Analysis of steam Power Plant cycle

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Outline

Chapter 1 Vapor Power Cycles

1.1. The Carnot Cycle

1.2. The Rankine Cycle

1.3. Deviation of Actual Vapor Power Cycles from Idealized Ones

1.4. The Ideal Reheat Rankine Cycle

1.5. The Ideal Regenerative Rankine Cycle

1.6. Types of Feed-Water Heaters

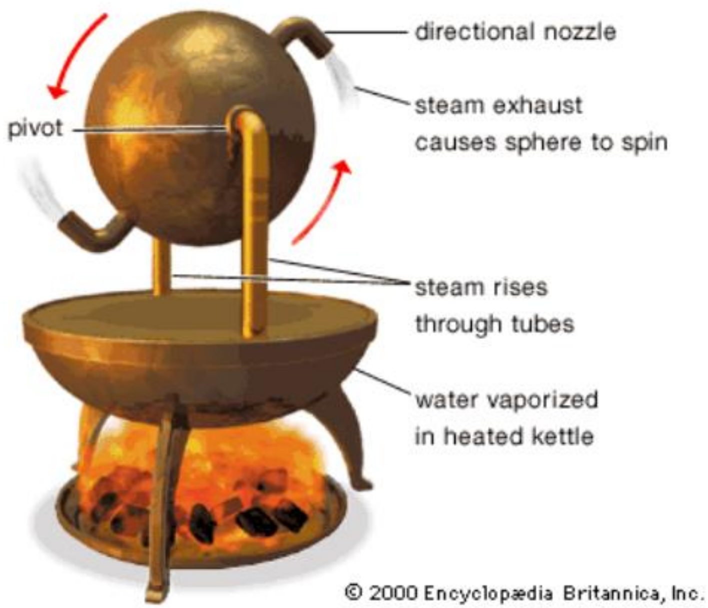
Introduction

Ancient History

Already the early humans learned how to boil water and found out that power can be extracted from the steam flow!

First documented steam power device by:

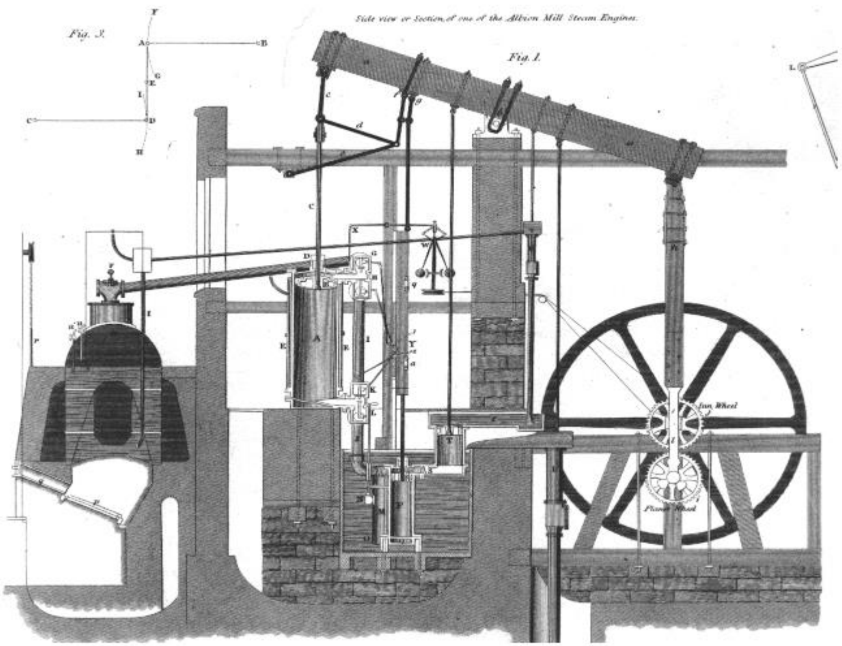
Heron of Alexandria, around year 60 A.D.



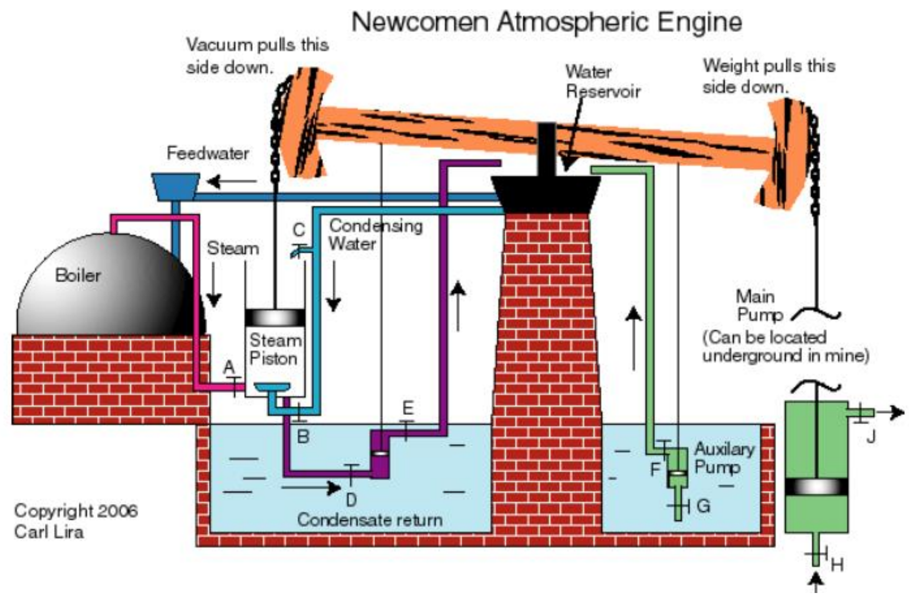
Introduction

The industrial revolution

Year 1690: Steam piston/cylinder patented by Denis Papin.
Year 1698: First patent of a steam pump by Thomas Savery.



The first large industrial steam engine by Thomas Newcomen, around year 1700…



Introduction

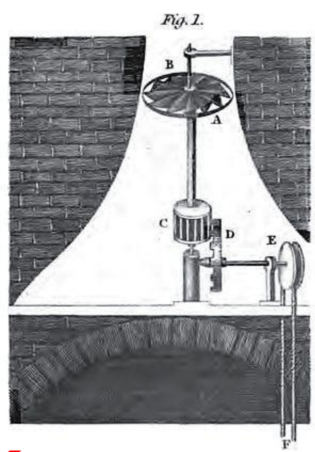
- Steam Engine: Definition
- A steam engine is a **heat** engine that converts **steam energy** into **mechanical motion**.
- Steam may be produced by combustion or no-combustion heat.
- **Combustion heat** sources include:
  - Hydrocarbon (Oil, Gas and Coal),
  - Biomass.
- **Non-combustion heat** sources include: Solar, Nuclear or Geothermal.

Evolution of Steam Engines

Hero's Aeolipile

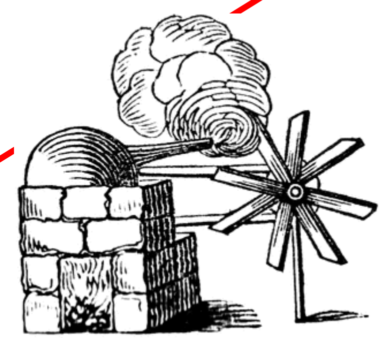
600 Years

100 BC:
the Aeolpile



1551
Taqi al-Din

Newton's Steam Wagon



1629
Giovanni Branca

State of the art Steam Turbine

100 BC: the Aeolpile

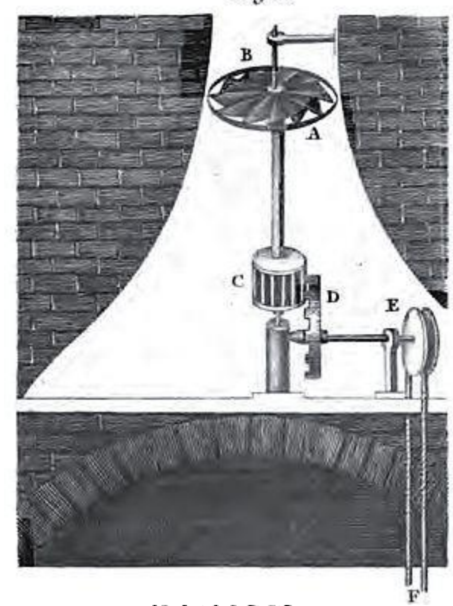
- Hero of Alexandria, developed the "aeolipile".
- It consisted of a boiler, two bent tubes mounted to a sphere.
- Steam coming from the boiler entered through the two tubes.
- The steam then exited through the bent tubes on the sphere, causing it to rotate.

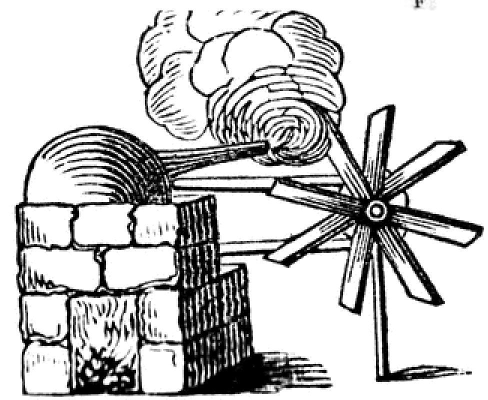
Hero's Aeolipile

Steam Engines by Taqi al-din and Branca

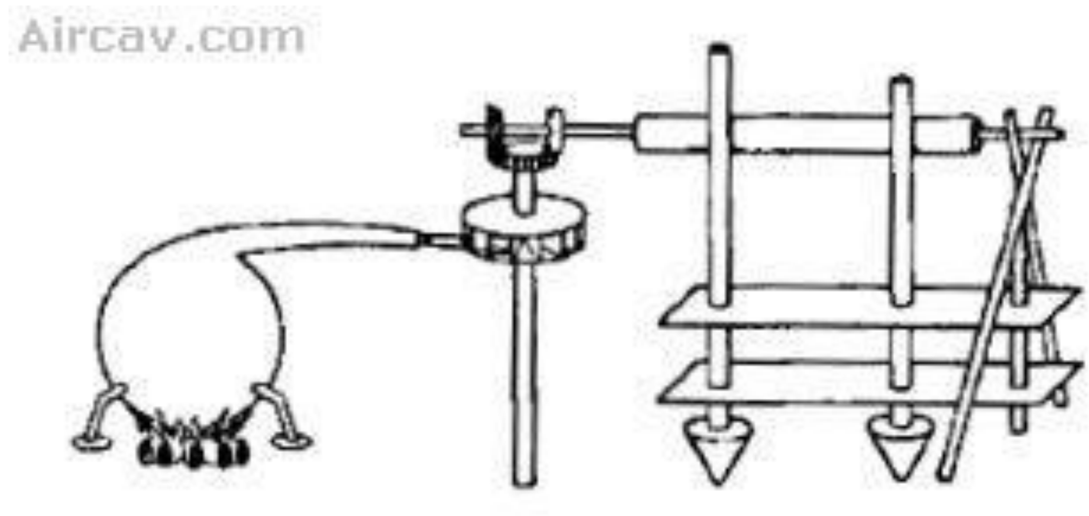
- In 1551, Taqi al-Din was the first to describe steam turbine device in his book “*Al-Turuq al-saniyyafi al-alat al-ruhaniyya*” (The Sublime Methods of Spiritual Machines).
- Later in 1629 Giovanni Branca described a related device.

Fig. 1.





1629: Branca's

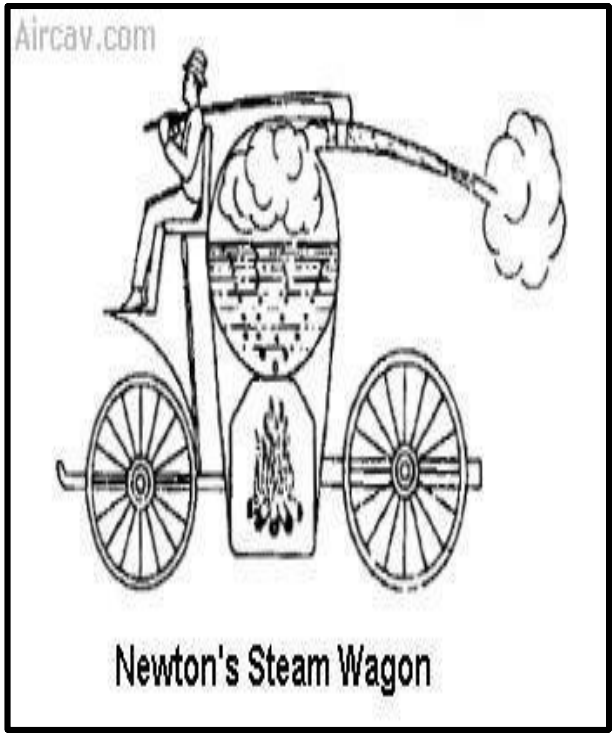


Branca's Jet Turbine

- In 1629 an Italian engineer, Giovanni Branca, described a stamping mill. A jet nozzle directed steam onto a horizontally mounted turbine wheel, which then turned an arrangement of gears that operated the stamping mill.

1687: Newton’s Wagon

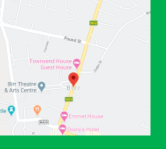
- Newton attempted to put his newly formulated laws of motion to the test.
- He tried to propel a wagon by directing steam through a nozzle pointed rearward
- Steam was produced by a boiler mounted on the wagon. Due to lack of power from the steam, this vehicle didn't operate.



But he conceived “moving” engine

Invention of steam turbine - *Charles Parsons*

- *A plentiful supply of cheap electricity, and much faster passenger steamships and military battleships. These were some of the things made possible by Charles Parsons, who grew up in Birr, and invented the steam turbine in 1887.*
- *Charles was born in 1854 and came from a brilliant scientific lineage. His father was the famous astronomer, William Parsons, who had built the world's largest telescope on the grounds of Birr Castle in the 1840s.*
- *The steam turbine invented by Charles, hugely increased the power that could be harnessed from a steam engine. The invention made him a rich man, and it changed the world.*



Introduction

- The industrial development was initiated with the steam engine
- Steam traction (locomotives and ships) and steam-engine drives (pumps, looms, presses, tooling machines, etc.) were completely ruling the industrial landscape until the 1920’s
- First thermal power plant –a coal-fired steam-engine driven, with overall efficiency of about 6 % (*Edison Electric Light Station - one in London and one in New York, year 1882)*

Introduction

- Modern application–advanced steam cycles for power production using steam turbines, whatever the source of heat, ranging from small-scale units and upto 1000+ MW per unit
- The fundamental steam cycle to day remains the same!

Introduction

- Steam cycles represent the reference technology for electric energy generation from cheap and low quality fuels (i.e. coal, oil, biomass, wastes); the thermodynamic cycle is in fact based on an external combustion process which allows decoupling the thermal power generation, which sustains the high temperature source of the cycle, from the working fluid of the thermodynamic process.

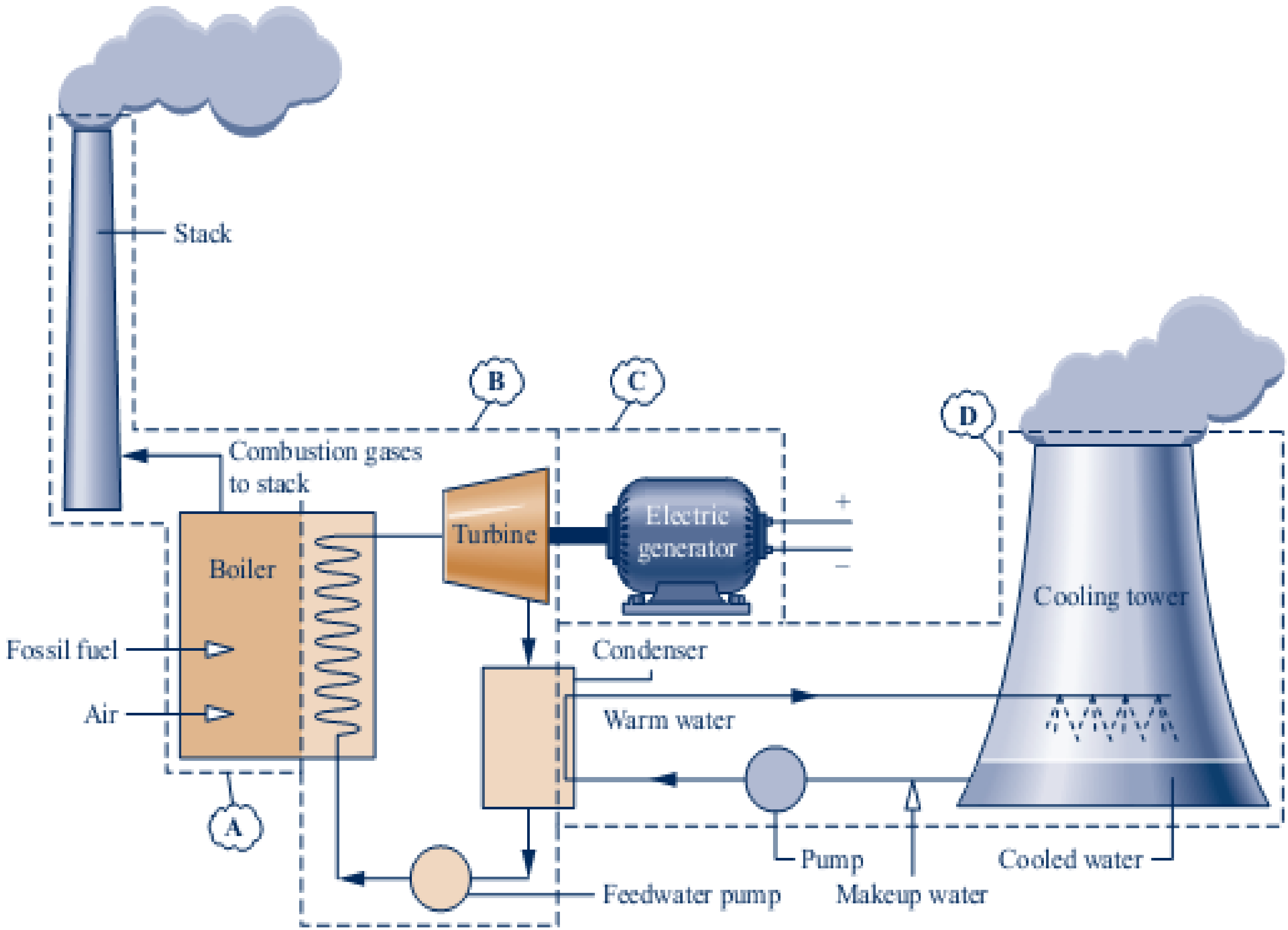
Introduction

- In other words, the exhaust gases of the combustion process do not directly contribute to the production of mechanical energy by passing through the machines of the thermodynamic cycle, but they exchange their thermal energy with the working fluid of the cycle. The main advantages and disadvantages of this technology shall be straighten out around this important characteristic

