INDUSTRIAL ENERGY MANAGEMENT

GAS TURBINE

Prof. Rodolfo Taccani

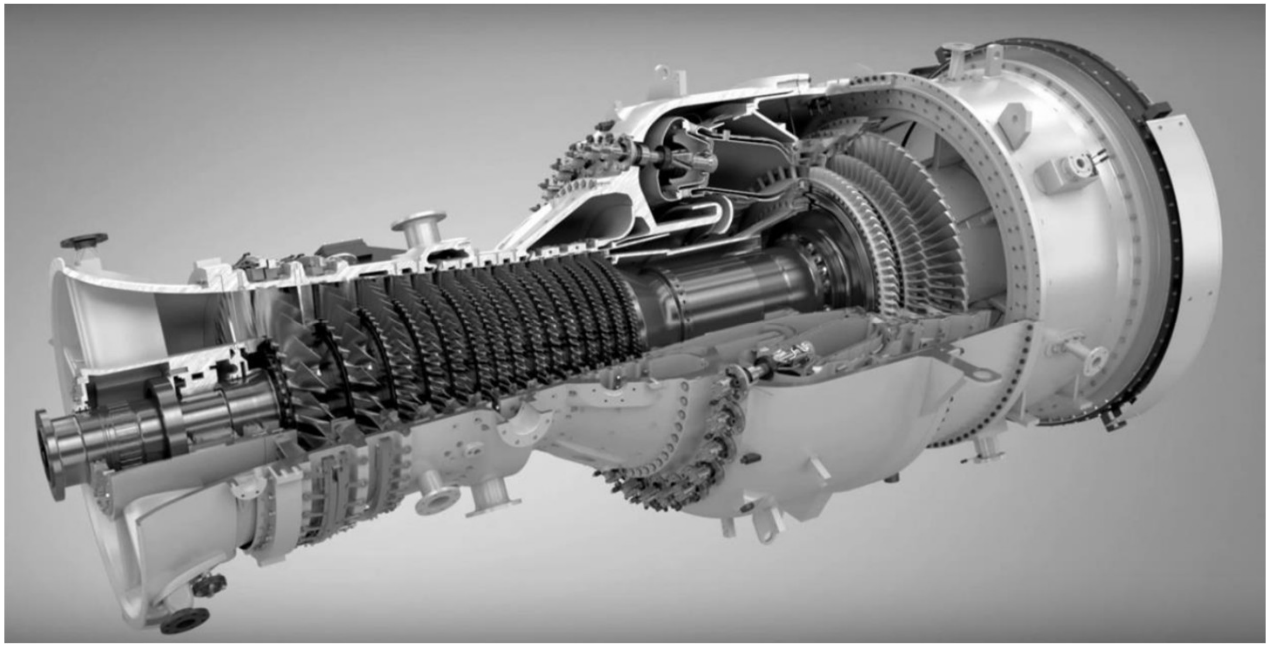
AY 2024 – 2025

Contents

Course materials

- Books:
  - Lozza G., Turbine a gas e cicli combinati 3rd edition, Società Editrice Esculapio, 2016.
  - Walsh P.P., Fletcher P., Gas turbine performance, Blackwell Science, 1998.
  - Kenneth C.W., Energy Conversion (E-book), 1992.
  - Moran M.J., Shapiro H.N., Boettner D.D., Bailey M.B., Fundamentals of Engineering Thermodynamics 8th edition, Wiley, 2014.
- Scientific paper and Others.

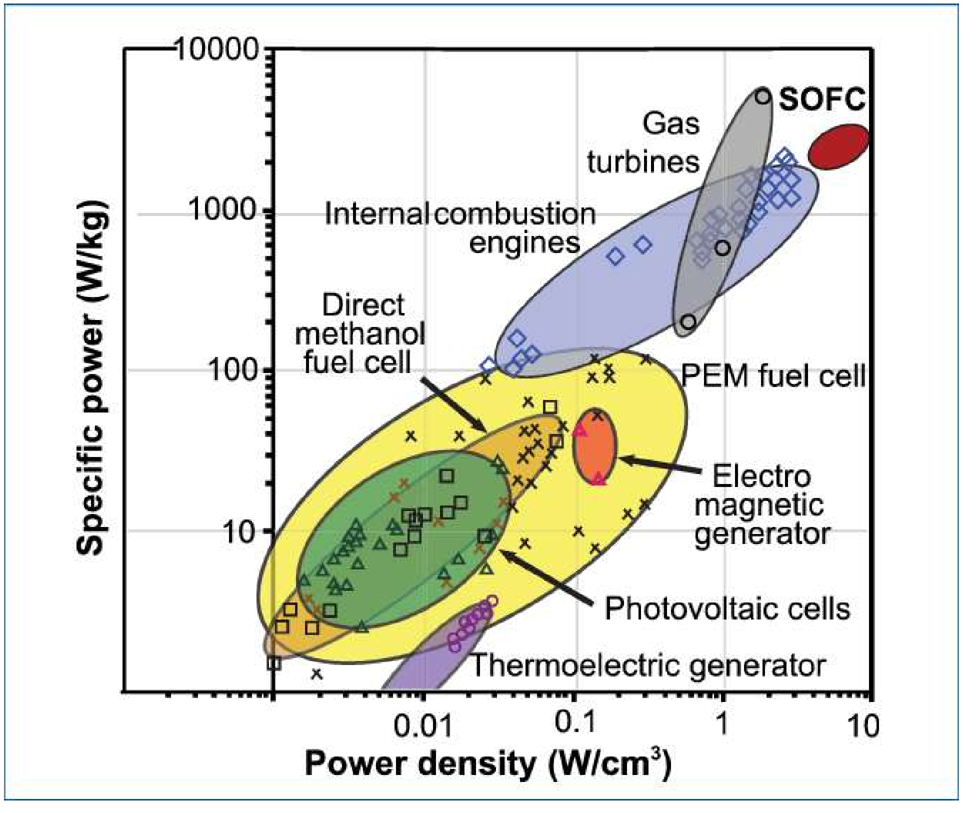
1. Introduction



Why do gas turbines have a higher specific power than Internal Combustion Engines (ICEs)?

A higher air flow rate can be processed since the flow is continuous:

- More air → more fuel
- More fuel → more heat released
- More heat released → more work (for the same efficiency)



Why do gas turbines have a higher specific power than ICEs?

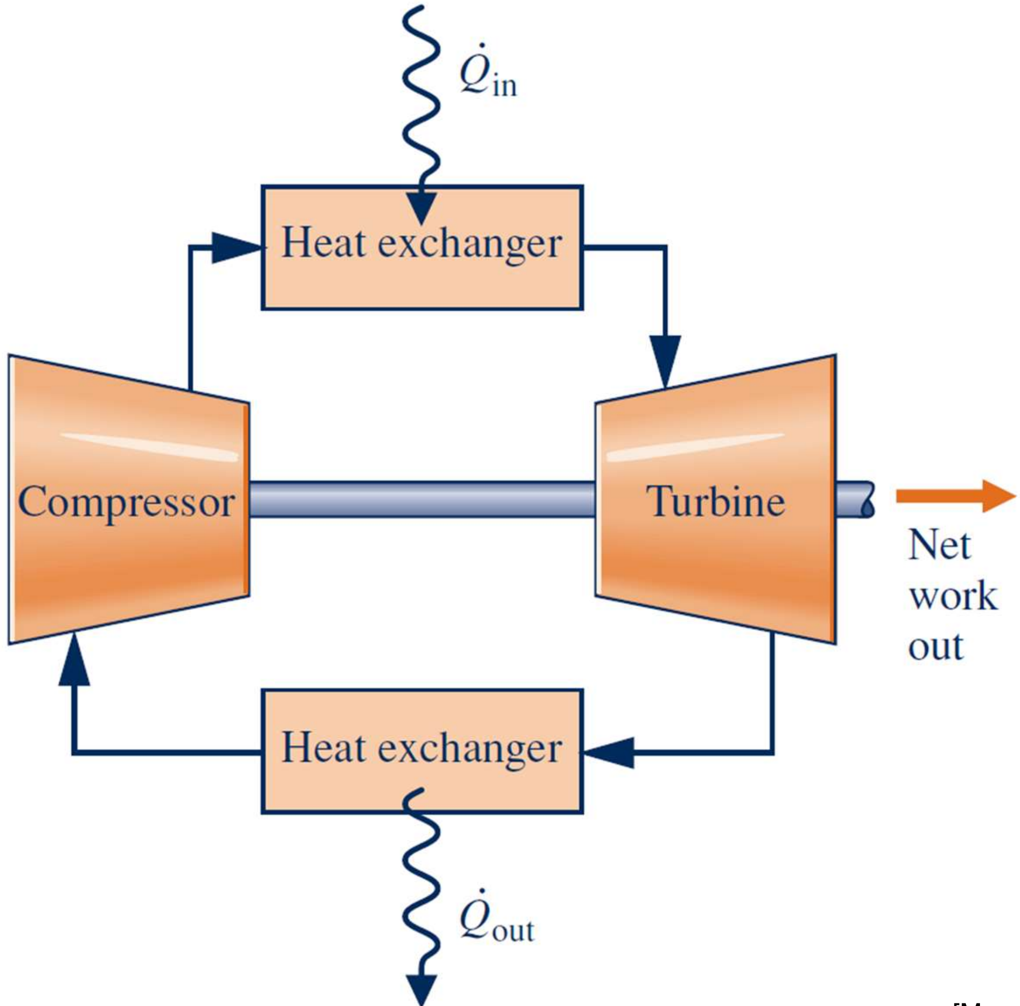
Disadvantages

- The compressor is a dynamic machine that moves air from a low-pressure zone to a high-pressure zone, without requiring a seal as in the piston/cylinder:
  - Very accurate fluid dynamic design is required
  - Blades need to rotate very fast to increase the energy transferred to the gas flow
  - Each stage can achieve compression ratios of up to 2:1 → many stages are required for high pressure differences
- Since the flow is continuous, every component downstream of the combustion chamber is subjected to high temperature. In addition, the expander rotates at high speed and, consequently, is subjected to high stress.
  - It is necessary to provide adequate (and expensive) cooling of the most stressed parts
  - The maximum temperature that can be reached by the turbine (approx. 1100 °C) places a limit on the fuel flow rate

The result is that:

→ Turbines are more expensive than ICEs at the same nominal power

Simple gas turbine closed cycle

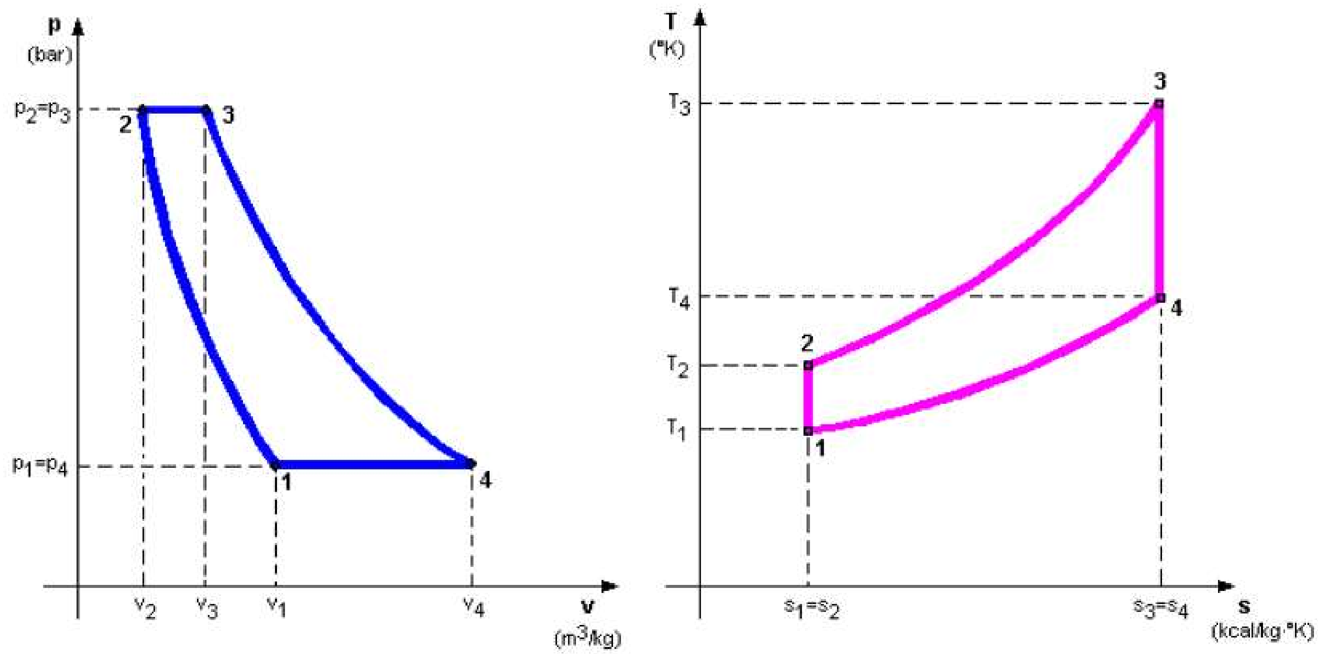


[Moran]

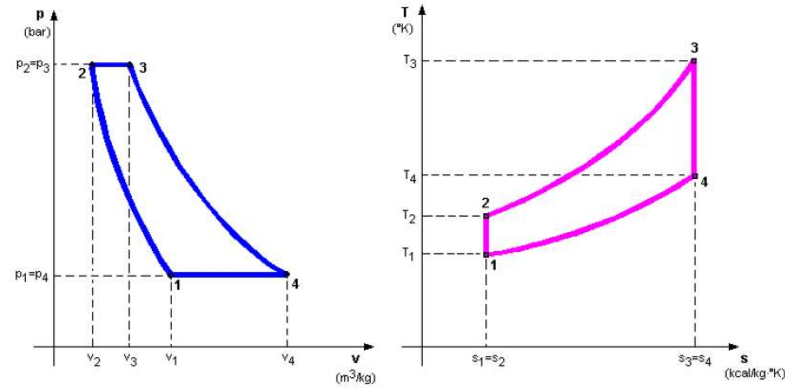
Ideal gas cycles

Gas turbine cycles generally perform **Brayton** (or **Joule**) cycle, consisting of the following transformations of a gas:

- 1-2 adiabatic compression;
- 2-3 isobaric heating;
- 3-4 adiabatic expansion;
- 4-5 isobaric cooling.



Efficiency



Energy contributions are defined as:

- Q_1 heat supplied between point 2 and point 3
- Q_2 heat supplied between point 4 and point 1
- L_C compression work between point 1 and point 2
- L_T expansion work in turbine between point 3 and point 4
- L_N useful work

Efficiency of the ideal cycle is calculated as follows:

$$\eta = \frac{L_N}{Q_1} = \frac{L_T - L_C}{Q_1} = \frac{(h_3 - h_4) - (h_2 - h_1)}{h_3 - h_2} = \frac{c_p(T_3 - T_4) - c_p(T_2 - T_1)}{c_p(T_3 - T_2)} = \frac{(T_3 - T_4) - (T_2 - T_1)}{(T_3 - T_2)}$$

$$\eta = \frac{T_3 \cdot \left(1 - \frac{T_4}{T_3}\right) - T_2 \cdot \left(1 - \frac{T_1}{T_2}\right)}{T_3 - T_2}$$

Efficiency

- In the adiabatic transformations (1-2) and (3-4) the following relationships apply:

$$\frac{T_4}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{k-1}{k}} \qquad \frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}}$$

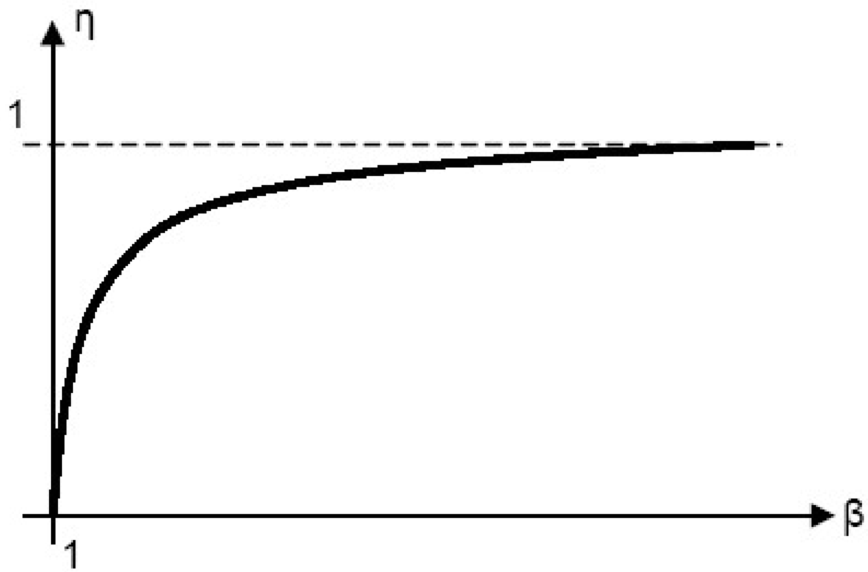
- k is the ratio between the specific heat at constant pressure (c_p) and the specific heat at constant volume (c_v). It is assumed that:

$$\beta = \frac{p_2}{p_1} = \frac{p_3}{p_4} \qquad \varphi = \frac{k-1}{k}$$

- It is obtained:

$$\eta = \frac{T_3 \cdot \left(1-\beta^{-\varphi}\right) - T_2 \cdot \left(1-\beta^{-\varphi}\right)}{T_3 - T_2} = 1 - \beta^{-\varphi}$$

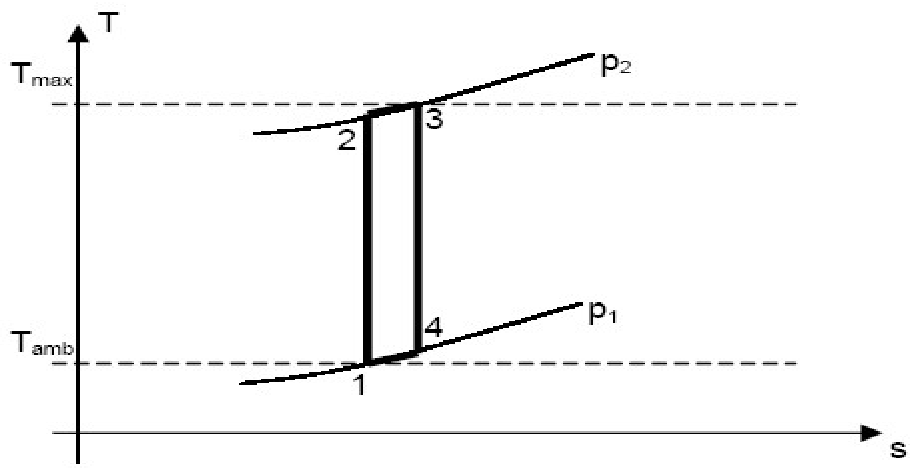
Efficiency



Efficiency increase with the increase of compression ratio

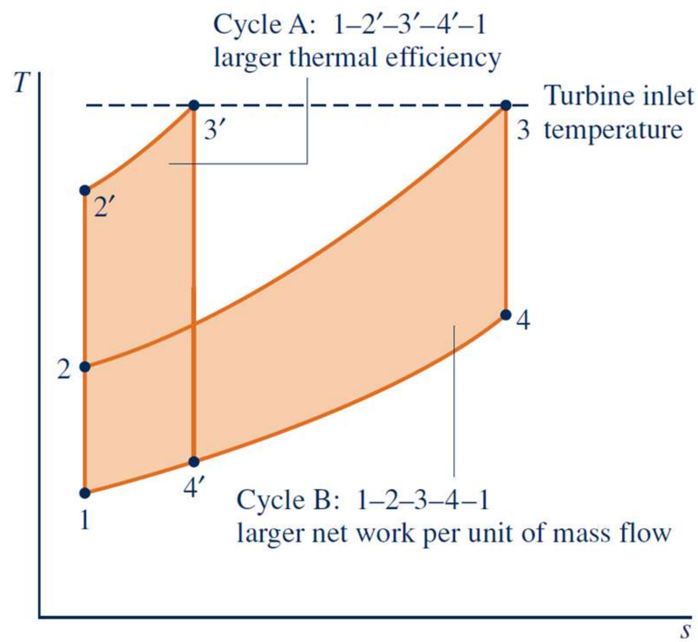
Minimum and maximum temperatures of the cycle are fixed. The minimum temperature is generally the ambient temperature, while the maximum temperature depends on the strength characteristics of the materials used. Therefore, the compression ratio cannot increase beyond a certain limit to not exceed the maximum allowed temperature.

Maximum specific work



- For a fixed T_3, a limiting compression ratio can be identified to which the maximum efficiency value corresponds.
- Gas systems have fixed costs that largely depend to the amount of air processed => therefore it is important to operate under conditions that have the maximum specific work

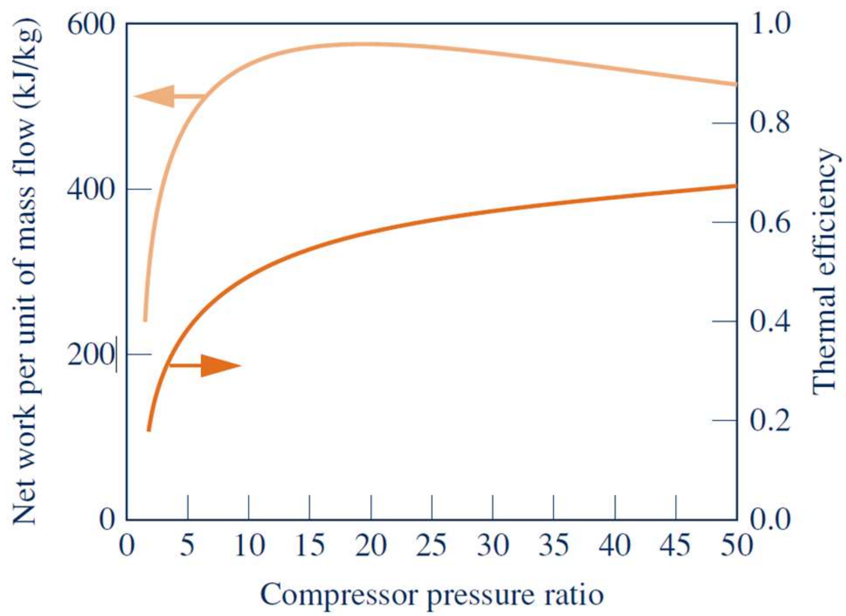
Maximum specific work



[Moran]

Cycle A has a higher compression ratio than cycle B and thus a higher thermal efficiency. However, cycle B has a larger closed area and therefore more net work developed per unit mass flow. Consequently, for cycle A to develop the same net power as cycle B, a higher mass flow rate would be required, which may require a larger system.

Mechanical and thermal efficiency



[Moran]

For mobility applications, it is desirable to operate close to the compression ratio that produces the maximum work per unit mass flow and not the compression ratio for maximum thermal efficiency. While thermal efficiency increases with compression ratio, the net work per unit of mass curve has a maximum value at a compression ratio of about 21. Also observe that the curve is relatively flat in the vicinity of the maximum. Therefore, for vehicle design purposes, a wide range of compression ratio values can be considered near-optimal from the point of view of maximum work per unit mass.

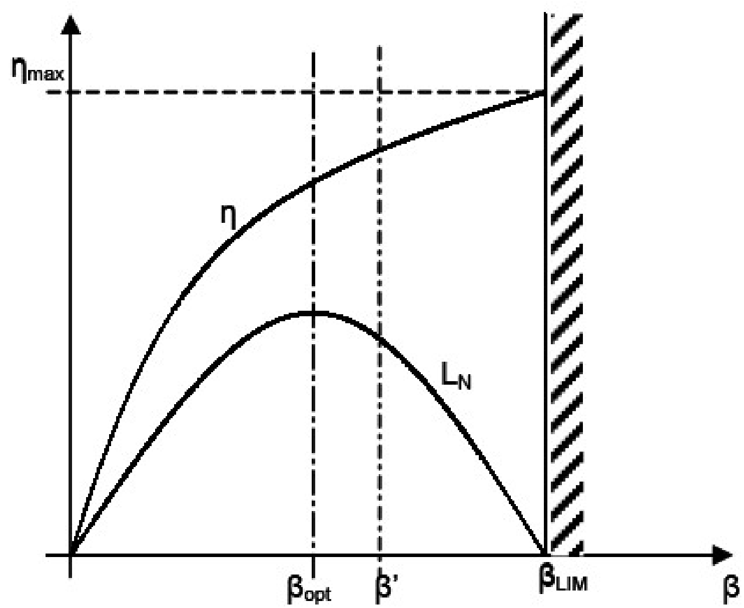
Maximum specific work

- The net work is defined as:

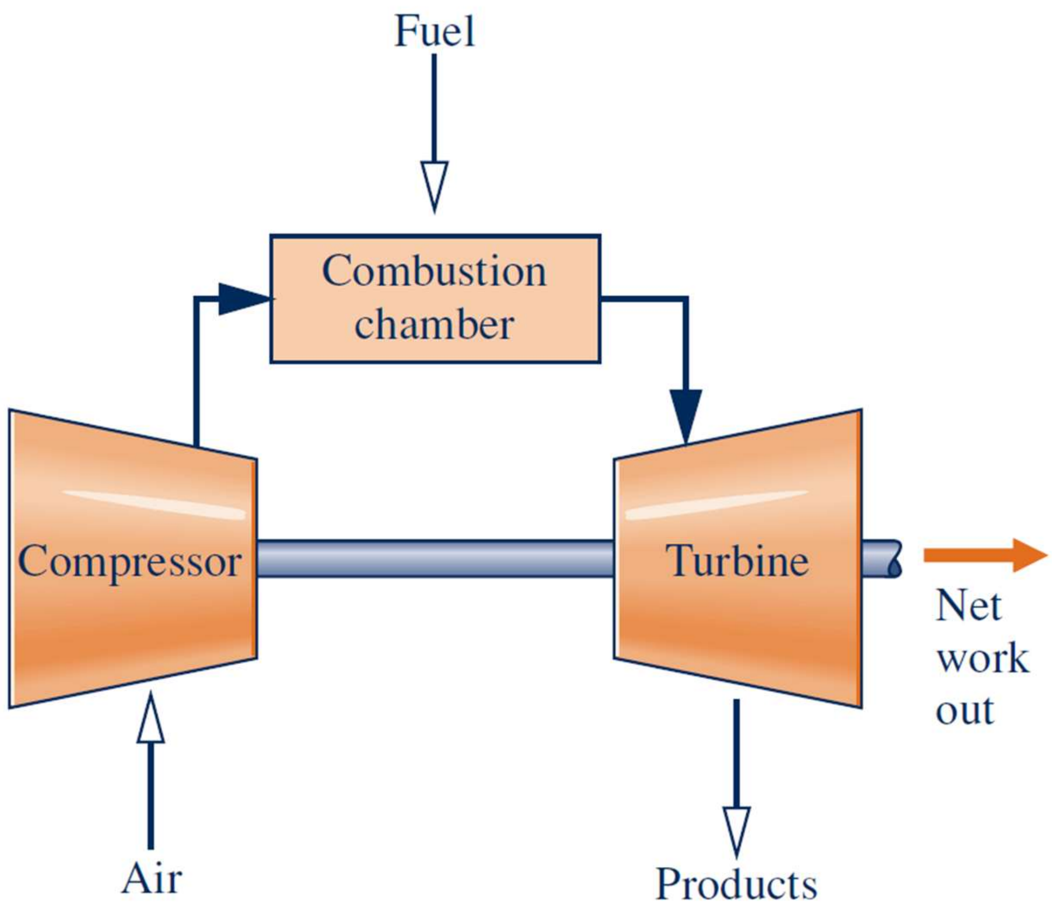
$$L_N = \eta \cdot Q_1 = \left(1-\beta^{-\varphi}\right)\cdot c_p \cdot (T_3 - T_2) = \left(1-\beta^{-\varphi}\right)\cdot c_p \cdot T_1 \cdot \left(\frac{T_3}{T_1} - \frac{T_2}{T_1}\right) = \left(1-\beta^{-\varphi}\right)\cdot c_p \cdot T_1 \cdot \left(\frac{T_3}{T_1} - \beta^{\varphi}\right)$$

- Therefore, the work generated is maximum for:

$$T_2 = T_4 = \sqrt{T_3 \cdot T_1}$$

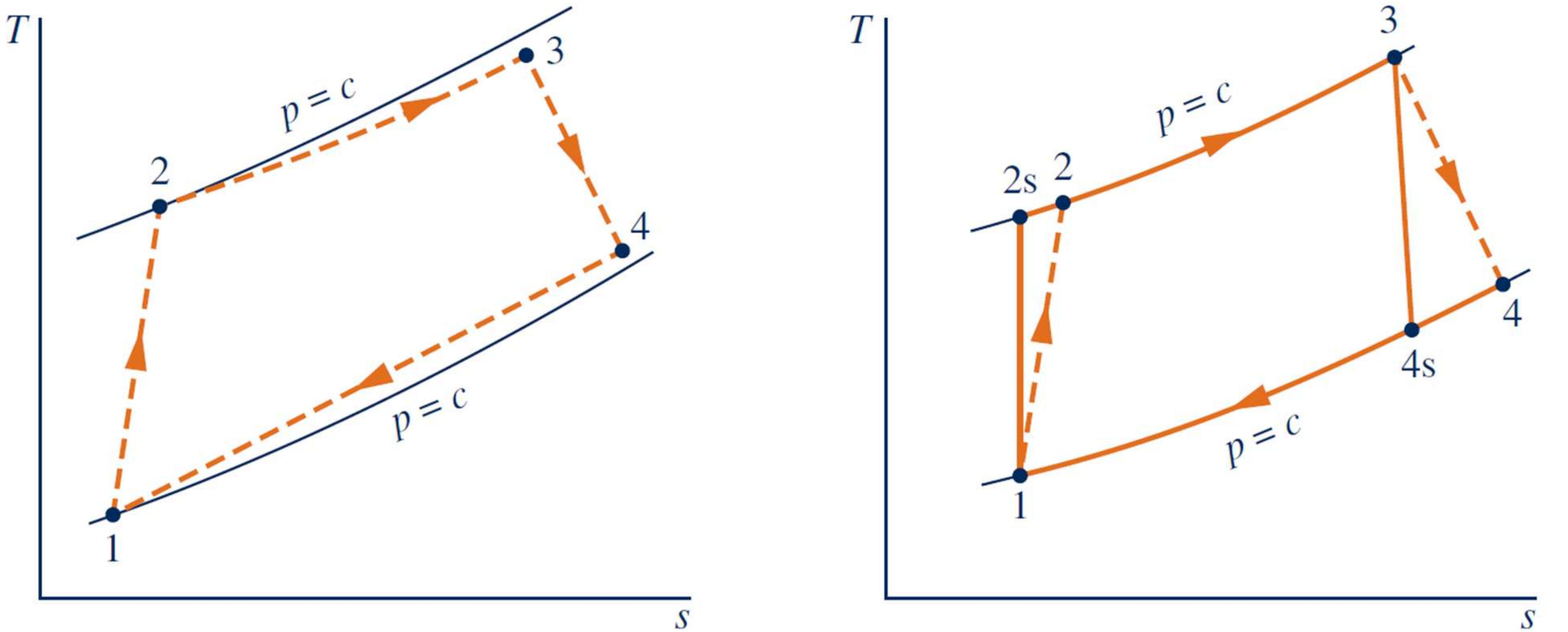


Real gas turbine open cycle



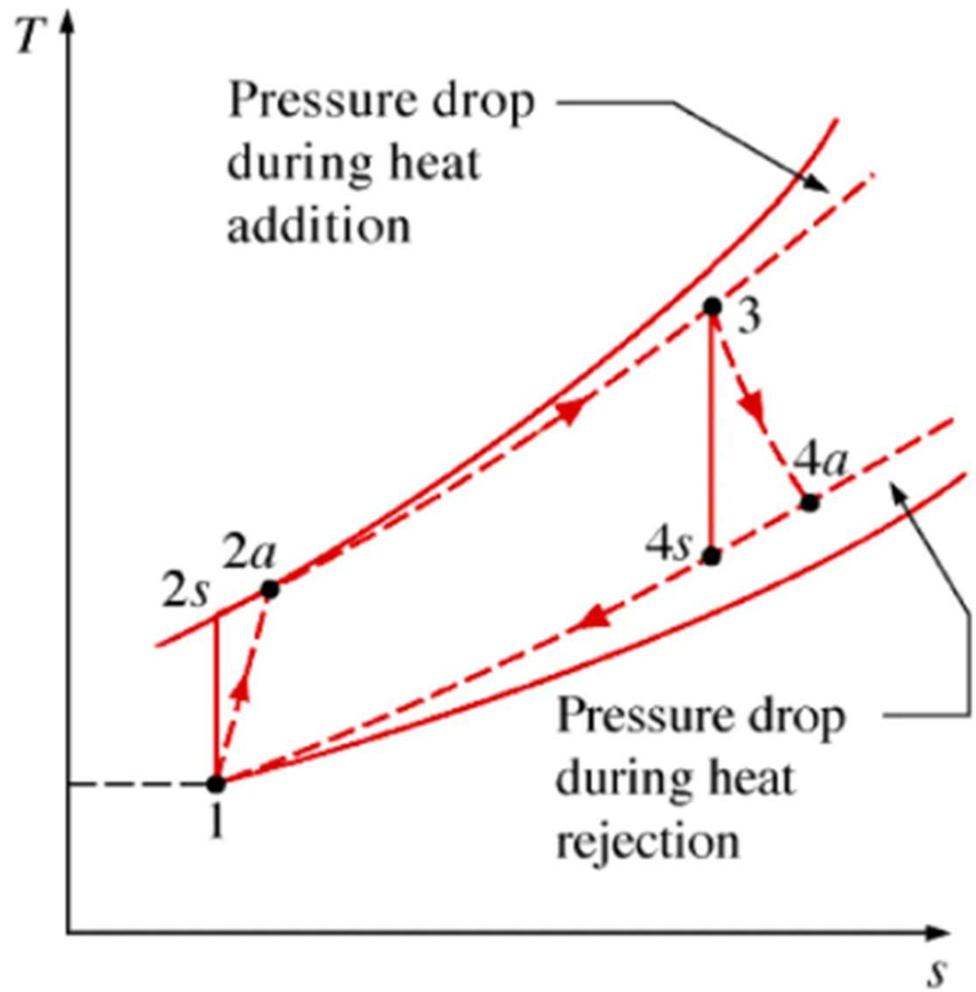
[Moran]

Effect of irreversibility on the air-standard gas turbine



[Moran]

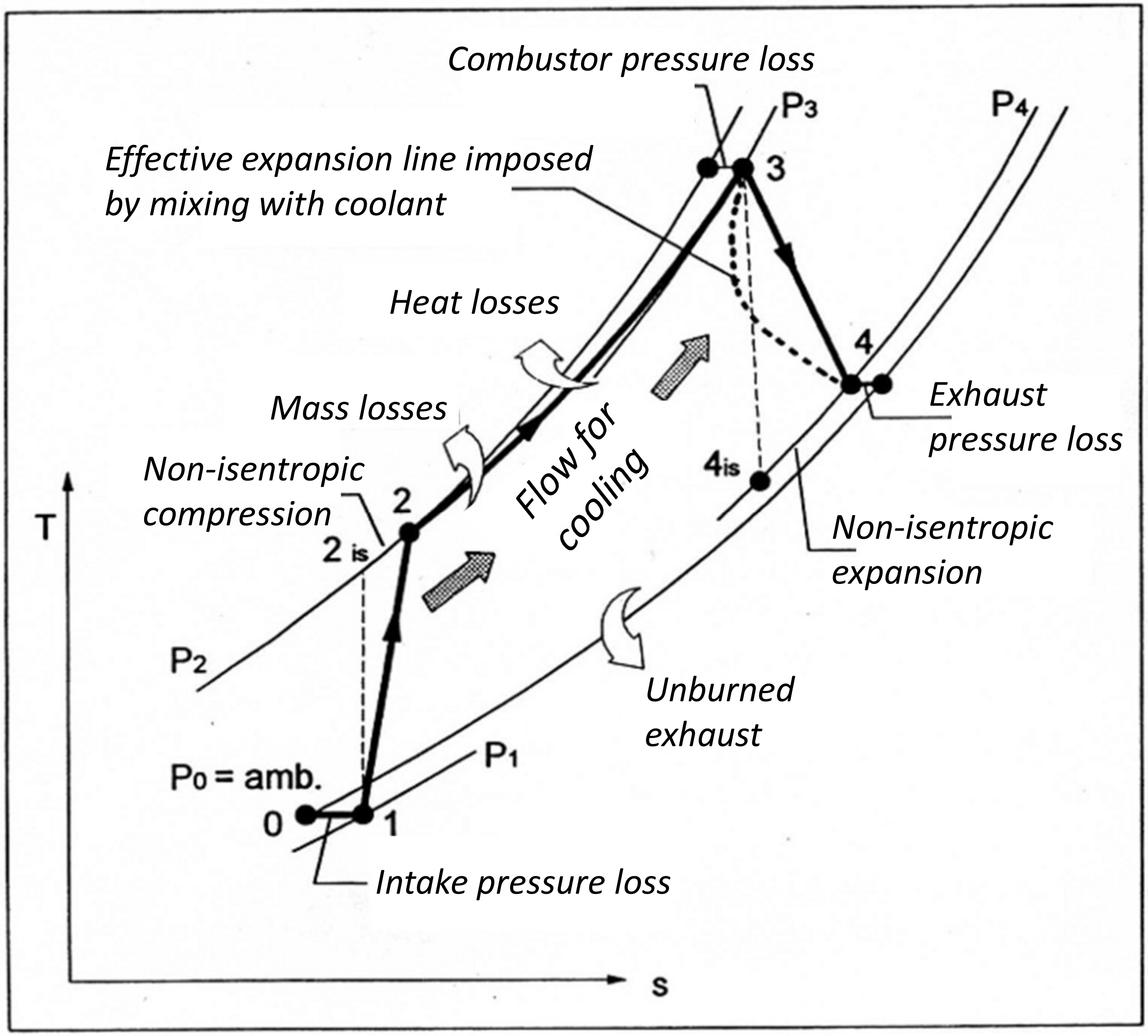
Comparison of ideal and real cycles



Dissipative processes in real cycles

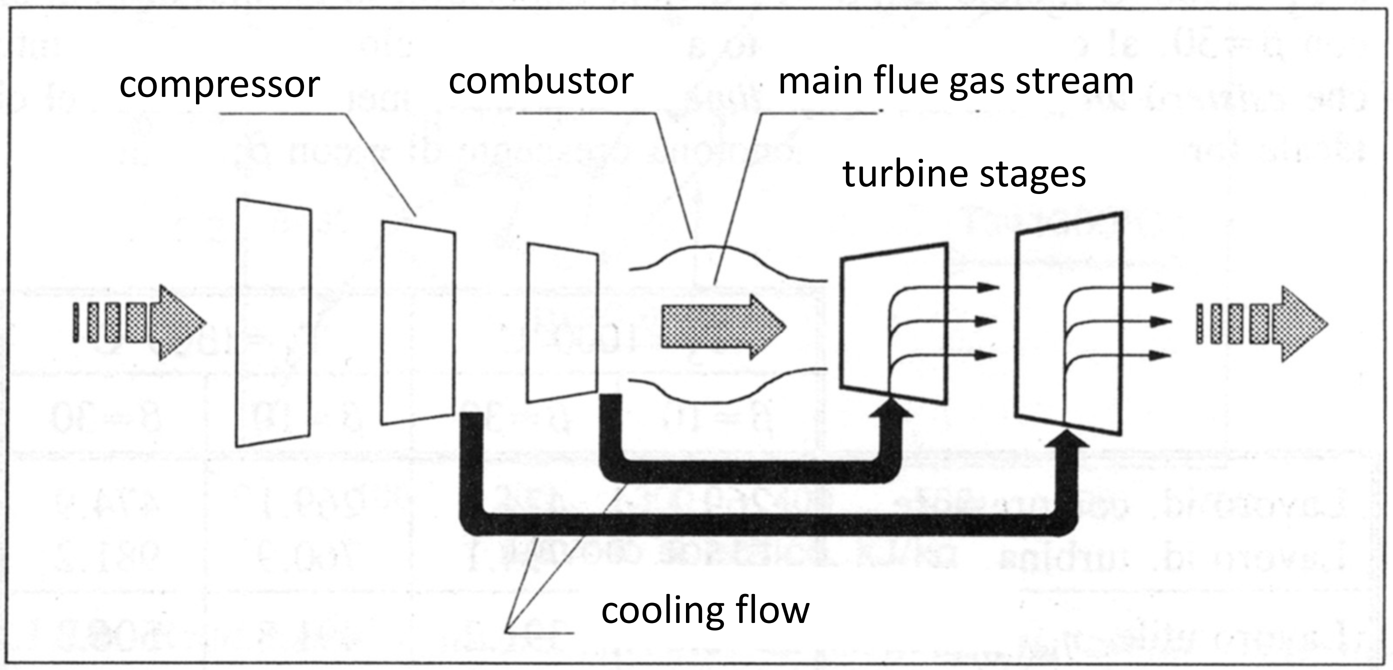
- Compression and expansion are not isentropic
- Transformations 2-3 and 4-1 are not isobaric, but there are numerous pressure losses:
  - Suction (filter and air ducts)
  - In the combustor and in the turbine adduction ducts
  - Downstream of the turbine (chimney, silencers, heat recovery unit components)
- Localized thermal losses in the various hot parts of the machine
- Chemical energy losses due to incomplete oxidation
- Mass losses (e.g. seals)
- Irreversible processes related to blade cooling
- Mechanical losses (due to ventilation of rotating parts, bearing friction, lubrication, auxiliaries, etc.)
- Losses due to conversion of mechanical energy into electric energy

Real open-cycle gas turbine



Blade cooling

Open-cycle cooling system for the blades. The air at the compressor intake is used as a coolant for the blades and is subsequently discharged into the main flue gas stream.

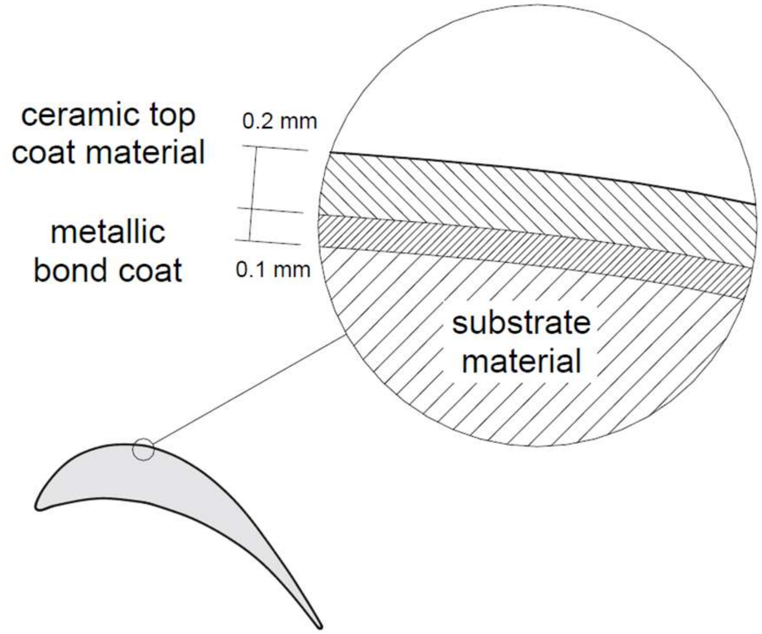


[Lozza]

Blade cooling

Current trends are:

- Improved materials (turbine blades)
- Use of high-temperature alloys: Inconel (rotors) and Cobalt alloys (stators)
- Development of reliable surface treatments and coatings ceramic
- Introduction of basic ceramic materials
- Improvement of cooling systems
- Development of new cooling techniques and their optimization
- Widespread extension of gas turbine monitoring techniques
- Improvement of reliability with forecasting and optimisation of maintenance operations, also according to the type of service



Influence of T and RC on performance

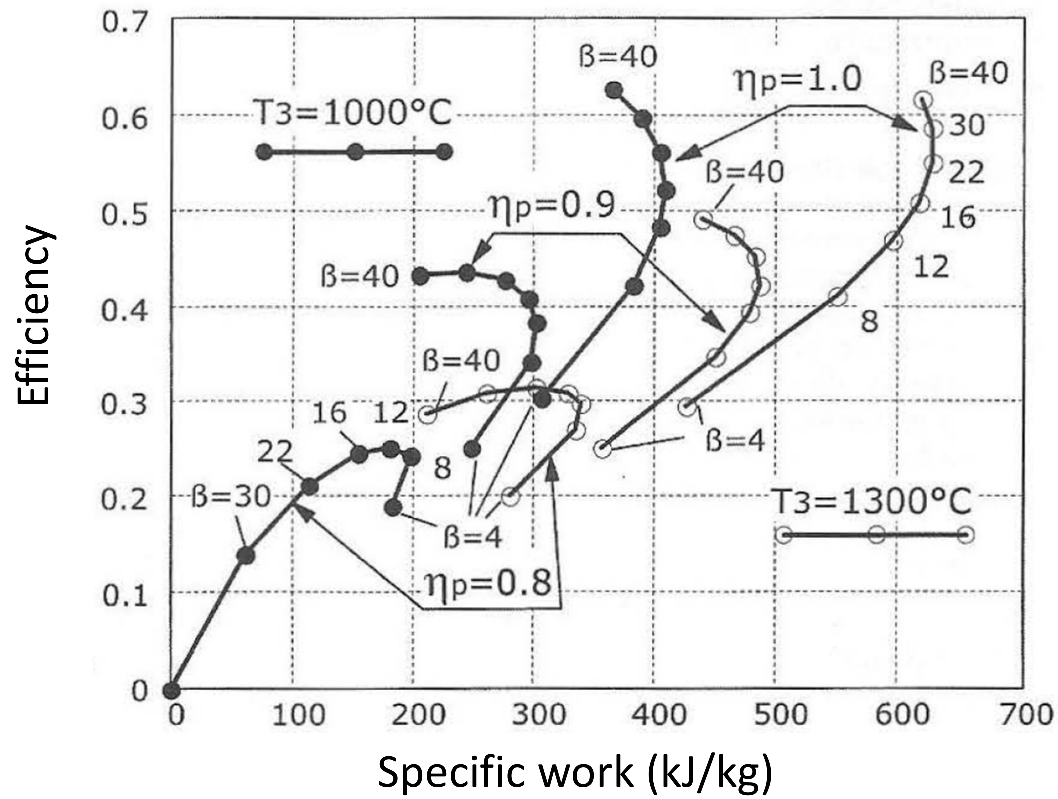
Influence of the adiabatic efficiency of the compressor and turbine on the work and efficiency of a closed cycle operating with a perfect diatomic gas (MM=29).

