Università degli studi di Padova

COGENERATION

Energy Management Course

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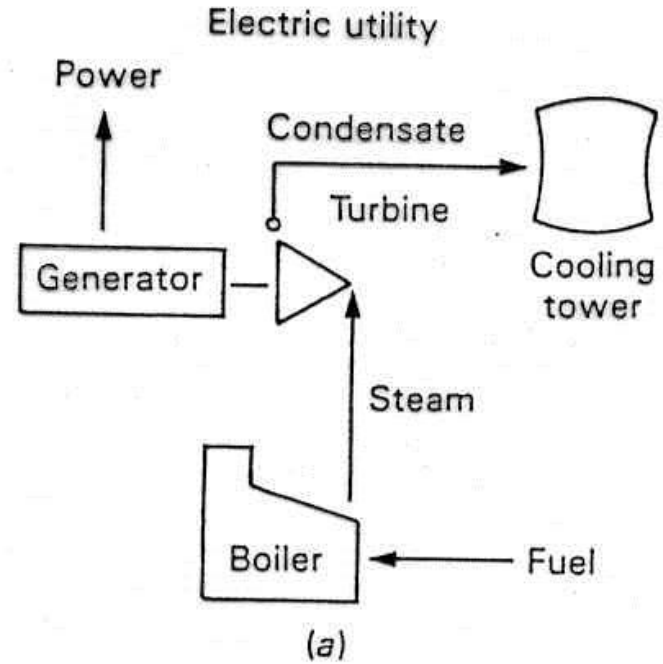
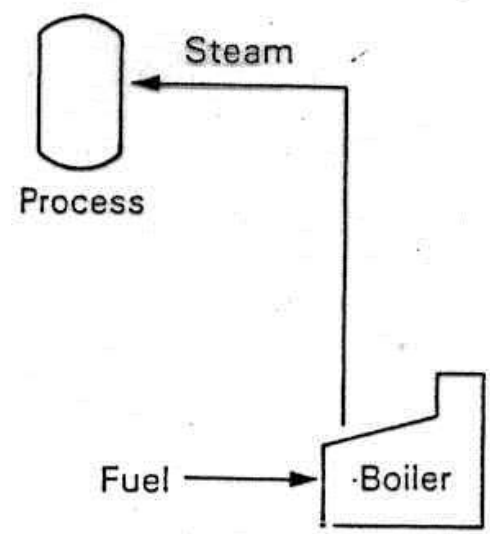
Cogeneration is the contemporary production of mechanical or electric power and of useful heat in a single integrated system.

Once the separate production of mechanical power and heat was realized by a steam power plant operating according to a *Hirn* cycle and by a boiler.
Let us consider such a plant in the figure. The plant provides 45.000 kg/h of steam at 10 ata pressure and 3.9 MW of mechanical power.

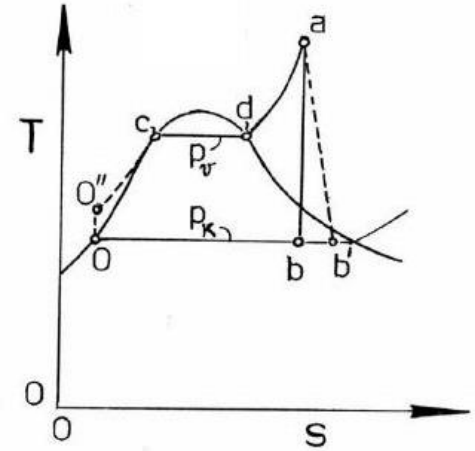


(a)



(b)