Introduction

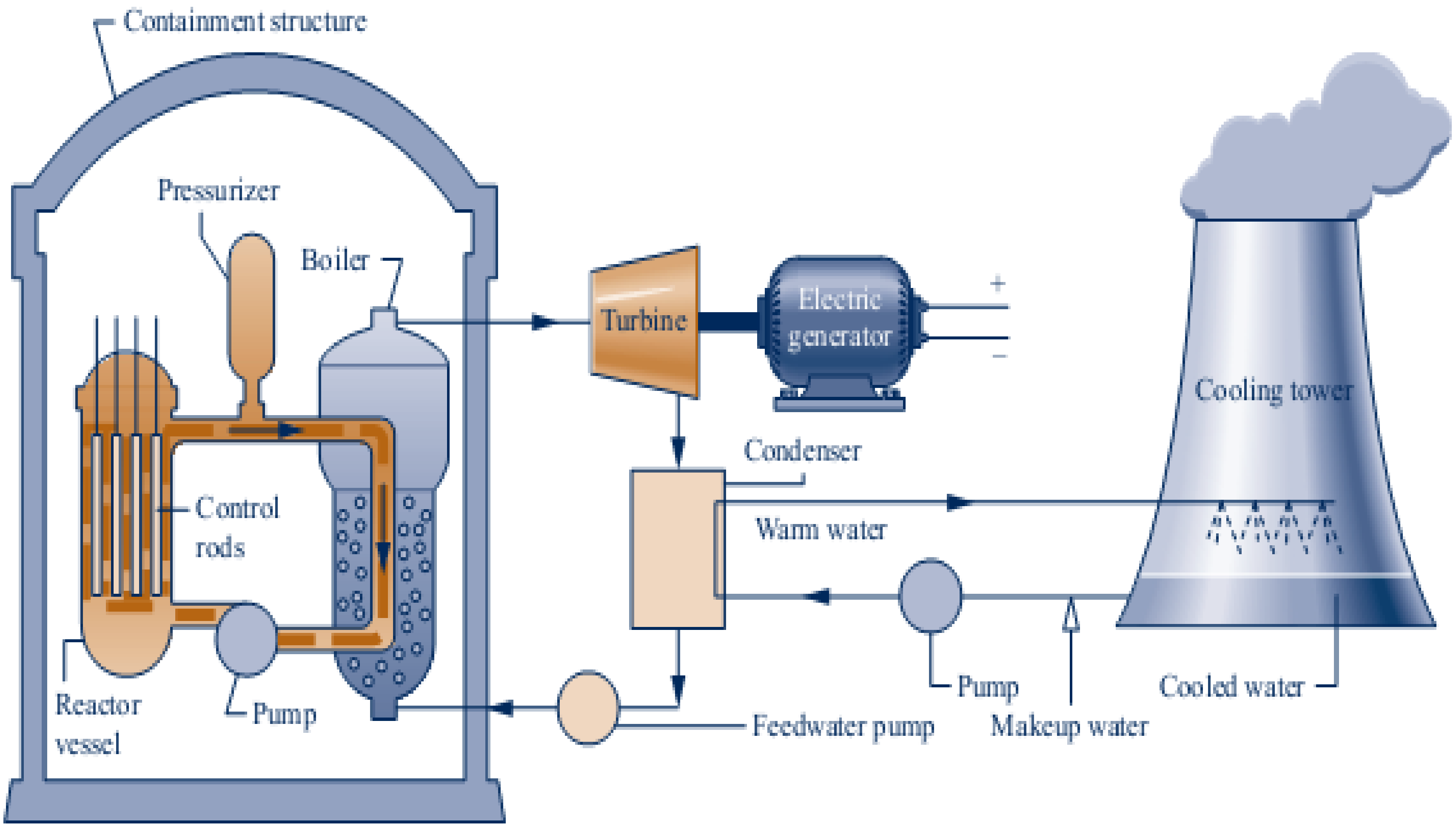
- Steam cycles have had a great importance in the transportation field; well-known examples of application are ships and trains moved by steam engines. Nowadays, however, these technologies have been overcome by others in this sector and are therefore mainly used for ***stationary power generation applications***. The thermodynamic cycle at the basis of steam power plants is the Rankine cycle

Fossil Fuel fired Power Plant



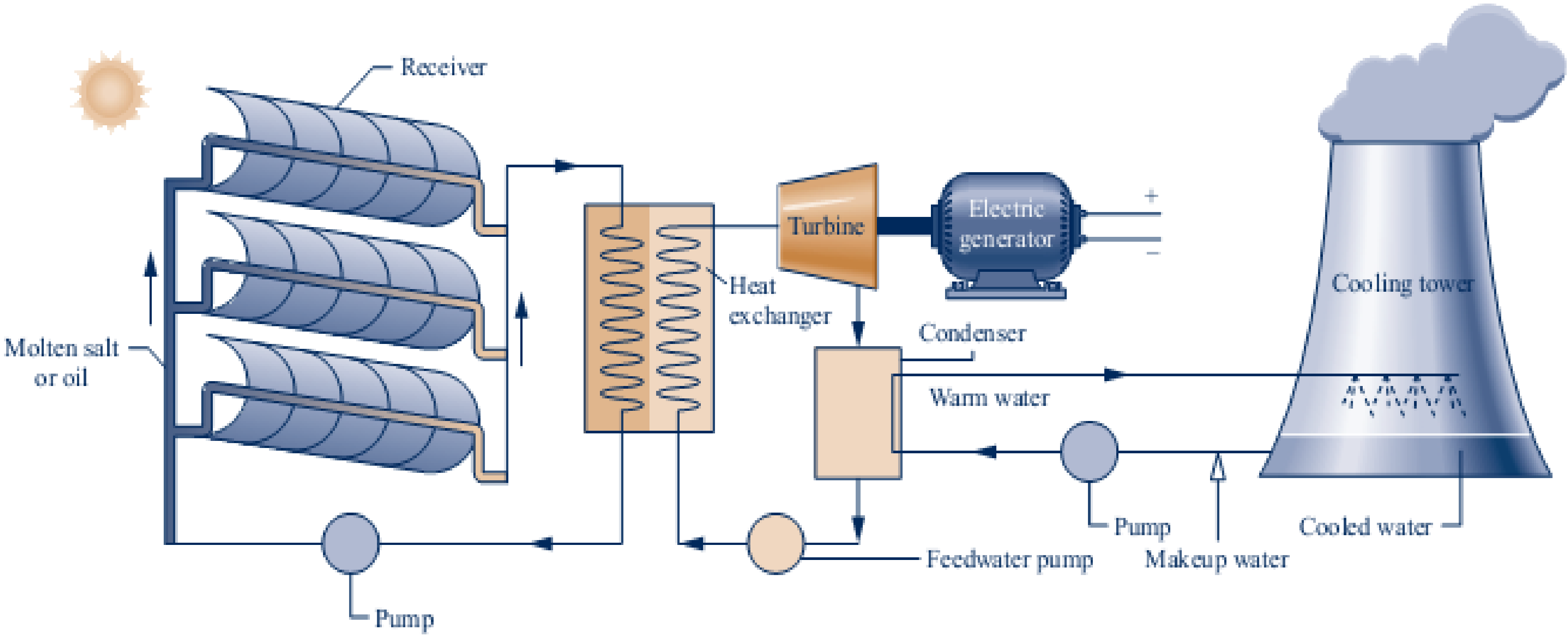
(*a*) Fossil-fueled vapor power plant.

Nuclear steam Power Plant



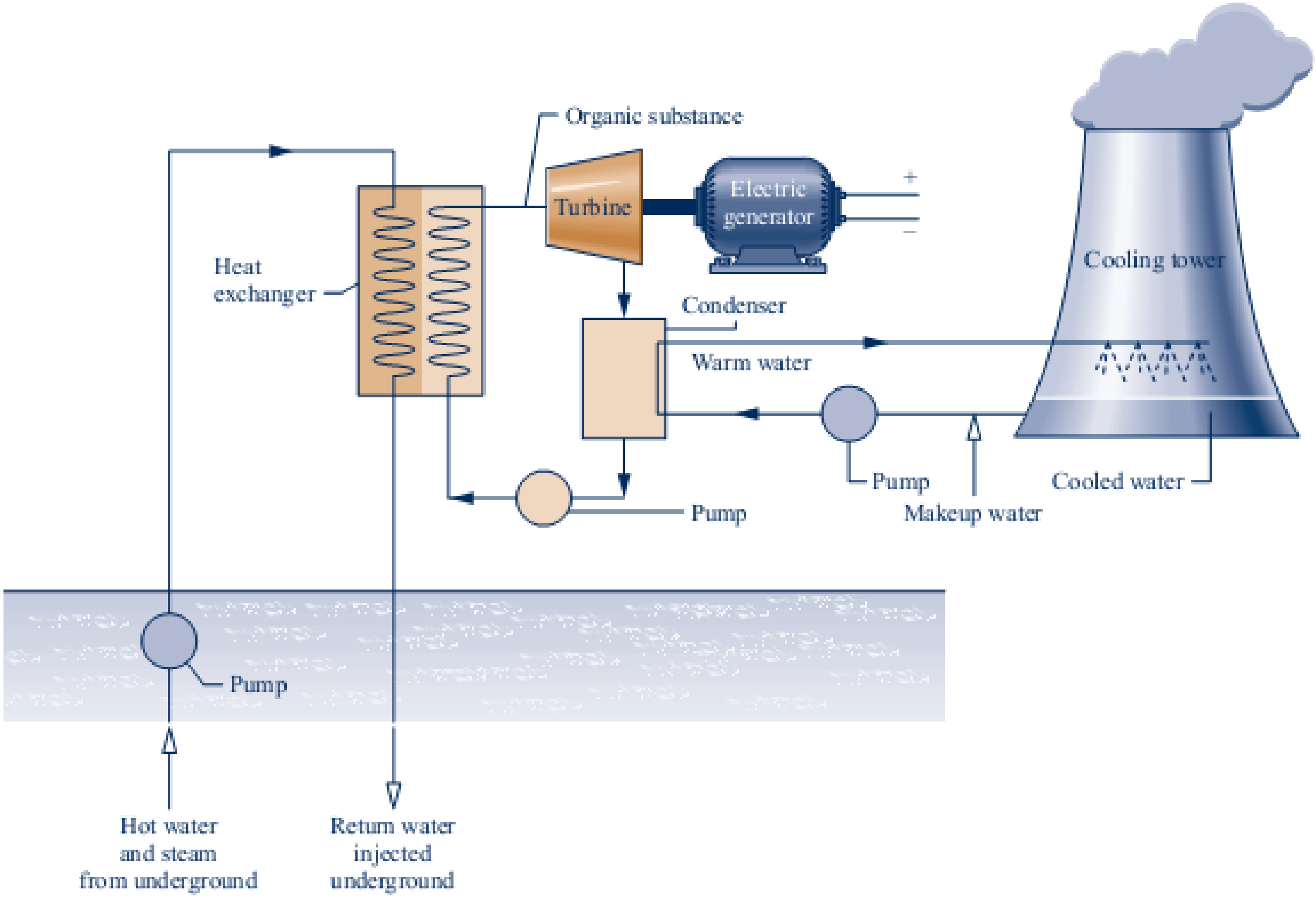
(*b*) Pressurized-water reactor nuclear vapor power plant.

Concentrating Solar Thermal Power Plant



(*c*) Concentrating solar thermal vapor power plant.

Geothermal Power Plant



(*d*) Geothermal vapor power plant.

Introduction

- Just to have an idea of point 1, let's consider a 1 GW fossil fuel fired steam power stations with $\eta_{\mathrm{net}} = 40\ \%$.

- From the definition: $\eta_{\mathrm{net}} = \dfrac{\dot{w}_{net}}{\dot{m}_{fuel} \times LHV}$

- So $\dot{Q}_{comb} = \dot{m}_{fuel} \times LHV = 2.5\ GW_{LHV}$

- The $\dot{Q}_{comb}$ needed to transfer to the cycle would be :

$$\dot{Q}_1 = \eta_{\mathrm{boiler}} \times \dot{m}_{fuel} \times LHV = 2.25\ GW_{LHV}$$

- Let's take $\eta_{\mathrm{boiler}} \approx 90\ \%$

- From heat transfer : $\dot{Q}_1 = \mathrm{U} \times A\ \Delta T_{LMTD} \times LHV$

Introduction

$$\frac{1}{U} = \frac{1}{h_e} + \frac{t_w}{k_w} + \frac{1}{h_i}$$

- In general:
- Note : Flue gas: h_e : low pressure and high T ----> low ρ ----> h_e $\downarrow\downarrow \approx (50 - 100)\frac{w}{m^2k}$
- Liquid: Pressurized --> low ρ --> h_i $\uparrow\uparrow \approx (1000 - 100{,}000)\ \frac{w}{m^2k}$
- $\frac{k_w}{t_w}$ is large, k_w is good and t_w is small.

Introduction

- $\frac{t_w}{k_w} + \frac{1}{h_i} \ll \frac{1}{h_e}$ Because
- $\frac{k_w}{t_w} \gg h_e \; and \; h_i \gg h_e$
- So it turns out that
- $\frac{1}{U} = \frac{1}{h_e}$
- So $U = h_e \approx 100 \frac{w}{m^2 k}$
- We are adding also radiation otherwise it will be far less. Then we expect a $\Delta T_{LMTD} \approx 400^o C$
- $\frac{t_w}{k_w} + \frac{1}{h_i} \ll \frac{1}{h_e}$ Because
- $\frac{k_w}{t_w} \gg h_e \; and \; h_i \gg h_e$

Introduction

- Finally we get: $A = \frac{2.5\ GW_{LHV}}{100\ \frac{w}{m^2 k} \times 400^o C} = 56{,}250\ m^{\ 2}$
- This is a key point why it is not suited for mobile application especially nowadays.

Selection of working Fluid

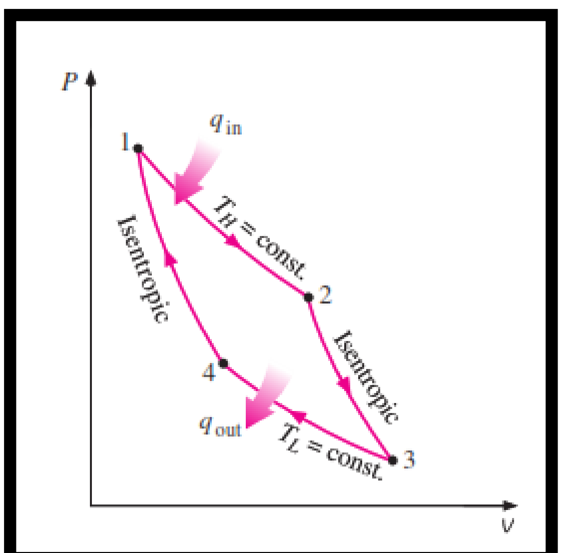
- **Water** : Another important point is the possibility to choose the working fluid. In general water is used
- Very common and readily available at low cost (Economic reason)
- $\frac{v_{vap}}{v_{liq}}$ is very high. This means a very high ratio of $\frac{w_{turbine}}{w_{pump}}$ (Thermodynamic reasons)
- Water given its ρ and κ transport properties has a very high convective heat transfer coefficient (*h*) (Heat Transfer reasons)

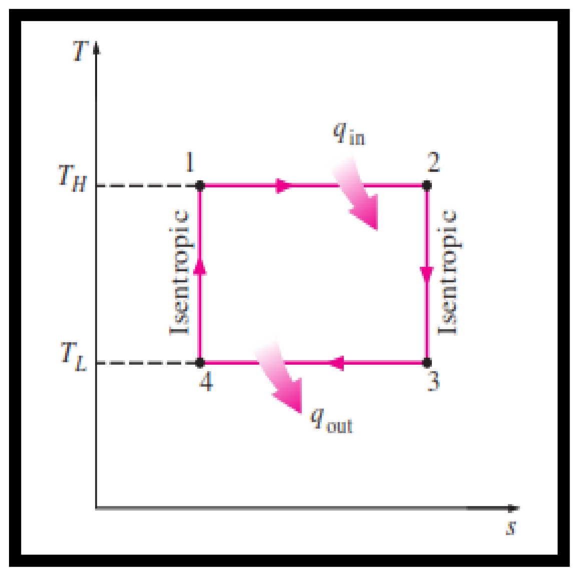
Carnot Cycle

- *Sadi Carnot was a French physicist who proposed an “ideal” cycle for a heat engine in 1824.*
- *Historical note – the idea of an ideal cycle came about because engineers were trying to develop a steam engine (a type of heat engine) where they could reject (waste) a minimal amount of heat.*
- *This would produce the best efficiency since* $\eta = 1 - (QL/QH)$.

THE CARNOT CYCLE

- The Carnot cycle is composed of four totally reversible processes: isothermal heat addition, isentropic expansion, isothermal heat rejection, and isentropic compression. The *P-v* and *T-s diagrams of a Carnot cycle are* plotted in Fig. below





Derivation of the Efficiency of Carnot Cycle

Show that the thermal efficiency of a Carnot cycle operating between the temperature limits of T_H and T_L is solely a function of these two temperatures and is given by Eq. 9–2.

Solution It is to be shown that the efficiency of a Carnot cycle depends on the source and sink temperatures alone.

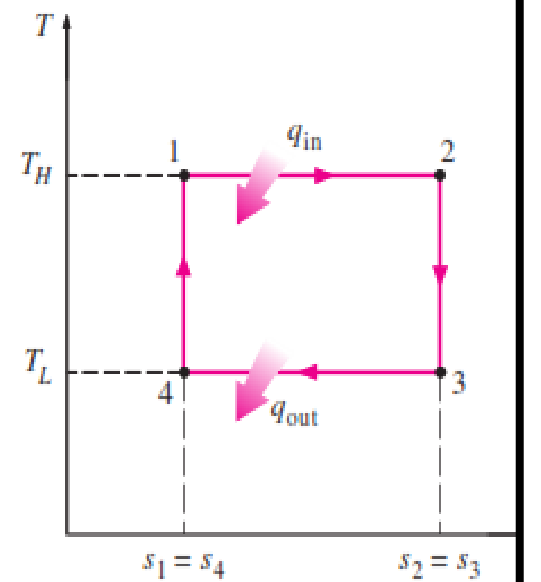
Analysis The *T-s* diagram of a Carnot cycle is redrawn in Fig. 9–8. All four processes that comprise the Carnot cycle are reversible, and thus the area under each process curve represents the heat transfer for that process. Heat is transferred to the system during process 1-2 and rejected during process 3-4. Therefore, the amount of heat input and heat output for the cycle can be expressed as

$$q_{in} = T_H(s_2 - s_1) \quad \text{and} \quad q_{out} = T_L(s_3 - s_4) = T_L(s_2 - s_1)$$

since processes 2-3 and 4-1 are isentropic, and thus $s_2 = s_3$ and $s_4 = s_1$. Substituting these into Eq. 9–1, we see that the thermal efficiency of a Carnot cycle is

$$\eta_{th} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_L(s_2 - s_1)}{T_H(s_2 - s_1)} = 1 - \frac{T_L}{T_H}$$

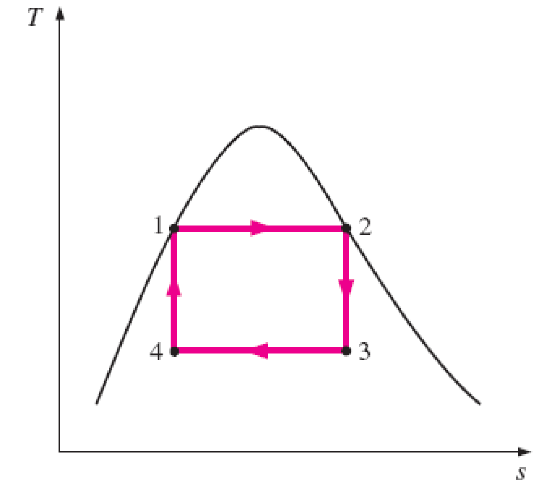
Discussion Notice that the thermal efficiency of a Carnot cycle is independent of the type of the working fluid used (an ideal gas, steam, etc.) or whether the cycle is executed in a closed or steady-flow system.



T-s diagram

Is Carnot Cycle Practical?

The Carnot cycle is NOT a suitable model for actual power cycles because of several impracticalities associated with it:



Process 1-2

Limiting the heat transfer processes to two-phase systems severely limits the 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**

Process 2-3

The turbine cannot handle steam with a high moisture content because of the impingement of liquid droplets on the turbine blades causing erosion and wear.

Process 4-1

It is not practical to design a compressor that handles two phases.

Several impracticalities are associated with this cycle

We can conclude that the Carnot cycle cannot be approximated in actual devices and is not a realistic model for vapor power cycles.

Exercise 1.1

- *A steady-flow Carnot cycle uses water as the working fluid. Water changes from saturated liquid to saturated vapor as heat is transferred to it from a source at 250°C. Heat rejection takes place at a pressure of 20 kPa.*

  (a) Show the cycle on a T-s diagram relative to the saturation lines, and determine

  (b) the thermal efficiency,

  (c) the amount of heat rejected, in kJ/kg, and (c) the net work output.

