Chapter 10

Vapor and Combined Power Cycles

Study Guide in PowerPoint

to accompany

Thermodynamics: An Engineering Approach, 7th edition
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Overview of a steam power plant

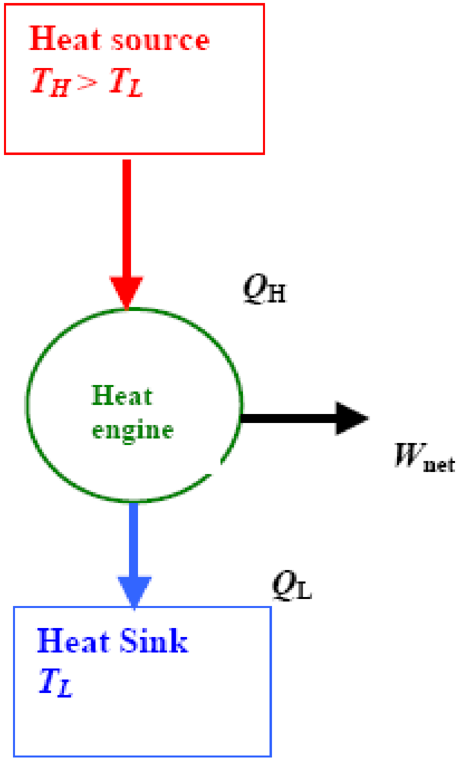
Photo courtesy of Progress Energy Carolinas, Inc.

We consider power cycles where the working fluid undergoes a phase change. The best example of this cycle is the steam power cycle where water (steam) is the working fluid.

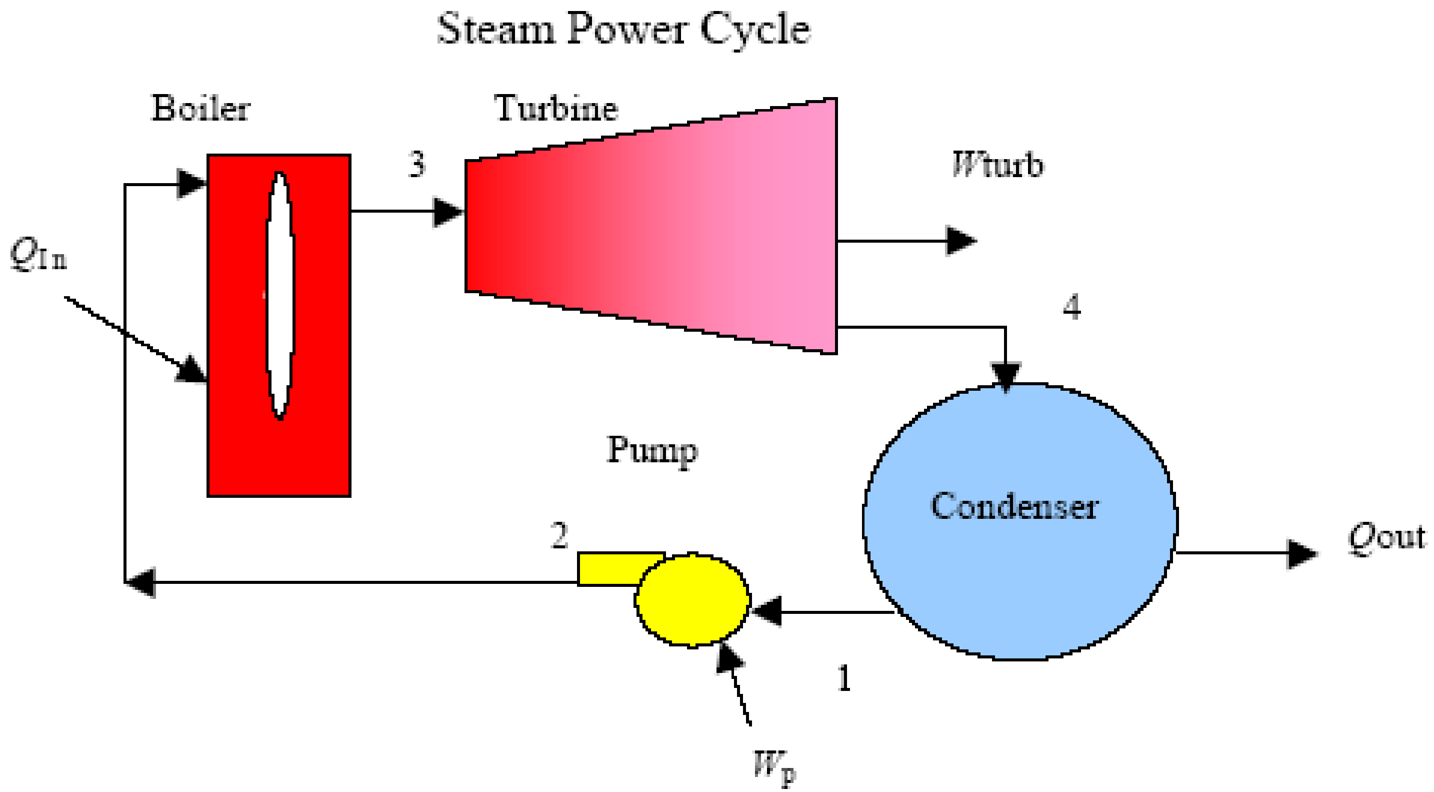
For more information and animations illustrating this topic visit the Animation Library developed by Professor S. Bhattacharjee, San Diego State University, at this link.

test.sdsu.edu/testhome/vtAnimations/index.html

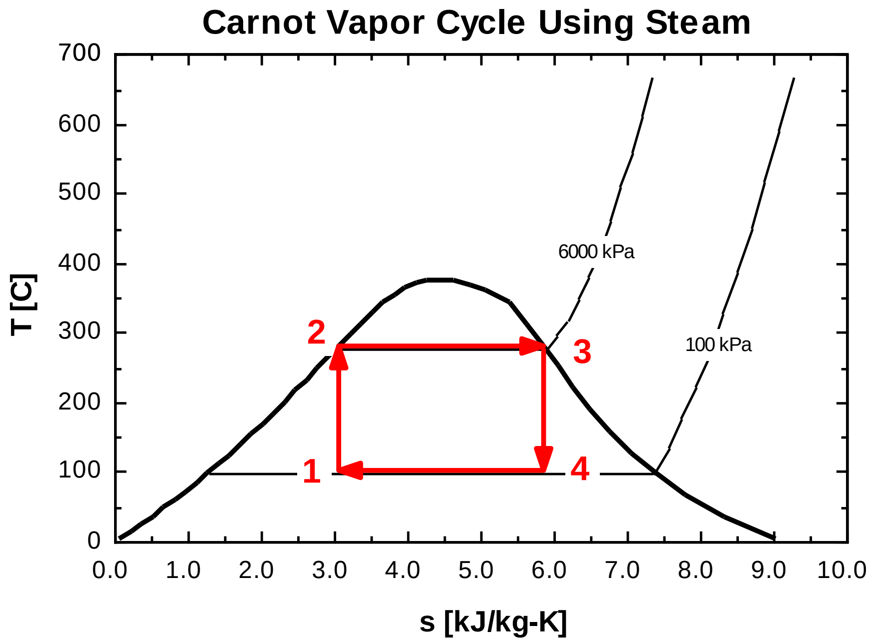
Carnot Vapor Cycle



The heat engine may be composed of the following components.



The working fluid, steam (water), undergoes a thermodynamic cycle from 1-2-3-4-1. The cycle is shown on the following *T-s* diagram.



The thermal efficiency of this cycle is given as

$$\eta_{th,\,Carnot} = \frac{W_{net}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}}$$

$$= 1 - \frac{T_L}{T_H}$$

Note the effect of T_H and T_L on $\eta_{th,\,Carnot}$.

- The larger the T_H the larger the $\eta_{th,\,Carnot}$
- The smaller the T_L the larger the $\eta_{th,\,Carnot}$

To increase the thermal efficiency in any power cycle, we try to increase the maximum temperature at which heat is added.

Reasons why the Carnot cycle is not used:
•Pumping process 1-2 requires the pumping of a mixture of saturated liquid and saturated vapor at state 1 and the delivery of a saturated liquid at state 2.
•To superheat the steam to take advantage of a higher temperature, elaborate controls are required to keep T_H constant while the steam expands and does work.

To resolve the difficulties associated with the Carnot cycle, the Rankine cycle was devised.

The steam power plant is composed of several distinct componets.

