Applied Thermo Fluids-II: (Autumn 2017)

Section-A: Thermal Power Plants

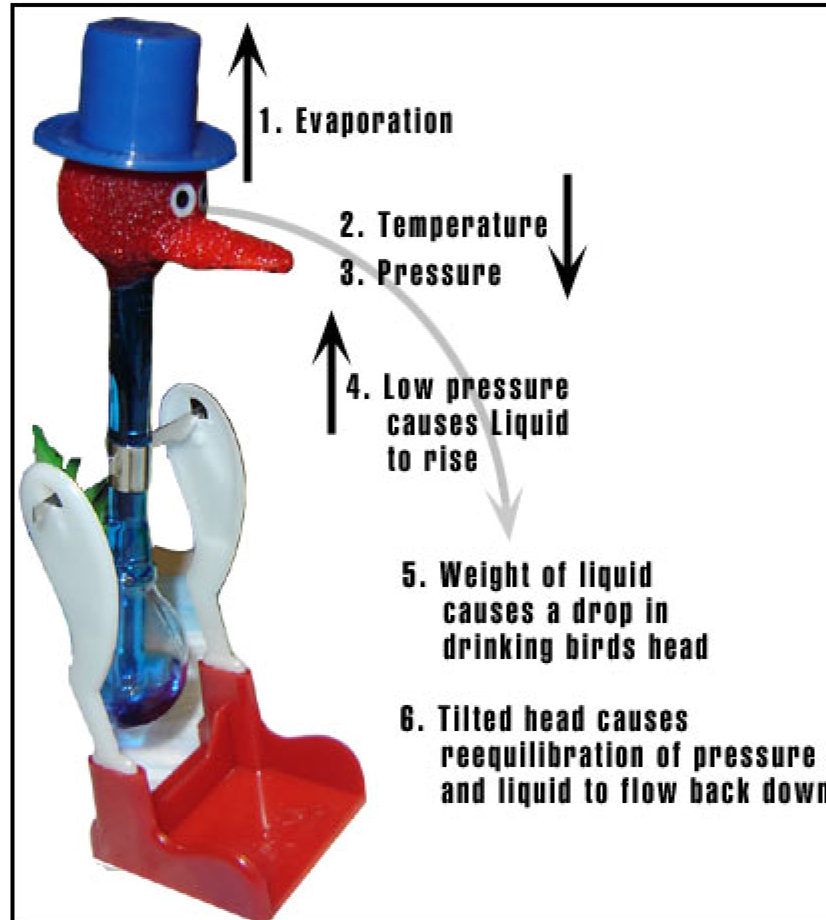
Module-1 (Introduction & Thermodynamics of thermal power plants)

Dr. M. Ramgopal, Mechanical Engineering, IIT Kharagpur

$$\eta_{Carnot} = \left(\frac{DBT - WBT}{DBT}\right) = \left(\frac{42 - 28}{42 + 273}\right) = \mathbf{0.0444} = 4.44\ \%$$

$$\frac{\eta_{actual}}{\eta_{Carnot}} \cong 0.02\ to\ 0.03\ \rightarrow \eta_{actual} = 0.09\ to\ 0.13\ \%!$$

Reference: Lily M Ng & Yvonne S Ng, The thermodynamics of the drinking bird toy, Phys. Edu, 28 (1993)



Course Structure

The course is divided into 5 modules:

Module 1: Introduction & Thermodynamics of thermal power plants

Module 2: **Power Plant Fuels & Combustion calculations**

Module 3: **Power Plant Steam Generators**

Module 4: Power plant prime movers

Module 5: **Power plant cooling systems**

Total Marks: 60

Mode of evaluation: Class test/Quiz/TA: 12 marks

Mid Sem: 18 marks

End Sem: 30 marks

Reference Books: 1) Power Plant Engineering, M. El Wakil, McGraw Hill, 2003

2) Power Plant Engineering, P.K. Nag, Tata McGraw Hill, 2008

Course Objectives

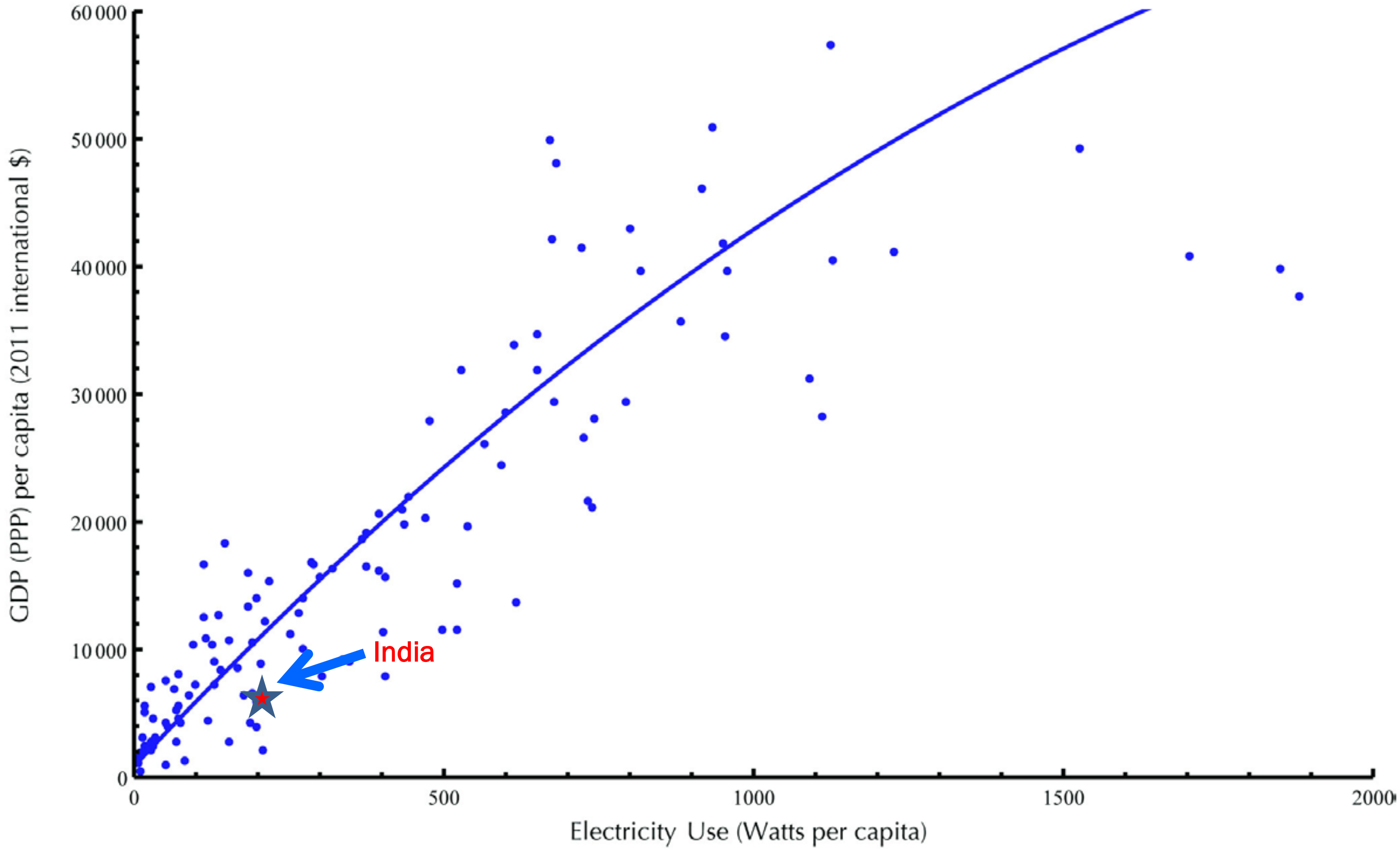
At the end of the course, the student should be able to:

1) Describe various cycles used in thermal power plants (Module 1)
2) Carry out thermodynamic analysis of the above cycles (Module 1)
3) Carry out combustion calculations of power plant fuels (Module 2)
4) Describe steam generators & steam heat exchangers (Module 3)
5) Carry out basic calculations of the above equipment (Module 3)
6) Describe various types of nozzles & steam turbines (Module 4)
7) Analyze steam nozzles and steam turbines (Module 4)
8) Describe heat rejection systems of steam power plants (Module 5)
9) Perform design and rating calculations of the above (Module 5)

Introduction

- Electrical energy is considered to be energy of highest grade as it:
  - can be converted into almost all other forms of energy easily and with very high efficiency
- Per capita consumption of electricity is considered to be an indication of the development of the country

Electricity Consumption vs GDP, per capita



Source: Jason Ross, Schiller Institute Conference, 7th April 2016, New York , USA

GDP represents the monetary value of all goods and services produced within a nation's geographic borders over a specified period of time.

GDP per capita (PPP based) is gross domestic product converted to international dollars using **purchasing power parity** rates and divided by total population. An international dollar has the same purchasing power over GDP as a U.S. dollar has in the United States.

Transmission & Distribution losses

	T & D Losses (%)		
Sl. No.	Name of the Country	2011	2012
1	Korea	3.57	3.47
2	Japan	4.98	4.79
3	Germany	4.70	4.46
4	Italy	6.46	6.61
5	Australia	5.94	5.68
6	South Africa	9.61	10.19
7	France	6.47	7.99
8	China	6.54	6.56
9	USA	6.41	6.73
10	Canada	6.27	8.19
11	UK	8.06	8.26
12	Russia	12.59	12.59
13	Brazil	16.08	16.63
14	India	23.97	23.65
15	World	8.90	8.89

Note :-
Basic data obtained from IEA Website (Except India)
* Per Capita Consumption= (Gross Electrical Energy Availability/Midyear Population).

Some country wise statistics (2014)

Country	Area 1000 km^2	Population (Millions)	Per capita consumption kWh/person	Total consumption GWh	% of total GWh
Iceland	103 (0.07%)	0.32 (0.005%)	**53568**	17142	0.08
Norway	385 (0.26%)	5.1 (0.07 %)	**24581**	125363	0.59
USA	9833 (6.6%)	319 (4.5%)	12271	**3914449**	**18.3**
Russia	17098 (11.5%)	143 (4.5%)	7475	1068925	5.0
Japan	377 (0.25%)	127 (2.0%)	7348	933196	4.4
Germany	357 (0.24%)	81 (1.13 %)	6581	533061	2.5
China	9598 (6.4%)	1356 (18.9 %)	4074	5524344	25.8
India	3287 (2.2%)	1236 (17.2%)	**787**	972732	4.55
Pakistan	882 (0.6%)	196 (2.7%)	418	81928	0.38
World	**148940**	**7175 (100%)**	**2980**	**2,13,81,500**	**100 %**

Comparison of Transmission & Distribution losses

	T & D Losses (%)		
Sl. No.	Name of the Country	2011	2012
1	Korea	3.57	3.47
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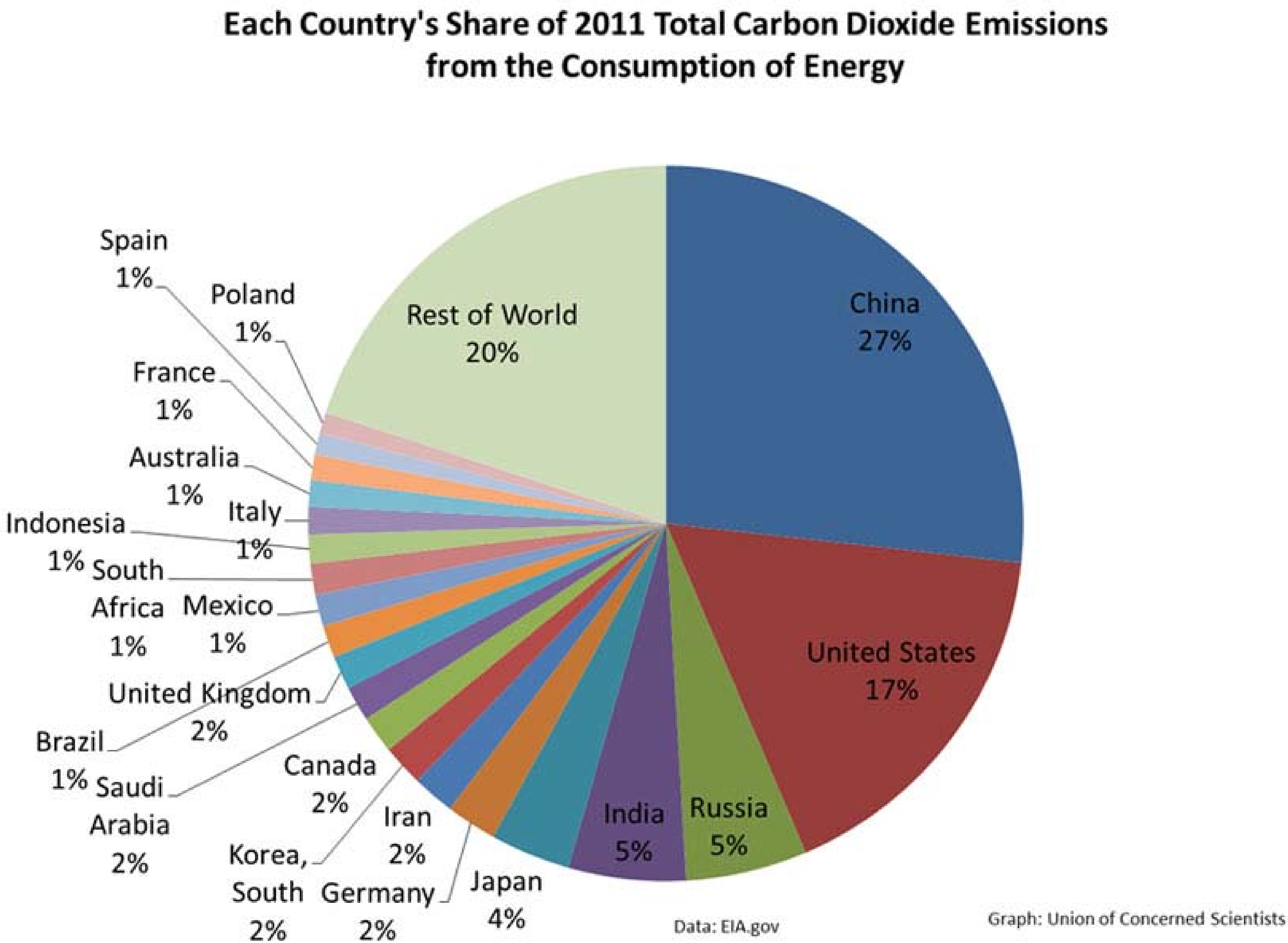
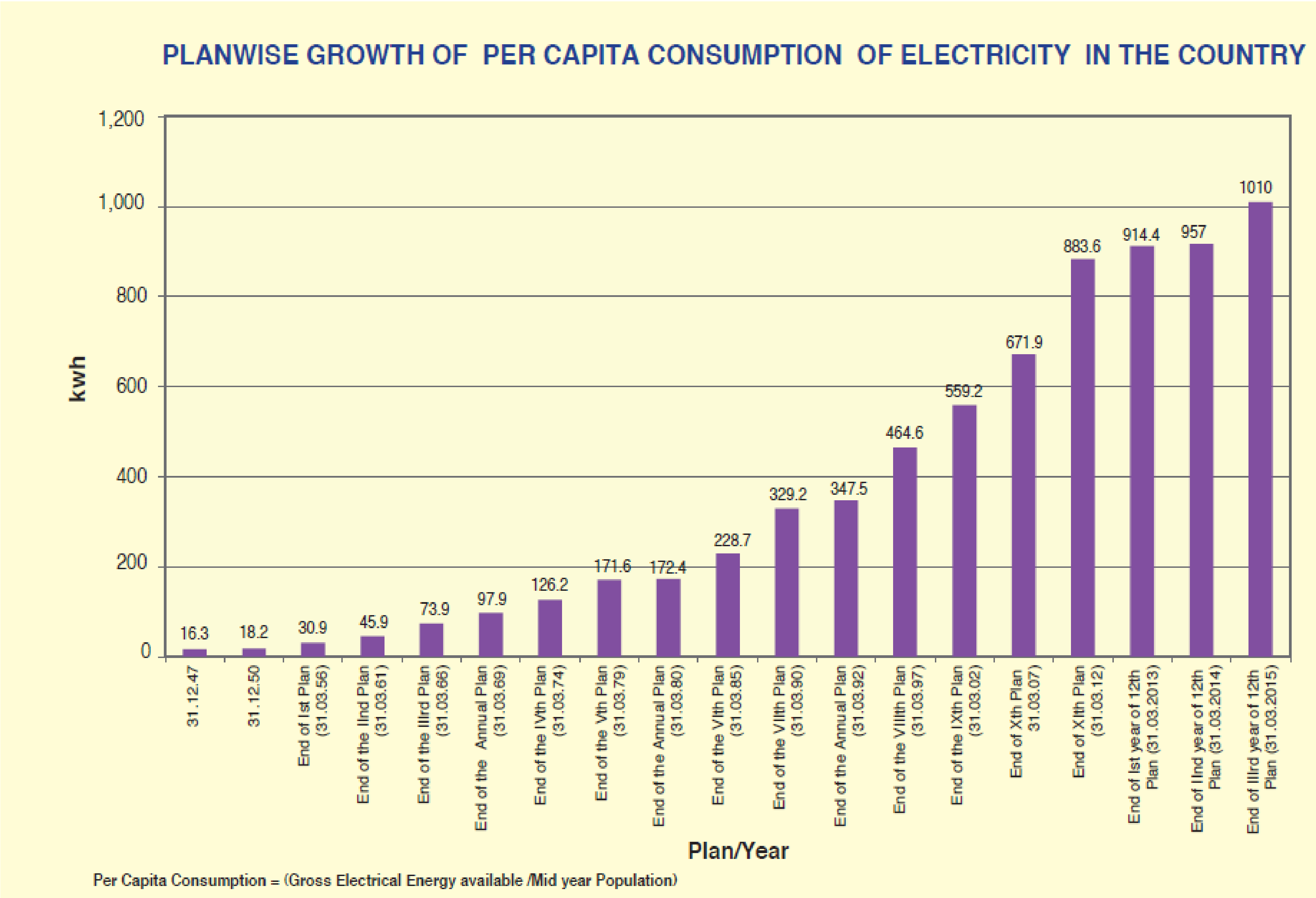

Each Country's Share of 2011 Total Carbon Dioxide Emissions
from the Consumption of Energy
China
27%
United States
17%
Russia
5%
India
5%
Japan
4%
Germany
2%
Iran
2%
Korea, South
2%
Canada
2%
Saudi Arabia
2%
United Kingdom
2%
Brazil
1%
Mexico
1%
South Africa
1%
Indonesia
1%
Italy
1%
Australia
1%
France
1%
Poland
1%
Spain
1%
Rest of World
20%
Data: EIA.gov
Graph: Union of Concerned Scientists

Electrical power scenario in India:

Some Statistics



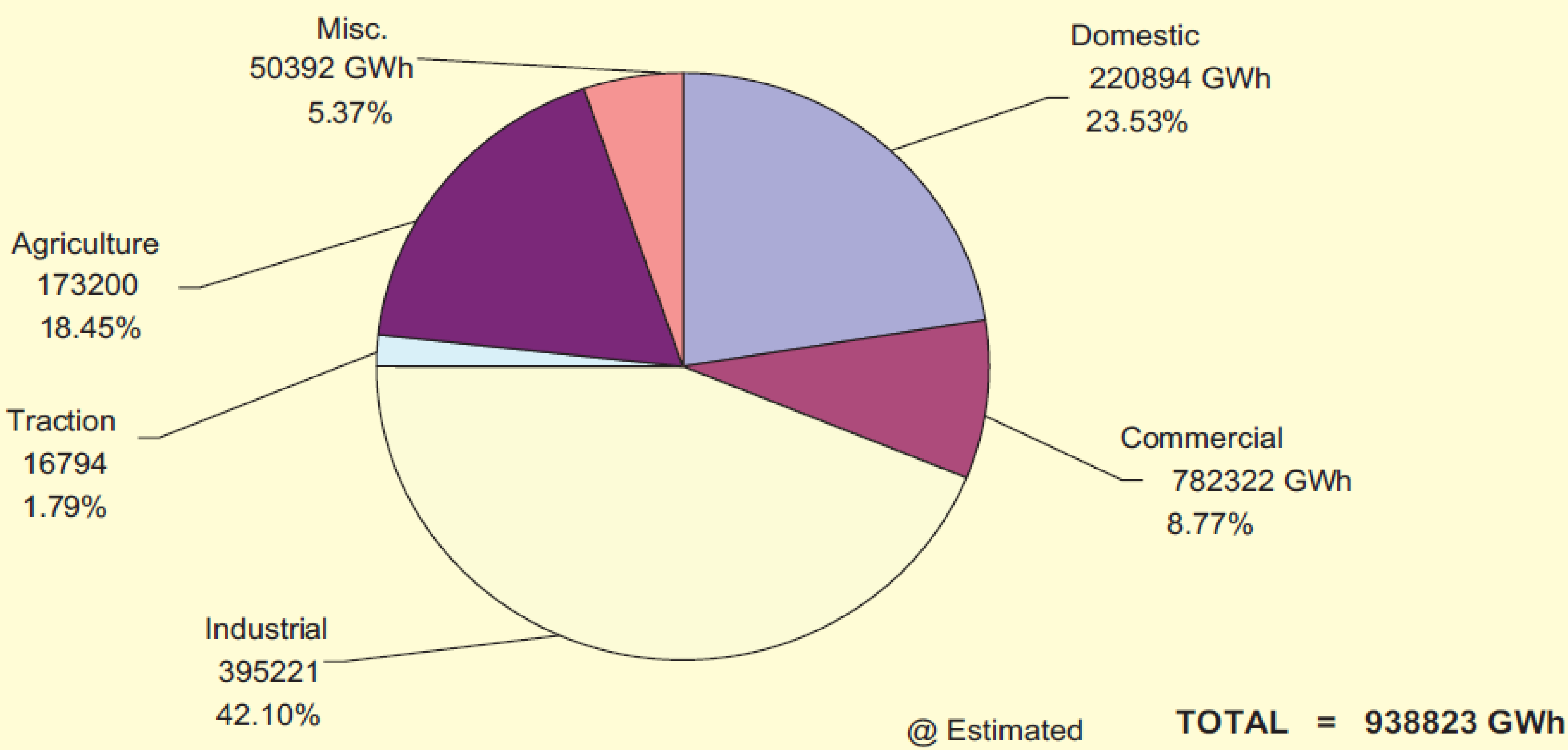
Source: Ministry of Power, Government of India

Pie Chart : 26

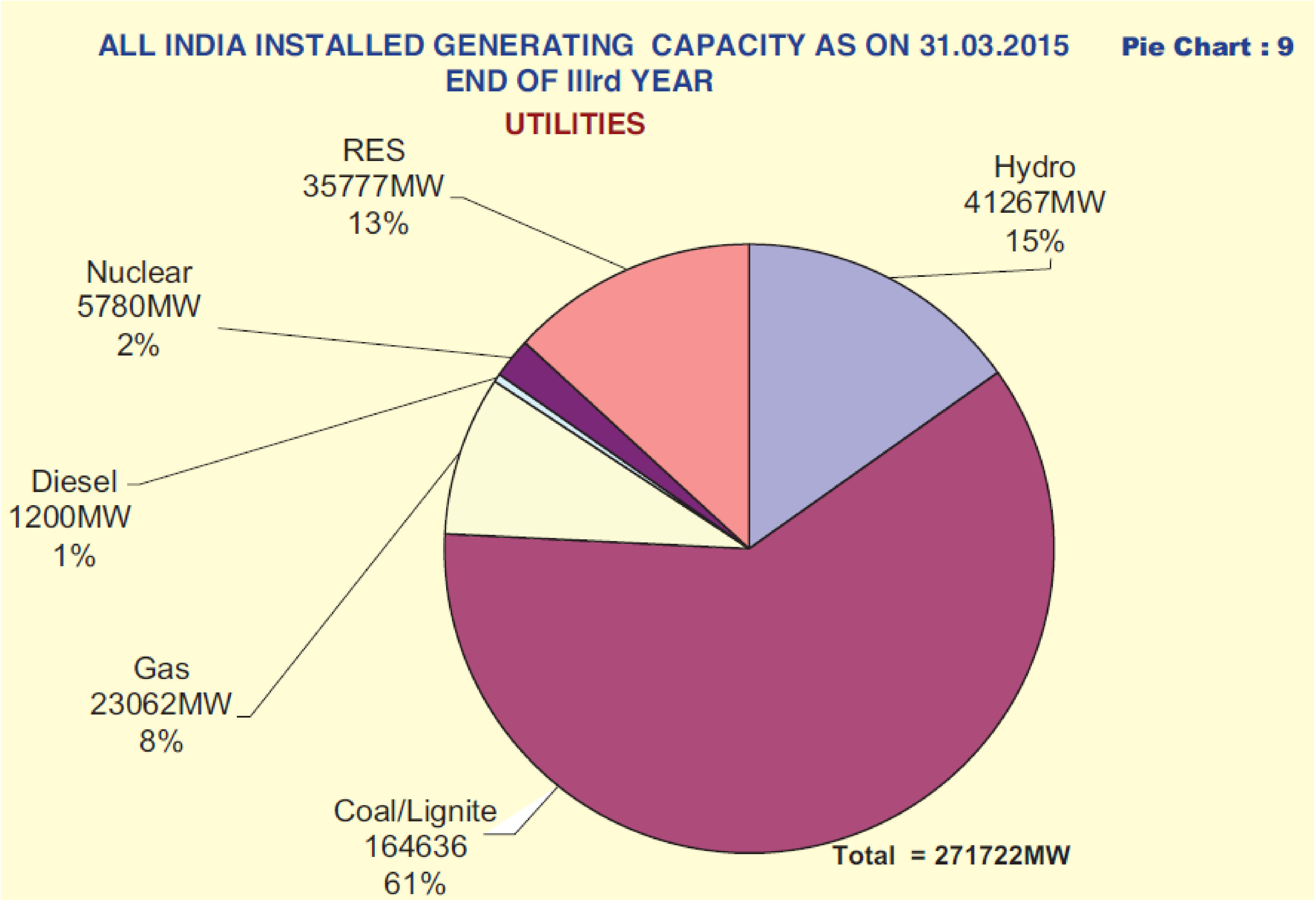
ALL INDIA ELECTRICITY CONSUMPTION SECTOR WISE- END OF IIIrd YEAR OF XIITH PLAN

UTILITIES & NON - UTILITIES

(31.03.2015) @



Source: Ministry of Power, Government of India

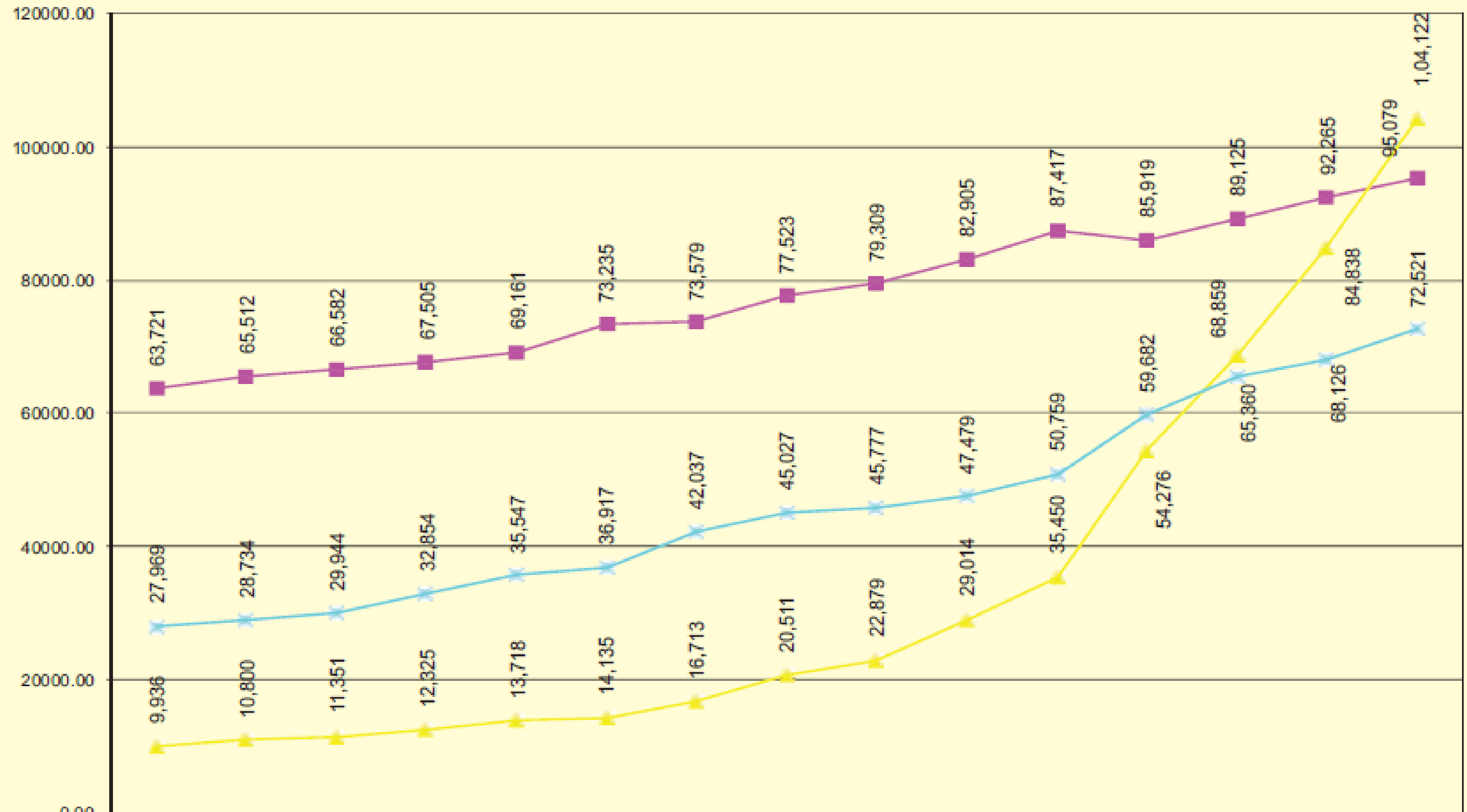


Source: Ministry of Power, Government of India

Chart : 8A

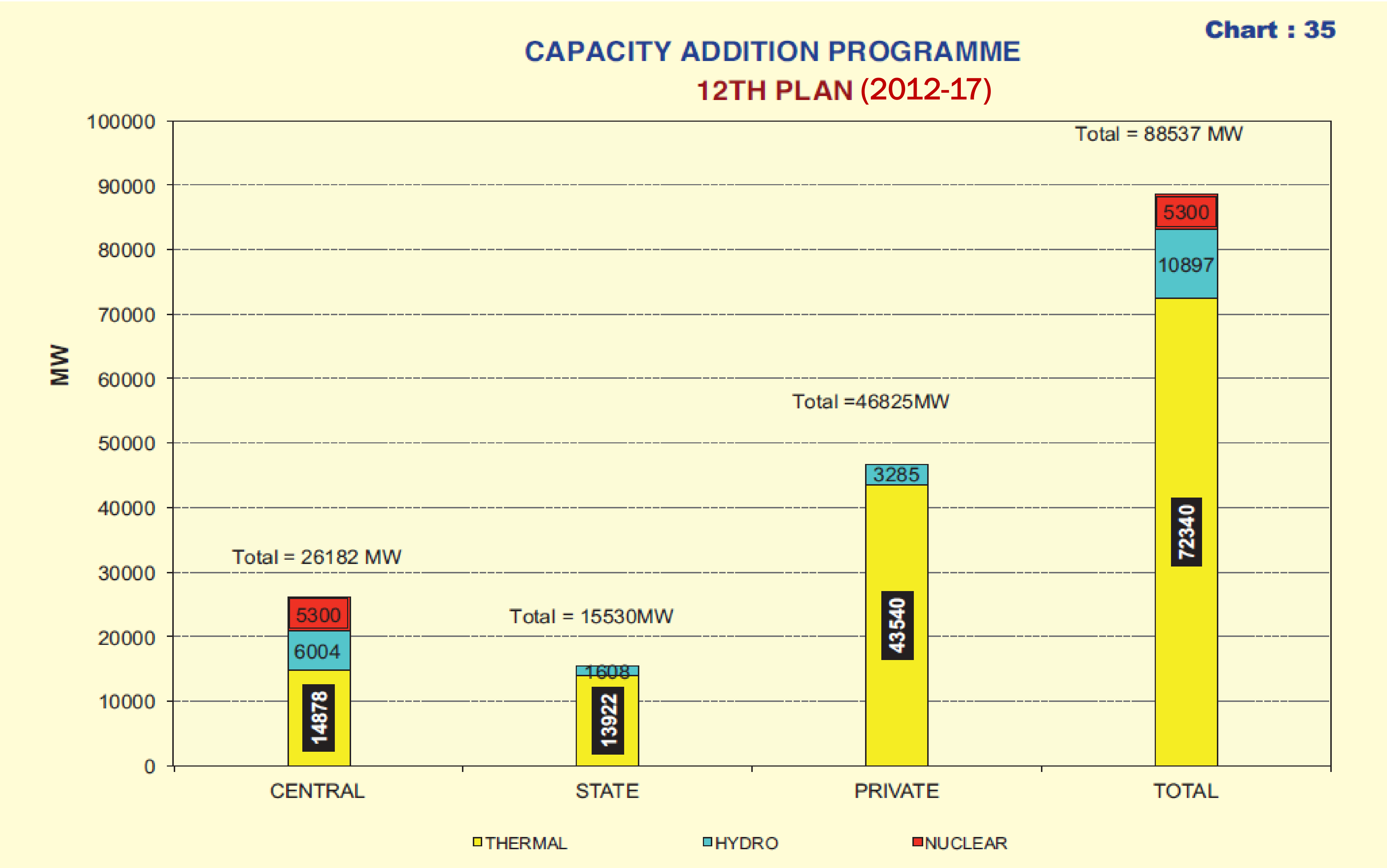
SECTORWISE GROWTH OF GENERATION CAPACITY (MW)

