Lecture # 15
Thermo-mechanical Conversion Gas Turbine Power Plants

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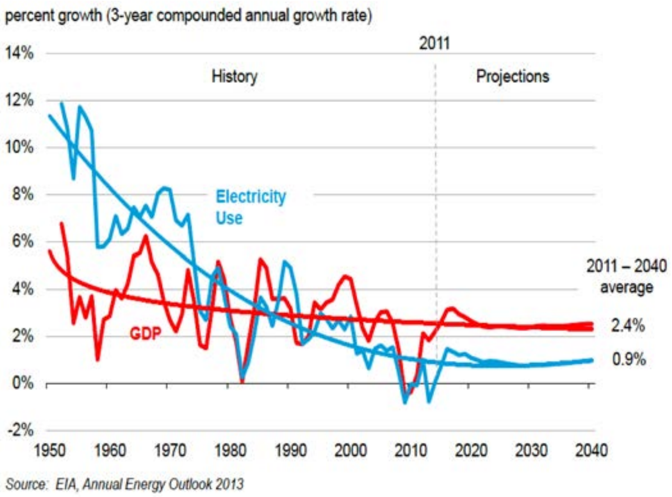
March 30, 2020

1. Why Gas Turbines
2. High *T* gas turbine cycles.
3. Recuperation.
4. Recovery of exhaust energy in HAT
5. Recovery of exhaust energy in chemical recuperation

Scenarios: Generation in kWh, now and in 2040

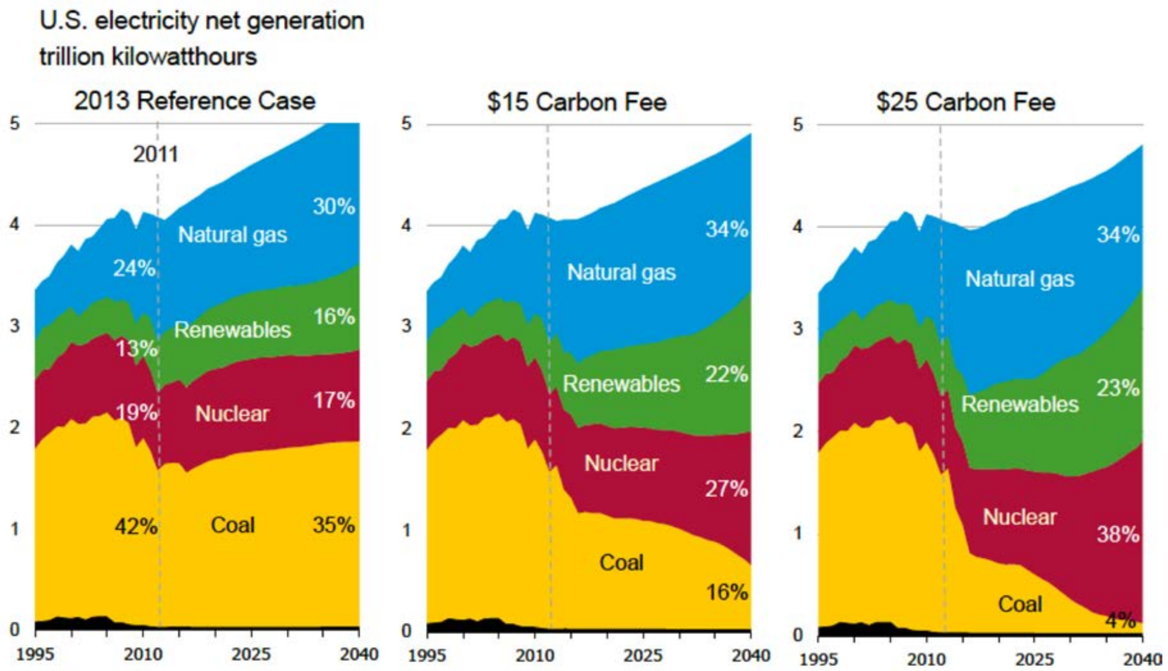
How to achieve certain targets (total electricity production) given constraints.
Without CO_2 constraints, coal remains the largest source for electricity production but NG grows significantly.
Renewables (hydropower, wind and solar) grow.
With CO_2 constraints, coal dies and NG and renewables grow much faster, with added nuclear.



Source: EIA, Annual Energy Outlook 2013

Image courtesy of Energy Information Administration (EIA).

US annual growth of electricity demand and GDP, indicating significant efficiency improvement



Source: EIA, Annual Energy Outlook 2013

Image courtesy of Energy Information Administration (EIA).

NG, Nuclear and renewables benefit significantly from CO_2 prices

MARCH 8, 2019

New U.S. power plants expected to be mostly natural gas combined-cycle and solar PV

Source: U.S. Energy Information Administration, *Annual Energy Outlook 2019*

EIA's long-term projections show that most of the electricity generating capacity additions installed in the United States through 2050 will be natural gas combined-cycle and solar photovoltaic (PV). Onshore wind looks to be competitive in only a few regions before the legislated phase-out of the production tax credit (PTC), but it becomes competitive later in the projection period as demand increases and the cost for installing wind turbines continues to decline.

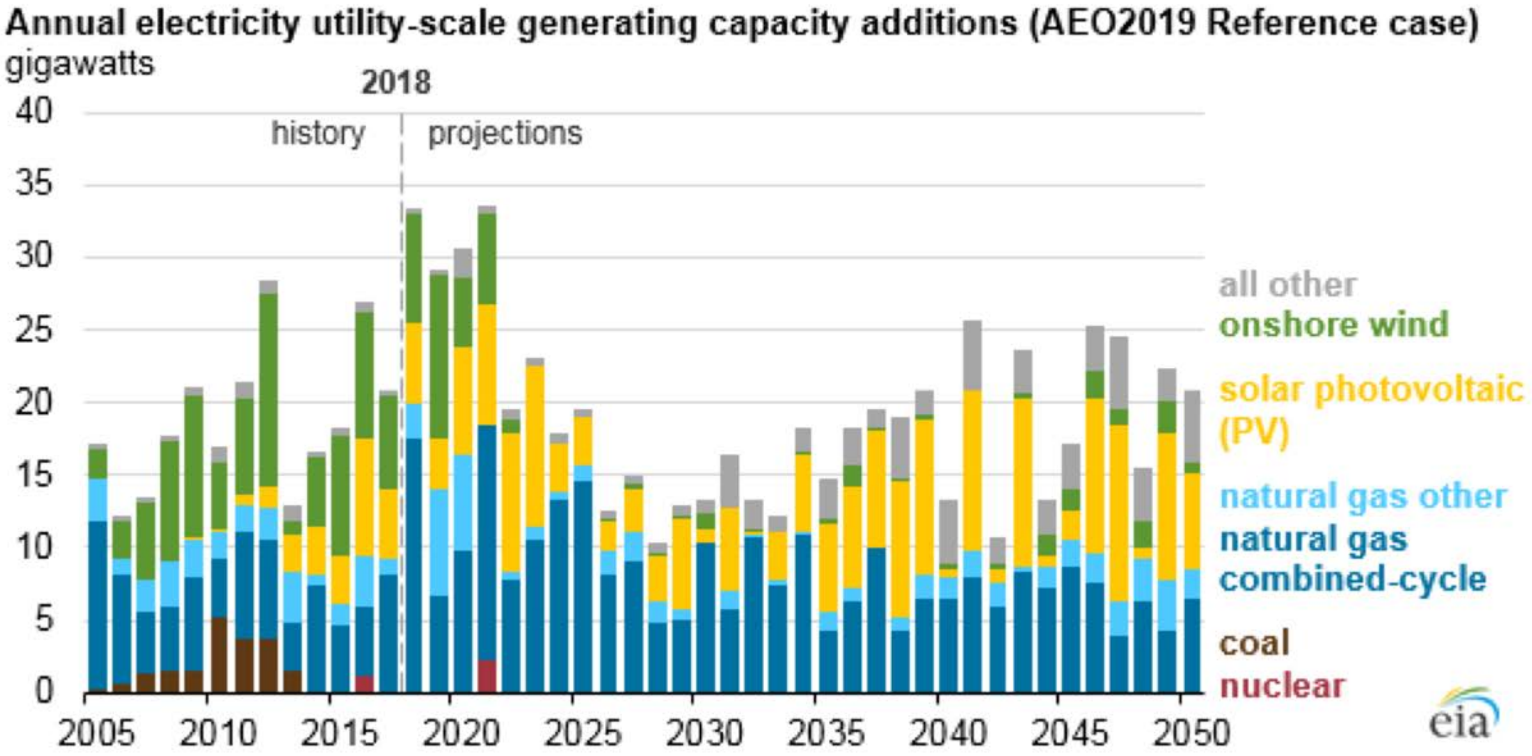


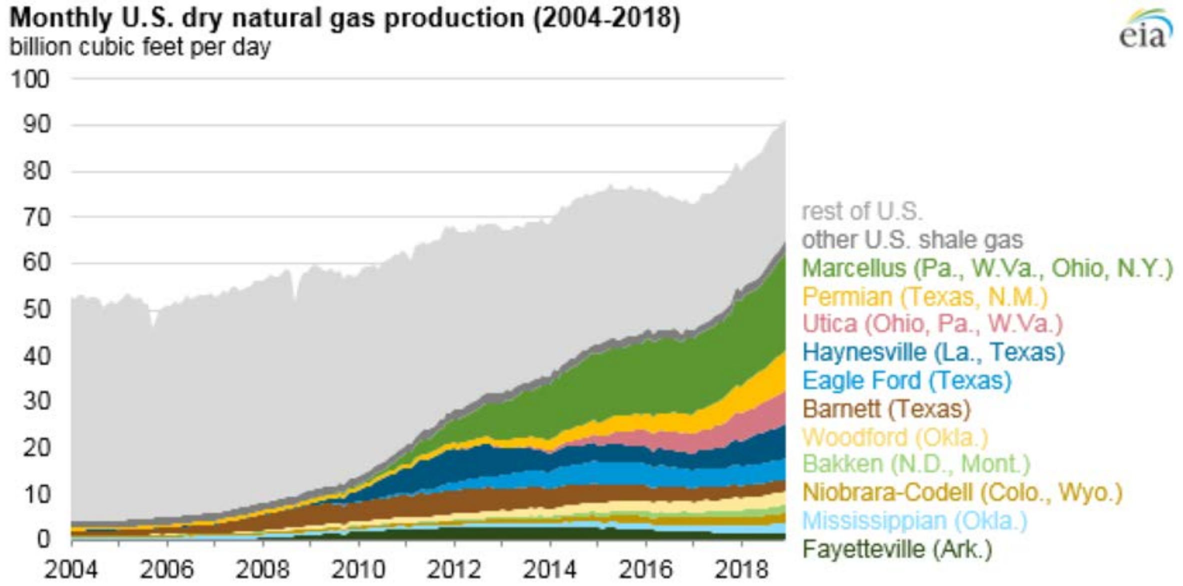
Image courtesy of Energy Information Administration (EIA).

https://www.eia.gov/todayinenergy/detail.php?id=38612

[EIA adds new play production data to shale gas and tight oil reports]

Source: U.S. Energy Information Administration, *Natural Gas Monthly*, *Petroleum Supply Monthly*, and *Short-Term Energy Outlook*, and DrillingInfo

Impact of fracking on US oil and gas production



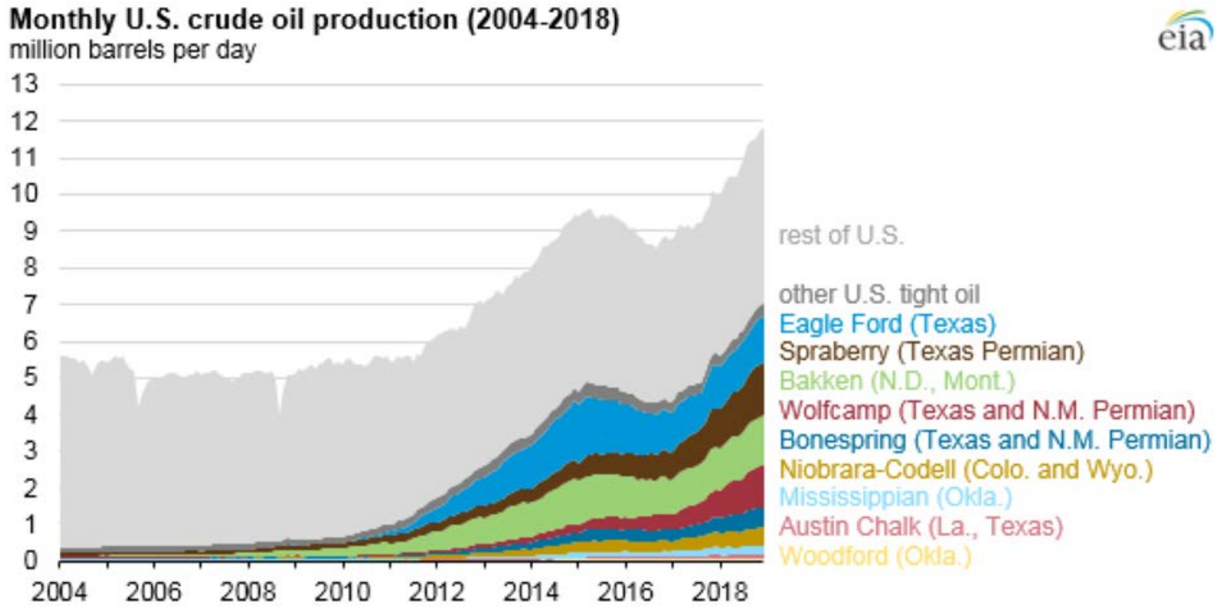
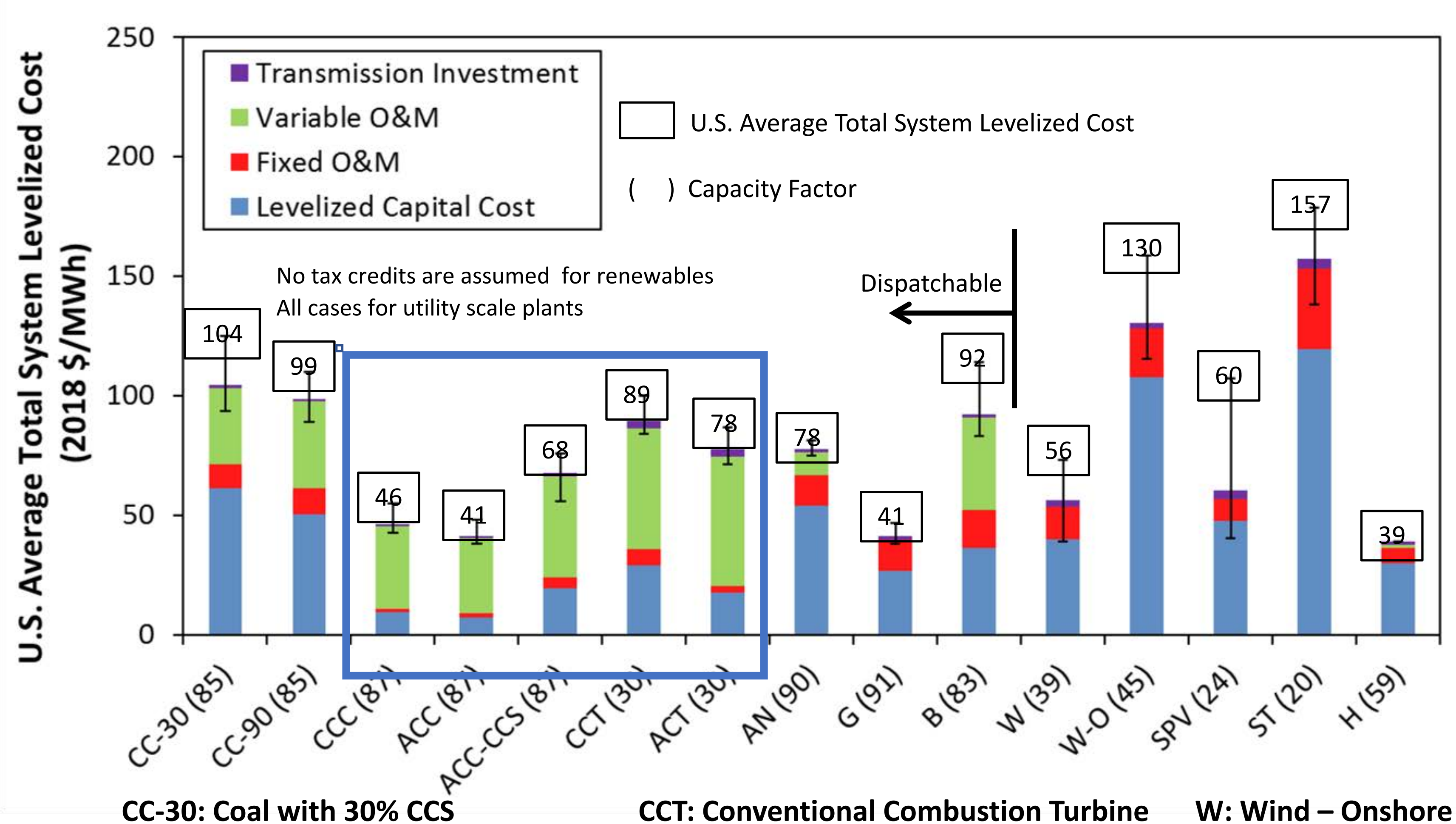


