Lecture 5: Regenerative Rankine Cycle

Course: MECH-422 – Power Plants

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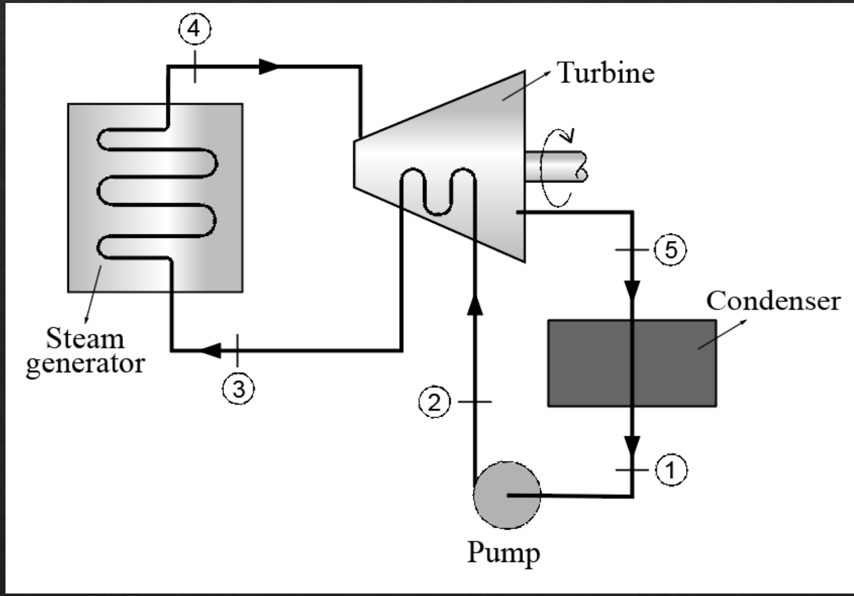
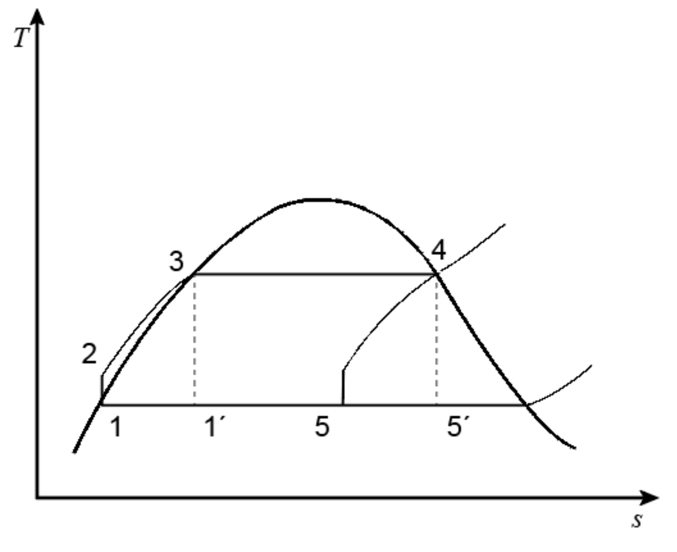
Term: Fall 2021

BUITEMS – DEPARTMENT OF MECHANICAL ENGINEERING

Regenerative Rankine cycle



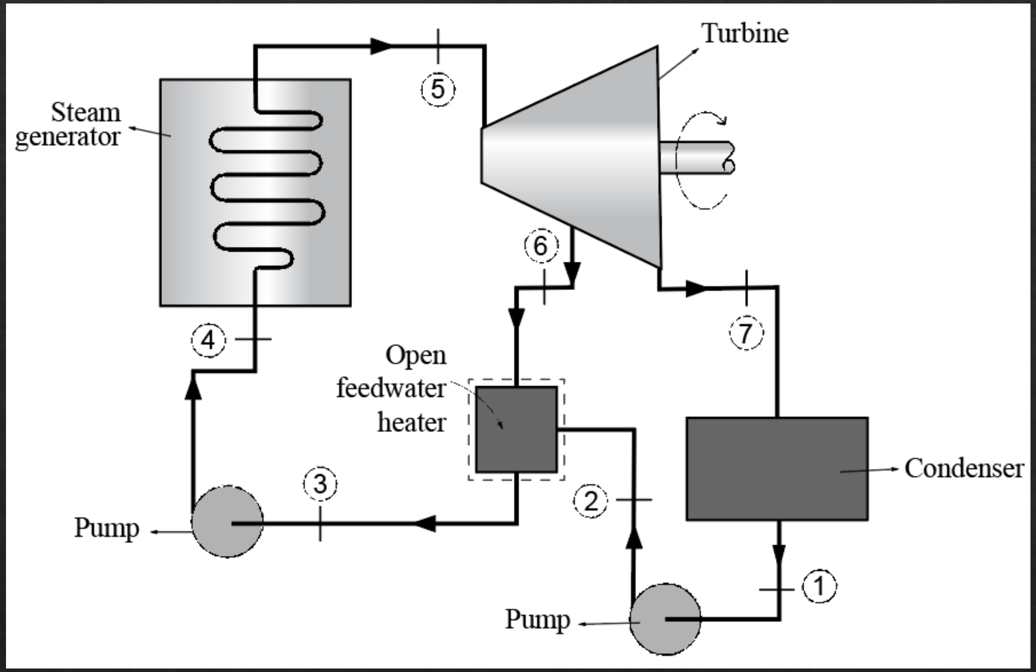


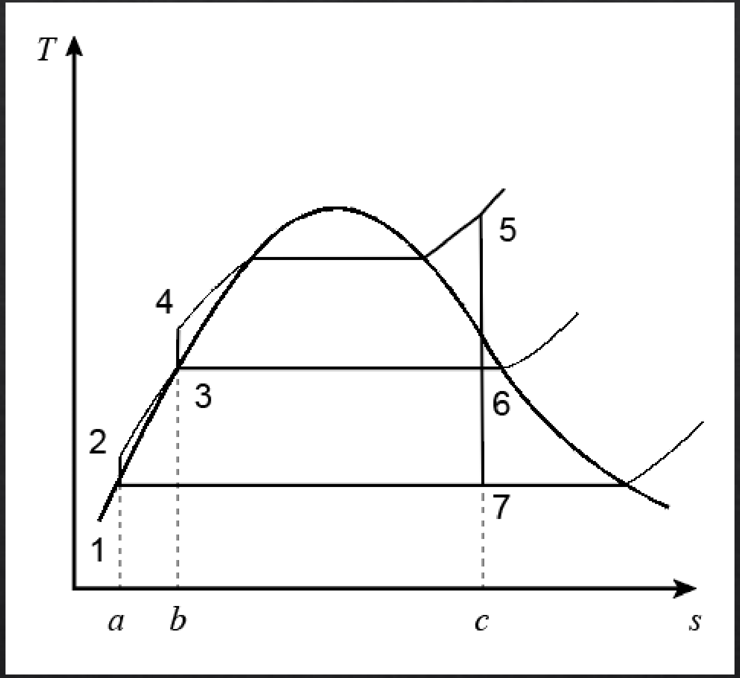
An ideal regenerative Rankine cycle

The idea of a regenerative cycle is to increase the temperature of the steam generator inlet flow from state 2 to as close to state 3 as possible to increase the average temperature at which heat is supplied to the cycle. This also reduces external irreversibilities due to the temperature difference between the working fluid and the flue gas.

