POWER PLANT ENGINEERING

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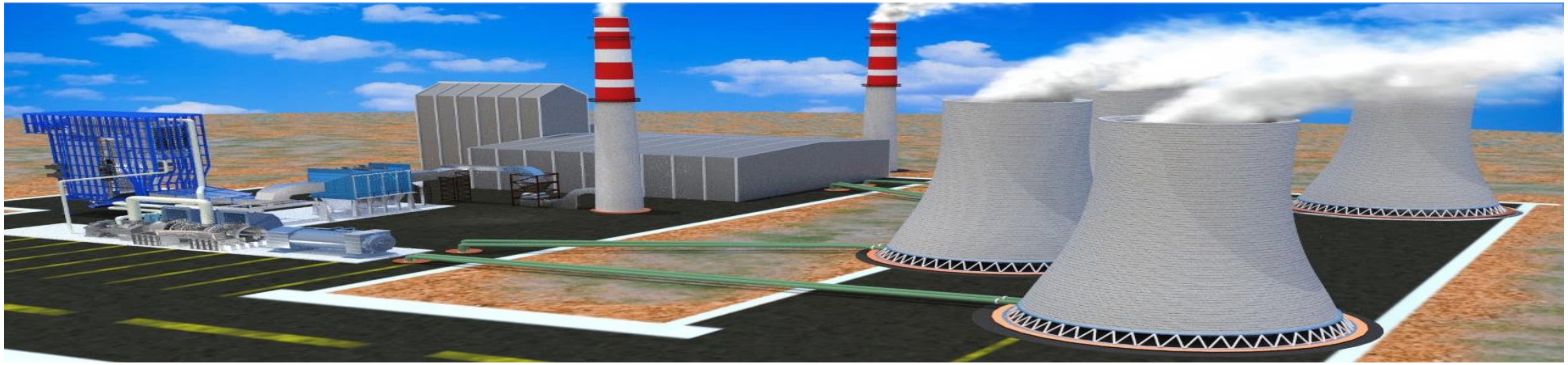


3. Steam Power Plants: Cycles and Materials

The aim of the lecture is to introduce the basic thermodynamic steam cycles employed in modern steam power plants.

3. Steam Power Plants: Cycles and Materials

3.1 Introduction

- They are the main sources of electricity and heat
- Coal is the main source of fuel, though oil and gas can also be used
- Their efficiency is low, determined by thermodynamic limits
- They are responsible for most greenhouse gas emission

Schwarze Pumpe PP, 2 x 813 MW_{th}

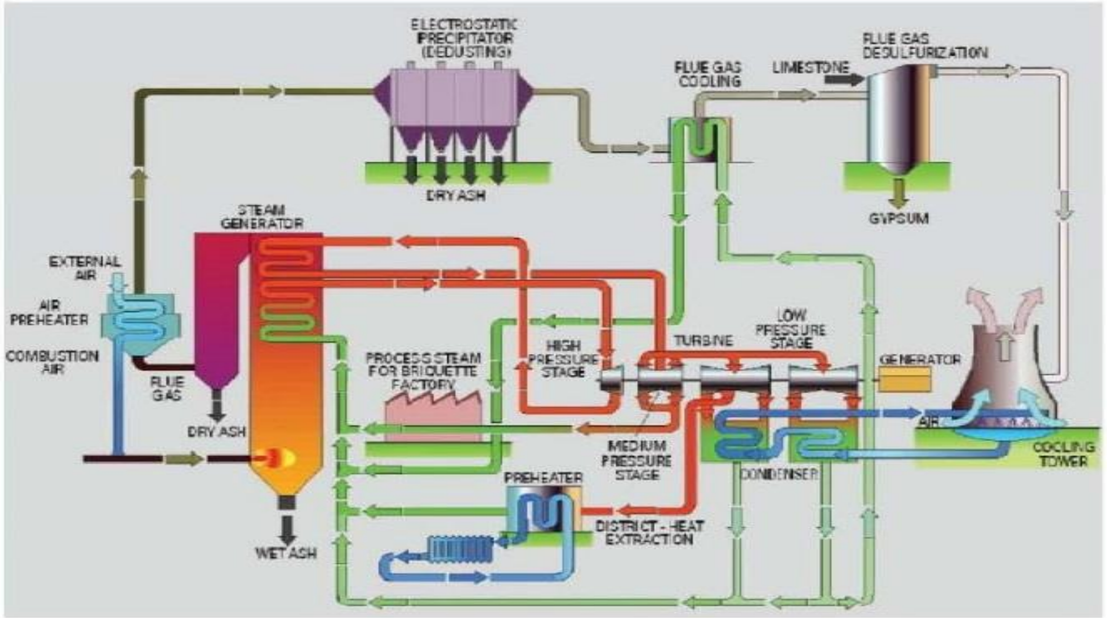
Jänschwalde PP, 6 x 500 MW_{th}

3. Steam Power Plants: Cycles and Materials

Steam turbine power plant cycle and components

Main components

- furnace or combustion chamber
- steam generator (economizer, boiler, superheater)
- power generating unit (turbine, generator)
- feedwater and steam circuit (feed pumps, feedwater heaters)
- air and gas circuit (air preheater, flue gas cleaning, chimeny)
- cooling water circuit (condenser, pumps, cooling tower)
- coal and ash circuit
- pipings

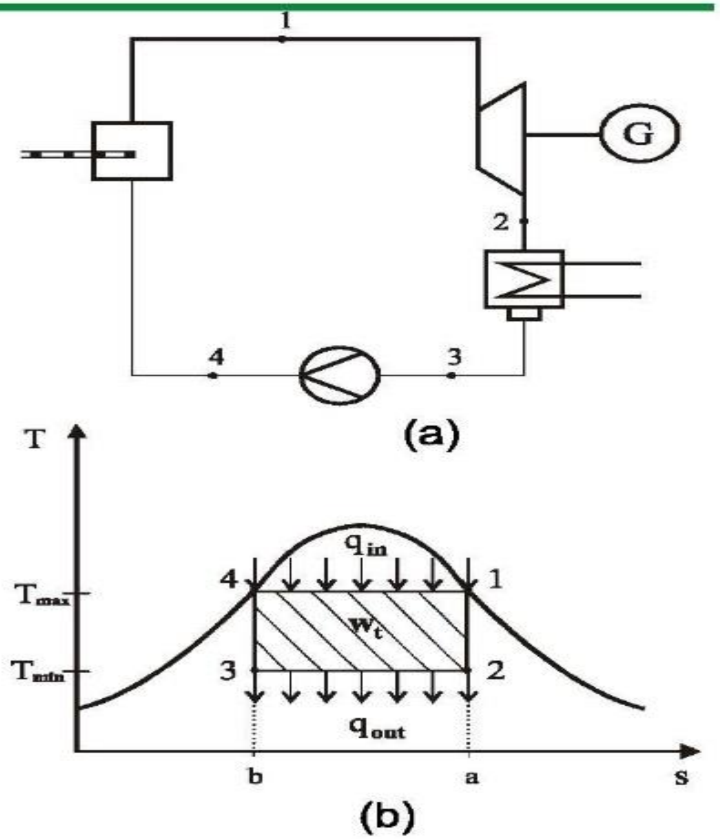


1. How does a thermal power plant work? 2.Alabama Coal Fired Power Plant Video

3.2 The Carnot Cycle

Assumptions:

- No mechanical irreversibilities occurred, i.e. motion is frictionless and no internal flow losses
- No net increase in entropy of the universe occurred
- The process could be reversed and returned everything to the original state