Image courtesy of Energy Information Administration (EIA).

https://www.eia.gov/todayinenergy/detail.php?id=38372

Estimated (in 2019) Levelized Cost of Electricity Generation Plants in 2023



CC-30: Coal with 30% CCS
CC-90: Coal with 90% CCS
CCC: Conventional Combined Cycle
ACC: Advanced Combined Cycle
ACC-CCS: Advanced CC with CCS

CCT: Conventional Combustion Turbine
ACT: Advanced Combustion Turbine
AN: Advanced Nuclear
G: Geothermal
B: Biomass

W: Wind – Onshore
W-O: Wind – Offshore
SPV: Solar PV
ST: Solar Thermal
H: Hydroelectric

Image courtesy of Energy Information Administration (EIA).

Source: U.S. Energy Information Administration, Annual Energy Outlook 2019, Feb 2019.

Carbon dioxide production in electricity generation: for each mole of fuel we produce:

$\left|\Delta\hat{h}_{R,f}\right|$ MJ of thermal energy, $\eta_e\left|\Delta\hat{h}_{R,f}\right|$ MJ_eelectricity and $\nu_{CO_2}M_{CO_2}$ kg-CO_2

or in short: $\dfrac{\nu_{CO_2}M_{CO_2}}{\eta_e\left|\Delta\hat{h}_{R,f}\right|}$ $\mathrm{kgCO_2/MJ_e}$

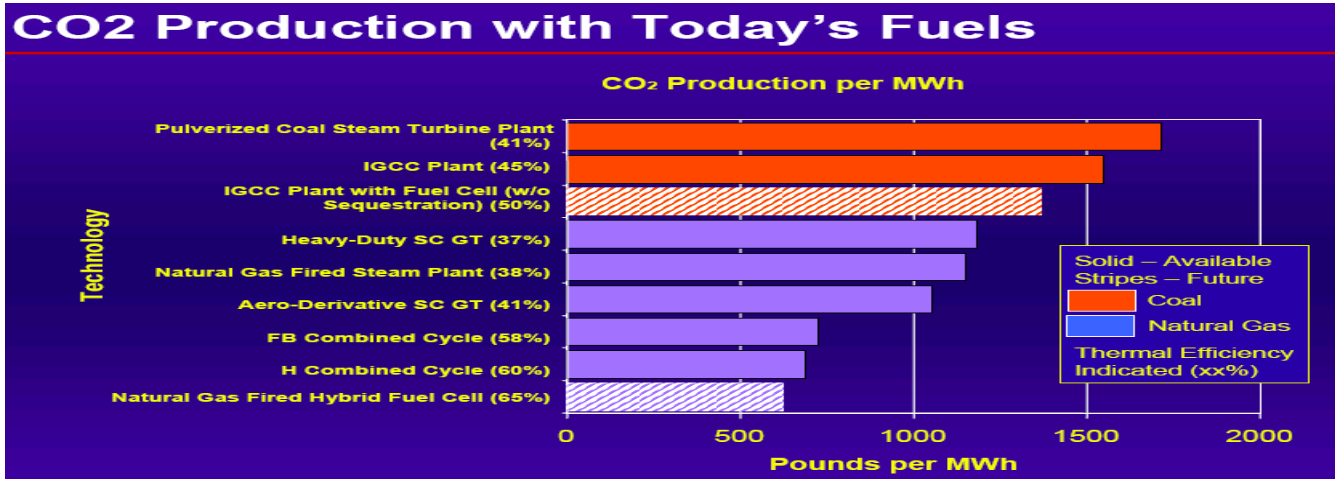
ν_{CO_2} number of moles of CO_2 per mole of fuel burned (=1 for coal or methane),

$M_{CO_2} = 44$ molecular weight of CO_2,

η_e is the plant efficiency (0.4 for coal and 0.55 for methane),

$\Delta\hat{h}_{R,f}$ the molar enthalpy of reaction of the fuel (~360 for coal and 800 for methane).

For methane, in a combined gas-steam cycle with 55% efficiency, 0.1 $\mathrm{kgCO_2/MJ_e}$.
For coal, in a simple steam cycle with 35% efficiency, 0.3 $\mathrm{kgCO_2/MJ_e}$.

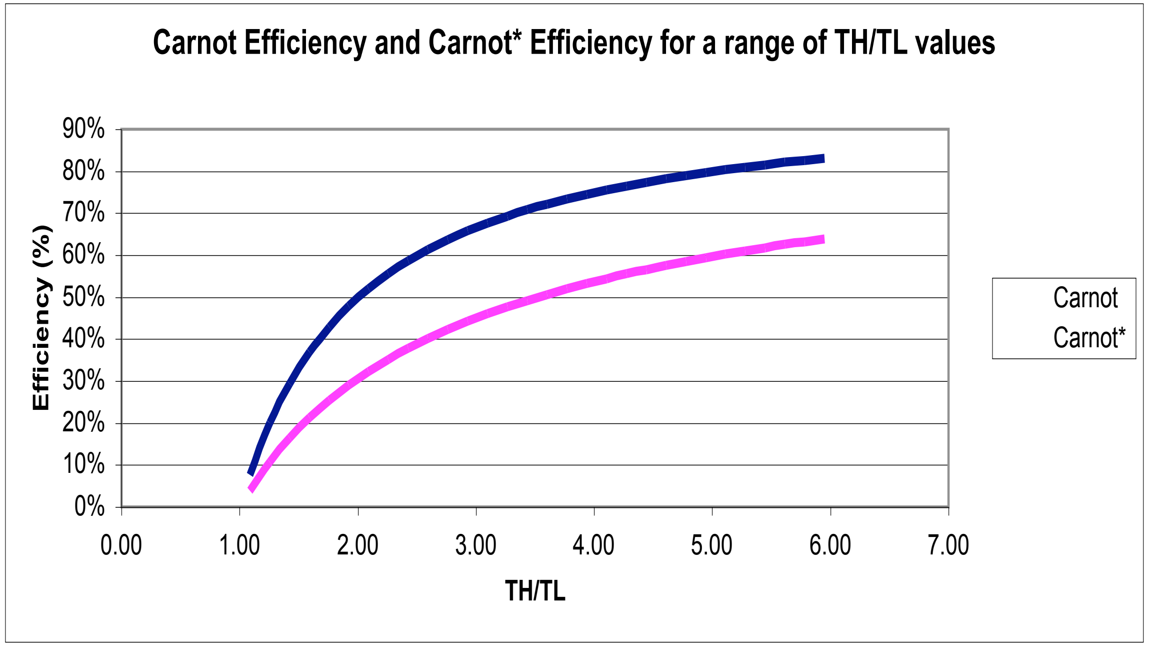


To convert to $\mathrm{kgCO_2/MJ_e}$, multiply the number given in the plot by $0.12\ 10^{-3}$

Alan Walker, GE Power Systems, H2 Symposium, MIT, Sep 2003. SC: Simple Cycle.

Thermomechanical efficiency depends on “heat source” Temperature

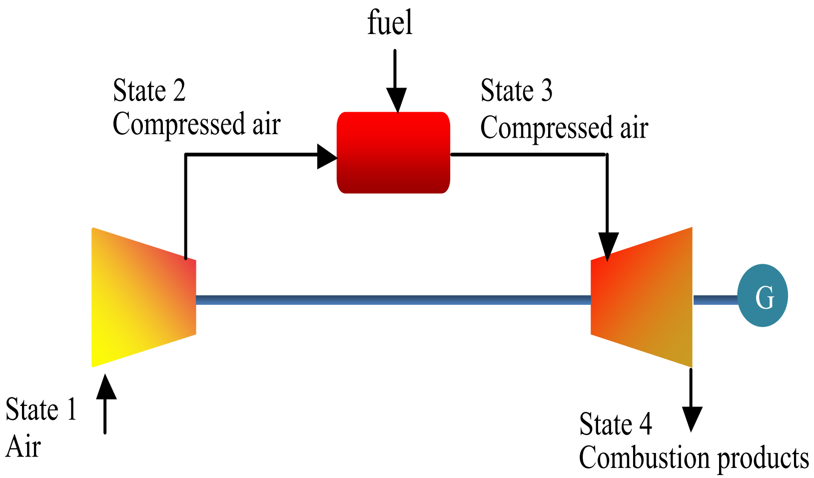
Power plant	**T_H in C**	**T_H/T_L**
Pressurized heavy water reactor (PHWR)	260-280	1.8-2.0
Boiling water reactors (BWR),	280-290	1.8-2
Pressurized water reactors (PWR	300-350	2.0-2.1
Metal cooled reactors	550	3
Compressed gas reactors (CGR	700-800	3-4
Solar thermal with troughs	280-350	2-2.2
Solar thermal with towers	Up to 500	3
Solar thermal with dishes	750	3.5
Geothermal plants	100-200	1.5
Gas turbine with NG	900-1400	4-5



Thermodynamic Models of Gas Turbine
Brayton Open Cycles

$$\eta_I = \frac{\text{net work out}}{\text{heat transfer in}} \quad \text{and} \quad \eta_{II} = \frac{\text{net work out}}{\text{maximum work}}$$

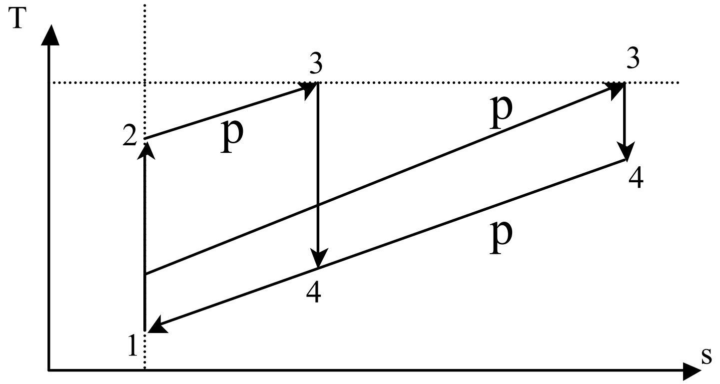
$$\eta_{fuel-utilization} = \frac{\wp}{\dot{m}_f \bullet \xi_f^o} \approx \frac{\wp}{\dot{m}_f \bullet LHV}$$



Ideal simple Brayton cycle: $\eta_I = 1 - \left(\frac{1}{\pi_P}\right)^{(k-1)/k} = 1 - \frac{1}{\vartheta_{2s}}$

π_p = pressure ratio across compressor, $\vartheta_{2s} = T_{2s} / T_1$.

at higher π_p, less fraction of the heat is rejected (see schematic)