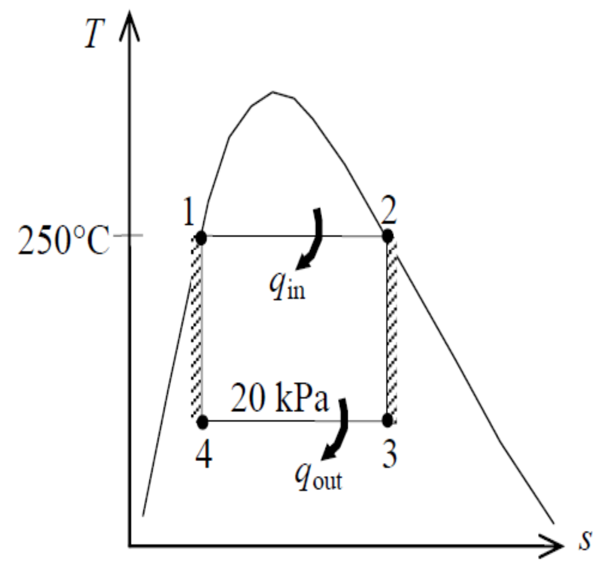
Solution 1.1

A steady-flow Carnot engine with water as the working fluid operates at specified conditions. The thermal efficiency, the amount of heat rejected, and the net work output are to be determined.

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

Analysis (*a*) Noting that $T_H = 250°C = 523$ K and $T_L = T_{sat\ @\ 20\ kPa} = 60.06°C = 333.1$ K, the thermal efficiency becomes

$$\eta_{th,C} = 1 - \frac{T_L}{T_H} = 1 - \frac{333.1\text{ K}}{523\text{ K}} = 0.3632 = \mathbf{36.3\%}$$



(*b*) The heat supplied during this cycle is simply the enthalpy of vaporization,

$$q_{in} = h_{fg@\ 250^\circ C} = 1715.3\text{ kJ/kg}$$

Thus,

$$q_{out} = q_L = \frac{T_L}{T_H} q_{in} = \left(\frac{333.1\text{ K}}{523\text{ K}}\right)(1715.3\text{ kJ/kg}) = \mathbf{1092.3\ kJ/kg}$$

(*c*) The net work output of this cycle is

$$w_{net} = \eta_{th} q_{in} = (0.3632)(1715.3\text{ kJ/kg}) = \mathbf{623.0\ kJ/kg}$$

Exercise

- Consider a 210-MW steam power plant that operates on a simple **ideal Rankine cycle**. Steam enters the turbine at 10 MPa and 500°C and is cooled in the condenser at a pressure of 10 kPa. Show the cycle on a *T-s diagram* with respect to saturation lines, and determine:

  A. The quality of the steam at the turbine exit,

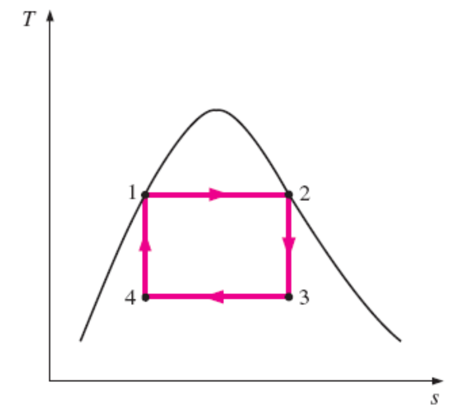
  B. The thermal efficiency of the cycle, and

  C. The mass flow rate of the steam.

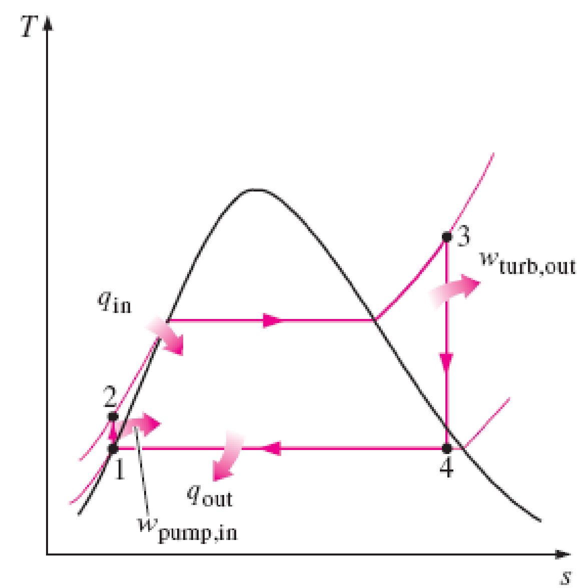
- *Answers: (a) 0.793, (b) 40.2 percent, (c) 165 kg/s*

1.2. The Rankine Cycle

Many of the impracticalities associated with the Carnot cycle can be eliminated by: (a) **superheating** the steam in the boiler, and (b) **condensing the steam completely in the condenser.**



The modified Carnot cycle is called the Rankine cycle, where the isothermal processes are replaced with constant pressure processes to facilitate doing (a) and (b) above. This is the ideal and practical cycle for vapor power plants (Figure).



This ideal cycle does not involve any internal irreversibilities.

Sequence of Processes

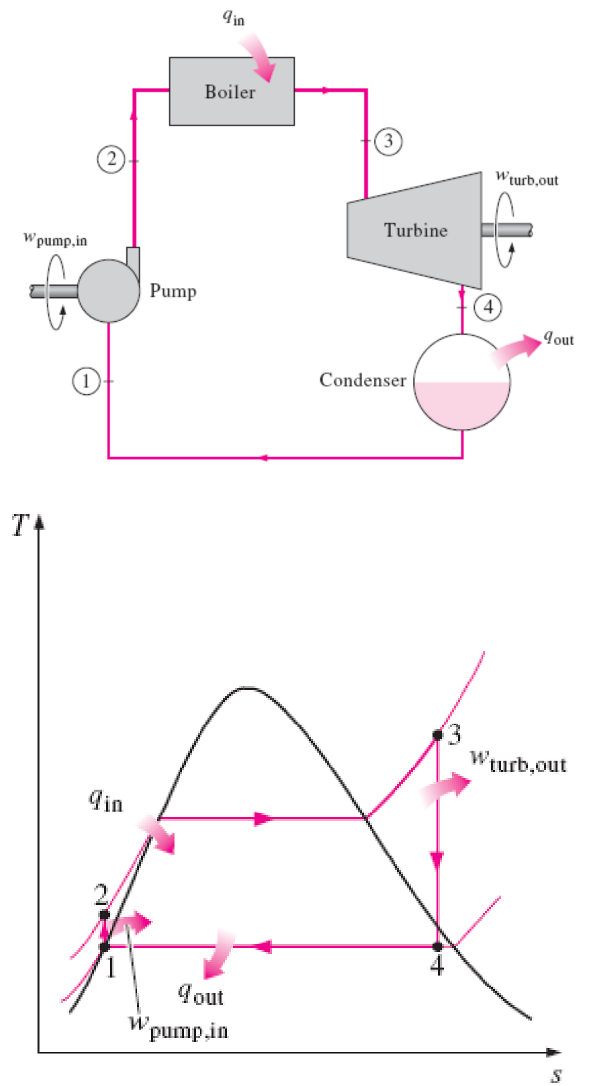
The ideal Rankine cycle consists of four processes:

1-2 Isentropic compression in a water 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.



The simple ideal Rankine cycle.

Energy Analysis of Rankine Cycle

- All four components associated with the Rankine cycle (the pump, boiler, turbine, and condenser) are steady-flow devices, and thus all four processes that make up the Rankine cycle can be analyzed as steady-flow processes.
- The kinetic and potential energy changes of the steam are usually small relative to the work and heat transfer terms and are therefore usually neglected.

Energy Analysis of Rankine Cycle

- Rankine cycle can be analyzed as steady-flow processes.
- Neglecting changes in kinetic and potential energies.
- The ***steady-flow energy equation*** per unit mass of steam

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_e - h_i$$

- The boiler and the condenser do not involve any work,
- The pump and the turbine are assumed to be isentropic.
- The conservation of energy relation for each device can be expressed as follows:
- $$Pump(q = 0): \qquad w_{pump,in} = h_2 - h_1$$

If we consider the fluid to be incompressible

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

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

Energy Analysis of Rankine Cycle

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

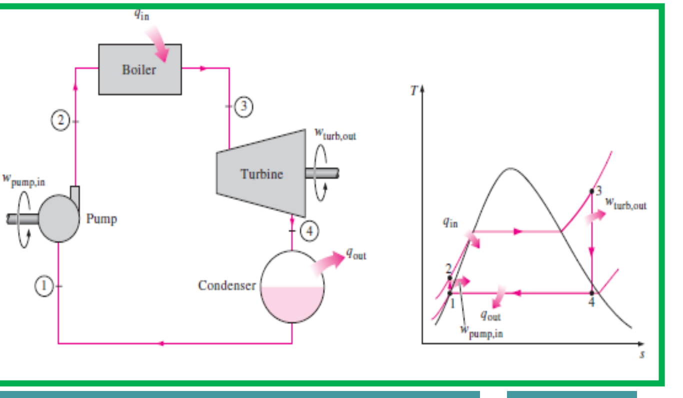
$$Turbine\,(q=0): \qquad w_{turb,out} = h_3 - h_4$$

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

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

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

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



Summery of Energy analysis

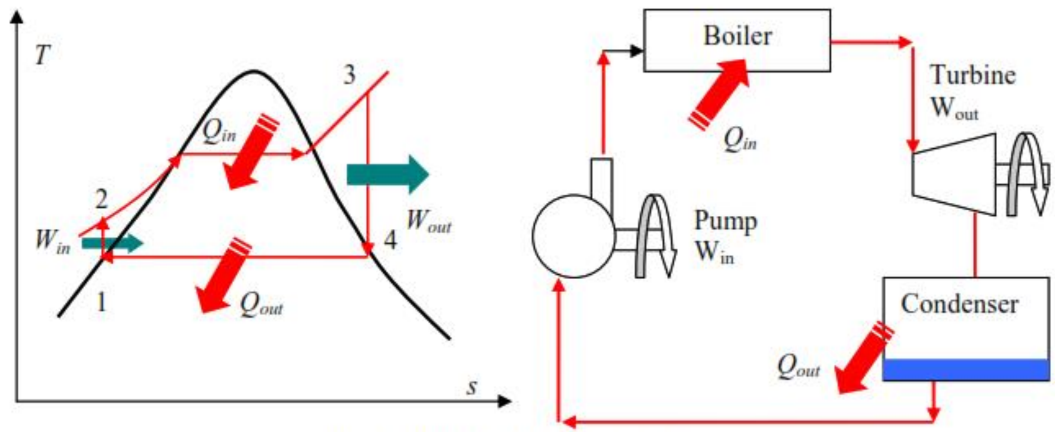


Fig. 2: The ideal Rankine cycle.

Energy Analysis for the Cycle

All four components of the Rankine cycle are steady-state steady-flow devices. The potential and kinetic energy effects can be neglected. The first law per unit mass of steam can be written as:

Pump	$q = 0$	$w_{pump,in} = h_2 - h_1$
Boiler	$w = 0$	$q_{in} = h_3 - h_2$
Turbine	$q = 0$	$w_{turbine,out} = h_3 - h_4$
Condenser	$w = 0$	$q_{out} = h_4 - h_1$

The thermal efficiency of the cycle is determined from:

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

where

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

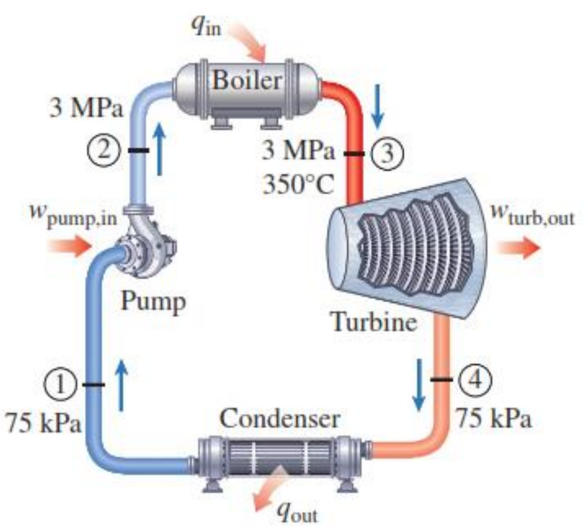
If we consider the fluid to be incompressible, the work input to the pump will be:

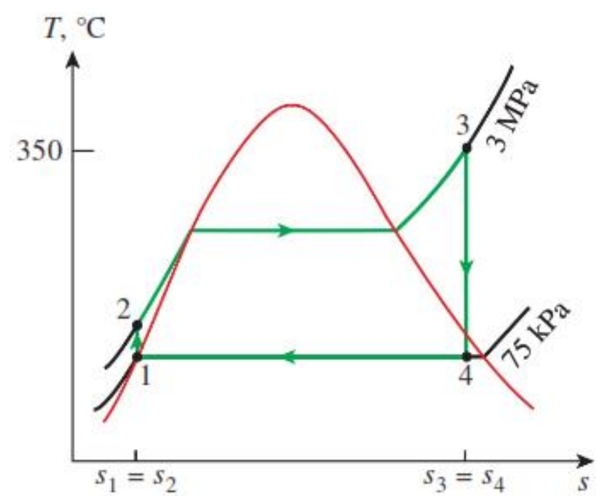
$$(h_2 - h_1) = v(P_2 - P_1)$$

where $h_1 = h_{f@P1}$ & $v = v_1 = v_{f@P1}$

The Simple Ideal Rankine Cycle

- Consider a steam power plant operating on the simple ideal Rankine cycle. Steam enters the turbine at 3 MPa and 350^{0}C and is condensed in the condenser at a pressure of 75 kPa. Determine the thermal efficiency of this cycle.





Solution

State 1: $P_1 = 75$ kPa, Sat. liquid $\Big\}$ $h_1 = h_{f\,@\,75\,\text{kPa}} = 384.44$ kJ/kg; $v_1 = v_{f\,@\,75\,\text{kPa}} = 0.001037\ \text{m}^3/\text{kg}$

State 2: $P_2 = 3$ MPa

$s_2 = s_1$

$$w_{\text{pump,in}} = v_1(P_2 - P_1) = (0.001037\ \text{m}^3/\text{kg})[(3000 - 75)\ \text{kPa}]\left(\frac{1\ \text{kJ}}{1\ \text{kPa·m}^3}\right)$$

$$= 3.03\ \text{kJ/kg}$$

$$h_2 = h_1 + w_{\text{pump,in}} = (384.44 + 3.03)\ \text{kJ/kg} = 387.47\ \text{kJ/kg}$$

State 3: $P_3 = 3$ MPa, $T_3 = 350°\text{C}$ $\Big\}$ $h_3 = 3116.1$ kJ/kg; $s_3 = 6.7450$ kJ/kg·K

State 4: $P_4 = 75$ kPa (sat. mixture)

$s_4 = s_3$

$$x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.7450 - 1.2132}{6.2426} = 0.8861$$

$$h_4 = h_f + x_4 h_{fg} = 384.44 + 0.8861(2278.0) = 2403.0\ \text{kJ/kg}$$

Thus,

$$q_{\text{in}} = h_3 - h_2 = (3116.1 - 387.47)\ \text{kJ/kg} = 2728.6\ \text{kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = (2403.0 - 384.44)\ \text{kJ/kg} = 2018.6\ \text{kJ/kg}$$

and

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2018.6\ \text{kJ/kg}}{2728.6\ \text{kJ/kg}} = \mathbf{0.260}\ \text{or}\ \mathbf{26.0\%}$$

The thermal efficiency could also be determined from

$$w_{\text{turb,out}} = h_3 - h_4 = (3116.1 - 2403.0)\ \text{kJ/kg} = 713.1\ \text{kJ/kg}$$

$$w_{\text{net}} = w_{\text{turb,out}} - w_{\text{pump,in}} = (713.1 - 3.03)\ \text{kJ/kg} = 710.1\ \text{kJ/kg}$$

or

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = (2728.6 - 2018.6)\ \text{kJ/kg} = 710.0\ \text{kJ/kg}$$

and

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{710.0\ \text{kJ/kg}}{2728.6\ \text{kJ/kg}} = \mathbf{0.260}\ \text{or}\ \mathbf{26.0\%}$$

That is, this power plant converts 26 percent of the heat it receives in the boiler to net work. An actual power plant operating between the same temperature and pressure limits will have a lower efficiency because of the irreversibilities such as friction.

Discussion Notice that the back work ratio ($r_{bw} = w_{in}/w_{out}$) of this power plant is 0.004, and thus only 0.4 percent of the turbine work output is required to operate the pump. Having such low back work ratios is characteristic of vapor power cycles. This is in contrast to the gas power cycles, which typically involve very high back work ratios (about 40 to 80 percent).

It is also interesting to note the thermal efficiency of a Carnot cycle operating between the same temperature limits

$$\eta_{\text{th,Carnot}} = 1 - \frac{T_{\text{min}}}{T_{\text{max}}} = 1 - \frac{(91.76 + 273)\ \text{K}}{(350 + 273)\ \text{K}} = 0.415$$

The difference between the two efficiencies is due to the large external irreversibility in the Rankine cycle caused by the large temperature difference between steam and combustion gases in the furnace.

Exercise 1.2.1

Consider a steam power plant that operates on a simple ideal Rankine cycle and has a net power output of 45 MW. Steam enters the turbine at 7 MPa and 500 ^{0}C and is cooled in the condenser at a pressure of 10 kPa by running cooling water from a lake through the tubes of the condenser at a rate of 2000 kg/sec.

(a) Show the cycle on a T-s diagram with respect to saturation lines, and determine

(b) the thermal efficiency of the cycle

(c) the mass flow rate of the steam, and

(d) the temperature rise of the cooling water.