	Unit	T_3=1000°C		T_3=1300°C	
		B=10	B=30	B=10	B=30
Ideal work compressor	kJ/kg	269.1	474.9	269.1	474.9
Ideal work turbine	kJ/kg	615.8	794.1	760.9	981.2
Net work $\eta_{is_{(t,c)}}$=1	kJ/kg	346.7	391.2	491.8	506.3
Net work $\eta_{is_{(t,c)}}$=0.85	kJ/kg	206.9	116.2	330.2	275.3
Efficiency $\eta_{is_{(t,c)}}$=1	%	48.21	62.16	48.21	62.16
Efficiency $\eta_{is_{(t,c)}}$=0.85	%	28.76	22.63	32.36	33.80

[Lozza]

Efficiency and specific work

Note: T_3 strongly influences real cycle efficiency.



[Lozza]

Efficiency and specific work of open cycles, having as the only deviation from the ideal cycle the polytropic efficiency of the turbine and the compressor (set at 0.8, 0.9, 1). Different compression ratios (from 4 to 40) have been considered for two maximum temperature values T_3.

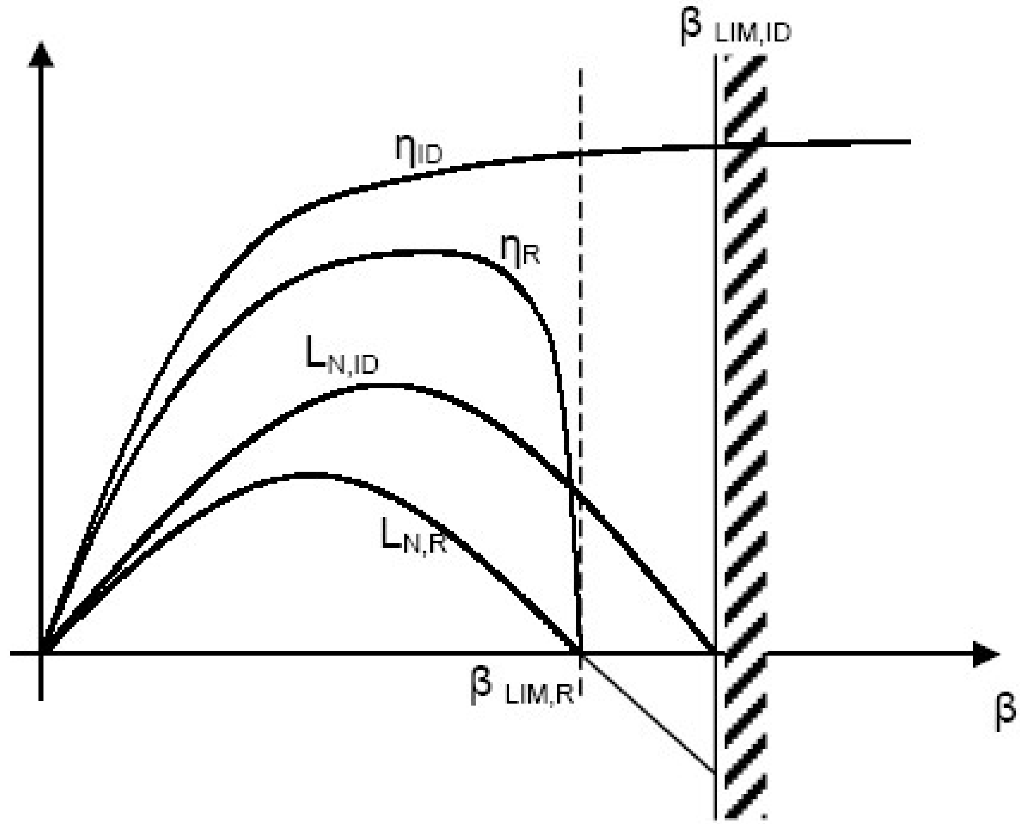
Specific work of a real cycle

$$L_{C,R} = h_2 - h_1 = \frac{h_{2,IS} - h_1}{\eta_{C,IS}} = c_p \cdot (T_2 - T_1) = c_p \cdot \frac{T_{2,IS} - T_1}{\eta_{C,IS}} = \frac{c_p \cdot T_1 \cdot (\beta^{\varphi} - 1)}{\eta_{C,IS}}$$

$$L_{T,R} = h_3 - h_4 = (h_3 - h_{4,IS}) \cdot \eta_{T,IS} = c_p \cdot (T_3 - T_4) = c_p \cdot (T_3 - T_{4,IS}) \cdot \eta_{T,IS} = c_p \cdot T_3 \cdot (1 - \beta^{\varphi}) \cdot \eta_{T,IS}$$

$$\eta_R = \frac{L_{T,R} - L_{C,R}}{Q_{1,R}}$$

Losses on real cycles

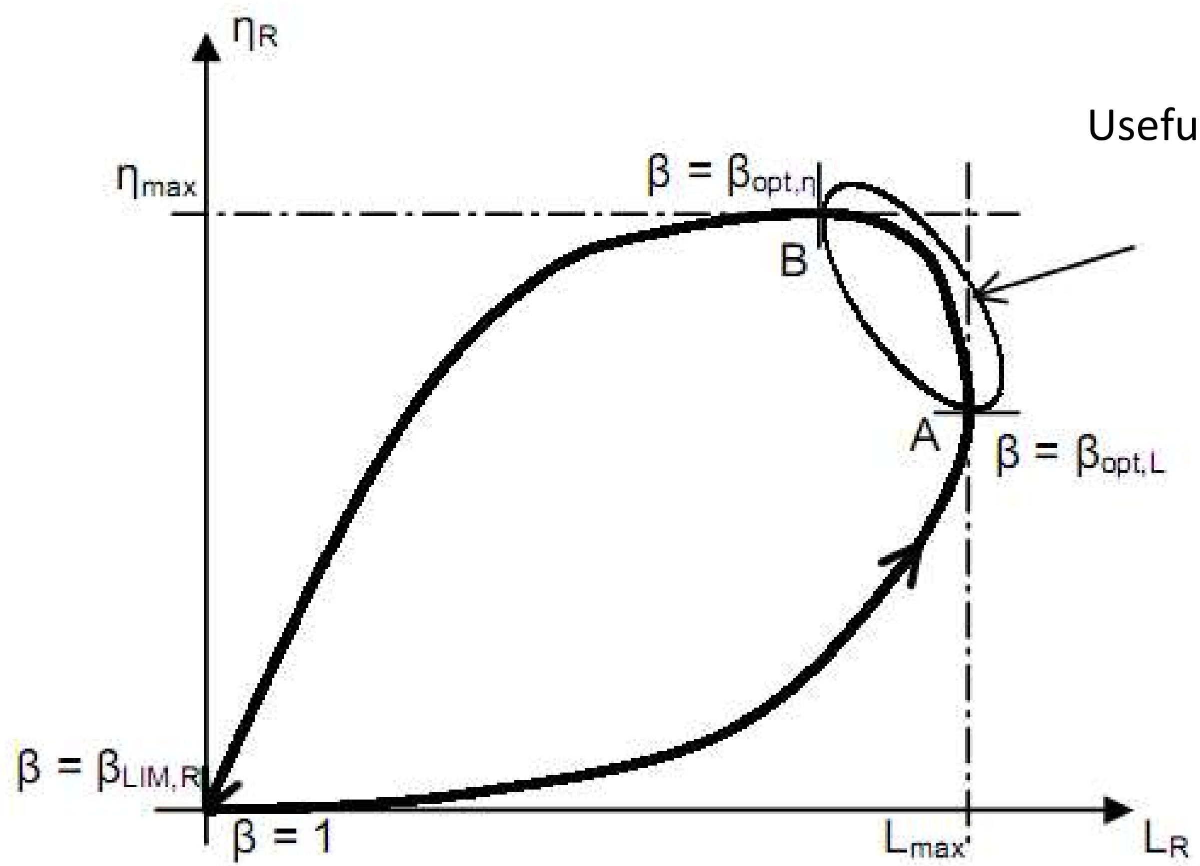


Efficiency and specific work

Gas turbines operating at **maximum efficiency** are usually used to supply the base energy load, so that the high fixed costs are amortised with a high number of operating hours, while the operating costs weigh less.

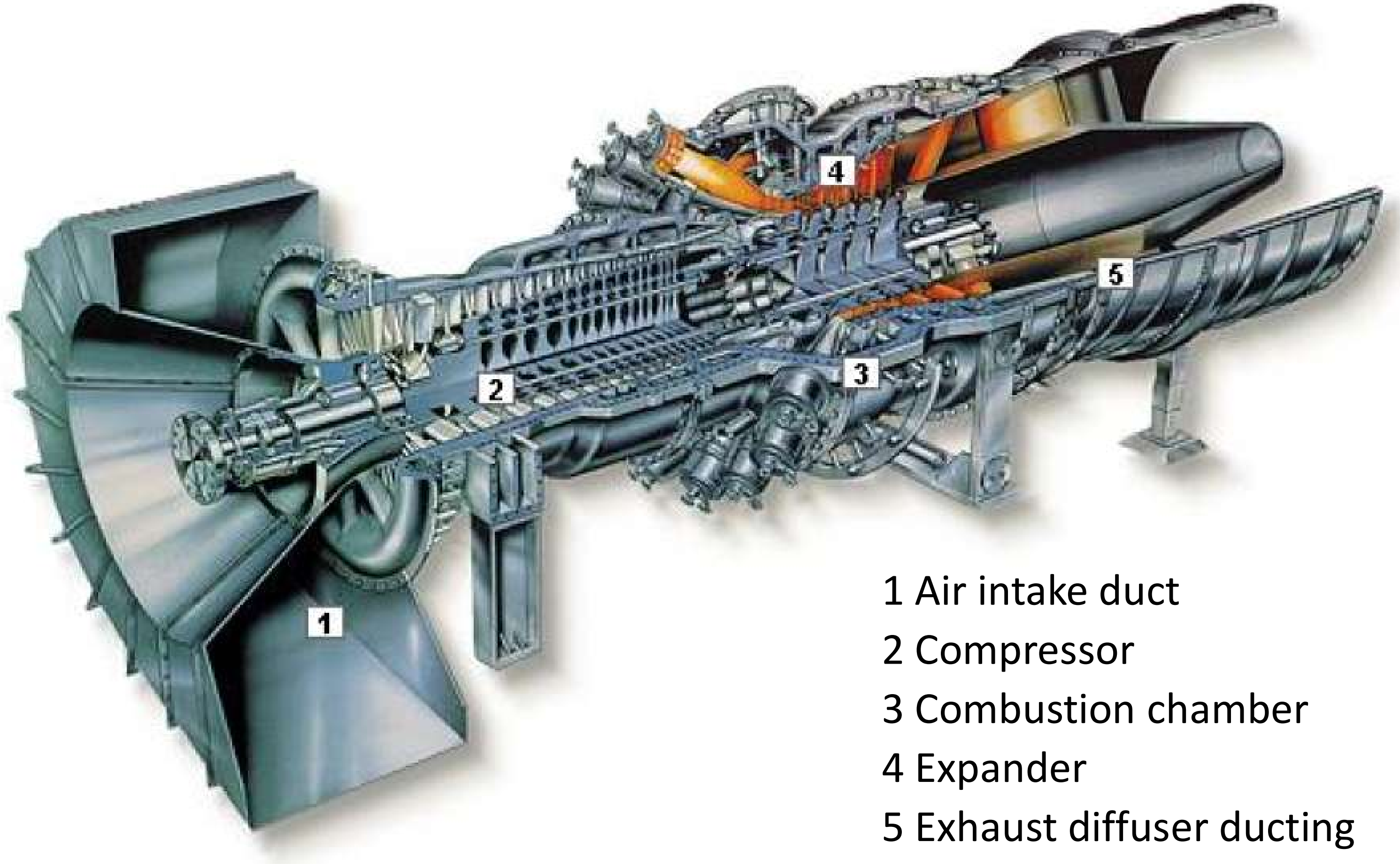
Differently, gas turbines that optimise **specific work** will be used to carry out peak load tasks with a low number of operating hours, having lower fixed costs but lower efficiency. The low number of operating hours means that having a higher specific fuel consumption than other machines has little impact on the overall economic balance.

Efficiency and specific work



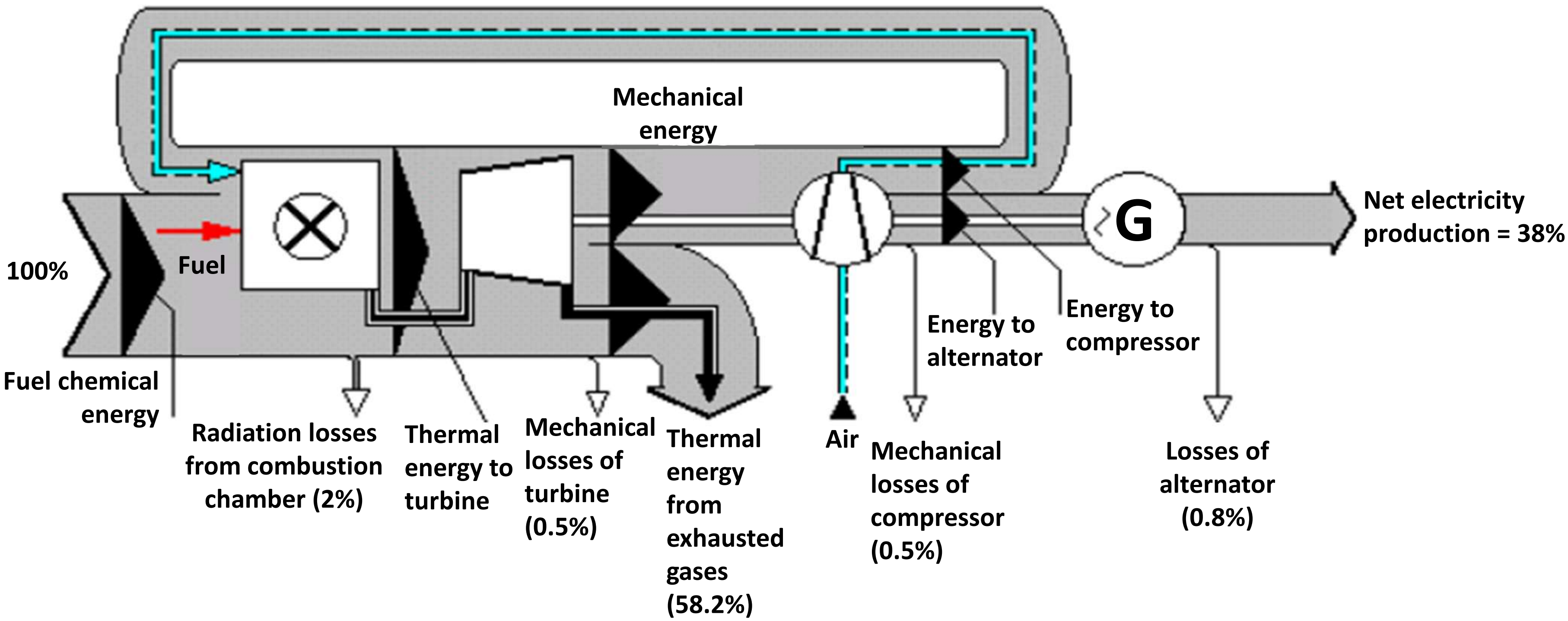
Useful operating area

Gas turbine operation



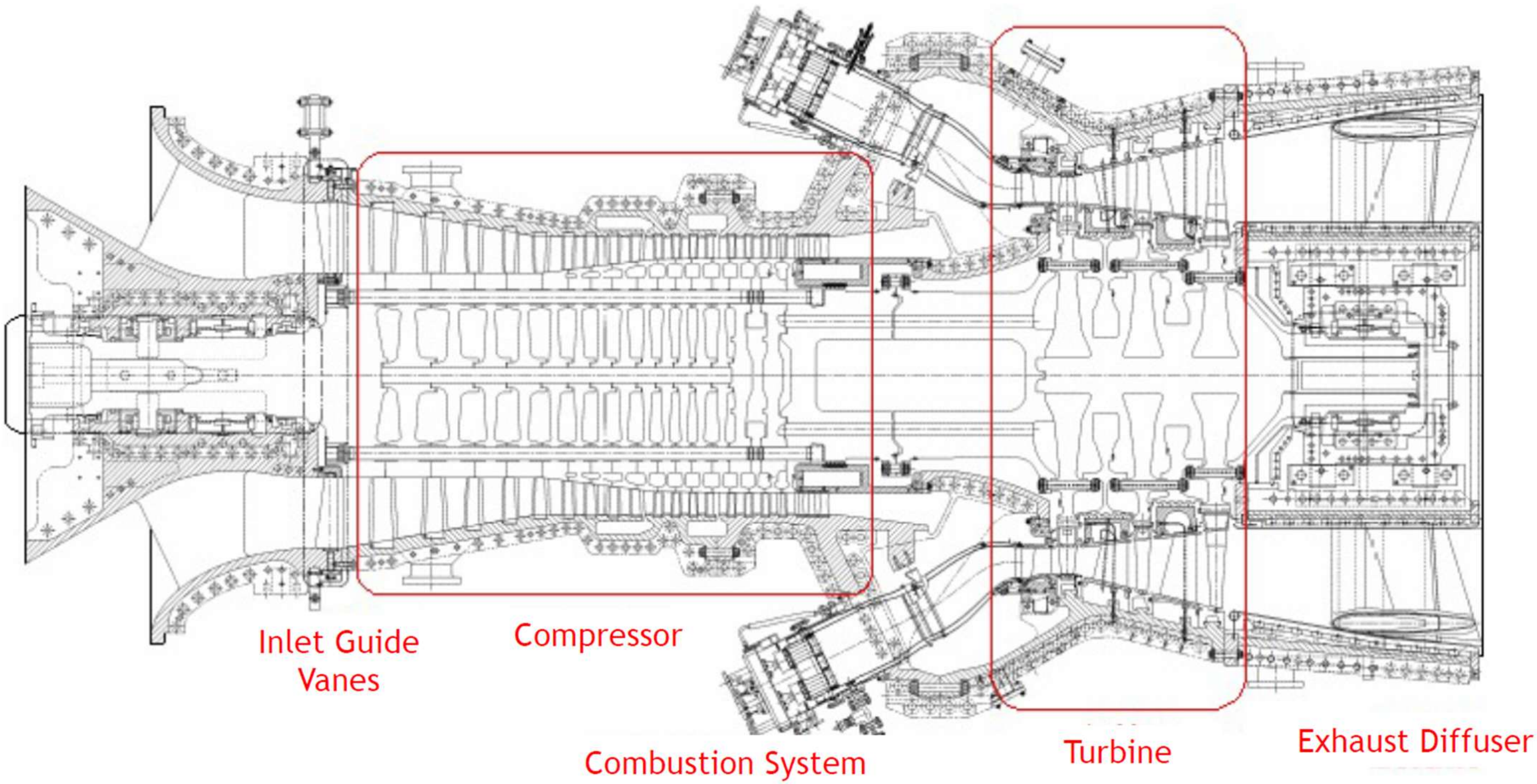
1 Air intake duct
2 Compressor
3 Combustion chamber
4 Expander
5 Exhaust diffuser ducting

Energy balance



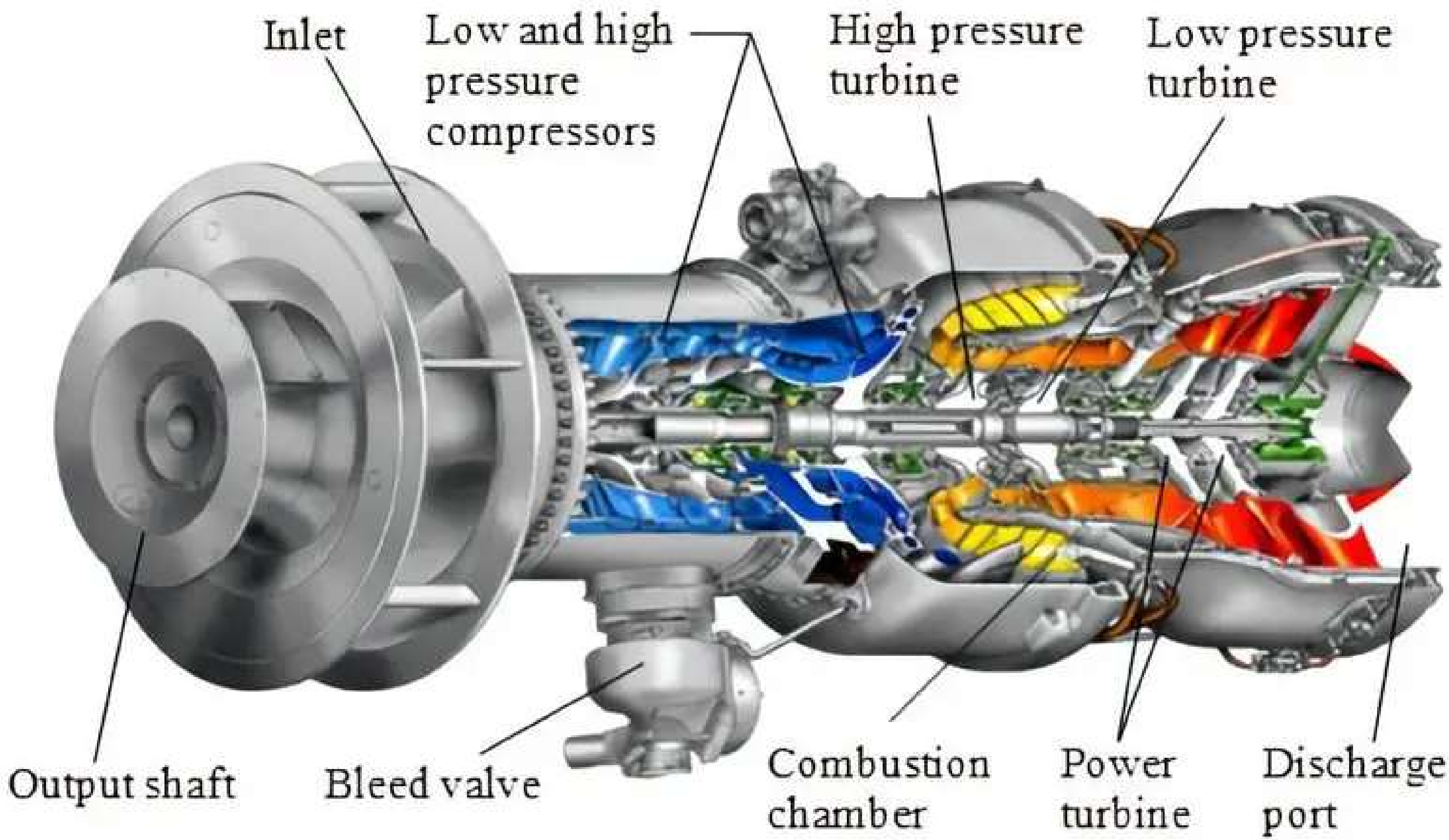
2. Gas turbine components

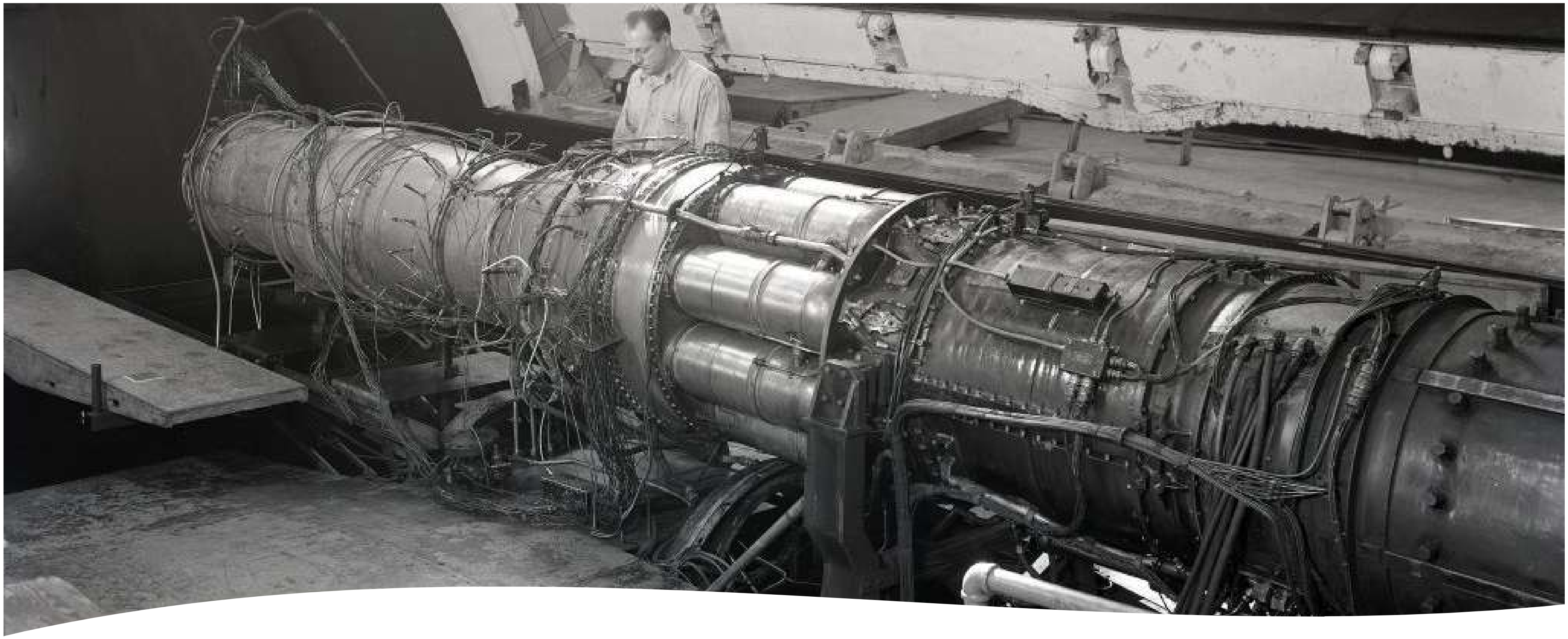
Components overview



Compressor

- Generally axial
- Fluid dynamics design
- Stalling and pumping



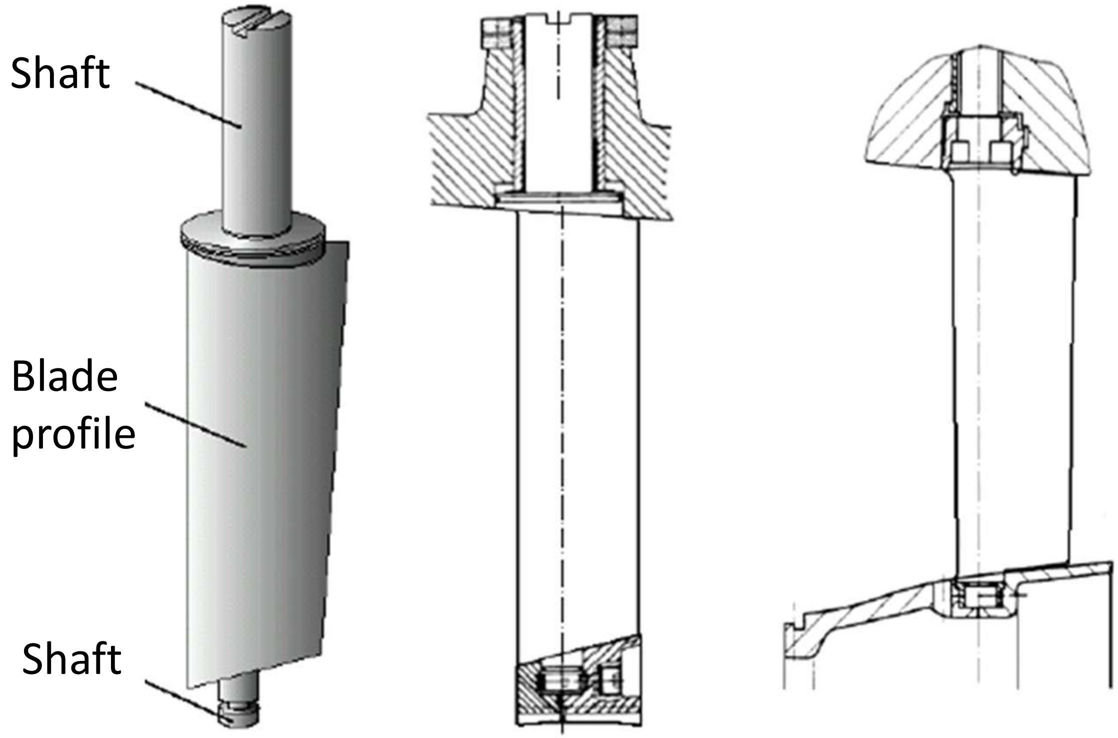


Dynamic compressor

The development of dynamic compressors began in the late 1800s, early 1900s. The first to be designed and used with some success were centrifugal compressors. In axial compressors, obtained by reversing the flow of steam turbines of the same type, there were major instability problems, which were only studied and solved from the 1950s onwards, in parallel with the development of the gas turbine as a machine for aircraft propulsion. This was possible thanks to fluid-dynamic studies both experimental, on profiles and blade arrays, and theoretical, thanks to the development of the first calculators that allowed the Navier-Stokes equations governing fluid motion to be solved numerically.

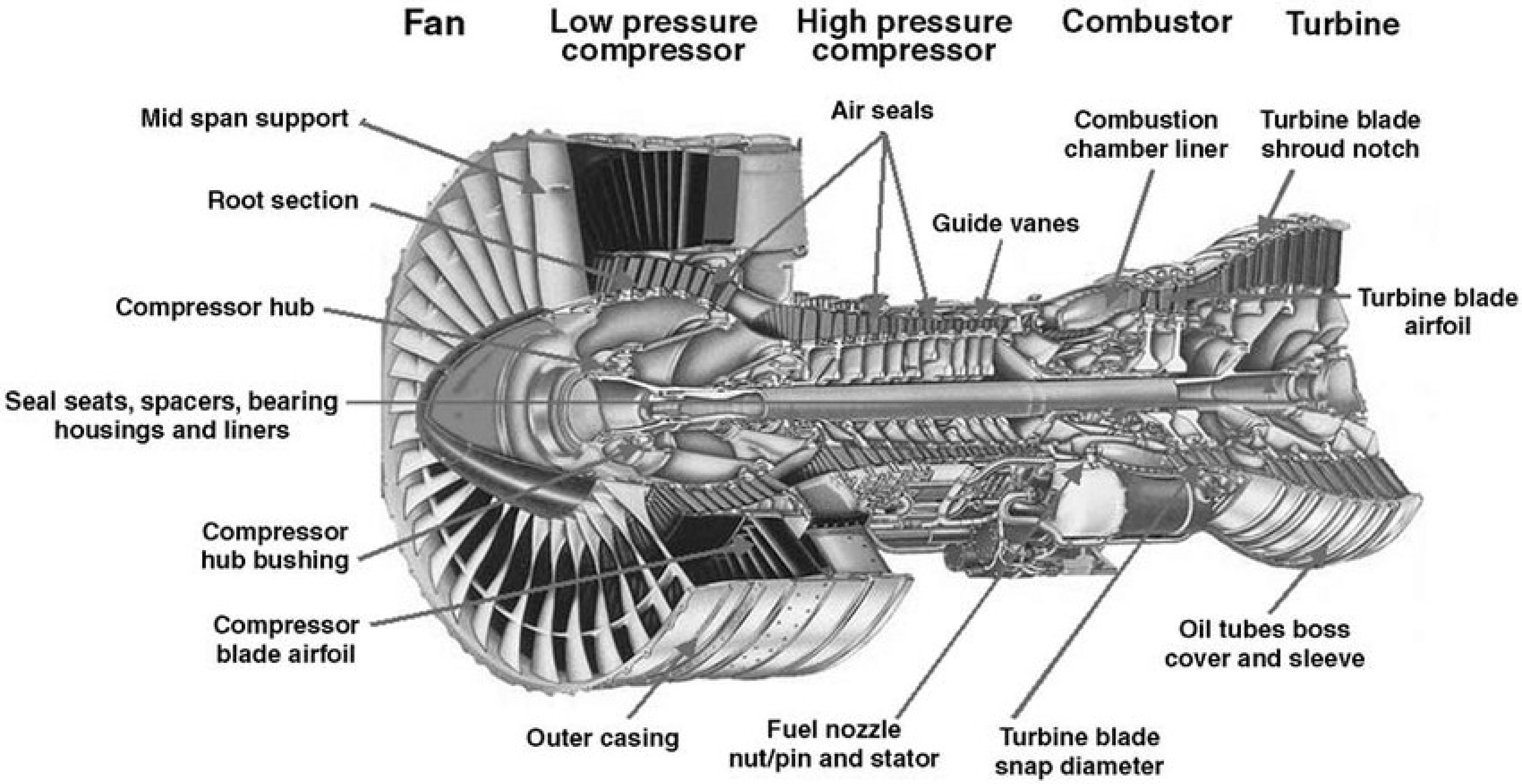
Photo of TG-190: Researchers study suction performance, an experimental fuel and afterburner configurations (31/8/1948).

Design features - Compressor



Inlet Guide Vanes

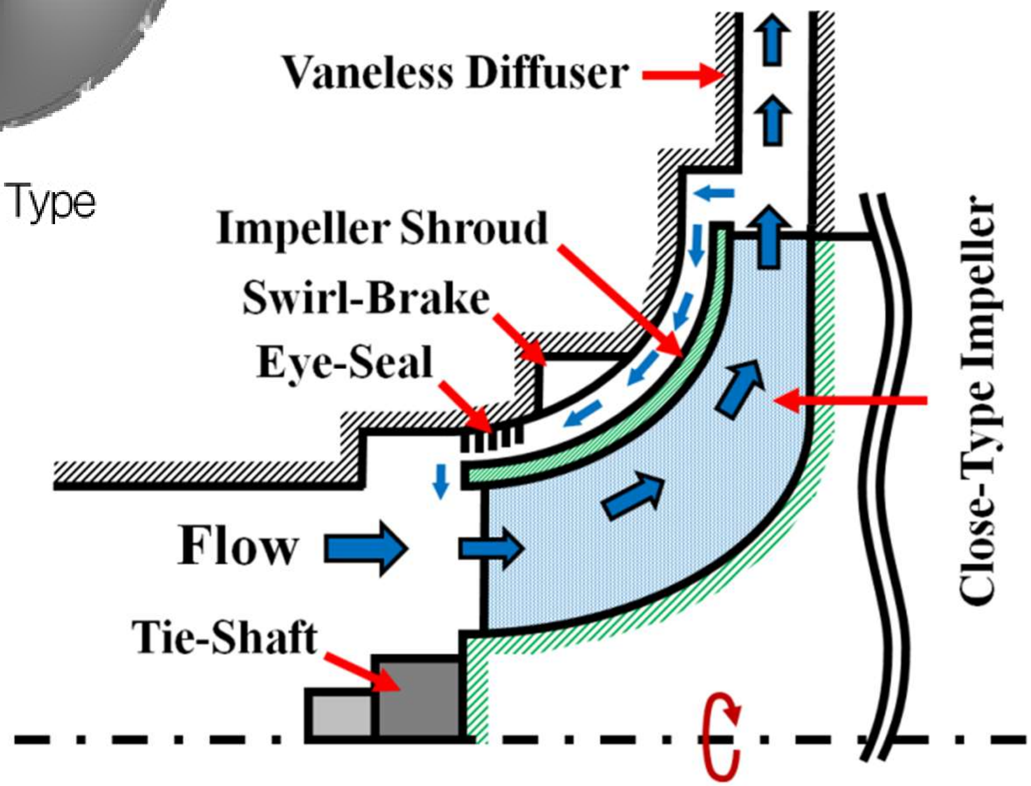
Design features - Compressor



Centrifugal compressors

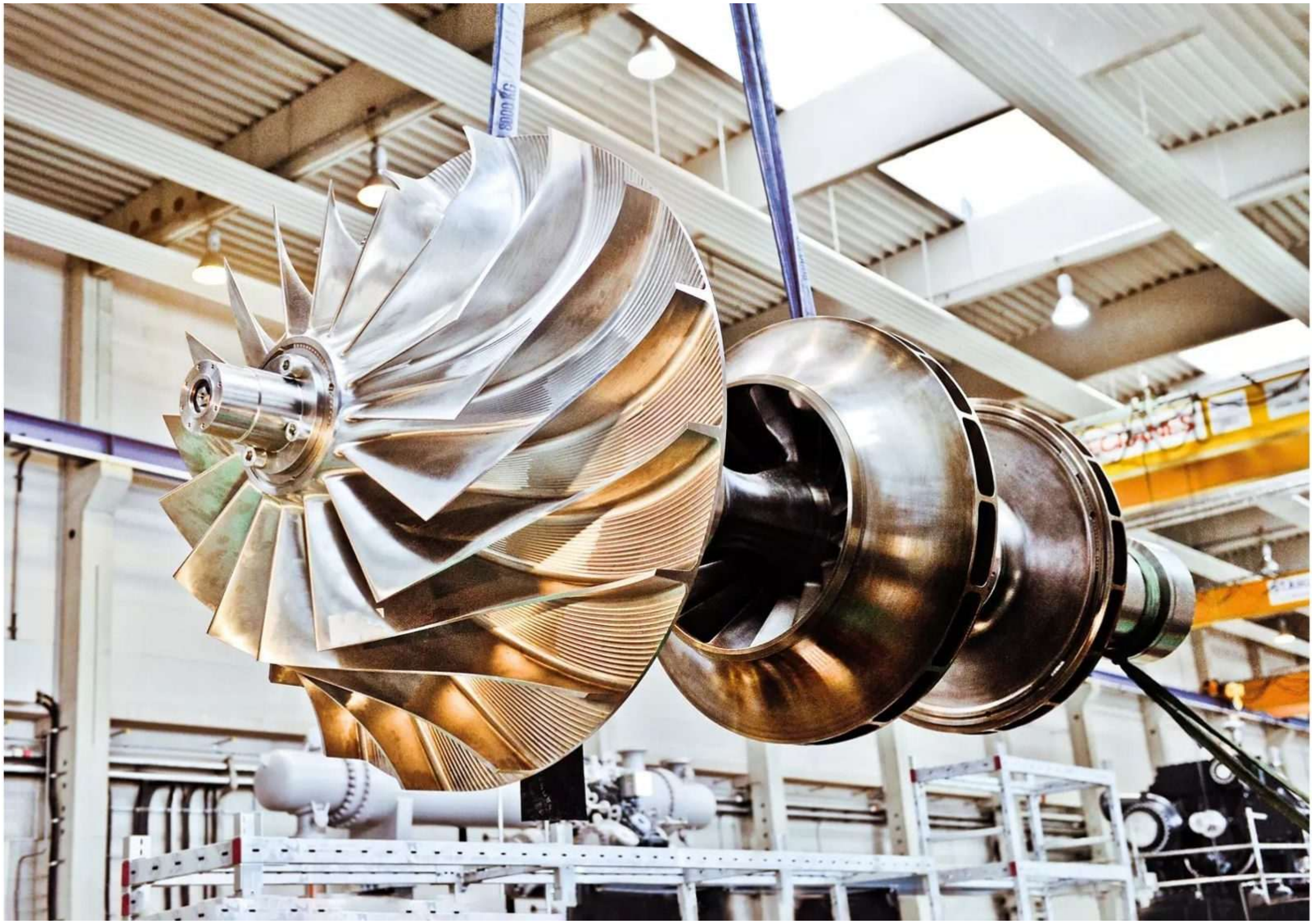
(a) Open Type

(b) Close Type



[http://dx.doi.org/10.6108/KSPE.2017.21.1.026]

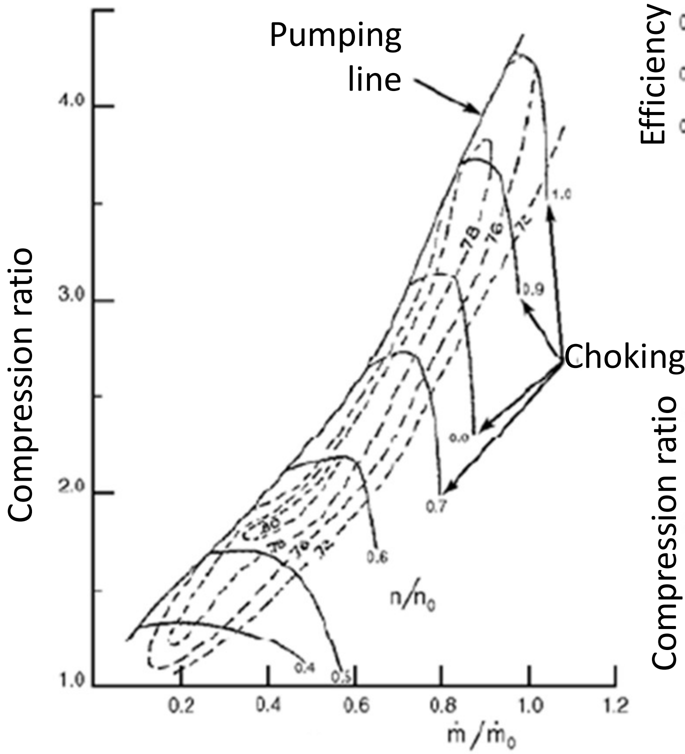
Centrifugal compressors



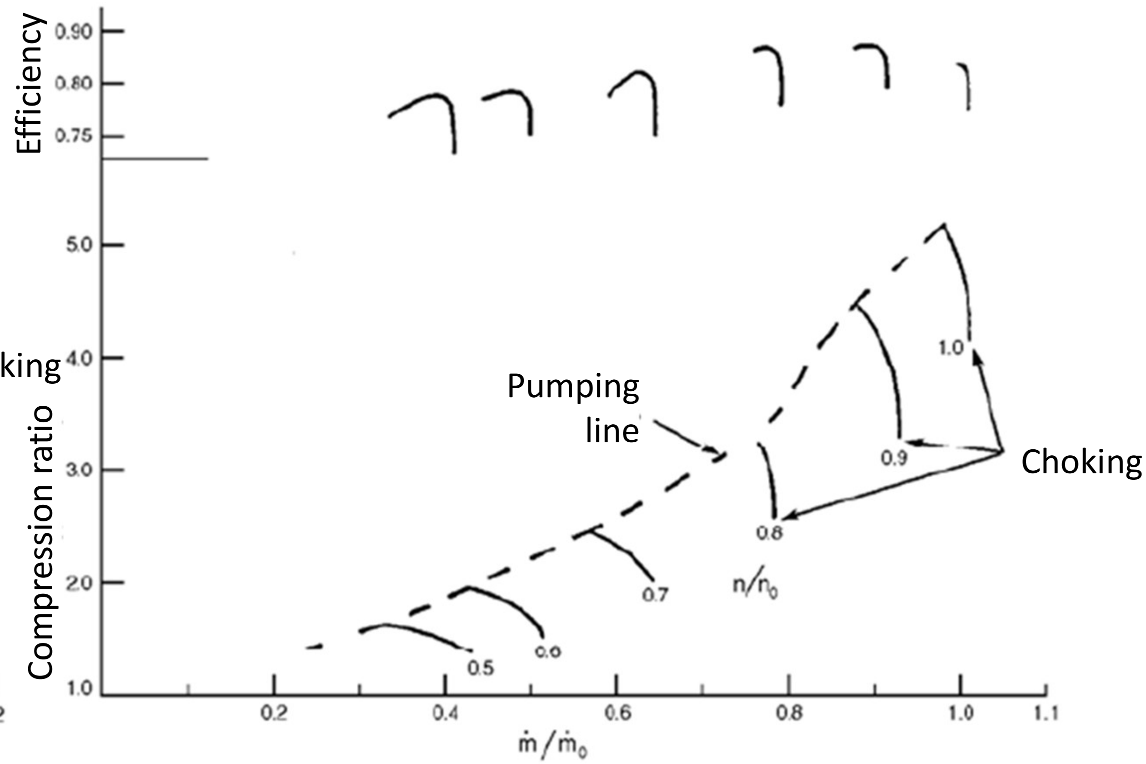
[www.controlgear.net]

Stall and pumping

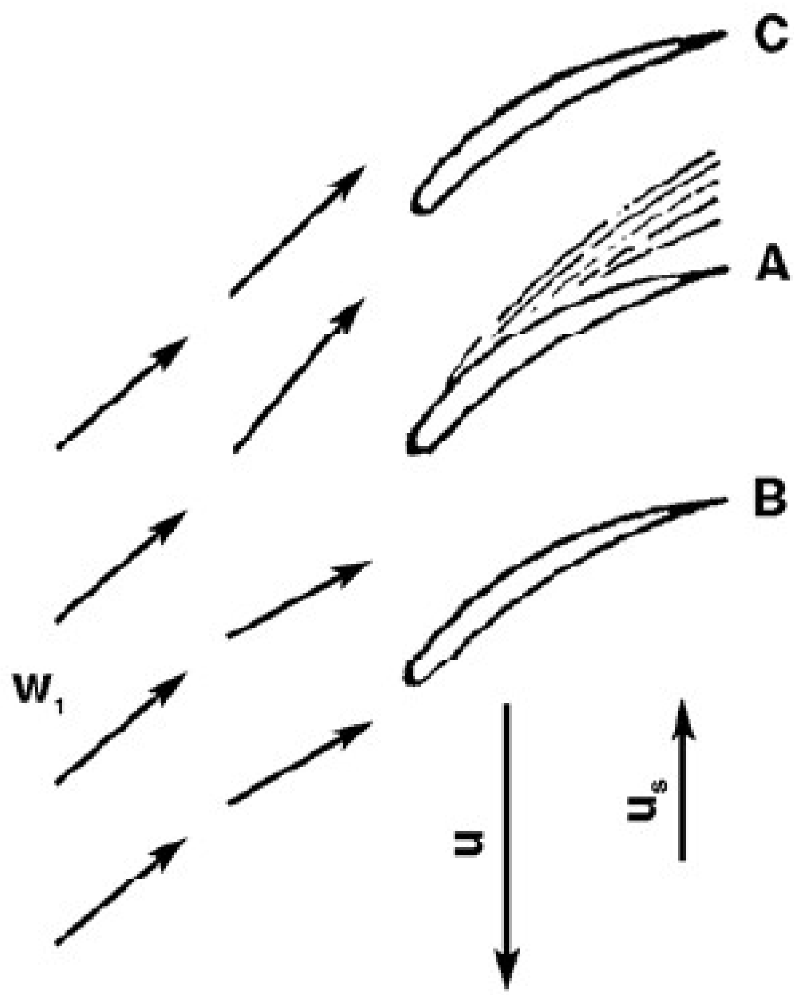
Centrifugal compressor



Axial compressor



Stall and pumping



Combustion chamber

The combustor is used to increase the temperature of working fluid by means of the heat generated by the fuel oxidation reactions. Since the temperature reached by the combustion gases is limited by the resistance of the materials, the quantity of fuel used is considerably less than that corresponding to the stoichiometric combustion of the air leaving the compressor.