Ideal gases, air standard cycle …

$$\frac{T_{2s}}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}, \text{ and } \eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{T_{2s} - T_1}{T_2 - T_1},$$

hence: $T_2 = T_1 + \dfrac{T_{2s} - T_1}{\eta_C}$

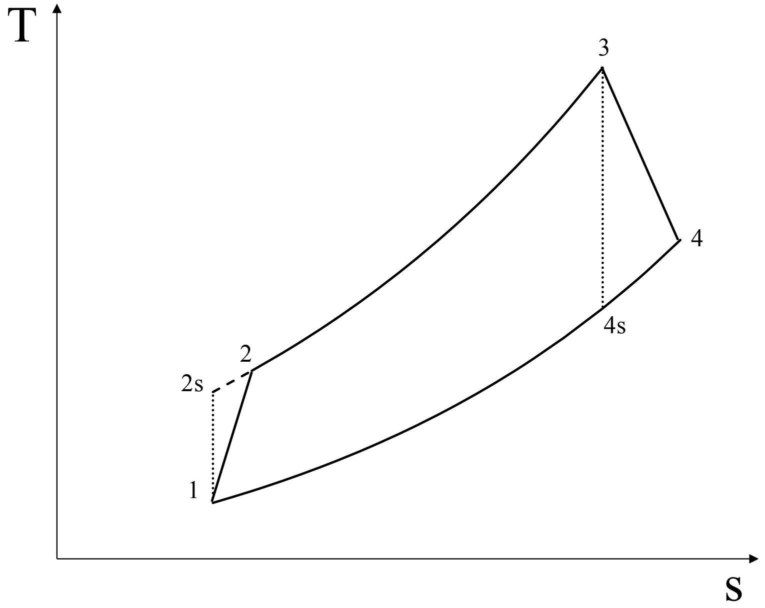
$$\frac{T_3}{T_{4s}} = \left(\frac{p_3}{p_4}\right)^{\frac{k-1}{k}} = \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}, \text{ and } \eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{T_3 - T_4}{T_3 - T_{4s}},$$

hence: $T_4 = T_3 - \eta_T \left(T_3 - T_{4s}\right)$

$$Q_{in} = h_3 - h_2 = c_p \left(T_3 - T_2\right)$$

$$W_{net} = \left(h_3 - h_4\right) - \left(h_2 - h_1\right) = c_p \left[\left(T_3 - T_4\right) - \left(T_2 - T_1\right)\right]$$

$$\eta_{cycle} = \frac{W_{net}}{Q_{in}}$$



Compressor efficiency is key …
T_{max} < 1000 C, Modern designs, T_{max} ~ 1400 C.

compressor work: $$w_c = \frac{c_p T_1}{\eta_C}\left[\pi_p^{(k-1)/k} - 1\right]$$

Turbine work: $$w_t = \eta_T\, c_p T_3 \left[1 - \frac{1}{\pi_p^{(k-1)/k}}\right]$$

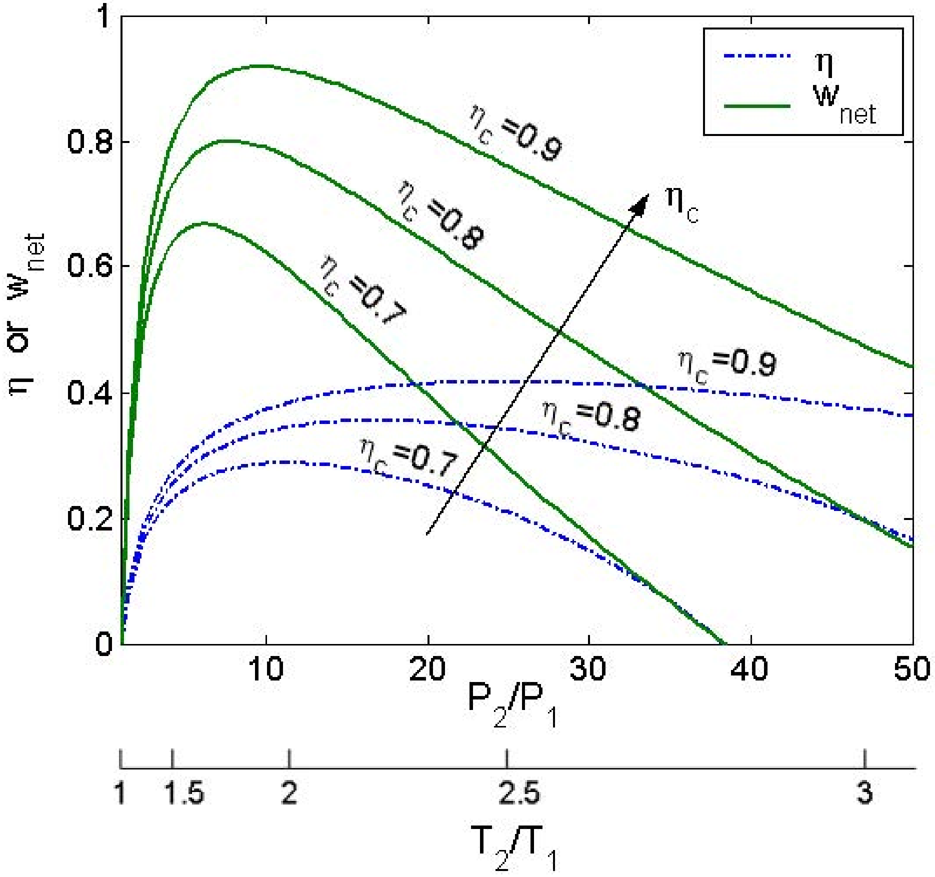
Air, $\pi_p = 4$	W_C	W_T	W_{net}	Q_H	η	T_4 (K)
Ideal	143.0	352.3	209.3	640.2	0.327	722.1
Real, η_c=0.85	168.2	317.1	148.9	614.9	0.242	757.2
Real, η_c=0.65	219.9	317.1	97.2	563.2	0.173	757.2
With regeneration η_c=0.85					0.412	

Air, $\pi_p = 8$	W_C	W_T	W_{net}	Q_H	η	T_4 (K)
Ideal	238.7	482.6	243.9	544.4	0.448	592.3
Real, η_c=0.85	280.8	434.3	153.5	502.3	0.306	640.4
Real, η_c=0.65	367.2	434.3	67.1	415.9	0.161	640.4
With regeneration, η_c=0.85					0.345	

Tables are for the following:
T_{min}= 20C, T_{max}=800C,
Carnot efficiency = 62.5%
Turbine isentropic efficiency = 90%

Figure 4. The impact of the compressor efficiency on the Brayton cycle efficiency and specific work, for $\vartheta_3 = 4.5, \eta_T = 90\%, \beta = 1$.

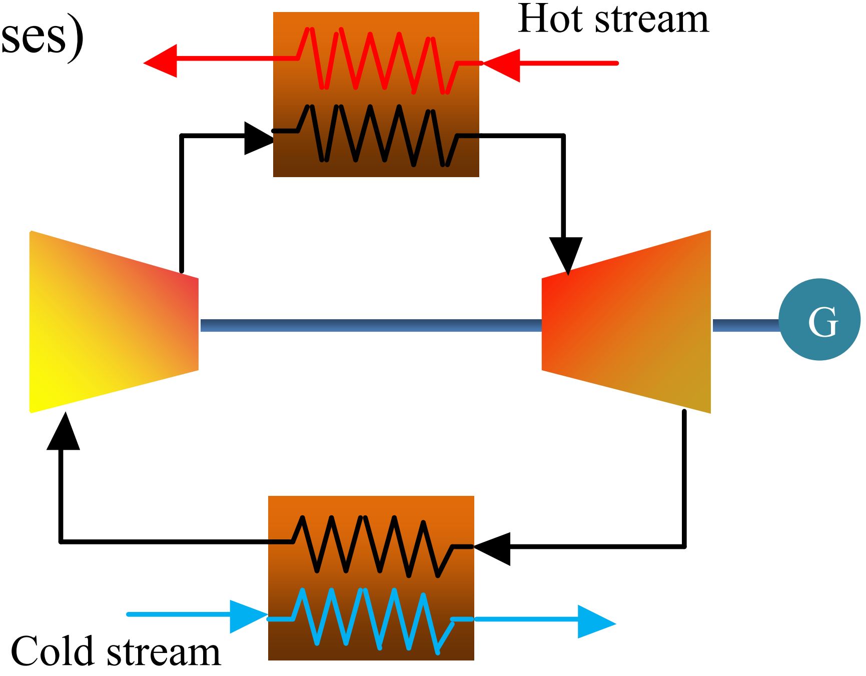
$$\vartheta \equiv T / T_1$$



- Compressor performance, or isentropic efficiency, is key.
- Note how the the specific power peaks at a certain pressure ratio.
- Also the efficiency peaks more sharply as the compressor efficiency decreases.

Closed Cycles: Not currently in use …

- ▲ Closed cycles allow flexbility in choosing working fluid;
- ▲ they need cooling, and turbine can exhaust at p lower than atmosphere (this may not be an advantage since compressor work increases)
- ▲ They can also use "dirty fuels" or nuclear (or renewable) heat.



$$\eta_I = 1 - \left(\frac{1}{\pi_P}\right)^{(k-1)/k}$$

Helium has:

higher $k = 1.67$, higher temperature @ low pressure ratio

Thus higher efficiency.

higher heat capacity, $c_{p,He} / c_{p,air} = 5$

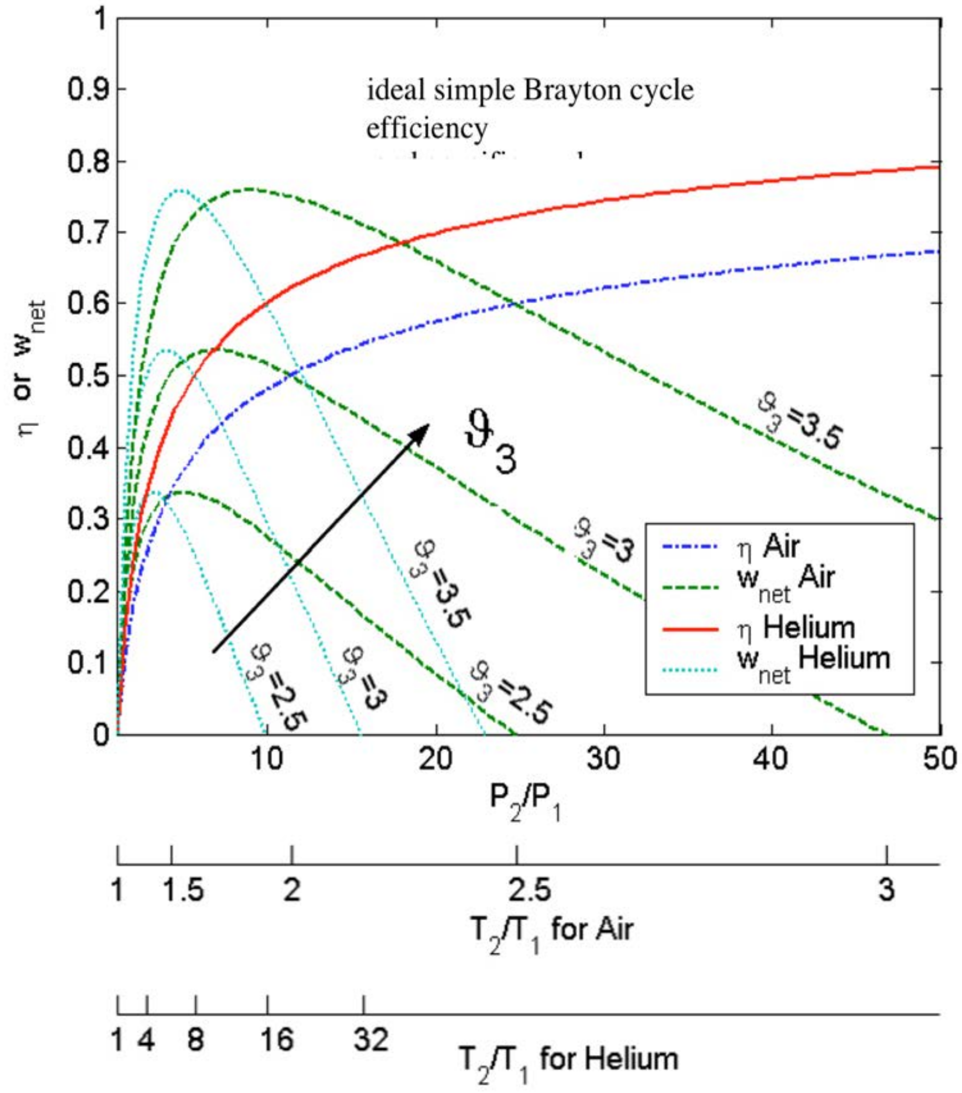
but it is less dense, i.e., high flow velocities needed.

Ideal cycle performance:

Impact of maximum turbine temperature on specific work:

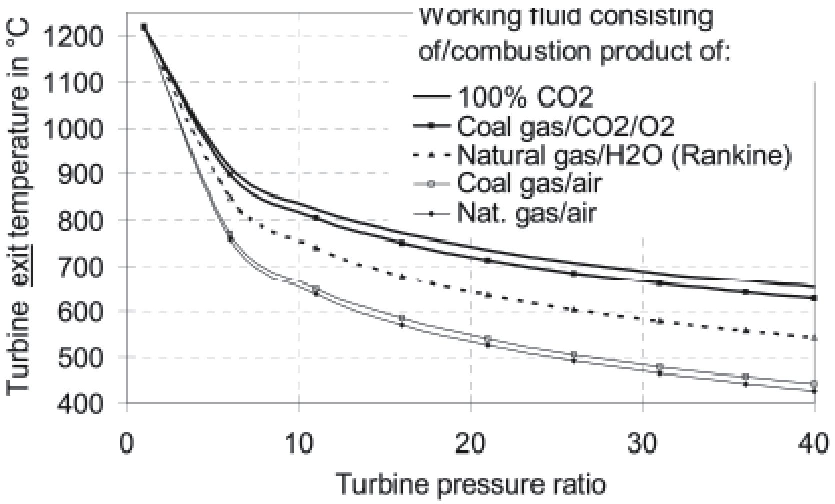
Impact of working fluid on efficiency and maximum work conditions.

- Choice of design point.
- Compromise between hardware cost (initial and running), and fuel cost, CO_2 emissions, etc.



$$\frac{T_3}{T_{4s}} = \left(\frac{p_3}{p_4}\right)^{\frac{k-1}{k}} = \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}},$$

$$\eta_I = 1 - \left(\frac{1}{\pi_P}\right)^{(k-1)/k}$$



2.15: Turbine exit temperature calculated for different working fluids in dependence on pressure ratio at a turbine inlet temperature of 1200ºC[22].

The gas turbine exit temperature calculated for different working fluids and pressure ratios across the turbine, for turbine inlet temperature of 1200 C. The working fluid consists of either pure CO2, or is the combustion products of the fuel and the oxidizer list in the figure, with stoichiometry adjusted to give the specified inlet temperature. The essential difference between the different gases is the effective isentropic index. The curve for helium is lower than that for NG/air because of the higher isentropic index of helium. Lower exit temperatures for the working fluid lead to higher overall cycle efficiency. But regeneration and combined cycles can be used to correct that!

