Lecture # 16
Thermomechanical Conversion II
Two-Phase Cycles and Combined Cycles

Ahmed Ghoniem
April 1, 2020

Rankine Cycle: two phase region
Superheat and Ultra-superheat Cycles. Reheating. Recuperation.
Supercritical Cycles. Hypercritical Cycles (CO_2 as working fluid)
Water requirements.

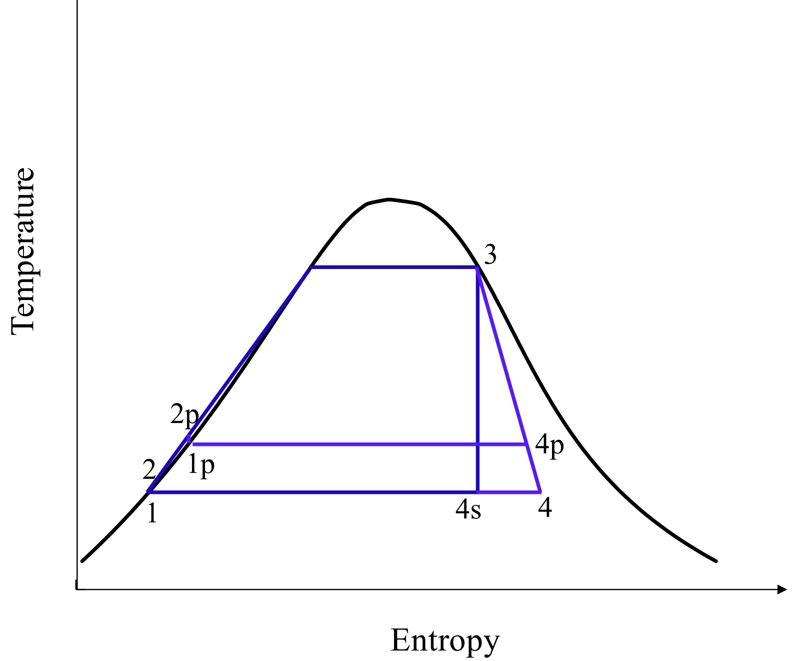
Simple Rankine Cycles: open and closed

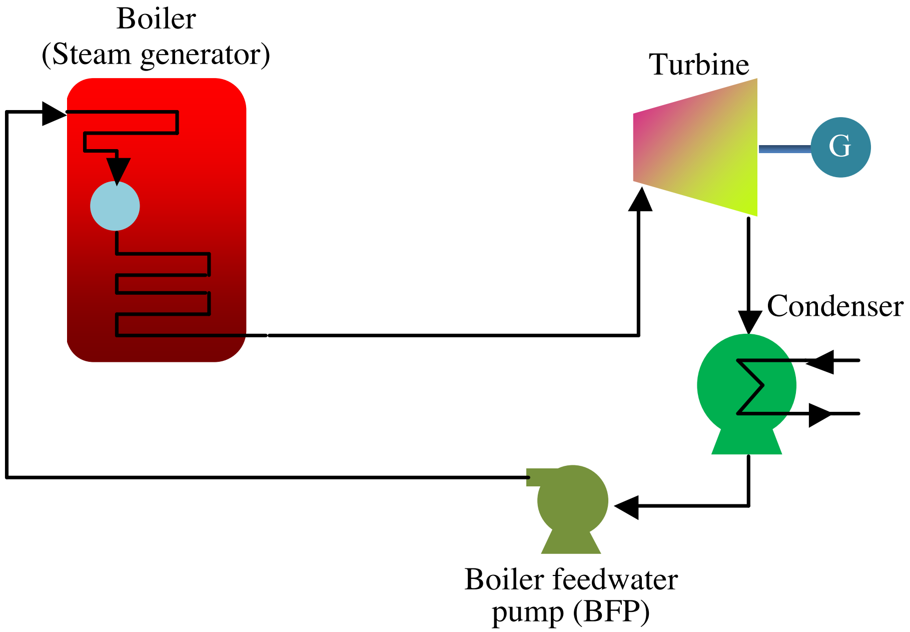
Critical point for Water: $T_c = 374 \text{ C}, p_c = 22.088 \text{ MPa}$

@ $T = 15 \text{ C}, p_{sat} = 5.63 \text{ kPa}$

@ $\text{p} = 1 \text{ atm}, T_{sat} = 100 \text{ C}.$

- Rankine cycles operate at relatively lower high temperature.
- They take advantage of the low pumping work of an incompressible liquid and high expansion work of the compressible gas.
- Operating in a closed cycle (to recirculate the working fluid), the turbine exhausts into vacuum, the pressure is determine by the cold temperature (condensation).
- Otherwise the efficiency is unacceptably low.





Simple ideal Rankine Cycle:

$$w_{pump,ideal} = h_{2s} - h_1 = v\left(p_2 - p_1\right)$$

$$w_{T,ideal} = h_3 - h_{4s}$$

$$q_H = h_3 - h_2$$

$$\eta_I = \frac{w_T - w_{pump}}{q_H}$$

In a real cycle:

$$w_{pump} = \frac{v\left(p_2 - p_1\right)}{\eta_{is}}$$

$$w_T = \eta_{T,is}\left(h_3 - h_{4s}\right)$$

$$q_H = h_3 - h_2$$

Simple saturated cycle efficiency,
Pressure Ratio = 8,
Pump = 65%, turbine 90%.

	Conventional	
	Tmin=20 Closed cycle	Pmin=1atm Open cycle
w_{pump} (kJ/kg)	1.23	1.12
w_t (kJ/kg)	736	316
w_{net} (kJ/kg)	735	315
η	27.4 %	13.4%
η_{ideal}	30.4%	14.9%
η_{car}	33.9%	15.8%
X_4	0.794	0.8856

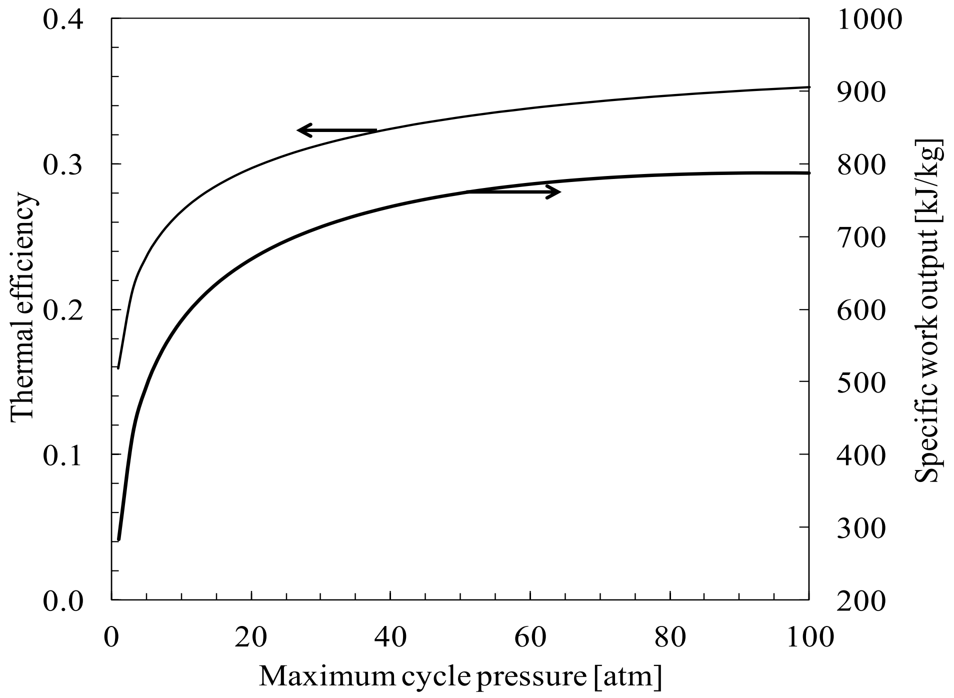
- low pumping work, for an incompressible fluid; $\Delta h = v \Delta p$ (the fluid temperature does not rise).

- Generally, lower *high T* requirements (compatible with nuclear, solar thermal, geothermal and lower quality fuel sources) but needs *high p*. Also good for waste heat recovery (using organic working fluids)

- Good efficiency: small pumping work and near isothermal heat interactions.

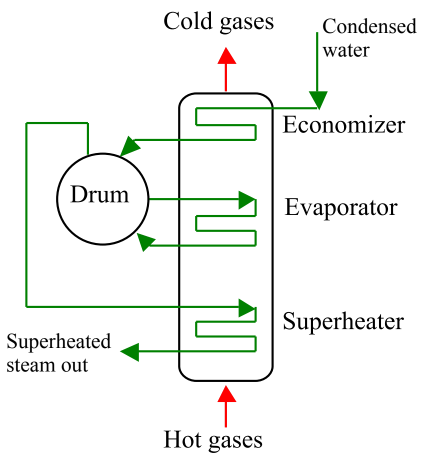
- Large heat transfer (latent heat).

Simple closed cycle efficiency, saturated state
Pump = 65%, turbine 90%, condenser T = 30 ^{o}C



Both work and efficiency increase monotonically because of small pumping work

Superheat Cycles ….



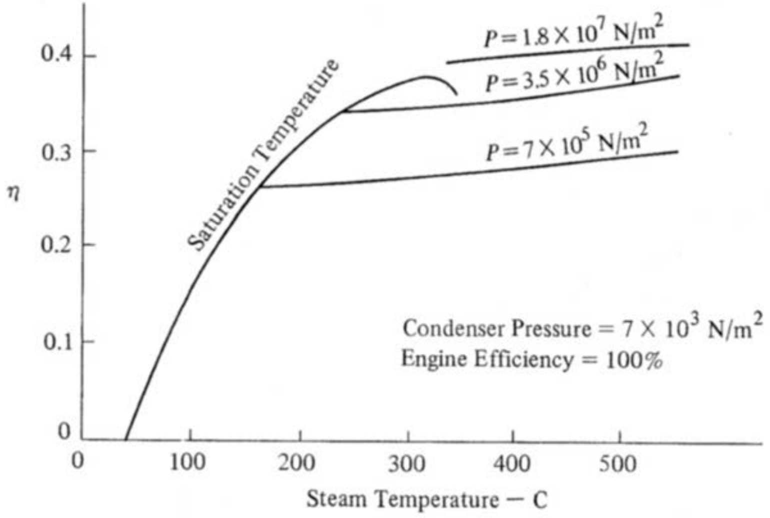
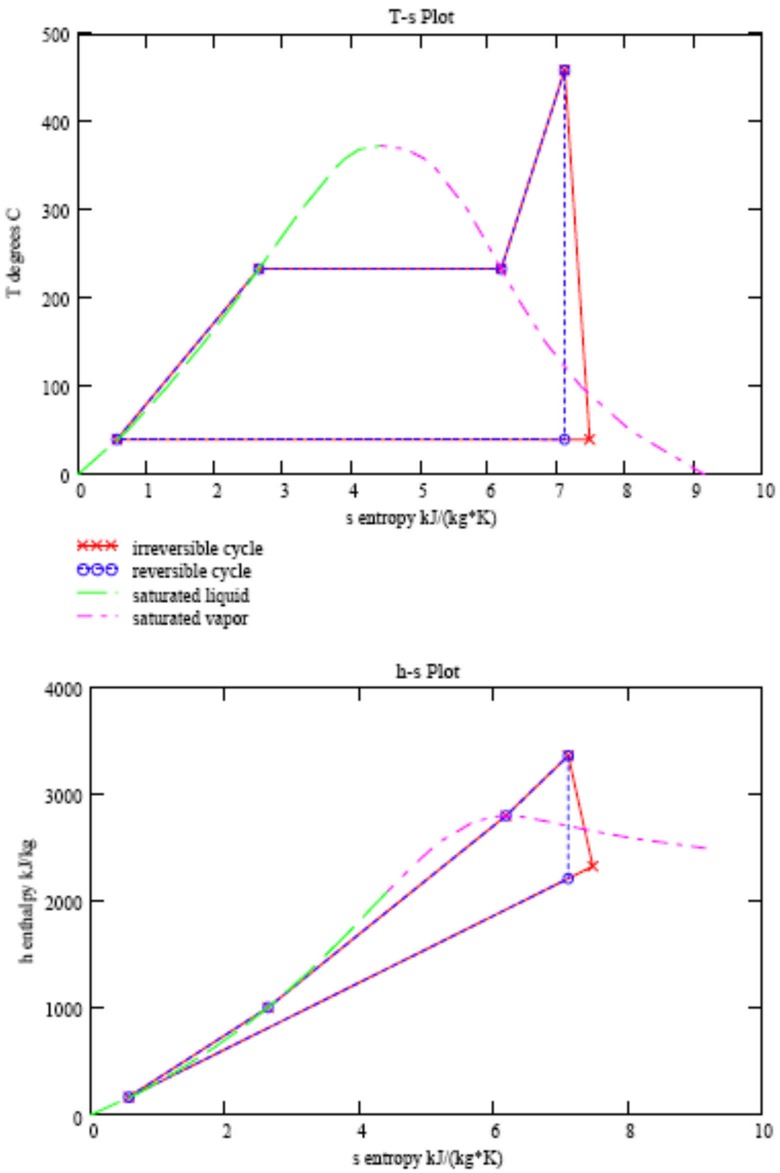


