Thermodynamics (Mechanical) 4

Rankine Cycle

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Overview for this Unit

Content

- Introduction to vapour power plants
- Ideal Rankine cycle and deviations from it
- Options for improving Rankine cycle performance
- Introduction to cogeneration and combined power cycles

Aim

- To provide you with underpinning knowledge needed to successfully analyse these cycles

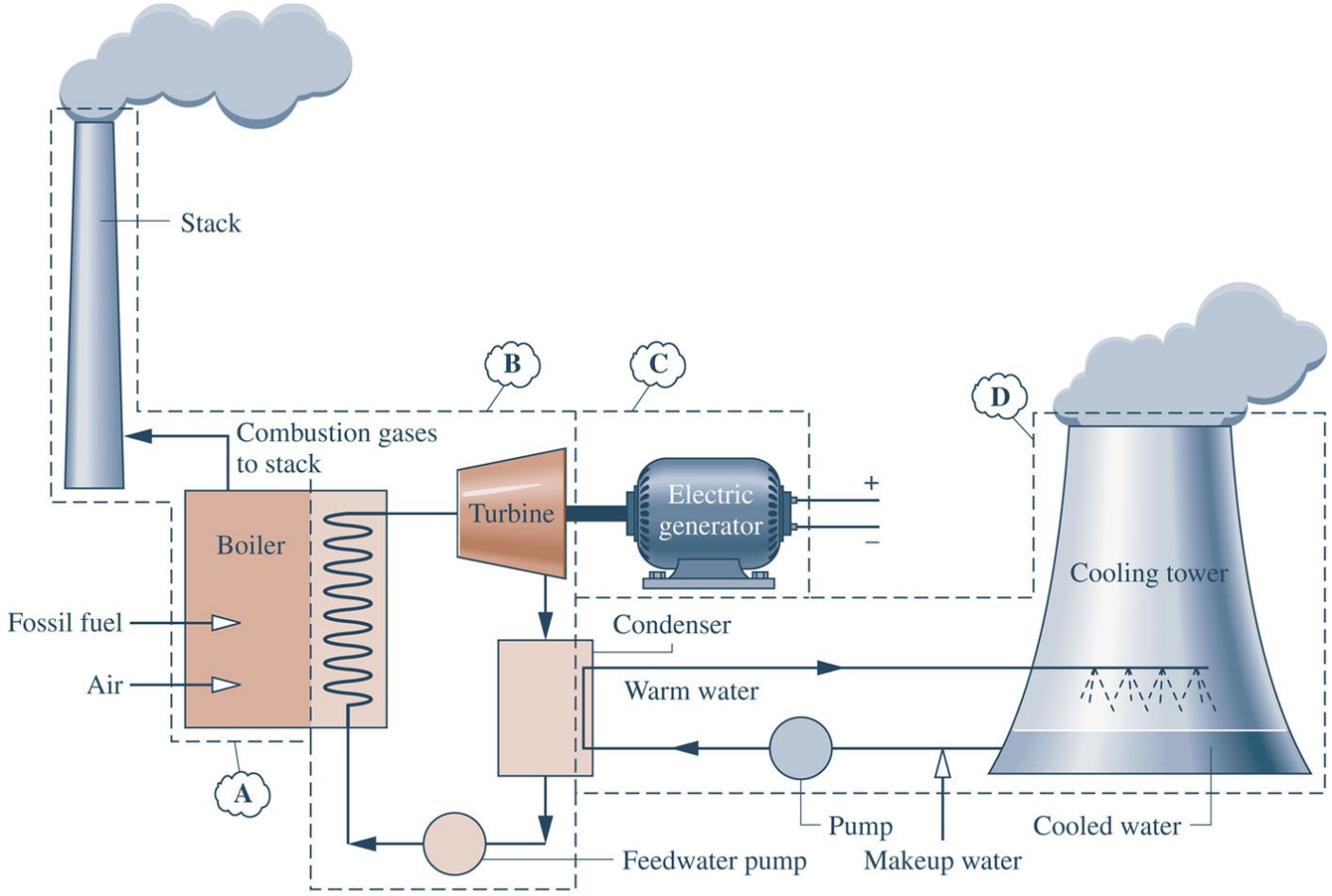
Borgnakke and Sonntag (8^{th} or 10^{th} ed) Sections 9.2–9.7

Introducing Vapour Power Plants

- The components of four alternative vapour power plant configurations are shown schematically in the next slides. They are
  - Fossil-fueled vapour power plants
  - Pressurised-water reactor nuclear vapour power plants
  - Concentrating solar thermal vapour power plants
  - Geothermal vapour power plants
- In each of the four types of vapour power plant, a working fluid is alternately vaporised and condensed. A key difference among the plants is the origin of the energy required to vaporise the working fluid.

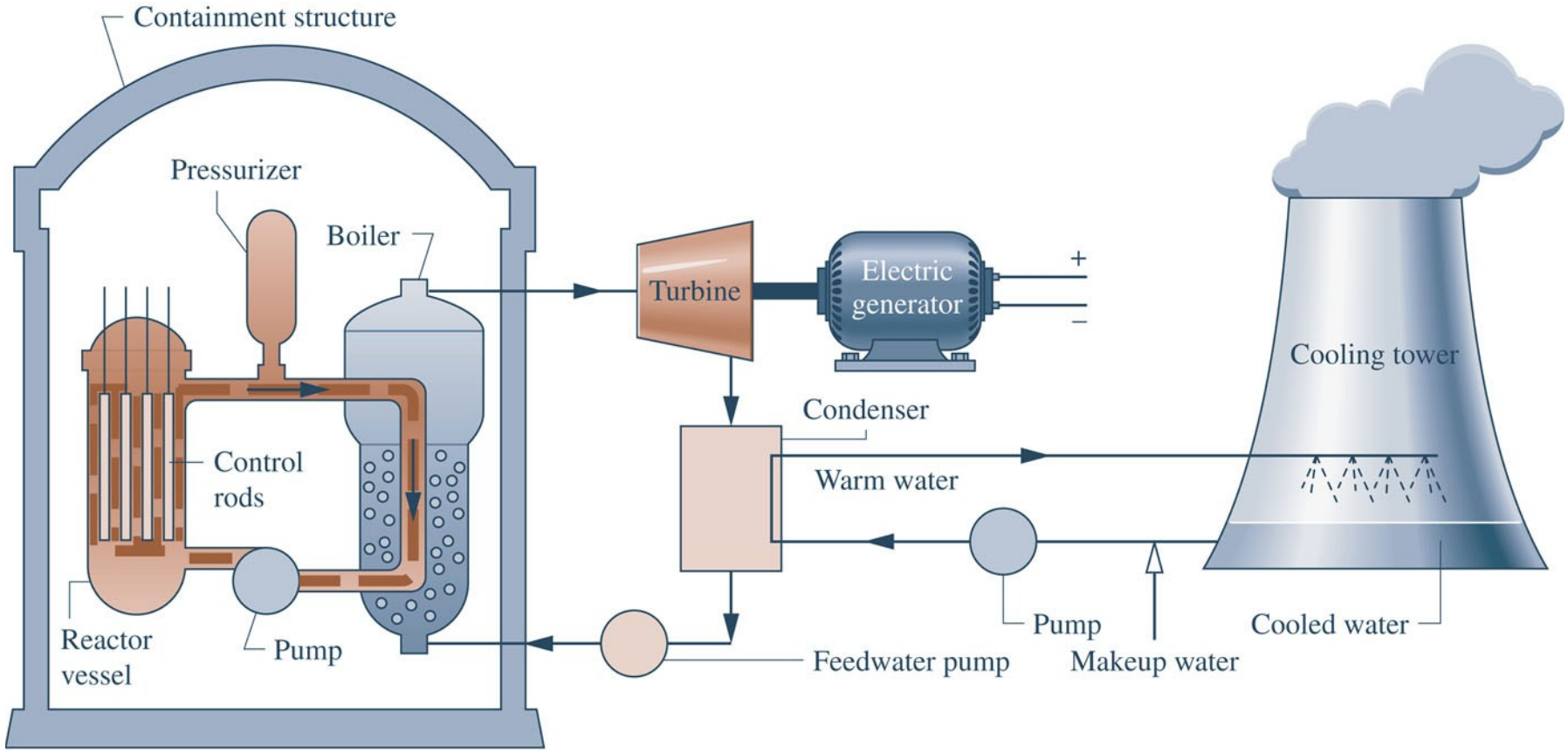
Introducing Vapour Power Plants

- In **fossil-fueled plants**, the energy required for vaporisation originates in combustion of the fuel.



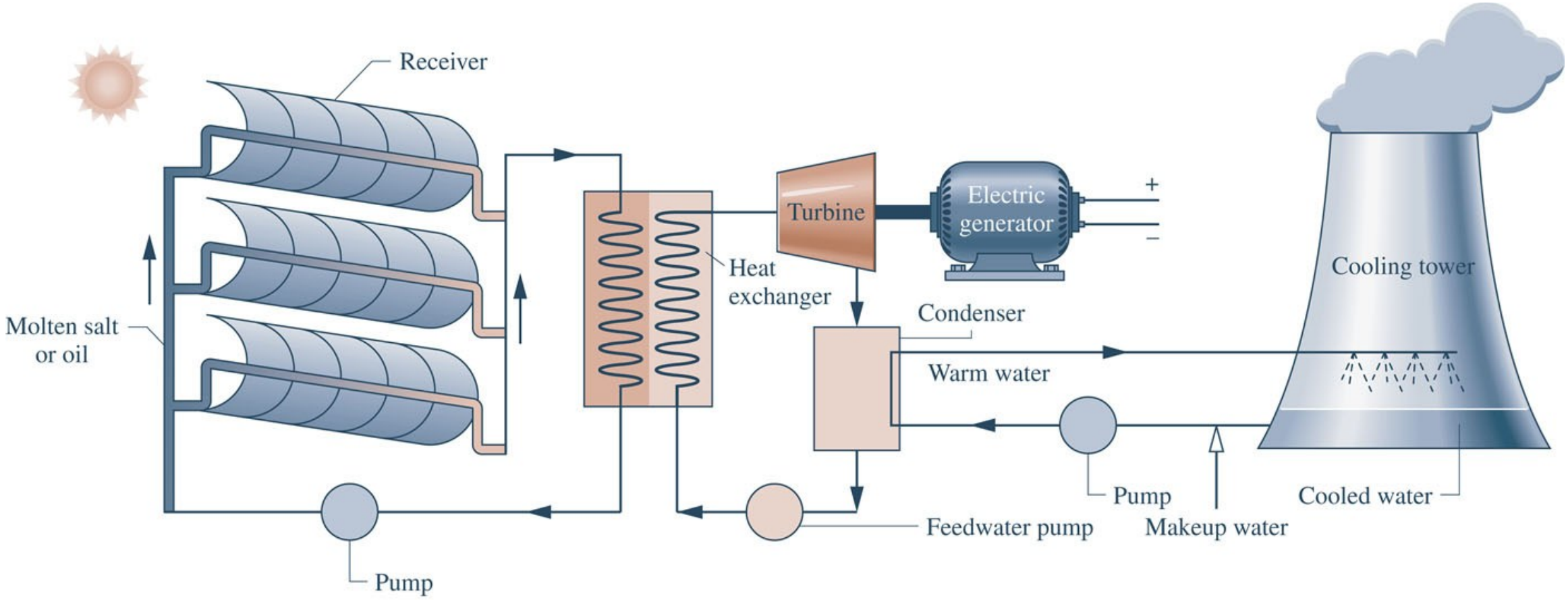
Introducing Vapour Power Plants

- In **nuclear plants**, the energy required for vaporisation originates in a controlled nuclear reaction.



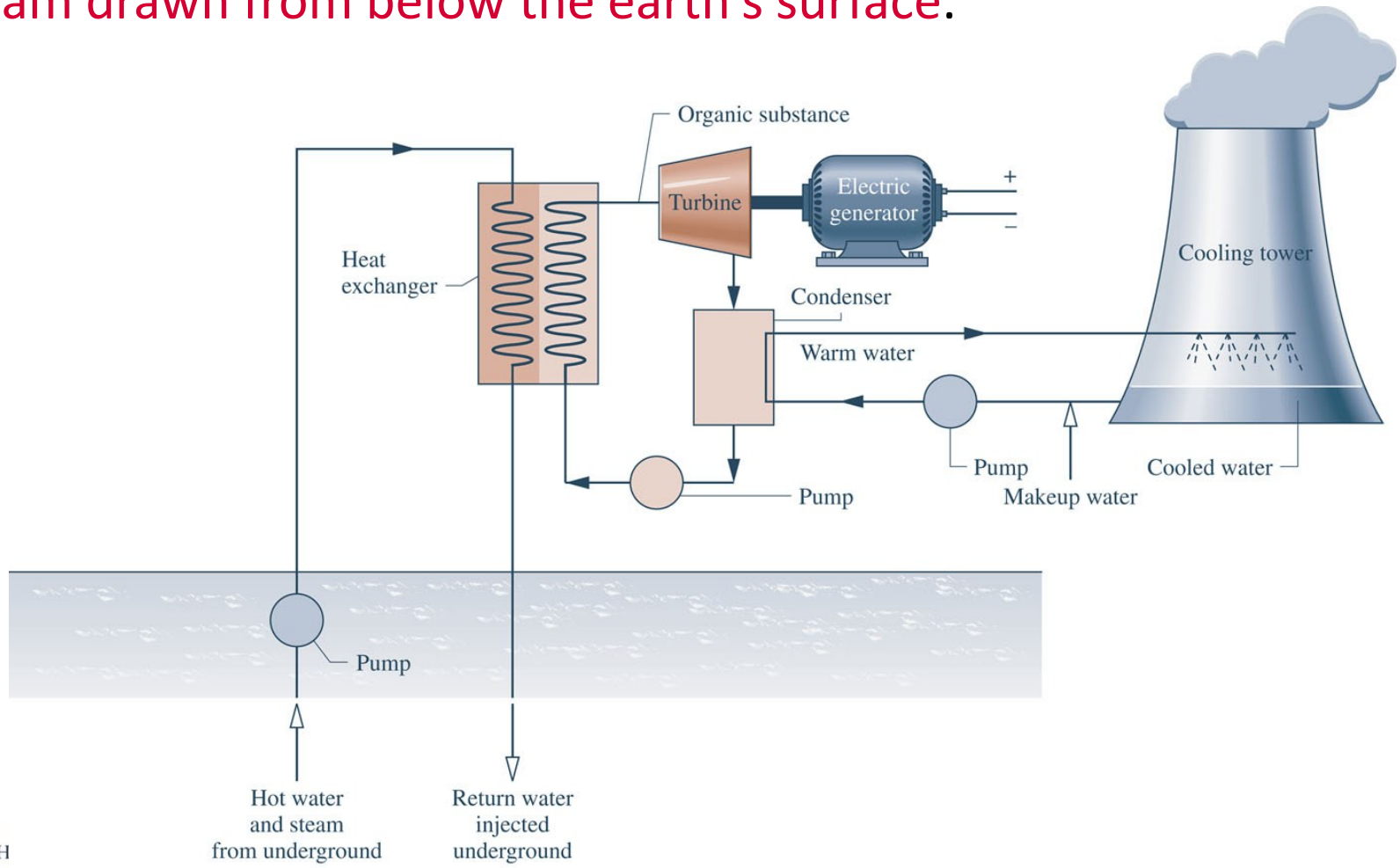
Introducing Vapour Power Plants

- In **solar plants**, the energy required for vaporisation originates in collected and concentrated solar radiation.

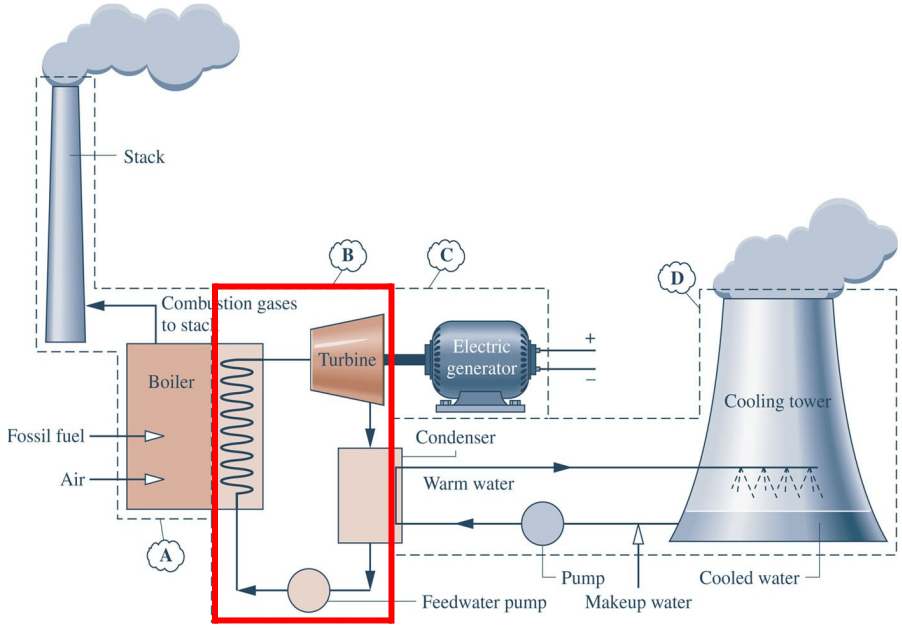


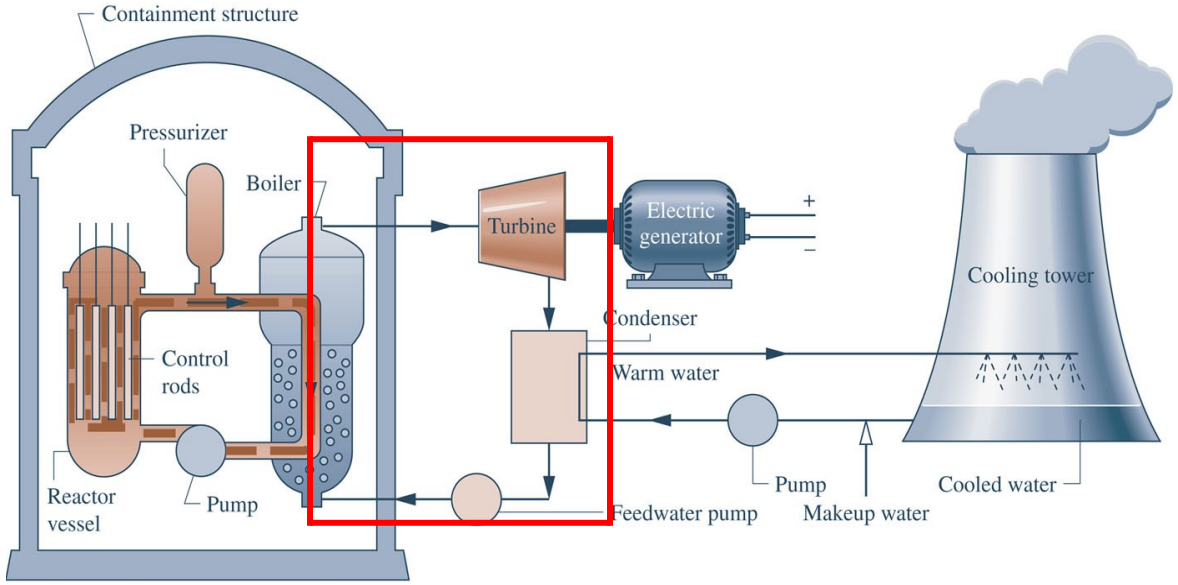
Introducing Vapour Power Plants

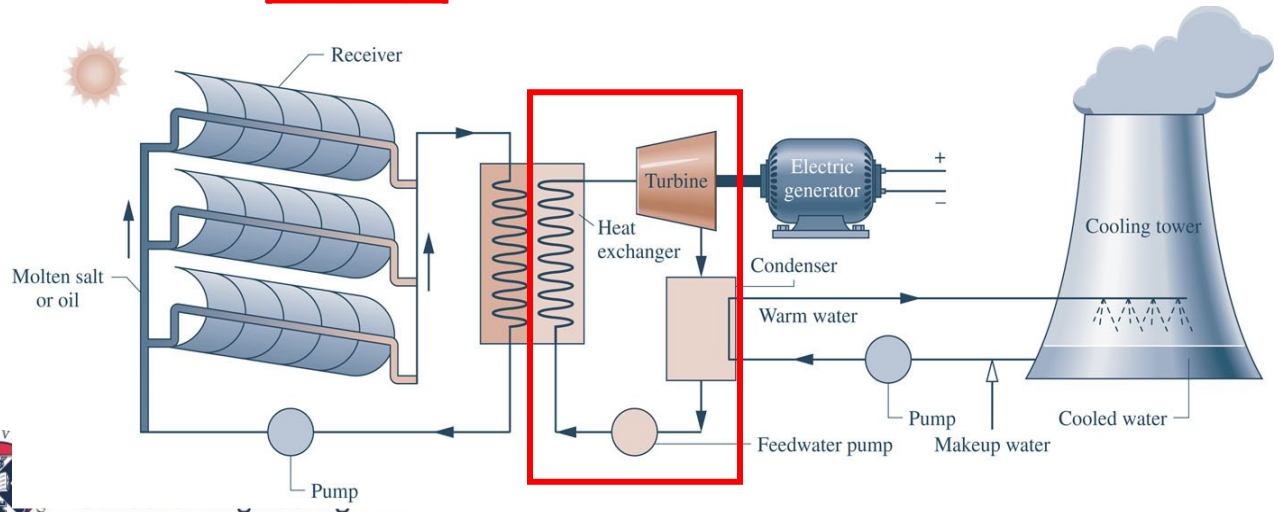
- In **geothermal plants**, the energy required for vaporisation originates in hot water and/or steam drawn from below the earth's surface.

