CHAPTER 10

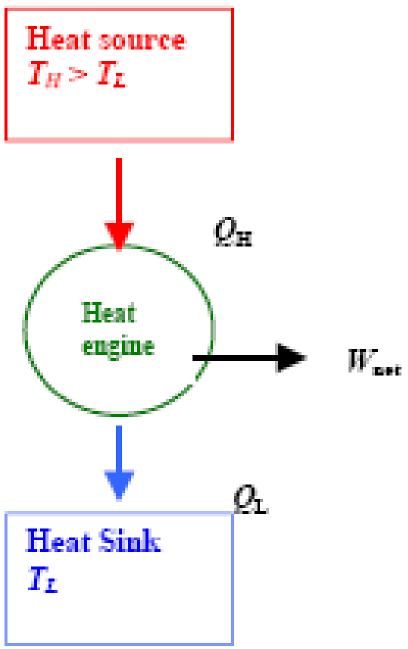
Vapor and Combined Power Cycles

1. The Carnot Vapor Cycle
2. Rankine Cycle : The Ideal Cycle for Vapor Power Cycles
3. Deviation of Actual Vapor Power Cycles
4. How Can We Increase The Efficiency of Cycle
5. The Ideal Reheat Rankine Cycle
6. The Ideal Regenerative Rankine Cycle
7. Second-Law Analysis of Vapor Power Cycles
8. Cogeneration
9. Combined Gas-Vapor Power Cycles

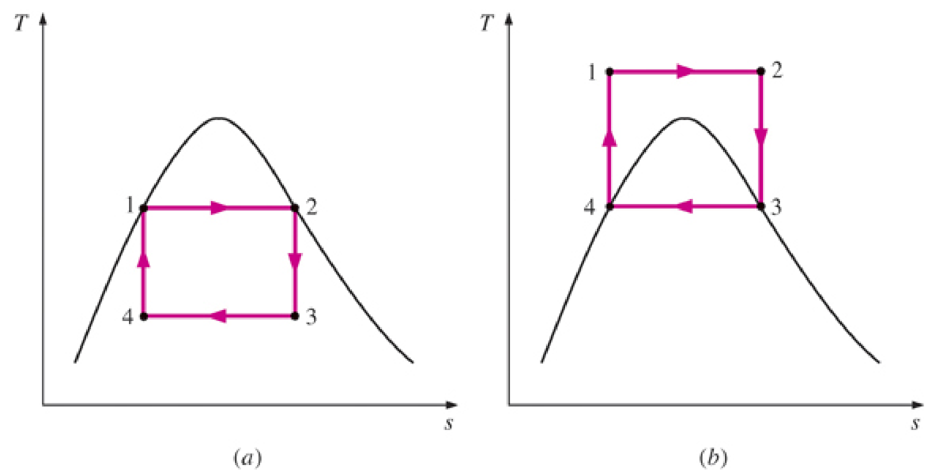
에너지변환시스템연구실(ECOS)
Energy Conversion System Lab.

- Evaluate the performance of gas power cycles for which the working fluid remains a gas throughout the entire cycle.
- Analyze vapor power cycles in which the working fluid is alternately vaporized and condensed.
- Analyze power generation coupled with process heating called *cogeneration*.
- Investigate ways to modify the basic Rankine vapor power cycle to increase the cycle thermal efficiency.
- Analyze the reheat and regenerative vapor power cycles.
- Analyze power cycles that consist of two separate cycles known as combined cycles.

1. The Carnot Vapor Cycle



***T-s* diagram of two Carnot vapor cycles**



The Carnot cycle is the most efficient cycle operating between two specified temperature limits but it is not a suitable model for power cycles. Because:

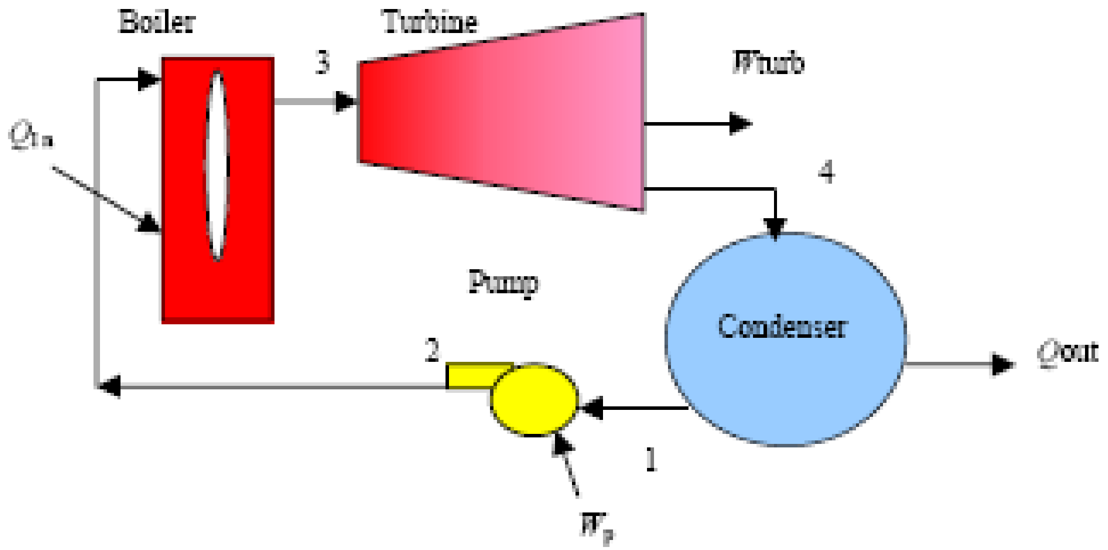
Process 1-2 Limiting the heat transfer processes to two-phase systems severely limits the maximum temperature that can be used in the cycle (374°C for water)

Process 2-3 The turbine cannot handle steam with a high moisture content because of the impingement of liquid droplets on the turbine blades causing erosion and wear.

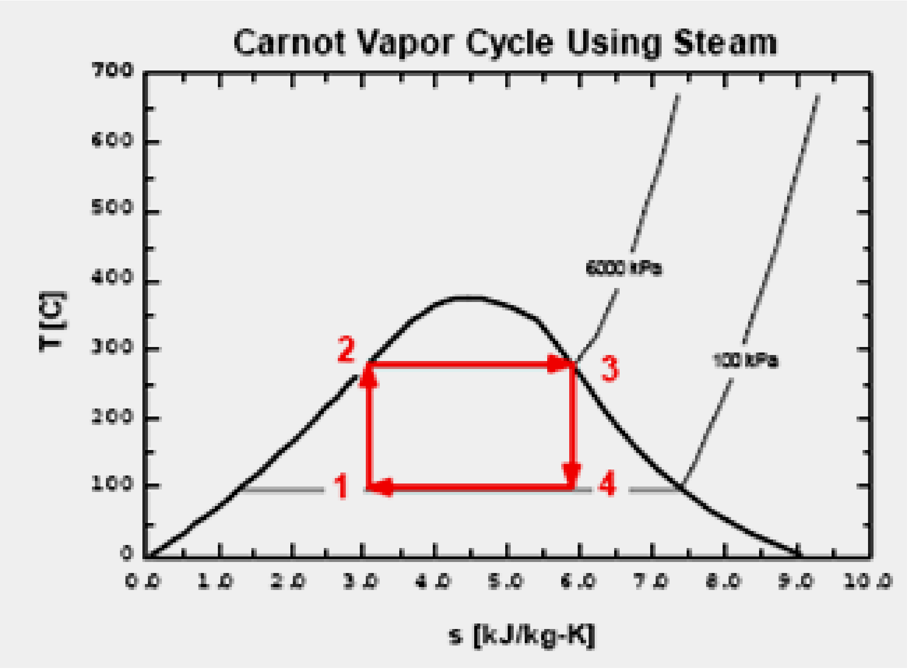
Process 4-1 It is not practical to design a compressor that handles two phases.

The cycle in (b) is not suitable since it requires isentropic compression to extremely high pressures and isothermal heat transfer at variable pressures.

Steam Power Cycle



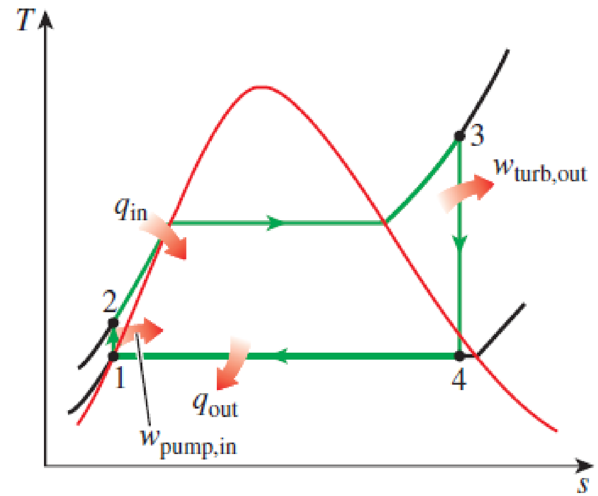
$$\eta_{th,\,Carnot} = \frac{W_{net}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}}$$
$$= 1 - \frac{T_L}{T_H}$$

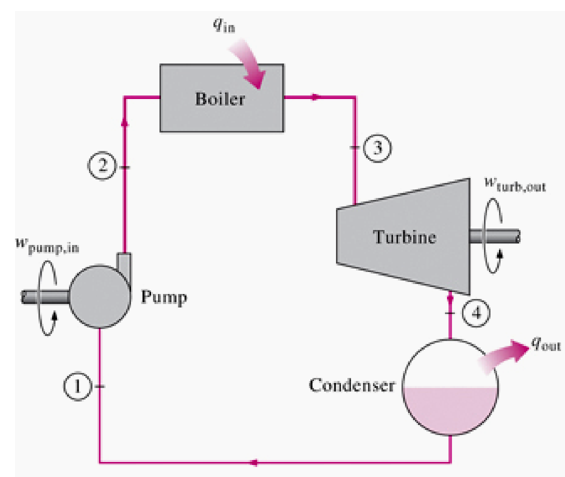


2. Rankine Cycle : Ideal Cycle for Vapor Power Cycles

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.

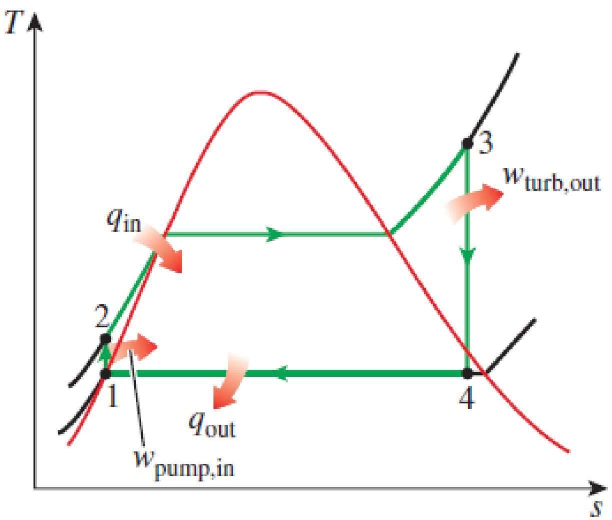
The cycle that results is the **Rankine cycle**, which is the ideal cycle for vapor power plants. The ideal Rankine cycle does not involve any internal irreversibilities.





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



Steady-flow energy equation

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

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

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

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

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

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

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

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

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

The thermal efficiency can be interpreted as the ratio of the area enclosed by the cycle on a *T-s* diagram to the area under the heat-addition process.

The efficiency of power plants in the U.S. is often expressed in terms of **heat rate**, which is the amount of heat supplied, in Btu's, to generate 1 kWh of electricity.