Pressure losses during combustion can impact efficiency:

$$\beta_H = p_3 / p_2, \beta_L = p_1 / p_4, \beta * = \left(\beta_L \beta_H\right)^{\frac{k-1}{k}}, \pi_P^* = \pi_P^{(k-1)/k}$$

$$\eta = \frac{\eta_T \vartheta_{\max}\left(1 - \frac{1}{\beta * \pi_C^*}\right) - \frac{1}{\eta_C}\left(\pi_c^* - 1\right)}{\vartheta_{\max} - \left(1 + \frac{\pi_c^* - 1}{\eta_C}\right)}$$

$$w_{net} = \vartheta_{\max}\eta_T\left(1 - \frac{1}{\beta * \pi_c^*}\right) - \frac{1}{\eta_C}\left(\pi_c^* - 1\right)$$

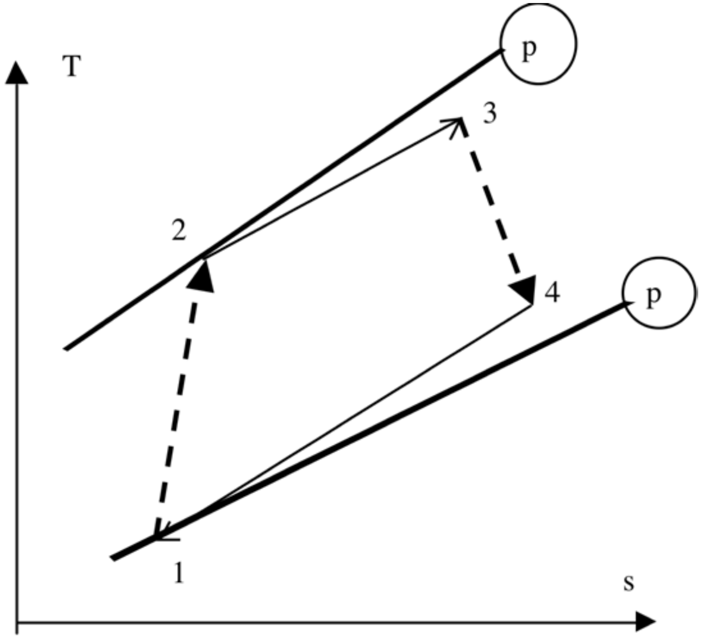
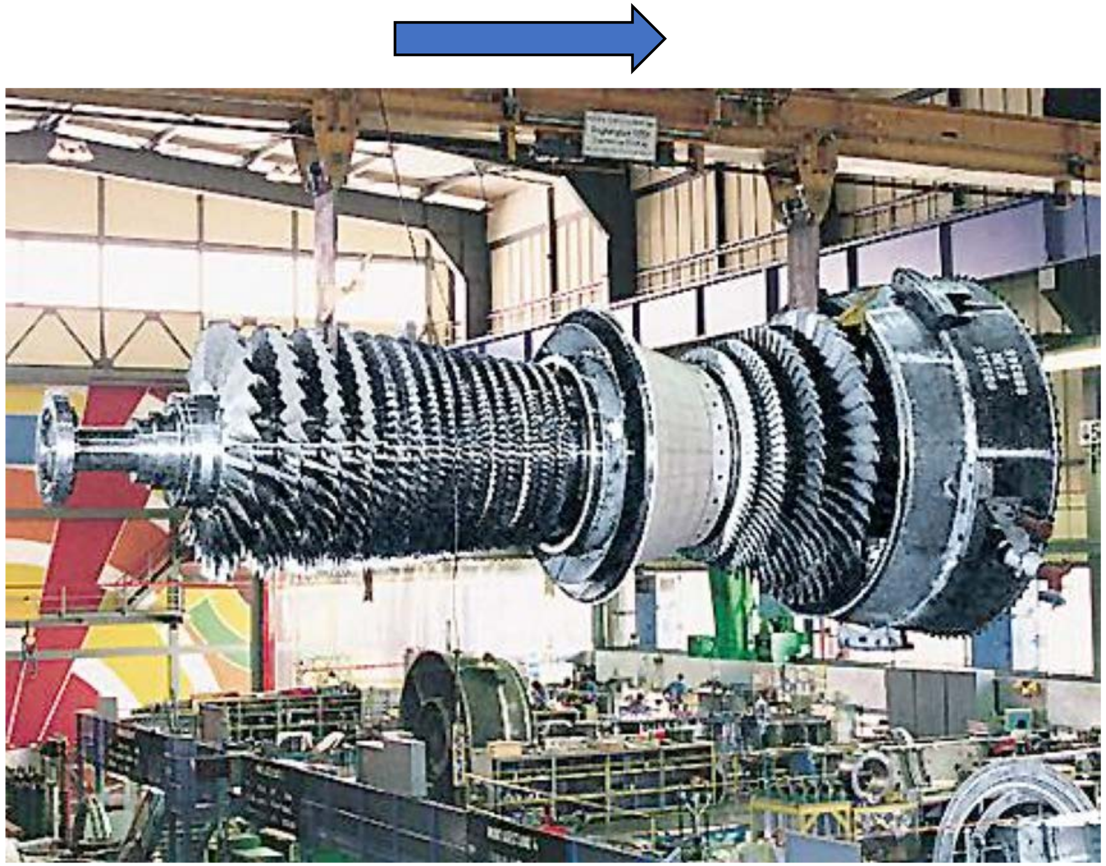


Figure 6. The temperature entropy diagram of a simple Brayton cycle, with isentropic efficiencies for the work transfer components, and pressure drop across the heat transfer components.

- Annular, walk-in combustion chamber with 24 hybrid burners
- Advanced cooling technology
- Ceramic combustion chamber tiles
- Optional multiple fuels capability
- 15-stage axial flow compressor with optimized flow distribution (controlled diffusion airfoils)
- Low-NOx combustion system
- Single-crystal turbine blades with thermal barrier coating and film cooling



SIEMENS: SGT5-4000F
(278 MW, 50Hz)

THE USE OF GAS TURBINES IN POWER GENERATION INCREASED FIVE FOLDS BETWEEN 1990 AND 2000, . Why?

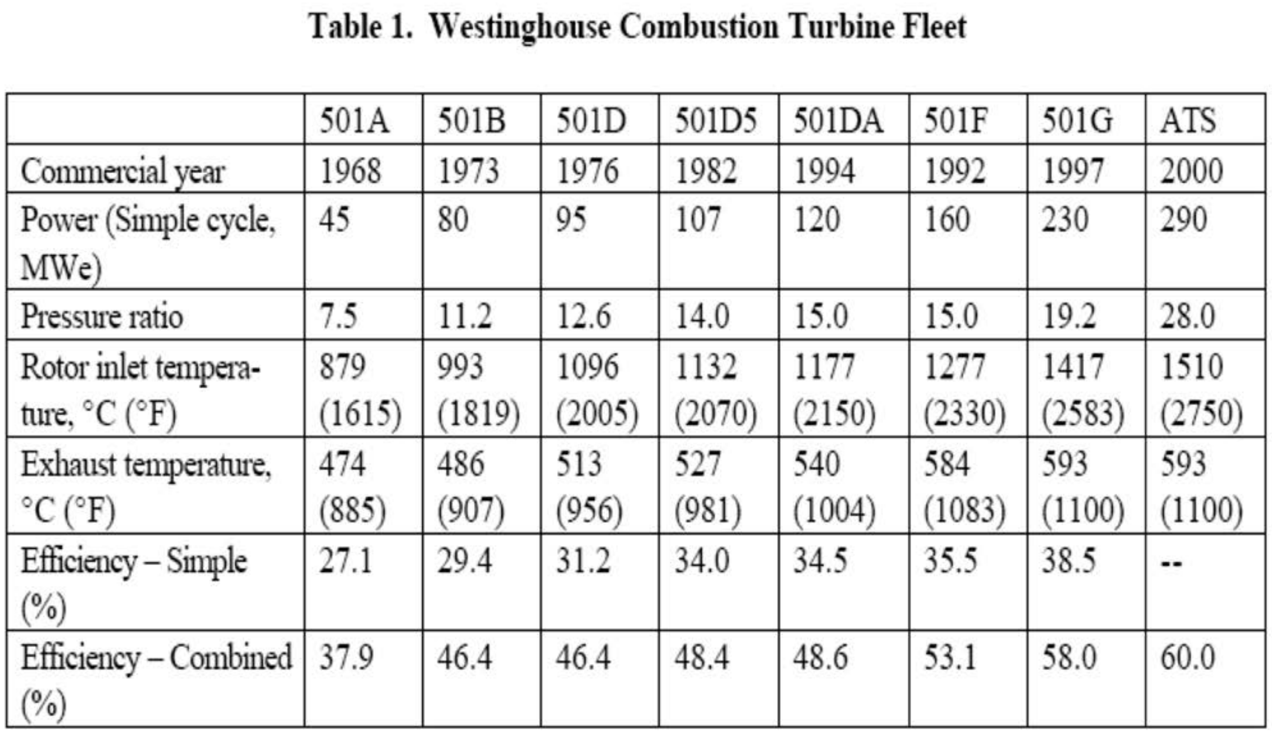
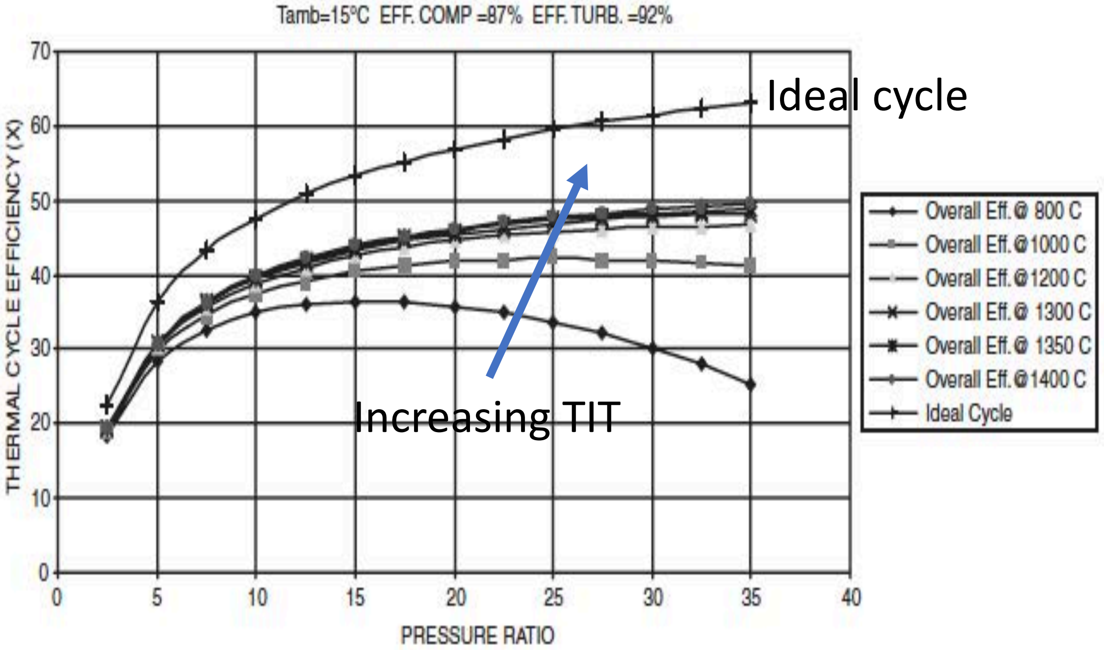


Table 1. Westinghouse Combustion Turbine Fleet

	501A	501B	501D	501D5	501DA	501F	501G	ATS
Commercial year	1968	1973	1976	1982	1994	1992	1997	2000
Power (Simple cycle, MWe)	45	80	95	107	120	160	230	290
Pressure ratio	7.5	11.2	12.6	14.0	15.0	15.0	19.2	28.0
Rotor inlet temperature, °C (°F)	879 (1615)	993 (1819)	1096 (2005)	1132 (2070)	1177 (2150)	1277 (2330)	1417 (2583)	1510 (2750)
Exhaust temperature, °C (°F)	474 (885)	486 (907)	513 (956)	527 (981)	540 (1004)	584 (1083)	593 (1100)	593 (1100)
Efficiency – Simple (%)	27.1	29.4	31.2	34.0	34.5	35.5	38.5	--
Efficiency – Combined (%)	37.9	46.4	46.4	48.4	48.6	53.1	58.0	60.0

"Advanced NG Fire Gas Turbine Systems ….." DOE contract DE-FG21-95MC32071, Westinghouse Electric .

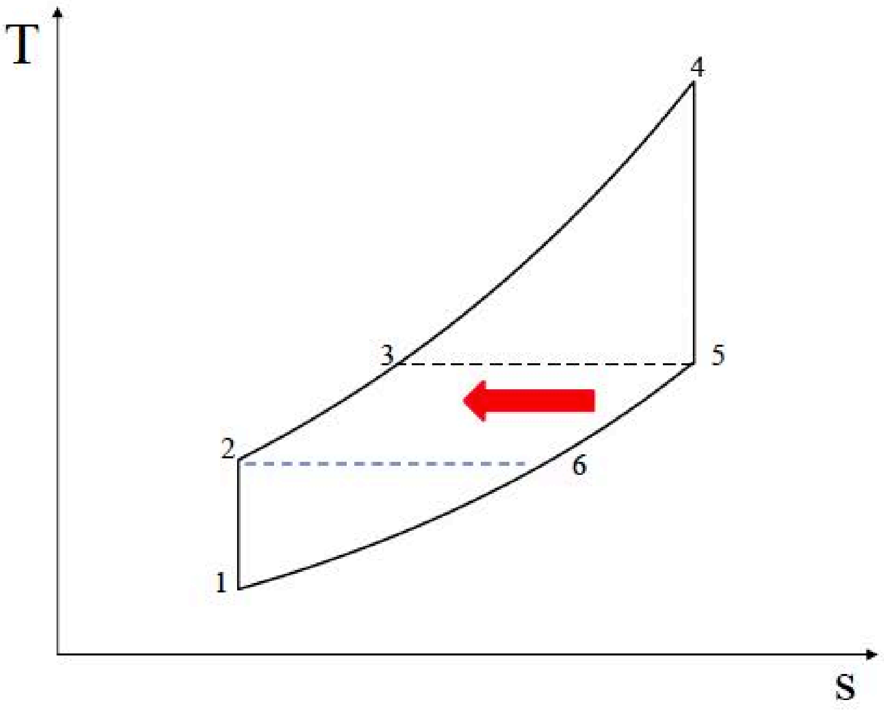
Image courtesy of DOE.



Impact of pressure ratio and turbine inlet temperature on overall cycle efficiency.

Boyce, *Gas turbine Handbook, 2nd Edition*. 2002.