As a result of this reversible heat transfer from the steam expanding in the turbine, the outlet flow of the turbine would be at state 5 instead of state 5′.

Open feedwater heaters



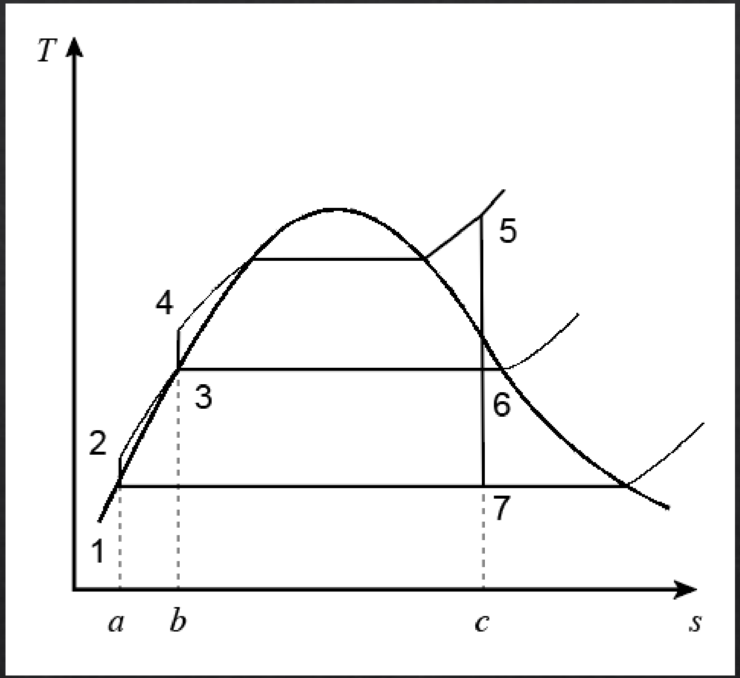


A regenerative Rankine cycle with an open feedwater heater

Open feedwater heaters

Mass balance: $\dot{m}_2 + \dot{m}_6 = \dot{m}_3$

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



$$\frac{\dot{m}_2}{\dot{m}_3} + \frac{\dot{m}_6}{\dot{m}_3} = 1$$

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

$$\frac{\dot{m}_6}{\dot{m}_3} = \frac{\dot{m}_{ext}}{\dot{m}_t} = \lambda$$

$$\frac{\dot{m}_2}{\dot{m}_3} = 1 - \lambda$$

$$(1 - \lambda) h_2 + \lambda h_6 = h_3$$

$$\lambda = \frac{h_3 - h_2}{h_6 - h_2}$$

$$\dot{W}_{Turb} = \dot{m}_t(h_5 - h_6) + (\dot{m}_t - \dot{m}_6)(h_6 - h_7)$$

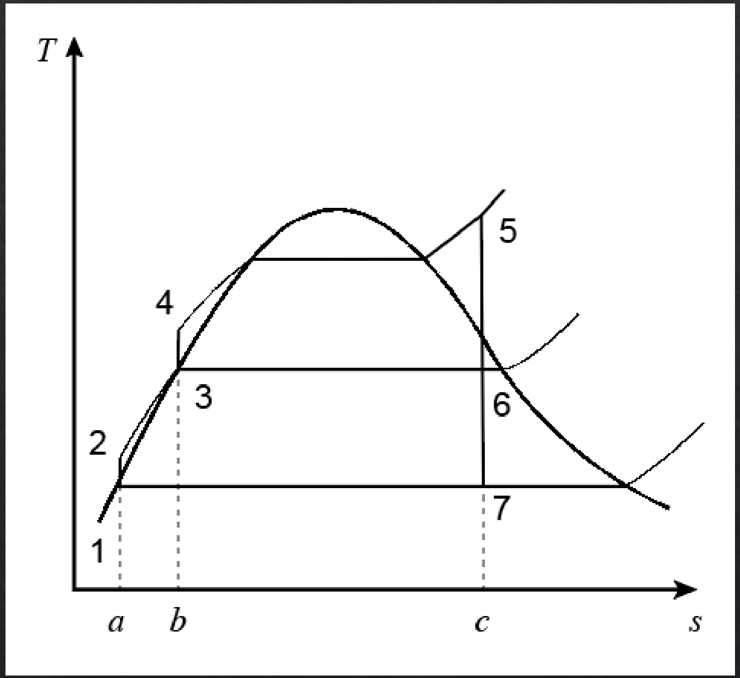
$$w_{Turb} = \frac{\dot{W}_{Turb}}{\dot{m}_t} = (h_5 - h_6) + (1 - \lambda)(h_6 - h_7)$$

$$\dot{W}_{pump} = (\dot{m}_t - \dot{m}_6)(h_2 - h_1) + \dot{m}_t(h_4 - h_3)$$

$$w_{pump} = \frac{\dot{W}_{pump}}{\dot{m}_t} = (1 - \lambda)(h_2 - h_1) + (h_4 - h_3)$$

$$\dot{Q}_{SG} = \dot{m}_t(h_5 - h_4) q_{SG} = \frac{\dot{Q}_{SG}}{\dot{m}_t} = h_5 - h_4$$

$$\dot{Q}_{Cond} = (\dot{m}_t - \dot{m}_6)(h_7 - h_1) \quad q_{Cond} = (1 - \lambda)(h_7 - h_1)$$



Example 2.10

In an ideal regenerative Rankine cycle, the operating pressure of the steam generator and the condenser are 10,000 kPa and 10 kPa, respectively, and the turbine inlet steam is superheated at the temperature of 600°C (similar to Example 2.6). In the turbine, some steam is extracted at the pressure of 1,000 kPa and diverted to an open feedwater heater. If the outlet of the open feedwater is saturated liquid water, determine the specific heat transfers in the steam generator and the condenser, the specific work involved in the turbine and the pumps, and the thermal efficiency of the cycle.

