ESO201A : THERMODYNAMICS
2021-22 Ist semester
IIT Kanpur

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Lecture 28

VAPOR POWER CYCLES

In this and subsequent lectures, we will consider vapor power cycles in which working fluid is alternatively vaporized and condensed.

The continued quest for higher thermal efficiencies has resulted in some innovative modifications to the basic vapor power cycle. Among these, we discuss the *reheat* and *regenerative cycles*.

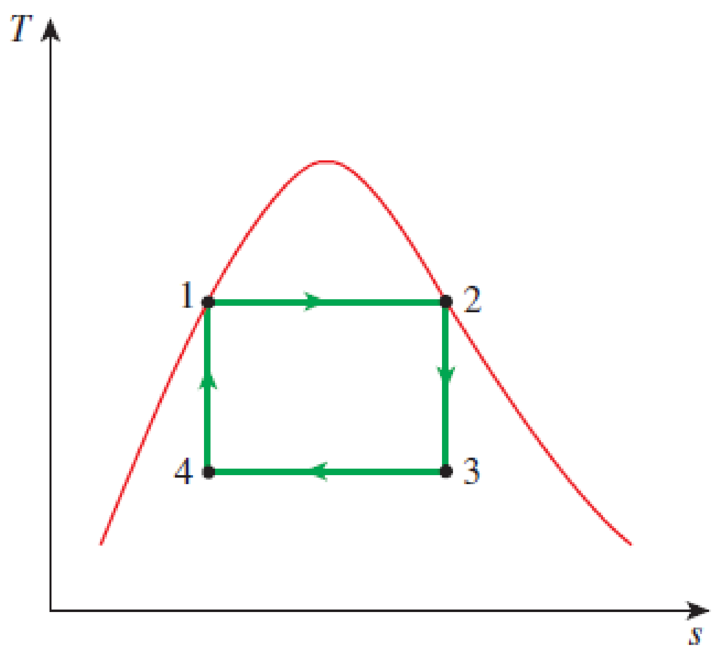
Steam is the most common working fluid used in vapor power cycles because of its many desirable characteristics, such as low cost, availability, and high enthalpy of vaporization.

Steam power plants are commonly referred to as *coal plants*, *nuclear plants*, or *natural gas plants*, depending on the type of fuel used to supply heat to the steam. However, the steam goes through the same basic cycle in all of them. Therefore, all can be analyzed in the same manner.

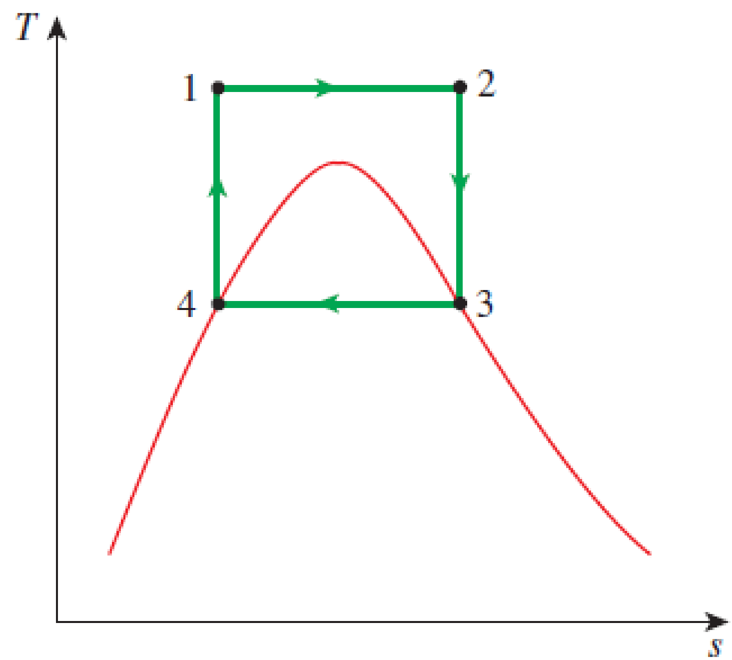
Ref. Cengel and Boles, 8th Edition (2015)

Is Carnot cycle suitable as ideal cycle for steam power plants ?

We have mentioned repeatedly that the Carnot cycle is the most efficient cycle operating between two specified temperature limits. Thus it is natural to look at the Carnot cycle first as a prospective ideal cycle for vapor power plants. If we could, we would certainly adopt it as the ideal cycle.



Prospective ideal cycle - 1



Prospective ideal cycle - 2

Both of the above prospective cycles are **impractical** !

