

Chapter 12

Vapor Engines

12.1 Boiler Exhaust Regeneration

The discussion of losses in combustion driven systems in the last chapter has shown that regeneration, i.e., use of exhaust energy by means of heat exchange within the system, can yield dramatic improvement of engine efficiency. In direct continuation of the argument, we first discuss regeneration in steam cycles, which rely on external combustion. For this, we need to consider not only the steam cycle, but also its heat source, which is hot combustion air.

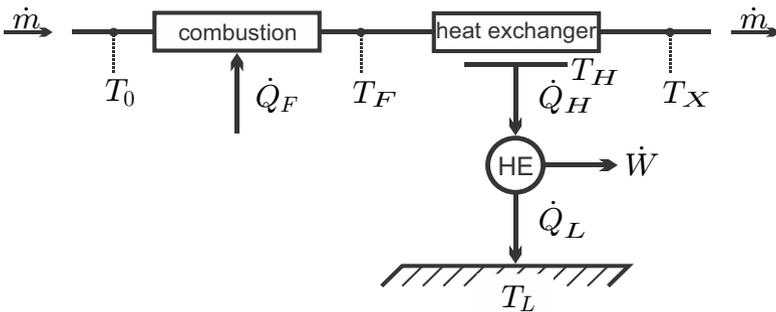


Fig. 12.1 Heat engine driven by external combustion without exhaust regeneration

Figure 12.1 shows a heat engine (HE) which is driven by heat exchange with a hot combustion product. Air at T_0 , flowing at rate \dot{m} , is mixed with fuel and burned so that the combustion product has the temperature T_F . The heat supplied to the air from the combustion is (air standard approximation, i.e., fuel mass ignored)

$$\dot{Q}_F = \dot{m} (h_F - h_0) . \tag{12.1}$$

The hot gas runs through the heat exchanger which it leaves at temperature T_X , so the heat supplied to the heat engine is

$$\dot{Q}_H = \dot{m} (h_F - h_X) . \quad (12.2)$$

Since the exhaust leaves at temperature T_X , the heat

$$\dot{Q}_E = \dot{m} (h_X - h_0) = \dot{Q}_F - \dot{Q}_H \quad (12.3)$$

remains unused; this is just the heat added to the environment when the exhaust equilibrates.

Earlier, we have discussed this set up when the heat engine is a Carnot engine, and have found the exhaust temperature T_X for optimum work output, see Sec. 11.7. The discussion showed that the simplest way to utilize the exhaust heat \dot{Q}_E is regeneration by preheating the air before combustion.

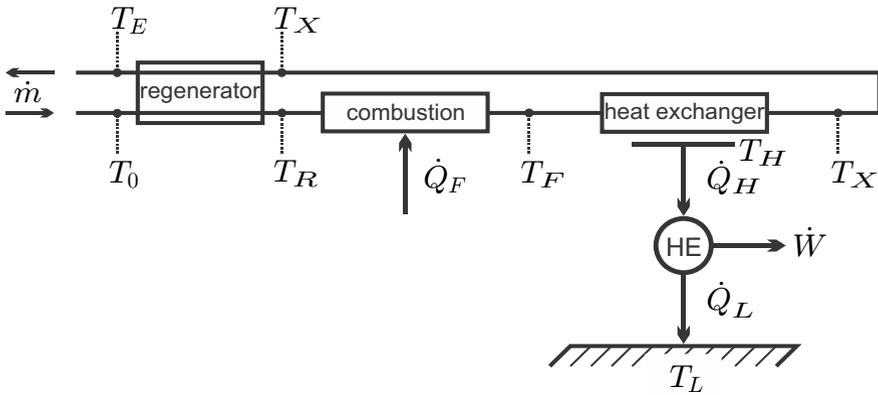


Fig. 12.2 Heat engine driven by external combustion with exhaust regeneration

Figure 12.2 shows the system with an added regenerator for preheating the air. The heat exchange in the regenerator is, from the first law,

$$\dot{Q}_R = \dot{m} (h_R - h_0) = \dot{m} (h_X - h_E) , \quad (12.4)$$

where T_E is the final exhaust temperature, and T_R the preheat temperature. In a perfect regenerator the preheat temperature would be T_X , and the exhaust would leave at T_0 . Accordingly, the regenerator effectiveness is defined as the ratio between the heat *used* for preheating, $(h_R - h_0)$, and the heat *available* for preheating, $(h_X - h_0)$, that is

$$\eta_{reg} = \frac{h_R - h_0}{h_X - h_0} . \quad (12.5)$$

With regenerator, the heat addition from the fuel becomes

$$\dot{Q}_F = \dot{m} (h_F - h_R) = \dot{Q}_H + (1 - \eta_{reg}) \dot{m} (h_X - h_0) , \quad (12.6)$$

where $\dot{Q}_H = \dot{m}(h_F - h_X)$ is the heat supplied to the heat engine as before. Thus, with a perfect regenerator ($\eta_{reg} = 1$), we have $\dot{Q}_H = \dot{Q}_F$, i.e., all the heat provided from the fuel arrives in the engine.

A realistic regenerator has effectiveness of about 80%, and still leads to a much better fuel usage compared to direct exhaust into the environment. It must be noted that for several reasons a somewhat elevated exhaust temperature T_E is beneficial: The combustion of fossil fuels generates water and sulfur oxides; the exhaust temperature must be high enough to avoid water condensation and subsequent formation of sulfuric acid. Also, the combustion air must be moved through the system, either by means of fans, or by natural draught chimneys, which rely on the buoyancy of warm air (Sec. 13.8). Since effective natural draught requires relatively warm exhaust, there is a marked loss. Therefore, modern power plants use fans.

From our previous discussion of heat engines we know that efficiency is high when heat is added at larger temperatures. Thus, for the heat engine one will aim at having the average temperature for heat addition T_H as high as possible. The temperature T_H is limited by the temperature-pressure characteristics of the working fluid and the materials used for construction. The maximum steam temperature in steam cycles using steel pipes in the steam generator is 560 °C. The regenerative steam cycles discussed below aim at raising the average temperature for heat addition, and thus increasing efficiency.

External (to the heat engine) irreversibilities occur in the combustion chamber, and in heat transfer to the heat engine. Our discussion of combustion processes in Chapter 25 will show that combustion irreversibility decreases with increasing flame temperature T_F . On the other hand, heat transfer irreversibility grows with the temperature difference between combustion product (T_F) and the heat engine (T_H). If T_H is limited, as is the case in steam power plants, reduction of T_F decreases heat transfer irreversibility, but increases combustion irreversibility, with the total irreversibility staying relatively constant. Heat is transferred more easily at larger temperature differences, and one can adjust T_F for efficient heat transfer. More efficient use of the fuel is made when the heat engine temperature T_H is increased, as in the combined cycle of Sec. 13.6.

12.2 Regenerative Rankine Cycle

With a high average temperature of the combustion gas in the boiler possible by exhaust regeneration, we now turn to the question of how to raise the average temperature during heat addition in a steam power plant. The basic Rankine cycle was already discussed in Sec. 10.1, where we introduced reheating between turbine stages as one means to this end, Fig. 12.3 repeats the T-s-diagram and the schematic, with air preheater added to the sketch. A higher pressure in the steam generator implies higher average temperature.

However, direct expansion from high pressure into the condenser (3-4') results in unacceptably low values for steam quality at the turbine exit, and turbine blade damage through droplet formation. Reheat at intermediate pressure is used so that the quality at turbine exit is larger, which implies fewer droplet, and low blade damage.

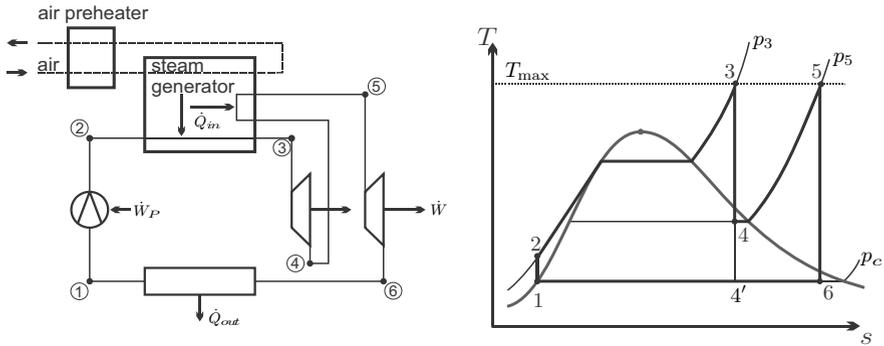


Fig. 12.3 Schematic and T-s-diagram of Rankine cycle with reheat and preheating of combustion air

Heat is added between states 2 and 3, with rather low temperatures in the liquid region right after the pump, the boiler feedwater. In a regenerative Rankine cycle internal heat exchange is used to preheat the feedwater between pump and boiler. For this, some steam is bled-off after the high pressure turbine stage (state 4), and then used to heat the feedwater.