2000-01 TO 2014-15



	2001	2002	2003	2004	2005	2006	2007	2008	2009	2010	2011	2012	2013	2014	2015
State	63721	65512	66582	67505	69161	73235	73579	77523	79309	82905	87417	85919	89125	92265	95079
Private	9936	10800	11351	12325	13718	14135	16713	20511	22879	29014	35450	54276	68859	84838	104122
Central	27969	28734	29944	32854	35547	36917	42037	45027	45777	47479	50759	59682	65360	68126	72521

Source: Ministry of Power, Government of India



Source: Ministry of Power, Government of India

Some observations based on statistics

In India, a major part of electricity is generated in coal based thermal power plants

It is expected that these thermal power plants will continue to dominate the energy sector in the coming decades also

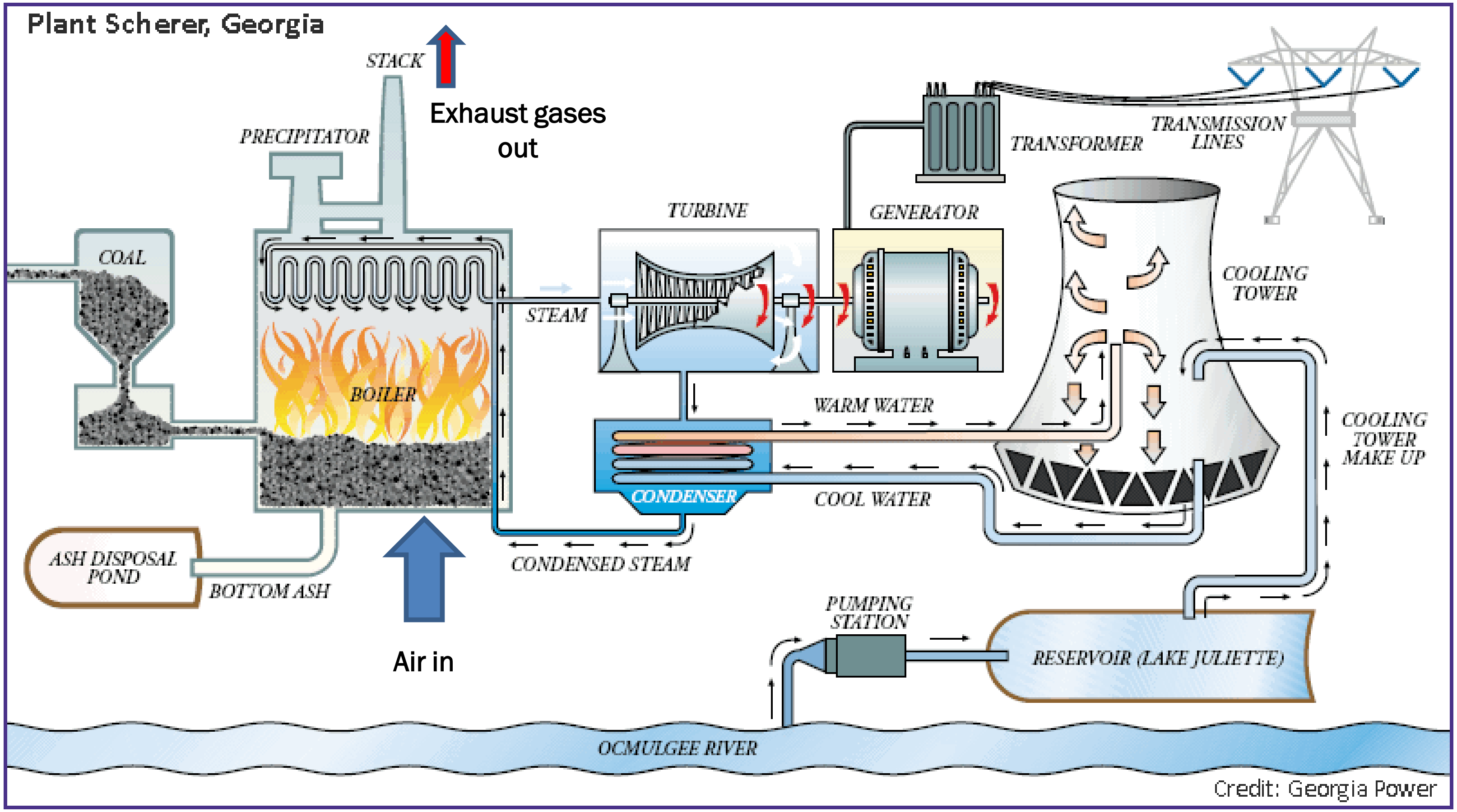
However, Indian coal has **low calorific value** and **high ash content**, as a result, per kWh consumption of coal is higher in India (**≈ 0.7 kg/kWh**) compared to other countries (**≈ 0.45 kg/kWh** for US plants)

The **poor quality of coal** affects both the **plant's thermal performance** as well as **emissions**

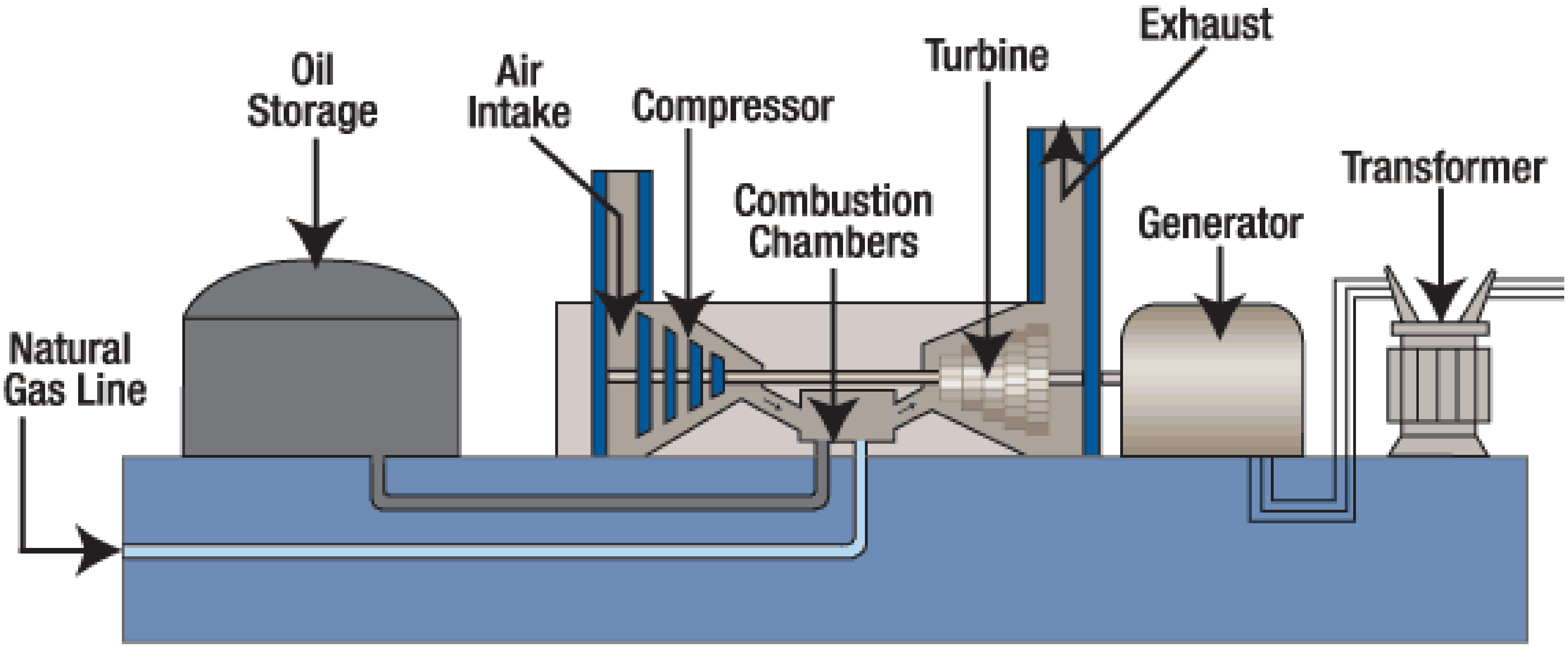
Advanced and innovative technologies are needed to address these issues

Thermal Power Plants – Some basics

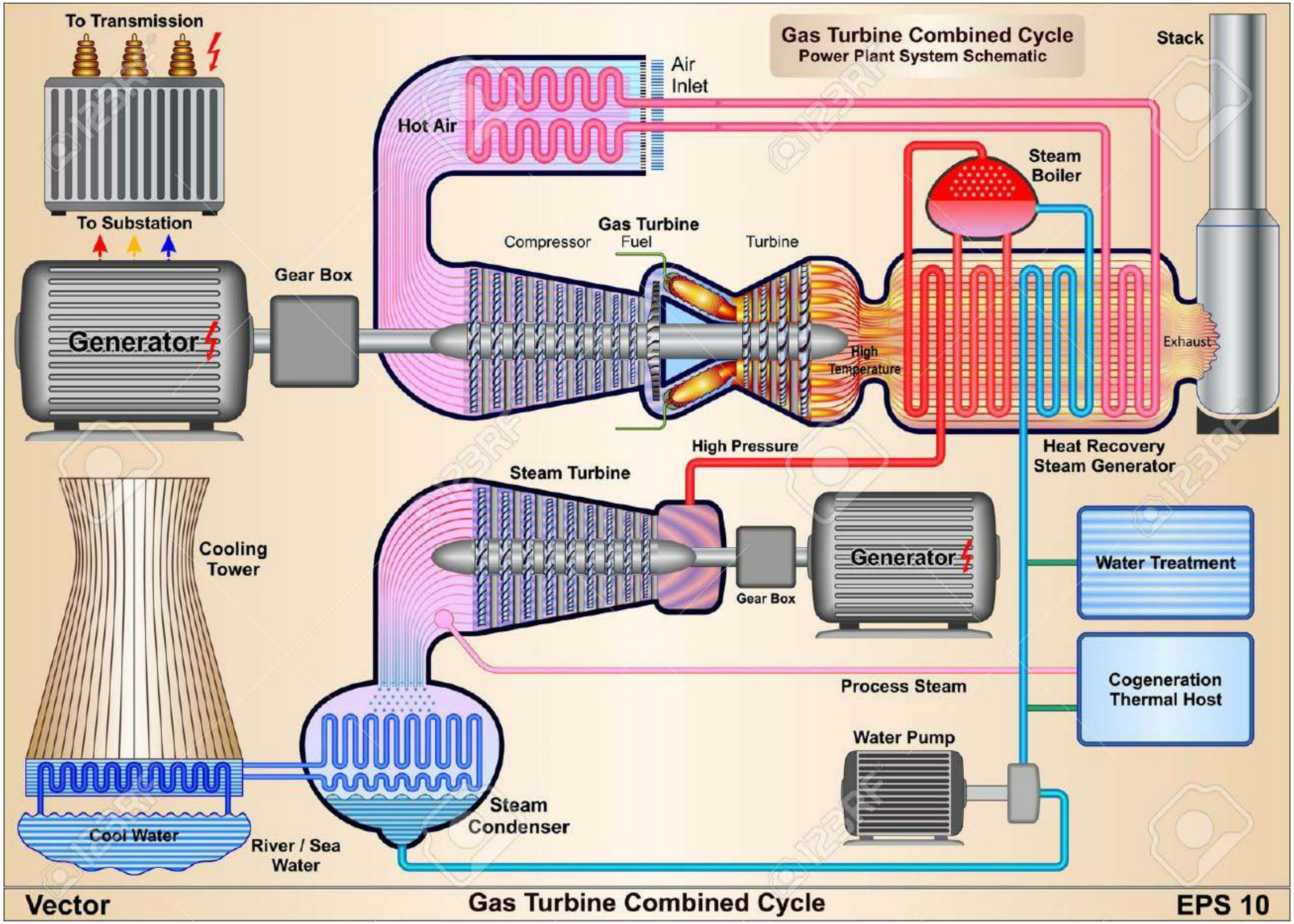
- Thermal power plants use a variety of fuels/energy sources such as:
  - Coal
  - Natural Gas
  - Various types of petroleum products such as diesel
  - Nuclear fuels
  - Solar energy
  - Geothermal energy
  - Ocean Thermal Energy etc.
- All the thermal power plants employ a thermodynamic cycle that **continuously** converts the thermal energy into mechanical or electrical energy
- Hence all these power plants are subjected to the **fundamental laws of thermodynamics**



A typical coal based steam power plant

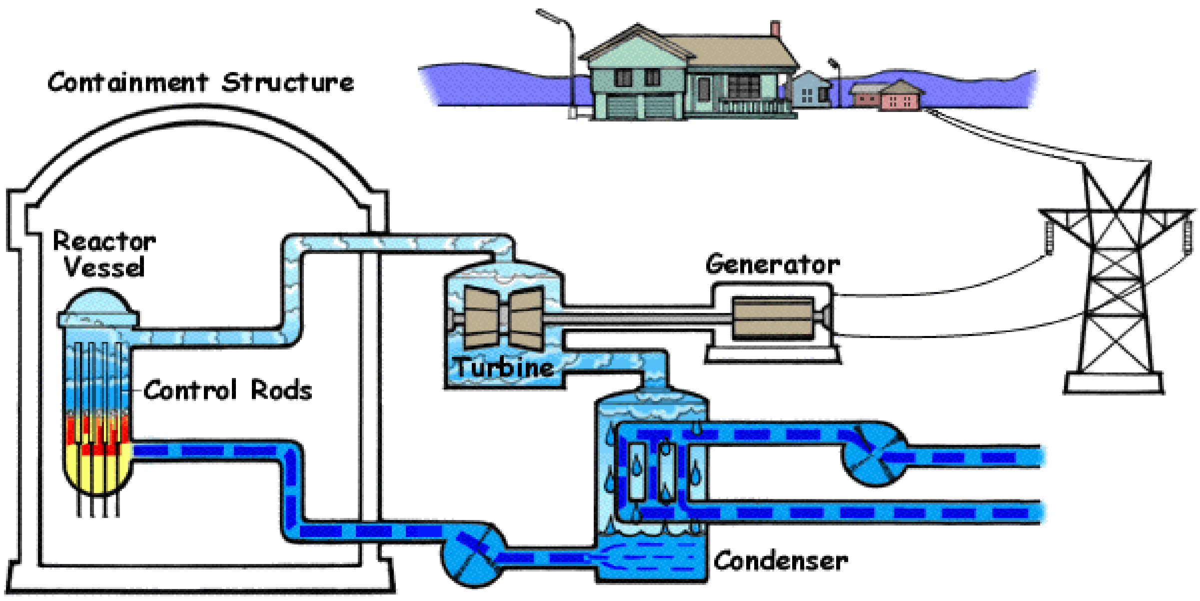


A typical dual fuel based gas turbine power plant

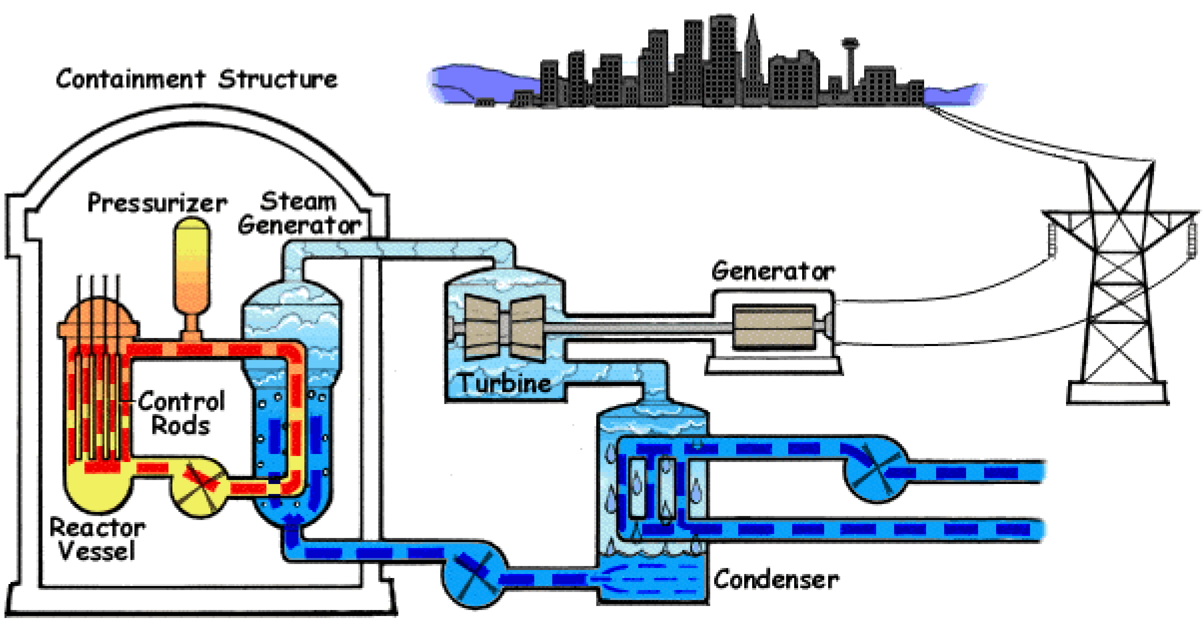


A typical combined cycle thermal power plant

Nuclear Energy based thermal power plant

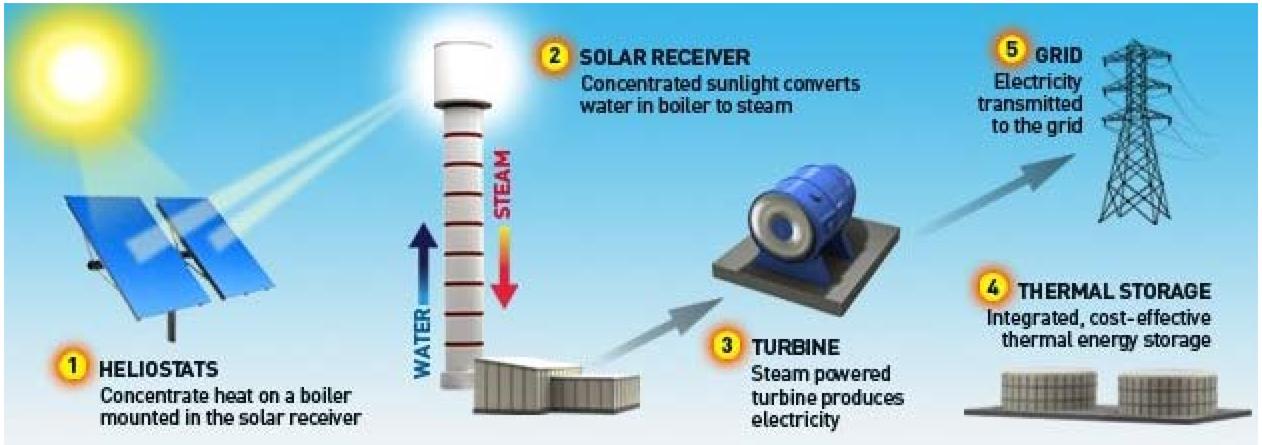


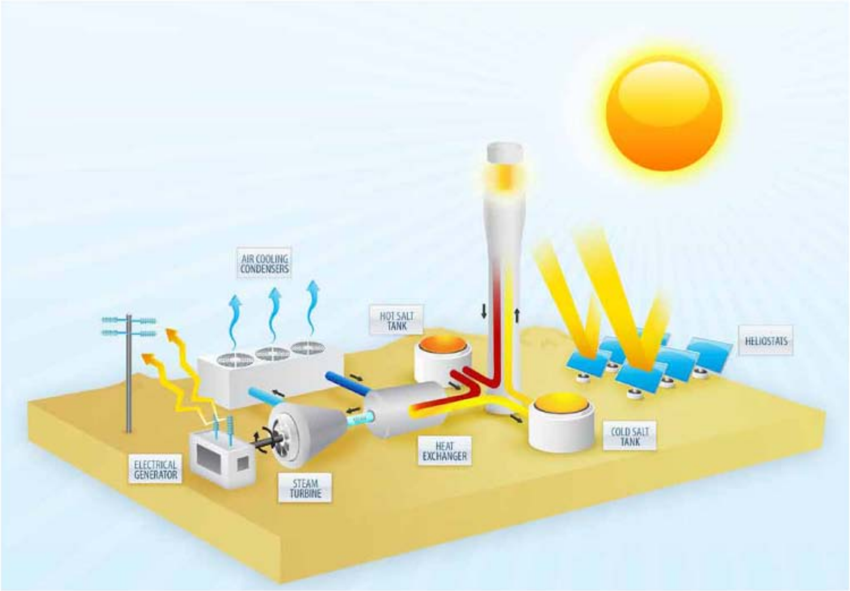
A typical boiling water reactor based nuclear power plant



A typical pressurised water reactor based nuclear power plant

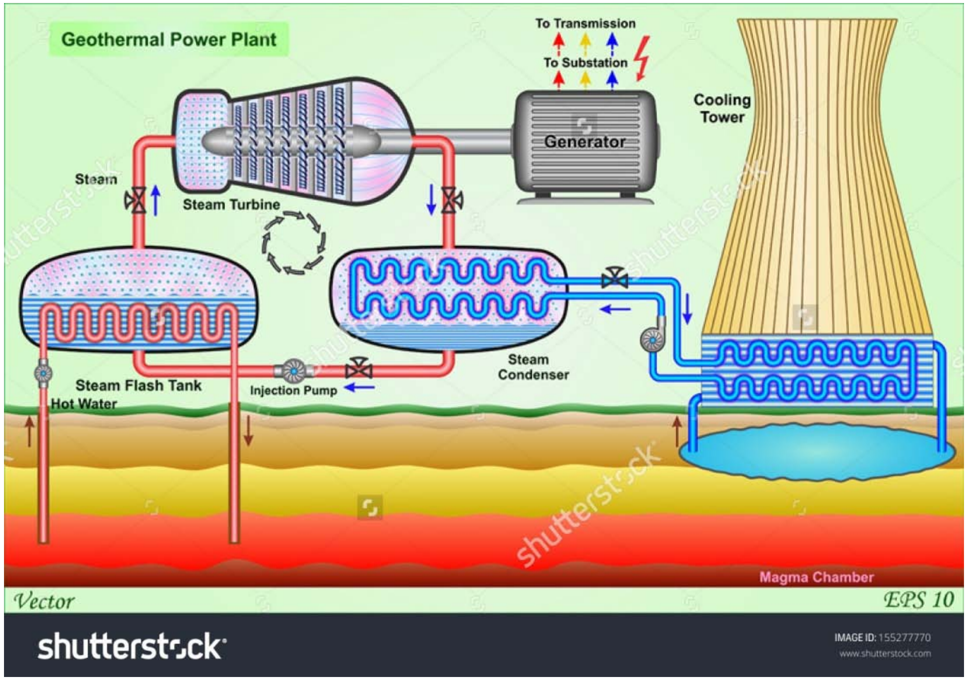
Solar Energy based thermal power plant



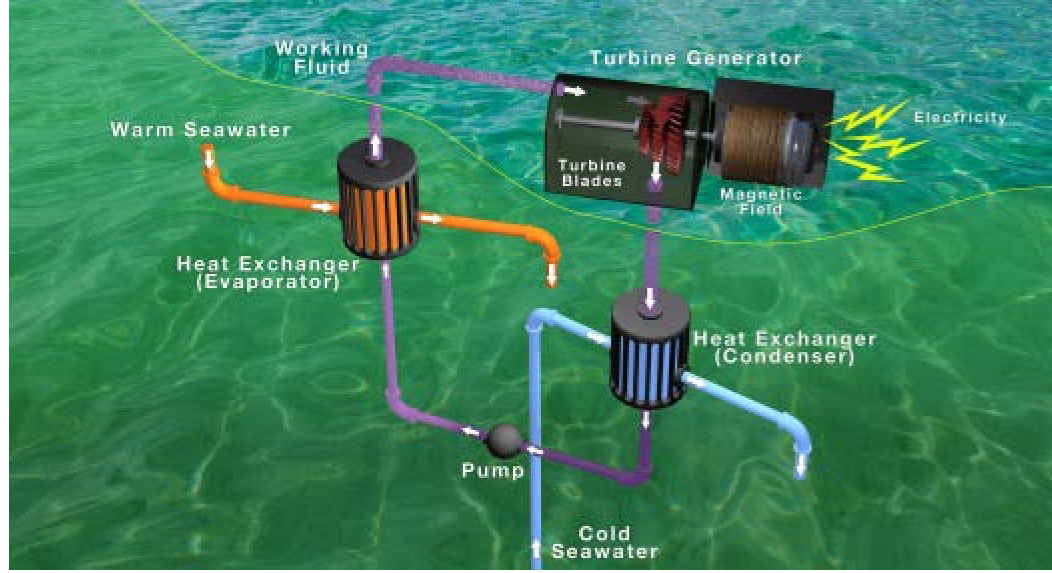


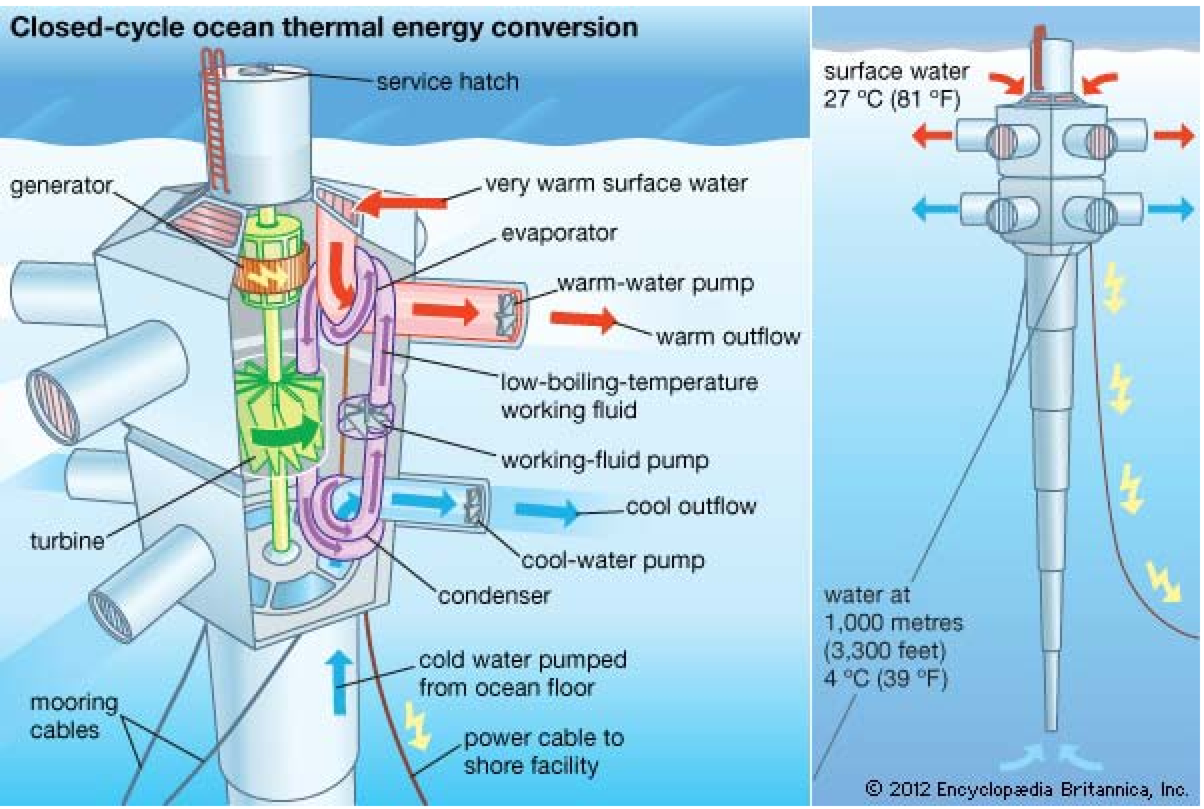
Solar Energy based thermal power plant with storage

Geothermal Energy based thermal power plant



Ocen Thermal Energy based thermal power plant

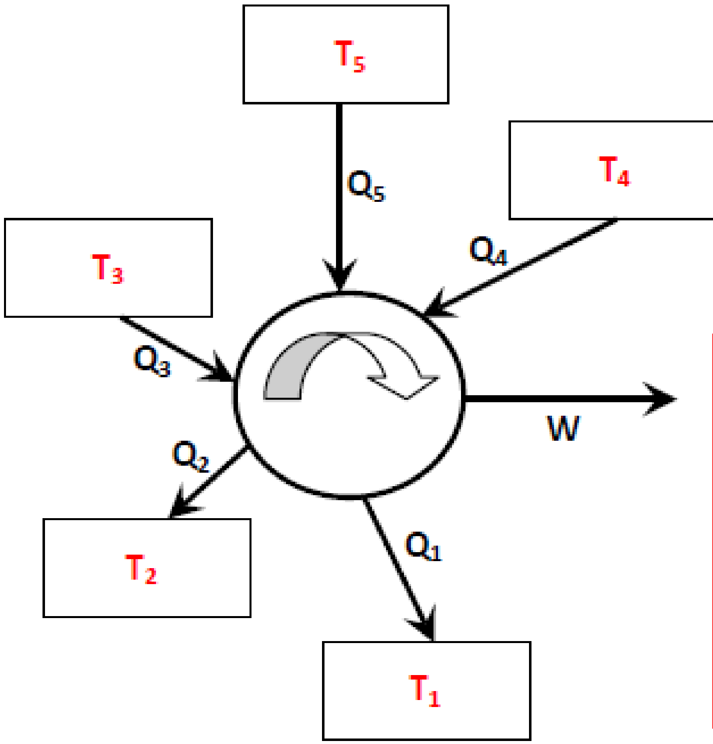




Thermal Power Plants

- All the thermal power plants employ a thermodynamic cycle that **continuously** converts the thermal energy into mechanical or electrical energy
- Hence all these power plants are subjected to the **fundamental laws of thermodynamics**
- Hence **understanding** of these thermodynamic laws is essential