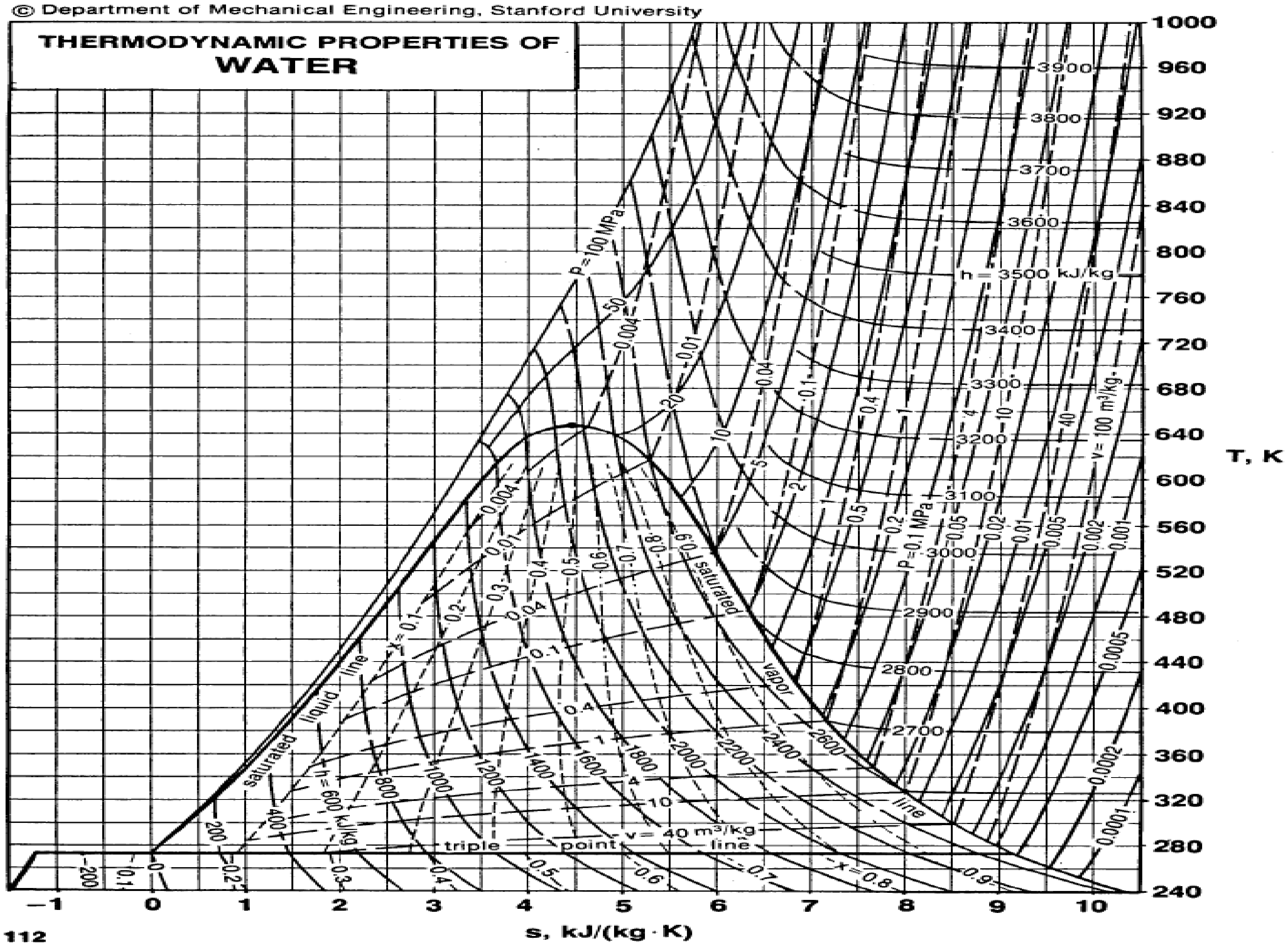
Figure 12.4 Influence of superheat on Rankine cycle efficiency

From Dave Burke, 2.611



From Smith and Cravalho,
Engineering Thermodynamics

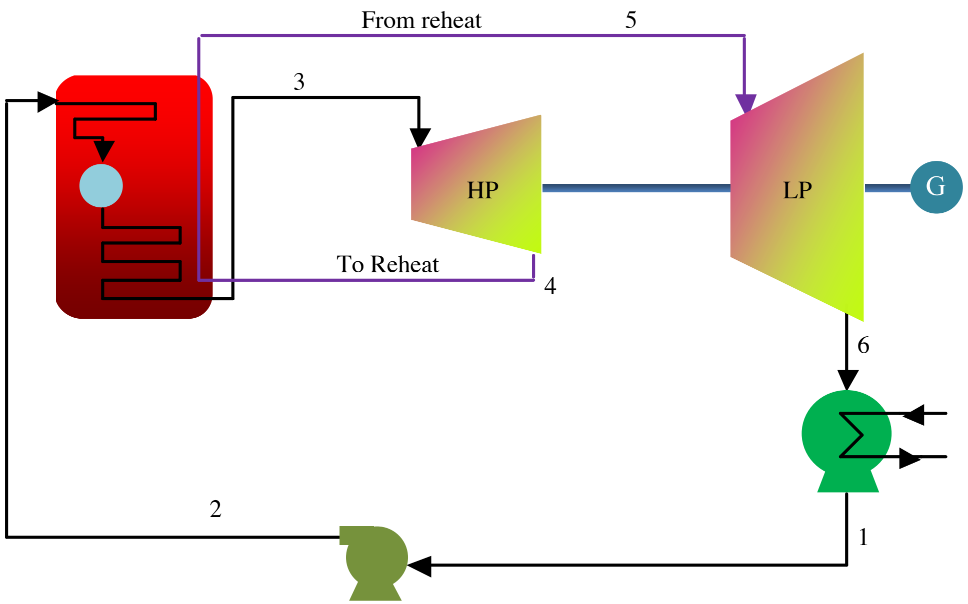
		Superheat +100
	Tmin=20	
w_{pump} (kJ/kg)	1.23	1.23
w_t (kJ/kg)	736	818
w_{net} (kJ/kg)	735	817
η	27.4 %	28.1%
η_{ideal}	30.4%	
η_{car}	33.9%	46.0%
X_4	0.794	0.8517

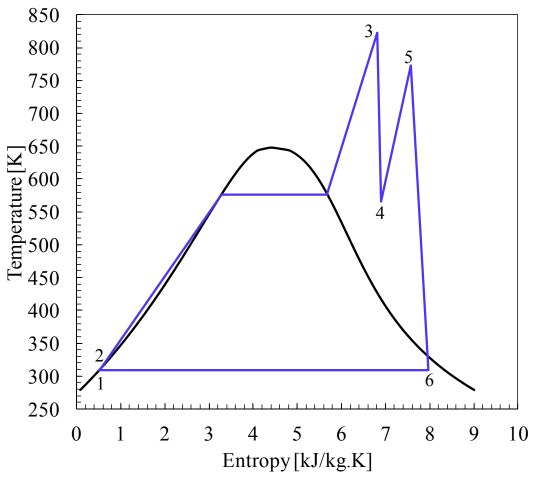

© Department of Mechanical Engineering, Stanford University
THERMODYNAMIC PROPERTIES OF
WATER
T, K
1000
960
920
880
840
800
760
720
680
640
600
560
520
480
440
400
360
320
280
240
s, kJ/(kg · K)
−1
0
1
2
3
4
5
6
7
8
9
10
112
P = 100 MPa
50
20
10
5
2
1
0.5
0.2
P = 0.1 MPa
0.05
0.02
0.01
0.005
0.002
0.001
0.0005
0.0002
0.0001
0.4
0.1
0.04
0.01
0.004
v = 100 m³/kg
40
10
4
1
h = 3500 kJ/kg
3900
3800
3700
3600
3400
3300
3200
3100
3000
2900
2800
2700
2600
2400
2200
2000
1800
1600
1400
1200
1000
800
h = 600 kJ/kg
400
200
0
−200
x = 0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
saturated liquid line
saturated vapor line
triple point line
v = 40 m³/kg
0.004
0.01
0.04
0.1
0.4
1
4
10

Concentrated solar thermal (CSP) and hybrid concentrated solar thermal (Hy-CS) power plants

Florida Power (FPL) is adding 75 MW (peak) solar increment to its 3800 MG NG plant (Hybrid Concentrated Solar, or HyCS) to boost the fraction of renewable energy generation. HyCS reduces the cost and does not require storage, another costly item in solar plants.