One distinguishes between open feedwater heaters, in which vapor and water are mixed isobarically, and closed feedwater heaters, where vapor and feedwater are running through a heat exchanger at different pressures. In both cases modifications of the cycle are necessary, in order to feed the bled-off flow back into the cycle, and to adjust pressures. Real life power plants employ an array of closed and open feedwater heaters to optimize efficiency and thus increase fuel utilization as much as possible.

12.2.1 Open Feedwater Heater

A reheat steam cycle with a single open feedwater heater is depicted in Fig. 12.4 together with the corresponding T-s-diagram. In the open feedwater heater, compressed liquid water coming from the low pressure pump (P1, state 2) is mixed isobarically with some of the steam leaving the high pressure turbine (T1, state 6). The mixing ratio is adjusted such that the resulting mixture (state 3) is saturated liquid at the mixing pressure $p_3 = p_2 = p_6$. The high pressure pump (P2) compresses this liquid to p_4 before it is fed into the boiler where it is heated, evaporated and superheated to state 5.

$$y = \frac{\dot{m}_B}{\dot{m}_A + \dot{m}_B} = \frac{h_3 - h_2}{h_6 - h_2} . \tag{12.9}$$

Together, the two turbines produce the power

$$\dot{W}_T = (\dot{m}_A + \dot{m}_B) (h_5 - h_6) + \dot{m}_A (h_7 - h_8) > 0 , \tag{12.10}$$

while the pumps consume

$$\dot{W}_P = \dot{m}_A (h_1 - h_2) + (\dot{m}_A + \dot{m}_B) (h_3 - h_4) < 0 . \tag{12.11}$$

Accordingly, the thermal efficiency of the cycle is

$$\eta = \frac{\dot{W}}{\dot{Q}_{in}} = \frac{(1 - y) (h_1 - h_2 + h_7 - h_8) + h_3 - h_4 + h_5 - h_6}{h_5 - h_4 + (1 - y) (h_7 - h_6)} . \tag{12.12}$$

12.2.2 Closed Feedwater Heater

With closed feedwater heaters the streams that exchange heat can be at different pressures, which gives some additional flexibility for process design. The bled-off steam must re-enter the main flow after the feedwater heater, and this can be done either by pumping it into the boiler flow, as depicted in Fig. 12.5, or by throttling it to lower pressure, for instance into the condenser as shown in Fig. 12.6. The latter solution is less efficient, since throttling is highly irreversible.

We consider the case with pump in detail. Steam is bled-off after the high pressure turbine (state 6) and runs through the closed feedwater heater (cfwh)

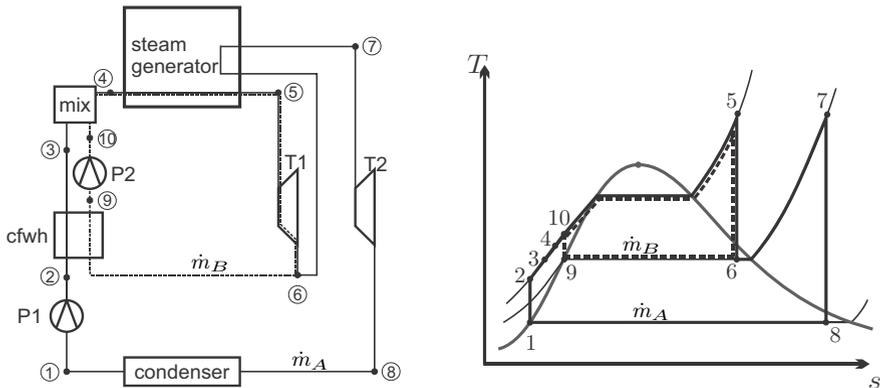


Fig. 12.5 Schematic and T-s diagram for a steam power plant with reheat and one closed feedwater heater (cfwh). Pump 2 only compresses the bled-off flow before it is mixed into the main feedwater flow in the mixing chamber (mix). Combustion air flow is not shown.

where it exchanges heat with the pressurized feedwater, and condenses (state 9). The condensate is brought to the turbine inlet pressure by the second pump (P2, state 10), and then mixed into the main flow to state 4.

Again, the working fluid undergoes two different cycles. The main stream \dot{m}_A runs through the full cycle 1-2-3-4-5-6-7-8-1 (continuous line), while the mass flow \dot{m}_B bled-off after the high pressure turbine runs through the cycle 9-10-4-5-6-9 (dashed line).

The energy balance for the (adiabatic) feedwater heater reads

$$\dot{m}_A (h_3 - h_2) = \dot{m}_B (h_6 - h_9) , \quad (12.13)$$

so that the mass flow ratio is

$$y = \frac{\dot{m}_B}{\dot{m}_A + \dot{m}_B} = \frac{h_3 - h_2}{h_6 - h_9 + h_3 - h_2} . \quad (12.14)$$

Note that the temperature of the main flow after the feedwater heater, T_3 , will be below the temperature of the condensing heating flow, T_9 .

The energy balance for the adiabatic mixing chamber reads

$$\dot{m}_A h_3 + \dot{m}_B h_{10} = (\dot{m}_A + \dot{m}_B) h_4 . \quad (12.15)$$

Heat input, turbine power and pump power are

$$\begin{aligned} \dot{Q}_{in} &= (\dot{m}_A + \dot{m}_B) (h_5 - h_4) + \dot{m}_A (h_7 - h_6) , \\ \dot{W}_T &= (\dot{m}_A + \dot{m}_B) (h_5 - h_6) + \dot{m}_A (h_7 - h_8) > 0 , \\ \dot{W}_P &= \dot{m}_A (h_1 - h_2) + \dot{m}_B (h_9 - h_{10}) < 0 . \end{aligned} \quad (12.16)$$

and thus the thermal efficiency is

$$\eta = \frac{(1 - y) (h_1 - h_2 + h_7 - h_8) + y (h_9 - h_{10}) + (h_5 - h_6)}{h_5 - h_4 + (1 - y) (h_7 - h_6)} . \quad (12.17)$$

If the bled-off flow is simply throttled into the condenser, the total mass flow $\dot{m}_A + \dot{m}_B$ must be pumped up and preheated in the feedwater heater. In this case the energy balance for the feedwater heater is

$$(\dot{m}_A + \dot{m}_B) (h_3 - h_2) = \dot{m}_B (h_5 - h_8) ,$$

which gives the mass flow ratio as

$$y = \frac{\dot{m}_B}{\dot{m}_A + \dot{m}_B} = \frac{h_3 - h_2}{h_5 - h_8} .$$

The thermal efficiency for this cycle can be read from the schematic as

$$\eta_{cwfht} = \frac{h_1 - h_2 + h_4 - h_5 + (1 - y) (h_6 - h_7)}{h_4 - h_3 + (1 - y) (h_6 - h_5)} .$$

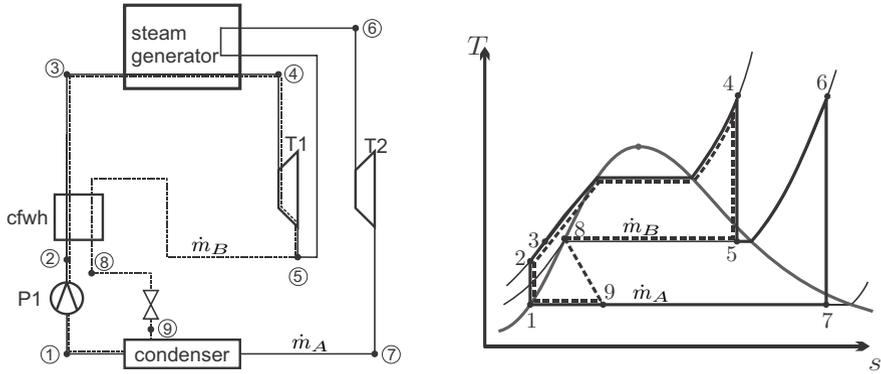


Fig. 12.6 Schematic and T-s diagram for a steam power plant with reheat and one closed feed water heater (cfwh). The bled-off flow is throttled into the condenser.

12.2.3 Several Feedwater Heaters

The example below will give further evidence that regeneration through feedwater heaters yields improvement of thermal efficiency. Already small improvements of thermal efficiency lead to significant savings (or increased profit), and therefore one aims for optimum process configurations. Modern power plants employ arrays of open and closed feedwater heaters operating at various pressures.

Feedwater heaters reduce the external irreversibility in the heat transfer between the combustion gas and the working fluid in the boiler steam generators. However, feedwater heaters add internal irreversibilities, due to heat transfer over finite temperature differences, mixing, and throttling. The subsequent examples will show that the overall irreversibilities of cycles with feedwater heaters are smaller. With multiple feedwater heaters, the internal irreversibilities become smaller, since the temperature differences for heat exchange become smaller. Since irreversibilities imply work loss, plants with multiple feedwater heaters have higher thermal efficiency, due to smaller irreversibilities. While efficiency grows with the number of feedwater heaters, the increase slows down with the number of heaters. Above a certain number of heaters, the small increase of efficiency cannot offset the cost for construction and maintenance, and thus one will limit their number.

The design of a large power plant with multiple feedwater heaters requires optimization of the process to determine the number of feedwater heaters, and the optimal values for the pressures and mass flow rates of the bled-off flows. This multi-parameter optimization is done by means of computer programs.