$$\eta_{th} = \frac{3412\ (Btu/kWh)}{\text{Heat rate}\ (Btu/kWh)}$$

Energy Analysis of the Ideal Rankine Cycle

1. pump

$$\dot m_1 = \dot m_2 = \dot m$$

$$\dot m_1 h_1 + \dot W_{pump} = \dot m_2 h_2$$

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

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

$$dh = v\,dP$$

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

$$v \cong v_1 = const.$$

$$h_2 - h_1 \cong v_1(P_2 - P_1)$$

$$\dot W_{pump} = \dot m(h_2 - h_1) \cong \dot m v_1(P_2 - P_1)$$

$$w_{pump} = \frac{\dot W_{pump}}{\dot m} = v_1(P_2 - P_1)$$

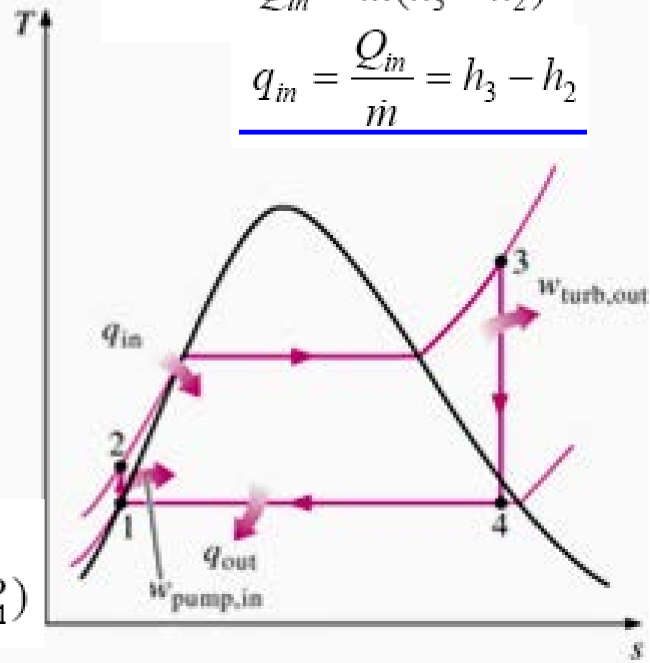
2. boiler

$$\dot m_2 = \dot m_3 = \dot m$$

$$\dot m_2 h_2 + \dot Q_{in} = \dot m_3 h_3$$

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

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



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

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

3. turbine

$$\dot m_3 = \dot m_4 = \dot m$$

$$\dot m_3 h_3 = \dot W_{turb} + \dot m_4 h_4$$

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

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

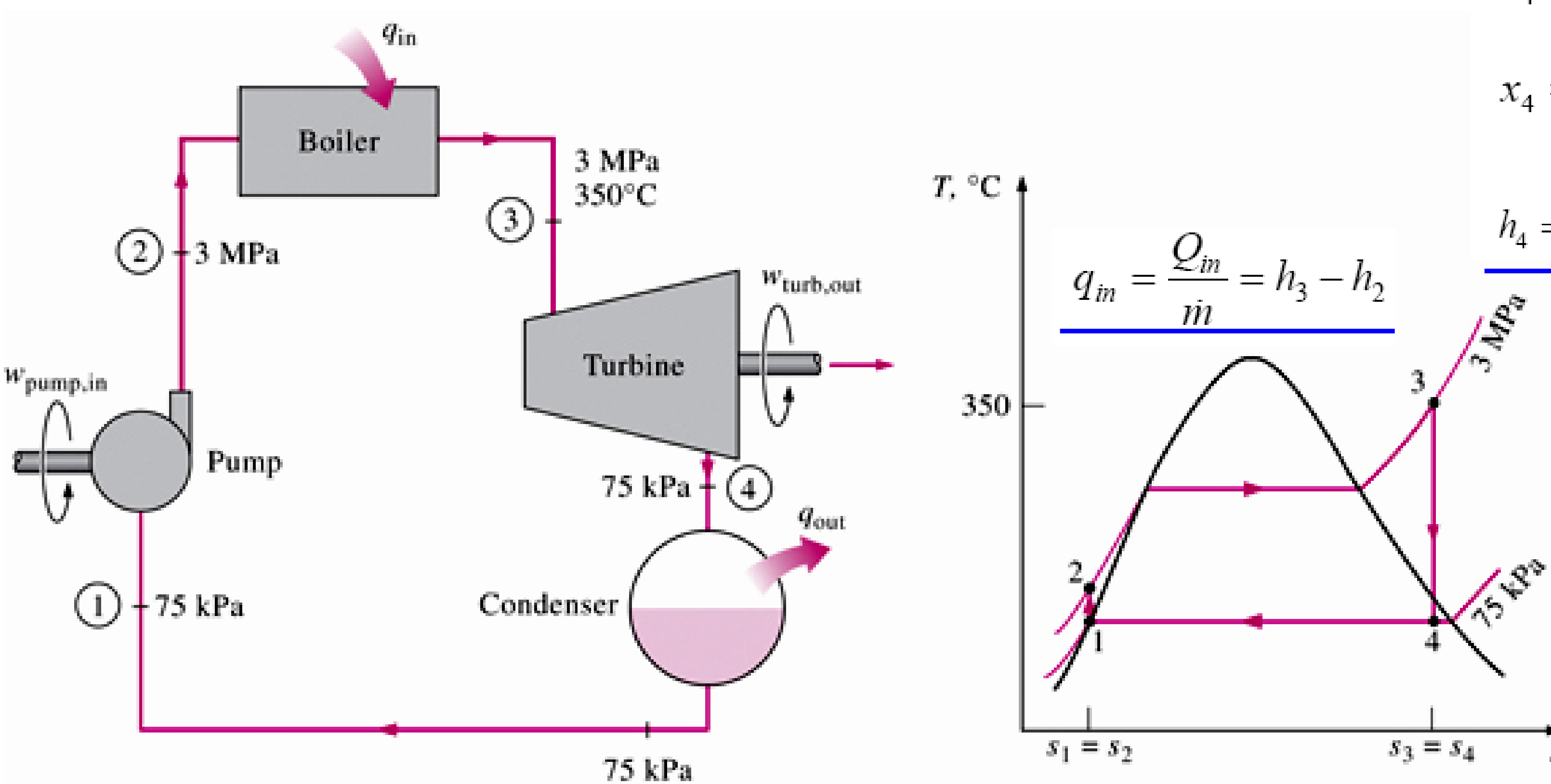
4. condenser

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

$$x_4 = \frac{s_4 - s_f}{s_{fg}} =$$

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

Ex 10-1, The Simple Ideal Rankine Cycle



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

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

$$x_4 = \frac{s_4 - s_f}{s_{fg}} =$$

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

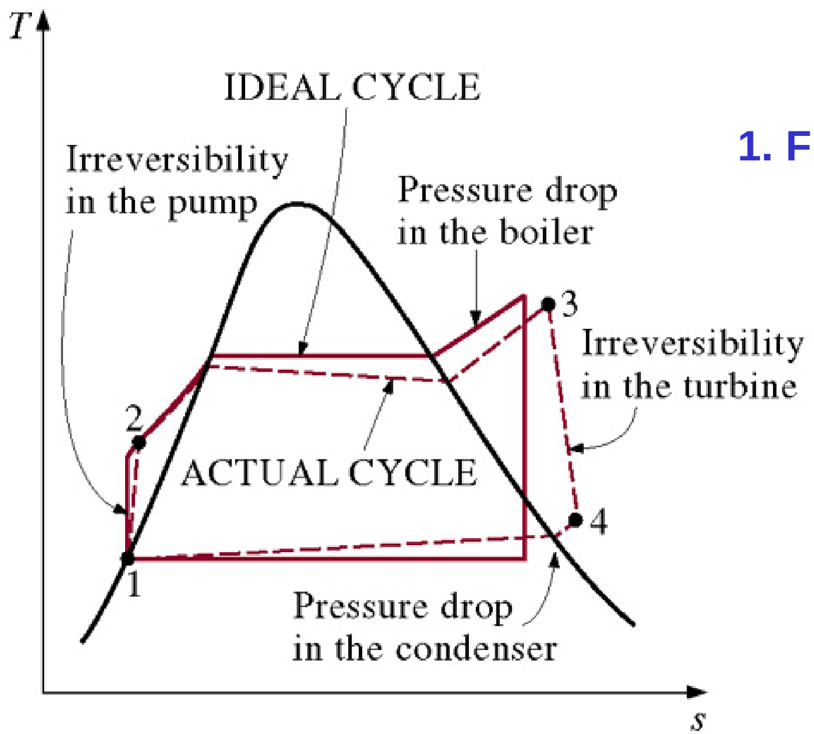
$$v \cong v_1 = const.$$

$$h_2 - h_1 \cong v_1 (P_2 - P_1)$$

3. Deviation of Actual Vapor Power Cycles from the Ideal 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.



1. Fluid friction causes pressure drops

2. Heat loss from steam to surrounding

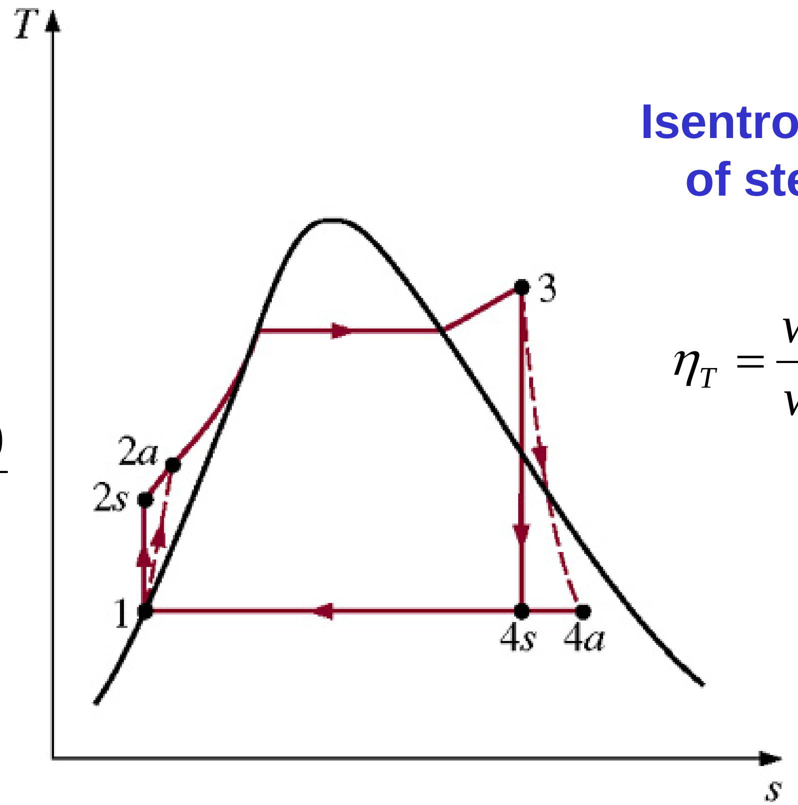
3. Deviation of Actual Vapor Power Cycles from the Ideal Ones

(b) The effect of pump and turbine irreversibilities on the ideal Rankine cycle.

Isentropic efficiency of pump

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

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



Isentropic efficiency of steam turbine

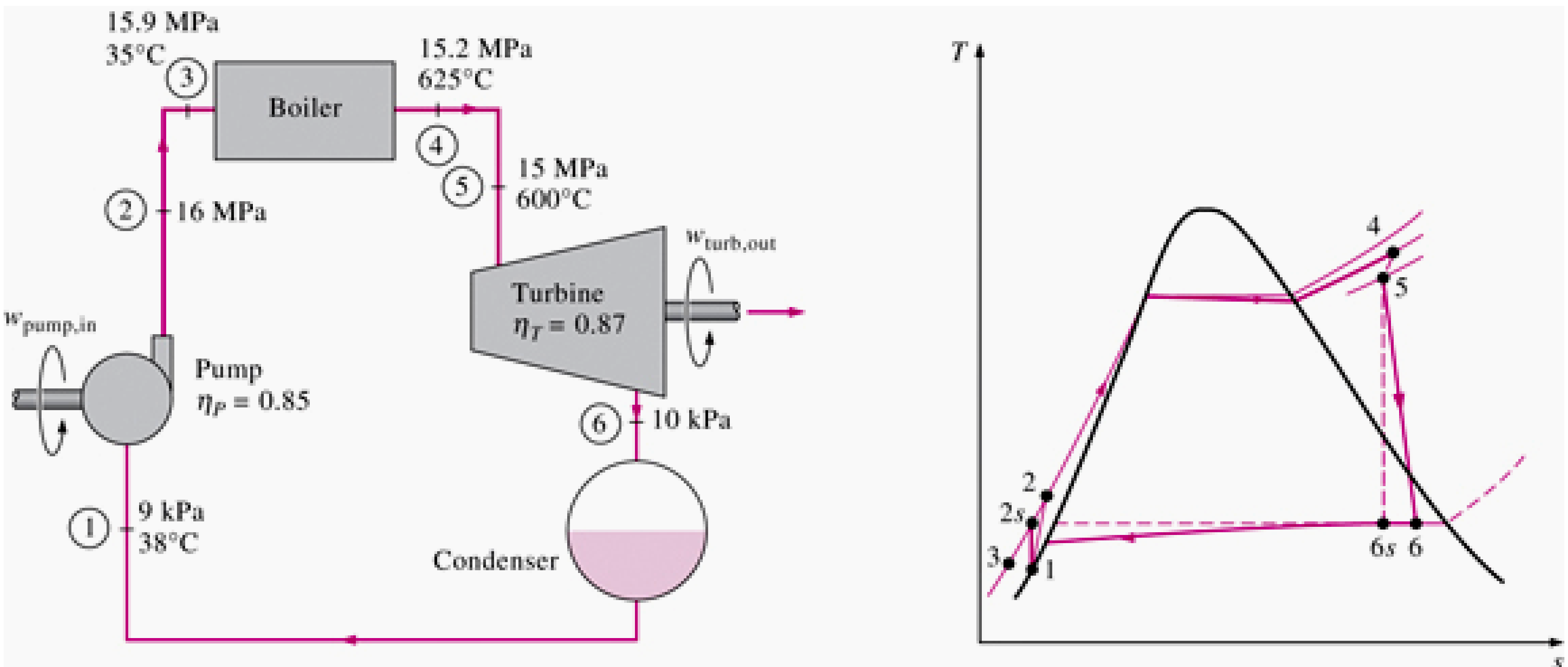
$$\eta_T = \frac{w_a}{w_s} = \frac{h_3 - h_{4a}}{h_3 - h_{4s}}$$

Ex 10-2, An Actual Steam Power Cycle

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

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

$$w_{turbine,out} = \eta_T w_{s,turbine,out}$$



4. How Can We Increase the Efficiency of the Rankine Cycle?

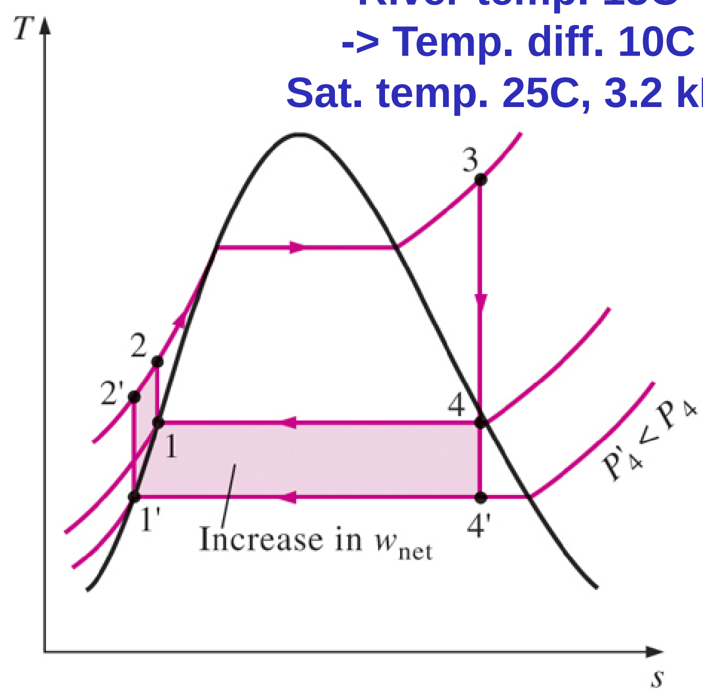
The basic idea behind all the modifications to increase the thermal efficiency of a power cycle is the same: *Increase the average temperature at which heat is transferred to the working fluid in the boiler, or decrease the average temperature at which heat is rejected from the working fluid in the condenser.*

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

To take advantage of the increased efficiencies at low pressures, the condensers of steam power plants usually operate well below the atmospheric pressure. There is a lower limit to this pressure depending on the temperature of the cooling medium

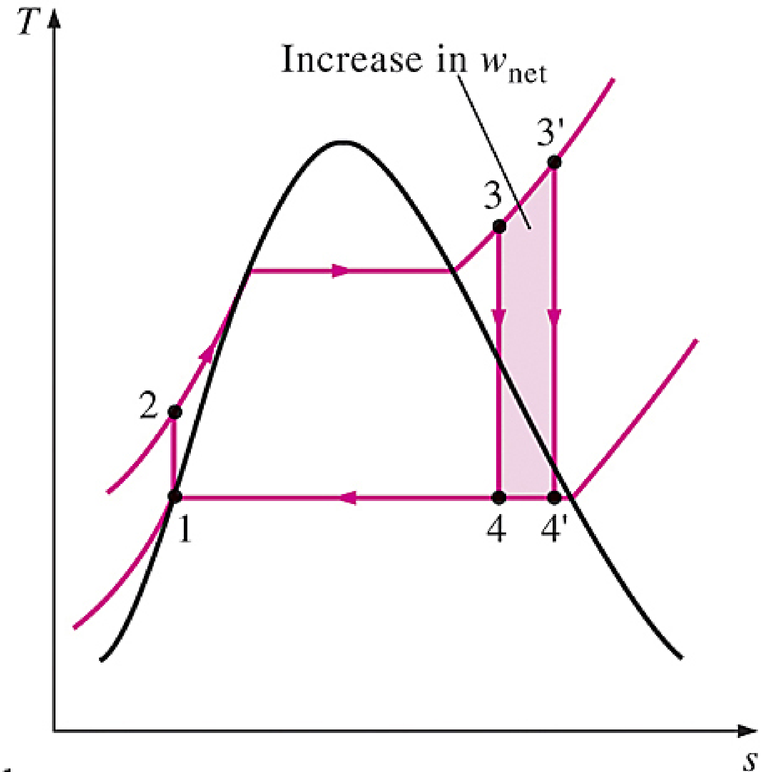
Side effect: Lowering the condenser pressure increases the moisture content of the steam at the final stages of the turbine.

River temp. 15C
-> Temp. diff. 10C
Sat. temp. 25C, 3.2 kPa



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

Both the net work and heat input increase as a result of superheating the steam to a higher temperature. The overall effect is an increase in thermal efficiency since the average temperature at which heat is added increases.



Superheating to higher temperatures decreases the moisture content of the steam at the turbine exit, which is desirable.

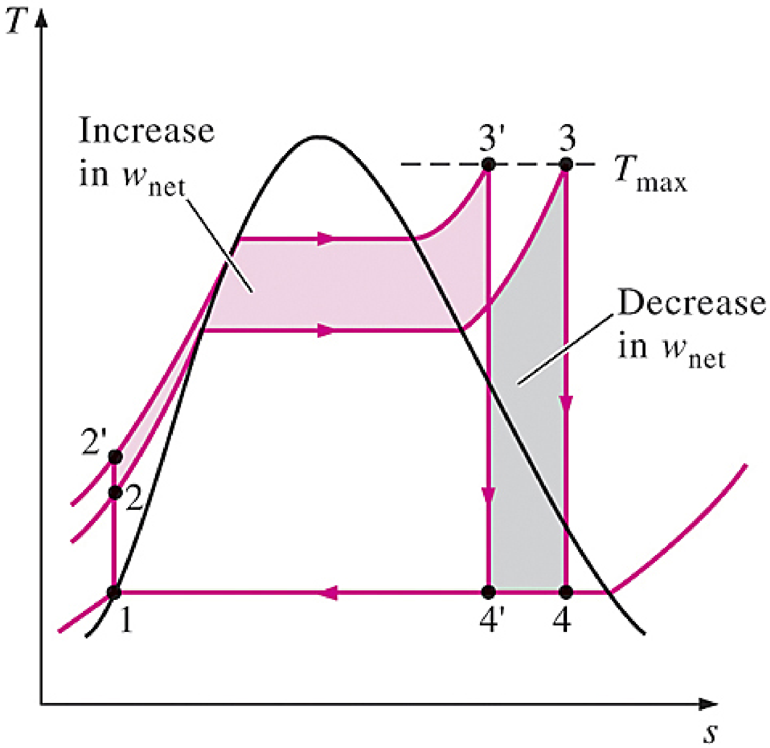
- Superheat the vapor
  - Average temperature is higher during heat addition.
  - Moisture is reduced at turbine exit (we want x_4 in the above example > 85 percent).

The temperature is limited by metallurgical considerations. Presently the highest steam temperature allowed at the turbine inlet is about 620°C in USC Bolier

Increasing the Boiler Pressure (*Increases* $T_{high,avg}$)

For a fixed turbine inlet temperature, the cycle shifts to the left and the moisture content of steam at the turbine exit increases. This side effect can be corrected by reheating the steam.

-> Reheating has both benefits

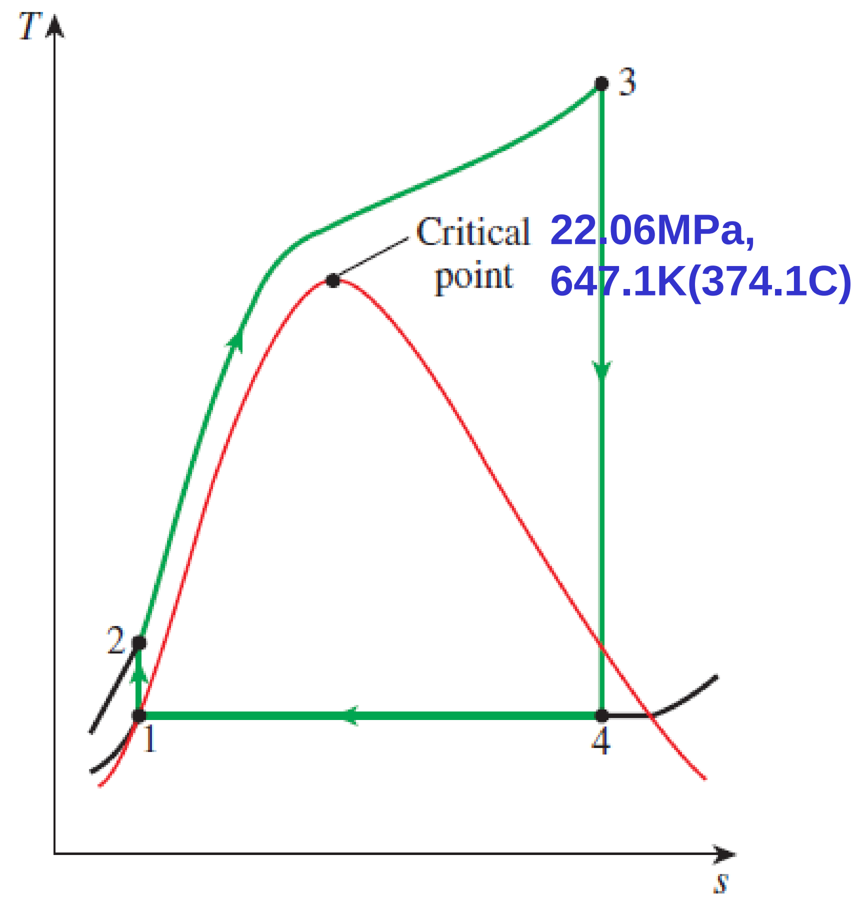


- Increase boiler pressure (for fixed maximum temperature)
  Availability of steam is higher at higher pressures.
  Moisture is increased at turbine exit.

A supercritical Rankine cycle

Today many modern steam power plants operate at supercritical pressures (P > 22.06 MPa) and have thermal efficiencies of about 40% for fossil-fuel plants and 34% for nuclear plants.

