Engineering Thermodynamics

Module -3.3

Compiled by

Prof. Chinmaya Mund

Prof. Dambarudhar patel

Prof. Chandra Sekhar Saran

EXAMPLE 10–4 The Ideal Reheat Rankine Cycle

Consider a steam power plant operating on the ideal reheat Rankine cycle. Steam enters the high-pressure turbine at 15 MPa and 600°C and is condensed in the condenser at a pressure of 10 kPa. If the moisture content of the steam at the exit of the low-pressure turbine is not to exceed 10.4 percent, determine (*a*) the pressure at which the steam should be reheated and (*b*) the thermal efficiency of the cycle. Assume the steam is reheated to the inlet temperature of the high-pressure turbine.

Solution A steam power plant operating on the ideal reheat Rankine cycle is considered. For a specified moisture content at the turbine exit, the reheat pressure and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–13. We note that the power plant operates on the ideal reheat Rankine cycle. Therefore, the pump and the turbines are isentropic, there are no pressure drops in the boiler and condenser, and steam leaves the condenser and enters the pump as saturated liquid at the condenser pressure.

(*a*) The reheat pressure is determined from the requirement that the entropies at states 5 and 6 be the same:

State 6: $P_6 = 10\ \text{kPa}$

$x_6 = 0.896 \quad (\text{sat. mixture})$

$s_6 = s_f + x_6 s_{fg} = 0.6492 + 0.896(7.4996) = 7.3688\ \text{kJ/kg} \cdot \text{K}$

Also,

$$h_6 = h_f + x_6 h_{fg} = 191.81 + 0.896(2392.1) = 2335.1\ \text{kJ/kg}$$

Thus,

State 5: $T_5 = 600°\text{C}$, $s_5 = s_6$ $\Rightarrow$ $P_5 = \mathbf{4.0\ MPa}$, $h_5 = 3674.9\ \text{kJ/kg}$

Therefore, steam should be reheated at a pressure of 4 MPa or lower to prevent a moisture content above 10.4 percent.

(b) To determine the thermal efficiency, we need to know the enthalpies at all other states:

State 1: $P_1 = 10\ \text{kPa}$, Sat. liquid $\Rightarrow$ $h_1 = h_{f@\,10\text{kPa}} = 191.81\ \text{kJ/kg}$, $v_1 = v_{f@\,10\text{kPa}} = 0.00101\ \text{m}^3/\text{kg}$

State 2: $P_2 = 15\ \text{MPa}$, $s_2 = s_1$

$$w_{\text{pump,in}} = v_1(P_2 - P_1) = (0.00101\ \text{m}^3/\text{kg}) \times [(15{,}000 - 10)\text{kPa}]\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = 15.14\ \text{kJ/kg}$$

$$h_2 = h_1 + w_{\text{pump,in}} = (191.81 + 15.14)\ \text{kJ/kg} = 206.95\ \text{kJ/kg}$$

State 3: $P_3 = 15\ \text{MPa}$, $T_3 = 600^\circ\text{C}$ $\Rightarrow$ $h_3 = 3583.1\ \text{kJ/kg}$, $s_3 = 6.6796\ \text{kJ/kg}\cdot\text{K}$

State 4: $P_4 = 4\ \text{MPa}$, $s_4 = s_3$ $\Rightarrow$ $h_4 = 3155.0\ \text{kJ/kg}$ $(T_4 = 375.5^\circ\text{C})$

Thus

$$q_{in} = (h_3 - h_2) + (h_5 - h_4) = (3583.1 - 206.95)\ \text{kJ/kg} + (3674.9 - 3155.0)\ \text{kJ/kg} = 3896.1\ \text{kJ/kg}$$