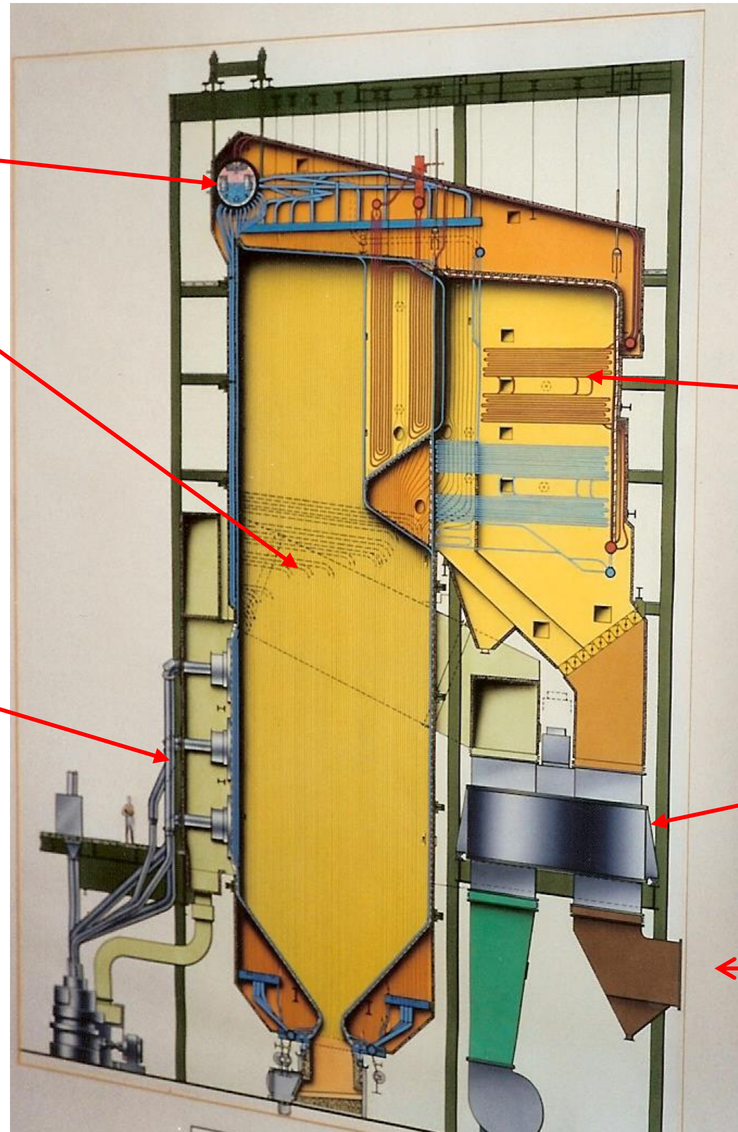
Steam generator or boiler
Turbine and electric generator
Condenser and cooling water system
Pumps

Steam generator or boiler.

Steam drum

Down comer and riser tubes

Pulverized Coal and air injectors



Heat exchangers for superheating, reheating, and water economizers

Air preheater

Exhaust

Photo courtesy of Progress Energy Carolinas, Inc.

Steam Drum

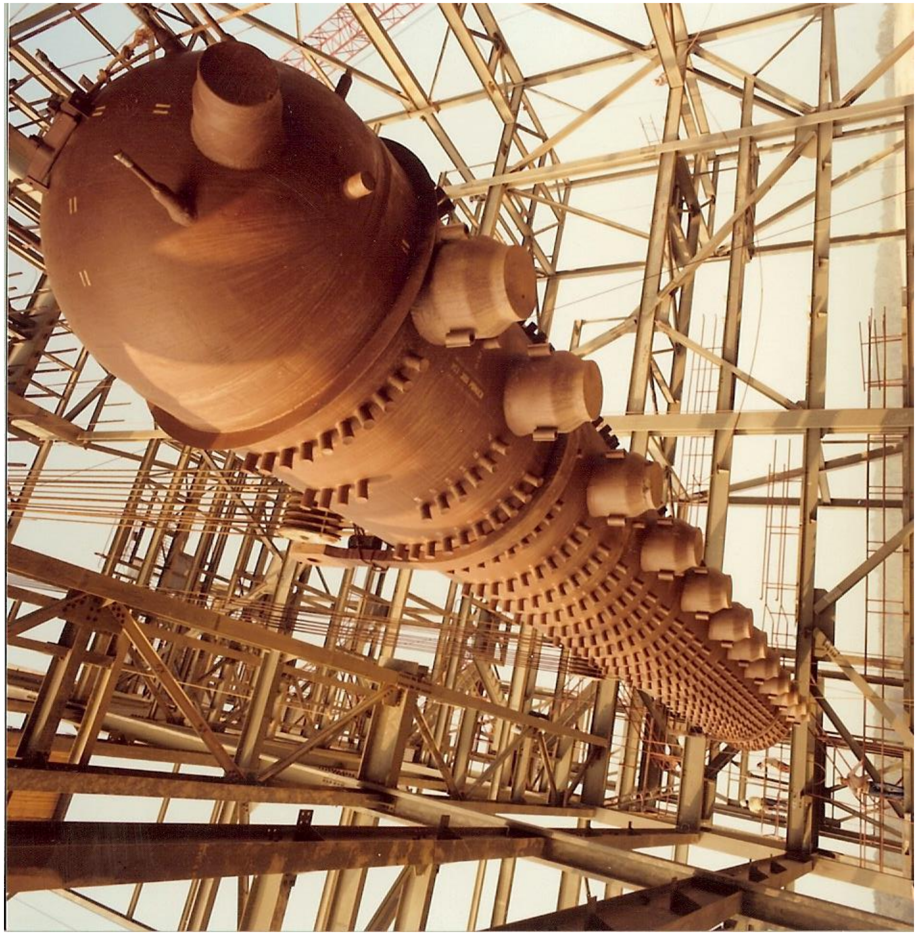


Photo courtesy of Progress Energy Carolinas, Inc.

