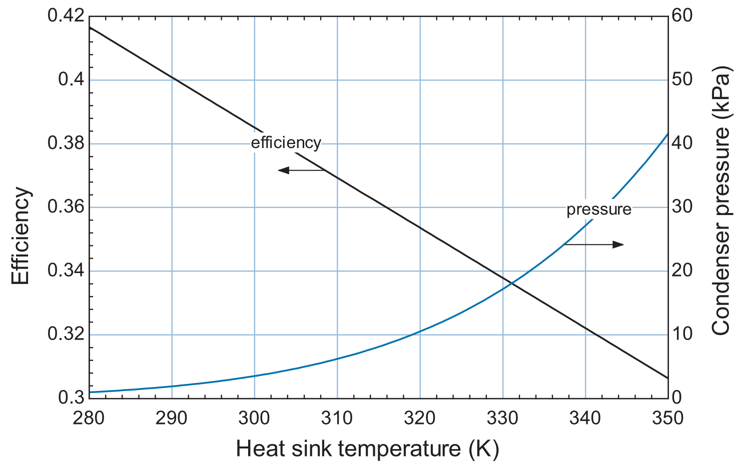
T199



T218

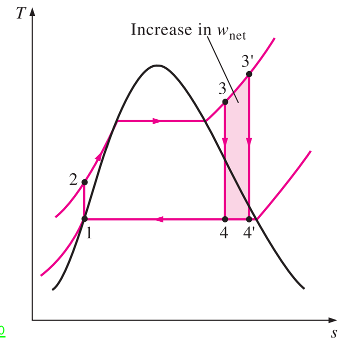
- $P_{cond} = P_{sat}(T_{cond})$: $T_{cond} - T_{atm} \simeq 10 - 15°C$.
- $P_{cond} \downarrow \Rightarrow w_{net} \uparrow$, $\eta_{th} \uparrow$ & $x_4 \downarrow$. Higher moisture decreases turbine efficiency and erodes its blades. In general, $x_4 \geqslant 0.9$ is maintained. Lower P_{cond} promotes leakage.

$P_B = 2.64$ MPa, $T_H = 800$ K

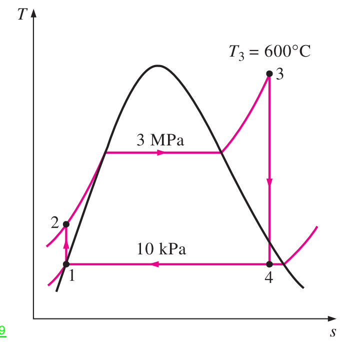


T217

Effect of Super-heating Steam to Higher Temperature



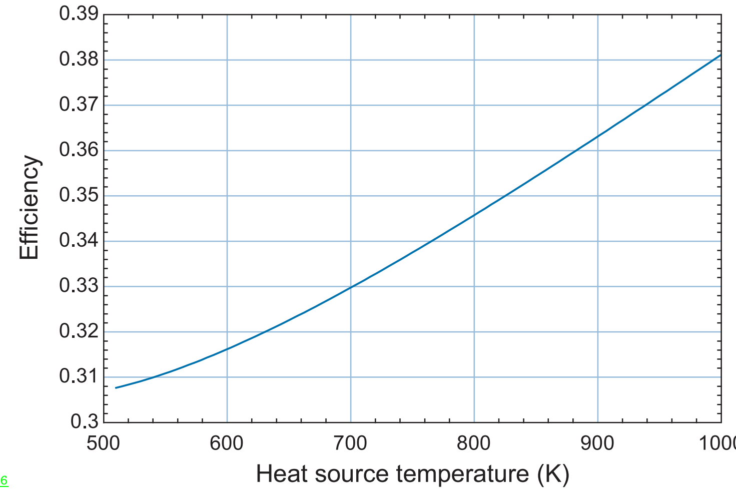
T200



T219

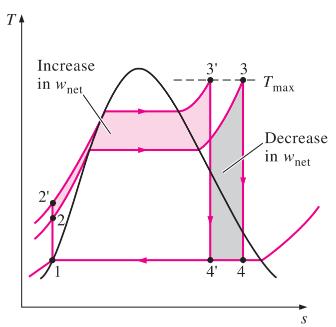
- $T_{max} \uparrow \Rightarrow w_{net} \uparrow, \eta_{th} \uparrow$ & $x_4 \uparrow$.
- Higher average temperature of heat addition increases η_{th}. T_{max} is limited by metallurgical considerations. In general, $T_{max} = 620^oC$.