Given: An ideal regenerative Rankine cycle with the working fluid of water operates at the given operating conditions. The cycle also has an open feedwater system operating at the given pressure.

Find: The specific heat transfers in the steam generator and the condenser, the specific work involved in the turbine and the pumps, and the thermal efficiency of the cycle.

Assumptions:

1. The cycle is an ideal regenerative Rankine cycle with the working fluid of water.
2. The steady state control volume mass and energy balance analyses with no kinetic and potential energy changes are applied to each component.
3. There is no pressure drop due to friction and no heat loss to the surrounding in the components and piping systems, including in the open feedwater heater.

State 1:

$h_1 = 191.83\ kJ/kg$

$v_1 = 0.001010\, m^3/kg$

State 2:

$h_2 = w_{Pump1} + h_1 = v_1(P_2 - P_1) + h_1 = 0.001010\ \ m^3/kg \times$

$(1000 - 10)kPa\ \left[\frac{1000\ N/m^2}{1\ kPa}\right]\left[\frac{1\ kJ}{1000\ N.m}\right] + 191.83\ kJ/kg =$

$192.83\ kJ/kg$

State 3:

$h_3 = h_{f\ @\ 1000\ kPa} = 762.81\ kJ/kg$

$v_3 = v_{f\ @\ 1000\ kPa} = 0.001127\, m^3/kg$

State 4:

$$h_4 = w_{Pump2} + h_3 = v_3(P_4 - P_3) + h_3 =$$

$$0.001127\ m^3/kg\ \ (10000 - 1000)kPa\ \left[\frac{1000\ N/m^2}{1\ kPa}\right]\left[\frac{1\ kJ}{1000\ N.m}\right] +$$

$$762.81\ kJ/kg = 772.95\ kJ/kg$$

State 5:

$$h_5 = 3625.3\ kJ/kg$$

$$s_5 = 6.9029\ kJ/kg\ K$$

State 6:

$$s_5 = s_6 = 6.9029\ kJ/kg\ K$$

$$h_6 = \ 2931.8\ kJ/kg$$

State 7:

$s_5 = s_7 = 6.9029\ kJ/kg\ K$

$s_7 = s_{f@10\ kPa} + x_7 s_{fg@10\ kPa} \rightarrow 6.9029 \ = 0.6493 +$

$x_7(7.5009) \ \rightarrow x_7 = 0.8337\ or\ 83.37\%$

$h_7 = h_{f@10\ kPa} + x_7 h_{fg@10\ kPa} = 191.83 + 0.8337 \times 2392.8 =$

$2186.7\ kJ/kg$

$\lambda = \frac{h_3 - h_2}{h_6 - h_2} = \frac{762.81 - 192.83}{2931.8 - 192.83} = 0.2081$

$$w_{Turb} = (h_5 - h_6) + (1 - \lambda)(h_6 - h_7) = (3625.3 - 2931.8) + (1 - 0.2081)(2931.8 - 2186.7) = 1283.5\ kJ/kg$$

$$w_{Pump} = (1 - \lambda)(h_2 - h_1) + (h_4 - h_3) = (1 - 0.2081)(192.83 - 191.83) + (772.95 - 762.81) = 10.93\ kJ/kg$$

$$q_{SG} = h_5 - h_4 = 3625.3 - 772.95 = 2852.3\ kJ/kg$$

$$q_{Cond} = (1 - \lambda)(h_7 - h_1) = (1 - 0.2081)(2186.7 - 191.83) = 1579.7\ kJ/kg$$

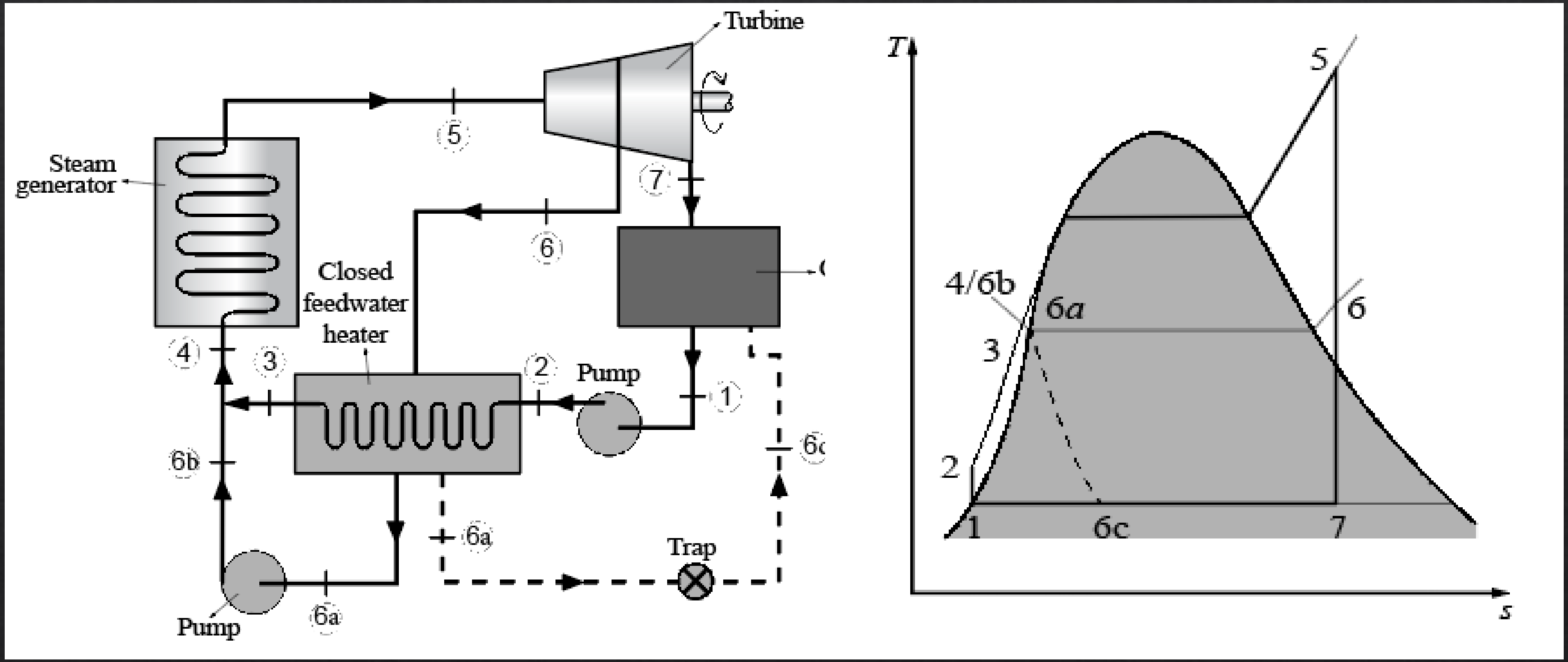
$$w_{Net} = w_{Turb} - w_{Pump} = 1283.5 - 10.93 = 1272.6\ \ kJ/kg$$

$$\eta_{Th} = \frac{w_{Net}}{q_{SG}} = \frac{1272.6}{2852.3} = 0.4462\ \text{ or } 44.62\%$$