Reheat Cycle

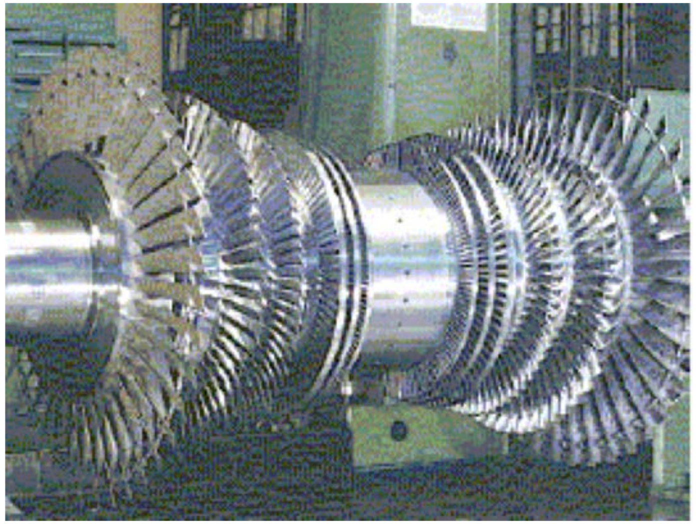


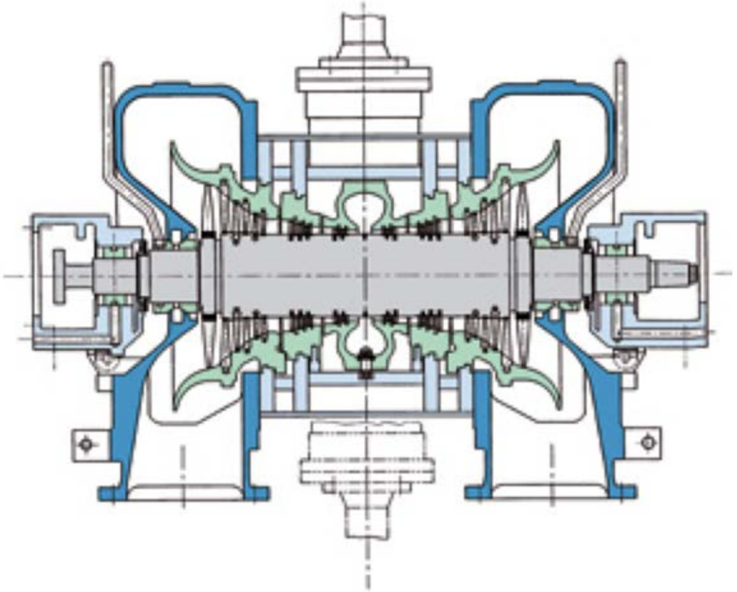


		Reheat Cycle		
	Tmin=20	+100	+200	+300
w_{pump} (kJ/kg)	1.23	1.23	1.23	1.23
w_t (kJ/kg)	736	947.2	1086	1400
w_{net} (kJ/kg)	735	946	1085	1398
η	27.4 %	28.1%	30.3%	35.5%
η_{ideal}	30.4%			
η_{car}	33.9%	46.0%	54.4%	60.6%
X_6	0.794	0.9583	Vapor	Vapor

Better efficiency and steam quality at end of expansion

SIEMENS SST-500





Technical Data

All data are approximate and project-related

Output range	up to	85 MW
live steam conditions		
temperature	up to	540C / 1000F
pressure	up to	140 bar / 2000 ps
Bleed	up to	2 at various pressure level
Controlled extraction		
temperature	up to	350C / 662F
pressure	up to	30 bar / 435 ps

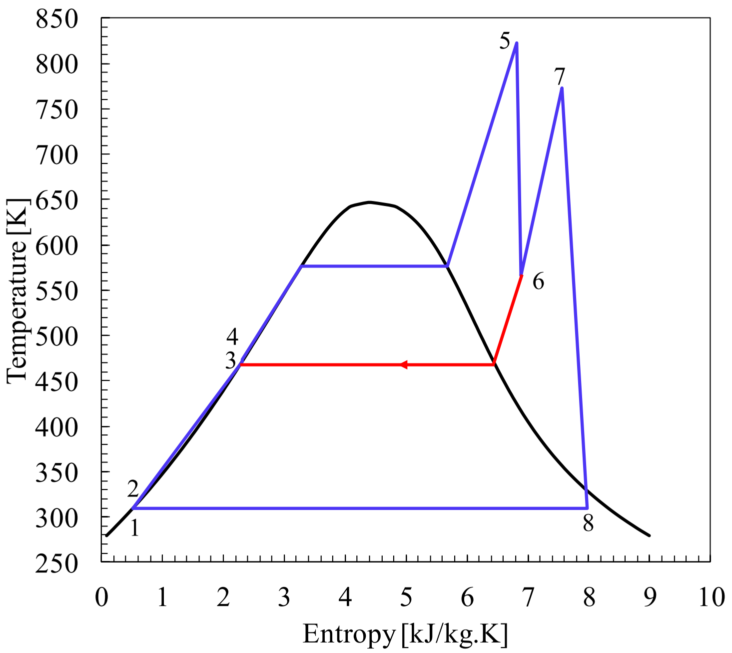
Typical plant layout for a SST-500 Steam Turbine

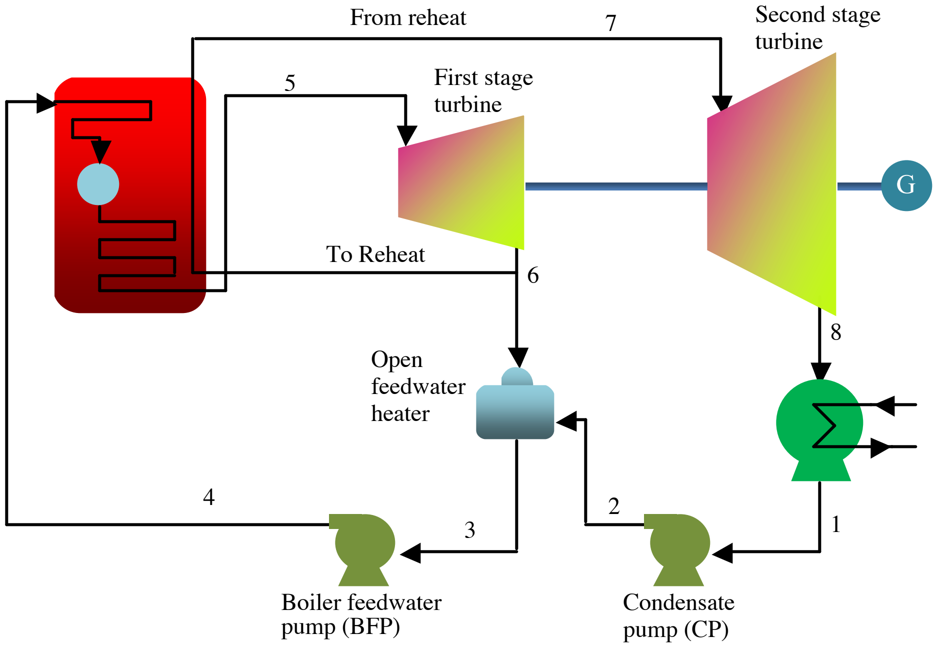
Dimensions

Length (L) 10m/32.8 ft. to 19m/62.3 ft.
Width (W) 4.0m/13.1 ft. to 6.0m/19.7 ft.
Height (H) 3.5m/11.5 ft. to 5.0m/16.5 ft

Regenerative Cycles

1. Direct Contact (open) Feedwater Heater





Best feedwater heater arrangement from the efficiency viewpoint, but requires an extra pump.

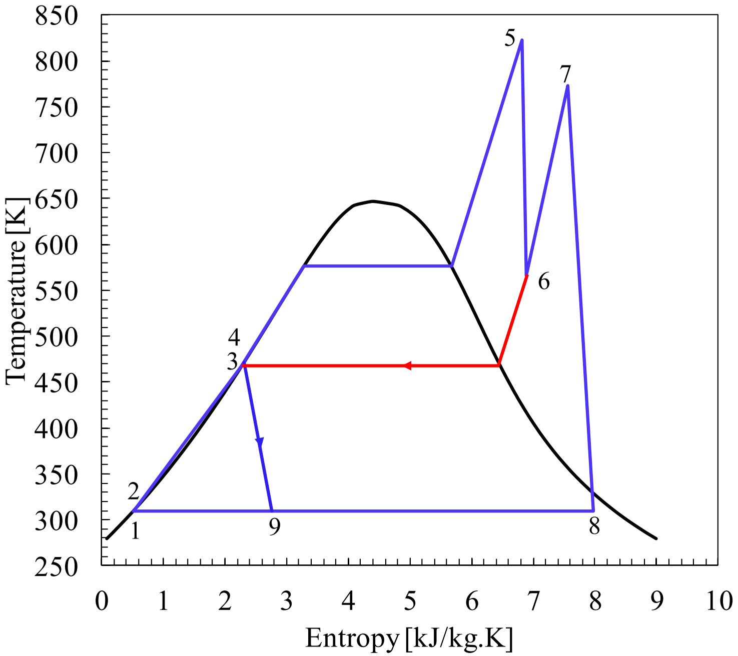
α is extracted from the turbine at state 6 is given by: $\alpha h_6 + (1-\alpha) h_2 = h_3$

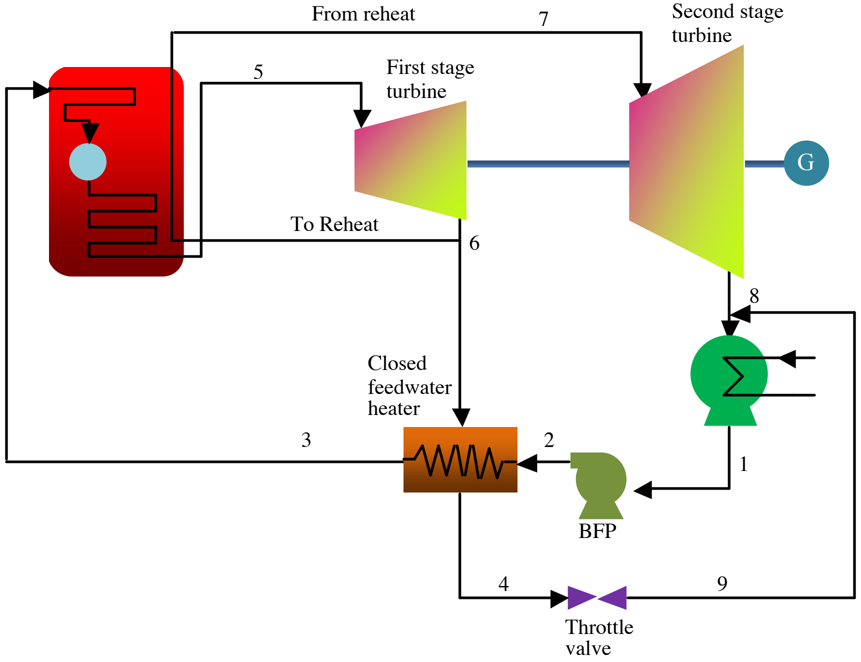
condenser sees only $(1-\alpha)$ of the flow

Pump = 65%, turbine 90%, Pressure ratio = 8

		Reheat Cycle	Regenerative Cycle +100
	Tmin=20	+100	
w_{pump} (kJ/kg)	1.23	1.23	1.26
w_t (kJ/kg)	736	947.2	774
w_{net} (kJ/kg)	735	946	773
η	27.4 %	28.1%	29.4%
η_{ideal}	30.4%		
η_{car}	33.9%	46.0%	46.0%
X_4	0.794	0.9583	N/A

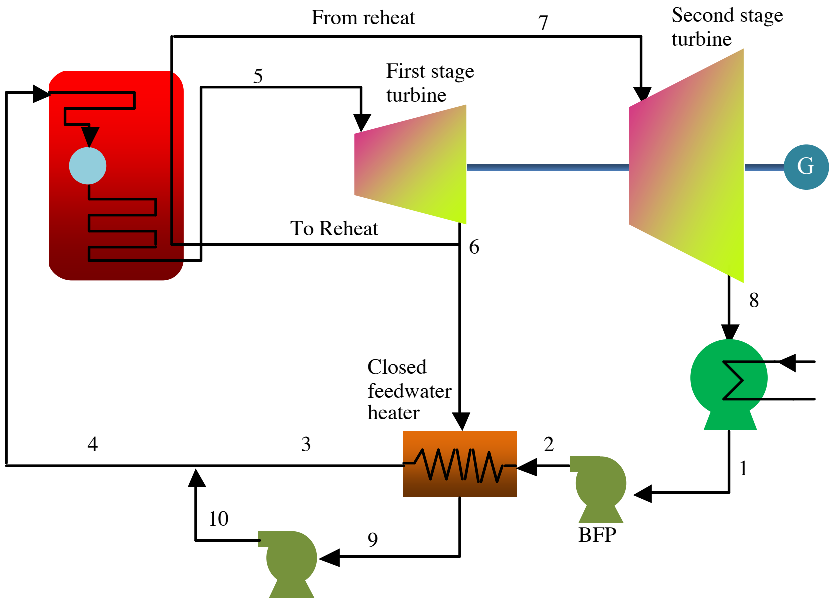
2. Cascading Backward, Closed Feedwater Heater

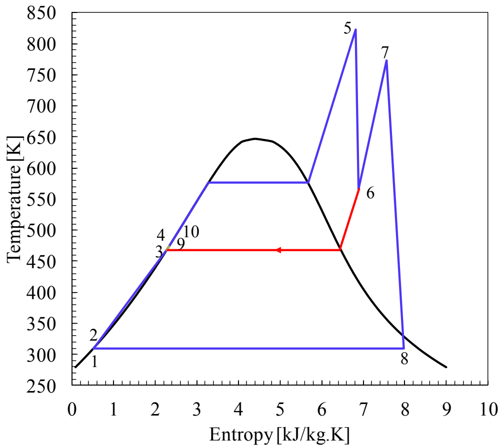




Less efficient because of throttling and some heat rejection in condenser, but only one pump is required .