Basic thermodynamics of thermal power plant cycles



$$1st\ Law: \sum_{i=1}^{i=n} Q_i = W$$

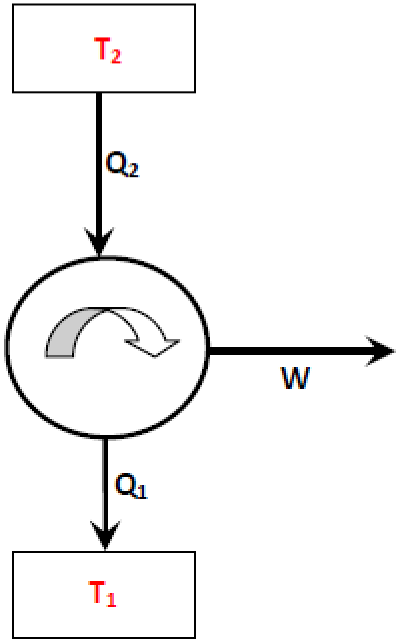
$$2nd\ Law: \sum_{i=1}^{i=n} \frac{Q_i}{T_i} \leq 0$$

$$Entropy\ generation, S_{gen} = -\sum_{i=1}^{i=n} \frac{Q_i}{T_i} \geq 0$$

$$T_5 > T_4 > T_3 > T_2 > T_1$$

Sign Convention: Heat supplied and work produced are positive

Thermal power plant cycle with two thermal reservoirs



$$1st\ Law: Q_2 - Q_1 = W$$

$$2nd\ Law: \frac{Q_2}{T_2} - \frac{Q_1}{T_1} \leq 0$$

$$\eta_{th} = \frac{W}{Q_2} = \left(\frac{Q_2 - Q_1}{Q_2}\right) \leq \left(\frac{T_2 - T_1}{T_2}\right)$$

$$Entropy\ generation, S_{gen} = \frac{Q_1}{T_1} - \frac{Q_2}{T_2} \geq 0$$

$$T_1 . S_{gen} = \left[Q_2\left(\frac{T_2 - T_1}{T_2}\right) - W\right] = (W_{rev} - W) = W_{loss}$$

Example

Given: **T_1 = 40°C** = 313 K; **T_2 = 560°C** = 833 K;

W = 1000 MW; η_{th} = **0.40**

Find: **a)** Power lost due to irreversibilities (MW),

b) entropy generation rate (MW/K)

Ans.:

$$Q_2 = \frac{W}{\eta_{th}} = \frac{1000}{0.4} = 2500\ MW$$

$$Q_1 = Q_2 - W = 1500\ \text{MW}$$

$$\boldsymbol{W_{loss}} = \left[\boldsymbol{Q_2}\left(\frac{\boldsymbol{T_2 - T_1}}{\boldsymbol{T_2}}\right) - \boldsymbol{W}\right] = \left[2500\left(\frac{833-313}{833}\right) - 1000\right]$$

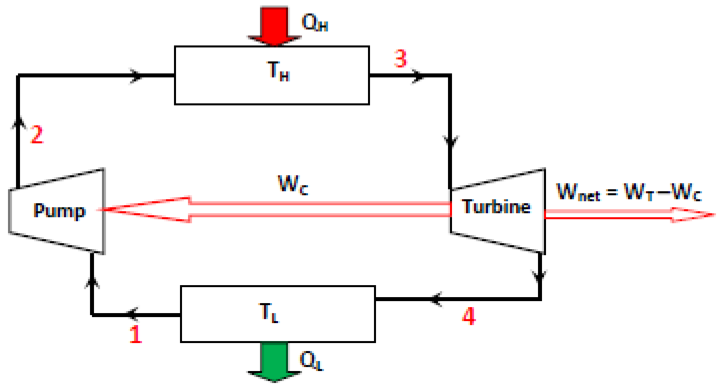
$$= \mathbf{560.62}\ \boldsymbol{MW}$$

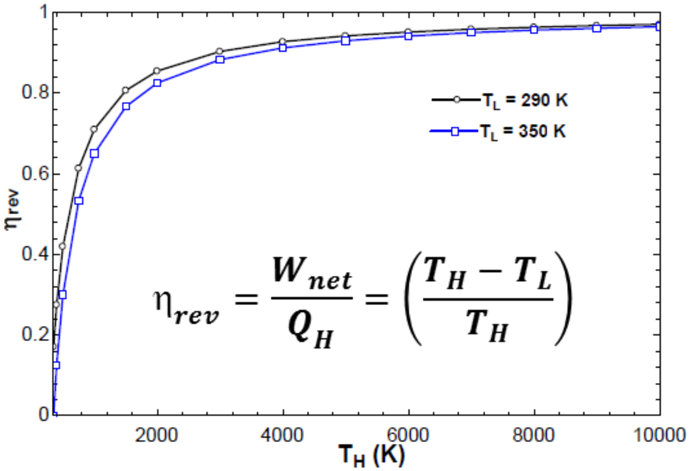
$$\boldsymbol{S_{gen}} = \frac{W_{loss}}{T_1} = \mathbf{1.7911}\ \boldsymbol{MW/K}$$

Carnot power cycle

Sadi Carnot (1796-1832)

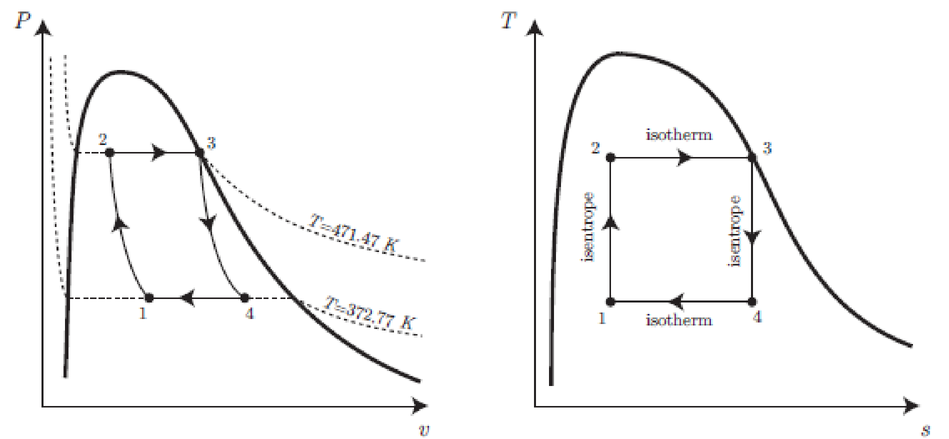
Carnot cycle is a **completely reversible**, but **hypothetical** cycle, that is generally used as an ideal for **2-temperature power cycle**





$$\eta_{rev} = \frac{W_{net}}{Q_H} = \left(\frac{T_H - T_L}{T_H}\right)$$

Thermal efficiency of a reversible (e.g. Carnot) cycle



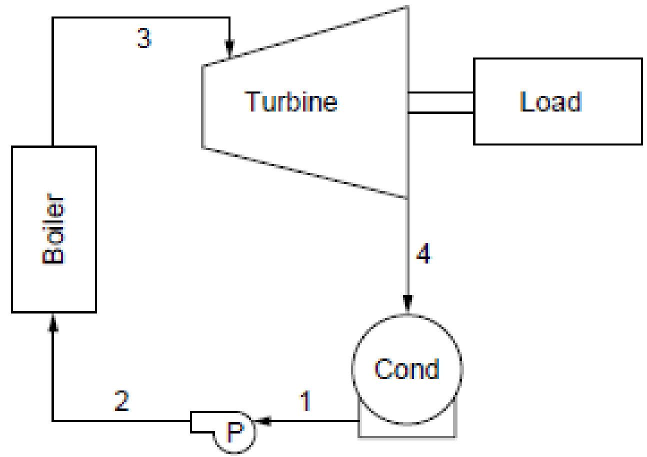
Carnot Vapour Power cycle with water as the working fluid

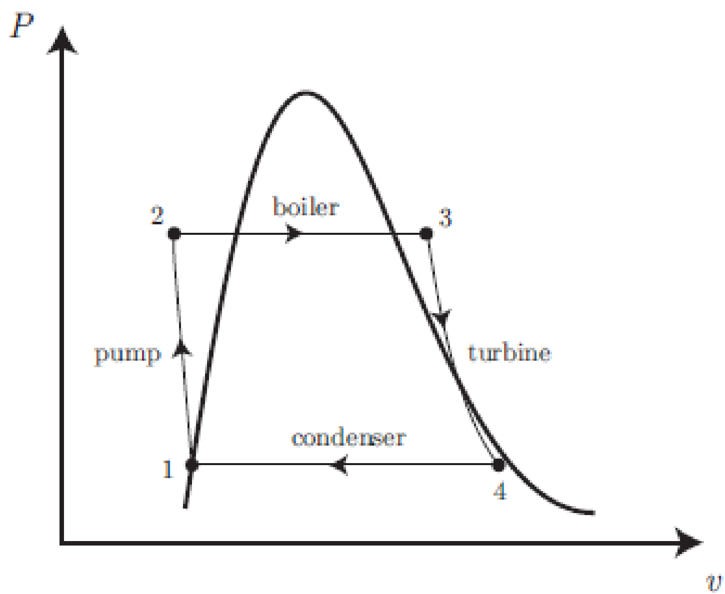
Carnot cycle and practical problems

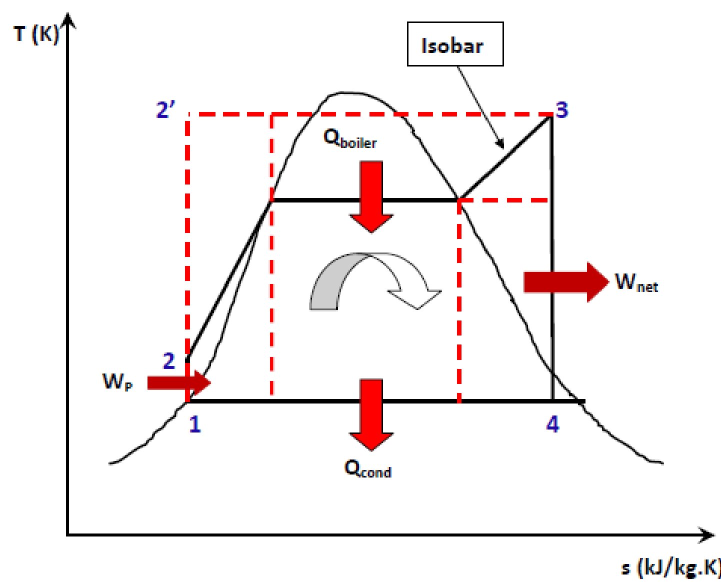
- **Performance of Carnot cycle** is a function of temperatures only, and is **independent of working fluid**
- Hence, theoretically Carnot cycle can be a vapour cycle or a gas cycle
- Carnot gas cycles are almost impossible to develop as they require isothermal heat addition and heat rejection
- Using the process of phase change, nearly isothermal heat transfer can be achieved $\Rightarrow$ Vapour cycles that resemble Carnot cycle are feasible
- Due to heat transfer and fluid friction, it is not possible achieve **reversible, adiabatic compression and expansion processes** in pumps and turbines, respectively
- A **finite temperature difference** is required for transferring heat at both high and low temperature ends $\Rightarrow$ Cycle is **externally irreversible**
- Need for avoiding presence of two-phase mixture in turbine and pump, calls for non-isothermal heat transfer

Rankine cycle – Basis for most of the thermal power plants

William Rankine (1820-1872)





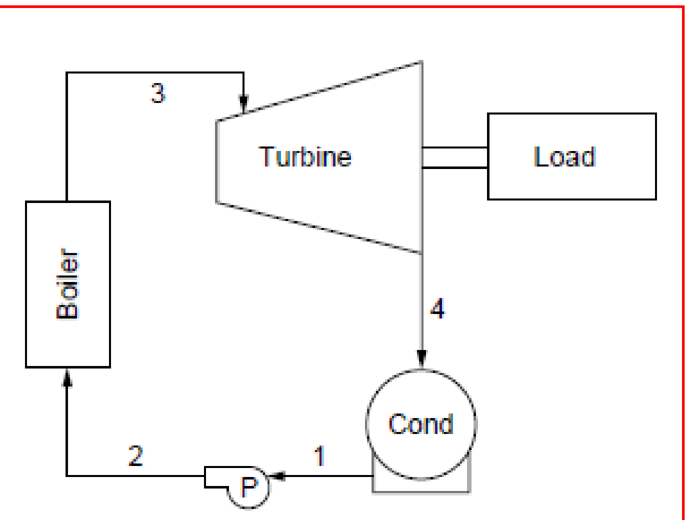


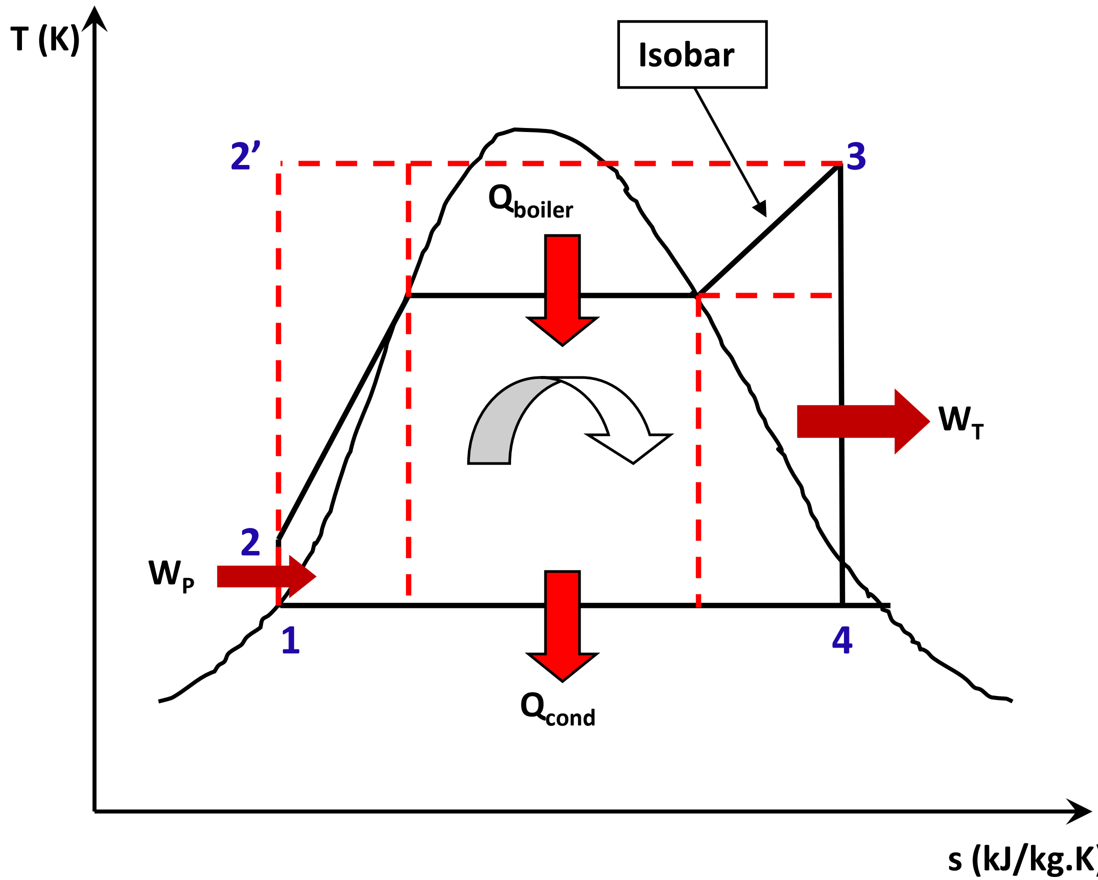
Simple Rankine cycle

Simple Rankine cycle

- The simple Rankine cycle deviates from the Carnot cycle as the **heat addition process** in the boiler is **no longer isothermal**
- This is because, an **isothermal heat addition** requires, **compression/expansion of two-phase mixture** or **compression** of condensed liquid **to very high pressure** followed by non-isobaric heat addition. Both these processes are either not desirable or extremely difficult to achieve in practice.
- In view of the above, in Rankine cycle a **compromise** is made between **efficiency** and **practical problems**
- Organic Rankine cycles are low temperature cycles in which the working fluid is not water

Simple Rankine cycle





Analysis of simple, ideal Rankine cycle

A simple, steady state analysis of the cycle **yields** **useful information** related to:

a) **Mass flow rate of working fluid** for a given power output and operating conditions (assuming that the working fluid is fixed, i.e., water)

b) **Heat transfer rates** across boiler and condenser

c) **Power** output from turbine and power input to pump

d) **Cycle efficiency** and sources of **losses** (?)

e) **Effects** of working fluid and operating conditions on **cycle performance**

Simplifying assumptions:

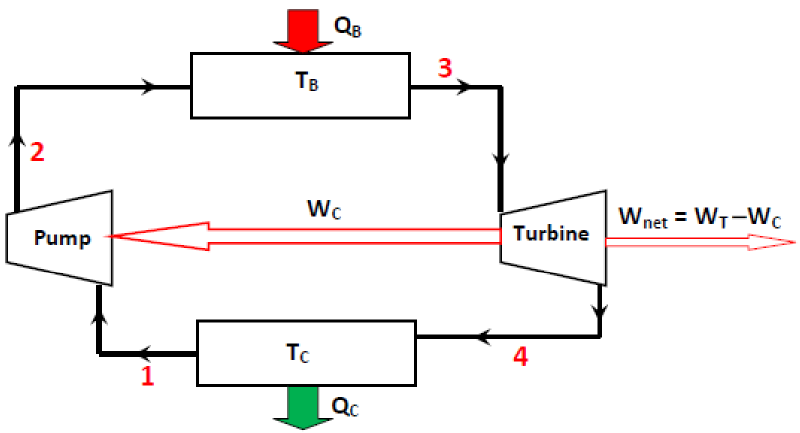
1) The cycle is **internally reversible**

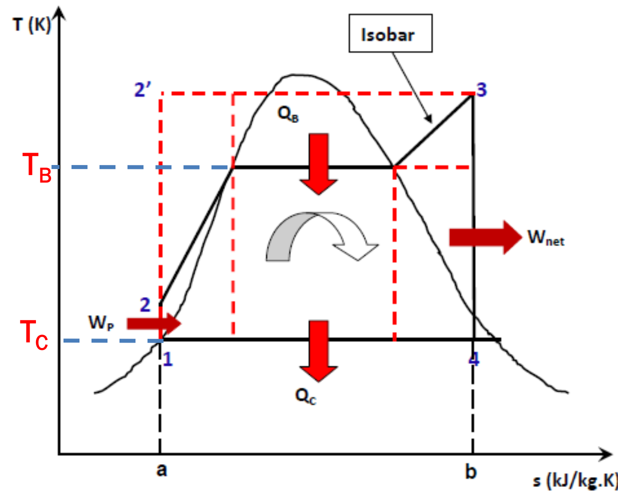
2) The system is operating in **steady state**

3) The **potential and kinetic energy changes** across any component are **negligible** compared to work and/or heat transfer across the component

4) The **working fluid** circulating through the system is a **pure fluid** (water)

Analysis of simple, ideal Rankine cycle (contd.)





Steady State, Steady Flow Energy equation (one inlet and one outlet):

$$\boxed{\dot{m}(h + \frac{V^2}{2} + gZ)_i + Q = \dot{m}(h + \frac{V^2}{2} + gZ)_i + W}$$

Turbine (process 3-4; assumed to be reversible and adiabatic):

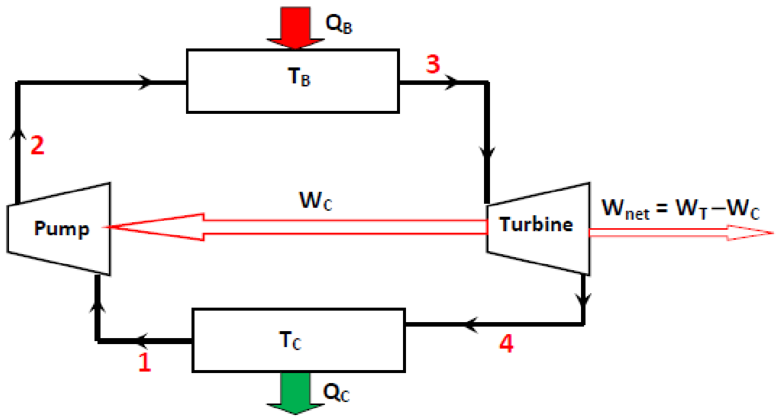
$$\dot{m}(h_3 - h_4) = W_T$$

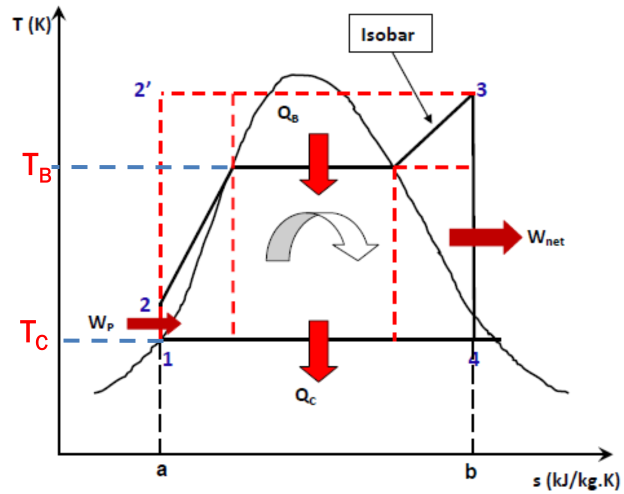
$$s_3 = s_4$$

Condenser (process 4-1, assumed to be isobaric): $p_C = p_{sat}(T_C)$

$$\boxed{\dot{m}(h_4 - h_1) = Q_C}$$

Analysis of simple, ideal Rankine cycle (contd.)





Pump (process 1-2; assumed to be reversible and adiabatic):

$$\dot{m}(h_2 - h_1) = W_P$$

$$Tds = dh - vdP$$

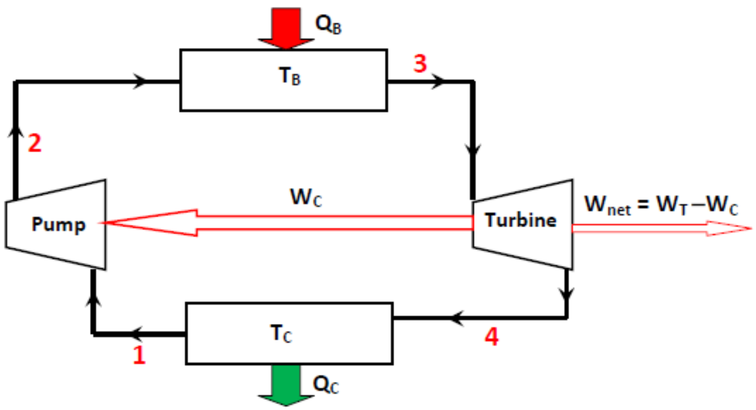
$$\Rightarrow \int_1^2 dh = h_2 - h_1 = \int_1^2 vdP \cong v(p_B - p_C)$$

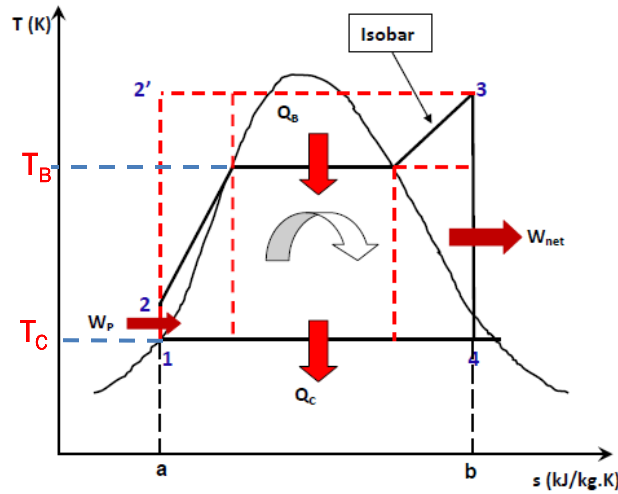
$$\Rightarrow W_P \cong \dot{m}v(p_B - p_C)$$

Boiler (process 2-3; assumed to be isobaric): $p_B = p_{sat}(T_B)$

$$\dot{m}(h_3 - h_2) = Q_B$$

Analysis of simple, ideal Rankine cycle (contd.)





Overall energy balance for the cycle:

$$\boxed{Q_B - Q_C = W_{net} = W_T - W_P}$$

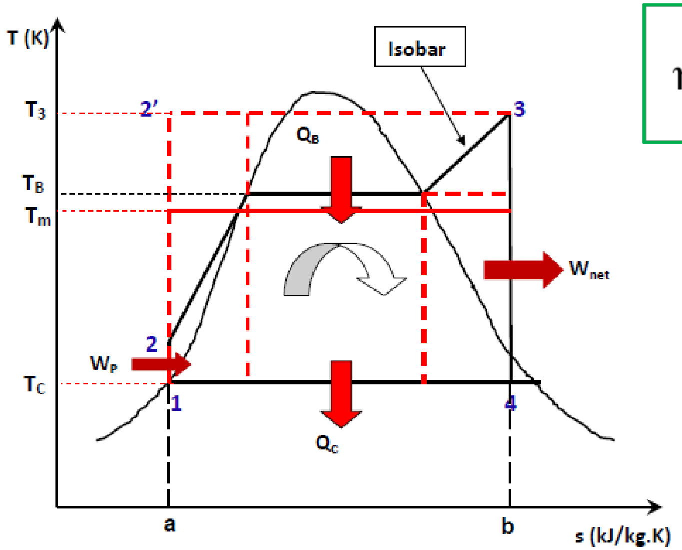
Thermal efficiency of the cycle, η_{th} is defined as:

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

$$= \frac{Area\ 1-2-3-4}{Area\ a-1-2-3-4-b}$$

$$Work\ Ratio, WR = \frac{W_T - W_P}{W_T}$$

Analysis of simple, ideal Rankine cycle (contd.)



$$\eta_{Carnot} = \frac{T_3 - T_C}{T_3}$$

In terms of **mean temperature** T_m, the **thermal efficiency** of the cycle, is given by:

$$\eta_{th} = \frac{W_{net}}{Q_B} = \frac{(h_3 - h_2) - (h_4 - h_1)}{(h_3 - h_2)} = \frac{T_m - T_C}{T_m}$$

$$T_m = \frac{(h_3 - h_2)}{(s_3 - s_2)} \Rightarrow Q_B = \dot{m}(h_3 - h_2) = \dot{m}T_m(s_3 - s_2)$$

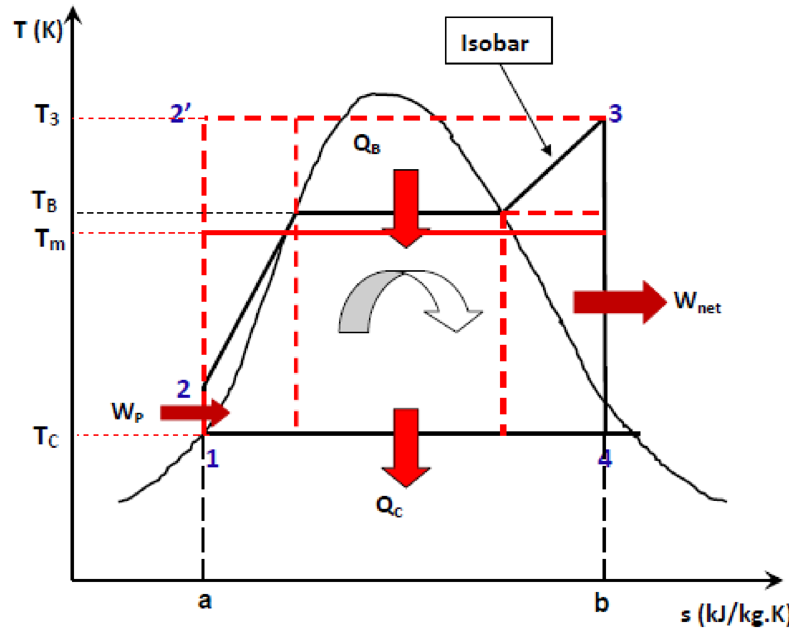
Since **T_m is less than T_3**, for same maximum and minimum temperatures, the **efficiency of Rankine cycle** is **always less than** that of **Carnot cycle!**

Second law or **exergetic efficiency** of the cycle, η_{2nd} is defined as:

$$\eta_{2nd} = \frac{\eta_{th}}{\eta_{Carnot}} = \frac{W_{net}}{Q_B\left(\frac{T_3 - T_C}{T_3}\right)} < 1.0$$

Example Problem on Simple Rankine cycle

Given:
Boiler Pressure = **163 bar**
Condenser pressure = **0.07 bar**
Max. heat addition temp., = **538 °C**
Net power output = **500 MW**



Parameter	Simple cycle
Net power output (MW)	500
Steam flow rate (kg/s)	359.8
Turbine power (MW)	505.9
Pump Power (MW)	5.9
Work Ratio	0.9884
Thermal efficiency (%)	43.14
T_{mean} (°C)	275.8
2nd law efficiency (%)	70.11

Results

From property data (EES)

State	1	2	3	4
p (bar)	0.07	163	163	0.07
T (°C)	39.01	39.48	538	39.01
ρ (kg/m^3)	992.5	999.4	-	-
h (kJ/kg)	163.4	179.7	3401	1995
s (kJ/kg.K)	0.559	0.559	6.428	6.428
x	0	-	-	0.7607

Improving efficiency of Rankine cycle

$$\eta_{th} = \frac{W_{net}}{Q_B} = \frac{T_m - T_C}{T_m} = 1 - \frac{T_c}{T_m}$$

Where: T_m is the entropic mean heat addition temperature

T_c is the heat sink temperature

Improving efficiency of Rankine cycle

Rankine cycle efficiency can be increased either by **increasing** the mean temperature of heat addition (T_m) and/or **decreasing** the temperature of heat rejection **(T_c)**

Decreasing T_c significantly is **not possible** due to the **constraint** imposed by the available heat sink