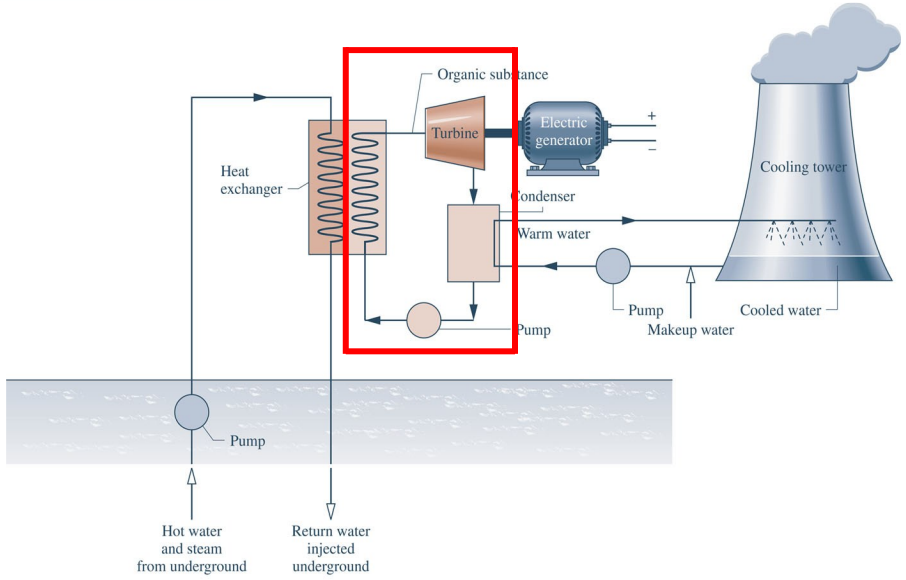


Introducing Vapour Power Plants



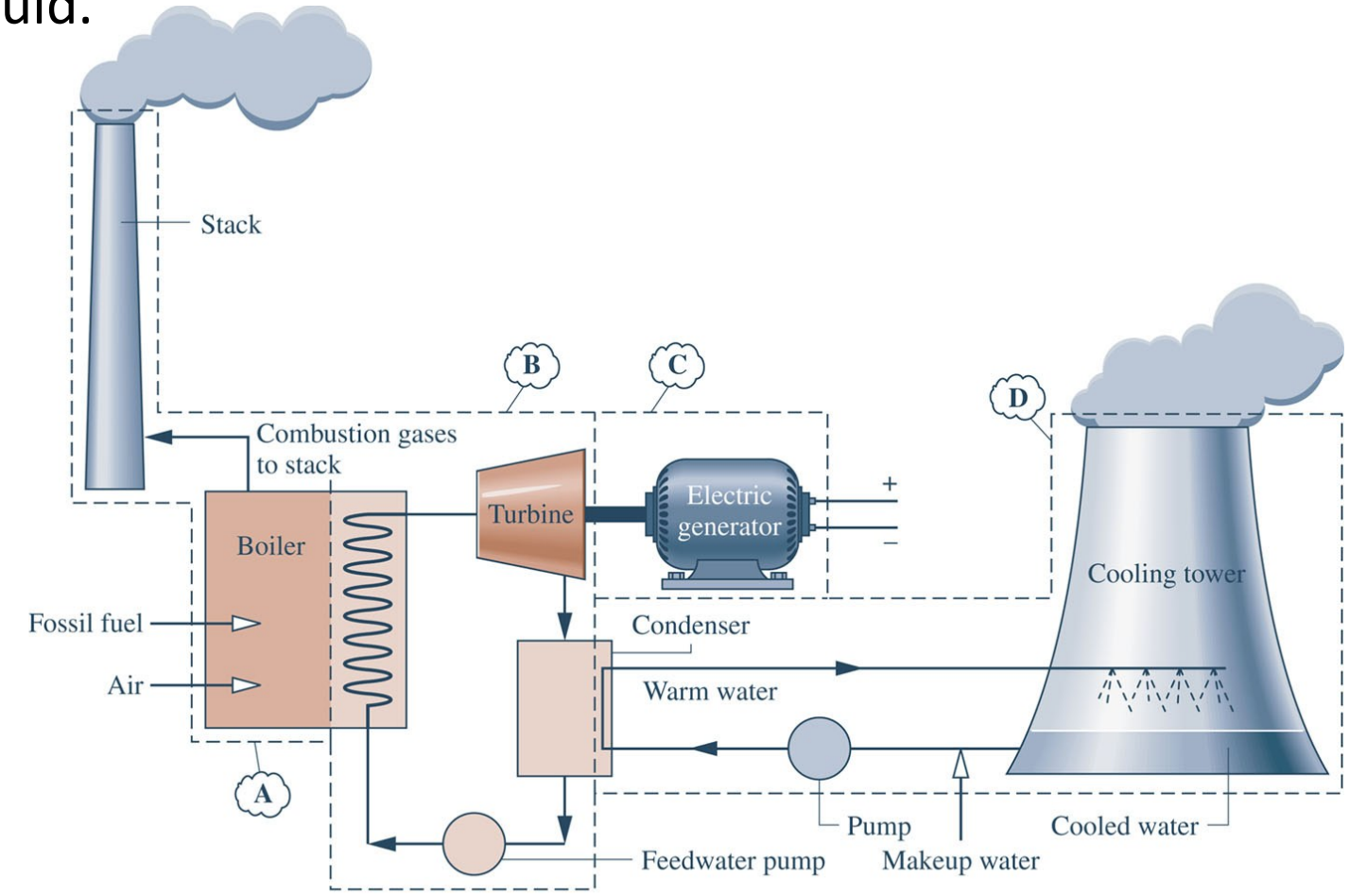






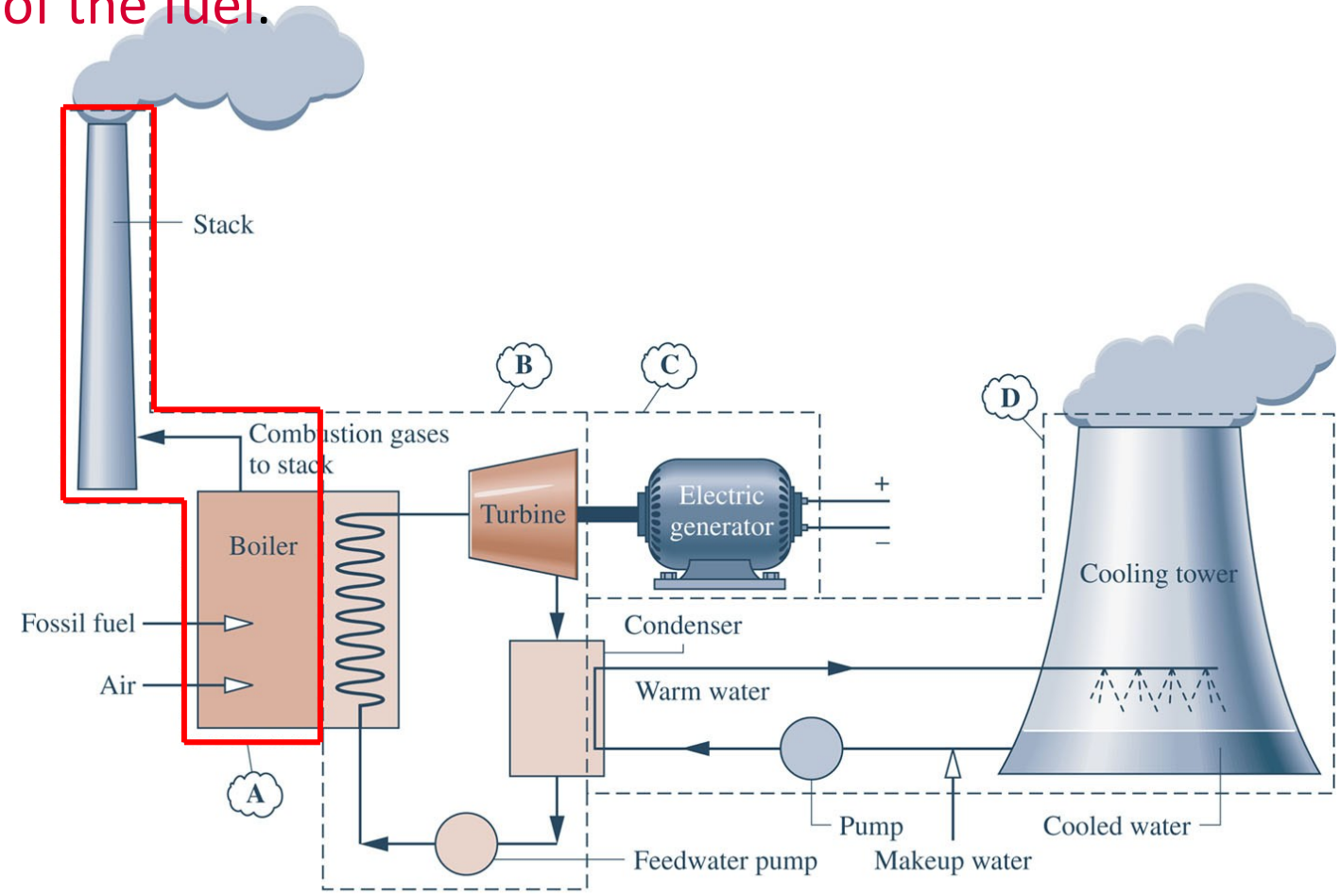
Introducing Vapour Power Plants

- Let us consider the **fossil-fueled vapour power** plant as representative. The overall plant can be broken into four major subsystems identified by A, B, C, and D. Water is the working fluid.



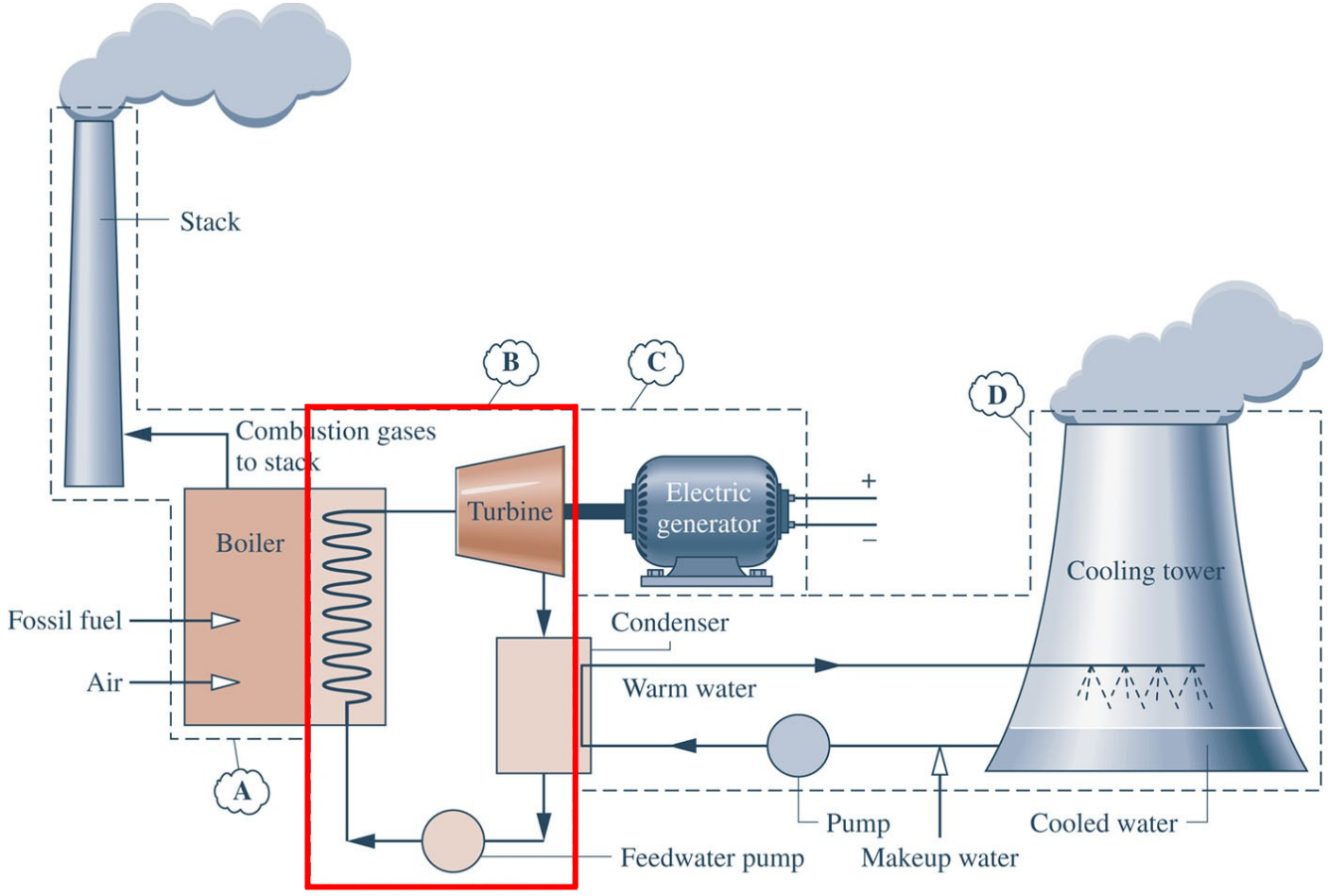
Introducing Vapour Power Plants

- Subsystem A provides the heat transfer of energy needed to vaporise water circulating in subsystem B. In fossil-fueled plants this heat transfer has its origin in the combustion of the fuel.



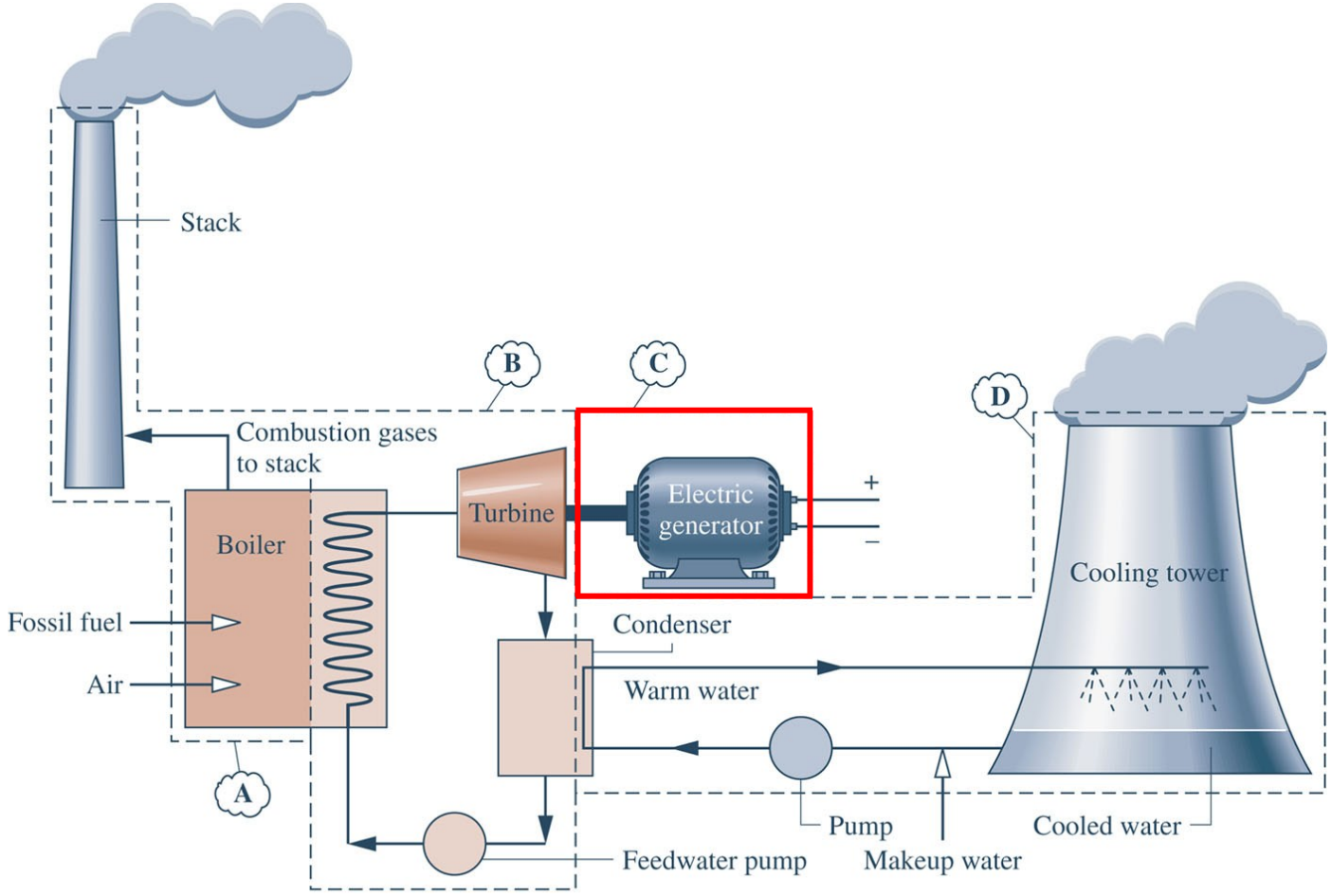
Introducing Vapour Power Plants

- In subsystem B, the water vapour expands through the turbine, developing power. The water then condenses and returns to the boiler.



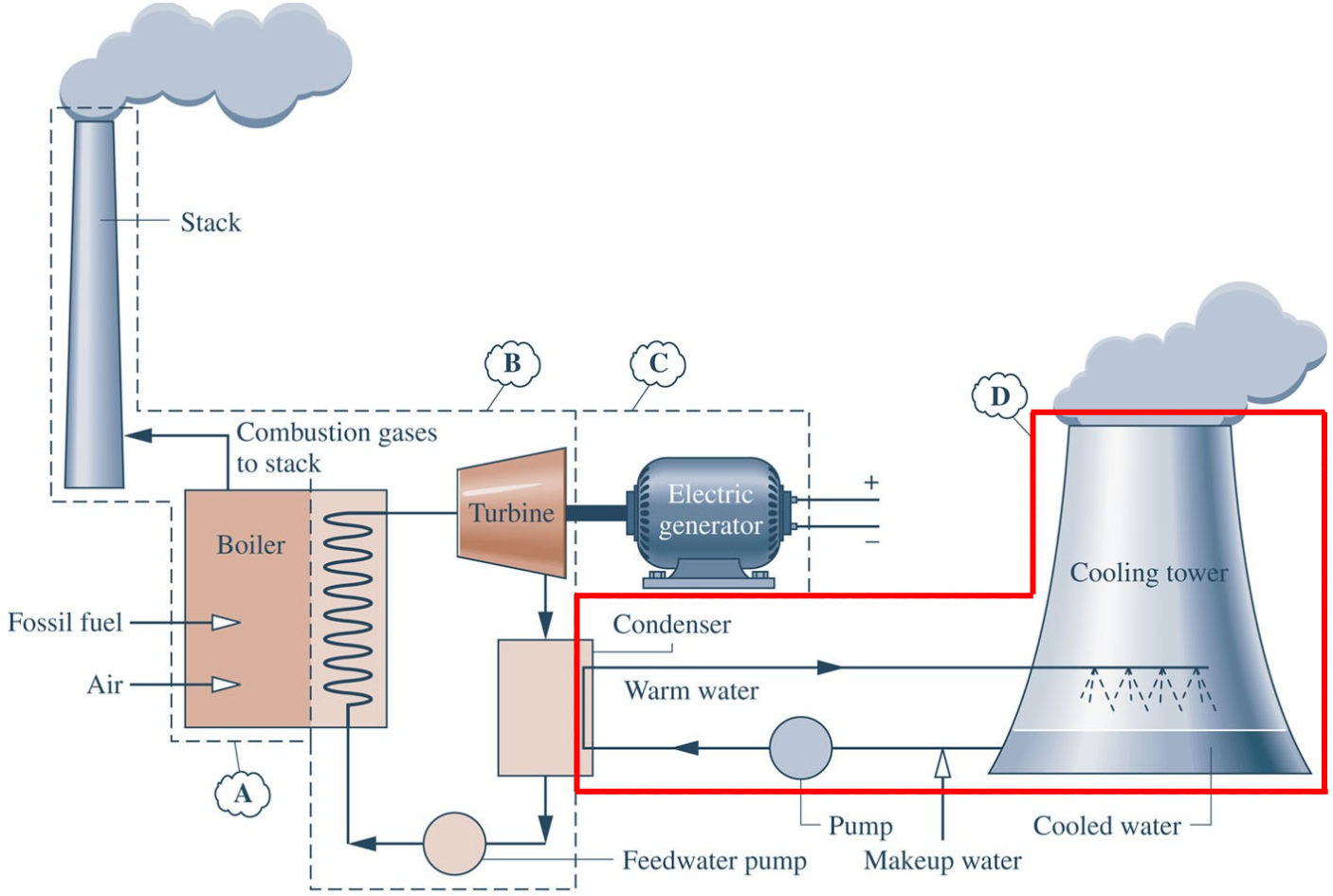
Introducing Vapour Power Plants

- In subsystem C, power developed by the turbine drives an electric generator.



