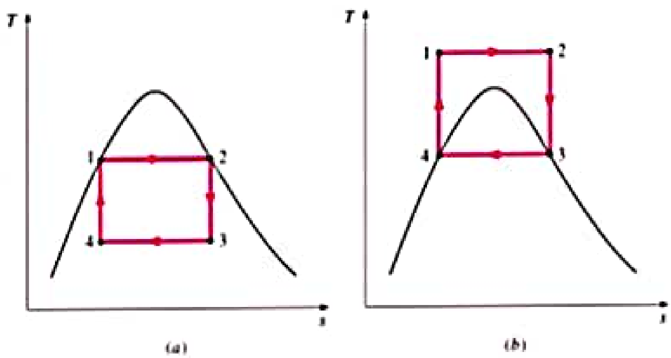
Carnot Vapor Cycle:



- The fluid is heated reversibly and isothermally in a boiler (process 1-2),
- Expanded isentropically in a turbine (process 2-3),
- Condensed reversibly and isothermally in a condenser (process 3-4),
- Compressed isentropically by a compressor to the initial state (process 4-1)

Limitations Carnot Vapor Cycle:

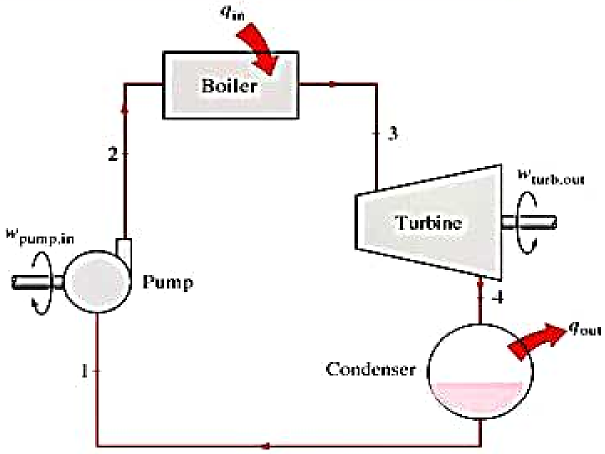
1. **Processes 1-2 and 3-4:** Maximum temperature that can be used in the cycle (374°C for water). Limiting the maximum temperature in the cycle also limits the thermal efficiency. Any attempt to raise the maximum temperature in the cycle involves heat transfer to the working fluid in a single phase, which is not easy to accomplish isothermally.
2. **Process 2-3:** Quality of the steam decreases during this process. Thus the turbine has to handle steam with low quality, that is, steam with a high moisture content. The impingement of liquid droplets on the turbine blades causes erosion and is a major source of wear. Thus steam with qualities less than about 90 percent cannot be tolerated in the operation of power plants.
3. **Process 4-1:** Involves the compression of a liquid–vapor mixture to a saturated liquid. It is not easy to control the condensation process so precisely as to end up with the desired quality at state 4. Second, it is not practical to design a compressor that handles two phases.

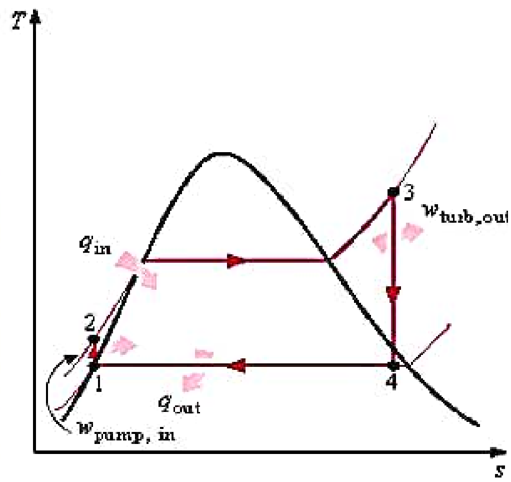
Simple Ideal Rankine Cycle:

Many of the impracticalities associated with the Carnot cycle can be eliminated by superheating the steam in the boiler and condensing it completely in the condenser. The cycle that results is the Rankine cycle, which is the ideal cycle for vapor power plants.

The ideal Rankine cycle does not involve any internal irreversibilities and consists of the following four processes:

- 1-2 Isentropic compression in a pump
- 2-3 Constant pressure heat addition in a boiler
- 3-4 Isentropic expansion in a turbine
- 4-1 Constant pressure heat rejection in a condenser





Steady-flow energy equation per unit mass of steam reduces to

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_e - h_i \qquad \text{(kJ/kg)}$$

Conservation of energy relation for each device

$$\textit{Pump } (q = 0): \qquad w_{pump,in} = h_2 - h_1$$

$$w_{pump,in} = v(P_2 - P_1)$$

Vapor Power Cycles

where

$$h_1 = h_{f\,@\,P_1} \quad \text{and} \quad v \cong v_1 = v_{f\,@\,P_1}$$

Boiler ($w = 0$): $$q_{in} = h_3 - h_2$$

Turbine ($q = 0$): $$w_{turb,out} = h_3 - h_4$$

Condenser ($w = 0$): $$q_{out} = h_4 - h_1$$

The *thermal efficiency* of the Rankine cycle is determined from

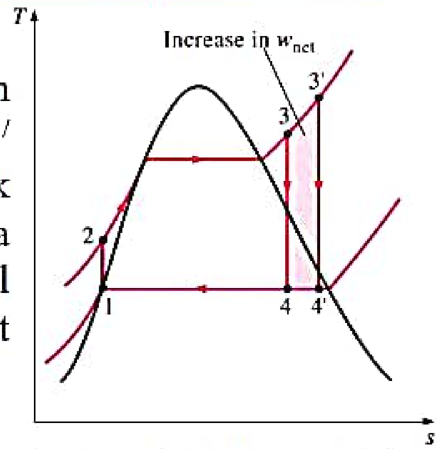
$$\eta_{th} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}}$$

where

$$w_{net} = q_{in} - q_{out} = w_{turb,out} - w_{pump,in}$$

Method of Increasing Efficiency Of Rankine Cycle

Superheating the Steam to High Temperatures:



The colored area on this diagram represents the increase in the net work. The total area under the process curve 3-3′ represents the increase in the heat input. Thus both the net work and heat input increase as a result of superheating the steam to a higher temperature. The overall effect is an increase in thermal efficiency, however, since the average temperature at which heat is added increases.