Comparison of the performance of the Rankine cycle with the open feedwater heater and the Rankine cycle without one in Examples 2.10 and 2.6

Type of Cycle	P_2 (kPa)	P_1 (kPa)	x_4 or x_7	q_{SG} (kJ/kg)	q_{Cond} (kJ/kg)	w_{Pump} (kJ/kg)	w_{Turb} (kJ/kg)	w_{Net} (kJ/kg)	η_{Th} (%)
Simple Rankine cycle without feedwater heater (Example 2.6)	10,000	10	0.834	3423.4	1994.9	10.1	1438.6	1428.5	41.73
Rankine cycle with open feedwater heater (Example 2.10)	10,000	10	0.834	2852.3	1579.7	10.9	1283.5	1272.6	44.62

Closed feedwater heaters

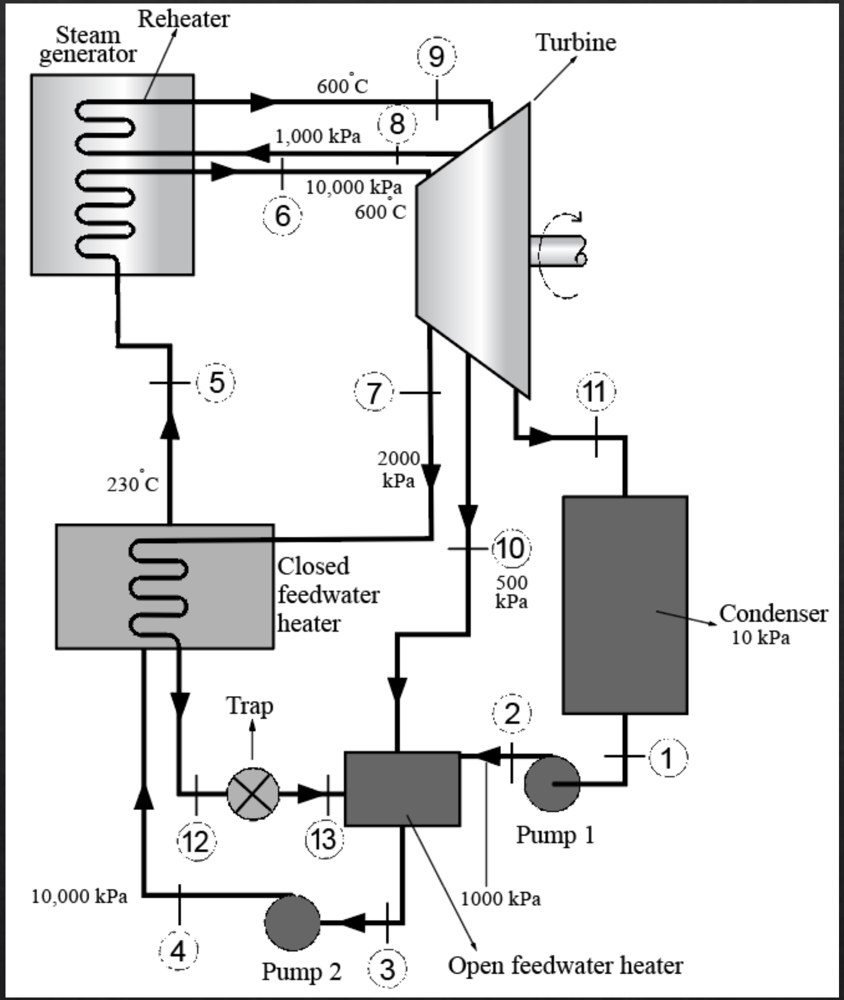


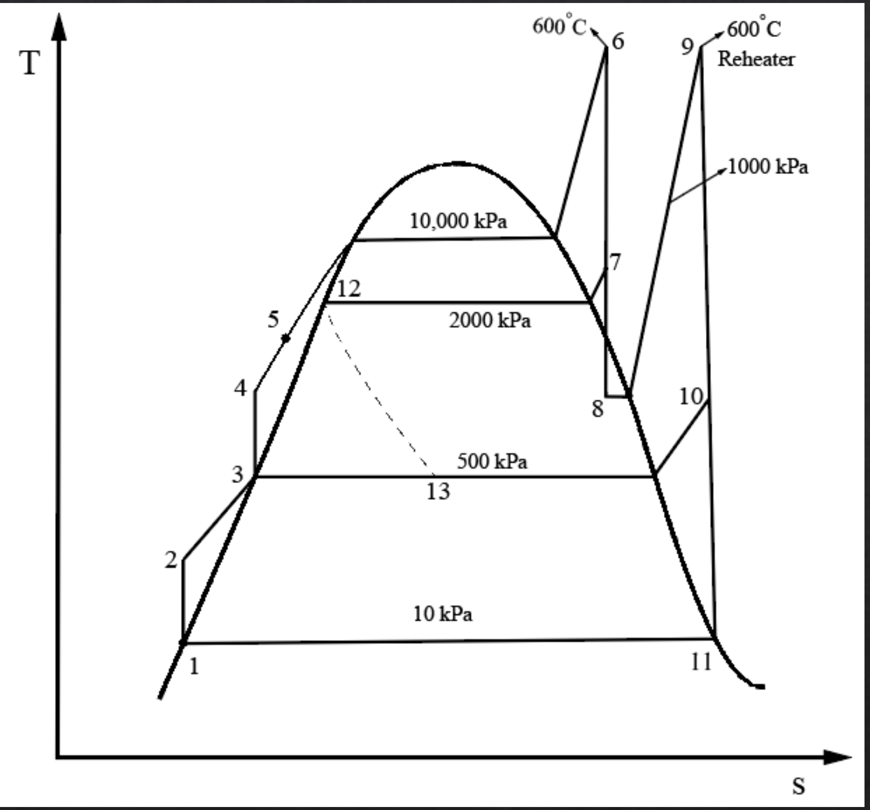
A regenerative Rankine cycle with a closed feedwater heater