Combustion chamber

- For example, assuming an air temperature at the compressor outlet of 400°C and using methane as fuel, the ideal stoichiometric combustion would bring the combustion gases to a temperature of around 2430°C.

→ This temperature is far beyond the current gas turbine construction technology.

Combustion chamber

Reaching a temperature of 1300°C, on the other hand, requires 45 kg of air per 1 kg of methane, compared to a stoichiometric ratio of 17.235 kg of air per kg of methane. In fact, to limit the increase in temperature, gas turbines generally operate with an equivalence ratio λ (amount of actual air to stoichiometric air) between 2.5 and 3.5. Consequently, the oxygen content in the flue gases is very high (on average around 15% by volume) and makes it possible to use the exhaust of these gases as an oxidiser in further combustion processes.

Combustion chamber

Maintaining combustion in the presence of a high excess of air can be problematic due to the lower flammability limit of the air/fuel mixture. Therefore, it is necessary to create a zone in the combustion chamber, known as the primary zone, in which only part of the combustion air flow reacts with the fuel.

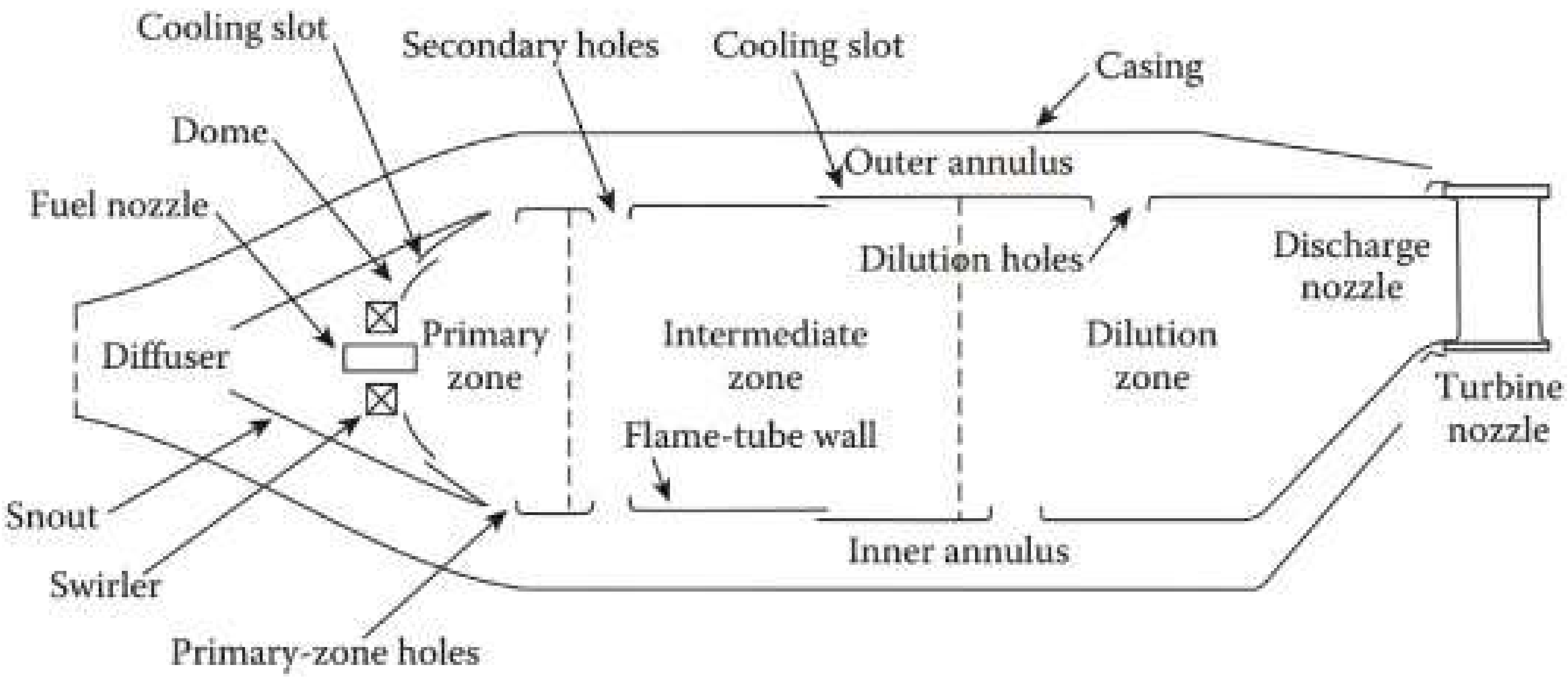
- In the **primary zone**, a sub-stoichiometric air/fuel ratio is achieved to reduce nitrogen oxides.
- The remaining air flow is introduced into the **secondary zone**, where the oxidation of the fuel is completed.
- Finally, in the **dilution zone**, air is added to obtain an adequate temperature of the exhaust gases to be fed into the turbine.

Combustion chamber

This is achieved by a **liner**, i.e. a perforated cylinder, which first contains the flame and then allows the dilution air to pass through its holes.

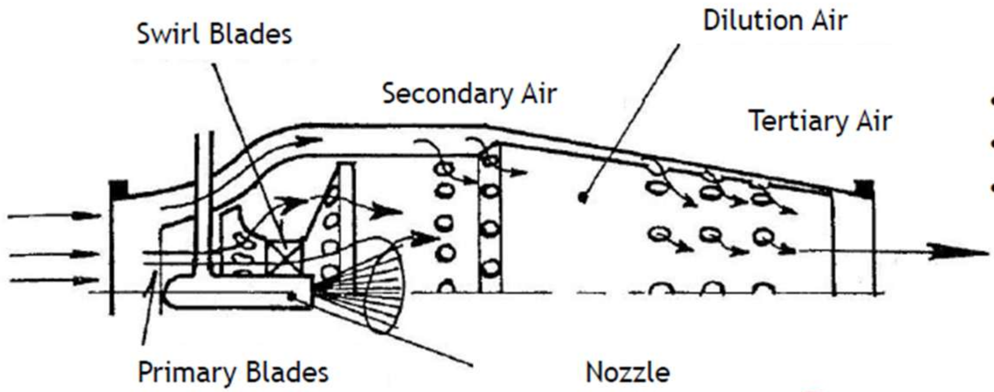
The liner is cooled on its outside by the flow of dilution air.

Combustion chamber

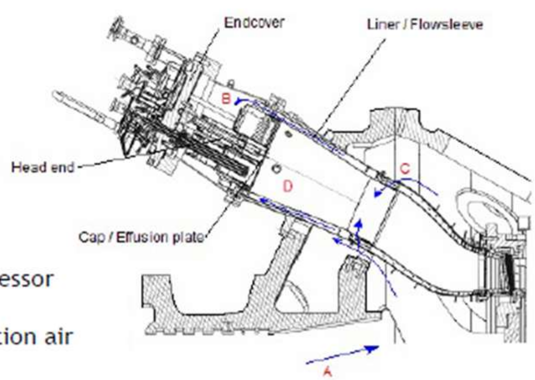


[https://doi.org/10.1016/j.paerosci.2023.100927]

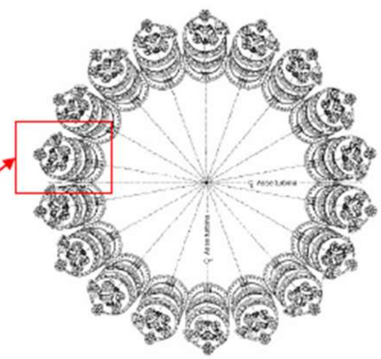
Combustion chamber



- Primary Air: 15-20%
- Secondary Air:30%
- Dilution Air: enough to reach 100% dilution

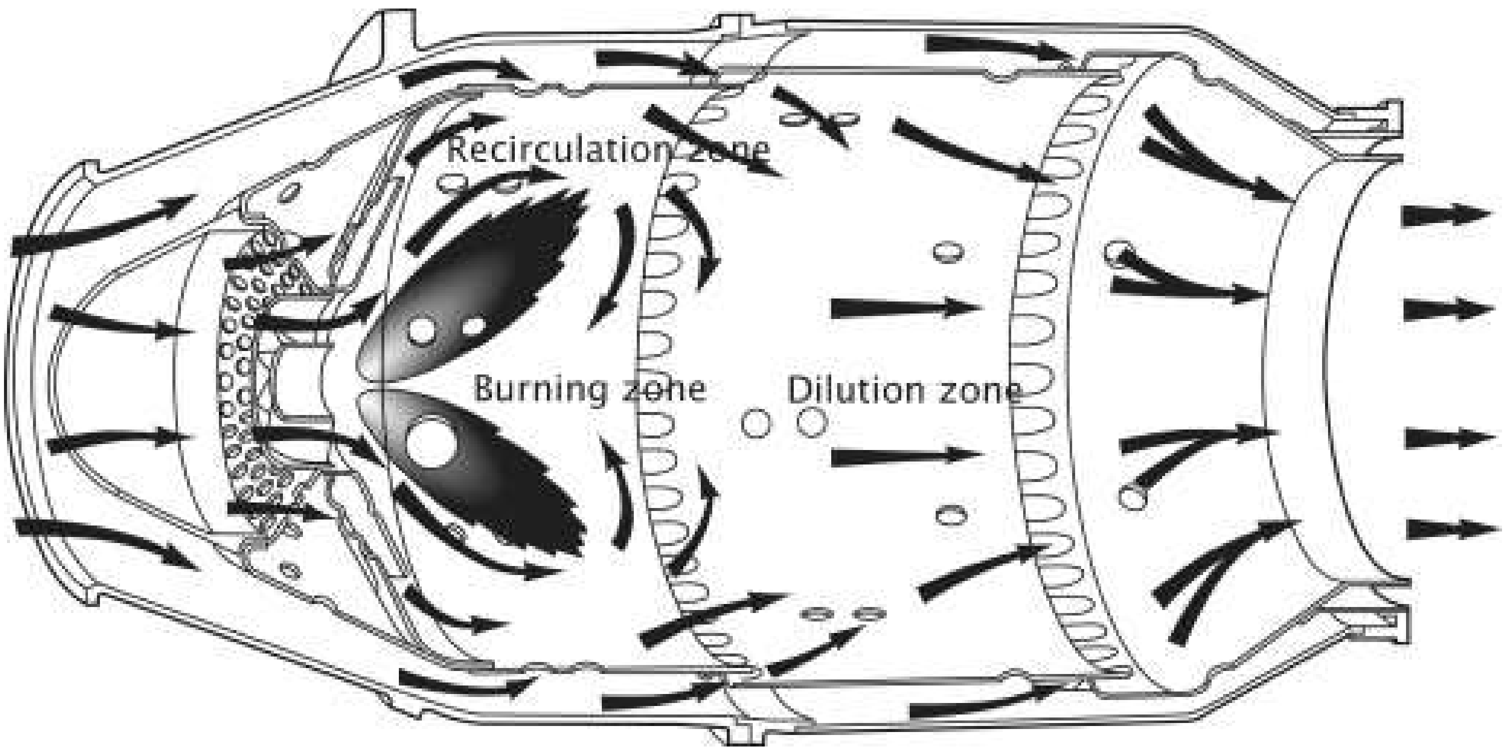


A: Air from the Compressor
B: Primary Air
C: Secondary and dilution air
D: Flame tube



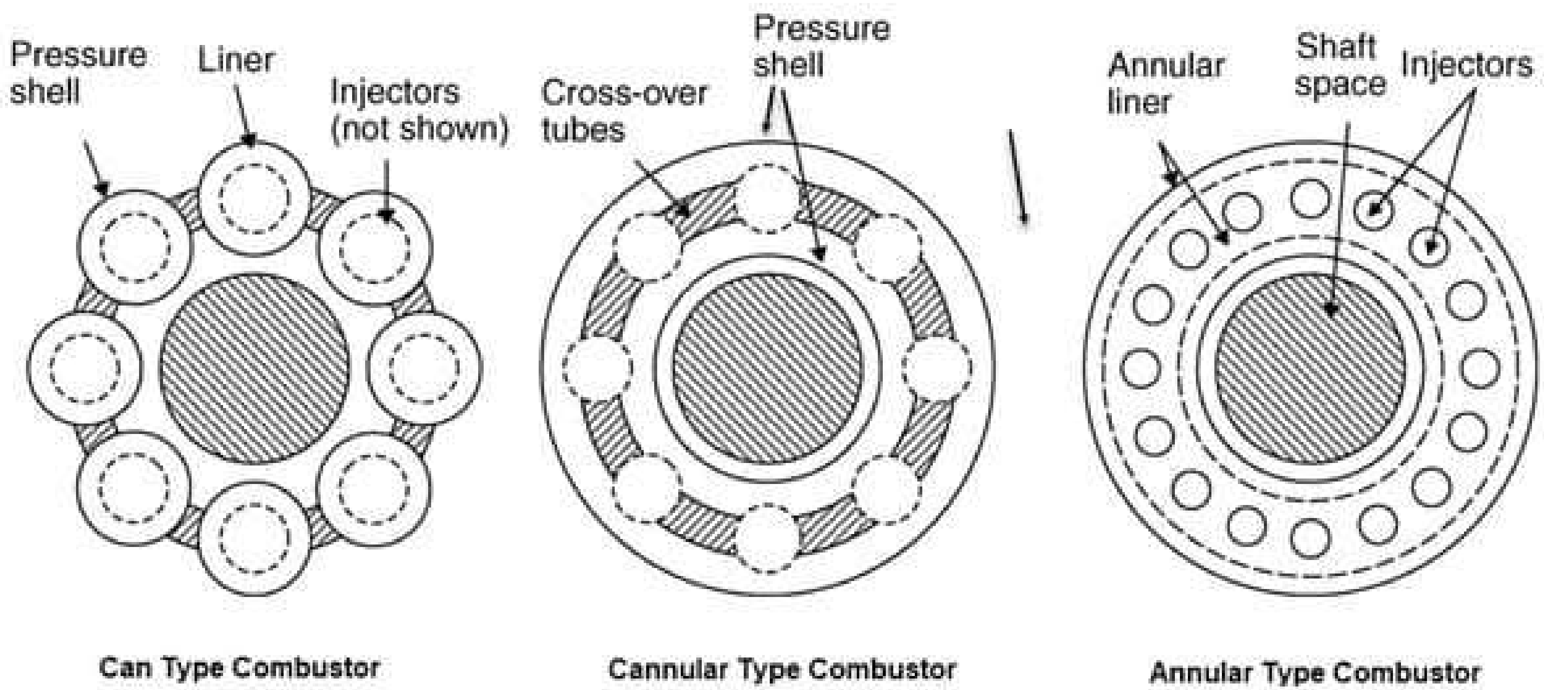
- Gas Burner
- About 20 Burner circularly positioned around the shaft. Each of them is composed by:
    - 1 Pilot Injector
    - 8 Main Injectors
- 3 indipendent supply line
- 2 injector are equipped with spark plugs while the others are ignited by cross flames
- Ultraviolet Flame sensors
- In each combustion chamber swirl blades are present to stabilitze the flame

Combustion chamber



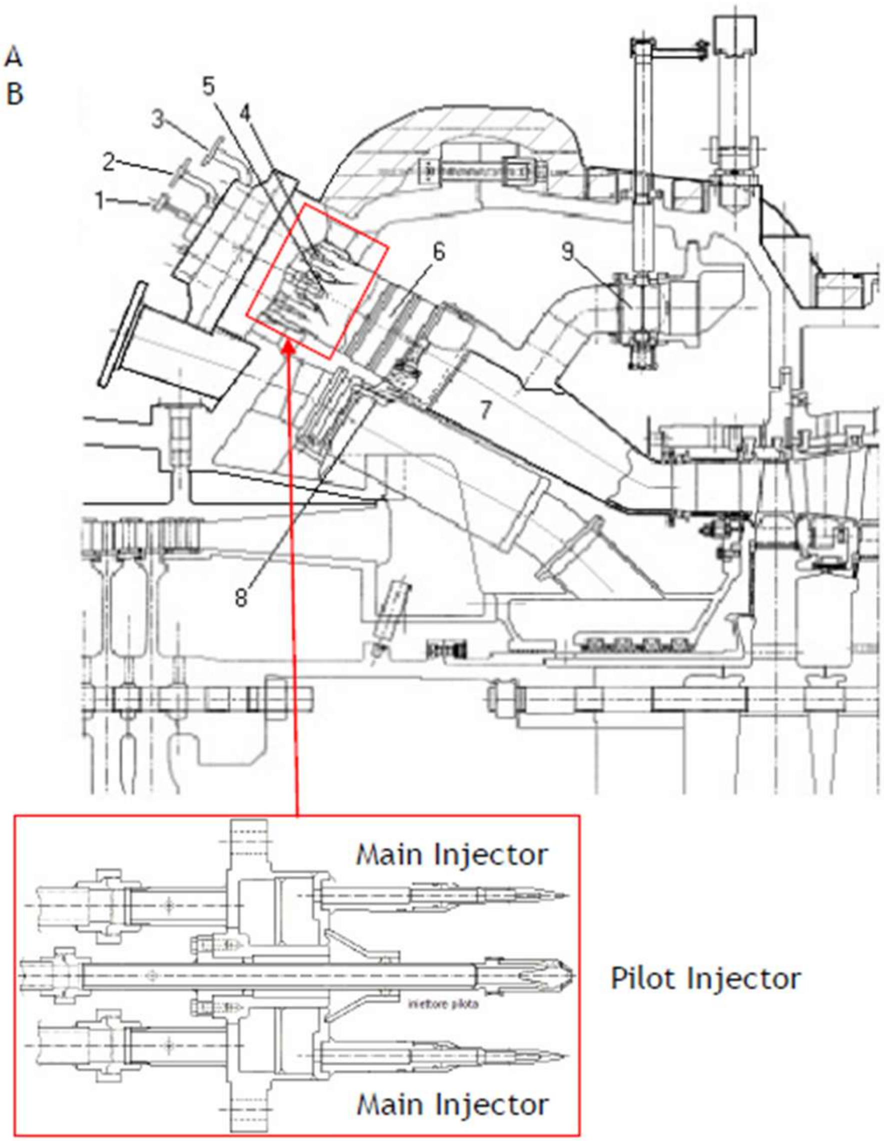
[https://doi.org/10.1016/B978-0-12-383842-1.00010-X]

Combustion chamber

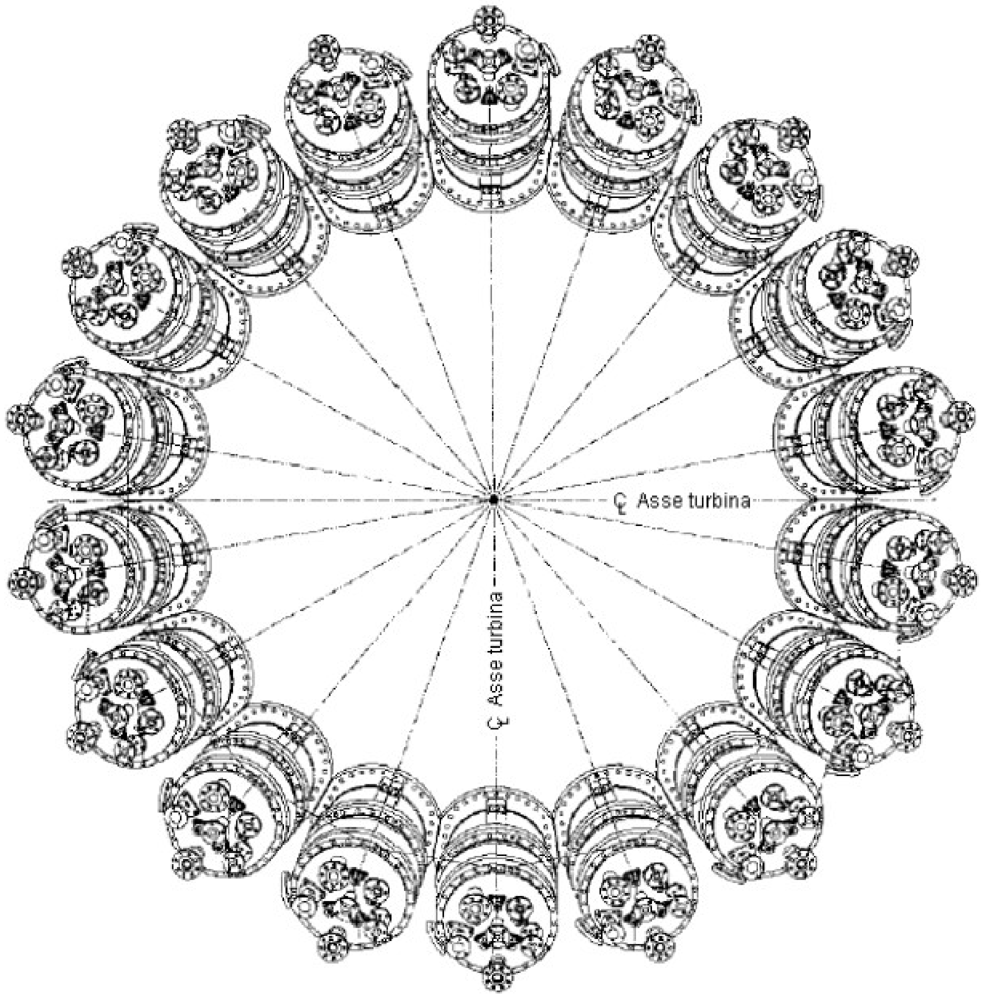


Design features - Combustors

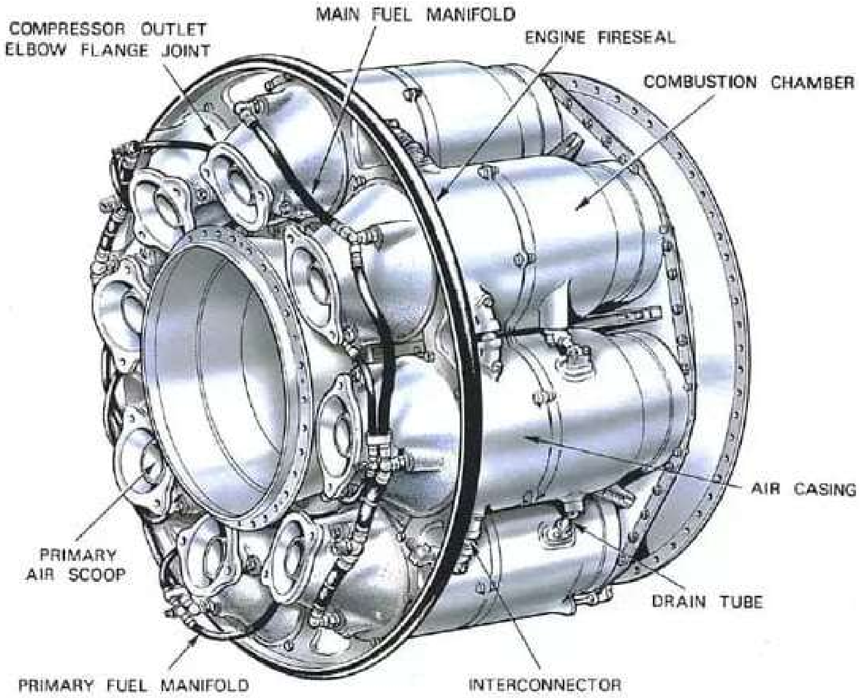
1. Pilot Injector Supply
2. Main Injector Supply A
3. Main Injector Supply B
4. Main Injector
5. Pilot Injector
6. Comustion Chamver
7. Transition Piece
8. Burner Bearing
9. Bypass Valve

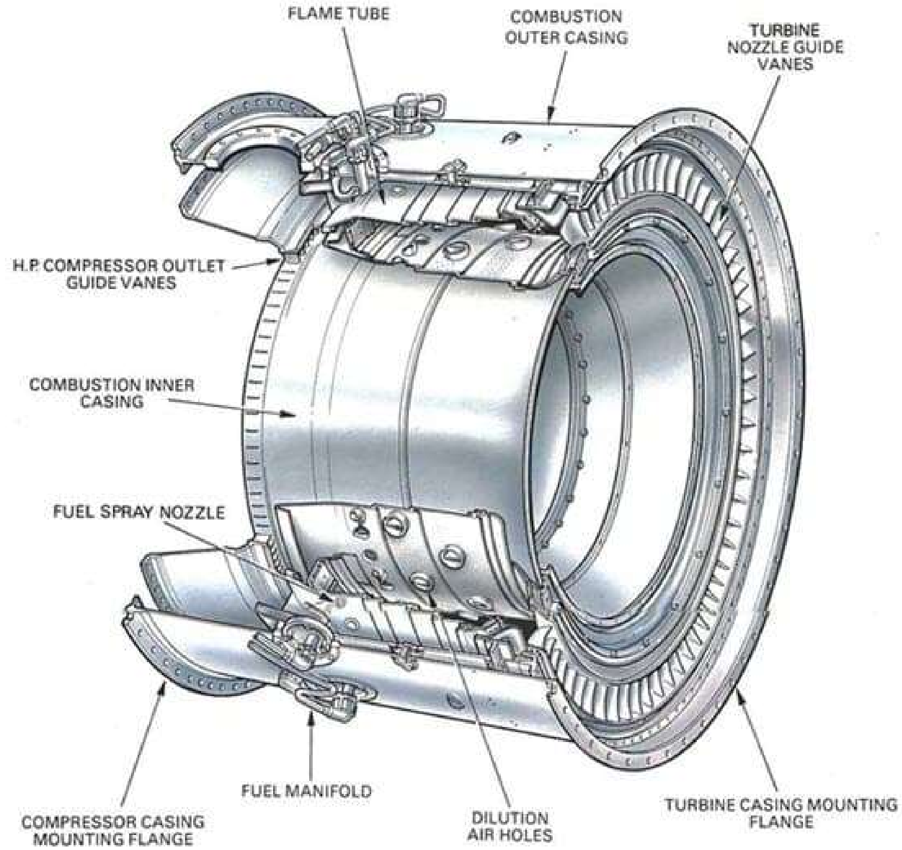


Design features - Combustors



Design features - Combustors





Fuels for gas turbine

- Gas turbines are internal combustion engines. It is therefore necessary that the gases are not chemically and physically aggressive towards the turbine and combustor, in particular, they do not lead to:
  - Corrosion
  - Erosion
  - Fouling and clogging

Fuels for gas turbine

The main elements considered harmful to the turbine are:

- Ash: mainly metallic compounds found in coal and heavy oils (t melting point 1200 °C). They tend to resolidify on the blades
- Vanadium: corrosion phenomena even with few ppm
- Alkali metals: (sodium and potassium) present in the form of salts. They are corrosive
- Sulphur: corrosive in the form of H_2S
- Heavy hydrocarbons (asphaltenes, gummy compounds): give clogging problems in fuel supply systems

Fuels for gas turbine

Positioning of fuels for direct use in TG:

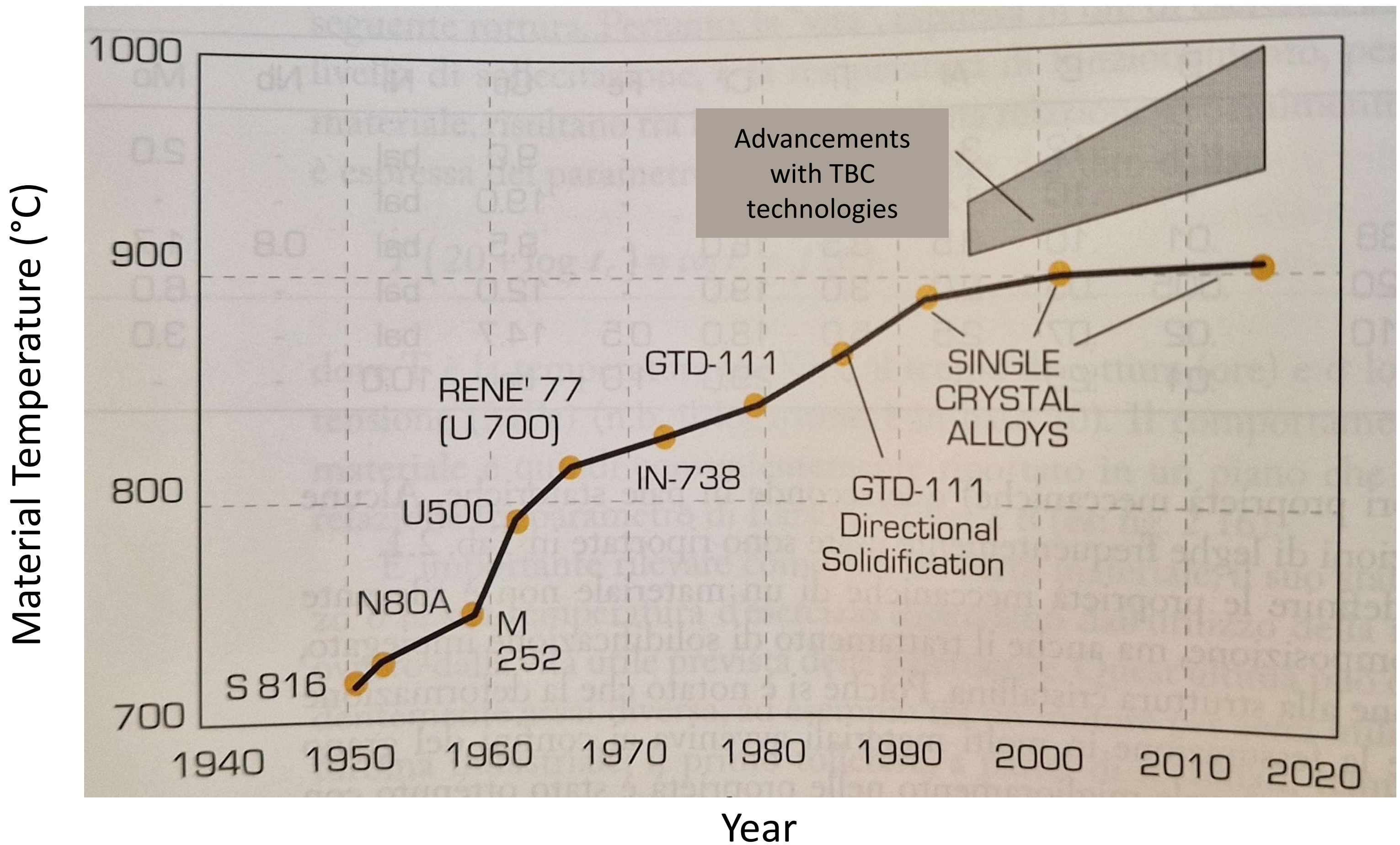
- Coal: not possible.
- Heavy fuel oils (crude oil processing residues) and crude oil are possible to be used if:
  - Additivation + frequent flushing
  - Adequate fuel system
  - Accept derating
- Distillates (naphtha, kerosene, diesel): suitable for aviation propulsion.
- Natural gas: excellent fuel for stationary applications
- Liquefied petroleum gases (propane, butane): excellent fuel for TG but high cost
- Synthetic gases produced from coal, biomass, coke oven gas: require adequate filtering system

Fuels for gas turbine

- Combustor Outlet Temperature (COT) or “Firing Temperature”, is the average total temperature of the combustion gases leaving the combustion chamber as they hit the first stator array.
- Turbine Inlet Temperature (TIT): is the temperature obtained by mixing the gases coming from the combustor and the cooling air of the first stator array; physically it is the temperature felt by the leading edge of the first rotor with an isentropic flow stop.
- Turbine Inlet Temperature according to ISO (TITiso): this is the temperature obtained by mixing the gases coming from the combustor with all the cooling flows of the turbine blades; it has no precise physical meaning and can only be obtained theoretically.

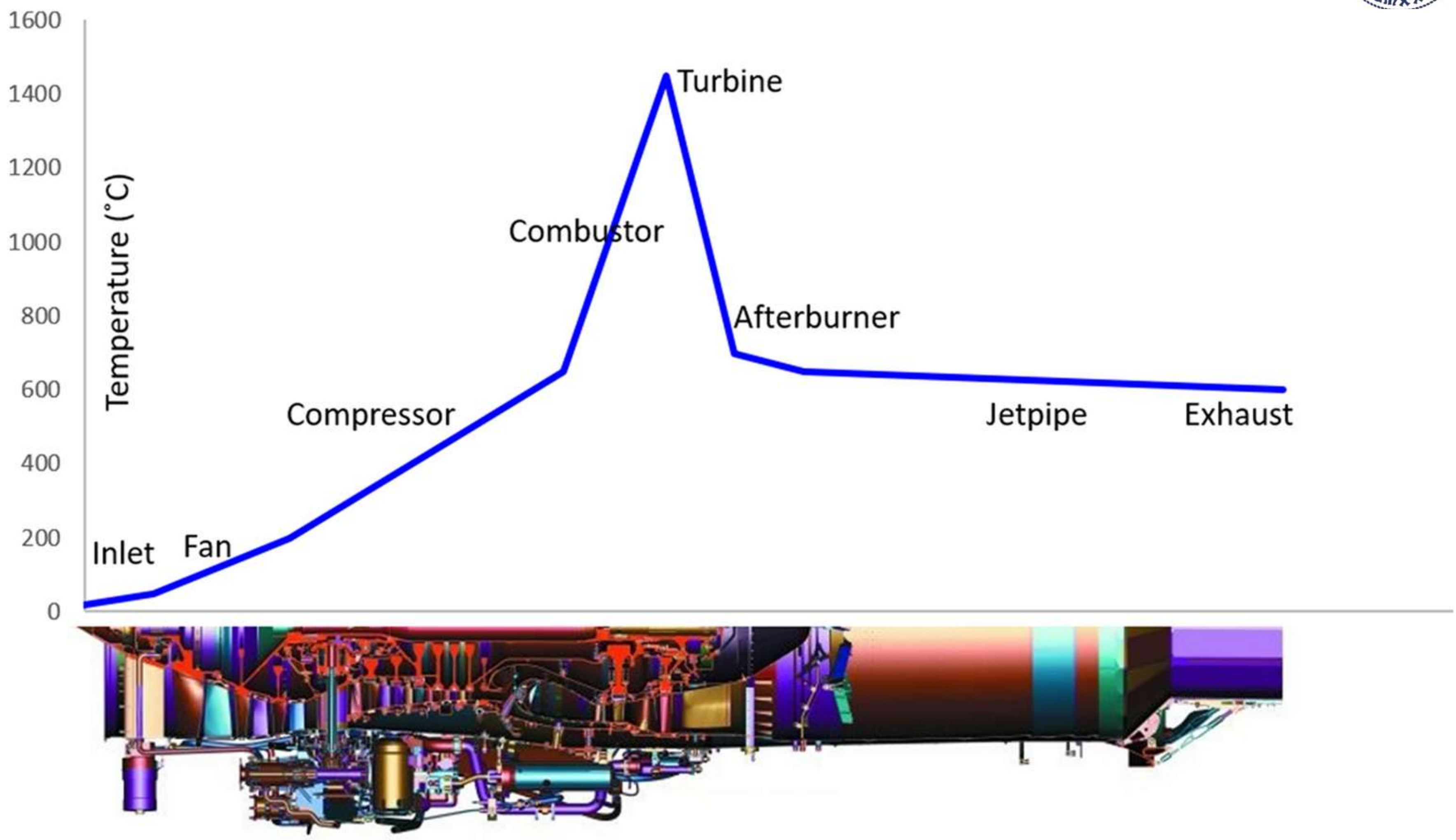
COT>TIT>TITiso

Temperature trend of materials



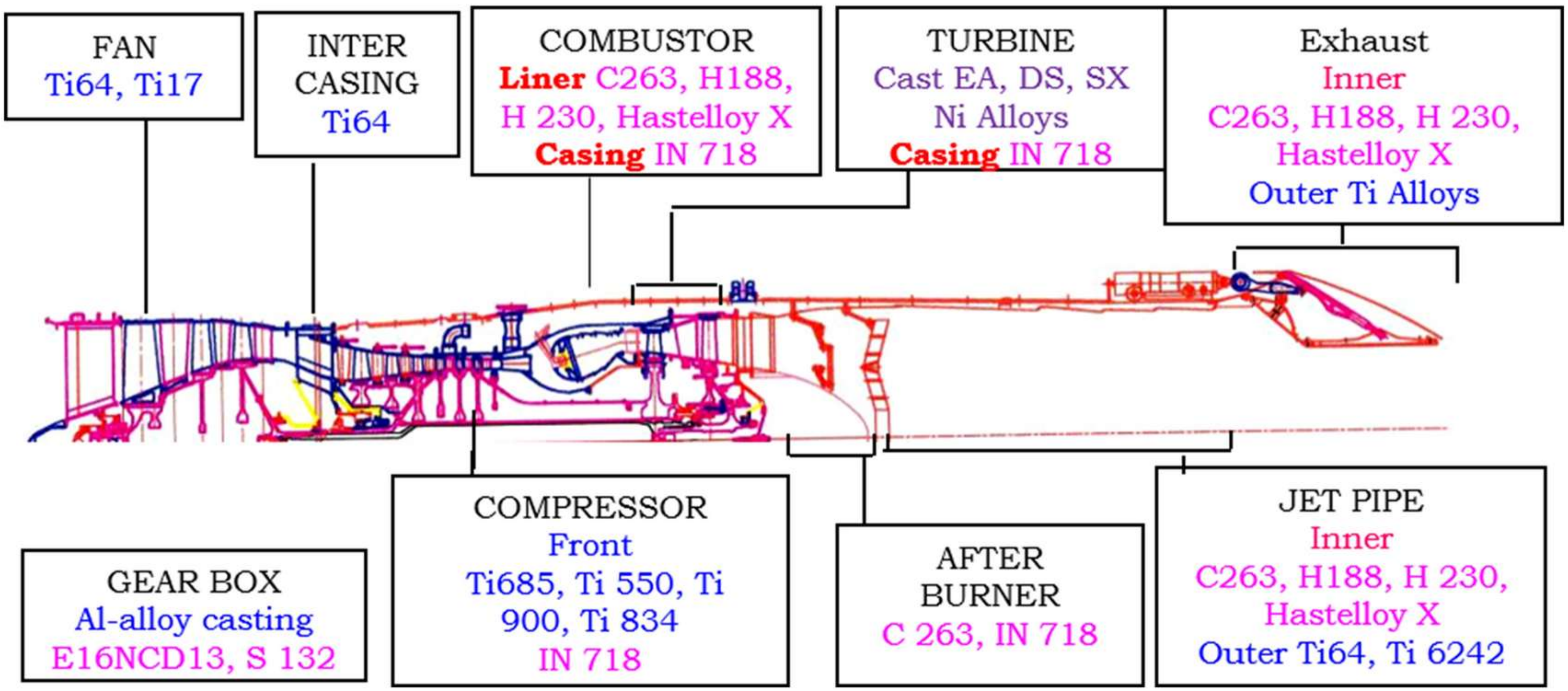
[Lozza]

Temperature trend of materials



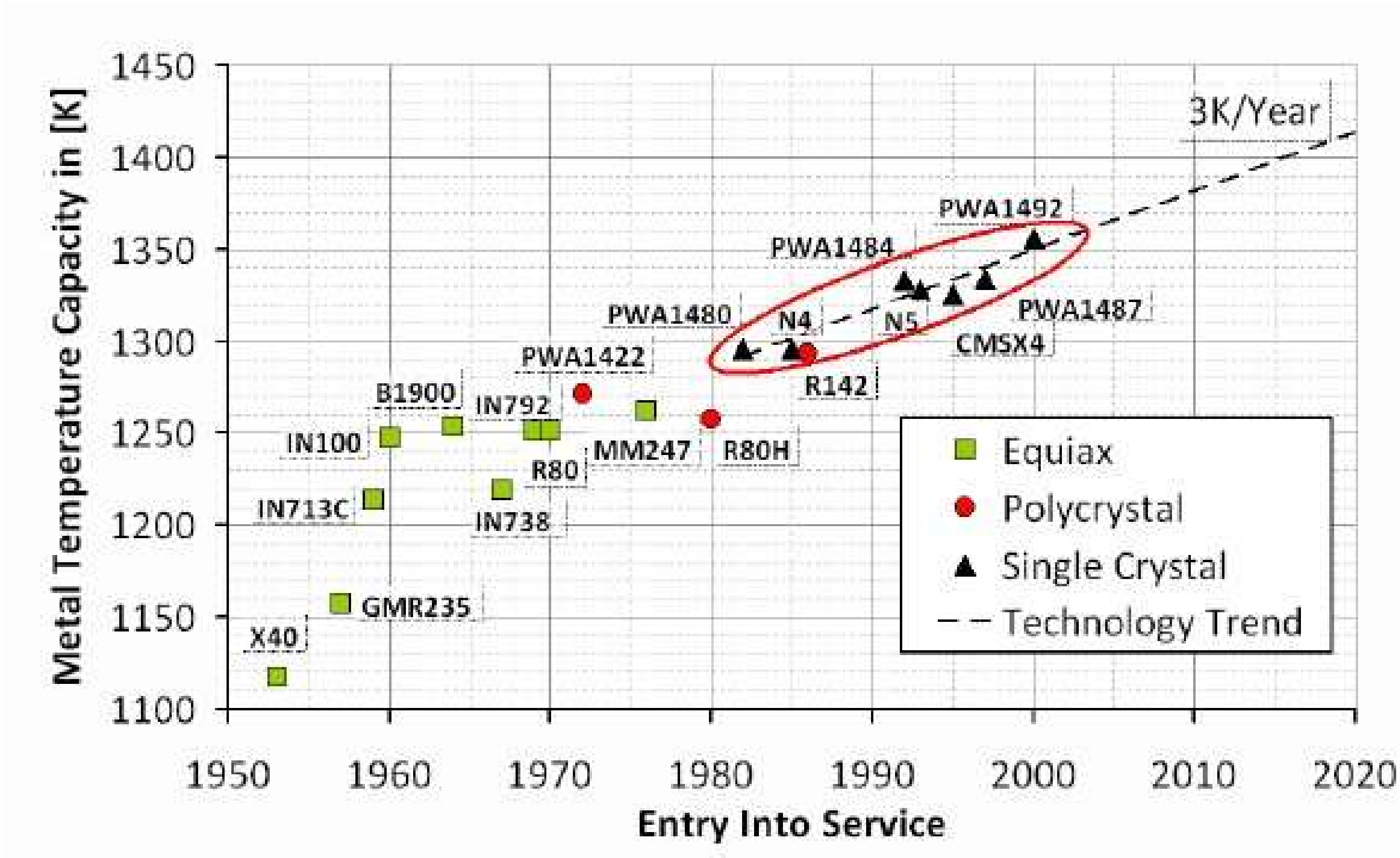
[https://doi.org/10.1007/s41745-022-00295-z]

Materials used in the different components



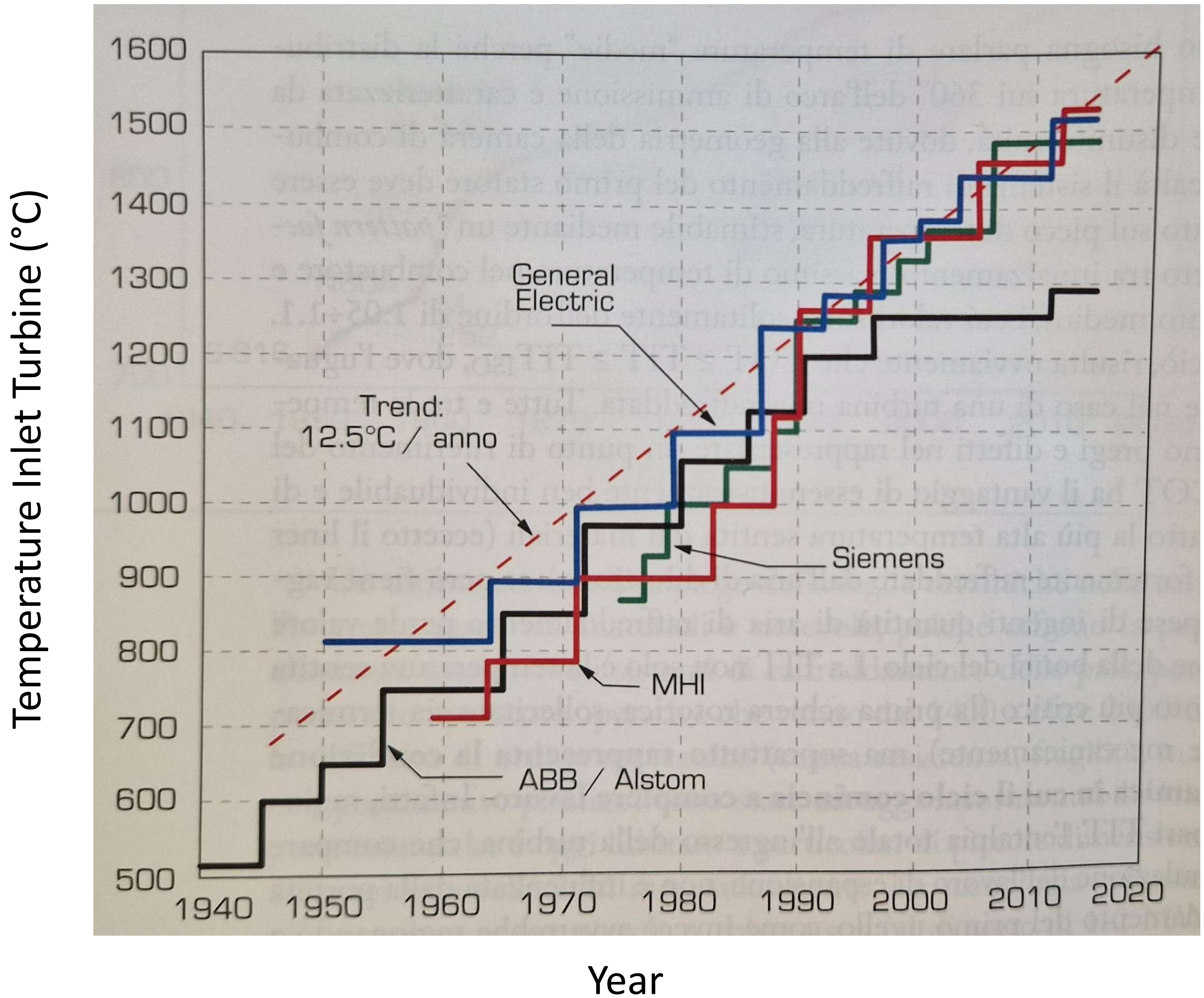
[https://doi.org/10.1007/s41745-022-00295-z]