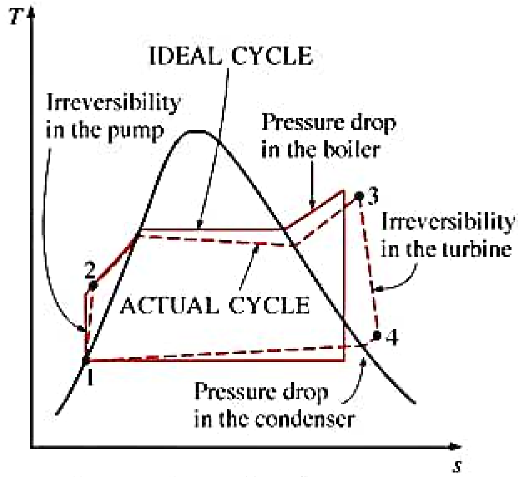
Superheating the steam to higher temperatures has another very desirable effect: It decreases the moisture content of the steam at the turbine exit.

The temperature to which steam can be superheated is limited, however, by metallurgical considerations. Presently the highest steam temperature allowed at the turbine inlet is about 620°C (1150°F). Any increase in this value depends on improving the present materials or finding new ones that can withstand higher temperatures. Ceramics are very promising in this regard.

Actual Vapor Power Cycle

The actual vapor power cycle differs from the ideal Rankine cycle, as a result of irreversibilities in various components. Fluid friction and heat loss to the surroundings are the two common sources of irreversibilities.

- Fluid friction causes pressure drops in the boiler, the condenser, and the piping between various components. As a result, steam leaves the boiler at a somewhat lower pressure.
- Also, the pressure at the turbine inlet is somewhat lower than that at the boiler exit due to the pressure drop in the connecting pipes.
- The pressure drop in the condenser is usually very small. To compensate for these pressure drops, the water must be pumped to a sufficiently higher pressure than the ideal cycle calls for. This requires a larger pump and larger work input to the pump.



Deviation of actual vapor power cycle from the ideal Rankine cycle.

- The other major source of irreversibility is the heat loss from the steam to the surroundings as the steam flows through various components. To maintain the same level of net work output, more heat needs to be transferred to the steam in the boiler to compensate for these undesired heat losses. As a result, cycle efficiency decreases.

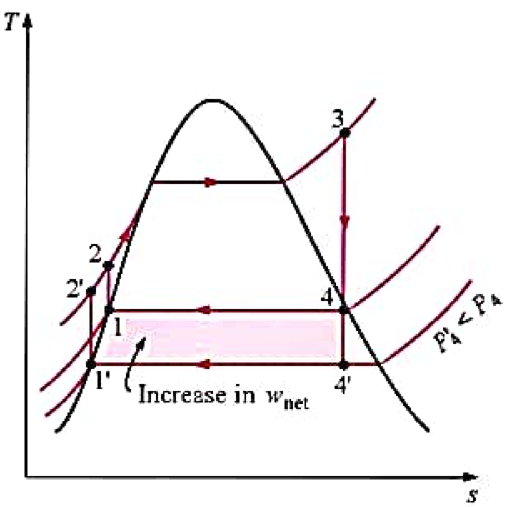
Method of Increasing Efficiency Of Rankine Cycle

- Increase the average temperature at which heat is transferred to the working fluid in the boiler
- Decrease the average temperature at which heat is rejected from the working fluid in the condenser

Effect of Lowering Condenser Pressure:

Lowering the operating pressure of the condenser automatically lowers the temperature of the steam, and thus the temperature at which heat is rejected

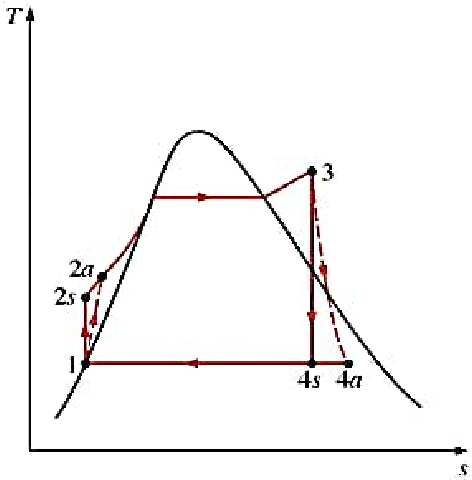
For comparison purposes, the turbine inlet state is maintained the same. The colored area on this diagram represents the increase in net work output as a result of lowering the condenser pressure from P_4 to P_4'. The heat input requirements also increase (represented by the area under curve $2'$ to 2), but this increase is very small. Thus the overall effect of lowering the condenser pressure is an increase in the thermal efficiency of the cycle.



Actual Vapor Power Cycle

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

A pump requires a greater work input, and a turbine produces a smaller work output as a result of irreversibilities. Under ideal conditions, the flow through these devices is isentropic. The deviation of actual pumps and turbines from the isentropic ones can be accounted for by utilizing isentropic efficiencies, defined as



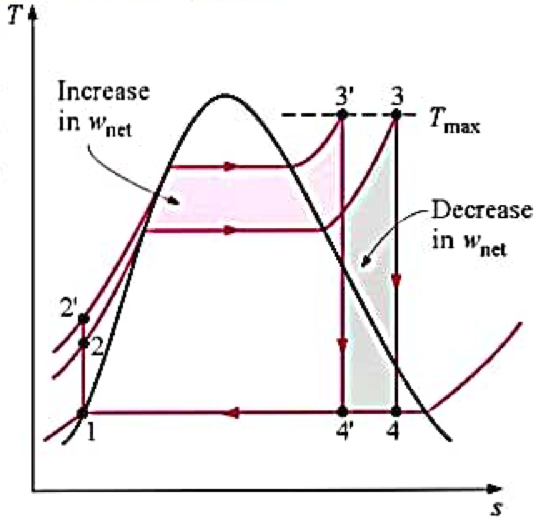
$$\eta_P = \frac{w_s}{w_a} = \frac{h_{2s} - h_1}{h_{2a} - h_1} \qquad \eta_T = \frac{w_a}{w_s} = \frac{h_3 - h_{4a}}{h_3 - h_{4s}}$$

- In actual condensers, for example, the liquid is usually subcooled to prevent the onset of cavitation, the rapid vaporization and condensation of the fluid at the low-pressure side of the pump impeller, which may damage it.
- Additional losses occur at the bearings between the moving parts as a result of friction.
- Steam that leaks out during the cycle and air that leaks into the condenser represent two other sources of loss.
- Finally, the power consumed by the auxiliary equipment such as fans that supply air to the furnace should also be considered in evaluating the overall performance of power plants.