$$q_{out} = h_6 - h_1 = (2335.1 - 191.81)\ \text{kJ/kg} = 2143.3\ \text{kJ/kg}$$

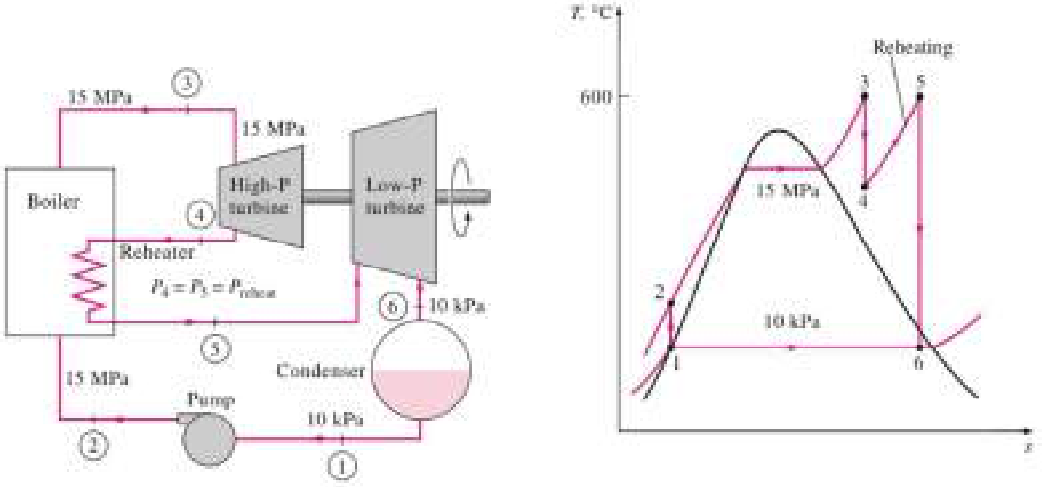


FIGURE 10–13

Schematic and *T-s* diagram for Example 10–4.

and

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{2143.3\ \text{kJ/kg}}{3896.1\ \text{kJ/kg}} = \mathbf{0.450\ or\ 45.0\%}$$

Discussion This problem was solved in Example 10–3*c* for the same pressure and temperature limits but without the reheat process. A comparison of the two results reveals that reheating reduces the moisture content from 19.6 to 10.4 percent while increasing the thermal efficiency from 43.0 to 45.0 percent.

EXAMPLE 10–5 The Ideal Regenerative Rankine Cycle

Consider a steam power plant operating on the ideal regenerative Rankine cycle with one open feedwater heater. Steam enters the turbine at 15 MPa and 600°C and is condensed in the condenser at a pressure of 10 kPa. Some steam leaves the turbine at a pressure of 1.2 MPa and enters the open feedwater heater. Determine the fraction of steam extracted from the turbine and the thermal efficiency of the cycle.

Solution A steam power plant operates on the ideal regenerative Rankine cycle with one open feedwater heater. The fraction of steam extracted from the turbine and the thermal efficiency are to be determined.

Assumptions 1 Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–18. We note that the power plant operates on the ideal regenerative Rankine cycle. Therefore, the pumps and the turbines are isentropic; there are no pressure drops in the boiler, condenser, and feedwater heater; and steam leaves the condenser and the feedwater heater as saturated liquid. First, we determine the enthalpies at various states:

State 1: $P_1 = 10$ kPa, Sat. liquid $\Rightarrow$ $h_1 = h_{f\,@\,10\,kPa} = 191.81$ kJ/kg, $v_1 = v_{f\,@\,10\,kPa} = 0.00101$ m³/kg

State 2: $P_2 = 1.2$ MPa

$$s_2 = s_1$$

$$w_{pump\,I,in} = v_1(P_2 - P_1) = (0.00101\ \text{m}^3/\text{kg})[(1200 - 10)\ \text{kPa}]\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)$$

$$= 1.20\ \text{kJ/kg}$$

$$h_2 = h_1 + w_{pump\,I,in} = (191.81 + 1.20)\ \text{kJ/kg} = 193.01\ \text{kJ/kg}$$