Solution 1.2.1

Analysis (*a*) From the steam tables

$$h_1 = h_{f@\,10\text{ kPa}} = 191.81\text{ kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\,10\text{ kPa}} = 0.00101\ \text{m}^3/\text{kg}$$

$$w_{p,\text{in}} = \boldsymbol{v}_1(P_2 - P_1)$$

$$= (0.00101\ \text{m}^3/\text{kg})(7{,}000 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

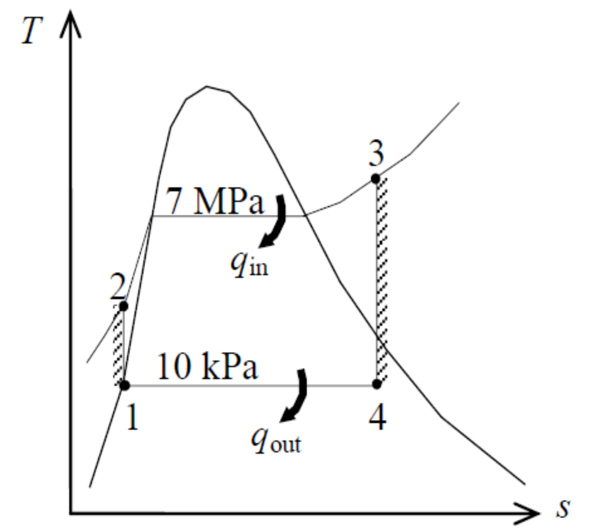
$$= 7.06\text{ kJ/kg}$$

$$h_2 = h_1 + w_{p,\text{in}} = 191.81 + 7.06 = 198.87\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 7\text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3411.4\text{ kJ/kg} \\ s_3 = 6.8000\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 10\text{ kPa} \\ s_4 = s_3 \end{array}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.8000 - 0.6492}{7.4996} = 0.8201$$

$$h_4 = h_f + x_4 h_{fg} = 191.81 + (0.8201)(2392.1) = 2153.6\text{ kJ/kg}$$



Solution 1.2.1

Thus,

$$q_{in} = h_3 - h_2 = 3411.4 - 198.87 = 3212.5 \text{ kJ/kg}$$
$$q_{out} = h_4 - h_1 = 2153.6 - 191.81 = 1961.8 \text{ kJ/kg}$$
$$w_{net} = q_{in} - q_{out} = 3212.5 - 1961.8 = 1250.7 \text{ kJ/kg}$$

and

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{1250.7 \text{ kJ/kg}}{3212.5 \text{ kJ/kg}} = \mathbf{38.9\%}$$

(*b*) $$\dot{m} = \frac{\dot{W}_{net}}{w_{net}} = \frac{45{,}000 \text{ kJ/s}}{1250.7 \text{ kJ/kg}} = \mathbf{36.0 \text{ kg/s}}$$

(*c*) The rate of heat rejection to the cooling water and its temperature rise are

$$\dot{Q}_{out} = \dot{m} q_{out} = (35.98 \text{ kg/s})(1961.8 \text{ kJ/kg}) = 70{,}586 \text{ kJ/s}$$
$$\Delta T_{coolingwater} = \frac{\dot{Q}_{out}}{(\dot{m}c)_{coolingwater}} = \frac{70{,}586 \text{ kJ/s}}{(2000 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot°\text{C})} = \mathbf{8.4°C}$$

Matlab Code for Exercise 1.1

```
clear all
close all
clc
% Input Variables
fprintf('Input Variables \n');
P1= 0.1;     % 10 kpa
fprintf('P1=%4.2f Bar \n',P1);
P2=70;       % 7 MPa
fprintf('P2=%4.2f Bar \n',P2);
P3=70;       % 7 MPa
fprintf('P3=%4.2f Bar \n',P3');
T3= 500;     % Degree centigrade
% Thermodynamics Properties at Point 1
fprintf('Thermodynamics Properties at Point 1 \n');
hf1=XSteam('hL_p',P1);
h1=hf1;
fprintf('hf1=%4.2f kj/kg \n',hf1);
vf1=XSteam('vL_p',P1);
fprintf('vf1=%4.3f m3/kg \n',vf1);
wpin = vf1*(P2-P1)*100;
fprintf('wpin=%4.3f kj/kg \n',wpin);
% Thermodynamics Properties at Point 2
fprintf('Thermodynamics Properties at Point 2 \n');
h2= hf1+ wpin;
fprintf('h2=%4.2f kj/kg \n',h2)
```

Matlab Code for Exercise 1.1

```
% Thermodynamics Properties at Point 3
fprintf('Thermodynamics Properties at Point 3 \n');
h3= XSteam('h_pT',P3,T3);
fprintf('h3=%4.2f kj/kg \n',h3)
s3= XSteam('s_pT',P3,T3);
fprintf('s3=%4.2f kj/kg \n',s3)
% Thermodynamics Properties at Point 4
fprintf('Thermodynamics Properties at Point 4 \n');
s4=s3;
fprintf('s4=%4.2f kj/kg.k \n',s4)
p4=0.1;   % bar
fprintf('p4=%4.2f Bar \n',p4)
x4= XSteam('x_ps',p4,s4);
fprintf('x4=%4.3f \n',x4)
h4= XSteam('h_ps',p4,s4);
fprintf('h4=%4.3f kj/kg \n',h4)
% Overall performance parameters
qin = h3- h2;
fprintf('qin=%4.3f kj/kg \n',qin)
qout = h4- h1;
fprintf('qout=%4.3f kj/kg \n',qout)
wnet=qin-qout;
fprintf('wnet=%4.3f kj/kg \n',wnet)
eff=wnet/qin*100;
fprintf('Answer for Question No.1 \n')
```

Matlab Code for Exercise 1.1

```
fprintf('efficiency=%4.3f  \n',eff)
power =45000;
massflow=power/wnet;
fprintf('Answer for Question No.2 \n')
fprintf('MassFlow=%4.2f kg/s \n', massflow)
Qoutdot= massflow*qout;
DT_Cooling_Water = Qoutdot/(2000*4.18);
fprintf('Answer for Question No.3 \n')
fprintf('DT_Cooling_Water= %4.3f OC \n',DT_Cooling_Water)
```

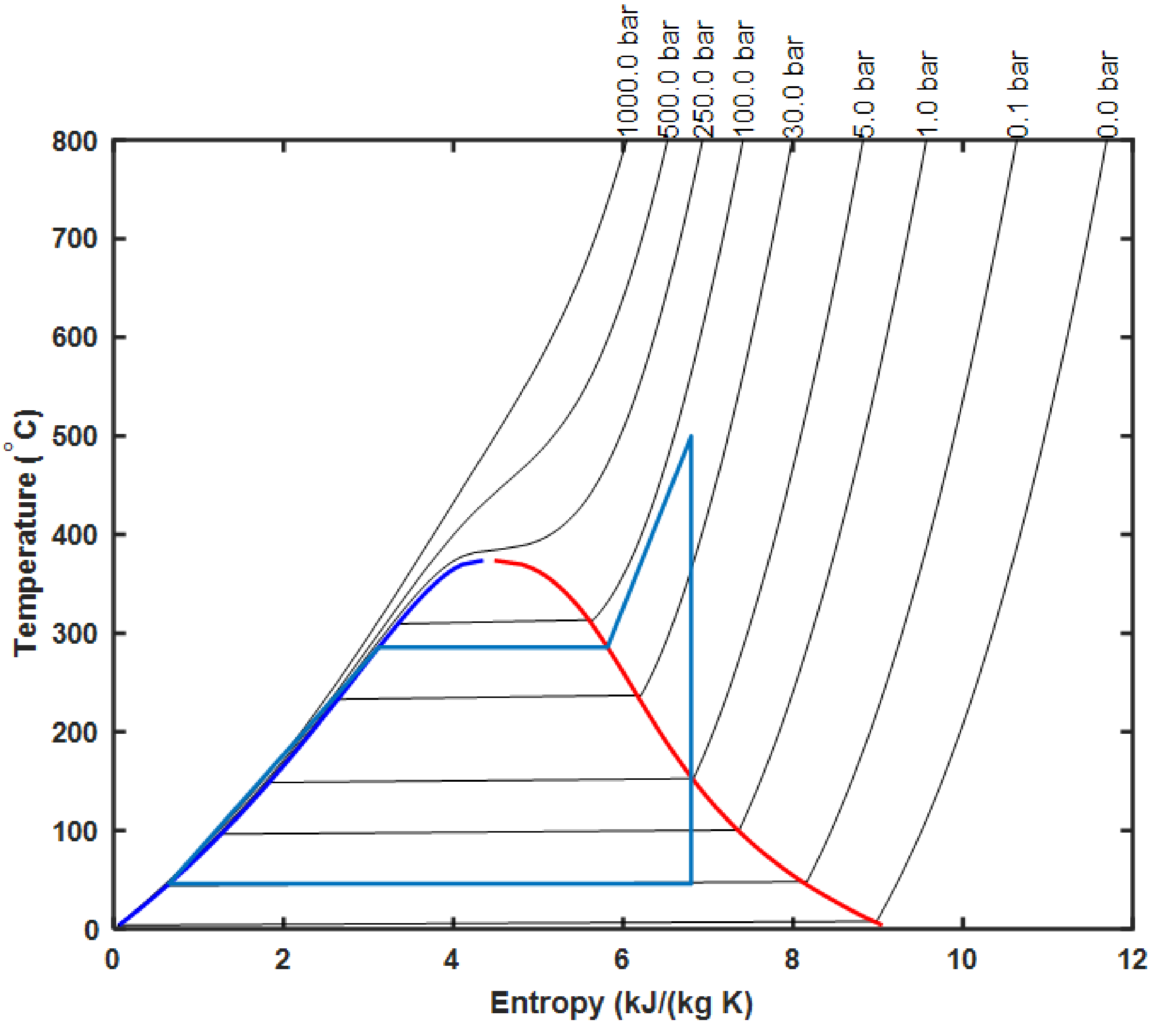
Matlab Code for Exercise 1.1

```
T = linspace(0,800,200); % range of temperatures
fh = figure; % store file handle for later
hold on
% we need to compute S-T for a range of pressures.
pressures = [0.01 0.1 1 5 30 100 250 500 1000]; % bar
for P = pressures
    % XSteam is not vectorized, so here is an easy way to compute a
    % vector of entropies
    S = arrayfun(@(t) XSteam('s_PT',P,t),T);
    plot(S,T,'k-')
    text(S(end),T(end),sprintf('%1.1f bar',P),'rotation',90)
end
% adjust axes positions so the pressure labels don't get cutoff
set(gca,'Position',[0.13 0.11 0.775 0.7])

% plot saturated vapor and liquid entropy lines
svap = arrayfun(@(t) XSteam('sV_T',t),T);
sliq = arrayfun(@(t) XSteam('sL_T',t),T);

plot(svap,T,'r-','LineWidth',1.5)
plot(sliq,T,'b-','LineWidth',1.5)
xlabel('Entropy (kJ/(kg K)','FontSize',10,'FontWeight', 'bold')
ylabel('Temperature (^\circC)','FontSize',10,'FontWeight', 'bold')
set(gca,'FontSize',10,'FontWeight','bold','linewidth',1.5)
box on
Tsat3=XSteam('TSat_p',P2);
ssat3v=XSteam('sV_T',Tsat3);
ssat3l=XSteam('sL_T',Tsat3);
hold on
s=[s1 s2 ssat3l ssat3v s3 s4 s1];
T=[TE1 TE2 Tsat3 Tsat3 TE3 TE4 TE1];
plot(s,T,'LineWidth',1.5)
```

Output from Matlab

Input Variables
P1=0.10 Bar
P2=70.00 Bar
P3=70.00 Bar
Thermodynamics Properties at Point 1
hf1=191.81 kj/kg
vf1=0.001 m3/kg
wpin=7.062 kj/kg
TE1=45.820 OC
S1=0.649 kj/kg.k
Thermodynamics Properties at Point 2
h2=198.87 kj/kg
TE2=46.048 OC
S2=0.649 kj/kg.k
Thermodynamics Properties at Point 3
h3=3411.25 kj/kg
s3=6.80 kj/kg
Thermodynamics Properties at Point 4
s4=6.80 kj/kg.k
p4=0.10 Bar
x4=0.820
h4=2153.557 kj/kg
TE4=45.808 OC
qin=3212.376 kj/kg
qout=1961.745 kj/kg
wnet=1250.632 kj/kg
Answer for Question No.1
efficiency=38.932
Answer for Question No.2
MassFlow=35.98 kg/s
Answer for Question No.3
DT_Cooling_Water= 8.443 OC

Output from Matlab



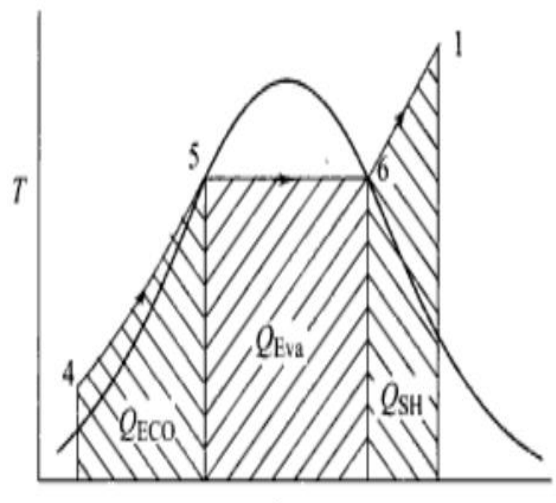
Fraction of total heat absorbed in economizer, evaporator and superheater

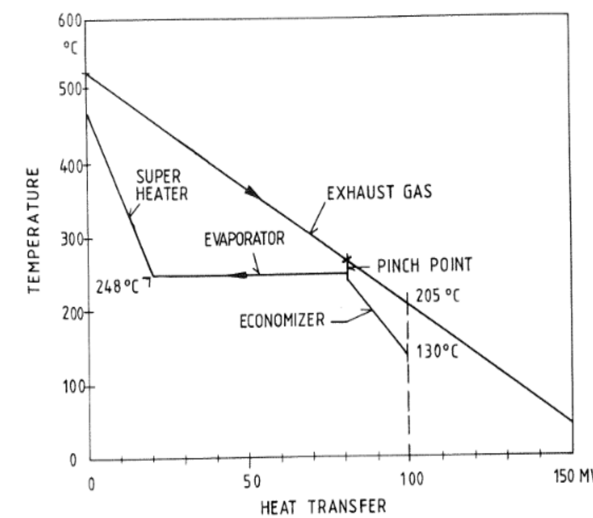
- *Note : The fraction of the total heat transfer absorbed in the economizer, evaporator and superheater are given by:*

$$\frac{Q_{Eco}}{Q_1} = \frac{h_5 - h_4}{h_1 - h_4} = \frac{\text{area under } 4-5}{\text{area under } 4-1}$$

$$\frac{Q_{Eva}}{Q_1} = \frac{h_6 - h_5}{h_1 - h_4} = \frac{\text{area under } 5-6}{\text{area under } 4-1}$$

$$\frac{Q_{SH}}{Q_1} = \frac{h_1 - h_6}{h_1 - h_4} = \frac{\text{area under } 6-1}{\text{area under } 4-1}$$





- *As the pressure increases, the latent heat decreases and so the heat absorbed in the evaporator decreases and the fraction of the total heat absorbed in the superheater increases. In high pressure boilers, more than 40 % of the total heat is absorbed in superheater.*

Exercise 1.2.2

- Steam at 40 bar; 500 °C flowing at the rate of 5500 kg/hr expands in a h.p. turbine to 2 bar with an isentropic efficiency of 83 %. A continuous supply of steam at 2 bar, 0.87 quality and a flow rate of 2700 kg/h is available from a geothermal energy source. This steam is mixed adiabatically with the h.p. turbine exhaust Steam and the combined flow then expands in a L.P turbine to 0.1 bar with an isentropic efficiency of 78 %. Determine the power output and the thermal efficiency of the plant. Assume that 5500 kg/h of steam is generated in the boiler at 40 bar, 500 °C from the saturated feed water at 0.1 bar.
- Had the geothermal steam not been added, what would have been the power output and efficiency of the plant? Neglect pump work

Solution 1.2.2

$h_1 = 3445.3$ kJ/kg, $s_1 = 7.0901$ kJ/kg $K = 1.5301 + x_{2s} \times 5.5970$

$$x_{2s} = \frac{5.5600}{5.5970} = 0.9934$$

$$h_{2s} = 504.7 + 0.9934 \times 2201.9 = 2692.04 \text{ kJ/kg K}$$

$$h_1 - h_2 = 0.83(3445.3 - 2692.04) = 625.21 \text{ kJ/kg}$$

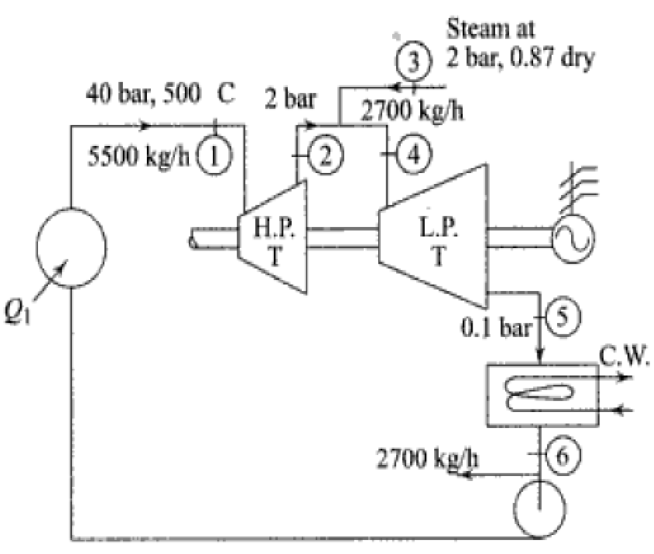
$$h_2 = 3445.3 - 625.21 = 2820.09 \text{ kJ/kg}$$

$$h_3 = 504.7 + 0.87 \times 2201.9 = 2420.4 \text{ kJ/kg}$$

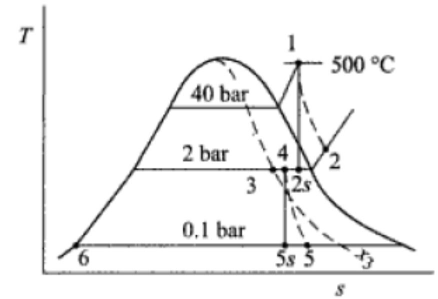
$$2700 h_3 + 5500 h_2 = (2700 + 5500) h_4$$

$$h_4 = \frac{2700 \times 2420.4 + 5500 \times 2820.09}{8200} = 796.96 + 1891.52$$

$$= 2688.48 \text{ kJ/kg}$$



(a)



(b)

Fig. E2.1

$$h_4 = 504.7 + x_4 \times 2201.9 = 2688.48; \; x_4 = \frac{2183.78}{2201.9} = 0.9917$$

$$s_4 = 1.5301 + 0.9917 \times 5.5970 = 7.0806 \text{ kJ/kg–K}$$

$$= s_{5s} = 0.6493 + x_{5s} \times 7.5009$$

$$x_{5s} = 0.8574; \quad h_{5s} = 191.84 + 0.8574 \times 2392.8 = 2243.44 \text{ kJ/kg}$$

$$h_4 - h_5 = 0.78(2688.48 - 2243.44) = 347.1 \text{ kJ/kg}$$

$$h_6 = 191.83 \text{ kJ/kg}$$

$$\dot{W} = 5500(h_1 - h_2) + 8200\,(h_4 - h_5) = 5500 \times 625.21 + 8200 \times 347.1$$

$$= 6284875 \text{ kJ/h} = 1745.8 \text{ kW} \quad (Ans.)$$

$$Q_1 = 5500(h_1 - h_6) = 5500\,(3445.3 - 191.8) \times \frac{1}{3600} = 4970.63 \text{ kW}$$

$$\eta_{cycle} = \frac{1745.8}{4970.63} = 0.353 \quad \text{or} \quad 35.3\%$$

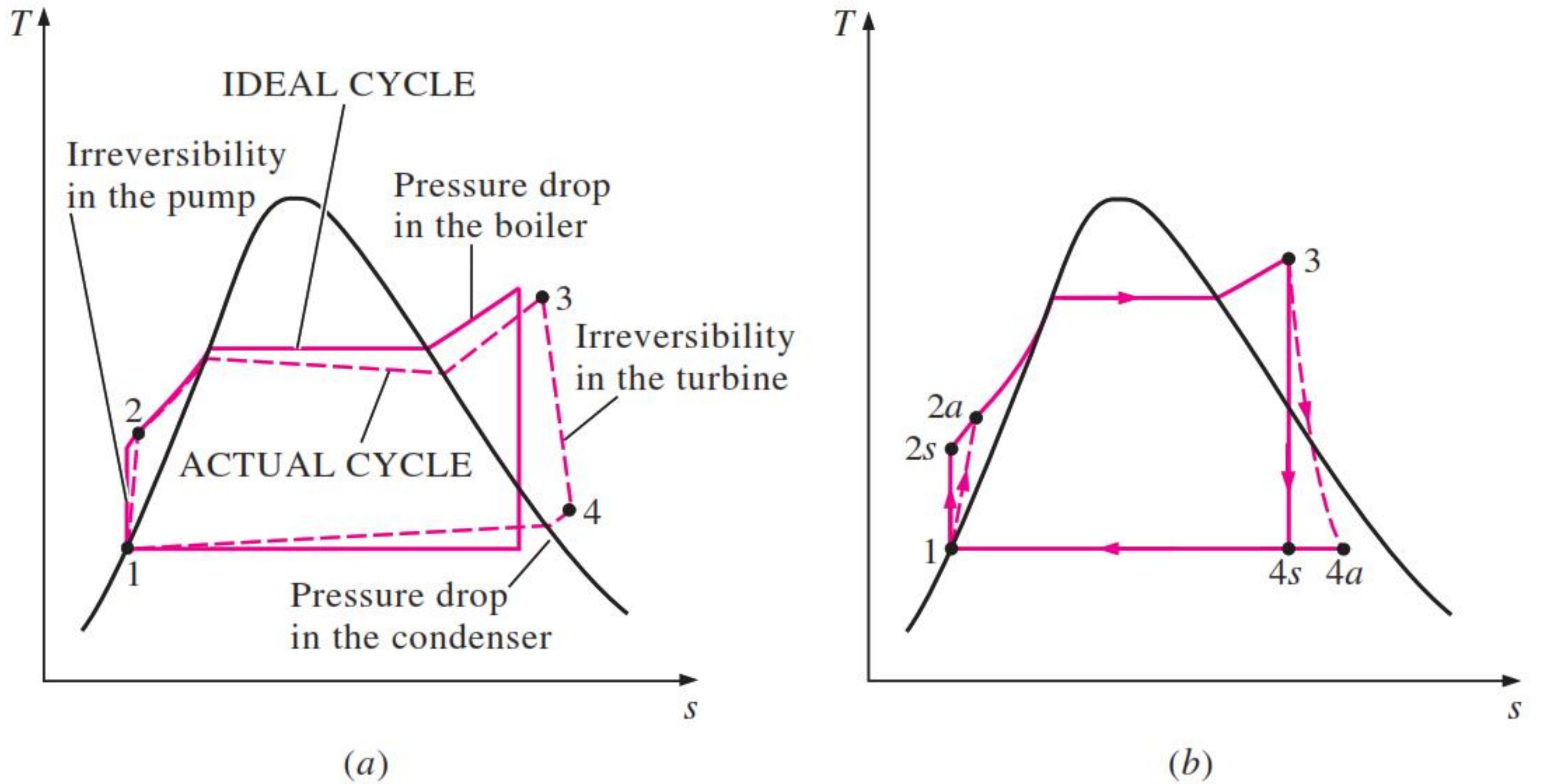
Without geothermal heat supply:

$$W_T = 5500\,(h_1 - h_2) = 955.18 \text{ kW} \quad (Ans.)$$

$$Q_1 = 5500\,(h_1 - h_6) = 4970.63 \text{ kW}$$

$$\eta_{cycle} = \frac{955.18}{4970.63} = 0.1922 \quad \text{or} \quad 19.22\% \quad (Ans.)$$

Deviation Of Actual Vapor Power Cycles From Idealized Ones



(a) Deviation of actual vapor power cycle from the ideal Rankine cycle.
(b) The effect of pump and turbine irreversibilities on the ideal Rankine cycle.