Temperature trend of materials



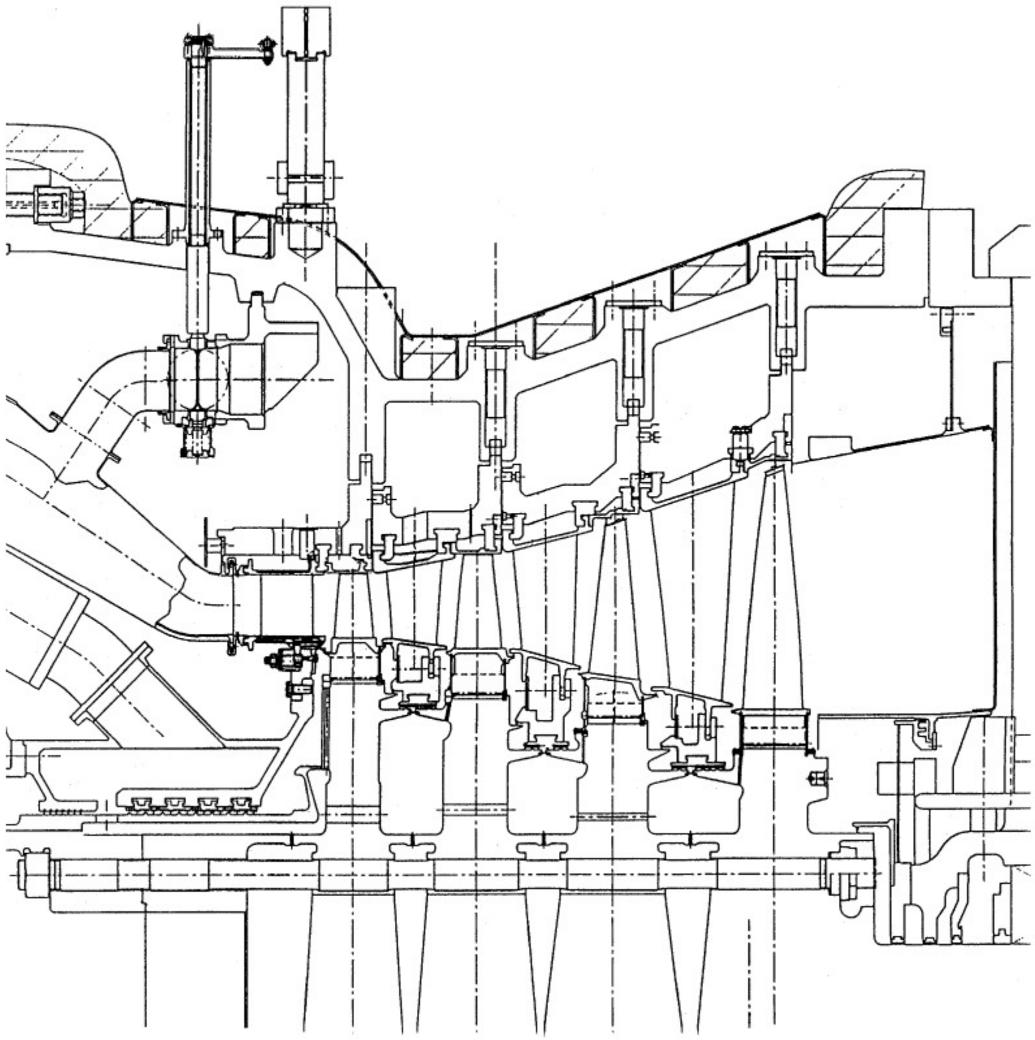
[10.5772/19689]

TIT trend

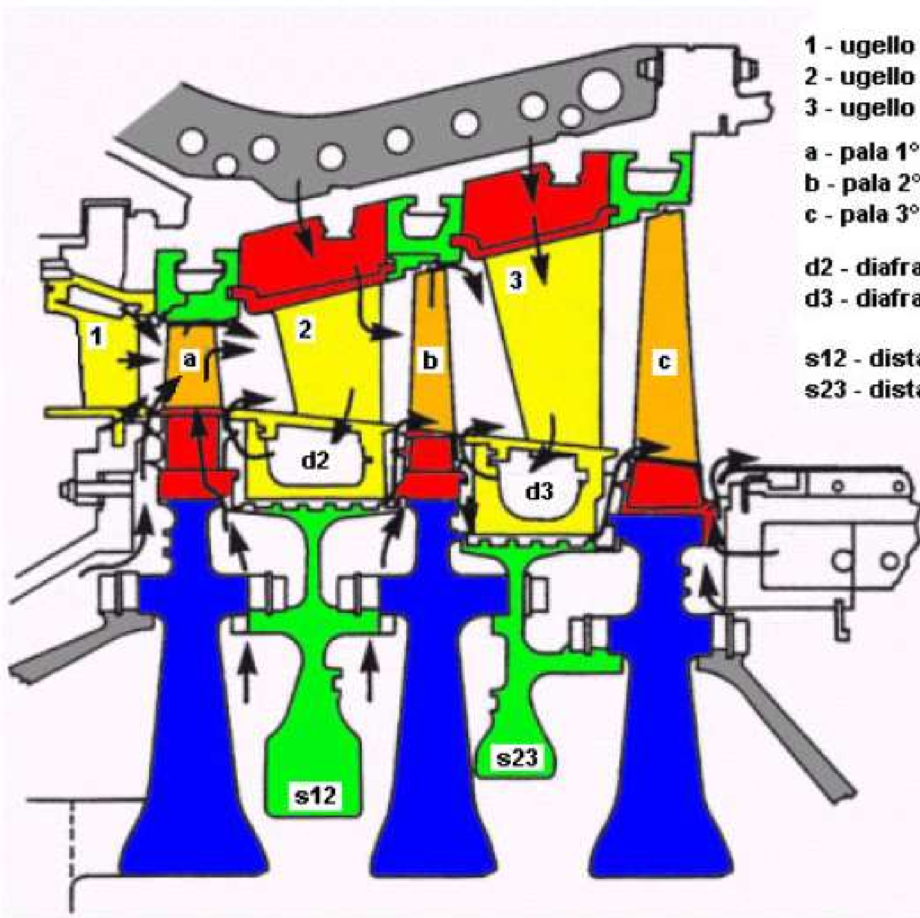


[Lozza]

Design features – Expander



Design features – Expander

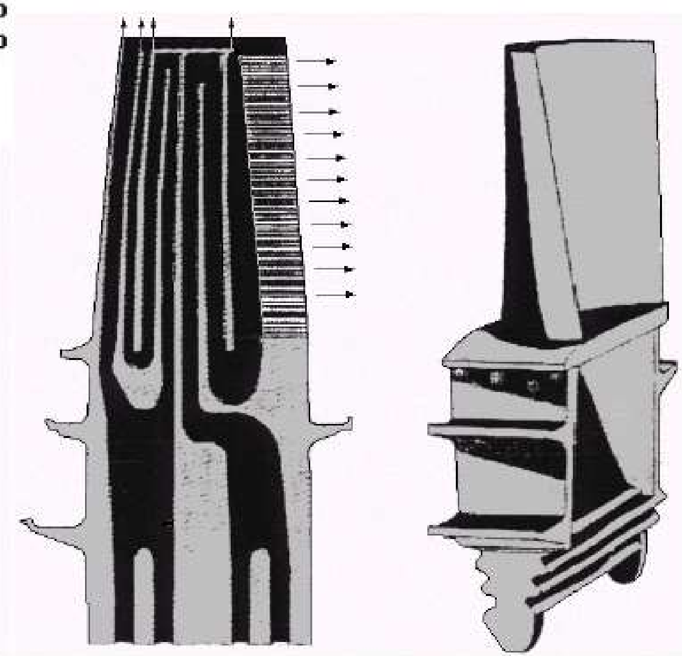


1 - ugello 1° stadio
2 - ugello 2° stadio
3 - ugello 3° stadio

a - pala 1° stadio
b - pala 2° stadio
c - pala 3° stadio

d2 - diaframma 2° stadio
d3 - diaframma 3° stadio

s12 - distanziatore 1-2
s23 - distanziatore 2-3



Blade design

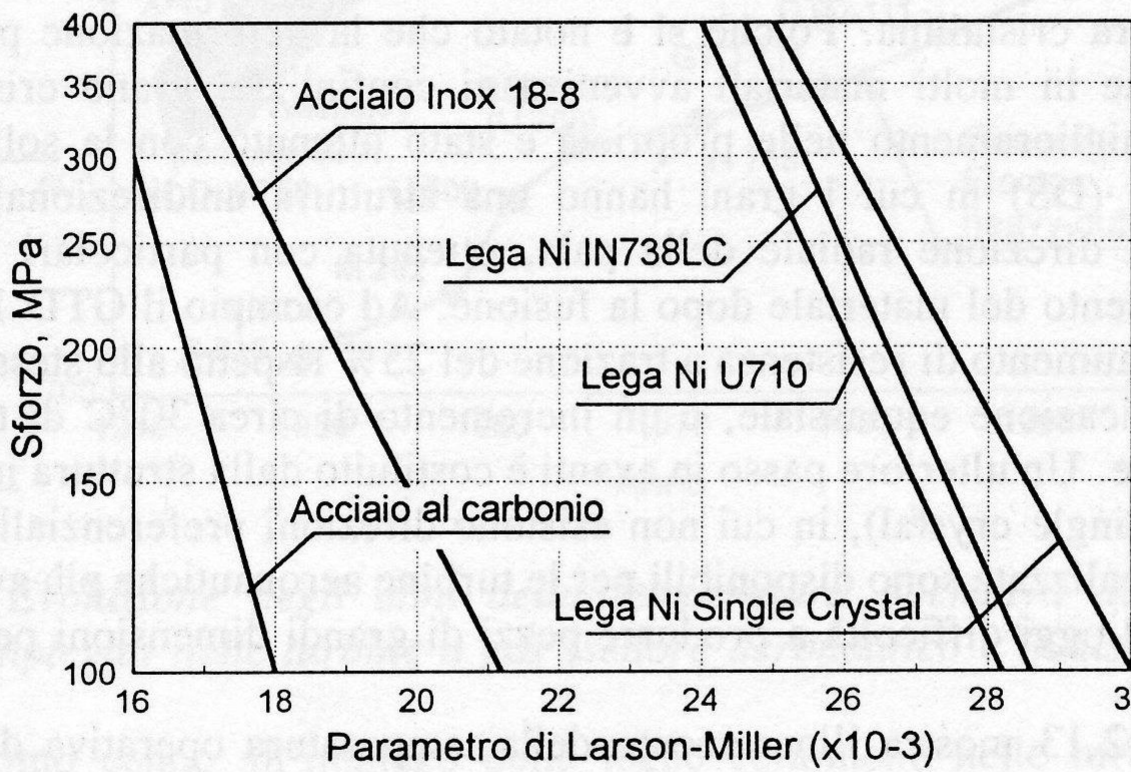
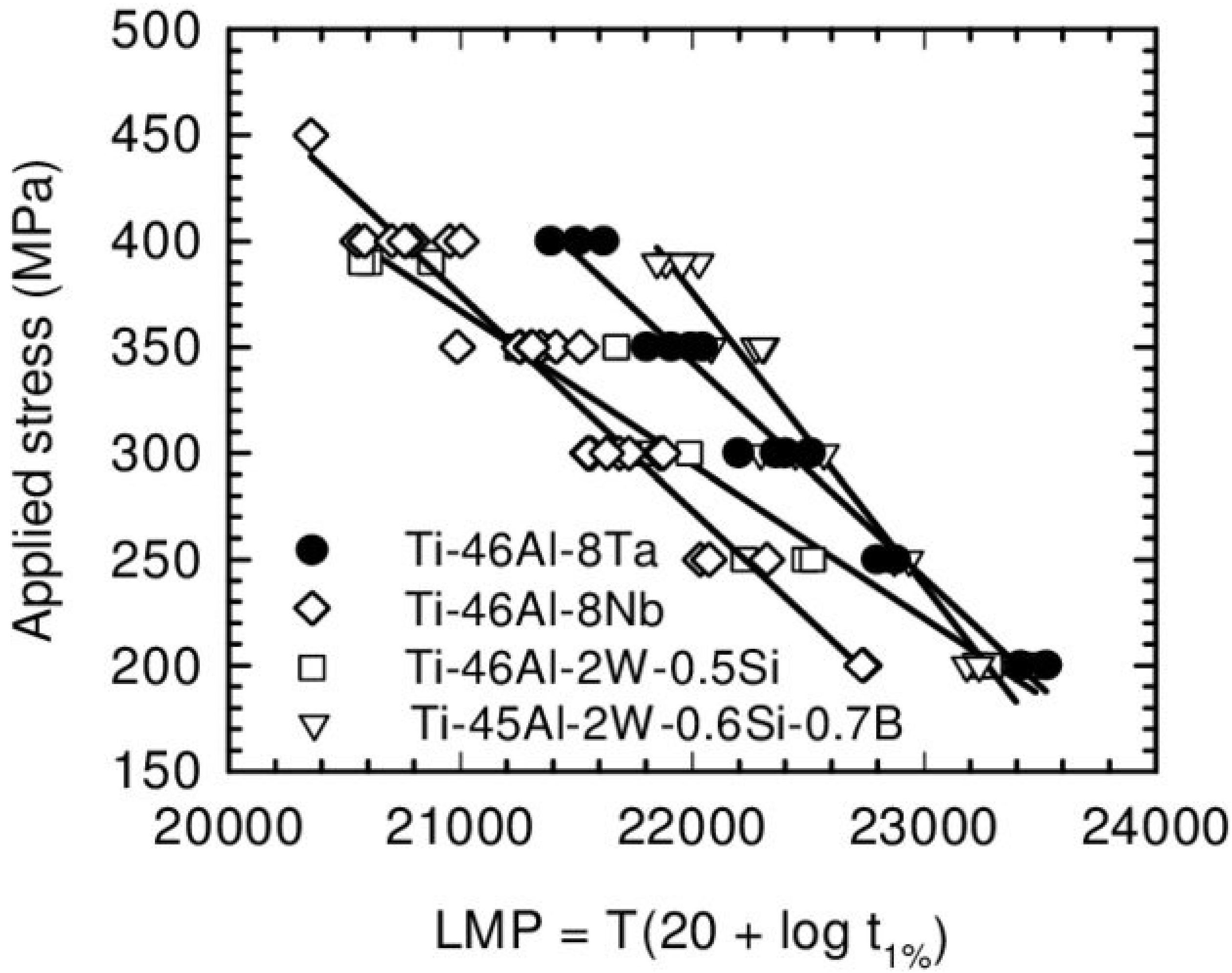


Fig.2.12: *Curve di Larson-Miller per acciai convenzionali e per materiali avanzati impiegati nelle turbine a gas.*

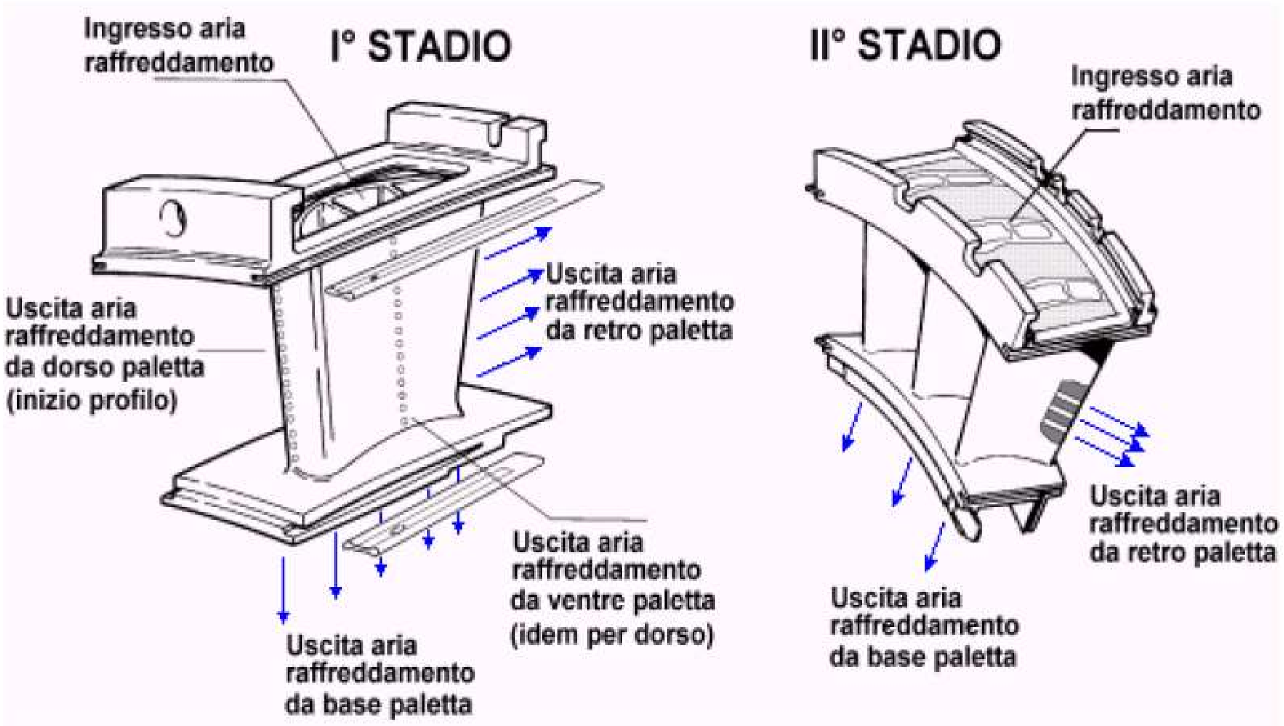
[Lozza]

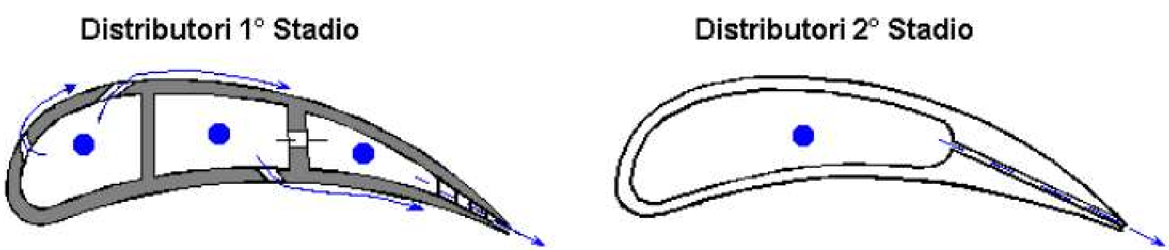
Blade design



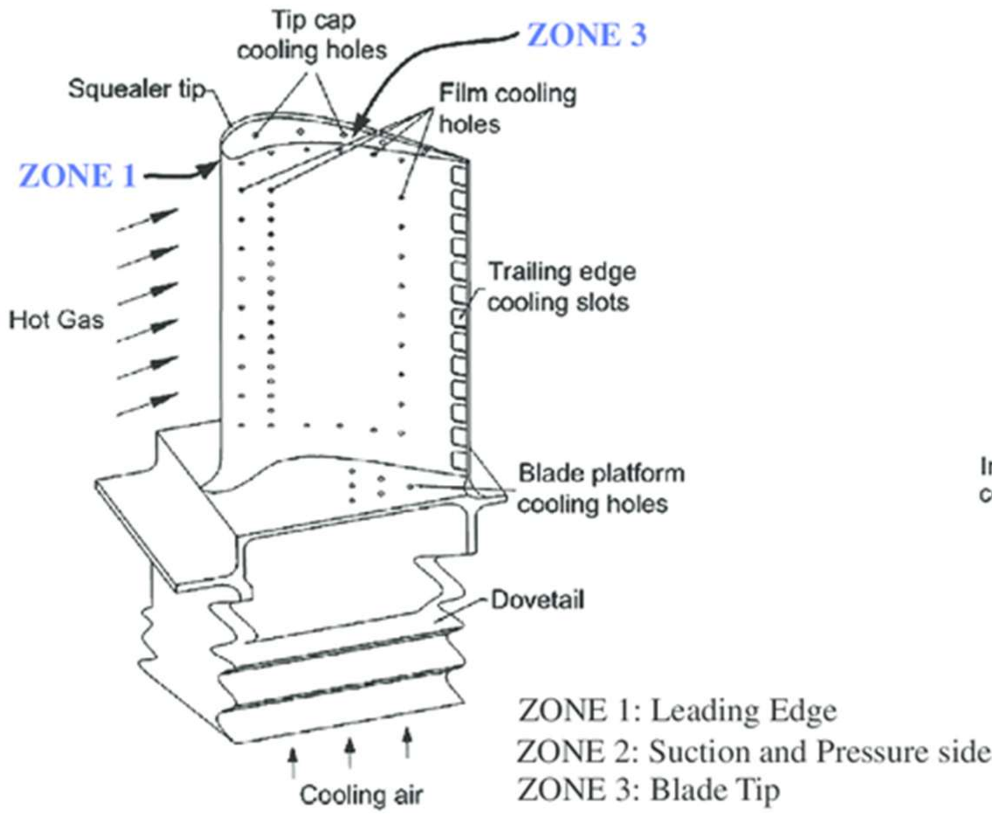
[METAL 2009 19. – 21. 5. 2009, Hradec, Moravicí]

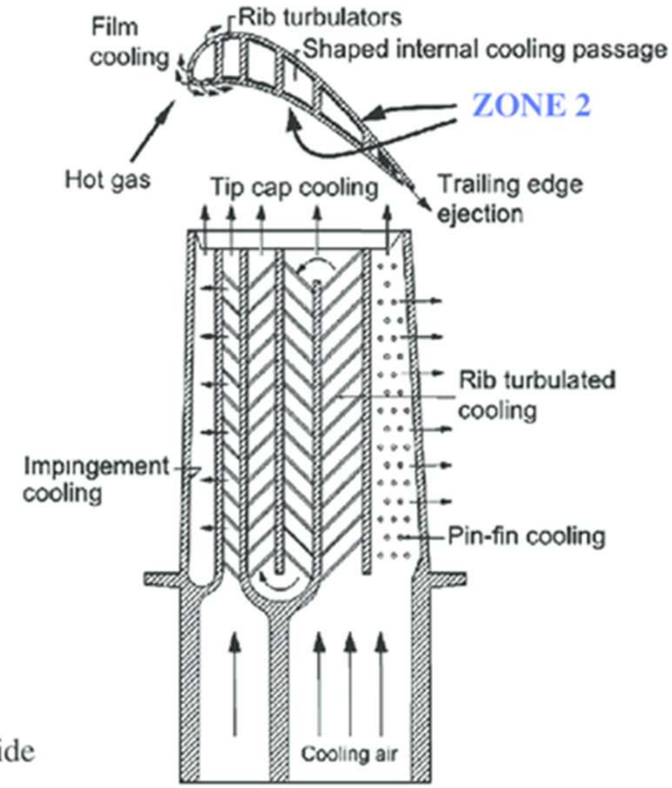
Design features – Expander





Design features – Expander

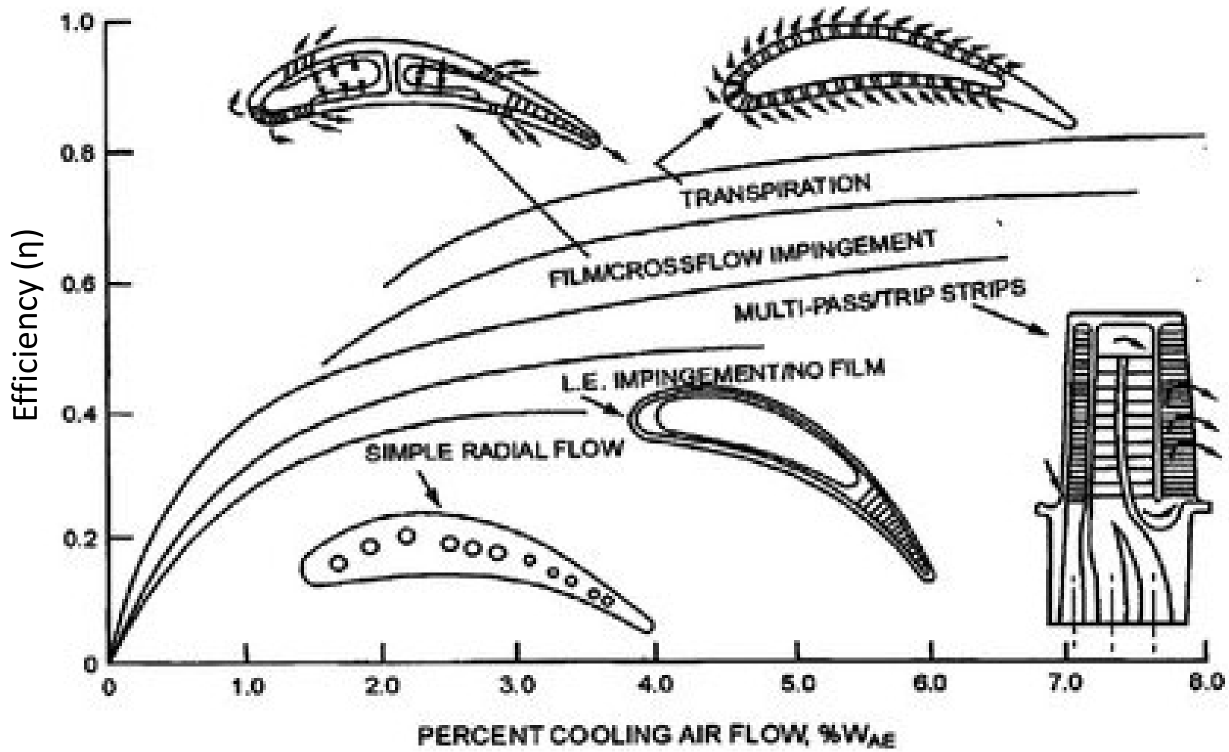




[10.1051/smdo/2014001]

Design features – Expander

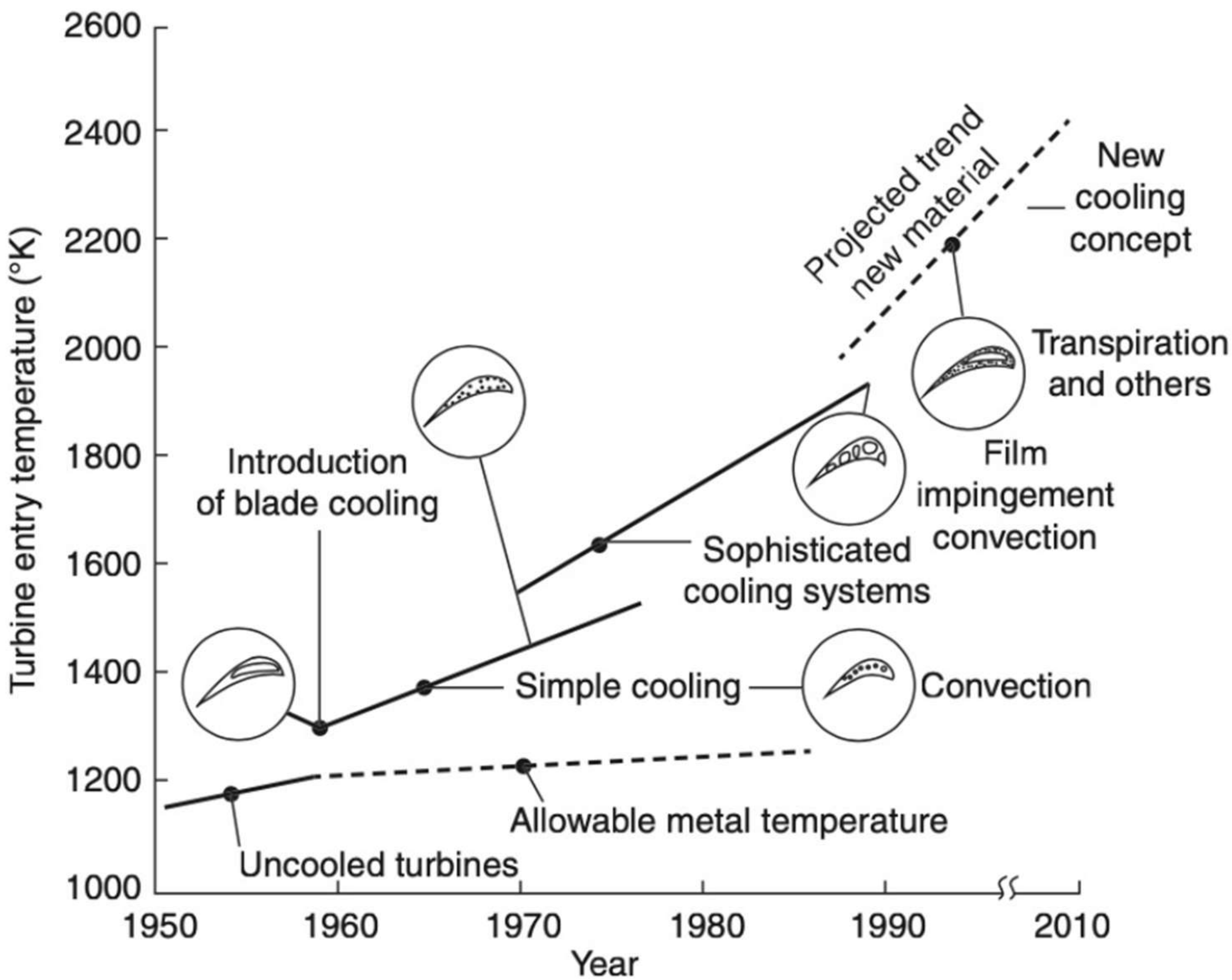
Cooling effectiveness of different blade cooling design vs. cooling air flow



[10.1051/smdo/2014001]

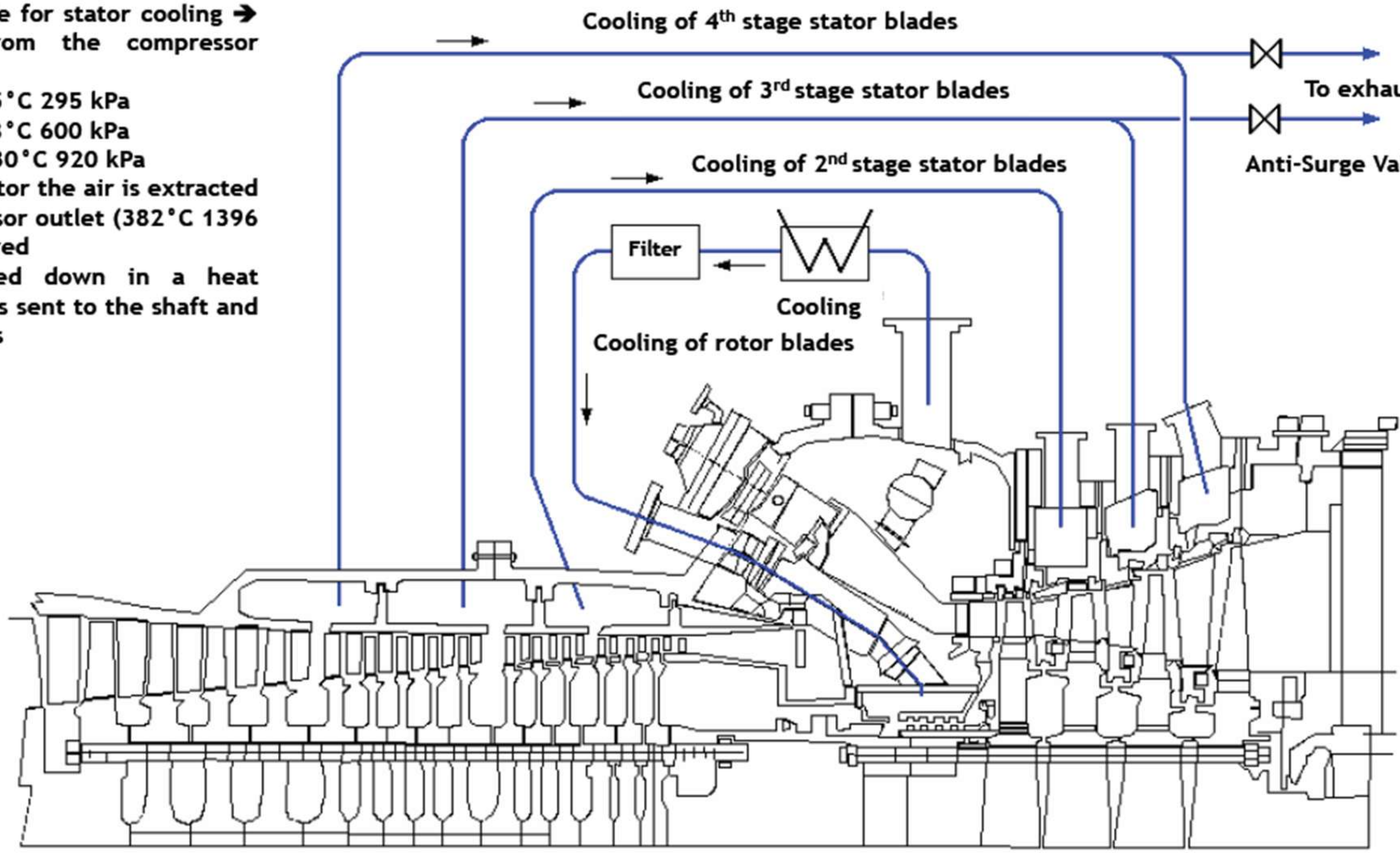
Design features – Expander

Aero-engine cooling technology

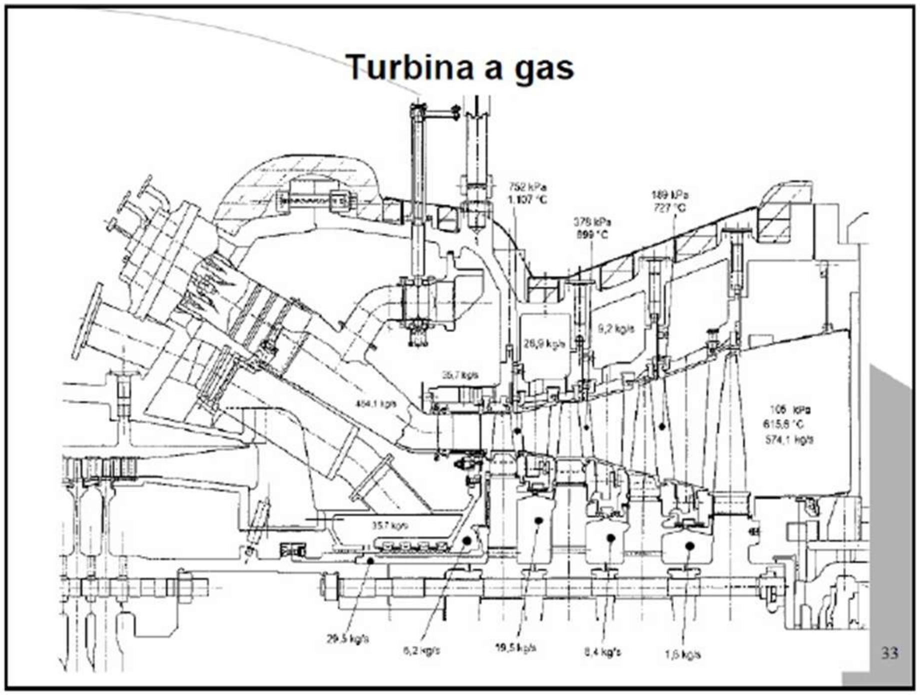


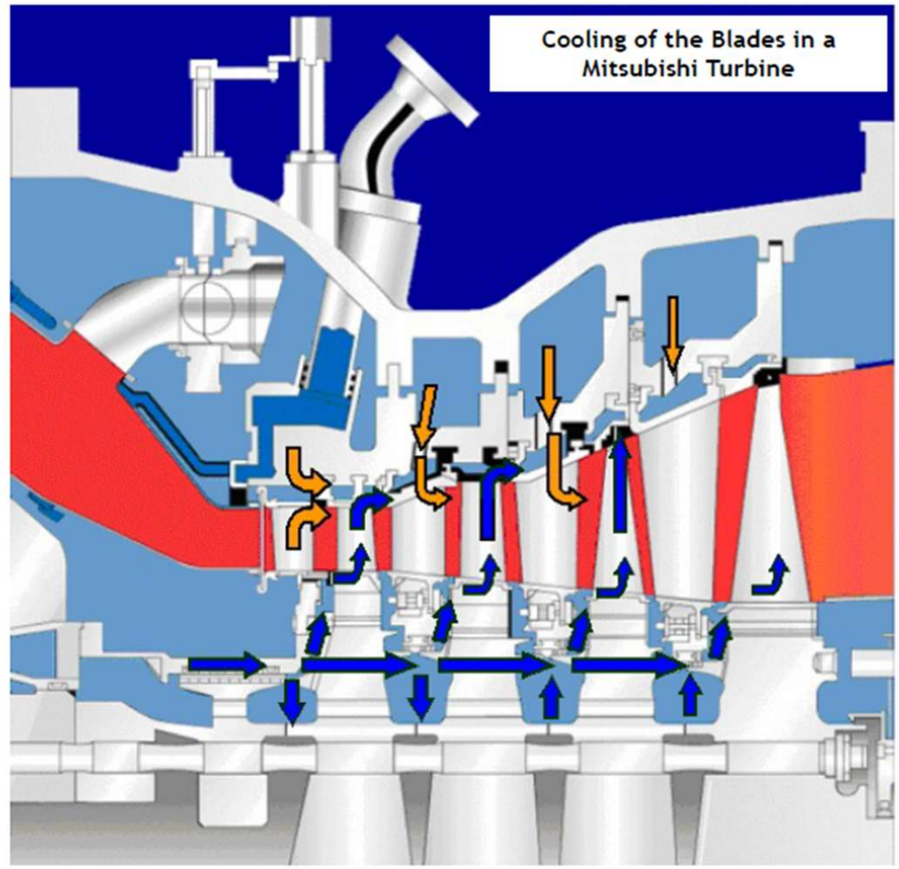
Design features – Expander

- Distinction Nozzle for stator cooling → air extracted from the compressor without filters:
    - VI Stage 155°C 295 kPa
    - IX Stage 258°C 600 kPa
    - XiV Stage 330°C 920 kPa
- For the turbine rotor the air is extracted from the compressor outlet (382°C 1396 kPa) and it is filtered
- The air is cooled down in a heat exchanger and it is sent to the shaft and to the rotor blades



Design features – Expander





3. Gas turbine performances

Analysis of real cycle

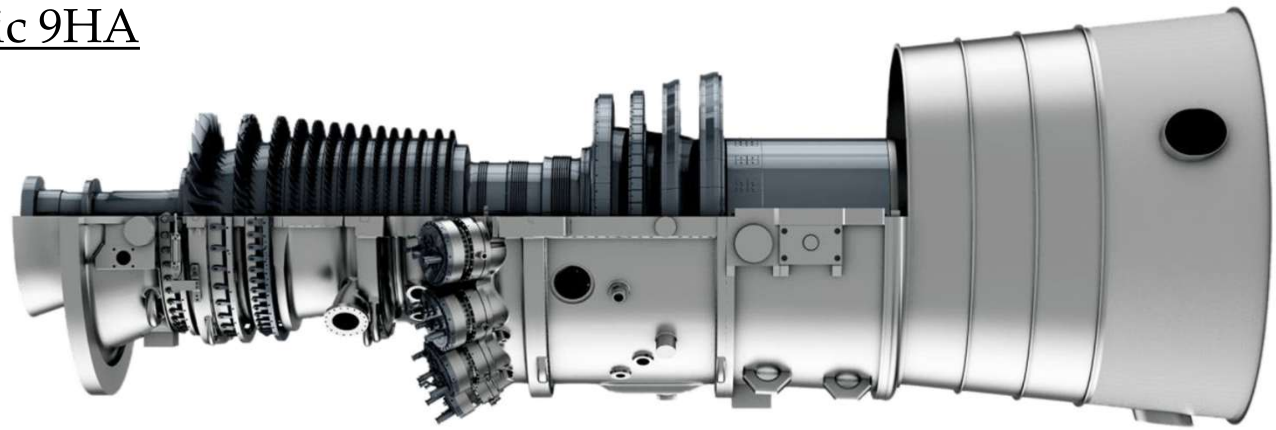
Category	Parameter	Value
Environmental conditions	Temperature and relative humidity	15°C, 60%
	Ambient pressure	101325 Pa
Compressor	Pressure drop, intake filter Δp	1 kPa
	Isentropic enthalpy change per stage Δh_{is}	27 kJ/kg
	Leakage mass at compressor outlet	0.80%
	Polytropic efficiency (SP < 1)	$\eta_p = 0.895 \cdot [1 - 0.07108 \cdot \log_{10}^2(SP)]$
	Polytropic efficiency (SP ≥ 1)	$\eta_p = 0.895$
	Organic efficiency	99.70%
Combustor	Fuel (natural gas)	93% CH_4 - LHV = 44.14 MJ/kg)
	Fuel temperature	15°C
	Fuel pressure	30 bar
	$\Delta p/p$ fuel (min.)	33%
	$\Delta p/p$ air	3%
	Heat losses (% of heat generated)	0.40%
	Total temperature at 1st rotor inlet (TIT)	1280°C

Category	Parameter	Value
Turbine	Isentropic enthalpy change, cooled stages Δh_{is}	300 kJ/kg
	Isentropic enthalpy change, uncooled stages Δh_{is}	100 kJ/kg
	Polytropic efficiency (SP < 1)	$\eta_p = \eta_{p,\infty} \cdot [1 - 0.02688 \cdot \log_{10}^2(SP)]$
	Polytropic efficiency (SP ≥ 1)	$\eta_p = \eta_{p,\infty}$
	$\eta_{p,\infty}$ for cooled stages	0.89
	$\eta_{p,\infty}$ for uncooled stages	0.925
	Polytropic efficiency, 1st nozzle	0.95
	Organic efficiency	0.997
	Max temperature, 1st nozzle vanes	830°C
	Max temperature, other blade rows	800°C
	Average refrigerant $\Delta p/p$	40%
	Axial Mach number at outlet	0.45
	Diffuser efficiency	0.5
	Outlet pressure	1 kPa

The parameter SP used in evaluating η_p is defined as $V^{0.5} / \Delta h_{is}^{0.25}$, where V is the volumetric flow rate at the outlet for the turbine stages and the average volumetric flow rate for the compressor.