Figure 12.7 shows a schematic of a 750MW power plant in Germany. The plant employs three turbine stages at high pressure (HP), intermediate pressure (MP) and low pressure (LP), which drive the generator. The power

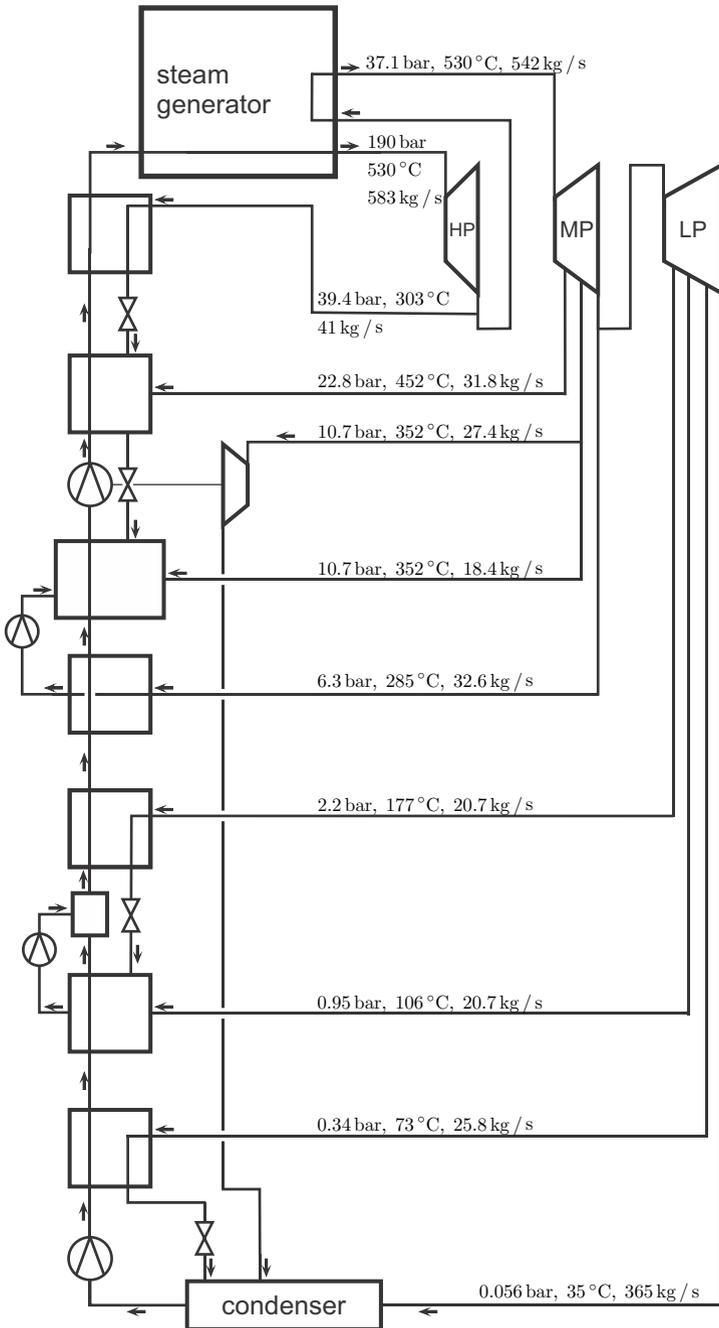


Fig. 12.7 Schematic of 750 MW power plant Bexbach, Germany. The plant has several feed water heaters, and three turbine stages. Note the extra turbine to drive the pump [simplified from H.D. Baehr: Thermodynamik, Springer 1996].

for the feedwater pumps is provided by a smaller turbine. Steam is bled-off from the turbines at a variety of pressures and routed through six feedwater heaters. The condensate leaving the low pressure feedwater heaters 2 and 4 is pumped up into the main flow, while for the other feedwater heaters the condensate is throttled into the lower heaters, or the condenser.

12.3 Example: Steam Cycles with Feedwater Heaters

We consider the thermal efficiencies for the cycles depicted in Figs. 12.3-12.6 with the following base data:

condenser pressure:	7.5 kPa
boiler pressure:	125 bar
reheat pressure:	10 bar
high pressure turbine inlet temperature:	500 °C
low pressure turbine inlet temperature:	500 °C
boiler temperature T_B :	900 K
standard environmental temperature T_0 :	298 K

The boiler temperature will be required to estimate the entropy generation in the heat transfer between boiler and steam cycle. Since all cycles operate between the boiler temperature T_B and the environmental temperature T_0 , their efficiencies must be compared to the corresponding Carnot efficiency $\eta_C = 1 - \frac{T_0}{T_B} = 0.669$.

To simplify the discussion, we shall assume that all pumps and turbines are reversible. Realistic pumps and turbines are irreversible, and add another source of irreversibility to the discussed cycles. Since all pumps and turbines are considered to be adiabatic, their entropy generation rates are $\dot{S}_{gen} = \dot{m}(s_{out} - s_{in})$, where the appropriate mass flow and entropy values must be used. The latter would be obtained from pressure and enthalpy data, based on the isentropic irreversibilities.

It is left to the reader to verify the data and calculations in detail.

12.3.1 No Feedwater Heater

We first examine the basic reheat cycle as shown in Fig. 12.3, for which we find the following property values (point 4 is in the two-phase region)

$$\begin{aligned}
 h_1 &= h_f(7.5 \text{ kPa}) = 168.8 \frac{\text{kJ}}{\text{kg}}, & s_1 &= s_f(7.5 \text{ kPa}) = 0.5764 \frac{\text{kJ}}{\text{kg K}}, \\
 h_2 &= h(125 \text{ bar}, s_1) = 181.4 \frac{\text{kJ}}{\text{kg}}, & s_2 &= s_1, \\
 h_3 &= h(125 \text{ bar}, 500 \text{ °C}) = 3342 \frac{\text{kJ}}{\text{kg}}, & s_3 &= s(125 \text{ bar}, 500 \text{ °C}) = 6.462 \frac{\text{kJ}}{\text{kg K}}, \\
 h_4 &= h(10 \text{ bar}, s_3) = 2722 \frac{\text{kJ}}{\text{kg}}, & s_4 &= s_3, \\
 h_5 &= h(10 \text{ bar}, 500 \text{ °C}) = 3479 \frac{\text{kJ}}{\text{kg}}, & s_5 &= s(10 \text{ bar}, 500 \text{ °C}) = 7.762 \frac{\text{kJ}}{\text{kg K}}, \\
 h_6 &= h(7.5 \text{ kPa}, s_5) = 2421 \frac{\text{kJ}}{\text{kg}}, & s_6 &= s_5.
 \end{aligned}$$

The work per unit mass is

$$\frac{\dot{W}}{\dot{m}} = h_1 - h_2 + h_3 - h_4 + h_5 - h_6 = 1665.4 \frac{\text{kJ}}{\text{kg}},$$

and the heat intake is

$$\frac{\dot{Q}_{in}}{\dot{m}} = h_3 - h_2 + h_5 - h_4 = 3917.6 \frac{\text{kJ}}{\text{kg}},$$

so that the thermal efficiency of this simple reheat cycle is

$$\eta_R = \frac{\dot{W}}{\dot{Q}_{in}} = 0.425.$$

The entropy generation rates for the two heat transfer steps in the boiler follow from the second law as

$$\dot{S}_{gen,23} = \dot{m}(s_3 - s_2) - \frac{\dot{Q}_{23}}{T_B} = \dot{m} \left[s_3 - s_2 - \frac{h_3 - h_2}{T_B} \right],$$

$$\dot{S}_{gen,45} = \dot{m}(s_5 - s_4) - \frac{\dot{Q}_{45}}{T_B} = \dot{m} \left[s_5 - s_4 - \frac{h_5 - h_4}{T_B} \right].$$

The entropy generation in the condenser due to heat transfer to the standard environment at T_0 is

$$\dot{S}_{gen,61} = \dot{m}(s_1 - s_6) - \frac{\dot{Q}_{61}}{T_0} = \dot{m} \left[s_1 - s_6 - \frac{h_1 - h_6}{T_0} \right].$$