Method of Increasing Efficiency Of Rankine Cycle

Effect of Increasing Boiler Pressure on the Ideal Rankine cycle:

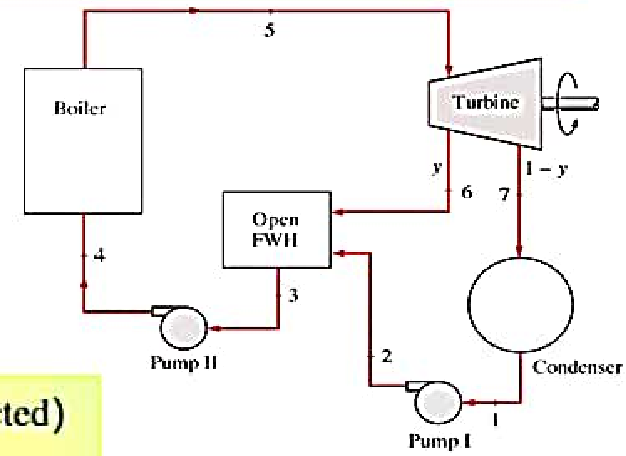
This raises the average temperature at which heat is transferred to the steam and thus raises the thermal efficiency of the cycle. Notice that for a fixed turbine inlet temperature, the cycle shifts to the left and the moisture content of steam at the turbine exit increases. This undesirable side effect can be corrected, however, by reheating the steam.

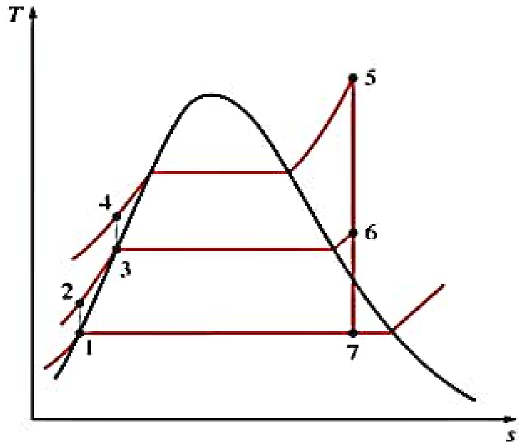


Ideal Regenerative Rankine Cycle with Open Feedwater Heater

$$q_{in} = h_5 - h_4$$
$$q_{out} = (1 - y)(h_7 - h_1)$$
$$w_{turb,out} = (h_5 - h_6) + (1 - y)(h_6 - h_7)$$
$$w_{pump,in} = (1 - y)w_{pump\ I,in} + w_{pump\ II,in}$$

$$y = \dot{m}_6/\dot{m}_5 \qquad \text{(fraction of steam extracted)}$$
$$w_{pump\ I,in} = v_1(P_2 - P_1)$$
$$w_{pump\ II,in} = v_3(P_4 - P_3)$$

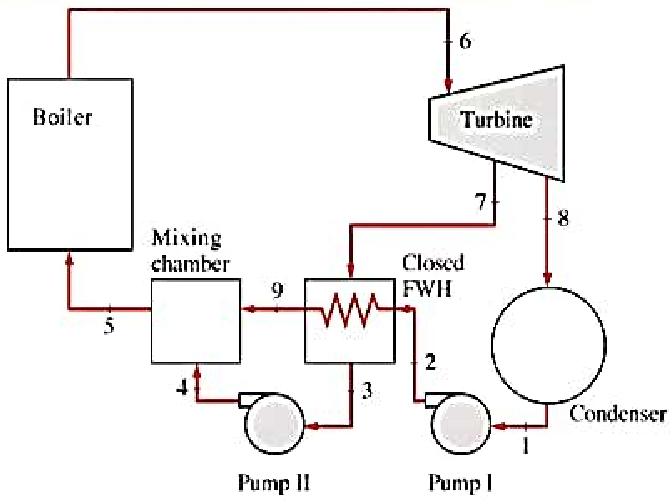


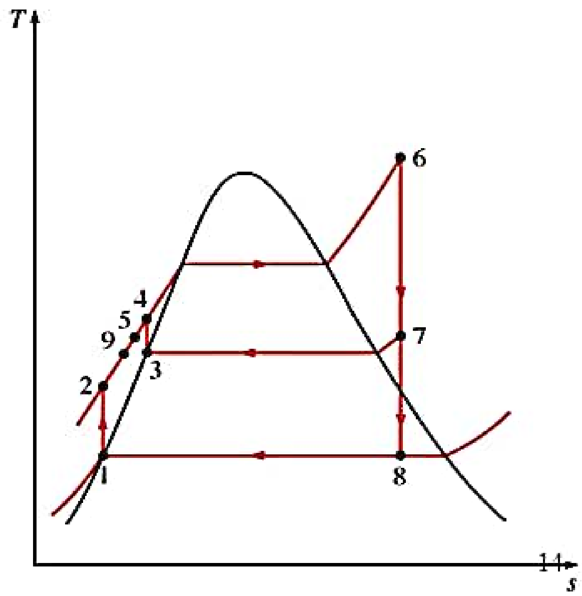


Ideal Regenerative Rankine Cycle with Closed Feedwater Heater

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

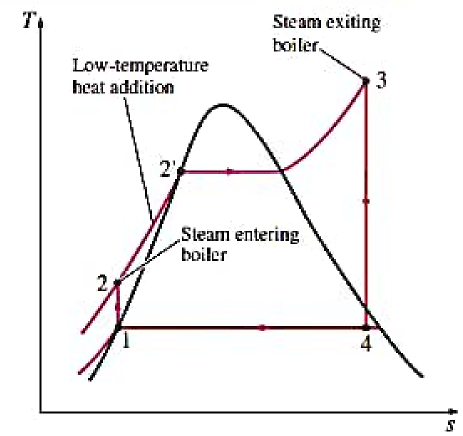
The closed feedwater heaters are more complex because of the internal tubing network, and thus they are more expensive. Heat transfer in closed feedwater heaters is also less effective since the two streams are not allowed to be in direct contact. However, closed feedwater heaters do not require a separate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





Regeneration

A practical regeneration process in steam power plants is accomplished by extracting, or "bleeding," steam from the turbine at various points. The device where the feedwater is heated by regeneration is called a **regenerator, or a feedwater heater (FWH).**



- Deaerating the feedwater to prevent corrosion in the boiler
- Control the large volume flow rate of the steam at the final stages of the turbine

A feedwater heater is basically a heat exchanger where heat is transferred from the steam to the feedwater either by mixing the two fluid streams (**open feedwater heaters**) or without mixing them (**closed feedwater heaters**).