Down comer and riser tubes forming the combustion chamber

Combustion chamber

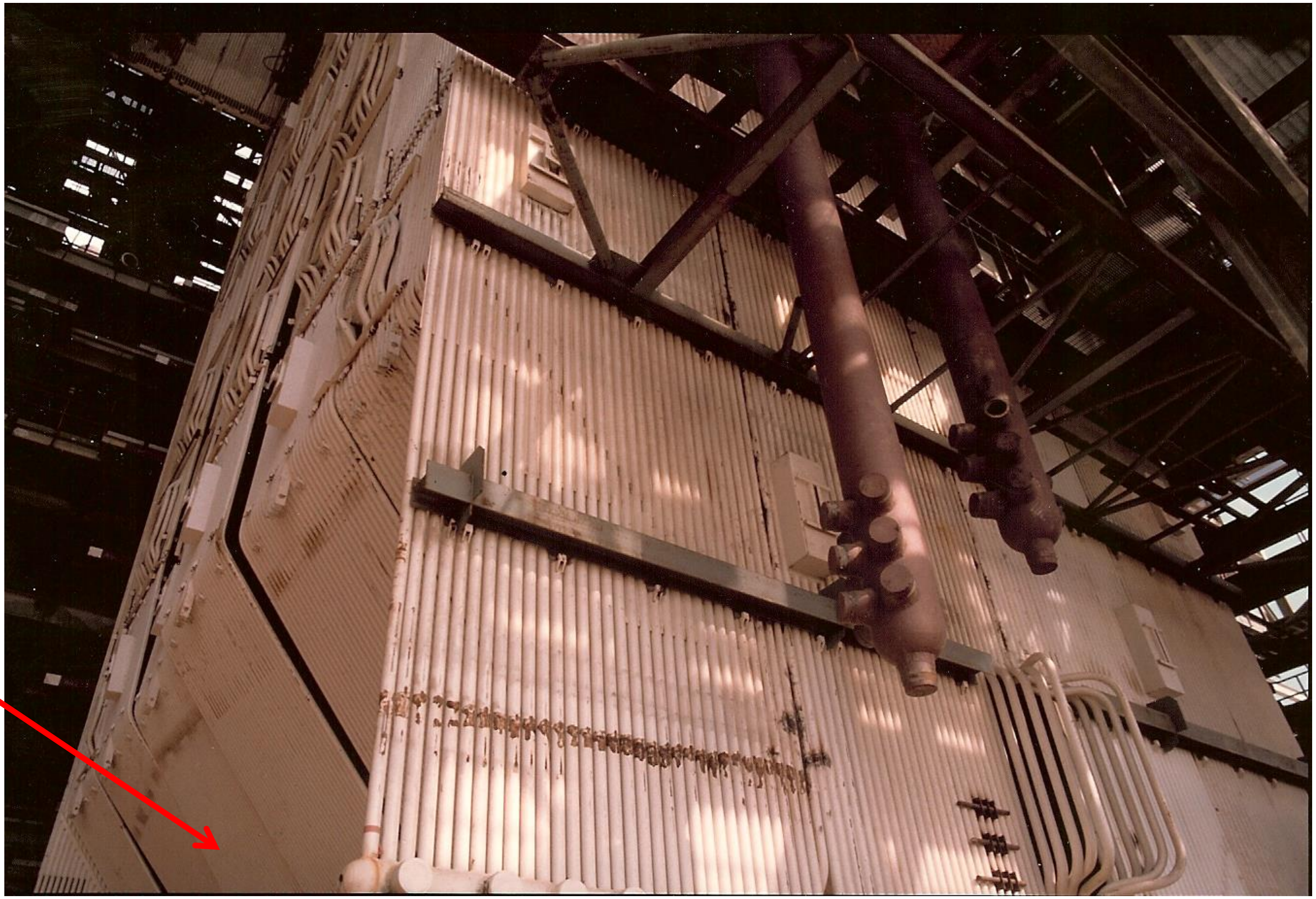


Photo courtesy of Progress Energy Carolinas

1. HP Turbine
2. IP Turbine
3. LP Turbine
4. Generator

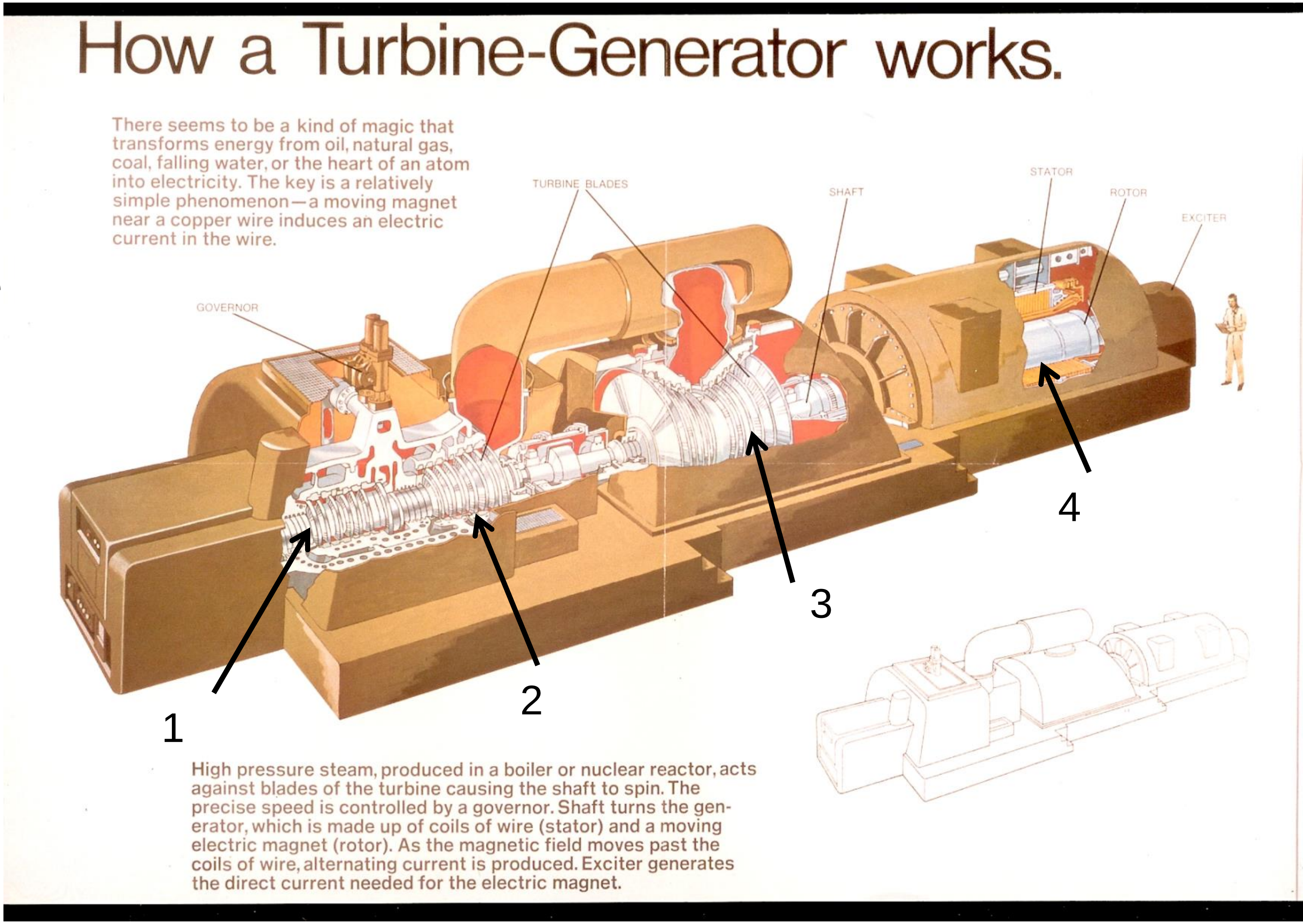


Photo courtesy of Progress Energy Carolinas, Inc.

Turbine bed for steam power plant.

Photo courtesy of Progress Energy Carolinas, Inc.

Installation of low pressure turbine into turbine bed

Photo courtesy of Progress Energy Carolinas, Inc.

Inspection of low pressure turbine blades

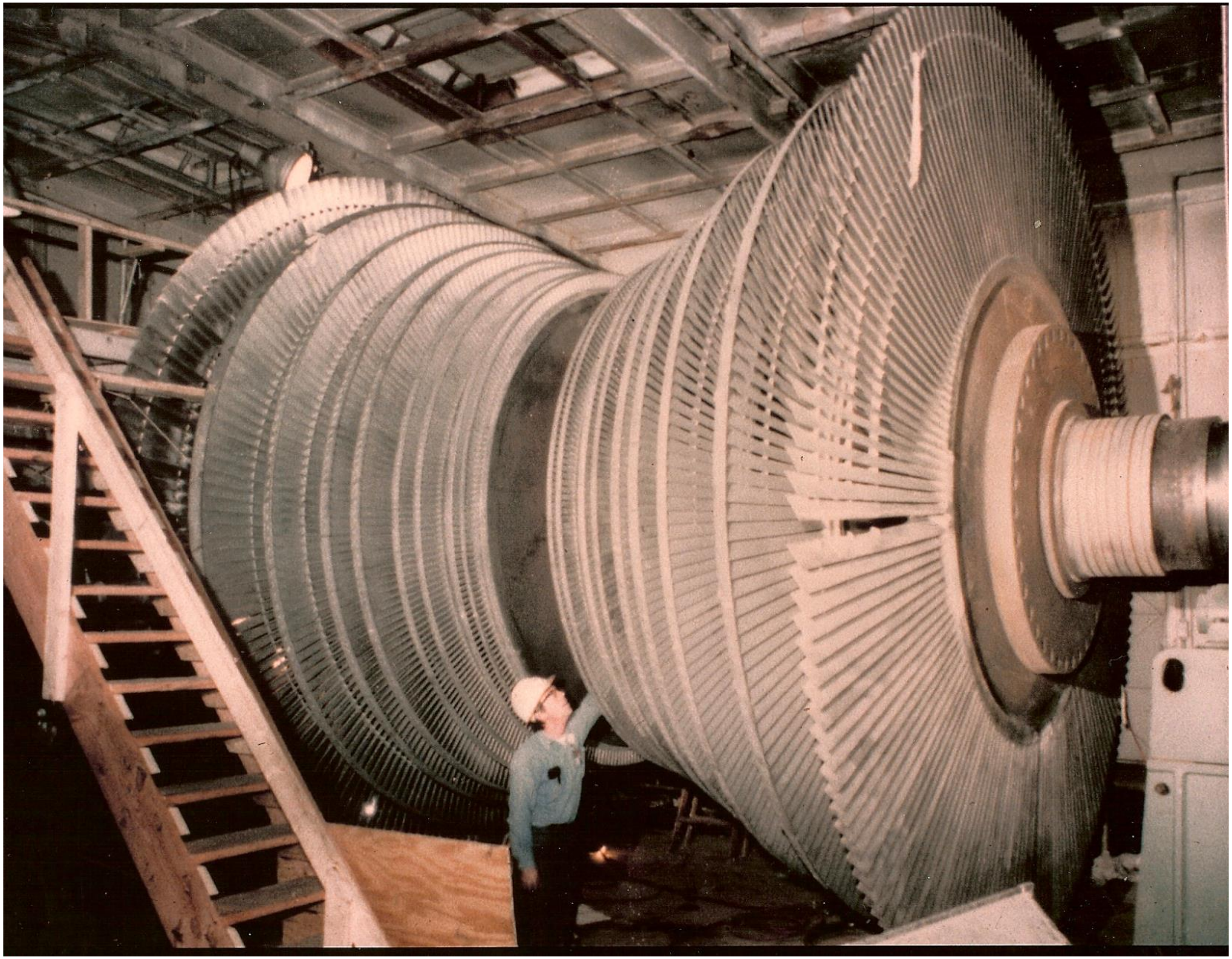


Photo courtesy of Progress Energy Carolinas, Inc.

End view of the cooling water tubes in a condenser

Photo courtesy of Progress Energy Carolinas, Inc.

Nuclear power plant using a cooling tower to supply cooling water to the condenser

Photo courtesy of Progress Energy Carolinas, Inc.

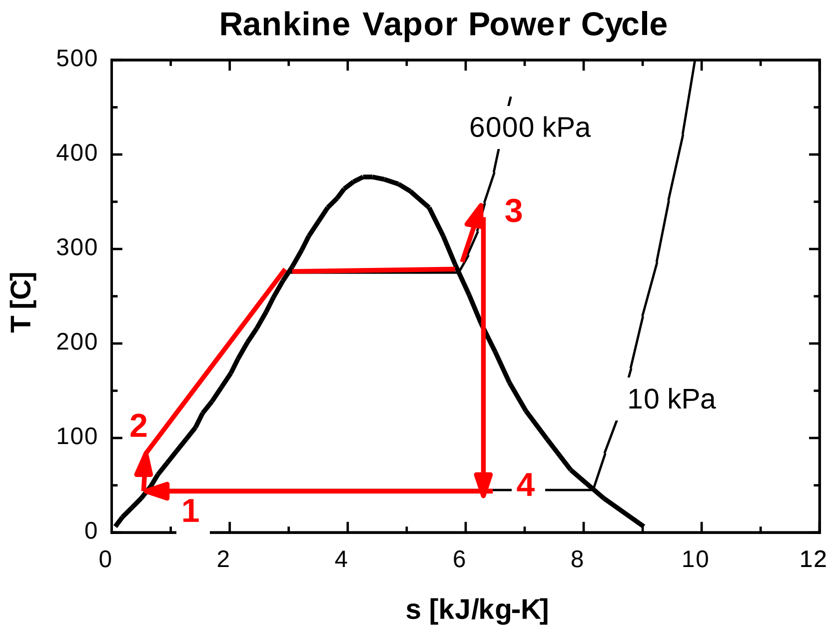
Rankine Cycle

The simple Rankine cycle has the same component layout as the Carnot cycle shown above. The simple Rankine cycle continues the condensation process 4-1 until the saturated liquid line is reached.

Ideal Rankine Cycle Processes

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

The *T-s* diagram for the Rankine cycle is given below. Locate the processes for heat transfer and work on the diagram.



Example 10-1

Compute the thermal efficiency of an ideal Rankine cycle for which steam leaves the boiler as superheated vapor at 6 MPa, 350°C, and is condensed at 10 kPa.

We use the power system and *T-s* diagram shown above.

$P_2 = P_3 = 6$ MPa = 6000 kPa

$T_3 = 350^oC$

$P_1 = P_4 = 10$ kPa

Pump

The pump work is obtained from the conservation of mass and energy for steady-flow but neglecting potential and kinetic energy changes and assuming the pump is adiabatic and reversible.