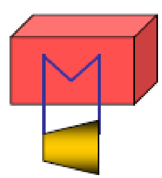
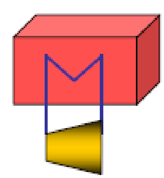
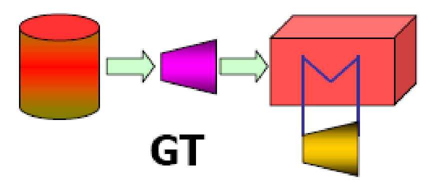
- 2.7MPa in 1992 to over 30MPa today
- output 1000MW
- 150 Super-critical Plants



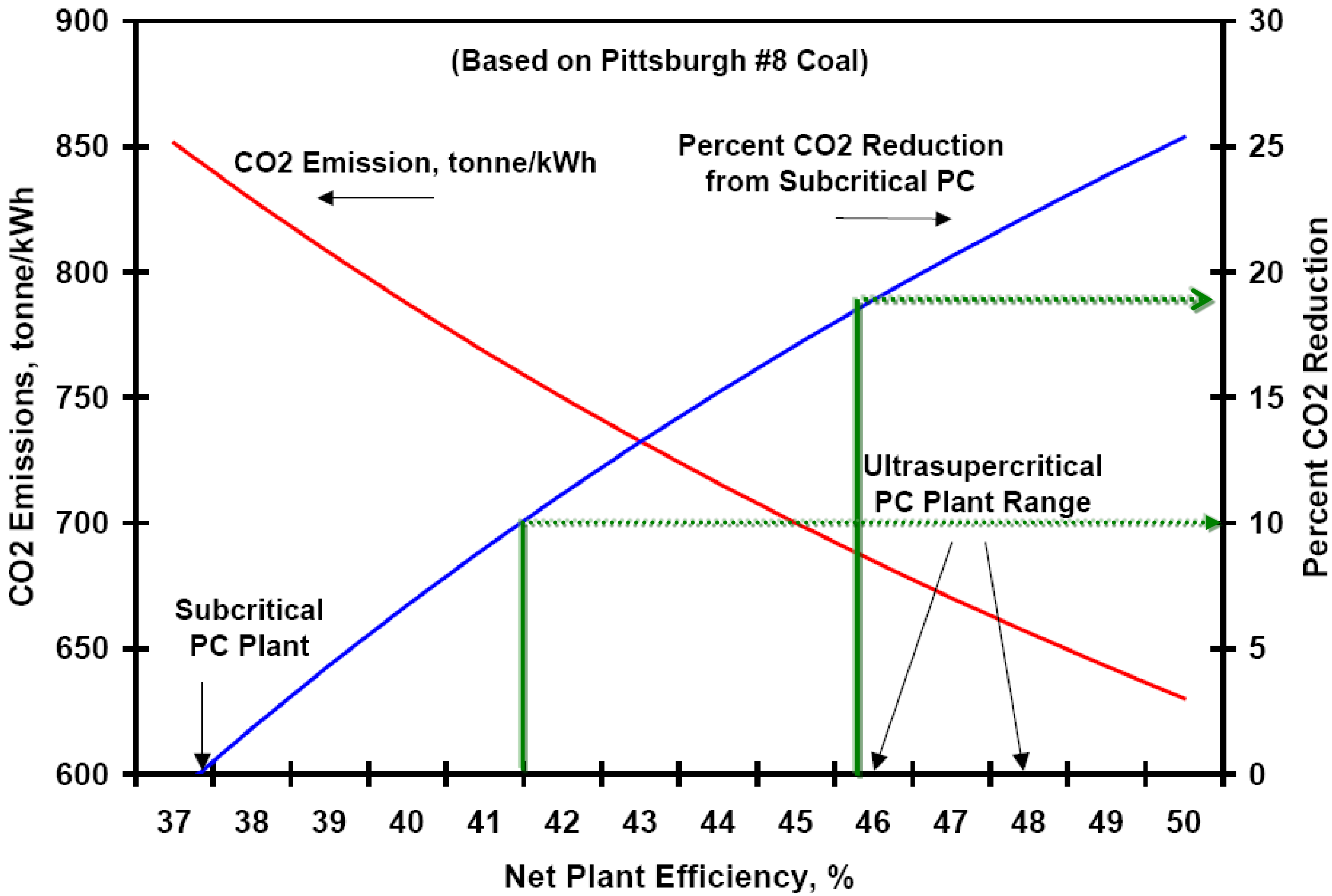
Clean Coal Technology in power generation sector

○ **Future development of the high-efficiency coal fired thermal power generation**

- **A-USC: Advanced ultra super critical pressure power generation**
- **IGCC: Integrated Coal Gasification Combined Cycle**

	USC	A-USC	IGCC(1,500degC)
Configuration			
Thermal Efficiency	42%	46%	46~48%
CO2 Emission Reduction	Base	▲11%	▲13%

저탄소 발전시대에 따른 발전설비의 열효율 관리

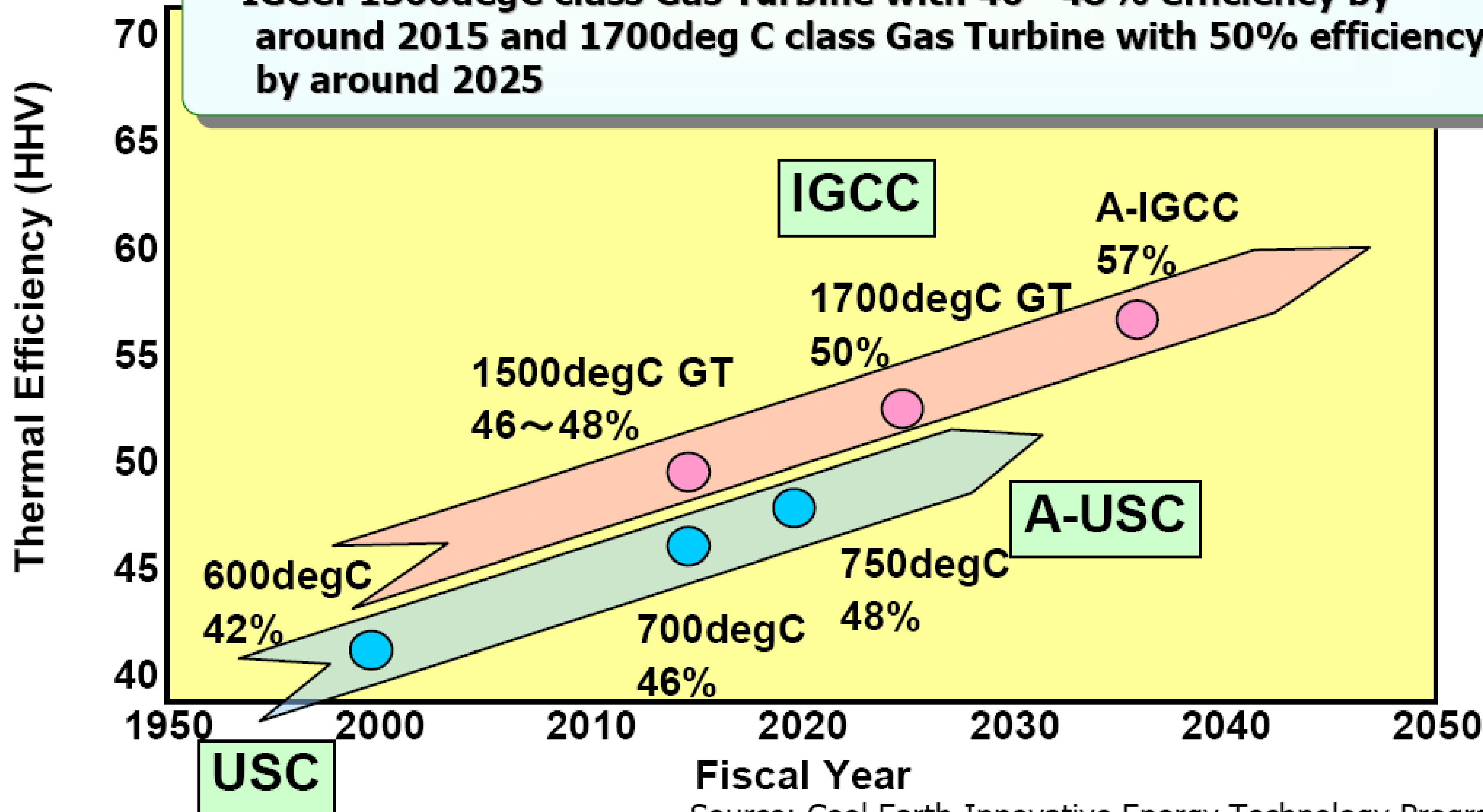


Combustion Technology University Alliance Workshop, August 4, 2003, Columbus, OH

EPRI

Roadmap for High-efficiency coal-fired power generation- Japan

- **A-USC: 700deg C class Turbine with 46% efficiency by around 2015 and 48% efficiency by around 2020**
- **IGCC: 1500degC class Gas Turbine with 46～48% efficiency by around 2015 and 1700deg C class Gas Turbine with 50% efficiency by around 2025**



Source: Cool Earth-Innovative Energy Technology Program, March 2008 Ministry of Economy, Trade and Industry

Roadmap for High-efficiency coal-fired power generation- EPRI

PC Plant Evolution to 2020 and Beyond

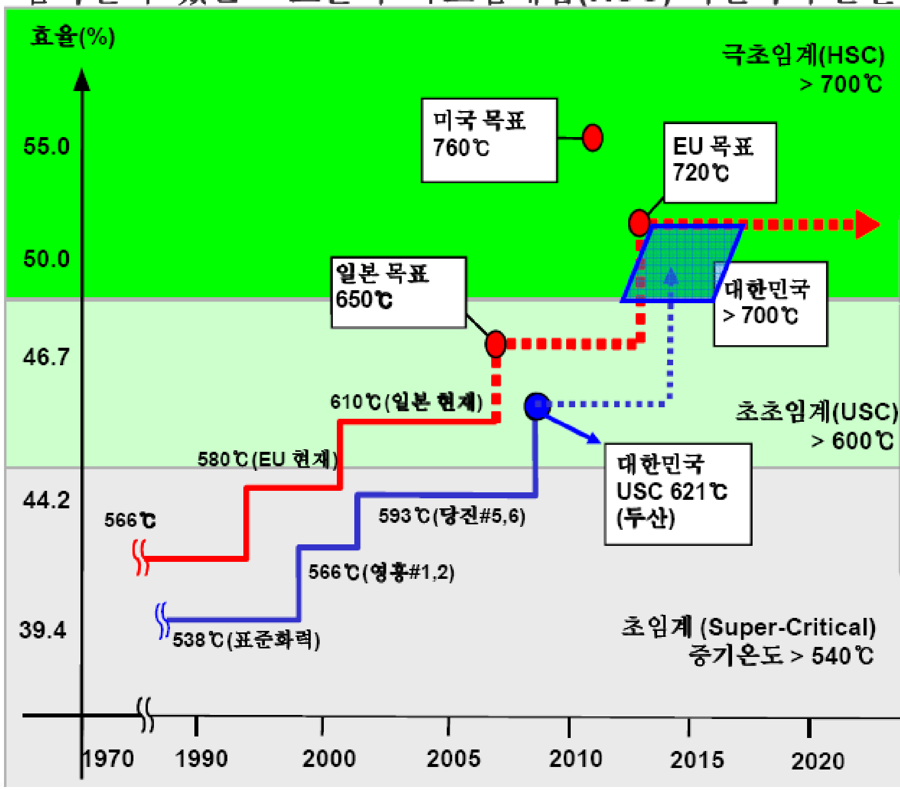
NOW	*NOW*	*NOW* *to 2010*	*FUTURE* *2010–20*	*FUTURE* *2020+*
• **Subcritical** • **Supercritical**	• **Subcritical** • **Supercritical**	• **USC** • **800 MW±**	• **USC** • **800 MW±**	• **USC** • **800 MW±**
Existing PC Plants w/ Changes 537°C–565°C →	**Current-Generation PC Plant Designs 537°C–582°C** →	**Advanced PC Plant 600°C/ 241 Barg** →	**Advanced PC Plant 650°C/ 276 Barg** →	**Advanced PC Plant 700°C/ 310 Barg+**
• CO_2 Capture Issue • Emissions Issue	• SOA Design • CO_2 Capture <u>Accommodation</u> • Future Emissions Reductions	• Advanced Ferritic Materials • CO_2 Capture <u>Accommodation</u> • Future Emissions Reductions	• Nickel / Austenitic Materials • CO_2 Capture <u>Ready</u> • Advanced Emission Controls	• Nickel Alloys • CO_2 Capture <u>Capable</u> • Near-Zero Emissions

EPRI ELECTRIC POWER RESEARCH INSTITUTE

Roadmap for High-efficiency coal-fired power generation- Korea

HSC(Hyper Super Critical) 발전기술

두산중공업은 기 확보한 USC 발전 기술을 바탕으로 기존 표준화력발전 대비 온실가스 배출량을 22%
감축할 수 있는 고효율의 극초임계압(HSC) 석탄화력 발전기술을 개발 예정임



추진현황

- 타당성 연구 및 개발전략 수립
- 해외 Network을 통한 정보수집
- HSC 소재기술개발 계획 수립

향후 계획

- 소재기술 : '11.01 ~ '16.12
- 보일러, 터빈 등 주기기 기술개발
  : '11.01 ~ 14.12
- 실증 Test

- 고효율 친환경 석탄화력발전 기술확보로 국내 발전소 CO_2 배출량 감축 목표 달성
- 선진국 수준의 독자기술로 해외시장에서 Global Leader 지위 획득

Source : 두산중공업 동남권 기업간담회 발표자료, 김정태 상무

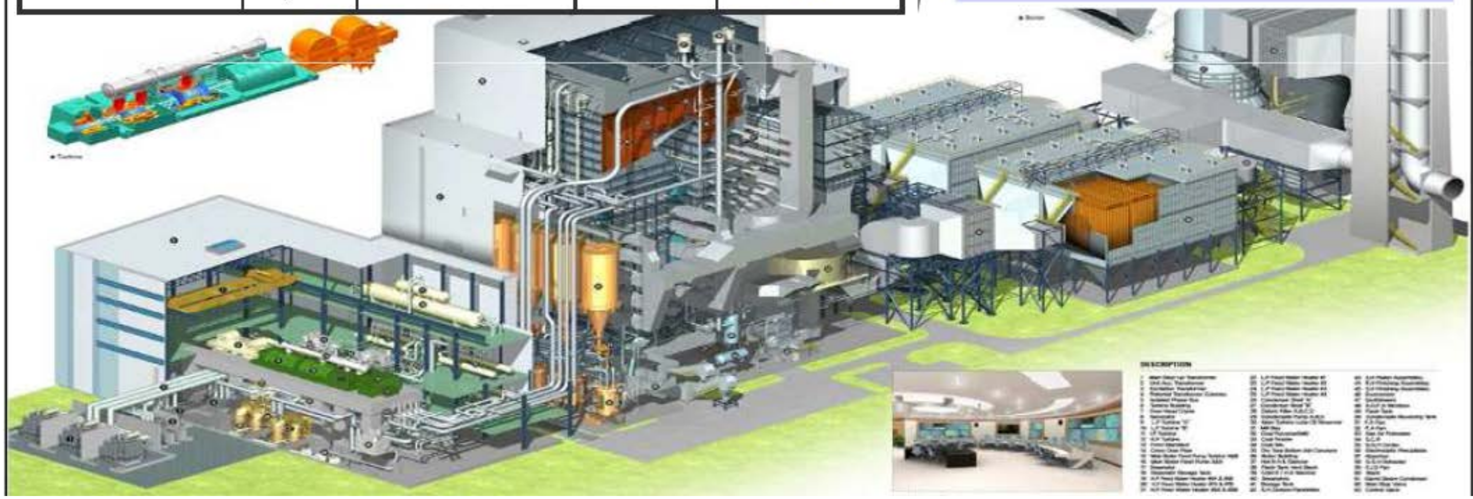
Roadmap for High-efficiency coal-fired power generation- Korea

USC(Ultra Super Critical) 발전기술

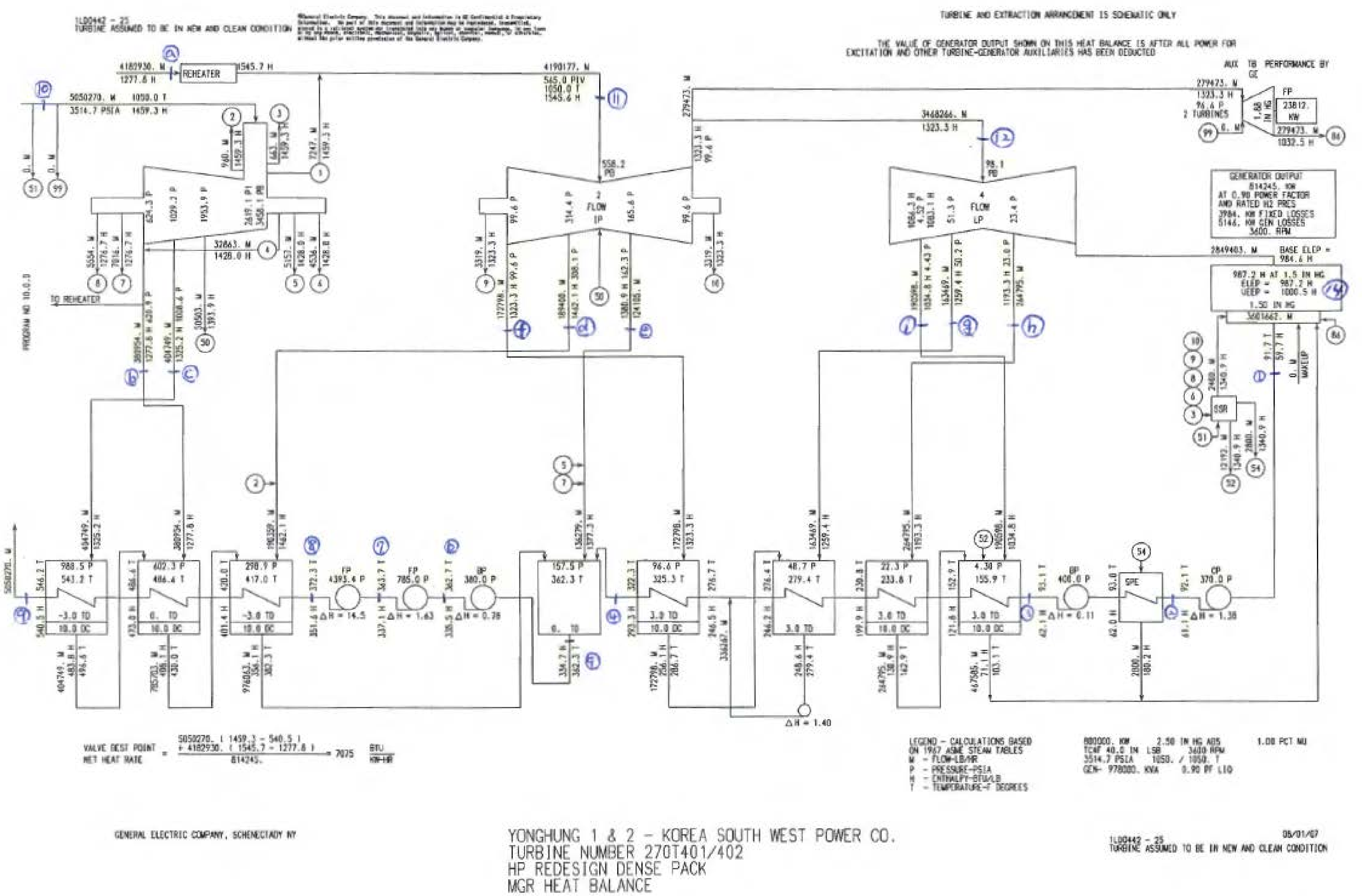
두산중공업은 국책과제를 통해 세계 최고수준의 친환경, 고효율, 대용량 1000MW급 초초임계압(USC) 석탄화력발전 주기기 설계 및 제작 기술의 독자 개발에 성공하여 상용화 준비 중임

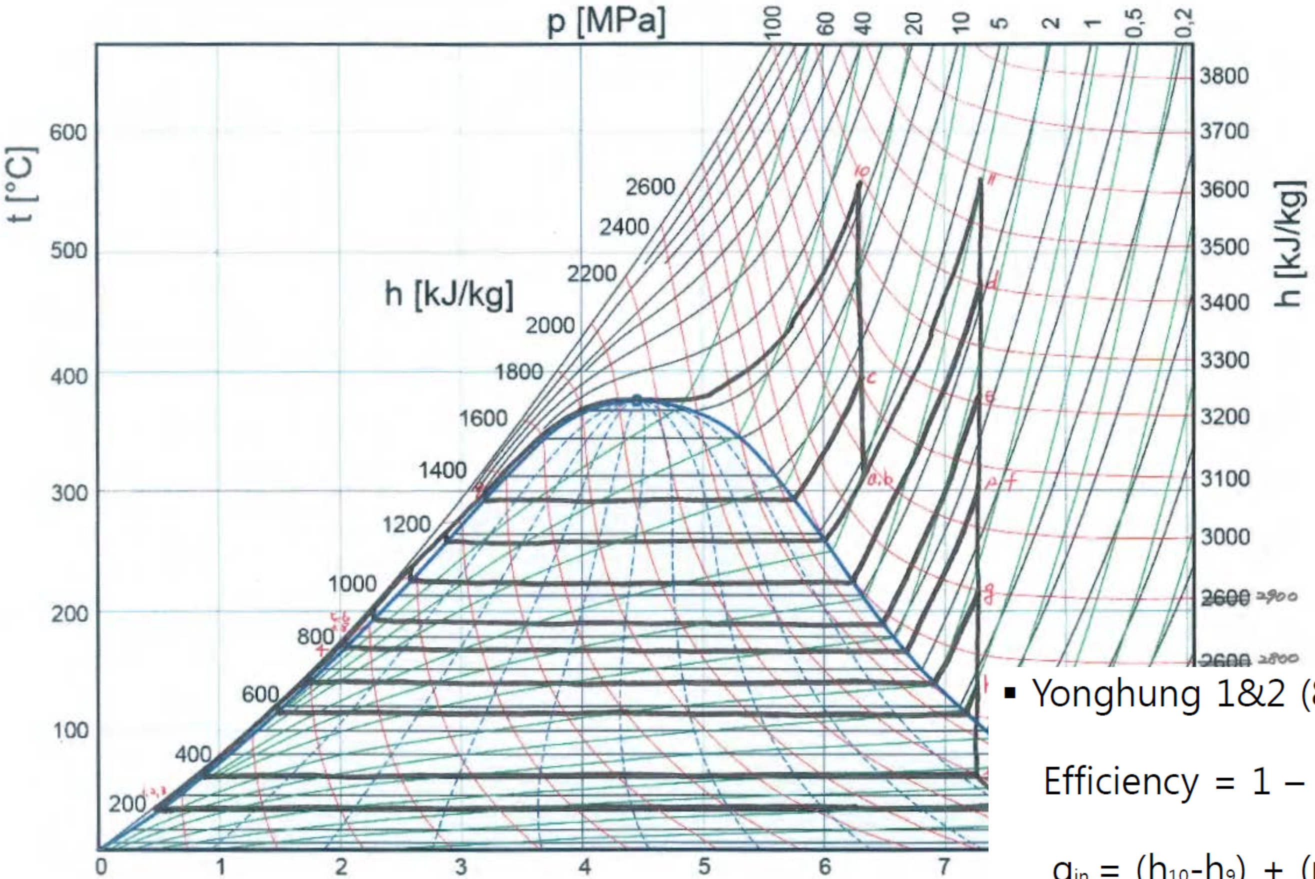
경쟁사 비교	발전용량 (MW)	증기조건[1] (kg/cm2/℃/℃/℃)	발전효율[2] (%)	CO2 배출량[3] (만톤/yr)
국내 개발 기술	1,000	265/610/621	Min. 44.4	Base
MHI	1,050	256/600/610	42.1	+32
ALSTOM	1,000	278/580/600	42~43	+16