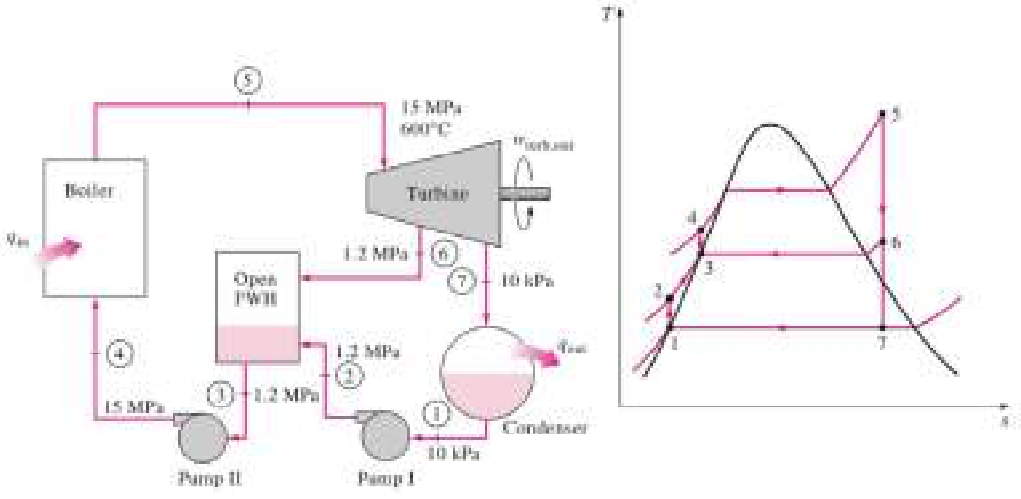


FIGURE 10–18

Schematic and *T-s* diagram for Example 10–5.

State 3: $P_3 = 1.2\ \text{MPa}$, Sat. liquid $\Rightarrow$ $v_3 = v_{f\,@\,1.2\,\text{MPa}} = 0.001138\ \text{m}^3/\text{kg}$, $h_3 = h_{f\,@\,1.2\,\text{MPa}} = 798.33\ \text{kJ/kg}$

State 4: $P_4 = 15\ \text{MPa}$

$$s_4 = s_3$$

$$w_{\text{pump II,in}} = v_3(P_4 - P_3)$$

$$= (0.001138\ \text{m}^3/\text{kg})[(15{,}000 - 1200)\ \text{kPa}]\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)$$

$$= 15.70\ \text{kJ/kg}$$

$$h_4 = h_3 + w_{\text{pump II,in}} = (798.33 + 15.70)\ \text{kJ/kg} = 814.03\ \text{kJ/kg}$$

State 5: $P_5 = 15\ \text{MPa}$, $T_5 = 600°\text{C}$ $\Rightarrow$ $h_5 = 3583.1\ \text{kJ/kg}$, $s_5 = 6.6796\ \text{kJ/kg}\cdot\text{K}$

State 6: $P_6 = 1.2\ \text{MPa}$, $s_6 = s_5$ $\Rightarrow$ $h_6 = 2860.2\ \text{kJ/kg}$, $(T_6 = 218.4°\text{C})$

State 7: $P_7 = 10\ \text{kPa}$

$$s_7 = s_3 \qquad x_7 = \frac{s_7 - s_f}{s_{fg}} = \frac{6.6796 - 0.6492}{7.4996} = 0.8041$$

$$h_7 = h_f + x_7 h_{fg} = 191.81 + 0.8041(2392.1) = 2115.3\ \text{kJ/kg}$$

The energy analysis of open feedwater heaters is identical to the energy analysis of mixing chambers. The feedwater heaters are generally well insulated ($\dot{Q} = 0$), and they do not involve any work interactions ($\dot{W} = 0$). By neglecting the kinetic and potential energies of the streams, the energy balance reduces for a feedwater heater to

$$\dot{E}_{in} = \dot{E}_{out} \rightarrow \sum_{in} \dot{m}h = \sum_{out} \dot{m}h$$

or

$$yh_6 + (1 - y)h_2 = 1(h_3)$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_6/\dot{m}_5$). Solving for y and substituting the enthalpy values, we find

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{798.33 - 193.01}{2860.2 - 193.01} = \mathbf{0.2270}$$