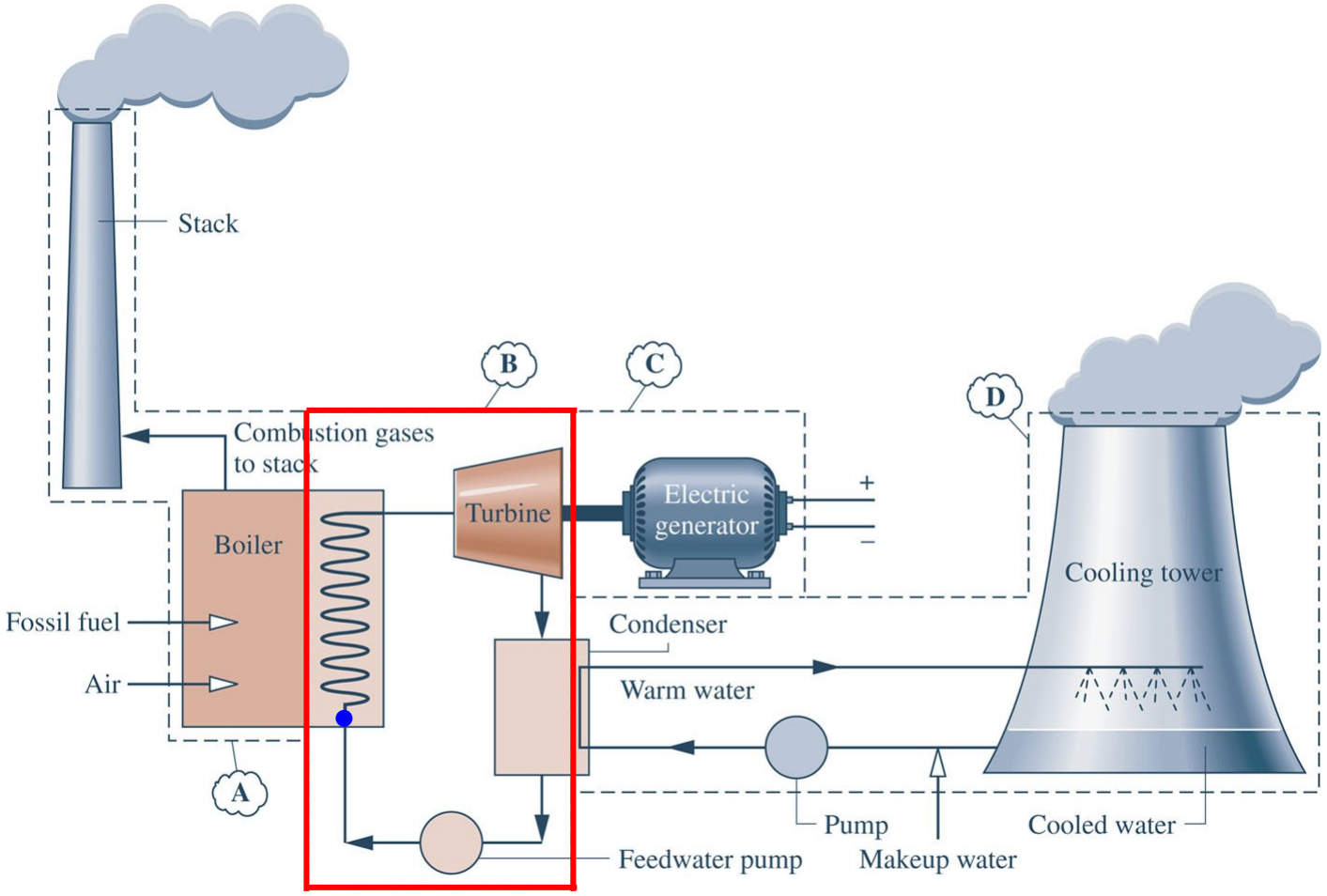
Introducing Vapour Power Plants

- Subsystem D removes energy by heat transfer arising from steam condensing in subsystem B.



Introducing Vapour Power Plants

- Each unit of mass of water (or working fluid) periodically undergoes a thermo-dynamic cycle as it circulates through the components of subsystem B. This cycle is the *Rankine cycle*.



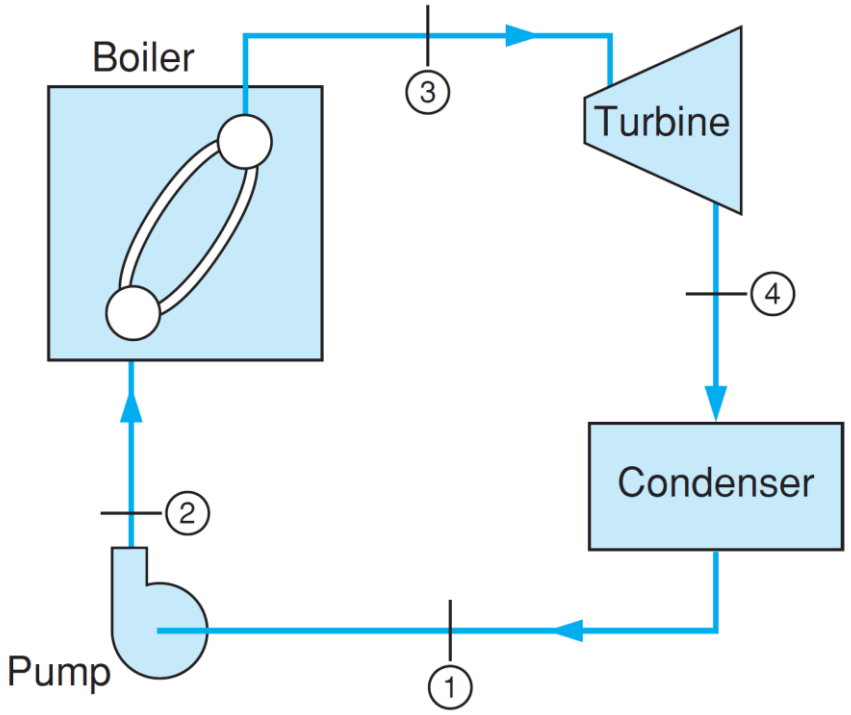
Rankine Cycle

- A bit of history:
  - The basic Rankine cycle is named after Edinburgh-born William John Macquorn Rankine, a 19th century Scottish mechanical engineer. Rankine studied at the University of Edinburgh from 1836 to 1838 and is a pioneer in the field of thermodynamics.
  - Rankine was a founding contributor, with Rudolf Clausius and William Thomson (Lord Kelvin), to the science of thermodynamics, particularly focusing on its First Law.



Source: www.wikipedia.org

Ideal Rankine Cycle



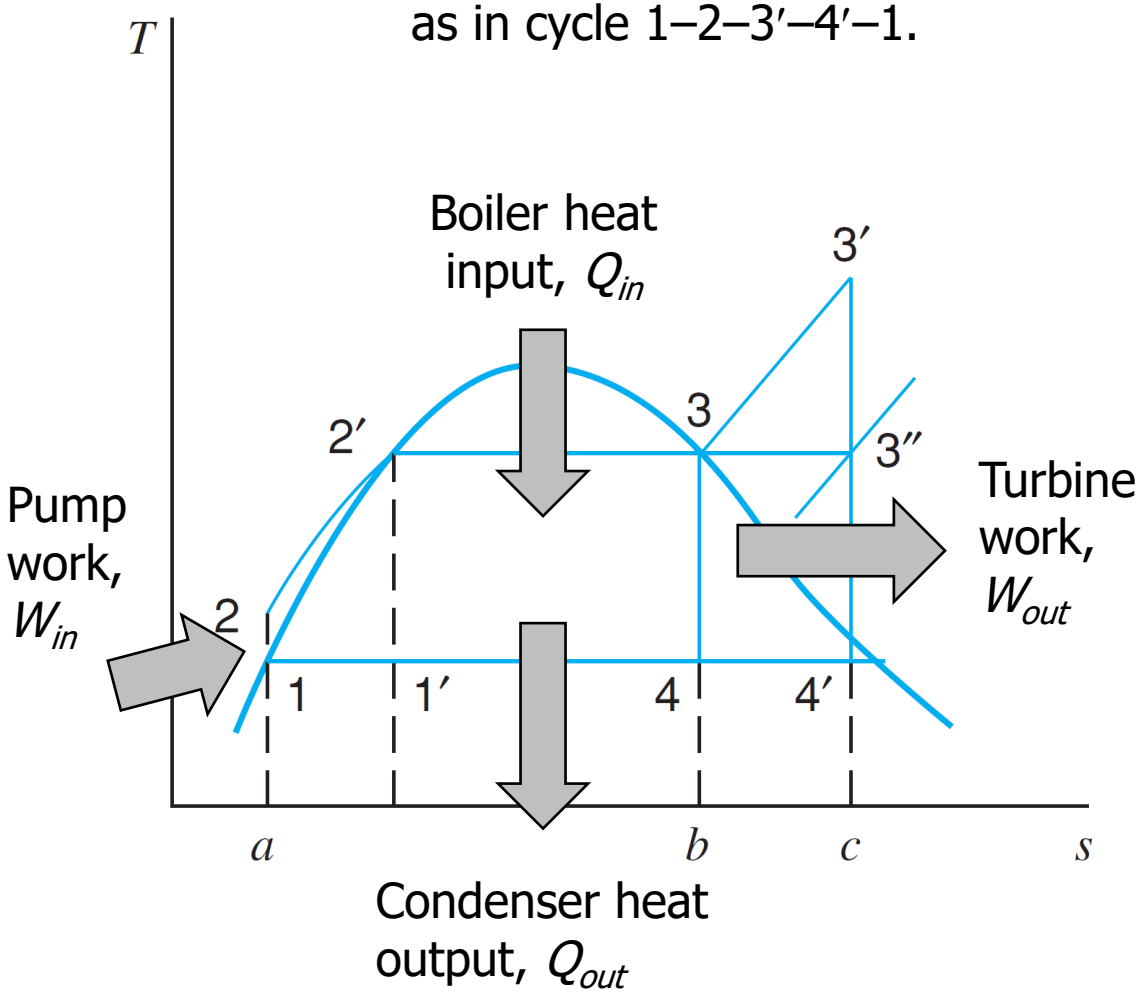
1–2: Reversible adiabatic pumping process in the pump

2–3: Constant-pressure heat transfer in the boiler

3–4: Reversible adiabatic expansion in the turbine

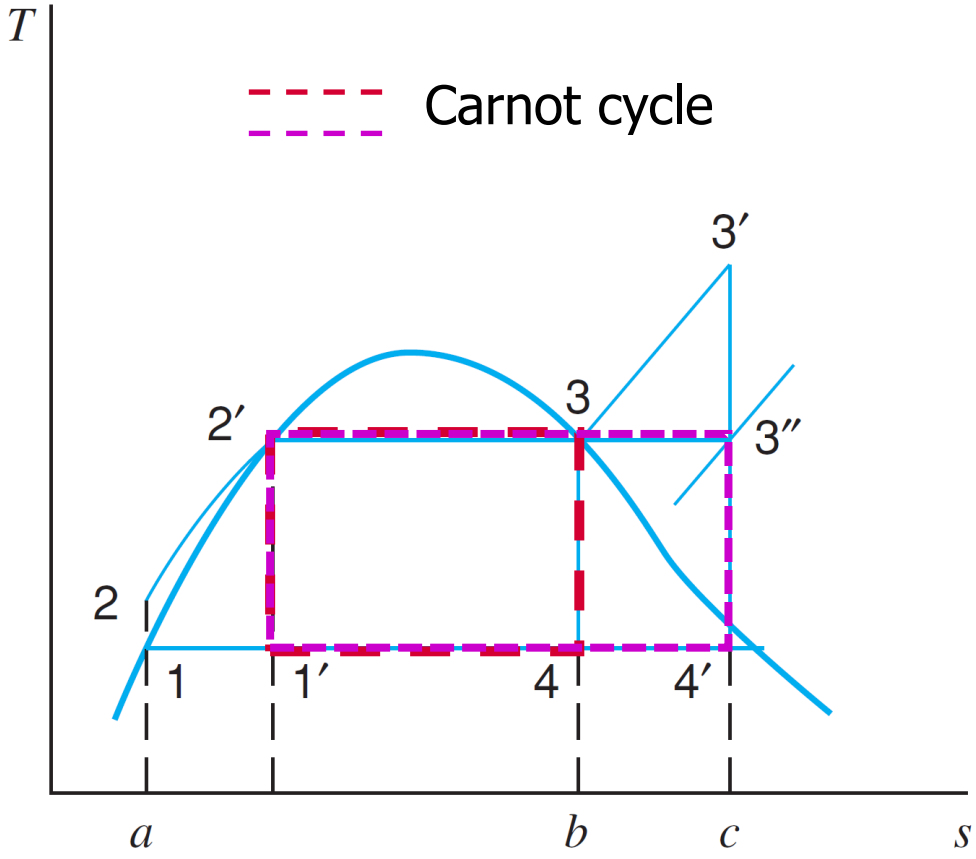
4–1: Constant-pressure transfer of heat in the condenser

The Rankine cycle can also include superheating the vapour, as in cycle 1–2–3′–4′–1.



Ideal Rankine Cycle

- It is readily evident that the Rankine cycle has lower efficiency than a Carnot cycle with the same maximum and minimum temperatures as the Rankine cycle because the average temperature between 2 and 2′ is less than the temperature during evaporation.
- We might well ask, why choose the Rankine cycle as the ideal cycle?
- Why not select the Carnot cycle 1′–2′–3–4–1′ or 1′–2′–3″–4′–1′?
  - At least two reasons can be given. The first reason concerns the pumping process. The second reason concerns steam quality for the turbine (it cannot handle $x < 0.9$).



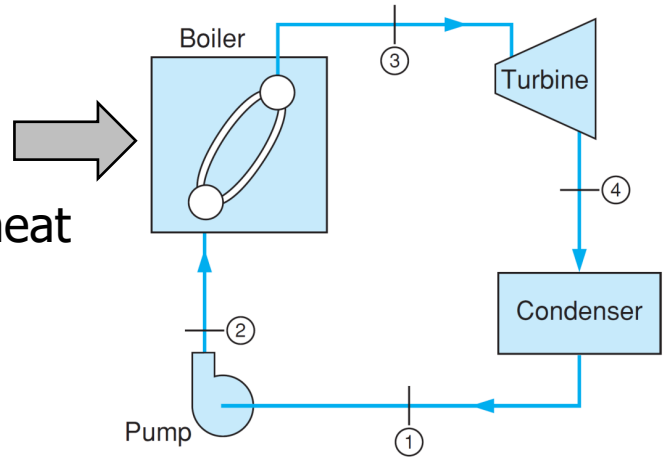
Carnot cycle cannot be approximated in actual devices and is not a realistic model for vapour power cycles.

Ideal Rankine Cycle Energy Analysis - Boiler

- At high pressures and temperatures, water can NO LONGER be considered an IDEAL GAS. Tabulated steam tables and diagrams need to be used.
- Applying the steady-flow conservation of mass and energy to the boiler, we can find the heat supplied to the boiler.
- If we neglect the potential and kinetic energies ($\Delta PE = 0$ & $\Delta KE = 0$) and note that no work is done on the steam in the boiler, then

1st Law: **Q – W = ΔH + ΔKE + ΔPE**

Boiler heat input, $\dot{Q}_{in}$



$$\dot{m}_2 = \dot{m}_3 = \dot{m}$$

$$\dot{m}_2 h_2 + \dot{Q}_{in} = \dot{m}_3 h_3 \quad \left(\because \dot{E}_{in} = \dot{E}_{out}\right)$$

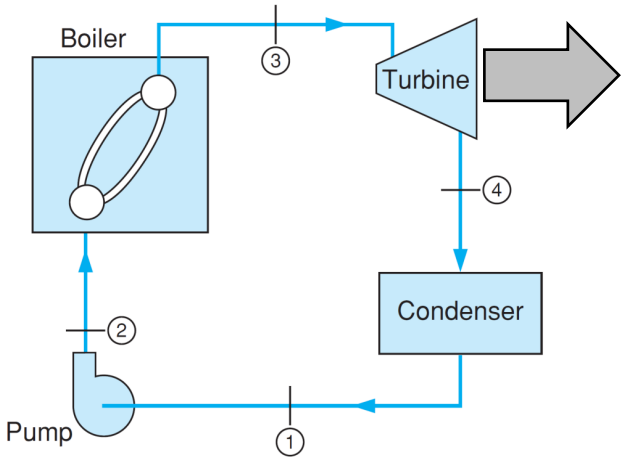
$$\dot{Q}_{in} = \dot{m}(h_3 - h_2)$$

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

Ideal Rankine Cycle Energy Analysis - Turbine

- Applying the steady-flow conservation of mass and energy to the turbine
  - if we assume that the expansion in the turbine is adiabatic and reversible, and
  - if we neglect any change in potential energy and kinetic energy
- The work output of the turbine is

1st Law: **Q – W = ΔH + ΔKE + ΔPE**



Turbine work, $\dot{W}_{\text{turbine}}$

$$\dot{m}_3 = \dot{m}_4 = \dot{m}$$

$$\dot{m}_3 h_3 = \dot{W}_{\text{turbine}} + \dot{m}_4 h_4 \qquad \left(\because \dot{E}_{\text{in}} = \dot{E}_{\text{out}}\right)$$

$$\dot{W}_{\text{turbine}} = \dot{m}(h_3 - h_4)$$

$$w_{\text{turbine}} = h_3 - h_4$$

Condensing Turbine

- In a Rankine cycle, steam exits the turbine as a two-phase mixture (a gas with water droplets in suspension)

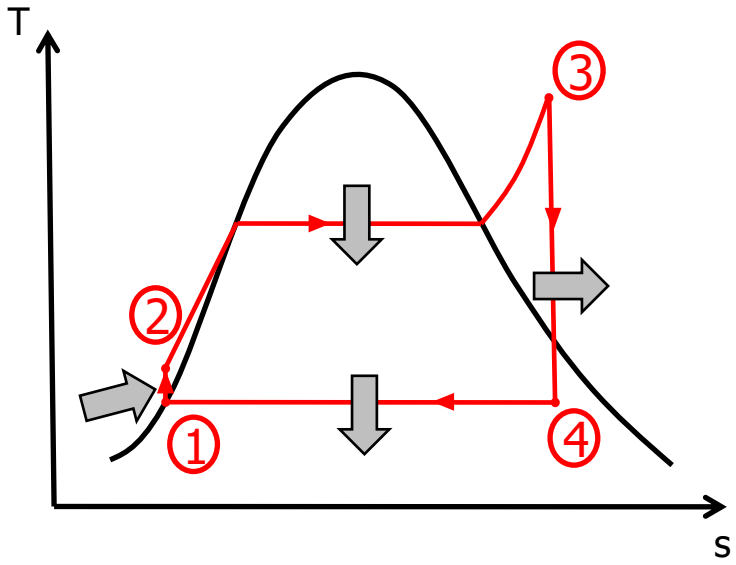
- To calculate the turbine outlet enthalpy, we need to introduce the dryness fraction, x

$$x = (s - s_{liq}) / (s_{vap} - s_{liq})$$

$$x = (h - h_{liq}) / (h_{vap} - h_{liq})$$

s = *entropy of the fluid*
s_{liq} = *entropy of saturated liquid at the same pressure as the fluid*
s_{vap} = *entropy of saturated vapour at the same pressure as the fluid*

h = *enthalpy of the fluid*
h_{liq} = *enthalpy of saturated liquid at the same pressure as the fluid*
h_{vap} = *enthalpy of saturated vapour at the same pressure as the fluid*



Condensing Turbine

- The dryness fraction at the turbine outlet is, for a reversible expansion,

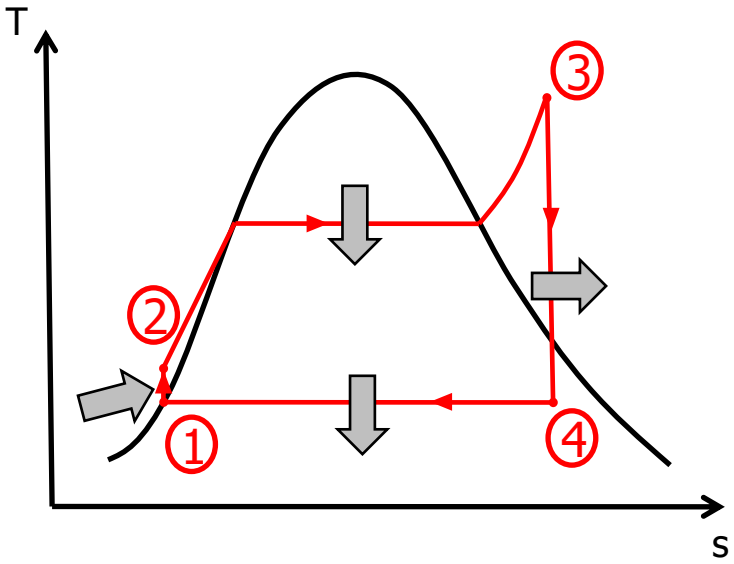
$$x_{4s} = (s_{4s} - s_{liq}) / (s_{vap} - s_{liq})$$

- However, $s_3 = s_{4s}$

$$x_{4s} = (s_3 - s_{liq}) / (s_{vap} - s_{liq})$$

- Once x_{4s} is found, we can calculate the outlet enthalpy for a reversible expansion

$$h_{4s} = x_{4s} \times h_{vap} + (1 - x_{4s}) \times h_{liq}$$



Ideal Rankine Cycle Energy Analysis - Pump

- The pump work is obtained from the conservation of mass and energy for steady-flow. Neglecting potential and kinetic energy changes and assuming the pump is adiabatic and reversible, the pump work can be written as:

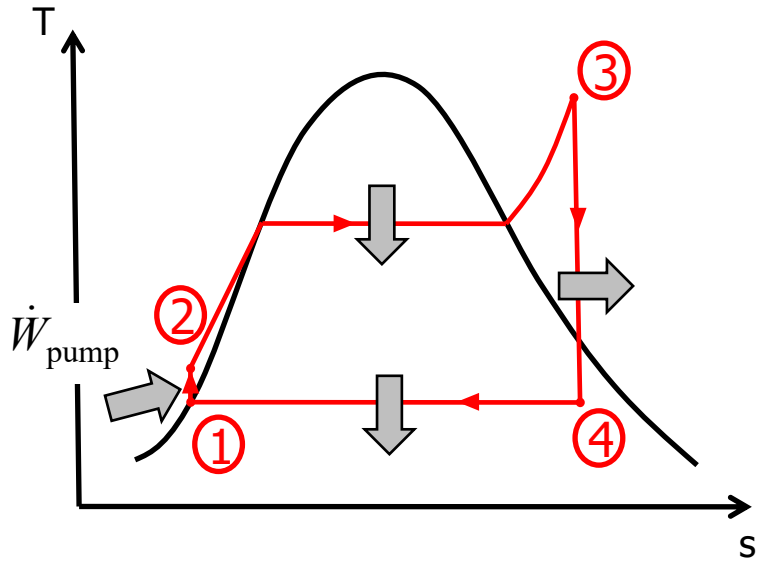
$$\dot{m}_1 = \dot{m}_2 = \dot{m}$$

$$\dot{m}_1 h_1 + \dot{W}_{\text{pump}} = \dot{m}_2 h_2 \quad \Rightarrow \quad \dot{W}_{\text{pump}} = \dot{m}(h_2 - h_1)$$