The Carnot cycle (a) the heat cycle diagram (b) the T-s diagram

1 → 2 Isentropic expansion in the turbine

2 → 3 Isothermal condensation in the condenser

3 → 4 Isentropic compression in a working machine

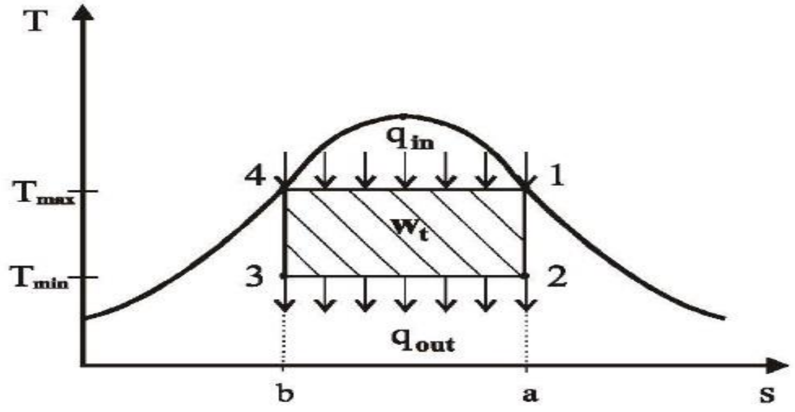
4 → 1 Isothermal heat addition in the steam generator

Efficiency of the Carnot Cycle

$$\eta_{th,C} = \frac{net\,work\,output}{total\,energy\,input} = \frac{w_{net}}{q_{in}} = \frac{q_{in} - q_{out}}{q_{in}}$$

$$\eta_{th,C} = \frac{T_{\max} \cdot (s_1 - s_4) - T_{\min} \cdot (s_2 - s_3)}{T_{\max} \cdot (s_1 - s_4)}$$

$$\eta_{th,C} = \frac{T_{max} - T_{min}}{T_{max}} = 1 - \frac{T_{min}}{T_{max}}$$

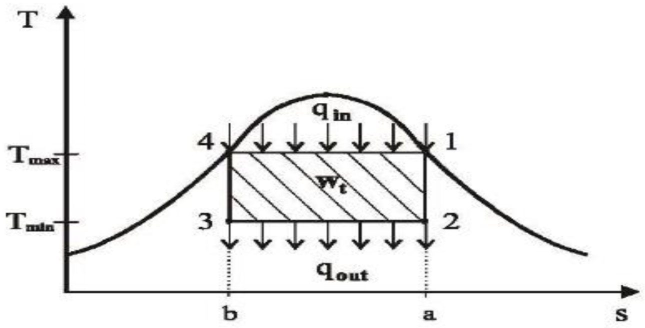


Remark:

- The Carnot cycle is the most efficient thermodynamic power cycle
- The efficiency of the Carnot cycle doesn't depend on the working substance

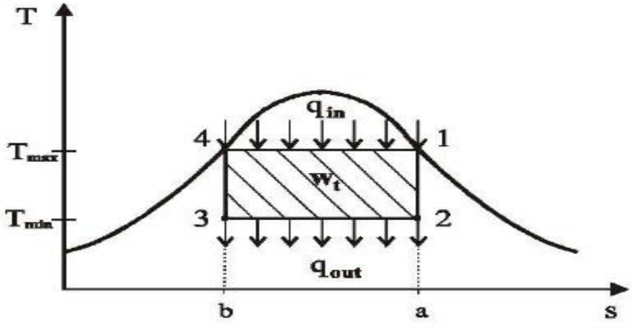
Limitations and impracticalities of the Carnot cycle

1. Isothermal heat transfer to or from a two-phase system is not difficult to achieve in practice
   - Hence Process 4-1 and 2-3 can be approached closely in actual boilers and condensers.
   - Limiting the heat transfer processes to two-phase systems, however, severely limits the maximum temperature that can be used in the cycle (it has to remain under the critical-point value, which is 374°C for water).
   - Limiting the maximum temperature in the cycle also limits the thermal efficiency.
   - Any attempt to raise the maximum temperature in the cycle involves heat transfer to the working fluid in a single phase, which is not easy to accomplish isothermally.



3. Steam Power Plants: Cycles and Materials

2. After isentropic expansion process (1-2), the turbine has to handle steam with low quality, that is, steam with a high moisture content.
   - The impingement of liquid droplets on the turbine blades causes erosion and is a major source of wear.
3. The isentropic compression process (process 4-1) involves the compression of a liquid–vapor mixture to a saturated liquid. There are three difficulties associated with this process.
   - First, it is not easy to control the condensation process so precisely as to end up with the desired quality at state 4.
   - Second, a compressor of size and cost comparable with the turbine is needed
   - Third, it is not practical to design a pump/compressor that handles two phases.

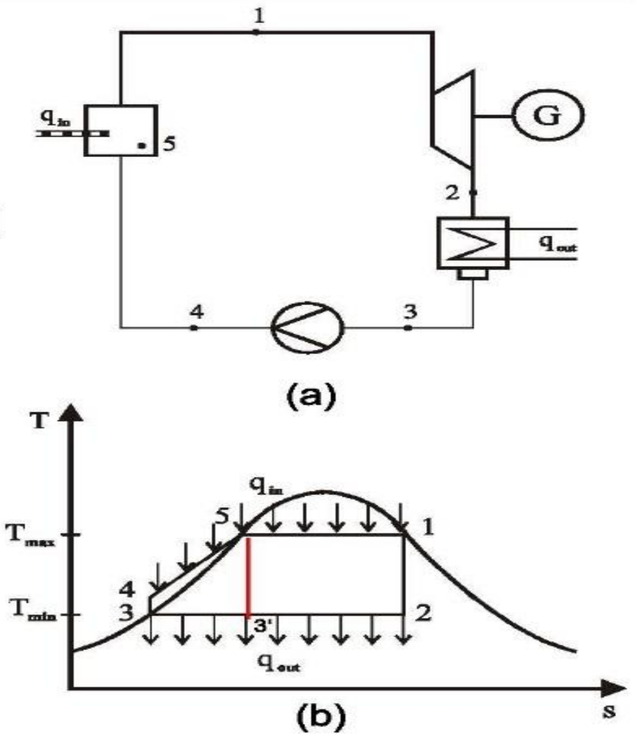


3.3 The Clausius-Rankine Saturated Steam Cycle

- Many of the impracticalities associated with the Carnot cycle can be eliminated by superheating the steam in the boiler and condensing it completely in the condenser.

- The cycle that resulted is the **Rankine Cycle.**

  Here the condensation is performed until state 3 (saturated liquid state), thus requiring less energy input to compress to higher pressure (state 4)



The Rankine saturated steam cycle (a) the heat cycle diagram (b) the T-s diagram

The ideal saturated steam Rankine cycle consists:

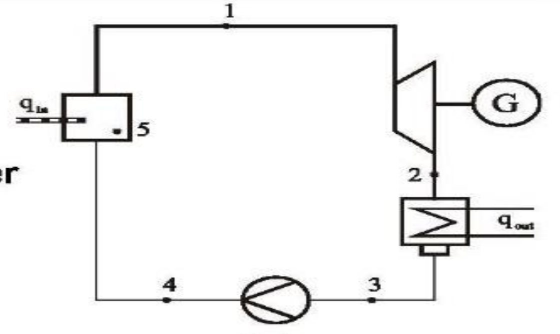
1 → 2 Isentropic expansion, i.e. adiabatic reversible in the turbine

2 → 3 Constant pressure (Isothermal) condensation in the condenser

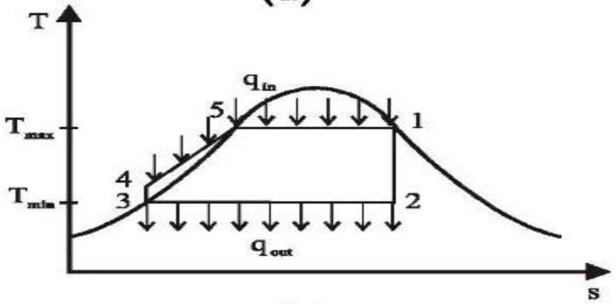
3 → 4 Isentropic compression in the pump