Thus,

$$q_{in} = h_5 - h_4 = (3583.1 - 814.03)\ \text{kJ/kg} = 2769.1\ \text{kJ/kg}$$

$$q_{out} = (1 - y)(h_7 - h_1) = (1 - 0.2270)(2115.3 - 191.81)\ \text{kJ/kg}$$

$$= 1486.9\ \text{kJ/kg}$$

and

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{1486.9\ \text{kJ/kg}}{2769.1\ \text{kJ/kg}} = \mathbf{0.463\ \text{or}\ 46.3\%}$$

Discussion This problem was worked out in Example 10–3*c* for the same pressure and temperature limits but without the regeneration process. A comparison of the two results reveals that the thermal efficiency of the cycle has increased from 43.0 to 46.3 percent as a result of regeneration. The net work output decreased by 171 kJ/kg, but the heat input decreased by 607 kJ/kg, which results in a net increase in the thermal efficiency.

EXAMPLE 10–6 The Ideal Reheat–Regenerative Rankine Cycle

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.

Solution A steam power plant operates on the ideal reheat–regenerative Rankine cycle with one open feedwater heater, one closed feedwater heater, and one reheater. The fractions of steam extracted from the turbine and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** In both open and closed feedwater heaters, feedwater is heated to the saturation temperature at the feedwater heater pressure. (Note that this is a conservative assumption since extracted steam enters the closed feedwater heater at 376°C and the saturation temperature at the closed feedwater pressure of 4 MPa is 250°C).

Analysis The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–19. The power plant operates on the ideal reheat–regenerative Rankine cycle and thus the pumps and the turbines are isentropic; there are no pressure drops in the boiler, reheater, condenser, and feedwater heaters; and steam leaves the condenser and the feedwater heaters as saturated liquid.

The enthalpies at the various states and the pump work per unit mass of fluid flowing through them are

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

The fractions of steam extracted are determined from the mass and energy balances of the feedwater heaters:

Closed feedwater heater:

$$\dot{E}_{in} = \dot{E}_{out}$$

$$y h_{10} + (1 - y) h_4 = (1 - y) h_5 + y h_8$$

$$y = \frac{h_5 - h_4}{(h_{10} - h_8) + (h_5 - h_4)} = \frac{1087.4 - 643.92}{(3155.8 - 1087.4) + (1087.4 - 643.92)} = \mathbf{0.1766}$$

Open feedwater heater:

$$\dot{E}_{in} = \dot{E}_{out}$$

$$z h_{12} + (1 - y - z) h_3 = (1 - y) h_2$$

$$z = \frac{(1 - y)(h_3 - h_2)}{h_{12} - h_2} = \frac{(1 - 0.1766)(640.09 - 192.30)}{3014.8 - 192.30} = \mathbf{0.1306}$$

The enthalpy at state 8 is determined by applying the mass and energy equations to the mixing chamber, which is assumed to be insulated:

$$\dot{E}_{in} = \dot{E}_{out}$$

$$(1) h_8 = (1 - y) h_7 + y h_9$$

$$h_8 = (1 - 0.1766)(1087.4)\ \text{kJ/kg} + 0.1766(1101.2)\ \text{kJ/kg}$$

$$= 1089.8\ \text{kJ/kg}$$

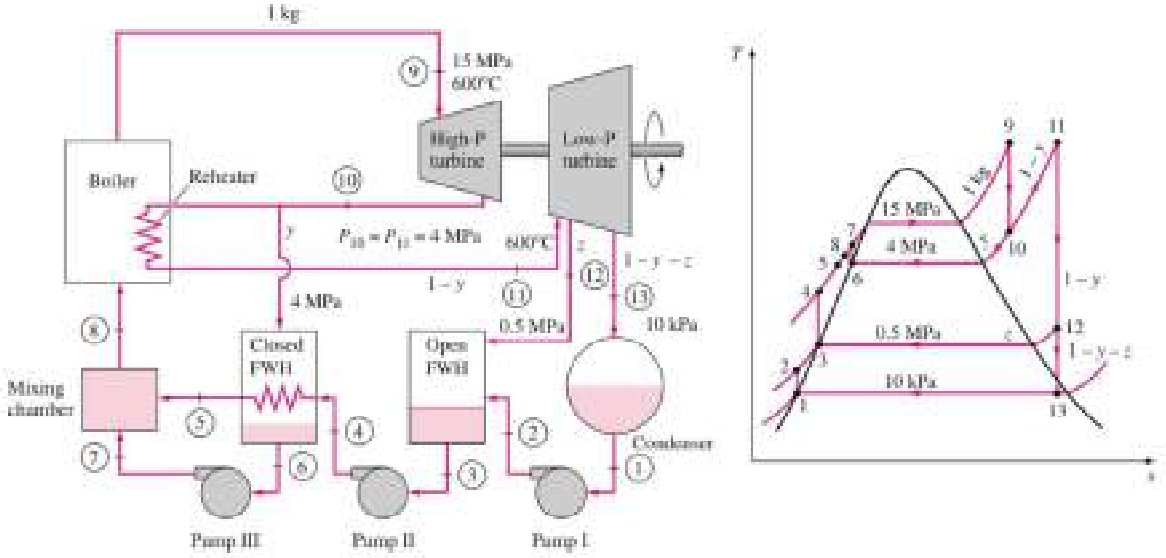


FIGURE 10–19

Schematic and *T-s* diagram for Example 10–6.

Thus,

$$\begin{aligned} q_{\text{in}} &= (h_9 - h_8) + (1 - y)(h_{11} - h_{10}) \\ &= (3583.1 - 1089.8)\,\text{kJ/kg} + (1 - 0.1766)(3674.9 - 3155.0)\,\text{kJ/kg} \\ &= 2921.4\,\text{kJ/kg} \\ q_{\text{out}} &= (1 - y - z)(h_{13} - h_1) \\ &= (1 - 0.1766 - 0.1306)(2335.7 - 191.81)\,\text{kJ/kg} \\ &= 1485.3\,\text{kJ/kg} \end{aligned}$$

and

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1485.3\,\text{kJ/kg}}{2921.4\,\text{kJ/kg}} = \mathbf{0.492\ or\ 49.2\%}$$

Discussion This problem was worked out in Example 10–4 for the same pressure and temperature limits with reheat but without the regeneration process. A comparison of the two results reveals that the thermal efficiency of the cycle has increased from 45.0 to 49.2 percent as a result of regeneration.

The thermal efficiency of this cycle could also be determined from

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{w_{\text{turb,out}} - w_{\text{pump,in}}}{q_{\text{in}}}$$

where

$$w_{\text{turb,out}} = (h_9 - h_{10}) + (1 - y)(h_{11} - h_{12}) + (1 - y - z)(h_{12} - h_{13})$$

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

Also, if we assume that the feedwater leaves the closed FWH as a saturated liquid at 15 MPa (and thus at $T_5 = 342°\text{C}$ and $h_5 = 1610.3$ kJ/kg), it can be shown that the thermal efficiency would be 50.6.

Engineering Thermodynamics

Module -3.4

Compiled by

Prof. Chinmaya Mund

Prof. Dambarudhar patel

Prof. Chandra Sekhar Saran

10-8 COGENERATION

- Many systems or devices, however, require energy input in the **form of heat, called process heat**
- Some industries that rely heavily on process heat are chemical, pulp and paper, oil production and refining, steel making
- The temperature in furnaces is typically very high (around 1400°C), and thus the energy in the furnace is of very high quality.
- This **high-quality energy is transferred to water to produce steam at about 200°C** or below (a highly irreversible process).
- Associated with this irreversibility is, of course, a **loss in exergy or work potential**.

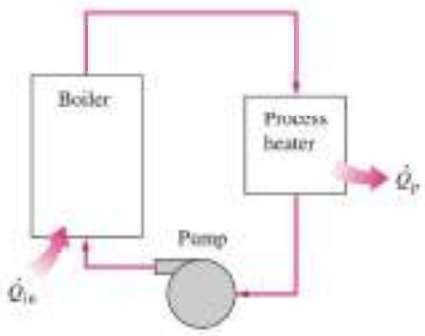


FIGURE 10–20
A simple process-heating plant.

- It is simply not wise to use high-quality energy to accomplish a task that could be accomplished with low-quality energy.
- cogeneration is the production of more than one useful form of energy (such as process heat and electric power) from the same energy source.