Comparison of the performance of the Rankine cycle with the closed feedwater heater, with the open feedwater heater, and without a feedwater heater in Examples 2.11, 2.10, and 2.6

Type of Cycle	P_2 (kPa)	P_1 (kPa)	x_4, x_7, or x_9	q_{SG} (kJ/kg)	q_{Cond} (kJ/kg)	w_{Pumps} (kJ/kg)	w_{Turb} (kJ/kg)	w_{Net} (kJ/kg)	η_{Th} (%)
Simple Rankine cycle without feedwater heater (Example 2.6)	10,000	10	0.834	3423.4	1994.9	10.1	1438.6	1428.5	41.73
Rankine cycle with open feedwater heater (Example 2.10)	10,000	10	0.834	2852.3	1579.7	10.9	1283.5	1272.6	44.62
Rankine cycle with closed feedwater heater (Example 2.11)	10,000	10	0.834	2964.6	1661.7	10.1	1314.2	1304.1	43.99

Reheat-Regenerative Rankine Cycle with Open & Closed Feedwater Heater



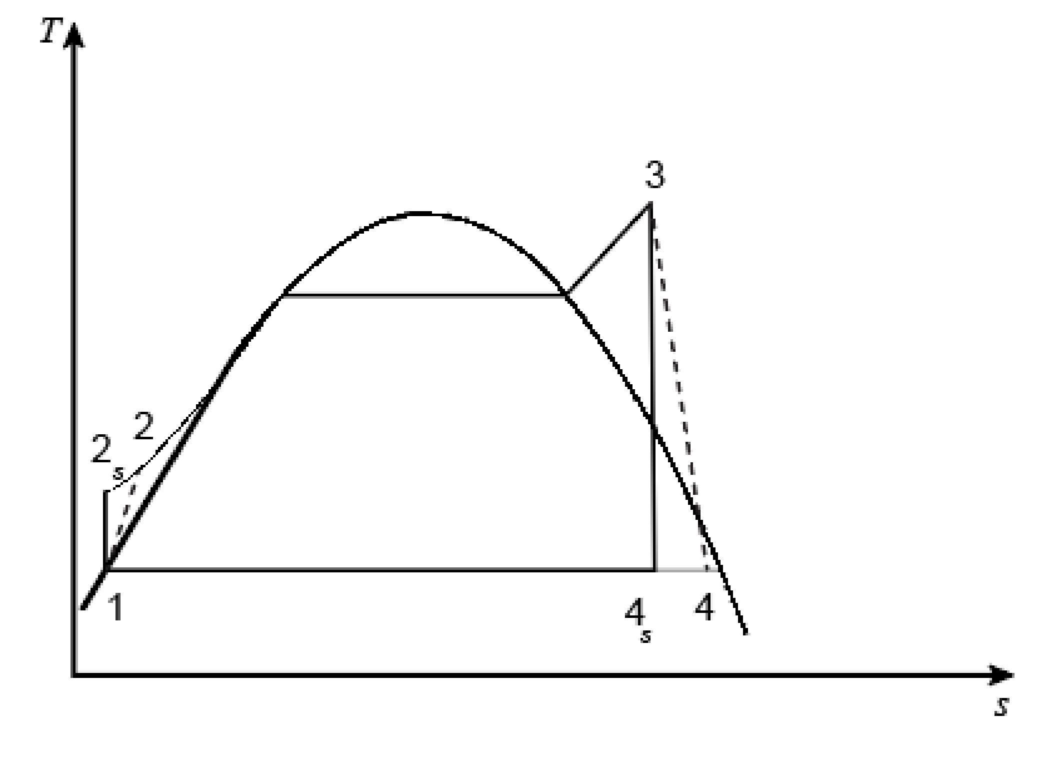


Schematic and T-s diagrams of the reheat-regenerative Rankine cycle in Example 2.12

Comparison of the performance of the Rankine cycle with various configurations (Examples 2.12, 2.11, 2.10, 2.9, 2.6, and 2.4)

Type of Cycle	P_2 (kPa)	P_1 (kPa)	x_4, x_7, x_9, or x_{11}	q_{SG} (kJ/kg)	q_{Cond} (kJ/kg)	w_{Pump} (kJ/kg)	w_{Turb} (kJ/kg)	w_{Net} (kJ/kg)	η_{Th} (%)
Simple Rankine –Turbine inlet: saturate steam (Example 2.4)	10,000	10	0.662	2522.8	1583.8	10.1	949.1	939.0	37.22
Simple Rankine –Turbine inlet: superheated at 600°C (Example 2.6)	10,000	10	0.834	3423.4	1994.9	10.1	1438.6	1428.5	41.73
Reheat Rankine cycle (Example 2.9)	10,000	10	0.984	4189.5	2354.1	10.1	1845.5	1835.4	43.81
Rankine cycle with open feedwater heater (Example 2.10)	10,000	10	0.834	2852.3	1579.7	10.9	1283.5	1272.6	44.62
Rankine cycle with closed feedwater heater (Example 2.11)	10,000	10	0.834	2964.6	1661.7	10.1	1314.2	1304.1	43.99
Reheat Rankine cycle with open and closed feedwater heaters (Example 2.12)	10,000	10	0.984	3281.8	1744.9	10.7	1547.66	1536.9	46.83

Deviation of actual steam cycle from ideal Rankine cycle



Losses associated with expansion and compression processes (in the turbine and the pump, respectively) in a Rankine cycle

- The primary cause of turbine losses is due to the flow of the working fluid through blades and passages.
- As a result of these losses, the expansion process is not isentropic, and the entropy increases in the process (in Figure 2.25, process 3-4 instead of state 3-4s).

- Irreversibilities in pumps are similar to those in turbines in nature but much smaller (Figure 2.25).