4 → 5 Constant pressure sensible heat addition (Economizer)

5 → 1 Constant pressure evaporation(latent heat addition) in the boiler



(a)



(b)

The Rankine saturated steam cycle (a) the heat cycle diagram (b) the T-s diagram

Efficiency of the Rankine Cycle

$$\eta_{th} = \frac{net\ work\ output}{total\ energy\ input} = \frac{w_{net}}{q_{in}} = \frac{q_{in} - q_{out}}{q_{in}}$$

$$\eta_{th,R.} = \frac{(h_1 - h_4) - (h_2 - h_3)}{(h_1 - h_4)}$$

$$w_{net} = w_t - w_p = (h_1 - h_2) - (h_4 - h_3)$$

$$= h_1 - h_2 + h_3 - h_4 = (h_1 - h_4) - (h_2 - h_3) = q_{in} - q_{out}$$

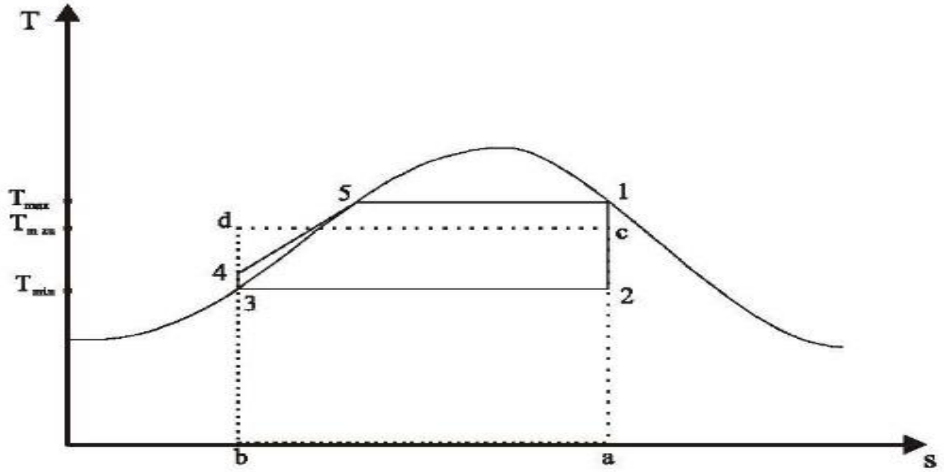
The ***mean temperature*** T_m at which heat is supplied to the working substance is

$$T_{m,in} = \frac{h_1 - h_4}{s_c - s_d} = \frac{h_1 - h_4}{s_1 - s_4}$$

The thermal efficincy of the Rankine cycle may be approximated

$$\eta_{th,R.} = \frac{T_{m,in} - T_{\min}}{T_{m,in}} = 1 - \frac{T_{\min}}{T_{m,in}}$$

Therefore, the basic idea to increase the thermal efficiency of the Rankine cycle is to increases the mean temperature at which heat is supplied to the working fluid in the boiler or to decreases the temperature at which heat is rejected from the working fluid in the condenser.



3.4 The Clausius-Rankine Superheated Steam Cycle

The cycle consists of the saturated cycle plus:

$6 \rightarrow 1$ Supply of sensible heat in the superheater (Point 1 = superheated steam)

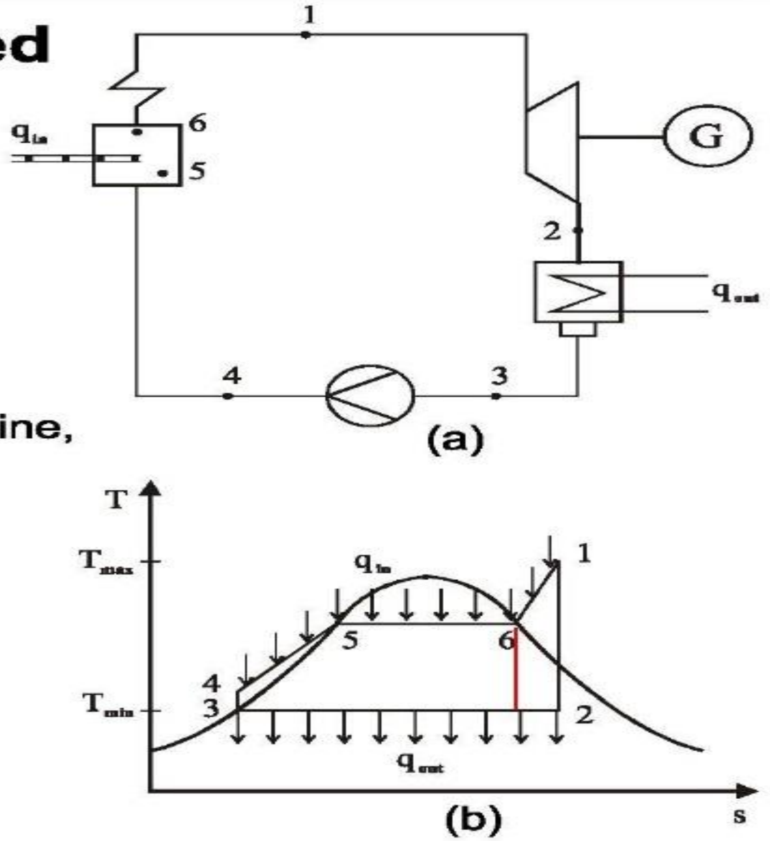
Advantages:

- increase the dryness fraction of the steam in the turbine, hence reduce turbine blade erosion and corrosion
- increase the operating temparutre, thus increase the thermal efficiency

Thermal efficiency:

$$\eta_{th} = \frac{net\ work\ output}{total\ energy\ input} = \frac{w_{net}}{q_{in}} = \frac{q_{in} - q_{out}}{q_{in}}$$

$$\eta_{th} = \frac{(h_1 - h_4) - (h_2 - h_3)}{(h_1 - h_4)}$$



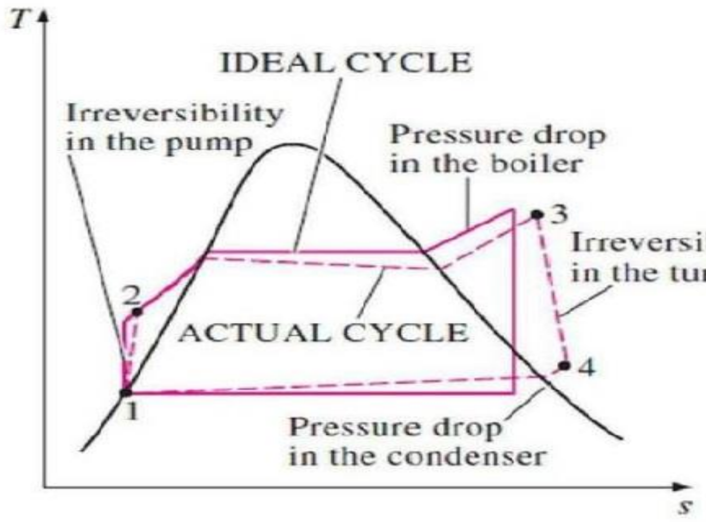
The Rankine superheated steam cycle (a) the heat cycle diagram (b) the T-s diagram