- Probably the most striking feature of the ideal steam-turbine cogeneration plant shown in Fig. 10–21 is the **absence of a condenser**
- Thus **no heat is rejected from this plant as waste heat**
- In other words, all the energy transferred to the steam in the boiler is utilized as either process heat or electric power.

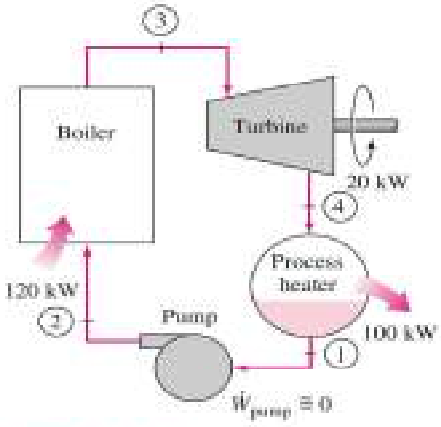


FIGURE 10–21
An ideal cogeneration plant.

Thus it is appropriate to define a **utilization factor** ε_u for a cogeneration plant as

$$\epsilon_u = \frac{\text{Net work output + Process heat delivered}}{\text{Total heat input}} = \frac{\dot{W}_{net} + \dot{Q}_p}{\dot{Q}_{in}} \quad \text{(10–23)}$$

or

$$\epsilon_u = 1 - \frac{\dot{Q}_{out}}{\dot{Q}_{in}} \quad \text{(10–24)}$$

where $\dot{Q}_{out}$ represents the heat rejected in the condenser. Strictly speaking,

- 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 (PRV)** to the extraction pressure P_6 and is directed to the process-heating unit
- **Maximum process heating** is realized when all the steam leaving the boiler passes through the PRV (**m_5= m_4**)
    - No power is produced in this mode

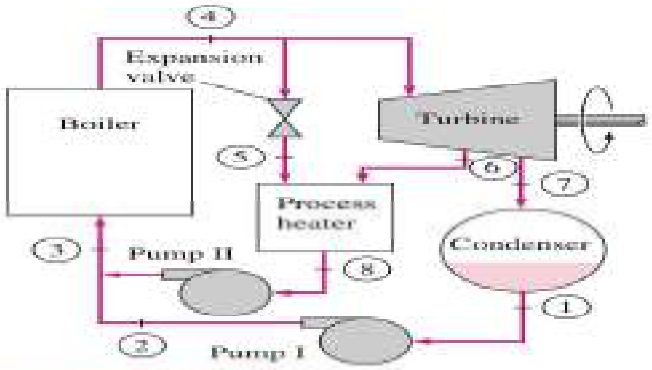


FIGURE 10–22
A cogeneration plant with adjustable loads.

- When there is **no demand for process heat**, all the steam passes through the turbine and the condenser (**m_5= m_6= 0**)
    - 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:

$$\dot{Q}_{in} = \dot{m}_3(h_4 - h_3) \quad \textbf{(10–25)}$$

$$\dot{Q}_{out} = \dot{m}_7(h_7 - h_1) \quad \textbf{(10–26)}$$

$$\dot{Q}_p = \dot{m}_5 h_5 + \dot{m}_6 h_6 - \dot{m}_8 h_8 \quad \textbf{(10–27)}$$

$$\dot{W}_{turb} = (\dot{m}_4 - \dot{m}_5)(h_4 - h_5) + \dot{m}_7(h_5 - h_7) \quad \textbf{(10–28)}$$

- Under **optimum conditions**, a cogeneration plant simulates the ideal cogeneration plant discussed earlier.
    - That is, **all the steam expands in the turbine to the extraction pressure and continues to the process-heating unit.**
    - **No steam passes through the PRV or the condenser;** thus, no waste heat is rejected (**m_4=m_6 and m_5= m_7= 0**).