Ref. Cengel and Boles, 8th Edition (2015)

RANKINE CYCLE: THE IDEAL CYCLE FOR VAPOR POWER CYCLES

Ref. Cengel and Boles, 8th Edition (2015)

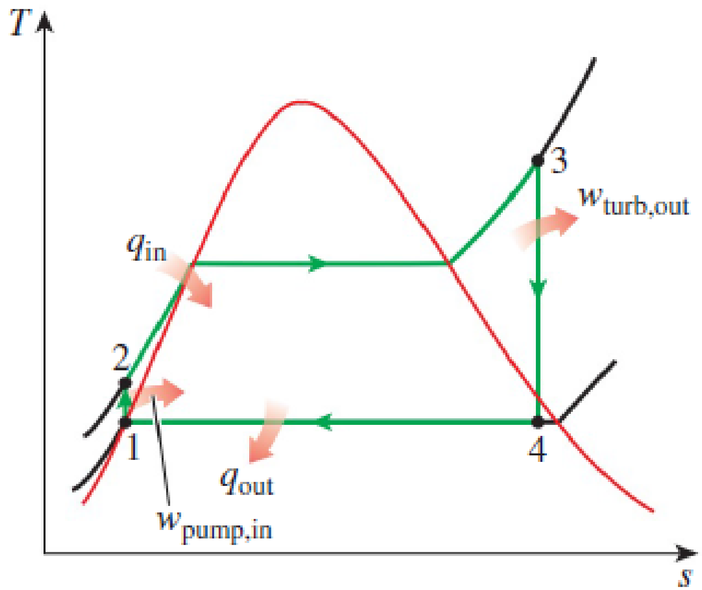
Many of the impracticalities associated with the Carnot cycle can be eliminated by superheating the steam in the boiler and condensing it completely in the condenser, as shown schematically on a *T-s* diagram in Fig.   The cycle that results is the **Rankine cycle**, which is the ideal cycle for vapor power plants. The ideal Rankine cycle does not involve any internal irreversibilities and consists of the following four processes:

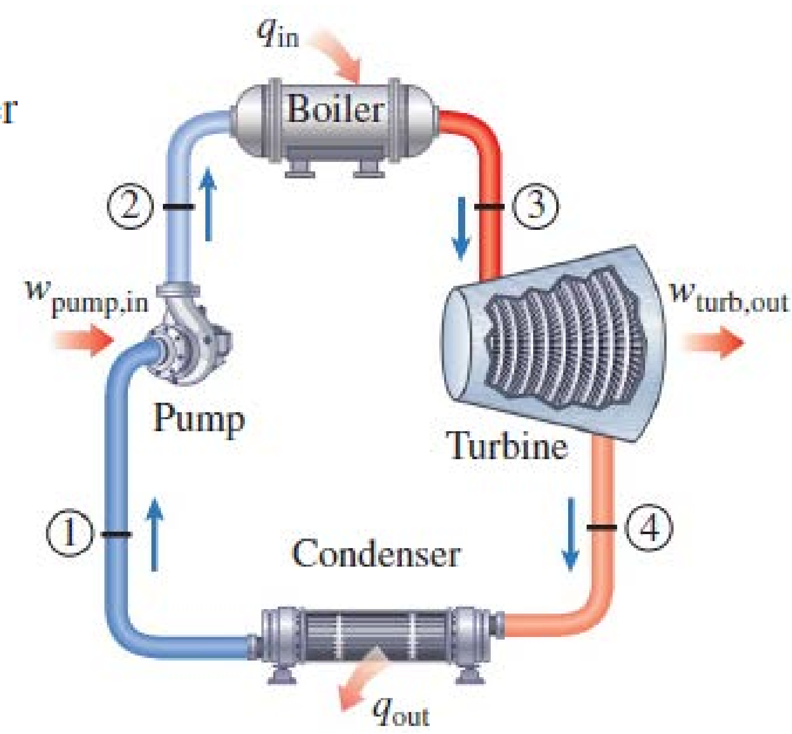
1-2 Isentropic compression in a pump

2-3 Constant pressure heat addition in a boiler

3-4 Isentropic expansion in a turbine

4-1 Constant pressure heat rejection in a condenser





Energy Analysis of the Ideal Rankine Cycle

Ref. Cengel and Boles, 8th Edition (2015)

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. Then the *steady-flow energy equation* per unit mass of steam reduces to