- 일본과 유럽에서 가동중인 600°C급 USC 석탄화력발전에 비해 효율이 높아, 연간 32만 톤의 CO_2 배출량 감소가 예상됨



YONGHUNG 1,2 Unit, 800MW





- Yonghung 1&2 (800MW) Efficiency

Efficiency = 1 – q_{out} / q_{in}

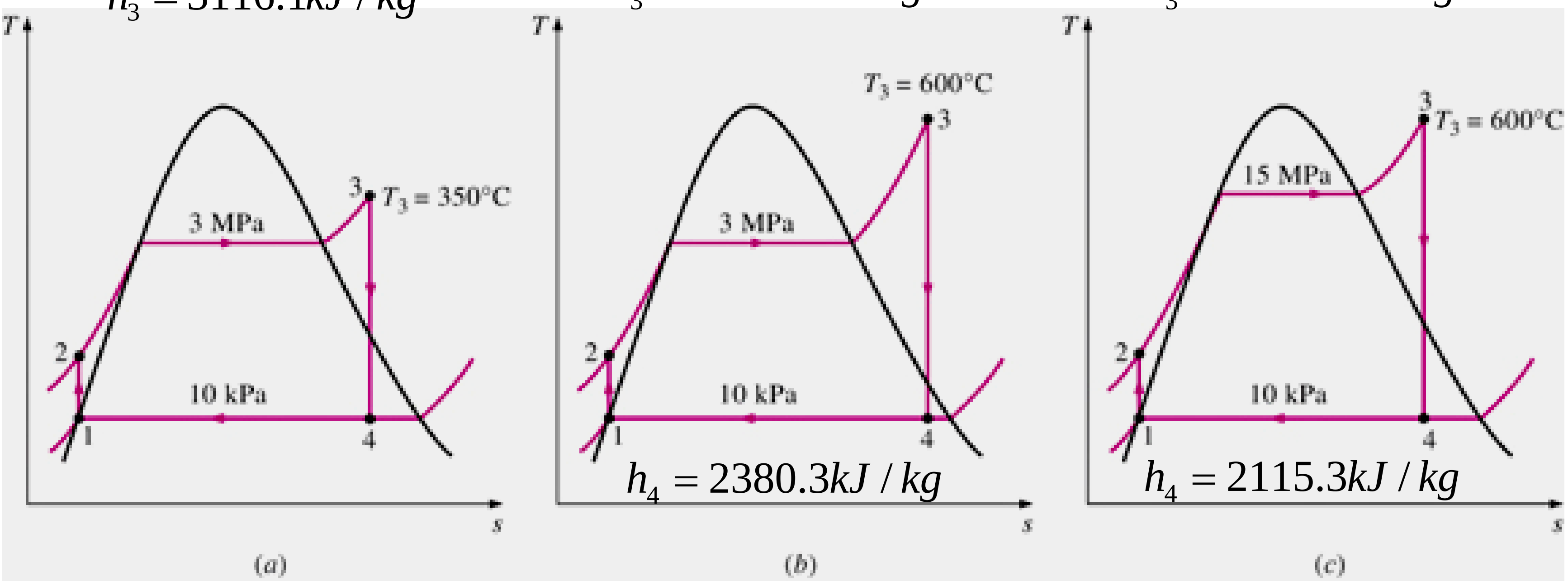
q_{in} = (h_{10}-h_9) + (Reheater유량 / Boiler유량) X (h_{11}-h_a)
= (1459.3-540.5) + (4182930/ 5050270) X (1545.6-1277.8)
= 1140.61 Btu/lb

q_{in} = (Conderser유량 / Boiler유량) X (h_{14}-h_1)
= (3601662/ 5050270) X (984.6-59.7)
= 659.60 Btu/lb

Efficiency = 1 – (659.6/1140.61) = 0.42

Ex 10-3, Effect of Boiler Pressure and Temperature on Efficiency

$h_3 = 3116.1 kJ/kg$ $h_3 = 3682.8 kJ/kg$ $h_3 = 3583.1 kJ/kg$



$h_4 = 2380.3 kJ/kg$ $h_4 = 2115.3 kJ/kg$

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 0.334$$

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 0.373$$

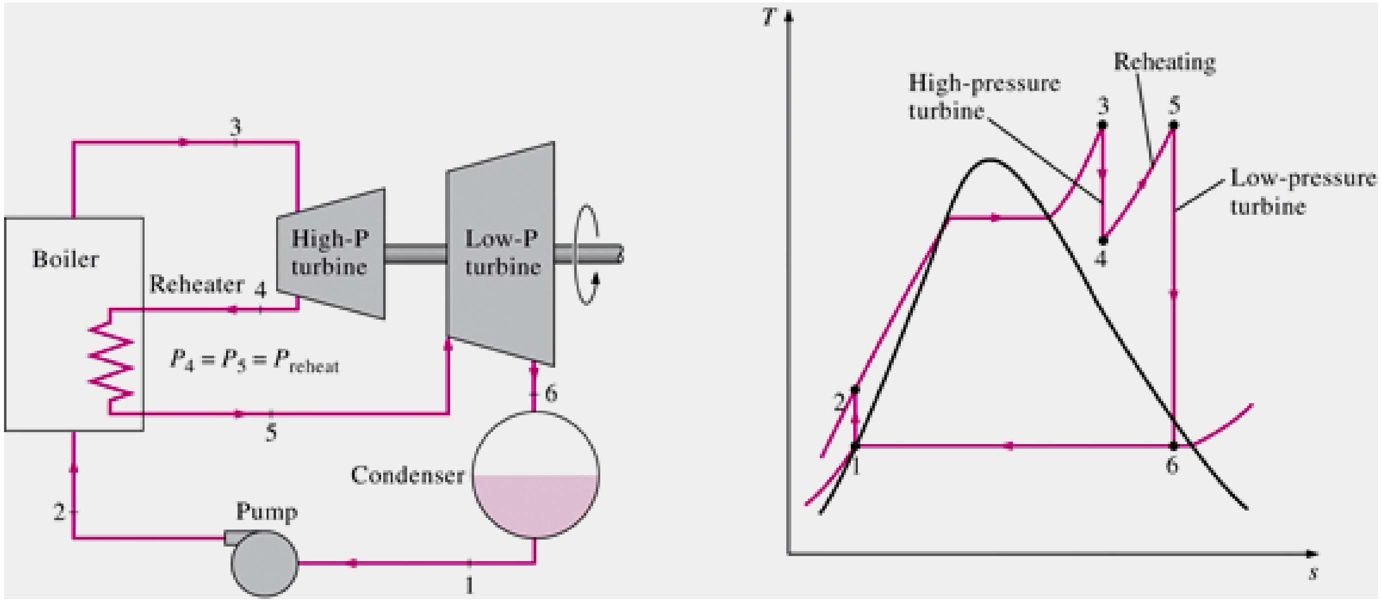
$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 0.430$$

5. The Ideal Reheat Rankine Cycle

How can we take advantage of the increased efficiencies at higher boiler pressures without facing the problem of excessive moisture at the final stages of the turbine?

1. Superheat the steam to very high temperatures. It is limited metallurgically.
2. Expand the steam in the turbine in two stages, and reheat it in between (**reheat**)

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

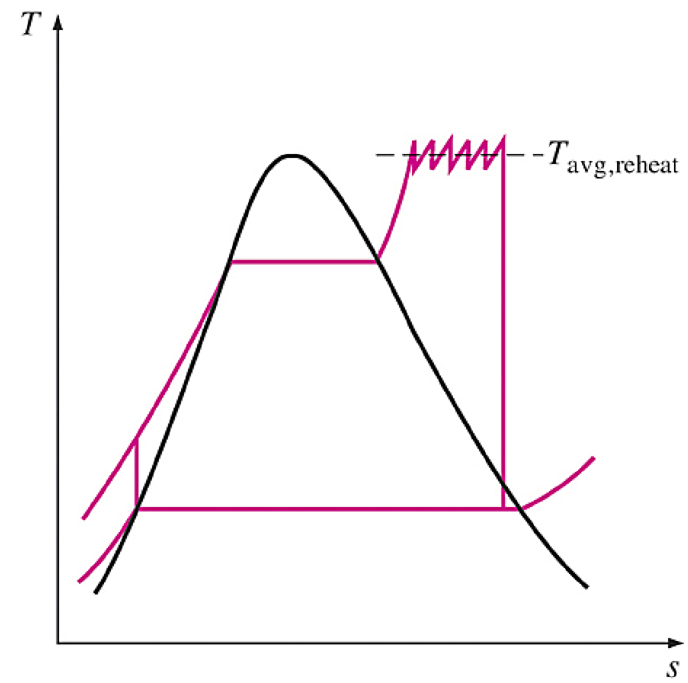


The single reheat in a modern power plant improves the cycle efficiency by 4 to 5% by increasing the average temperature at which heat is transferred to the steam.

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.



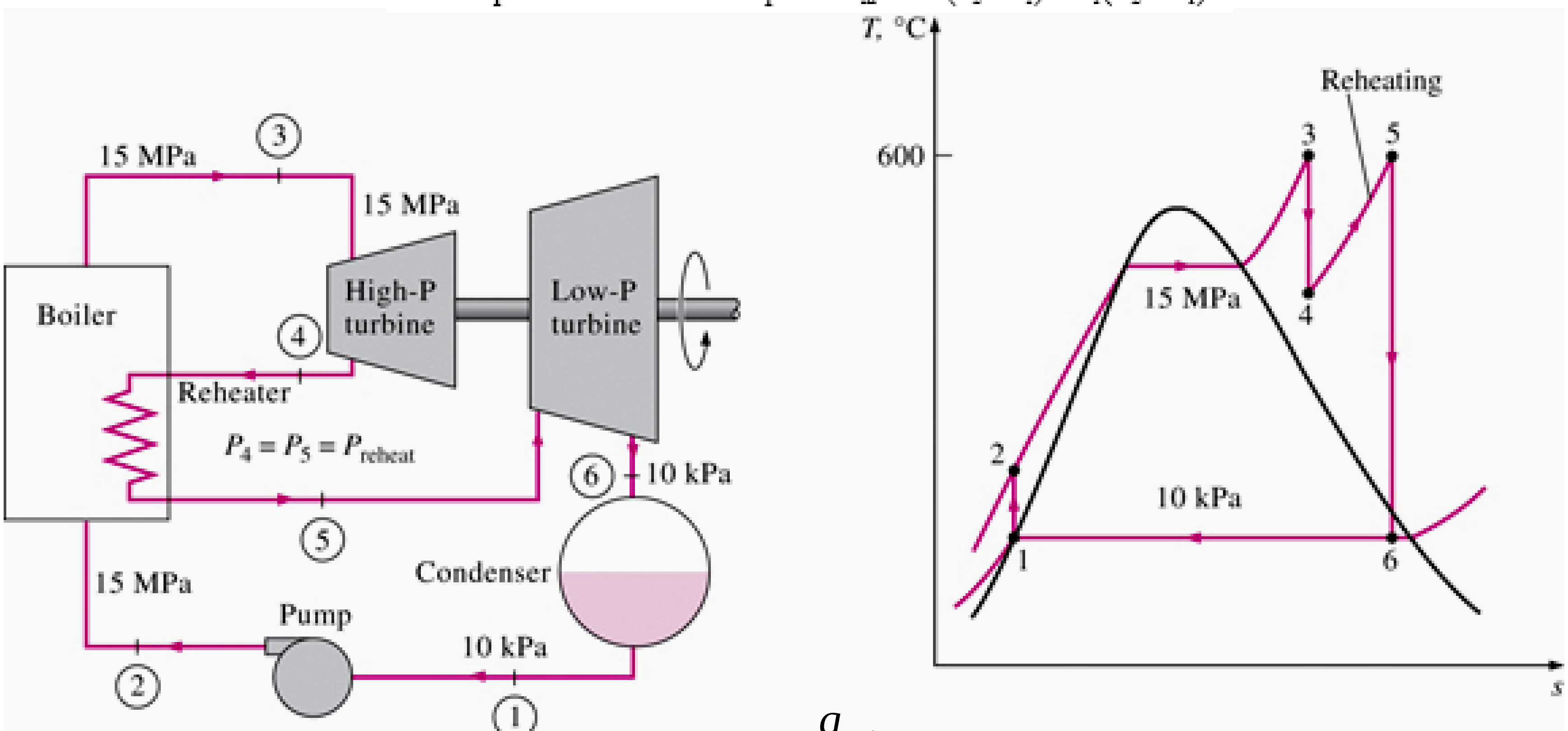
The average temperature at which heat is transferred during reheating increases as the numbero f reheat stages is increased

-> More than two reheat stages is not practical

Ex 10-4, The Ideal Reheat Cycle

Rankine Cycle with Reheat

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



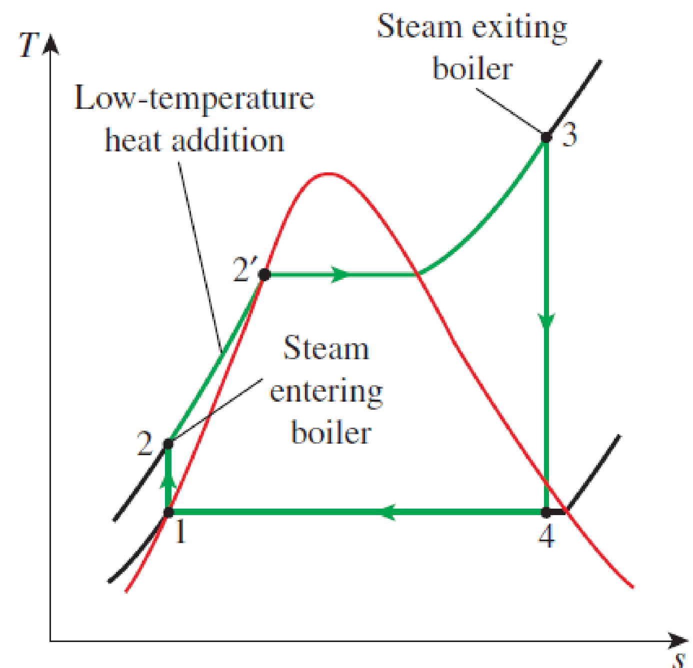
$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 0.450$$

6. The Ideal Regenerative Rankine Cycle

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

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

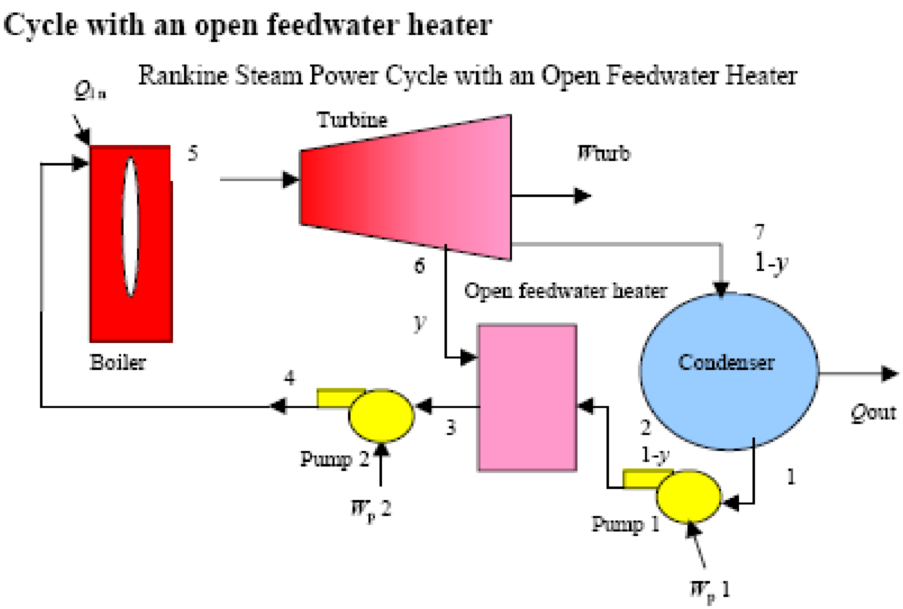
A feedwater heater is basically a heat exchanger where heat is transferred from the steam to the feedwater either by mixing the two fluid streams (**open feedwater heaters**) or without mixing them (**closed feedwater heaters**).

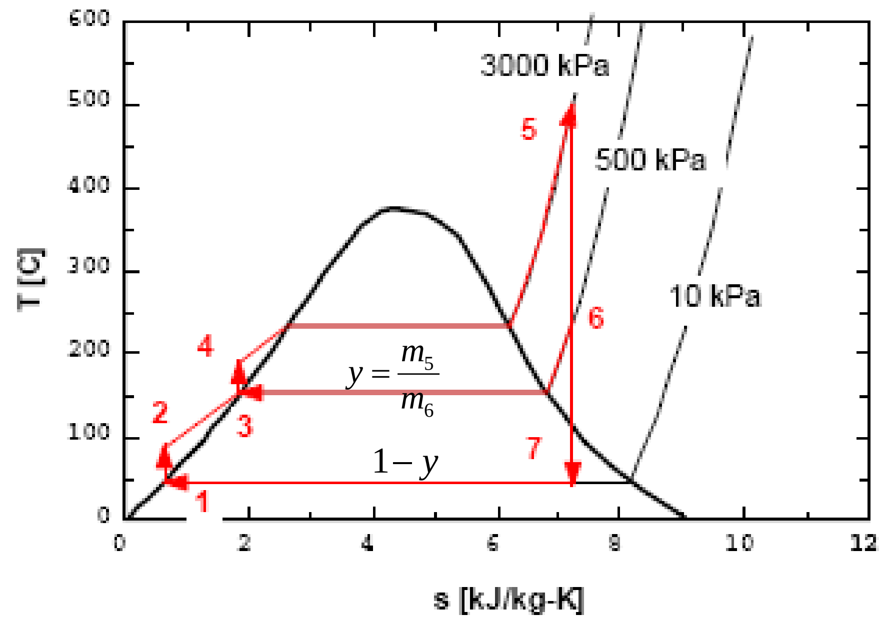


The first part of the heat-addition process in the boiler takes place at relatively low temperatures.