Commercial turbines

General Electric 9HA



Simple cycle:

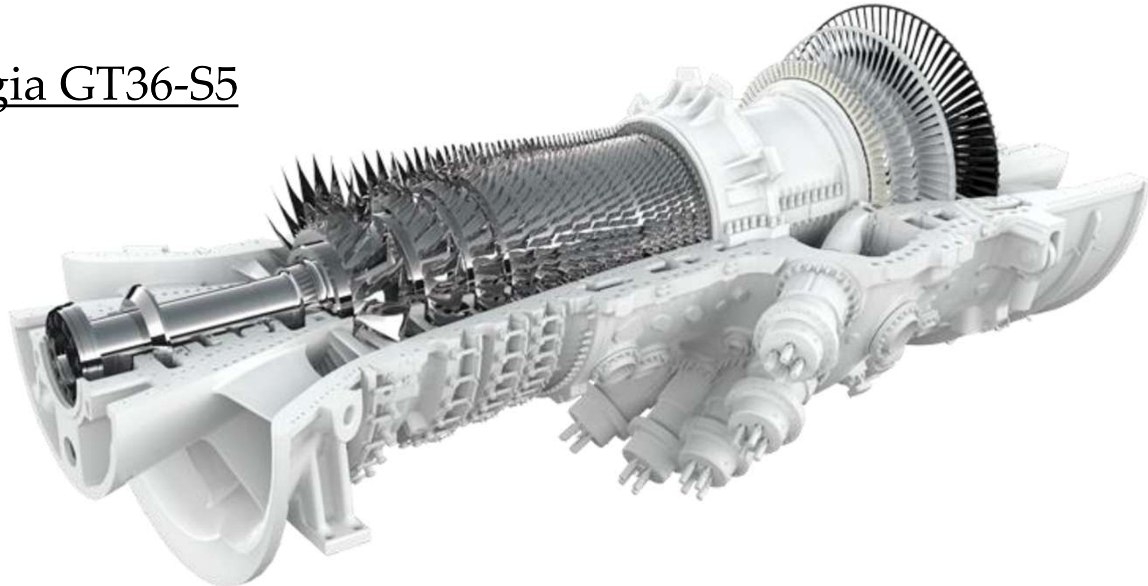
SC Net output (MW)	448
SC Net Heat Rate (kJ/kWh, LHV)	8398
SC Net Efficiency (% LHV)	42.9%
Frequency (Hz)	50

Combined cycle:

CC Net output (MW)	680
CC Net Heat Rate (kJ/kWh, LHV)	5661
CC Net Efficiency (% LHV)	63.7%
Plant Turndown – Minimum Load (%)	33.0%
Ramp Rate (MW/min)	65
Startup Time (RR Hot, Minutes)	<30

Commercial turbines

Ansaldo Energia GT36-S5



Simple cycle:

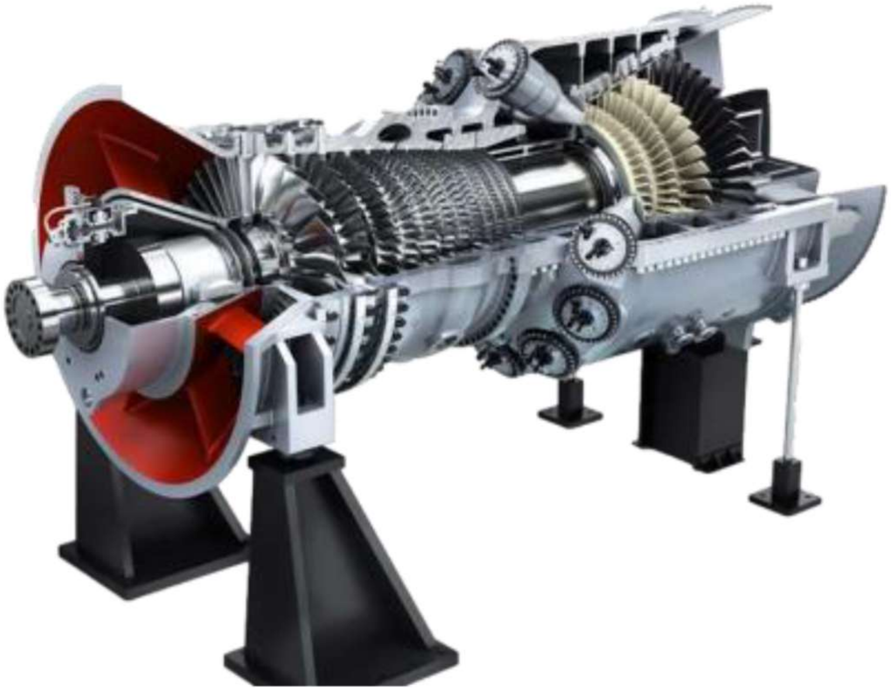
Net Power Output (MW)	538
Frequency (Hz)	50
Efficiency (%, LHV)	42.8%
Exhaust Mass Flow (kg/s)	1020
Exhaust Gas Temperature (°C)	624
NOx Emissions (mg/Nm³)	<50
CO Emissions (mg/Nm³)	<10

Combined cycle:

Net Power Output (MW)	760
Efficiency (%, LHV)	62.6%

Commercial turbines

Siemens SGT5-9000HL



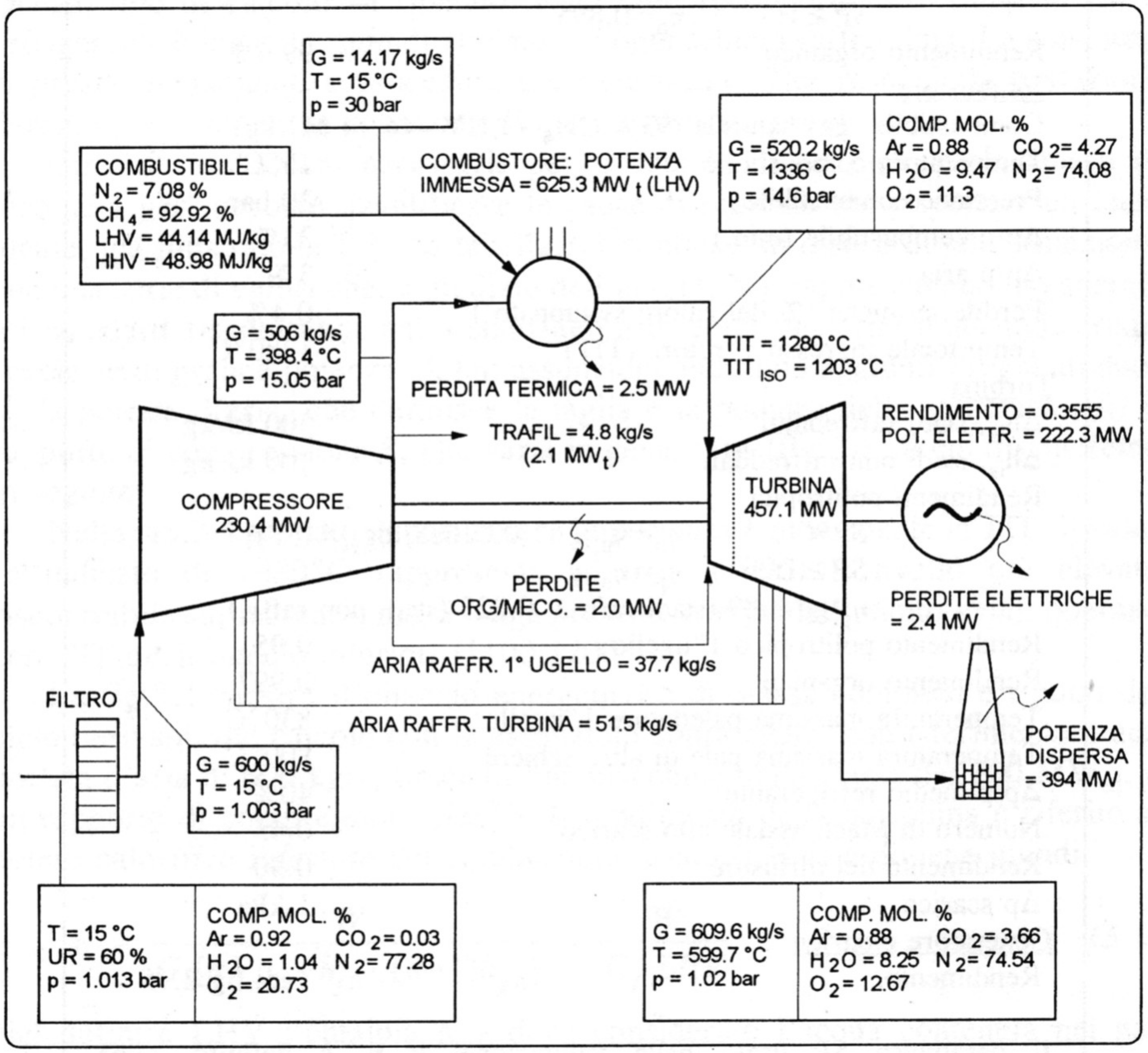
Simple cycle:

Net Power Output (MW)	593
Frequency (Hz)	50
Efficiency (%, LHV)	> 43%
Exhaust Mass Flow (kg/s)	1050
Exhaust Gas Temperature (°C)	670
NOx Emissions (ppmvd)	2
CO Emissions (ppm)	10

Combined cycle:

Net Power Output (MW)	880
Efficiency (%, LHV)	> 64%

Real cycle



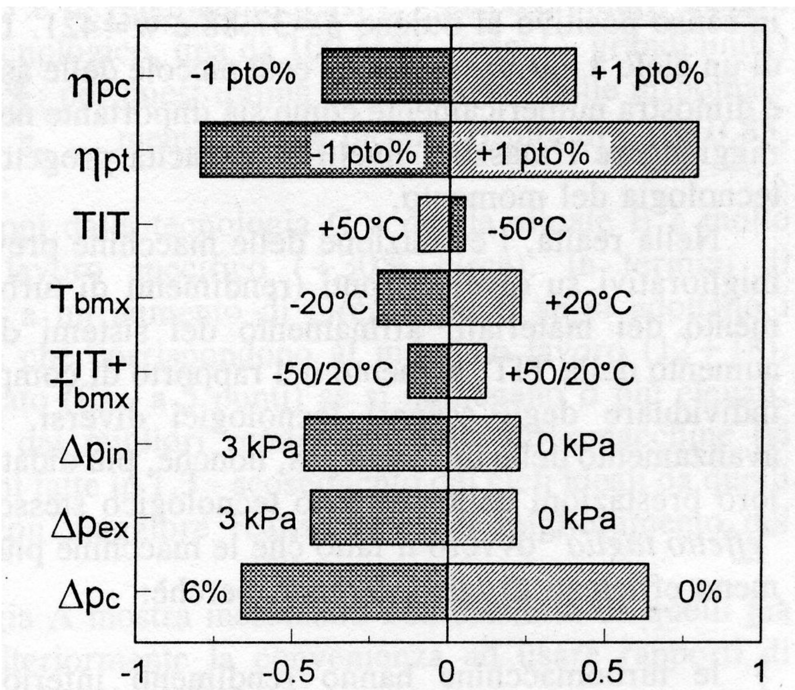
G = 14.17 kg/s
T = 15 °C
p = 30 bar
COMBUSTIBILE
N 2 = 7.08 %
CH 4 = 92.92 %
LHV = 44.14 MJ/kg
HHV = 48.98 MJ/kg
COMBUSTORE: POTENZA IMMESSA = 625.3 MW t (LHV)
G = 520.2 kg/s
T = 1336 °C
p = 14.6 bar
COMP. MOL. %
Ar = 0.88
CO 2 = 4.27
H 2O = 9.47
N 2 = 74.08
O 2 = 11.3
G = 506 kg/s
T = 398.4 °C
p = 15.05 bar
TIT = 1280 °C
TIT ISO = 1203 °C
PERDITA TERMICA = 2.5 MW
TRAFIL = 4.8 kg/s
(2.1 MW t)
RENDIMENTO = 0.3555
POT. ELETTR. = 222.3 MW
COMPRESSORE
230.4 MW
TURBINA
457.1 MW
PERDITE
ORG/MECC. = 2.0 MW
PERDITE ELETTRICHE
= 2.4 MW
ARIA RAFFR. 1° UGELLO = 37.7 kg/s
ARIA RAFFR. TURBINA = 51.5 kg/s
FILTRO
G = 600 kg/s
T = 15 °C
p = 1.003 bar
POTENZA
DISPERSA
= 394 MW
T = 15 °C
UR = 60 %
p = 1.013 bar
COMP. MOL. %
Ar = 0.92
CO 2 = 0.03
H 2O = 1.04
N 2 = 77.28
O 2 = 20.73
G = 609.6 kg/s
T = 599.7 °C
p = 1.02 bar
COMP. MOL. %
Ar = 0.88
CO 2 = 3.66
H 2O = 8.25
N 2 = 74.54
O 2 = 12.67

Observations

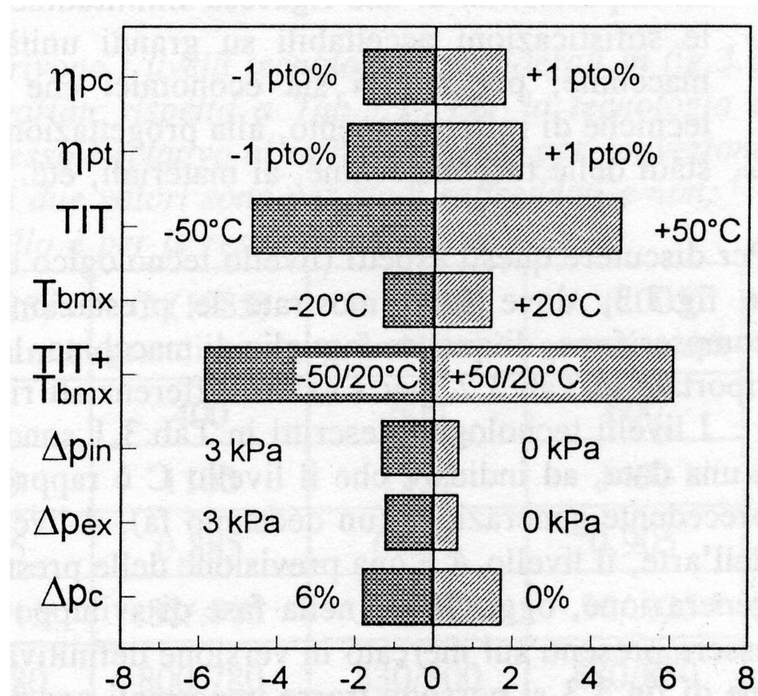
- The power developed by the turbine is approximately double that absorbed by the compressor
- The thermal power developed by the combustor and not converted into useful work is almost entirely contained in the exhaust gases
- Potential heat recovery → for example in combined cycles
- Cooling flow rates make up almost 15% of the air flow rate
- COT=1336°C, TIT=1280°C, TITiso=1203°C
- Exhaust gas 12% O_2

Sensitivity analysis

Sensitivity analysis of efficiency and specific work of the gas turbine to changes in the calculation assumptions shown in the previous table



Efficiency variation (%)



Specific work variation (%)

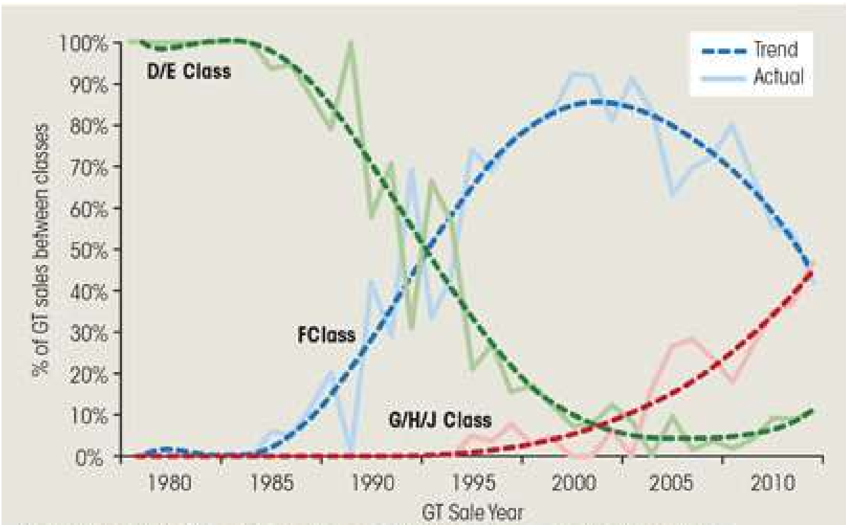
Observations

- Just a 1% variation of eta_c and eta_t has important effects on total efficiency and work
- With the increase of TIT (with the same cooling technology) increases work but reduces efficiency due to higher cooling air flow rates. Differently, if Tbmx increases, the opposite effect occurs
- Intake and exhaust pressure losses are important

If all **negative** effects occur → eta=32.5%, w=318kJ/kg

If all **positive** effects occur → eta=37.8%, w=421kJ/kg

Turbogas - Categories



North Amreica Market Share Evolution between D/E-Class, F-Class, and G/H/J-Class Turbines.

Source: 2014 McCoy Power Report

Gas turbine technology level is commonly identified by letter designation. Historically, gas turbine frame types were defined by

- ➢ Output power,
- ➢ Firing temperatures
- ➢ Compression Ratio

• **D-E-class** is in 75–110 MW range. Products include:

  - GE's 7E.03
  - Siemens SGT6-2000E
  - Mitsubishi Hitachi's H-100

• **F-class** is in 170-230 MW range. Products include:

  - GE's 7F.03-.05 models
  - Siemen's SGT6-5000F,
  - Mitsubishi Hitachi's M501F.

• **The advanced class turbines G-H-J** is in 275–350 MW range. These include:

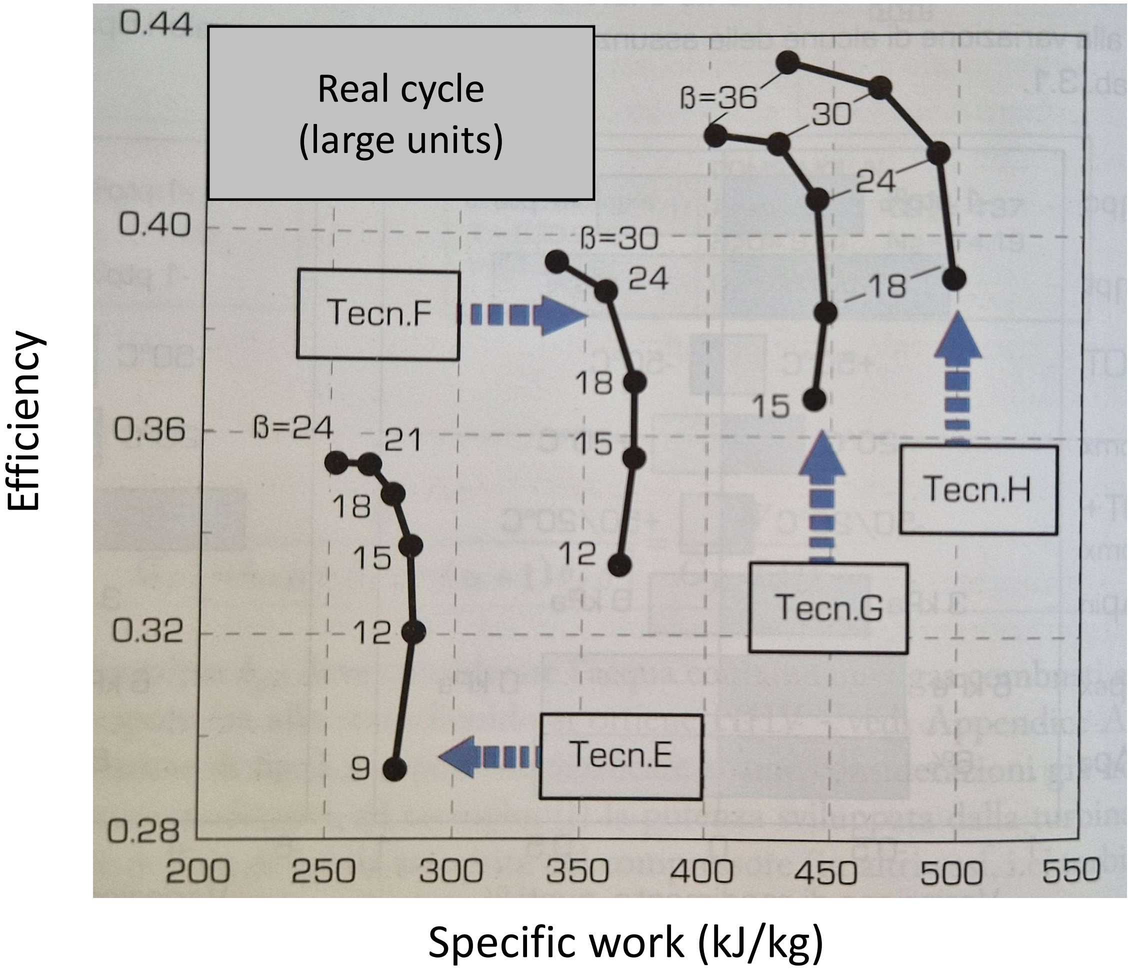
  - Mitsubishi Hitachi's M501J and M501G machines,
  - Siemens SGT6-8000H
  - GE's 7HA.01 and .02.

Assumptions for real cycles analysis

Technology	Unit	E	F	G	H
Air flow rate	kg/s	400	600	700	800
TIT	°C	1100	1280	1400	1600
Compr. efficiency	%	88.5	89.5	90.5	91.5
Turbine efficiency	%	91.5	92.5	92.5	94.0
Metal temperature*	°C	800/780	830/800	880/830	900/850
Film cooling	-	No	1° nozzle	1° stage	1°-2°stage
Thermal Barrier coating	-	No	No	Yes	Yes

*Note: temperature difference between 1° nozzle and last cooled stage

Analysis of real cycle



[Lozza]

Entropy analysis

- Using the I and II Laws of the Thermodynamic, it is allowed to express the efficiency as a function of the losses caused by entropic production in the irreversible processes occurring within the cycle.

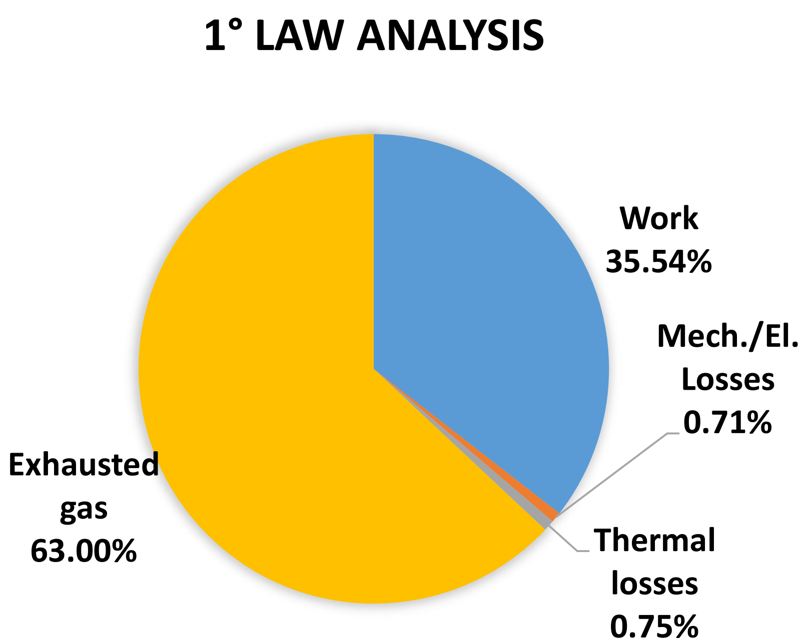
$$\eta_{II} = W/W_{rev} = (Wrev - T_0\Delta S)/W_{rev} = 1 - \Sigma_i \Delta\, \eta_{IIi}$$

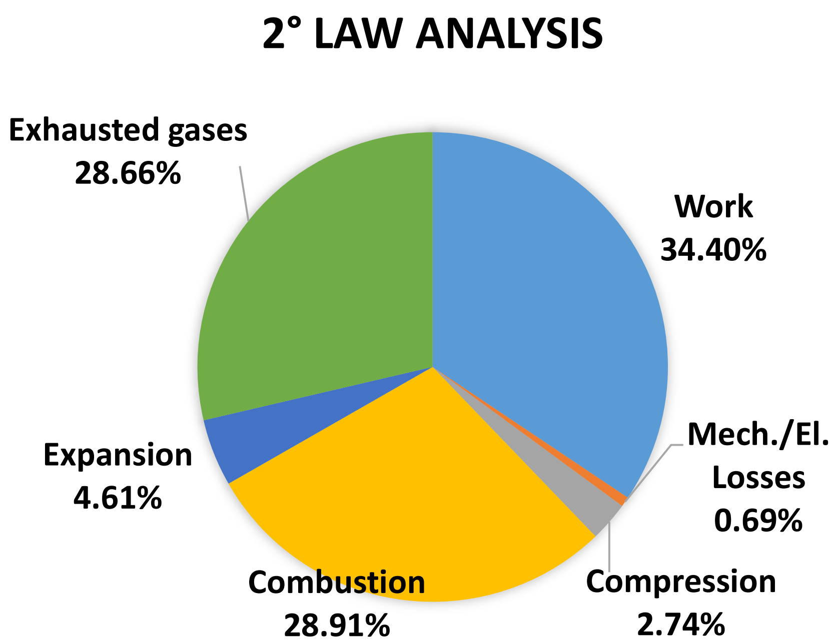
W_rev

For example, assuming to use Methane (LHV= 44137 kJ/kg, HHV= 48978 kJ/kg)

- W_{rev} =45606 kJ/kg (Revesible work of fuels → mixing work contribution is not considered)
- ex_f =46359 [kJ/kg]

Analysis of First and Second Laws





Entropy analysis between cycles with different β

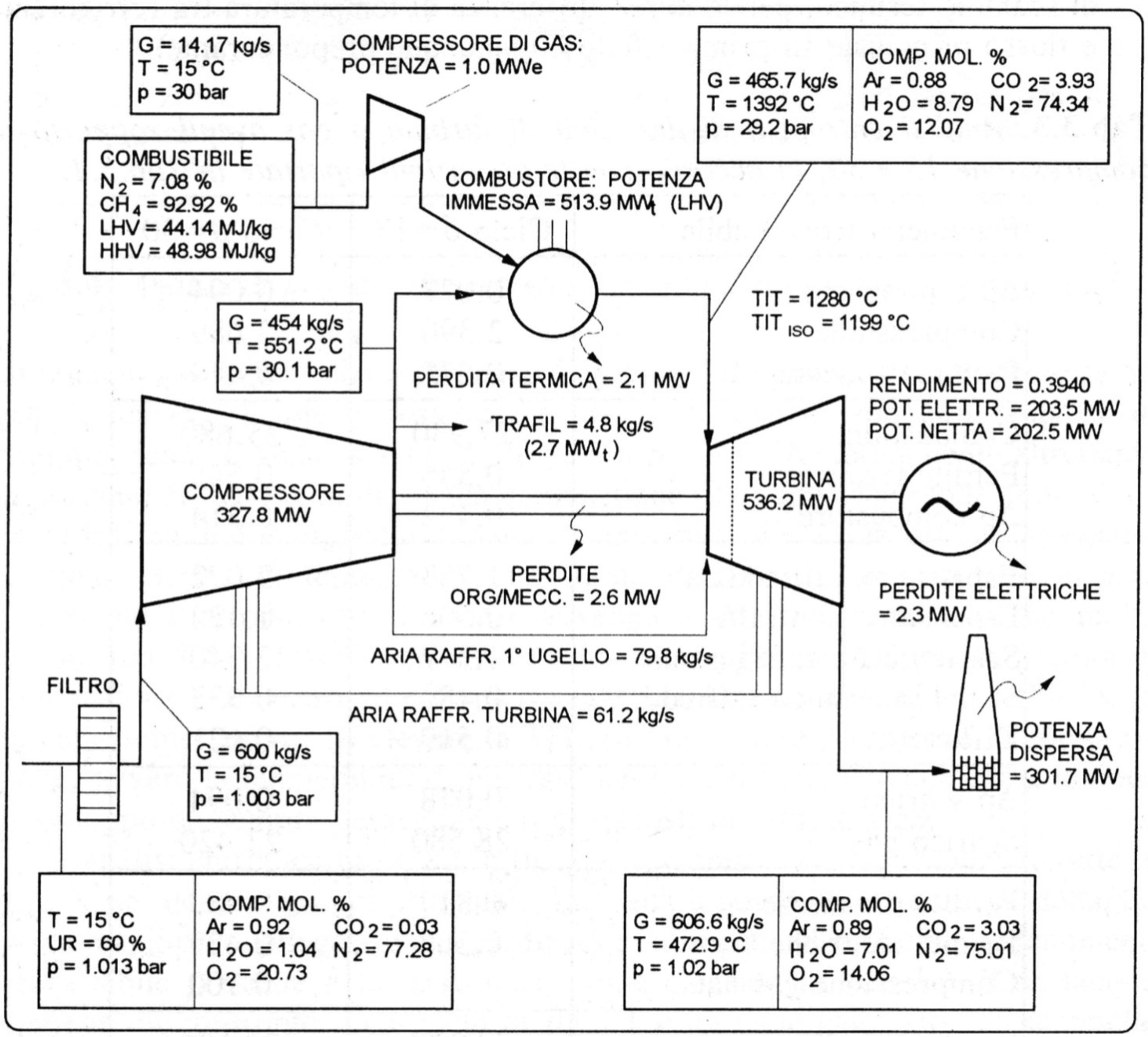
Irreversible phenomenon	Cycle β=15	Cycle β=30
Intake pressure drop Δp	0.077	0.094
Compression	2.39	3.689
Mass loss	0.275	0.479
Combustion	27.93	25.68
Heat losses	0.556	0.545
Combustor pressure drop Δp	0.422	0.314
Cooled expansion	1.753	2.072
Uncooled expansion	0.506	1.172
Exhaust & refrigerant Δp	1.347	2.34
Cooled heat exchange	0.482	0.332
Diffuser	0.519	0.621
Exhaust pressure drop Δp	0.078	0.094
Gas exhaust	28.58	23.42
Organic losses	0.319	0.489
Electrical losses	0.368	0.429
Natural gas compression	-	0.1
Second law efficiency	34.398	38.13

Comparison between I and II Laws – Exhaust leakage

Δη resulted from the entropic analysis:

→ is not an increase of efficiency due to the irreversibility removal, but is proportional to the work losses due to an irreversible transformation

Thermal balance of a GT with β=30



Pag. 90

Comparison between cycles with different β

- Due to the higher temperature of the comburent (551°C vs. 398°C) combustion losses decrease by 2 points
- Due to the lower temperature of the exhausted gases (473°C versus 600°C), relative losses decrease by 5 points
- Compression and expansion losses increase
- Cooling losses change:
  - refrigerant discharge losses of the main flow increase, due to the increase of injected mass
  - heat exchange losses decrease, due to the lower temperature difference between refrigerant and main flow

Market of gas turbines

TG much more compact than Steam–electric power plant. The entire TG system is as large as the steam turbine

Advantages:

- Possibility of installation in limited space.
- Easily transportable.
- Ease of installation (factory-assembled, not site-assembled).
- Short installation times (TG 1y, steam power plant 5y)
- Limited investment costs (TG 350$/kW, steam power plant 1000$/kW)

In the past, there were obstacles to TG deployment related to performance and reliability: these have now been overcome

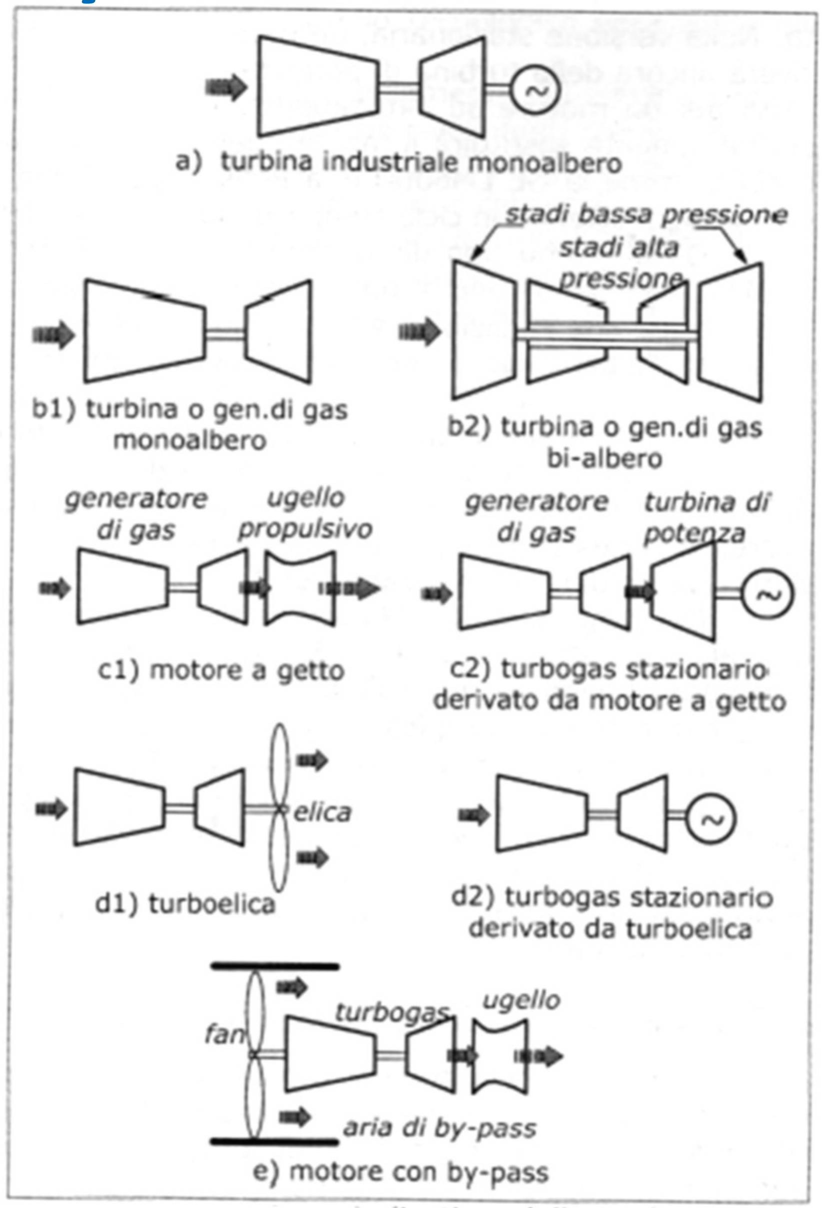
Problem:

➢ Fuel is expensive!

Heavy Duty e Aeroderivative turbines

- Heavy duty or industrial: these are designed and developed exclusively for industrial use and, mainly, for electric power production
- Aero-Derivative: these are derived, with as few modifications as possible, from engines designed and developed for aircraft propulsion

Heavy Duty e Aeroderivative turbines



Heavy Duty e Aeroderivative turbines

HD: characterised by more basic and ‘heavy’ design. Normally single-shaft. Lower than optimal compression ratios:

- To reduce costs, specific work is maximized
- To reduce costs, the number of TM stages is minimized
- High thermal content of exhaust gases is not a problem (CC)

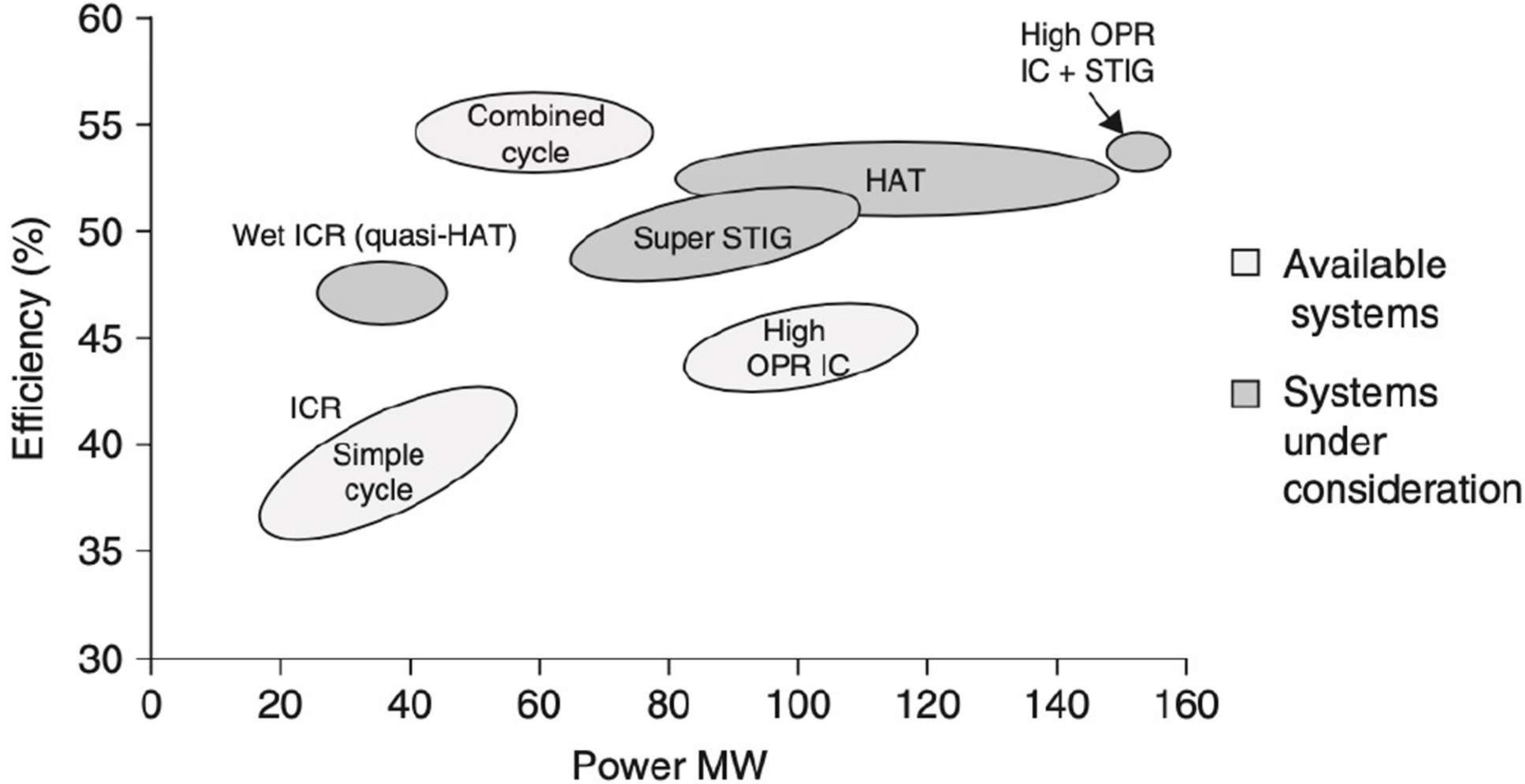
Heavy Duty e Aeroderivative turbines

- AD: lighter (pay load), smaller front section
- Higher efficiency (weight and cost)
- More expensive
- Higher compression ratios and more sophisticated cooling techniques
- Multi-shaft configuration: optimisation for a specific RPM and thus improve efficiency

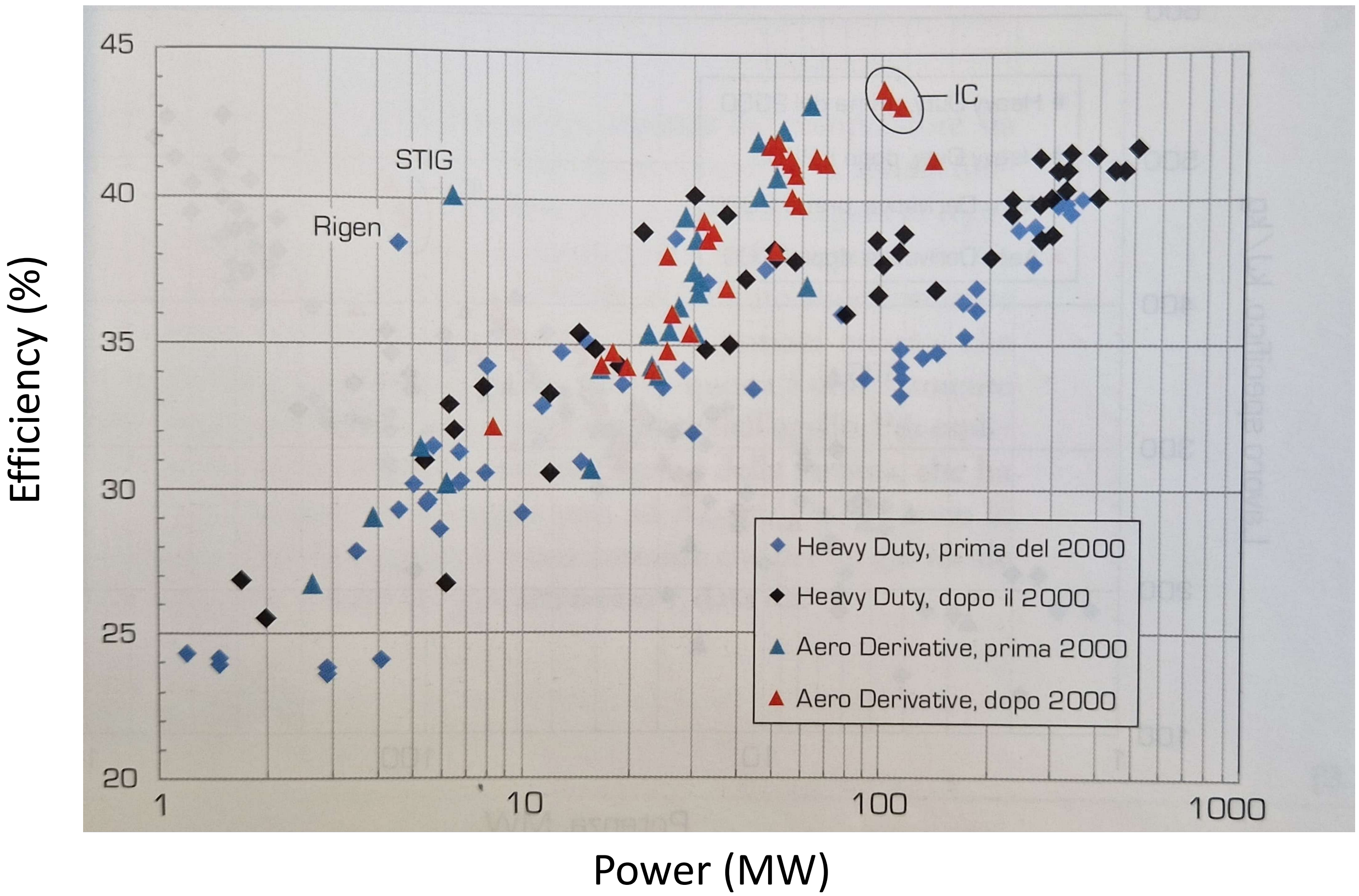
<u>Conversion</u>

- Jet engine - TG does not generate power but only gas: a power turbine must be inserted
- Turbo propeller - the propeller is replaced by a generator

Cycle options for aero-derivative-based GTCC plants



Efficiency



[Lozza]

Operating performance: influence of external conditions

- ISO standards
  - T amb. 15°C
  - P amb. 101325 Pa
  - No intake and exhaust leaks
  - Natural combustion at sufficient pressure
  - New, clean machine

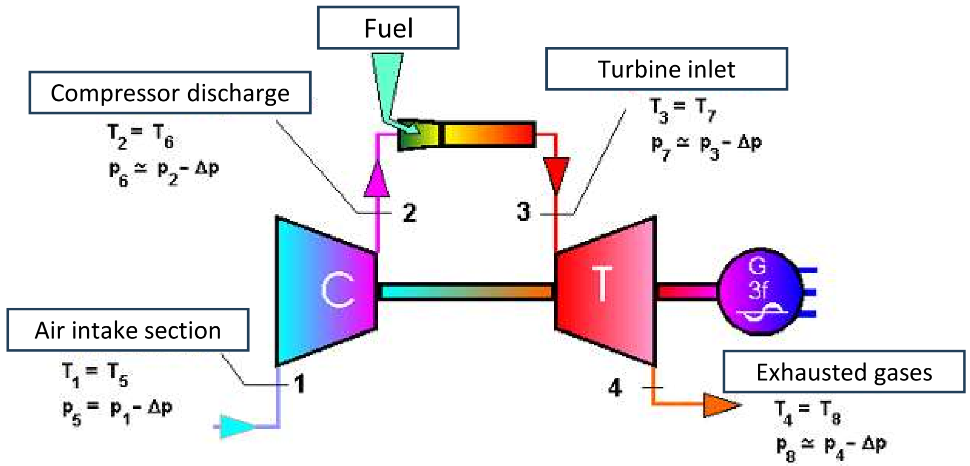
Operating performance: influence of external conditions

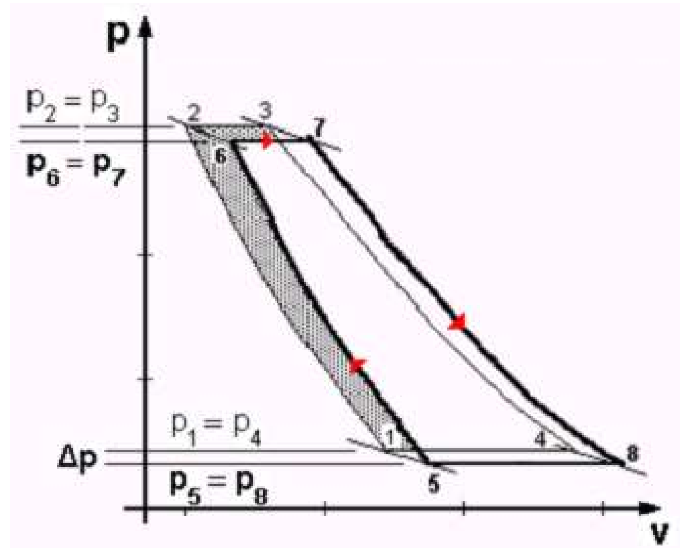
Assumptions:

- $F_{in,c}$ = cost (Volumetric flow rate)
- $F_{in,t}$ = cost (Volumetric flow rate)
- $T_{in,t}$= cost (TIT)

- $G_{in,t} = k^* (p_{in,t} / T_{in,t}^{1/2})$ (Turbine mass flow rate)

Ambient pressure variation

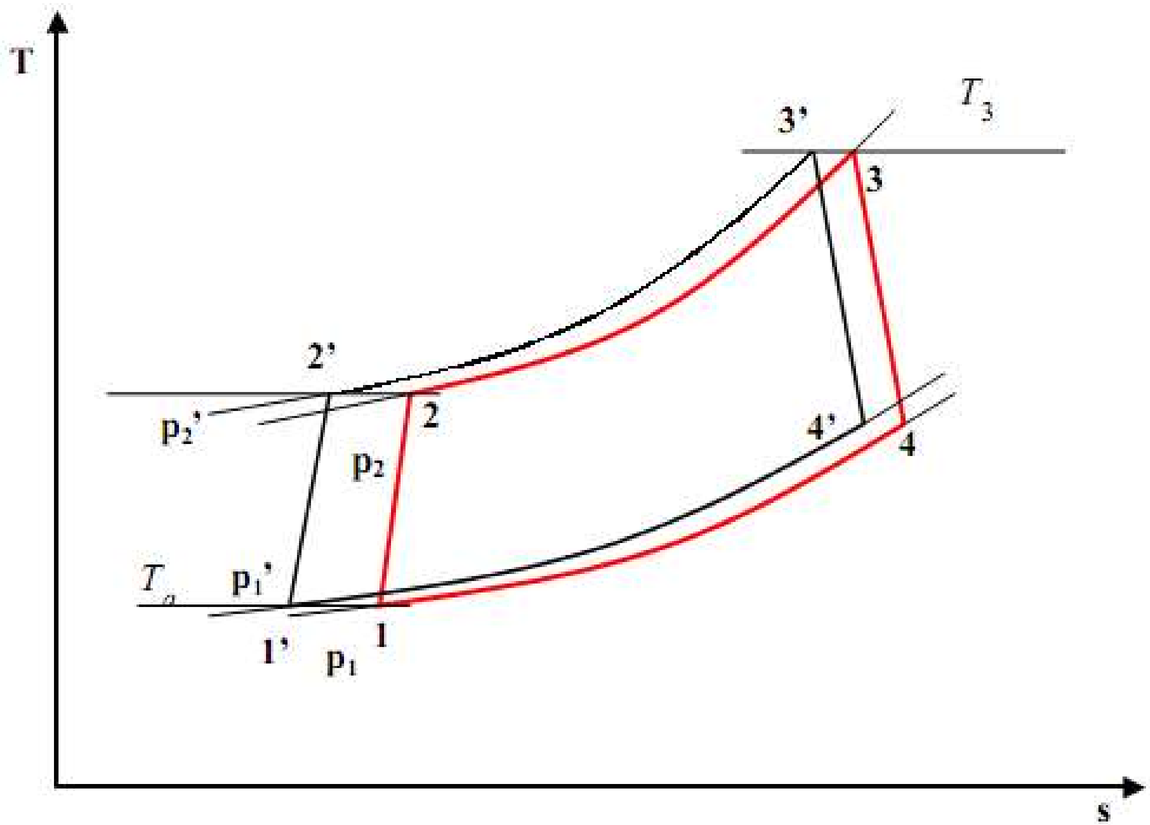




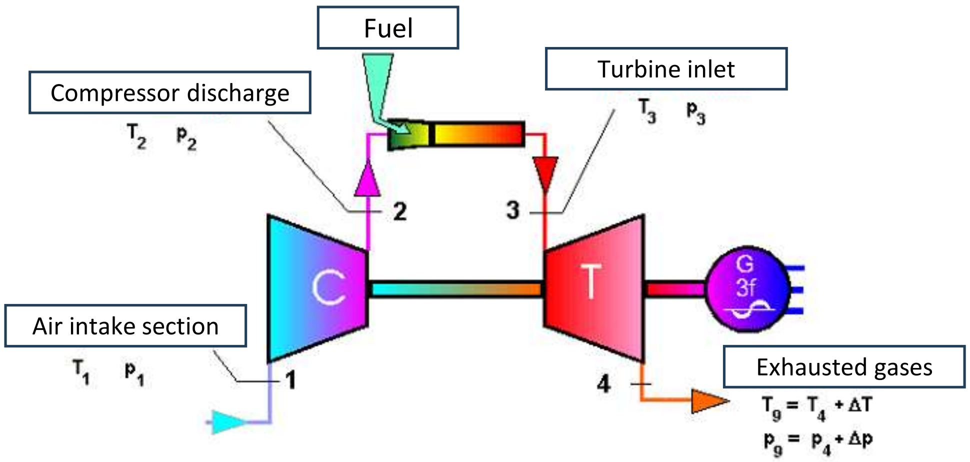
$$F_c = \frac{G_c \cdot R_c \cdot T_1}{p_1} = costant$$

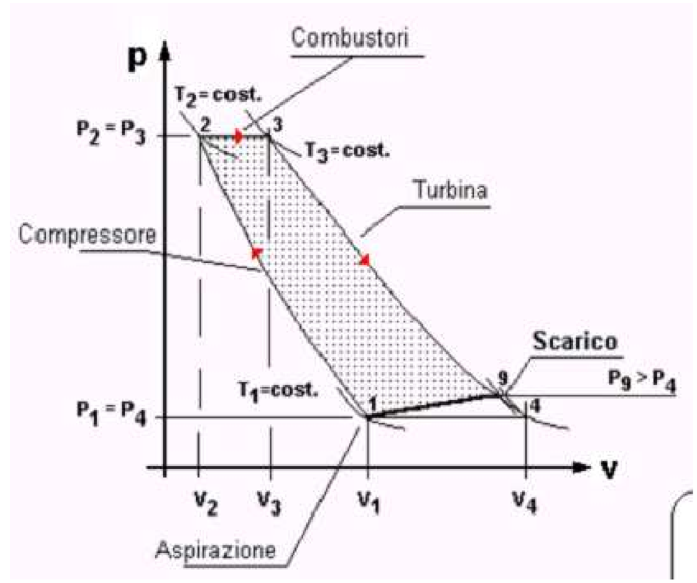
Ambient pressure decreases from p1 to p5 → Air density and air mass flow rate decrease → Power generated decrease $P = P_5 \cdot G_c$