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

$$\dot{m}_1 h_1 + \dot{W}_{pump} = \dot{m}_2 h_2$$

$$\dot{W}_{pump} = \dot{m}(h_2 - h_1)$$

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

Recall the property relation:

$$dh = T\,ds + v\,dP$$

Since the ideal pumping process 1-2 is isentropic, $ds = 0$.

$$dh = v\,dP$$

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

The incompressible liquid assumption allows

$$v \cong v_1 = const.$$
$$h_2 - h_1 \cong v_1(P_2 - P_1)$$

The pump work is calculated from

$$\dot{W}_{pump} = \dot{m}(h_2 - h_1) \cong \dot{m}v_1(P_2 - P_1)$$
$$w_{pump} = \frac{\dot{W}_{pump}}{\dot{m}} = v_1(P_2 - P_1)$$

Using the steam tables

$$\left.\begin{matrix} P_1 = 10\ kPa \\ Sat.\ liquid \end{matrix}\right\} \begin{cases} h_1 = h_f = 191.81 \dfrac{kJ}{kg} \\ v_1 = v_f = 0.00101 \dfrac{m^3}{kg} \end{cases}$$

$$\begin{aligned} w_{pump} &= v_1(P_2 - P_1) \\ &= 0.00101 \frac{m^3}{kg}(6000 - 10)\ kPa \frac{kJ}{m^3 kPa} \\ &= 6.05 \frac{kJ}{kg} \end{aligned}$$

Now, h_2 is found from

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

$$= 6.05\frac{kJ}{kg} + 191.81\frac{kJ}{kg}$$

$$= 197.86\frac{kJ}{kg}$$

Boiler

To find the heat supplied in the boiler, we apply the steady-flow conservation of mass and energy to the boiler. If we neglect the potential and kinetic energies, and note that no work is done on the steam in the boiler, then

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

$$\dot{m}_2 h_2 + \dot{Q}_{in} = \dot{m}_3 h_3$$

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

We find the properties at state 3 from the superheated tables as

$$\left.\begin{array}{l} P_3 = 6000\ kPa \\ T_3 = 350^o C \end{array}\right\} \begin{cases} h_3 = 3043.9 \dfrac{kJ}{kg} \\ s_3 = 6.3357 \dfrac{kJ}{kg \cdot K} \end{cases}$$

The heat transfer per unit mass is

$$q_{in} = \frac{\dot{Q}_{in}}{\dot{m}} = h_3 - h_2$$

$$= (3043.9 - 197.86) \frac{kJ}{kg}$$

$$= 2845.1 \frac{kJ}{kg}$$

Turbine

The turbine work is obtained from the application of the conservation of mass and energy for steady flow. We assume the process is adiabatic and reversible and neglect changes in kinetic and potential energies.

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

$$\dot{m}_3 h_3 = \dot{W}_{turb} + \dot{m}_4 h_4$$

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

We find the properties at state 4 from the steam tables by noting $s_4 = s_3 = 6.3357$ kJ/kg-K and asking three questions.

$$at\ \ P_4 = 10 kPa: s_f = 0.6492 \frac{kJ}{kg \cdot K};\ s_g = 8.1488 \frac{kJ}{kg \cdot K}$$

$$is\ \ s_4 < s_f\,?$$

$$is\ \ s_f < s_4 < s_g\,?$$

$$is\ \ s_g < s_4\,?$$

$$s_4 = s_f + x_4 s_{fg}$$

$$x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.3357 - 0.6492}{7.4996} = 0.758$$

$$h_4 = h_f + x_4 h_{fg}$$

$$= 191.81\frac{kJ}{kg} + 0.758(2392.1)\frac{kJ}{kg}$$

$$= 2005.0\frac{kJ}{kg}$$

The turbine work per unit mass is

$$w_{turb} = h_3 - h_4$$

$$= (3043.9 - 2005.0)\frac{kJ}{kg}$$

$$= 1038.9\frac{kJ}{kg}$$

The net work done by the cycle is

$$w_{net} = w_{turb} - w_{pump}$$

$$= (1038.9 - 6.05)\frac{kJ}{kg}$$

$$= 1032.8\frac{kJ}{kg}$$

The thermal efficiency is

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{1032.8\frac{kJ}{kg}}{2845.1\frac{kJ}{kg}}$$

$$= 0.363 \; or \; 36.3\%$$

Ways to improve the simple Rankine cycle efficiency:

- Superheat the vapor
  Average temperature is higher during heat addition.
  Moisture is reduced at turbine exit (we want x_4 in the above example > 85 percent).

- Increase boiler pressure (for fixed maximum temperature)
  Availability of steam is higher at higher pressures.
  Moisture is increased at turbine exit.

- Lower condenser pressure
  Less energy is lost to surroundings.
  Moisture is increased at turbine exit.

Extra Assignment

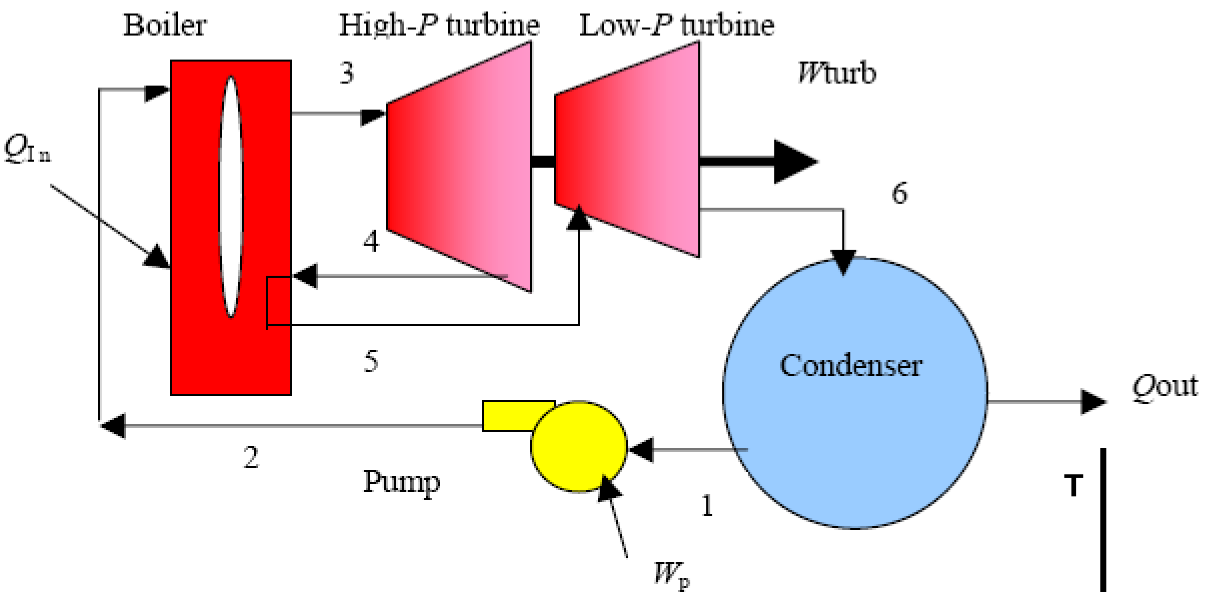
For the above example, find the heat rejected by the cycle and evaluate the thermal efficiency from

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

Reheat Cycle

As the boiler pressure is increased in the simple Rankine cycle, not only does the thermal efficiency increase, but also the turbine exit moisture increases. The reheat cycle allows the use of higher boiler pressures and provides a means to keep the turbine exit moisture (x > 0.85 to 0.90) at an acceptable level.

Steam Power Cycle with Reheat



Let's sketch the *T-s* diagram for the reheat cycle.

Rankine Cycle with Reheat

Component	Process	First Law Result
Boiler	Const. P	$q_{in} = (h_3 - h_2) + (h_5 - h_4)$
Turbine	Isentropic	$w_{out} = (h_3 - h_4) + (h_5 - h_6)$
Condenser	Const. P	$q_{out} = (h_6 - h_1)$
Pump	Isentropic	$w_{in} = (h_2 - h_1) = v_1(P_2 - P_1)$

The thermal efficiency is given by

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

$$= \frac{(h_3 - h_4) + (h_5 - h_6) - (h_2 - h_1)}{(h_3 - h_2) + (h_5 - h_4)}$$

$$= 1 - \frac{h_6 - h_1}{(h_3 - h_2) + (h_5 - h_4)}$$

Example 10-2

Compare the thermal efficiency and turbine-exit quality at the condenser pressure for a simple Rankine cycle and the reheat cycle when the boiler pressure is 4 MPa, the boiler exit temperature is 400°C, and the condenser pressure is 10 kPa. The reheat takes place at 0.4 MPa and the steam leaves the reheater at 400°C.