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

The boiler and the condenser do not involve any work, and the pump and the turbine are assumed to be isentropic. Then the conservation of energy relation for each device can be expressed as follows:

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

or,

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

where

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

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

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

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

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

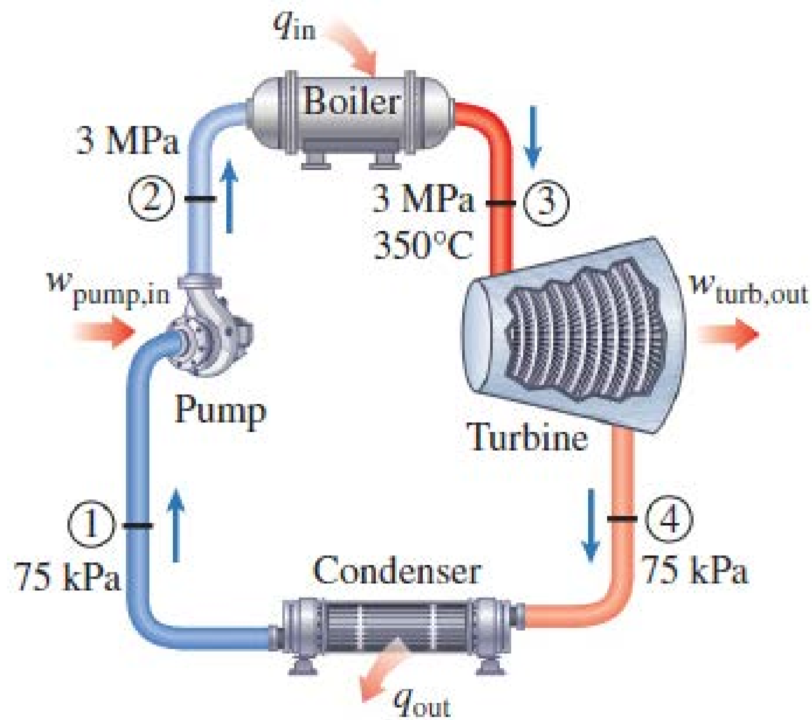
$$\eta_{th} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}}$$

where

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

Ref. Cengel and Boles, 8th Edition (2015)

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



SOLUTION A steam power plant operating on the simple ideal Rankine cycle is considered. The thermal efficiency of the cycle is to be determined.

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

Ref. Cengel and Boles, 8th Edition (2015)

First we determine the enthalpies at various points in the cycle, using data from steam tables (Tables A–4, A–5, and A–6):

State 1: $\left.\begin{array}{l} P_1 = 75 \text{ kPa} \\ \text{Sat. liquid} \end{array}\right\} \begin{array}{l} h_1 = h_{f\,@\,75\text{ kPa}} = 384.44 \text{ kJ/kg} \\ v_1 = v_{f\,@\,75\text{ kPa}} = 0.001037 \text{ m}^3\text{/kg} \end{array}$

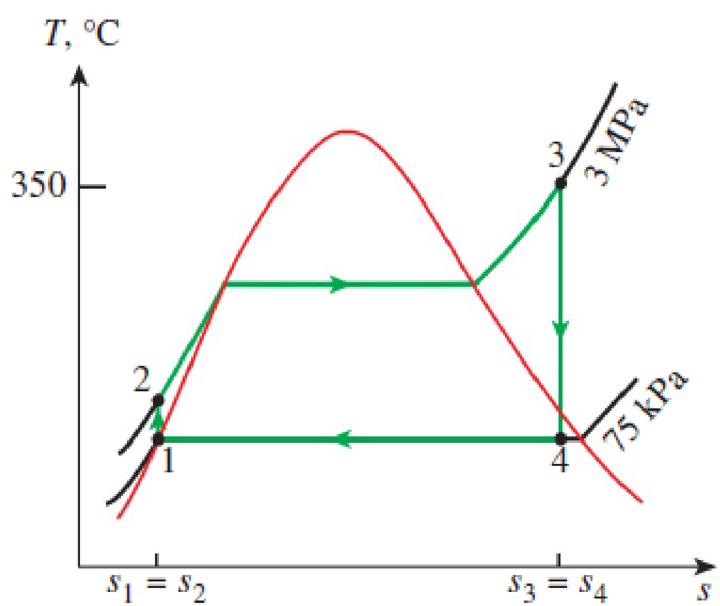
State 2: $P_2 = 3 \text{ MPa}$

$s_2 = s_1$

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

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

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



Ref. Cengel and Boles, 8th Edition (2015)

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

State 4:
$$P_4 = 75\text{ kPa} \quad \text{(sat. mixture)}$$

$$s_4 = s_3$$

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

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

Thus,

$$q_{in} = h_3 - h_2 = (3116.1 - 387.47)\text{ kJ/kg} = 2728.6\text{ kJ/kg}$$
$$q_{out} = h_4 - h_1 = (2403.0 - 384.44)\text{ kJ/kg} = 2018.6\text{ kJ/kg}$$

and

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

Ref. Cengel and Boles, 8th Edition (2015)

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).