Second-law Analysis of Vapor Power Cycles

The exergy destruction for a steady-flow system can be expressed, in the rate form, as

$$\dot{X}_{dest} = T_0\dot{S}_{gen} = T_0(\dot{S}_{out} - \dot{S}_{in}) = T_0\left(\sum_{out}\dot{m}s + \frac{\dot{Q}_{out}}{T_{b,out}} - \sum_{in}\dot{m}s - \frac{\dot{Q}_{in}}{T_{b,in}}\right) \qquad \text{(kW)}$$

or on a unit mass basis for a one-inlet, one-exit, steady-flow device as

$$x_{dest} = T_0 s_{gen} = T_0\left(s_e - s_i + \frac{q_{out}}{T_{b,out}} - \frac{q_{in}}{T_{b,in}}\right) \qquad \text{(kJ/kg)}$$

where $T_{b,in}$ and $T_{b,out}$ are the temperatures of the system boundary where heat is transferred into and out of the system, respectively.

The exergy destruction associated with a *cycle* depends on the magnitude of the heat transfer with the high- and low-temperature reservoirs involved, and their temperatures. It can be expressed on a unit mass basis as

$$x_{dest} = T_0\left(\sum\frac{q_{out}}{T_{b,out}} - \sum\frac{q_{in}}{T_{b,in}}\right) \qquad \text{(kJ/kg)}$$

For a cycle that involves heat transfer only with a source at T_H and a sink at T_L , the exergy destruction becomes

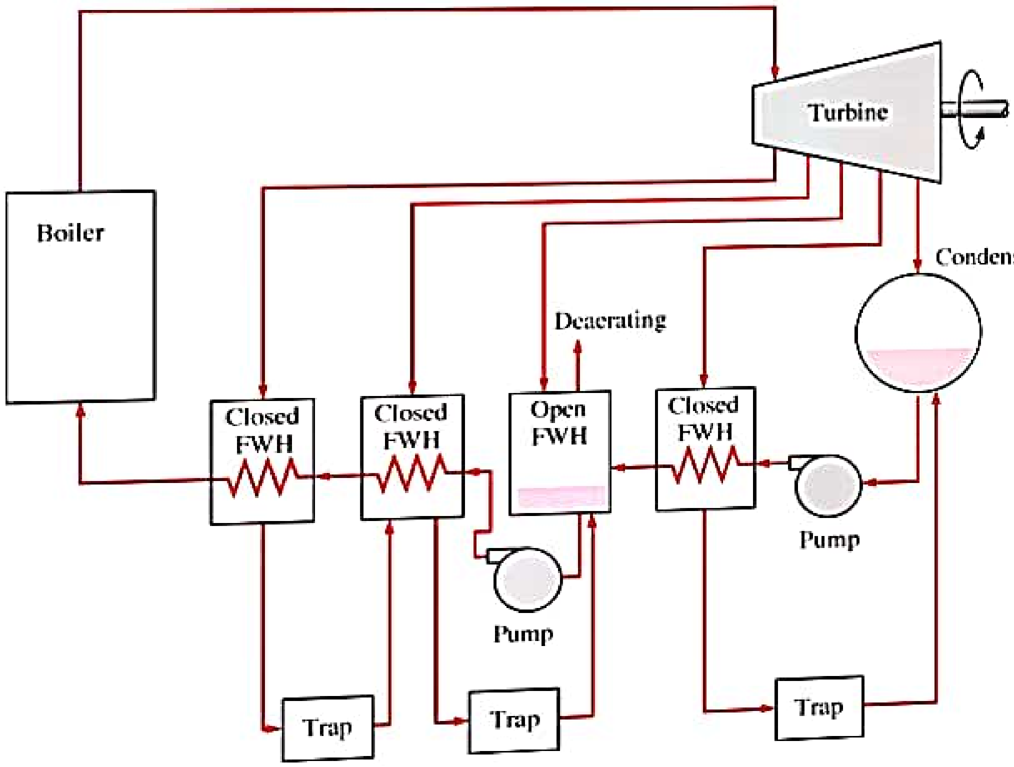
$$x_{dest} = T_0\left(\frac{q_{out}}{T_L} - \frac{q_{in}}{T_H}\right) \qquad \text{(kJ/kg)}$$

The exergy of a fluid stream c at any state can be determined from

$$\psi = (h - h_0) - T_0(s - s_0) + \frac{V^2}{2} + gz \qquad \text{(kJ/kg)}$$

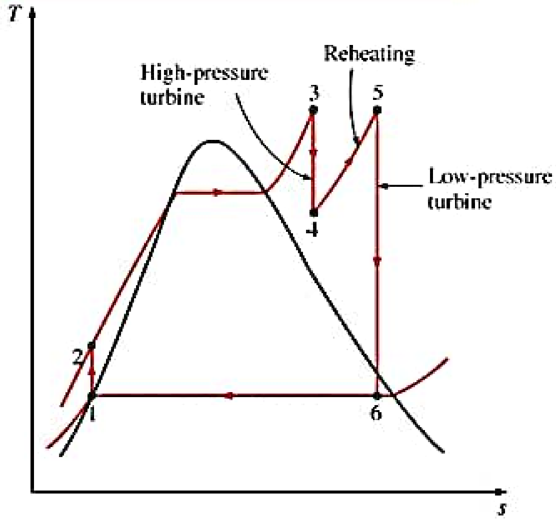
where the subscript “0” denotes the state of the surroundings.

Ideal Regenerative Rankine Cycle with Open & Closed Feedwater Heater



Ideal Reheat Rankine Cycle

How can we take advantage of the increased efficiencies at higher boiler pressures without facing the problem of excessive moisture at the final stages of the turbine?



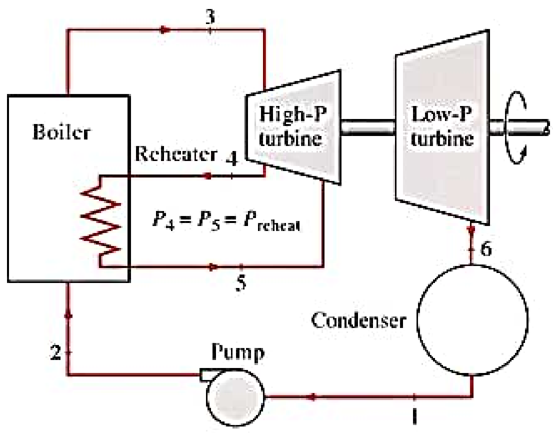
Two possibilities come to mind:

1. Superheat the steam to very high temperatures before it enters the turbine. This would be the desirable solution since the average temperature at which heat is added would also increase, thus increasing the cycle efficiency.
This is not a viable solution, however, since it requires raising the steam temperature to metallurgically unsafe levels.
2. Expand the steam in the turbine in two stages, and reheat it in between. In other words, modify the simple ideal Rankine cycle with a **reheat** process. Reheating is a practical solution to the excessive moisture problem in turbines, and it is commonly used in modern steam power plants.