	η_{th}	$x_{turb\ exit}$
No Reheat	35.3%	0.8159
With Reheat	35.9%	0.9664

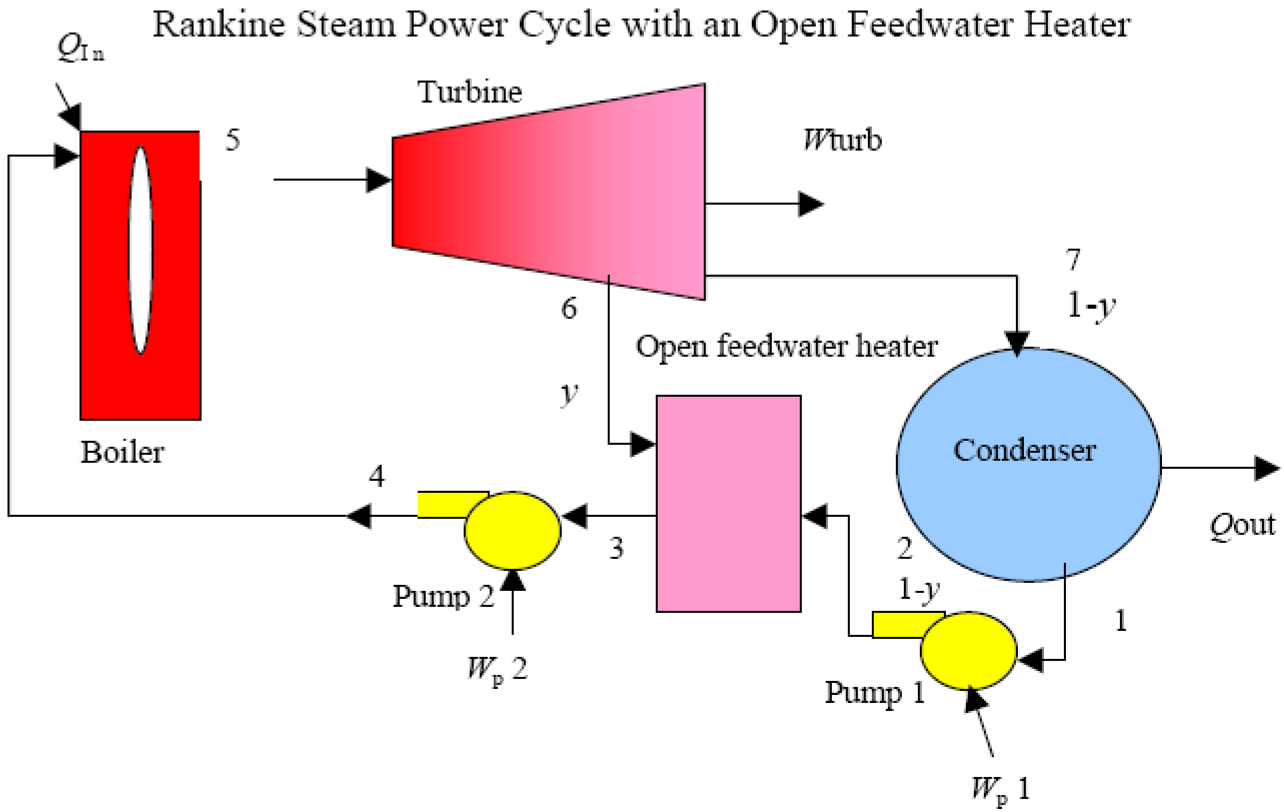
Regenerative Cycle

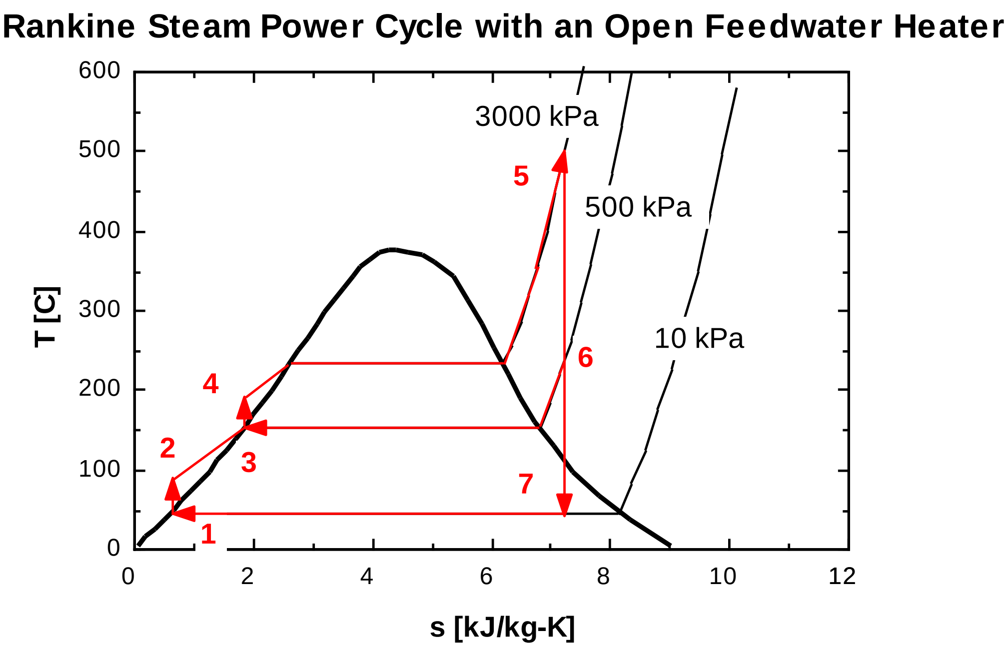
To improve the cycle thermal efficiency, the average temperature at which heat is added must be increased.

One way to do this is to allow the steam leaving the boiler to expand the steam in the turbine to an intermediate pressure. A portion of the steam is extracted from the turbine and sent to a regenerative heater to preheat the condensate before entering the boiler. This approach increases the average temperature at which heat is added in the boiler. However, this reduces the mass of steam expanding in the lower-pressure stages of the turbine, and, thus, the total work done by the turbine. The work that is done is done more efficiently.

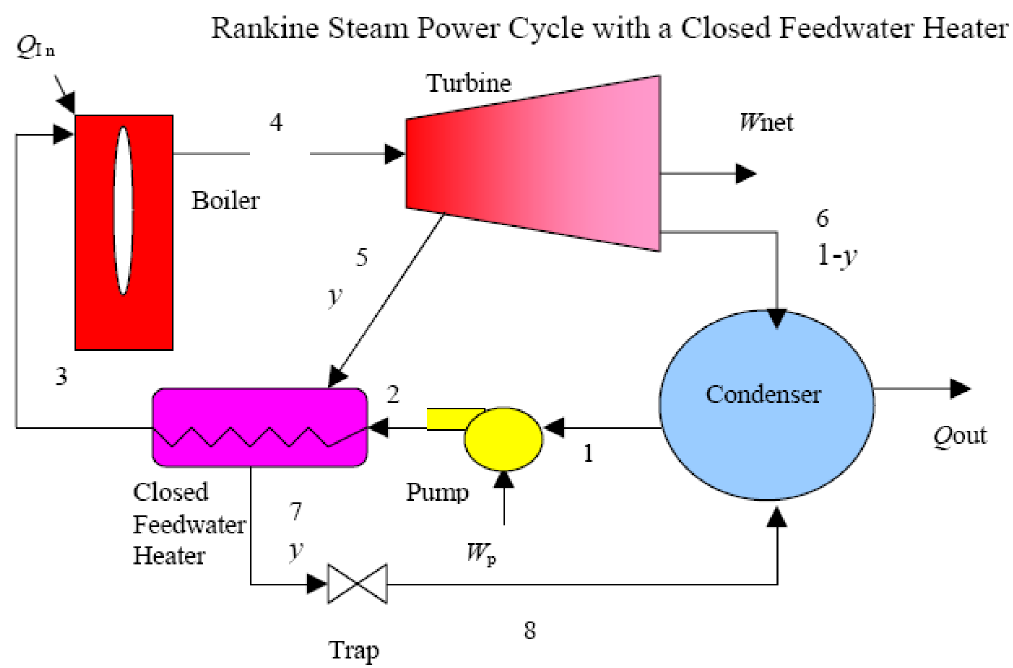
The preheating of the condensate is done in a combination of open and closed heaters. In the open feedwater heater, the extracted steam and the condensate are physically mixed. In the closed feedwater heater, the extracted steam and the condensate are not mixed.