- Since the pumping process involves an incompressible liquid, state 2 is in the compressed liquid region, we use a second method to find the pump work or the Δh across the pump.
  - Recall the property relation for an adiabatic reversible process

$$dh = vdP$$

$$\Delta h = h_2 - h_1 = \int_1^2 vdP$$



Ideal Rankine Cycle Energy Analysis - Pump

- Assuming that the liquid is incompressible

$$v \cong v_1 = const.$$

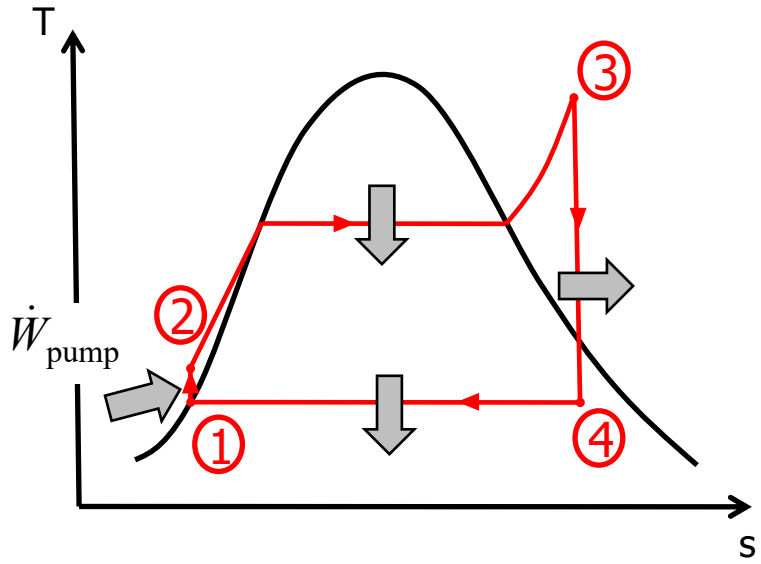
$$h_2 - h_1 \cong v_1(P_2 - P_1)$$

$$\Delta h = h_2 - h_1 = \int_1^2 v dP$$

- The pump work is calculated from

$$\dot{W}_{\text{pump}} = \dot{m}(h_2 - h_1) @ \dot{m} v_1 (P_2 - P_1)$$

$$w_{\text{pump}} = \frac{\dot{W}_{\text{pump}}}{\dot{m}} = v_1(P_2 - P_1)$$

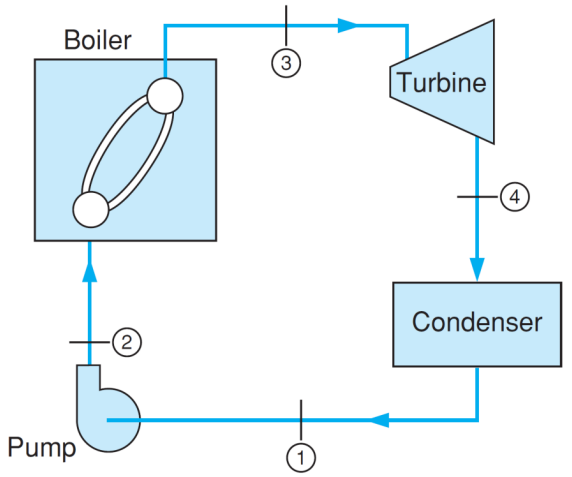


- The enthalpy at the pump outlet is thus:

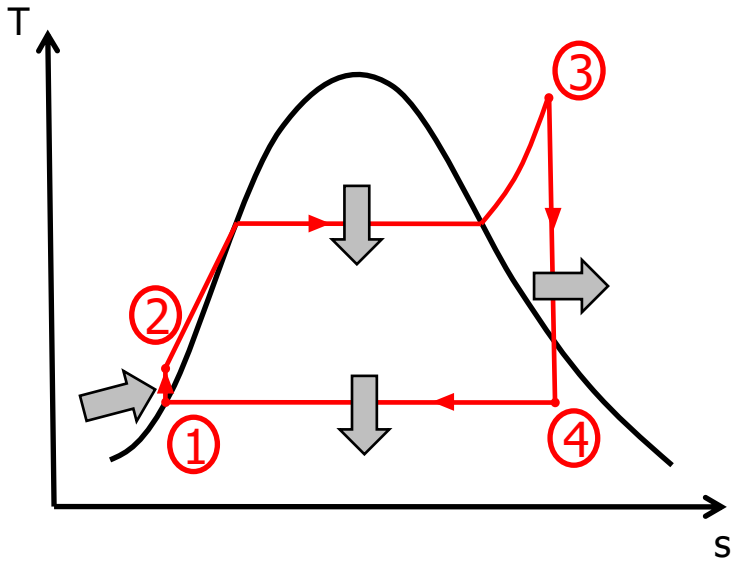
$$h_2 = w_{\text{pump}} + h_1$$

Ideal Rankine Cycle Efficiency

- Pump ($Q = 0$) $w_{\text{pump}} = h_2 - h_1 = v_1 (P_2 - P_1)$
- Boiler ($W = 0$) $q_{\text{boiler}} = h_3 - h_2$
- Turbine ($Q = 0$) $w_{\text{turbine}} = h_3 - h_4$
- Condenser ($W = 0$) $q_{\text{condenser}} = h_4 - h_1$



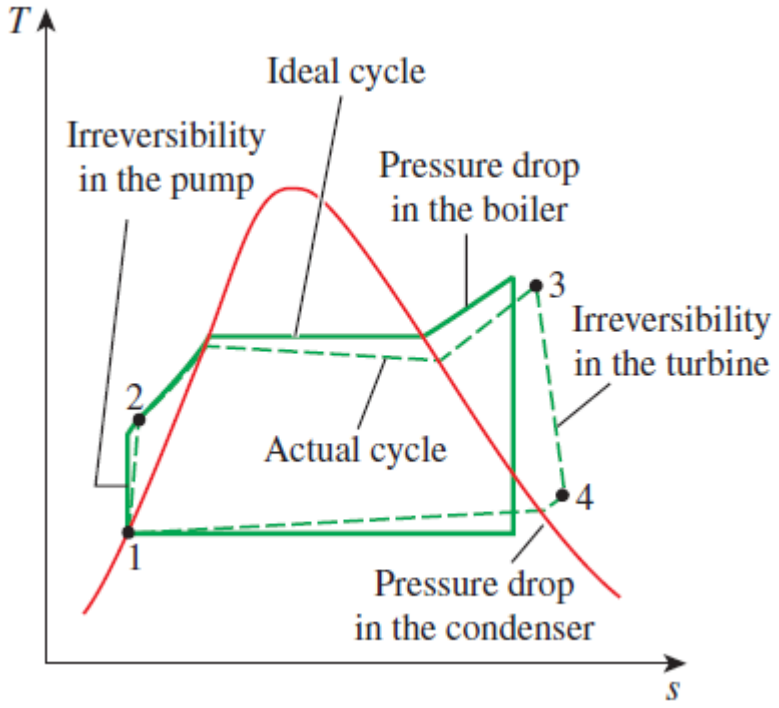
- The thermal efficiency of the Rankine cycle is the ratio between the net work output and the heat input into the cycle
  - $\eta = w_{\text{net}} / q_{\text{in}}$
  - $\eta = (w_{\text{turbine}} - w_{\text{pump}}) / q_{\text{boiler}}$
  - $\eta = (h_3 - h_4 - h_2 + h_1)/(h_3 - h_2)$



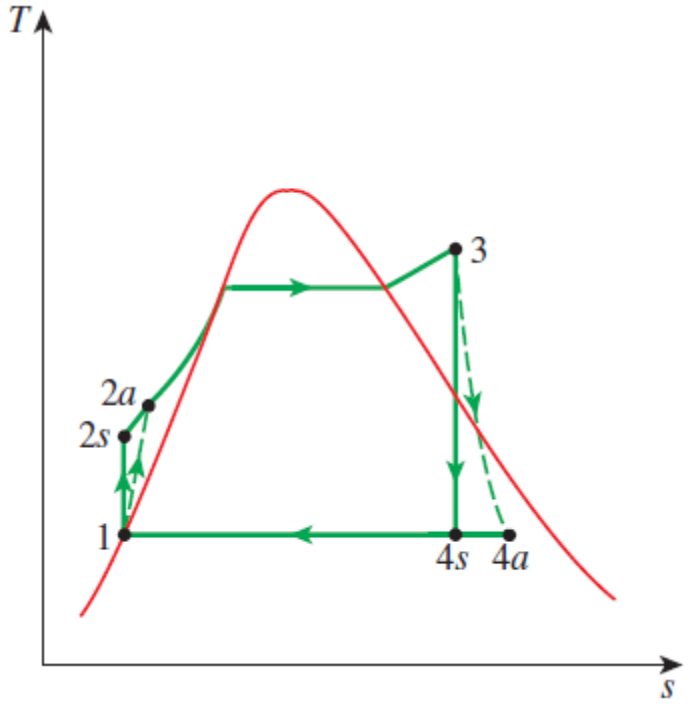
Deviation from the Ideal Rankine Cycle

- The actual Rankine 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.
- Fluid friction causes pressure drops in the boiler, the pipes etc.
- To compensate for pressure drops the pump must increase the water pressure to a sufficiently higher pressure than the ideal cycle calls for.
- This requires a larger work input to the pump.
- Irreversibilities due to fluid friction, fluid recirculation and vortexes also occur within the pump and the turbine.

Deviation from the Ideal Rankine Cycle



Deviation of actual vapour power cycle (dashed-line) from the ideal Rankine cycle (solid-line).



The effect of pump and turbine irreversibilities on the ideal Rankine cycle.

Isentropic efficiencies

$$\eta_P = \frac{w_s}{w_a} = \frac{h_{2s} - h_1}{h_{2a} - h_1}$$

$$\eta_T = \frac{w_a}{w_s} = \frac{h_3 - h_{4a}}{h_3 - h_{4s}}$$

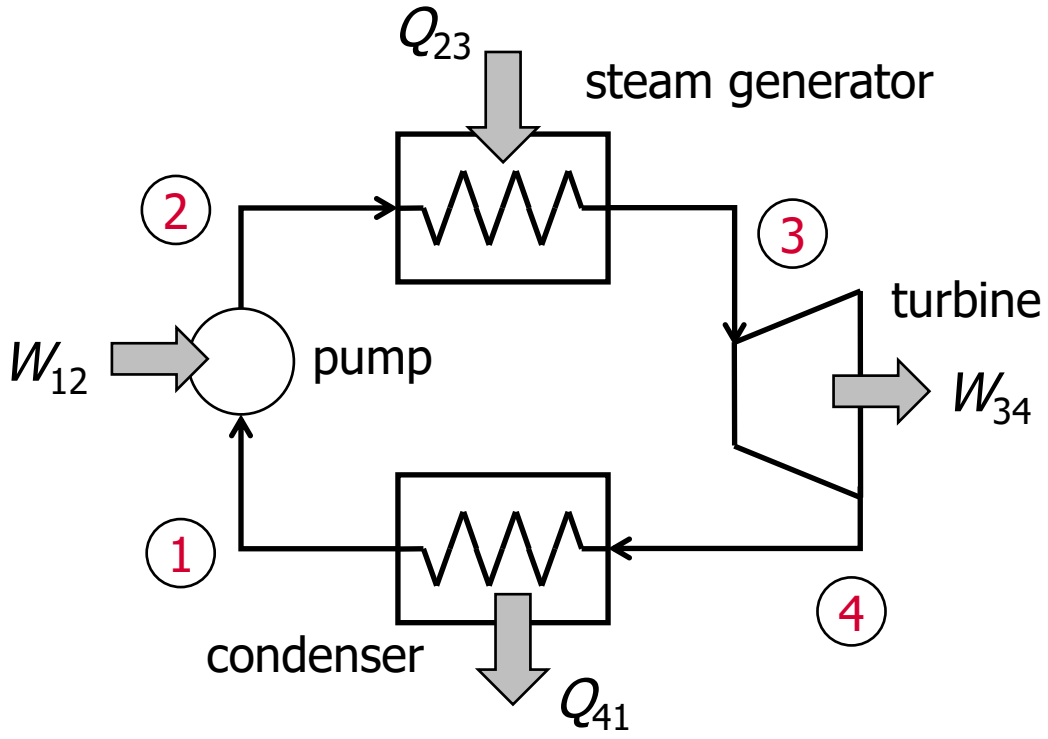
The pump outlet enthalpy is then calculated as follows:

$$h_{2a} = h_1 + v_1 (P_2 - P_1)/\eta_P$$

Example 4.1

Steam is generated at 40 bar and 600°C in a sub-critical boiler and is expanded into a turbine down to 0.1 bar. For a turbine isentropic efficiency of 90%. Calculate the turbine work output.

- Calculate the efficiency of the corresponding Rankine cycle, assuming a pressure drop in the boiler of 20 bar and a pump efficiency of 80%.
- Compare the steam cycle efficiency to the corresponding Carnot cycle efficiency.



Next

- How Can We Increase the Efficiency of the Rankine Cycle?
  - The basic idea behind all the modifications to increase the thermal efficiency of a vapour power cycle is the same:
    - Increase the average temperature at which heat is transferred to the working fluid in the boiler.
    - Decrease the average temperature at which heat is rejected from the working fluid in the condenser.
- Example problem
- Cogeneration and combined power cycles

Increasing Rankine Cycle Efficiency

- Three obvious ways to improve Rankine cycle efficiency following first principles:

  1. Increase the maximum temperature
  2. Decrease the cycle minimum temperature
  3. Increase the average temperature of heat addition

1. Increasing maximum temperature

Increase the maximum temperature by increasing the amount of superheating at the boiler outlet. This increases the efficiency of the corresponding Carnot Cycle.

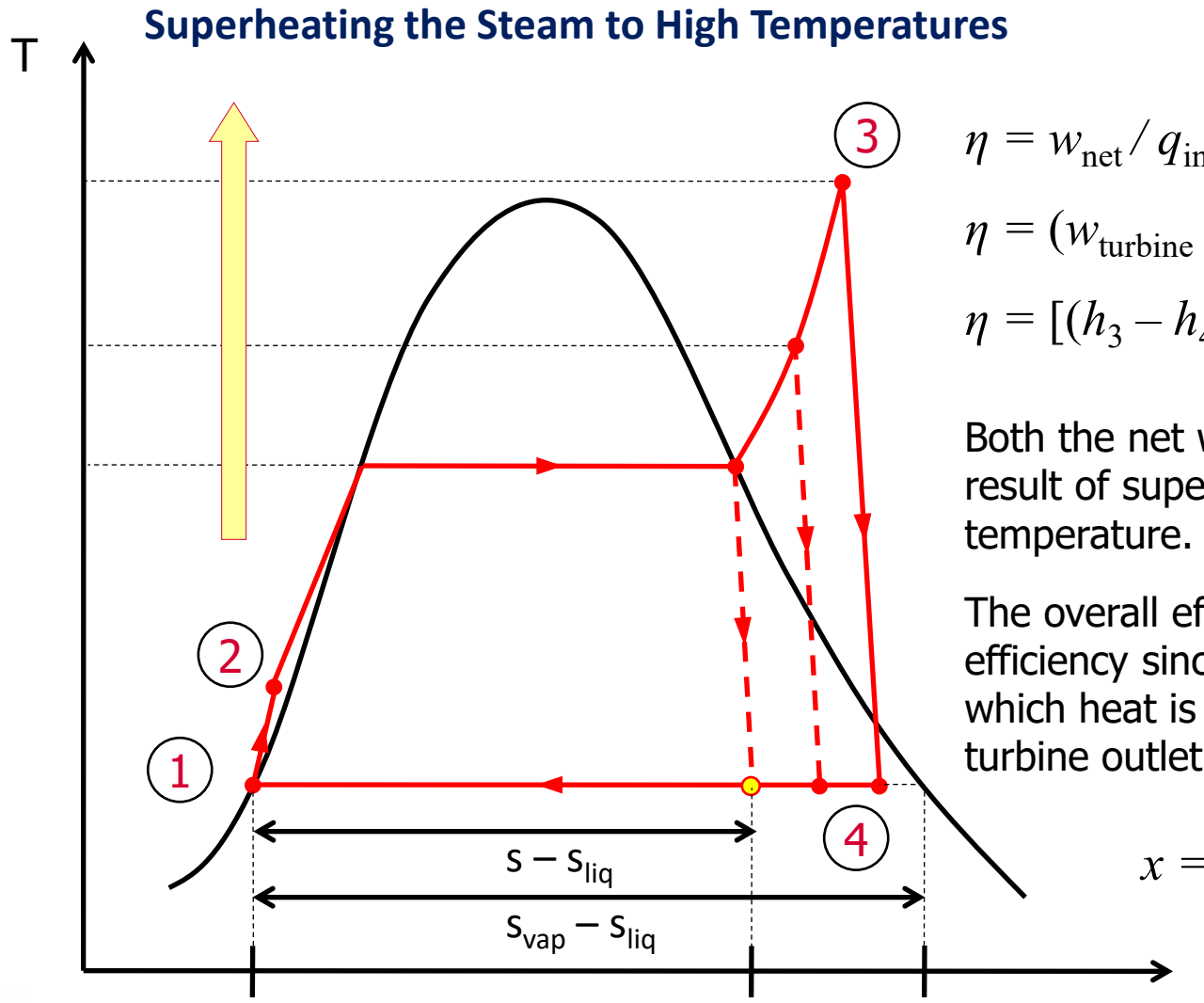
$$\eta_C = 1 - T_{\text{cold}}/T_{\text{hot}}$$

In practice, the maximum temperature achievable is limited by metallurgical considerations. Around 600°C is the normal maximum achievable for state-of-the-art alloys used by boiler manufacturers.

There is another benefit associated with increased temperature:

- An increase in the turbine outlet dryness fraction, as shown on the *T-s* diagram (next slide); this helps to limit the amount of droplets in suspension and reduce possible erosion damage to the turbine blades caused by impact with water droplets.

1. Increasing maximum temperature



$$\eta = w_{\text{net}} / q_{\text{in}}$$

$$\eta = (w_{\text{turbine}} - w_{\text{pump}}) / q_{\text{boiler}}$$

$$\eta = [(h_3 - h_4) - (h_2 - h_1)]/(h_3 - h_2)$$

Both the net work and heat input increase as a result of superheating the steam to a higher temperature.

The overall effect is an increase in thermal efficiency since the average temperature at which heat is added increases; also increases turbine outlet dryness fraction.

$$x = (s - s_{liq}) / (s_{vap} - s_{liq})$$

2. Decrease the cycle minimum temperature

Decrease the cycle minimum temperature by reducing the condenser pressure. Again, this increases the efficiency of the corresponding Carnot Cycle.

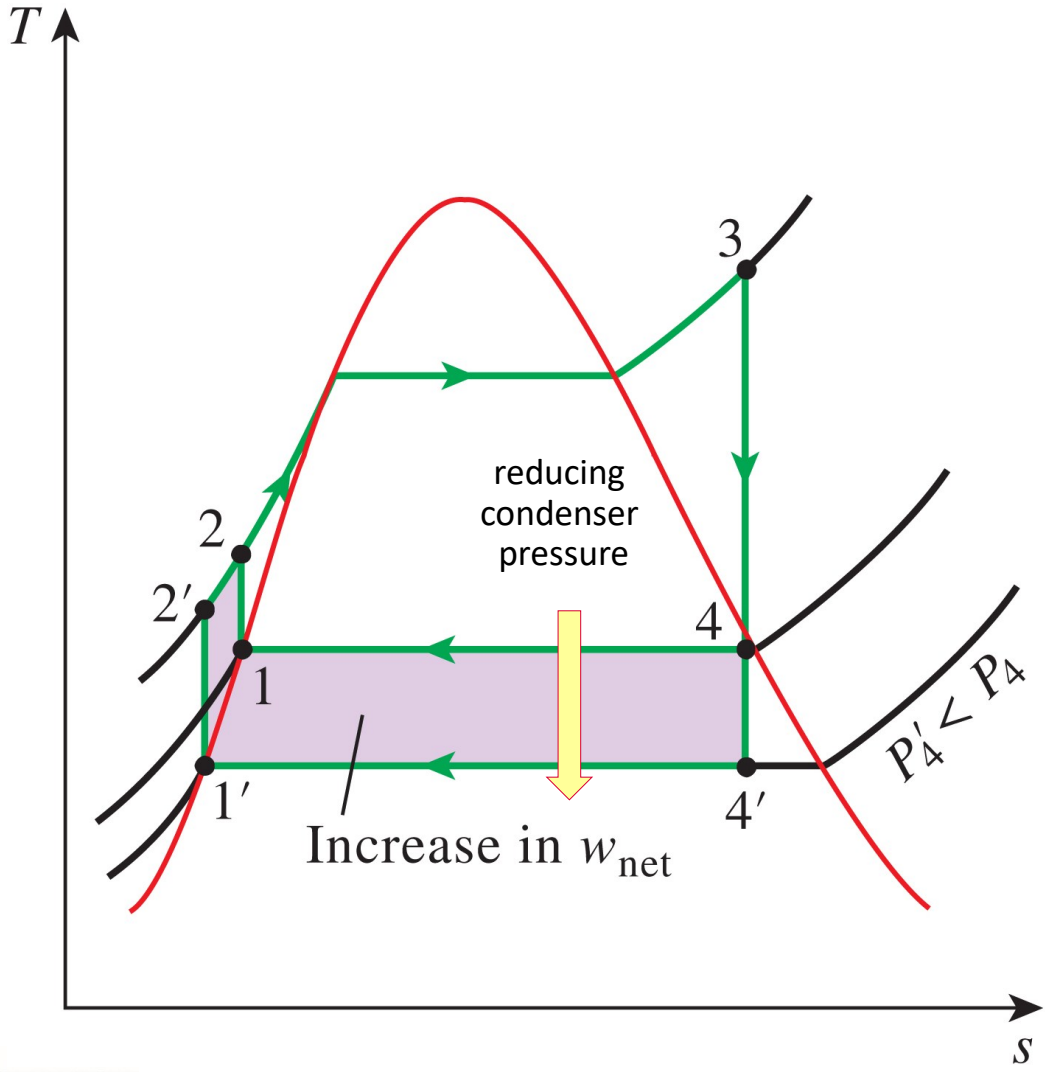
$$\eta_C = 1 - T_{\text{cold}}/T_{\text{hot}}$$

This is limited by the available temperature for cooling water. For example, sea-water cooling is better than river water cooling and air-cooling.