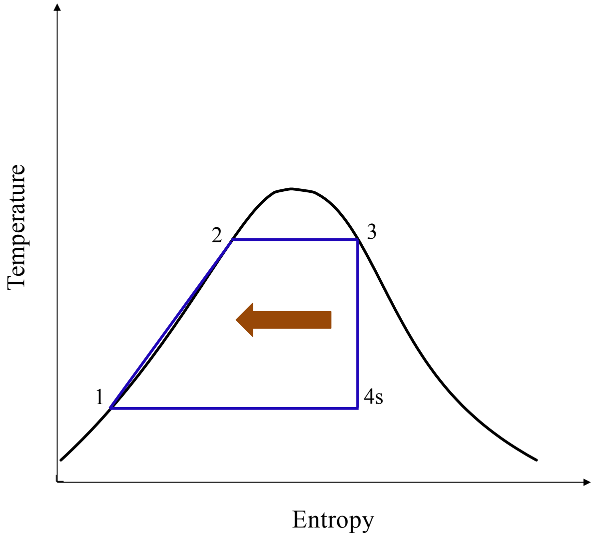
3. Cascading Forward, Closed Feedwater Heater





Using a small pump after heater avoids rejecting extra heat in extracted steam

Ultimate Regenerative Cycle:



Ultimate Regenerative Cycle:

1. Internally heat the feedwater using extracted steam.
2. Amount of extracted steam is small, latent heat >> sensible heat.
3. External heat transfer is isothermal.
4. Cycle efficiency = Carnot efficiency.

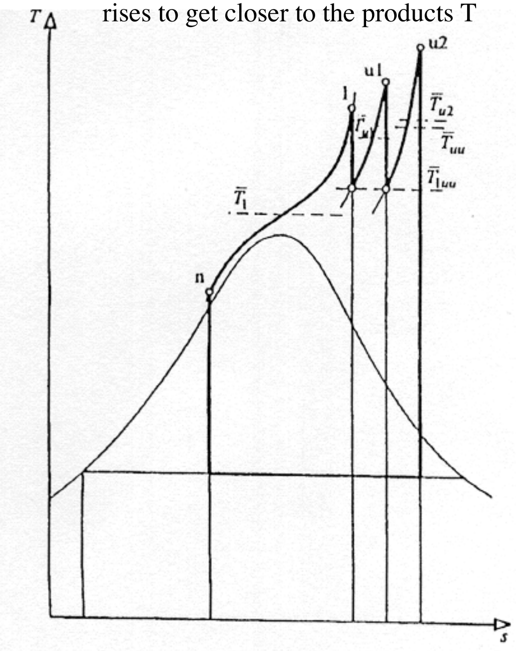
SUPERCRITICAL CYCLES

$p_{boiler} > p_c$, for Water: $T_c = 374\ \mathrm{C}, p_c = 22.088\ \mathrm{MPa}$

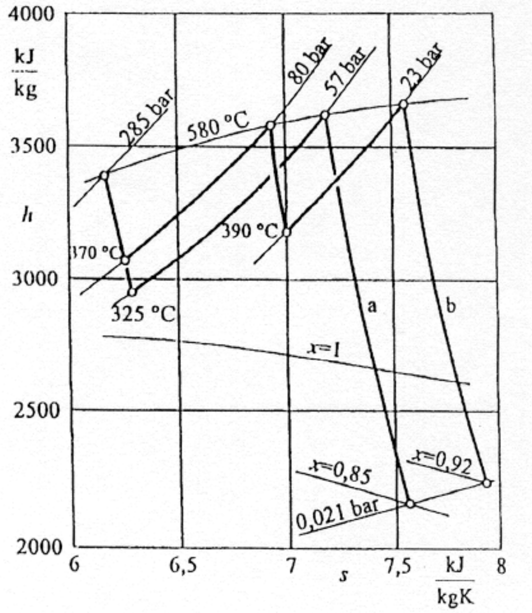
$>$ Raises the cycle temperature

$>$ Reduces ΔT between source and steam

Products T line, with reheat steam T rises to get closer to the products T



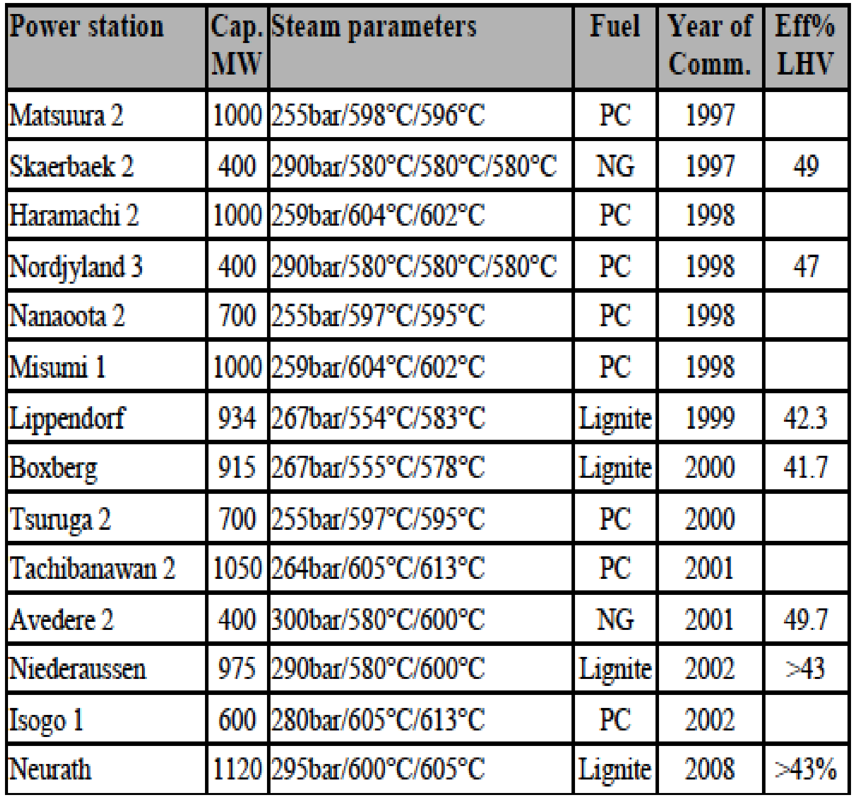
REVERSIBLE SUPERCRITICAL STEAM CYLE WITH DOUBBLE REHEAT



SUPERCRITICAL STEAM TURBINE EXPANSION LINES a.)Single, and b)Doubble reheat

T-s and h-s diagram representations of supercritical steam cycle with reheat (3.Büki G.,Magyar Energiatechnika 1998;6:33-42)

Table 1.3 USC steam plants in service or under construction globally



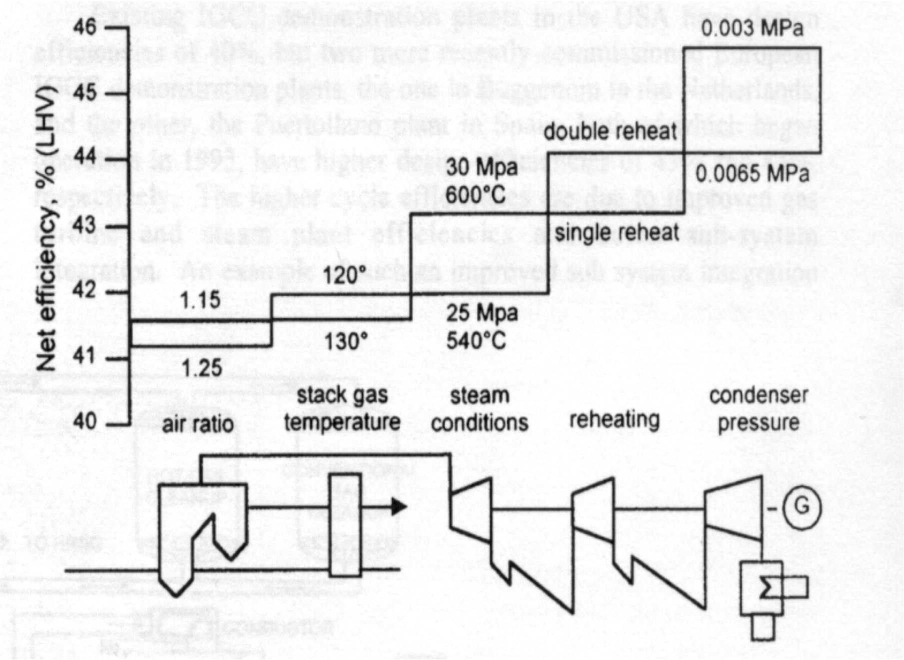
Power station	Cap. MW	Steam parameters	Fuel	Year of Comm.	Eff% LHV
Matsuura 2	1000	255bar/598°C/596°C	PC	1997	
Skaerbaek 2	400	290bar/580°C/580°C/580°C	NG	1997	49
Haramachi 2	1000	259bar/604°C/602°C	PC	1998	
Nordjyland 3	400	290bar/580°C/580°C/580°C	PC	1998	47
Nanaoota 2	700	255bar/597°C/595°C	PC	1998	
Misumi 1	1000	259bar/604°C/602°C	PC	1998	
Lippendorf	934	267bar/554°C/583°C	Lignite	1999	42.3
Boxberg	915	267bar/555°C/578°C	Lignite	2000	41.7
Tsuruga 2	700	255bar/597°C/595°C	PC	2000	
Tachibanawan 2	1050	264bar/605°C/613°C	PC	2001	
Avedere 2	400	300bar/580°C/600°C	NG	2001	49.7
Niederaussen	975	290bar/580°C/600°C	Lignite	2002	>43
Isogo 1	600	280bar/605°C/613°C	PC	2002	
Neurath	1120	295bar/600°C/605°C	Lignite	2008	>43%

(Blum and Hald and others)

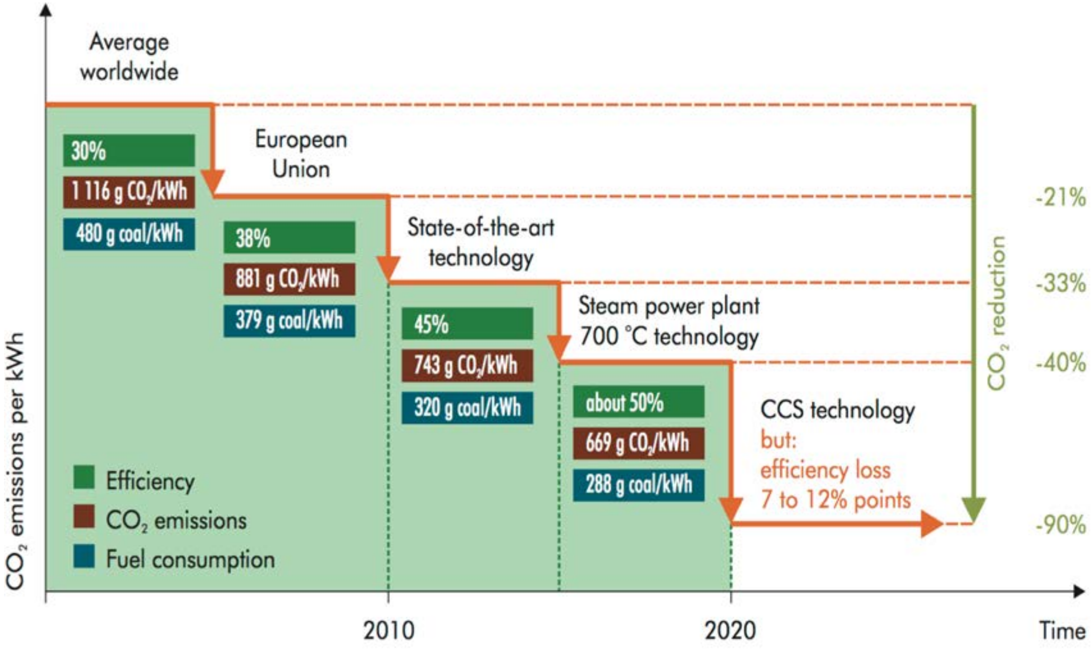
Coal plans are less efficient than NG plants because of exhaust gas clean up