- Turbine work output 3-4s (Isentropic) > 3-4 (Actual)

- Pump work input 1-2s (Isentropic) < 1-2 (Actual)

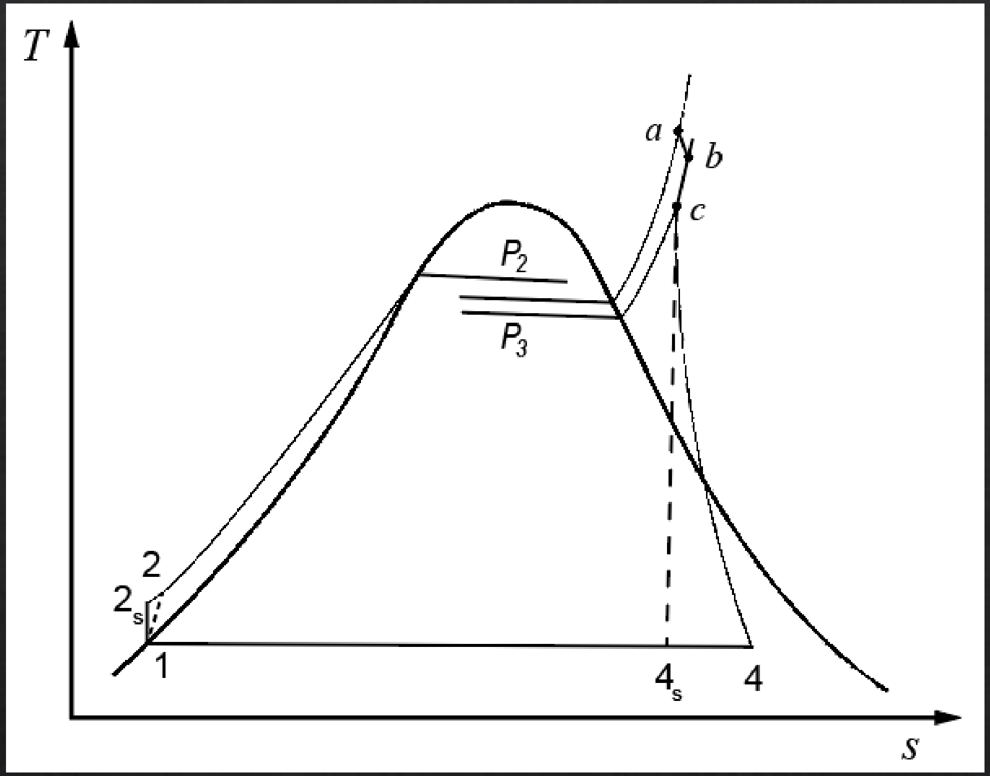
Deviation of actual steam cycle from ideal Rankine cycle

$$\eta_{Turb} = \frac{\dot{W}_{Turb}}{\left(\dot{W}_{Turb}\right)_s} = \frac{w_{Turb}}{(w_{Turb})_s} = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

$$\eta_{Pump} = \frac{\left(\dot{W}_{Pump}\right)_s}{\dot{W}_{Pump}} = \frac{\left(w_{Pump}\right)_s}{w_{Pump}} = \frac{h_{2s} - h_1}{h_2 - h_1}$$

where subscript s represents the isentropic expansion process

Deviation of actual steam cycle from ideal Rankine cycle - *Piping Losses*



Deviation of actual steam power plants from Rankine cycle

- In every steam power plant, extensive piping systems are required to transport the working fluid to and from components.
- Piping losses are mainly a result of pressure drops in pipes, valves, bends due to friction as well as heat losses.
- For instance, in the steam transfer from the steam generator to the turbine, in Figure 2.26, the process a-b represents the pressure drop and the heat loss is represented by process b-c.
- Figure 2.26 illustrates all the losses in the working fluid from the pump outlet to the turbine inlet due to pressure drops and heat losses in various heat-transfer surfaces within the steam generator, feedwater heaters, and the piping systems.

Example 2.13

Reconsider the Rankine cycle in Example 2.6, where the operating pressures of the steam generator and the condenser are 10,000 and 10 kPa, respectively, and the turbine inlet steam is superheated at the temperature of 600°C. If the isentropic efficiencies of the turbine and pump are both 90%, determine the specific heat transfers in the steam generator and the condenser, the specific work involved in the turbine and the pump, and the thermal efficiency and the back work ratio of the cycle. Also, if the power plant produces 250 MW power, determine the mass flow rate of the cycle's working fluid.

Given: A non-ideal Rankine cycle with the working fluid of water operates at the given operating pressure of the steam generator and the condenser. The turbine inlet flow is superheated steam and the condenser outlet flow is saturated liquid. The power output is also given. The processes in the turbine and pump are not isentropic and the isentropic efficiencies of the turbine and the pump are given.

Find: The specific heat transfers in the steam generator and the condenser, the specific work involved in the turbine and the pump, the thermal efficiency and the back work ratio of the cycle, and the mass flow rate of the working fluid.

Assumptions:

1. The cycle is a non-ideal Rankine steam cycle. The expansion and compression processes are adiabatic but not isentropic with the isentropic efficiencies of 90%.
2. The steady state control volume mass and energy balance analyses with no kinetic and potential energy changes are applied to each component.
3. There is no pressure drop due to friction and no heat loss to the surrounding in the components and piping systems.

$$h_1 = 191.83\ kJ/kg$$

$$h_{2s} = 201.92\ kJ/kg$$

$$h_3 = 3625.3\ kJ/kg$$

$$h_{4s} = 2186.7\ kJ/kg$$

$$\eta_{Pump} = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$0.9 = \frac{201.92 - 191.83}{h_2 - 191.83}$$

$$h_2 = 203.04\ kJ/kg$$

$$\eta_{Turb} = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

$$0.9 = \frac{3625.3 - h_4}{3625.3 - 2186.7}$$

$$h_4 = 2330.56\ kJ/kg$$

$$h_4 = h_{f@10\ kPa} + x_4 h_{fg@10\ kPa}$$

$$2330.56 = 191.83 + x_4 \times 2392.8$$

$$x_4 = 0.8938\ or\ 89.38\%$$

$$w_{Pump} = h_2 - h_1 = 203.04\ - 191.83 = 11.21\ kJ/kg$$

$$q_{SG} = h_3 - h_2 = 3625.3\ - 203.04\ = 3422.3\ kJ/kg$$

$$w_{Turb} = h_3 - h_4 = 3625.3\ - 2330.56\ = 1294.7\ kJ/kg$$

$$q_{Cond} = h_4 - h_1 = 2330.56\ - 191.83 = 2138.7\ kJ/kg$$

$$w_{Net} = w_{Turb} - w_{Pump} = 1294.7\ - 11.21 = 1283.5\ kJ/kg$$

$$\eta_{Th} = \frac{w_{net}}{q_{SG}} = \frac{1283.5}{3422.3} = 0.3750\ or\ 37.50\%$$

$$BWR = \frac{w_{Pump}}{w_{Turb}} = \frac{11.21}{1294.7} = 0.0087\ or\ 0.87\%$$

$$\dot{W}_{Net} = \dot{m}\, w_{Net} \text{ so } \dot{m} = \frac{\dot{W}_{Net}}{w_{Net}} = \frac{250{,}000\ kJ/s}{1283.5\ \ kJ/kg} = 194.78\ kg/s =$$

$$7.01 \times 10^5\ kg/h$$

Effects of the turbine and pump irreversibilities on the performance characteristics of the Rankine cycles in Examples 2.6 and 2.13