The turbine exit dryness fraction is reduced, so more risk of droplet damage.

The other limiting factor is the size of the condenser.

Reducing condenser pressure

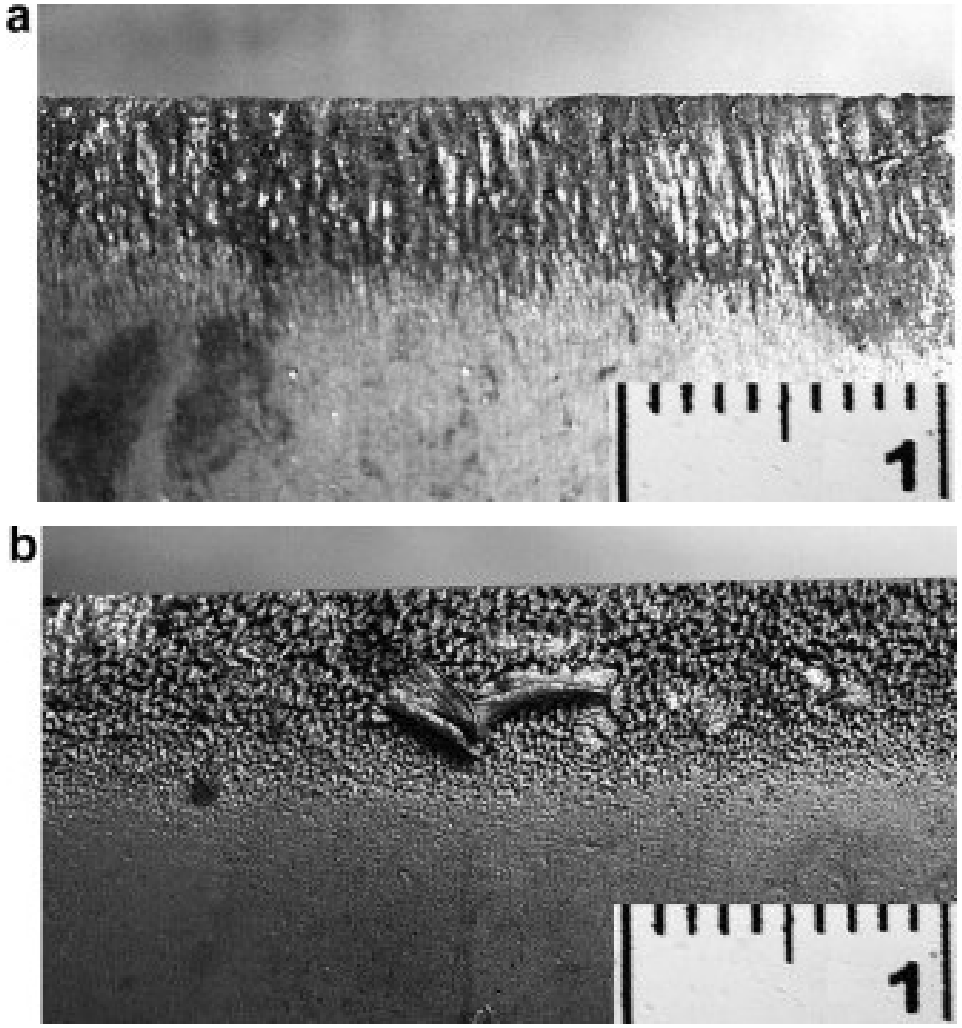


Side effect: Lowering the condenser pressure increases the moisture content of the steam at the final stages of the turbine.

$$x = (s - s_{liq}) / (s_{vap} - s_{liq})$$

$$x_{4'} < x_4$$

Example of blade erosion



Application of Repair Technique After Tip Removal



Erosion on the blade of a condensing turbine caused by water droplets

Mitchell and Martin (2007) LP Higg Twist LSB Refurbishment, Steam Turbine Users Group, IMechE

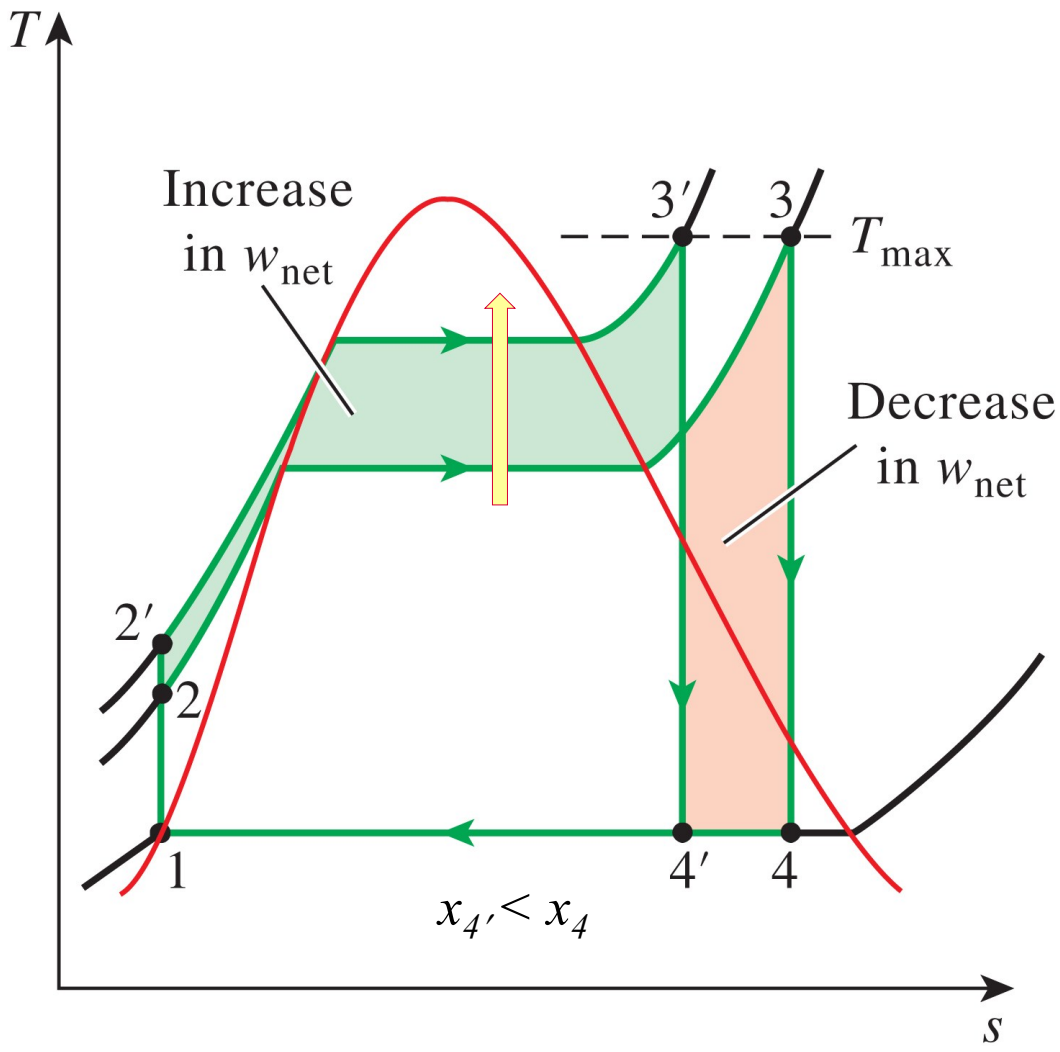
Azevedo et al. (2009) Erosion-fatigue of steam turbine blades, *Engineering Failure Analysis*, 16, 2290-2303. https://doi.org/10.1016/j.engfailanal.2009.03.007.

3. Increasing average temperature of heat addition

Increase the average temperature of heat addition by increasing the boiler pressure and hence the pump work.

It has an adverse effect on turbine exit dryness fraction. For a fixed turbine inlet temperature, the cycle shifts to the left and the moisture content of steam at the turbine exit increases.

This side effect can be corrected by reheating the steam.



Increasing Rankine Cycle Efficiency

- Three obvious ways to improve Rankine cycle efficiency following first principles:

  1. Increase the maximum temperature
  2. Decrease the cycle minimum temperature
  3. Increase the average temperature of heat addition

- Other methods to increase the average temperature of heat addition are:
  - reheat cycle
  - regenerative feed water heating

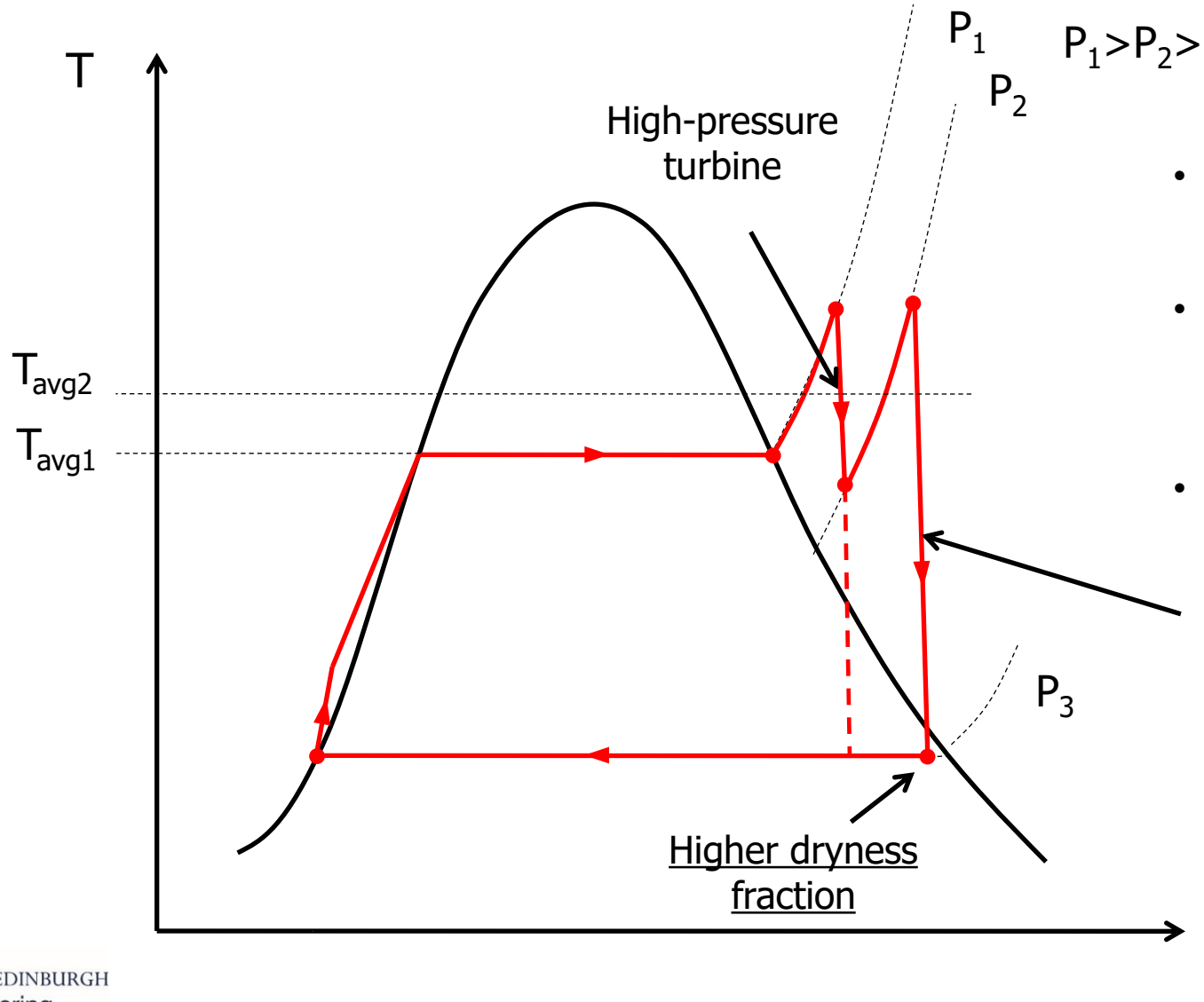
Reheat Cycle

- The reheat cycle has two purposes:
  - increase exit dryness fraction, i.e. reduce the moisture content, at the final stages of the turbine expansion process to allow for higher boiler pressure and lower condenser pressure, and
  - increase the average temperature of heat addition by adding more heat to superheated steam.
- Steam is now expanded twice in two separate turbines. They are synchronised to the same shaft.
- The single reheat in a modern power plant improves the cycle efficiency by 4 to 5%.

Reheat Cycle

- Metallurgical considerations, notably corrosion problems at high temperature and pressure, are a limiting factor for high temperatures and pressure in steam boilers. State-of-the-art plants use steam conditions around 290 bar and 600°C.
- Metallurgical considerations also limit turbine exit dryness fraction. Blade erosion reduce the operating life of the last stages of condensing turbines. Exit dryness fraction are typically kept above 88-90% moisture.
- To benefit from higher boiler conditions and limit turbine blade erosion steam is expanded in two stages with an intermediate reheat stage. This is a practical solution to excessive moisture, and is frequently used in modern steam power plants.
- The use of more than two reheat stages is not practical. The theoretical improvement in efficiency from the second reheat is about half of that which results from a single reheat. So, double reheat system are less common since the resulting savings are typically more than offset by the increased capital cost, but can be justified with very low condenser pressure, i.e. if very cold water is available.

Single Reheat Cycle



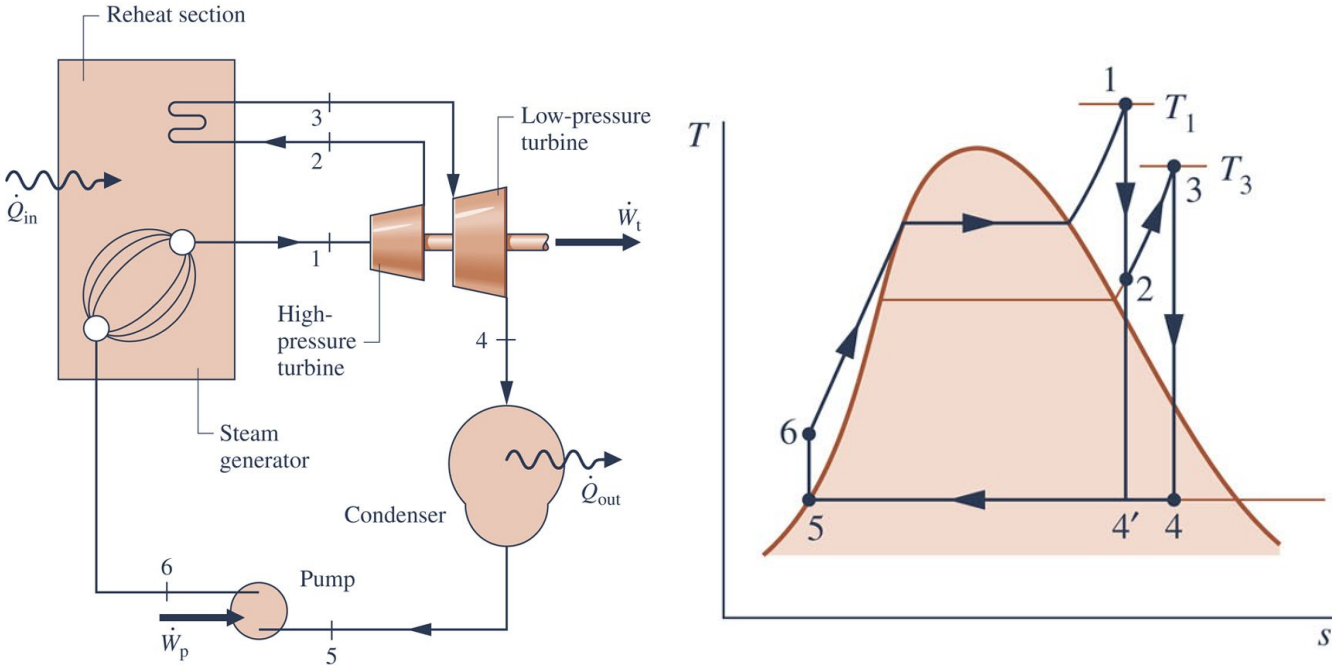
- Average temperature of heat addition into reheater: T_{avg2}
- Average temperature of heat addition into main part of the boiler: T_{avg1}
- $T_{avg2} > T_{avg1}$

Single Reheat Cycle

- When computing the thermal efficiency of a reheat cycle, it is necessary to account for the
  - Total work developed in the two turbine stages: processes **1-2** and **3-4**.
  - Total heat added during vaporization/superheating and reheating: processes **6-1** and **2-3**.

$$\eta = \frac{w_{\text{t, HP}} + w_{\text{t, LP}} - w_{\text{p}}}{q_{6\text{-}1} + q_{2\text{-}3}}$$

Ideal Reheat Rankine Cycle

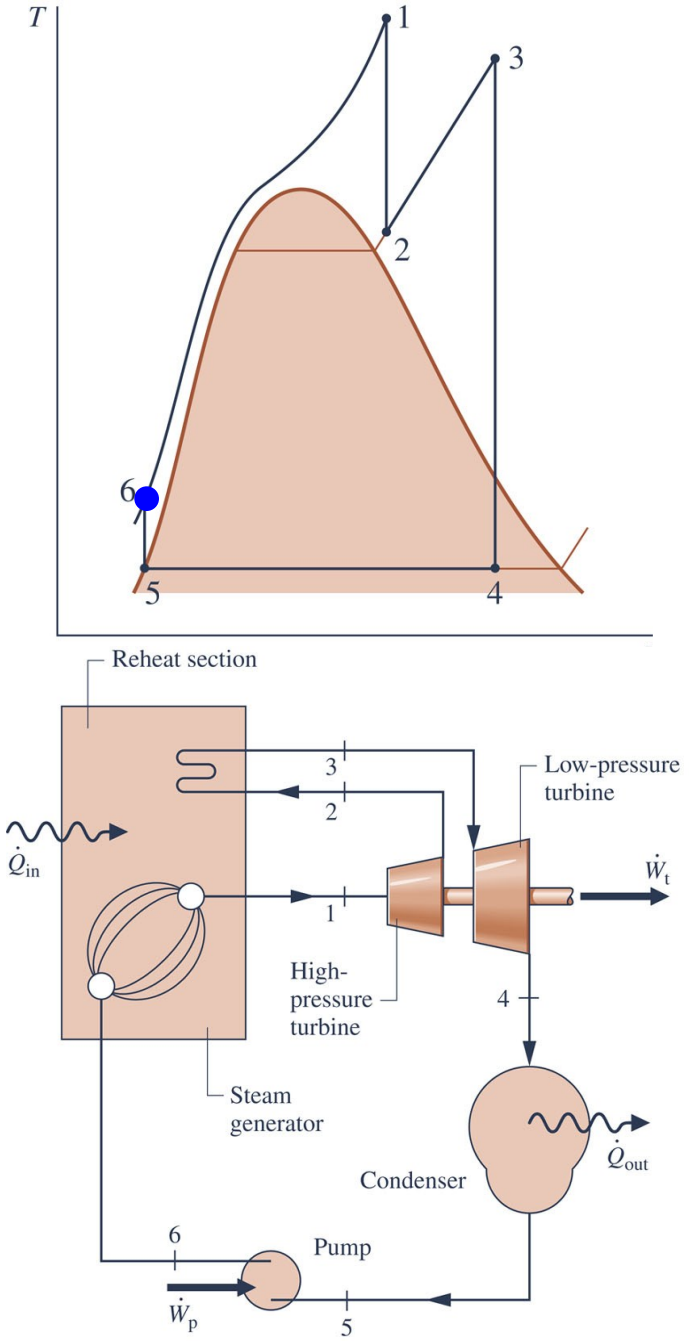


$$q_{\text{in}} = q_{\text{primary}} + q_{\text{reheat}} = q_{6\text{-}1} + q_{2\text{-}3} = (h_1 - h_6) + (h_3 - h_2)$$

$$w_{\text{t}} = w_{\text{t, HP}} + w_{\text{t, LP}} = (h_1 - h_2) + (h_3 - h_4)$$

Supercritical Reheat Cycle

- Steady improvements over many decades in materials and fabrication methods have permitted significant increases in maximum allowed temperatures and steam generator pressures.
- These efforts are embodied in the ***supercritical*** reheat cycle.
- Steam generation occurs at a pressure greater than the critical pressure of water: Process **6-1**.
- No phase change occurs during process **6-1**. Instead, water flowing through tubes is heated from liquid to vapor without the bubbling associated with boiling.
- Today's supercritical plants achieve thermal efficiencies up to 47%. ***Ultra-supercritical*** plants capable of operating at still higher temperatures and steam generator pressures will have thermal efficiencies exceeding 50%.