Efficiency Improvements and CO_2 Emissions

Effect of various measures for improving the efficiency (LHV) of pulverized coal fired power generating plant
(Schilling,H.D.:VGB Kraftwerkstechnik 1993;73(8)pp.564-76 (English Edition)



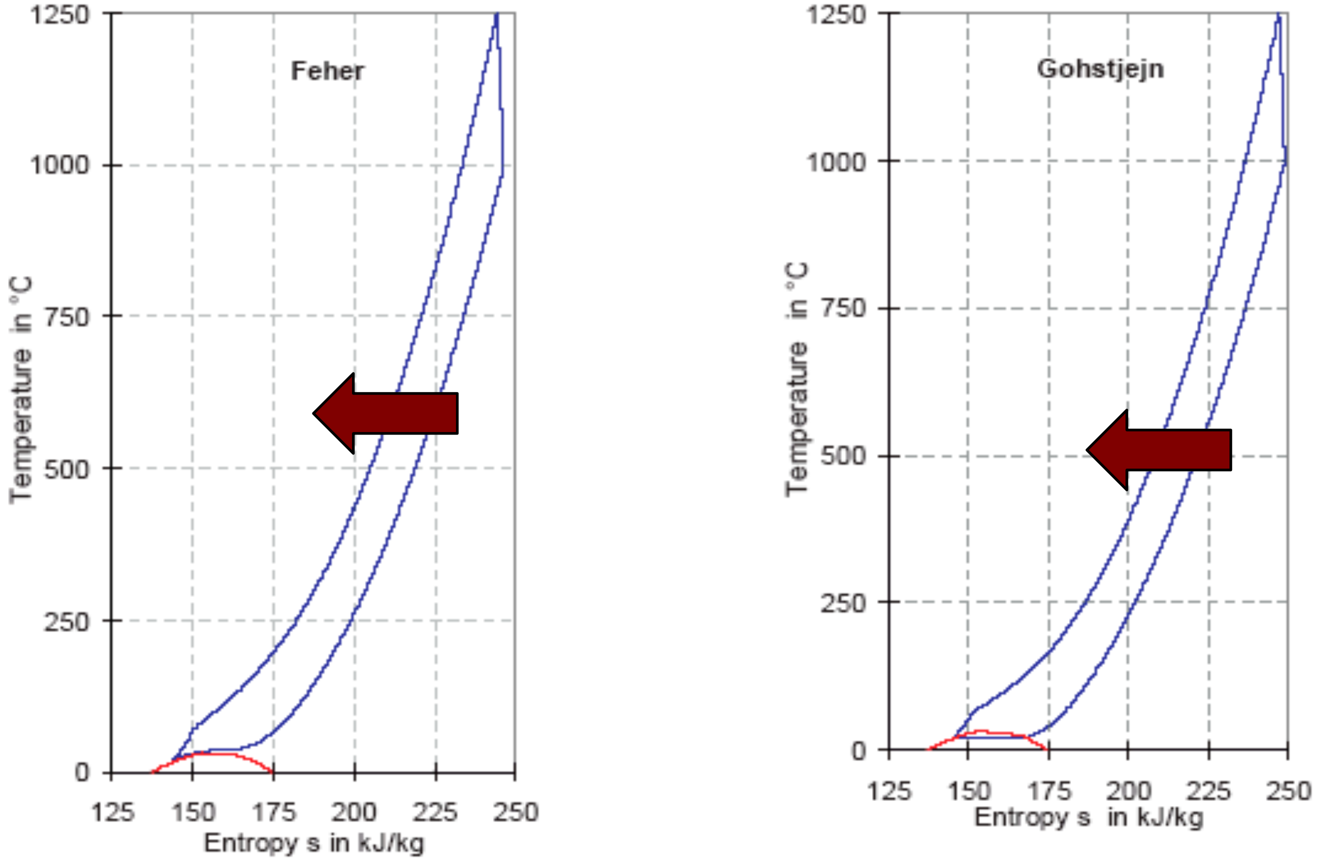
Limits of Efficiency improvement on CO_2 emissions and Role of CCS



Source: VGB (2009). Reprinted by permission of the publisher. © VGB PowerTech e.V., 2009.

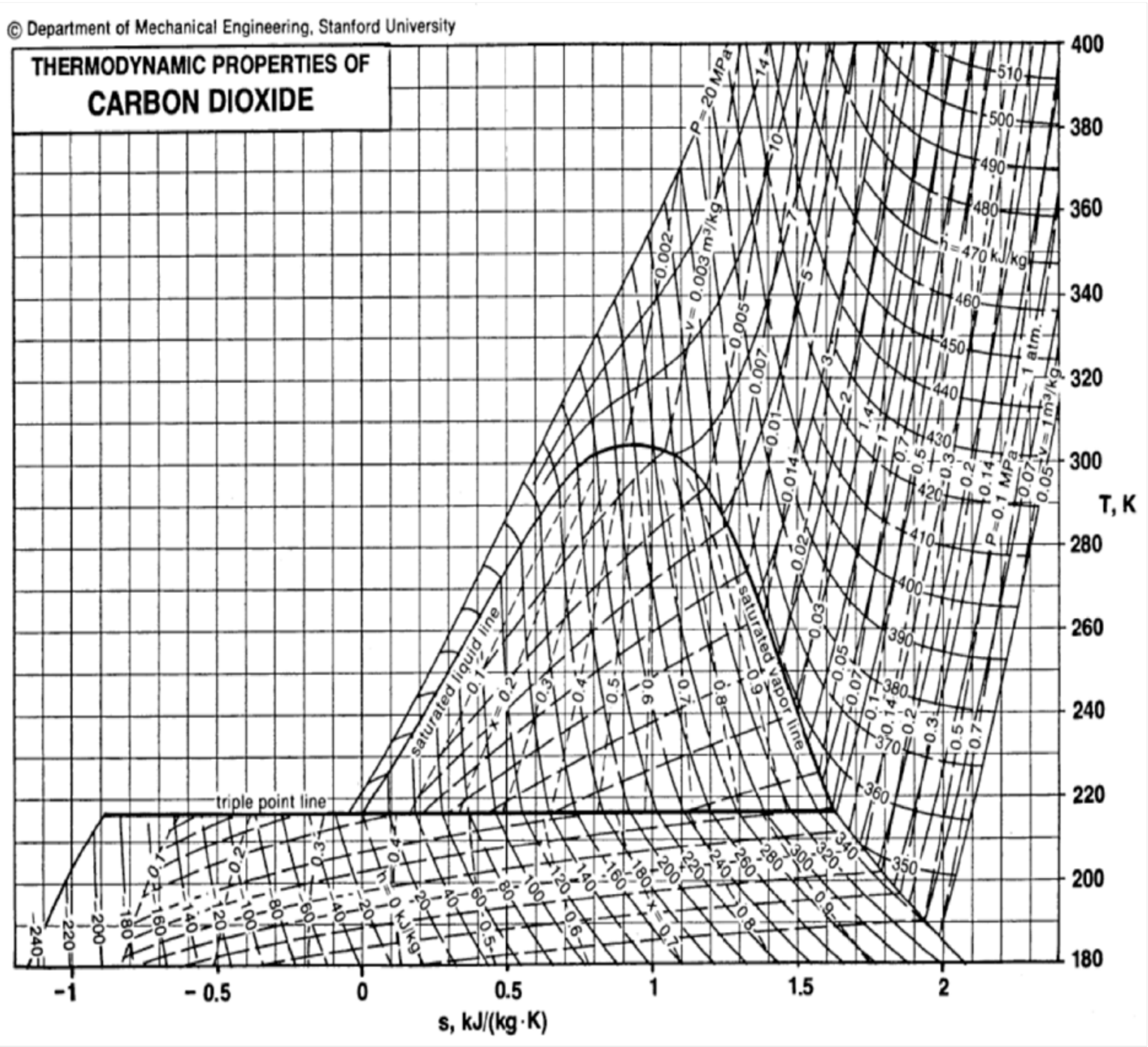
Hypercritical closed CO_2 "Gas" Cycles

- $p_{crit} = 7.39$ MPa, and $T_{crit} = 30.4$ C.
- Can take advantage of benefits of supercritical cycles without the need for very high *p* (typical pressure ratio is 4 but can go up to 10)).
- High T is used to improve efficiency.
- Regeneration improves the efficiency significantly, see diagrams.
- Low compression work (near critical point, more pumping than compression)
- Under consideration for nuclear plants.
- Also for oxy-combustion cycles

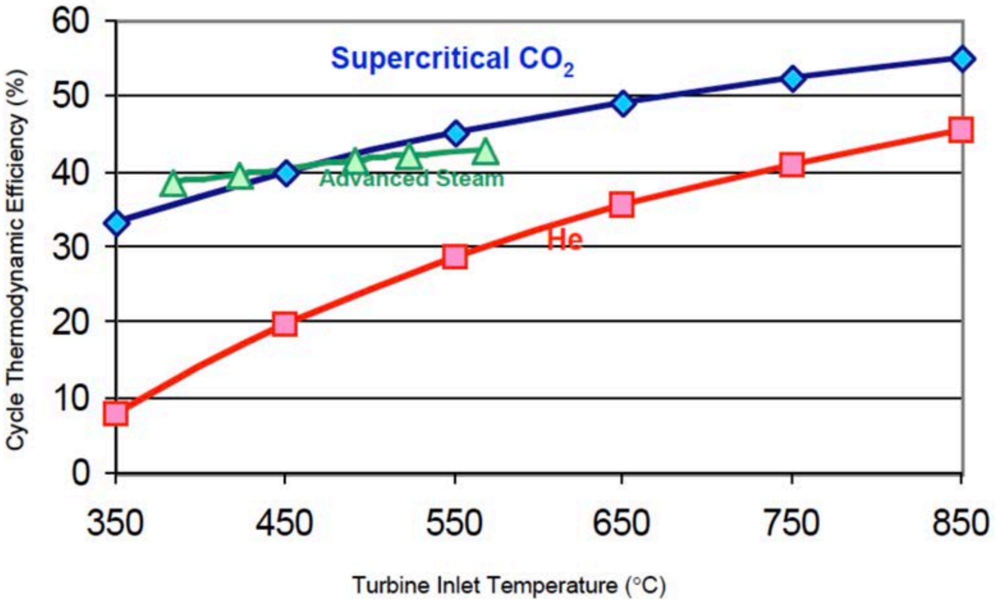


Courtesy of USGS.

From Gottlicher, the Energetics of Carbon Sequestration in Power Plants



See Chapter 6 for more detail, especially the impact of regeneration and split compression to achieve impressive efficiency in CO_2 cycles



- Thermal efficiency of a number of cycles within the low temperature range.
- Helium cycles are Brayton cycles, which can only achieve low efficiency at these low temperatures.
- Advanced steam cycles are superheated or supercritical steam cycles.
- Supercritical CO_2 are “hypercritical” cycles.