The total entropy generation for the system is

$$\dot{S}_{gen} = \dot{S}_{gen,23} + \dot{S}_{gen,45} + \dot{S}_{gen,61},$$

and from the data we find the entropy generation per unit as

$$\frac{\dot{S}_{gen}}{\dot{m}} = 3.20 \frac{\text{kJ}}{\text{kg K}}.$$

As a measure for the associated work loss, we consider the standard work loss $T_0 \dot{S}_{gen}$ relative to the power output:

$$\frac{\dot{W}_{loss}}{\dot{W}} = \frac{T_0 \dot{S}_{gen}}{\dot{W}} = 57\%.$$

12.3.2 Open Feedwater Heater

We next examine the reheat cycle with open feedwater heater as shown in Fig. 12.4, for which we find

$$\begin{aligned}
 h_1 &= h_f(7.5 \text{ kPa}) = 168.8 \frac{\text{kJ}}{\text{kg}} & , & \quad s_1 = s_f(7.5 \text{ kPa}) = 0.5764 \frac{\text{kJ}}{\text{kg K}} , \\
 h_2 &= h(10 \text{ bar}, s_1) = 169.9 \frac{\text{kJ}}{\text{kg}} & , & \quad s_2 = s_1 , \\
 h_3 &= h_f(10 \text{ bar}) = 762.8 \frac{\text{kJ}}{\text{kg}} & , & \quad s_3 = s_f(10 \text{ bar}) = 2.139 \frac{\text{kJ}}{\text{kg K}} , \\
 h_4 &= h(125 \text{ bar}, s_3) = 775.8 \frac{\text{kJ}}{\text{kg}} & , & \quad s_4 = s_3 , \\
 h_5 &= h(125 \text{ bar}, 500^\circ\text{C}) = 3342 \frac{\text{kJ}}{\text{kg}} & , & \quad s_5 = s(10 \text{ bar}, 500^\circ\text{C}) = 6.462 \frac{\text{kJ}}{\text{kg K}} , \\
 h_6 &= h(10 \text{ bar}, s_5) = 2722 \frac{\text{kJ}}{\text{kg}} & , & \quad s_6 = s_5 , \\
 h_7 &= h(10 \text{ bar}, 500^\circ\text{C}) = 3479 \frac{\text{kJ}}{\text{kg}} & , & \quad s_7 = s(10 \text{ bar}, 500^\circ\text{C}) = 7.762 \frac{\text{kJ}}{\text{kg K}} , \\
 h_8 &= h(7.5 \text{ kPa}, s_5) = 2421 \frac{\text{kJ}}{\text{kg}} & , & \quad s_8 = s_7 .
 \end{aligned}$$

The mass flow ratio is adjusted such that the mixed state (state 3) is saturated liquid. From applying the first law to the open feedwater heater, we find the mass flow ratio

$$y = \frac{\dot{m}_B}{\dot{m}_A + \dot{m}_B} = \frac{h_3 - h_2}{h_6 - h_2} = 0.23 .$$

Net work and heat in per unit mass are

$$\begin{aligned}
 \frac{\dot{W}}{\dot{m}_A + \dot{m}_B} &= (1 - y)(h_1 - h_2 + h_7 - h_8) + h_3 - h_4 + h_5 - h_6 = 1418.4 \frac{\text{kJ}}{\text{kg}} , \\
 \frac{\dot{Q}_{in}}{\dot{m}_A + \dot{m}_B} &= h_5 - h_4 + (1 - y)(h_7 - h_6) = 3147.3 \frac{\text{kJ}}{\text{kg}} ,
 \end{aligned}$$

so that the thermal efficiency of the cycle is

$$\eta_{ofwh} = \frac{\dot{W}}{\dot{Q}_{in}} = 0.451 .$$

Compared to the standard reheat cycle, the single open feedwater heater improves thermal efficiency by 2.6%. The following calculation shows that this improvement is due to decreased irreversible losses.

The entropy generation rates for the two heat transfer steps in the boiler follow from the second law as

$$\begin{aligned}
 \dot{S}_{gen,45} &= (\dot{m}_A + \dot{m}_B) \left[s_5 - s_4 - \frac{h_5 - h_4}{T_B} \right] , \\
 \dot{S}_{gen,67} &= \dot{m}_A \left[s_7 - s_6 - \frac{h_7 - h_6}{T_B} \right] ,
 \end{aligned}$$

and the entropy generation in the condenser is

$$\dot{S}_{gen,81} = \dot{m}_A \left[s_1 - s_8 - \frac{h_1 - h_8}{T_0} \right] .$$

Moreover, entropy is generated in the adiabatic feedwater heater through mixing:

$$\dot{S}_{gen,ofwh} = \dot{m}_A (s_3 - s_2) + \dot{m}_B (s_3 - s_6) .$$

The total entropy generation for the system is

$$\dot{S}_{gen} = \dot{S}_{gen,45} + \dot{S}_{gen,67} + \dot{S}_{gen,81} + \dot{S}_{gen,ofwh} ,$$

with a value per unit mass of

$$\frac{\dot{S}_{gen}}{\dot{m}_A + \dot{m}_B} = 2.3 \frac{\text{kJ}}{\text{kg K}} .$$

Now, the standard work loss $T_0 \dot{S}_{gen}$ relative to the power output is

$$\frac{\dot{W}_{loss}}{\dot{W}} = \frac{T_0 \dot{S}_{gen}}{\dot{W}} = 48.4\% .$$

12.3.3 Closed Feedwater Heater (with Pump)

Now we examine the reheat cycle with closed feedwater heater and pump as shown in Fig. 12.5, where

$$\begin{aligned} h_1 &= h_f(7.5 \text{ kPa}) = 168.8 \frac{\text{kJ}}{\text{kg}} , & s_1 &= s_f(7.5 \text{ kPa}) = 0.5764 \frac{\text{kJ}}{\text{kg K}} , \\ h_2 &= h(125 \text{ bar}, s_1) = 181.4 \frac{\text{kJ}}{\text{kg}} , & s_2 &= s_1 , \\ h_3 &\simeq h_f(T_9) = 762.8 \frac{\text{kJ}}{\text{kg}} , & s_3 &\simeq s_f(T_9) = 2.139 \frac{\text{kJ}}{\text{kg K}} , \\ h_4 &= 765.8 \frac{\text{kJ}}{\text{kg}} , & s_4 &\simeq s_f(T_4) = 2.145 \frac{\text{kJ}}{\text{kg K}} , \\ h_5 &= h(125 \text{ bar}, 500^\circ\text{C}) = 3342 \frac{\text{kJ}}{\text{kg}} , & s_5 &= s(125 \text{ bar}, 500^\circ\text{C}) = 6.462 \frac{\text{kJ}}{\text{kg K}} , \\ h_6 &= h(10 \text{ bar}, s_5) = 2722 \frac{\text{kJ}}{\text{kg}} , & s_6 &= s_5 , \\ h_7 &= h(10 \text{ bar}, 500^\circ\text{C}) = 3479 \frac{\text{kJ}}{\text{kg}} , & s_7 &= s(10 \text{ bar}, 500^\circ\text{C}) = 7.762 \frac{\text{kJ}}{\text{kg K}} , \\ h_8 &= h(7.5 \text{ kPa}, s_7) = 2421 \frac{\text{kJ}}{\text{kg}} , & s_8 &= s_7 , \\ h_9 &= h_f(10 \text{ bar}) = 762.8 \frac{\text{kJ}}{\text{kg}} , & s_9 &= s_f(10 \text{ bar}) = 2.139 \frac{\text{kJ}}{\text{kg K}} , \\ h_{10} &= h(125 \text{ bar}, s_9) = 775.8 \frac{\text{kJ}}{\text{kg}} , & s_{10} &= s_9 . \end{aligned}$$

State 3 is chosen by assuming perfect heat exchange in the feedwater heater, so that $T_3 = T_9 = T_{sat}(10 \text{ bar}) = 179.9^\circ\text{C}$. With that, the mass flow ratio is

$$y = \frac{\dot{m}_B}{\dot{m}_A + \dot{m}_B} = \frac{h_3 - h_2}{h_6 - h_9 + h_3 - h_2} = 0.229 .$$

The boiler feed state (state 4) has the enthalpy

$$h_4 = (1 - y) h_3 + y h_{10} .$$

Net work and heat in per unit mass are

$$\begin{aligned} \frac{\dot{W}}{\dot{m}_A + \dot{m}_B} &= (1 - y) (h_1 - h_2 + h_7 - h_8) + y (h_9 - h_{10}) + (h_5 - h_6) \\ &= 1423.2 \frac{\text{kJ}}{\text{kg}} , \end{aligned}$$

$$\frac{\dot{Q}_{in}}{\dot{m}_A + \dot{m}_B} = h_5 - h_4 + (1 - y) (h_7 - h_6) = 3160.0 \frac{\text{kJ}}{\text{kg}} ,$$

so that the thermal efficiency of the cycle is

$$\eta_{cfwh,p} = \frac{\dot{W}}{\dot{Q}_{in}} = 0.450 .$$

Compared to the standard reheat cycle, the single closed feedwater heater improves thermal efficiency by 2.5%, slightly below the cycle with open feedwater heater.