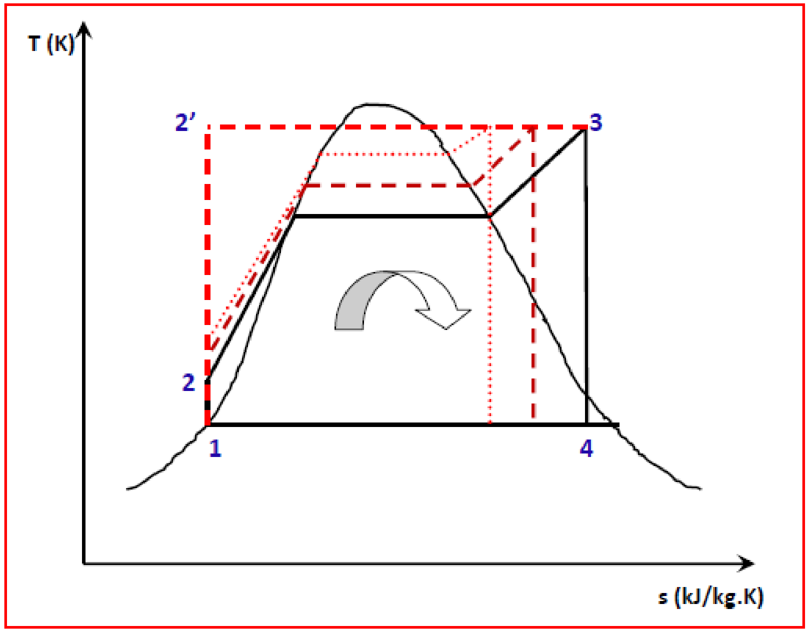
Increasing T_m is possible by using either reheat and/or regeneration

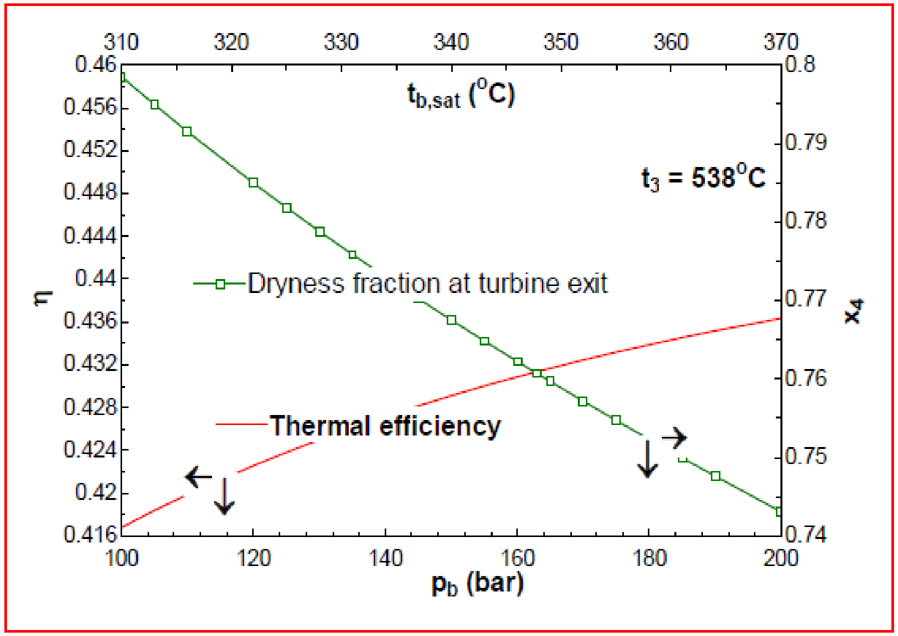
In **actual power plant** cycles, both reheat and regeneration are used to maximize the efficiency subject to economic constraints

Reheat is also beneficial as it minimizes wet expansion and also provides an opportunity for increasing the boiler pressure

Effect of increasing boiler pressure

The mean temperature of **heat addition (T_m)** can be **increased** by **increasing the boiler pressure** $\Rightarrow$ Thermal efficiency increases for given heat source temperature



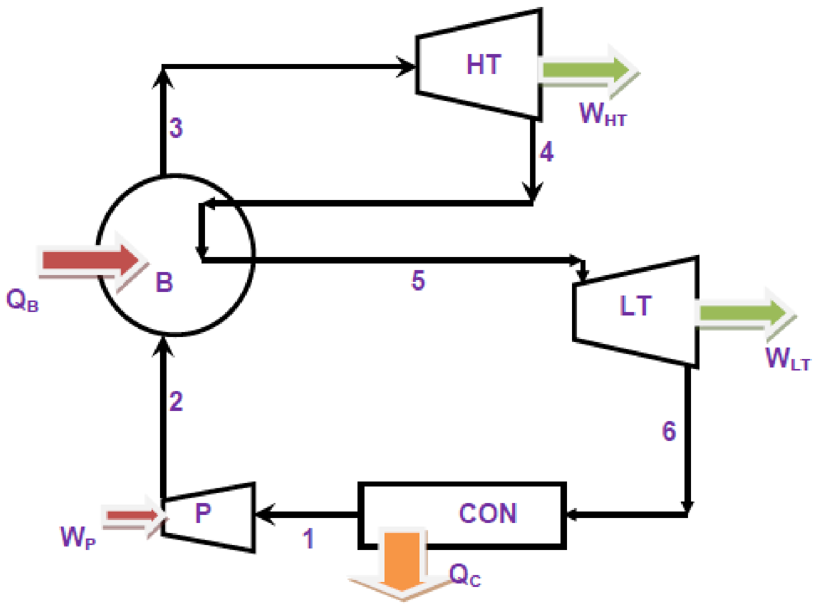


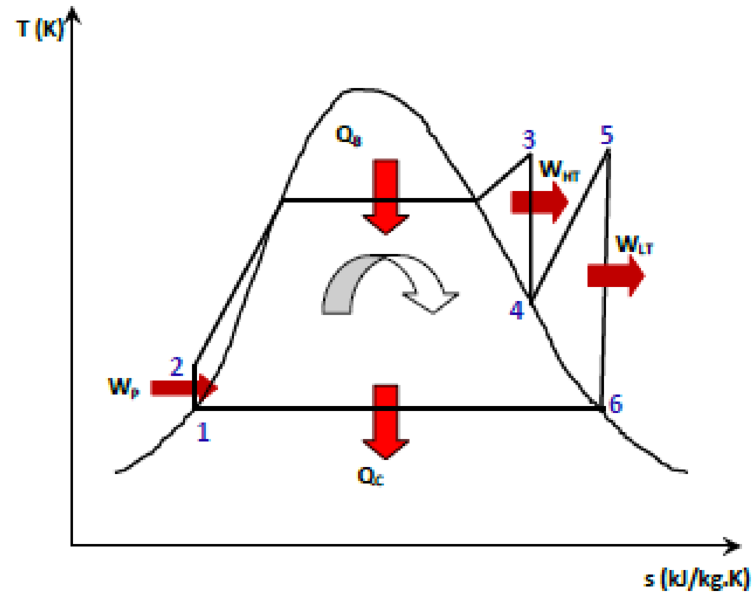
Though the **efficiency increases by about 2%,** for an **increase in pressure of 100 bar**, increased boiler pressure **decreases the dryness fraction** at turbine exit $\Rightarrow$ **Not desirable**
$\Rightarrow$ **For operational reasons, the dryness fraction should be more than 0.9**

Hence operating the cycle at **very high pressure** in a **simple Rankine cycle** is **not very beneficial**

Increased boiler pressure together with **reheat** results in **better performance**

Rankine cycle with single reheat





$$W_{HT} = \dot{m}(h_3 - h_4) \qquad W_{LT} = \dot{m}(h_5 - h_6)$$

$$Q_B = \dot{m}[(h_3 - h_2) + (h_5 - h_4)] \qquad W_P = \dot{m}(h_2 - h_1)$$

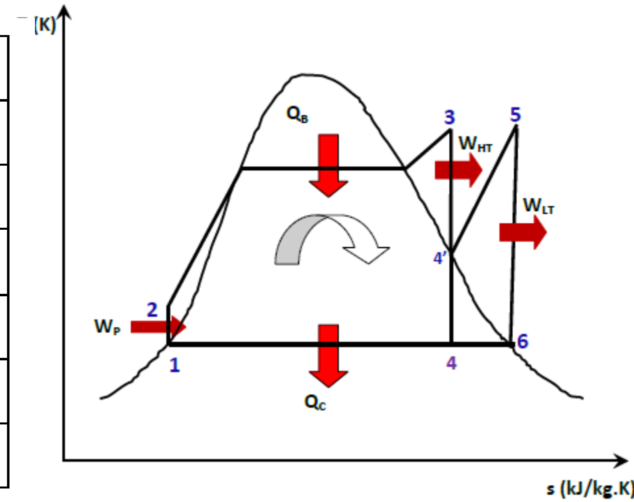
$$\boxed{\eta_{th,rh} = \frac{(h_3 - h_4) + (h_5 - h_6) - (h_2 - h_1)}{(h_3 - h_2) + (h_5 - h_4)}}$$

Performance comparison with and without reheat

Given data: Boiler pressure = **163 bar**, Condenser pressure = **0.07 bar**

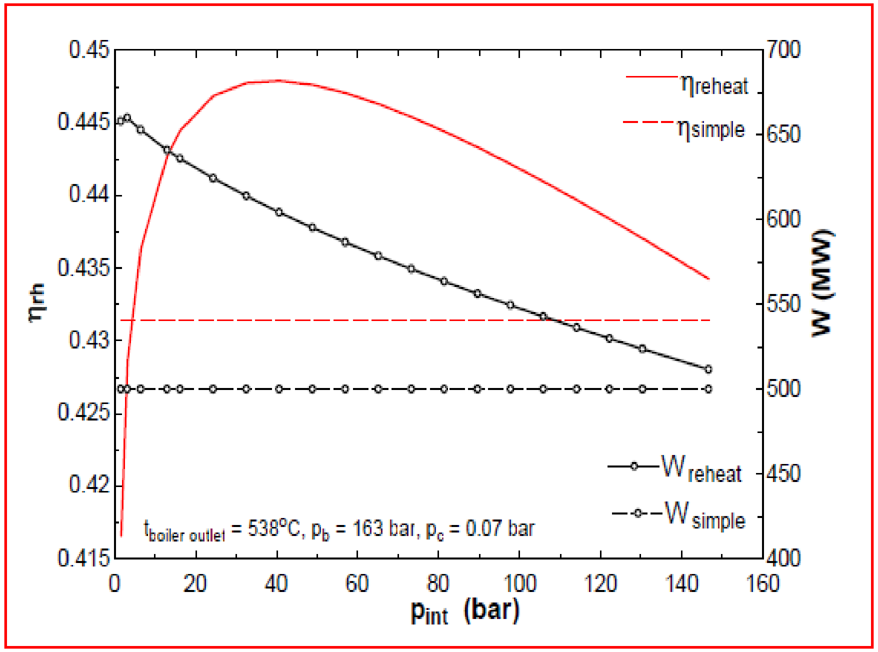
Highest temperature of heat addition (T_3) = **538°C, x=1.0 at HT exit**

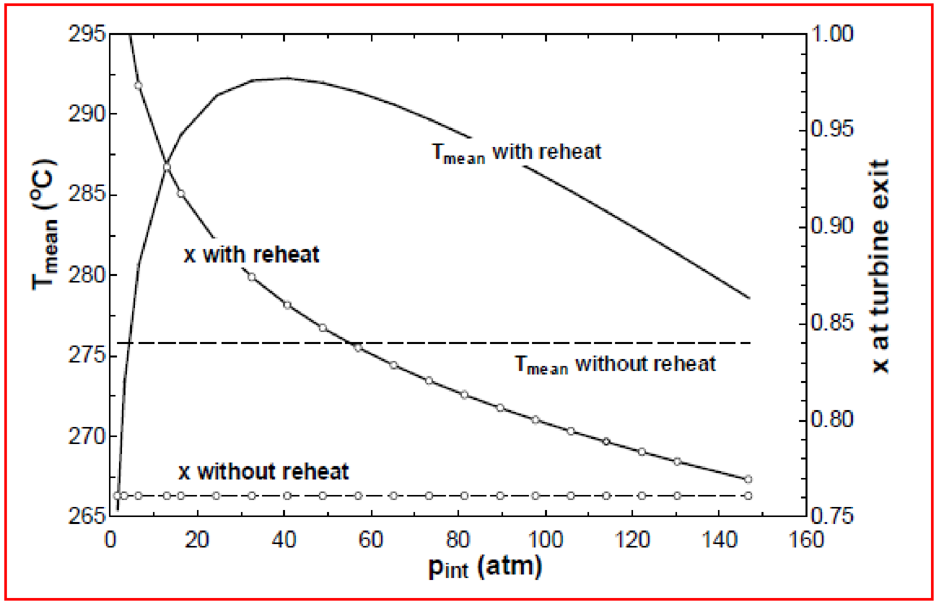
State	1	2	3	4′	5	4	6
p (bar)	0.07	163	163	40.77	40.77	0.07	0.07
T (°C)	39.01	39.48	538	251.5	538	39.01	39.01
ρ (kg/m³)	992.5	999.4	-	-	-	-	-
h (kJ/kg)	163.4	179.7	3401	3004	3532	1995	2233
s (kJ/kg.K)	0.559	0.559	6.428	6.428	7.19	6.428	7.19
x	0	-	-	1	-	0.7607	0.8596



Parameter	Simple cycle	Reheat cycle
Net power output (MW)	500	604.2
Steam flow rate (kg/s)	359.8	359.8
Turbine power (MW)	505.9	610.1 (143.1 + 467.1)
Pump Power (MW)	5.9	5.9
Work Ratio	0.9884	0.9904
Thermal efficiency (%)	43.14	44.79
T_{mean} (°C)	275.8	292.3
2nd law efficiency	70.11	72.79

Performance comparison with and without reheat





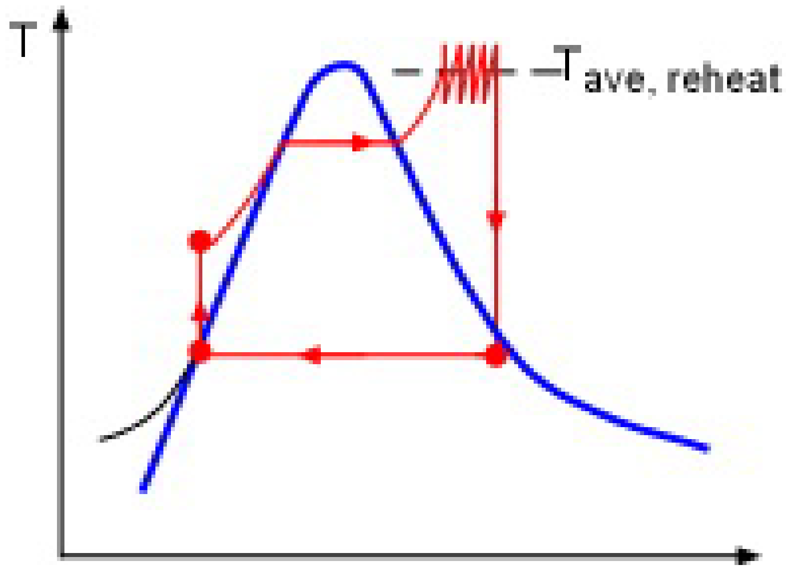
Given data:

Boiler pressure = **163 bar**, Condenser pressure = **0.07 bar**

Highest temperature of heat addition (T_3) = 538°C

Results show that for given boiler and condenser pressures and heat addition temperature, there is an optimum intermediate pressure at which the efficiency reaches a maximum . A general rule of thumb is : $p_{opt,int} = p_{boiler}/(4 \text{ to } 6)$

By **increasing the number of reheat stages**, the heat addition process can approach **an isothermal heat addition** process



However, due to economic, design and operational regions, no. of reheat stages is limited to 2

Possibility of employing superheat

Whether **superheat** is possible or not **depends** upon the **type of external heat transfer fluid** used in the boiler and the **boiler pressure**

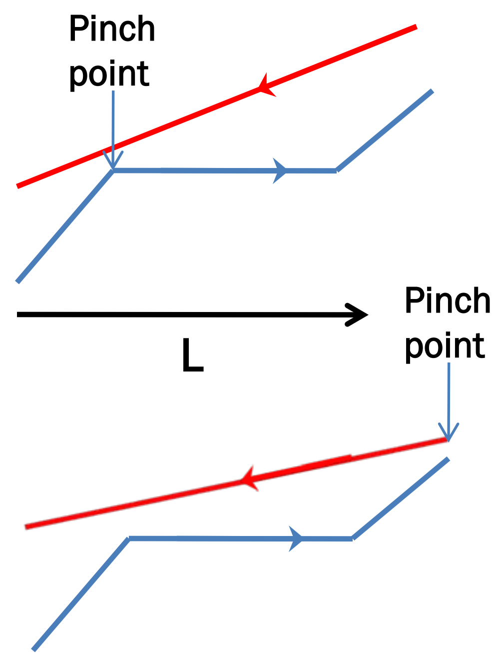
For **higher performance**, generally a near **counterflow** type arrangement is used in the boiler of the power plant

The **point** where the **temperature difference** between the **external fluid and steam** reaches a **minimum value** is called as a **pinch point**

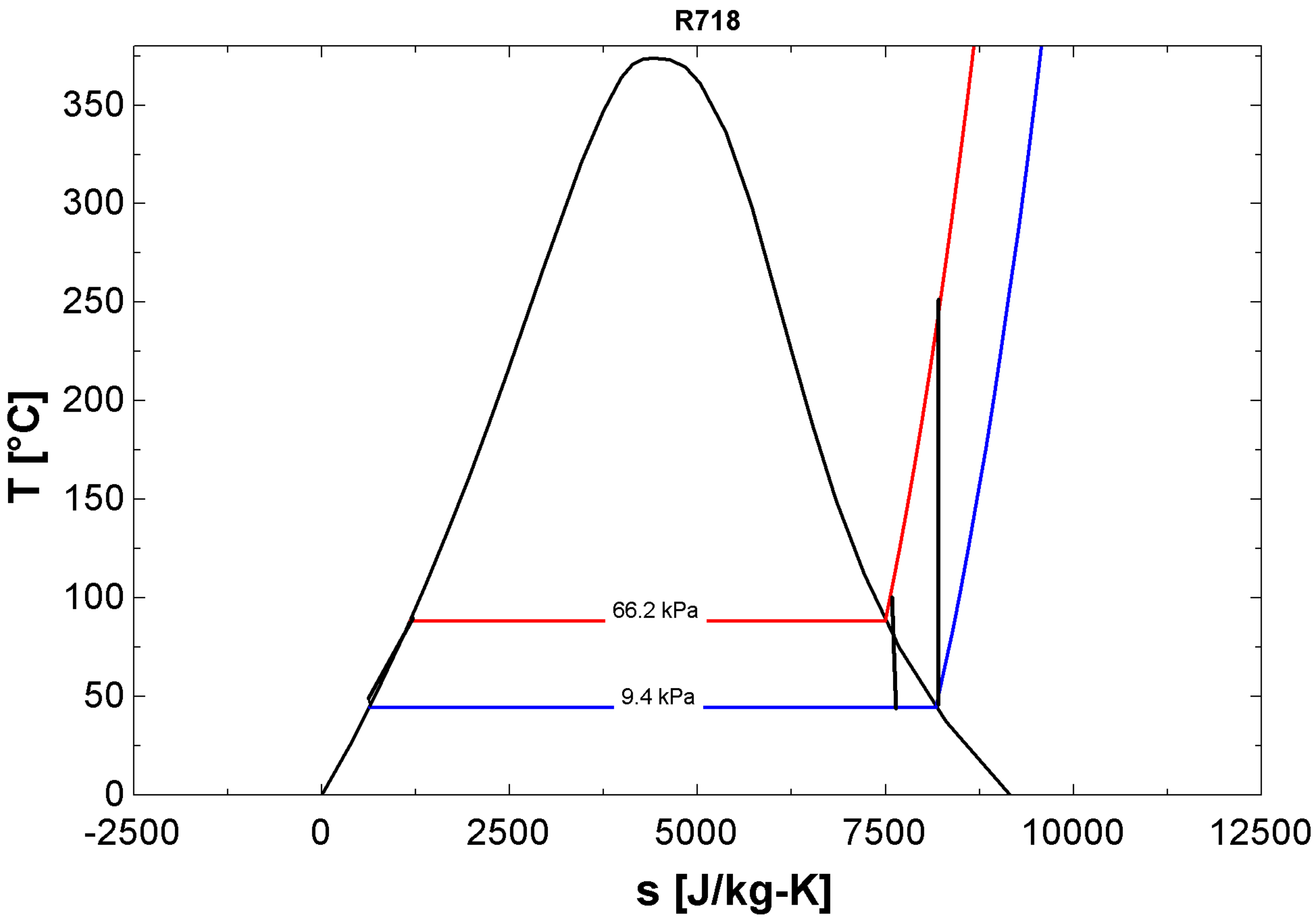
For **a given pinch point temperature difference**, the slope of the external fluid temperature depends upon its thermal capacity, i.e.,

$$\left(\frac{dT}{dL}\right)_{ext} = \frac{1}{(\dot{m}c_p)_{ext}}$$

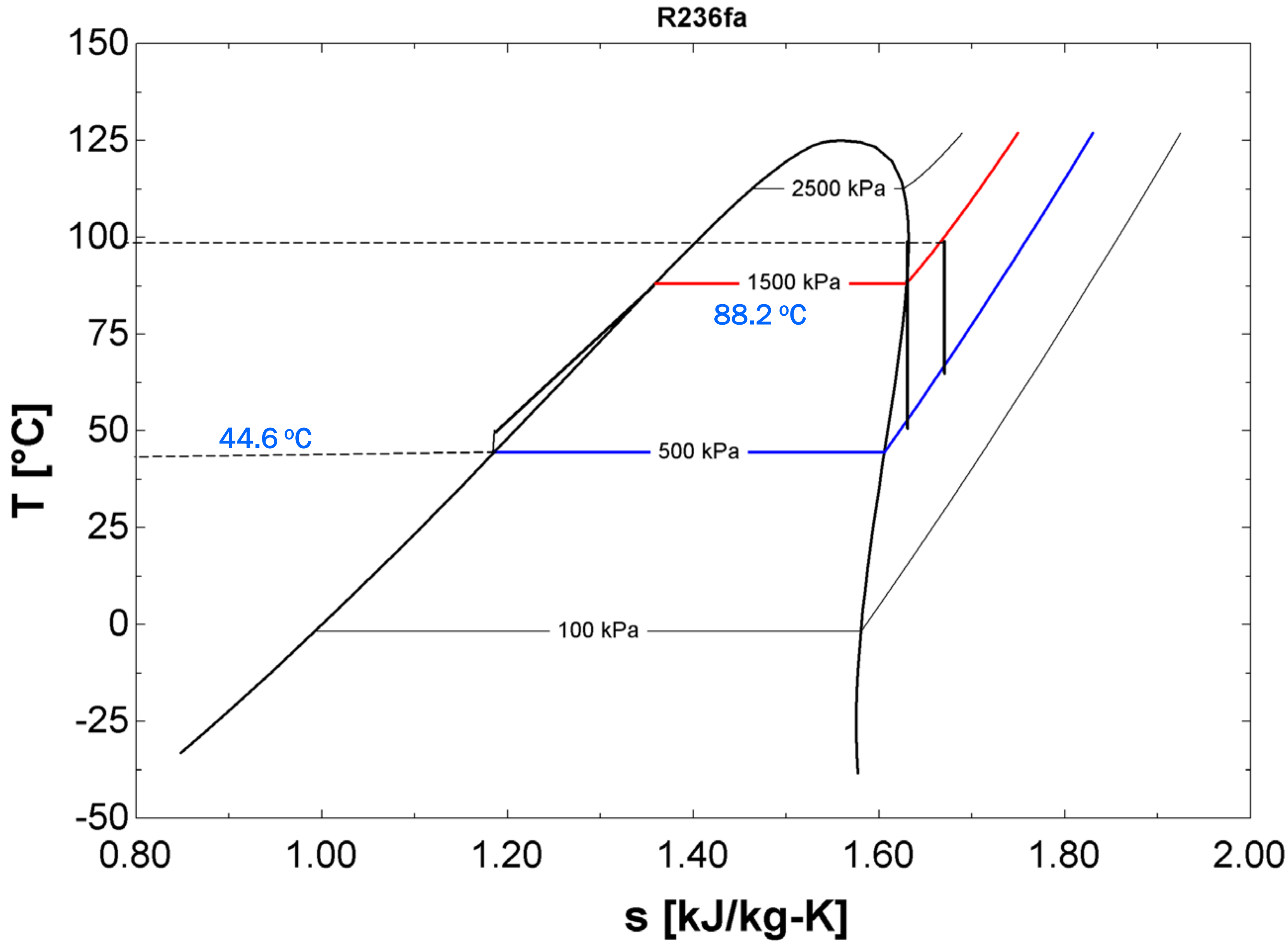
In systems where either the **mass flow rate** of the external fluid and/or its **specific heat** is very large, then the slope is small.
This puts a **constraint** on the amount of **superheat** that can be employed **for a given heat transfer rate.**



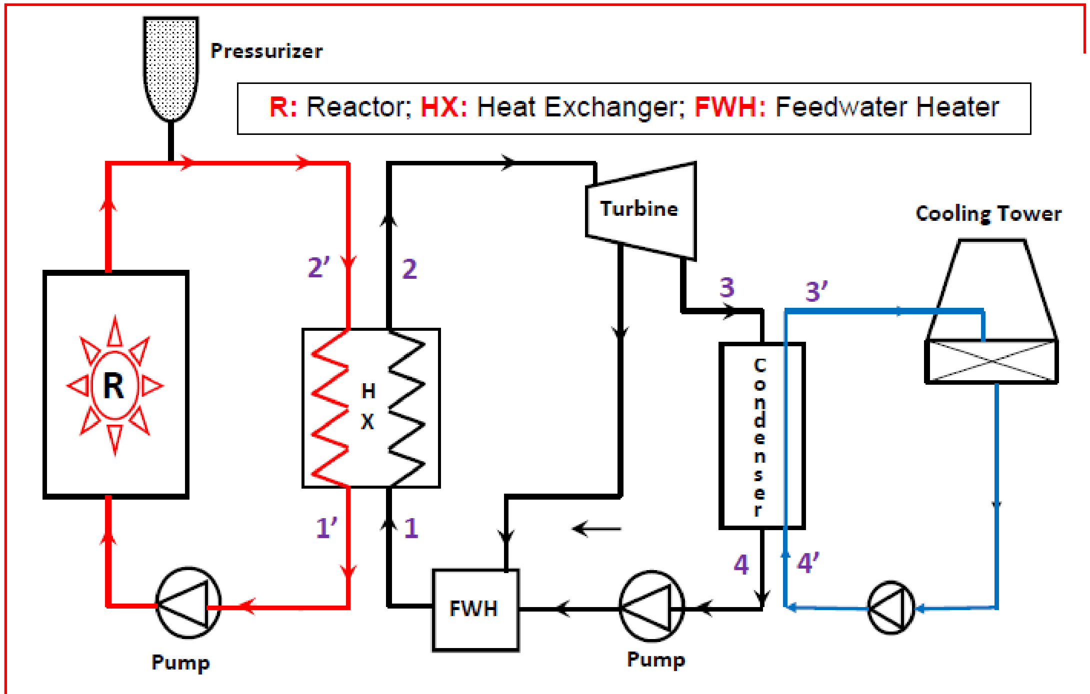
Cycles with low heat source temperature



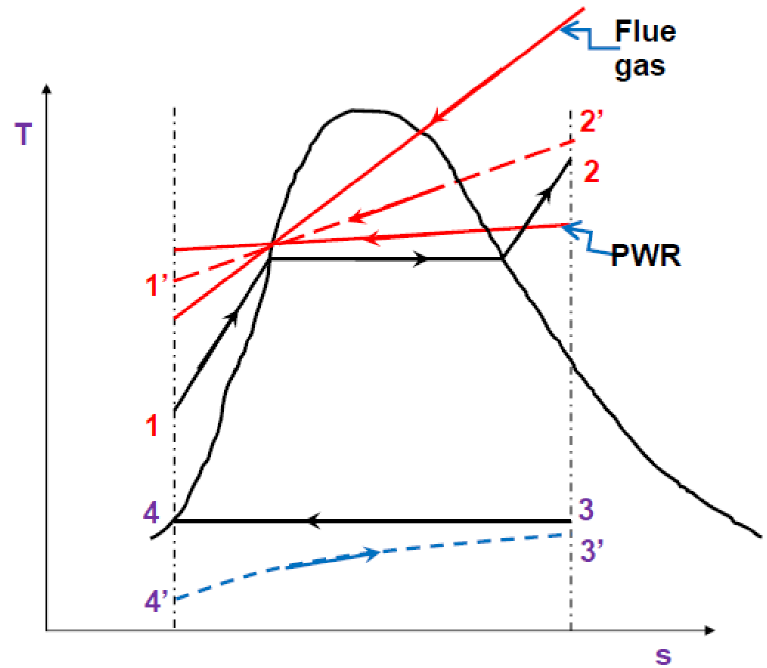
Working fluid: R236fa



Pressurized water reactor system



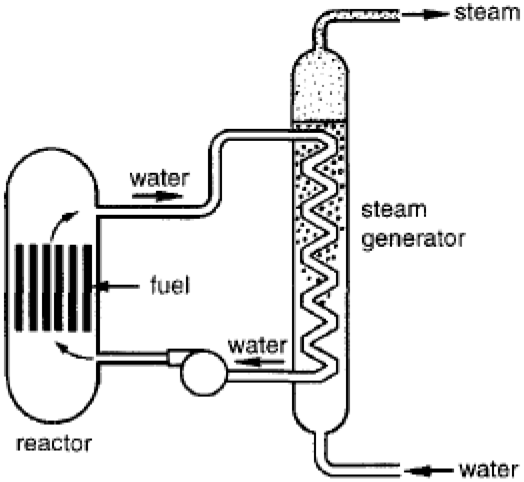
Schematic of a Steam power plant based on Pressurized Water

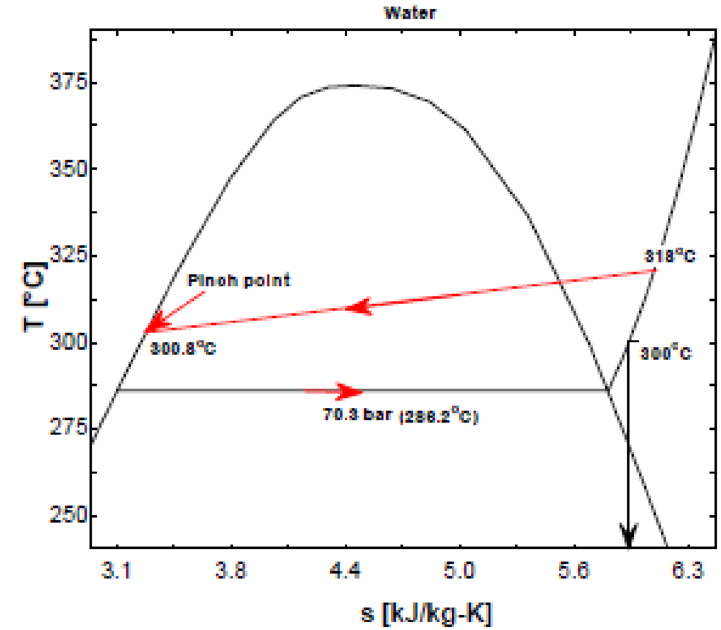


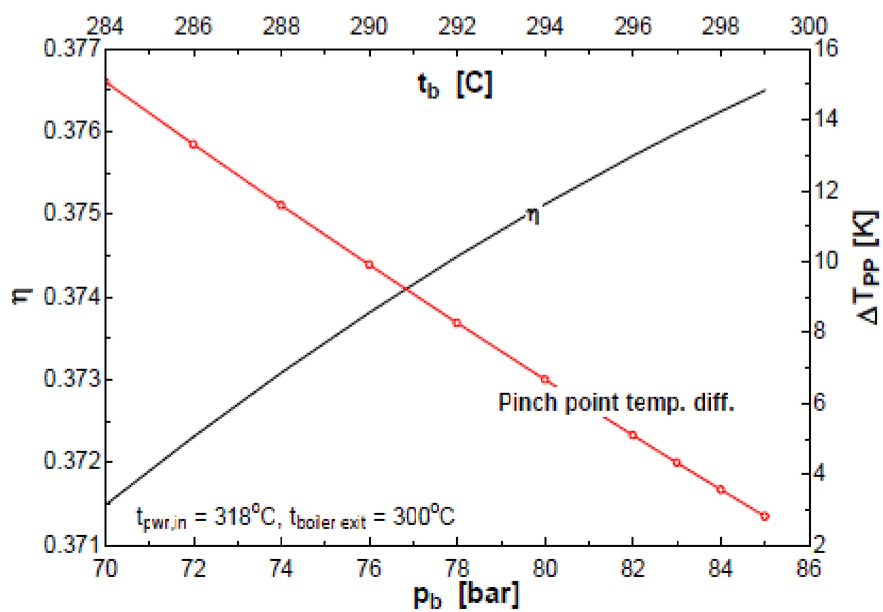
In a **PWR based power plant**, due to operational constraints the **maximum temperature** of the pressurized water **is limited.** In addition, since the **c_p** value of **water** is **very high**, the **temperature variation is small** ⇒ Limited scope for superheat ⇒ steam at turbine inlet is close to saturation