Deviation Of Actual Vapor Power Cycles From Idealized Ones

- The actual vapor power cycle differs from the ideal Rankine cycle as a result of irreversibility 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.

Deviation Of Actual Vapor Power Cycles From Idealized Ones

- *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.*
- *The other major source of irreversibility is the heat loss from the steam to the steam in the boiler to compensate for these undesired heat losses. As a result, cycle efficiency decreases.*
- *Of particular importance are the irreversibilities occurring within the pump and the turbine.*
- *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.*

Deviation Of Actual Vapor Power Cycles From Idealized Ones

- The deviation of actual pumps and turbines from the isentropic ones can be accounted for by utilizing *isentropic efficiencies, defined as* :

$$\eta_P = \frac{w_s}{w_a} = \frac{h_{2s} - h_1}{h_{2a} - h_1}$$

$$\eta_T = \frac{w_a}{w_s} = \frac{h_3 - h_{4a}}{h_3 - h_{4s}}$$

- Other factors also need to be considered in the analysis of actual vapor power cycles.
- 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.

Example 1.2

- A steam power plant operates on the cycle shown in Fig. below. If the isentropic efficiency of the turbine is 87 percent and the isentropic efficiency of the pump is 85 percent, determine (a) the thermal efficiency of the cycle and (b) the net power output of the plant for a mass flow rate of 15 kg/s .

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Solution

(*a*) The thermal efficiency of a cycle is the ratio of the net work output to the heat input, and it is determined as follows:

Pump work input:

$$w_{\text{pump,in}} = \frac{w_{s,\text{pump,in}}}{\eta_P} = \frac{v_1(P_2 - P_1)}{\eta_P}$$

$$= \frac{(0.001009\ \text{m}^3/\text{kg})[(16{,}000 - 9)\ \text{kPa}]}{0.85}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)$$

$$= 19.0\ \text{kJ/kg}$$

Turbine work output:

$$w_{\text{turb,out}} = \eta_T w_{s,\text{turb,out}}$$

$$= \eta_T(h_5 - h_{6s}) = 0.87(3583.1 - 2115.3)\ \text{kJ/kg}$$

$$= 1277.0\ \text{kJ/kg}$$

Boiler heat input: $q_{\text{in}} = h_4 - h_3 = (3647.6 - 160.1)\ \text{kJ/kg} = 3487.5\ \text{kJ/kg}$

Thus,

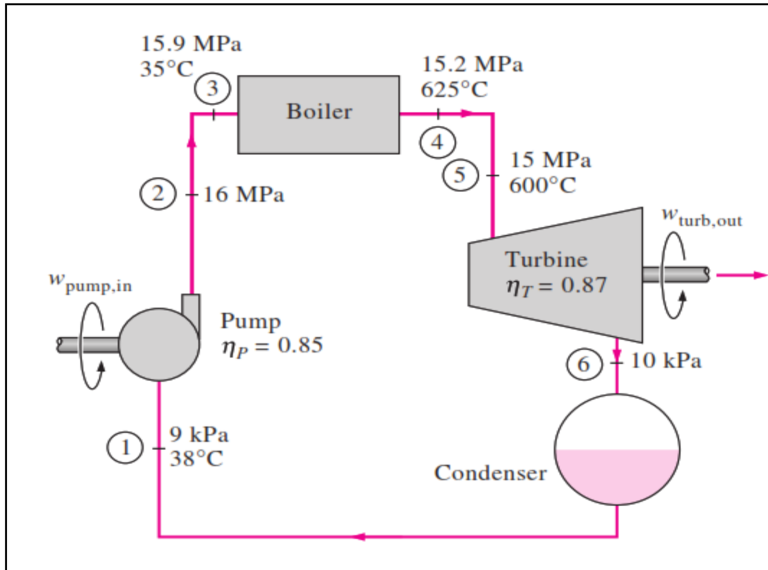
$$w_{\text{net}} = w_{\text{turb,out}} - w_{\text{pump,in}} = (1277.0 - 19.0)\ \text{kJ/kg} = 1258.0\ \text{kJ/kg}$$

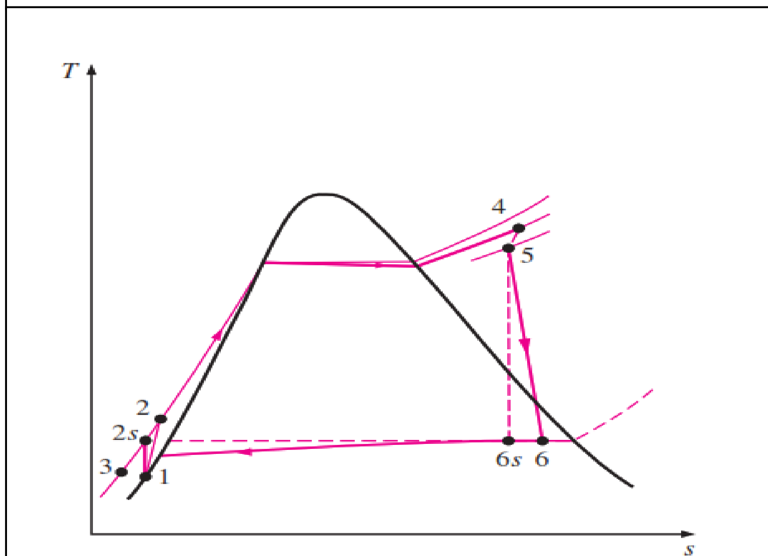
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1258.0\ \text{kJ/kg}}{3487.5\ \text{kJ/kg}} = \mathbf{0.361\ or\ 36.1\%}$$

(*b*) The power produced by this power plant is

$$\dot{W}_{\text{net}} = \dot{m}(w_{\text{net}}) = (15\ \text{kg/s})(1258.0\ \text{kJ/kg}) = \mathbf{18.9\ MW}$$

Discussion Without the irreversibilities, the thermal efficiency of this cycle would be 43.0 percent





Exercise

- Consider a steam power plant that operates on a **reheat Rankine cycle** and has a net power output of 80 MW. Steam enters the high-pressure turbine at 10 MPa and 500°C and the low-pressure turbine at 1 MPa and 500°C. Steam leaves the condenser as a saturated liquid at a pressure of 10 kPa. The isentropic efficiency of the turbine is 80 percent, and that of the pump is 95 percent. Show the cycle on a *T-s diagram* with respect to saturation lines, and determine:

  A. The quality of the steam at the turbine exit,

  B. The thermal efficiency of the cycle, and

  C. The mass flow rate of the steam.

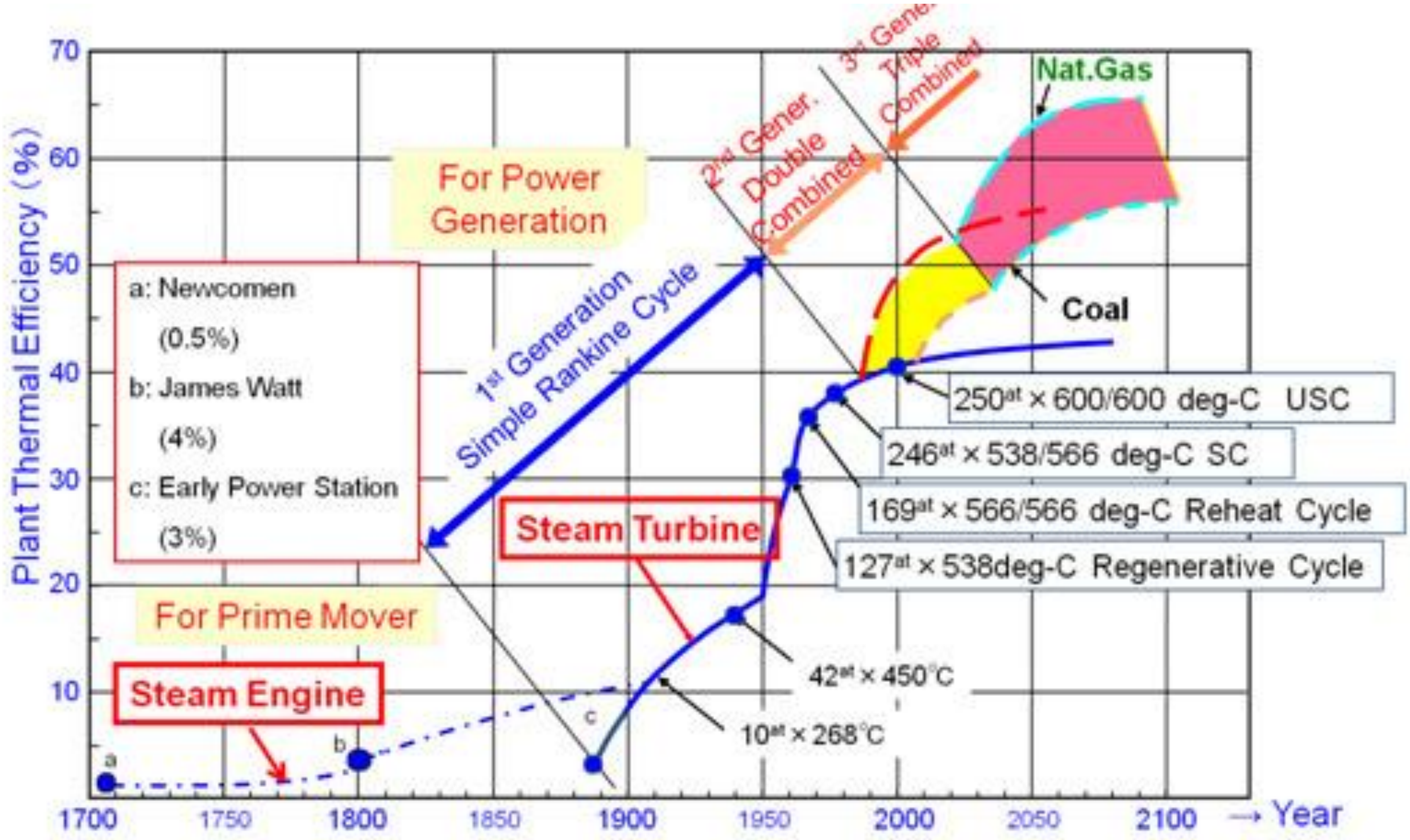
- *Answers: (a) 88.1 (b) 34.1 percent, (c) 62.7 kg/s*

How can we increase the efficiency of the rankine cycle?

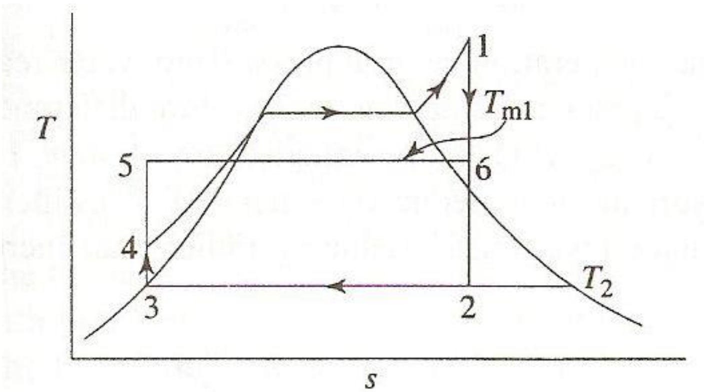
- Steam power plants are responsible for the production of most electric power in the world, and ***even small increases in thermal efficiency can mean large savings from the fuel requirements***. Therefore, every effort is made to improve the efficiency of the cycle on which steam power plants operate.The basic idea behind all the modifications to increase the thermal efficiency of a power cycle is the same :
  - *Increase the average temperature at which heat is transferred to the working fluid in the boiler, or decrease the average temperature at which heat is rejected from the working fluid in the condenser. That is, the average fluid temperature should be as high as possible* during heat addition and as low as possible during heat rejection.

- The basic idea behind all the modifications to increase the thermal efficiency of a power cycle is the same :
  - ***Lowering the condenser pressure***
  - ***Superheating the steam to high temperatures***
  - ***Increasing the boiler pressure***

How can we increase the efficiency of the rankine cycle ?



How can we increase the efficiency of the rankine cycle ?



Mean Temperature of Heat Addition

- Heat added reversibly at a constant pressure at 5-6 and T_{m1} is called mean temperature of heat addition. So, the total area under 5-6 is equal to the area under 4-1. Then, heat added is:

$$Q_1 = h_1 - h_4 = T_{m1}(s_1 - s_4)$$

Since heat rejected $Q_2 = h_2 - h_3 = T_2(s_1 - s_4)$

Mean Temperature of Heat Addition

$$\eta_{Rankine} = 1 - \frac{Q_2}{Q_1} = 1 - \frac{T_2(s_1 - s_4)}{T_{m1}(s_1 - s_4)} = 1 - \frac{T_2}{T_{m1}}$$

$$\eta_{Rankine} = f(T_{m1})$$

Cycle efficiency increases when the mean temperature of heat addition increases

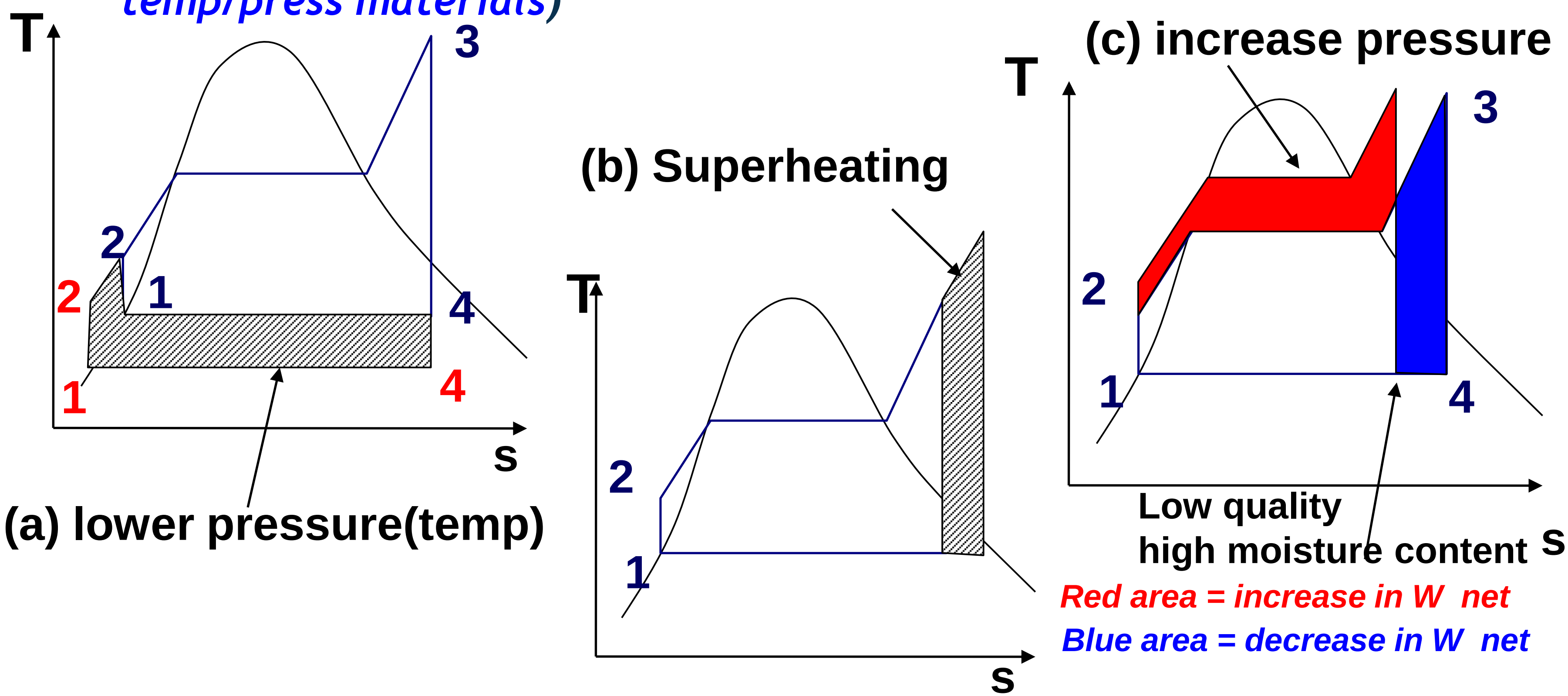
How can we increase the efficiency of the rankine cycle?

Thermal efficiency can be improved by manipulating the temperatures and/or pressures in various components

(a) Lowering the condensing pressure (*lowers* T_L, *but decreases quality*, x_4)

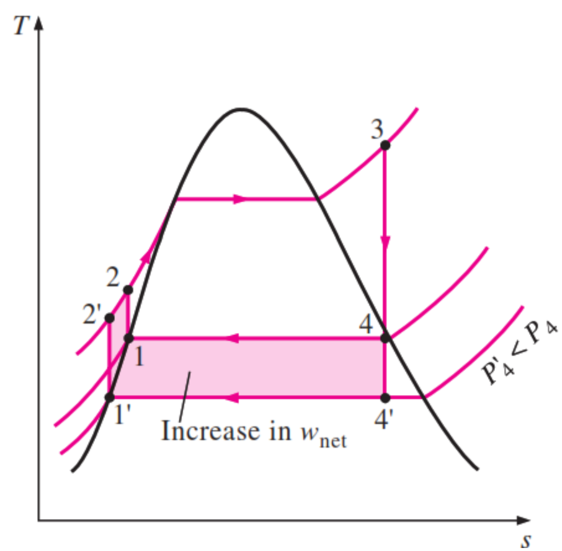
(b) Superheating the steam to a *higher temperature* (*increases* T_H *but requires higher temp materials*)

(c) Increasing the boiler pressure (*increases* T_H *but requires higher temp/press materials*)



Lowering the Condenser Pressure (Lowers $T_{low,av}$)

- 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 , but this increase is very small.
- The ***overall effect*** of lowering the condenser pressure is an ***increase*** in the ***thermal efficiency*** of the cycle.