Rankine cycles:

1. Fuel flexible, works well with coal and other dirty fuels (closed cycle).
2. Have high efficiency, low pumping power.
3. Require lower flow rate (latent enthalpy).
4. Run at lower high T (work well with renewable sources), but high p.
5. Works well with nuclear energy:

BUT …

1. High inertia, good for base load but not for load following.
2. Require cooling, big condensers, .. Water …

- Condenser adds cost, needs vacuum and allows air leakage.
- Must remove air to maintain low pressure in condenser.
- Condenser needs large surface area and large water flow.
- Superheat increases efficiency and specific work.
- Superheat improves steam quality in late stages of turbine, reduces material damage.
- Reheat helps efficiency and steam quality.
- Recuperation increases efficiency at the cost of hardware complexity.

COMBINED CYCLES

$$Q_{in} = Q_{GT},$$

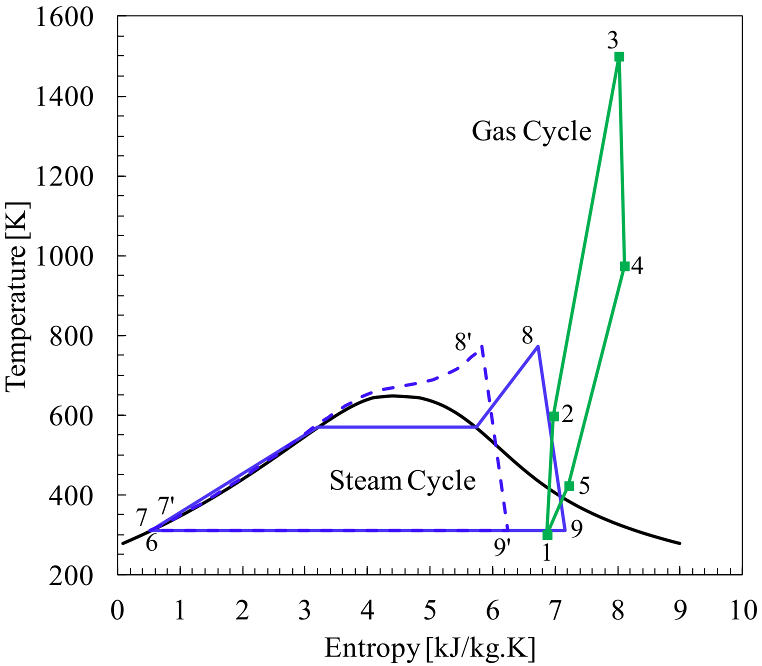
$$W_{GT} = \eta_{GT} Q_{GT},$$

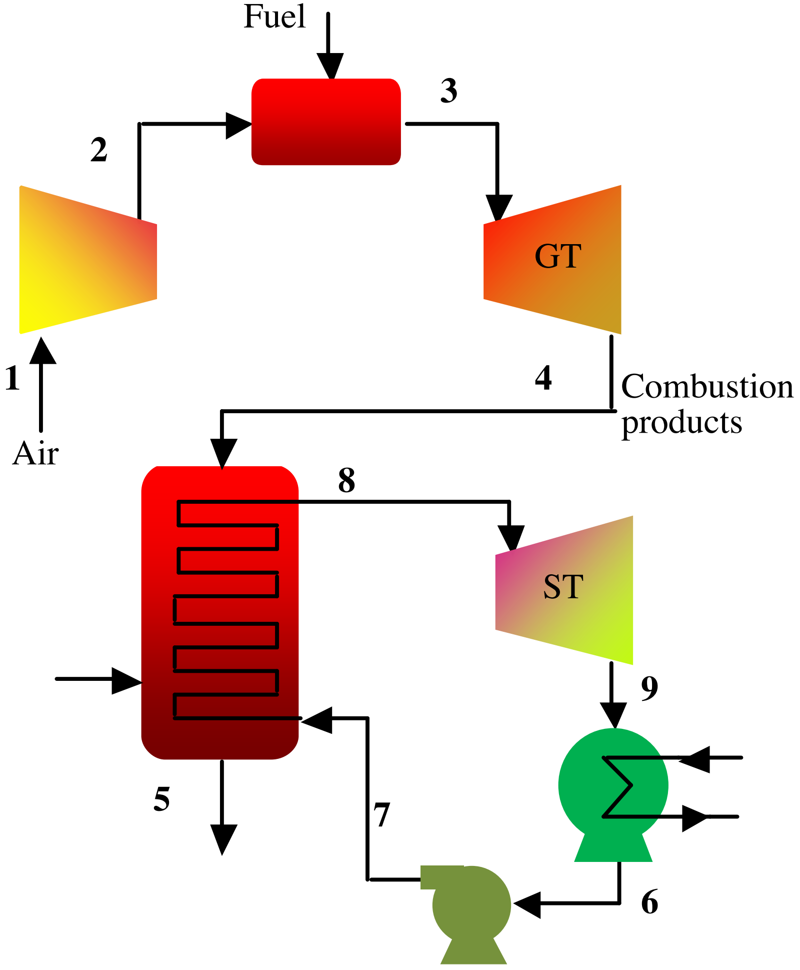
$$Q_{ST} \approx (1-\eta_{GT}) Q_{GT},$$

$$W_{ST} = \eta_{ST} (1-\eta_{GT}) Q_{GT}$$

$$W = W_{GT} + W_{ST},$$

$$\eta_{CC} = \eta_{GT} + \eta_{ST} (1-\eta_{GT})$$





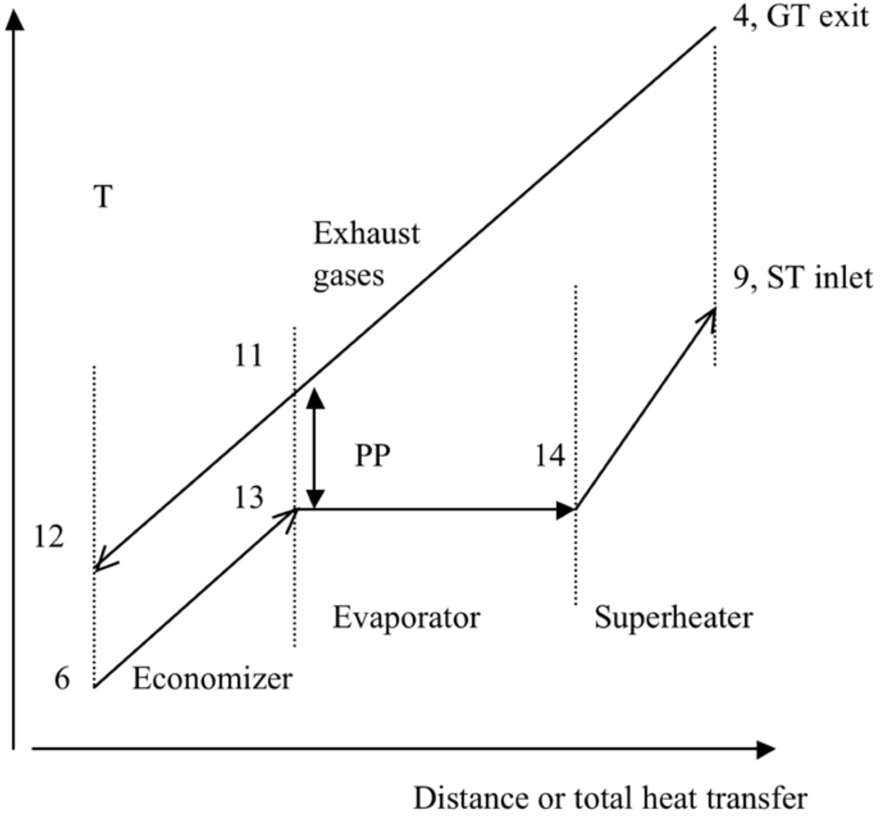
$\eta_{GT} = 0.25$, and $\eta_{ST} = 0.4$, $\eta_{CC} = 0.55$

$\eta_{GT} = 0.3$, and $\eta_{ST} = 0.28$, $\eta_{CC} = 0.5$

$\eta_{GT} = 0.38$, and $\eta_{ST} = 0.25$, $\eta_{CC} = 0.535$

$\eta_{GT} = 0.38$, and $\eta_{ST} = 0.40$, $\eta_{CC} = 0.628$

Mass flow rates are not arbitrary, Pinch-point analysis and impact on efficiency:



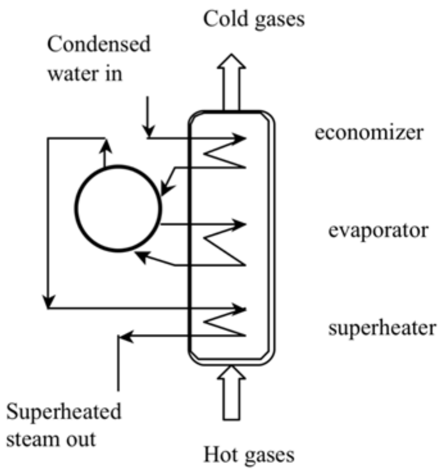
T_4 is determined by gas turbine exit conditions

T_{13} is determined by steam cycle high pressure

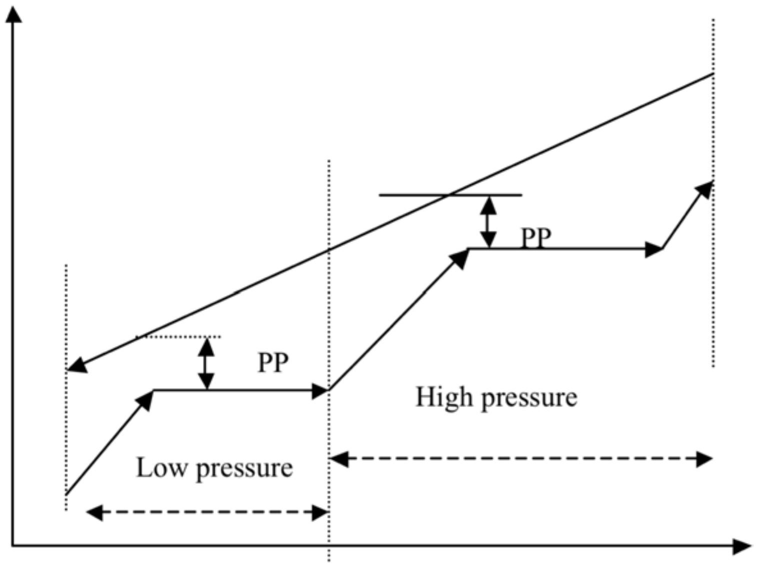
$T_{11} = T_{13} + PP$ for good heat transfer rates: $PP = O(10-15\text{ C})$