Thus the total heat input and the total turbine work output for a reheat cycle become

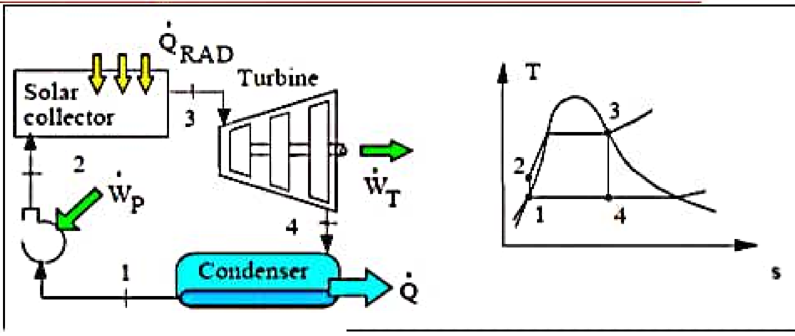
$$q_{in} = q_{primary} + q_{reheat} = (h_3 - h_2) + (h_5 - h_4)$$

$$w_{turb,out} = w_{turb,I} + w_{turb,II} = (h_3 - h_4) + (h_5 - h_6)$$



Simple Rankine Cycle

Consider a solar-energy-powered ideal Rankine cycle that uses water as the working fluid. Saturated vapor leaves the solar collector at 1000 kPa, and the condenser pressure is 10 kPa. Determine the thermal efficiency of this cycle.



Solution:

State 3: 1000 kPa, sat vap. $h_3 = 2778.08$ kJ/kg, $s_3 = 6.5864$ kJ/kg K

CV Turbine adiabatic and reversible so second law gives

$$s_4 = s_3 = 6.5864 = 0.6492 + x_4 \times 7.5010 \quad => \quad x_4 = 0.7915$$

$$h_4 = 191.81 + 0.7915 \times 2392.82 = 2085.73 \text{ kJ/kg}$$

The energy equation gives

$$w_T = h_3 - h_4 = 2778.08 - 2085.73 = 692.35 \text{ kJ/kg}$$

C.V. pump and incompressible liquid gives work into pump

$$w_P = v_1(P_2 - P_1) = 0.00101(1000 - 10) = 1.0 \text{ kJ/kg}$$

$$h_2 = h_1 + w_P = 191.81 + 1.0 = 192.81 \text{ kJ/kg}$$

C.V. boiler gives the heat transfer from the energy equation as

$$q_H = h_3 - h_2 = 2778.08 - 192.81 = 2585.3 \text{ kJ/kg}$$

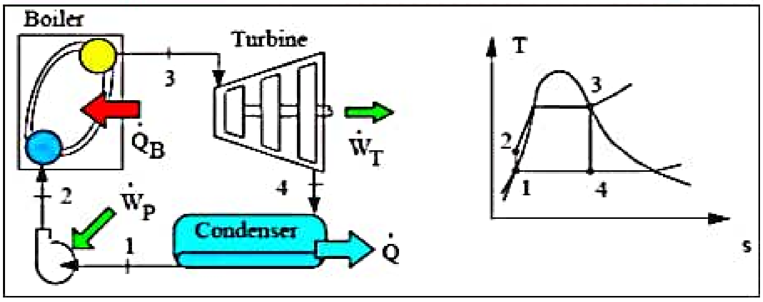
The cycle net work and efficiency are found as

$$w_{NET} = w_T - w_P = 692.35 - 1.0 = 691.35 \text{ kJ/kg}$$

$$\eta_{TH} = w_{NET}/q_H = 691.35/2585.3 = \mathbf{0.267}$$

Simple Rankine Cycle

A steam power plant operating in a Rankine cycle has saturated vapor at 3.0 MPa leaving the boiler. The turbine exhausts to the condenser operating at 10 kPa. Find the specific work and heat transfer in each of the ideal components and the cycle efficiency.



Solution:

Cycle is determined by: (P_3, x_3, $P_1 = P_4$, assume $x_1 = 0$)

C.V. Pump Reversible and adiabatic.

Energy: $w_p = h_2 - h_1$; Entropy: $s_2 = s_1$

since incompressible it is easier to find work (positive in) as

$$w_p = \int v\, dP = v_1 (P_2 - P_1) = 0.00101 (3000 - 10) = 3.02 \text{ kJ/kg}$$

$$\Rightarrow h_2 = h_1 + w_p = 191.81 + 3.02 = 194.83 \text{ kJ/kg}$$

C.V. Boiler : $q_H = h_3 - h_2 = 2804.14 - 194.83 = \mathbf{2609.3\ kJ/kg}$

C.V. Turbine : $w_T = h_3 - h_4$: $s_4 = s_3$

$$s_4 = s_3 = 6.1869 = 0.6492 + x_4 (7.501) \Rightarrow x_4 = 0.7383$$

$$\Rightarrow h_4 = 191.81 + 0.7383 (2392.82) = 1958.34 \text{ kJ/kg}$$

$$w_T = 2804.14 - 1958.34 = \mathbf{845.8\ kJ/kg}$$

C.V. Condenser : $q_L = h_4 - h_1 = 1958.34 - 191.81 = \mathbf{1766.5\ kJ/kg}$

$$\eta_{cycle} = w_{net} / q_H = (w_T + w_p) / q_H = (845.8 - 3.0) / 2609.3 = \mathbf{0.323}$$

Simple Rankine Cycle

The power plant in the previous problem is augmented with a natural gas burner to superheat the water to 300°C before entering the turbine. Find the cycle efficiency with this configuration and the specific heat transfer added by the natural gas burner.

Solution: C.V. H_2O ideal Rankine cycle

Cycle is determined by: (P_3, T_3, $P_1 = P_4$, assume $x_1 = 0$)

State 3: 1000 kPa, 300°C, $h_3 = 3051.15$ kJ/kg, $s_3 = 7.1228$ kJ/kg K

CV Turbine adiabatic and reversible so second law gives

$$s_4 = s_3 = 7.1228 = 0.6492 + x_4 \times 7.5010 \quad => \quad x_4 = 0.86303$$

$$h_4 = 191.81 + 0.86303 \times 2392.82 = 2256.88 \text{ kJ/kg}$$

The energy equation gives

$$w_T = h_3 - h_4 = 3051.15 - 2256.88 = 794.27 \text{ kJ/kg}$$

C.V. pump and incompressible liquid gives work into pump

$$w_P = v_1(P_2 - P_1) = 0.00101(1000 - 10) = 1.0 \text{ kJ/kg}$$

$$h_2 = h_1 + w_P = 191.81 + 1.0 = 192.81 \text{ kJ/kg}$$

C.V. boiler gives the heat transfer from the energy equation as

$$q_H = h_3 - h_2 = 3051.15 - 192.81 = 2858.3 \text{ kJ/kg}$$

$$q_{H\,gas} = h_3 - {}_{g\ 1000\ kPa} = 3051.15 - 2778.08 = \mathbf{273.07\ kJ/kg}$$