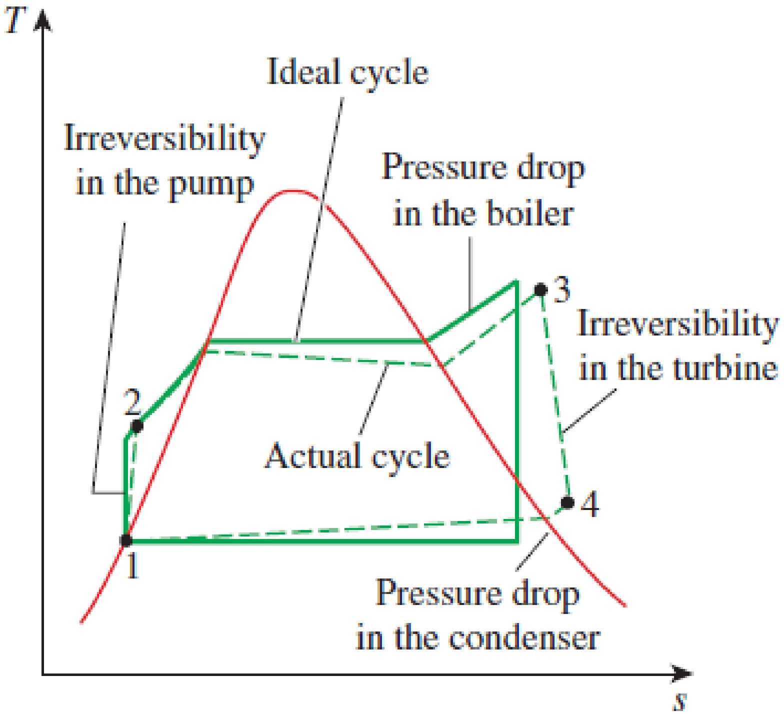
It is also interesting to note the thermal efficiency of a Carnot cycle operating between the same temperature limits

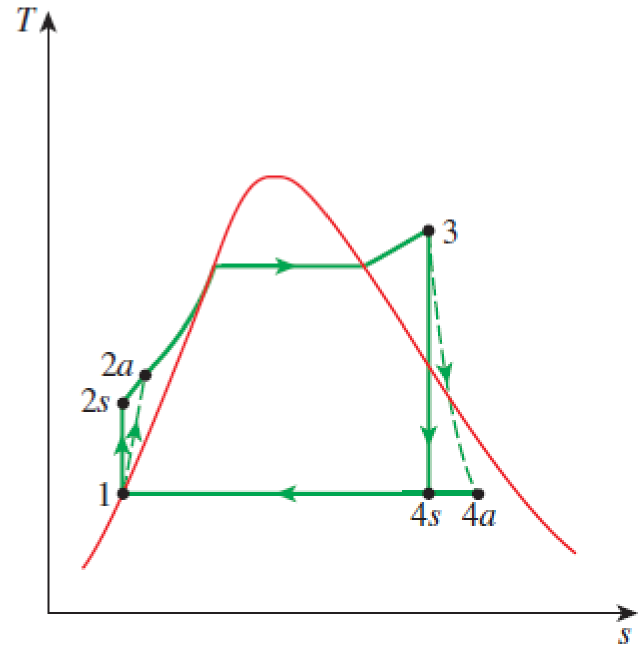
$$\eta_{\text{th,Carnot}} = 1 - \frac{T_{\text{min}}}{T_{\text{max}}} = 1 - \frac{(91.76 + 273)\text{ K}}{(350 + 273)\text{ K}} = 0.415$$

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.