Cycle with an open feedwater heater

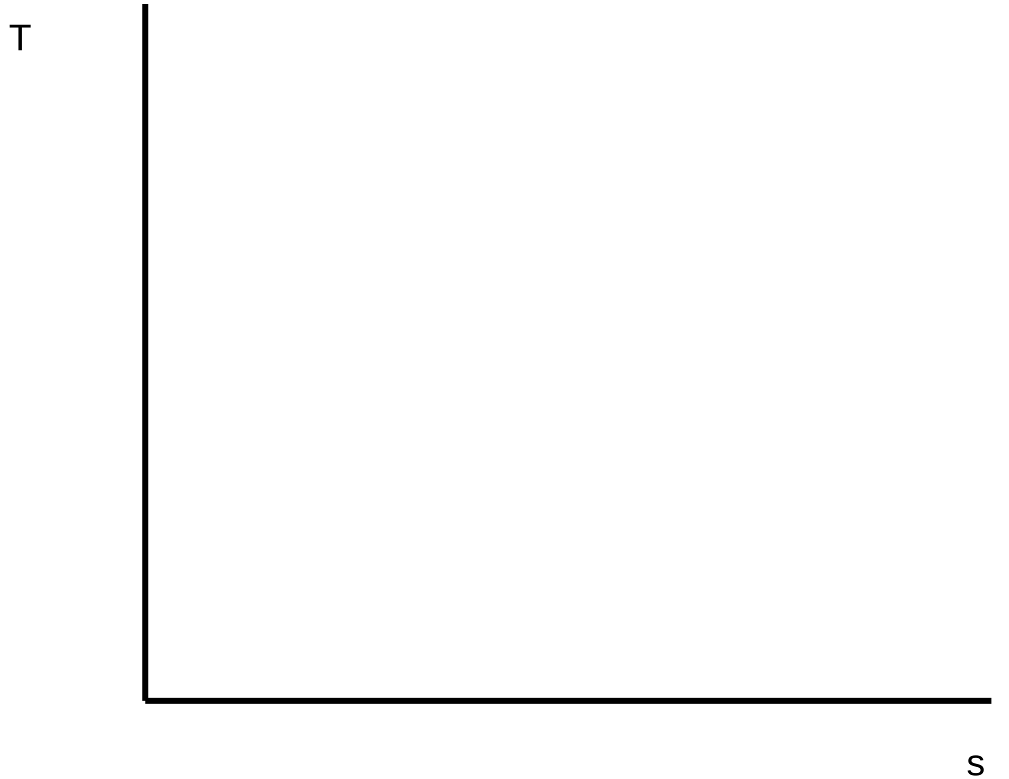




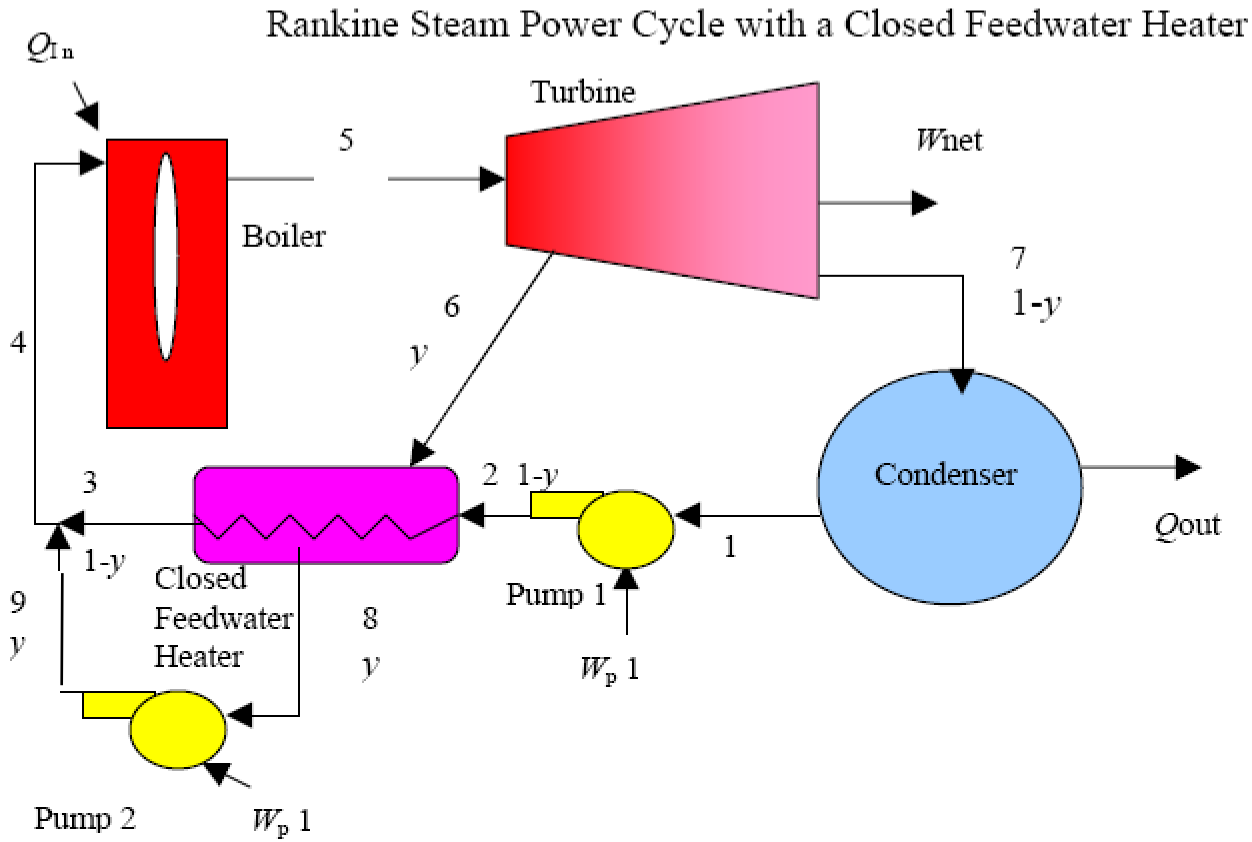
Cycle with a closed feedwater heater with steam trap to condenser



Let’s sketch the *T-s* diagram for this closed feedwater heater cycle.



Cycle with a closed feedwater heater with pump to boiler pressure



Let’s sketch the *T-s* diagram for this closed feedwater heater cycle.

Consider the regenerative cycle with the open feedwater heater.

To find the fraction of mass to be extracted from the turbine, apply the first law to the feedwater heater and assume, in the ideal case, that the water leaves the feedwater heater as a saturated liquid. (In the case of the ideal closed feedwater heater, the feedwater leaves the heater at a temperature equal to the saturation temperature at the extraction pressure.)

Conservation of mass for the open feedwater heater:

Let $y = \dot{m}_6 / \dot{m}_5$ be the fraction of mass extracted from the turbine for the feedwater heater.

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

$$\dot{m}_6 + \dot{m}_2 = \dot{m}_3 = \dot{m}_5$$

$$\dot{m}_2 = \dot{m}_5 - \dot{m}_6 = \dot{m}_5 (1 - y)$$

Conservation of energy for the open feedwater heater:

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

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

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

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

Example 10-3

An ideal regenerative steam power cycle operates so that steam enters the turbine at 3 MPa, 500°C, and exhausts at 10 kPa. A single open feedwater heater is used and operates at 0.5 MPa. Compute the cycle thermal efficiency.

The important properties of water for this cycle are shown below.

States with selected properties				
State	P kPa	T °C	h kJ/kg	s kJ/kg-K
1	10			
2	500			
3	500			
4	3000			
5	3000	500	3457.2	7.2359
6	500		2942.6	7.2359
7	10		2292.7	7.2359

Selected saturation properties			
P kPa	T_{sat} °C	v_f m^3/kg	h_f kJ/kg
10	45.81	0.00101	191.8
500	151.83	0.00109	640.1
3000	233.85	0.00122	1008.3

The work for pump 1 is calculated from

$$w_{pump\,1} = v_1(P_2 - P_1)$$

$$= 0.00101\frac{m^3}{kg}(500-10)\ kPa\,\frac{kJ}{m^3 kPa}$$

$$= 0.5\frac{kJ}{kg}$$

Now, h_2 is found from

$$h_2 = w_{pump\,1} + h_1$$

$$= 0.5\frac{kJ}{kg} + 191.8\frac{kJ}{kg}$$

$$= 192.3\frac{kJ}{kg}$$

The fraction of mass extracted from the turbine for the open feedwater heater is obtained from the energy balance on the open feedwater heater, as shown above.

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{(640.1 - 192.3)\dfrac{kJ}{kg}}{(2942.6 - 192.3)\dfrac{kJ}{kg}} = 0.163$$

This means that for each kg of steam entering the turbine, 0.163 kg is extracted for the feedwater heater.

The work for pump 2 is calculated from

$$\begin{aligned} w_{pump\,2} &= v_3(P_4 - P_3) \\ &= 0.00109\frac{m^3}{kg}(3000 - 500)\ kPa\frac{kJ}{m^3 kPa} \\ &= 2.7\frac{kJ}{kg} \end{aligned}$$

Now, h_4 is found from the energy balance for pump 2 for a unit of mass flowing through the pump.

$$E_{out} = E_{in}$$

$$h_4 = w_{pump\ 2} + h_3$$

$$= 2.7\frac{kJ}{kg} + 640.1\frac{kJ}{kg}$$

$$= 642.8\frac{kJ}{kg}$$

Apply the steady-flow conservation of energy to the isentropic turbine.

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

$$\dot{m}_5 h_5 = \dot{W}_{turb} + \dot{m}_6 h_6 + \dot{m}_7 h_7$$

$$\dot{W}_{turb} = \dot{m}_5[h_5 - y h_6 - (1-y)h_7]$$

$$w_{turb} = \frac{\dot{W}_{turb}}{\dot{m}_5} = h_5 - y h_6 - (1-y)h_7$$

$$= [3457.2 - (0.163)(2942.1) - (1-0.163)(2292.7)]\frac{kJ}{kg}$$

$$= 1058.6\frac{kJ}{kg}$$

The net work done by the cycle is

$$\dot{W}_{net} = \dot{W}_{turb} - \dot{W}_{pump\,1} - \dot{W}_{pump\,2}$$

$$\dot{m}_5 w_{net} = \dot{m}_5 w_{turb} - \dot{m}_1 w_{pump\,1} - \dot{m}_3 w_{pump\,2}$$

$$\dot{m}_5 w_{net} = \dot{m}_5 w_{turb} - \dot{m}_5 (1-y) w_{pump\,1} - \dot{m}_5 w_{pump\,2}$$

$$w_{net} = w_{turb} - (1-y) w_{pump\,1} - w_{pump\,2}$$

$$= [1058.6 - (1-0.163)(0.5) - 2.7] \frac{kJ}{kg}$$

$$= 1055.5 \frac{kJ}{kg}$$

Apply the steady-flow conservation of mass and energy to the boiler.

$$\dot{m}_4 = \dot{m}_5$$

$$\dot{m}_4 h_4 + \dot{Q}_{in} = \dot{m}_5 h_5$$

$$\dot{Q}_{in} = \dot{m}_5 (h_5 - h_4)$$

$$q_{in} = \frac{\dot{Q}_{in}}{\dot{m}_5} = h_5 - h_4$$

The heat transfer per unit mass entering the turbine at the high pressure, state 5, is