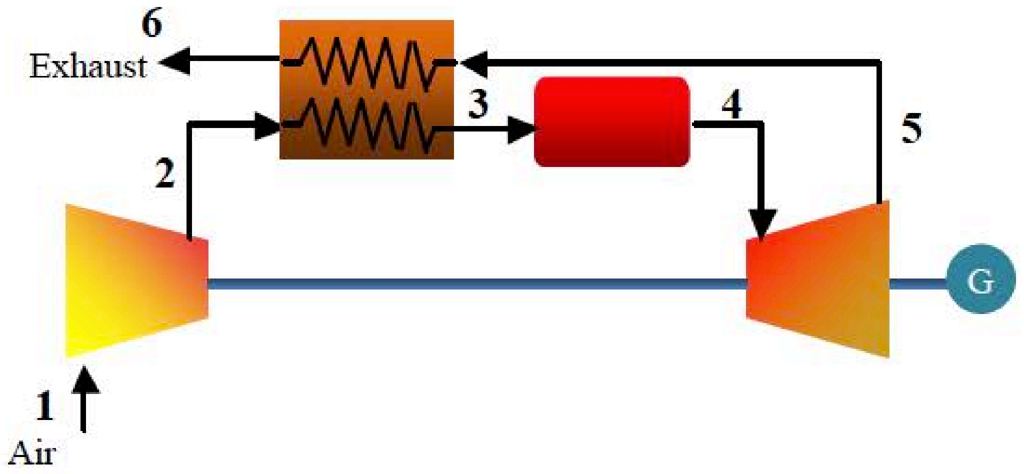
Exhaust heat recovery: Regenerative Cycles:



with ideal recuperation,

$T_3 = T_5$

$$\eta = 1 - \pi_P^{\frac{k-1}{k}} / \vartheta_4$$

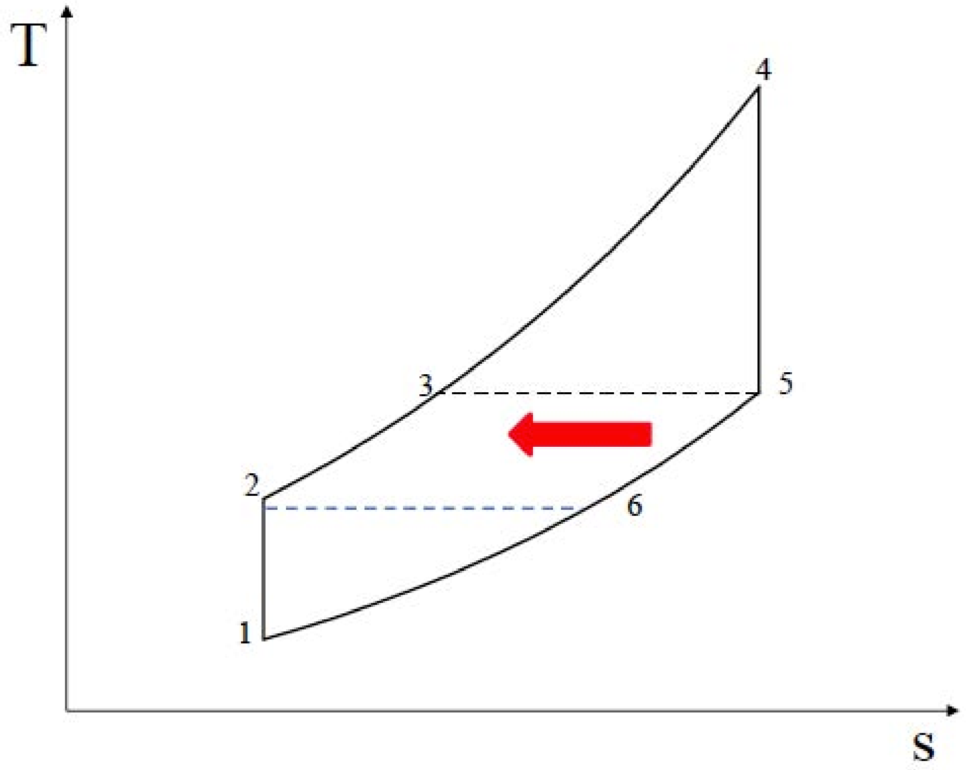


With 85% temperature recovery, $T_3 - T_{2s} = 0.85(T_{5s} - T_{2s})$, and 85% compressor and turbine efficiencies:

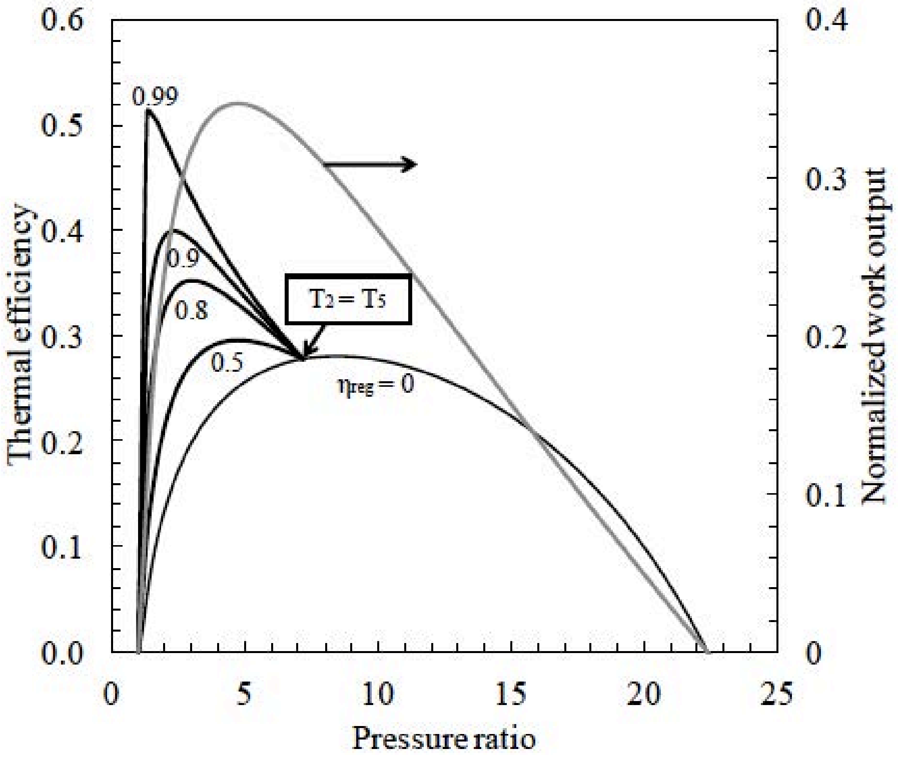
For $\pi_c = 4$: $\eta =$ 41.2% vs. 24.2% for a simple cycle.

For $\pi_c = 8$: $\eta =$ 34.5% vs. 30.6% for a simple cycle.

- Regeneration works best for low-p ratio, $T_5 >> T_2$.
- Intercooling and reheating improve performance.

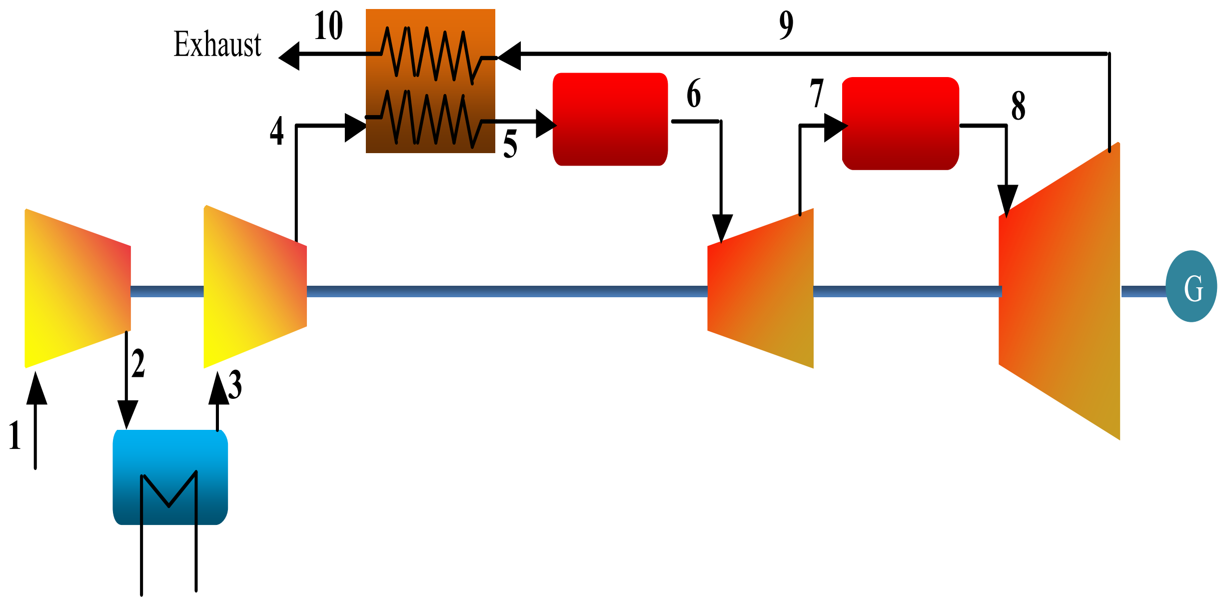


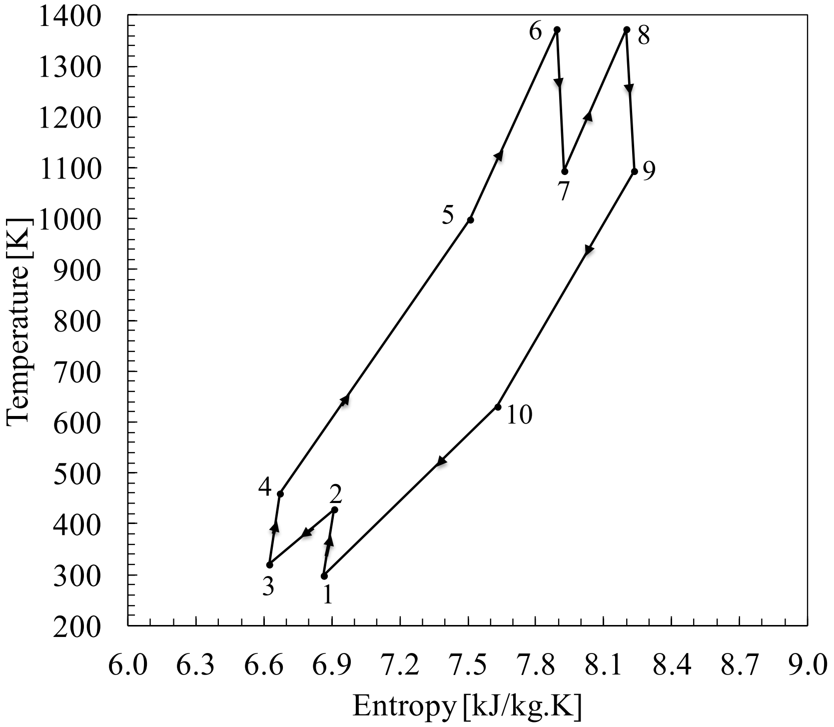
With regeneration, we recover some of the exhaust energy, making it possible to reduce the added heat (heat demand) and adding that heat at the highest possible T, resulting in a low exhaust temperature.



Impact of regeneration efficiency on the Brayton cycle efficiency, $\eta_T = \eta_C = 0.90$, $\vartheta_{\max} = 3$, $\beta = 1$. Numbers on the lines show the regeneration efficiency, defined as $((T_5\text{-}T_2)/T_5\text{-}T_6))$

Intercooling and Reheating:
Near Isothermal Heating and Cooling:





compressor work: $w_c = \frac{c_p T_1}{\eta_C}\left[\pi_p^{(k-1)/k} - 1\right]$ increases with T_1

Turbine work: $w_t = \eta_T\, c_p T_3 \left[1 - \frac{1}{\pi_p^{(k-1)/k}}\right]$ increases with T_3

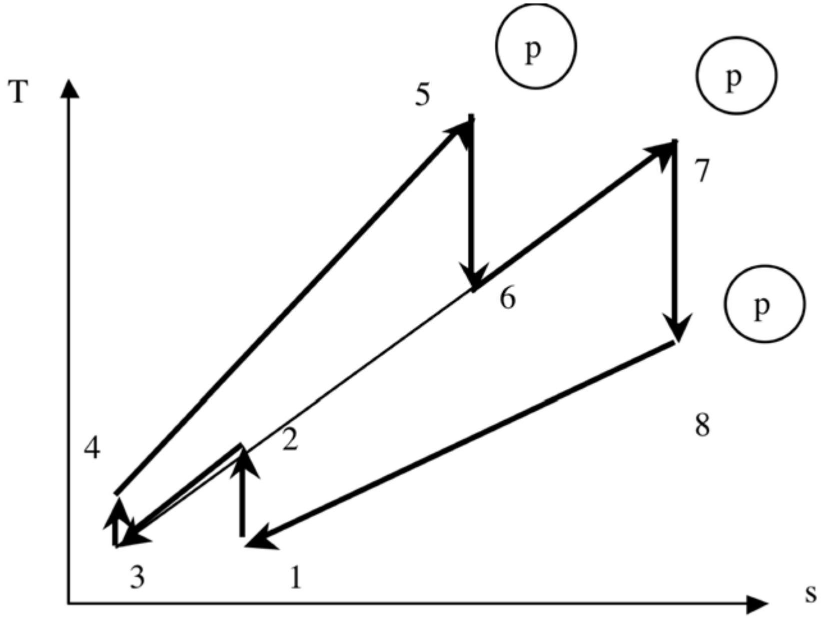


Figure 10. Ideal Brayton cycle with one intercooling stage and one reheating stage.