Open Feedwater Heaters

The ideal regenerative Rankine cycle with an open feedwater heater.





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

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

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

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

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

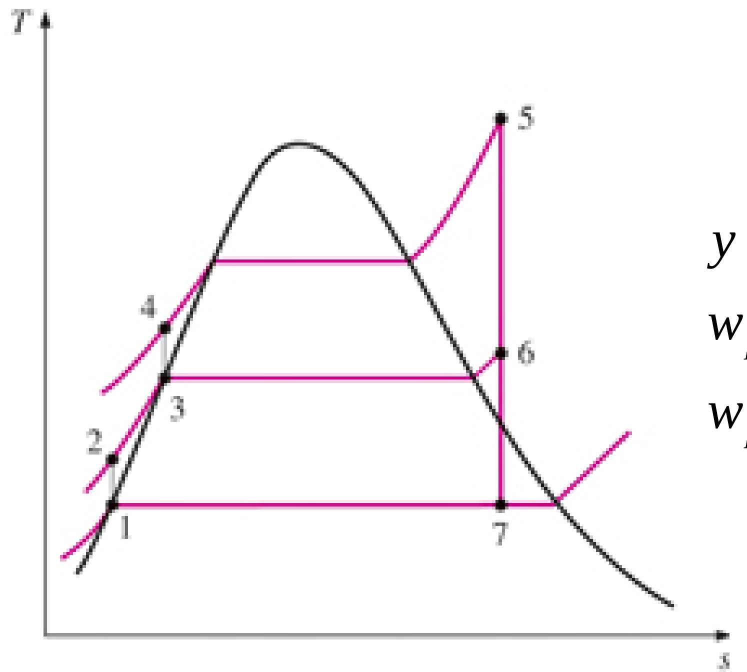
$$y\dot{m}_5 h_6 + (1-y)\dot{m}_5 h_2 = \dot{m}_5 h_3$$

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

Open Feedwater Heaters(FWH)

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

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



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

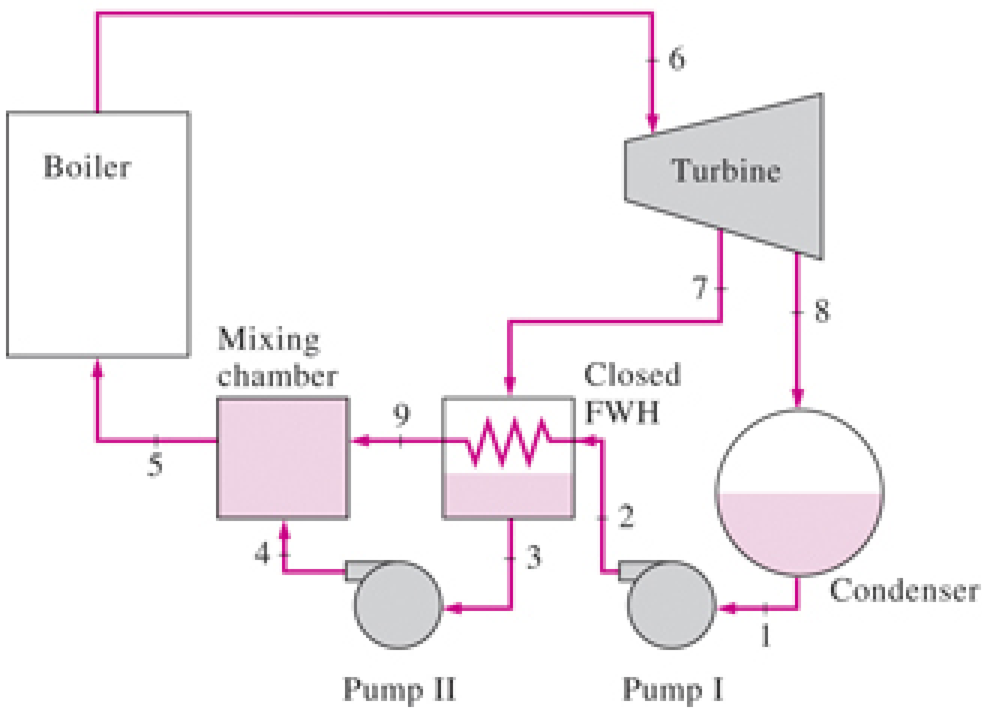
$$w_{pumpI,in} = v_1(P_2 - P_2)$$

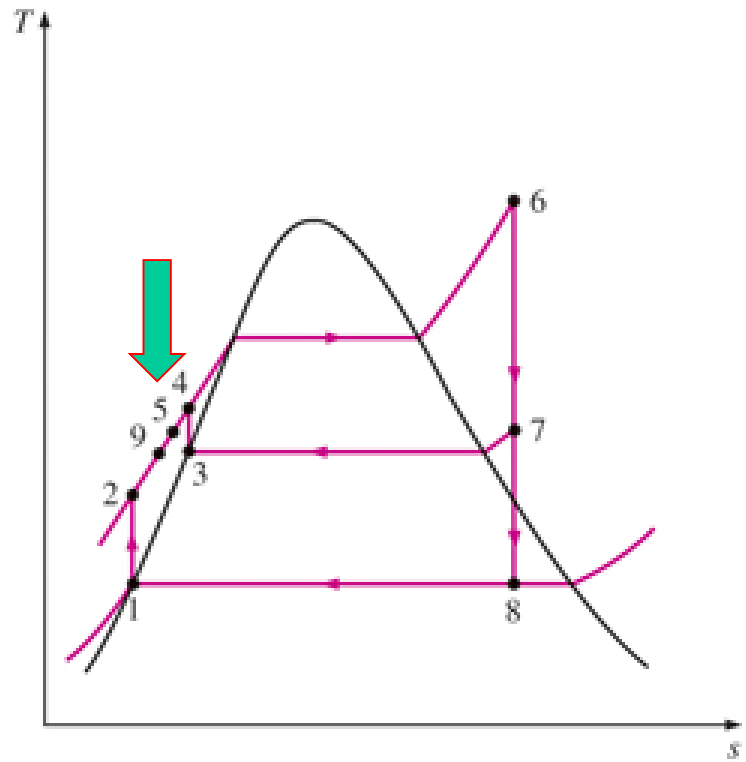
$$w_{pumpII,in} = v_3(P_4 - P_3)$$

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

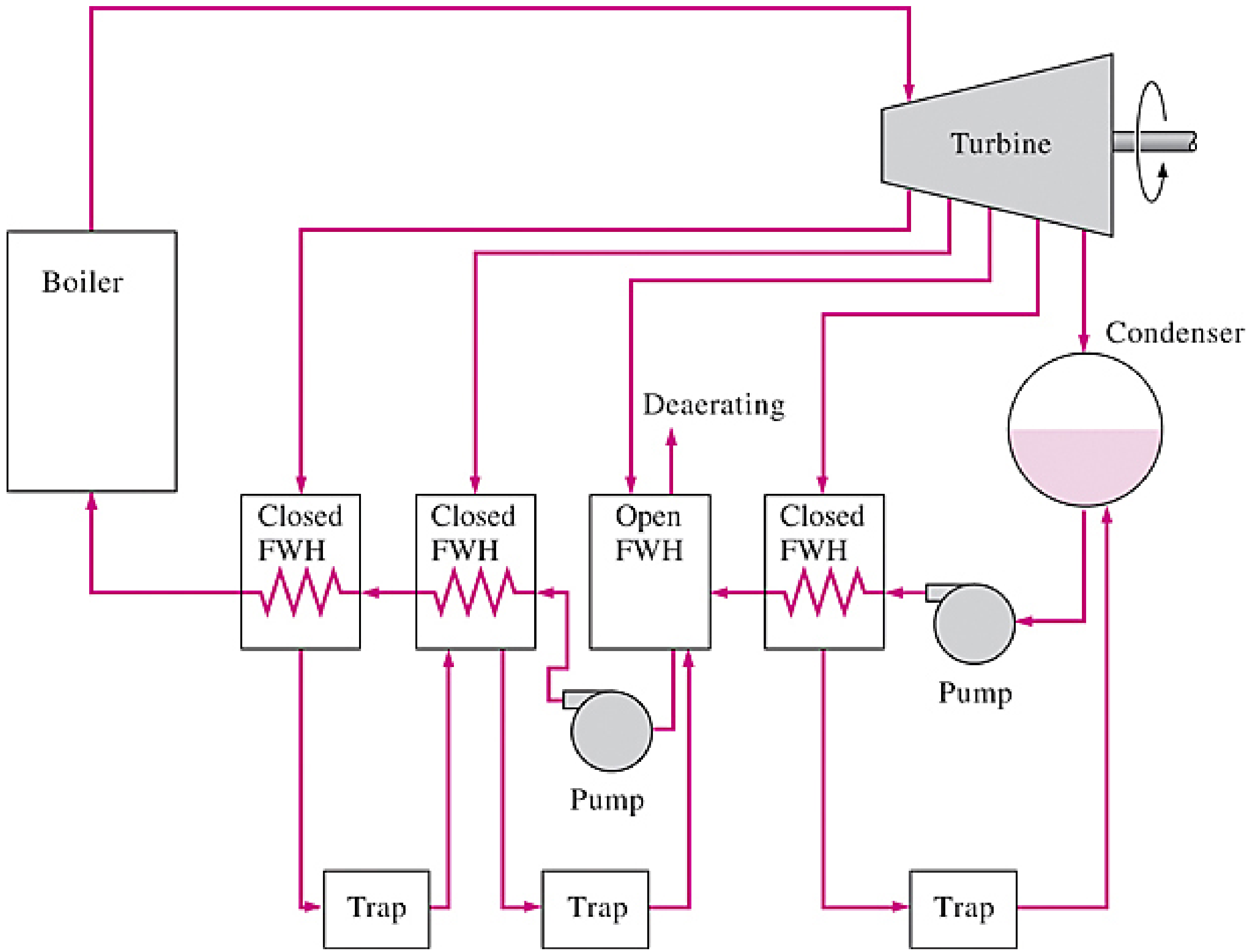
$$w_{pump,in} = (1-y)w_{pumpI,in} + w_{pumpII,in}$$

Closed Feedwater Heaters





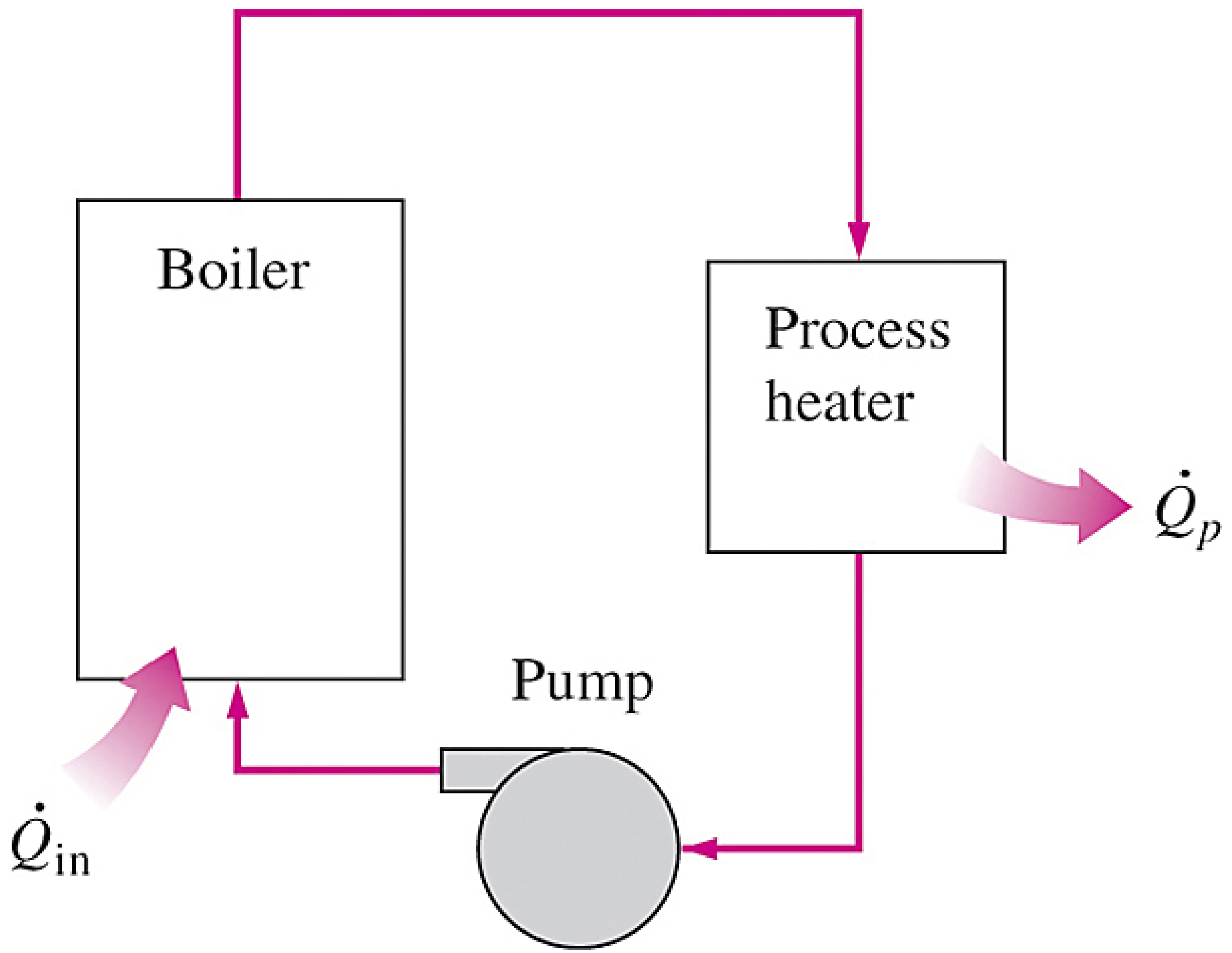
Closed Feedwater Heaters



8. Cogeneration

A simple process-heating plant

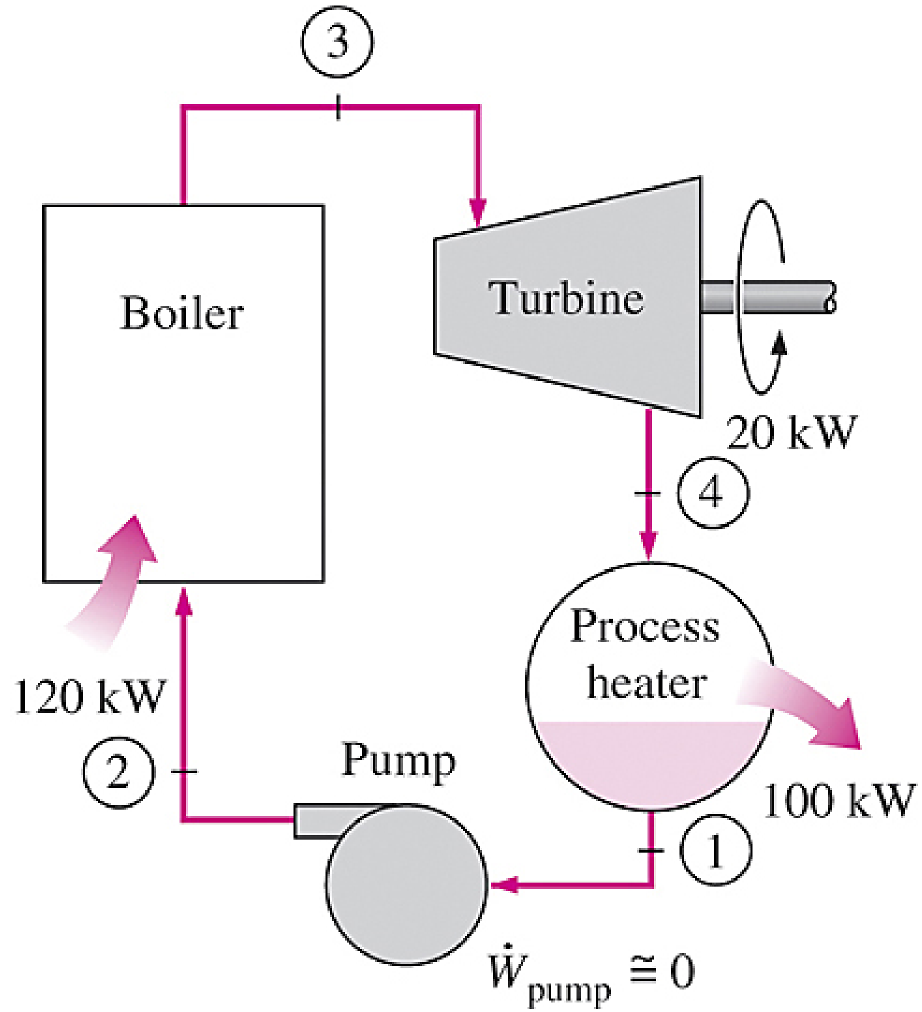
The production of more than one useful form of energy (such as process heat and electric power) from the same energy source



An ideal cogeneration plant

Utilization factor

$$\varepsilon_u = \frac{\dot{W}_{net} + \dot{Q}_P}{Q_{in}} = 1 - \frac{\dot{Q}_{out}}{Q_{in}} > 80$$