The cycle net work and efficiency are found as

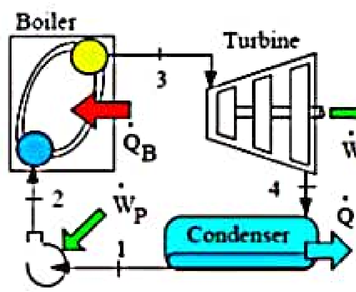
$$w_{NET} = w_T - w_P = 794.27 - 1.0 = 793.27 \text{ kJ/kg}$$

$$\eta_{TH} = w_{NET}/q_H = 793.27/2858.3 = \mathbf{0.277}$$

Simple Rankine Cycle

A steam power plant has a high pressure of 3 MPa and it maintains 60°C in the condenser. A condensing turbine is used, but the quality should not be lower than 90% at any state in the turbine. For a turbine power output of 8 MW find the work and heat transfer in all components and the cycle efficiency.

State 1: Sat. liquid. $P_1 = 19.94$ kPa, $h_1 = 251.11$ kJ/kg, $v_1 = 0.001017$ m^3/kg



Consider C.V. pump

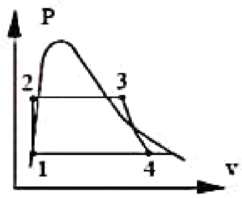
Energy: $h_2 - h_1 = w_P = v_1 (P_2 - P_1) = 0.001017 (3000 - 19.94) = 3.03$ kJ/kg

State 2: $P_2 = 3000$ kPa. $h_2 = h_1 + w_P = 251.11 + 3.03 = 254.1$ kJ/kg

State 4: $P_4 = P_1 = 19.94$ kPa, $x = 0.9$

$$s_4 = s_f + x_4\, s_{fg} = 0.8311 + 0.9 \times 7.0784 = 7.20166 \text{ kJ/kg-K}$$

$$h_4 = h_f + x_4\, h_{fg} = 251.11 + 0.9 \times 2358.48 = 2373.74 \text{ kJ/kg}$$



Consider the turbine for which $s_4 = s_3$.

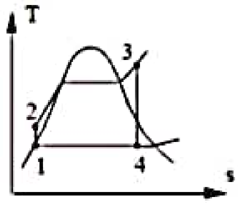
State 3: Table B.2.2 3000 kPa, $s_3 = 7.20166$ kJ/kg K => $h_3 = 3432.5$ kJ/kg

Turbine: $w_T = h_3 - h_4 = 3432.5 - 2373.74 = 1058.8$ kJ/kg

Boiler: $q_H = h_3 - h_2 = 3432.5 - 254.1 = 3178.4$ kJ/kg

Condenser: $q_L = h_4 - h = 2373.74 - 251.1 = 2122.6$ kJ/kg

Efficiency: $\eta_{TH} = w_{NET}/q_H = (w_T - w_P)/q_H = \dfrac{1058.8 - 3.03}{3178.4} = \mathbf{0.332}$



Scaling: $\dot{m} = \dot{W}_T/w_T = 8\,000 \text{ kW} / 1058.8 \text{ kJ/kg} = 7.555$ kg/s

$\dot{W}_P = \dot{m}\, w_P = \mathbf{22.9\ kW}$; $\dot{Q}_H = \dot{m}\, q_H = \mathbf{24\ MW}$, $\dot{Q}_L = \dot{m}\, q_L = \mathbf{16\ MW}$

Reheat Rankine Cycle

1: 45°C, x = 0: $h_1 = 188.42$ kJ/kg, $v_1 = 0.00101$ m³/kg, $P_{sat} = 9.59$ kPa

3: 3.0 MPa, 600°C: $h_3 = 3682.34$ kJ/kg, $s_3 = 7.5084$ kJ/kg K

6: 45°C, x = 1: $h_6 = 2583.19$ kJ/kg, $s_6 = 8.1647$ kJ/kg K

C.V. Pump Reversible and adiabatic.

Energy: $w_p = h_2 - h_1$; Entropy: $s_2 = s_1$

since incompressible it is easier to find work (positive in) as

$w_p = \int v\, dP = v_1 (P_2 - P_1) = 0.00101\,(3000 - 9.59) = 3.02$ kJ/kg

$h_2 = h_1 + w_p = 188.42 + 3.02 = 191.44$ kJ/kg

C.V. HP Turbine section

Entropy Eq.: $s_4 = s_3$ => $h_4 = 3093.26$ kJ/kg; $T_4 = 314°C$

C.V. LP Turbine section

Entropy Eq.: $s_6 = s_5 = 8.1647$ kJ/kg K => state 5

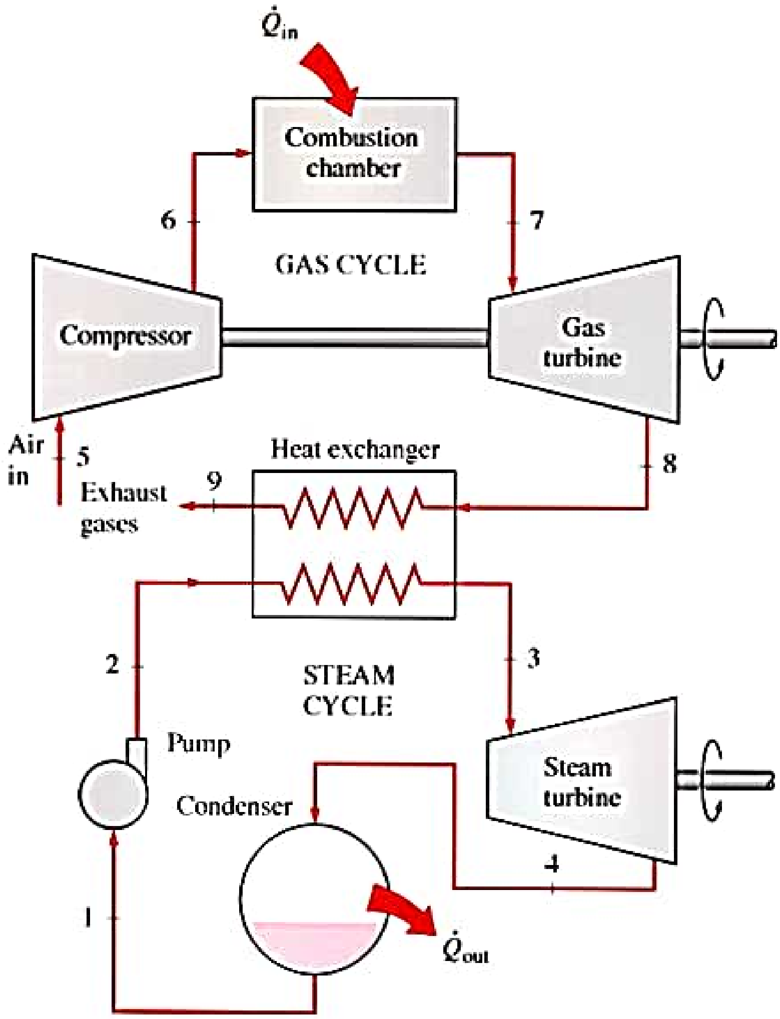
State 5: 500 kPa, s_5 => $h_5 = 3547.55$ kJ/kg, $T_5 = 529°C$