Example 4.2

On a *T*-*s* diagram, draw the steam cycle for a single reheat cycle with the following operating conditions:

- High-pressure (HP) turbine: 300 bar and 600°C at the inlet
- Low-pressure (LP) turbine: 60 bar and 600°C at the inlet, 0.05 bar at the outlet
- Turbine isentropic efficiencies: 90% and pump isentropic efficiency: 80%
- Reheater pressure drop: 10 bar; boiler pressure drop: 20 bar

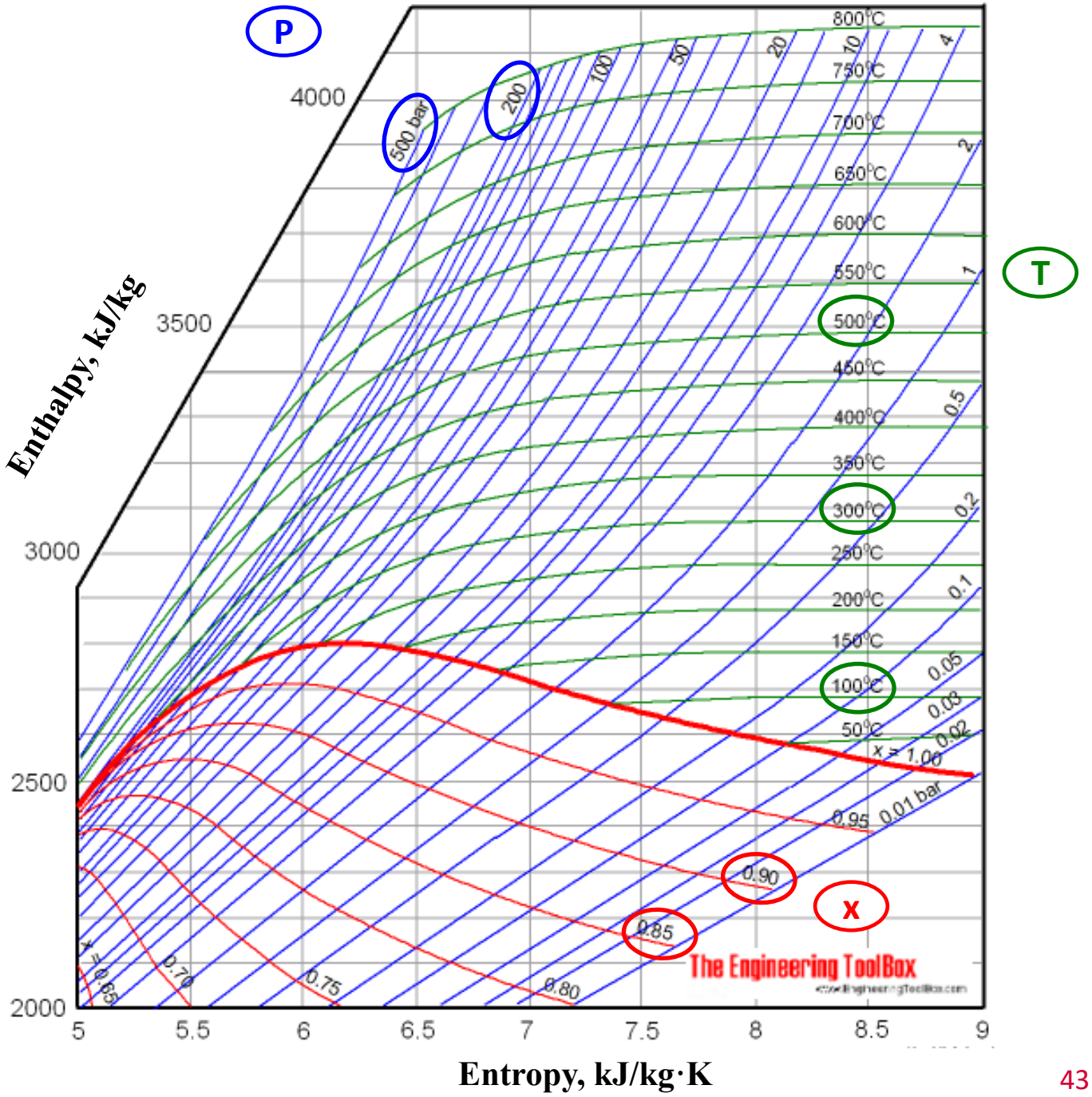
(i) Calculate the work of the LP turbine and the HP turbine. Verify your answer graphically on the *h*-*s* diagram (Mollier diagram) at the end of the steam tables.

(ii) Calculate the heat input into the steam cycle and the pump work.

(iii) Calculate the Rankine efficiency and compare it to the corresponding Carnot efficiency.

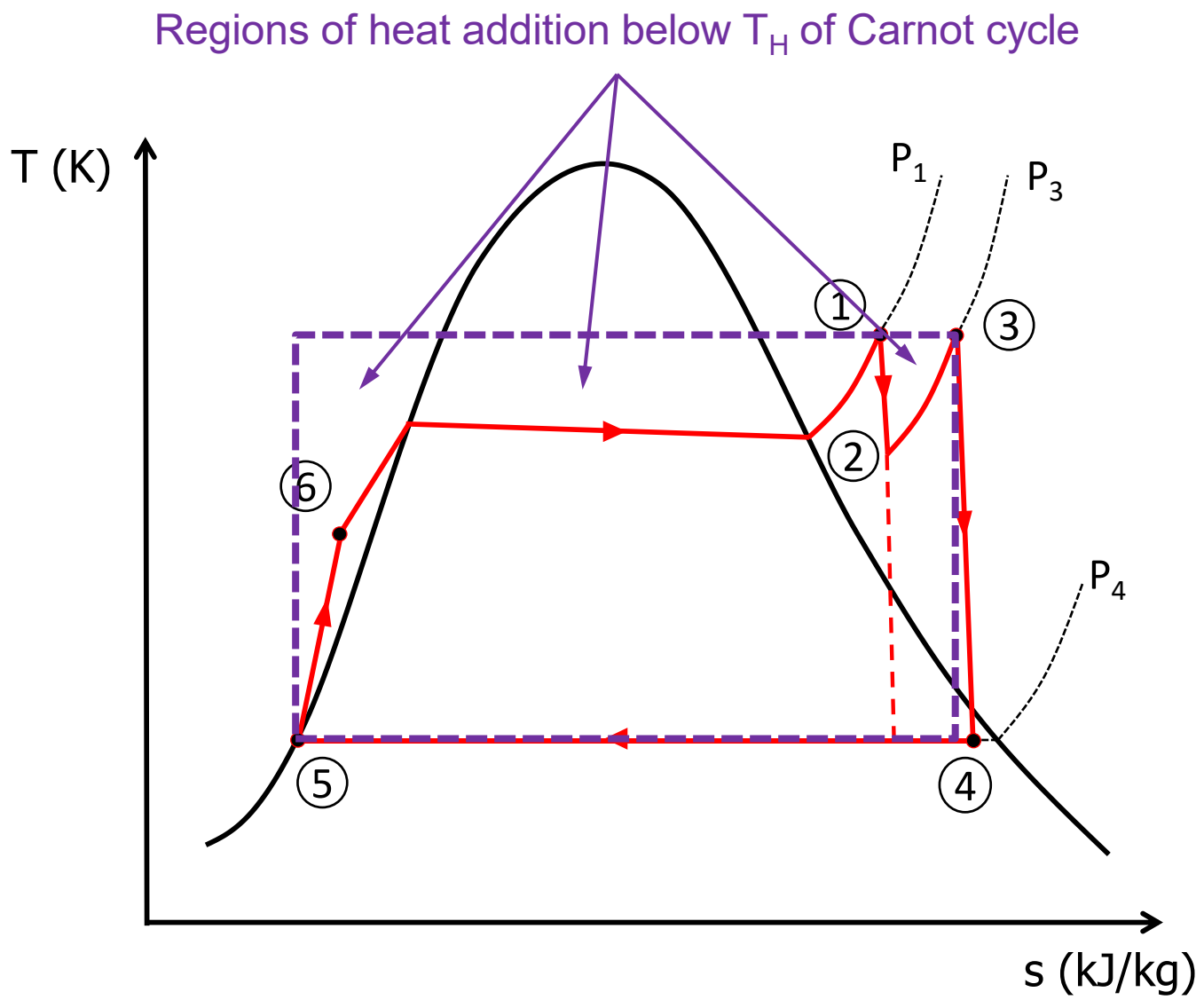
Mollier Diagram

- The diagram was created in 1904, when Richard Mollier (German Professor of Applied Physics and Mechanics) plotted the total heat (*H)* against entropy (*S*).
- At the 1923 Thermodynamics Conference held in Los Angeles, it was decided to name, in his honour, as a "Mollier diagram" any thermodynamic diagram using enthalpy as one of its axes.
- The work done in a process on vapour cycles is represented by the length of *h*, so it can be measured directly, whereas in a *T-s* diagram it has to be computed using thermodynamic relationships between thermodynamic properties.



Why the thermal efficiency of Example 4.2 is lower than the Carnot efficiency

- Irreversible heat transfer between the hot source of the Carnot Cycle and the working fluid
- Irreversible processes in turbines and pump



Regenerative Cycle and Feedwater Heaters

- Another method for increasing the thermal efficiency of vapour power plants is by the application of regenerative feedwater heaters, or simply *regeneration*.
- Regenerative feedwater heating increases the average temperature of heat addition, thereby increasing the thermal efficiency.
- In vapour power plants, both *open* and *closed* feedwater heaters are typically used.

Open Feedwater Heater

An open (or direct-contact) feedwater heater is basically a *mixing chamber,* where the steam extracted from the turbine mixes with the feedwater exiting the pump.

Ideally, the mixture leaves the feedwater heater (FWH) as a saturated liquid at the heater pressure.

- Referring to the ***T-s*** diagram, heat transfer to the cycle takes place from state **4** to state **5**, rather than from state **a** to state **5** as would be the case if there were no open feedwater heater.
- Accordingly, when an open feedwater heater is included, the average temperature of heat addition to the cycle increases; thus, thermal efficiency increases.

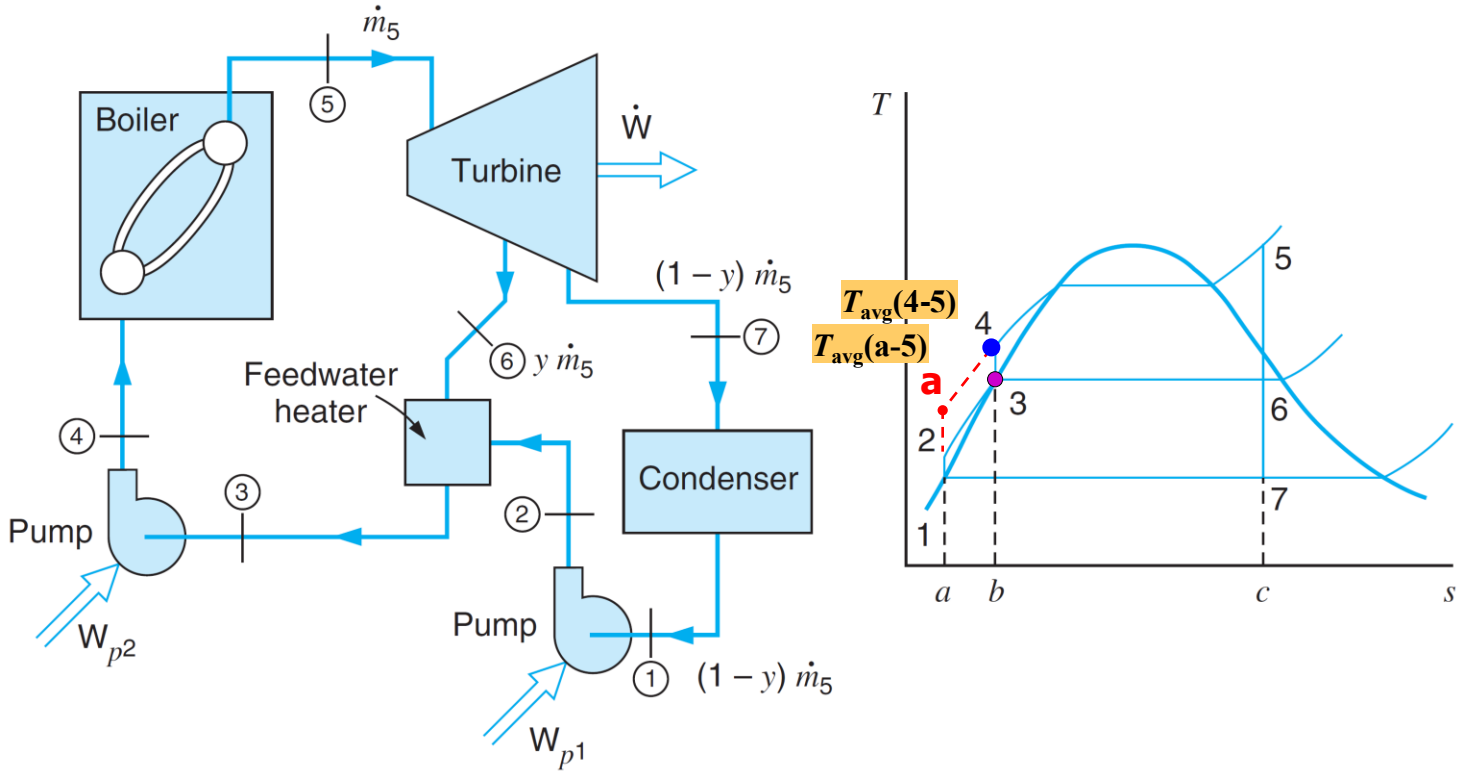


Image source: Figure 9.12: Regenerative cycle with an open FWH [Borgnakke and Sonntag (8th ed)]

Open Feedwater Heater

Referring to the schematic, let us follow a unit of mass as it moves through the cycle. The unit of mass is denoted in parentheses by **(1)**.

- The unit of mass **(1)** enters the turbine at state **5** and expands to state **6** where a fraction **(*y*)** is diverted into the open feedwater heater.
- The remaining fraction **(1-*y*)** expands through the second turbine stage to state **7**, is condensed to state **1**, and then pumped to state **2** where it enters the open feedwater heater.

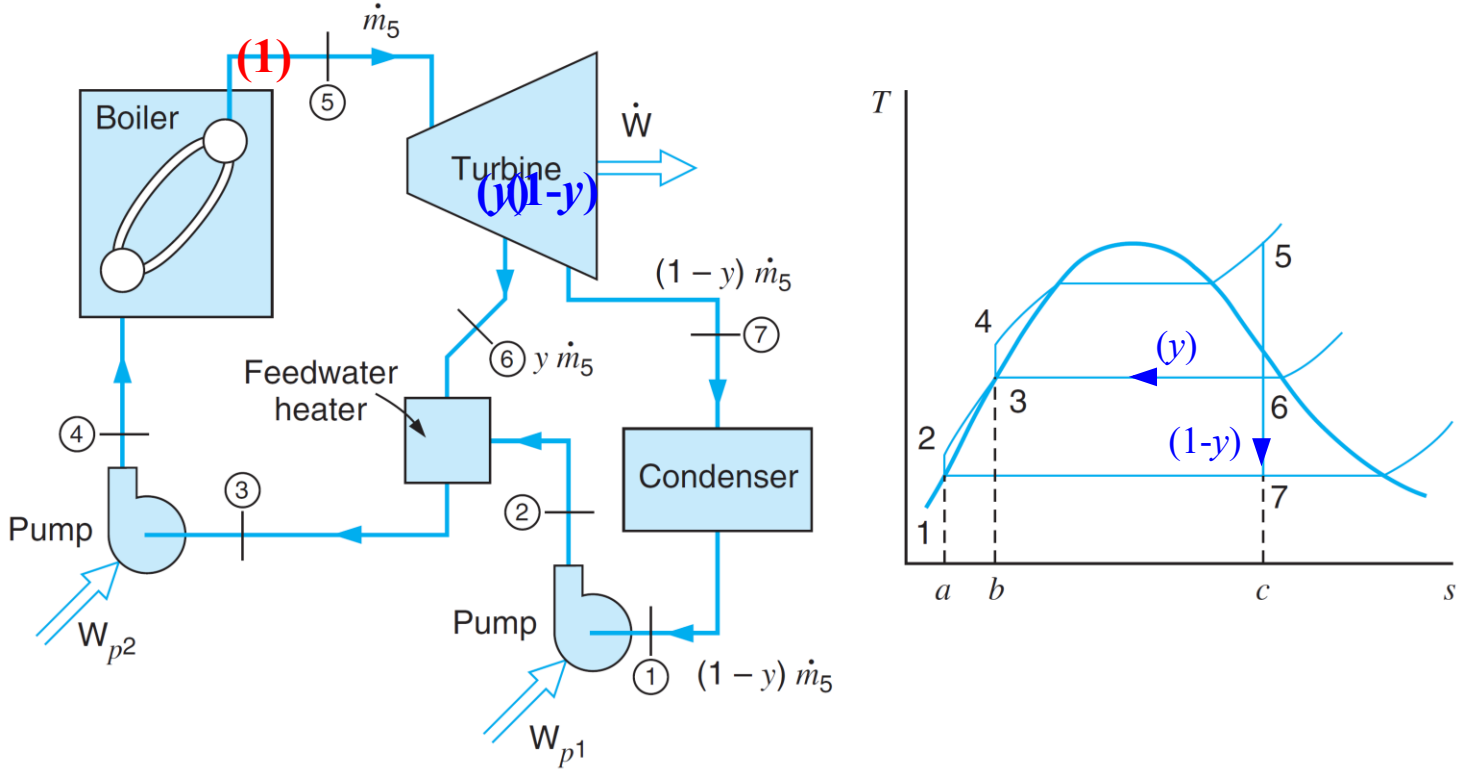


Image source: Figure 9.12: Regenerative cycle with an open FWH [Borgnakke and Sonntag (8th ed)]

Open Feedwater Heater

Referring to the schematic, let us follow a unit of mass as it moves through the cycle. The unit of mass is denoted in parentheses by **(1)**.

- The fractions **(*y*)** and **(1-*y*)** entering the feedwater heater at states **6** and **2**, respectively, mix within that component, giving a single stream at state **3** and recovering the unit of mass **(1)**.
- The unit of mass **(1)** at state **3** is pumped to state **4** where it enters the steam generator (boiler) and returns to state **5** after heating and then enter to turbine.

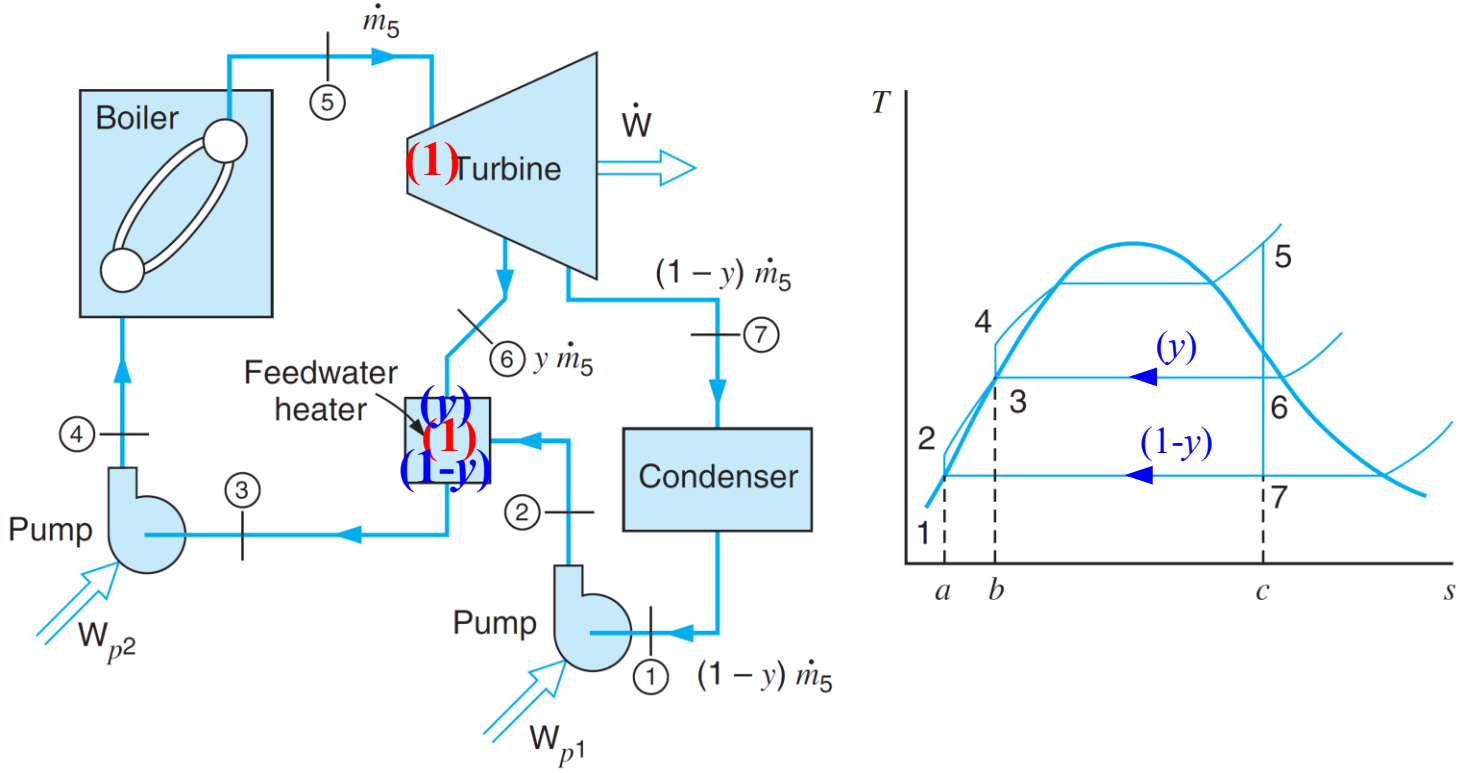


Image source: Figure 9.12: Regenerative cycle with an open FWH [Borgnakke and Sonntag (8th ed)]