$P_B = 2.64$ MPa, $T_L = 325$ K

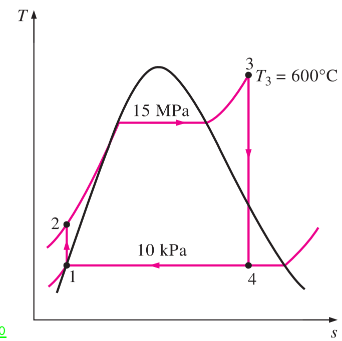


T216

Effect of Increasing Boiler Pressure



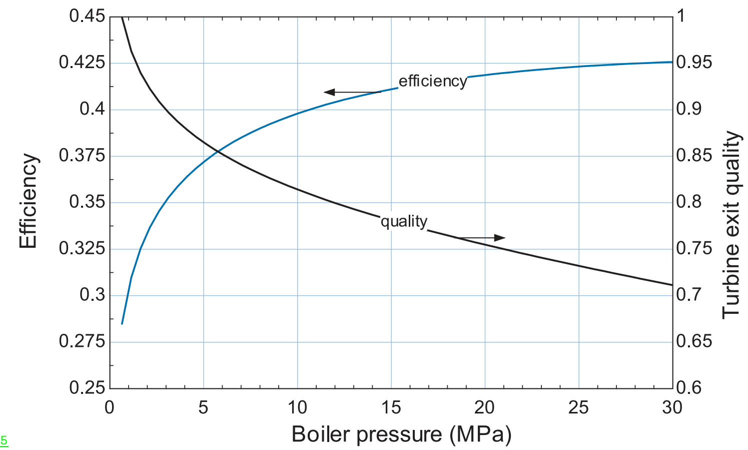
T201



T220

- For fixed T_{max} : $P_B \uparrow \Rightarrow \eta_{th} \uparrow$ & $x_4 \downarrow$. Higher η_{th} is achieved because of higher average temperature of heat addition.

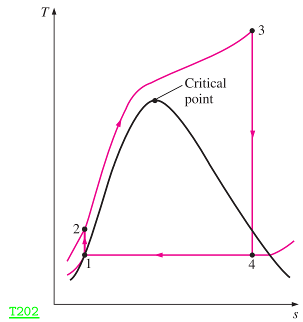
$T_H = 800$ K, $T_L = 325$ K



T215

Effects of Operating Parameters on Ideal Rankine Cycle Efficiency

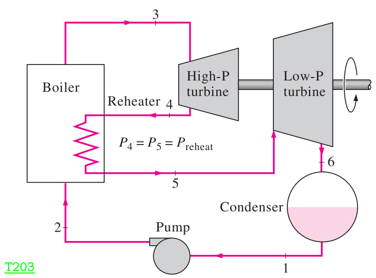
Boiler Pressure	[MPa]	3.0	3.0	**3.0**	**15.0**
Max. Temperature	[°C]	350	**350**	**600**	600
Cond. Pressure	[kPa]	**75**	**10**	10	10
Heat added	[kJ/kg]	2729	2921	3488	3376
Turbine work	[kJ/kg]	713	979	1302	1467
Pump work	[kJ/kg]	3.03	3.02	3.02	15.1
Thermal efficiency	[%]	26.0	33.4	37.3	43.0
x_4	[-]	0.886	0.812	0.915	0.804

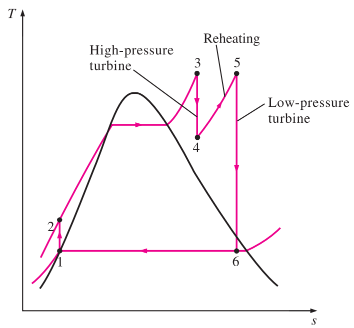


A supercritical Rankine cycle

- Some modern power plants operate at supercritical pressure ($P \approx 30$ MPa $> P_C = 22.06$ MPa) and have $\eta_{th} \sim 40\%$ for fossil-fuel plants and $\eta_{th} \sim 34\%$ for nuclear power plants.
- Lower η_{th} of nuclear power plants are due to lower maximum temperatures used due to safety reasons.