The entropy generation rates for the two heat transfer steps in the boiler follow from the second law as

$$\begin{aligned} \dot{S}_{gen,45} &= (\dot{m}_A + \dot{m}_B) \left[s_5 - s_4 - \frac{h_5 - h_4}{T_B} \right] , \\ \dot{S}_{gen,67} &= \dot{m}_A \left[s_7 - s_6 - \frac{h_7 - h_6}{T_B} \right] , \end{aligned}$$

and the entropy generation in the condenser is

$$\dot{S}_{gen,81} = \dot{m}_A \left[s_1 - s_8 - \frac{h_1 - h_8}{T_0} \right] .$$

Moreover, entropy is generated in the feedwater heater and mixing chamber:

$$\dot{S}_{gen,cfwh} = \dot{m}_A (s_4 - s_2) + \dot{m}_B (s_4 - s_6) .$$

The total entropy generation for the system is

$$\dot{S}_{gen} = \dot{S}_{gen,45} + \dot{S}_{gen,67} + \dot{S}_{gen,81} + \dot{S}_{gen,cfwh}$$

and its value per unit mass is

$$\frac{\dot{S}_{gen}}{\dot{m}_A + \dot{m}_B} = 2.32 \frac{\text{kJ}}{\text{kg K}} .$$

As a measure for the associated work loss, we consider the standard work loss $T_0 \dot{S}_{gen}$ relative to the power output:

$$\frac{\dot{W}_{loss}}{\dot{W}} = \frac{T_0 \dot{S}_{gen}}{\dot{W}} = 48.5\% .$$

12.3.4 Closed Feedwater Heater (with Throttle)

Finally we examine the reheat cycle with closed feedwater heater and throttling into the condenser as shown in Fig. 12.6, for which we find

$$\begin{aligned} h_1 &= h_f(7.5 \text{ kPa}) = 168.8 \frac{\text{kJ}}{\text{kg}} & , & \quad s_1 = s_f(7.5 \text{ kPa}) = 0.5764 \frac{\text{kJ}}{\text{kg K}} , \\ h_2 &= h(125 \text{ bar}, s_1) = 181.4 \frac{\text{kJ}}{\text{kg}} & , & \quad s_2 = s_1 , \\ h_3 &\simeq h_f(T_8) = 762.8 \frac{\text{kJ}}{\text{kg}} & , & \quad s_3 \simeq s_f(T_8) = 2.139 \frac{\text{kJ}}{\text{kg K}} , \\ h_4 &= h(125 \text{ bar}, 500^\circ\text{C}) = 3342 \frac{\text{kJ}}{\text{kg}} & , & \quad s_4 = s(125 \text{ bar}, 500^\circ\text{C}) = 6.462 \frac{\text{kJ}}{\text{kg K}} , \\ h_5 &= h(10 \text{ bar}, s_4) = 2722 \frac{\text{kJ}}{\text{kg}} & , & \quad s_5 = s_4 , \\ h_6 &= h(10 \text{ bar}, 500^\circ\text{C}) = 3479 \frac{\text{kJ}}{\text{kg}} & , & \quad s_6 = s(10 \text{ bar}, 500^\circ\text{C}) = 7.762 \frac{\text{kJ}}{\text{kg K}} , \\ h_7 &= h(7.5 \text{ kPa}, s_6) = 2421 \frac{\text{kJ}}{\text{kg}} & , & \quad s_7 = s_6 , \\ h_8 &= h_f(10 \text{ bar}) = 762.8 \frac{\text{kJ}}{\text{kg}} & , & \quad s_8 = s_f(10 \text{ bar}) = 2.139 \frac{\text{kJ}}{\text{kg K}} , \\ h_9 &= h_8 & , & \quad s_9 = s(7.5 \text{ kPa}, h_9) = 2.471 \frac{\text{kJ}}{\text{kg K}} . \end{aligned}$$

State 3 is chosen by assuming perfect heat exchange in the feedwater heater, so that $T_3 = T_8 = T_{sat}(10 \text{ bar}) = 179.9^\circ\text{C}$. The energy balance for the feedwater heater now is

$$(\dot{m}_A + \dot{m}_B)(h_3 - h_2) = \dot{m}_B(h_5 - h_8) .$$

With that, the mass flow ratio is

$$y = \frac{\dot{m}_B}{\dot{m}_A + \dot{m}_B} = \frac{h_3 - h_2}{h_5 - h_8} = 0.297 .$$

Net work and heat in per unit mass are

$$\begin{aligned} \frac{\dot{W}}{\dot{m}_A + \dot{m}_B} &= h_1 - h_2 + h_4 - h_5 + (1 - y)(h_6 - h_7) = 1351.4 \frac{\text{kJ}}{\text{kg}} , \\ \frac{\dot{Q}_{in}}{\dot{m}_A + \dot{m}_B} &= h_4 - h_3 + (1 - y)(h_6 - h_5) = 3111.6 \frac{\text{kJ}}{\text{kg}} , \end{aligned}$$

so that the thermal efficiency of the cycle is

$$\eta_{cwfth.th} = \frac{\dot{W}}{\dot{Q}_{in}} = 43.4\% .$$

Compared to the standard reheat cycle, the single closed feedwater with throttling into the condenser improves thermal efficiency by only $\sim 1\%$. The improvement is significantly lower than for the case where the bled-off flow is pumped to boiler pressure, since a considerable amount of work is lost in the irreversible expansion through the throttling valve.

The entropy generation rates for the two heat transfer steps in the boiler follow from the second law as

$$\begin{aligned}\dot{S}_{gen,34} &= (\dot{m}_A + \dot{m}_B) \left[s_4 - s_3 - \frac{h_4 - h_3}{T_B} \right], \\ \dot{S}_{gen,56} &= \dot{m}_A \left[s_6 - s_5 - \frac{h_6 - h_5}{T_B} \right],\end{aligned}$$

and the entropy generation in the condenser is

$$\dot{S}_{gen,cond} = \dot{m}_A \left[s_1 - s_7 - \frac{h_1 - h_7}{T_0} \right] + \dot{m}_B \left[s_1 - s_9 - \frac{h_1 - h_9}{T_0} \right].$$

Moreover, entropy is generated in the feedwater heater and in throttling:

$$\begin{aligned}\dot{S}_{gen,cfwh} &= (\dot{m}_A + \dot{m}_B) (s_3 - s_2) + \dot{m}_B (s_8 - s_5), \\ \dot{S}_{gen,th} &= \dot{m}_B (s_9 - s_8).\end{aligned}$$

The total entropy generation is

$$\dot{S}_{gen} = \dot{S}_{gen,34} + \dot{S}_{gen,45} + \dot{S}_{gen,cond} + \dot{S}_{gen,cfwh} + \dot{S}_{gen,th}$$

and its value per unit mass is

$$\frac{\dot{S}_{gen}}{\dot{m}_A + \dot{m}_B} = 2.45 \frac{\text{kJ}}{\text{kg K}}.$$

As a measure for the associated work loss, we consider the standard work loss $T_0 \dot{S}_{gen}$ relative to the power output:

$$\frac{\dot{W}_{loss}}{\dot{W}} = \frac{T_0 \dot{S}_{gen}}{\dot{W}} = 54.0\%.$$

12.3.5 Summary

The analysis of the four configurations, one without and three with feedwater heaters, shows that incorporation of feedwater heaters improves thermal efficiency of steam cycles. Due to the feedwater heaters, the average temperature for heat transfer between the working fluid and the boiler is smaller, hence there is less irreversibility for the boiler processes. All feedwater heaters are associated with irreversibilities through heat exchange over finite temperature

difference, mixing, or throttling. The reduction of irreversibility in the boiler processes is larger than the additional irreversibility associated with the feedwater heaters, so that overall the irreversibility is reduced, and the process performance is improved. For this example, the improvement through open feedwater heater and closed feedwater heater with pump is similar. The system with open feedwater heater and throttling into the condenser yields a smaller improvement, due to the additional irreversibility in the throttling process; however, this system is cheaper to build and maintain. With more feedwater heaters employed, the temperature differences for heat transfer, and the pressure differences for throttling decrease, and thus irreversible losses are reduced.

Finally, it must be noted that losses occurring in irreversible pumps and turbines were ignored, to not overwhelm the computations with more detail.

12.4 Cogeneration Plants

12.4.1 *Process Heat*

Many industries require process heat and electrical power, for instance the chemical industry, pulp and paper plants, refineries, textile production, etc. Cogeneration power plant are variants of the Rankine cycle, which produce power and provide heat at the required temperature level. Adjustments in the process allow to provide variable amounts of process heat, depending on demand. If excess electrical power is produced, this can be sold to the grid, if more electrical power is needed, it must be purchased from the grid.

As an example we study a cogeneration plant that produces power and steam at intermediate pressures and temperatures (say, 5 – 7 bar, 150 – 200 °C). Depending on the demand for process heat, steam can be routed into the process heater either directly from the steam generator (with throttling to lower the pressure), or it can be drawn out of the turbine at intermediate pressure, see Fig. 12.8.