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

$$= (3457.2 - 642.8)\frac{kJ}{kg} = 2814.4\frac{kJ}{kg}$$

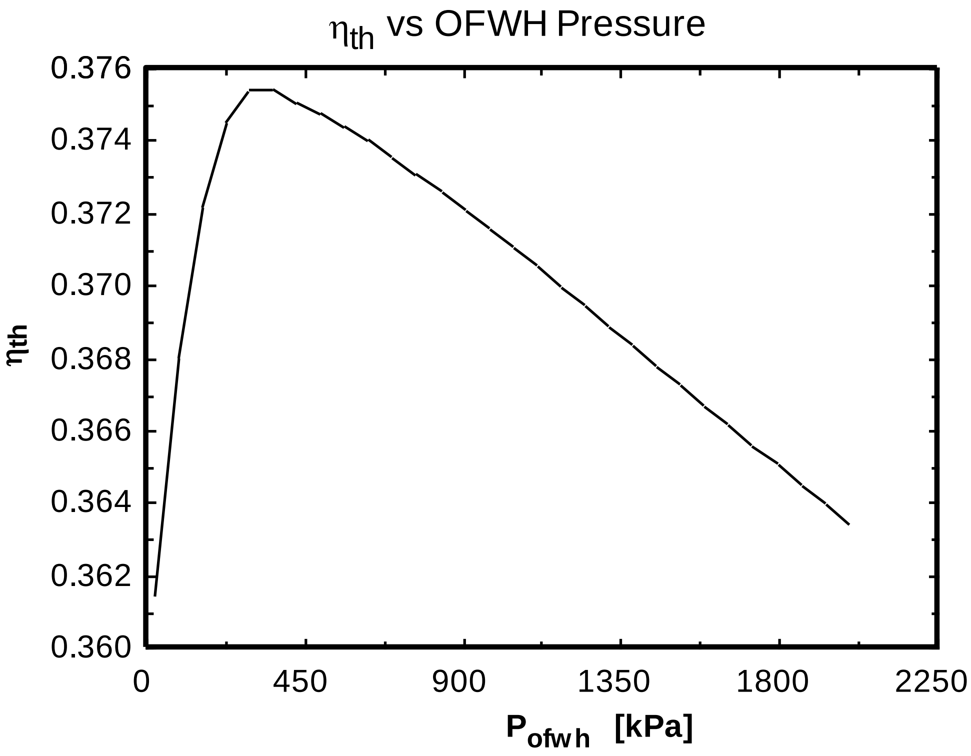
The thermal efficiency is

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{1055.5\frac{kJ}{kg}}{2814.4\frac{kJ}{kg}}$$

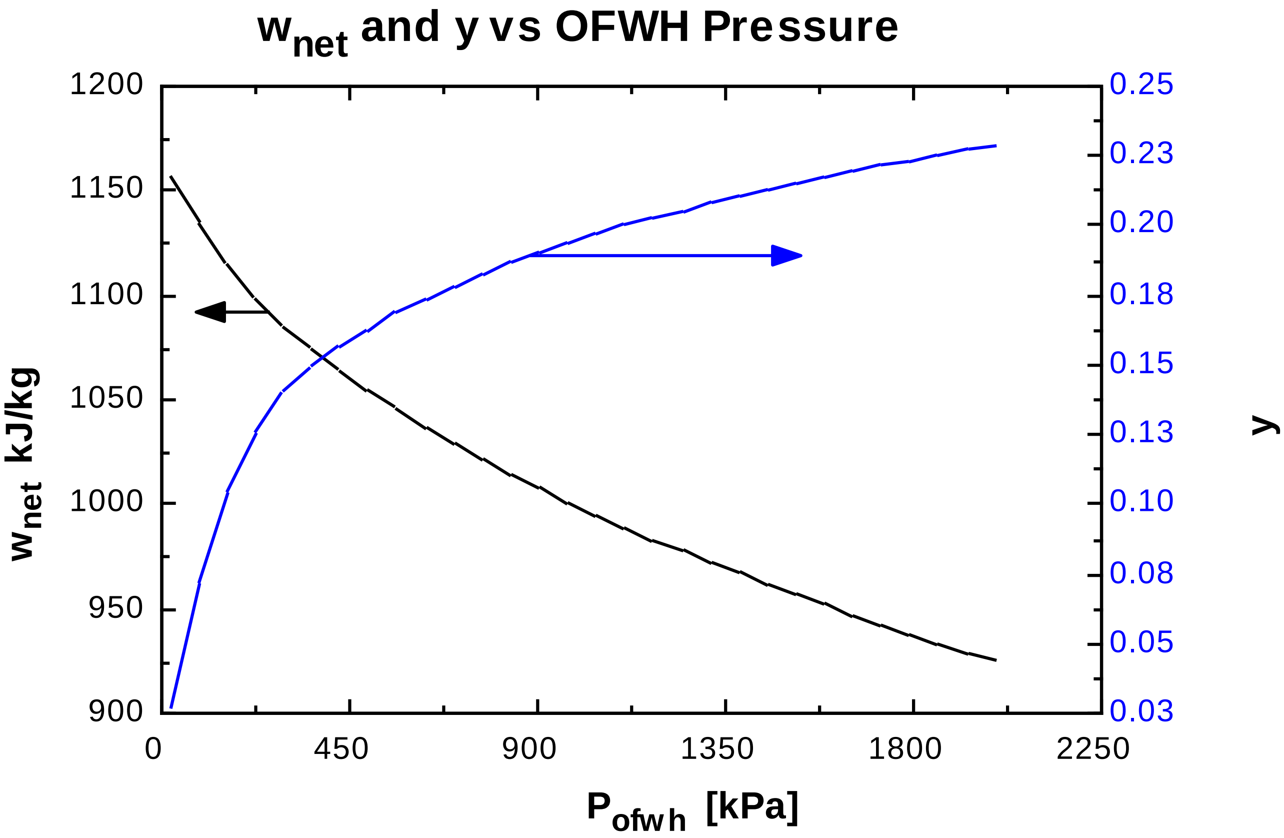
$$= 0.375 \;\; or \;\; 37.5\%$$

If these data were used for a Rankine cycle with no regeneration, then η_{th} = 35.6 percent. Thus, the one open feedwater heater operating at 0.5 MPa increased the thermal efficiency by 5.3 percent. However, note that the mass flowing through the lower-pressure turbine stages has been reduced by the amount extracted for the feedwater and the net work output for the regenerative cycle is about 10 percent lower than the standard Rankine cycle based on a unit of mass entering the turbine at the highest pressure.

Below is a plot of cycle thermal efficiency versus the open feedwater heater pressure. The feedwater heater pressure that makes the cycle thermal efficiency a maximum is about 400 kPa.



Below is a plot of cycle net work per unit mass flow at state 5 and the fraction of mass *y* extracted for the feedwater heater versus the open feedwater heater pressure. Clearly the net cycle work decreases and the fraction of mass extracted increases with increasing extraction pressure. Why does the fraction of mass extracted increase with increasing extraction pressure?



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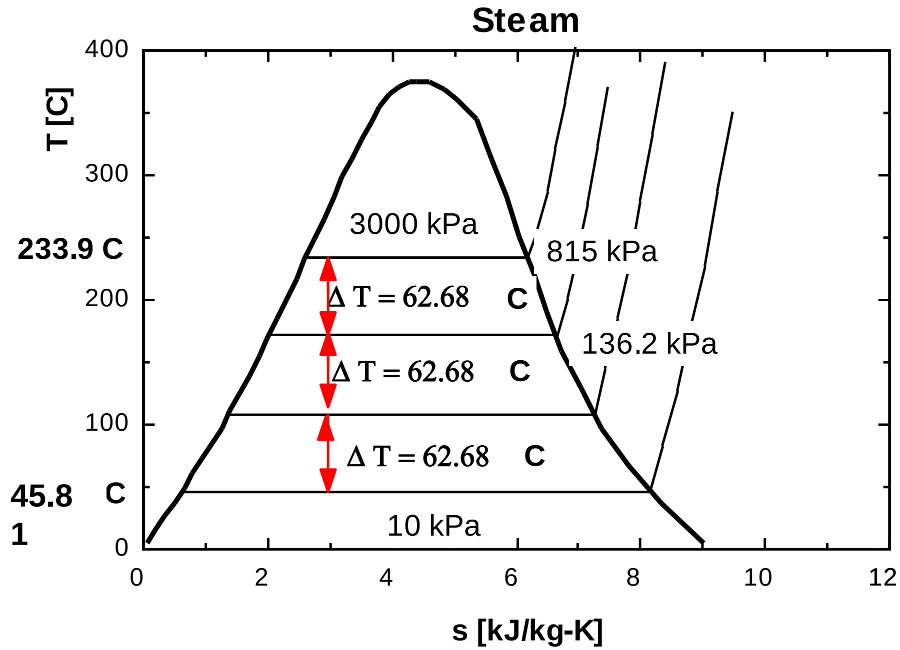
Placement of Feedwater Heaters

The extraction pressures for multiple feedwater heaters are chosen to maximize the cycle efficiency. As a rule of thumb, the extraction pressures for the feedwater heaters are chosen such that the saturation temperature difference between each component is about the same.

$$\Delta T_{cond\ to\ FWH} = \Delta T_{boiler\ to\ FWH},\ etc.$$

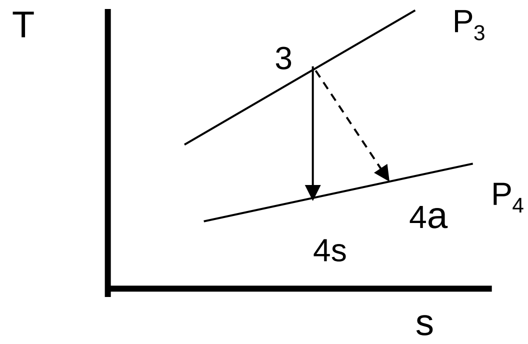
Example 10-4

An ideal regenerative steam power cycle operates so that steam enters the turbine at 3 MPa, 500°C, and exhausts at 10 kPa. Two closed feedwater heaters are to be used. Select starting values for the feedwater heater extraction pressures.



Deviation from Actual Cycles

- Piping losses--frictional effects reduce the available energy content of the steam.
- Turbine losses--turbine isentropic (or adiabatic) efficiency.