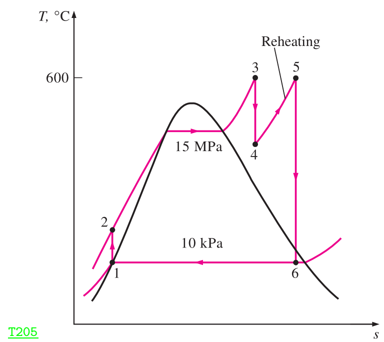
Ideal Reheat Rankine Cycle





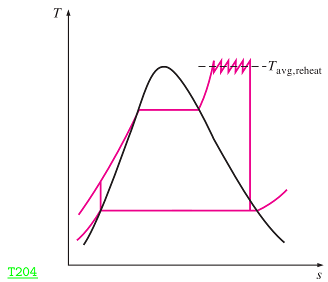
- $q_{in} = q_{23} + q_{45} = (h_3 - h_2) + (h_5 - h_4)$
- $w_{turb} = w_{34} + w_{56} = (h_3 - h_4) + (h_5 - h_6)$
- Average temperature of heat addition is increased $\Rightarrow \eta_{th} \uparrow$. Moisture quality at turbine exit is also improved.

$P_{reheat} = 4.0$ MPa

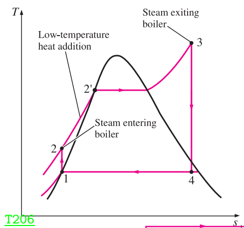


Heat added	=	3897.2 [kJ/kg]
Turbine work	=	1768.5 [kJ/kg]
Pump work	=	-15.1 [kJ/kg]
Thermal efficiency	=	45.0 [%]

Average temperature at which heat is transferred during reheating increases as the number of reheat stages is increased.

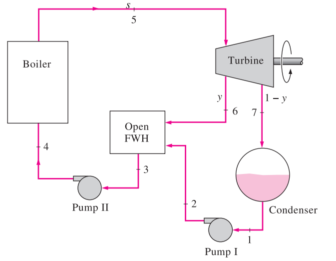


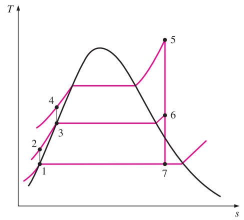
Ideal Regenerative Rankine Cycle with Open FWH



T206

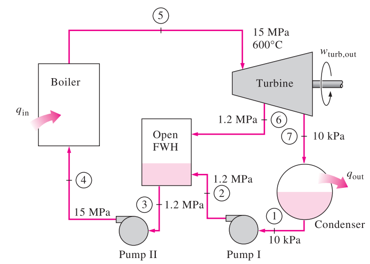
$T_{av}(2 \to 2')$ is low $\Rightarrow \eta_{th} \downarrow$. A practical regeneration process in steam power plants is accomplished by extracting, or bleeding, steam from the turbine at various points. The device where the feed-water is heated by regeneration is called a **regenerator**, or a **feed-water heater (FWH)**.

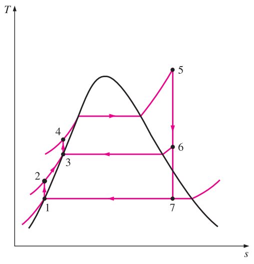




T207

Cengel 10.5: ▷ Ideal Regenerative Rankine Cycle with Open FWH.