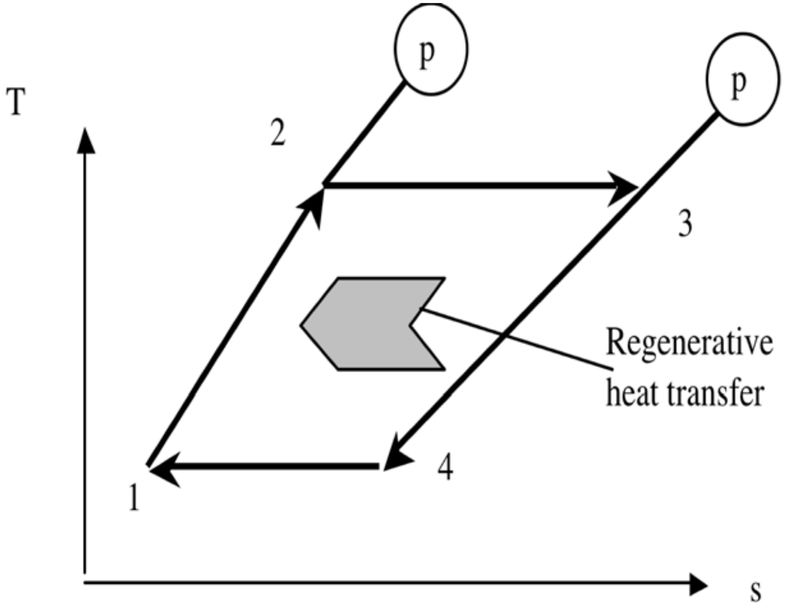
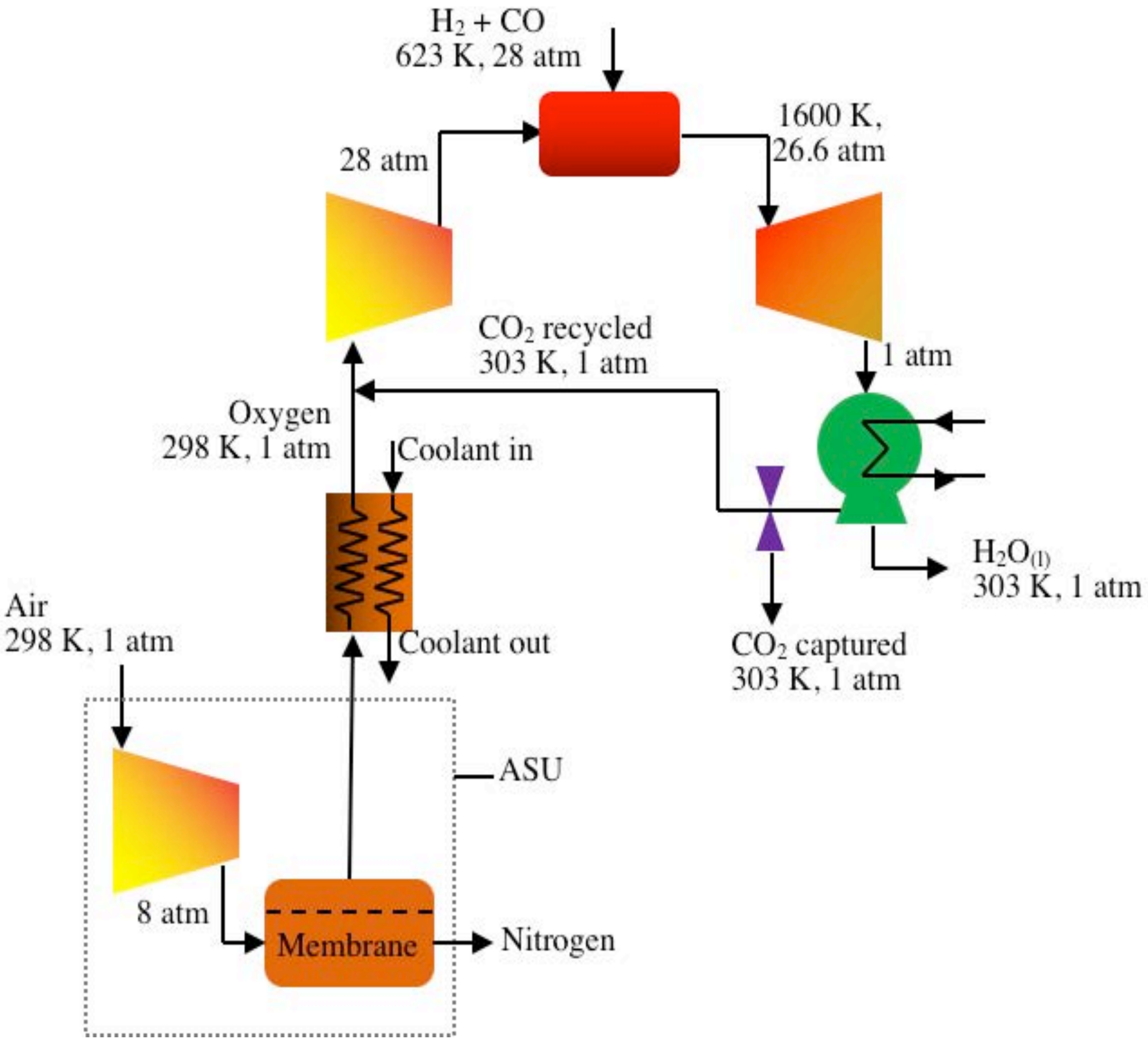


Figure 11. The ideal Ericsson cycle.

- **Intercooling** decreases compressor work, asymptotes to isothermal compression .. Minimum compression work
- **Reheat** increases power output and efficiency (work at high T).
- Both work better with **regeneration** (high turbine exit T and lower compressor inlet T).
- Asymptotes to **Ericsson Cycle,** has Carnot cycle efficiency (but with regeneration in the constant p processes)**.**

Example 5.3 A gas turbine power plant operates with oxy-fuel combustion and uses syngas (a mixture of 1 mole of hydrogen and 1 mole of carbon monoxide) as a fuel. Air at 25 °C and 1 atm is pressurized to 8 atm within an ASU, which produces oxygen at 1 atm. Oxygen is cooled to 30 °C (303 K not 298 K shown in fig.) before mixing with recycled CO_2. The mixture of oxygen and carbon dioxide is compressed to 28 atm. The syngas is burned adiabatically (and completely), and the products exit at 1600 K. The pressure drop within the combustor is 5%. The combustion products expand in the turbine whose isentropic efficiency is 90%. The turbine exhaust is cooled to 30 °C to condense water. Some of the CO_2 is recycled. Assume an isentropic efficiency of 80% for the compressors. How much CO_2 recycle is needed? Calculate the net power and thermal efficiency of the plant.



Solution is in notes

We begin the analysis from the first compressor where air is compressed for separation within the ASU. The air temperature at the compressor outlet is

$$T_1^{'} = T_0\left[1+\frac{\left(p_1^{'}/p_0\right)^{k-1/k}-1}{\eta_c}\right] = 298\times\left[1+\frac{8^{0.4/1.4}-1}{0.8}\right] = 600.3K$$

Next, we calculate the temperature of the oxygen and carbon dioxide mixture at the exit of the gas turbine cycle compressor. We assume that the specific heat ratio of the O_2-CO_2 mixture is that of carbon dioxide and that they are at the same temperature (the figure shows oxygen at 298 K incorrectly). This will be verified later. Hence,

$$k_{mix} = k_{CO_2} = 1.289$$

$$T_2 = T_1\left[1+\frac{\left(p_2/p_1\right)^{k-1/k}-1}{\eta_c}\right] = 303\times\left[1+\frac{28^{0.342/1.342}-1}{0.8}\right] = 723.7K$$

The combustion reaction can be written as:

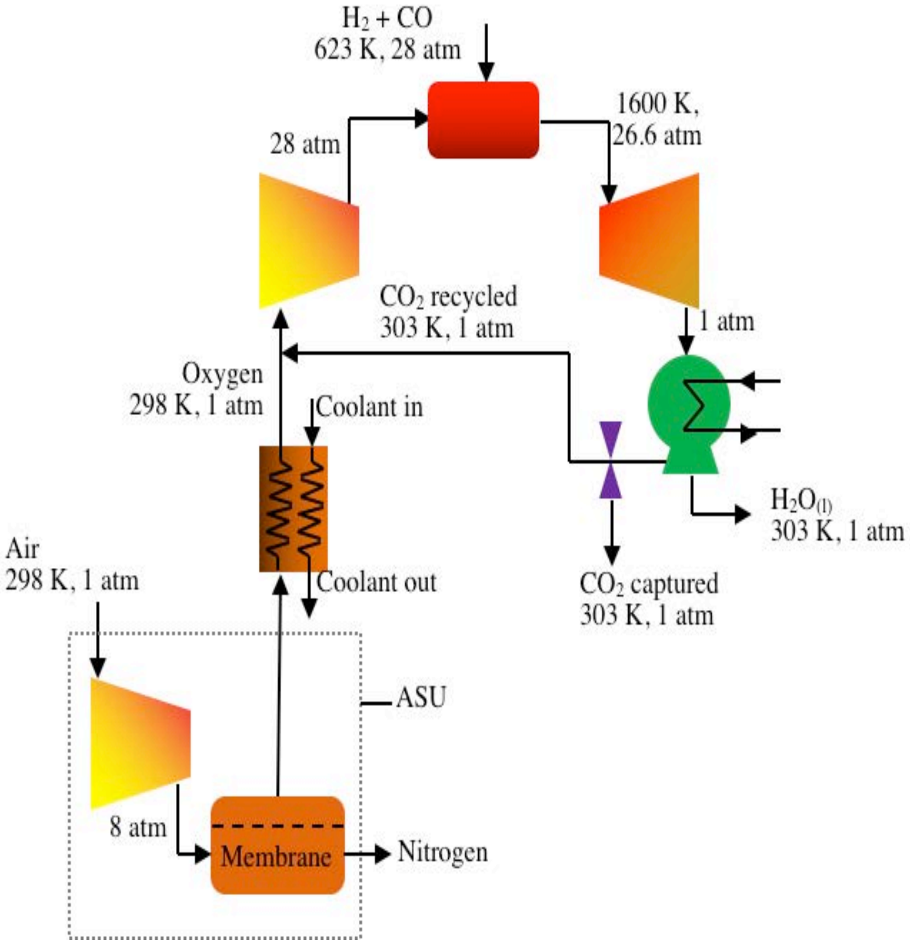
$$\underbrace{H_2+CO}_{syngas}+n_{O_2}O_2+n_{CO_{2,r}}CO_2 \rightarrow \left(n_{CO_{2,r}}+1\right)CO_2+H_2O$$

where $n_{CO_{2,r}}$ is the number of CO_2 moles recycled.

From oxygen balance, we find $n_{O_2}=1$. Applying energy conservation to the adiabatic combustor,

$$\hat{h}_{H_2}^{623K}+\hat{h}_{CO}^{623K}+\hat{h}_{O_2}^{723.7K}n_{CO_{2,r}}\hat{h}_{CO_2}^{723.7K} = \left(n_{CO_{2,r}}+1\right)\hat{h}_{CO_2}^{1600}+\hat{h}_{H_2O}^{1600}$$

The enthalpies of gases are calculated as follows.



$$\hat{h}_{H_2}^{623K} = \hat{h}_{f,H_2}^{0} + \hat{c}_{p,H_2}\left(623-298\right) = 0 + 28.6x325 = 9295\,kJ/kmol$$

Similarly: $\hat{h}_{CO}^{623K} = \hat{h}_{f,CO}^{0} + \hat{c}_{p,CO}\left(623-298\right) = -101072.5\,kJ/mol$

$\hat{h}_{O_2}^{723.7K} = 12517\,kJ/kmol$, $\hat{h}_{CO_2}^{809.7K} = -377963\,kJ/kmol$

$\hat{h}_{H_2O}^{1600} = 202419.2\,kj/kmol$, $\hat{h}_{CO_2}^{1600} = 345365.6\,kJ/kmol$

Substituting into the equation above and solving for $n_{CO_2,r}$, we find $n_{CO_2,r} = 14.4$

Next, we calculate the exit temperature of the turbine.

$$T_4 = T_3\left\{1-\eta_t\left[1-\left(p_4/p_3\right)^{k-1/k}\right]\right\} = 1600\left\{1-0.9\left[1-\left(1/26.6\right)^{\frac{0.289}{1.289}}\right]\right\} = 850.1K$$

The specific heat ratio of CO2 is used because over 90% of the mixture is carbon dioxide. work of the gas turbine is obtained as follows.

$$W_t = \left(n_{CO_2}\hat{c}_{p,CO_2} + n_{H_2O}\hat{c}_{p,O_2}\right)\left(1600-850.1\right) = \left(15.4x37.2+1x30.4\right)\left(1600-850.1\right)$$

$$= 451.668\,kJ$$

Similarly, the work of the O2-CO2 compressor is calculated

$$W_c = \left(n_{CO_{2,r}}\hat{c}_{p,CO_2} + n_{O_2}\hat{c}_{p,O_2}\right)\left(723.7-303\right) = \left(14.4x37.2+1x29.4\right)x420.7 = 237.333kJ$$