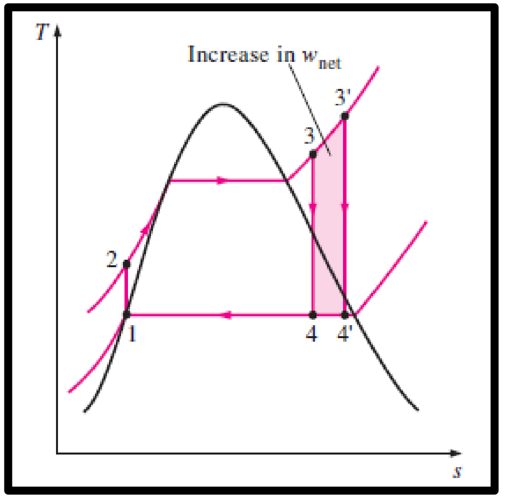
Lowering the Condenser Pressure (Lowers $T_{low,av}$)

Decrease condenser pressure

Good news	Bad news
- Increase in the net Work. - Increase in the efficiency.	- Increase in Q_H (but this is small compared to the increase in the net work). - Limited by the saturation pressure corresponding to the cooling medium (ex: river, sea, lake) - More moisture within the turbine.

Superheating the Steam to High Temperatures (Increases $T_{high,av}$)

- 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***.
- The temperature to which ***steam*** can be ***superheated*** is limited by ***metallurgical considerations***.
- Presently, the ***highest steam temperature allowed*** at the turbine inlet is about 650 ^{0}C.



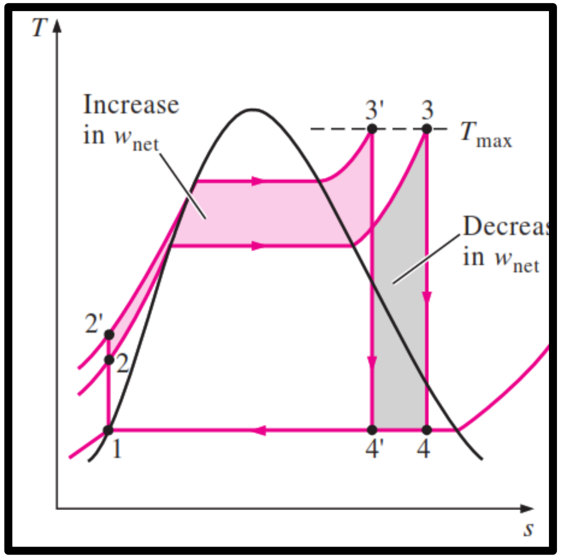
Superheating the Steam to High Temperatures (Increases $T_{high,av}$)

Superheating the steam

Good news	Bad news
- Increase in the net Work.	- Significant increase in Q_H.
- Increase in the efficiency.	- Metallurgical limitations ($T_3 < 620$ °C) to protect turbine blades.
- Decrease moisture.	

Increasing the Boiler Pressure (Increase $T_{high,av}$)

- Another way of increasing the average temperature during the heat-addition process is to increase the operating pressure of the boiler, which automatically raises the temperature at which boiling takes place.
- This, in turn, raises the average temperature at which heat is added to the steam and thus ***raises*** the ***thermal efficiency*** of the cycle.



Increasing the Boiler Pressure (Increase $T_{high,av}$)

Increase boiler pressure

Good news	Bad news
- Increase Temperature at which boiling take place (reduce Q_H).	- increase moisture.
- Increase in the efficiency.	

Exercise 1.3. Effect of Boiler Pressure and Temperature on Efficiency

- Consider a steam power plant operating on the ideal Rankine cycle. Steam enters the turbine at 3 MPa and 350°C and is condensed in the condenser at a pressure of 10 kPa. Determine

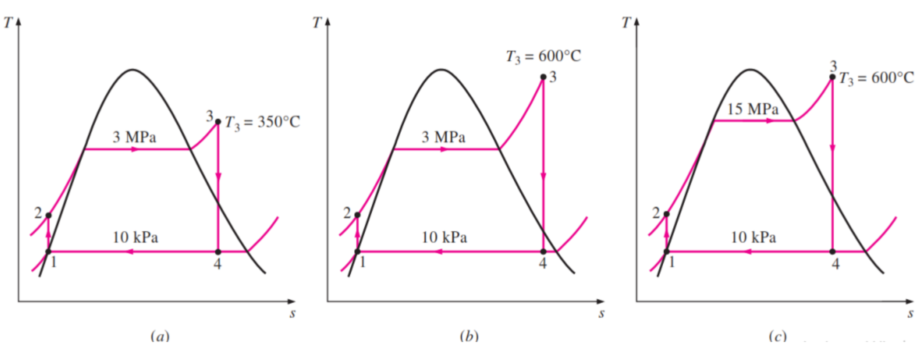
A. *The thermal efficiency of this power* plant

B. *The thermal efficiency if steam is superheated to 600°C instead of* 350°C

C. *The thermal efficiency if the boiler pressure is raised to 15* MPa while the turbine inlet temperature is maintained at 600°C.

D. *The thermal efficiency if the boiler pressure is raised to 15* MPa while the turbine inlet temperature is maintained at 600°C and condenser pressure is reduced to 5 kpa.

E. Repeat solving from A - D using isentropic pump and turbine efficiency of 86 and 88 % respectively.



Solution

(*a*) This is the steam power plant discussed in Example 10–1, except that the condenser pressure is lowered to 10 kPa. The thermal efficiency is determined in a similar manner:

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

State 2: $P_2 = 3\text{ MPa}$
$s_2 = s_1$

$$w_{\text{pump,in}} = v_1(P_2 - P_1) = (0.00101\text{ m}^3\text{/kg})[(3000 - 10)\text{ kPa}]\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= 3.02\text{ kJ/kg}$$

$$h_2 = h_1 + w_{\text{pump,in}} = (191.81 + 3.02)\text{ kJ/kg} = 194.83\text{ kJ/kg}$$

State 3: $\left.\begin{array}{l} P_3 = 3\text{ MPa} \\ T_3 = 350°\text{C} \end{array}\right\}$ $\begin{array}{l} h_3 = 3116.1\text{ kJ/kg} \\ s_3 = 6.7450\text{ kJ/kg}\cdot\text{K} \end{array}$

State 4: $P_4 = 10\text{ kPa}$ (sat. mixture)
$s_4 = s_3$

$$x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.7450 - 0.6492}{7.4996} = 0.8128$$

Thus,

$$h_4 = h_f + x_4 h_{fg} = 191.81 + 0.8128(2392.1) = 2136.1\text{ kJ/kg}$$

$$q_{\text{in}} = h_3 - h_2 = (3116.1 - 194.83)\text{ kJ/kg} = 2921.3\text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = (2136.1 - 191.81)\text{ kJ/kg} = 1944.3\text{ kJ/kg}$$

and

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1944.3\text{ kJ/kg}}{2921.3\text{ kJ/kg}} = \mathbf{0.334\text{ or }33.4\%}$$

Therefore, the thermal efficiency increases from 26.0 to 33.4 percent as a result of lowering the condenser pressure from 75 to 10 kPa. At the same time, however, the quality of the steam decreases from 88.6 to 81.3 percent (in other words, the moisture content increases from 11.4 to 18.7 percent).

(*b*) States 1 and 2 remain the same in this case, and the enthalpies at state 3 (3 MPa and 600°C) and state 4 (10 kPa and $s_4 = s_3$) are determined to be

$$h_3 = 3682.8\text{ kJ/kg}$$

$$h_4 = 2380.3\text{ kJ/kg} \quad (x_4 = 0.915)$$

Thus,

$$q_{\text{in}} = h_3 - h_2 = 3682.8 - 194.83 = 3488.0\text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = 2380.3 - 191.81 = 2188.5\text{ kJ/kg}$$

and

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2188.5\text{ kJ/kg}}{3488.0\text{ kJ/kg}} = \mathbf{0.373\text{ or }37.3\%}$$

Therefore, the thermal efficiency increases from 33.4 to 37.3 percent as a result of superheating the steam from 350 to 600°C. At the same time, the quality of the steam increases from 81.3 to 91.5 percent (in other words, the moisture content decreases from 18.7 to 8.5 percent).

(*c*) State 1 remains the same in this case, but the other states change. The enthalpies at state 2 (15 MPa and $s_2 = s_1$), state 3 (15 MPa and 600°C), and state 4 (10 kPa and $s_4 = s_3$) are determined in a similar manner to be

$$h_2 = 206.95\text{ kJ/kg}$$

$$h_3 = 3583.1\text{ kJ/kg}$$

$$h_4 = 2115.3\text{ kJ/kg} \quad (x_4 = 0.804)$$

Thus,

$$q_{\text{in}} = h_3 - h_2 = 3583.1 - 206.95 = 3376.2\text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = 2115.3 - 191.81 = 1923.5\text{ kJ/kg}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1923.5\text{ kJ/kg}}{3376.2\text{ kJ/kg}} = \mathbf{0.430\text{ or }43.0\%}$$

Discussion The thermal efficiency increases from 37.3 to 43.0 percent as a result of raising the boiler pressure from 3 to 15 MPa while maintaining the turbine inlet temperature at 600°C. At the same time, however, the quality of the steam decreases from 91.5 to 80.4 percent (in other words, the moisture content increases from 8.5 to 19.6 percent).

1.4. The Ideal Reheat Rankine Cycle

- ***Increasing the boiler pressure*** increases the thermal efficiency of the Rankine cycle, but it also ***increases the moisture content of the steam*** to unacceptable levels.
- Then, it is natural to ask the following question:
- 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 increases, thus increasing the cycle efficiency.
   - **This is not a viable solution, however, since it will require raising the steam temperature to metallurgically unsafe levels.**

1.4. The Ideal Reheat Rankine Cycle

2. Expand the steam in the turbine in two stages, and reheat it in between.

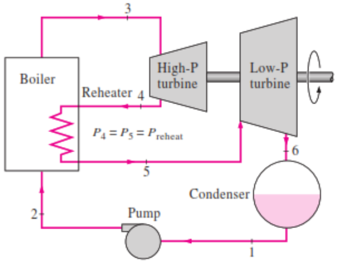
- In other words, ***modify the simple ideal Rankine cycle*** with a **reheat** process.
- Reheating is a practical solution to the excessive moisture problem in turbines, and it is used frequently in modern steam power plants.

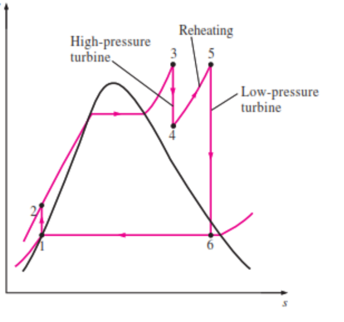
1.4. The Ideal Reheat Rankine Cycle

- *The ideal Reheat Rankine cycle differs from the simple ideal Rankine cycle in that the expansion process takes place in two stages.*
- *In the first stage (high-pressure turbine), steam is expanded isentropically to an intermediate pressure and sent back to the boiler where it is reheated at constant pressure, usually to the inlet temperature of the first turbine stage.*
- *Steam then expands isentropically in the second stage (low-pressure turbine) to the condenser pressure.*
- *The total heat input and the total turbine work output for a reheat cycle become :*

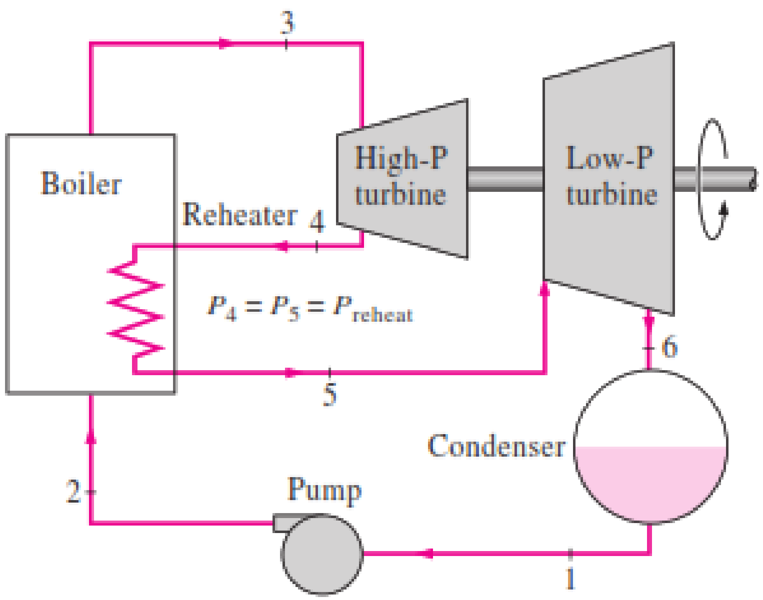
$$q_{in} = q_{primary} + q_{reheat} = (h_3 - h_2) + (h_5 - h_4)$$

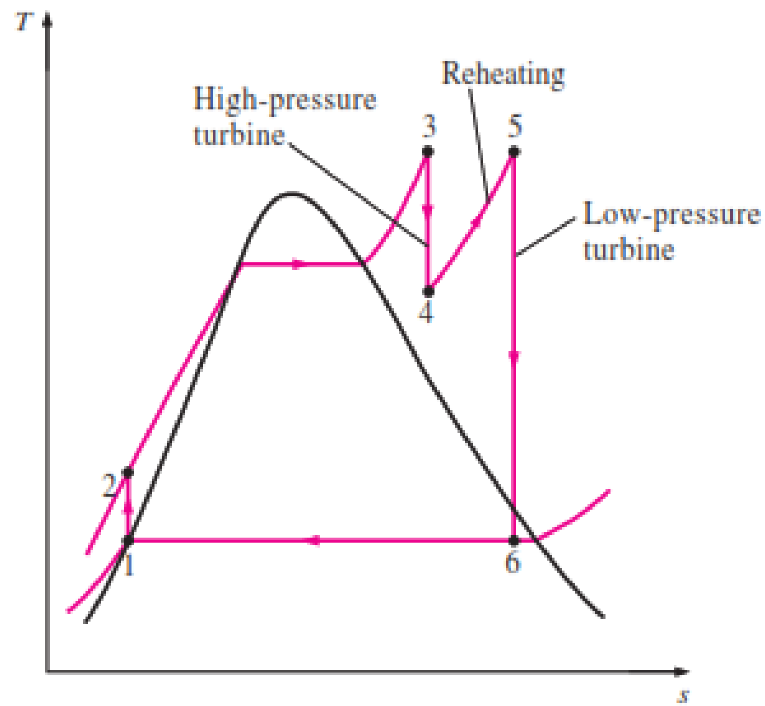
$$w_{turb,out} = w_{turb,I} + w_{turb,II} = (h_3 - h_4) + (h_5 - h_6)$$





1.4. The Ideal Reheat Rankine Cycle





Exercise - 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

(b) The thermal efficiency of the cycle. Assume the steam is reheated to the inlet temperature of the high-pressure turbine.

Solution

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

State 6: $P_6 = 10$ kPa

$x_6 = 0.896$ (sat. mixture)

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

Also,

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

Thus,

State 5: $\left.\begin{array}{l} T_5 = 600°\text{C} \\ s_5 = s_6 \end{array}\right\}$ $P_5 = \mathbf{4.0\ MPa}$, $h_5 = 3674.9$ 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: $\left.\begin{array}{l} P_1 = 10 \text{ kPa} \\ \text{Sat. liquid} \end{array}\right\}$ $h_1 = h_{f\,@\,10\text{ kPa}} = 191.81$ kJ/kg, $v_1 = v_{f\,@\,10\text{ kPa}} = 0.00101$ m³/kg

State 2: $P_2 = 15$ 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: $\left.\begin{array}{l} P_3 = 15 \text{ MPa} \\ T_3 = 600°\text{C} \end{array}\right\}$ $h_3 = 3583.1$ kJ/kg, $s_3 = 6.6796$ kJ/kg·K

State 4: $\left.\begin{array}{l} P_4 = 4 \text{ MPa} \\ s_4 = s_3 \end{array}\right\}$ $h_4 = 3155.0$ kJ/kg, ($T_4 = 375.5°$C)

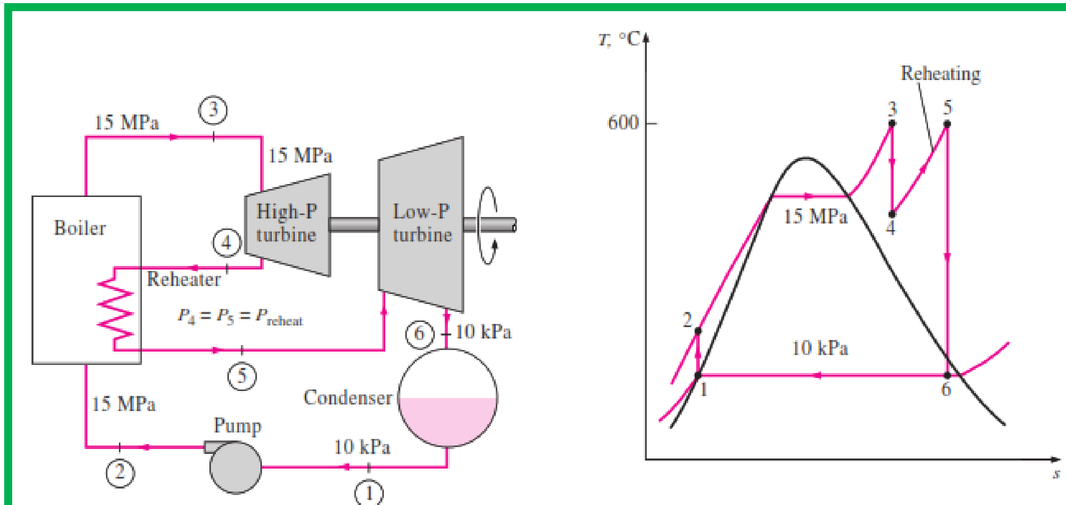
Thus

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4)$$
$$= (3583.1 - 206.95) \text{ kJ/kg} + (3674.9 - 3155.0) \text{ kJ/kg}$$
$$= 3896.1 \text{ kJ/kg}$$
$$q_{\text{out}} = h_6 - h_1 = (2335.1 - 191.81) \text{ kJ/kg}$$
$$= 2143.3 \text{ kJ/kg}$$

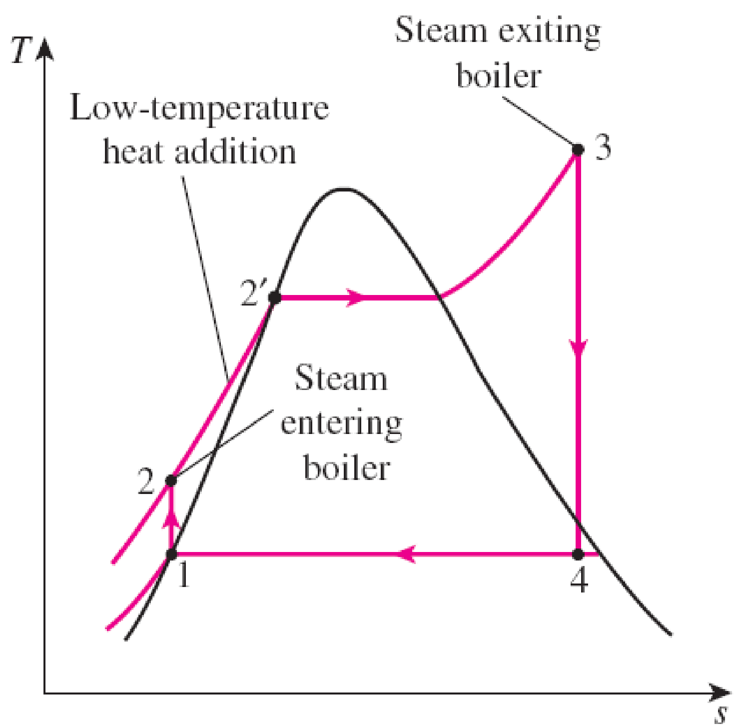
and

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

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.



THE IDEAL REGENERATIVE RANKINE CYCLE



The first part of the heat-addition process in the boiler takes place at relatively low temperatures.