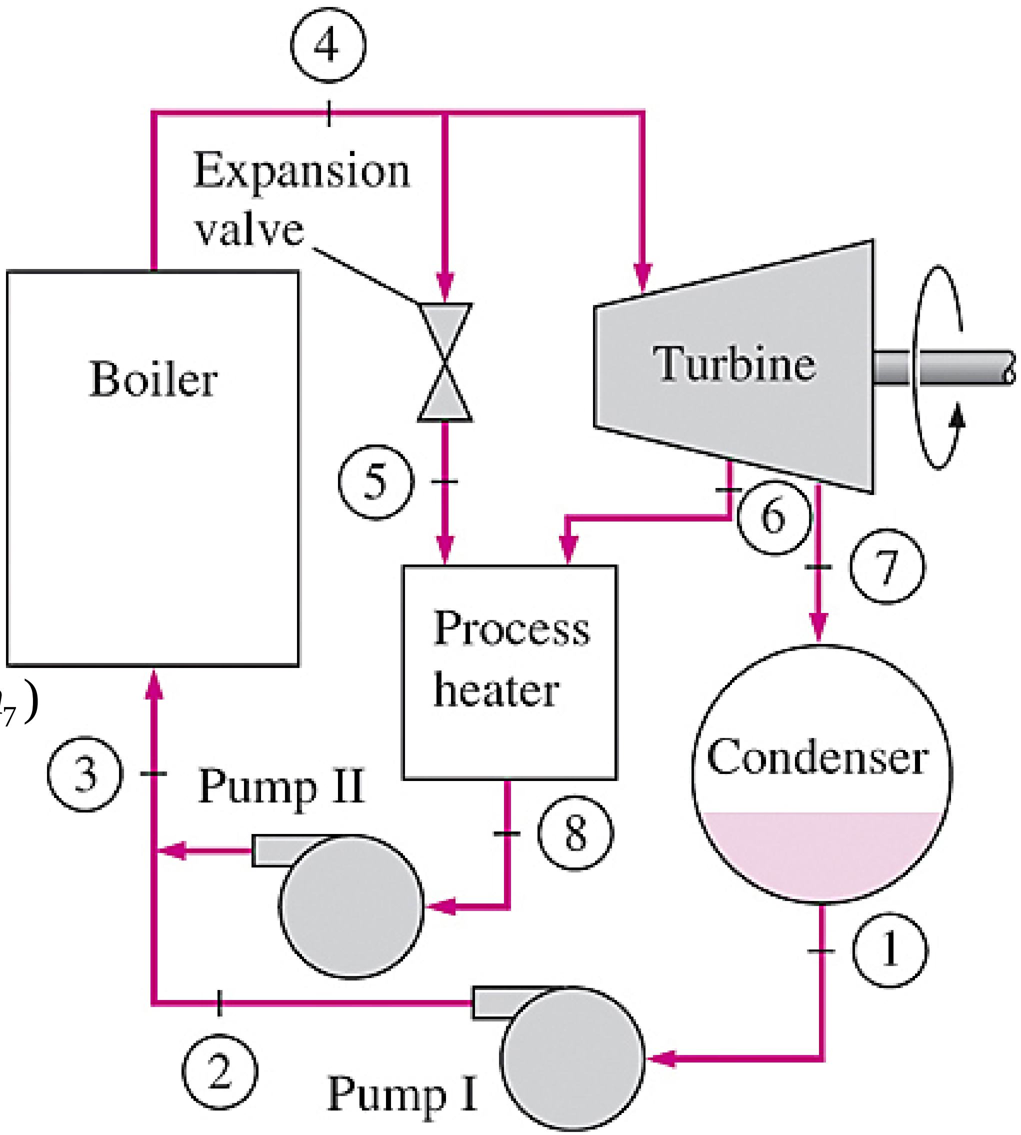
A cogeneration plant with adjustable loads

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

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

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

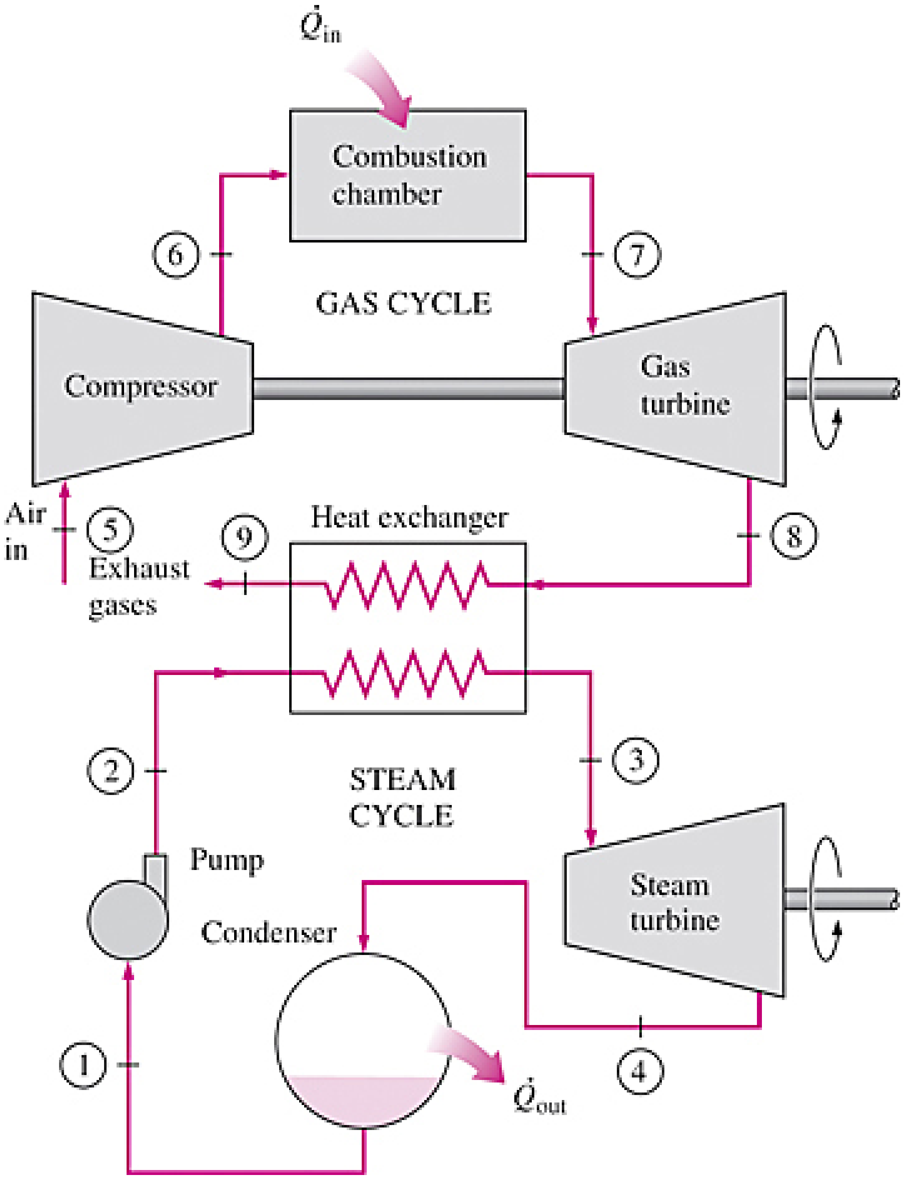
$$\dot{W}_{turb} = (\dot{m}_4 - \dot{m}_5)(h_4 - h_6) + \dot{m}_7(h_6 - h_7)$$

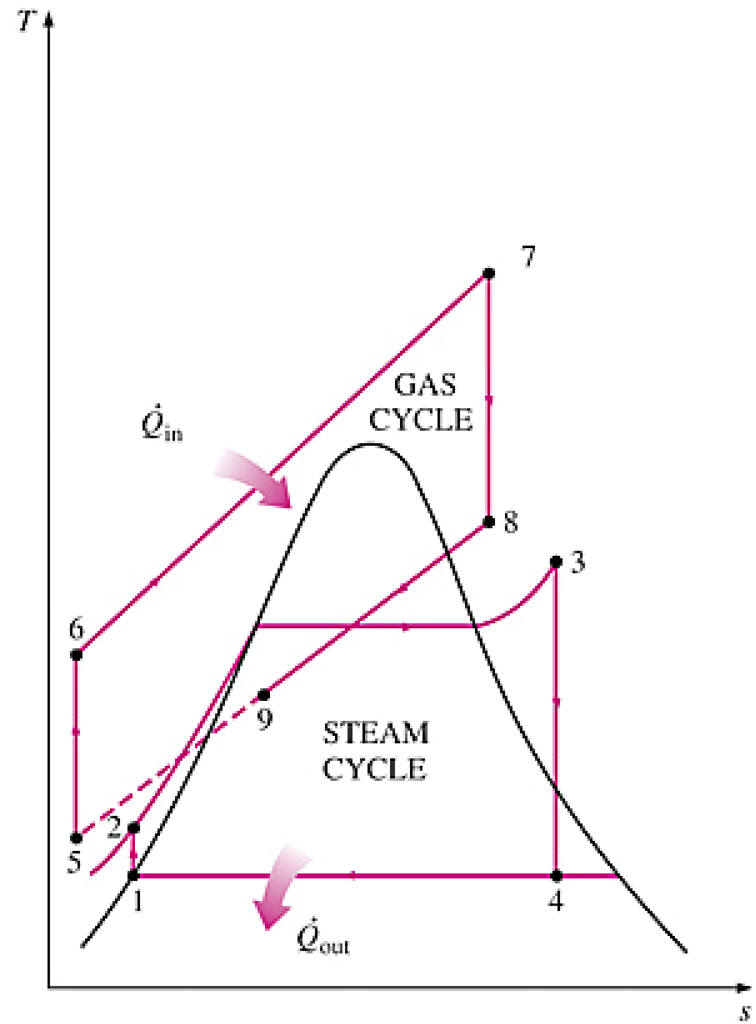


9. Combined Gas-Vapor Power Cycles

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

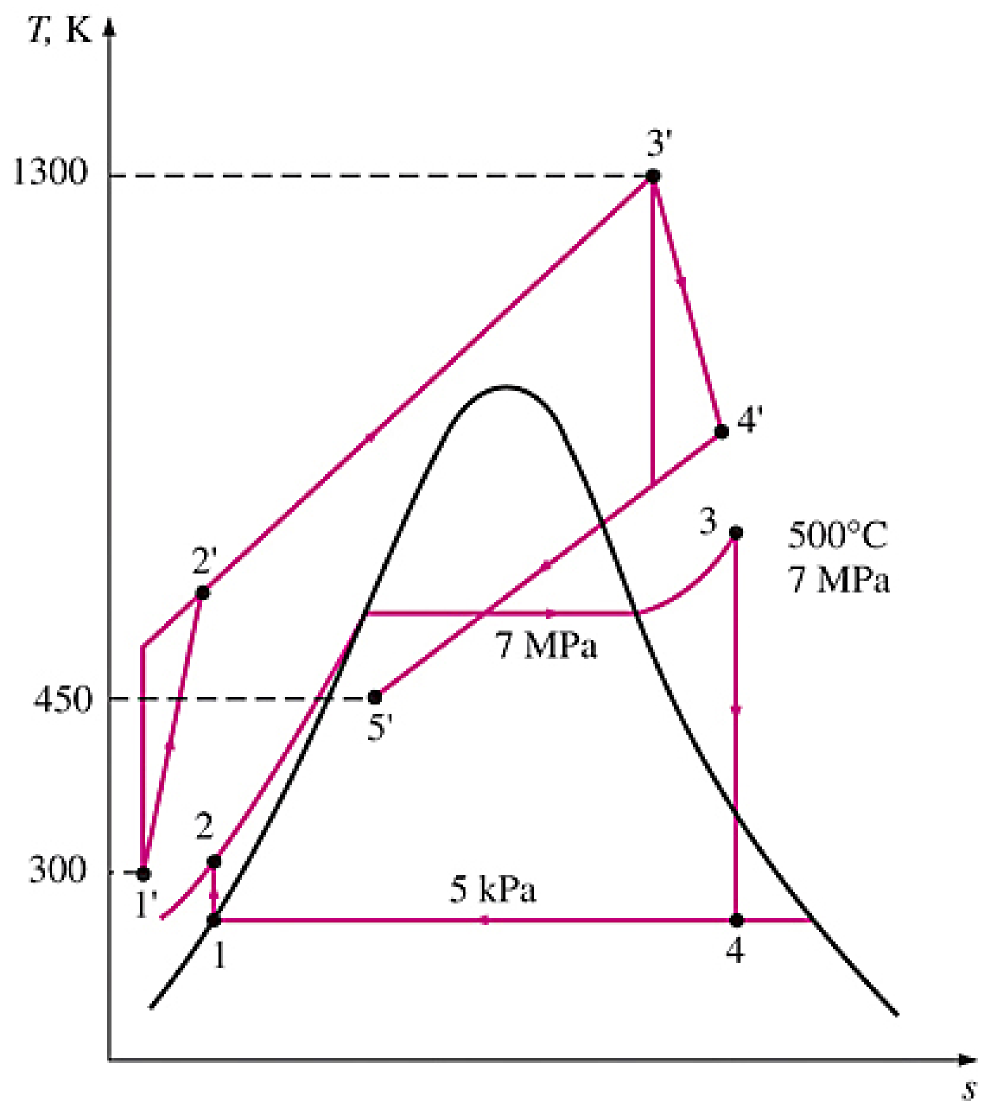
9. Combined Gas-Vapor Power Cycles



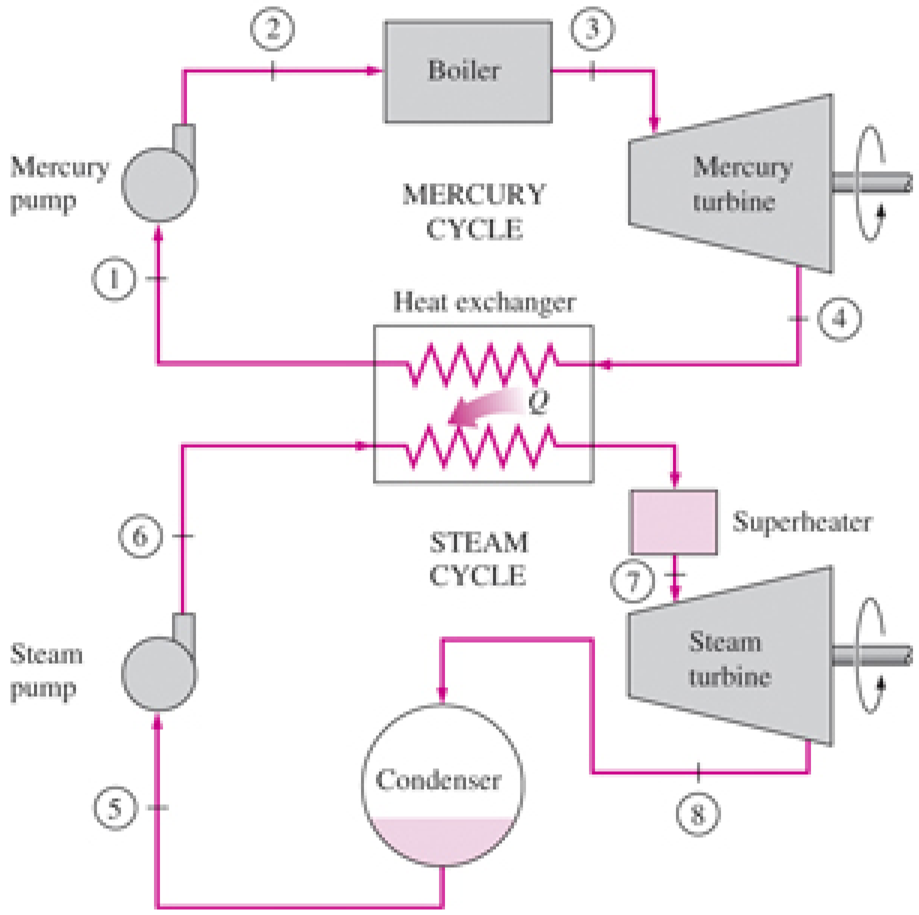


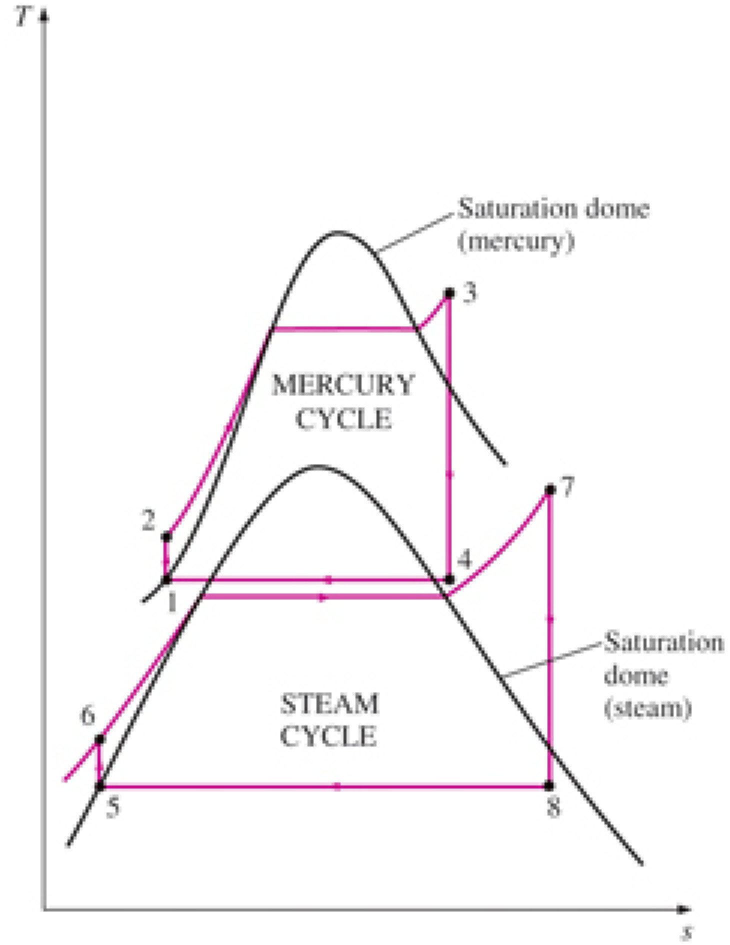
1090-MW Tohoku Combined Cycle

- 1985 Niigata Japan
- 44% thermal efficiency
- 191-MW steam turbines*2plant
- 118-MW gas turbines*6 plants
- 1154°C gas turbine inlet
- 500°C steam turbine inlet