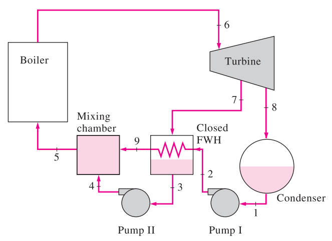


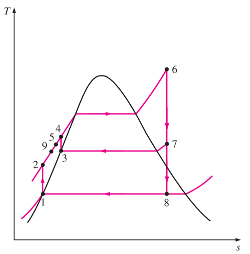
T210

- $q_{in} = h_5 - h_4 = 2769.2$ kJ/kg
- Energy balance (at FWH): $yh_6 + (1-y)h_2 = 1 \cdot h_3 \to y = 0.227$.
- $w_t = (h_5 - h_6) + (1-y)(h_6 - h_7) = 1299.1$ kJ/kg
- $w_P = (1-y)w_{P,I} + w_{P,II} =$ -16.58 kJ/kg

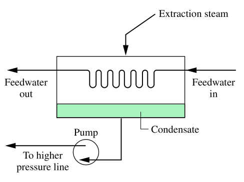
$\Rightarrow \eta_{th} = 46.3\%$ ◀

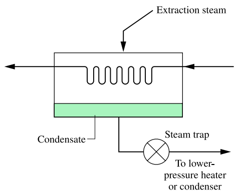
Closed Feed-water Heaters



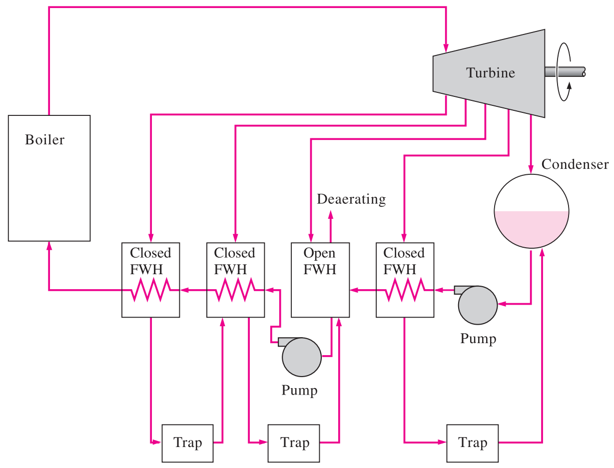


T208





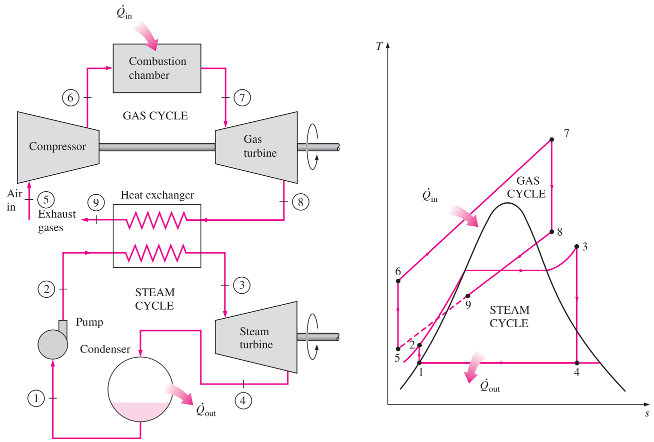
T221



T209

A steam power plant with one open FWH and three closed FWH.