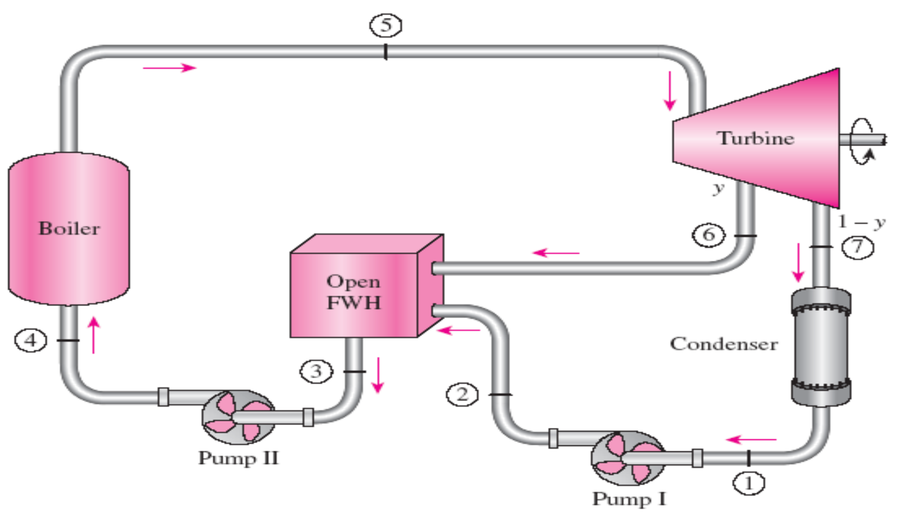
Heat is transferred to the working fluid during process 2-2 at a relatively low temperature. This lowers the average heat-addition temperature and thus the cycle efficiency.

In steam power plants, steam is extracted from the turbine at various points. This steam, which could have produced more work by expanding further in the turbine, is used to heat the feedwater instead. The device where the feedwater is heated by regeneration is called a regenerator, or a feedwater heater (FWH).

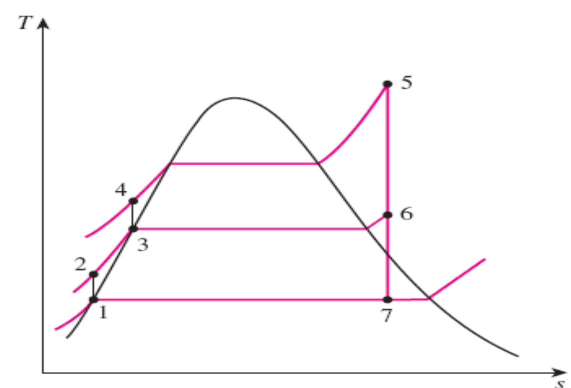
A feedwater heater is basically a heat exchanger where heat is transferred from the steam to the feedwater either by mixing the two fluid streams (open feedwater heaters) or without mixing them (closed feedwater heaters).

Open Feedwater heaters

An **open** (or **direct-contact**) **feedwater heater** is basically a ***mixing chamber***, **where the steam extracted from the turbine mixes with the feedwater exiting the pump. Ideally, the mixture leaves the heater as a saturated liquid at the heater pressure.**



The ideal regenerative Rankine cycle with an open feedwater heater.



Energy Analyses

The heat and work interactions in a regenerative Rankine cycle with one feedwater heater can be expressed (per unit mass of steam flowing through the boiler), as follows:

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

$$q_{out} = (1 - y)(h_7 - h_1)$$

$$w_{turb,out} = (h_5 - h_6) + (1 - y)(h_6 - h_7)$$

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

Mass fraction of steam extracted from the turbine,

$$y = \dot{m}_6/\dot{m}_5$$

Pump work input,

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

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

Mass of Steam Extracted

For each 1 kg of steam leaving the boiler, y kg expands partially in the turbine and is extracted at state 6.

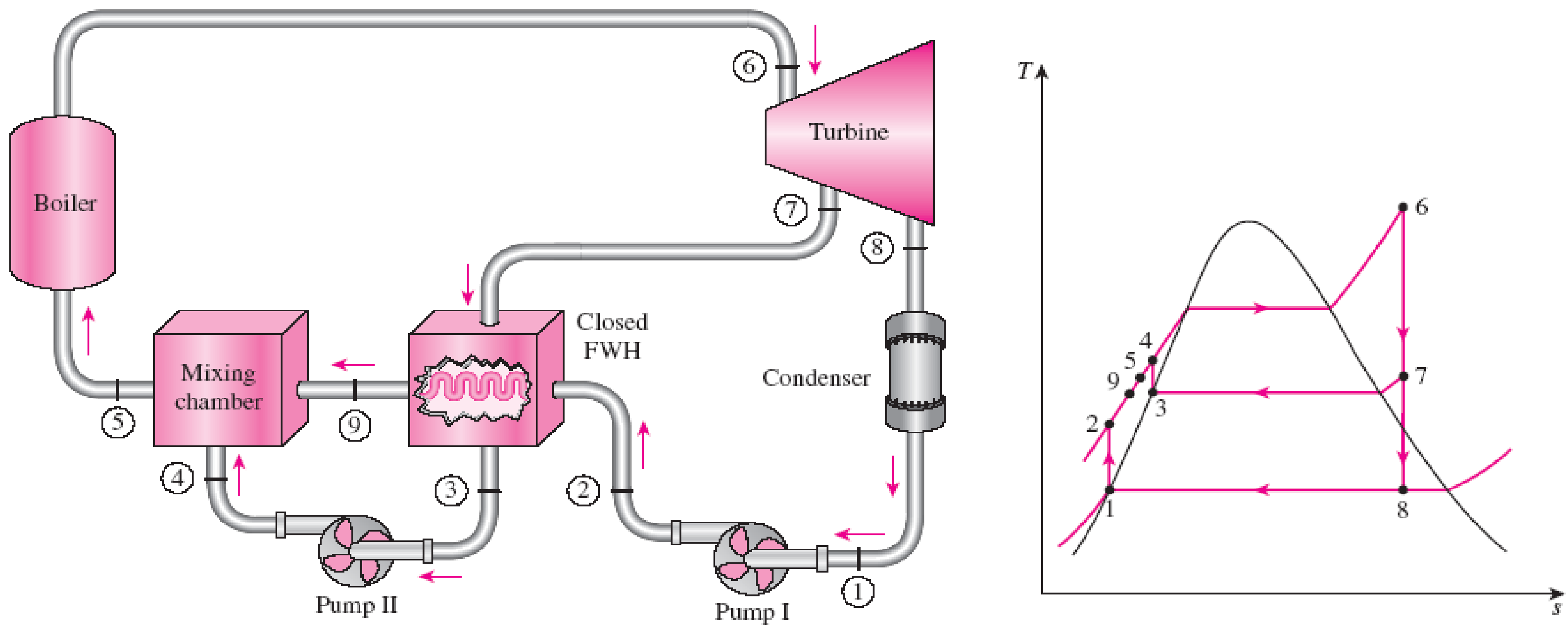
The remaining (1-y) kg of the steam expands to the condenser pressure.

Therefore, the mass flow rates of the steam will be different in different components.

Note: The cycle efficiency increases further as the number of feedwater heaters is increased.

Closed Feedwater Heaters

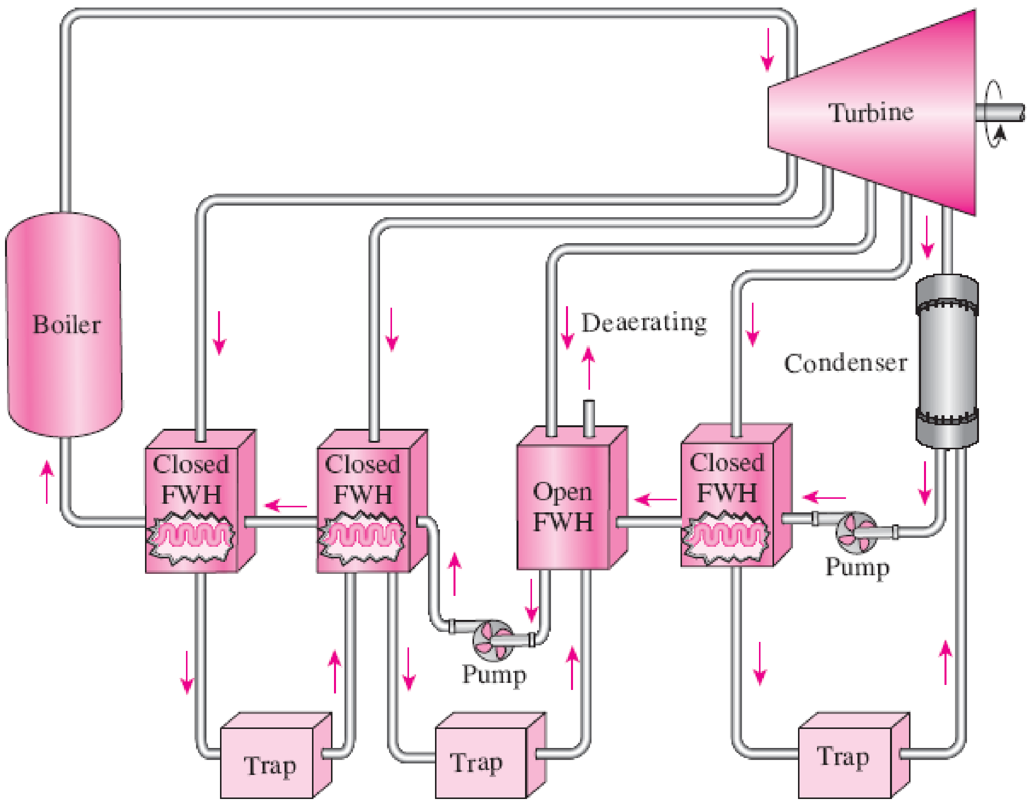
Another type of feedwater heater frequently used in steam power plants is the closed feedwater heater, in which heat is transferred from the extracted steam to the feedwater without any mixing taking place. The two streams now can be at different pressures, since they do not mix.



The ideal regenerative Rankine cycle with a closed feedwater heater.

Closed Feedwater Heaters

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 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.



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

Most steam power plants use a combination of open and closed feedwater heaters.

Open vs. Closed Feedwater Heater

Open FWHs

Open feedwater heaters are simple and inexpensive. They have good heat transfer characteristics.

For each feedwater heater used, additional feedwater pump is required.

Closed FWHs

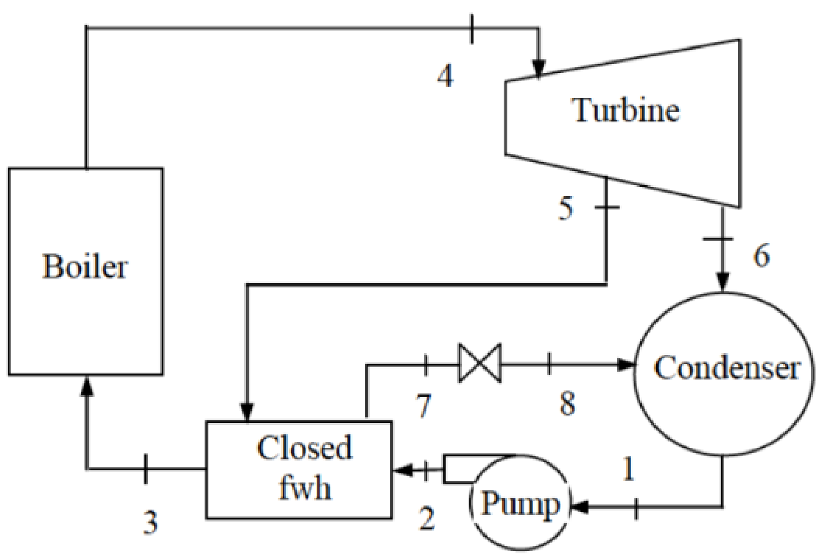
The closed feedwater heaters are more complex because of the internal tubing network. Thus they are more expensive.

Heat transfer in closed feedwater heaters is less effective since the two streams are not allowed to be in direct contact.

The closed feedwater heaters do not require a separate pump for each FWH since the extracted steam and the feedwater can be at different pressures.

Exercise

- Consider a steam power plant that operates on the ideal regenerative Rankine cycle with a closed feedwater heater as shown in the figure. The plant maintains the turbine inlet at 3000 kPa and 350 ^oC; and operates the condenser at 20 kPa. Steam is extracted at 1000 kPa to serve the closed feedwater heater, which discharges into the condenser after being throttled to condenser pressure. Calculate the work produced by the turbine, the work consumed by the pump, and the heat supply in the boiler for this cycle per unit of boiler flow rate.

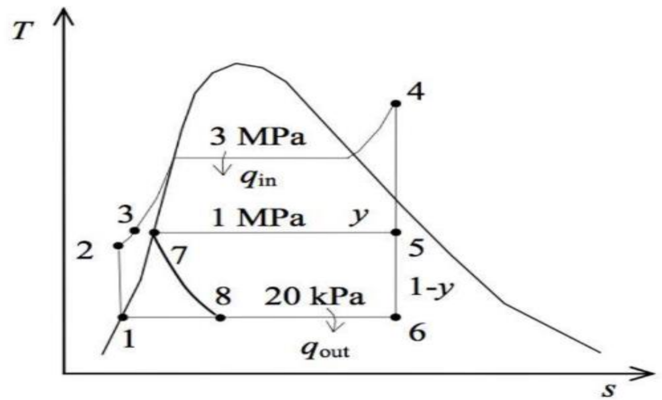


Solution

Analysis From the steam tables

$$h_1 = h_{f@\,20\,\text{kPa}} = 251.42\ \text{kJ/kg}$$

$$v_1 = v_{f@\,20\,\text{kPa}} = 0.001017\ \text{m}^3/\text{kg}$$



$$\left.\begin{array}{l} P_6 = 20\ \text{kPa} \\ s_6 = s_4 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{6.7450 - 0.8320}{7.0752} = 0.8357 \\ h_6 = h_f + x_6 h_{fg} = 251.42 + (0.8357)(2357.5) = 2221.7\ \text{kJ/kg} \end{array}$$

For an ideal closed feedwater heater, the feedwater is heated to the exit temperature of the extracted steam, which ideally leaves the heater as a saturated liquid at the extraction pressure

$$\left.\begin{array}{l} P_7 = 1000\ \text{kPa} \\ x_7 = 0 \end{array}\right\} \begin{array}{l} h_7 = 762.51\ \text{kJ/kg} \\ T_7 = 179.9°\text{C} \end{array}$$

$$h_8 = h_7 = 762.51\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 3000\ \text{kPa} \\ T_3 = T_7 = 209.9°\text{C} \end{array}\right\} h_3 = 763.53\ \text{kJ/kg}$$

$$\begin{aligned} w_{p,in} &= v_1(P_2 - P_1) \\ &= (0.001017\ \text{m}^3/\text{kg})(3000 - 20)\text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 3.03\ \text{kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,in} = 251.42 + 3.03 = 254.45\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 3000\ \text{kPa} \\ T_4 = 350°\text{C} \end{array}\right\} \begin{array}{l} h_4 = 3116.1\ \text{kJ/kg} \\ s_4 = 6.7450\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_5 = 1000\ \text{kPa} \\ s_5 = s_4 \end{array}\right\} h_5 = 2851.9\ \text{kJ/kg}$$

An energy balance on the heat exchanger gives the fraction of steam extracted from the turbine ($= \dot{m}_5/\dot{m}_4$) for closed feedwater heater:

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$$

$$\dot{m}_5 h_5 + \dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_7 h_7$$

$$y h_5 + 1h_2 = 1h_3 + y h_7$$

Rearranging,

$$y = \frac{h_3 - h_2}{h_5 - h_7} = \frac{763.53 - 254.45}{2851.9 - 762.51} = 0.2437$$

Then,

$$w_{T,out} = h_4 - h_5 + (1-y)(h_5 - h_6) = 3116.1 - 2851.9 + (1 - 0.2437)(2851.9 - 2221.7) = \mathbf{740.9\ kJ/kg}$$

$$w_{P,in} = \mathbf{3.03\ kJ/kg}$$

$$q_{in} = h_4 - h_3 = 3116.1 - 763.53 = \mathbf{2353\ kJ/kg}$$

Also, $w_{net} = w_{T,out} - w_{P,in} = 740.9 - 3.03 = 737.8\ \text{kJ/kg}$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{737.8}{2353} = 0.3136$$

Exercise

- A steam power plant operates on an ideal reheat-regenerative Rankine cycle and has a net power output of 80 MW. Steam enters the high-pressure turbine at 10 MPa and 550°C and leaves at 0.8 MPa. Some steam is extracted at this pressure to heat the feedwater in an open feedwater heater. The rest of the steam is reheated to 500°C and is expanded in the low-pressure turbine to the condenser pressure of 10 kPa. Show the cycle on a T-s diagram and determine:

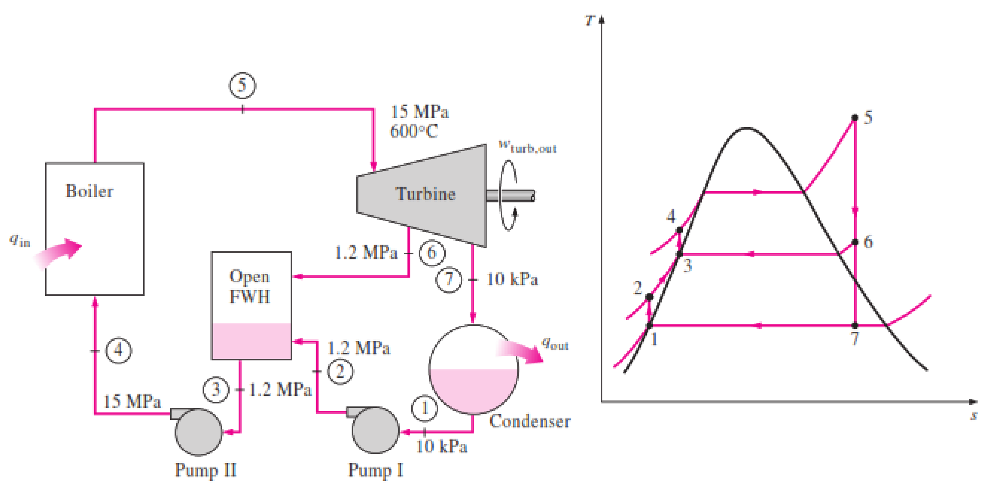
(a) The mass flow rate of steam through the boiler, and

(b) thermal efficiency of the cycle.

Answers: (a) 54.5 kg/s, (b) 44.4 percent

Exercise

- 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

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

State 2: $P_2 = 1.2 \text{ MPa}$

$s_2 = s_1$

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

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

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

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

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

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

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

$$s_7 = s_5 \qquad x_7 = \frac{s_7 - s_f}{s_{fg}} = \frac{6.6796 - 0.6492}{7.4996} = 0.8041$$

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

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

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

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,

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

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

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

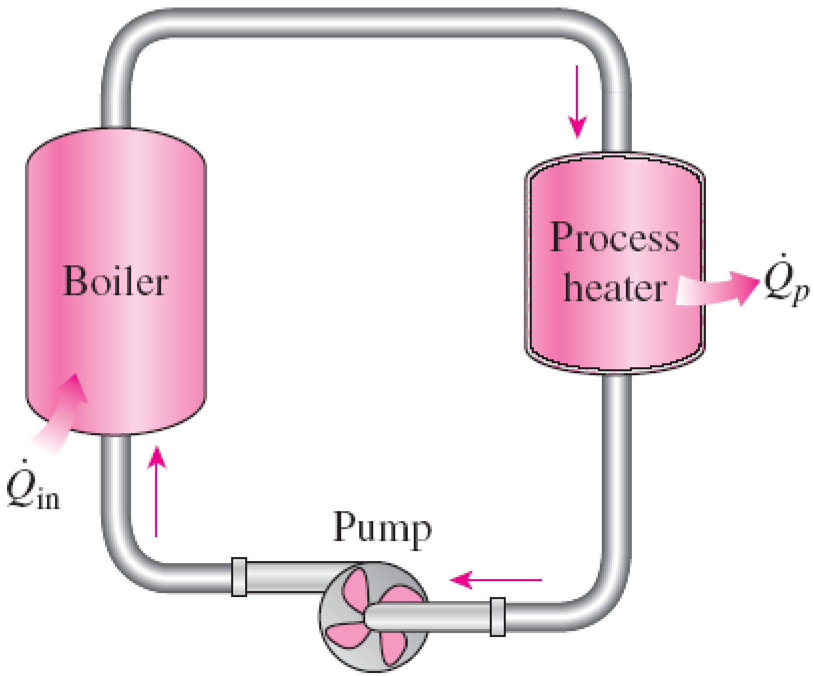
and

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

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.