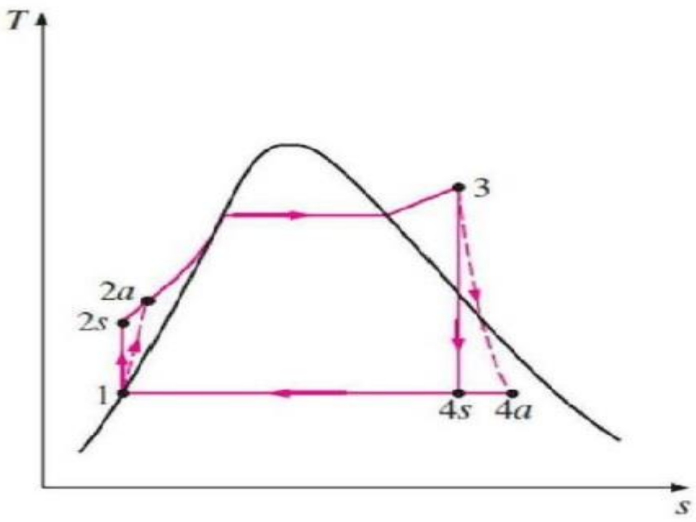
3.5 The actual (irreversible) Rankine cycle

- The actual vapor power cycle differs from the ideal Rankine cycle, as a result of inherent irreversibilities in various components.
- Fluid friction and heat loss to the surroundings are the two common sources of irreversibilities.
- *Fluid friction* causes pressure drops in the boiler, the condenser, and the piping between various components. As a result:
  - steam leaves the boiler at a somewhat lower pressure.
  - the pressure at the turbine inlet is somewhat lower than that at the boiler exit
  - To compensate for these pressure drops, the water must be pumped to a sufficiently higher pressure than the ideal cycle.
  - This requires a larger pump and larger work input to the pump.

3. Steam Power Plants: Cycles and Materials

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

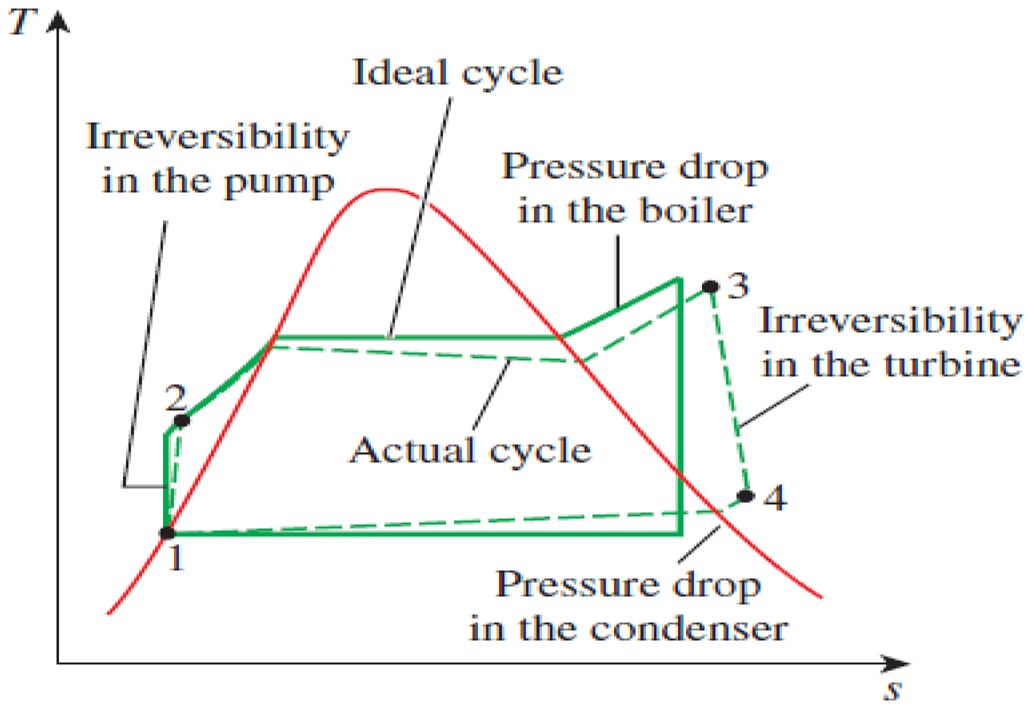


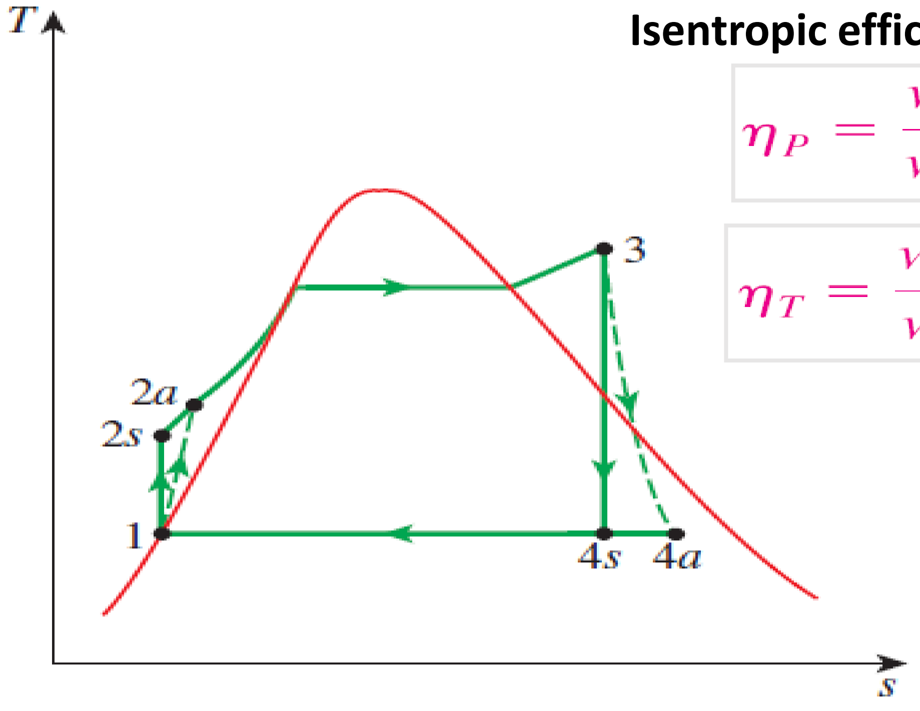


DEVIATION OF ACTUAL VAPOR POWER CYCLES FROM IDEALIZED ONES

The actual vapor power cycle differs from the ideal Rankine cycle as a result of irreversibilities in various components.

Fluid friction and heat loss to the surroundings are the two common sources of irreversibilities.





Isentropic efficiencies

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

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

(*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.

3. Steam Power Plants: Cycles and Materials

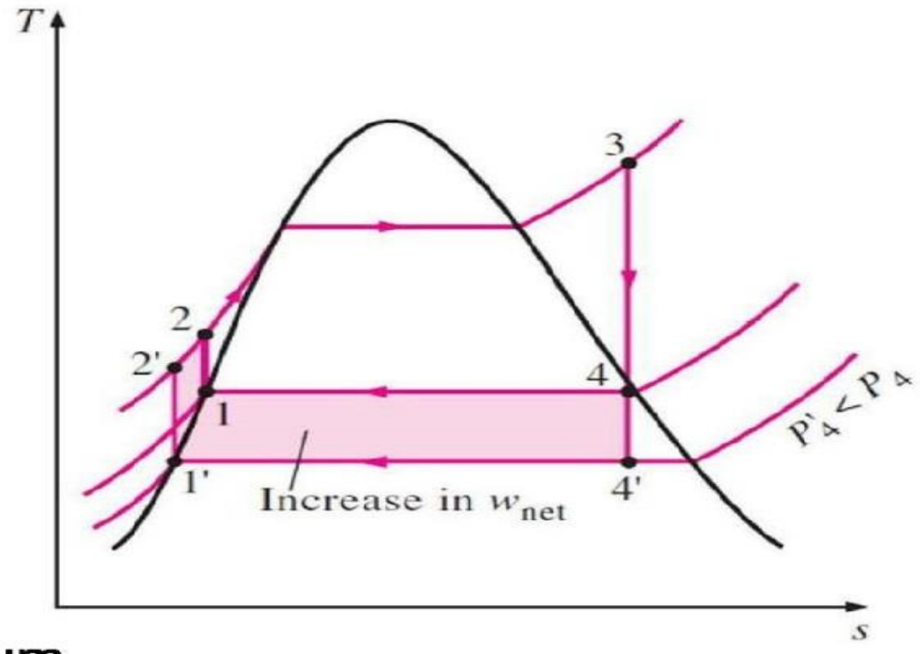
Other factors also need to be considered in the analysis of actual vapor power cycles include:

- 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.
- 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.