Open Feedwater Heater

Applying mass and energy balance, it is possible to find out the amount of steam required for feedwater heating

$m_{water_in} + m_{steam} = m_{water_out}$ and $m_{water_in} \times h_{water_in} + m_{steam} \times h_{steam} = m_{water_out} \times h_{water_out}$

$$q_{\text{in}} = h_5 - h_4$$

$$q_{\text{out}} = (1-y)(h_7 - h_1)$$

$$w_{\text{turb, out}} = (h_5 - h_6) + (1-y)(h_6 - h_7)$$

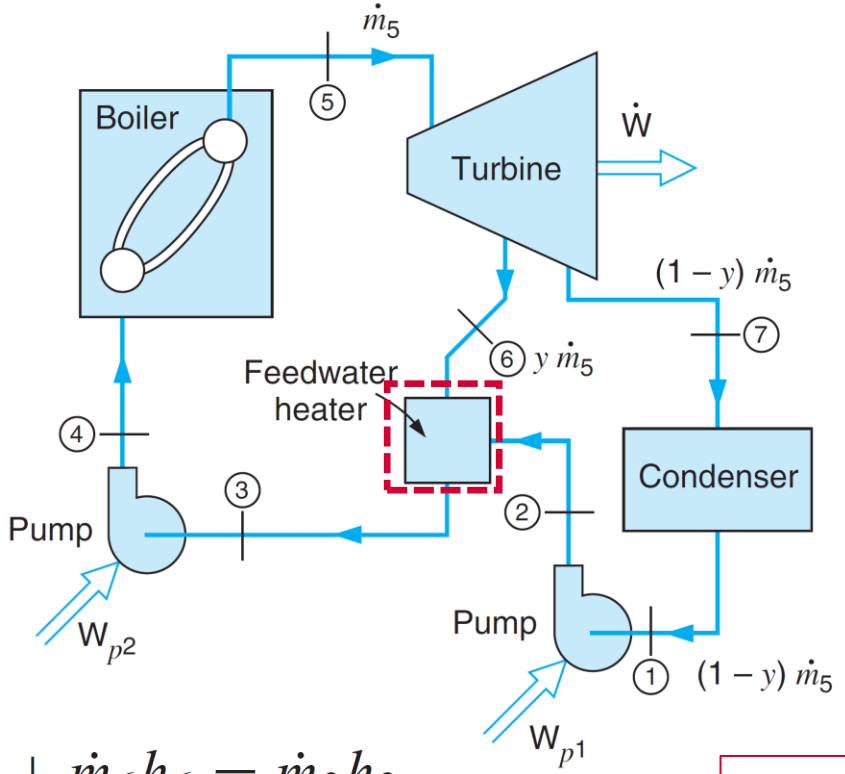
$$w_{\text{pump, in}} = (1-y)w_{\text{pump I, in}} + w_{\text{pump II, in}}$$

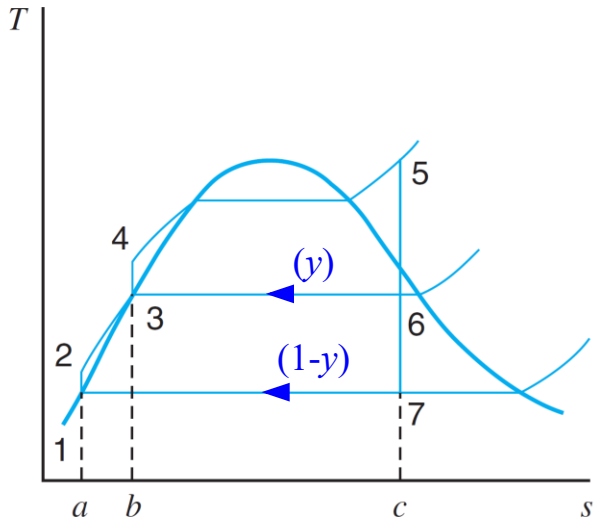
$$w_{\text{pump I, in}} = \text{v}_1(P_2 - P_1)$$

$$w_{\text{pump II, in}} = \text{v}_3(P_4 - P_3)$$

$y = \dot{m}_6 / \dot{m}_5$

(fraction of stream extracted)





$$\dot{m}_2 h_2 + \dot{m}_6 h_6 = \dot{m}_3 h_3$$

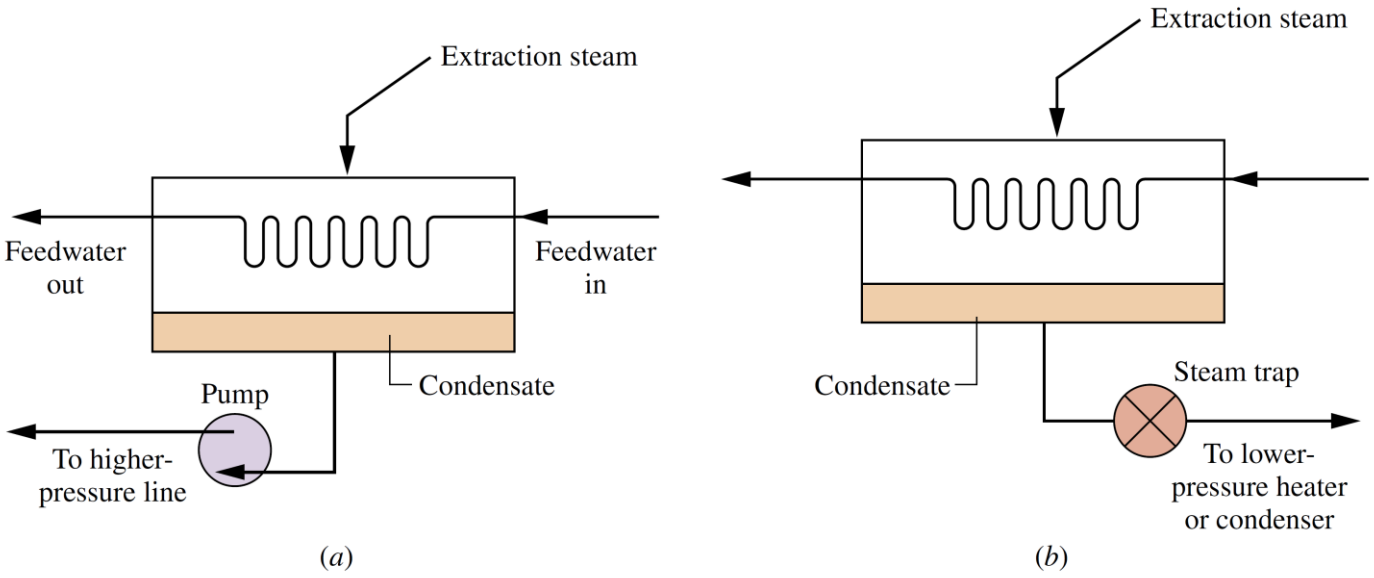
$$\Rightarrow (1-y)\dot{m}_5 h_2 + y\dot{m}_5 h_6 = \dot{m}_5 h_3 \quad \Rightarrow \quad \boxed{y = \frac{h_3 - h_2}{h_6 - h_2}}$$

Closed Feedwater Heater

Another type of feedwater heater frequently used in steam power plants is the closed feedwater heater, in which heat is transferred from the extracted steam to the feedwater without any mixing taking place. The two streams now can be at different pressures, since they do not mix.

- Boiler feedwater is heated up by heat transfer through a shell and tube heat exchanger.
- Steam enters as superheated, condenses and leaves as a subcooled liquid.
- A pinch analysis of the heat transfer is necessary to work out the inlet and outlet conditions for closed FWH.
- The point where the temperature difference between the hot fluid and the cold fluid is at a minimum is called the **pinch point**.

Examples of closed feedwater heaters

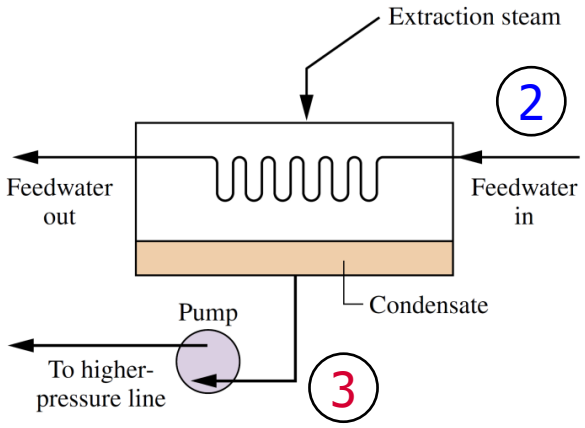


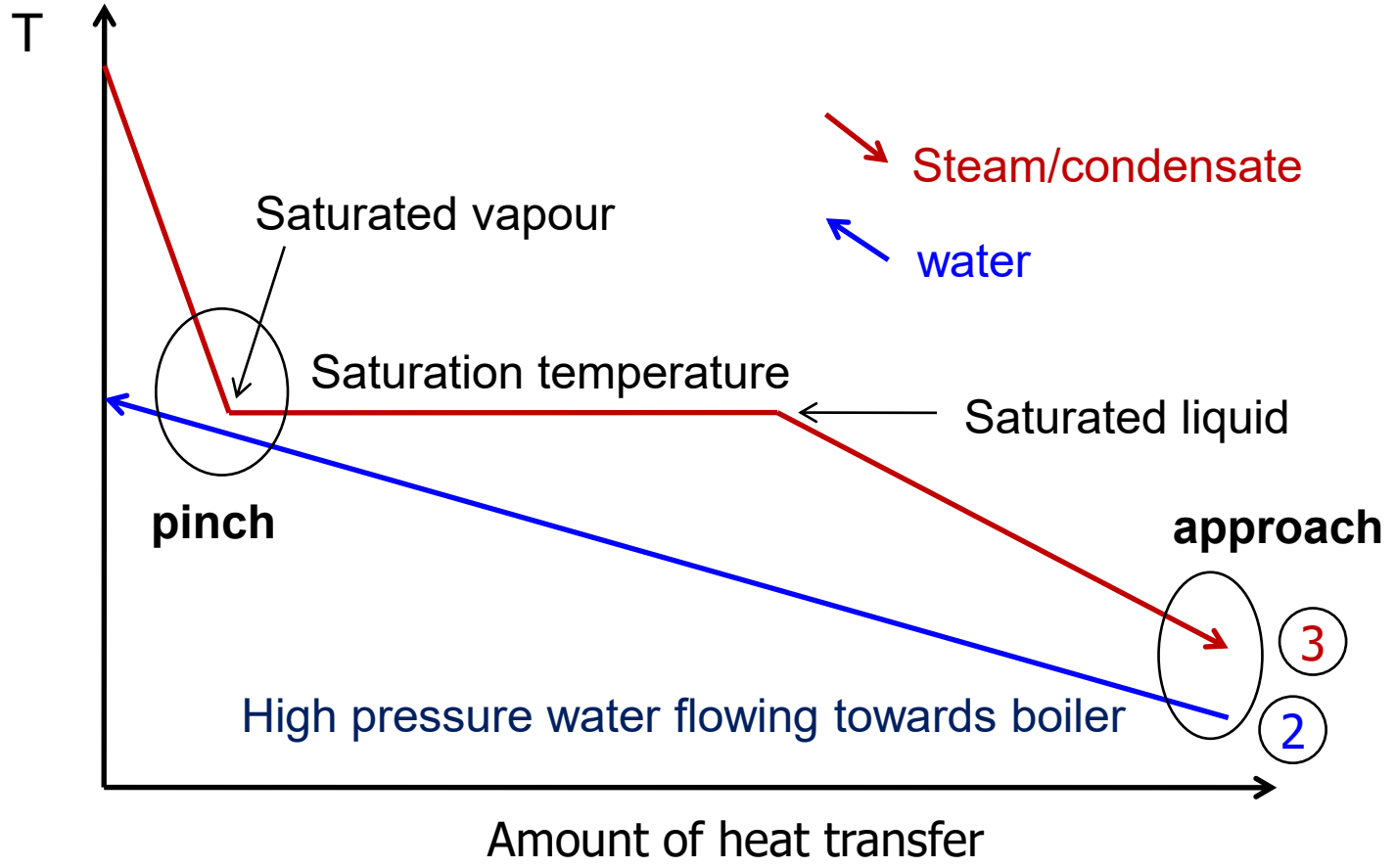
The condensed steam is either pumped to the feedwater line (left figure) or routed to another heater or to the condenser through a device called a trap (right figure).

Closed Feedwater Heater Pinch Diagram

Pinch analysis is a methodology for minimising energy consumption of a process by calculating thermodynamically feasible energy targets (or minimum energy consumption) and achieving them by optimising heat recovery systems, energy supply methods and process operating conditions.

Heat transfer versus Temperature diagram



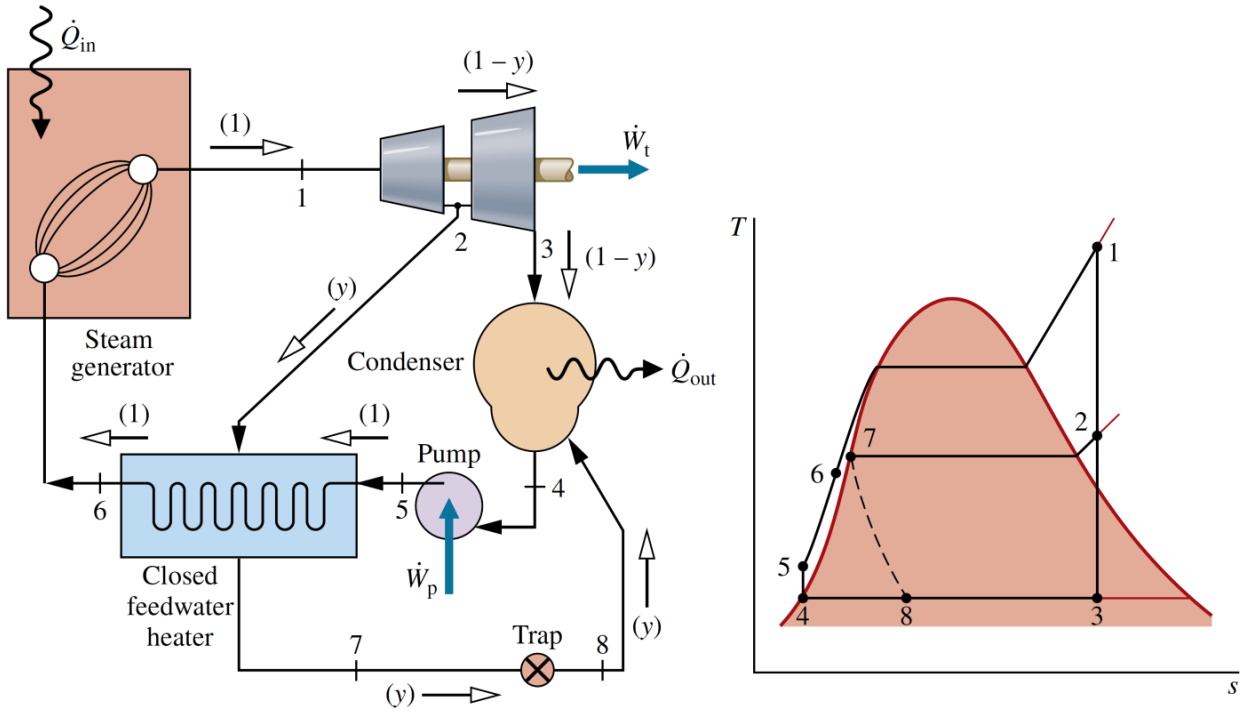


Closed Feedwater Heater with Trap

Closed feedwater heaters are more complex because of the internal tubing network, and thus they are more expensive. Heat transfer in closed feedwater heaters is less effective since the two streams are not allowed to be in direct contact. However, closed feedwater heaters do not require a separate pump for each heater since the extracted steam and the feedwater can be at different pressures.

Similar to open FWH, an energy balance gives the fraction of steam extracted from the turbine for the closed FWH.

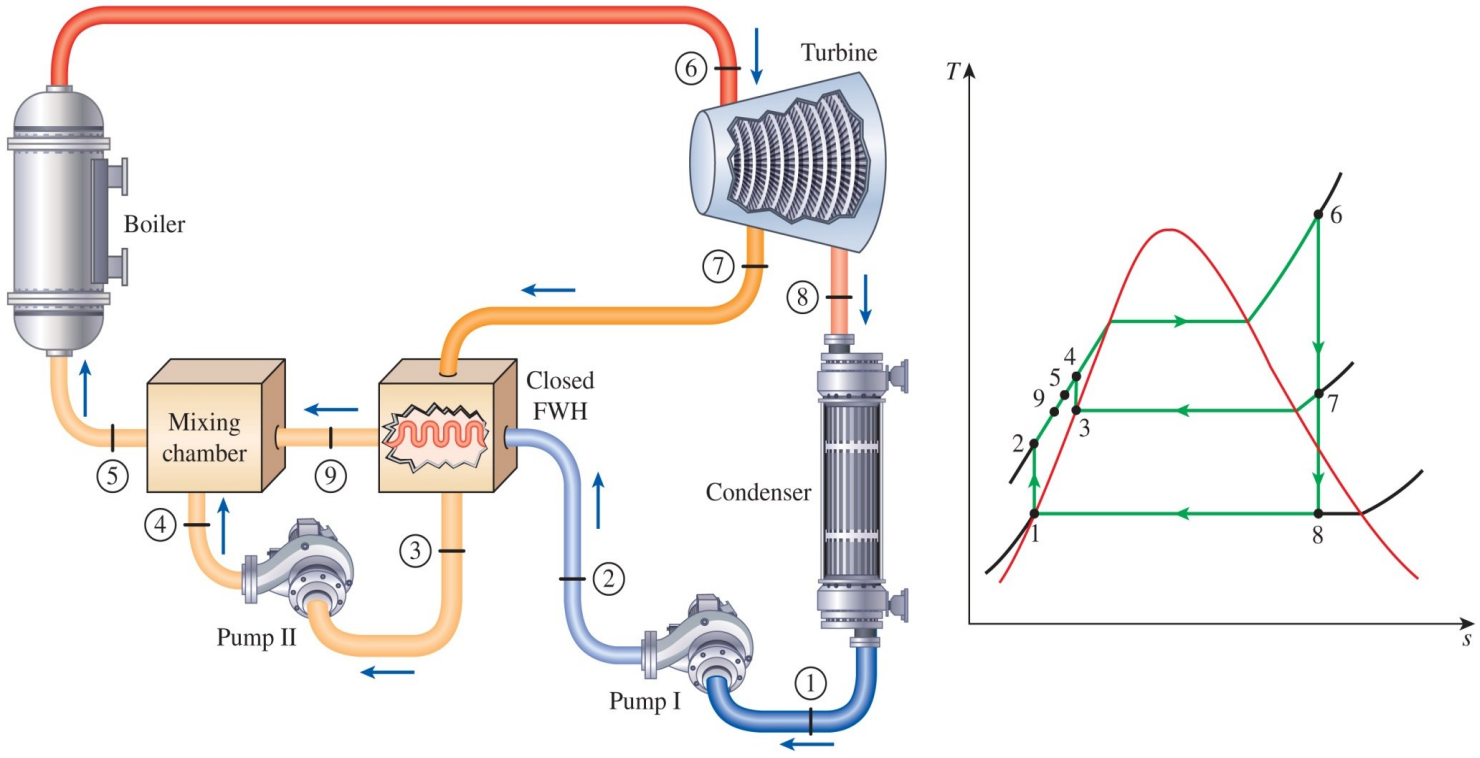
Regenerative vapour power cycle with one closed feedwater heater with a steam trap.



The condensed steam is routed to the condenser through a steam trap.

Closed Feedwater Heater with Pump

The condensed steam is can be pumped to the feedwater line as shown below.



The condensed steam is pumped to the feedwater line.

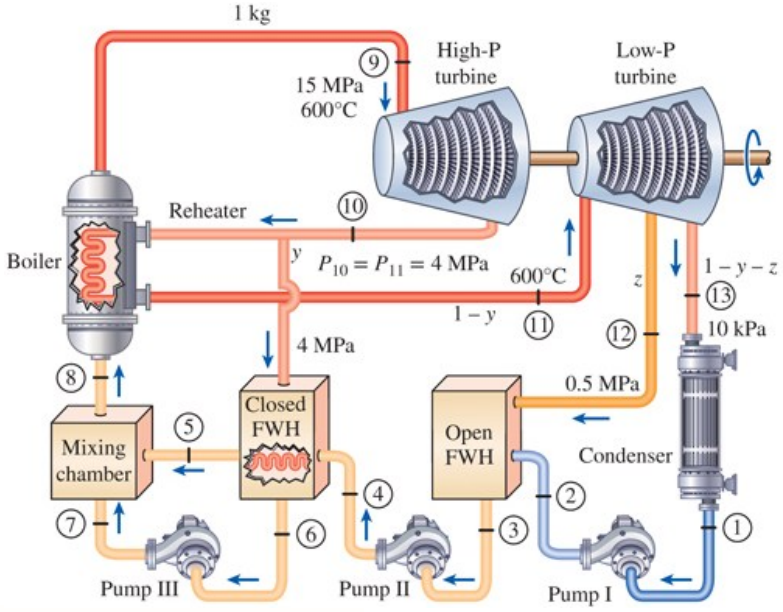
For an ideal closed feedwater heater, the feedwater is heated to the exit temperature of the extracted steam ($T_9 = T_3$), which ideally leaves the heater as a saturated liquid at the extraction pressure.