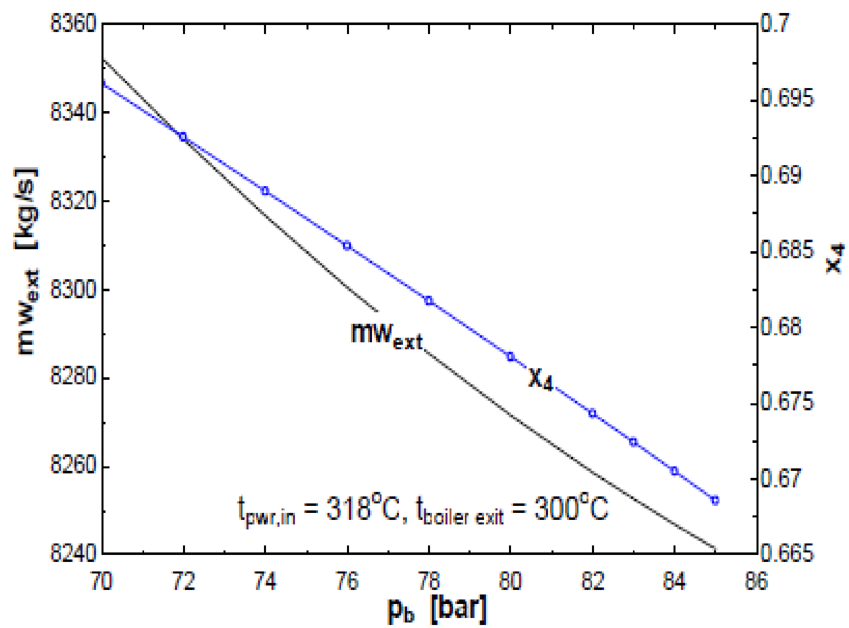
For the **same pinch point temperature difference**, in a **gas cooled reactor** or in a **conventional coal based power plant**, the **temperature gradient is very steep**, hence it is **possible to employ superheat/reheat** in these systems

Effect of pressure for plants with low external temperature variation in boiler









- Studies on actual power plants show that:
- **Effect of boiler pressure:**

1. Upto about 250 bar pressure, the net efficiency increases by 0.01% per bar and between **250 to 300 bar**, the improvement is about 0.008% per bar.
2. There is not much gain with further increase in pressure

- **Effect of boiler exit/reheater temperature:**

1. For temperature range of upto 600ºC, the net efficiency increases by about 0.02% per ºC rise in turbine inlet temperature
2. In the temperature range of 600 to 700ºC, the net efficiency increases by about 0.016% per ºC in turbine inlet temperature

- The maximum allowable temperature and pressure are limited by available construction materials

Worked out example: Steam power plant connected to a PWR

Given:

W_{net} = **500 MW**

Condenser Pressure = **0.07 bar**, boiler pressure = **75 bar**

Inlet temperature of heat source (pressurized water) = **318^{o}C**

Outlet temperature of heat source (pressurized water) = **289^{o}C**

Temperature difference between heat source (inlet) and boiler exit = **18 K**

Condenser water inlet temperature = **30^{o}C**

Condenser water outlet temperature = **35^{o}C**

Find:

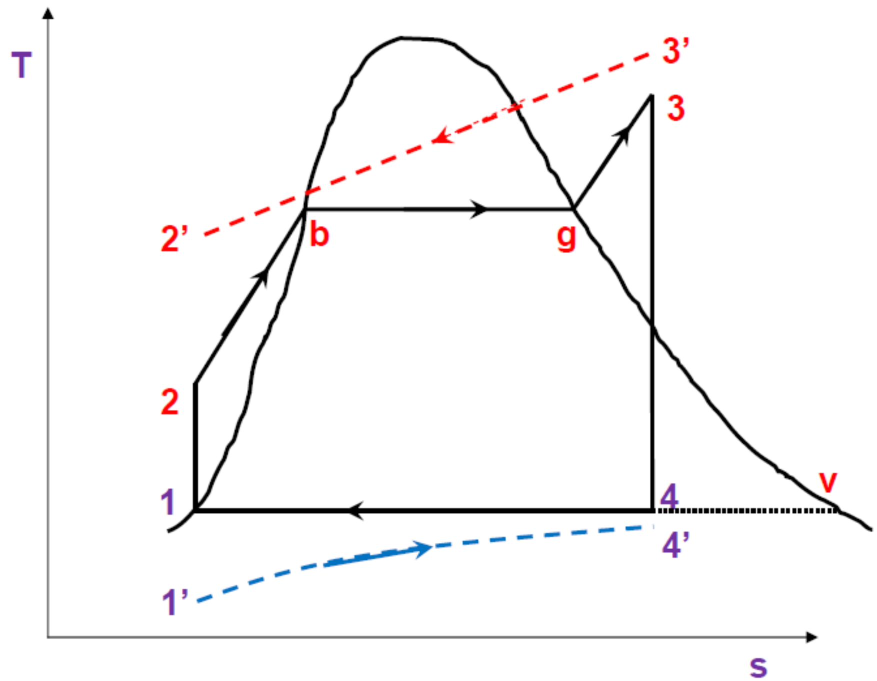
1) Thermal efficiency of the plant

2) Flow rates of steam, pressurized water and cooling water in condenser

3) Pinch point location and the temperature difference at pinch point

4) Entropy generation (total, condenser and boiler)

5) Carnot efficiency



Property data:

	1	2	b	g	3	4	v
t (°C)	39.01	39.22	290.6	290.6	300	39.01	39.01
p (bar)	0.07	75	75	75	75	0.07	0.07
h (kJ/kg)	163.4	170.9	1292	2765	2812	1818	2572
s (kJ/kg.K)	0.559	0.559	?	?	5.861	5.861	8.274

Results:

1) Thermal efficiency of the plant = 37.35 %

2) Flow rates:

   a) Steam = 506.9 kg/s

   b) Pressurized water = 8308 kg/s

   c) Cooling water in condenser = 40136 kg/s

3) Temperature difference at pinch point = 10.74 K (at sat. liquid)

4) Entropy generation:

   a) Total = **422.3 kW/K** $\Rightarrow$ Lost work = 129 MW

   b) In condenser* = 57.9 kW/K

   c) In boiler* = 364.4 kW/K

5) Carnot efficiency = 47%

* From entropy balance across condenser and boiler

Distribution of total entropy generation

	Heat transfer rate, kW	% of total	Entropy generation (kW/K)	% of total
Total	-	-	423.0	100
Condenser	838342	100	58.6	13.9
Boiler (total)	1338842	100	364.4	86.1
Subcooled region of boiler	568378	42.4	320.4	75.7
Saturated region of boiler	746534	55.8	42.4	10.0
Superheated region of boiler	23930	1.8	1.6	0.4

Entropy generation rate in a particular zone $\dot{S}_{gen}$ (kW/K) is given by:

$$\dot{S}_{gen} = \left(\dot{m}c_p\right)_{ext} ln\left(\frac{T_{ext,out}}{T_{ext,in}}\right) + \dot{m}_{wf}\left(s_{out} - s_{in}\right)$$

Where:

$\left(\dot{m}c_p\right)_{ext}$ is the thermal capacity of the external fluid (kW/K)

$T_{ext,out}$ is the exit temperature of the external fluid (K)

$T_{ext,in}$ is the inlet temperature of the external fluid (K)

$\dot{m}_{wf}$ is the mass flow rate of the working fluid (kg/s)

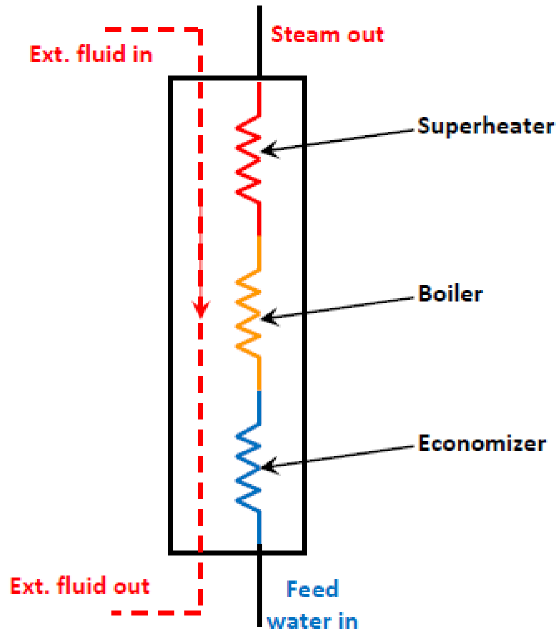
s_{out} is the exit entropy of the working fluid, (kJ/kg.K)

s_{in} is the inlet entropy of the working fluid, (kJ/kg.K)

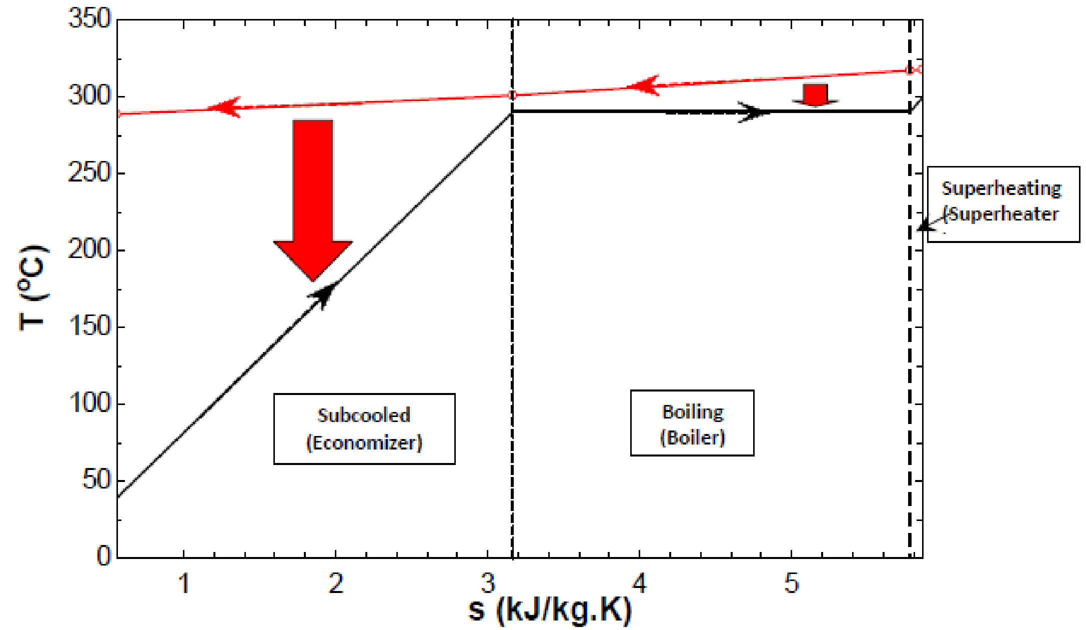
The example shows that **of the total entropy generation** in the power plant, almost **76%** is generated in the sub-cooled region of the steam generator itself, even though the **heat transfer rate** in this region is about **42%** of the total input.

This is obviously **due to heat transfer** taking place over a very **large temperature difference** in this region.

This remains true for all the external heat sources (e.g. PWR or flue gas based)



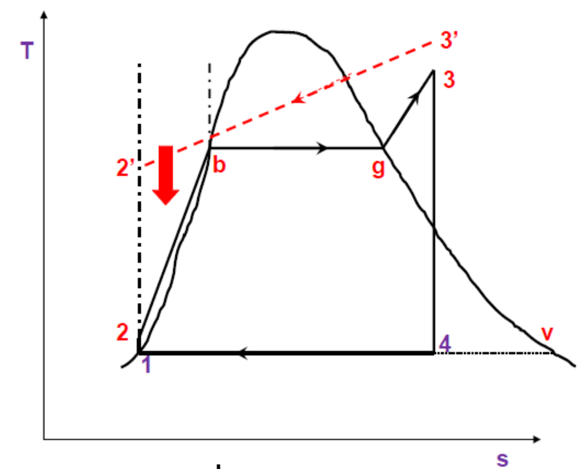
The 3 zones in a steam generator



Temperature profile in steam generator

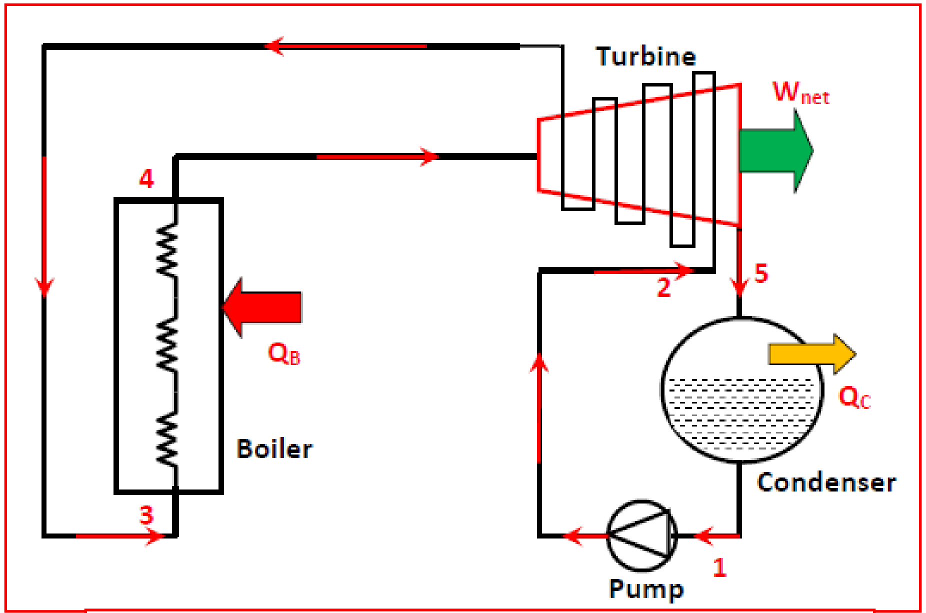
The concept of regenerative feedwater heating

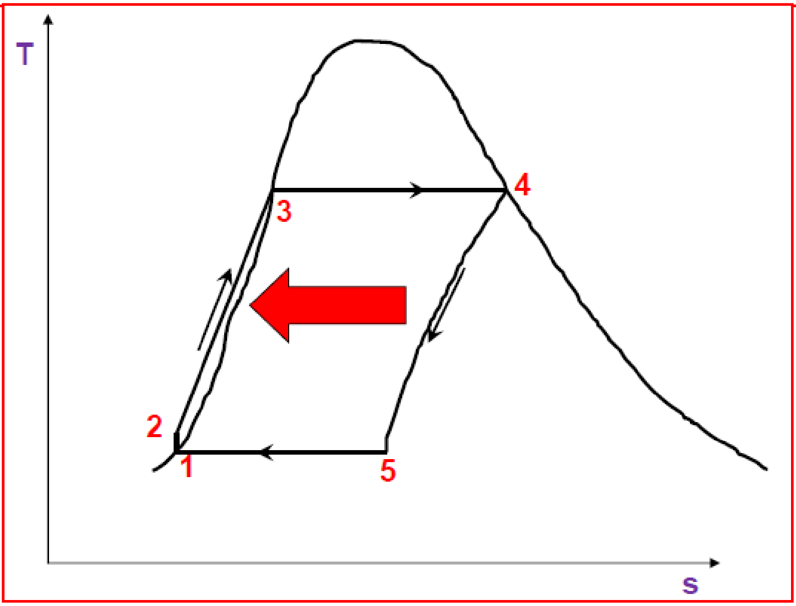
- Analysis of simple Rankine cycle shows that the **irreversibility due to heat transfer is very high in the subcooled liquid region** due to the **large temperature difference** between the heat source and the working fluid (feed water)
- Ideally, **this irreversibility can be eliminated** if the feed water enters the boiler **at point b, instead of point 2** ⇒ **Elimination of subcooled region**
- This can be done using **regenerative feed water heaters**
- Conceptually, in **regenerative feed water heat**ing, the **feed water is heated internally** by **extracting heat** from **the expanding steam in the turbine**



Ideal, regenerative feed water heating

- **Under ideal conditions**, the feed water from the pump (2) is **heated reversibly** by the steam that is expanding in the turbine such that it is **saturated** at the **inlet to the boiler (3)** ⇒ The **economizer** is integrated with the **turbine**!
- The **resulting cycle** will have **completely isothermal heat addition** and **heat rejection**
- ⇒ **If** there are **no other** internal or external **irreversibilities**, then the **efficiency** of this cycle is **same** as that of a **Carnot cycle**!
- However, it is impossible in practice to construct such a system in which there is reversible heat transfer from the high speed vapour flowing through the turbine blades to the feed water
- In addition, the amount of **liquid that forms during the expansion** process will be **unacceptably high**!



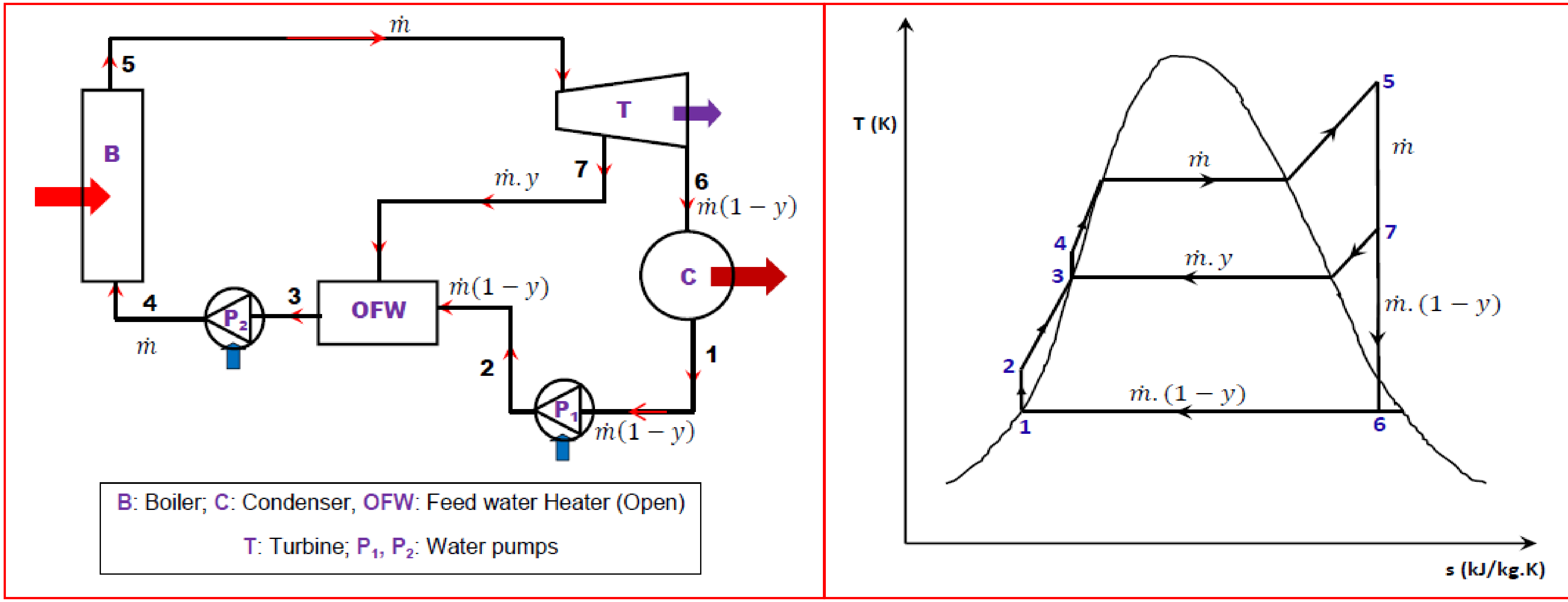


Saturated Rankine cycle with ideal regeneration

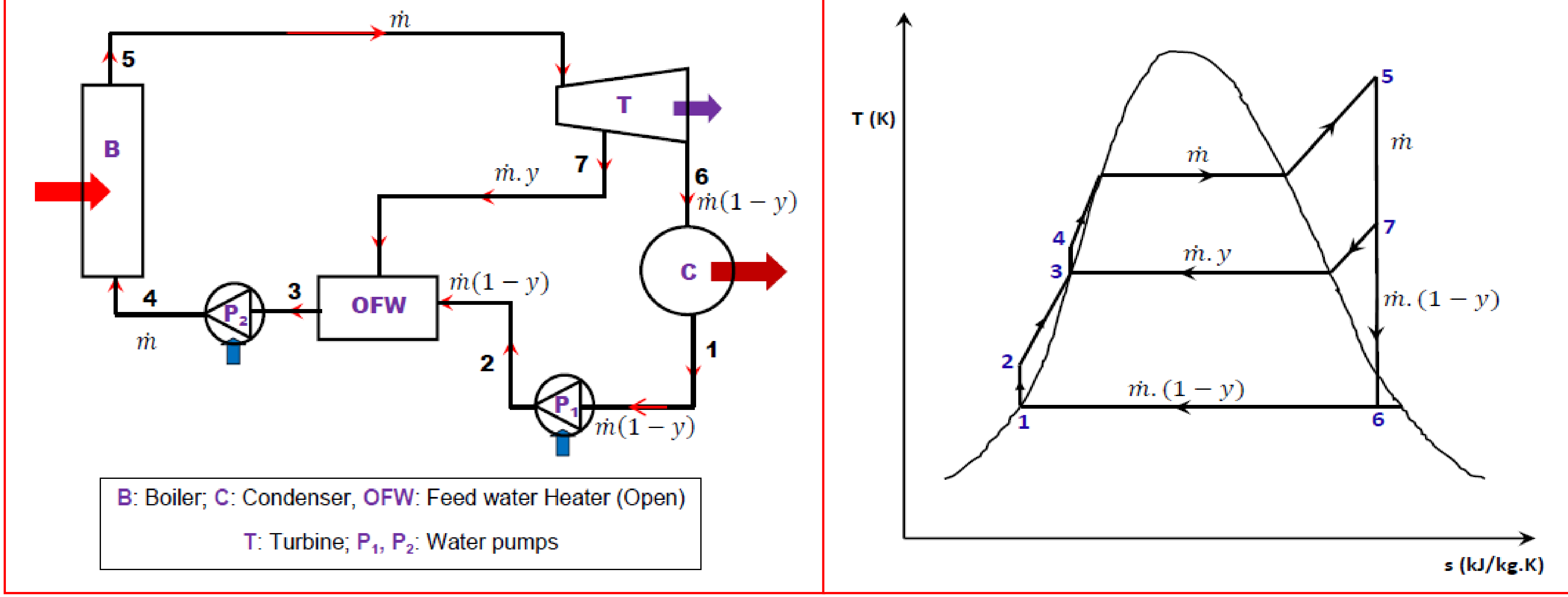
Regenerative feedwater heating

- Since it is **not possible** to **heat** the feed water **reversibly** by direct exchange of heat with the expanding steam in the turbine, **in practice, separate feedwater heaters** are used in all steam power plants
- Unlike ideal regeneration, **use of feedwater heaters does not completely eliminate** the **external irreversibility** but minimizes it
- Depending upon the type, **feedwater heaters** can be **classified as:**
  - **Open or direct contact type feedwater heaters**
  - **Closed or indirect contact type feedwater heaters**
    - Drain cascaded backward
    - Drain cascaded forward
- **In actual power plants,** the feedwater is heated internally using **as many as 5 to 6** feedwater heaters, out of which at least **one** is an open feedwater heater.

System with one open or direct contact type feedwater heater



1. **No. of pumps required** = No. of Open Feedwater heaters + 1
2. The **pressure** at the **exit** of low stage pump **P_1** (2) cannot be higher than the pressure at which steam is extracted (7), otherwise there will be reverse flow of condensate water into turbine
3. Mass fraction of extraction steam (y) should be such that the state of the mixture at the exit of the OFW (3) is either saturated or subcooled liquid.
4. If the extraction steam flow rate is more than required, then there will be:
    1. Loss of turbine power, and
    2. Inlet condition for high stage pump (P_2) will be in 2-phase region



From 1st and 2nd law of thermodynamics across each component:

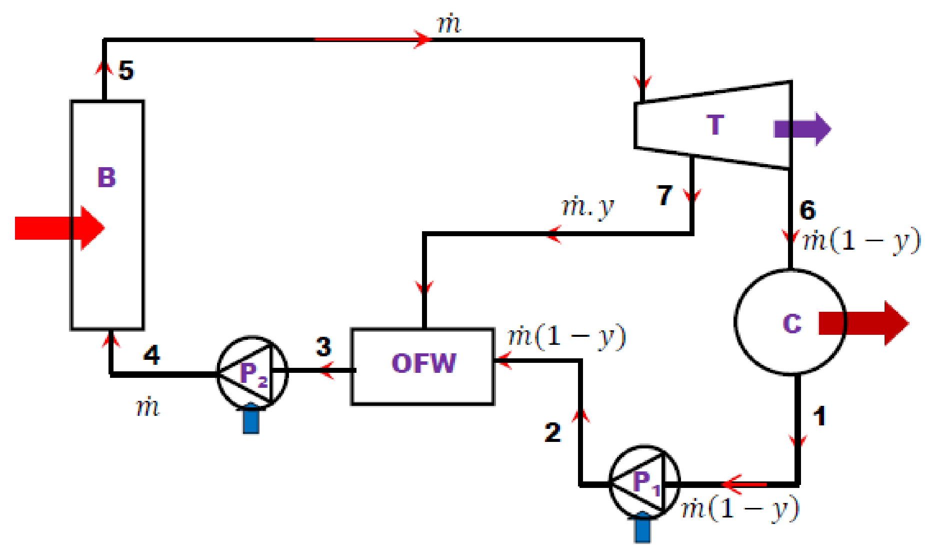
$$W_{turbine} = \dot{m}\{y(h_5 - h_7) + (1-y)(h_5 - h_6)\} = \dot{m}\{h_5 - y.h_7 - (1-y)h_6\}$$

$$W_{pump} = \dot{m}\{(h_4 - h_3) + (1-y)(h_2 - h_1)\}$$

$$Q_{boiler} = \dot{m}\{(h_5 - h_4)\}$$

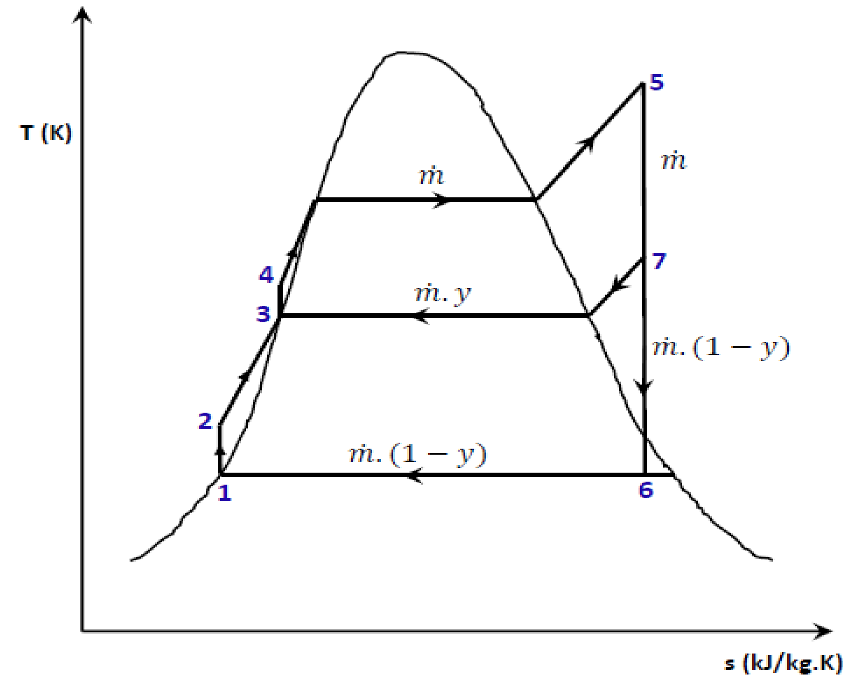
$$Q_{condenser} = \dot{m}\{(1-y)(h_6 - h_1)\}$$

Open Feed water heater (OFW): $\boxed{\dot{m}.h_3 = \dot{m}\{y.h_7 + (1-y)h_2\}}$



B: Boiler; C: Condenser, OFW: Feed water Heater (Open)

T: Turbine; P_1, P_2: Water pumps



From 1st and 2nd law of thermodynamics across each component:

$$\eta_{thermal} = \frac{W_{turbine} - W_{pump}}{Q_{boiler}} = \frac{Q_{boiler} - Q_{condenser}}{Q_{boiler}}$$

$$S_{gen,total} = \underbrace{\{S_{gen,boiler} + S_{gen,condenser}\}}_{\text{External irreversibility}} + \underbrace{S_{gen,OFW}}_{\text{Internal irreversibility}}$$

Open Feed water heater (OFW): $\boxed{S_{gen,OFW} = \dot{m}\{s_3 - y.s_7 - (1-y)s_2\}}$

Worked out example: Steam power plant with an open feedwater heater

Given:

W_{net} = **500 MW**

Pressures: Condenser = **0.07 bar**, Boiler = **75 bar**, Feedwater heater = **35 bar**

Heat source: Inlet temperature = **318°C**, Outlet temperature = **289°C**

Temperature difference between heat source (inlet) and boiler exit = **18 K**

Heat sink: water inlet temperature = **30°C** , water outlet temperature = **35°C**

Find:

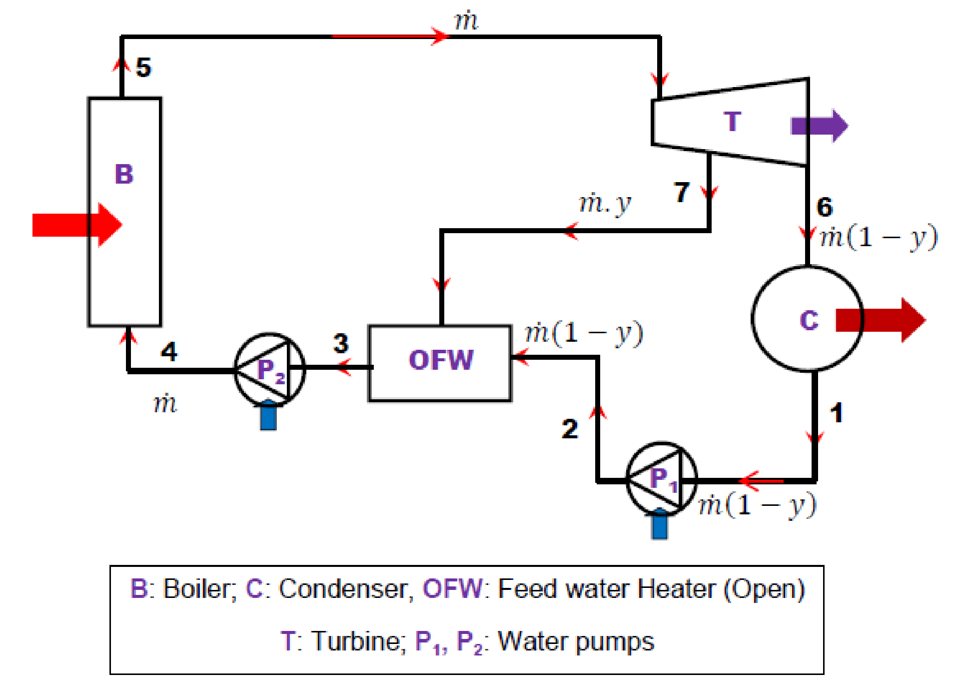
1) Thermal efficiency of the plant

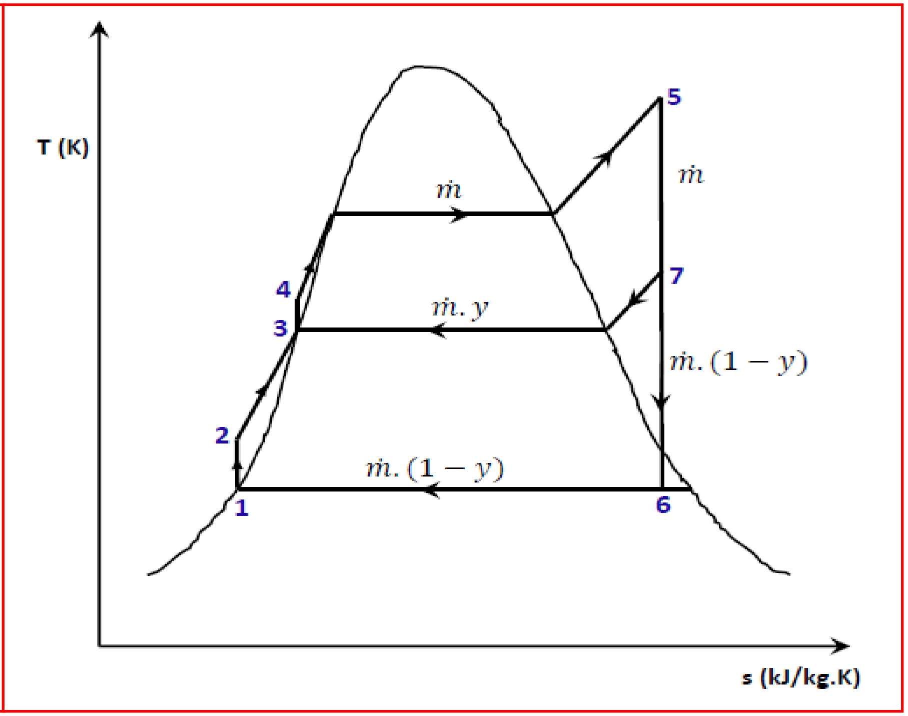
2) Turbine and pump power, heat transfer in boiler and condenser

3) Mass flow rate of steam through boiler and steam extraction fraction

3) Entropy generation (total, condenser, boiler and feedwater heater)

Worked out example (contd.)