In normal mode, all generated steam is supplied to the turbine, and only a portion of the steam is extracted from the turbine, that is

$$\dot{m}_6 = 0 \quad , \quad \dot{m}_5 \neq 0 \quad , \quad \dot{m}_4 \neq 0 \quad .$$

If the demand for process heat is larger, all steam is taken from the turbine, so that

$$\dot{m}_6 = 0 \quad , \quad \dot{m}_5 \neq 0 \quad , \quad \dot{m}_4 = 0 \quad .$$

This decreases the power generation. Only in case of extremely high demand one will bypass the turbine entirely, so that

$$\dot{m}_6 \neq 0 \quad , \quad \dot{m}_5 = 0 \quad , \quad \dot{m}_4 = 0 \quad .$$

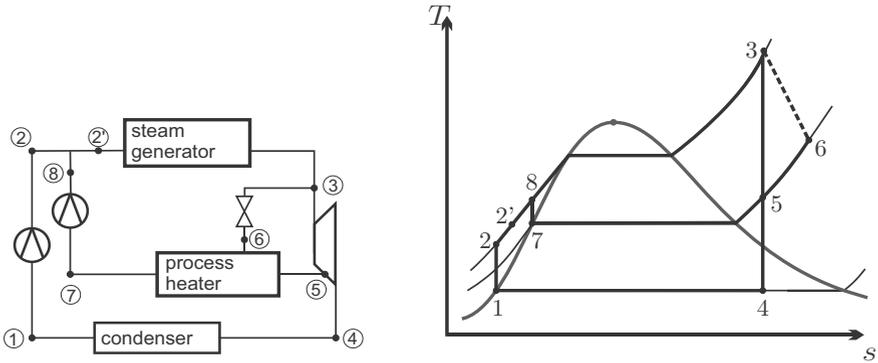


Fig. 12.8 Cogeneration power plant. The plant generates power and process heat. Depending on demand, steam for the process heater is supplied directly from the turbine feed flow (State 3), or extracted from the turbine (State 5).

In this case, only process heat is provided, no electrical power is generated.

From the diagram, we read off the pump and turbine work as

$$\begin{aligned} \dot{W}_P &= \dot{m}_1 (h_1 - h_2) + (\dot{m}_5 + \dot{m}_6) (h_7 - h_8) , \\ \dot{W}_T &= \dot{m}_1 (h_3 - h_4) + \dot{m}_5 (h_3 - h_5) , \end{aligned}$$

and the net work of the plant is

$$\dot{W}_{net} = \dot{W}_P + \dot{W}_T .$$

The process heat is

$$\left| \dot{Q}_{proc} \right| = \dot{m}_5 (h_5 - h_7) + \dot{m}_6 (h_6 - h_7) ,$$

and the heat addition in the steam generator is

$$\dot{Q}_{in} = \dot{m}_1 (h_3 - h_2) + (\dot{m}_5 + \dot{m}_6) (h_3 - h_8) = (\dot{m}_1 + \dot{m}_5 + \dot{m}_6) (h_3 - h_2) .$$

The so-called utilization factor of the plant is

$$\varepsilon_u = \frac{\dot{W}_{net} + \left| \dot{Q}_{proc} \right|}{\dot{Q}_{in}} .$$

Note that the utilization factor favors use of process heat: For the case where all heat is used in the process ($\dot{m}_5 = \dot{m}_6 = 0$), the utilization factor is unity. However, for the case where no process heat is used ($\dot{m}_5 = \dot{m}_6 = 0$), the utilization factor is equal to the thermal efficiency of the plant. Hence, as

with all efficiency measures, one has to be careful in the interpretation of the utilization factor.

12.4.2 *District Heating*

Since heat cannot be completely converted to work, a power plant has to reject heat, typically to the environment. To maximize the work output, the temperature for heat rejection should be as close as possible to the local environmental temperature T_0 . Heat transferred at environmental temperature has no further use, and thus has no value. For space heating in winter, one needs heat at somewhat elevated temperature. This heat can be provided from the heat rejection of power plants in district heating systems. For this, at times when space heating is required, the plant is operated at elevated condenser temperature. The condenser heat is rejected into a water circuit that then is used to bring the heat into buildings. In summer, when no heating is required, the condenser operates at lower temperatures, and the heat is rejected directly to the environment. District heating works best when the distance for heat transmission is short, so that little heat is lost to the environment in transmission. Hence, for combined heat and power plants, it is best to build smaller local power stations, which provide power and heat for the closer neighborhood, rather than large plants far from consumers. Utilization factors for combined heat and power plants can be defined similar to those for process heating.

12.5 Refrigeration Systems

We return to the discussion of vapor refrigeration and heat pump cycles. The basic cycle, consisting of compressor, condenser, throttle, and evaporator was discussed in 10.3. Refrigerators and heat pumps draw heat from a cold environment (T_L) and reject heat into a warm environment (T_H). This transfer of heat from cold to warm requires work, which in vapor and gas refrigeration systems is supplied to the compressor. Efficient refrigeration and heat pump cycles requires: (a) small temperature differences between the working fluid and the respective environments, and (b) efficient compressors.

Typically, compressors are adiabatic, which leads to relatively high temperature of the compressed vapor, and therefore to large temperature differences between the working gas and the warm environment. This implies large entropy generation in heat transfer, hence more work is required to overcome irreversibilities. Intercooling during compression lowers the work requirement for compressors (see also Sec. 13.3). Intercooling also leads to lower temperatures after compression, which reduces the work loss to irreversibilities. There are several ways to incorporate intercooling into refrigeration cycles, which we now discuss.

Multi-stage refrigeration systems are particularly useful when one has to bridge a large temperature difference between the cold and warm environment, as, e.g., in the production of frozen food. Heat pumps normally operate on smaller temperature differences, and multi-stage compression is not used.

12.5.1 Cascade Refrigeration System

In a cascade refrigeration system two or more refrigeration cycles operate on top of each other. Figure 12.9 shows an example with two cycles *A* and *B* connected by a closed heat exchanger, which serves as evaporator for cycle *A*, and as condenser for cycle *B*.

With the closed heat exchanger, the working fluids of the two cycles remain separated. Thus, the two cycles can employ different cooling fluids, and operate at different pressures; cooling fluids can be chosen with the best temperature-pressure characteristics in the respective temperature ranges. For simplicity, the T-s-diagram is drawn for the case that both cycles use the same working fluid, and for the case of reversible compressors.

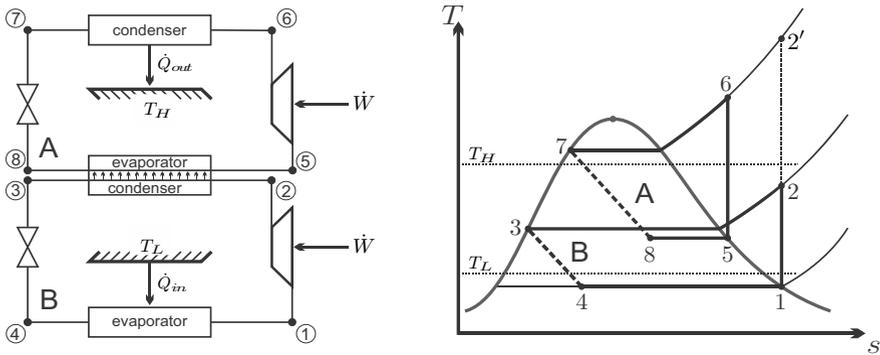


Fig. 12.9 Schematic and T-s-diagram for a cascade refrigeration system: A closed heat exchanger between two cycles serves as condenser for the low pressure cycle and as evaporator for the high pressure cycle

The energy balance for the heat exchanger relates the mass flows of both cycles,

$$\dot{m}_A (h_5 - h_8) = \dot{m}_B (h_2 - h_3) \implies \frac{\dot{m}_A}{\dot{m}_B} = \frac{h_2 - h_3}{h_5 - h_8} . \quad (12.18)$$

The cooling power of the cycle is

$$\dot{Q}_{in} = \dot{m}_B (h_1 - h_4) , \quad (12.19)$$

and the two compressors consume the power

For air liquefaction, the process runs as shown in Fig. 12.11: Air at environmental conditions ($T_1 = 298 \text{ K}$, $p_1 = 1 \text{ bar}$) is compressed in a multi-stage compressor with intercooling to a high pressure $p_2 \simeq 200 \text{ bar}$. Intercooling is necessary to reduce the work for compression as much as possible; for simplicity the T-s-diagram in the figure shows isothermal compression at $T_2 = T_1$. Next the compressed air is pre-cooled isobarically in the regenerator to a temperature T_3 , and then throttled to p_1 . The temperature T_3 must be sufficiently low, so that state 4 after throttling is in the two-phase region. In a flash chamber the flow is divided into saturated liquid—the desired product P—and saturated vapor (state 5) which is routed through the pre-cooler. Obviously, the conditions for this process are such that air cannot be described as an ideal gas; proper tables for air as superheated vapor and saturated liquid-vapor mix are required.

Due to the use of throttling, the Linde process is inherently irreversible, and thus requires more work than a reversible process for the same task. As we have seen, all refrigeration systems suffer from irreversibilities, and there is little alternative to this process for large scale gas liquefaction.

Problems

12.1. Regenerative Boiler

The boiler for a power plant is fitted with a regenerator of effectiveness 81.2% to preheat the incoming air before it is heated by burning of coal. Specifically, the system draws environmental air at $T_0 = 7^\circ\text{C}$, the flame temperature is 827°C , and the boiler exhaust is at 667°C .

1. Determine the preheat temperature, the final exhaust temperature, the work potential of the boiler exhaust, and the work potential of the final exhaust.
2. Determine the entropy generation in the regenerator, and the corresponding work loss.