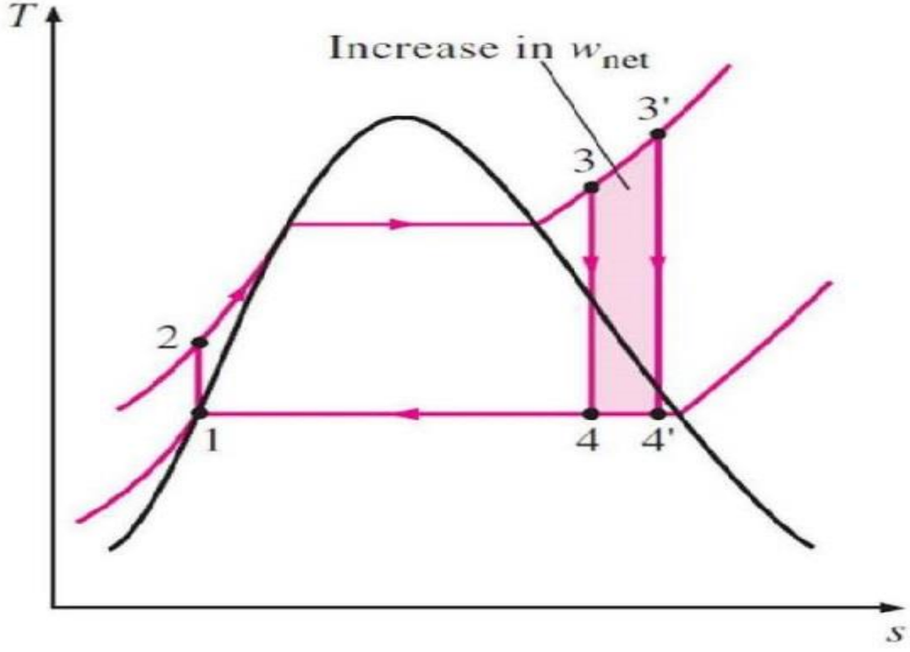
3.6 Increasing the Efficiency of the Rankine Cycle

1. Lowering the Condenser Pressure (*Lowers* $T_{m,out}$)

- **Area 1-4-4'-1'** is the increase in work output
- **Area 1-1'-2'-2** is the increase in heat input
- But Area 1-1'-2'-2 is much smaller than area 1-4-4'-1'
- Thus the net effect is Increase in $\mathbf{W_{net}}$ hence the **efficiency**
- The lower limit of the condenser pressure is set by the saturation pressure of the corresponding to the temperature of the cooling medium.



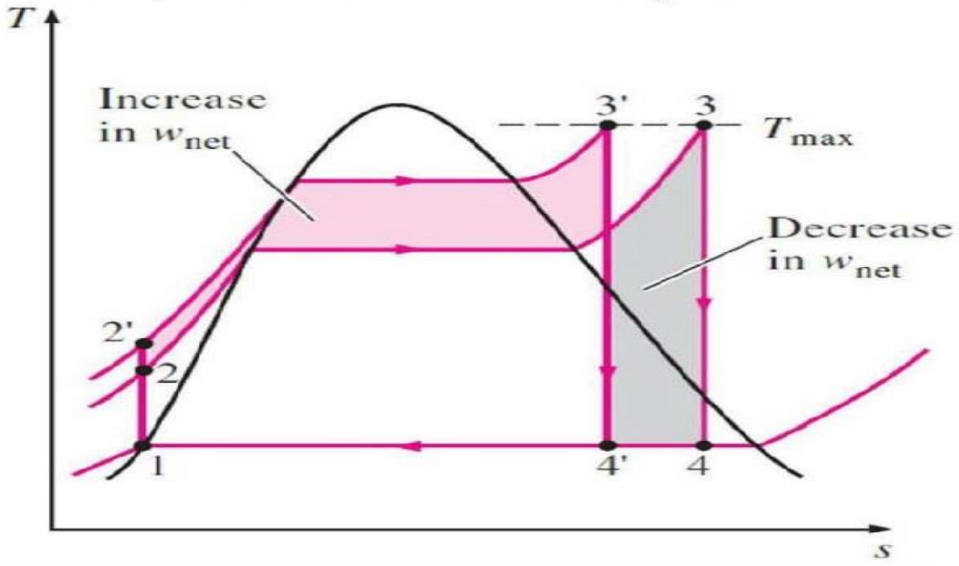
2. Superheating the Steam to High Temperatures (*Increases $T_{m,in}$*)



- In this case both the work output and net work increased
- Thus, the efficiency increased
- Another very desirable effect: It decreases the moisture content of the steam at the turbine exit
- The temperature to which steam can be superheated is limited, however, by metallurgical considerations.
- Presently the highest steam temperature allowed at the turbine inlet is about 620°C

3. Increasing the Boiler Pressure (*Increases $T_{m,in}$*)

- Increase the operating pressure of the boiler raises the temperature at which boiling takes place. This, in turn, raises the mean temperature at which heat is transferred to the steam and thus raises the thermal efficiency of the cycle.
- But, tends to result in an increased turbine exhaust wetness fraction resulting In lower turbine efficiency and more erosion problems
- This undesirable effect can be Corrected by reheating the steam



Note: Related Problems are Numerical Problem 1-4 in problem sheet.

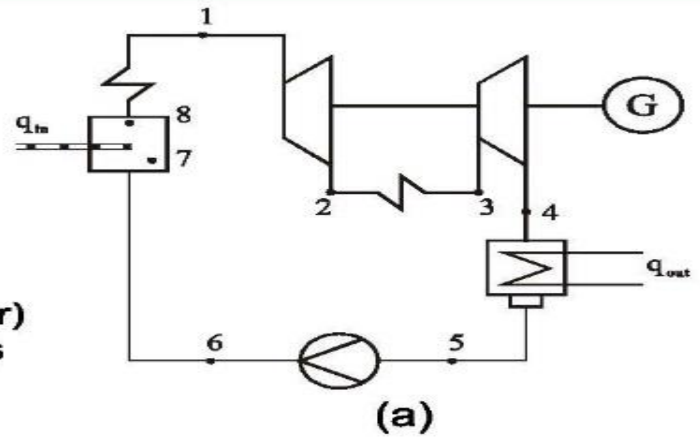
3.7 The Ideal Clausius-Rankine Cycle with Reheater

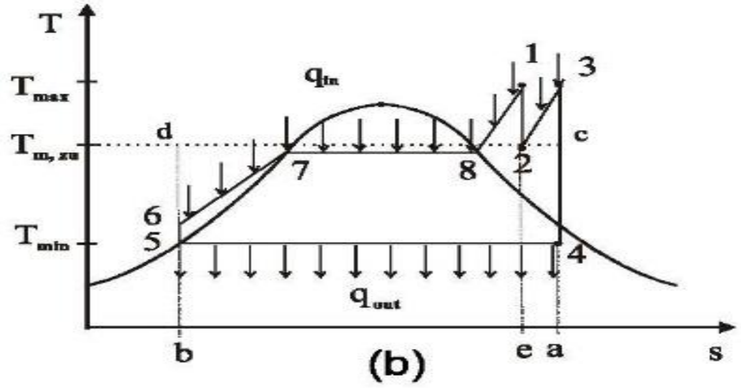
The cycle consists of the following processes:

$1 \rightarrow 2$	Isentropic expansion in the high pressure turbine
$2 \rightarrow 3$	Constant pressure Reheating in the steam generator (reheater)
$3 \rightarrow 4$	Isentropic expansion in the medium and low pressure turbines
$4 \rightarrow 5$	Constant pressure heat rejection
$5 \rightarrow 6$	Isentropic compression
$6 \rightarrow 7$	Constant pressure heat addition in the economizer
$7 \rightarrow 8$	Constant pressure evaporation in the boiler
$8 \rightarrow 1$	Constant pressure superheating in the super heater

Advantages:

- The mean temperature of heat addition is increased, thus increase thermal efficiecy
- Increases dryness fraction of steam at exhaust so that blade erosion due to impact of water particles is reduced
- Increases the work output per kg of steam and this results in reduced boiler size





The reheat Rankine cycle (a) the heat cycle diagram (b) the T-s diagram

3. Steam Power Plants: Cycles and Materials

Disadvantages:

- additional cost due to the reheater and its long connections
- increases condenser capacity due to increased dryness fraction

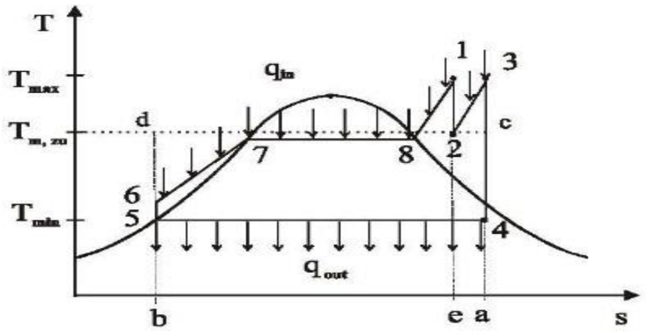
Thermal efficiency:

$$\eta_{th} = \frac{net\ work\ output}{total\ energy\ input} = \frac{w_{net}}{q_{in}} = \frac{q_{in} - q_{out}}{q_{in}}$$

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

$$w_{net} = w_t - w_p = (h_1 - h_2) + (h_3 - h_4) - (h_6 - h_5)$$

$$= h_1 - h_2 + h_3 - h_4 + h_5 - h_6$$



Note: The choice of the intermediate pressure for reheating is very important as reheating may reduce the overall net efficiency of the cycle. Thus there exist an intermediate pressure which gives the maximum possible net efficiency for a given boiler and condenser pressures.

The single reheat in a modern power plant improves the cycle efficiency by 4 to 5% by increasing the average temperature at which heat is transferred to the steam.

The average temperature during the reheat process can be increased by increasing the number of expansion and reheat stages. As the number of stages is increased, the expansion and reheat processes approach an isothermal process at the maximum temperature. The use of more than two reheat stages is not practical. The theoretical improvement in efficiency from the second reheat is about half of that which results from a single reheat.

The reheat temperatures are very close or equal to the turbine inlet temperature.

The optimum reheat pressure is about one-fourth of the maximum cycle pressure.

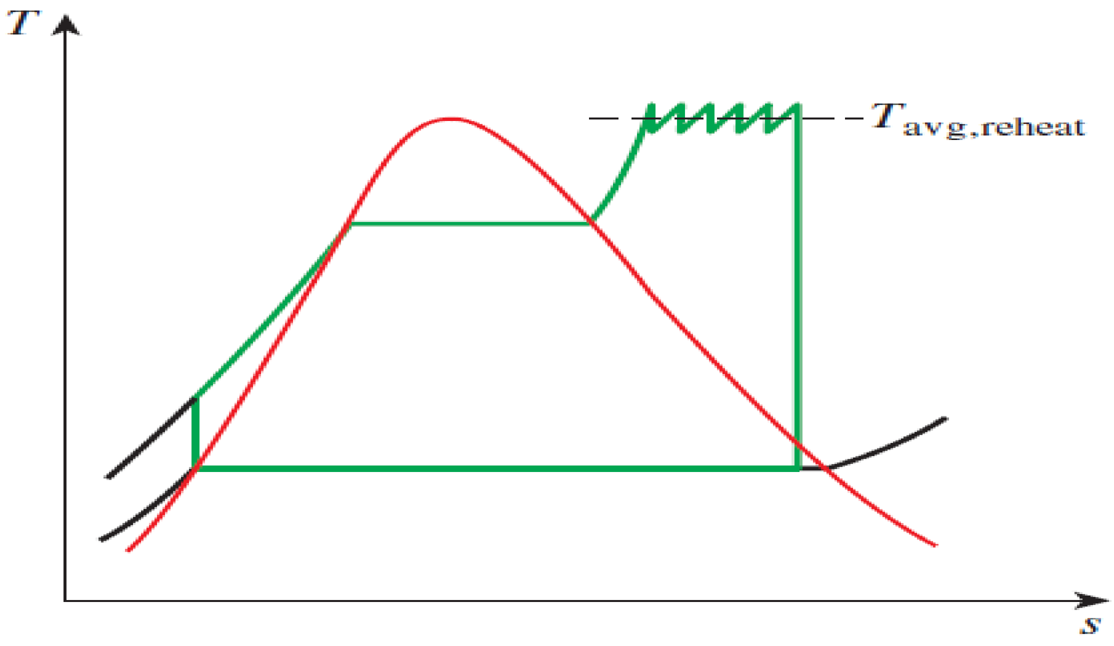


FIGURE 10–12
The average temperature at which heat is transferred during reheating increases as the number of reheat stages is increased.

Note: Related Problems are Numerical Problems 5-8 in problem sheet.

Specific Steam Consumption (SSC)

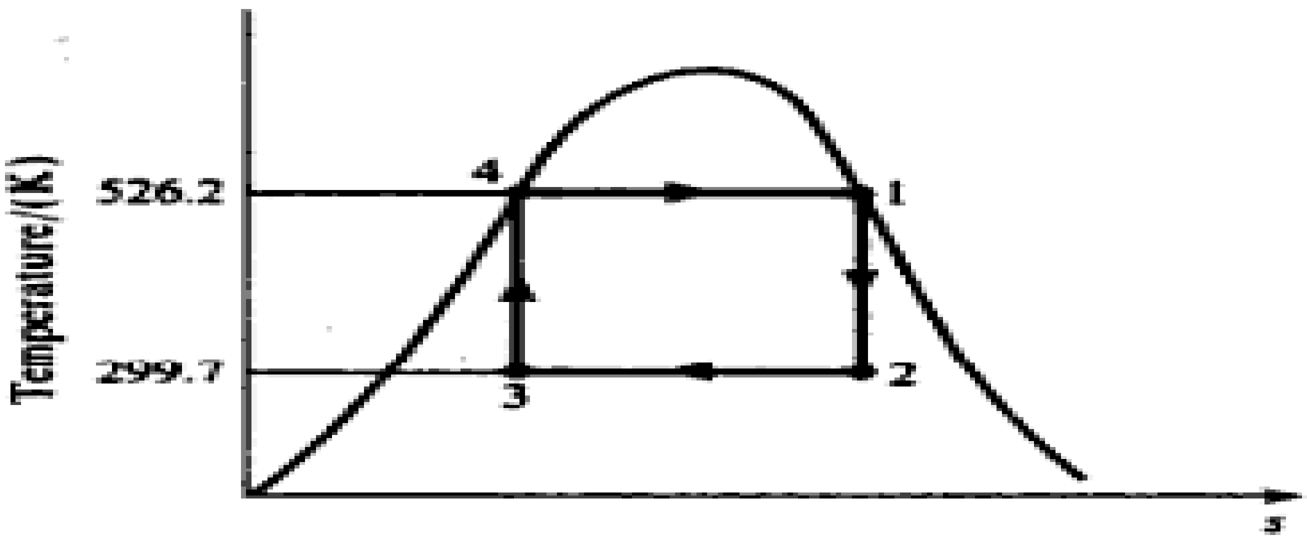
- SSC is the steam flow required to develop unit power output.
- SSC = 1/(net work output of Steam Power Plant)
- If feed water pump work is neglected then

$SSC = 1/(h_1-h_2)$

Class Problem 1

A steam power plant operates between a boiler pressure of 42 bar and a condenser pressure of 0.035 bar. Calculate for these limits the cycle efficiency, the work ratio, and the specific steam consumption:

(i) for a Carnot cycle using wet steam;
(ii) for a Rankine cycle with dry saturated steam at entry to the turbine;
(iii) for the Rankine cycle of (ii), when the expansion process has an isentropic efficiency of 80%.