State point	1	2	3	4	5	6	7
p (bar)	0.07	35	35	75	75	0.07	35
t (^oC)	39.01	39.11	242.6	243.6	300	39.01	242.6
h (kJ/kg)	163.4	166.9	1050	1055	2812	1818	2667
S (kJ/kg.K)	0.559	0.559	2.725	2.725	5.861	5.861	5.861

Worked out example: Results

1. Thermal efficiency, $\eta_{thermal}$ **= 39.09 %**

   (37.35 % without regeneration)

2. Total Turbine output, $W_{Turbine}$ **= 505.243 MW**

   Total Pump input, W_{Pump} **= 5.243 MW**

   Boiler input, Q_{boiler} **= 1279 MW**

   Condenser heat rejection, Q_{cond} **= 779 MW**

3. Mass flow rate of steam through boiler = **727.6 kg/s**

   Fraction of extracted steam, y **= 0.3531**

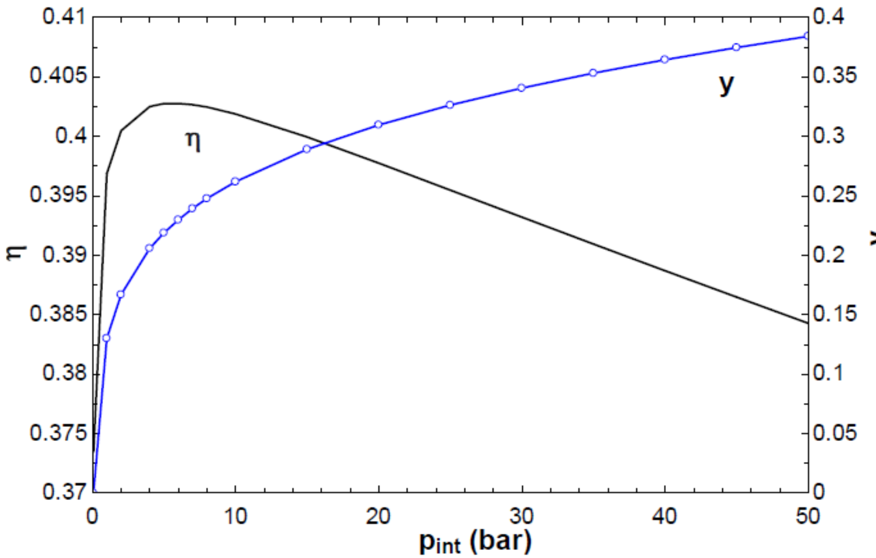
4. Entropy generation: Total: **330.9 kW/K** (422.3 kW/K without OFW)

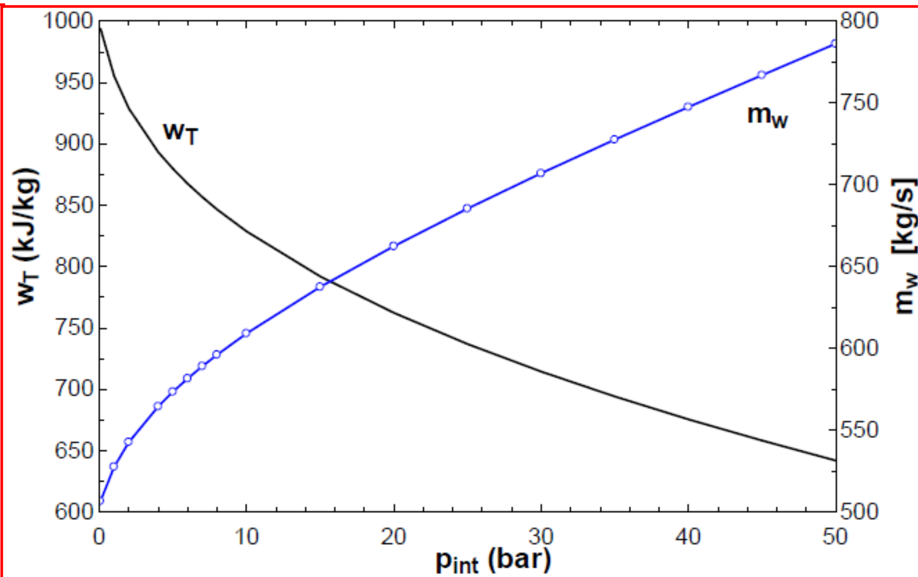
   Boiler: **62.55 kW**

   Condenser: **54.44 kW/K**

   Feedwater heater: **213.9 kW/K**

Effect of steam extraction pressure





As steam extraction is varied from condenser pressure:

Efficiency increases, reaches a peak and then starts decreasing

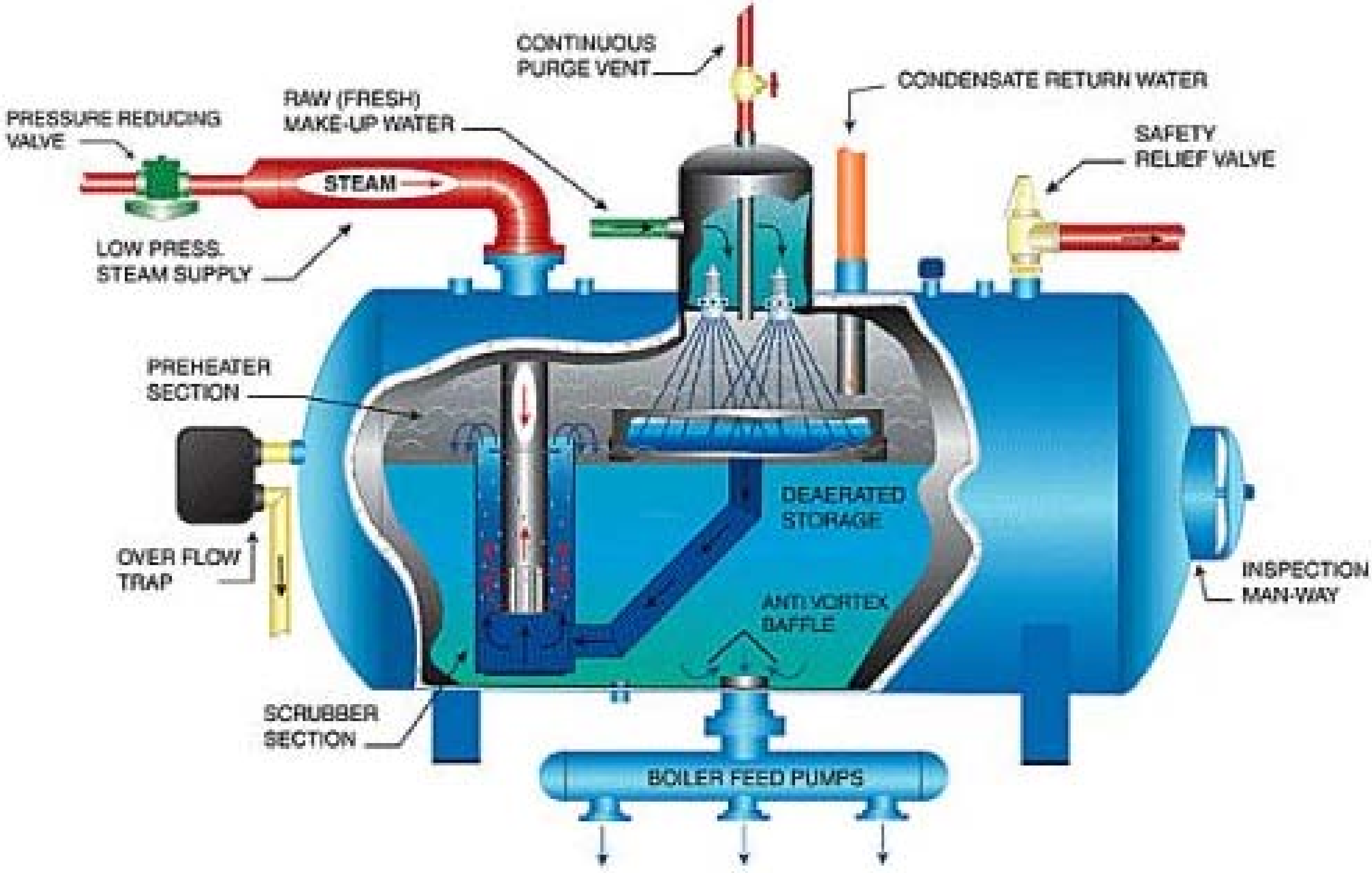
Fraction of steam extracted increases ⇒ **Specific turbine work output decreases**

⇒ Mass flow rate of working fluid (water) increases

It can be shown that the **efficiency is maximum** when the saturation temperature corresponding to the intermediate pressure is midway between the boiler and condenser pressures, i.e., $t_{sat}(p_{int}) \approx (t_{boiler} + t_{condenser})/2 \approx 165^oC$ ($p_{sat} \approx 7$ bar)

In general, in conventional power plants, **only one open feed water heater** is used, which also acts as a deaerator – **Hence it is also called as DA heater**

Open feedwater heater cum deaerator - inner details



www.crazyengineers.com

Solution with minimum data

- In an open feedwater (OFW) heater, saturated steam at 14.27 bar extracted from turbine is mixed with subcooled liquid from condensate pump. The state of water at the exit of OFW is saturated. Using the minimum property data provided below, find:

1) Fraction of steam that is extracted from turbine for OFW
2) Entropy generated in feedwater heater in kJ/kg.K

Given: a) Condenser pressure = 0.08 bar,

b) Latent heat of vapourization at 0.08 bar = 2402 kJ/kg

c) Latent heat of vapourization at 14.27 bar = 1957 kJ/kg

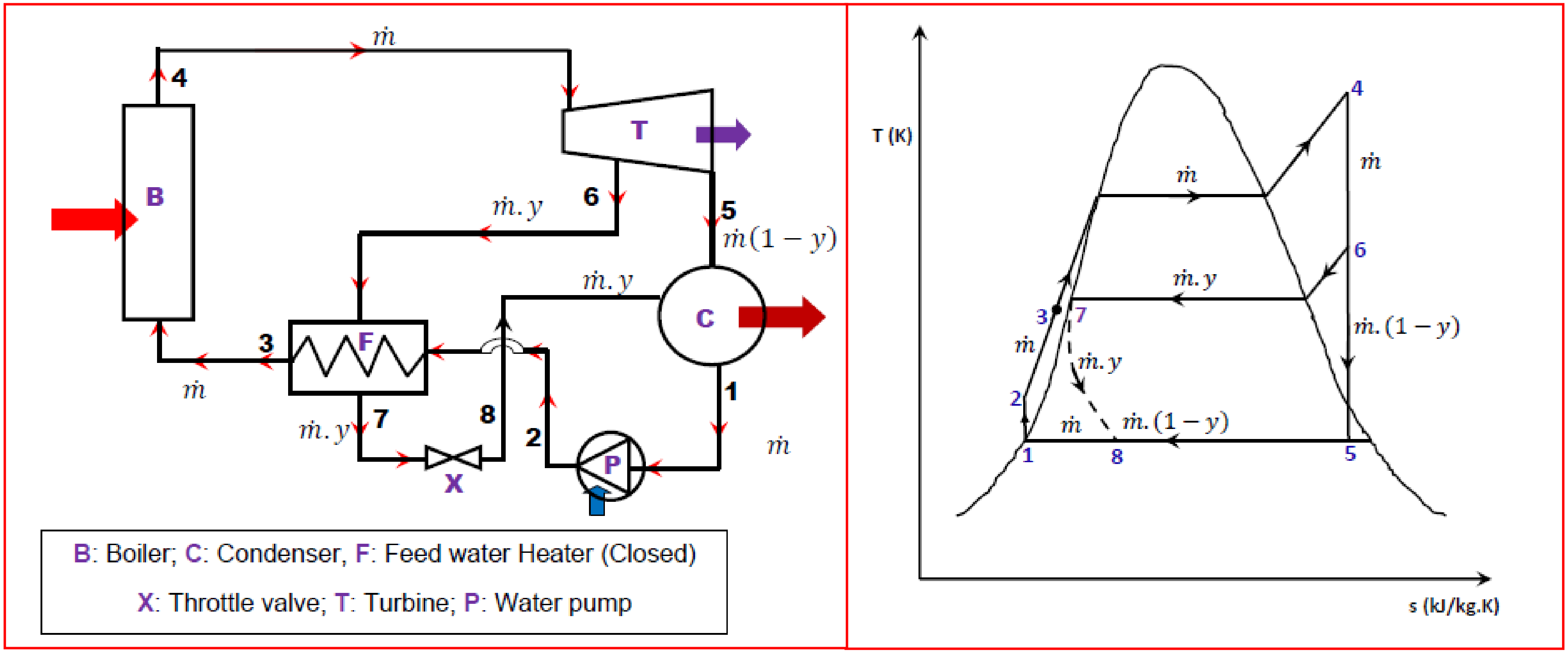
d) mean specific heat of liquid water = 4.325 kJ/kg

e) density of liquid water = 991.6 kg/m^3

Antoine equation for saturation pressure of water:

$$\ln(\boldsymbol{p_{sat}}) = 16.54 - \left(\frac{3985}{\boldsymbol{T} - 39}\right); \quad \boldsymbol{p_{sat}} \text{ in } \mathbf{kPa} \textit{ and } T \text{ in K}$$

System with one closed feed water heater (drain backward)



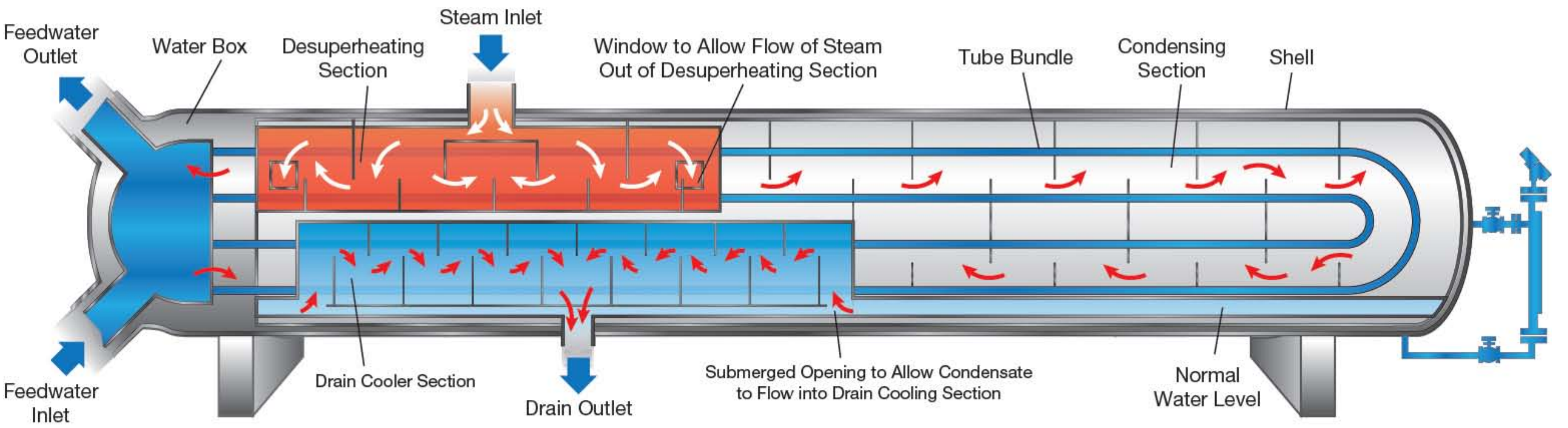
1. This is the **simplest and most commonly used** type of feedwater heater
2. This feedwater heater is **similar** to a **shell-and-tube type condenser**, wherein the **extracted steam condenses** in the shell, while the **feedwater** flowing through the tubes is **sensibly heated**
3. Depending upon the **condition of extracted steam** (6):

   t_3 can be equal to, higher or lower than t_{sat} (p_6)

4. Only a **single feedwater pump is required** in this system $\Rightarrow$ extracted steam condensing in the feedwater heater is fed back to the main condenser through a throttle valve = **Additional internal irreversibility!**

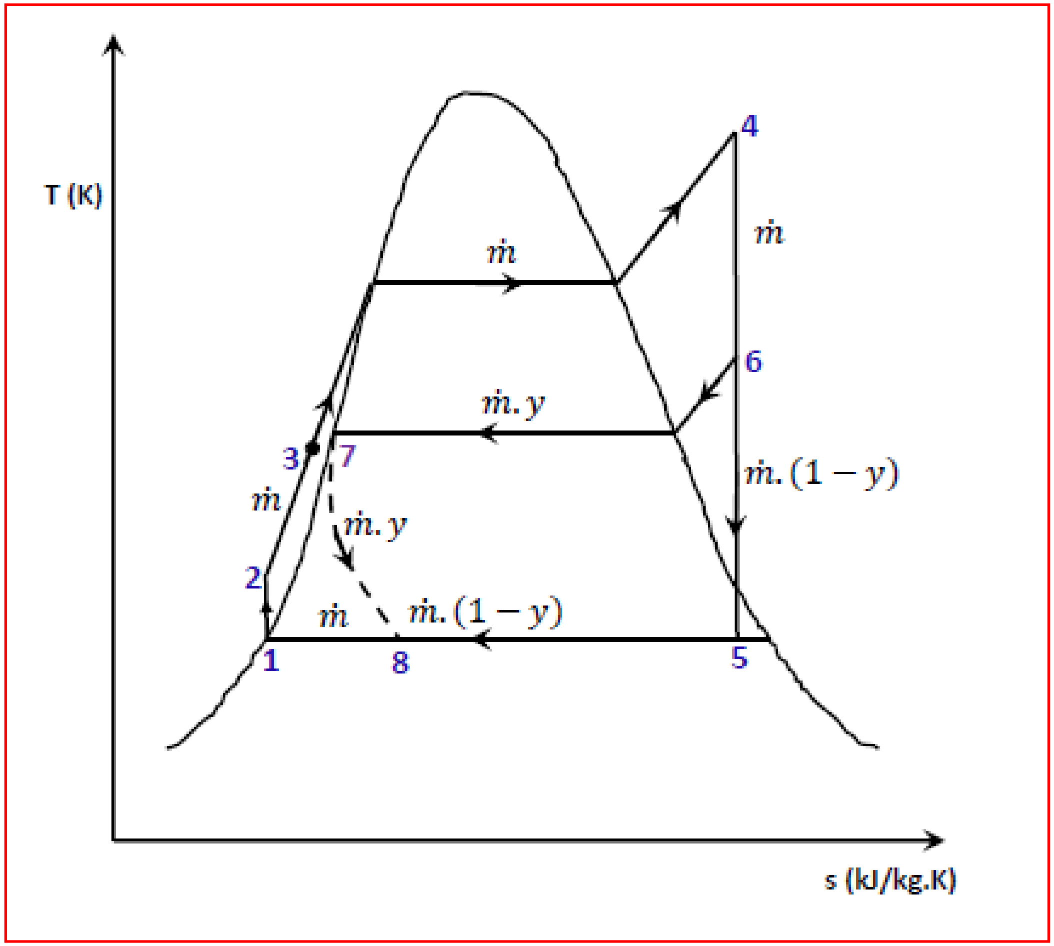
Inner details of a closed feed water heater

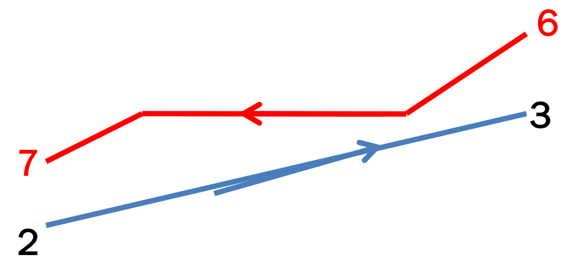
(www.levelandflowsolutions.magnetrol.com)



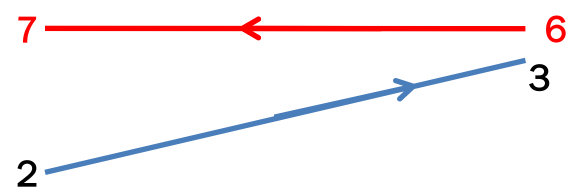
Closed feed water heater (drain backward) contd.

Condition of feedwater & drain water at feedwater heater exit:

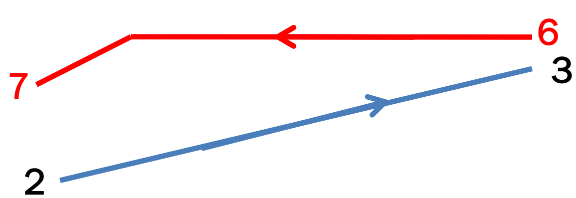




Case(i): Steam is extracted at high pressure

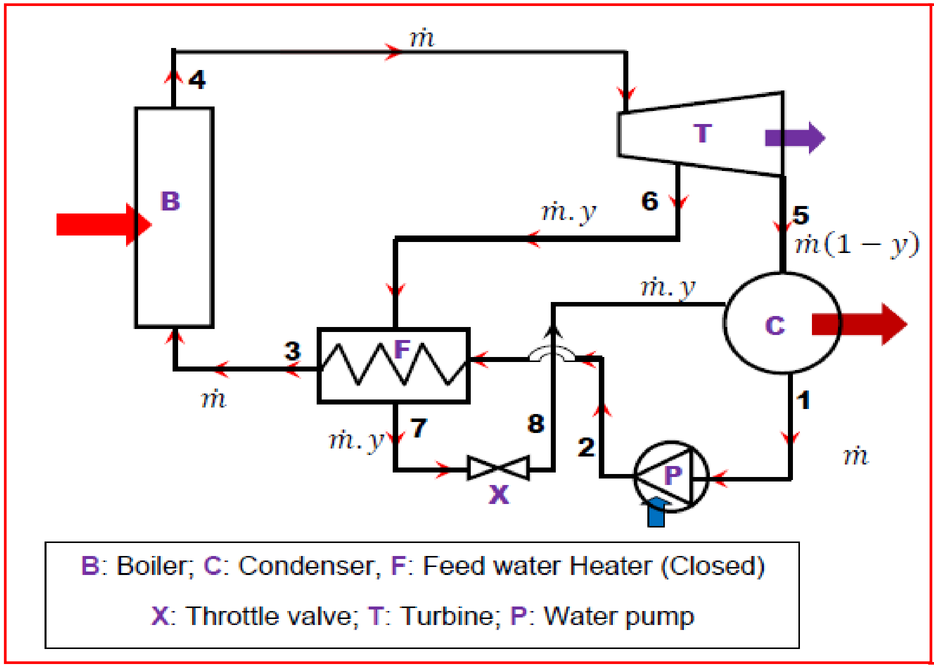


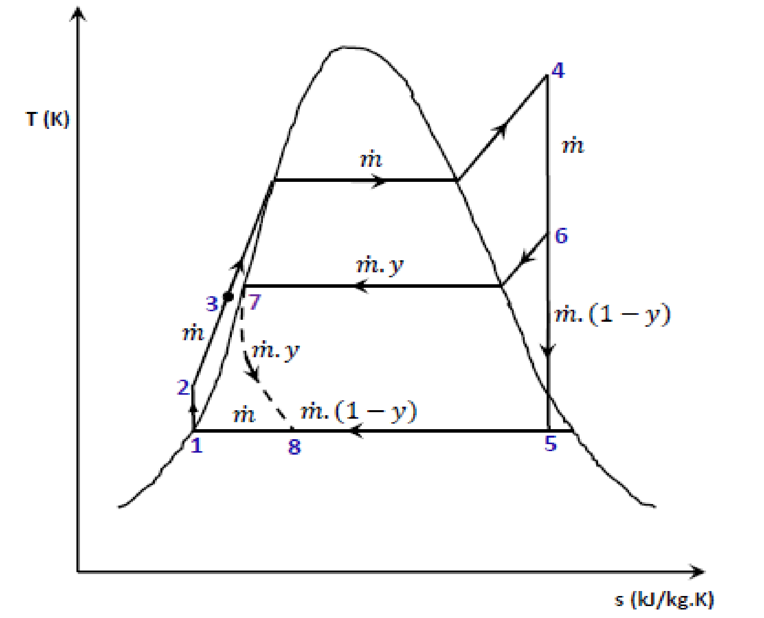
Case(ii): Steam is extracted at low pressure



Case(iii): Steam is extracted at low pressure

Closed feed water heater (drain backward) contd..





Governing equations:

1) Boiler:

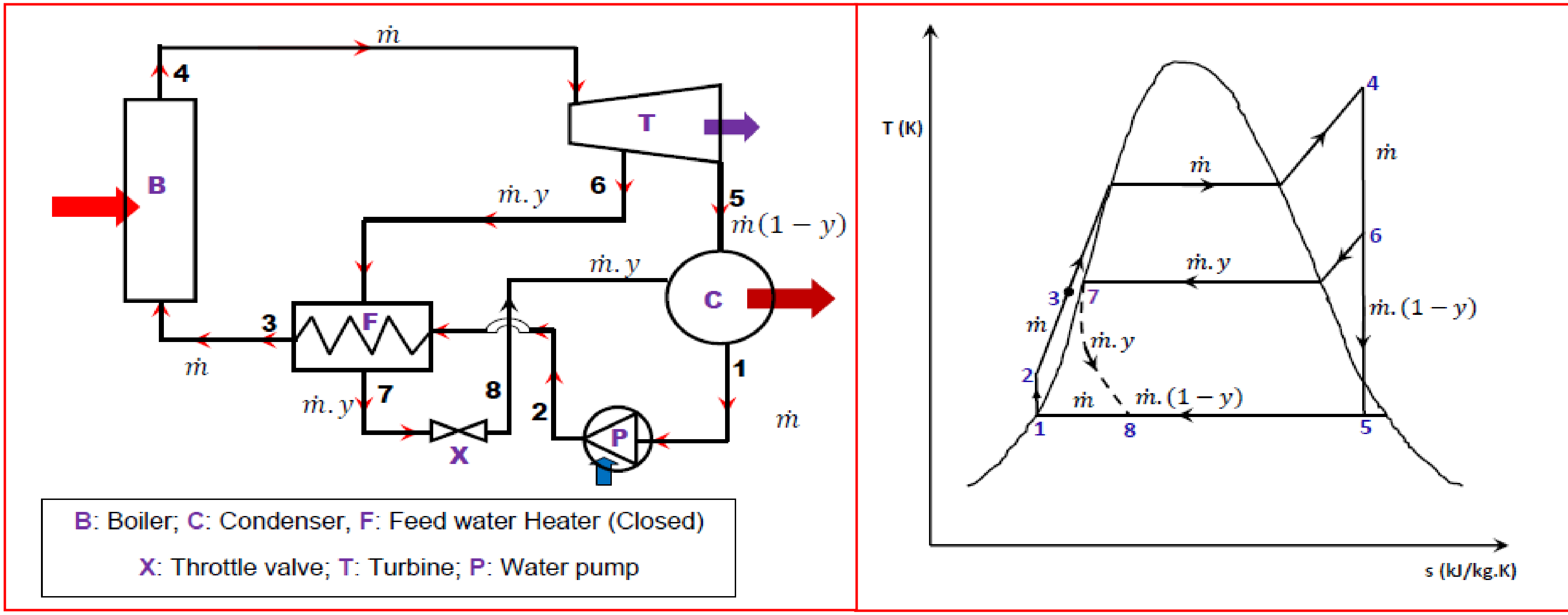
$$Q_{boiler} = \dot{m}\{(h_4 - h_3)\}$$

$$S_{gen,boiler} = \dot{m}\{(s_4 - s_3)\} - \left(\frac{Q_{boiler}}{T_{entropic,ext,boiler}}\right)$$

2) Turbine:

$$W_{turbine} = \dot{m}\{h_4 - y.h_6 - (1 - y)h_5\}$$

Closed feed water heater (drain backward) contd..