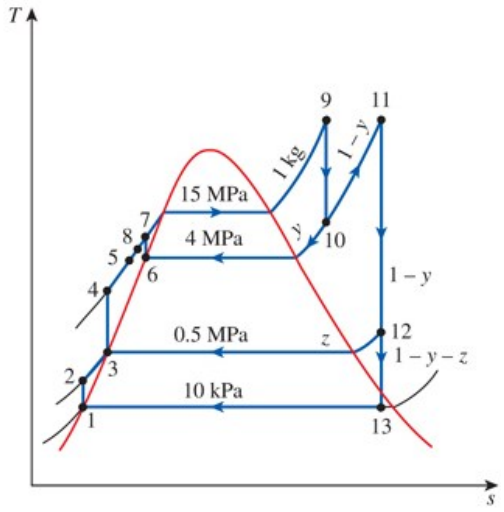
Open feedwater heaters are simple and inexpensive and have good heat transfer characteristics. For each heater, however, a pump is required to handle the feedwater.

Most steam power plants use **a combination of open and closed feedwater heaters.**

Example 4.3

Consider a steam power plant that operates on an ideal reheat–regenerative Rankine cycle with one open feedwater heater, one closed feedwater heater, and one reheater. Steam enters the turbine at 15 MPa and 600°C and is condensed in the condenser at a pressure of 10 kPa. Some steam is extracted from the turbine at 4 MPa for the closed feedwater heater, and the remaining steam is reheated at the same pressure to 600°C. The extracted steam is completely condensed in the heater and is pumped to 15 MPa before it mixes with the feedwater at the same pressure. Steam for the open feedwater heater is extracted from the low-pressure turbine at a pressure of 0.5 MPa. Determine the fractions of steam extracted from the turbine as well as the thermal efficiency of the cycle.





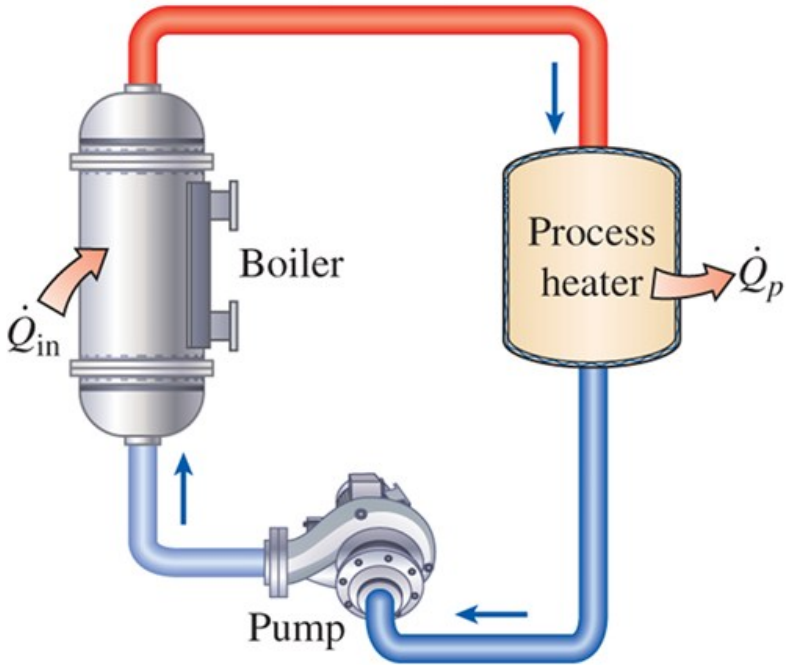
Source: Cengel (9th Ed.)

The enthalpies at the various states and the pump work per unit mass of fluid flow are:

$h_1 = 191.81$ kJ/kg
$h_2 = 192.30$ kJ/kg
$h_3 = 640.09$ kJ/kg
$h_4 = 643.92$ kJ/kg
$h_5 = 1087.4$ kJ/kg
$h_6 = 1087.4$ kJ/kg
$h_7 = 1101.2$ kJ/kg
$h_8 = 1089.8$ kJ/kg
$h_9 = 3155.0$ kJ/kg
$h_{10} = 3155.0$ kJ/kg
$h_{11} = 3674.9$ kJ/kg
$h_{12} = 3014.8$ kJ/kg
$h_{13} = 2335.7$ kJ/kg
$w_{\text{pump I,in}} = 0.49$ kJ/kg
$w_{\text{pump II,in}} = 3.83$ kJ/kg
$w_{\text{pump III,in}} = 13.77$ kJ/kg

Cogeneration Systems

- Many industries require energy input in the form of heat, called *process heat*. Process heat in these industries is usually supplied by steam at 5 to 7 atm and 150 to 200°C. Energy is usually transferred to the steam by burning coal, oil, natural gas, or another fuel in a furnace.



A simple process–heating plant.

- Industries that use large amounts of process heat also consume a large amount of electric power.
- It makes sense to use the already–existing work potential to produce power instead of letting it go to waste.
- The result is a plant that produces electricity while meeting the process–heat requirements of certain industrial processes (cogeneration plant).
- **Cogeneration:** The production of more than one useful form of energy (such as process heat and electric power) from the same energy source.

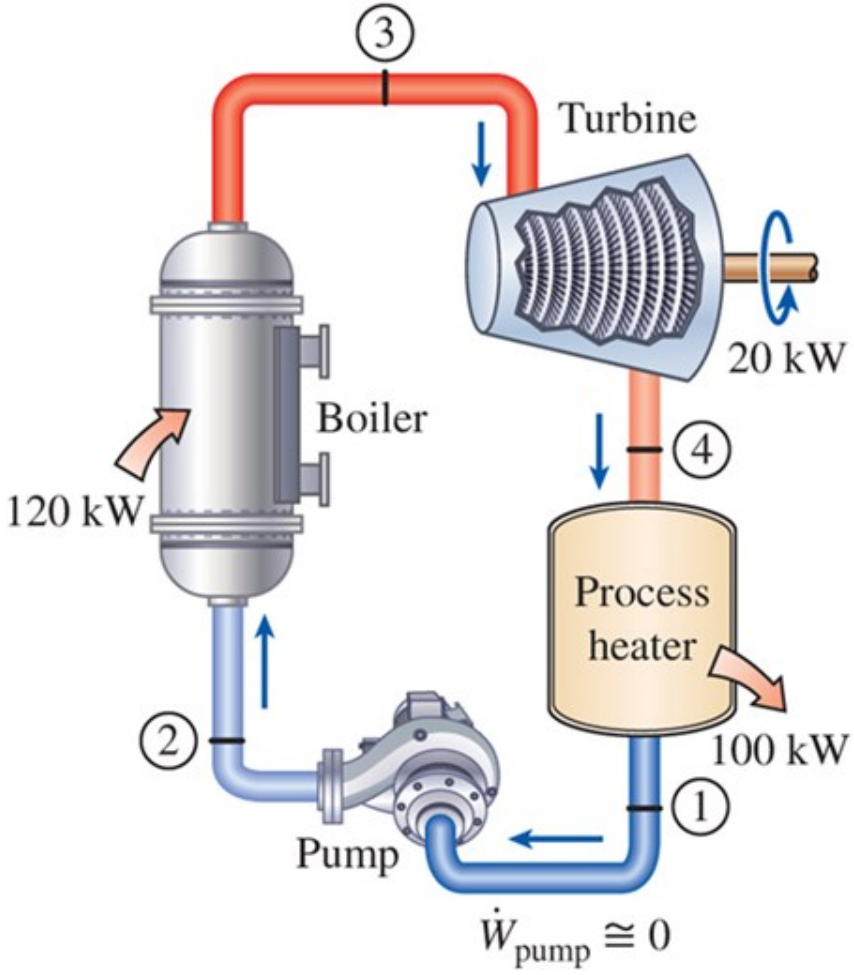
Cogeneration Systems

- Utilization factor

$$\varepsilon_u = \frac{\text{Net Power output} \ + \ \text{Process heat delivered}}{\text{Total heat input}}$$

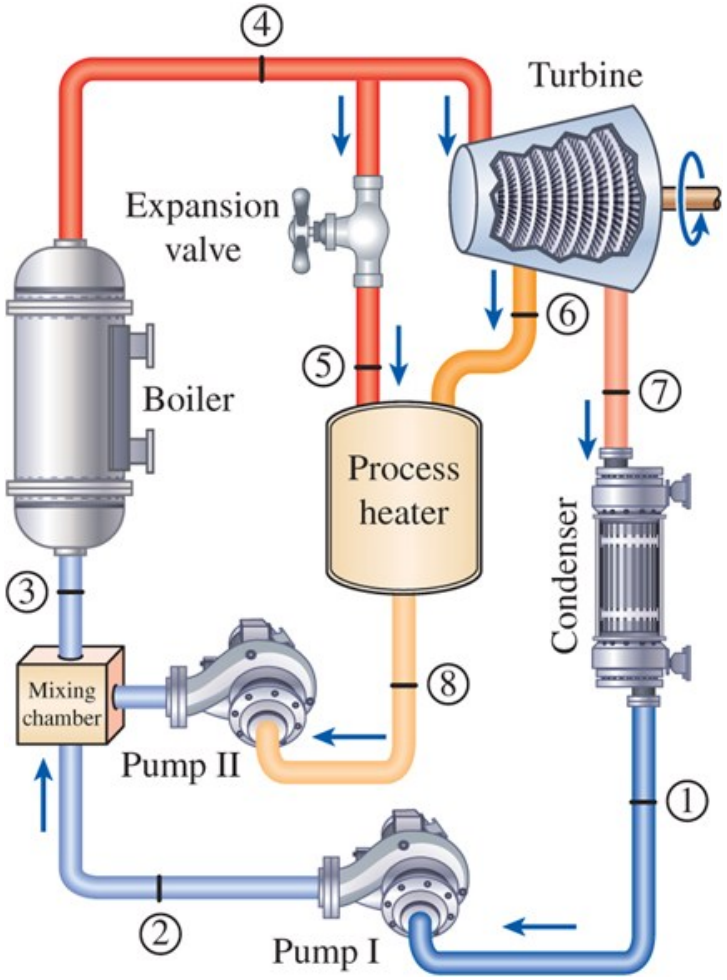
$$= \frac{\dot{W}_{net} + \dot{Q}_p}{\dot{Q}_{in}} = 1 - \frac{\dot{Q}_{out}}{\dot{Q}_{in}}$$

- The utilization factor of the ideal steam–turbine cogeneration plant is 100%.
- Actual cogeneration plants have utilization factors as high as 80%.
- Some recent cogeneration plants have even higher utilization factors.



An ideal cogeneration plant.

Cogeneration Plant with Adjustable Loads



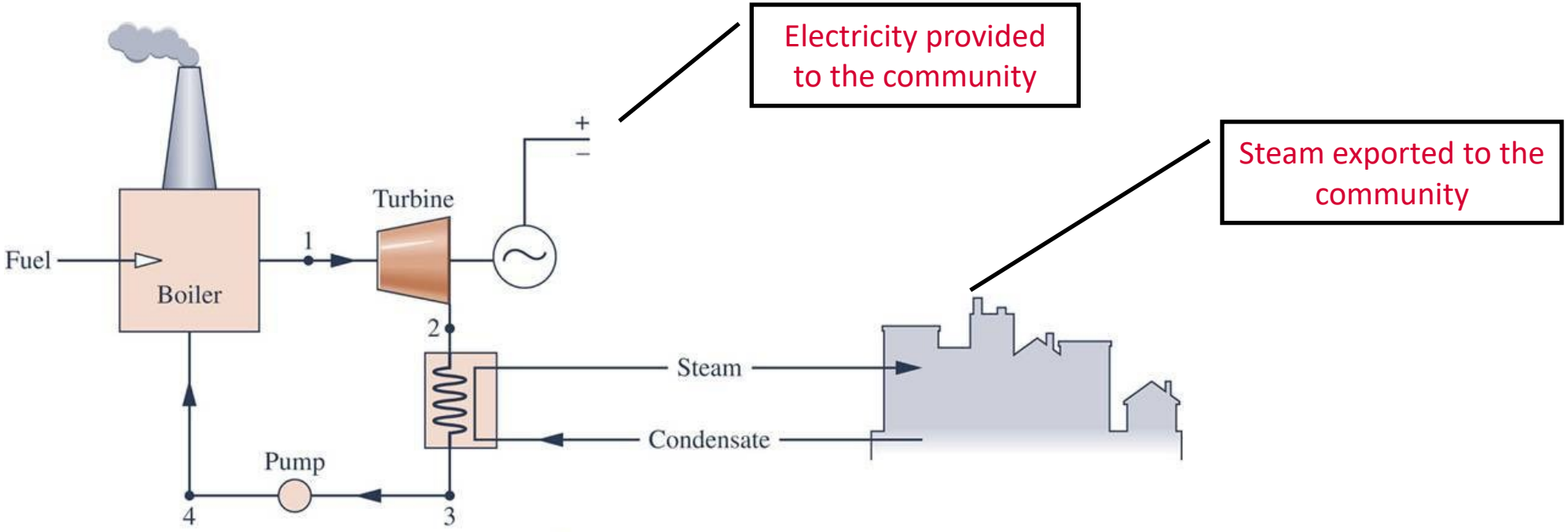
- At times of high demand for process heat, all the steam is routed to the process–heating units and none to the condenser $(m_7 = 0)$. The waste heat is zero in this mode.
- If this is not sufficient, some steam leaving the boiler is throttled by an expansion or pressure–reducing valve to the extraction pressure P_6 and is directed to the process–heating unit.
- Maximum process heating is realised when all the steam leaving the boiler passes through the valve $(m_5 = m_4)$. No power is produced in this mode.
- When there is no demand for process heat, all the steam passes through the turbine and the condenser $(m_5 = m_6 = 0)$ and the cogeneration plant operates as an ordinary steam power plant.
- The rates of heat input, heat rejected, and process heat supply as well as the power produced for this cogeneration plant can be expressed as follows:

$$\dot{Q}_{\text{in}} = \dot{m}_3 (h_4 - h_3) \qquad \dot{Q}_{\text{p}} = \dot{m}_5 h_5 + \dot{m}_6 h_6 - \dot{m}_8 h_8$$

$$\dot{Q}_{\text{out}} = \dot{m}_7 (h_7 - h_1) \qquad \dot{W}_{\text{turb}} = (\dot{m}_4 - \dot{m}_5)(h_4 - h_6) + \dot{m}_7 (h_6 - h_7)$$

Cogeneration Systems

- An application of cogeneration based on vapour power plants is district heating – providing steam or hot water for space heating together with electricity for domestic, commercial, and industrial use.



Combined Power Cycles

In Open Cycle Gas Turbines, the flue gas is sent to the stack at relatively high temperatures. The remaining heat from combustion is generally wasted.

The addition of a combined steam cycle to an open cycle gas turbine extract more energy from the flue gas and increase power output through the use of steam turbines. The principle is that the exhaust of one heat engine is used as the heat source for another. This is effectively a combination of a Brayton Cycle (gas turbine) and a Rankine Cycle (steam turbine). By combining both gas and steam cycles, high input temperatures and low output temperatures can be achieved, resulting in in improved overall efficiency.

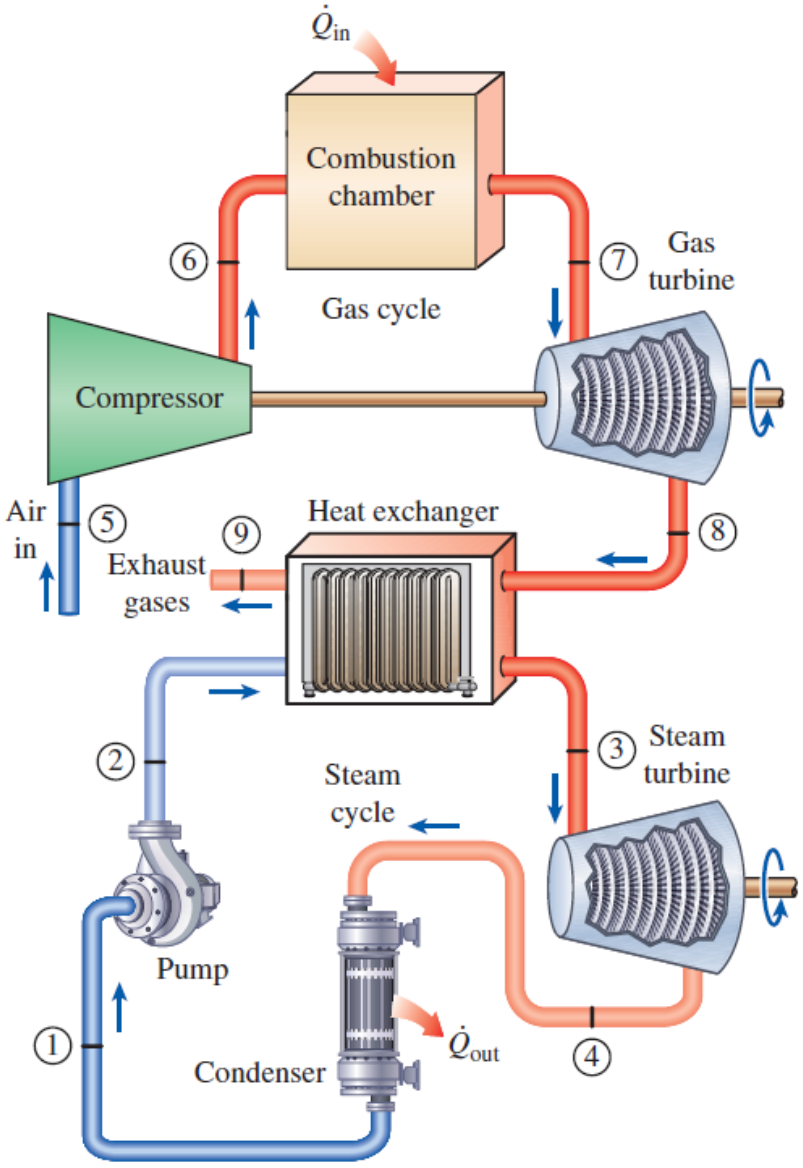
The combined cycle increases the efficiency without increasing the initial cost greatly. Consequently, many new power plants operate on combined cycles, and many more existing steam- or gas-turbine plants are being converted to combined-cycle power plants. Thermal efficiencies over 50% are reported.

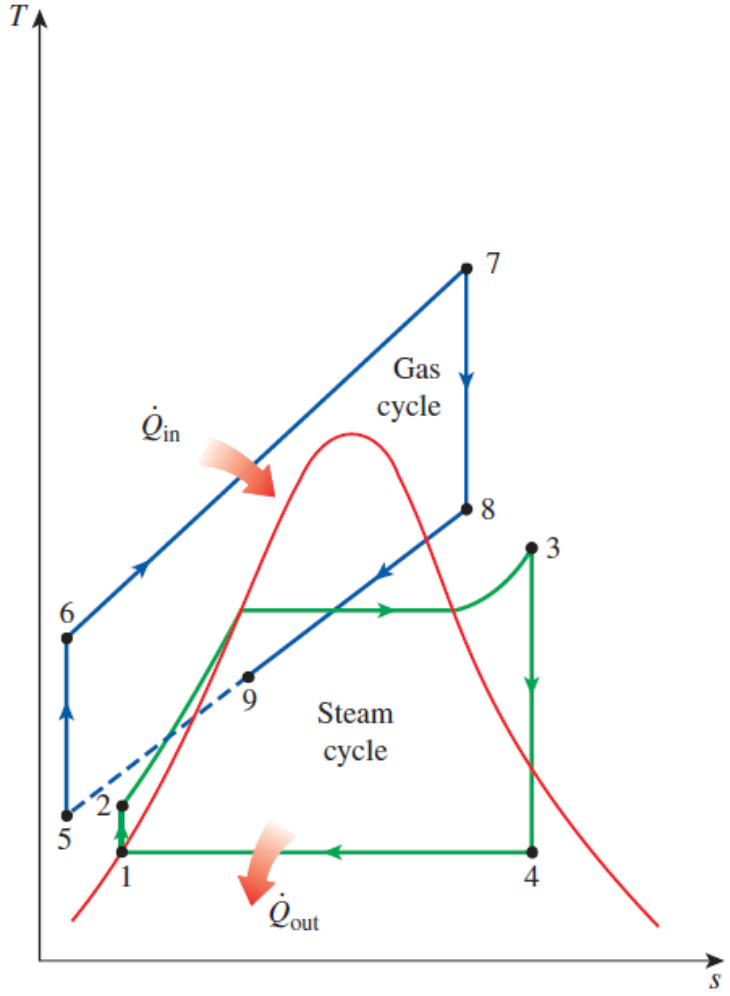
https://www.youtube.com/watch?v=ZHLTZP540_8&feature=related

https://www.youtube.com/watch?v=eeiu-wcyEbs (relevant until 1′ 58′′)

Combined Power Cycles

Image source: Cengel (9th Ed.) Section: 10-9





Combined gas–steam power plant

Combined Power Cycles

- A 1090-MW Tohoku combined plant that was put in commercial operation in 1985 in Niigata, Japan, is reported to operate at a thermal efficiency of 44%. This plant has two 191-MW steam turbines and six 118-MW gas turbines. Hot combustion gases enter the gas turbines at 1154°C, and steam enters the steam turbines at 500°C. Steam is cooled in the condenser by cooling water at an average temperature of 15°C. The compressors have a pressure ratio of 14, and the mass flow rate of air through the compressors is 443 kg/s.
- A 1350-MW combined-cycle power plant built in Ambarli, Turkey, in 1988 by Siemens is the first commercially operating thermal plant in the world to attain an efficiency level as high as 52.5% at design operating conditions. This plant has six 150-MW gas turbines and three 173-MW steam turbines.
- Another plant built by Siemens in Irsching, Germany, in 2011 reached a thermal efficiency of 60.8% with an electrical output of 578 MW.
- In 2016, General Electric reported 62.2% efficiency for its combined cycle plant in Bouchain, France with an output of 594 MW.
- The new target for thermal efficiency is 65%, approaching the Carnot limit.

Thanks for listening

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Next: **Ideal Gas Mixtures & Psychrometry**