Mercury–water binary vapor cycle



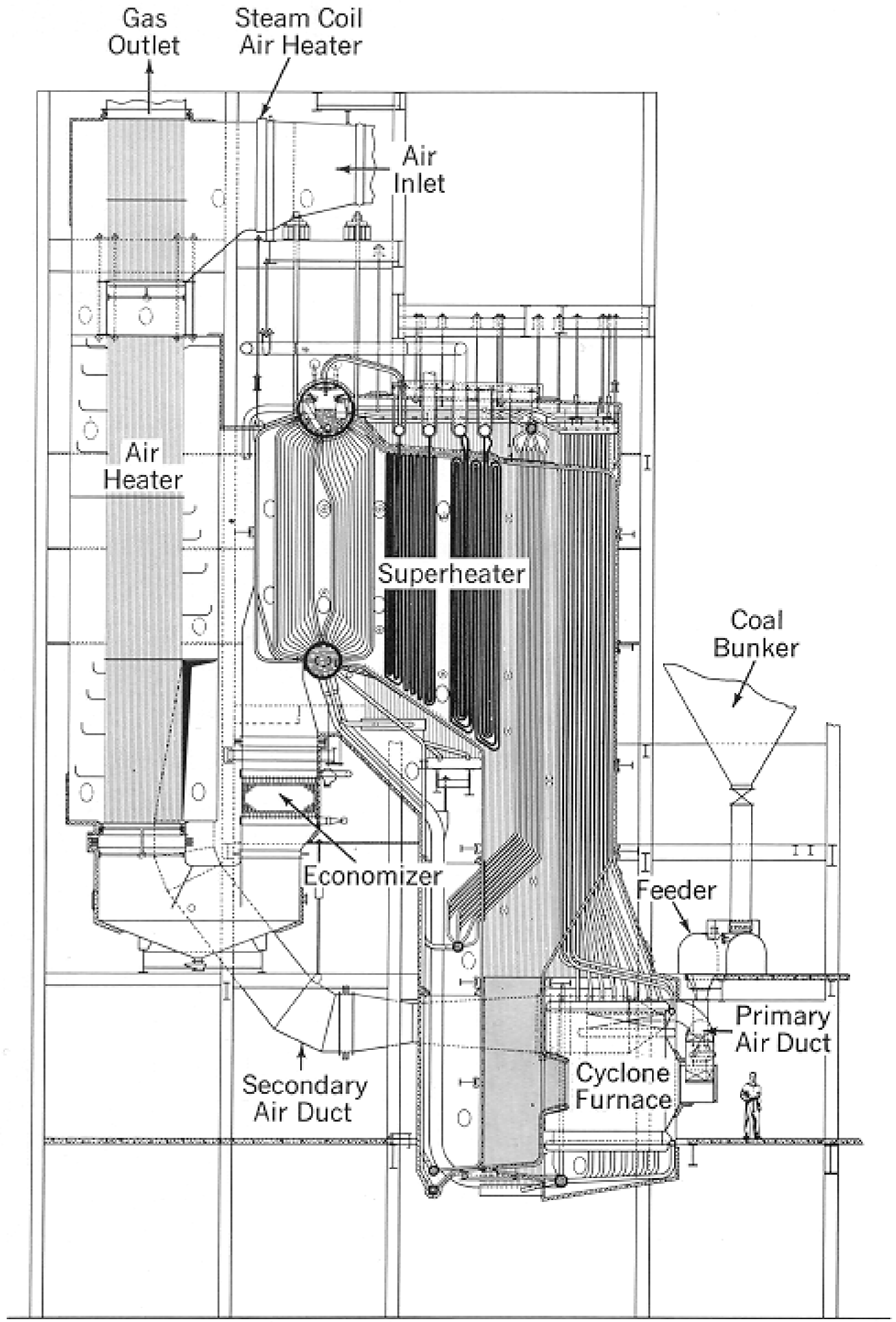


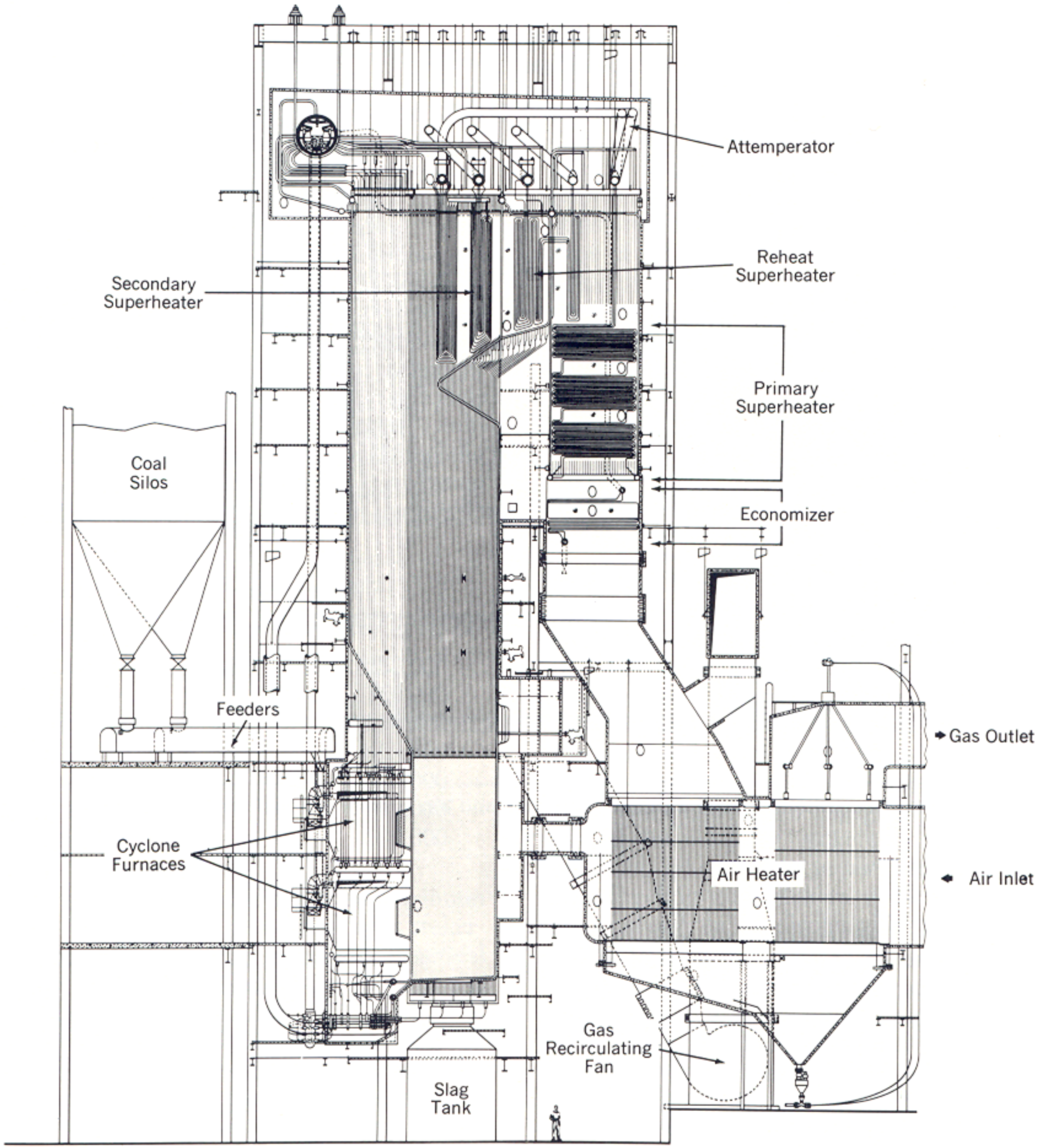
Summary

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

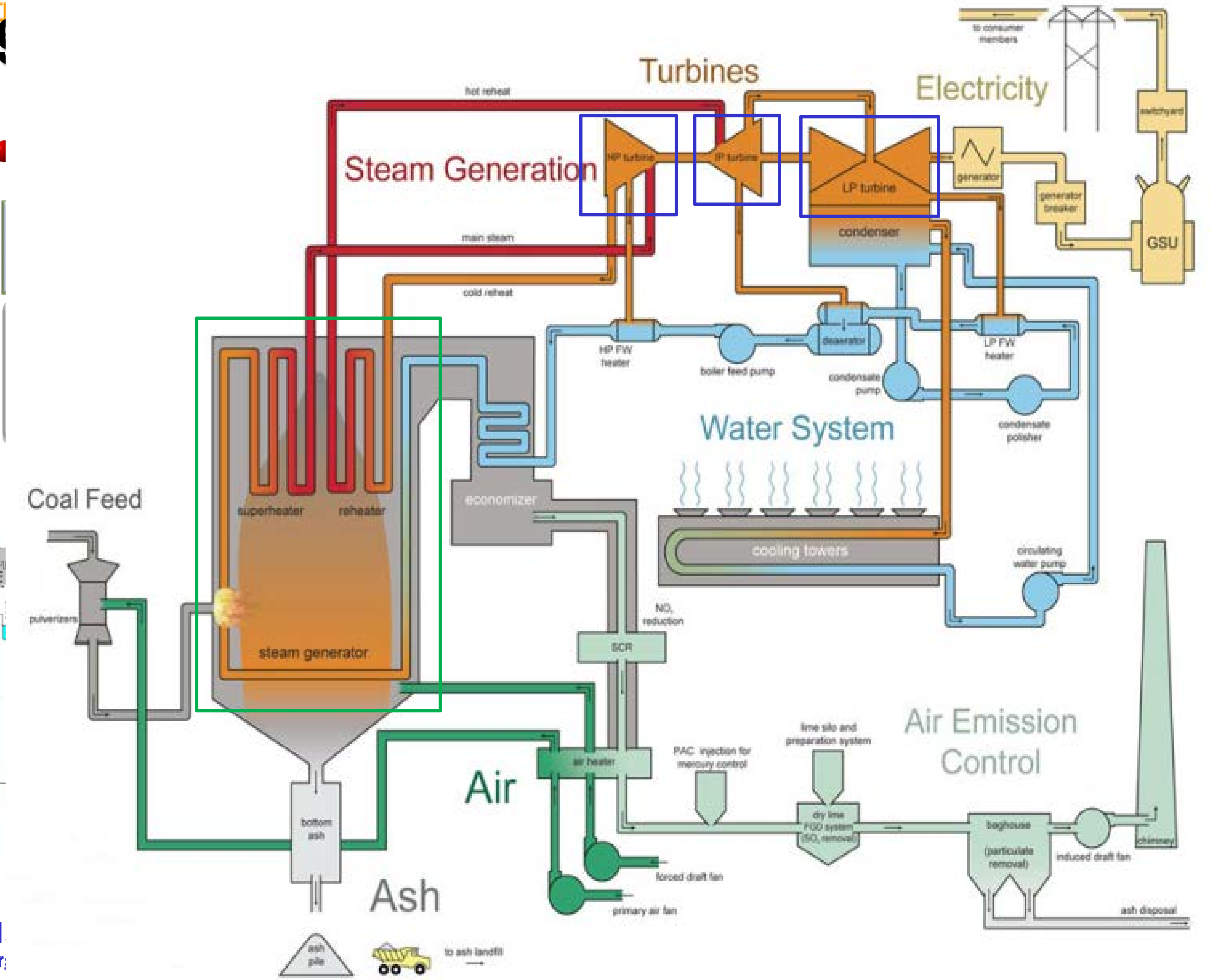
Chapter 10 Assessment

Examples and two problems in each subparts


Gas
Outlet
Steam Coil
Air Heater
Air
Inlet
Air
Heater
Superheater
Coal
Bunker
Economizer
Feeder
Primary
Air Duct
Secondary
Air Duct
Cyclone
Furnace

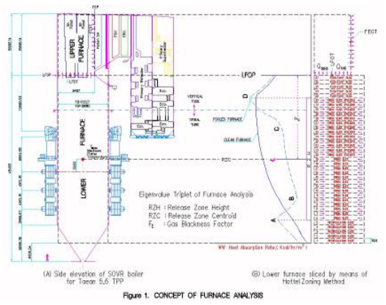


Radiant boiler with Cyclone Furnaces (one-wall) and bin system for coal preparation and feeding.

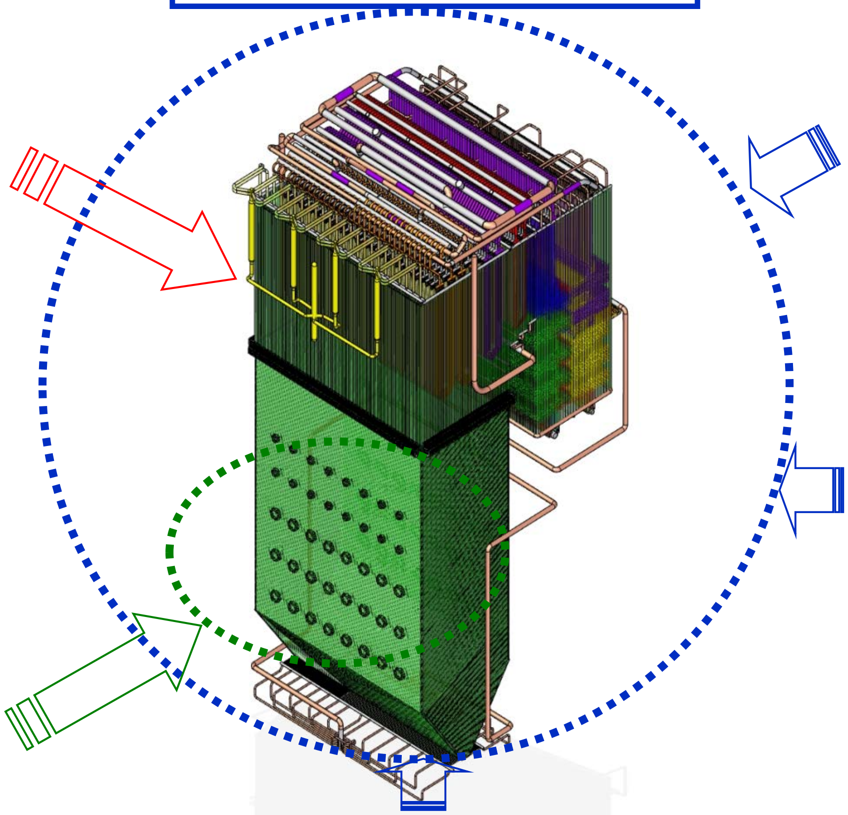

to consumer members
Turbines
Electricity
hot reheat
Steam Generation
HP turbine
IP turbine
LP turbine
generator
switchyard
generator breaker
GSU
condenser
main steam
cold reheat
HP FW heater
boiler feed pump
deaerator
LP FW heater
condensate pump
condensate polisher
Water System
superheater
reheater
economizer
cooling towers
circulating water pump
Coal Feed
pulverizers
steam generator
NOx reduction
SCR
Air Emission Control
air heater
PAC injection for mercury control
lime silo and preparation system
dry lime FGD system (SO2 removal)
baghouse (particulate removal)
induced draft fan
chimney
Air
forced draft fan
primary air fan
bottom ash
Ash
ash pile
to ash landfill
ash disposal

차세대 모델 개념/기본설계

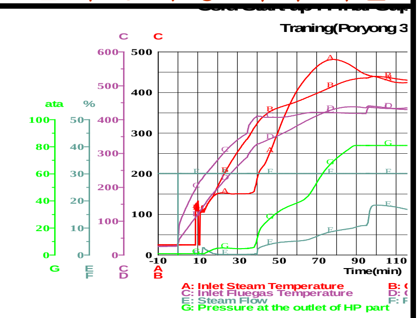
보일러 화로설계 기술



보일러 저NOx연소시스템

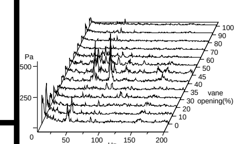


보일러 동특성 해석 기술



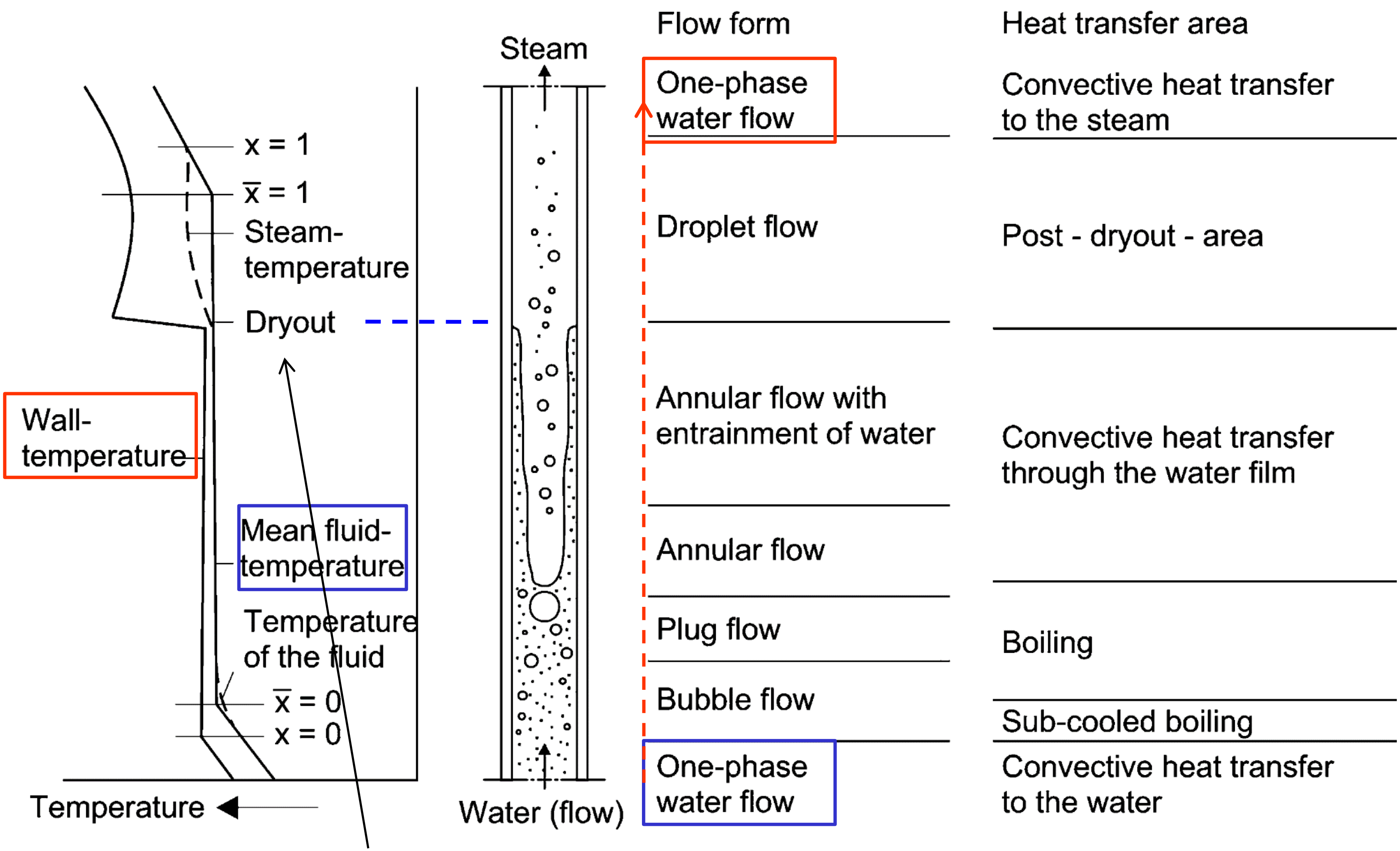
보일러 구조설계 기술

보일러 소음/진동해석기술



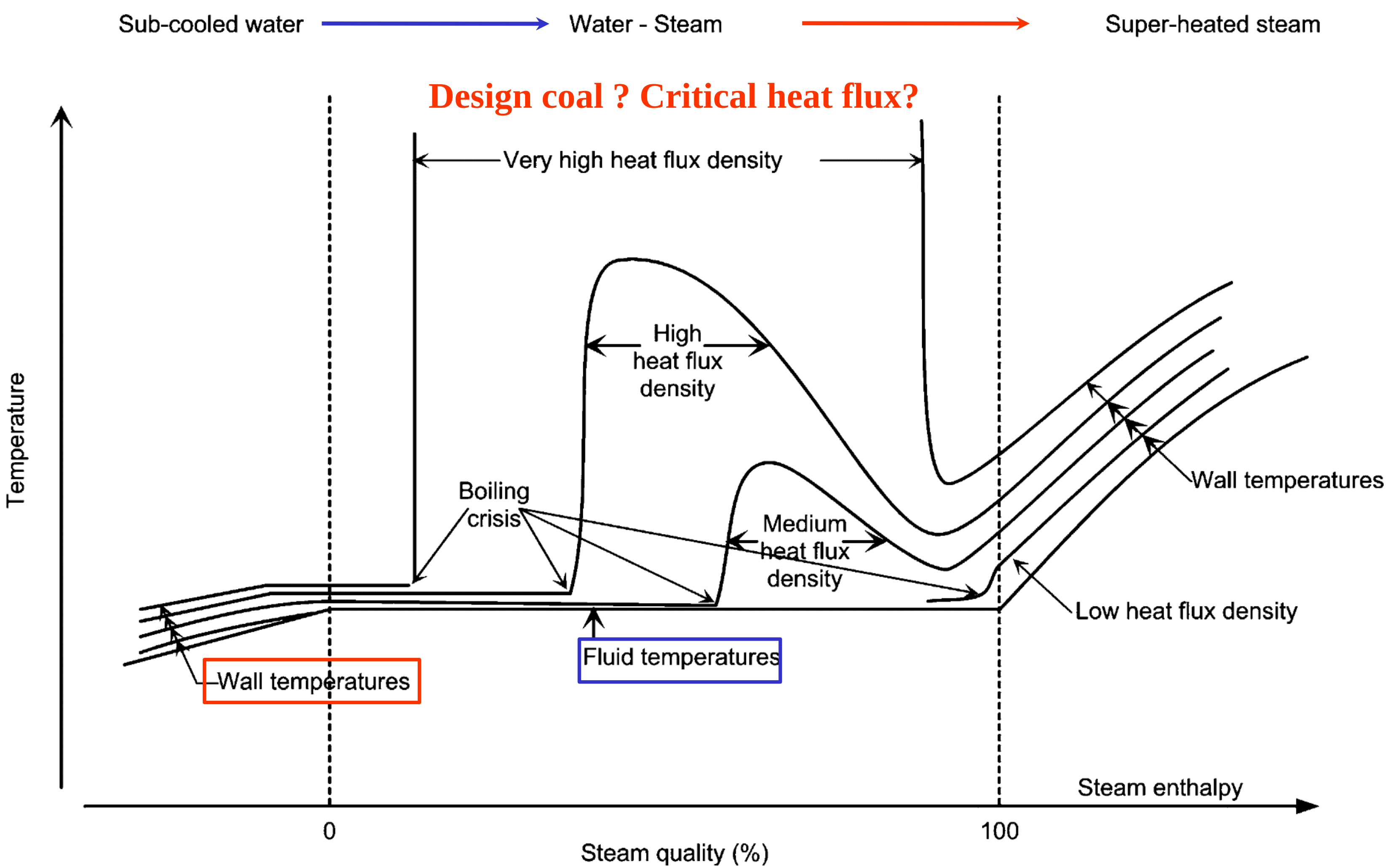
보일러 기본설계 시스템 구축

4.2.1 Flow and Heat Transfer Inside a Tube



The location of the boiling crisis and the level the wall temperature rises to depend on numerous factors, <u>such as the heat flux density, the mass flow density, the tube design and the steam quality.</u>

4.2.1 Flow and Heat Transfer Inside a Tube



For the design of steam generators, boiling crises are of great importance, because they can lead to excess temperatures in the tube walls, which have to be taken into account in the design stage.