COGENERATION

Many industries require energy input in the form of heat, called *process heat*. Process heat in these industries is usually supplied by steam at 5 to 7 atm and 150 to 200°C. Energy is usually transferred to the steam by burning coal, oil, natural gas, or another fuel in a furnace.



A simple process-heating plant.

Industries that use large amounts of process heat also consume a large amount of electric power.

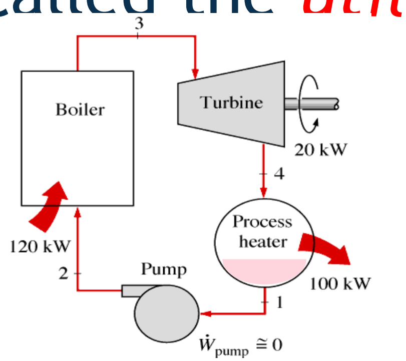
It makes sense to use the already-existing work potential to produce power instead of letting it go to waste.

The result is a plant that produces electricity while meeting the process-heat requirements of certain industrial processes (cogeneration plant)

Cogeneration: The production of more than one useful form of energy (such as process heat and electric power) from the same energy source.

COGENERATION

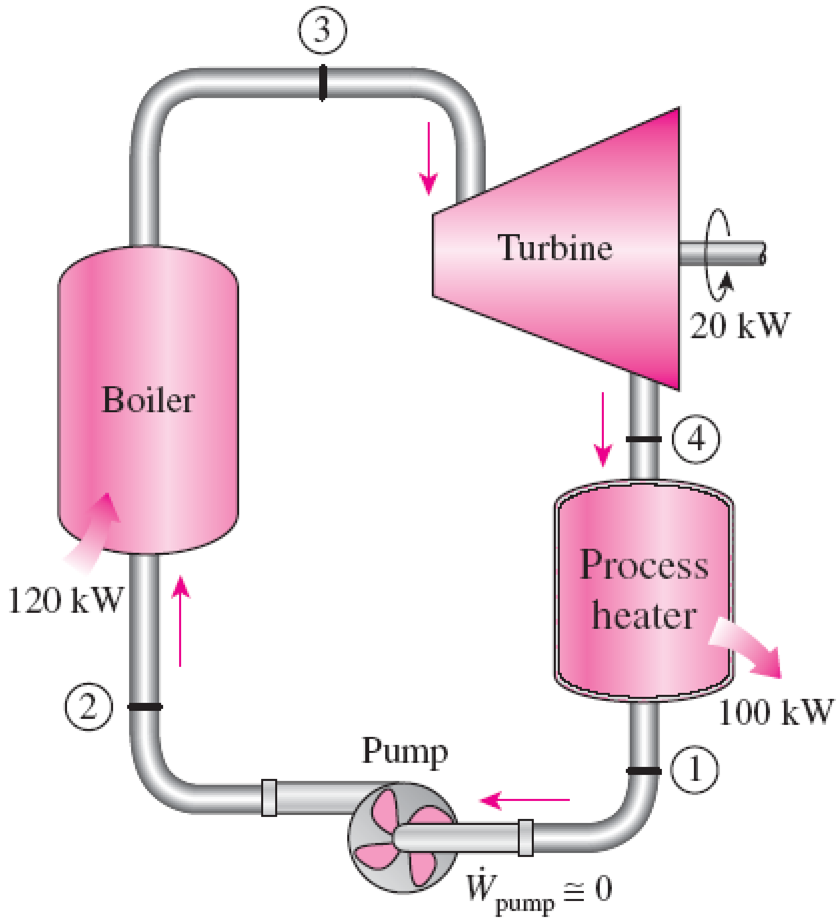
- The production of more than one useful form of energy (such as process heat and electric power) from the same energy source is called *cogeneration*.
- Cogeneration plants produce electric power while meeting the process heat requirements of certain industrial processes. This way, more of the energy transferred to the fluid in the boiler is utilized for a useful purpose.
- The fraction of energy that is used for either process heat or power generation is called the *utilization factor* of the cogeneration plant.



COGENERATION

Utilization factor

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

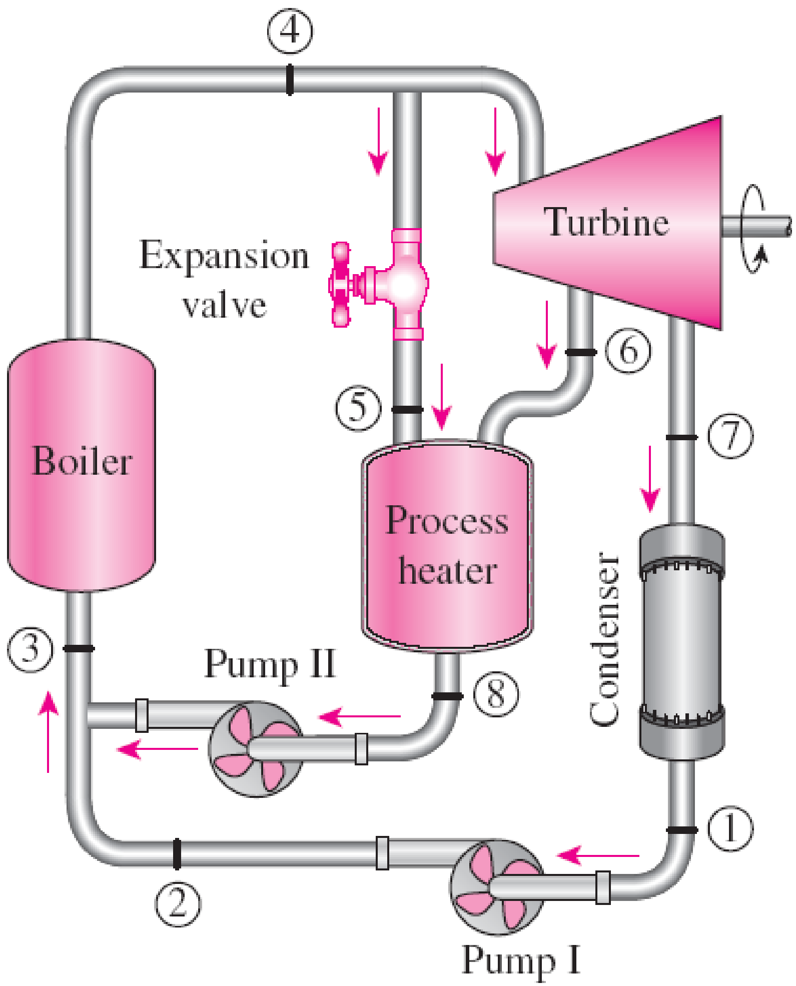


An ideal cogeneration plant.

$$\epsilon_u = 1 - \frac{\dot{Q}_{out}}{\dot{Q}_{in}}$$

- The utilization factor of the ideal steam-turbine cogeneration plant is 100%.
- Actual cogeneration plants have utilization factors as high as 80%.
- Some recent cogeneration plants have even higher utilization factors.

COGENERATION



A cogeneration plant with adjustable loads.

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 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.

When there is no demand for process heat, all the steam passes through the turbine and the condenser (m_5=m_6=0), and the cogeneration plant operates as an ordinary steam power plant.

$$\dot{Q}_{in} = \dot{m}_3(h_4 - h_3)$$

$$\dot{Q}_{out} = \dot{m}_7(h_7 - h_1)$$

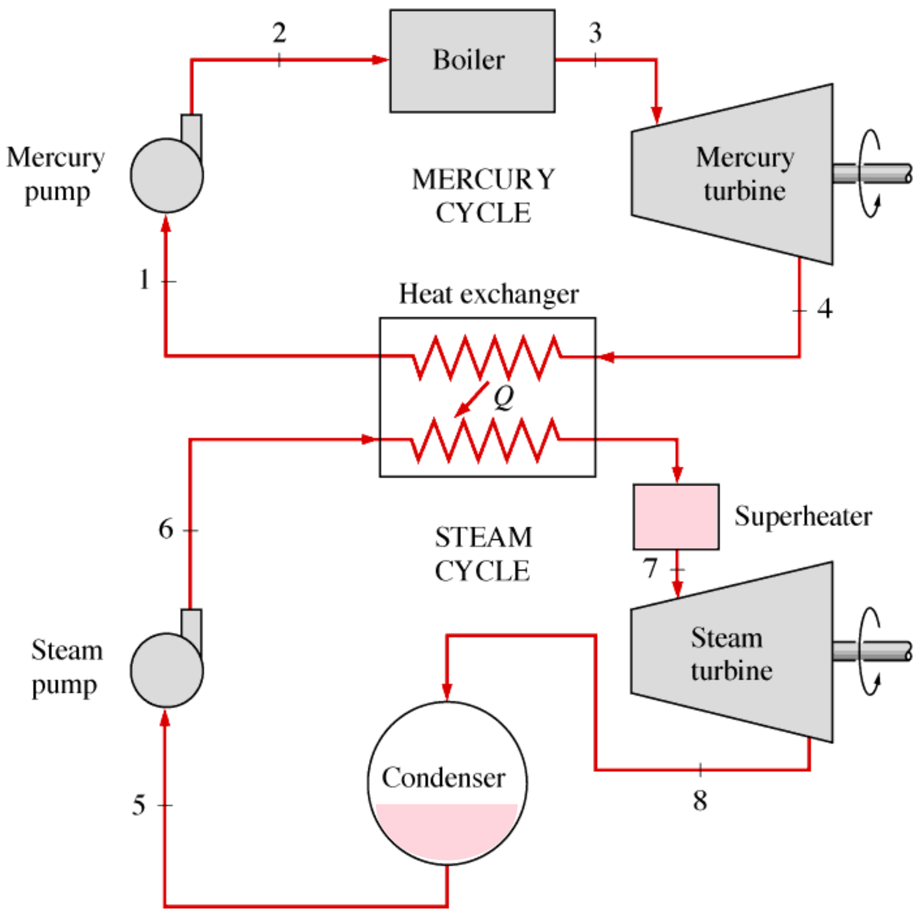
$$\dot{Q}_p = \dot{m}_5 h_5 + \dot{m}_6 h_6 - \dot{m}_8 h_8$$

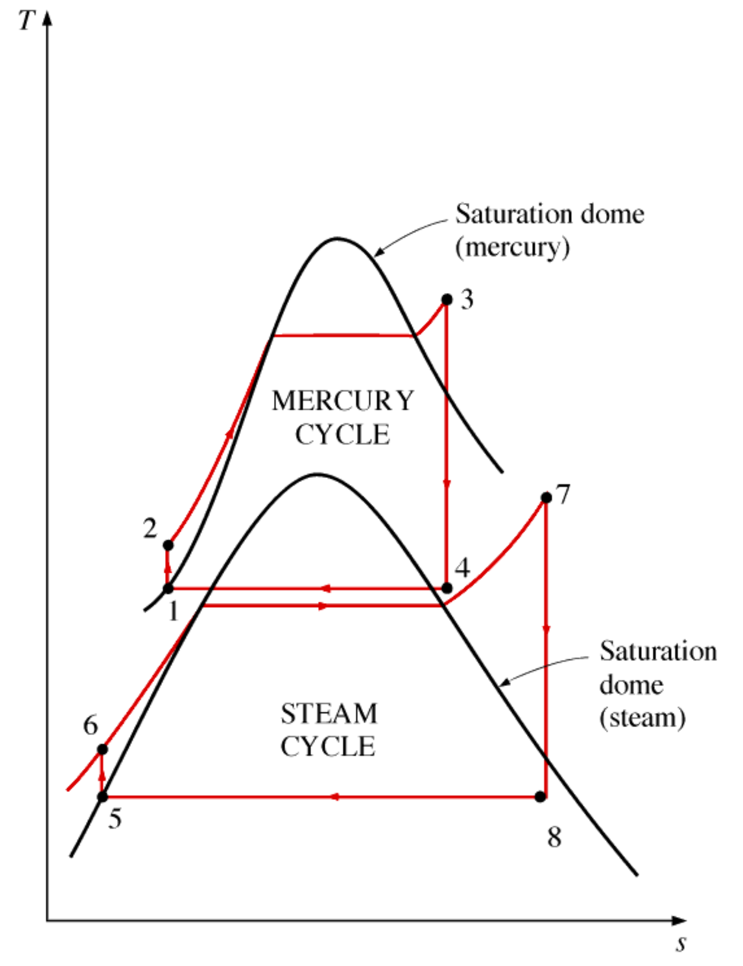
$$\dot{W}_{turb} = (\dot{m}_4 - \dot{m}_5)(h_4 - h_6) + \dot{m}_7(h_6 - h_7)$$

More Ways to Increase Power plant Thermal Efficiency

- The overall thermal efficiency of a power plant can be increased by using *binary cycles* or *combined cycles*.
- A binary cycle is composed of two separate cycles, one at high temperatures (topping cycle) and the other at relatively low temperatures.
- The most common combined cycle is the gas-steam combined cycle where a gas-turbine cycle operates at the high-temperature range and a steam-turbine cycle at the low-temperature range.
- Steam is heated by the high-temperature exhaust gases leaving the gas turbine. Combined cycles have a higher thermal efficiency than the steam- or gas-turbine cycles operating alone.

Mercury-Water Binary Vapor Cycle

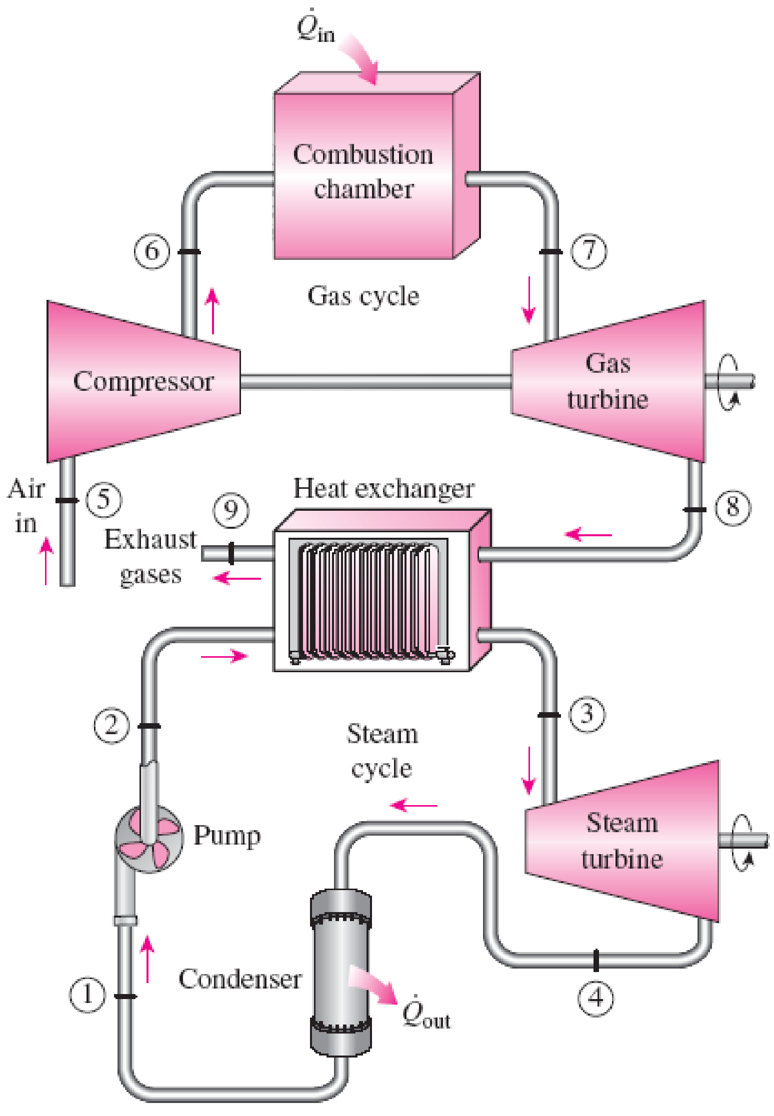


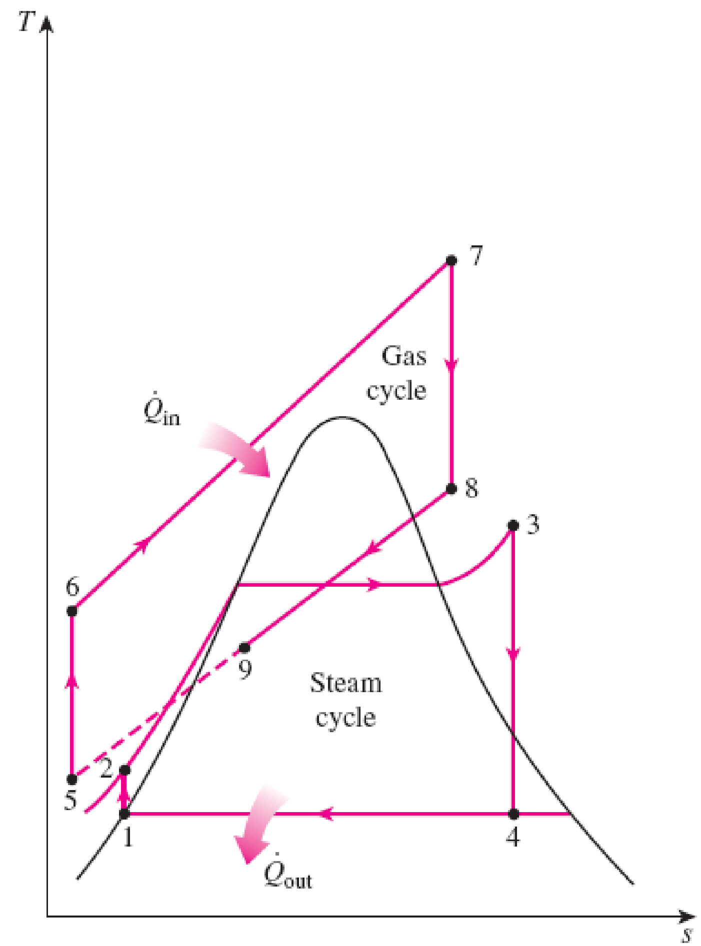


COMBINED GAS-VAPOR POWER CYCLES

- The continued quest for higher thermal efficiencies has resulted in rather innovative modifications to conventional power plants.
- A popular modification involves a gas power cycle topping a vapor power cycle, which is called the combined gas–vapor cycle, or just the combined cycle.
- The combined cycle of greatest interest is the gas-turbine (Brayton) cycle topping a steam-turbine (Rankine) cycle, which has a higher thermal efficiency than either of the cycles executed individually.
- It makes engineering sense to take advantage of the very desirable characteristics of the gas-turbine cycle at high temperatures *and* to use the high-temperature exhaust gases as the energy source for the bottoming cycle such as a steam power cycle. The result is a combined gas–steam cycle.
- Recent developments in gas-turbine technology have made the combined gas–steam cycle economically very attractive.
- The combined cycle increases the efficiency without increasing the initial cost greatly. Consequently, many new power plants operate on combined cycles, and many more existing steam- or gas-turbine plants are being converted to combined-cycle power plants.
- Thermal efficiencies over 50 % are reported.

Combined gas-steam power plant





Summary

- *The Carnot vapor cycle*
- *Rankine cycle: The ideal cycle for vapor power cycles*
  - *Energy analysis of the ideal Rankine cycle*
- *Deviation of actual vapor power cycles from idealized ones*
- *How can we increase the efficiency of the Rankine cycle?*
  - *Lowering the condenser pressure (Lowers $T_{low,avg}$)*
  - *Superheating the steam to high temperatures (Increases $T_{high,avg}$)*
  - *Increasing the boiler pressure (Increases $T_{high,avg}$)*
- *The ideal reheat Rankine cycle*
- *The ideal regenerative Rankine cycle*
  - *Open feedwater heaters and Closed feedwater heaters*
- *Cogeneration*
- *Combined gas-vapor power cycle*

Thank you