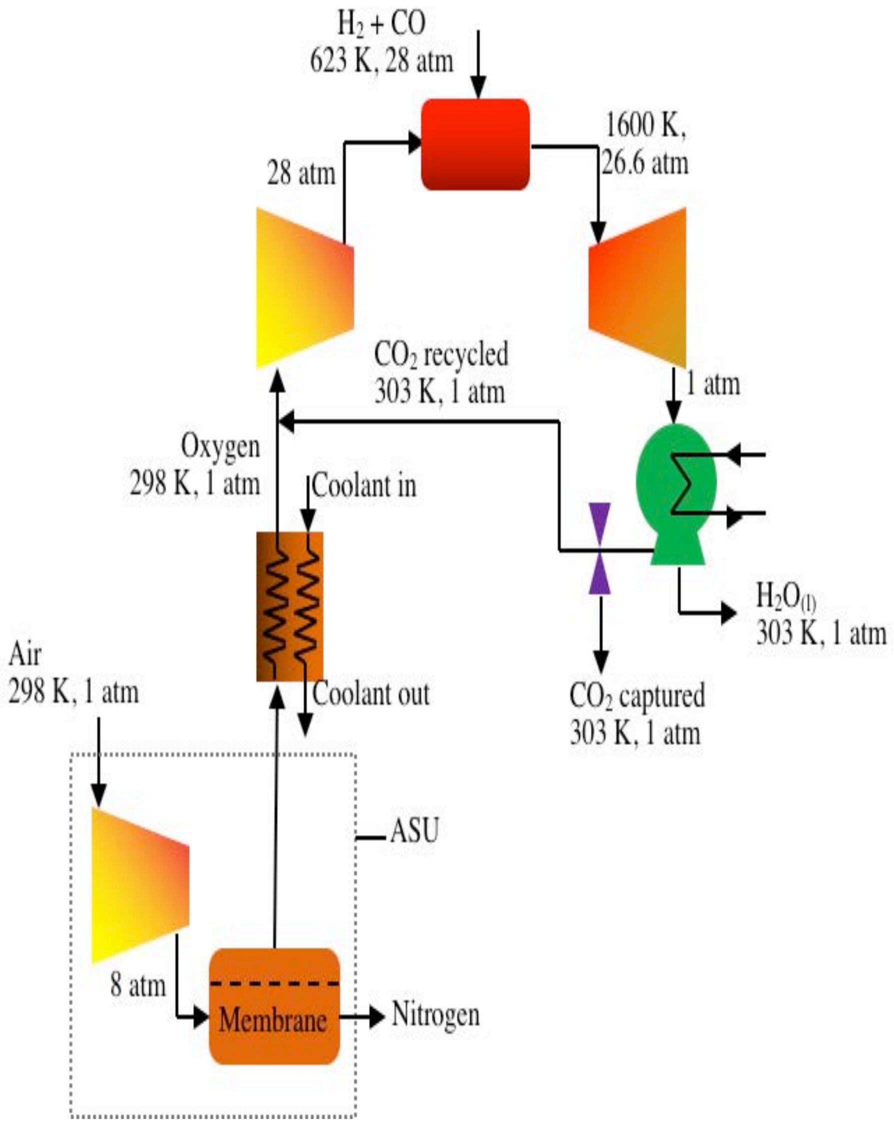
Moreover, the work requirement of the ASU compressor is

$$W_{c,ASU} = \left(29.4+3.76x29.1\right)\left(600.3-303\right) = 41.265\,kJ$$

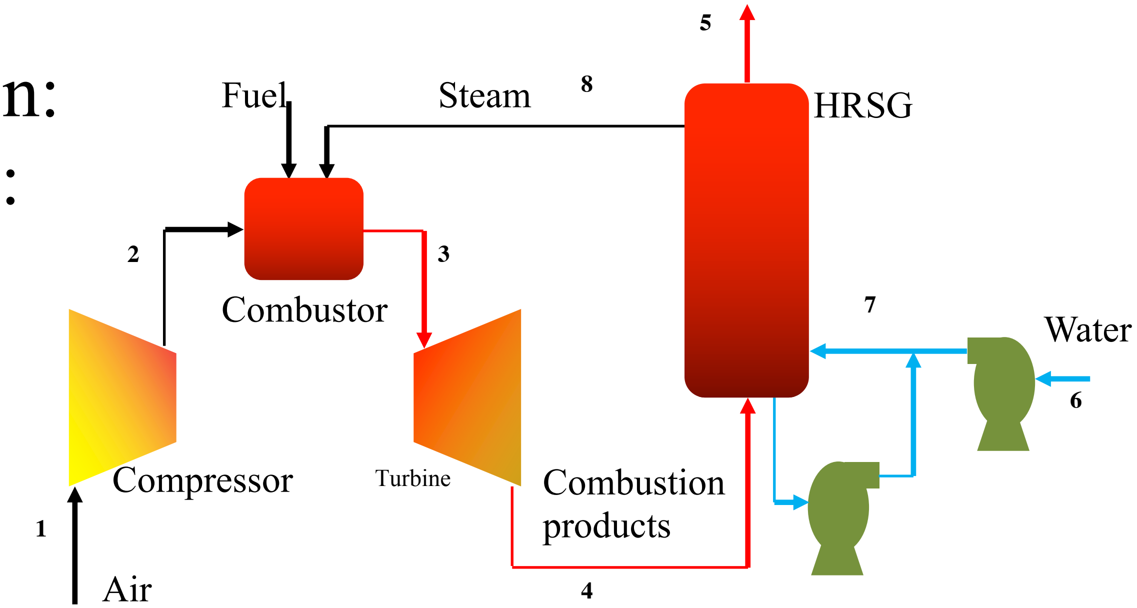
The net work produced by the power plant is therefore

$$W_{net} = W_t - W_c - W_{c,ASU} = 173.070\,kJ$$

The thermal efficiency of the power plant is $n_{th} = \dfrac{W_{net}}{LHV_{H_2}+LHV_{CO}} = \dfrac{173070}{242000+283270} = 0.33$



Exhaust Heat Recovery:
Humid Air Cycles, alternative to regeneration:
1. Steam injection and heat recovery cycle:

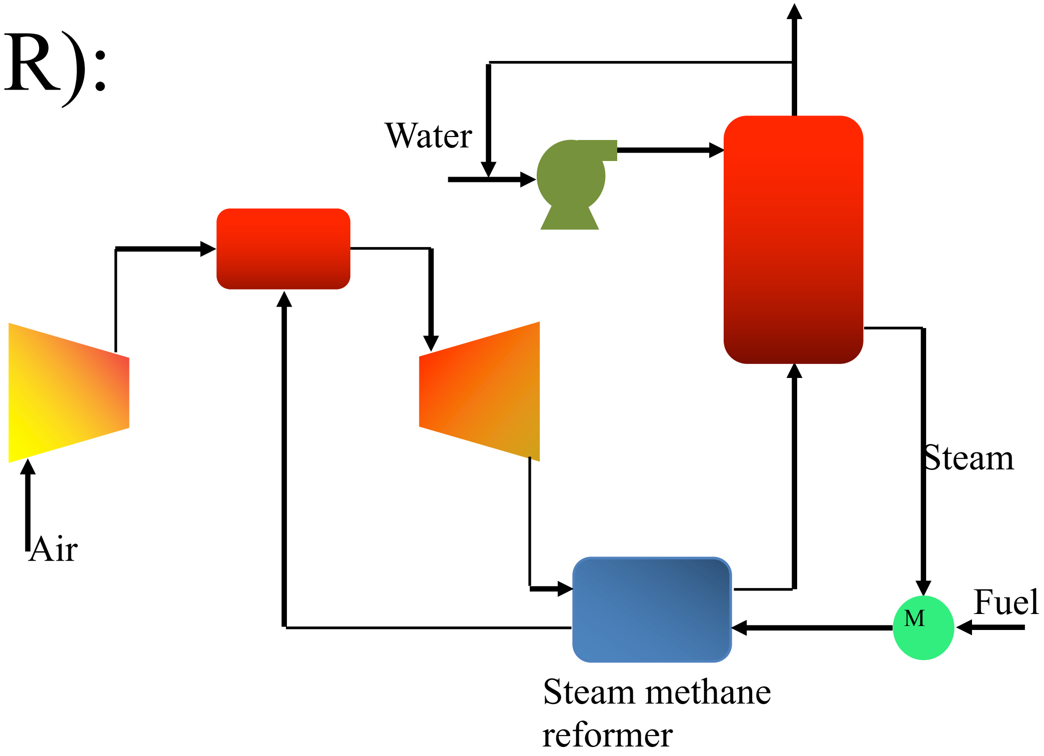


- Similar to regenerative cycles.
- Recovers some of the turbine exhaust energy.
- 20% of turbine mass flow is water.
- Limited by condensation pressure at turbine exit T.
- Needs purified water
- Has materials' issues.
- Can have NOx emissions' advantages.

Performance data	Simple cycle	CC	HAT
Gas turbine type		AD	
Pressure ratio		46	
TIT °C		1500	
Water consumption, kg/kWh		0.74	0.72
Efficiency, %			

Ad: aeroderivative
Water/air ~ 15% (water/(air+NG) ~ 13%)
In HAT water/products ~ 20%

2. Thermochemical Recuperation (TCR):



$$C_nH_m + n\ H_2O \rightarrow n\ CO + \left(n + \frac{m}{2}\right)H_2$$

for methane,

$$\Delta H_{reforming} = 226 \text{ kJ/mole of methane}$$

The HV of methane is ~ 800 MJ/kmol

thus reforming to syngas raises the HV by ~ 25%

further reforming is also possible:

$$CO + H_2O \rightarrow CO_2 + H_2$$

$$\Delta H_R = 41 \text{ kJ/mole}$$

3. Combined Cycle, next chapter

	TCR	SC	CC
Steam to NG ratio by mass		NA	
Air to NG ratio by mass		42.7	
Makeup water, kg/kWh		0	
Stack gas temperature, °C		590	
Net cycle power, MW		166	264
Cycle efficiency, %			

Water/air ~ 15% (water/(air+NG) ~ 13%)
SC: simple cycle,
CC Combined Cycle

Gas turbines have advantages in power generation:

- They operate at high temperatures.
- They can be started, turned down, and stopped relatively easily and within a short period of time, i.e. can load-follow and are capable of meeting peak load demands.
- They are compact and easy to operate, and they take advantage of ongoing developments in the aerospace, sea and some ground propulsion applications.
- They operate at relatively low pressures, compared to steam turbines, and this simplifies the plumbing of the plant.

Advantages of combustion turbines:

- Installations, for a wide range of loads, have been built and operated over the past couple of decades, mostly burning natural gas, or in dual fuel mode NG and oil.
- Gas turbines do not handle wet gases like steam turbines do, and are not as vulnerable to corrosion as steam turbines.
- Open cycle, or combustion gas turbines do not require heat transfer equipment on the low-temperature side, and no coolant either, and hence can be built and operated in hot dry areas.

limitations:

- They may have relatively low thermodynamic efficiency, the maximum temperature is limited by the blade material can handle, even with cooling.
- Their Second Law efficiency is low, because of the high compressor work; and the low efficiency of compressors.
- Open cycle turbines are limited by the relatively high exhaust pressure, which limits the work transfer of the turbine.
- They cannot be used with "dirty" fuels, e.g., coal, since sulfur oxides damage the blades.

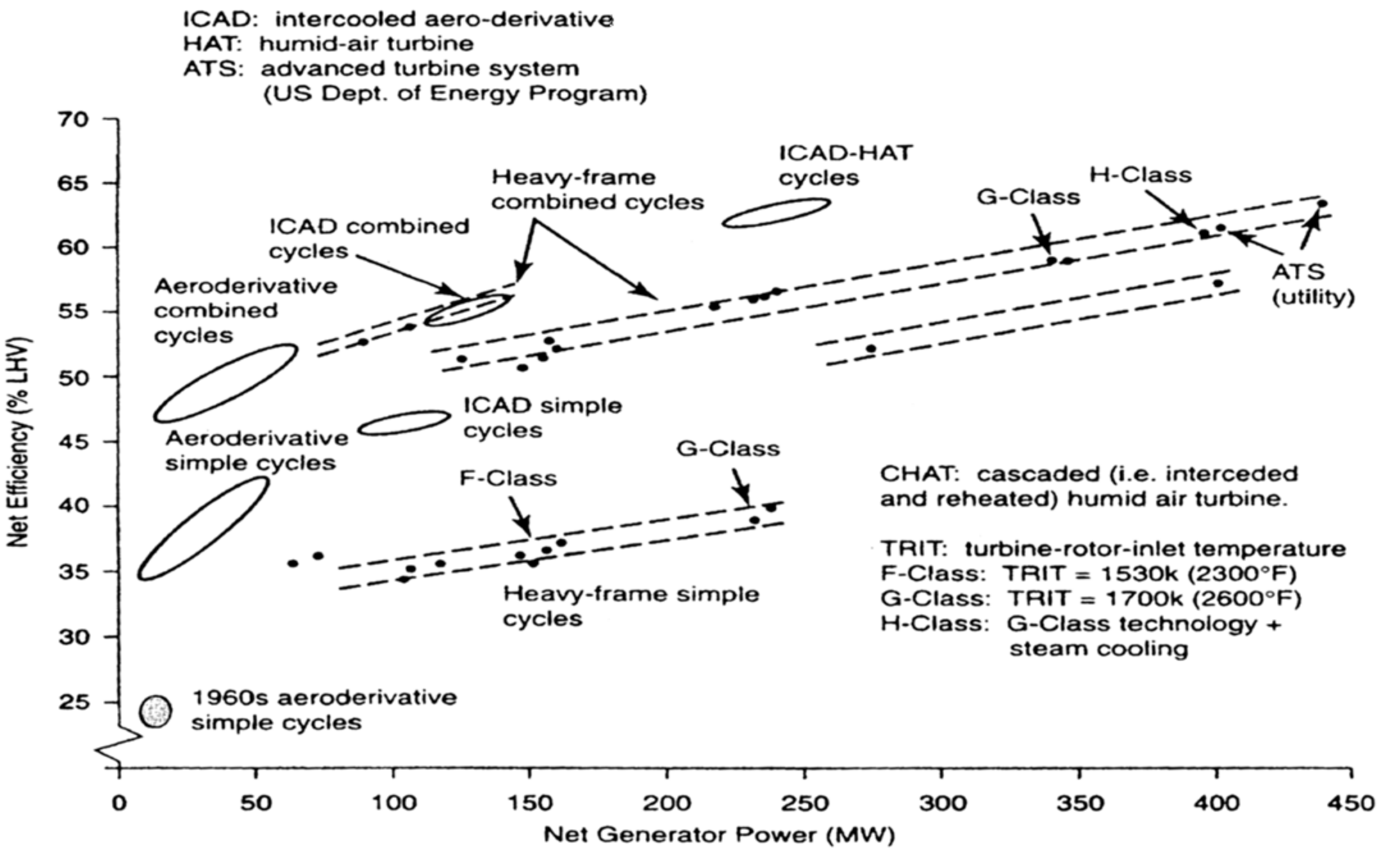
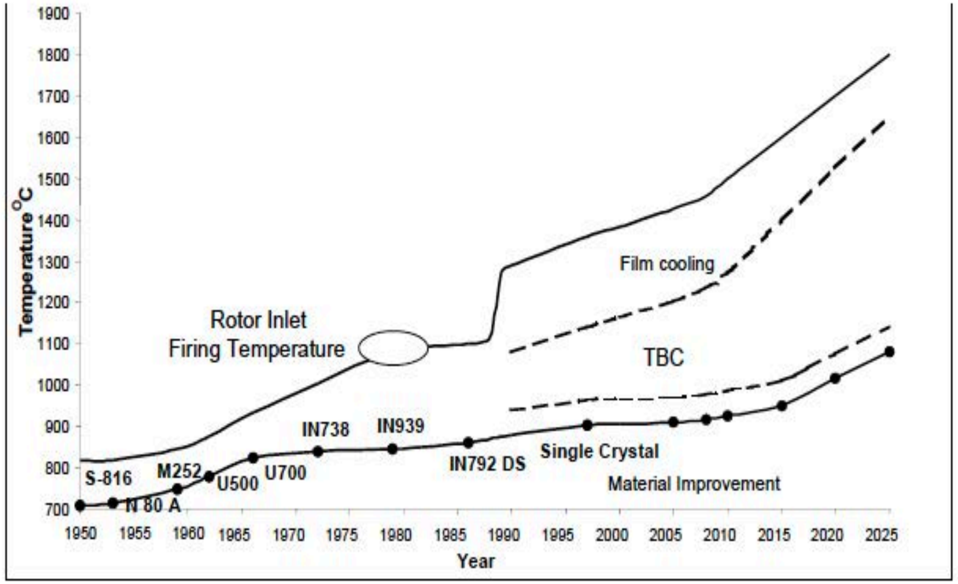
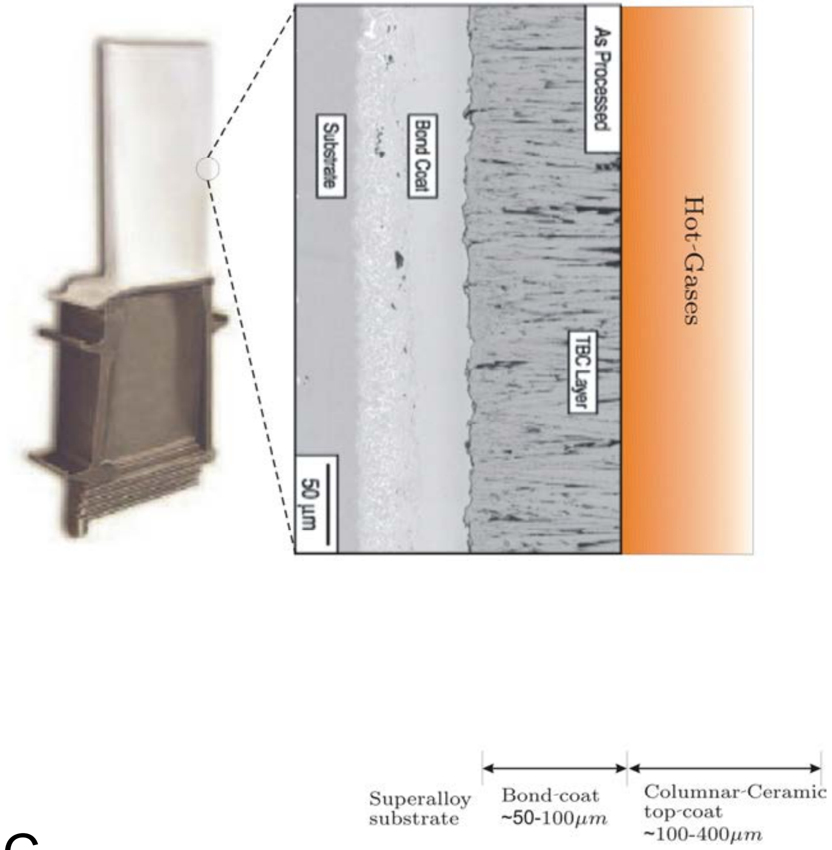