T_1 = saturation temperature at 42 bar

$= 253.2 + 273 = 526.2$ K

T_2 = saturation temperature at 0.035 bar

$= 26.7 + 273 = 299.7$ K

Then from equation (5.1)

$$\eta_{Carnot} = \frac{T_1 - T_2}{T_1} = \frac{526.2 - 299.7}{526.2} = 0.432 \quad \text{or} \quad 43.2\%$$

Also Heat supplied $= h_1 - h_4 = h_{fg}$ at 42 bar $= 1698$ kJ/kg

Then $\eta_{Carnot} = \dfrac{\text{Net work output, } -\sum W}{\text{Gross heat supplied}} = 0.432$

Therefore $-\sum W = 0.432 \times 1698$,

i.e. Net work output, $-\sum W = 734$ kJ/kg

To find the gross work of the expansion process it is necessary to calculate h_2, using the fact that $s_1 = s_2$.

From tables

$$h_1 = 2800 \text{ kJ/kg} \quad \text{and} \quad s_1 = s_2 = 6.049 \text{ kJ/kg K}$$

Using equation (4.10)

$$s_2 = 6.049 = s_{f_2} + x_2 s_{fg_2} = 0.391 + x_2 8.13$$

therefore

$$x_2 = \frac{6.049 - 0.391}{8.13} = 0.696$$

Then using equation (2.2)

$$h_2 = h_{f_2} + x_2 h_{fg_2} = 112 + (0.696 \times 2438) = 1808 \text{ kJ/kg}$$

Hence, from equation (8.2)

$$-W_{12} = h_1 - h_2 = 2800 - 1808 = 992 \text{ kJ/kg}$$

Therefore we have, using equation (8.13),

$$\text{Work ratio} = \frac{\text{net work output}}{\text{gross work output}} = \frac{734}{992} = 0.739$$

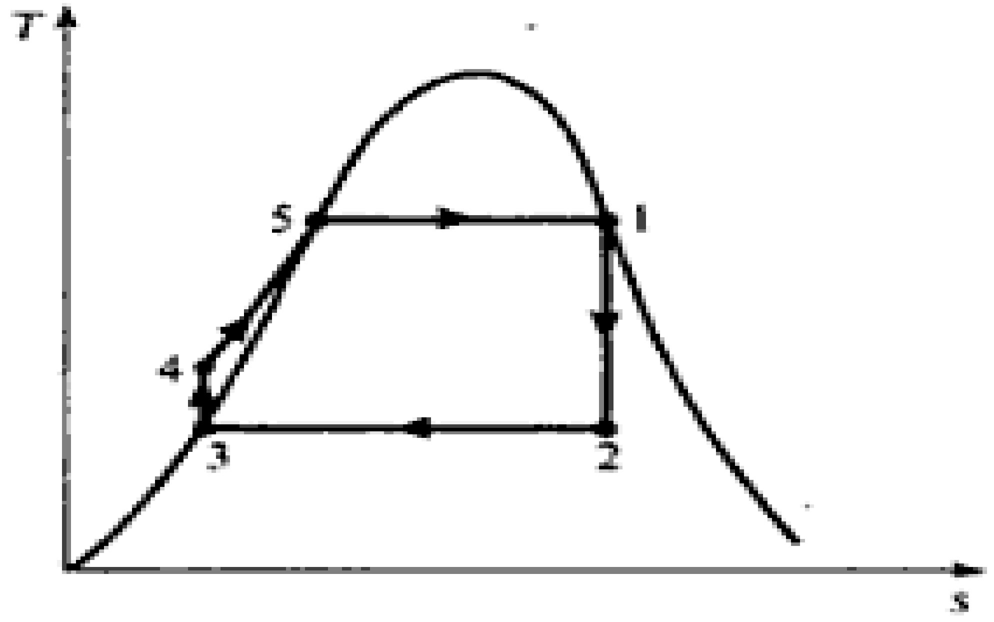
Using equation (8.14)

$$\text{ssc} = \frac{1}{734}$$

i.e. $\text{ssc} = 0.001\,36 \text{ kg/kW s}$

$$= 4.91 \text{ kg/kW h}$$

(ii) The Rankine cycle is shown in Fig.



As in part (i)

$$h_1 = 2800 \text{ kJ/kg} \quad \text{and} \quad h_2 = 1808 \text{ kJ/kg}$$

Also, $h_3 = h_f$ at 0.035 bar $= 112$ kJ/kg

Using equation (8.10), with $v = v_f$ at 0.035 bar

$$\text{Pump work} = v_f(p_4 - p_3) = 0.001 \times (42 - 0.035) \times \frac{10^5}{10^3}$$

$$= 4.2 \text{ kJ/kg}$$

Using equation (8.2)

$$-W_{12} = h_1 - h_2 = 2800 - 1808 = 992 \text{ kJ/kg}$$

Then using equation (8.8)

$$\eta_R = \frac{(h_1 - h_2) - (h_4 - h_3)}{(h_1 - h_3) - (h_4 - h_3)} = \frac{992 - 4.2}{(2800 - 112) - 4.2} = 0.368$$

i.e. $\eta_R = 36.8\%$

Using equation (8.13)

$$\text{Work ratio} = \frac{\text{net work output}}{\text{gross work output}} = \frac{992 - 4.2}{992} = 0.996$$

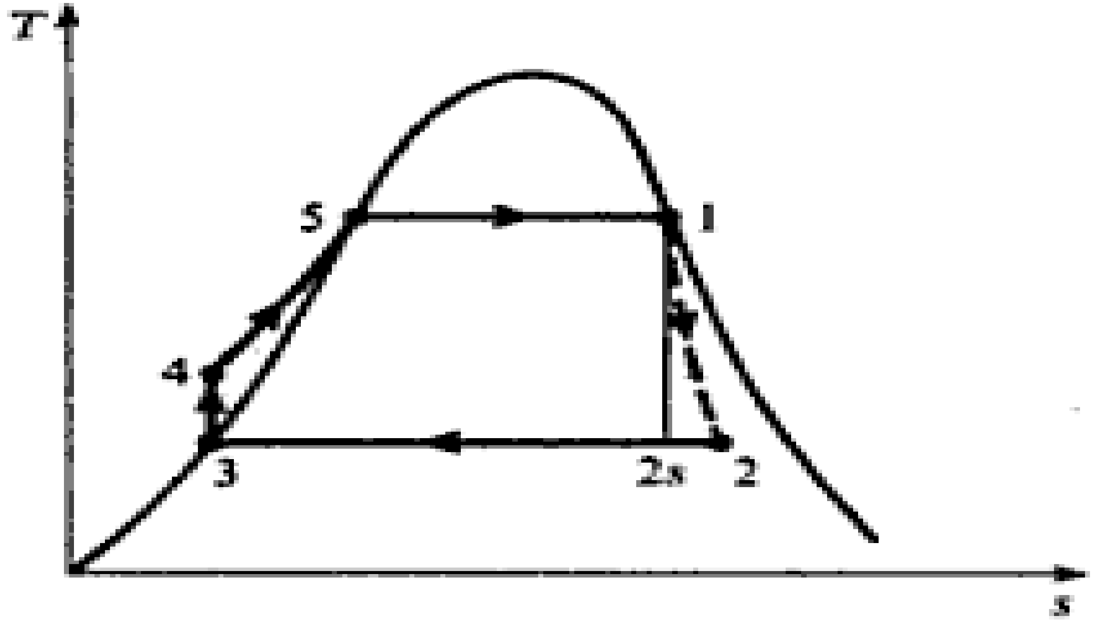
Using equation (8.14)

$$\text{ssc} = \frac{1}{-\sum W}$$

i.e. $\text{ssc} = \dfrac{1}{(992 - 4.2)} = 0.001\,01 \text{ kg/kW s} = 3.64 \text{ kg/kW h}$