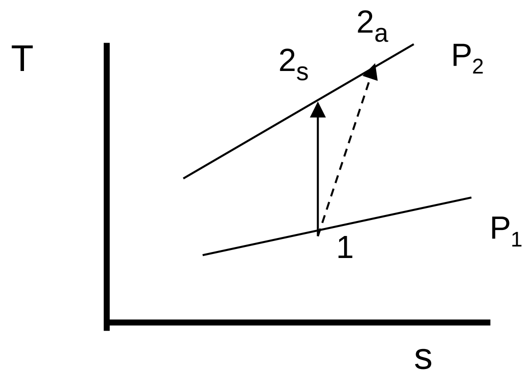


$$\eta_{turb} = \frac{w_{actual}}{w_{isentropic}} = \frac{h_3 - h_{4a}}{h_3 - h_{4s}}$$

The actual enthalpy at the turbine exit (needed for the energy analysis of the next component) is

$$h_{4a} = h_3 - \eta_{turb}(h_3 - h_{4s})$$

•Pump losses--pump isentropic (or adiabatic) efficiency.



$$\eta_{pump} = \frac{w_{isentropic}}{w_{actual}} = \frac{h_{2s} - h_1}{h_{2a} - h_1}$$

The actual enthalpy at the pump exit (needed for the energy analysis of the next component) is

$$h_{2a} = h_1 + \frac{1}{\eta_{pump}}(h_{2s} - h_1)$$

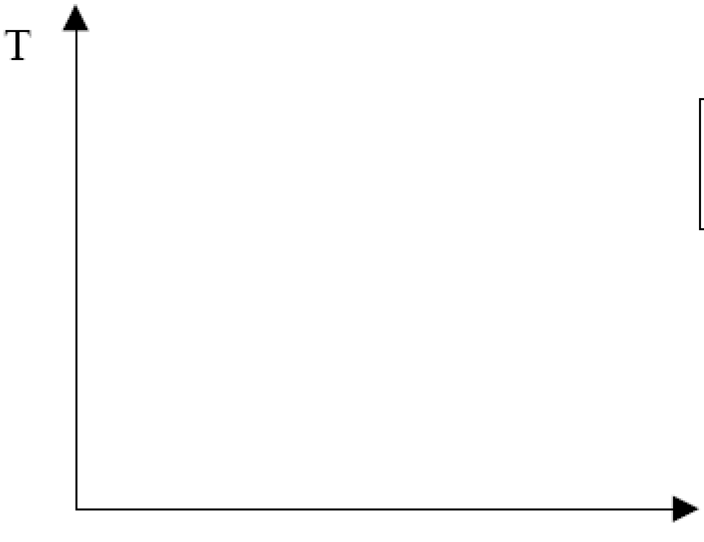
•Condenser losses--relatively small losses that result from cooling the condensate below the saturation temperature in the condenser.

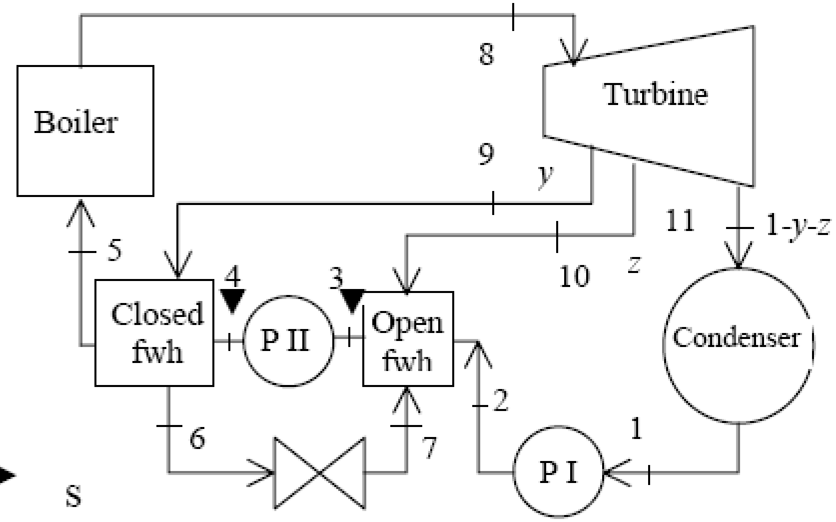
The following examples you should try on your own.

Regenerative Feedwater Heater problem

Consider an ideal steam regenerative Rankine cycle with two feedwater heaters, one closed and one open. Steam enters the turbine at 10 MPa and 500 C and exhausts to the condenser at 10 kPa. Steam is extracted from the turbine at 0.7 MPa for the closed feedwater heater and 0.3 MPa for the open one. The extracted steam leaves the closed feedwater heater and is subsequently throttled to the open feedwater heater. Show the cycle on a T-s diagram with respect to saturation lines, and using only the data presented in the data tables given below determine

a) the fraction of steam leaving the boiler that is extracted at 0.3 MPa z=0.1425
b) the fraction of steam leaving the boiler that is extracted at 0.7 MPa y=0.06213
c) the heat transfer from the condenser per unit mass leaving the boiler q_out=1509 kJ/kg
d) the heat transfer to the boiler per unit mass leaving the boiler q_in=2677 kJ/kg
e) the mass flow rate of steam through the boiler for a net power output of 250 MW m_dot=214.1 kg/s
f) the thermal efficiency of the cycle. Eta_th=0.4363





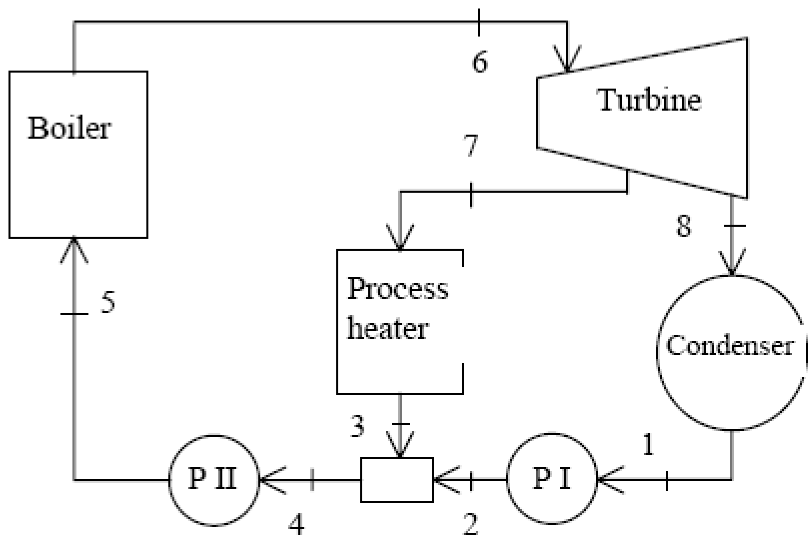
Selected Data Around the Cycle				
State	P kPa	T°C	h kJ/kg	s kJ/kg-K
1	10			
2	300		192	
3	300			
4	10000		571.9	
5	10000			
6	700			
7	300			
8	10000	500	3374	6.597
9	700		2715	6.597
10	300		2564	6.597
11	10		2089	6.597

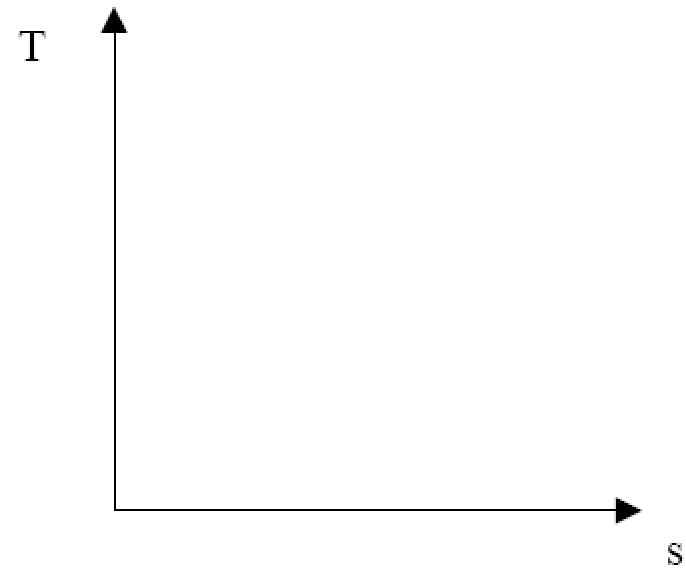
Saturation Data			
P kPa	T_{sat} °C	h_f kJ/kg	v_f m^3/kg
10	45.8	191.7	0.00101
300	133.6	561.5	0.00107
700	165	697.3	0.00111
10000	311	1407.6	0.00145

Cogeneration Plant

A cogeneration plant is to generate power and process heat. Consider an ideal cogeneration steam plant. Steam enters the turbine from the boiler at 7 MPa, 500 C and a mass flow rate of 30 kg/s. One-fourth of the steam is extracted from the turbine at 600-kPa pressure for process heating. The remainder of the steam continues to expand and exhausts to the condenser at 10 kPa. The steam extracted for the process heater is condensed in the heater and mixed with the feedwater at 600 kPa. The mixture is pumped to the boiler pressure of 7 MPa. Show the cycle on a T-s diagram with respect to saturation lines, and determine

a) the heat transfer from the process heater per unit mass leaving the boiler Qdot,process = 15,774 kW.
b) the net power produced by the cycle. Wdot,net = 32,848 kW.
c) the utilization factor of the plant Qdot,in = 92,753 kW, Utilization factor = 52.4%.





Selected Data Around the Cycle				
State	P kPa	T°C	h kJ/kg	s kJ/kg-K
1	10			
2	600		192	
3	600			
4	600			
5	7000		318.53	
6	7000	500	3410.3	6.7975
7	600		2773.7	6.7975
8	10		2153.2	6.7975

Saturation Data			
P kPa	T_{sat} °C	h_f kJ/kg	v_f m^3/kg
10	45.8	191.7	0.00101
600	158.8	670.6	0.00110
7000	285.9	1267	0.00135

Combined Gas-Steam Power Cycle

Example of the Combined Brayton and Rankine Cycles
(a) Explain whats happening in the various process for the hardware shown below.