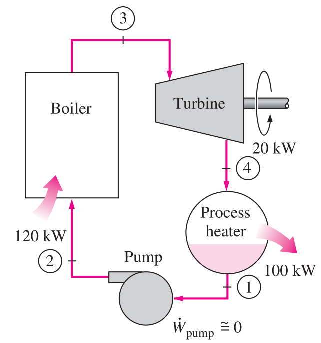
Combined Gas-Vapour Power Cycle



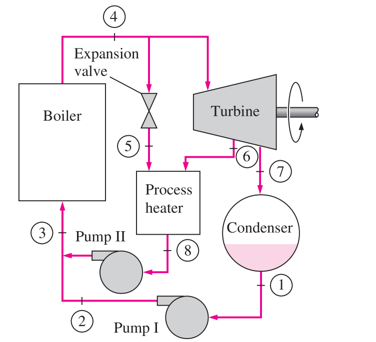
T214

Heat Exchanger → HRSG ≡ Heat Recovery Steam Generator

Cogeneration: CHP ≡ Combined Heating & Power



T211

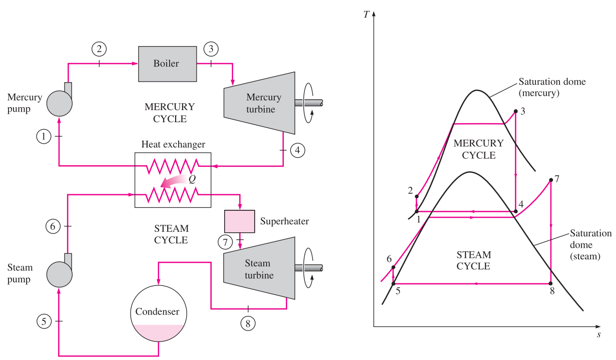


T212

- Cogeneration is the production of more than one useful form of energy (such as process heat and electric power) from the same energy source.
- Utilization factor, $\epsilon_u = \frac{\dot{W}_{net}+\dot{Q}_P}{\dot{Q}_{in}} = 1 - \frac{\dot{Q}_{out}}{\dot{Q}_{in}}$.

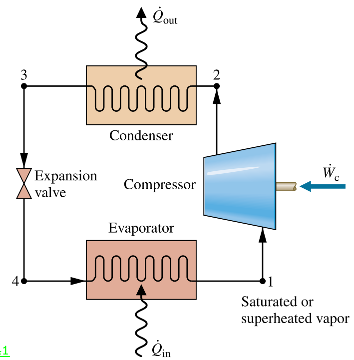
Binary Vapour Cycle

- For Mercury: $P_C = 18$ MPa, $T_C = 898$°C.
- At 0.07 kPa, $T_{sat} = 32$°C & at 7.0 kPa, $T_{sat} = 273$°C.
- $\eta_{th} \geqslant 50\%$ is possible with binary-vapour cycle.

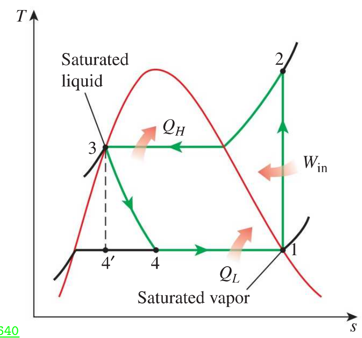


T213

Ideal Vapour Compression Refrigeration Cycle



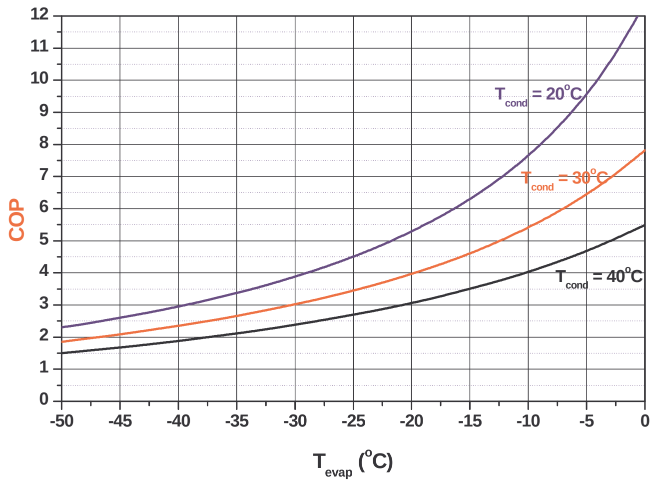
T641



T640

1 → 2: Isentropic compression, $P_{evap} \rightarrow P_{cond}$
2 → 3: Isobaric heat rejection, Q_H
3 → 4: Isenthalpic expansion, $P_{cond} \rightarrow P_{evap}$
4 → 1: Isobaric heat extraction, Q_L

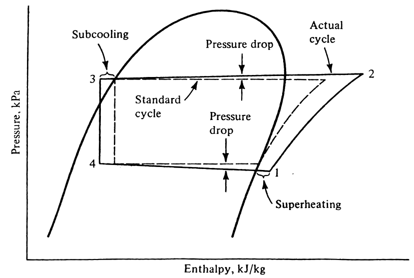
Effect of Evaporator & Condenser Temperatures