(iii) The cycle with an irreversible expansion process is shown in Fig. Using equation (8.12)

$$\text{Isentropic efficiency} = \frac{h_1 - h_2}{h_1 - h_{2s}} = \frac{-W_{12}}{-W_{12s}}$$



therefore

$$0.8 = \frac{-W_{12}}{992}$$

i.e. $\quad -W_{12} = 0.8 \times 992 = 793.6\ \mathrm{kJ/kg}$

Then the cycle efficiency is given by

$$\text{Cycle efficiency} = \frac{(h_1 - h_2) - (h_4 - h_3)}{\text{gross heat supplied}}$$

$$= \frac{793.6 - 4.2}{(2800 - 112) - 4.2} = 0.294$$

i.e. $\quad$ Cycle efficiency = 29.4%

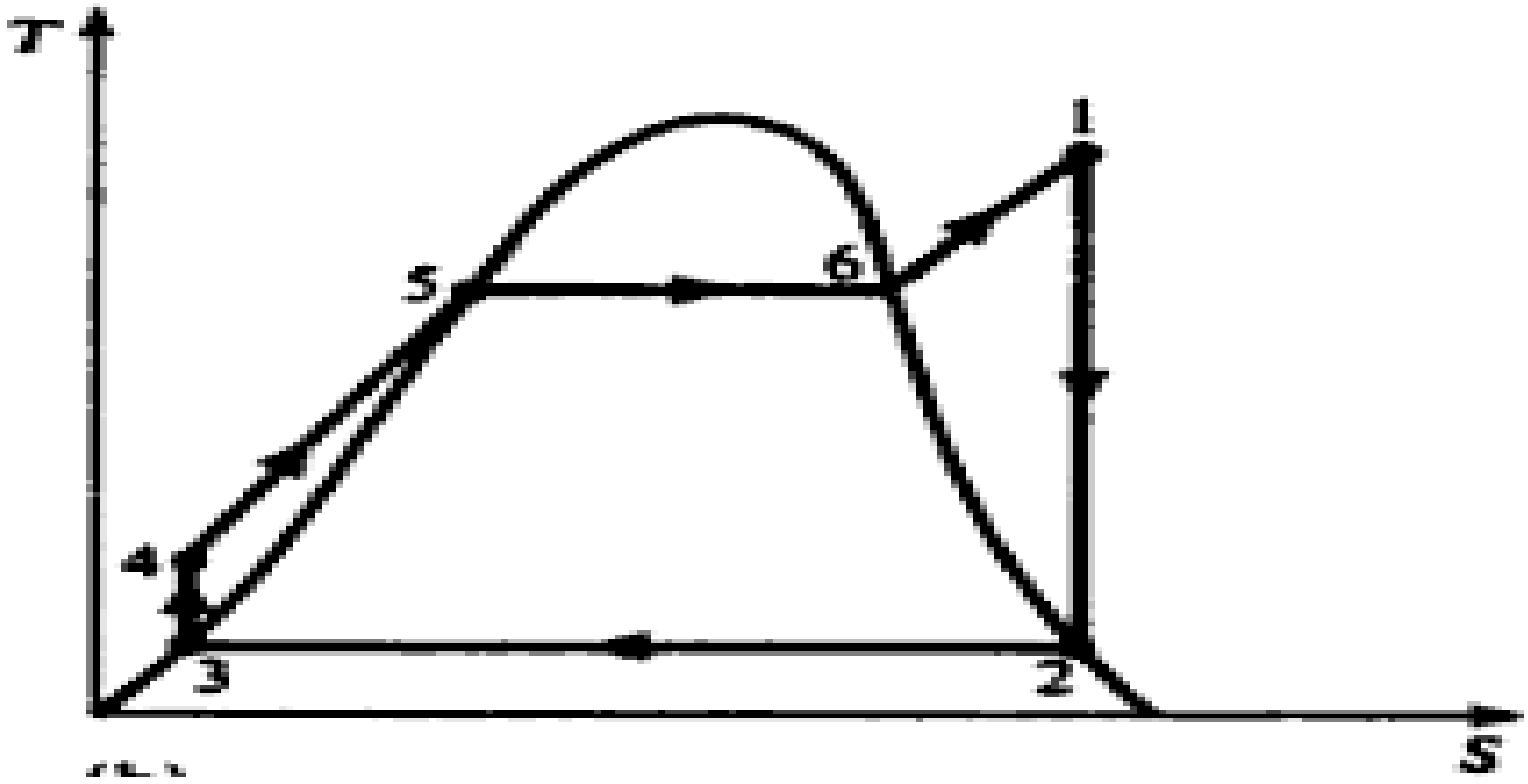
$$\text{Work ratio} = \frac{-W_{12} - \text{pump work}}{-W_{12}} = \frac{793.6 - 4.2}{793.6} = 0.995$$

Also

$$\text{ssc} = \frac{1}{793.6 - 4.2} = 0.001\,267\ \mathrm{kg/kW\,s} = 4.56\ \mathrm{kg/kW\,h}$$

Class Problem 2

- Compare the Rankine cycle performance of Class problem 1 with that obtained when the steam is superheated to 500 C. Neglect Feed pump work?



T
1
5
6
4
3
2
S

From tables, by interpolation, at 42 bar:

$$h_1 = 3442.6 \text{ kJ/kg} \quad \text{and} \quad s_1 = s_2 = 7.066 \text{ kJ/kg K}$$

Using equation (4.10)

$$s_2 = s_{f_2} + x_2 s_{fg_2} \quad \text{therefore } 0.391 + x_2 8.13 = 7.066$$

i.e. $\quad x_2 = 0.821$

Using equation (2.2)

$$h_2 = h_{f_2} + x_2 h_{fg_2} = 112 + (0.821 \times 2438) = 2113 \text{ kJ/kg}$$

From tables:

$$h_3 = 112 \text{ kJ/kg}$$

Then, using equation (8.2)

$$-W_{12} = h_1 - h_2 = 3442.6 - 2113 = 1329.6 \text{ kJ/kg}$$

$h_1 = 3445 \text{ KJ/kg}$ at $P_1 = 40 \text{ bar}$

$h_2 = 3493 \text{ KJ/kg}$ at $P_2 = 50 \text{ bar}$

$$\frac{P - P_1}{P_2 - P_1} = \frac{h - h_1}{h_2 - h_1}$$

$$\frac{42 - 40}{50 - 40} = \frac{h - 3445}{3433 - 3445}$$

$h = 3442.6 \text{ KJ/kg}$ at $P = 42 \text{ bar}$

Similarly $S_1 = S_2 = 7.066 \text{ KJ/kgK}$

Neglecting the feed-pump term, we have

$$\text{Heat supplied} = h_1 - h_3 = 3442.6 - 112 = 3330.6\ \text{kJ/kg}$$

Using equation (8.9)

$$\text{Cycle efficiency} = \frac{h_1 - h_2}{h_1 - h_3} = \frac{1329.6}{3330.6} = 0.399 \text{ or } 39.9\%$$

Also, using equation (8.14)

$$\text{ssc} = \frac{1}{h_1 - h_2} = \frac{1}{1329.6} = 0.000\,752\ \text{kg/kW s} = 2.71\ \text{kg/kW h}$$