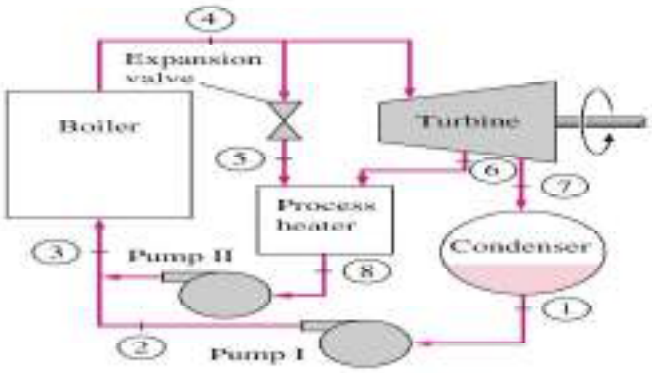


FIGURE 10–22
A cogeneration plant with adjustable loads.

- This condition may be difficult to achieve in practice because of the **constant variations in the process-heat and power loads**.
- But the plant should be designed so that the optimum operating conditions are approximated most of the time

EXAMPLE 10–8 An Ideal Cogeneration Plant

Consider the cogeneration plant shown in Fig. 10–23. Steam enters the turbine at 7 MPa and 500°C. Some steam is extracted from the turbine at 500 kPa for process heating. The remaining steam continues to expand to 5 kPa. Steam is then condensed at constant pressure and pumped to the boiler pressure of 7 MPa. At times of high demand for process heat, some steam leaving the boiler is throttled to 500 kPa and is routed to the process heater. The extraction fractions are adjusted so that steam leaves the process heater as a saturated liquid at 500 kPa. It is subsequently pumped to 7 MPa. The mass flow rate of steam through the boiler is 15 kg/s. Disregarding any pressure drops and heat losses in the piping and assuming the turbine and the pump to be isentropic, determine (*a*) the maximum rate at which process heat can be supplied, (*b*) the power produced and the utilization factor when no process heat is supplied, and (*c*) the rate of process heat supply when 10 percent of the steam is extracted before it enters the turbine and 70 percent of the steam is extracted from the turbine at 500 kPa for process heating.

Solution A cogeneration plant is considered. The maximum rate of process heat supply, the power produced and the utilization factor when no process heat is supplied, and the rate of process heat supply when steam is extracted from the steam line and turbine at specified ratios are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Pressure drops and heat losses in piping are negligible. **3** Kinetic and potential energy changes are negligible.

Analysis The schematic of the cogeneration plant and the *T-s* diagram of the cycle are shown in Fig. 10–23. The power plant operates on an ideal cycle and thus the pumps and the turbines are isentropic; there are no pressure drops in the boiler, process heater, and condenser; and steam leaves the condenser and the process heater as saturated liquid.

The work inputs to the pumps and the enthalpies at various states are as follows:

$$w_{\text{pump I,in}} = v_1(P_4 - P_5) = (0.001005\ \text{m}^3/\text{kg})[(7000 - 5)\ \text{kPa}]\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)$$

$$= 7.03\ \text{kJ/kg}$$

$$w_{\text{pump II,in}} = v_7(P_8 - P_7) = (0.001093\ \text{m}^3/\text{kg})[(7000 - 500)\ \text{kPa}]\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)$$

$$= 7.10\ \text{kJ/kg}$$

$$h_1 = h_2 = h_3 = h_4 = 3411.4\ \text{kJ/kg}$$

$$h_5 = 2739.3\ \text{kJ/kg}$$

$$h_6 = 2073.0\ \text{kJ/kg}$$

$$h_7 = h_{f@500\ \text{kPa}} = 640.09\ \text{kJ/kg}$$

$$h_9 = h_{f@5\ \text{kPa}} = 137.75\ \text{kJ/kg}$$

$$h_5 = h_9 + w_{\text{pump I,in}} = (137.75 + 7.03)\ \text{kJ/kg} = 144.78\ \text{kJ/kg}$$

$$h_{10} = h_7 + w_{\text{pump II,in}} = (640.09 + 7.10)\ \text{kJ/kg} = 647.19\ \text{kJ/kg}$$