4.2.1 Flow and Heat Transfer Inside a Tube

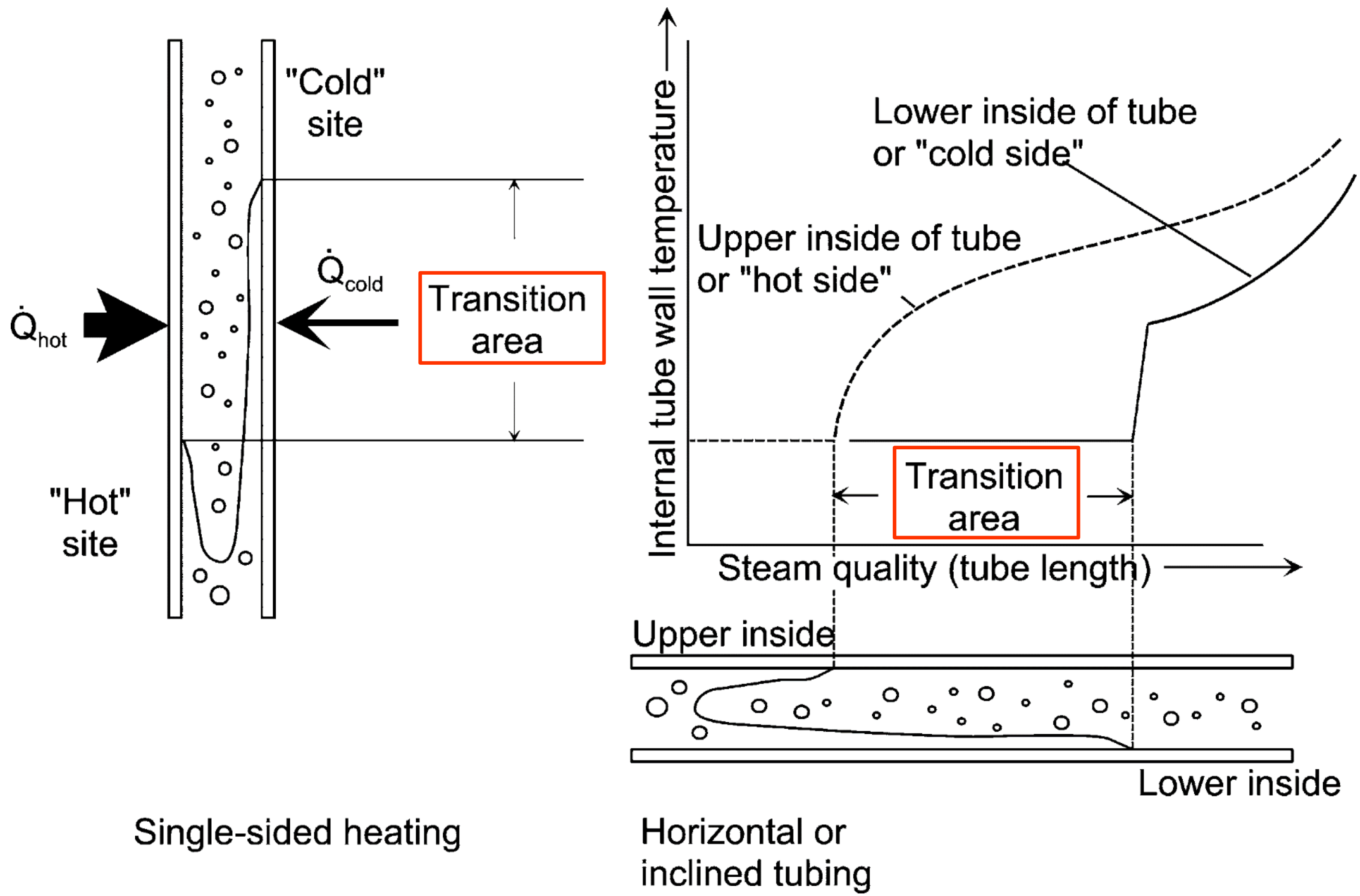
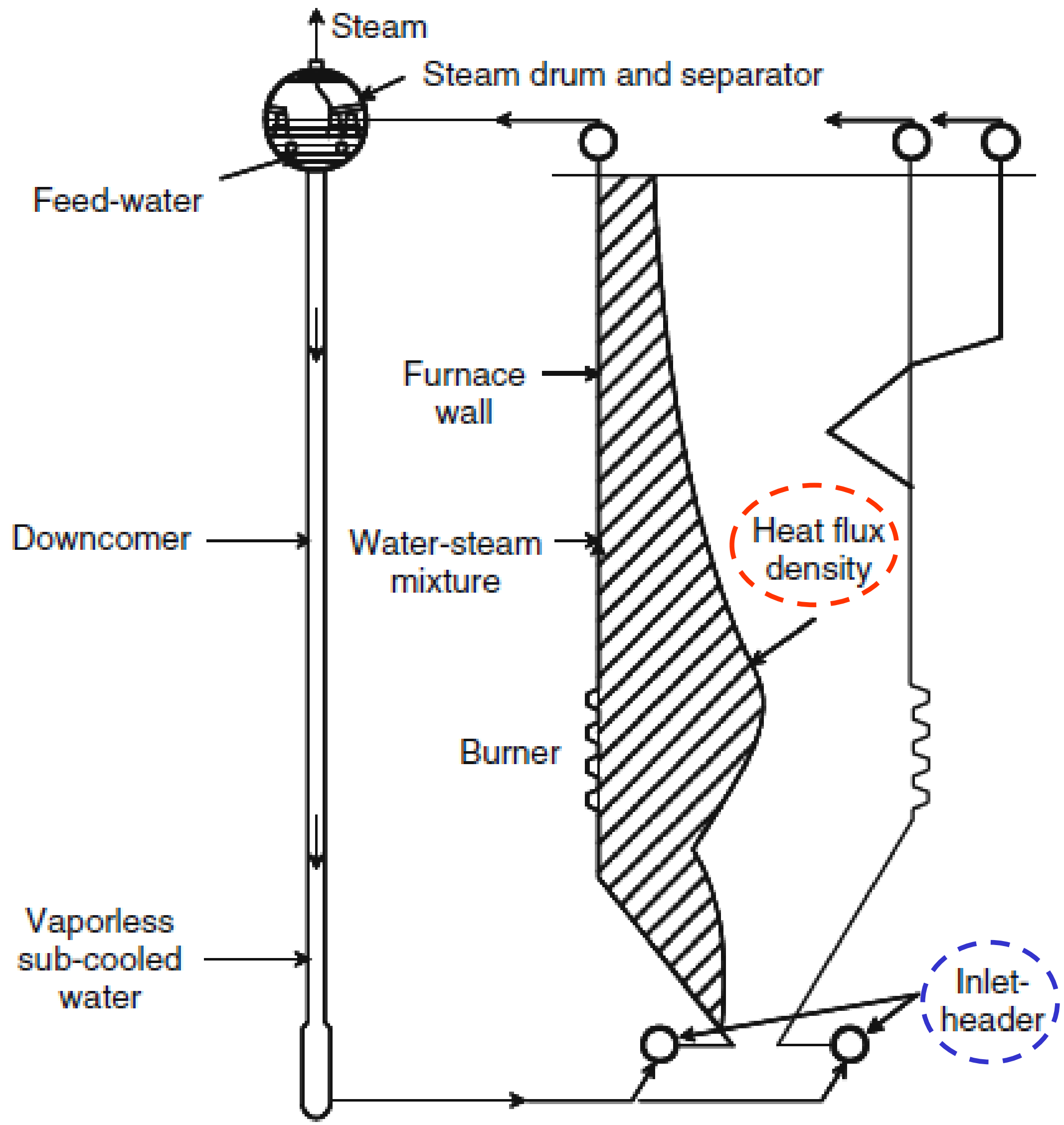


Fig. 4.12 Flow patterns and wall temperatures in a single-sided heated, horizontal or inclined evaporator tube (Kefer et al. 1990)

4.2.2.1 Natural Circulation

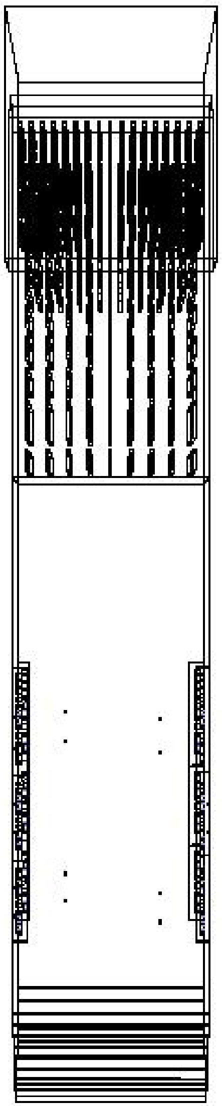


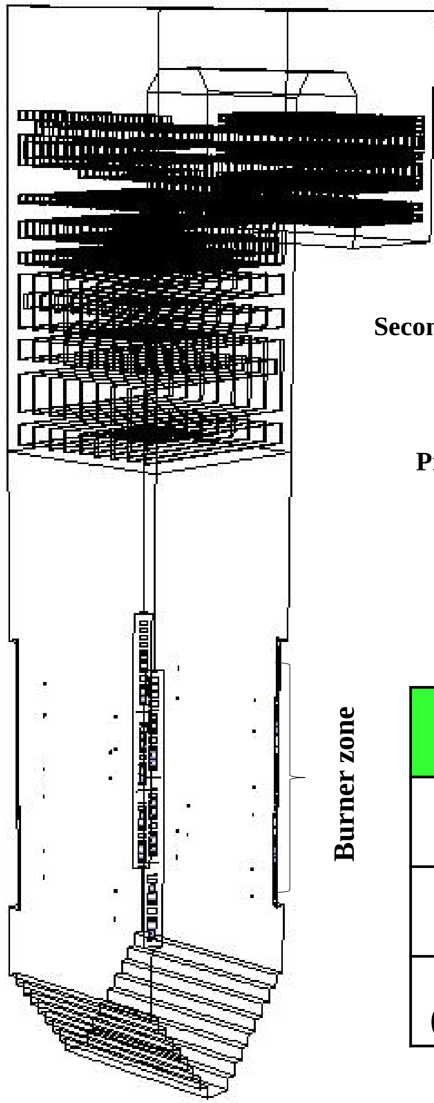
Schematic diagram of a natural-circulation steam generator . (Stultz and Kitto 1992)

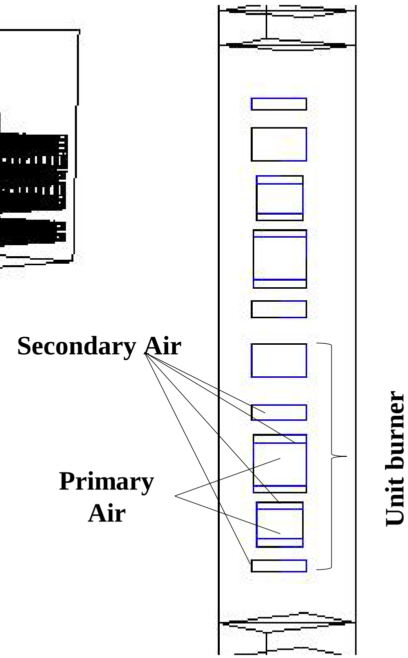
Natural-circulation steam generators typically consist of economisers and an evaporator with risers that form the heated furnace wall, a drum for the separation of water from steam and unheated down pipes and superheaters.

1차년도 개발 내용_ 입력조건들

- **500MW T-Firing boiler**







Build up

Aux air
Aux air
Weak coal (B, D, F burner)
Conc.coal (B, D, F burner)
Aux air
Oil
Aux air
Conc. Coal (A, C, E burner)
Weak coal (A, C, E burner)
Bottom air

Con : Weak = 7:3

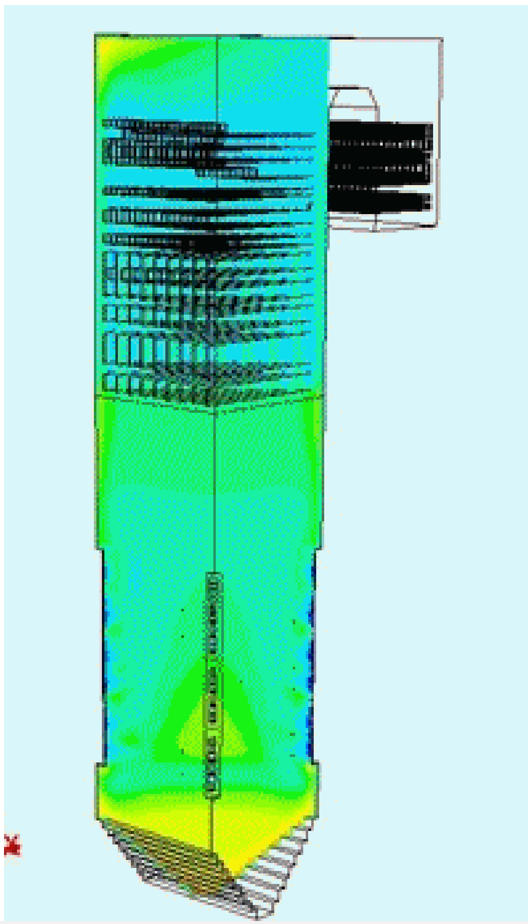
Aux air
Aux air
D Weak coal
D Conc.coal
Aux air
Oil
Aux air
C Conc. Coal
C Weak coal
Bottom air

CASE	500MW Design	550MW Design	550MW SUEK	550MW Mix	550MW Loa
H.V (kcal/kg)	6080	6080	5954	5300	5090
Coal (Ton/h)	193	208	212.4	238.61	248.46
Air flow (Nm^3/s)	375.6	404.8	403.7	453.3	472

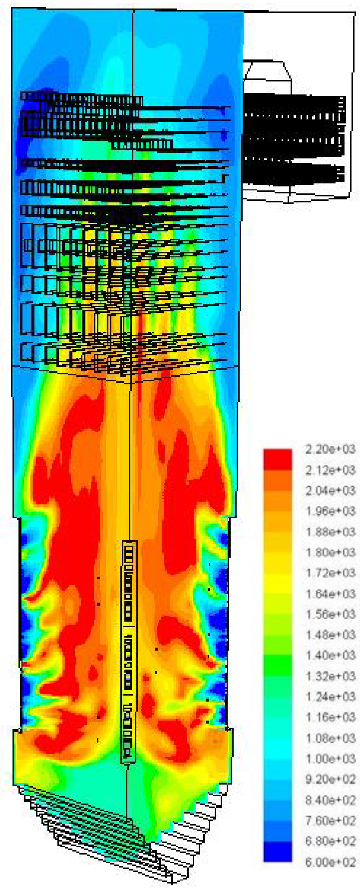
1차년도 개발 내용_ 모델 검증

- Validation as design coal

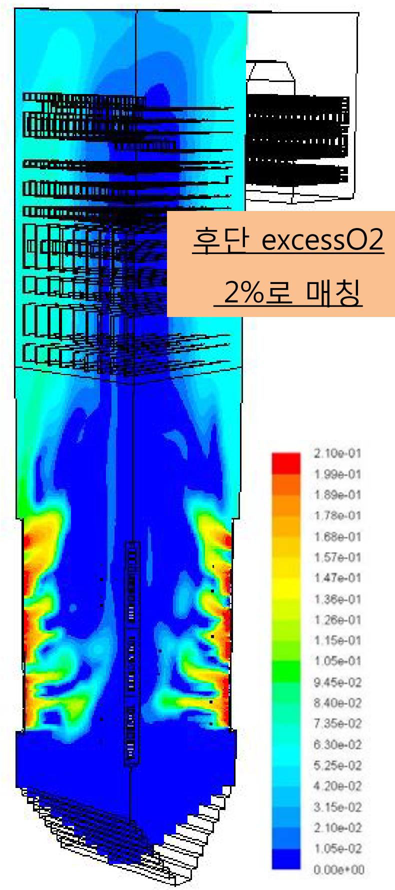
Iteration process



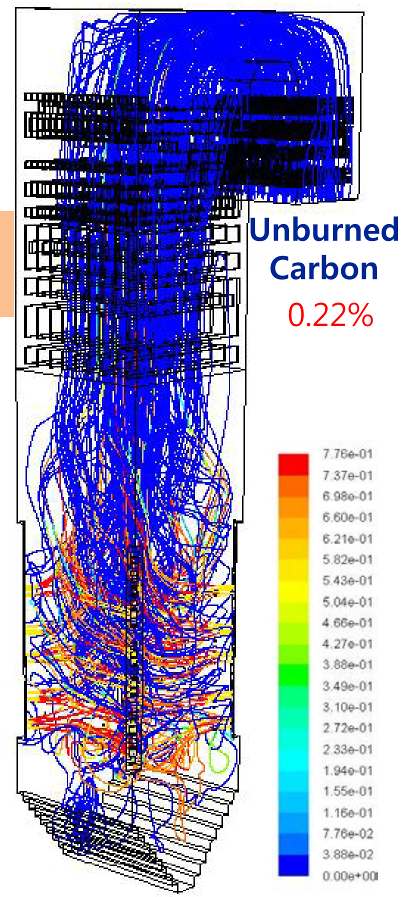
Final results



Temperature



O2 fraction



Particle trajectory