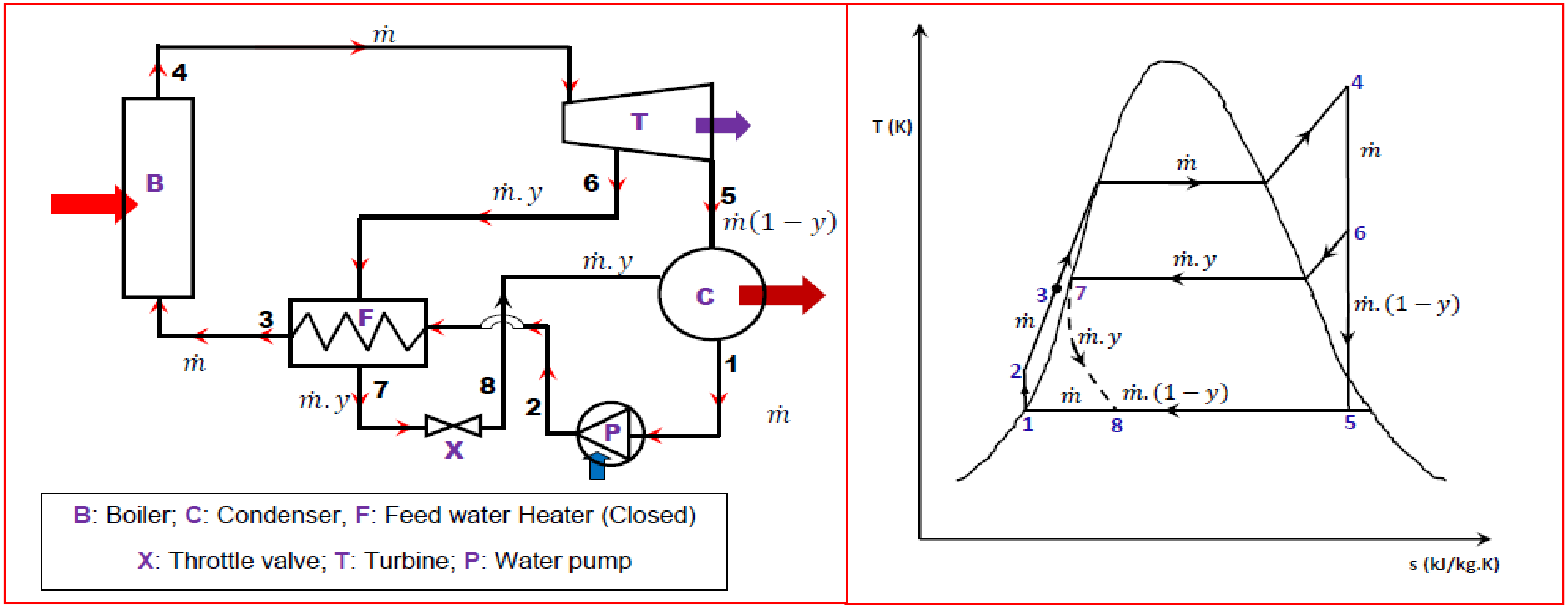
Governing equations:

3) Condenser: $$Q_{condenser} = \dot{m}\{y.h_8 + (1-y)h_5 - h_1\}$$

$$S_{gen,condenser} = \dot{m}\{s_1 - (1-y)s_5 - y.s_8\} + \left(\frac{Q_{condenser}}{T_{entropic,ext,condenser}}\right)$$

4) Pump: $$W_{pump} = \dot{m}\{(h_2 - h_1)\}$$

Closed feed water heater (drain backward) contd..



Governing equations:

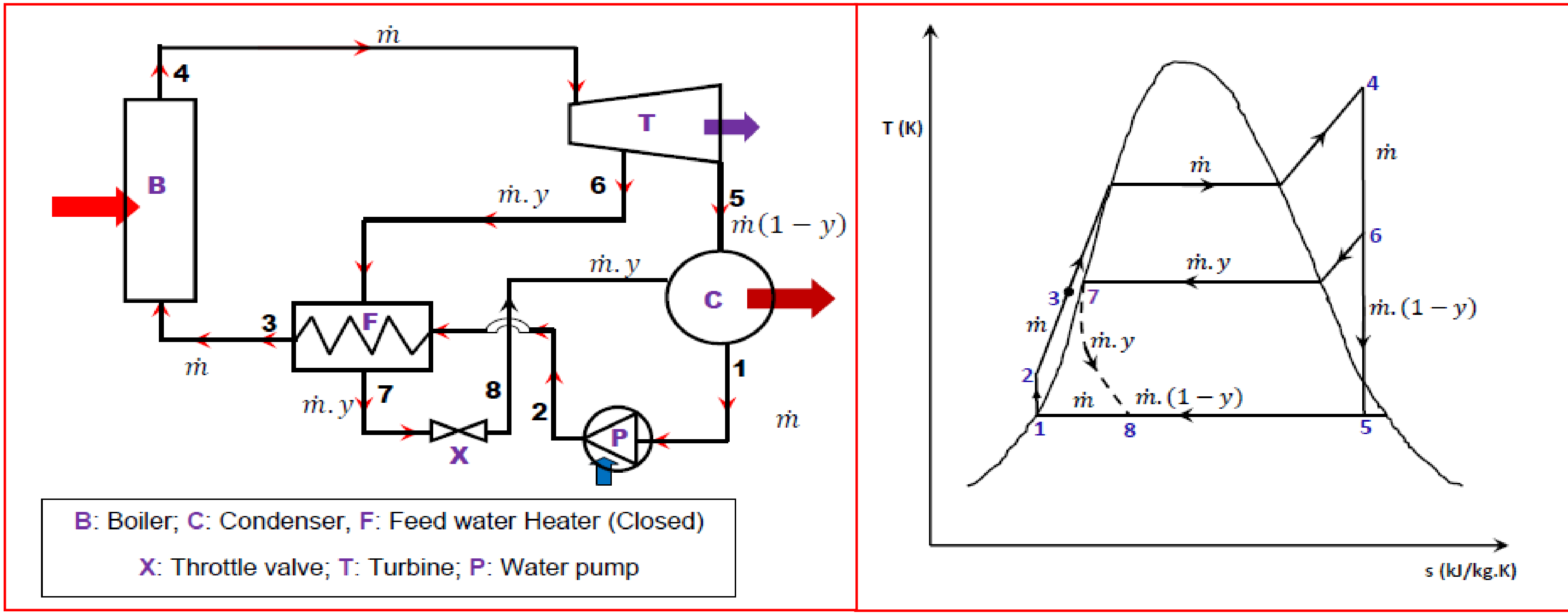
5) Closed feedwater heater: $\dot{m}.(h_3 - h_2) = \dot{m}.y\{h_6 - h_7\}$

$$S_{gen,CFW} = \dot{m}(s_3 - s_2) - \dot{m}y.(s_6 - s_7)$$

6) Throttle valve: $h_8 = h_7$

$$S_{gen,throttle} = \dot{m}\{(s_8 - s_7)\}$$

Closed feed water heater (drain backward) contd..



Governing equations:

$$\eta_{thermal} = \frac{W_{turbine} - W_{pump}}{Q_{boiler}} = \frac{Q_{boiler} - Q_{condenser}}{Q_{boiler}}$$

$$S_{gen,total} = \underbrace{\{S_{gen,boiler} + S_{gen,condenser}\}}_{} + \underbrace{S_{gen,CFW} + S_{gen,throttling}}_{}$$

To solve the problem, we need to specify the condition of feedwater (3) and bleed steam (7) by specifying **Terminal Temperature Differences (TTD) = ($t_7 - t_3$) & ($t_{sat,p6} - t_7$)**

Worked out example: Closed feedwater heater (drain backward)

Given:

W_{net} = **500 MW**

Pressures: Condenser = **0.07 bar**, Boiler = **75 bar**, Feedwater heater = **35 bar**

Heat source: Inlet temperature = **318ºC**, Outlet temperature = **289ºC**

Temperature difference between heat source (inlet) and boiler exit = **18 K**

Heat sink: water inlet temperature = **30ºC** , water outlet temperature = **35ºC**

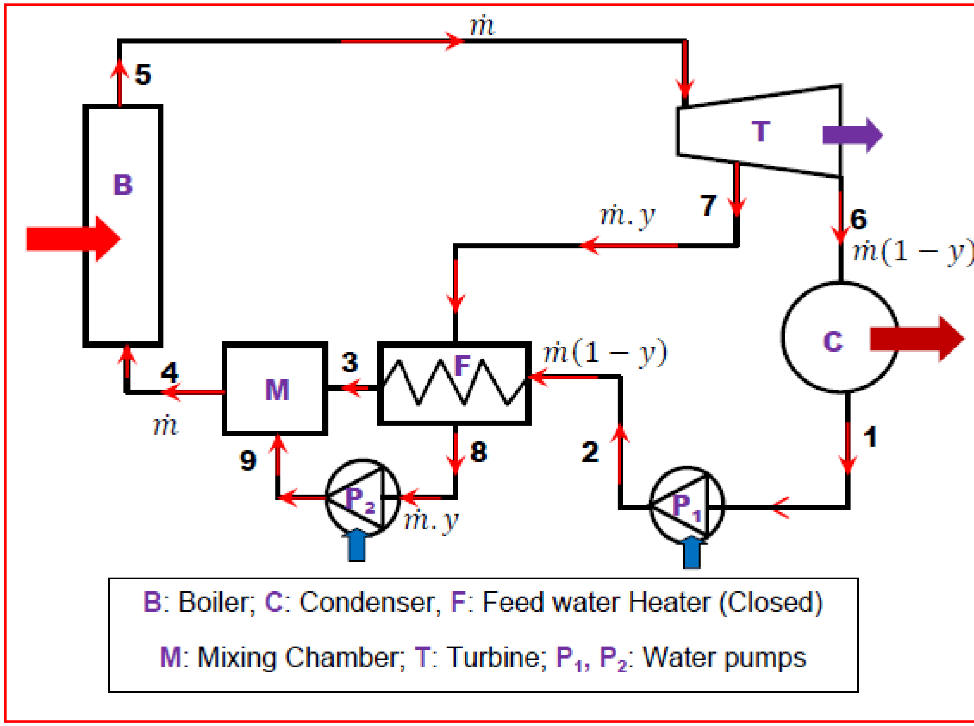
Terminal Temperature Difference: Feedwater = 3 K, **Drain cooler** = **0 K** (saturated)

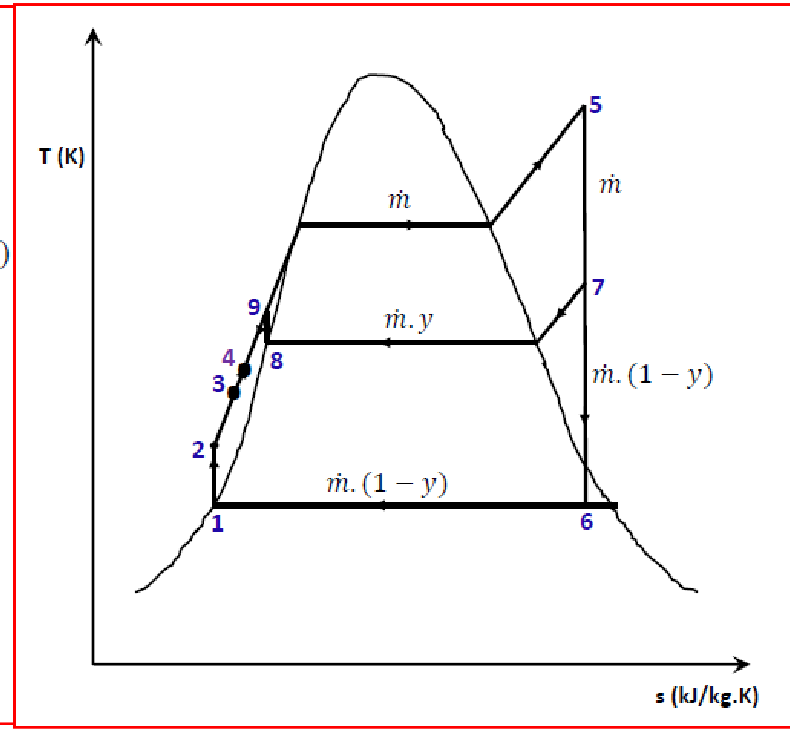
Find:

1) Thermal efficiency of the plant
2) Turbine and pump power, heat transfer in boiler and condenser
3) Mass flow rate of steam through boiler and steam extraction fraction

3) Entropy generation (total, condenser, boiler and feedwater heater)

System with one closed feed water heater (drain forward)





This is also similar in construction to a **shell-and-tube type condenser**

Throttling losses are eliminated by pumping the drain water to the boiler using a small drain water pump

Since drain water flow rate is smaller compared to condensate water, inlet condition to boiler (4) is closer to (3)

This system yields slightly better performance compared to drain backward

- Currently in large thermal power plants 6 to 9 feed water heaters with feed water outlet temperature between 250 to 300^{o}C is used
- **Higher the boiler pressure, higher is the feed water outlet temperature**
- However, for stability of the system, **boiling** should be **prevented** in the **economizer** section
- The common terminal temperature differences used in plants are:
  - Condensation equipment: 2 K (condensate water – cooling water exit)
  - Condensate drain cooler: 7 K (condensate water out – feed water in)
  - Desuperheater: 25 K (bleed steam – feed water out)

Performance of actual power plant cycles

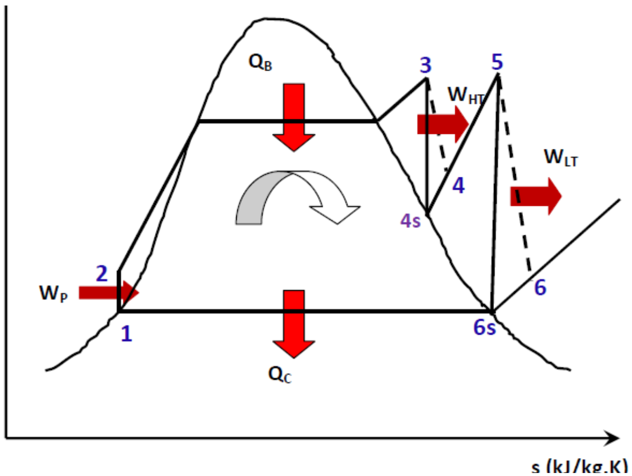
- The **efficiency** of an **actual power plant cycle** will be smaller than that of an **ideal Rankine cycle** due to:

1. Non-isentropic expansion in turbine
2. Non-isentropic compression in pumps
3. Pressure drop across heat exchangers and connecting pipe lines & heat losses

- The non-isentropic expansion/compression across turbine and compressor are indicated by an isentropic efficiency:

$$\eta_{is,turbine} = \frac{W_{act,turbine}}{W_{is,turbine}} = \frac{\Delta h_{act,turbine}}{\Delta h_{is,turbine}}$$

$$\eta_{is,pump} = \frac{W_{is,pump}}{W_{act,pump}} = \frac{\Delta h_{is,pump}}{\Delta h_{act,pump}}$$



- Modern power plant turbines and pumps are extremely well designed with efficiencies as high as 95 %!

Example

- In one stage of turbine, steam at **100 bar** and 380^{o}C expands to a pressure of **25 bar** and develops **25** MW power. If the state of steam at the exit of the turbine is saturated vapour, find:
- A) mass flow rate of steam through the turbine
- B) isentropic efficiency of turbine
- Use the following property data:

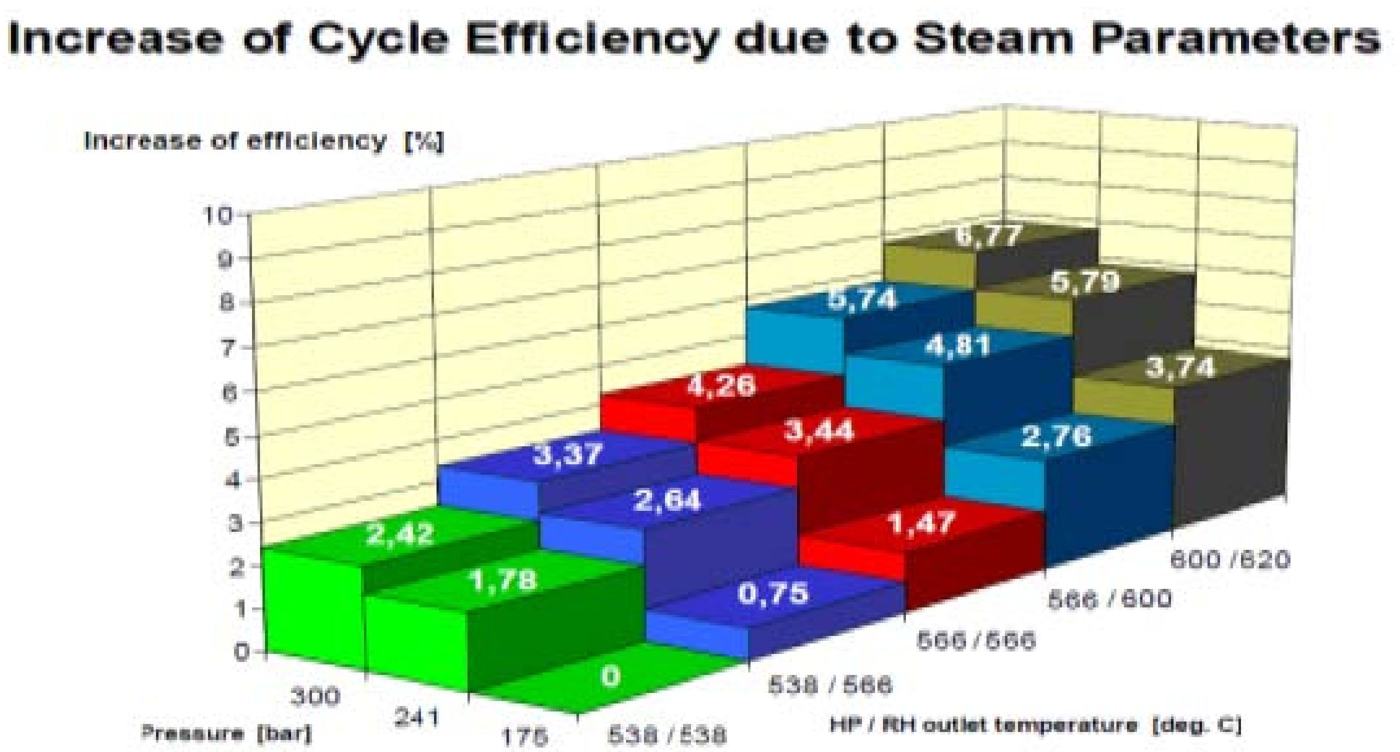
t, ^{o}C	P, bar	x	h, kJ/kg	s, kJ/kg.K
380	100	-	3032	6.114
224	25	1	2802	6.256
224	25	0	962	2.554

Ans.: A) 108.9 kg/s B) 76.5 %

Further improvements in power plant performance

Thermodynamic analysis shows that thermal efficiency of power plants can be increased by operating the plant at higher temperatures and pressures.

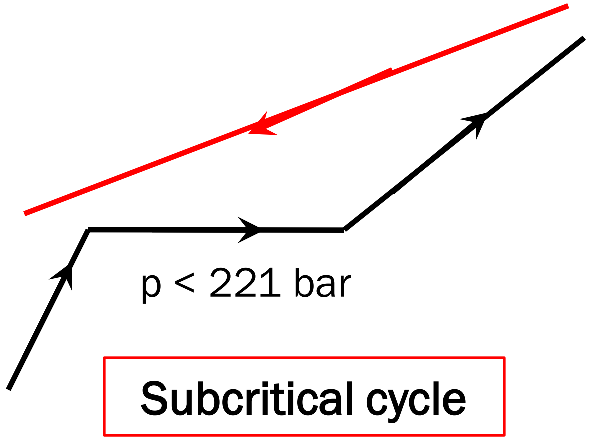
Studies show that with every 1 % increase in efficiency, the emission levels from the power plants can be reduced by 2 %.

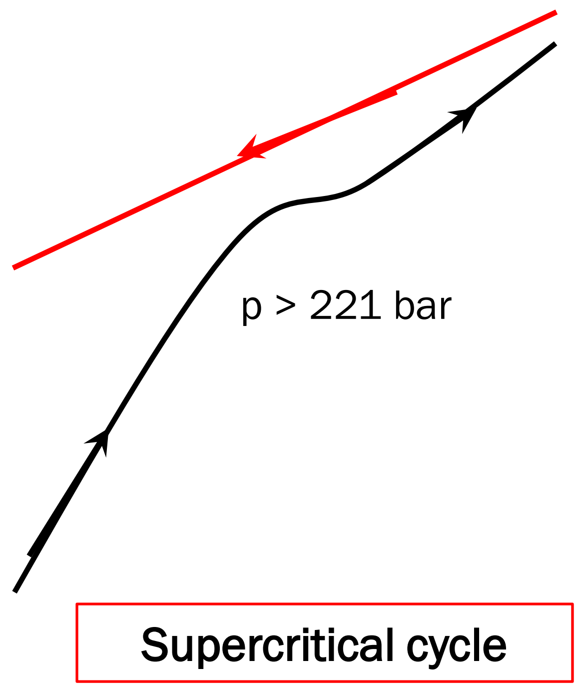


R.S. Yadav and Vaibhav Chauhan, **Supercritical Technology in Indian Power Sector**, National Seminar on Thermal Power Plant Performance Management (NSTPPPM), 2014

The Supercritical Cycle - Concepts

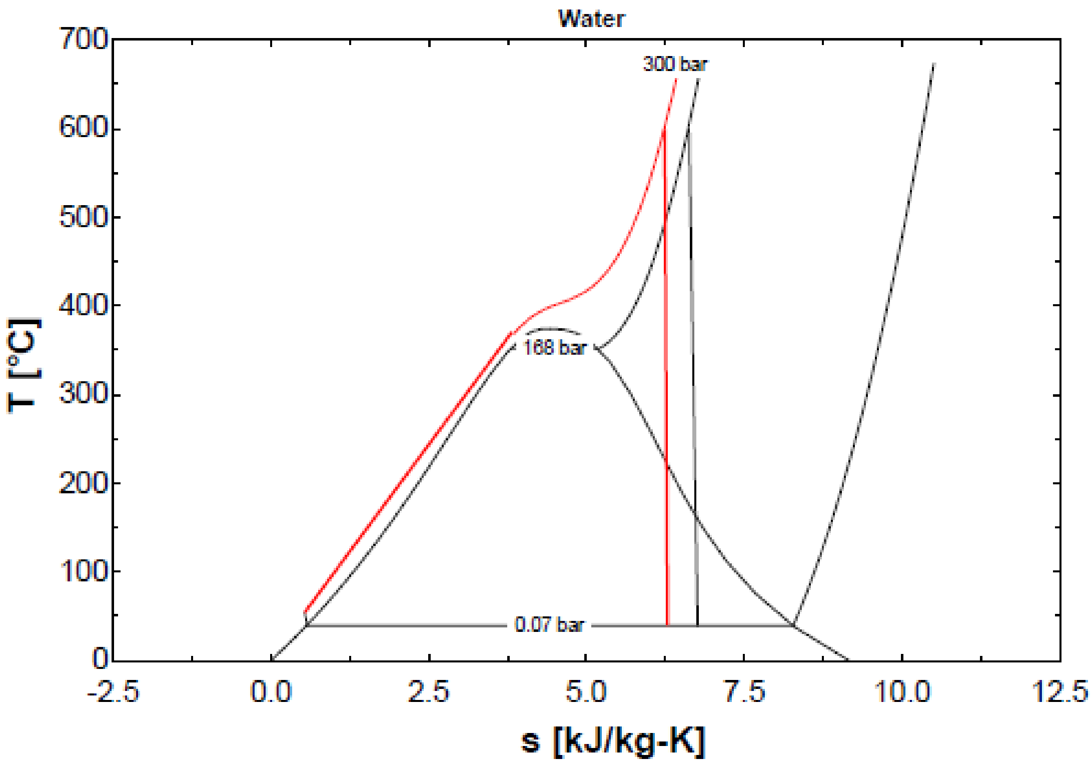
- The temperature at which water can boil is limited by its critical temperature (≈ 374ºC)
- When an external heat source temperature is much higher than 374ºC, **most of the heating** has to take place in the superheated zone
- The resulting **non-uniform temperature profile**, gives rise to a **lower mean temperature of heat addition**
- Under these conditions, higher mean temperature and hence higher efficiency can be obtained by **operating the steam generator in supercritical region** (pressures higher than, critical pressure of ≈ 221 bar)





Supercritical power cycle

- **In a supercritical steam generator** the **properties** of water **change gradually** without undergoing any sudden phase change!



- **Thermodynamic analysis** is **similar** to **standard Rankine cycle**, however, **actual, design, operating and performance characteristics etc. are different**

Example

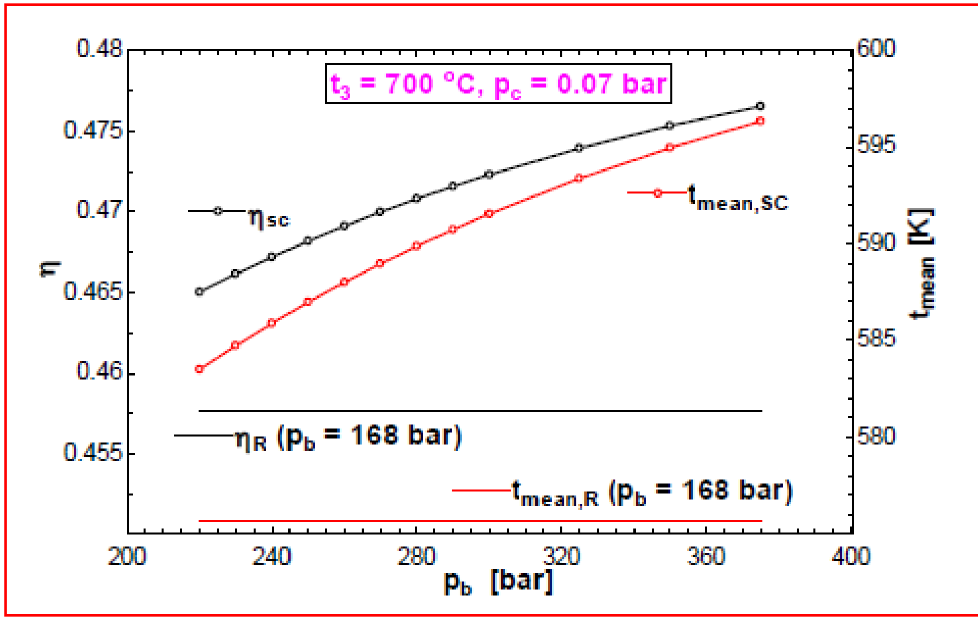
- For the same turbine inlet temperature of 560^{o}C and a condensing pressure of **7 kPa**, compare the thermal efficiency of a subcritical cycle that operates at a turbine inlet pressure of **168 bar** with a supercritical cycle that operates at a turbine inlet pressure of **300 bar**
- For both the cycles, neglect the pump work.
- Use the following property data:

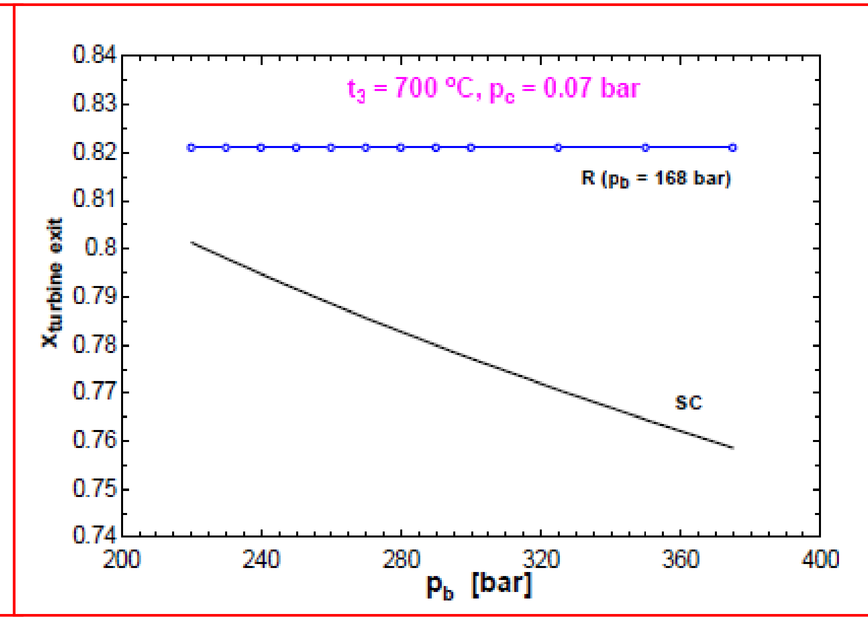
t, ^{o}C	P, bar	x	h, kJ/kg	s, kJ/kg.K
560	168	-	3457	6.483
560	300	-	3312	6.078
39.01	0.07	0	163.4	0.559
39.01	0.07	1	2572	8.274

Ans.: Subcritical: 43.9 %; Supercritical: 45.3 %

Supercritical power cycle – Performance comparison

Effect of boiler pressure





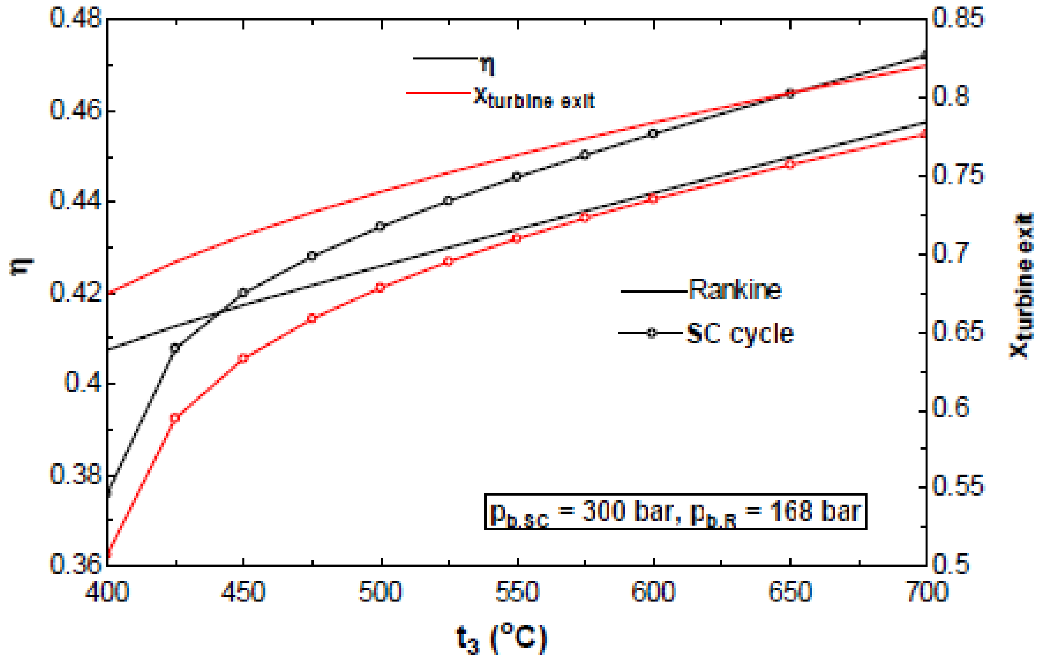
For a **given boiler exit temperature** (say 700°C), **efficiency of SC** cycle **increases** with **boiler pressure**, however,

The turbine **exit quality decreases** as the boiler pressure increases

Hence in **actual power plan**ts, **reheat is always employed** with SC cycle to reduce the liquid fraction in the turbine

Supercritical power cycle – Performance comparison

Effect of boiler exit temperature



It is seen that for **fixed boiler pressures, SC cycle performance** exceeds that of a subcritical **Rankine cycle** only when the **boiler exit temperature is above a certain value**