Type of Cycle	P_2 (kPa)	P_1 (kPa)	x_4, x_7, x_9, or x_{11}	q_{SG} (kJ/kg)	q_{Cond} (kJ/kg)	w_{Pump} (kJ/kg)	w_{Turb} (kJ/kg)	w_{Net} (kJ/kg)	η_{Th} (%)
Simple Rankine –Turbine inlet: saturate steam (Example 2.4)	10,000	10	0.662	2522.8	1583.8	10.1	949.1	939.0	37.22
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Reheat Rankine cycle with open and closed feedwater heaters (Example 2.12)	10,000	10	0.984	3281.8	1744.9	10.7	1547.66	1536.9	46.83

Effects of the turbine and pump isentropic efficiencies on the performance of the Rankine cycle in Example 2.13

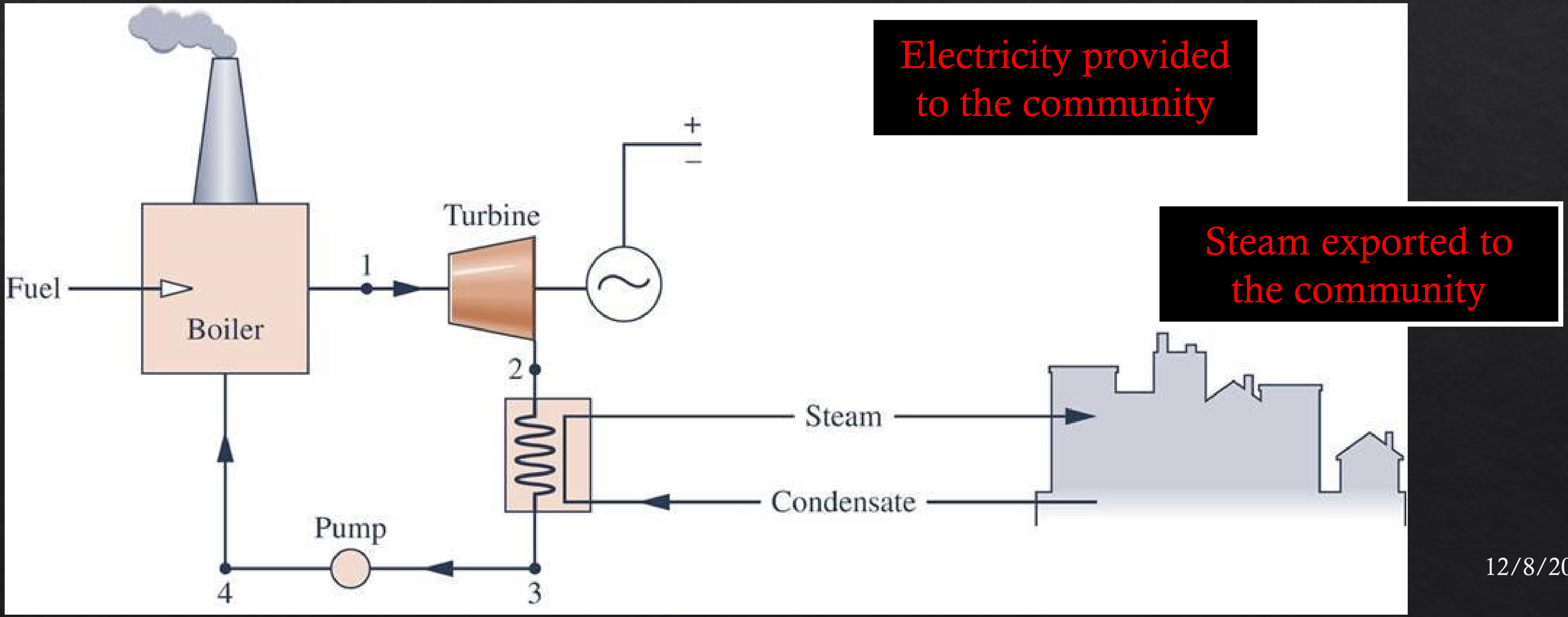
Isentropic Efficiency (%)	$P_2=P_3$ (kPa)	$P_1=P_4$ (kPa)	x_4	q_{SG} (kJ/kg)	q_{Cond} (kJ/kg)	w_{Pump} (kJ/kg)	w_{Turb} (kJ/kg)	w_{Net} (kJ/kg)	η_{Th} (%)
60	10,000	10	(1.07)	3416.7	2570.1	16.8	863.3	846.5	24.78
70	10,000	10	(1.01)	3419.1	2426.2	14.4	1007.2	992.8	29.04
80	10,000	10	0.95	3420.9	2282.4	12.6	1151.1	1138.5	33.28
90	10,000	10	0.89	3422.3	2138.7	11.2	1294.7	1283.5	37.50
100	10,000	10	0.83	3423.4	1994.9	10.1	1438.6	1428.5	41.73

Cogeneration Systems (1 of 3)

- ►Are integrated systems that simultaneously yield two valuable products, electricity and steam (or hot water) from a single fuel input.
- ►Typically provide cost savings relative to producing power and steam (or hot water) in separate systems.
- ►Are widely deployed in industrial plants, refineries, food processing plants, and other facilities requiring process steam, hot water, and electricity.
- ►Can be based on vapor power plants, gas turbine power plants, internal combustion engines, and fuel cells.