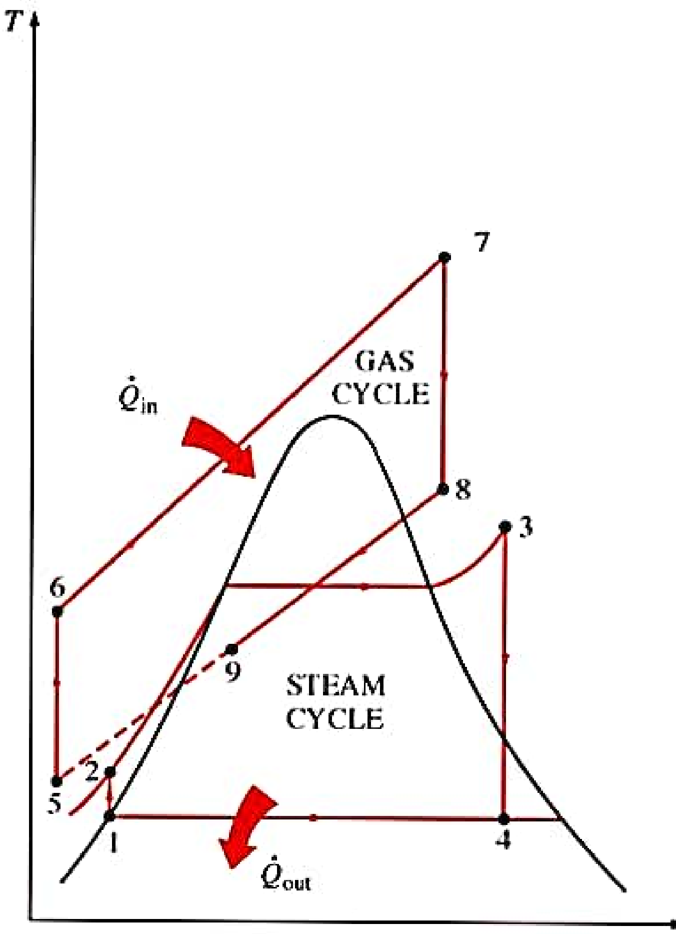
C.V. Condenser.

Energy Eq.: $q_L = h_6 - h_1 = h_{fg} = 2394.77$ kJ/kg

$\dot{m} = \dot{Q}_L / q_L = 10\,000 / 2394.77 = 4.176$ kg/s

Combined Gas-Steam Power Plant

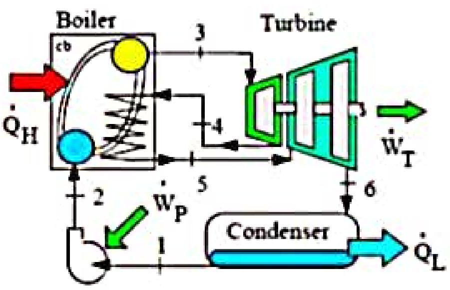


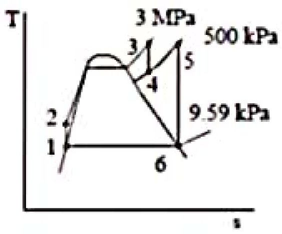


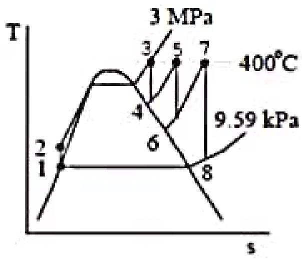
H/W: Example 10.1 – 10.9

Reheat Rankine Cycle

A smaller power plant produces steam at 3 MPa, 600°C in the boiler. It keeps the condenser at 45°C by transfer of 10 MW out as heat transfer. The first turbine section expands to 500 kPa and then flow is reheated followed by the expansion in the low-pressure turbine. (a) Find the reheat temperature so the turbine output is saturated vapor. For this reheat find the total turbine power output and the boiler heat transfer. (b) The effect of a number of reheat stages on the ideal steam reheat cycle is to be studied using two reheat stages, one stage at 1.2 MPa and the second at 0.2 MPa, instead of the single reheat stage at 0.8 MPa.







Reheat Rankine Cycle

Both turbine sections

$$\dot{W}_{T,tot} = \dot{m}w_{T,tot} = \dot{m}(h_3 - h_4 + h_5 - h_6)$$
$$= 4.176\ (3682.34 - 3093.26 + 3547.55 - 2583.19) = \mathbf{6487\ kW}$$

Both boiler sections

$$\dot{Q}_H = \dot{m}(h_3 - h_2 + h_5 - h_4)$$
$$= 4.176\ (3682.34 - 191.44 + 3547.55 - 3093.26) = \mathbf{16\ 475\ kW}$$

b.