12.2. Mixing Chamber

Steam at 100 bar, 600°C is throttled to 10 bar, and fed into an adiabatic mixing chamber where it is mixed with compressed liquid water at 10 bar, 50°C . The exiting mass flow is $100 \frac{\text{kg}}{\text{s}}$ of saturated liquid at 10 bar. For steady state operation, determine

1. The mass flows of steam and liquid that enter.
2. The rate of entropy generation due to throttling, and the rate of entropy generation due to mixing.
3. The associated work loss.

As always: draw a sketch and a T-s-diagram.

12.3. Closed Feedwater Heater

In a closed feedwater heater, a mass flow of 200t/h of compressed liquid water at 80 bar, 40 °C is heated by heat exchange with a stream of saturated liquid-vapor mix at $x = 0.95$, $p = 10$ bar.

For the case that both streams leave the heat exchanger in saturated liquid state determine the mass flow of liquid-vapor mix, the entropy generation rate, and the work loss.

12.4. Steam Power Plant with Regeneration and Reheat

A steam power plant operates on a reheat-regenerative Rankine cycle with an open feedwater heater. Steam enters the high-pressure turbine at 100 bar, 550 °C, and leaves at a pressure of 8 bar as saturated vapor. Some steam is extracted at this pressure to heat the feedwater in an open feedwater heater which provides saturated liquid. The rest of the steam is reheated to 500 °C and then expanded in the low pressure turbine to the condenser pressure of 10 kPa.

The isentropic efficiency of the low pressure turbine is 0.95; all pumps can be considered to operate reversibly.

1. Draw a schematic and a T-s-diagram of the process, numerate corresponding points in schematic and diagram, and name the different devices (pump, turbine, etc.)
2. Make a list with the values of enthalpy at all relevant points of the process.
3. Compute the ratio of mass flow diverted to the feedwater heater after the first turbine.
4. Compute the thermal efficiency of the cycle.
5. The net power output of the plant is 100 MW. Determine the mass flow through the high pressure turbine.
6. Determine total entropy generation rate and work loss of the cycle.

12.5. Vapor Power Plant with Regeneration

A power plant operates on a regenerative vapor power cycle with one closed feedwater heater according to the following process:

- 1-2: Isentropic compression of saturated water from condenser pressure 0.04 bar to 60 bar.
- 2-3: Isobaric heating in the closed feedwater heater to 141.3 °C.
- 3-4: Isobaric heating in the steam generator to 60 bar, 550 °C.
- 4-5: Isentropic expansion into the condenser.

Some steam is extracted from the turbine at 4 bar to heat feedwater in a closed feedwater heater. This part of the steam undergoes the following two processes:

- 6-7: Isobaric cooling and condensation at 4 bar of diverted steam to saturated liquid state.

7-8: Throttling of condensate exiting the feedwater heater into the condenser.

1. Draw a schematic and a T-s-diagram of the process.
2. Make a list of the values of enthalpies at the points 1 to 8.
3. Compute the percentage of mass flow diverted into the feedwater heater at point 6.
4. Determine the thermal efficiency of the cycle.
5. Determine the mass flow rate into the turbine, if the net power developed is 320 MW.
6. Determine total entropy generation rate and work loss of the cycle.

12.6. Steam Power Plant with Two Feedwater Heaters, One Open, One Closed

Consider an ideal steam regenerative Rankine cycle with one open and one closed feedwater heater. Steam enters the turbine at 12.5 MPa, 550 °C, the condenser pressure is 10 kPa. Steam for the closed feedwater heater is extracted from the turbine at 0.8 MPa and for the open feedwater heater at 0.3 MPa. The feedwater is heated to the condensation temperature of the stream for the closed feedwater heater. The extracted steam leaves the closed feedwater heater at saturated state and is throttled into the open feedwater heater.

1. Draw a schematic of the process, and the corresponding T-s-diagram.
2. For a power output of 250 MW determine the mass flow rate through the steam generator, and the mass flows into the feedwater heaters.
3. Determine the thermal efficiency of the cycle.
4. Determine the entropy generation in the throttle, and estimate the corresponding work loss.

12.7. Steam Power Plant with Reheat and Two Feedwater Heaters, One Closed, One Open

The boiler pressure in a reheat steam power plant is 150 bar, the reheat pressure is 14 bar, and the condenser pressure is 10 kPa. For both turbines, the inlet temperature is 500 °C. After the high pressure turbine, some steam is bled-off and routed to the closed feedwater heater where it is fully condensed, and then pumped into the boiler feedwater. The remaining steam is reheated, and then runs through the low pressure turbine. Part of the flow is bled-off from the turbine at a pressure of 4 bar, while the main flow expands into the condenser. The diverted flow is mixed in the open feedwater heater with the flow that is pumped in from the condenser. The resulting mixture, which is in the saturated liquid state, is then pumped to boiler pressure before it enters the closed feedwater heater.

1. Draw a schematic of the process. Use the following numbering of processes:
1-2: Low pressure feedwater pump (from condenser). 3-4: Second feedwater

pump (after open feedwater heater). 6-7: High pressure steam generator. 7-8: High pressure turbine. 8-9: Reheat. 9-11: Low pressure turbine. 10: Bled-off for open feedwater heater. 11-1: Condenser. 12-13: Third feedwater pump.

2. Draw the corresponding T-s-diagram. Use different colors (or different line styles) to show the process curves for the main flow and the two bled-off flows.
3. Determine the enthalpies at all relevant states, based on the following assumptions: Reversible pumps and turbines, exit of open feedwater heater is saturated liquid (state 3), perfect heat exchange in closed feedwater heater, so that $T_5 = T_{12}$.
4. Determine the thermal efficiency of the plant.
5. Determine the three mass flows when the plant delivers a power of 500 MW.
6. Determine the overall entropy generation of the system, and the work loss to irreversibilities.
7. Determine the thermal efficiency of a standard reheat plant with the same pressures and turbines. Explain why the feedwater heaters improve efficiency.

12.8. Steam Power Plant with Reheat and Two Feedwater Heaters, One Closed, One Open

Repeat the previous problem, now considering irreversible pumps (isentropic efficiency $\eta_P = 0.85$) and turbines (isentropic efficiency $\eta_T = 0.92$).

12.9. Steam Power Plant with Reheat and Two Feedwater Heaters, One Open, One Closed

A reheat steam power plant has one closed feedwater heater (c.f.w.h.) and one open feedwater heater (o.f.w.h.). The boiler pressure is 150 bar, the reheat pressure is 15 bar, and the condenser pressure is 10 kPa. For both turbines, the inlet temperature is 500 °C. After the high pressure turbine, some steam is bled-off and routed to the o.f.w.h. The remaining steam is reheated, and then runs through the low pressure turbine. Part of the flow is bled-off from the turbine at a pressure of 5 bar. This flow is further routed through the c.f.w.h., where it fully condenses, and is then pumped into the o.f.w.h. The main flow expands into the condenser, which it leaves as saturated liquid. This flow is pumped to the o.f.w.h. pressure, heated in the c.f.w.h., and then mixed with the other flows in the o.f.w.h. The resulting mixture in the o.f.w.h., which is in the saturated liquid state, is then pumped to boiler pressure.

1. Draw a schematic of the process. Use the following numbering of processes: 1-2: Low pressure feedwater pump (from condenser). 2-3 and 9-11: Closed f.w.h. 4-5: Second feedwater pump (after o.f.w.h.). 5-6: High pressure steam generator. 6-7: High pressure turbine. 7-8: Reheat. 8-10: Low pressure turbine. 9: Bled-off for closed f.w.h. 10-1: Condenser. 11-12: Third feedwater pump.

2. Draw the corresponding T-s-diagram. Use different colors (or different line styles) to show the process curves for the main flow and the two bled-off flows.
3. Determine the enthalpies at all relevant states, based on the following assumptions: Reversible pumps and turbines, exit of o.f.w.h. is saturated liquid (state 4), perfect heat exchange in closed feedwater heater, so that $T_3 = T_{11}$.
4. Determine the relative amounts of the relevant mass flows.
5. Determine the thermal efficiency of the plant.
6. Determine the three mass flows when the plant delivers a power of 500 MW.
7. Determine the overall entropy generation of the system, and the work loss to irreversibilities.
8. Determine the thermal efficiency of a standard reheat plant with the same pressures and turbines. Explain why the feedwater heaters improve efficiency.

12.10. Steam Power Plant with Reheat and Two Feedwater Heaters, One Open, One Closed

Repeat the previous problem, now considering irreversible pumps (isentropic efficiency $\eta_P = 0.85$) and turbines (isentropic efficiency $\eta_T = 0.92$).

12.11. Cogeneration Power Plant

A cogeneration power plant with reheat produces 3 MW of power and supplies 7 MW of process heat. Steam enters the isentropic high-pressure turbine at 8 MPa and 500 °C and expands to a pressure of 1 MPa. At this pressure, part of the steam is extracted from the turbine and routed to the process heater; this stream leaves the process heater as compressed liquid at 120 °C. The remaining steam is reheated to 500 °C and then expanded in the isentropic low-pressure turbine to the condenser pressure of 15 kPa. The condensate is pumped to 1 MPa and then mixed with the stream of compressed liquid that comes from the process heater. The mixture is then pumped to the boiler pressure.