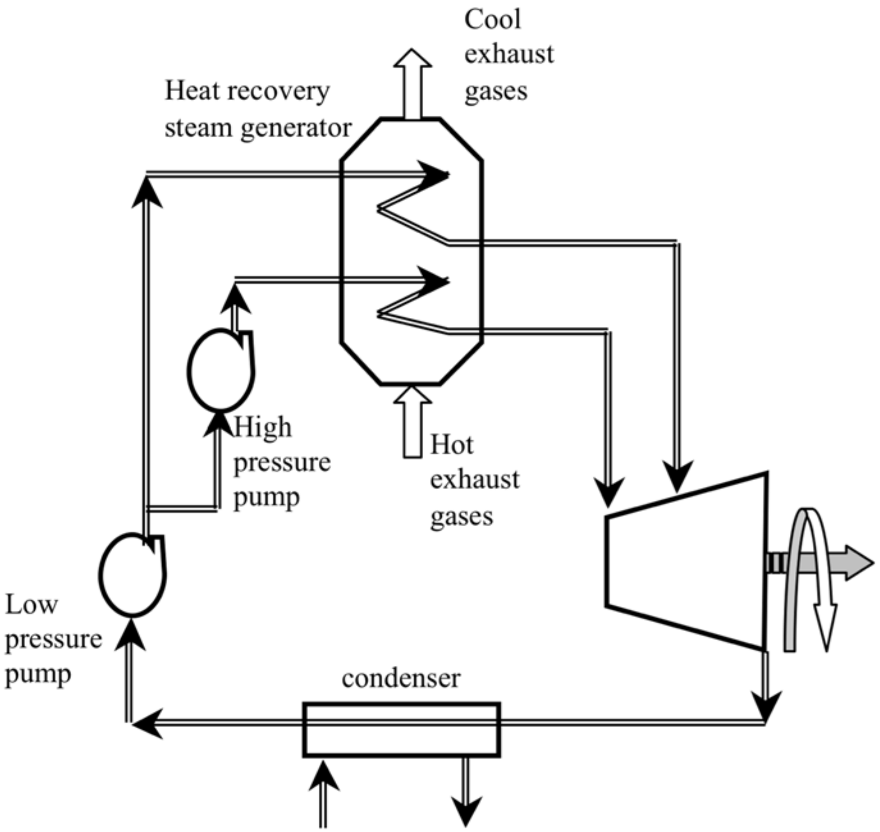
T_9 is determined by steam cycle design

Therefore: $$\dot{m}_{st} = \dot{m}_g \frac{c_{pg}\left(T_4 - T_{11}\right)}{\left(h_9 - h_{13}\right)}$$



Temperature difference between streams can be reduced by employing dual or triple pressure steam cycles:



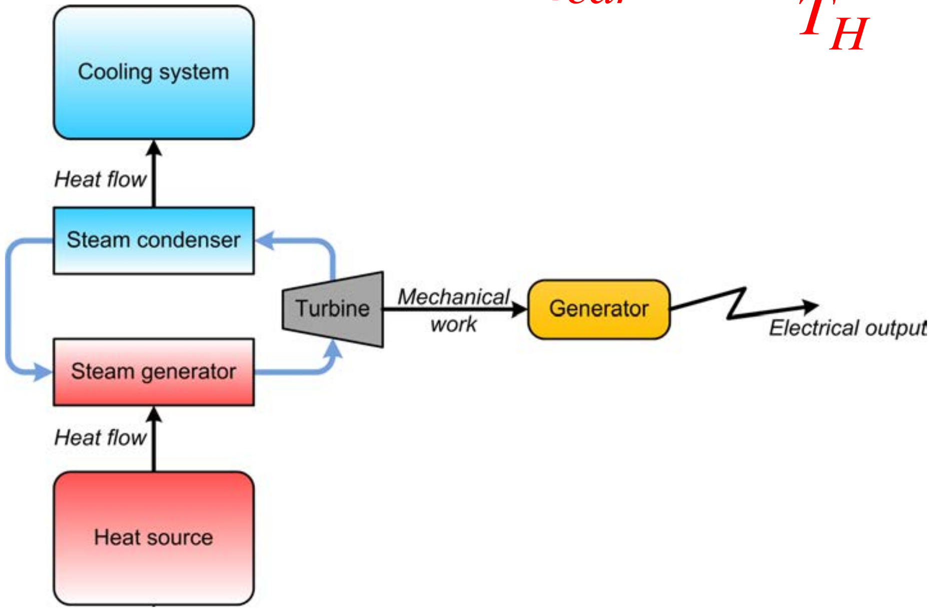


Leading to 2-3 percentage points in efficiency gain.

Steam power plant energy balance

more complex than just the efficiency

$$\eta_{car} = 1 - \frac{T_o}{T_H}$$



Cooling system

Heat flow

Steam condenser

Turbine

Mechanical work

Generator

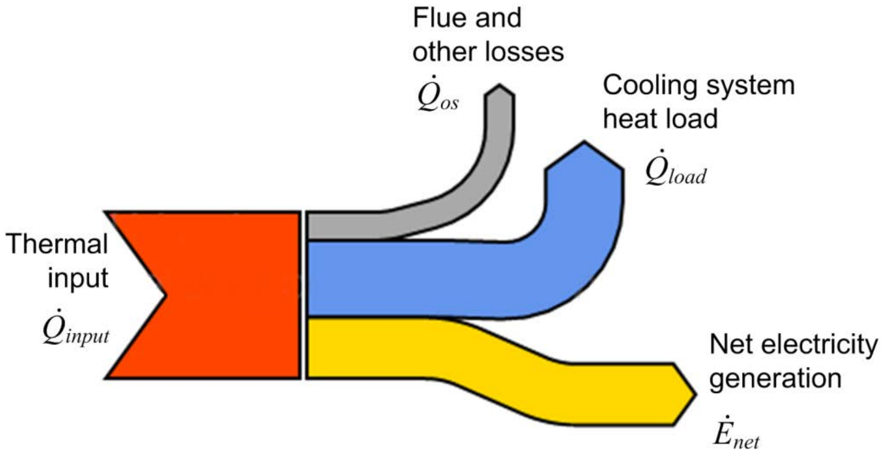
Electrical output

Steam generator

Heat flow

Heat source

***Solar thermal, geothermal and nuclear** plants run at lower temperatures than combustion plants, have lower thermal efficiencies and **higher water footprints – but lower carbon footprints**!*



Flue and other losses $\dot{Q}_{os}$

Cooling system heat load $\dot{Q}_{load}$

Thermal input $\dot{Q}_{input}$

Net electricity generation $\dot{E}_{net}$

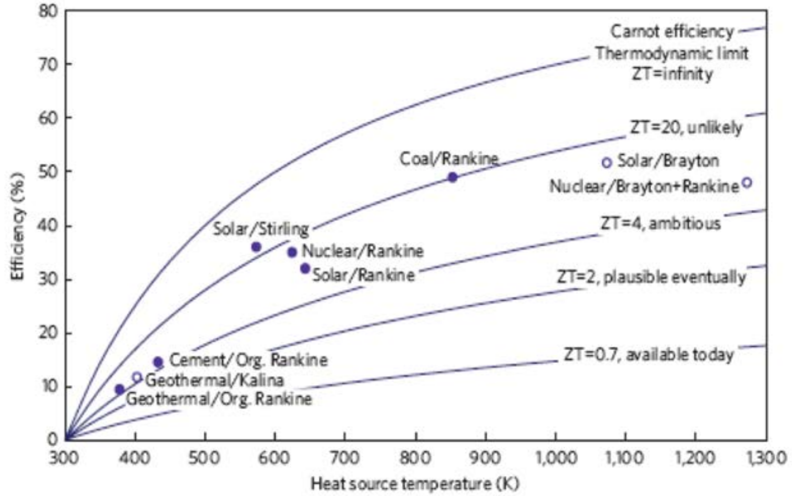
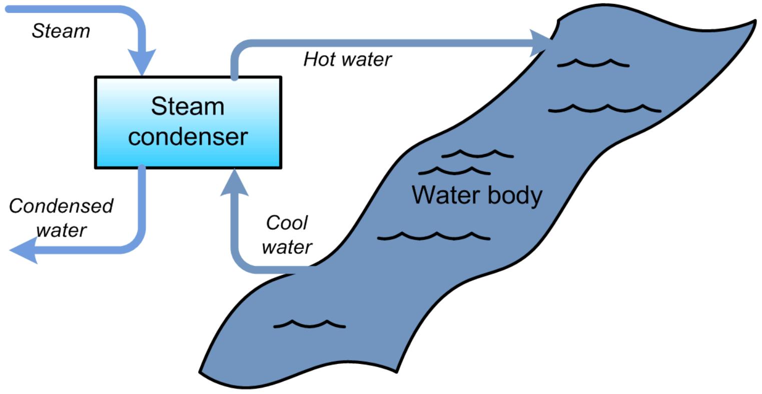


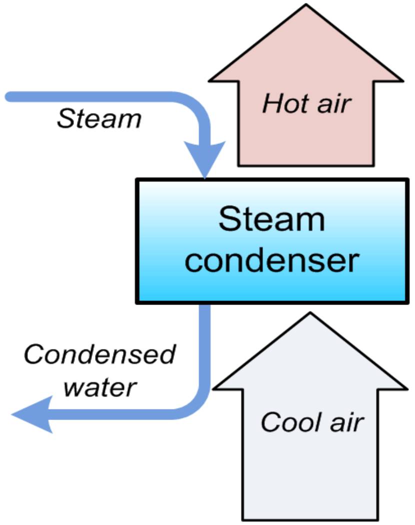
Figure 2 | Assessing thermoelectrics. Efficiency of 'best practice' mechanical heat engines compared with an optimistic thermoelectric estimate (see main text for description).

Cooling system types and tradeoffs

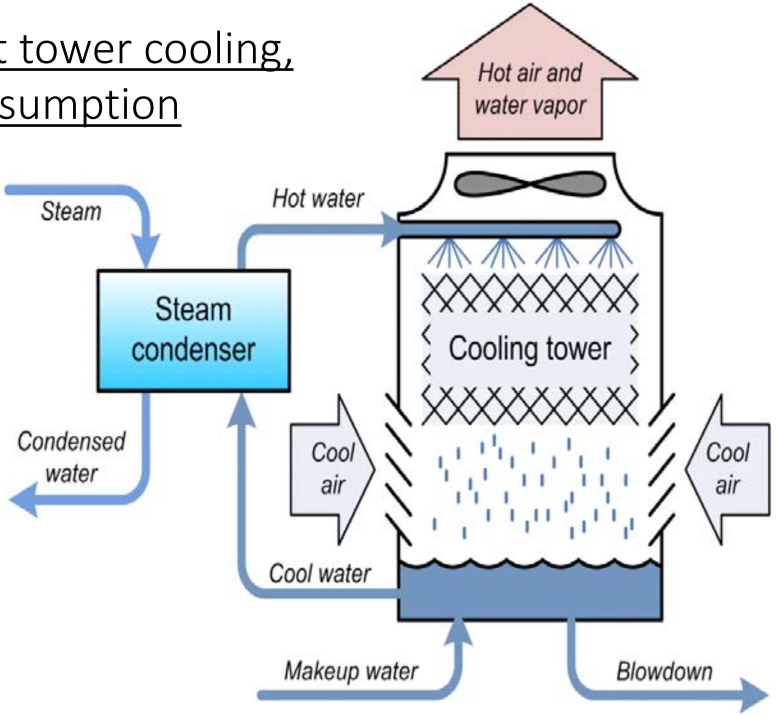
Once-through cooling, withdrawal



Dry cooling

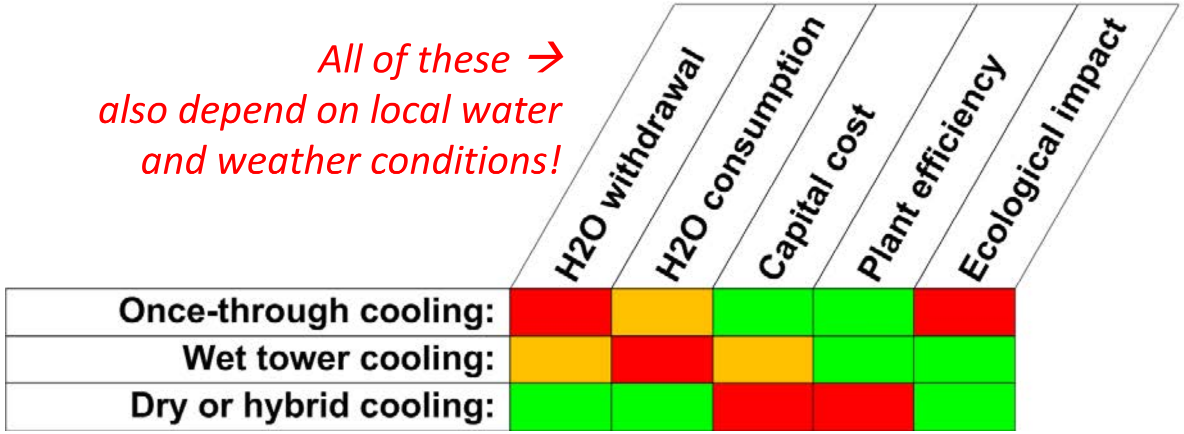


Wet tower cooling, consumption



exit T is the dew point of water at its partial p in the exit air.