Ref. Cengel and Boles, 8th Edition (2015)

DEVIATION OF ACTUAL VAPOR POWER CYCLES FROM IDEALIZED ONES





Ref. Cengel and Boles, 8th Edition (2015)

<u>Main factors</u> :

Fluid friction in boiler, condenser, and pipes connecting various parts

Heat loss from steam to surroundings

Irreversibilities in pump and turbine

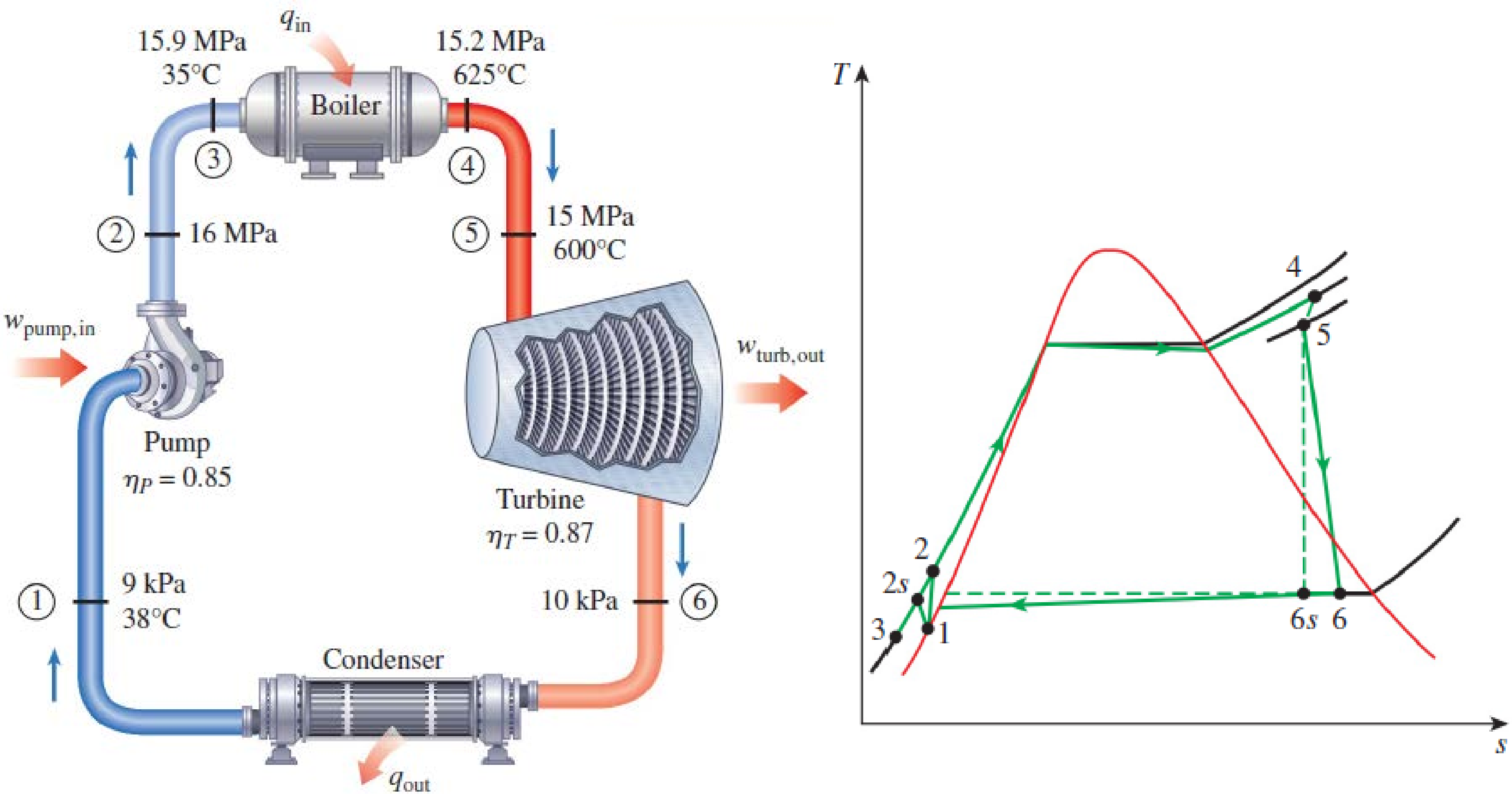
<u>Other additional factors</u> :

Subcooling of liquid in condensers

Losses at the bearings between the moving parts as a result of friction

Losses due to leakage of steam to surroundings and air into the condenser

A steam power plant operates on the cycle shown in Fig. 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.



Ref. Cengel and Boles, 8th Edition (2015)

(*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,

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

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

Ref. Cengel and Boles, 8th Edition (2015)

(b) The power produced by this power plant is

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

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

Ref. Cengel and Boles, 8th Edition (2015)