Increase of pressure



Exhaust pressure variation





$$F_c = \frac{G_t \cdot R_t \cdot T_4}{p_4} = costant$$

Discharge pressure increases –> p9>p4 → Temperature at discharge increases T9>T4 → Power generated decrease $P = P_S \cdot G_t$

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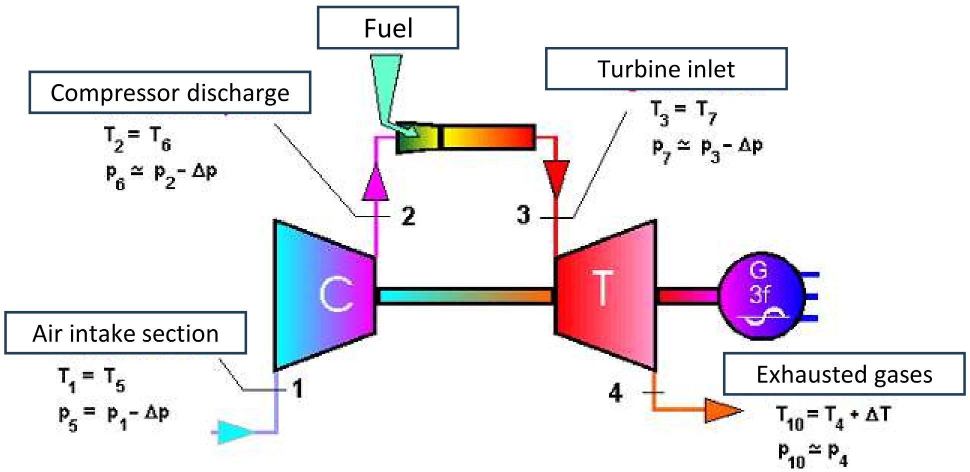
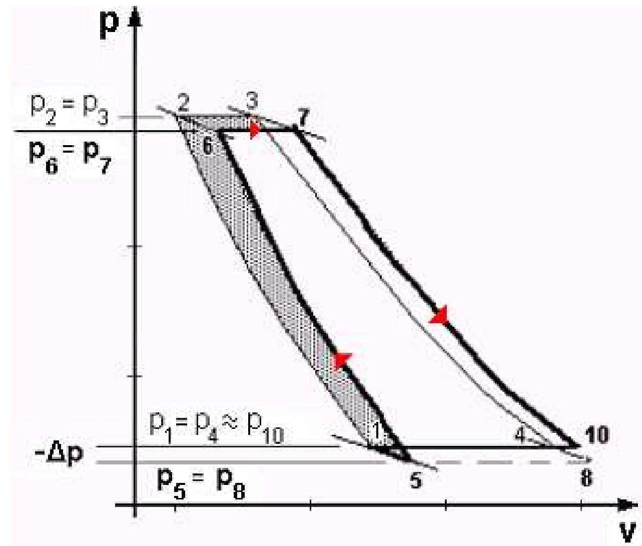
Suction pressure drop

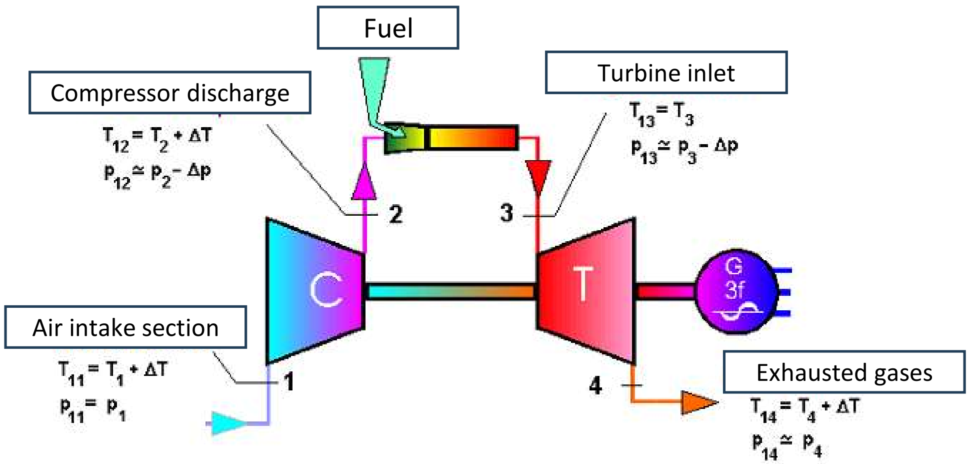


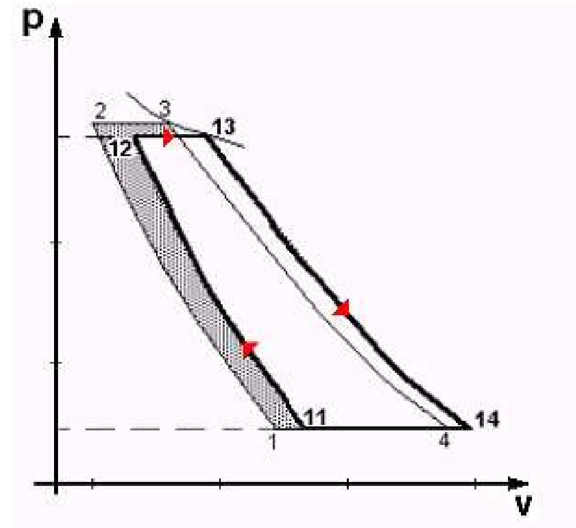


$$F_c = \frac{G_c \cdot R_c \cdot T_1}{p_5} = costant$$

Leaks at air intake section move the cycle starting point from 1 to 5 → Air mass flow rate at compressor decreases → Power generation and efficiency decrease

Air temperature variation

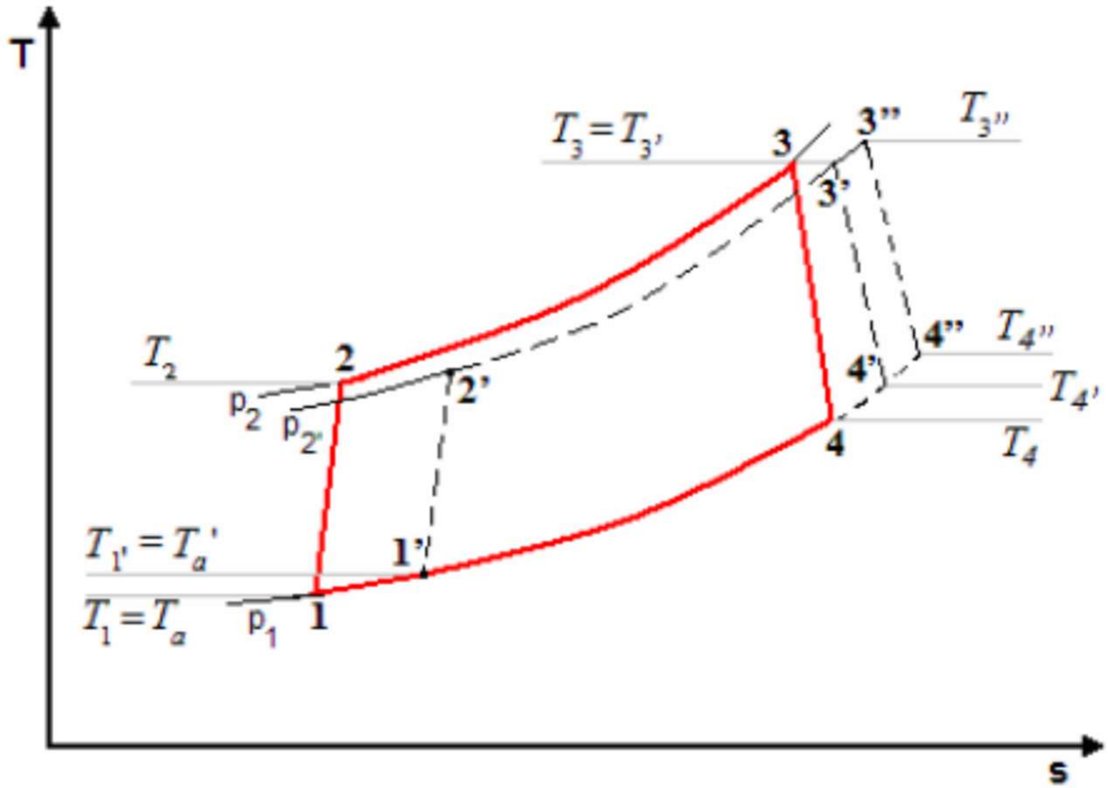




$$F_c = \frac{G_c \cdot R_c \cdot T_1}{p_1} = costant$$

Air temperature increases from T1 to T11 → Air density and air mass flow rate decrease; Temperature inlet turbine is constant but pressure decreases → Power and efficiency decrease

Air temperature influence



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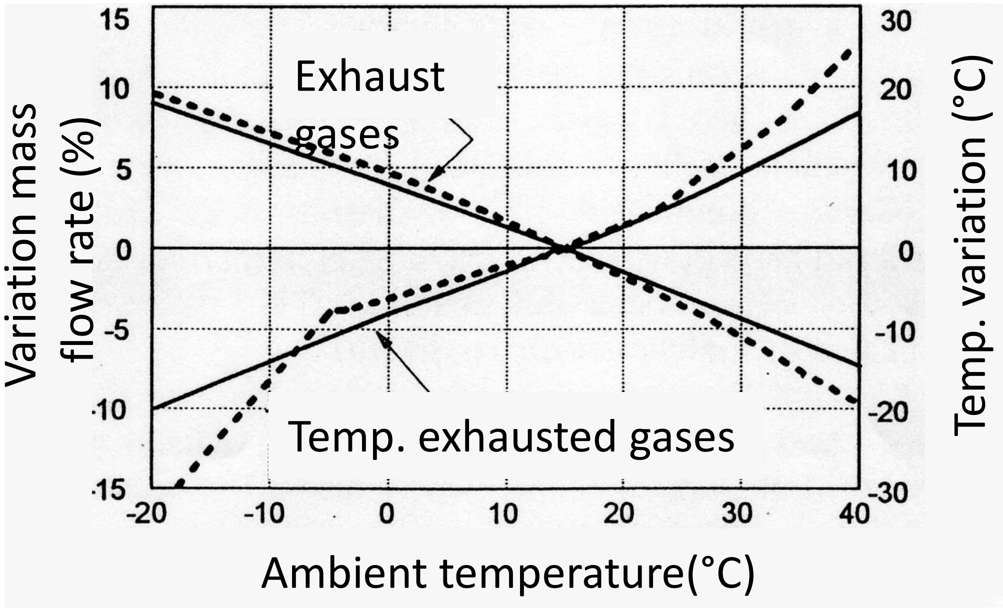
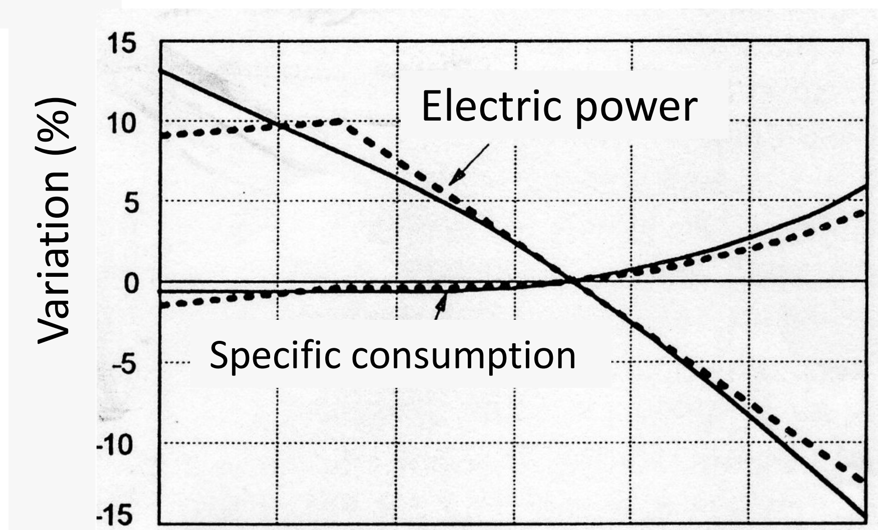
Outside temperature variation





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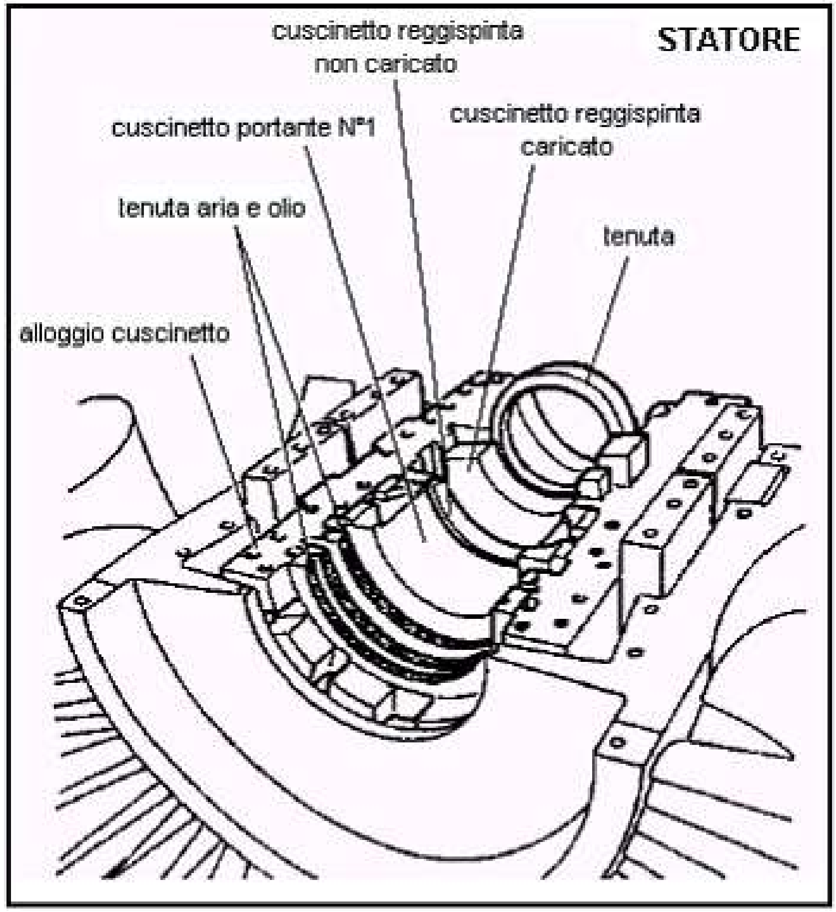
Compressor fouling

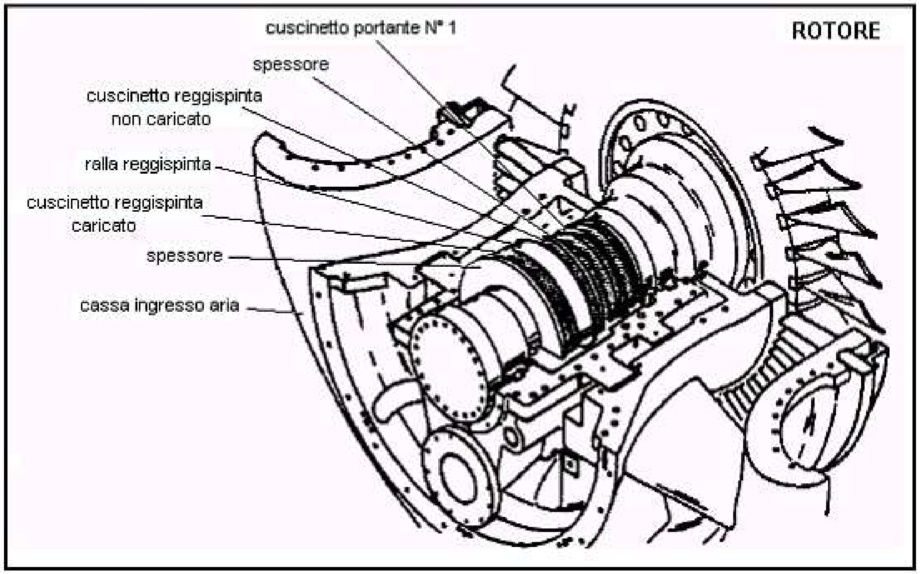
The compressor blade is subject to fouling due to deposits that accumulate during machine operation, consisting of contaminants partly from the external environment and partly from the machine itself (compressor bearing oil vapours).

With regard to oil leaks from the first bearing on the compressor side (bearing No. 1), special attention must be paid to the sealing systems as oil leaks along the shaft are directly sucked in by the compressor.

The oil vapours deposited on the surfaces not only directly cause fouling of the blades, but are also responsible for the retention of dust particles entering the machine.

Compressor fouling





Compressor fouling

The main types of contaminants that are responsible for compressor fouling are related to the environment in which the machine operates, for example:

- hard particles that cause both erosion and fouling (dust, sand, ash, rust, coal dust)
- light particles that cause fouling (soot, oil vapours, pollen, spores, insects)
- inorganic salts.

These contaminants, deposited on the fixed and mobile blades of the compressor, reduce its aerodynamic efficiency and the flow of air sucked in. This ultimately leads to a decrease in the performance of the gas turbine, in particular power and specific consumption.

Compressor fouling

There are now well-established compressor washing techniques, which can be carried out either with the machine off parallel at crank speed (off-line washing) or with the machine in parallel (on-line washing), with which the operator can intervene to counter compressor fouling and restore normal gas turbine efficiency without needing to open the machine.

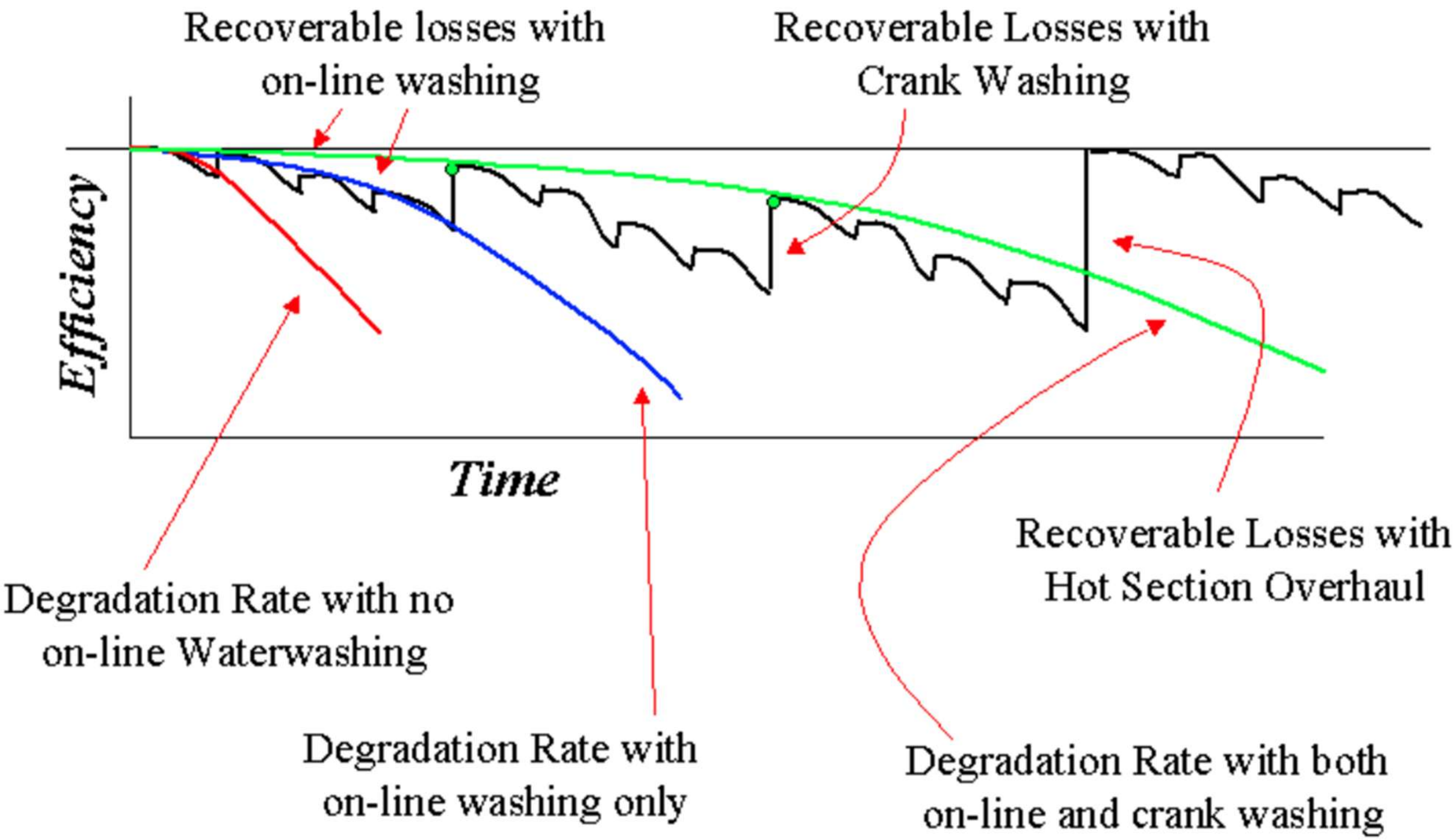
- Off-line washing, which takes place with the machine switched off and dragged by the launch motor, results in low air and wash solution velocities and achieves excellent degrees of cleanliness, sometimes up to 100% efficiency.
- On-line washing, which takes place with the machine running, can never be as thorough as off-line washing for 4 reasons:
    1. with the rotor spinning at 3000 rpm, centrifugal forces will push the washing liquid towards the periphery;
    2. the speed of the air is very high and allows a short residence time of the liquid in the compressor;
    3. turbulence is very high and much of the liquid flows over the walls of the inlet duct before entering the compressor;
    4. compression gives rise to an increase in temperature, which can cause the liquid to start vaporising halfway through the compressor.

Compressor fouling

Finally, there is the risk of erosion damage due to droplet impact on the vanes: this can be avoided by reducing the droplet diameter of the washing liquid by increasing its atomisation pressure (about 70 bar).

In high-pressure washing systems, the nozzles are installed near the compressor inlet and generate a high-speed atomisation: the droplets have about the same speed as air and are therefore more likely to penetrate through the compressor.

Fouling - ageing

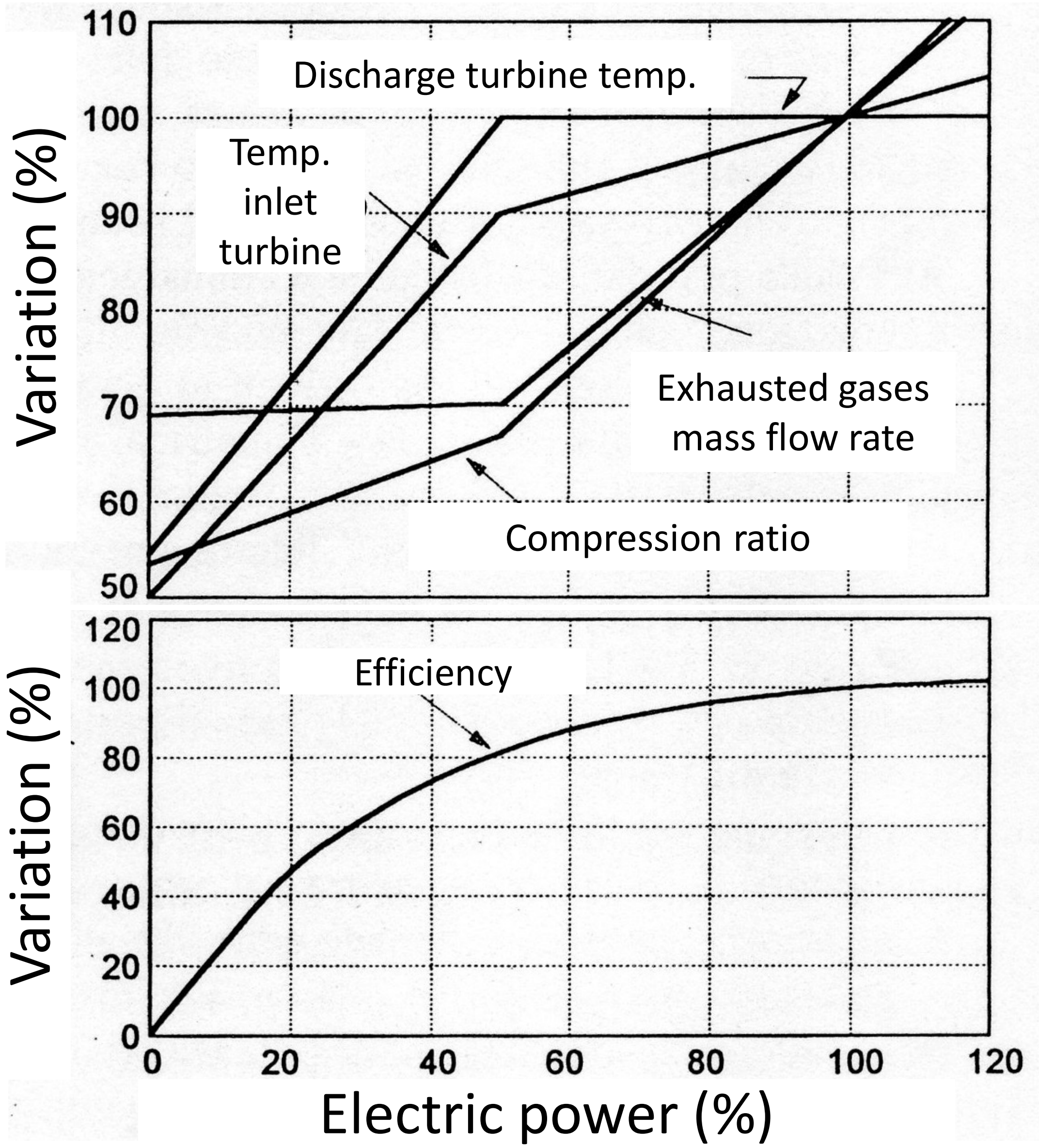


[https://doi.org/10.1115/2001-GT-0218]

Gas turbine regulation

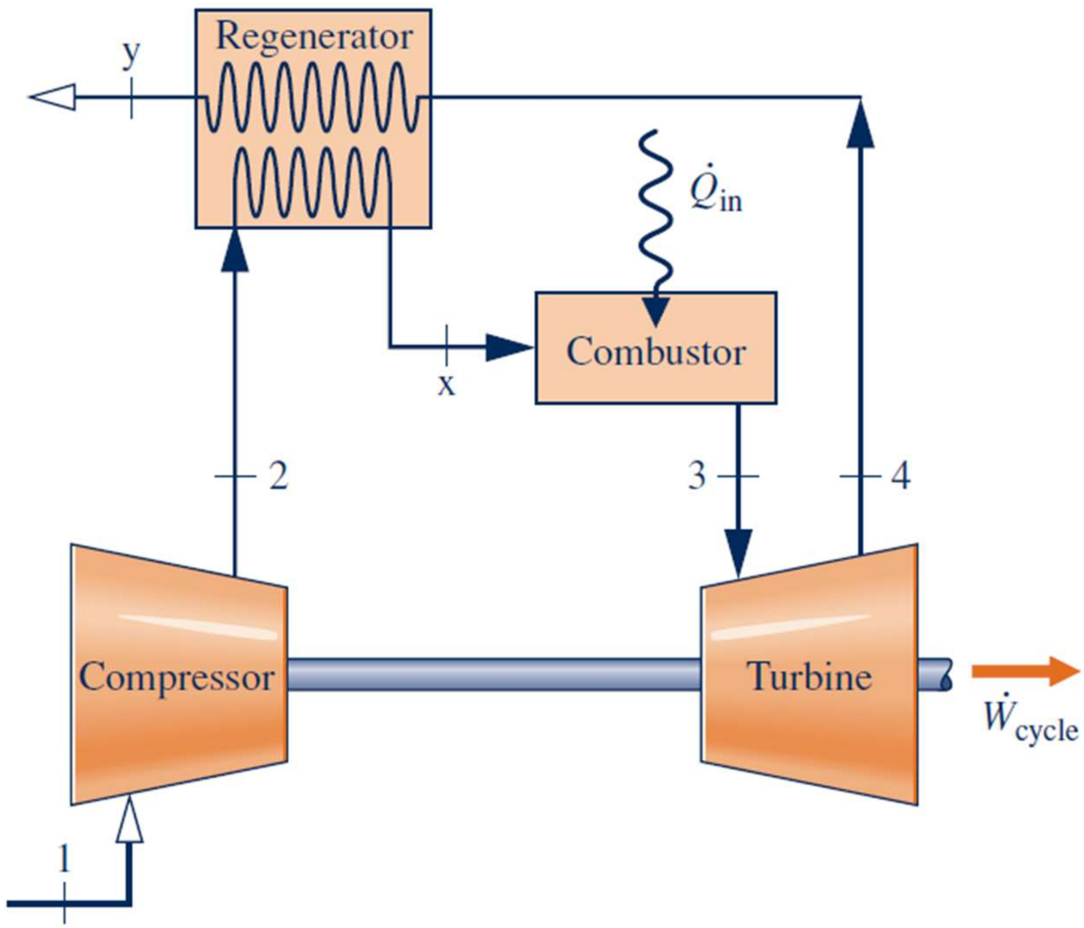
- Regulation by inlet guide vanes (IGV)
- Adjustment with intake lamination valve
- Speed adjustment

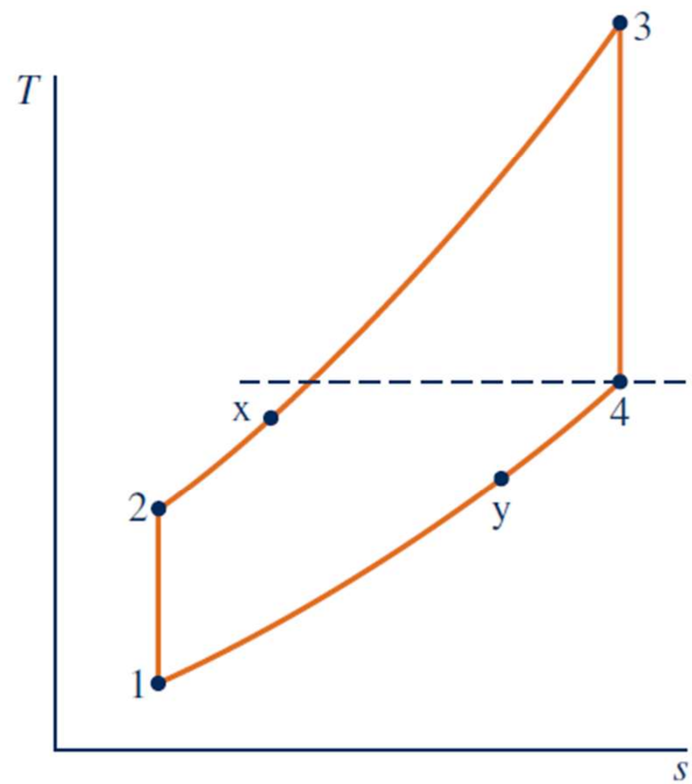
Gas turbine regulation



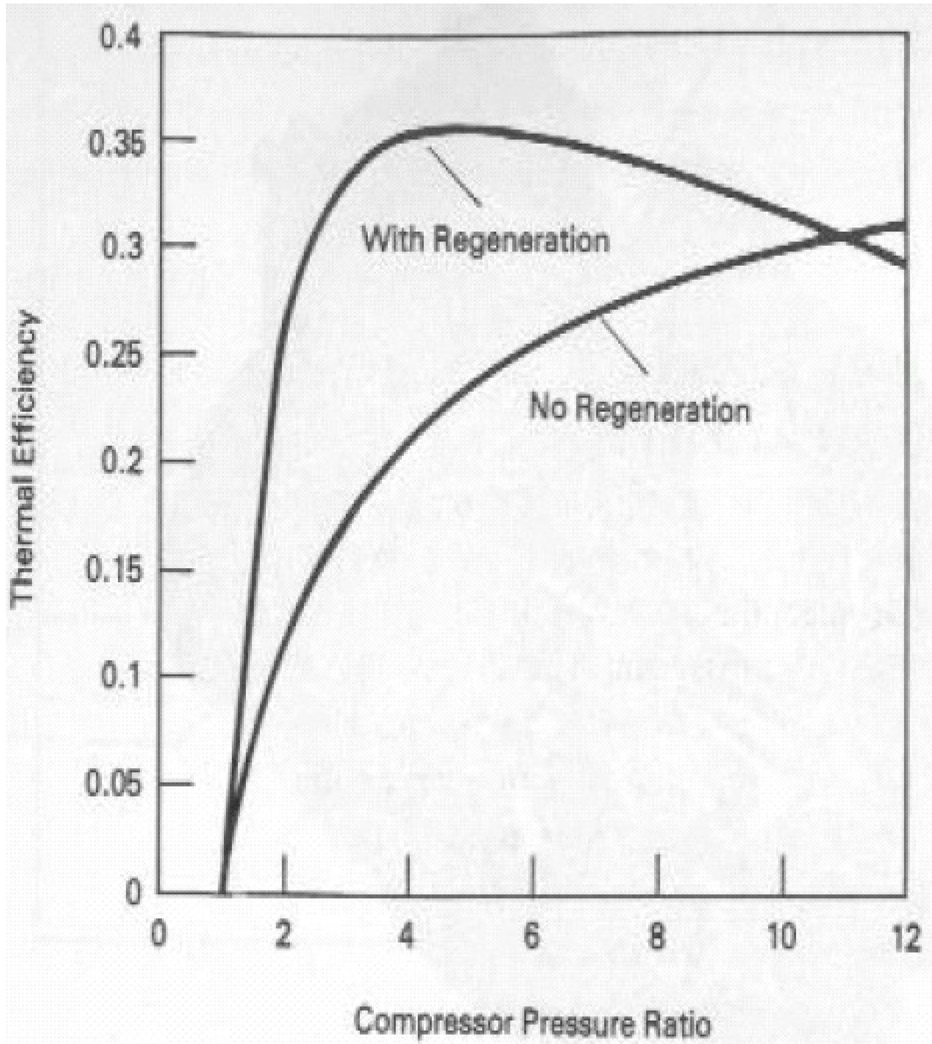
Gas cycle variants – Regeneration

The represented cycle is ideal because flow through the turbine and compressor occurs isentropically and there are no frictional pressure drops. Still, heat transfer between the counterflow streams of the regenerator is a source of irreversibility.





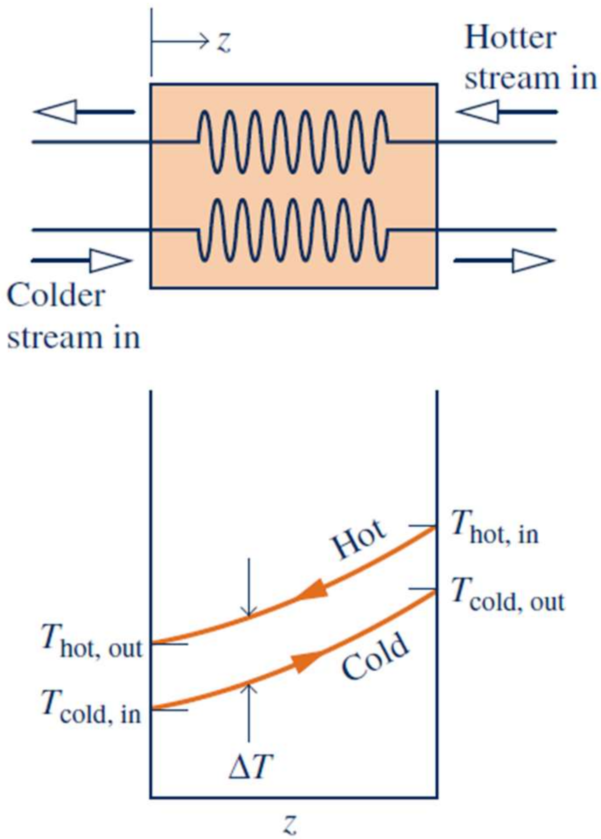
Gas cycle variants – Regeneration



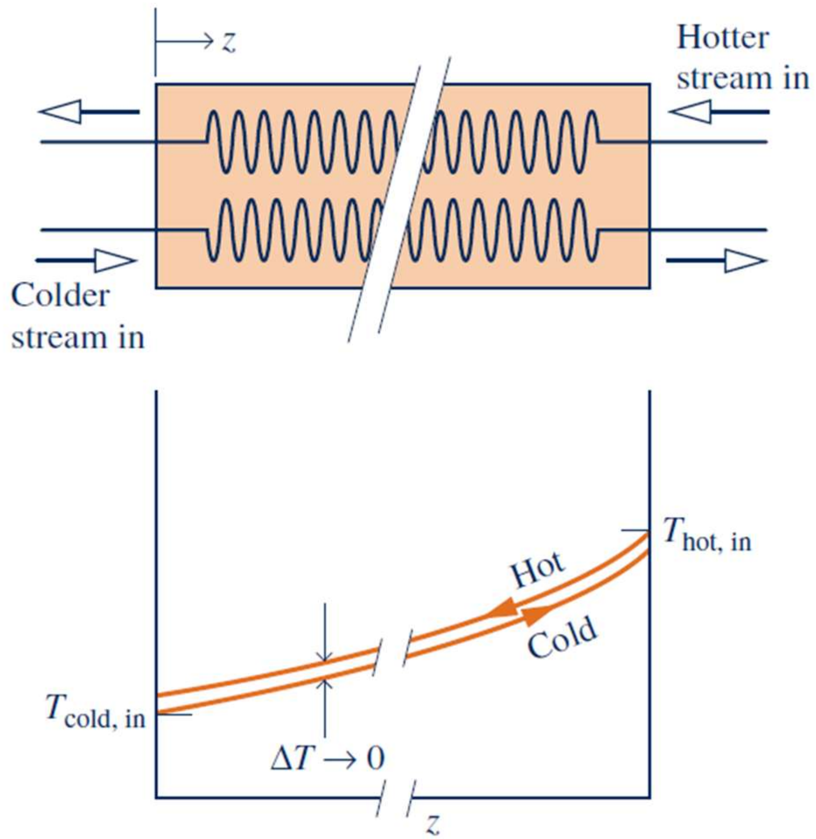
Gas cycle variants – Regeneration

Regenerator effectiveness

Actual

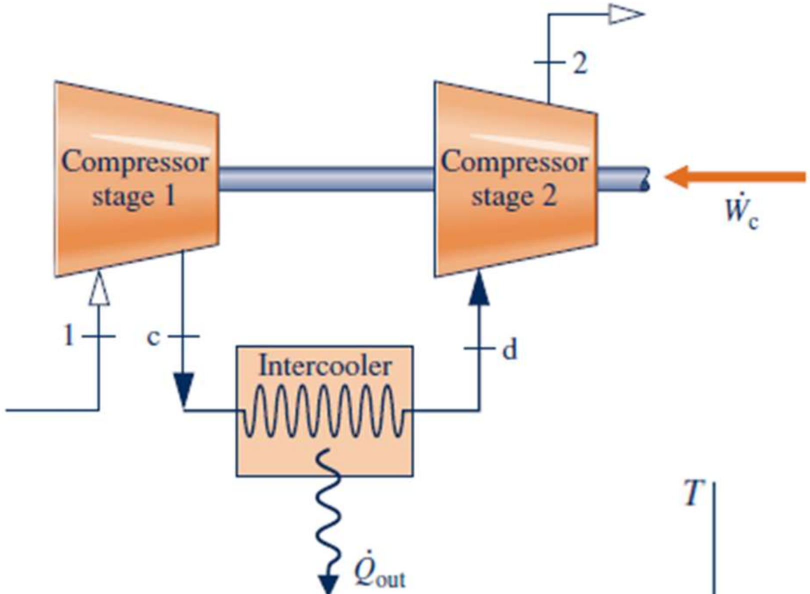


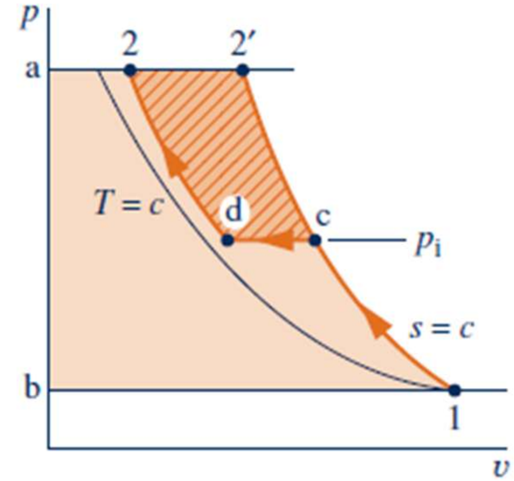
Reversible

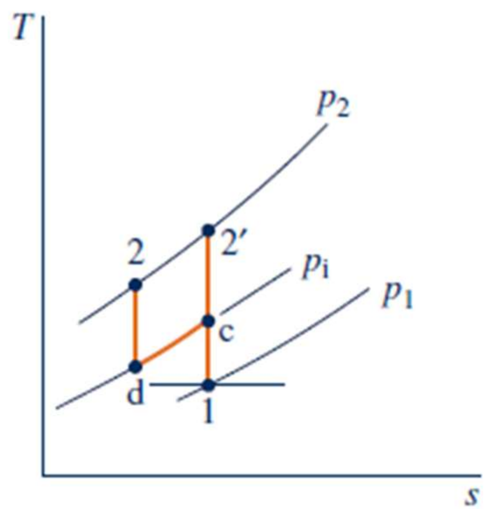


Gas cycle variants – Regeneration

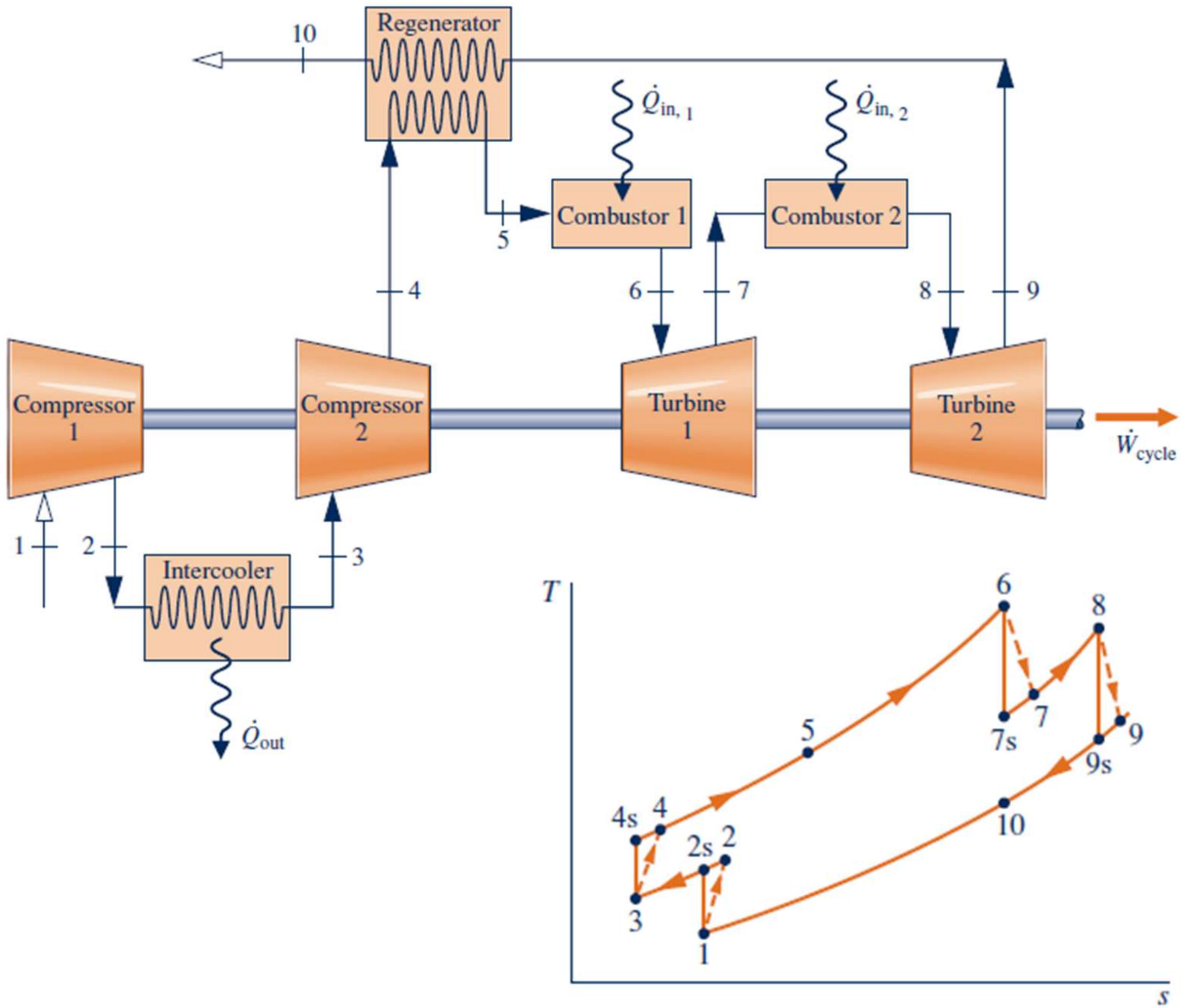
Intercooling





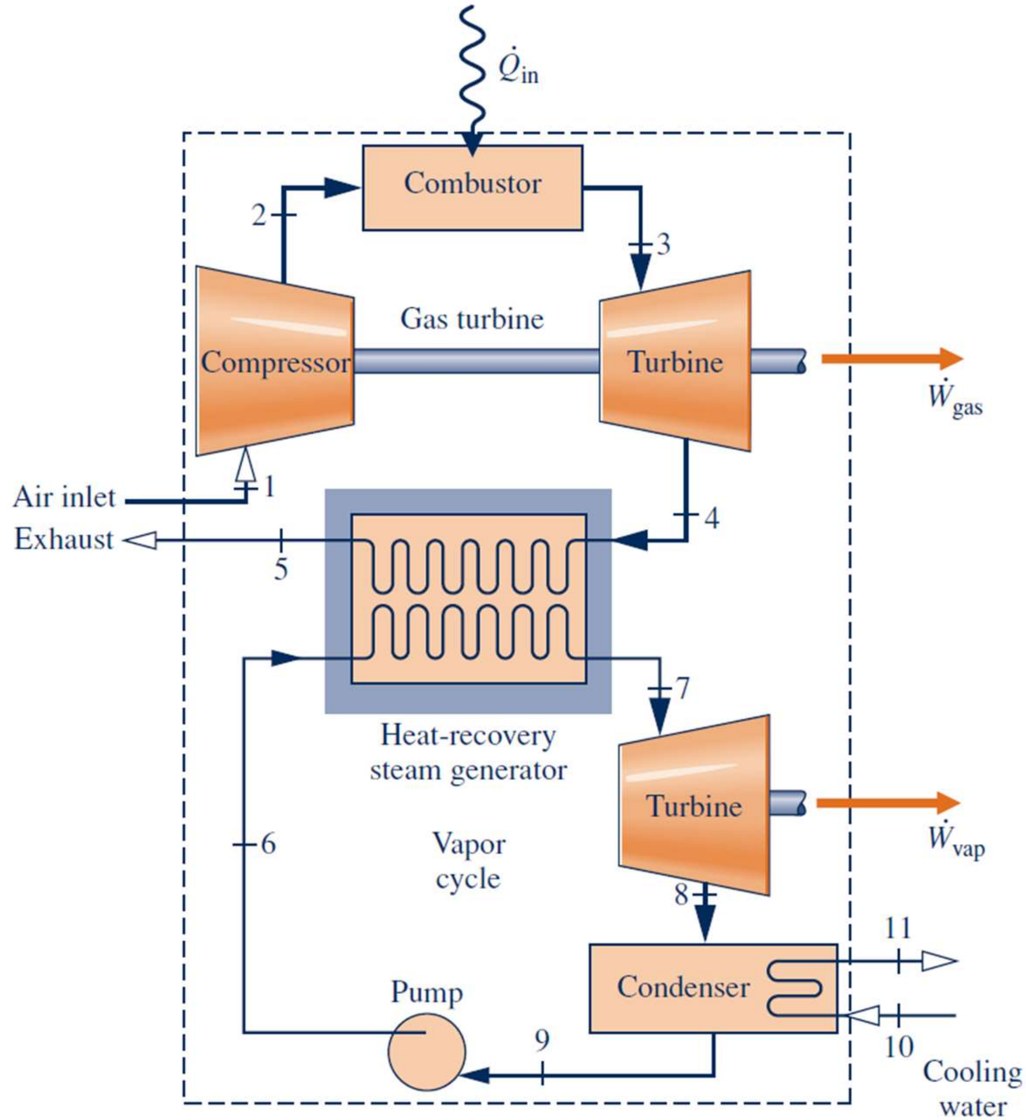


Gas cycle variants – Regeneration



4. Combined cycle

Combined cycles



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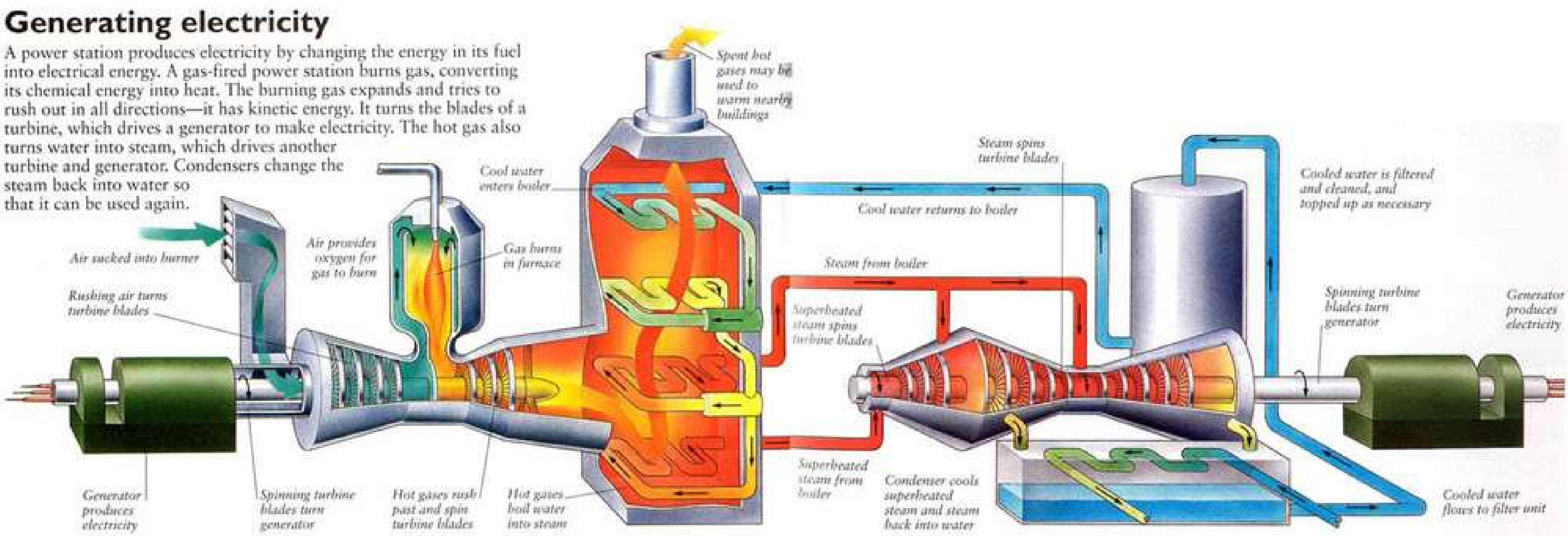
Combined cycles