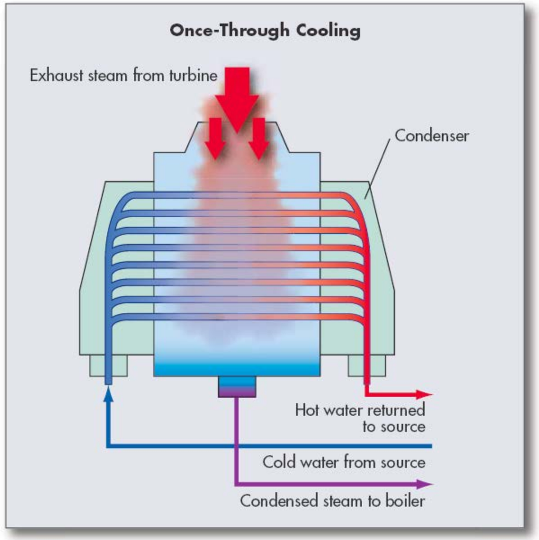
All of these →
also depend on local water
and weather conditions!



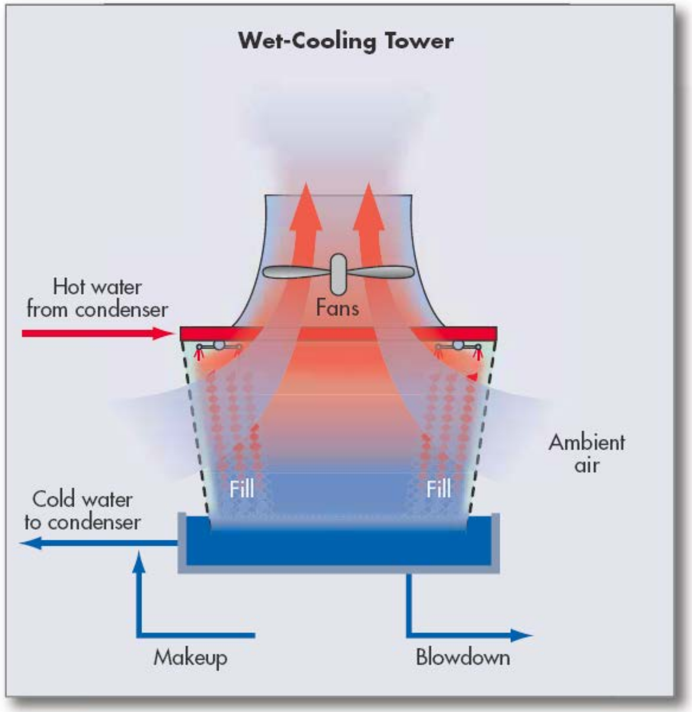
	H2O withdrawal	H2O consumption	Capital cost	Plant efficiency	Ecological impact
Once-through cooling:	🟥	🟧	🟩	🟩	🟥
Wet tower cooling:	🟧	🟥	🟧	🟩	🟩
Dry or hybrid cooling:	🟩	🟩	🟥	🟥	🟩

Cooling system types

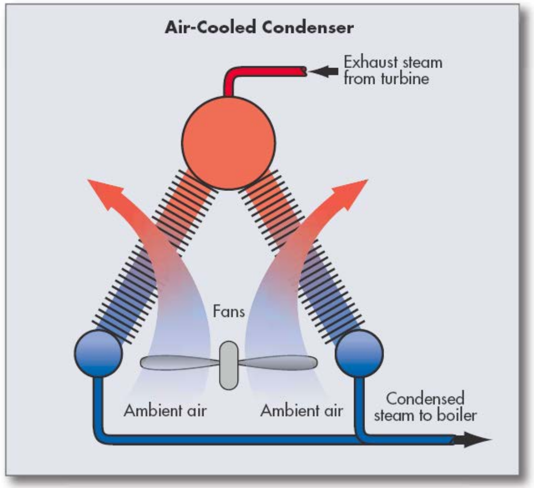
Image: EPRI Journal Summer 2007 "Running Dry at the Power Plant"



- Simple, low-cost
- Condensate temp approaches source temp
- High withdrawal but ow consumption, about 1% of withdrawal,
- Ecological issues: organism entrainment and impingement, hot effluent
- Use being phased out in the US under Clean Water Act

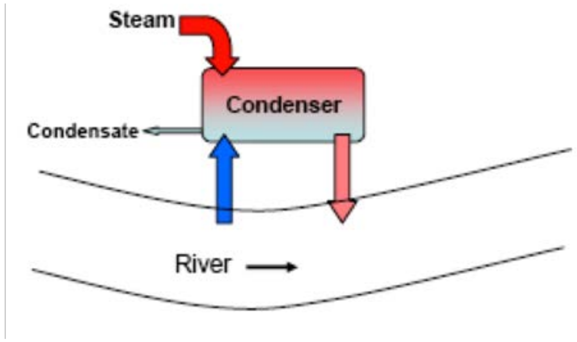


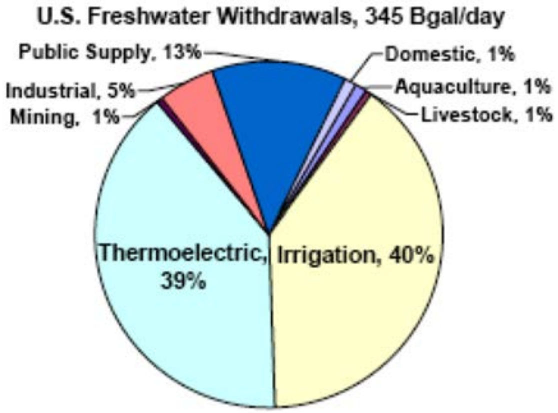
- More complex and costly
- Cooling water temp approaches ambient wet-bulb temp
- Lower withdrawal, 2 orders of magnitude less
- High consumption

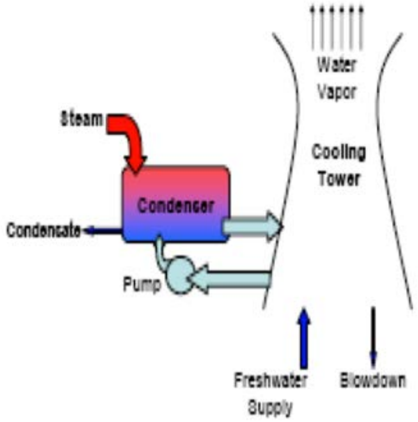


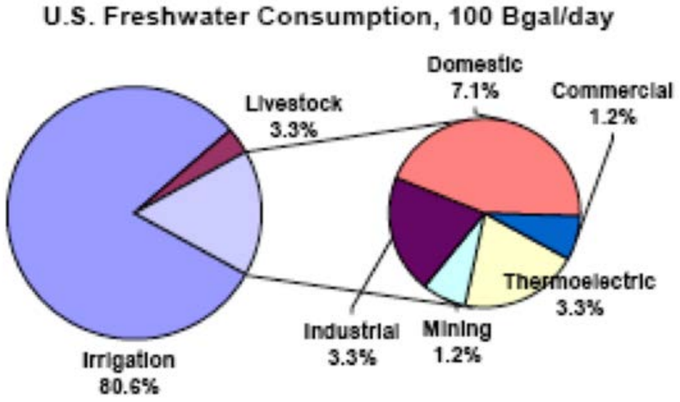
- Very expensive, 3-4x more than evaporative
- Condensate temp approaches ambient dry-bulb temp, poor efficiency on hot days
- Zero withdrawal and consumption

In late 90ies, 59 BGPD seawater and 136 BGPD fresh water were withdrawn for thermoelectric power plant* (39% of total)*, only 3.3 BGPD were consumed (~20% of non agri consumption)*, other returned at higher T (causes further evaporation, estimated at 1%). About 30 % of US plants use open loop cooling*. Most plants after 1970 utilize closed loop cooling









Electricity is as thirsty as the livestock or industrial use

2. HELLER System References – cont.

GEBZE & ADAPAZARI 3 x 777 MW$_e$ CCPP commissioned in 2002 (Turkey)
EPC-Contractor: BECHTEL-ENKA JV, End-user: INTERGEN
The world's largest dry cooled combined cycle power plant

The Advanced Heller System, by A. Balogh and Z. Szabo, EPRI Conference on Advanced Cooling Strategies/Technologies, June 2005, Sacramento, CA

Working fluids requirements:

1. High T_c for efficiency but low p_c for simplicity
2. Large enthalpy of evaporation
3. Non toxic, non flammable, non corrosive, cheap ..

Water: p_c=22.088 MPa T_c=374 C, most common
CO_2: p_c=7.39 MPa, T_c=30.4C (low p)

Can also use a bottoming cycle (Binary Cycle) to avoid strong vacuum, but need exotic fluids (mercury…)

Renewable sources (low to very low T for solar and geothermal):
Ammonia: p_c=11.63 MPa, T_c=132 C.
Propane: p_c= 4.26 MPa, T_c= 97 C
Isobutane, Freon

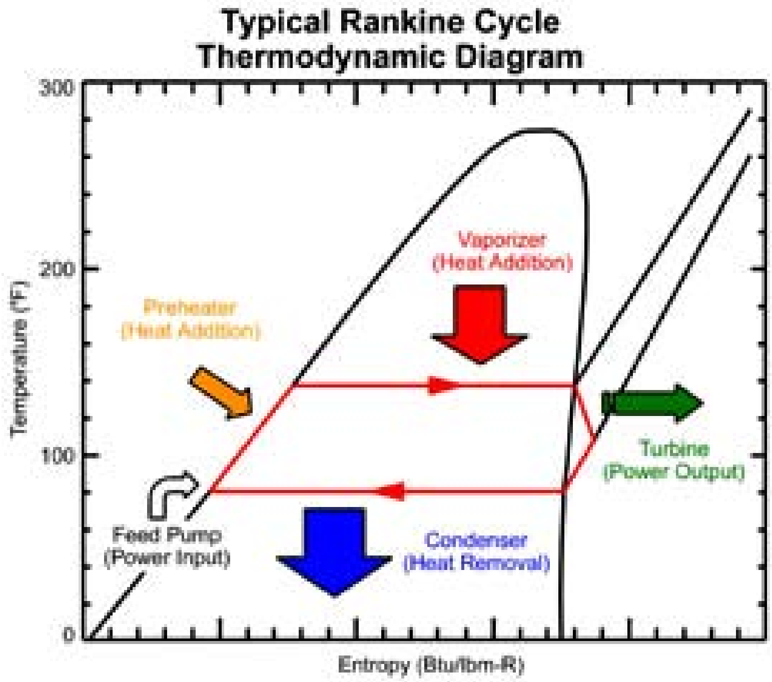
Organic Rankine Cycles

Solar Energy Applications:

When flat plate collectors are used, maximum heat transfer fluid temperature is ~ 150 C.

When geothermal heat sources are used, maximum temperature is below 200 C.

In both case, working fluid critical temperature should be lower. An example these “organic” working fluids, used in “Organic Rankine Cycles” is shown:



Ha Teboho Village, Lesotho
Matt Orosz, Liz Wyman et al.

2.60J Fundamentals of Advanced Energy Conversion
Spring 2020