T260

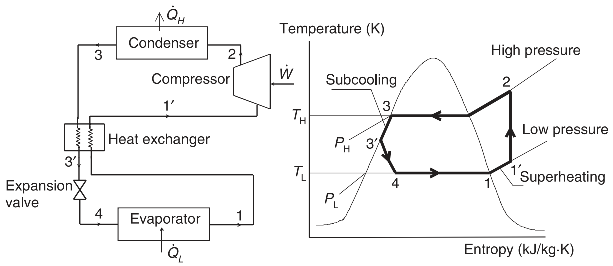
Deviations from Ideal Cycle

1. Refrigerant pressure drop in piping, evaporator, condenser, receiver tank, and through the valves and passages.
2. Sub-cooling of liquid leaving the condenser.
3. Super-heating of vapour leaving the evaporator.
4. Compression process is not isentropic.



T262

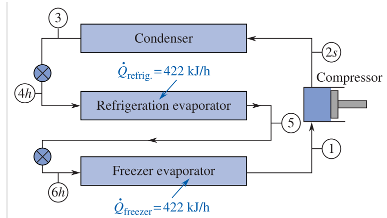
Super-heating & Sub-cooling

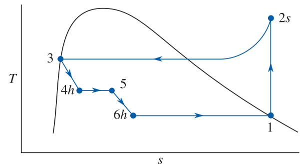


T263

- Sub-cooling of liquid serves a desirable function of ensuring that 100% liquid will enter the expansion device.
- Super-heating of vapour ensures no droplets of liquid being carried over into the compressor.
- Even through refrigeration effect is increased, compression work is greater & probably has negligible thermodynamic advantages.

Example: ▷ A dual-evaporator refrigeration system: T_F = -18.0°C, T_R = 4.0°C, T_c = 30.0°C & η_c = 80%.

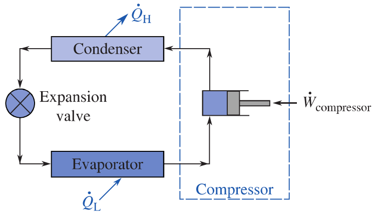




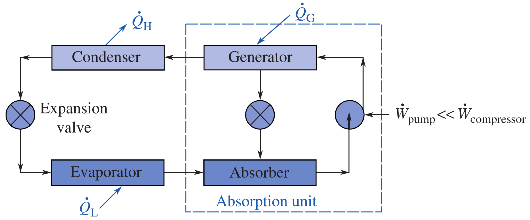
- $COP_R = ?\quad x_5 = ?$
- $\dot m = \frac{\dot Q_R + \dot Q_F}{h_1 - h_{4h}} = 1.62 \times 10^{-3}$ kg/s ◀
- $\dot W_c = \frac{\dot m (h_{2s} - h_1)}{\eta_c}$
- $COP_R = \frac{\dot Q_R + \dot Q_F}{\dot W_c} = 3.37$ ◀
- $\dot Q_R = \dot m (h_5 - h_{4h}) \rightarrow h_5$
- $h_5 \Rightarrow x_5 = 0.56$ ◀

T274

Vapour Absorption Refrigeration System

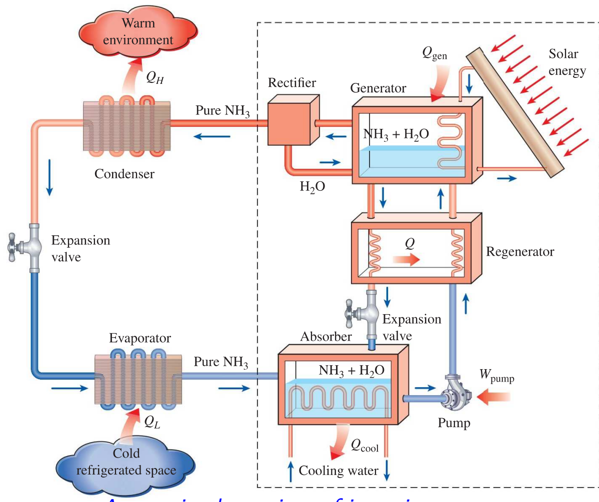


(a) Standard vapor-compression refrigeration.



(b) Absorption vapor-compression refrigeration.

T1388



T267

Ammonia absorption refrigeration system

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