Due to continuous improvement in materials and manufacturing technologies, it is now possible to **operate** coal based power plants **at much higher pressures and temperatures**

⇒ Supercritical cycles are becoming, a norm rather than an exception, especially when the coal is of high quality

Practical Supercritical cycles

To **reduce the liquid levels** in the turbine, **reheat is employed** in all practical Supercritical cycle based power plants

Often double or triple reheat is used

Due to **progressively lower operating pressures**, **reheat temperature** can be **higher than the boiler exit temperature**, yielding improved performance

Example problem

Given: Condenser pressure = **0.07 bar**

Boiler pressure = **300 bar**

1st Reheat pressure = **80 bar**

2nd Reheat pressure = **28 bar**

Turbine inlet temperature = **600 °C**

Find:

Mean temp. of heat addition: **609 K**

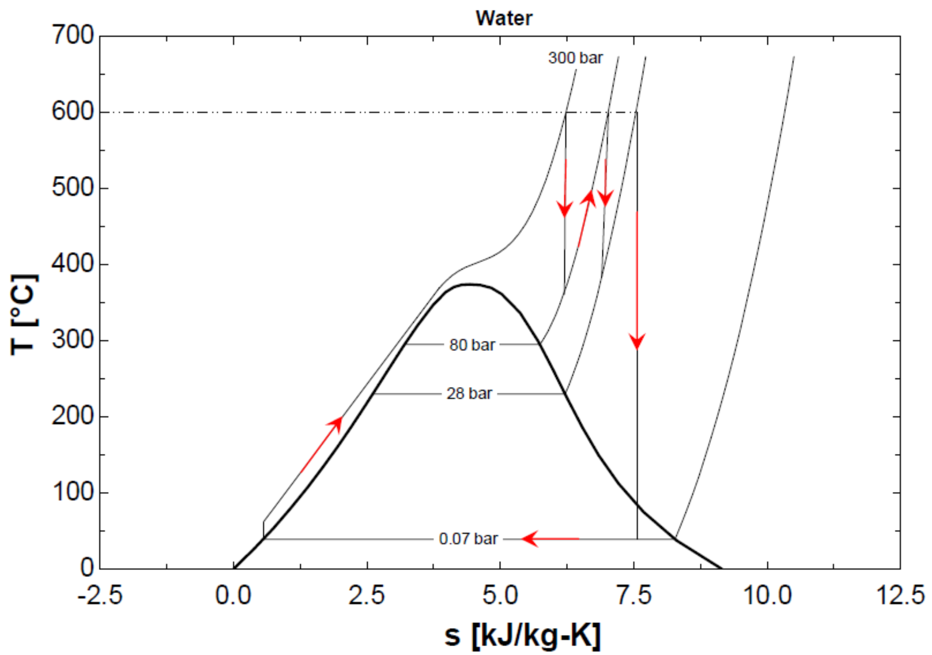
Turbine work: 391+424+1315 = **2130 kJ/kg**

Pump work: **30 kJ/kg**

Boiler heat input: **4309 kJ/kg**

Thermal efficiency: **48.73 %**

Carnot efficiency: **64.25 %**



Supercritical cycle with double reheat

SC power plant in India

Eight units of 660 MW under execution at Sipat and Barh

Two units of 800 MW under execution after 4 February 2015 at Krisnapattnam

Pressure: 246-250 kgf/cm^2

Temperature: 537-566 ^oC (660 MW in Sipat &Barh-I TPS)

565-593 ^oC (in 660 MW Barh-II & 2 x 800 MW in Krisnapattnam)

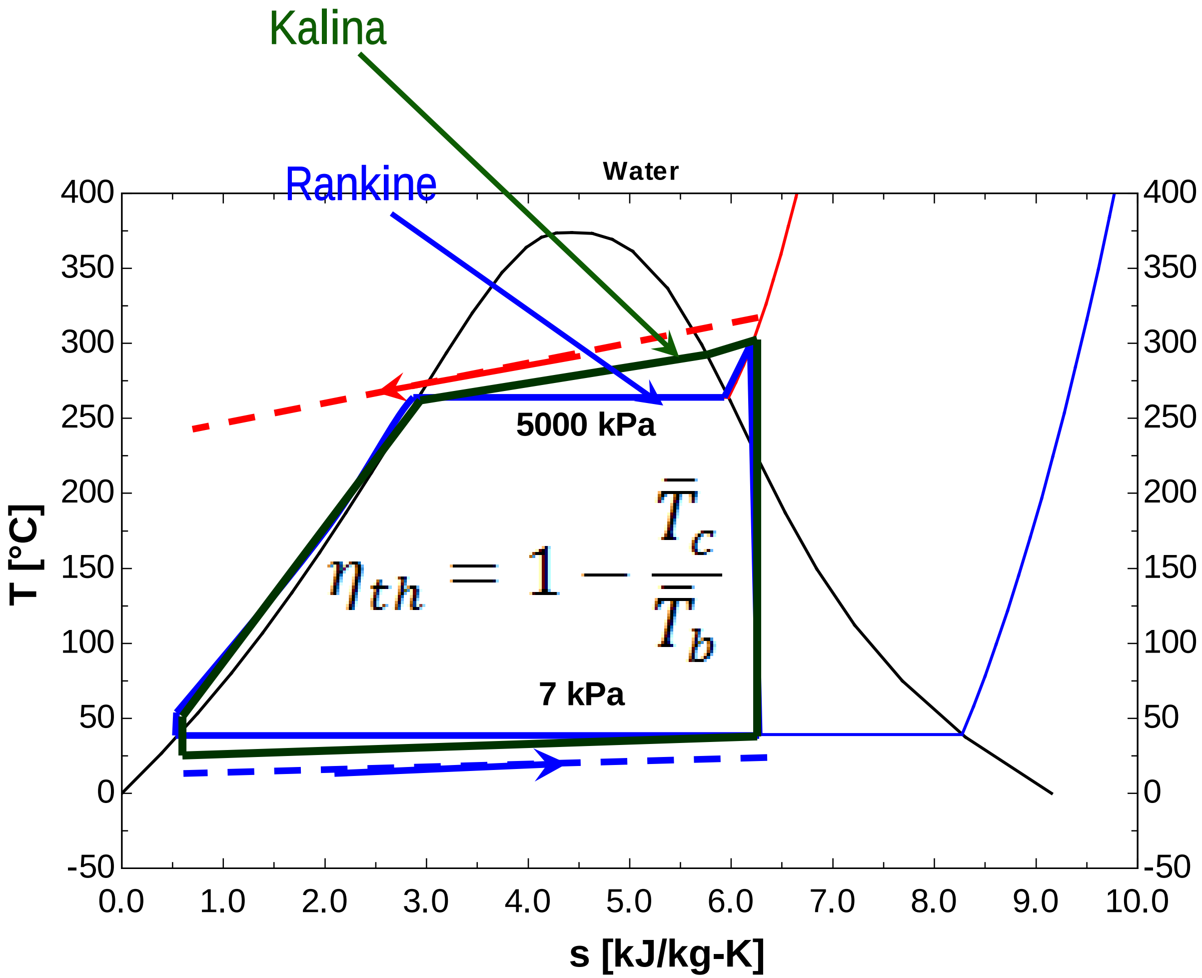
- Adani Power Maharashtra Limited (APML), a subsidiary of Adani Group is constructing India's first super-critical technology based thermal power plant at Tiroda, Maharashtra.
- Tiroda power plant will be the biggest power generation facility in Maharashtra, with an installed capacity of 3,300MW (5x660MW)

Typical operating conditions of modern steam power plants

Type	Operating conditions	Remarks
Subcritical	**163/168** bar, 538°C/538°C with single reheat	Efficiency ≈ **40 %**
Supercritical (SC)	**245** bar, 565°C/565°C with single reheat	Efficiency ≈ **45 %**
Ultra Supercritical (USC)	**300** bar, 600°C/600°C with single reheat	Efficiency ≈ **47 to 49 %**
Ultra Supercritical (USC)	**375** bar, 700°C/720°C with single reheat	Efficiency ≈ **50 %** (Expected)

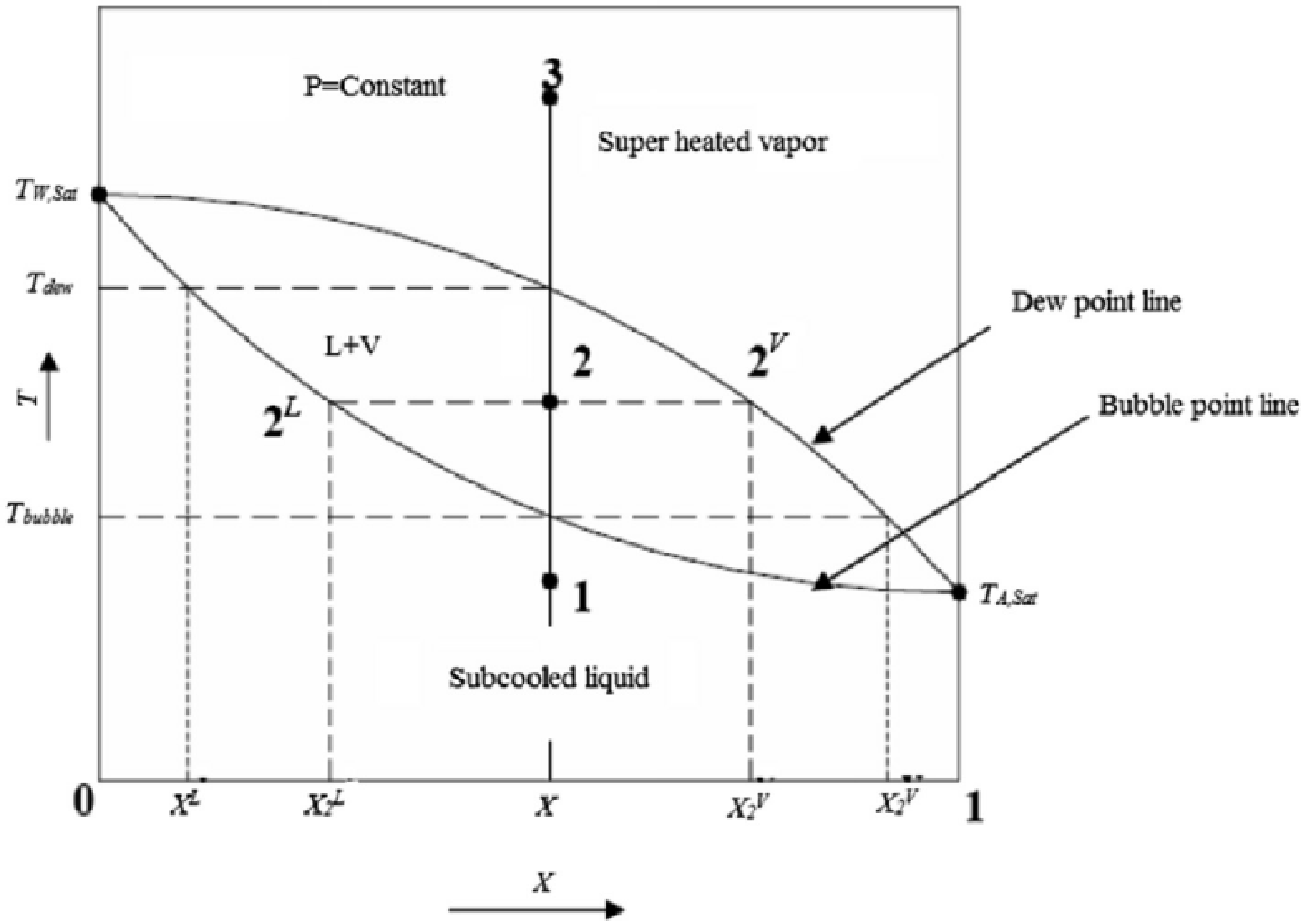
1) With Pulverized coal (PC) combustion
2) Efficiency is based on Lower Heating Value (LHV) of the fuel
3) Plant efficiency increases by about **1 % for every 20 K increase** in superheat temperature

Kalina Cycle



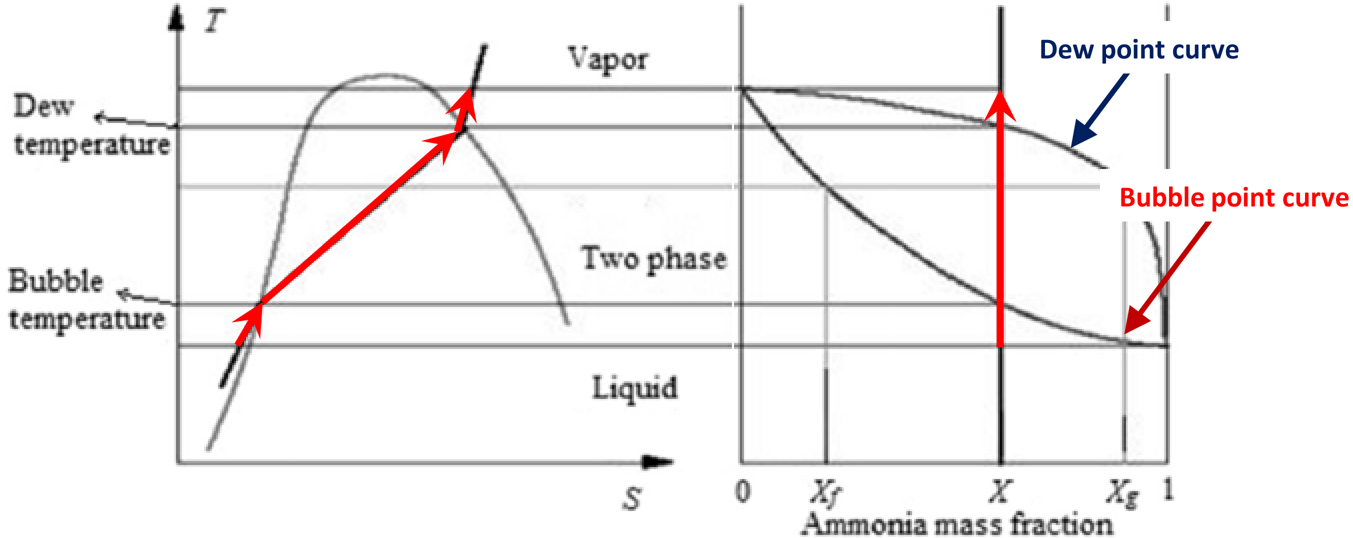
$\bar{\bar{T}}_{b,Kalina} > \bar{\bar{T}}_{b,Rankine}$ & $\bar{\bar{T}}_{c,Kalina} < \bar{\bar{T}}_{c,Rankine} \quad \Rightarrow \eta_{th,Kalina} > \eta_{th,Rankine}$

•Unlike Rankine cycle which uses a pure working fluid (water), the Kalina cycle uses a binary mixture, e.g., ammonia-water

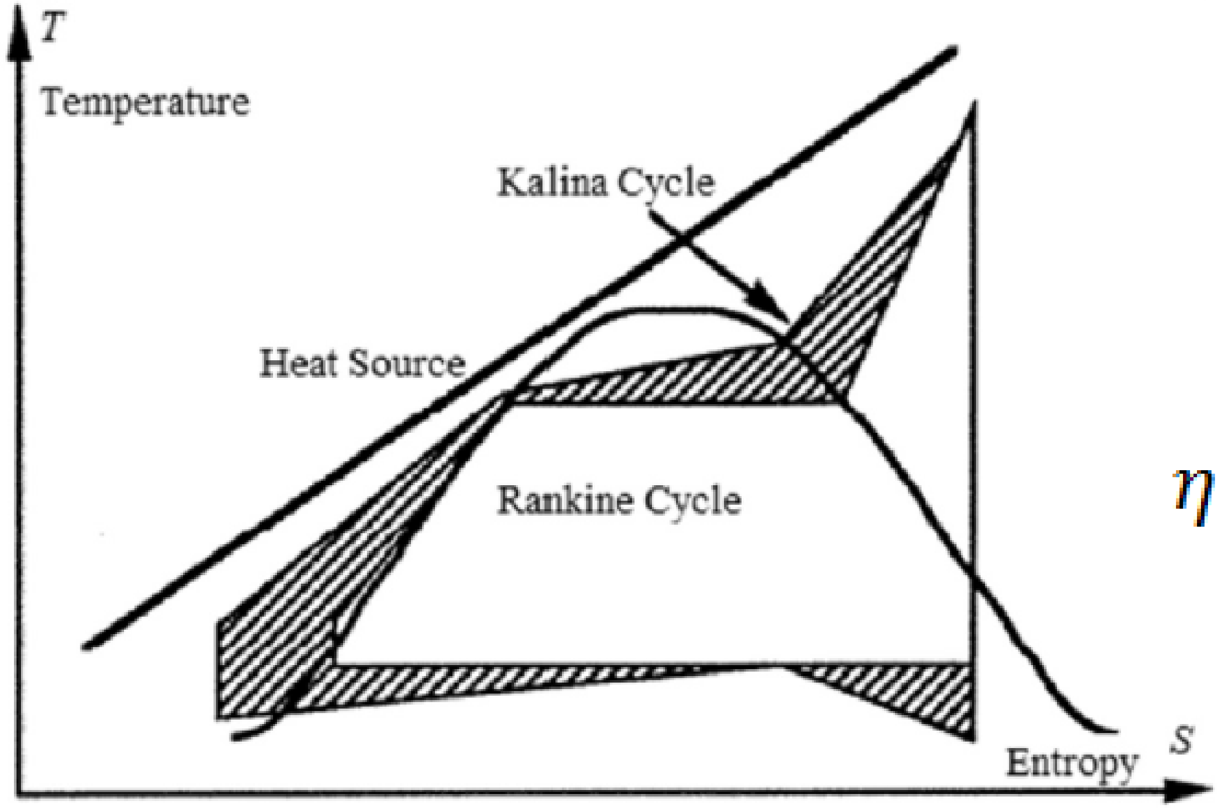


Phase diagram of a binary mixture

Kalina Cycle



Phase diagram of ammonia-water mixture at constant pressure



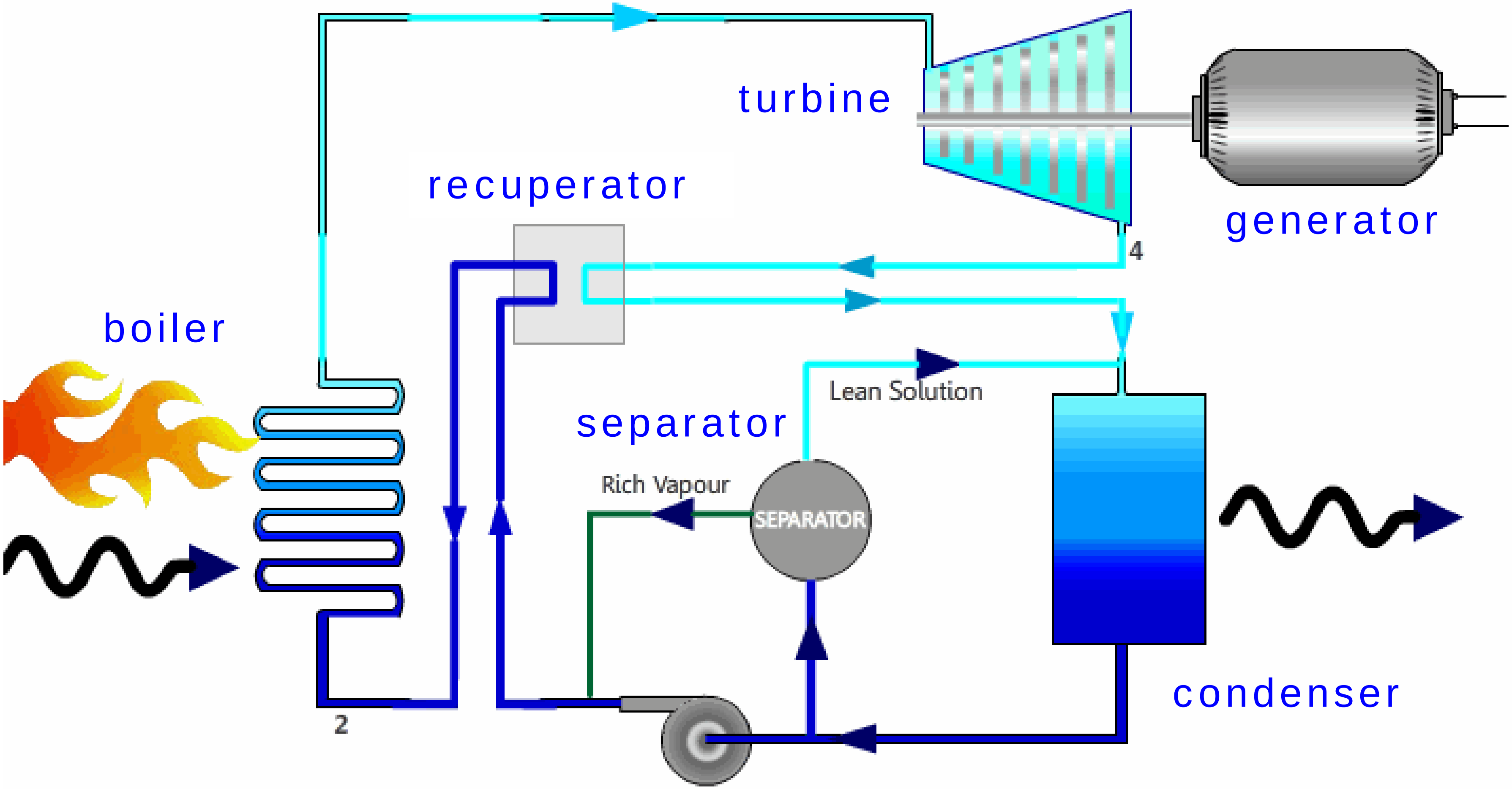
Comparison between Rankine & Kalina Cycles

$$\eta_{th} = 1 - \frac{\bar{T}_c}{\bar{T}_b}$$

$$\bar{T}_{b,Kalina} > \bar{T}_{b,Rankine} \;\&\; \bar{T}_{c,Kalina} < \bar{T}_{c,Rankine}$$

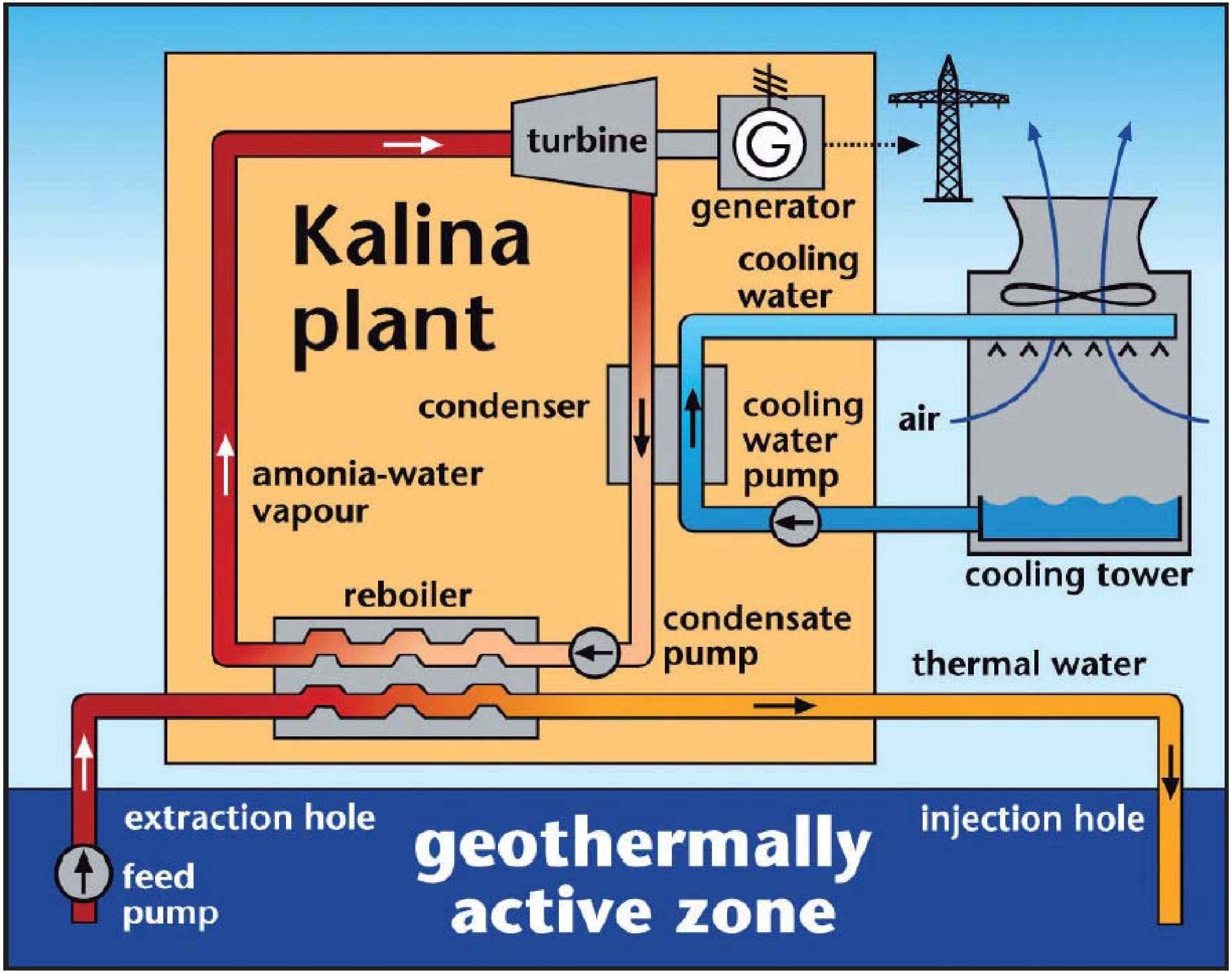
$$\Rightarrow \eta_{th,Kalina} > \eta_{th,Rankine}$$

Simple Kalina Cycle



Since mixture rich in ammonia requires much lower temperature for condensation, it is **diluted** before condensation **by mixing with lean mixture** coming from **separator**

Kalina Cycle for geothermal applications

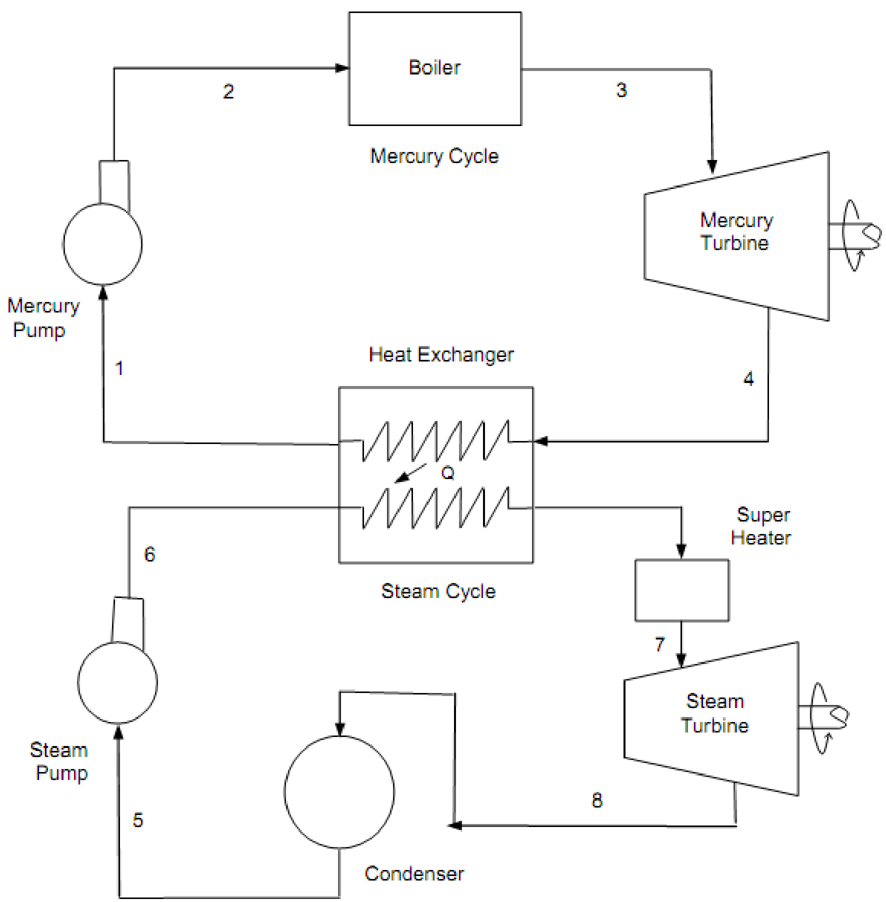


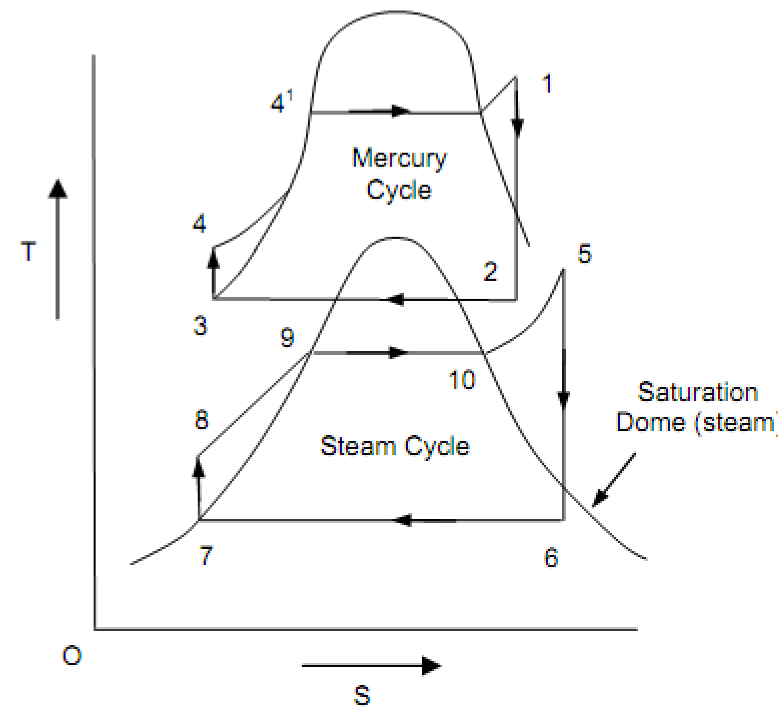
https://www.youtube.com/watch?v=7Z4-8nJQBbU

Binary Vapour Cycles (Topping and bottoming cycles)

- When **water is used as the working fluid in a Rankine cycle**:
  - The **boiler pressure** is **very high** at **high temperatures** (of the order of 100 bar)
  - The condenser pressure is **very low** at low condensing temperatures ($\approx$ 0.1 bar)
- Very high pressure in boiler and very low pressure in condenser are not desirable due to several practical problems
- When a heat source is available at a high temperatures, it is advantageous to use a working fluid with high boiling point, e.g., Mercury, Sodium, Potassium etc.
  - e.g. at 600^{o}C, Mercury has saturation pressure of about 12 bar!
- Similarly when a heat sink is available at a low temperatures, it is advantageous to use a working fluid with low boiling point, e.g., ammonia
  - e.g. at 40^{o}C, **Ammonia** has saturation pressure of 15.6 bar, while it is 0.07 bar for water
- The above facts, give rise to the concept of topping and bottoming cycles, in which a high boiling point temperature is used in the **topping cycle** and a low boiling point fluid is used as working fluid in the **bottoming cycle**

A binary (Topping) vapour cycle with mercury (www.expertsmind.com)





Evolution of Steam Power Stations Efficiency World wide