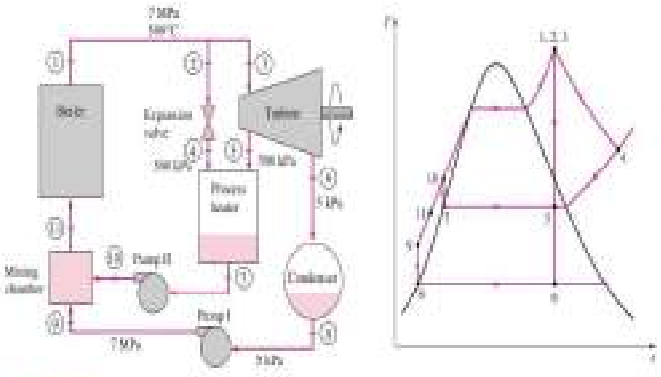


FIGURE 10–23
Schematic and *T-s* diagram for Example 10–8.

>>

(*a*) The maximum rate of process heat is achieved when all the steam leaving the boiler is throttled and sent to the process heater and none is sent to the turbine (that is, $\dot{m}_4 = \dot{m}_7 = \dot{m}_1 = 15$ kg/s and $\dot{m}_2 = \dot{m}_5 = \dot{m}_6 = 0$). Thus,

$$\dot{Q}_{p,max} = \dot{m}_1(h_4 - h_7) = (15\ \text{kg/s})[(3411.4 - 640.09)\ \text{kJ/kg}] = \mathbf{41{,}570\ kW}$$

The utilization factor is 100 percent in this case since no heat is rejected in the condenser, heat losses from the piping and other components are assumed to be negligible, and combustion losses are not considered.

(*b*) When no process heat is supplied, all the steam leaving the boiler passes through the turbine and expands to the condenser pressure of 5 kPa (that is, $\dot{m}_3 = \dot{m}_5 = \dot{m}_1 = 15$ kg/s and $\dot{m}_2 = \dot{m}_6 = 0$). Maximum power is produced in this mode, which is determined to be

$$\dot{W}_{turb,out} = \dot{m}(h_3 - h_4) = (15\ \text{kg/s})[(3411.4 - 2073.0)\ \text{kJ/kg}] = 20{,}076\ \text{kW}$$

$$\dot{W}_{pump,in} = (15\ \text{kg/s})(7.03\ \text{kJ/kg}) = 105\ \text{kW}$$

$$\dot{W}_{net,out} = \dot{W}_{turb,out} - \dot{W}_{pump,in} = (20{,}076 - 105)\ \text{kW} = 19{,}971\ \text{kW} = \mathbf{20.0\ MW}$$

$$\dot{Q}_{in} = \dot{m}_1(h_1 - h_{11}) = (15\ \text{kg/s})[(3411.4 - 144.78)\ \text{kJ/kg}] = 48{,}999\ \text{kW}$$

Thus,

$$\varepsilon_u = \frac{\dot{W}_{net} + \dot{Q}_p}{\dot{Q}_{in}} = \frac{(19{,}971 + 0)\ \text{kW}}{48{,}999\ \text{kW}} = \mathbf{0.408\ or\ 40.8\%}$$

That is, 40.8 percent of the energy is utilized for a useful purpose. Notice that the utilization factor is equivalent to the thermal efficiency in this case.

(*c*) Neglecting any kinetic and potential energy changes, an energy balance on the process heater yields

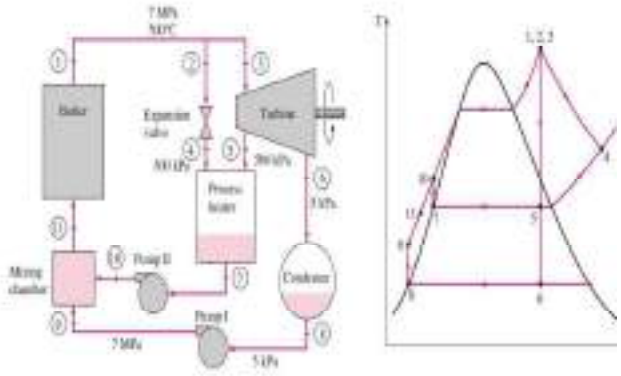


FIGURE 10-22
Schematic and T-s diagram for Example 10-4.