Cogeneration Systems (2 of 3)

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

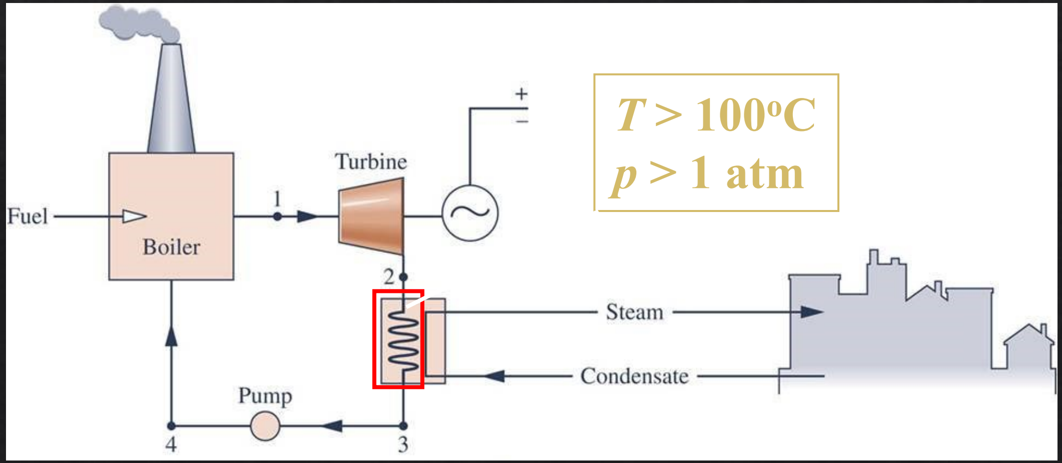


Cogeneration Systems (3 of 3)

► Exporting useful steam to the community limits the electricity that also can be provided from a given fuel input, however.

► For instance, to produce saturated vapor at 100°C (1 atm) for export to the community water circulating through the power plant will condense at a higher temperature and thus at a higher pressure.

► In such an operating mode thermal efficiency is less than when condensation occurs at a pressure below 1 atm, as in a plant fully dedicated to power production.



Exergy

Exergy, also called free energy or available energy, is that portion of any transferred energy either to or from the system that is available to perform thermodynamic work.

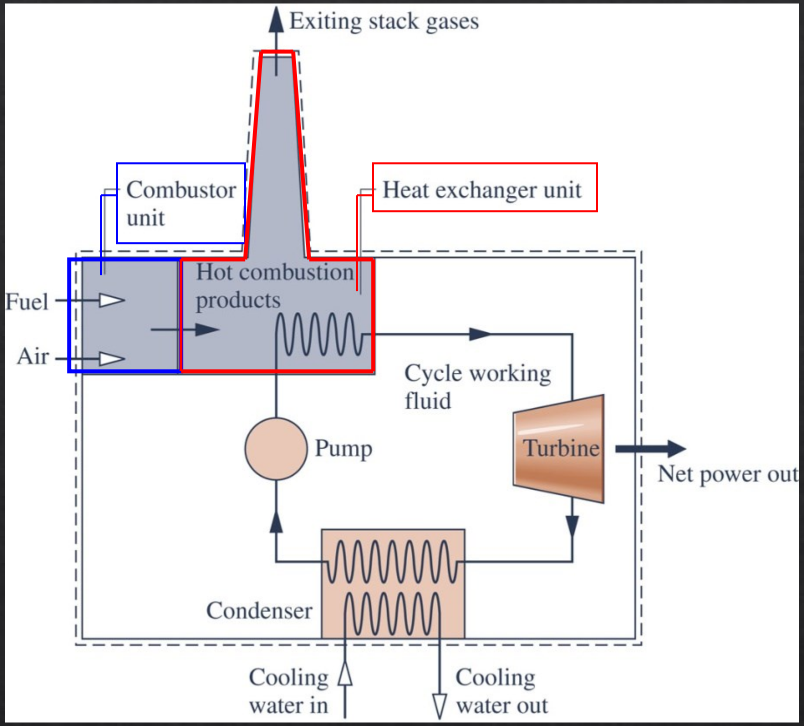
Exergy is that portion of energy that can be converted into useful work.

- Energy is neither created nor destroyed during a process. Energy changes from one form to another (First Law of Thermodynamics).
- In contrast, exergy is always destroyed when a process is irreversible, for example loss of heat to the environment (Second Law of Thermodynamics).
- This destruction is proportional to the entropy increase of the system together with its surroundings (Entropy production).
- The destroyed exergy is called anergy. For an isentropic process, exergy and energy are interchangeable terms, and there is no anergy.

Exergy Accounting of a Vapor Power Plant (1 of 3)

► Exergy analysis, provides insights about vapor power cycle performance beyond what is achievable using conservation of mass and conservation of energy principles.

► The value added by an exergy analysis is demonstrated by application to the vapor power plant shown in the figure.



► For simplicity, the boiler is assumed to have two units: a combustor where fuel and air burn to produce hot combustion gases, followed by a heat exchanger where water circulating through the closed loop of the Rankine cycle is vaporized by the hot gases.

Exergy Accounting of a Vapor Power Plant (2 of 3)

- ►An exergy accounting of the vapor plant is provided in the table.
- ►Table values are determined in Examples 8.7 through 8.9 or assumed for purposes of illustration (stack gas loss, exergy destruction in combustor).

TABLE 8.4 Vapor Power Plant Exergy Accounting[a]

Outputs	
Net power out[b]	30%
Losses	
Condenser cooling water[c]	1%
Stack gases (assumed)	1%
Exergy destruction	
Boiler	
Combustion unit (assumed)	30%
Heat exchanger unit[d]	30%
Turbine[e]	5%
Pump[f]	—
Condenser[g]	3%
Total	100%

[a]All values are expressed as a percentage of the exergy carried into the plant with the fuel. Values are rounded to the nearest full percent. Exergy losses associated with stray heat transfer from plant components are ignored.

[b]Example 8.8.
[c]Example 8.9.
[d]Example 8.7.
[e]Example 8.8.
[f]Example 8.8.
[g]Example 8.9.

Source: Engineering Thermodynamics by Michael J. Moran, 9th Edition

Exergy Accounting of a Vapor Power Plant (3 of 3)

► The table provides several insights about vapor power cycle performance.

- The net power developed is just 30% of the exergy entering with the fuel.
- The exergetic efficiency of the plant is thus 30%.
- Exergy destructions are far more significant than exergy losses.
- The largest portion of the exergy entering with the fuel is destroyed.
- The boiler is the site of greatest exergy destruction.
- Exergy loss via condenser cooling water and stack gases are each minor.

TABLE 8.4 Vapor Power Plant Exergy Accounting[a]

Outputs	
Net power out[b]	30%
Losses	
Condenser cooling water[c]	1%
Stack gases (assumed)	1%
Exergy destruction	
Boiler	
Combustion unit (assumed)	30%
Heat exchanger unit[d]	30%
Turbine[e]	5%
Pump[f]	—
Condenser[g]	3%
Total	100%

[a]All values are expressed as a percentage of the exergy carried into the plant with the fuel. Values are rounded to the nearest full percent. Exergy losses associated with stray heat transfer from plant components are ignored.

[b]Example 8.8.
[c]Example 8.9.
[d]Example 8.7.
[e]Example 8.8.
[f]Example 8.8.
[g]Example 8.9.

End of Lecture!