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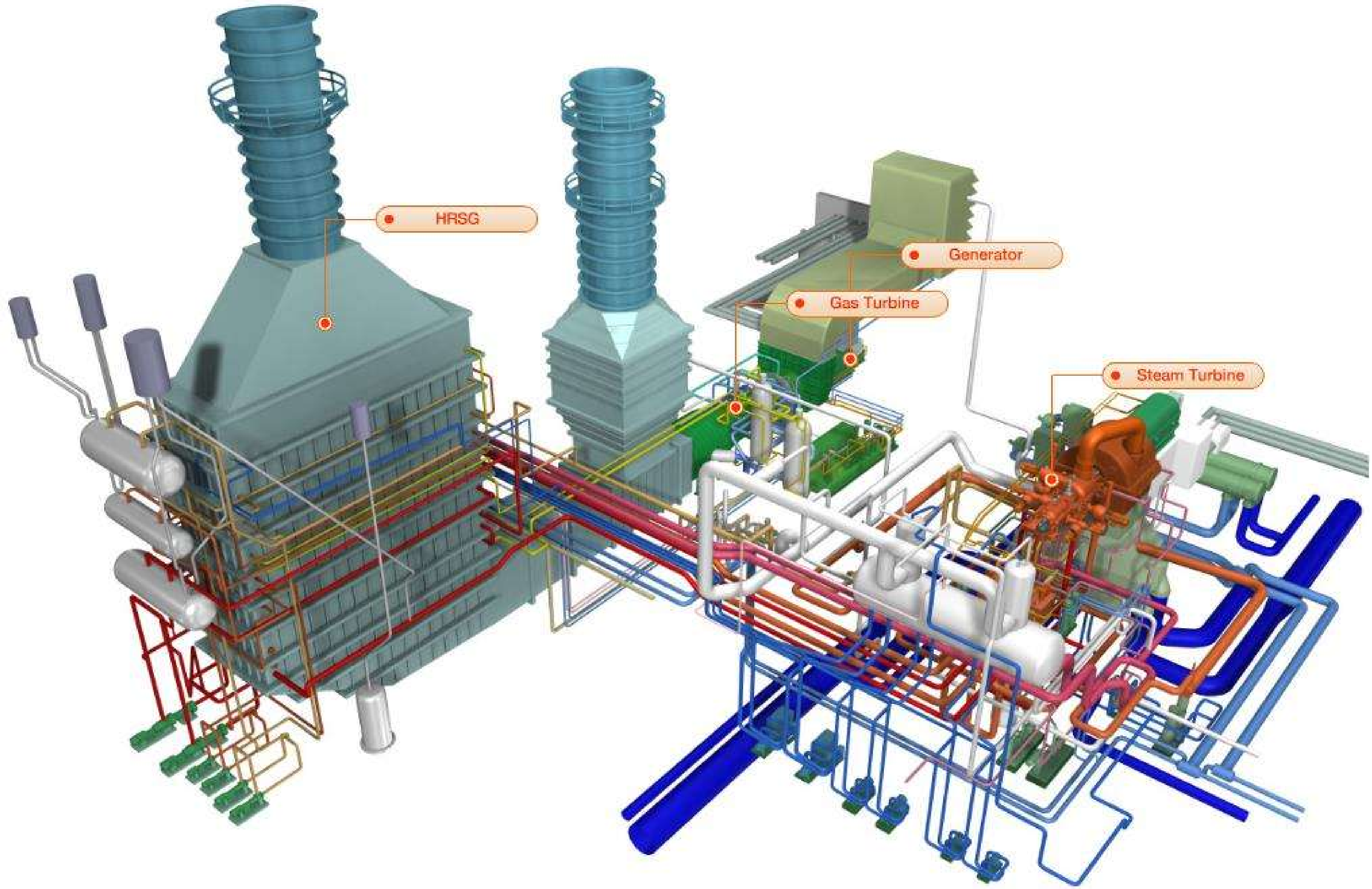
Combined cycles



Combined cycles

Due to the combined balance of 1st and 2nd Laws of an open system:

$$W_{rev} = G \cdot [(h - T_0 \cdot s) - (h_0 - T_0 \cdot s_0)] = G \cdot (\Delta h - T_0 \cdot \Delta s)$$

$$W_{rev} = G \cdot \Delta h \cdot (1 - T_0 \frac{\Delta s}{\Delta h}) = Q_{av} \cdot (1 - \frac{T_0}{\Delta h / \Delta s})$$

The quantity Q_{av} (equal to G*Δh) represents the thermal power 'available' in the gas stream, i.e. that which can be obtained from its complete cooling. This quantity is used to define the efficiency (1st Law) of recovery cycle. In the case of a reversible cycle, efficiency can be defined as:

$$\eta_{rev} = \frac{W_{rev}}{Q_{av}} = 1 - \frac{T_0}{\Delta h / \Delta s}$$

which in the particular case of a gas with constant specific heat becomes:

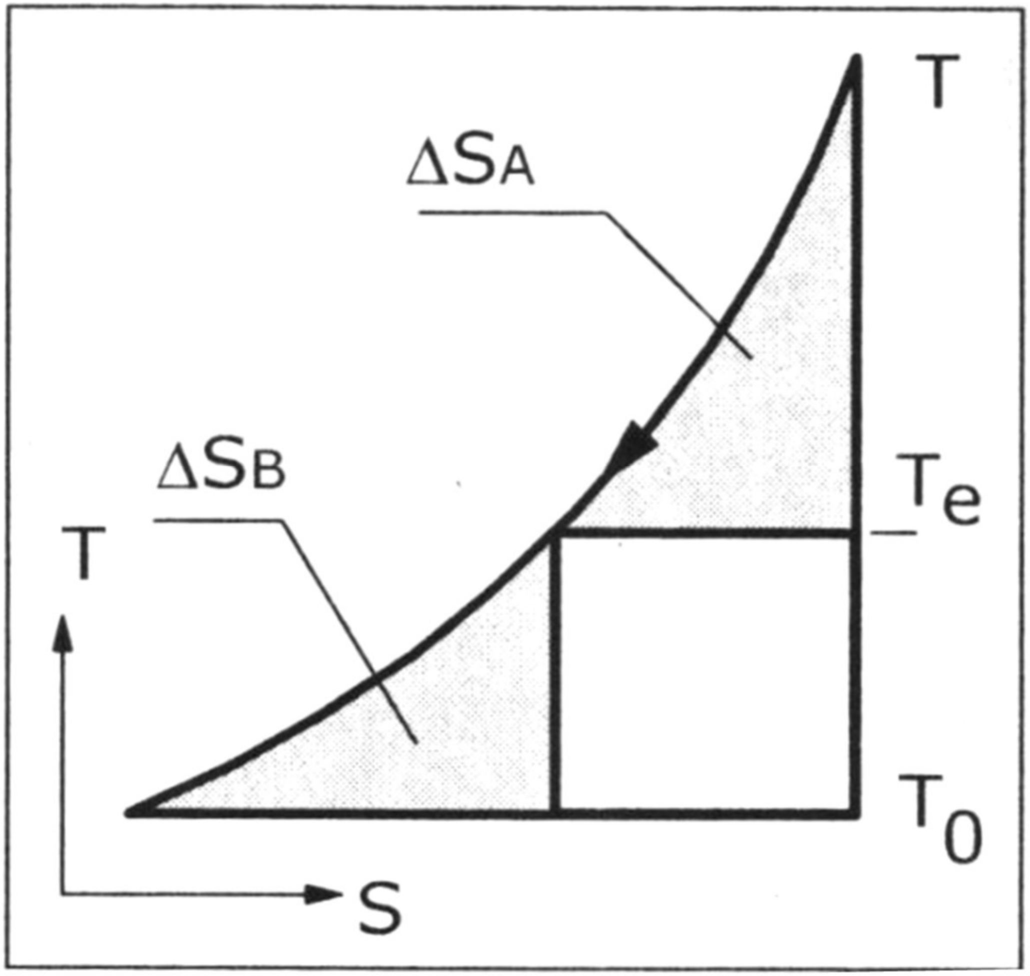
$$\eta_{rev} = 1 - \frac{T_0}{\left[\frac{c_p \cdot (T - T_0)}{c_p \cdot \ln(T/T_0)} \right]} = 1 - \frac{T_0}{T_{ml}} \qquad (T_{ml} = \frac{T - T_0}{\ln \frac{T}{T_0}})$$

Combined cycles

$$\eta_I = \frac{W}{Q_{av}} \;\; ; \;\; \eta_{II} = \frac{W}{W_{rev}} \;\; ; \;\; \eta_I = \eta_{II} \cdot \eta_{rev}$$

$$\eta_I = \frac{W}{Q_{av}} = \frac{W}{Q_{in}} \cdot \frac{Q_{in}}{Q_{av}} = \eta \cdot \chi = \eta_{rec}$$

Combined cycles



Combined cycles

$$\Delta s_A = \frac{c_p \cdot (T - T_e)}{T_e} - c_p \cdot \ln \frac{T}{T_e}$$

$$\Delta s_B = \frac{c_p \cdot (T_e - T_0)}{T_0} - c_p \cdot \ln \frac{T_e}{T_0}$$

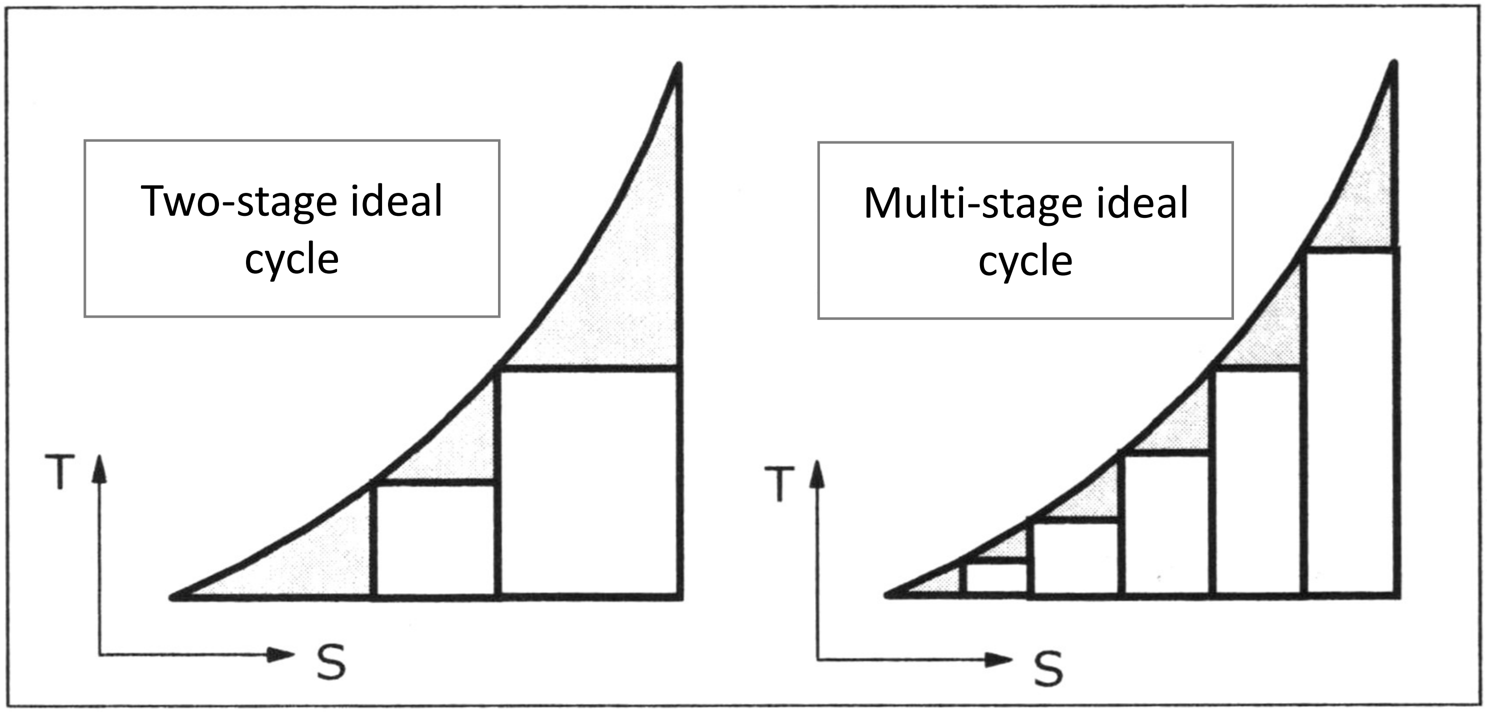
$$\Delta s_A + \Delta s_B = c_p \left[\frac{T}{T_e} - 1 + \frac{T_e}{T_0} - 1 - \ln\left(\frac{T}{T_e} \cdot \frac{T_e}{T_0}\right) \right]$$

$$\frac{\partial(\Delta s_A + \Delta s_B)}{\partial T_e} = c_p \left(-\frac{T}{T_e^2} + \frac{1}{T_0}\right) = 0 \;\rightarrow\; T_e = \sqrt{T \cdot T_0}$$

$$\eta = 1 - \frac{T_0}{T_e} = 0.3895 \;;\; \chi = \frac{T - T_e}{T - T_0} = 0.6209 \;;\; \eta_{rec} = \eta \cdot \chi = 0.2419$$

$$\eta_{rev} = 1 - \frac{T_0}{T_{ml}} = 0.4136 \quad (T_{ml} = 491.39K)$$

Temperature difference composite curves (TDCC) analysis

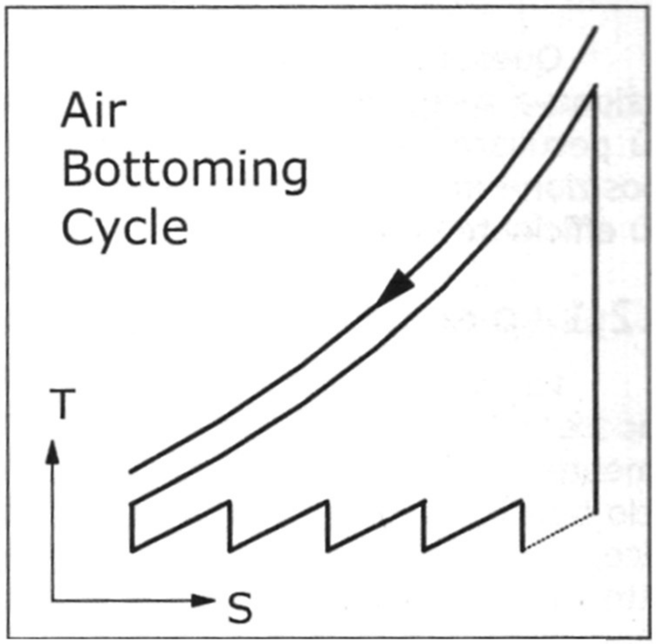


$$T_{e1} = T_0^{1/3} \cdot T^{2/3} \quad ; \quad T_{e2} = T_0^{2/3} \cdot T^{1/3}$$

$$\eta_{rec} = \left(1 - \frac{T_0}{T_{e1}}\right) \cdot \frac{T - T_{e1}}{T - T_0} + \left(1 - \frac{T_0}{T_{e2}}\right) \cdot \frac{T_{e1} - T_{e2}}{T - T_0}$$

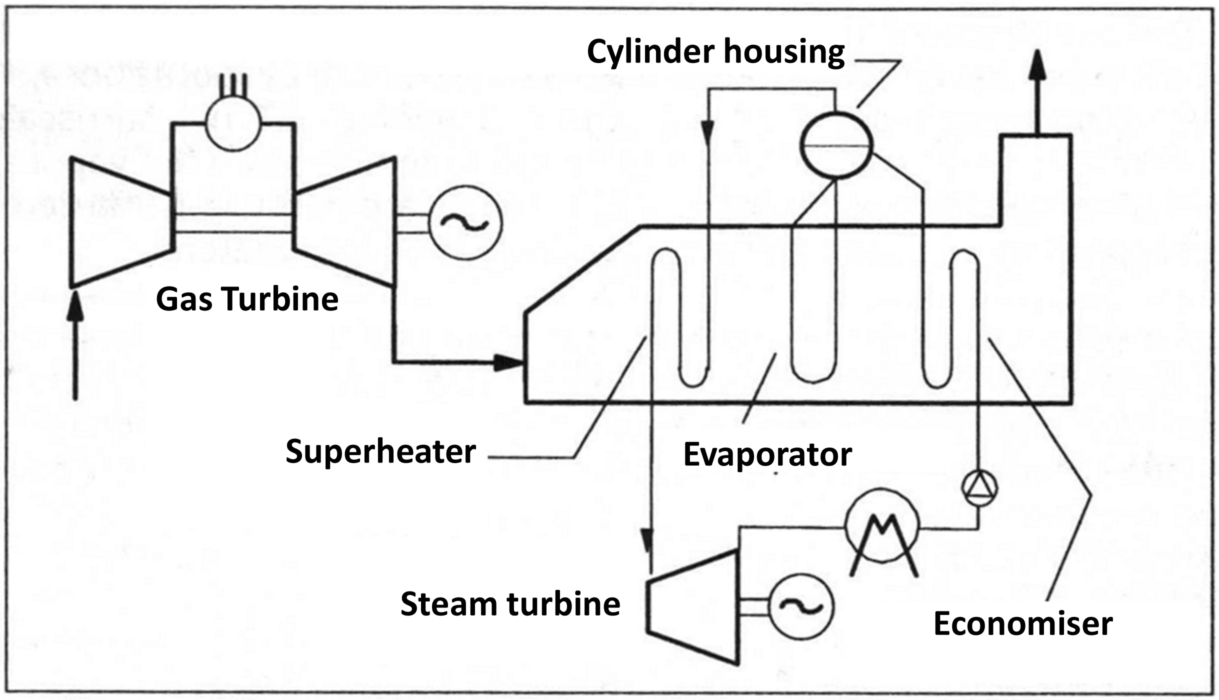
Other cycles

Heat recovery from gaseous stream with gas cycle equipped with multiple intercoolers

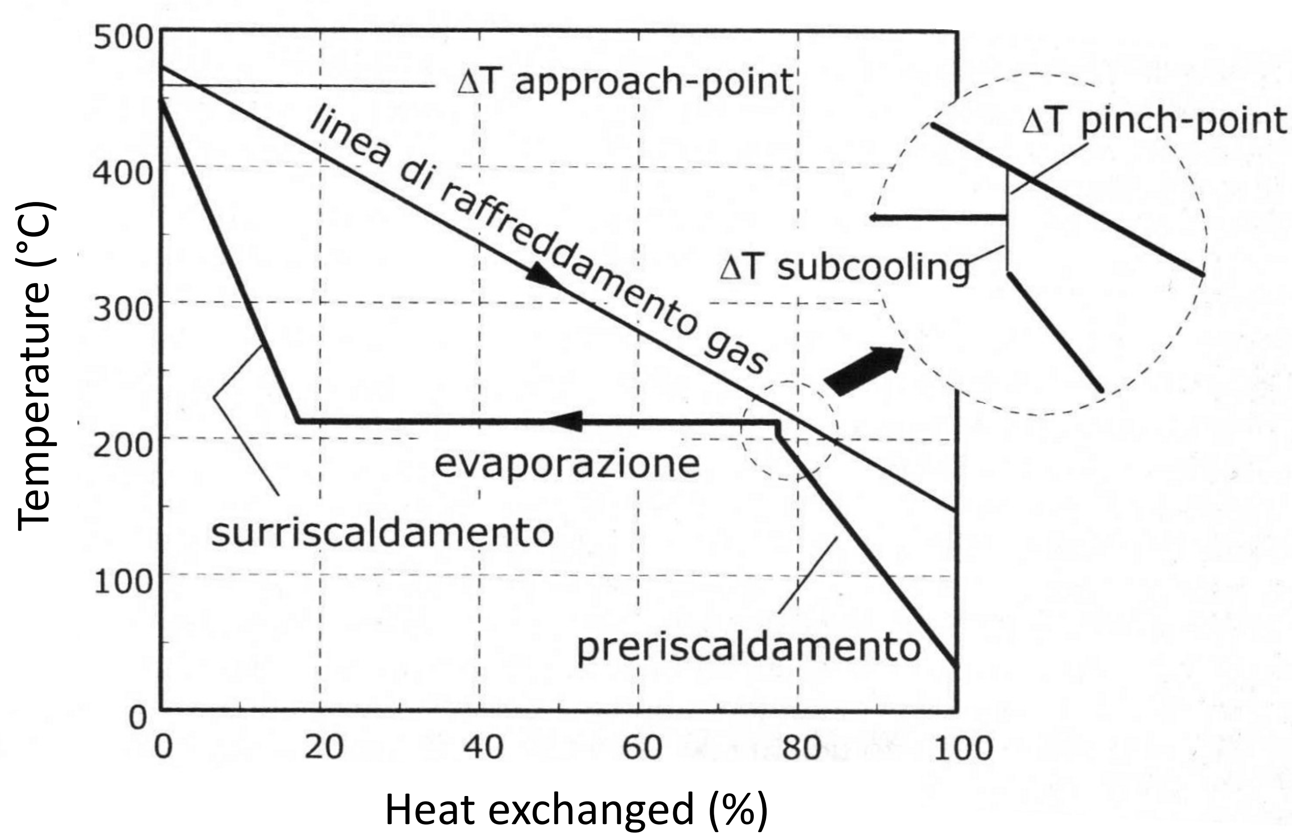


$$\eta_{CC} = \eta_{GT} + (1 - \eta_{GT} - \xi) \cdot \eta_{rec}$$

Heat exchange in the HRSG



Heat exchange in the HRSG



Heat exchange in the HRSG

Variations of performance and parameters of a one pressure level recuperative-cycle for a gas turbine. Different ΔT of pinch-point, approach-point and subcooling are compared with a baseline (ΔT_{PP}=10°C, ΔT_{AP}=25°C and ΔT_{SC}=10°C).

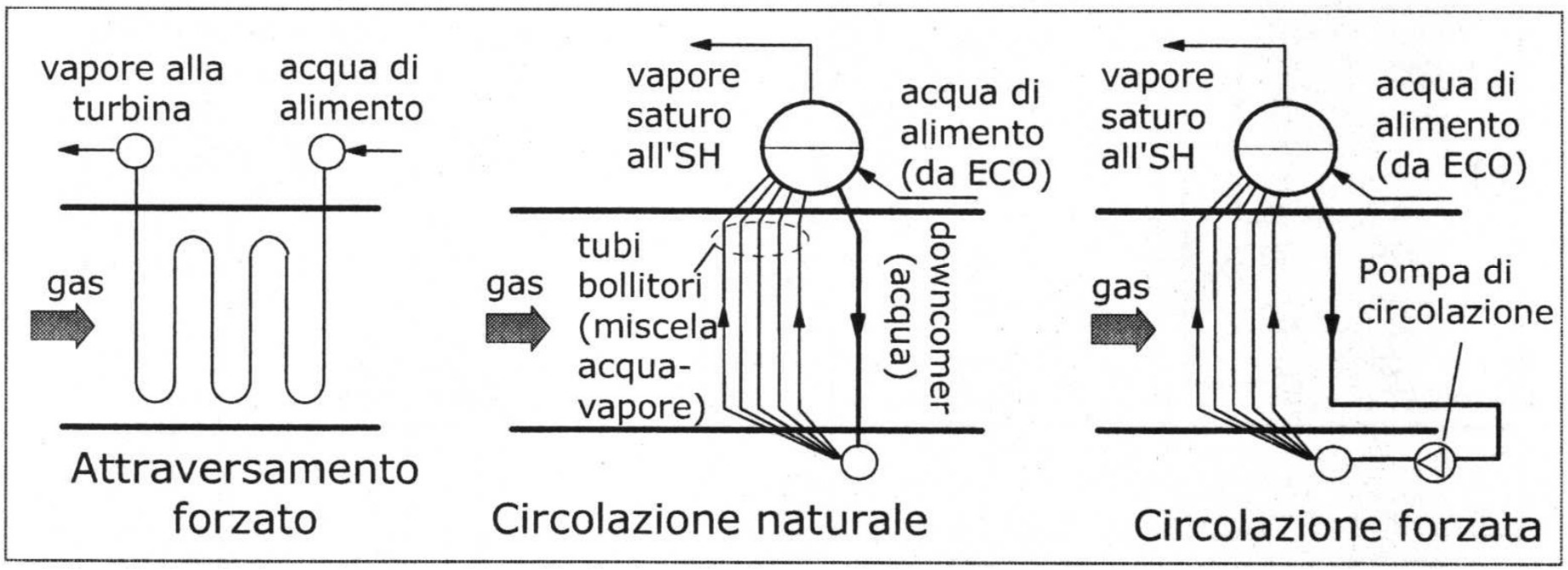
	Unit	Baseline	ΔT_{PP}		ΔT_{AP}		ΔT_{SC}	
			5°C	20°C	10°C	50°C	0°C	20°C
Steam turbine gross power	MW_{el}	65.20	66.52 (+2.0%)	62.62 (+4.0%)	65.66 (+0.7%)	64.46 (-1.1%)	64.46 (+1.9%)	64.03 (-1.8%)
Steam flow rate	kg/s	67.19	68.49	64.58	66.32	68.70	68.44	66.00
Flue gas temperature	°C	147.0	140.5	160.1	148.0	145.3	140.7	153.0
U·A	kW/K	3349	3971	2670	3496	3266	3742	3129

Heat exchange in the HRSG

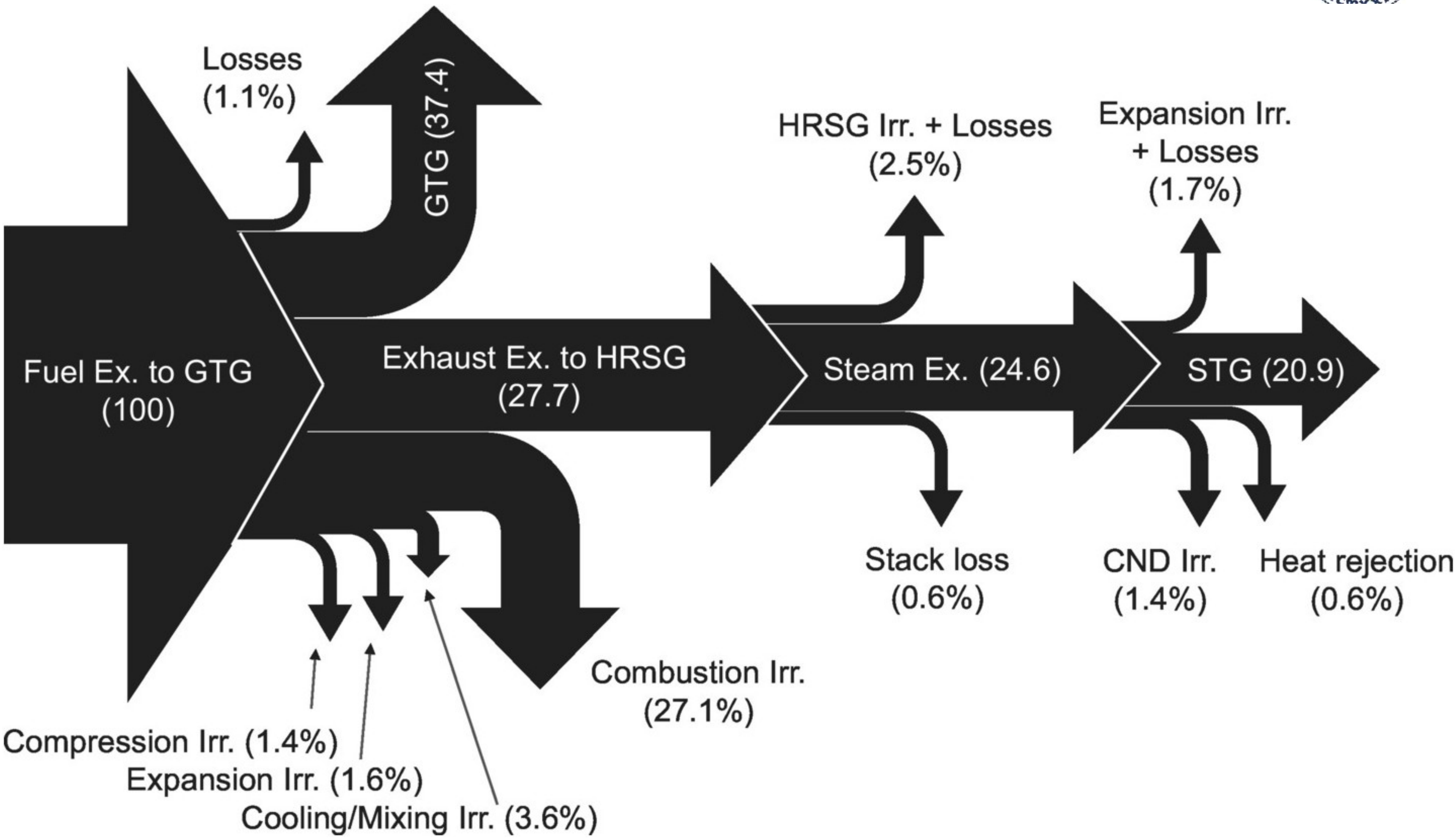
Design of a Heat Recovery Steam Generator referred to a two-stage cycle: the influence of different pinch-point temperatures on the investment and operating costs are reported.

	Unit	ΔT_{PP}		
		5°C	10°C	15°C
Net power generated by steam turbine	MW_{el}	74.764	73.461	72.144
HRSG U·A	kW/K	6673	5434	4684
Max cost HRSG	M€/y	1.044	0.394	0.000
Max cost other components	M€/y	0.138	0.069	0.000
Cost related to a lower power production respect ΔT_{PP}=5°C	M€/y	0.000	0.489	0.983
Total yearly cost	M€/y	1.182	0.952	0.983

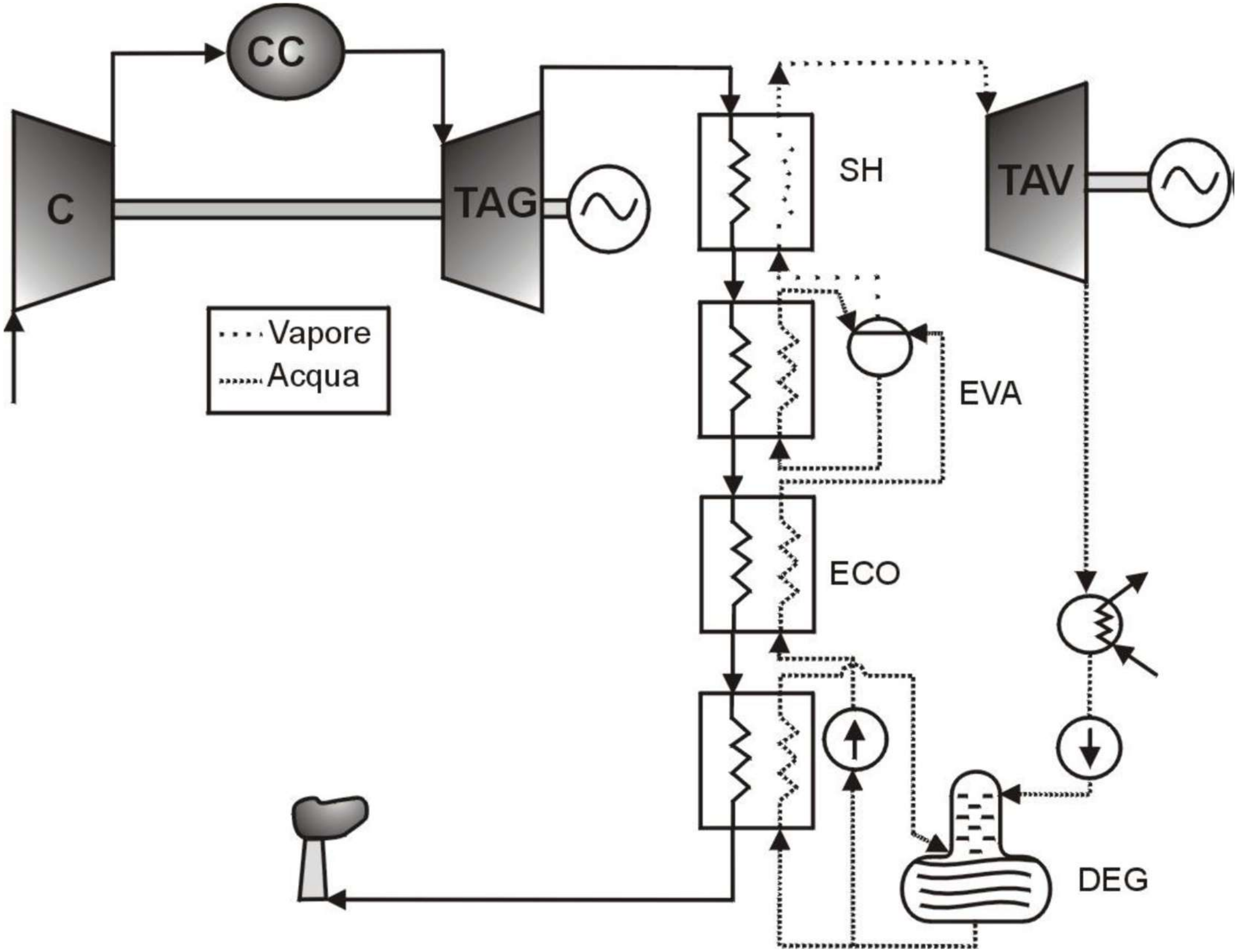
Steam circulation



CCGT energy balance



Degasator



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Degasator

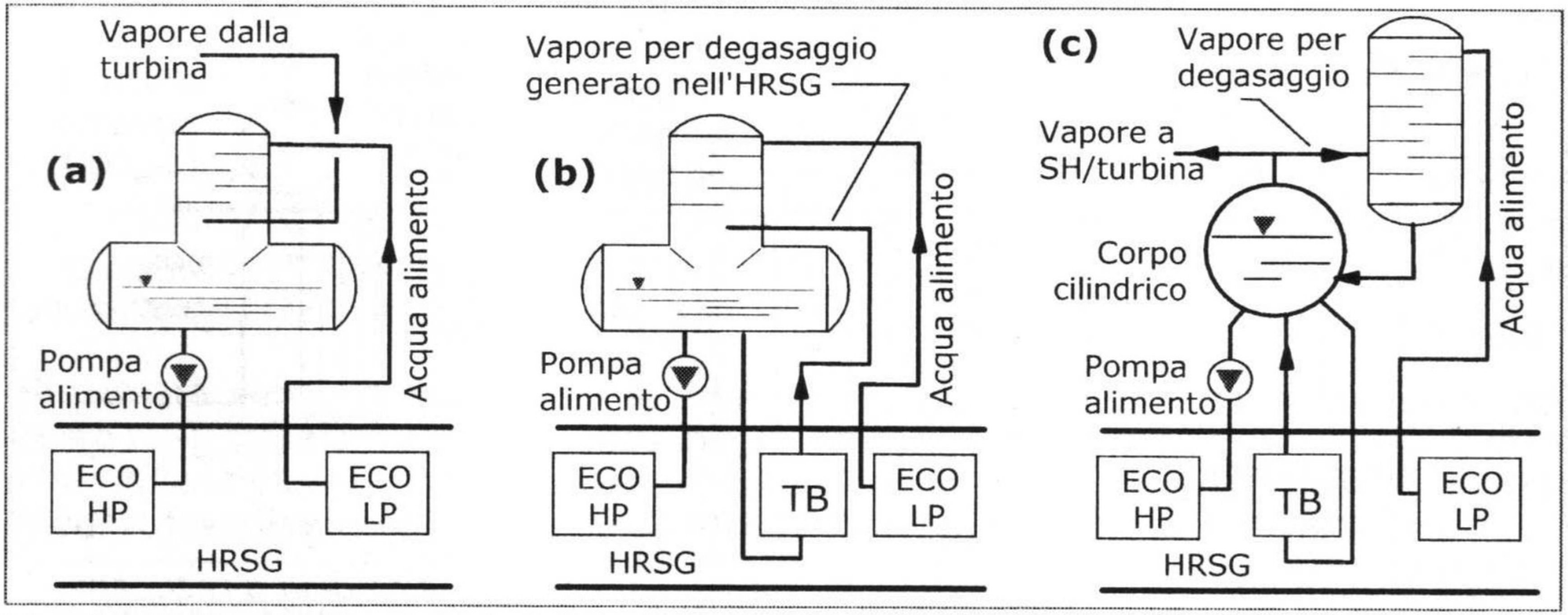
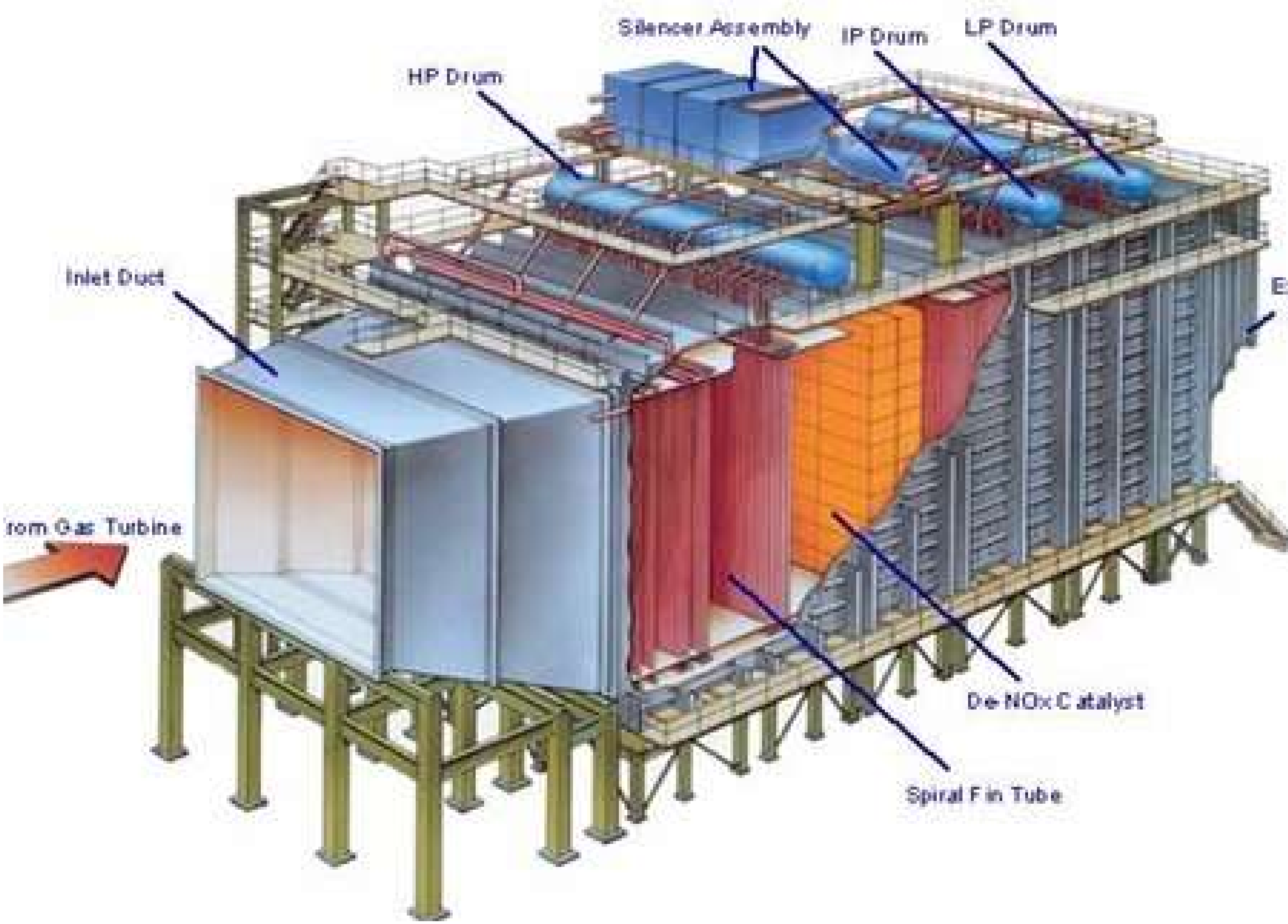


Fig.4.10: *Possibili schemi di funzionamento del degasatore, inserito nella caldaia a recupero di un ciclo combinato (TB: tubi bollitori per la generazione di vapore).*

[Lozza]

HRSG layout



Silencer Assembly
IP Drum
LP Drum
HP Drum
Inlet Duct
Exhaus
Duct
rom Gas Turbine
De-NOx Catalyst
Spiral Fin Tube