1. Make a schematic of the cycle, and draw the corresponding T-s-diagram.
2. Determine the heat input, the relative amount of steam running through the process heater, and the utilization factor.

12.12. Cogeneration Steam Power Plant with Regeneration

A small power plant that produces 30 MW of power operates on a regenerative vapor power cycle with one closed feedwater heater according to the following process:

Steam of 125 bar, 550 °C (state 1) enters the high pressure turbine where it is expanded isentropically to 10 bar (state 2). 50% of this steam are reheated to 500 °C (state 3) and then expanded in the low pressure turbine to the condenser pressure 0.075 bar (state 4). After condensation to saturated liquid state (state 5) this stream is pumped isentropically to 10 bar (state 6) and

routed into the open feedwater heater. Part of the steam extracted after the high pressure turbine (state 2) is used for process heating. For this, the steam passes through a heat exchanger which it leaves as compressed liquid at 60°C (state 7) that is fed into the open feedwater heater. The rest of the extracted steam of state 2 is directly routed into the feedwater heater. The water leaving the feedwater heater is in the saturated liquid state (state 8); an isentropic pump increases its pressure to the boiler pressure (state 9).

1. Draw a schematic and a T-s-diagram of the process.
2. Make a table with enthalpies at the relevant states of the process.
3. Determine the mass flows through the boiler and the process heater.
4. Determine the utilization factor of the plant.

12.13. District Heating

A 40 MW power plant is to be build to supply electrical power to a small town in the North where, due to the low average temperature, a large amount of space heating is required. One proposal suggest to set the condenser pressure a bit higher, so that the condenser is at temperature T_{C1} , and to use the removed heat for district heating. An alternative proposal suggests to set the condenser to the lower temperature T_{C2} so that the turbine delivers more work, which can then be used to run heat pumps between T_{C1} and T_{C2} . Discuss these proposals and make a recommendation to town council. Use thermodynamic arguments (of course!), it might be helpful to draw pictures with energy flows and temperatures.

12.14. Standard Vapor Cooling Cycle with Ammonia

A standard vapor refrigeration cycle operates with ammonia as cooling fluid. The maximum and minimum pressures reached are 1.5 atm and 10 atm, respectively. The adiabatic compressor draws saturated vapor, and has an isentropic efficiency of 0.9. The ammonia vapor leaving the compressor is cooled, condensed and further cooled to 20°C before it enters the throttling device.

Draw the process into the log p-h diagram for ammonia, and find the enthalpies and temperatures at all principal points. For a cooling power of 2 kW, determine the power consumption and the COP.

12.15. Two-Stage Refrigeration Cycle

1. Draw a schematic, and the corresponding T-s-diagram for a two-stage refrigeration cycle with an open heat exchanger.
2. Indicate all principal points in both diagrams by numbers, and indicate the different elements by name (compressor, throttle, etc.).
3. Compute the mass flow ratio between upper and lower cycle in terms of enthalpies.
4. Give the expression for the COP of the system in terms of enthalpies and mass flow ratio.

Note: The next three problems compare cooling cycles running between the same upper and lower pressures. There is some data overlap, and to simplify proceedings an irreversible compressor is considered only in the first cycle. For all three, start with drawing a sketch, and the T-s-diagram

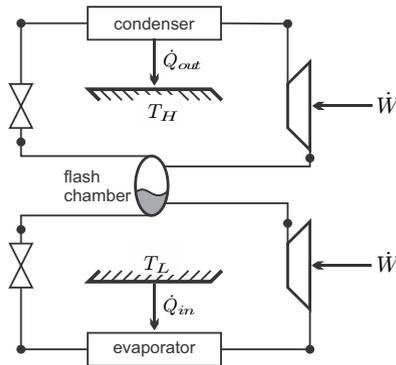
12.16. Standard Vapor Cooling Cycle with R134a

A standard vapor refrigeration cycles operates with R134a as cooling fluid between the pressures 1.2 MPa and 0.1 MPa, respectively. The adiabatic compressor draws saturated vapor, and has an isentropic efficiency of 0.9. The vapor leaving the compressor is cooled and fully condensed before it enters the throttling device. Draw a schematic and the T-s-diagram, and then determine:

1. The COP for the cycle with irreversible and with reversible compressor.
2. The mass flow rate and the power consumption for a cooling power of 200 kW.
3. Determine entropy generation rates for each process, the overall entropy generation rate, and work loss of the cycle. Assume $T_H = 30^\circ\text{C}$ and $T_L = -20^\circ\text{C}$.

12.17. Two-Stage Refrigeration Cycle with Flash Chamber

A two-stage compression refrigeration system operates with R134a between the pressures 1.2 MPa and 0.1 MPa. The refrigerant leaves the condenser as saturated liquid and is throttled to a flash chamber operating at 0.4 MPa. The refrigerant leaving the low-pressure compressor at 0.4 MPa is also routed to the flash chamber.



The saturated vapor leaving the flash chamber is compressed to the condenser pressure by the high-pressure compressor, while the saturated liquid leaving the flash chamber is throttled to the evaporator pressure. The refrigerant leaves the evaporator as saturated vapor and both compressors are isentropic. Draw a schematic and the T-s-diagram, and then determine:

1. The fraction of mass flows running through the two compressors.
2. The COP, and compare to that of the previous problem.
3. The two mass flow rates and the power consumption for a cooling power of 200 kW.
4. Determine entropy generation rates for each process, the overall entropy generation rate, and work loss of the cycle. Assume $T_H = 30^\circ\text{C}$ and $T_L = -20^\circ\text{C}$.

12.18. Two-Stage Refrigeration Cycle with Heat Exchanger

A two-stage cascade refrigeration system operates with R134a between the pressures 1.2 MPa and 0.1 MPa. Heat exchange between the two cycles takes place in an adiabatic counter-flow heat exchanger where the pressures are 0.32 and 0.4 MPa, respectively. In both cycles, the refrigerant is in saturated liquid state at the condenser exit and at saturated vapor state at the compressor inlet. Draw a schematic and the T-s-diagram, and then determine:

1. The fraction of mass flows running through the two compressors.
2. The coefficient of performance.
3. The two mass flow rates, and the power consumption for a cooling power of 200 kW.
4. The total entropy generation rate and work loss of the cycle. Assume $T_H = 30^\circ\text{C}$ and $T_L = -20^\circ\text{C}$.

12.19. Two-Stage Refrigeration Cycles

Repeat the previous two problems for the case where the compressors have an isentropic efficiency of 0.85.

12.20. Refrigeration Cycle with Intercooling

A vapor-compression refrigeration cycle operates at steady state with ammonia as working fluid according to the following cycle:

- 1-2: Adiabatic irreversible compression of saturated vapor at $p_1 = 1.75$ bar to $p_2 = 5$ bar, isentropic compressor efficiency is $\eta_C = 0.8$.
- 2-3: Isobaric cooling to 20°C .
- 3-4: Adiabatic irreversible compression to $p_4 = 12$ bar, isentropic compressor efficiency is $\eta_C = 0.8$.
- 4-5: Isobaric heat rejection in condenser; state 5 is saturated liquid.
- 5-6: Throttling into the evaporator, $p_6 = p_1$.
- 6-1: Isobaric evaporation to state 1.

1. Draw a schematic and plot the process in a T-s-diagram.
2. Find the enthalpies at points 1-6.
3. Determine the coefficient of performance.

12.21. Advanced Cooling Cycle

In a hot climate, a two-stage cascade refrigeration system operates with refrigerant R134a. The evaporator temperature of the low pressure stage is -20°C ,

and the condenser temperature of the high pressure stage is 50°C . Heat exchange between the cycles takes place in a counter-flow heat exchanger where the pressures are 0.4 and 0.5 MPa, respectively. In both cycles, the refrigerant is in saturated liquid state at the condenser exit, and in saturated vapor state at the compressor inlet. The isentropic efficiency of both compressors is 0.8.

1. Draw a T-s-diagram of the cycle with respect to saturation lines.
2. Make a list of the enthalpies and entropies at states 1 through 8.
3. Determine the ratio of mass flows entering the upper and the lower compressor.
4. Determine the COP of the cycle, and the power requirement for a cooling power of 120 kW.
5. Determine entropy generation rates for all processes, the overall entropy generation rate, and the work loss of the cycle. Assume $T_H = 40^\circ\text{C}$ and $T_L = -10^\circ\text{C}$.

12.22. Regenerative Gas Cooling System

A regenerative gas refrigeration cycle uses helium as working fluid. The helium enters the compressor at 100 kPa and -10°C and is compressed to 300 kPa. Then, it is cooled to 20°C by heat exchange with a cooling water flow. Next, the helium enters the regenerator where it is cooled further before it enters the turbine. Helium leaves the refrigerated space at -25°C and enters the regenerator. Assume isentropic operation of turbine and regenerator, and determine

1. The temperature at the turbine inlet.
2. The COP of the cycle.
3. The net power input required for a mass flow rate of $0.45 \frac{\text{kg}}{\text{s}}$.

Helium behaves as an ideal gas; it is monatomic, and thus has constant specific heat, $c_p = \frac{5}{2}R$.