Figure 3. Thermal efficiency versus power of different turbines, and combined cycles (Wilson, and Korakianitis, 1998).



Impact of turbine blade metal, thermal barrier coating (TBC) and film cooling on the turbine inlet temperature (A. Rao, “Advanced Bryton Cycles,” 2002.

Current GE H-System NGCC Turbine Technology
(Natural Gas Combined-Cycle Plants, 400MW ~60% Efficiency)



- Turbine inlet temperatures --- ~ 1430C
- Single Crystal superalloy blades. Melting temperature ~ 1300C
- Active cooling so that blade temperatures do not exceed ~ 1050C (~0.8 T_m of blade material)
- Ceramic thermal barrier coatings (TBCs) to accommodate blade surface temperatures of ~1275C

Advanced turbines are manufactured using composite materials and “superalloys” of nickel (Ni) and cobalt (Co), mixed with molybdenum, tungsten, titanium, aluminum (Al) and chromium (Cr). The blades are hollowed for cooling.

A combination of high temperature and oxygen-rich gases make gas turbine blade vulnerable to corrosion. The blades are coated with chromium, or at higher temperature, with XCrAlY, where X stands for cobalt or nickel, and Y is yttrium, mixed in a dense aluminum oxide layer on the blade surface. This is part of the thermal barrier coating (TBC) applied to the blade surface, which is often a ceramic layer of zirconia (ZrO2) stabilized with yttria and a bonding of a metallic layer of XCrAlY. The ceramic layer has low thermal conductivity. Advanced manufacturing techniques, including physical vapor deposition or plasma vapor deposition are used in applying these coats.

Cooling techniques are also used. These include air and steam cooling using jet impingement, inner extended surfaces and cooling films on the surface.

The latest generation of gas turbines offered by different manufacturers, showing pressure ratio, the maximum temperature and simple cycle efficiency

	Westinghouse Fiat, MHI			ABB				
Performance data	TG50D5S6	FMW701F	MW501F	GT13E2	GT11N2	GT26	GT24	V84.3A
Power output, MW	143	237	153	164	109	254	173	170
Simple cycle efficiency, %	38.5	37.2	35.3	35.7	34.2	38.3	38.0	38.0
Exhaust gas flow rate, kg/s	454	666		525	375	562	390	454
Turbine inlet temp, °C	1250	1350		1100	1085			
Exhaust gas temp, °C	528	550		525	524	608	610	
Compressor pressure ratio	14.1	15.9	16	15		30		

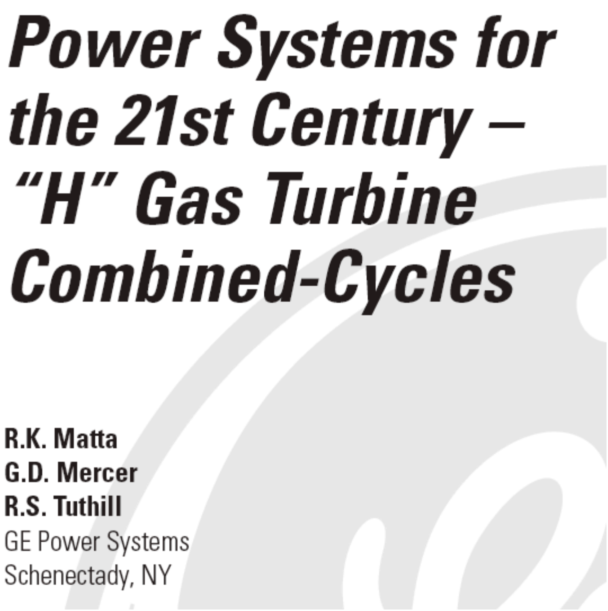
	General Electric Nuovo Pignone			Siemens Ansaldo	
Performance data	MS9001FA	MS7001FA	MS9001EC	V94.3A	V84.3A
Power output, MW	226.5	159	219	240	170
Simple cycle efficiency, %	35.7		34.9	38.0	38.0
Exhaust gas flow, kg/s	615		507	640	454
Turbine inlet temp, °C	1235		1290	1204-1340	
Exhaust gas temp, °C	589		558	562	
Compressor p. ratio	15		14.2	16	

Khartchenko, N.V., Advanced Energy Systems, Taylor & Francis, 1998, xix+218

Table 5.1. Operating data of aeroderivative based gas turbine power plants

Performance data	Value
Power output, MW	40-50
Turbine inlet temperature, °C	1280-1350
Compressor pressure ratio	30-60
Net specific work output, kJ/kg	350-370
Thermal efficiency, %	39.0-39.9
Air mass flow, kg/s	115-135
Gas turbine outlet temperature, °C	450-470

Khartchenko, N.V., Advanced Energy Systems, Taylor & Francis, 1998, xix+218



Power Systems for the 21st Century – "H" Gas Turbine Combined-Cycles

R.K. Matta
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GE Power Systems
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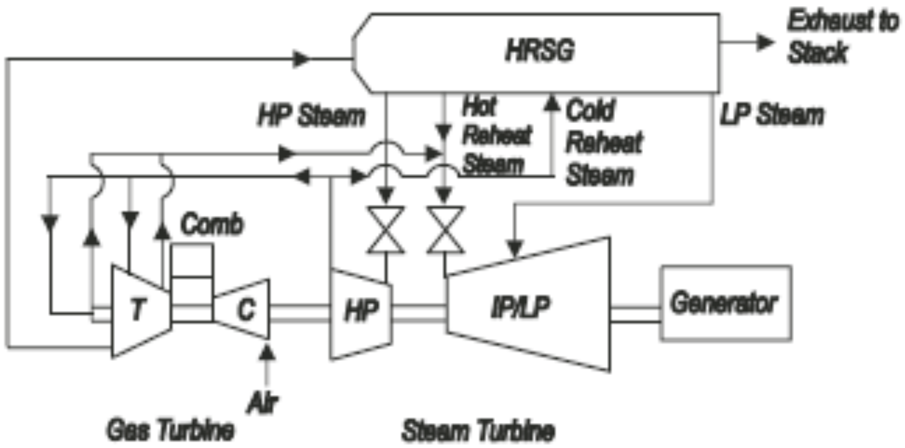
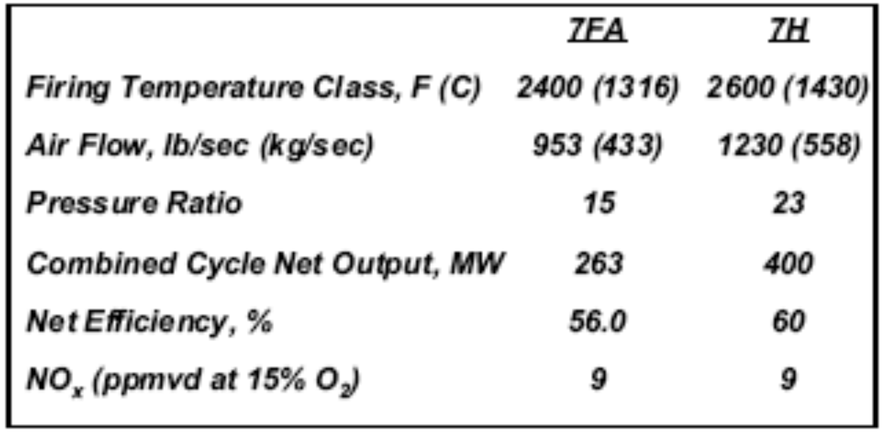


Figure 3. H Combined-cycle and steam description

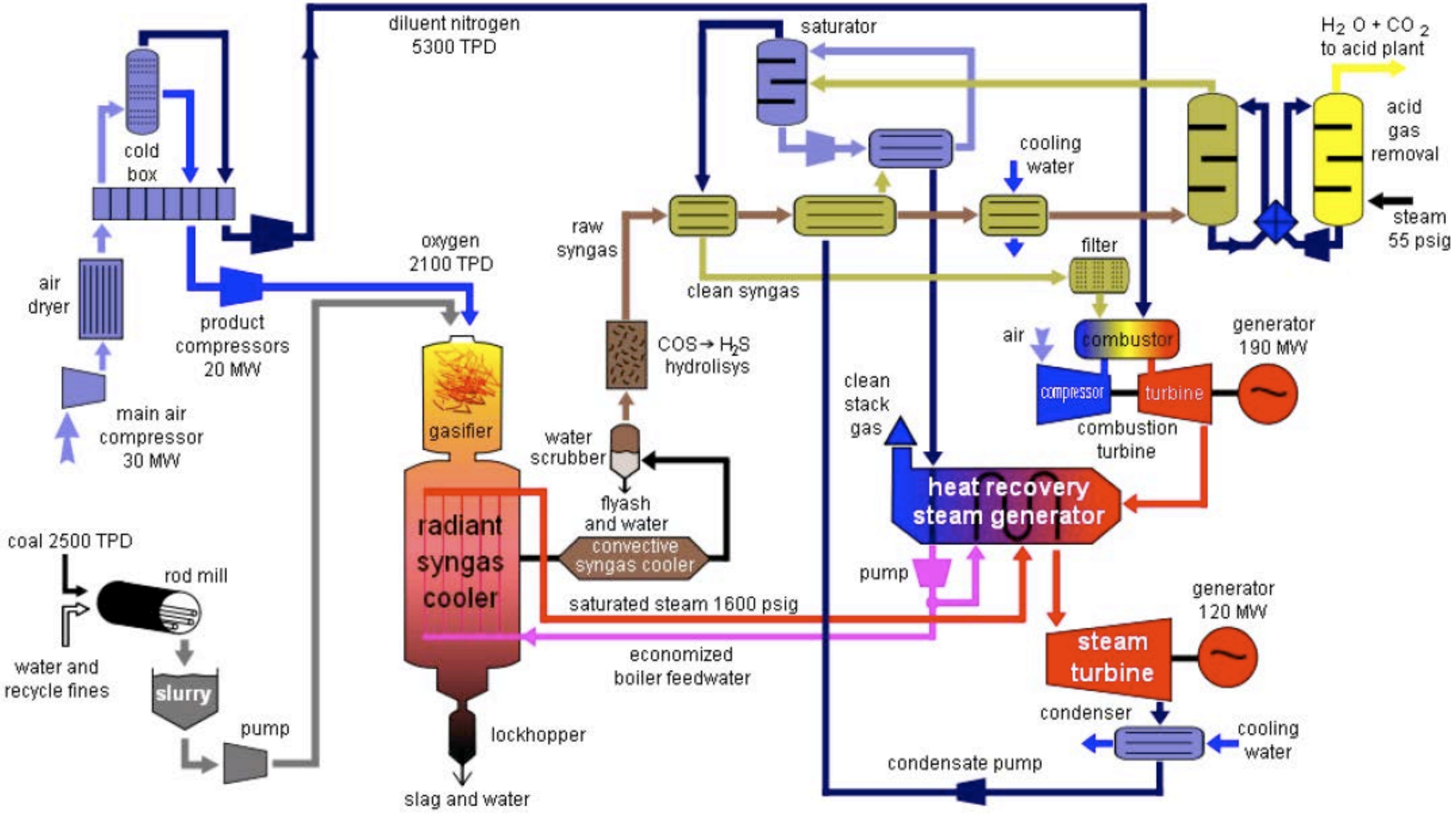
Some steam is injected into the GT to cool blade and increase power (HAT)



	7FA	7H
Firing Temperature Class, F (C)	2400 (1316)	2600 (1430)
Air Flow, lb/sec (kg/sec)	953 (433)	1230 (558)
Pressure Ratio	15	23
Combined Cycle Net Output, MW	263	400
Net Efficiency, %	56.0	60
NO_x (ppmvd at 15% O_2)	9	9

Table 2. H Technology performance characteristics (60 Hz)

Using Gas Turbines with Coal …. Gasification is necessary



2.60J Fundamentals of Advanced Energy Conversion
Spring 2020