Condenser

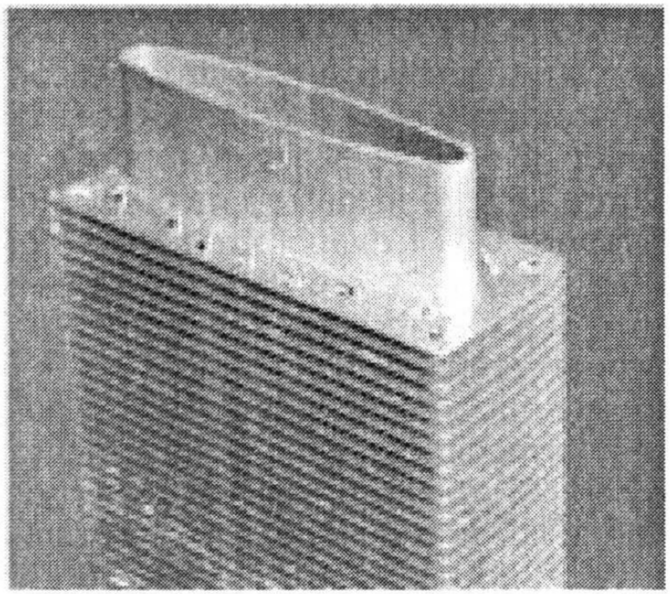
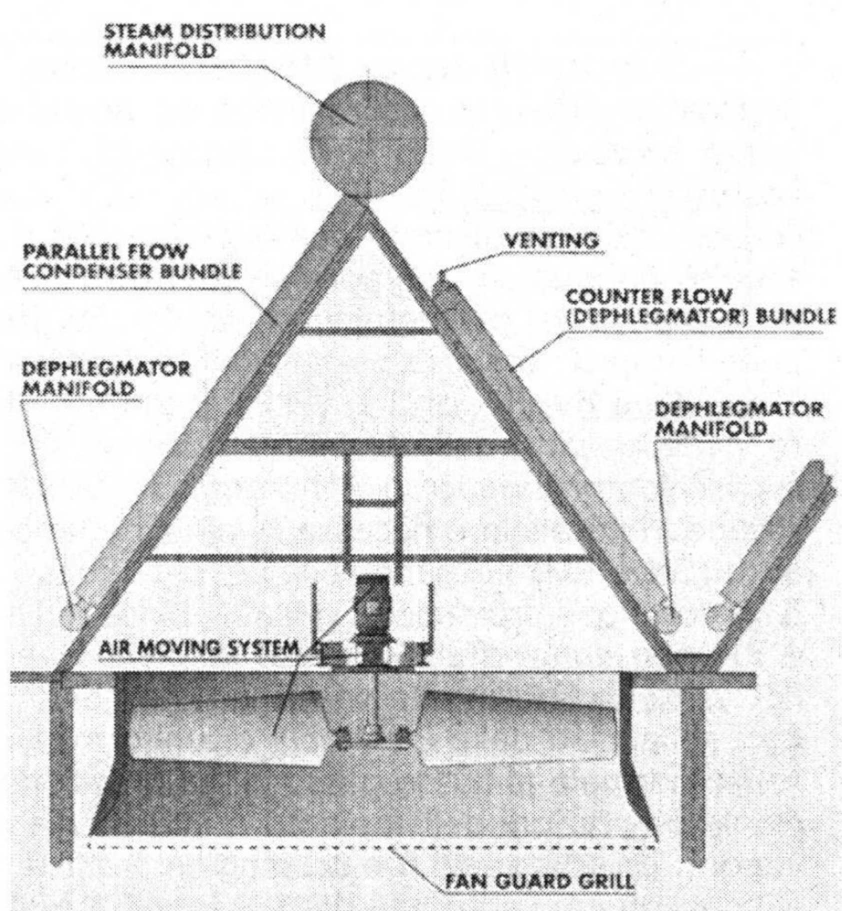


Fig.4.14: *Tubo alettato impiegato in un condensatore di vapore a secco (sopra) e struttura di un condensatore a circolazione forzata (a destra), con le batterie di scambio disposte a V rovesciata e ventilatore assiale (fonte: GEA).*



Plant layout

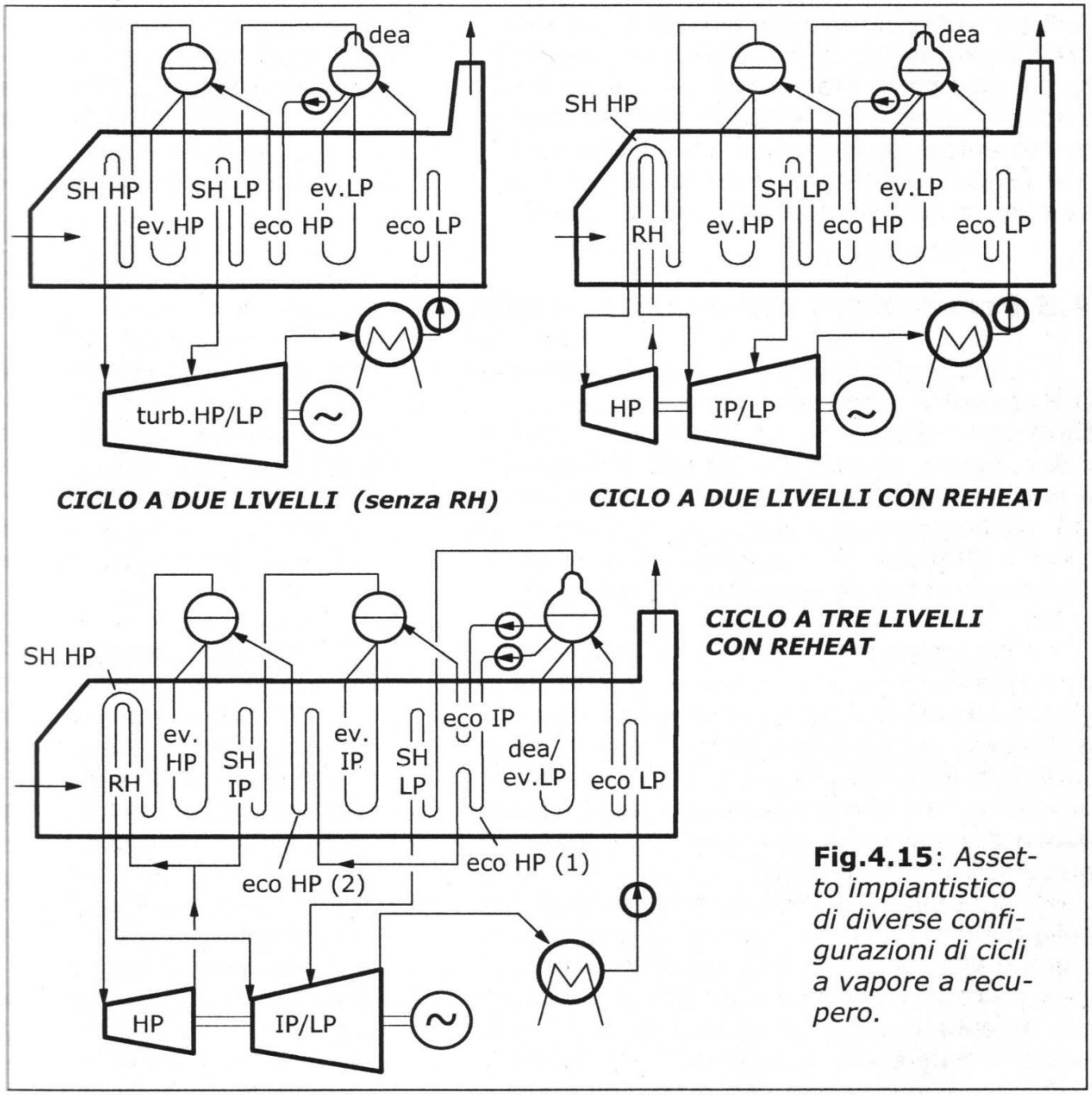


Fig.4.15: *Assetto impiantistico di diverse configurazioni di cicli a vapore a recupero.*

[Lozza]

Combined cycle performance

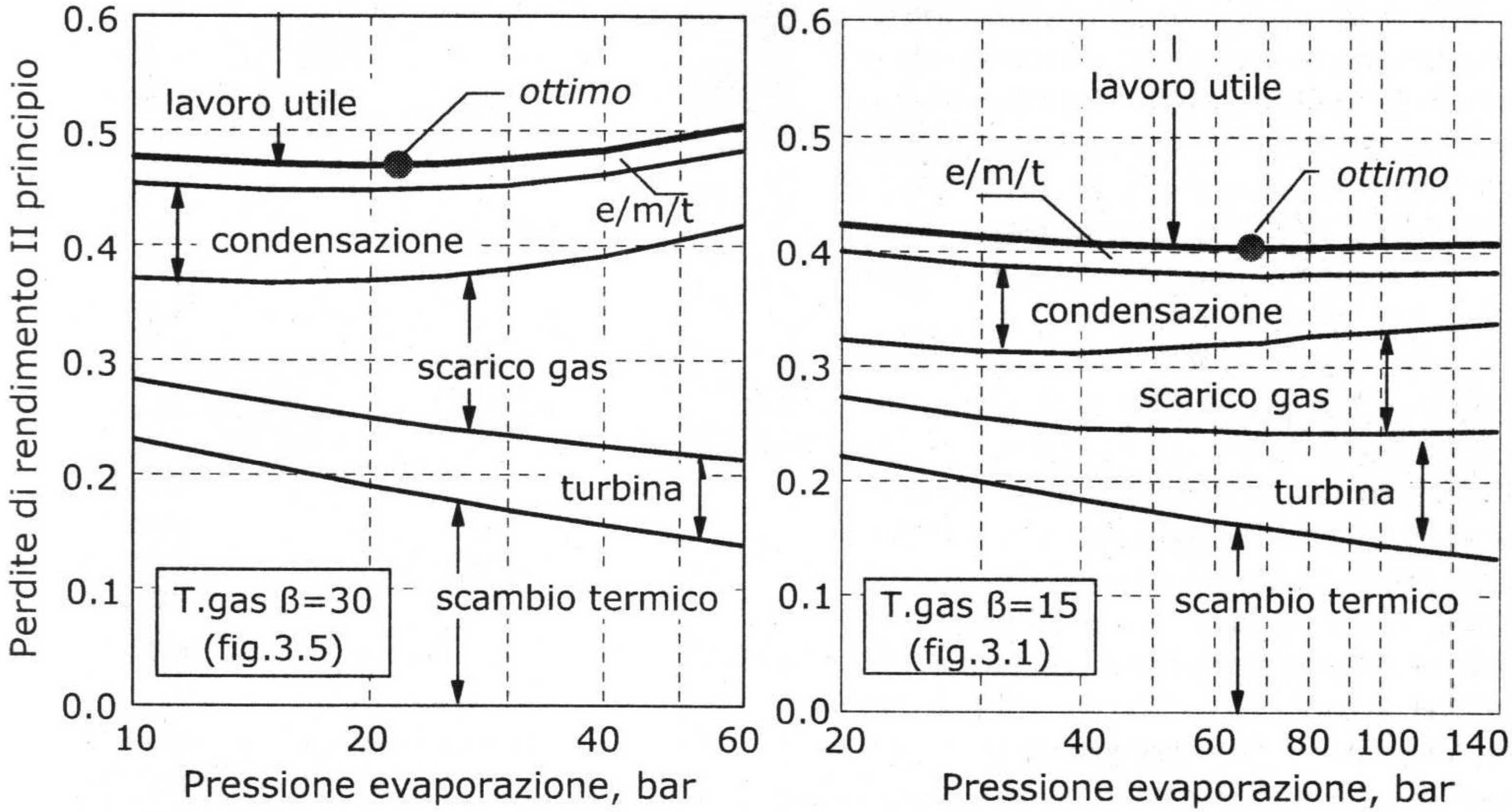
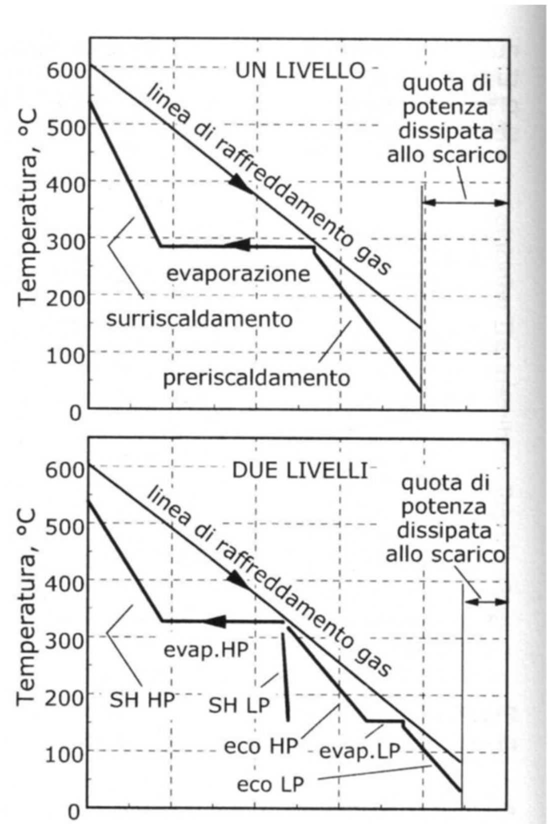


Fig.4.16: *Analisi entropica di cicli a recupero monolivello, al variare della pressione di evaporazione, per due casi con diversa temperatura dei gas.*

[Lozza]

Influence of plant layout



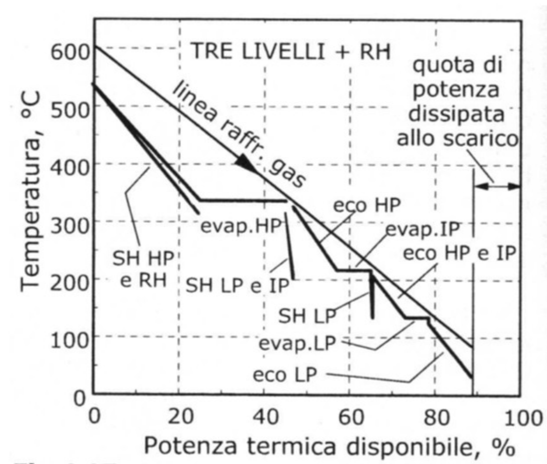


Fig.4.17: *Diagramma dello scambio termico per cicli a recupero ottimizzati per la turbina a gas di fig.3.1, nei casi: (a) a un livello di pressione, (b) a due livelli, (c) a tre livelli con RH.*

[Lozza]

Influence of plant layout

Type of heat recovery	Unit	1L	2L	3LR	3LR(H)
CC efficiency η_{CC}	%	57.96	59.08	60.25	61.21
Rec. efficiency η_{rec}	%	29.34	31.27	33.29	34.94
Net power	MW	166.20	177.15	188.57	197.93
Pressure	bar	80	110/5	140/30/4	140/30/4
Steam temperature	°C	538	538/260	538/538/280	590/590/300
Stam flow rate	kg/s	138	138/17	108/17/12	103/17/12
Exhausted gas temperature	°C	109	82	89	87
HRSG U·A	kW/K	6766	10130	11516	13766
2^{nd} Princ. efficiency η_{II}	%	62.18	66.28	70.55	74.05
HRSG heat transfer η_{II}	%	15.40	12.90	10.44	8.97
Chimney gas exhaust η_{II}	%	6.20	2.19	2.62	2.52
Steam turbine η_{II}	%	7.81	8.96	6.91	6.38
Condensator η_{II}	%	6.29	7.00	6.64	5.42
Press. and temp. losses η_{II}	%	0.69	1.11	1.25	1.04
Mech. and therm. losses η_{II}	%	1.43	1.56	1.59	1.62

Combined cycle performance

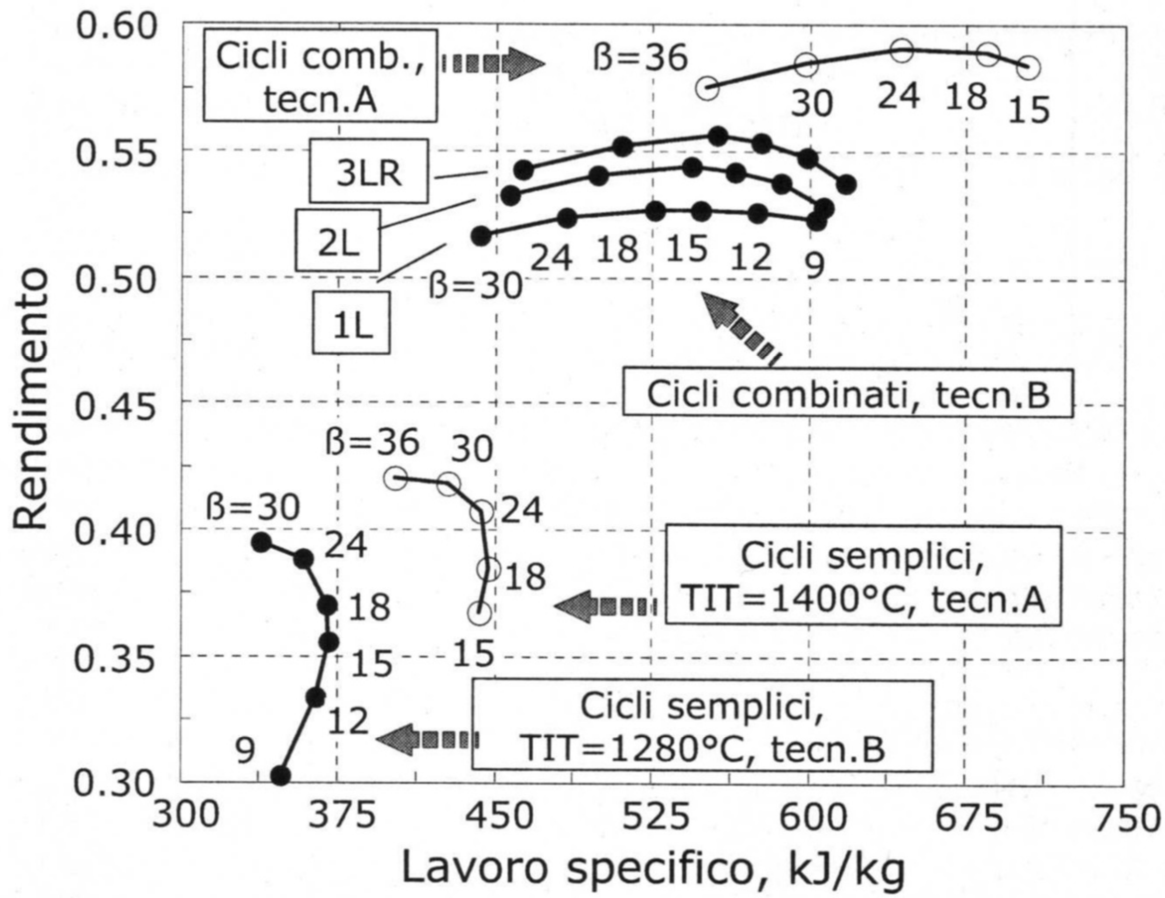
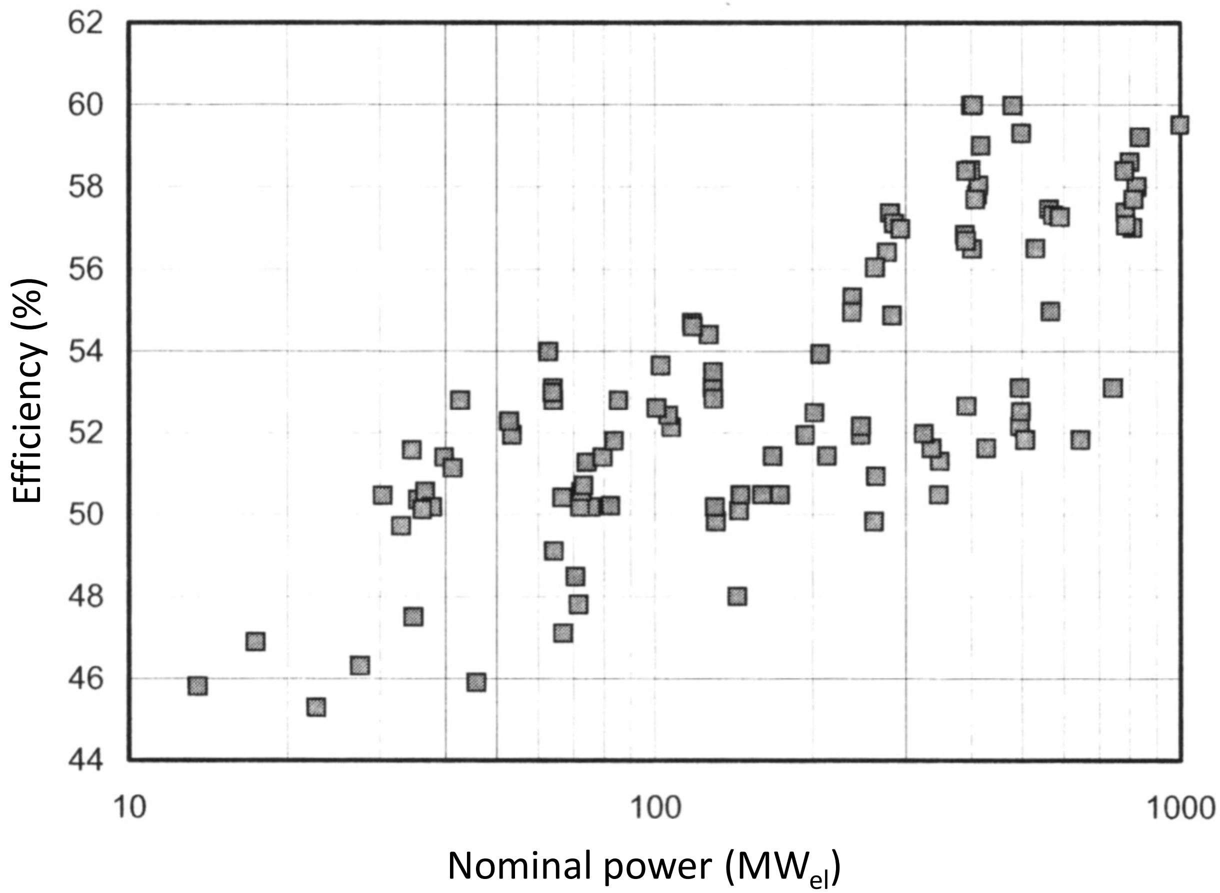


Fig.4.19: *Rendimenti e lavori specifici di cicli combinati, per diversi rapporti di compressione della turbina a gas, a confronto con i cicli semplici. Le tecnologie A e B sono definite in Tab.3.2; per i cicli combinati con tecn.B sono presentati i risultati con cicli a recupero a diversi livelli di pressione, mentre per quelli con tecn.A ci si riferisce solo a cicli 3LR (avanzati - Tab.4.4).*

[Lozza]

Combined cycle performance



[Lozza]

Cost of electricity generated

$$COE = C_{CAP} * CCR/H_{eq} + C_{OM} + C_{FUEL}/\eta$$

C_{CAP}:capital share specific to its nominal net power (€/MWe), reported at the start of commercial operations.

CCR: share of the capital cost to be charged to the annual budget (15% EPRI report).

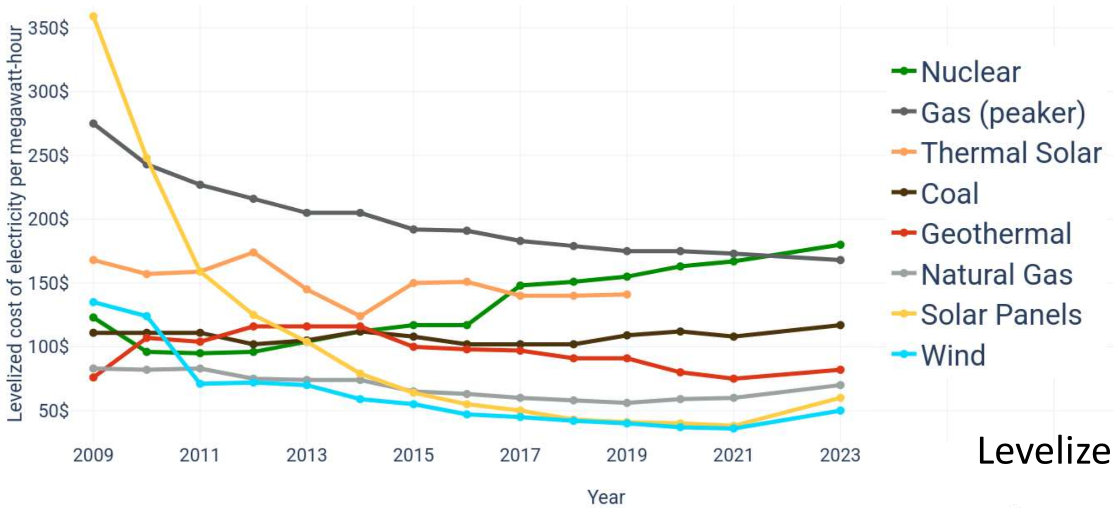
h_{eq}: number of annual equivalent operating hours at nominal power (MWh produced divided by the nominal power).

C_{OM}: Operating and maintenance costs.

C_{FUEL}: fuel share, specific fuel cost (€/MWh produced by combustion).

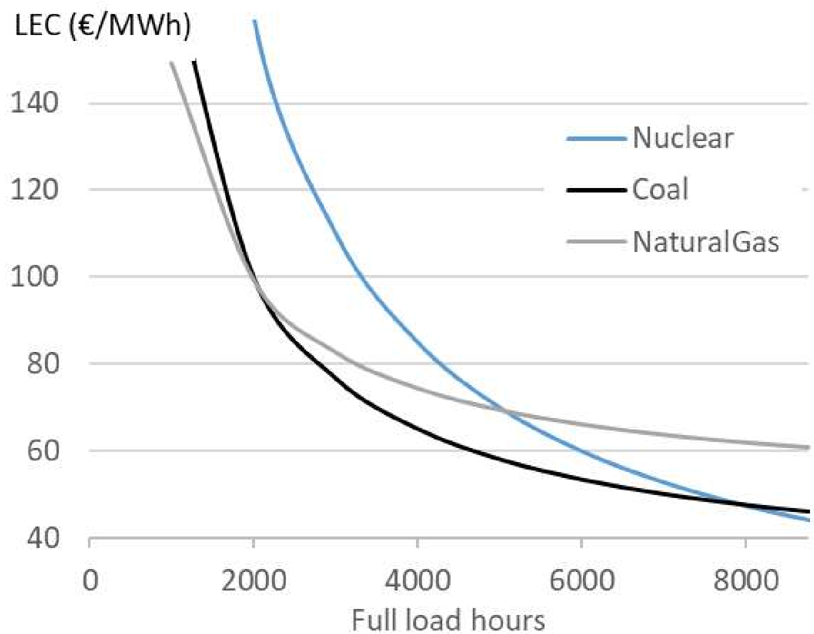
η: plant efficiency.

Cost of electricity generated



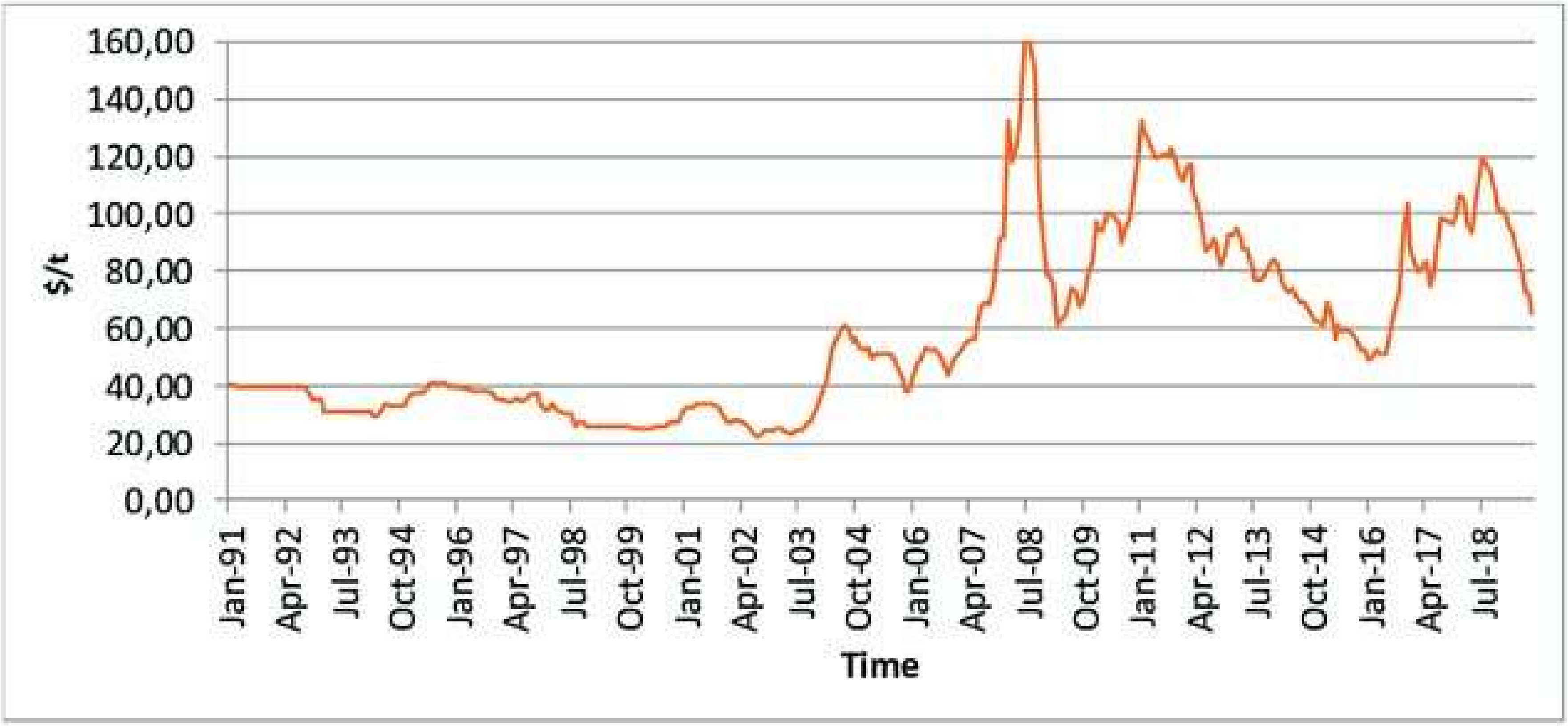
Data from Lazard

Levelized generation costs by technology



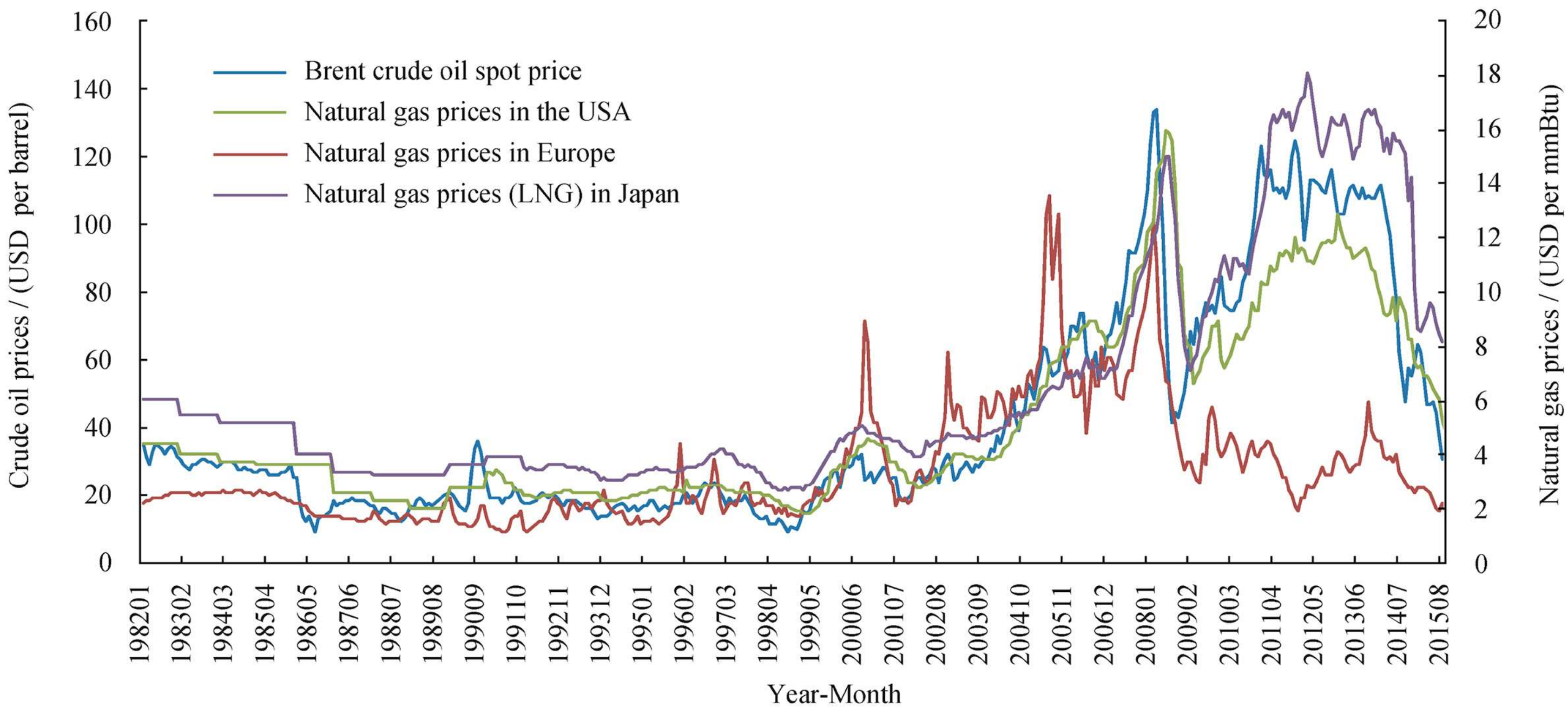
Data from Open Electricity Economics

Trends in coal price



[10.3390/en13082017]

Trends in fuel price



[https://doi.org/10.1016/j.ngib.2017.02.010]

5. Exercise

Exercise: Energy and Exergy Analyses of a CCGT power plant

A combined gas turbine–vapor power plant has a net power output of 45 MW. Air enters the compressor of the gas turbine at 100 kPa, 300 K, and is compressed to 1200 kPa. The isentropic efficiency of the compressor is 84%. The condition at the inlet to the turbine is 1200 kPa, 1400 K. Air expands through the turbine, which has an isentropic efficiency of 88%, to a pressure of 100 kPa. The air then passes through the interconnecting heat recovery steam generator and is finally discharged at 400 K. Steam enters the turbine of the vapor power cycle at 8 MPa, 400°C, and expands to the condenser pressure of 8 kPa. Water enters the pump as saturated liquid at 8 kPa. The turbine and pump of the vapor cycle have isentropic efficiencies of 90% and 80%, respectively.

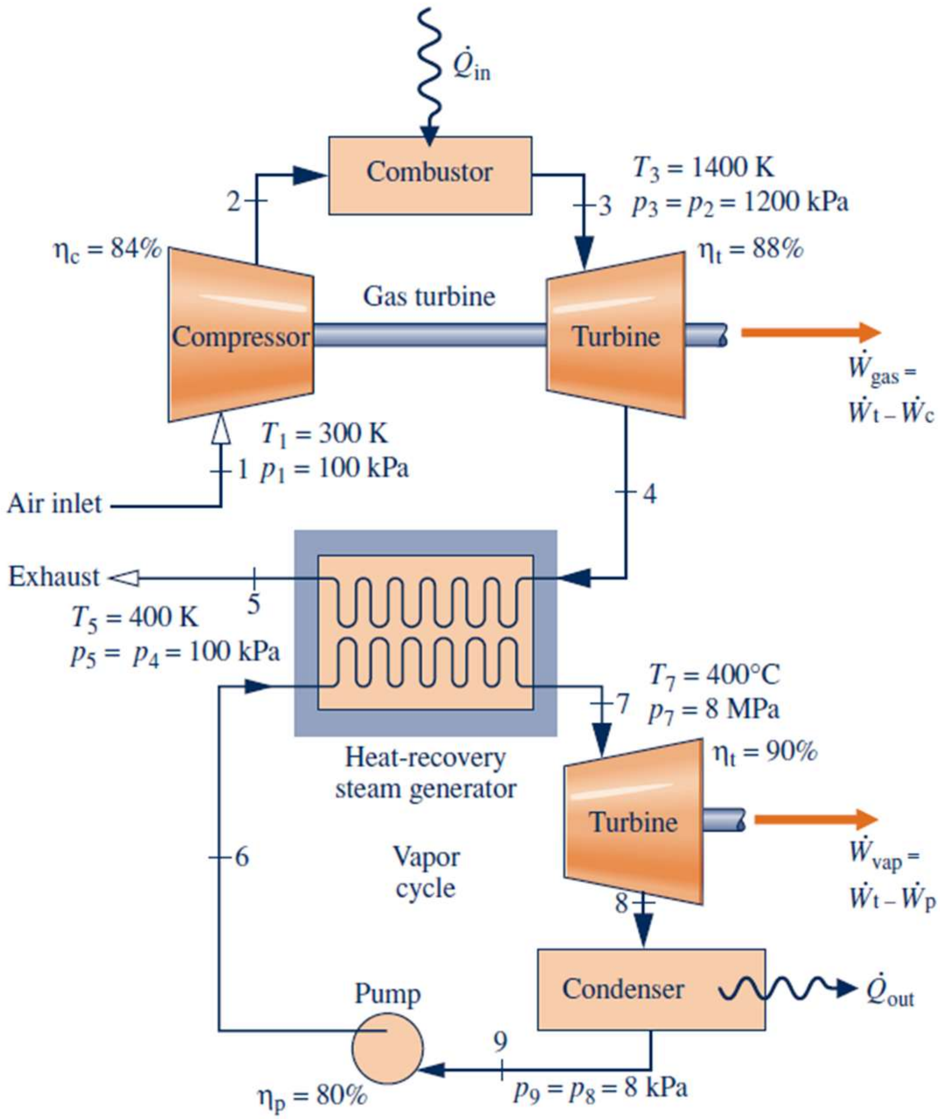
(a) Determine the mass flow rates of the air and the steam, each in kg/s; the net power developed by the gas turbine and vapor power cycle, each in MW; and the thermal efficiency.

(b) Develop a full accounting of the net rate of exergy increase as the air passes through the gas turbine combustor.

Exercise: Assumptions

- *Ambient condition:* T=300 K and p=100 kPa
- Each component on the accompanying sketch is analyzed as a control volume at steady state.
- The turbines, compressor, pump, and interconnecting heat-recovery steam generator operate adiabatically.
- Kinetic and potential energy effects are negligible.
- There are no pressure drops for flow through the combustor, heat-recovery steam generator, and condenser.
- An air-standard analysis is used for the gas turbine.

Exercise: Schematic and input data



Gas Turbine		
State	**h (kJ/kg)**	**s° (kJ/kgK)**
1	300.19	1.7020
2	669.79	2.5088
3	1515.42	3.3620
4	858.02	2.7620
5	400.98	1.9919
Vapor Cycle		
6	183.96	0.5975
7	3138.30	6.3634
8	2104.74	6.7282
9	173.88	0.5926

Energy Analysis

To determine the mass flow rates of the vapor, $\dot{m}_v$, and the air, $\dot{m}_g$, begin by applying mass and energy rate balances to the interconnecting heat-recovery steam generator to obtain:

$$0 = \dot{m}_g(h_4 - h_5) + \dot{m}_v(h_6 - h_7)$$

$$\Rightarrow \quad \frac{\dot{m}_v}{\dot{m}_g} = \frac{(h_4 - h_5)}{(h_7 - h_6)} = \frac{858.02 - 400.98}{3138.3 - 183.96} = 0.1547$$

Mass and energy rate balances applied to the gas turbine and vapor power cycles give the net power developed by:

$$\dot{W}_{gas} = \dot{m}_g[(h_3 - h_4) - (h_2 - h_1)]$$

$$\dot{W}_{vap} = \dot{m}_v[(h_7 - h_8) - (h_6 - h_9)]$$

$$\Rightarrow \quad \dot{W}_{net} = \dot{W}_{gas} + \dot{W}_{vap}$$

Energy Analysis

$\dot{W}_{net}$ can be defined as:

$$\dot{W}_{net} = \dot{m}_g \left\{ [(h_3 - h_4) - (h_2 - h_1)] + \frac{\dot{m}_v}{\dot{m}_g} [(h_7 - h_8) - (h_6 - h_9)] \right\}$$

$$\dot{m}_g = \frac{\dot{W}_{net}}{\{\dots\}} = 100.87 \text{ kg/s}$$

$$\dot{m}_v = \left(\frac{\dot{m}_v}{\dot{m}_g} \right) * \dot{m}_g = 15.60 \text{ kg/s}$$

Using these mass flow rate values and specific enthalpies from the table above, the net power developed by the gas turbine and vapor power cycles are:

$$\dot{W}_{gas} = 29.03\ MW$$

$$\dot{W}_{vap} = 15.97\ MW$$

$$\dot{Q}_{in} = \dot{m}_g (h_3 - h_2) = 85.30\ MW \quad \Rightarrow \quad \eta = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = 52.8\%$$

Exergy Analysis

The net rate of exergy increase of the air passing through the combustor is:

$$\dot{E}_{f3} - \dot{E}_{f2} = \dot{m}_g[h_3 - h_2 - T_0(s_3 - s_2)] = \dot{m}_g \left[h_3 - h_2 - T_0\left(s_3^o - s_2^o - Rln\left(\frac{p_3}{p_2}\right)\right)\right]$$

Assuming $p_3 = p_2$:

$$\dot{E}_{f3} - \dot{E}_{f2} = \cdots = 59.48\ MW$$

The net rate exergy is carried out of the plant by the exhaust air stream at 5 is:

$$\dot{E}_{f5} - \dot{E}_{f1} = \dot{m}_g \left[h_5 - h_1 - T_0\left(s_5^o - s_1^o - \cancel{Rln\left(\frac{p_5}{p_1}\right)}\right)\right] = 1.39\ MW$$

The net rate exergy is carried out of the plant as water passes through the condenser is:

$$\dot{E}_{f8} - \dot{E}_{f9} = \dot{m}_v[h_8 - h_9 - T_0(s_8 - s_9)] = 1.41\ MW$$

Exergy Analysis

The rates of exergy destruction for the air turbine, compressor, steam turbine, pump, and heat-recovery steam generator are evaluated using $\dot{E}_d = T_0\dot{\sigma}_{cv}$

Air turbine:

$$\dot{E}_d = \dot{m}_g T_0 (s_4 - s_3) = \dot{m}_g\, T_0 \left(s_4^o - s_3^o - R ln\left(\frac{p_4}{p_3}\right)\right) = 3.42\ MW$$

Compressor:

$$\dot{E}_d = \dot{m}_g T_0 (s_2 - s_1) = \dot{m}_g\, T_0 \left(s_2^o - s_1^o - R ln\left(\frac{p_2}{p_1}\right)\right) = 2.83\ MW$$

Steam turbine:

$$\dot{E}_d = \dot{m}_v T_0 (s_8 - s_7) = 1.71\ MW$$

Exergy Analysis

The rates of exergy destruction for the air turbine, compressor, steam turbine, pump, and heat-recovery steam generator are evaluated using $\dot{E}_d = T_0 \dot{\sigma}_{cv}$

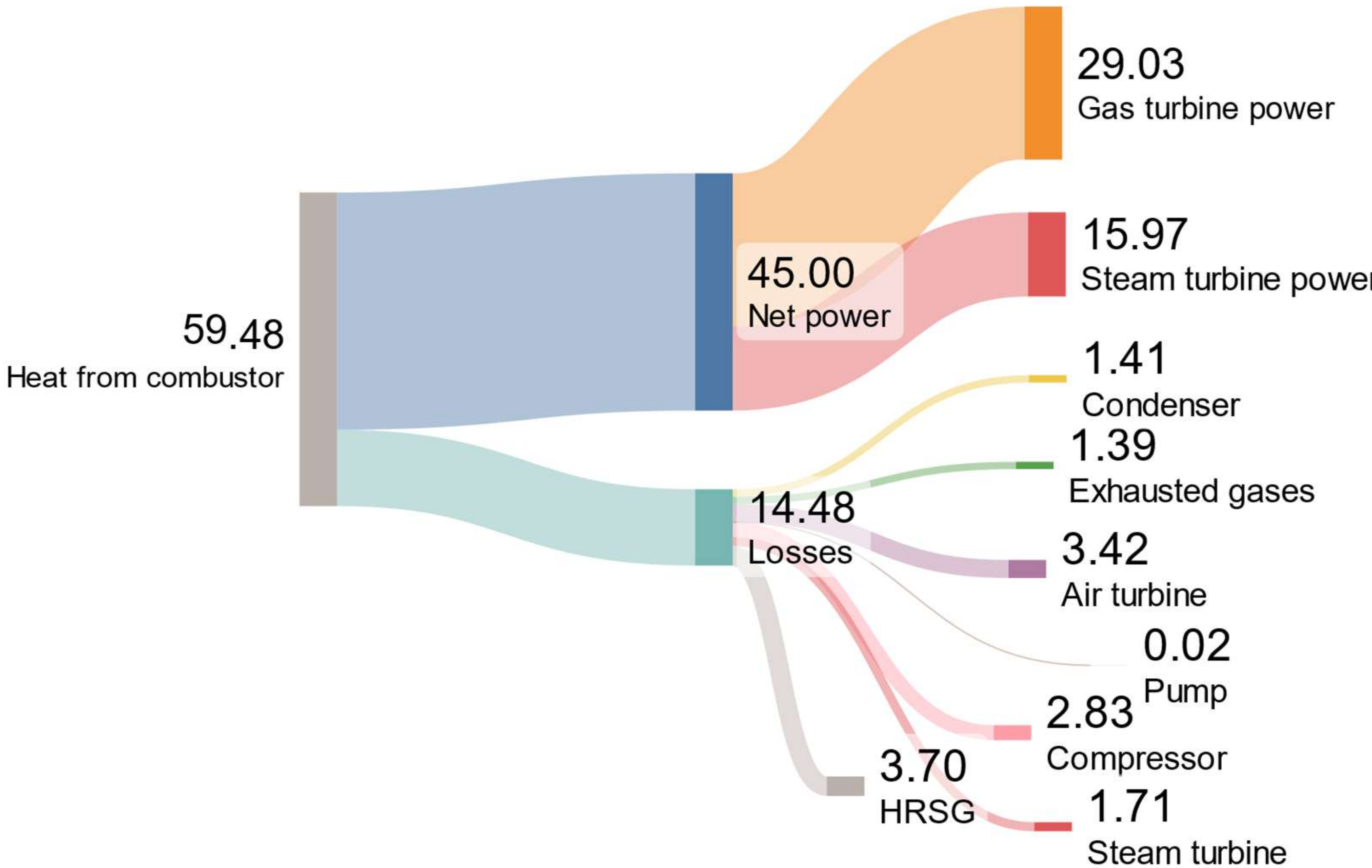
Pump:

$$\dot{E}_d = \dot{m}_v T_0 (s_6 - s_9) = 0.02\ MW$$

Heat-recovery steam generator:

$$\dot{E}_d = T_0 \left[\dot{m}_g (s_5 - s_4) + \dot{m}_v (s_7 - s_6) \right] = 3.68\ MW$$

Exergy Analysis



Note that the exergy destroyed due to combustion is approximately 30% of the exergy entering the combustor with the fuel, i.e. only 70% of the fuel exergy will used by the CCGT.

Thank you for your attention!

Prof. Rodolfo Taccani

e-mail: taccani@units.it