C.V. Pump reversible, adiabatic and assume incompressible flow, work in

$$w_P = v_1(P_2 - P_1) = 0.00101\ m^3/kg \times (3000 - 9.593)\ kPa = 3.02\ kJ/kg,$$
$$h_2 = h_1 + w_P = 188.42 + 3.02 = 191.44\ kJ/kg$$

$$P_4 = P_5 = 1.2\ MPa,\ \ P_6 = P_7 = 0.2\ MPa$$

3: $h_3 = 3230.82$ kJ/kg, $s_3 = 6.9211$ kJ/kg K

4: P_4, $s_4 = s_3 \Rightarrow$ sup. vap. $h_4 = 2985.3$

5: $h_5 = 3260.7$ kJ/kg, $s_5 = 7.3773$ kJ/kg K

6: P_6, $s_6 = s_5 \Rightarrow$ sup. vapor

$h_6 = 2811.2$ kJ/kg

7: $h_7 = 3276.5$ kJ/kg, $s_7 = 8.2217$ kJ/kg K

Reheat Rankine Cycle

8: P_8. $s_8 = s_7 \Rightarrow$ sup. vapor $h_8 = 2602$ kJ/kg   (used CATT3)

Total turbine work. same flow rate through all sections

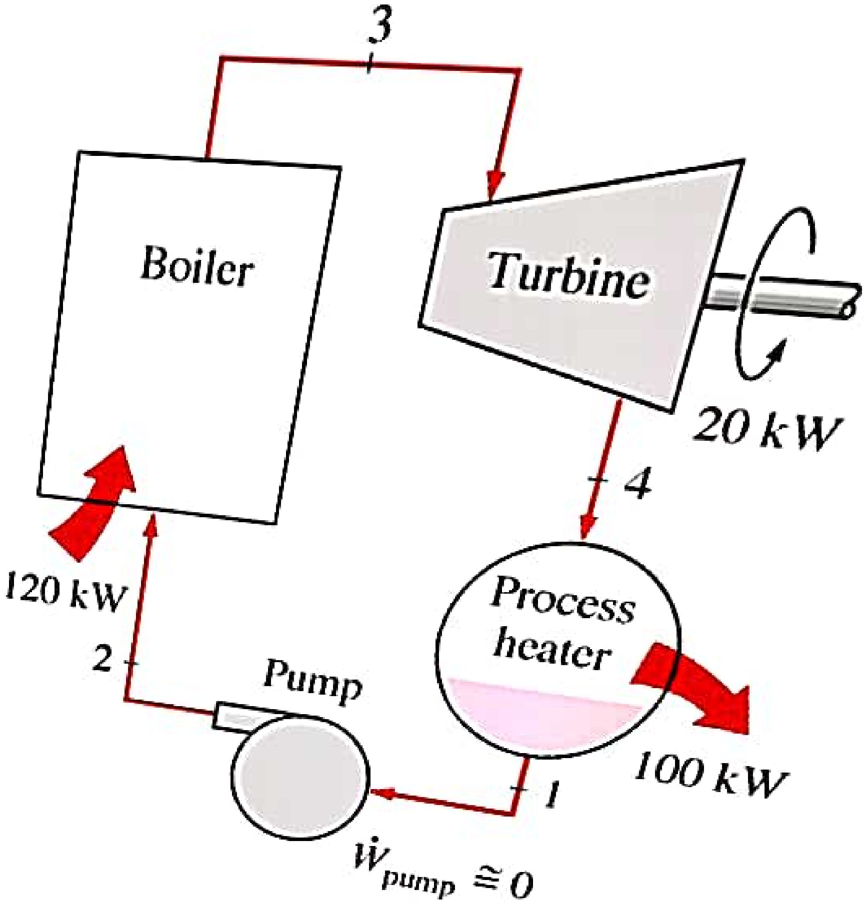
$$w_T = (h_3 - h_4) + (h_5 - h_6) + (h_7 - h_8) = 245.5 + 449.5 + 674.5 = 1369.5 \text{ kJ/kg}$$

Total heat transfer in boiler. same flow rate through all sections

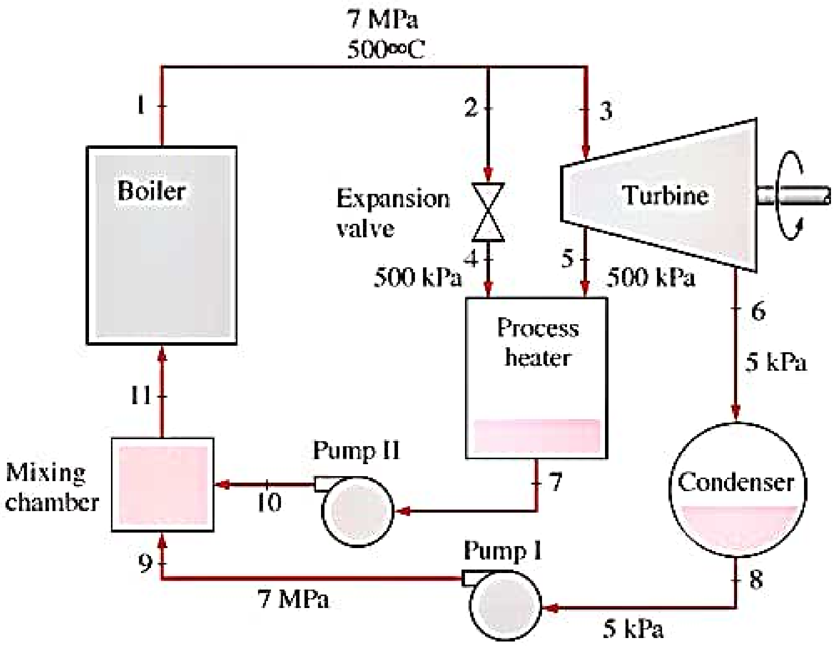
$$q_H = (h_3 - h_2) + (h_5 - h_4) + (h_7 - h_6) = 3039 + 319.8 + 465.3 = 3824.5 \text{ kJ/kg}$$

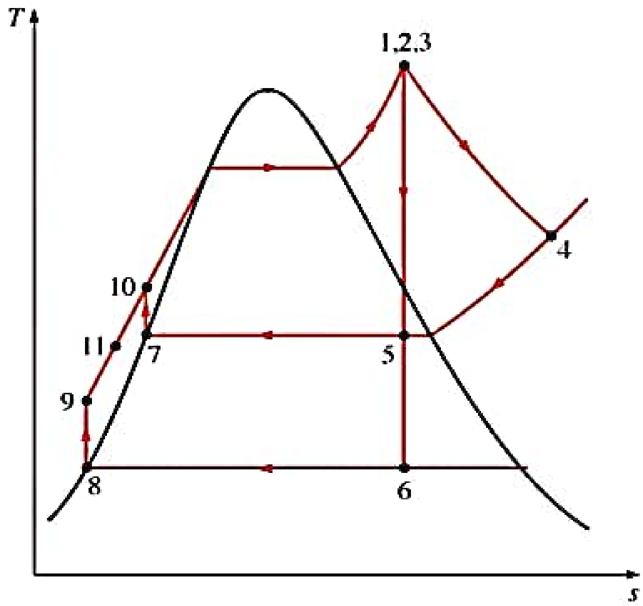
Cycle efficiency:
$$\eta_{TH} = \frac{w_T - w_P}{q_H} = \frac{1369.5 - 3.02}{3824.5} = \mathbf{0.357}$$

Ideal Cogeneration Plant



Schematic and T-s Diagram for Example 9-8





Mercury-Water Binary Vapor Cycle