A traditional steam plant of such a power might have had an overall efficiency of 34%, then requiring fuel for an equivalent power of:



$$q = \frac{P}{\eta} = \frac{3.9}{0.34} = 11.5 \text{ MW}$$

The heat power is easily estimated because the saturated steam enthalpy at that pressure is 671 kcal/kg with a saturation temperature of 184°C and an enthalpy of saturated liquid of 187 kcal/kg. Consequently the thermal required power is:

$$(671-187)\times 45,000 = 21,762 \text{ Mcal/h (25MW)}$$

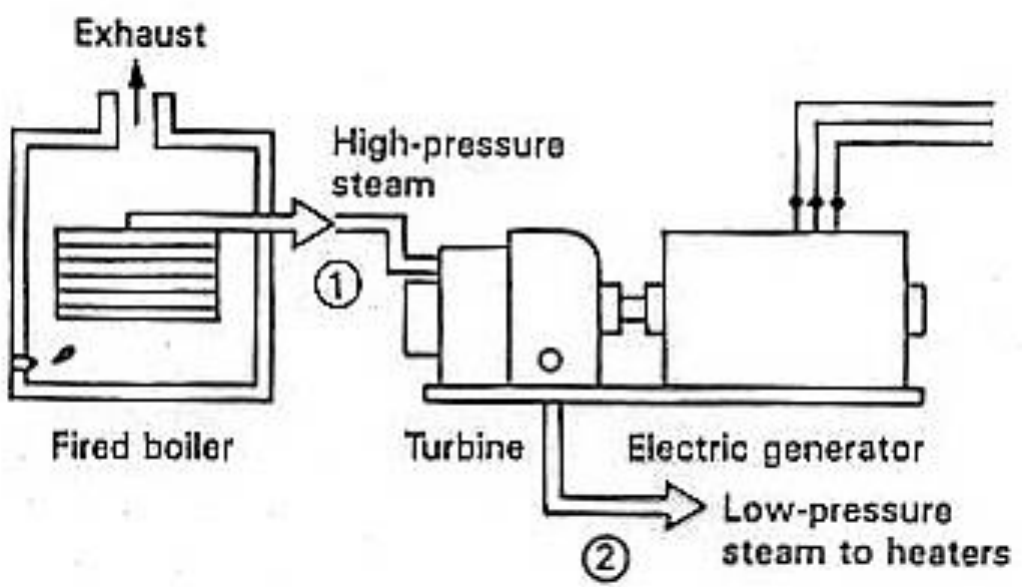
Supposing an 80% efficiency of the steam generator, the required fuel equivalent power is:

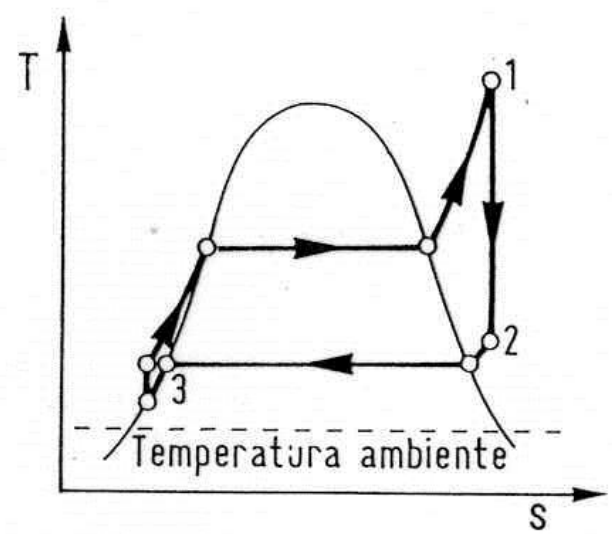
$$\frac{25}{0.8} = 31.2 \text{ MW}$$

Summing up the two separate contributions, the considered traditional system would require about:

$$11.5+31.2 = 42.7 \text{ MW}$$

Let us now consider a system where the steam is generated in the same amount of the previous example (45,000 kg/h) in a steam generator with an 80% efficiency but at a pressure of 52 ata and a temperature of 370°C. That steam can be expanded in a back-pressure turbine till a 10 ata pressure with an isentropic efficiency of 80%:





The enthalpy of overheated vapour at 370°C and 52 ata is 745 kcal/kg. With an isentropic expansion till 10 ata, the final enthalpy would be 652 kcal/kg. Then the real enthalpy drop is:

$$(745-652)\times 0.8 = 74.4 \text{ kcal/kg}$$

The mechanical power is:

$$74.4\times\frac{45,000}{860} = 3.893 \text{ kW}$$

Thermal power is just the same as before because the enthalpy after the back-pressure turbine is 671 kcal/kg.

The overall fuel requirement is here due only to steam production:

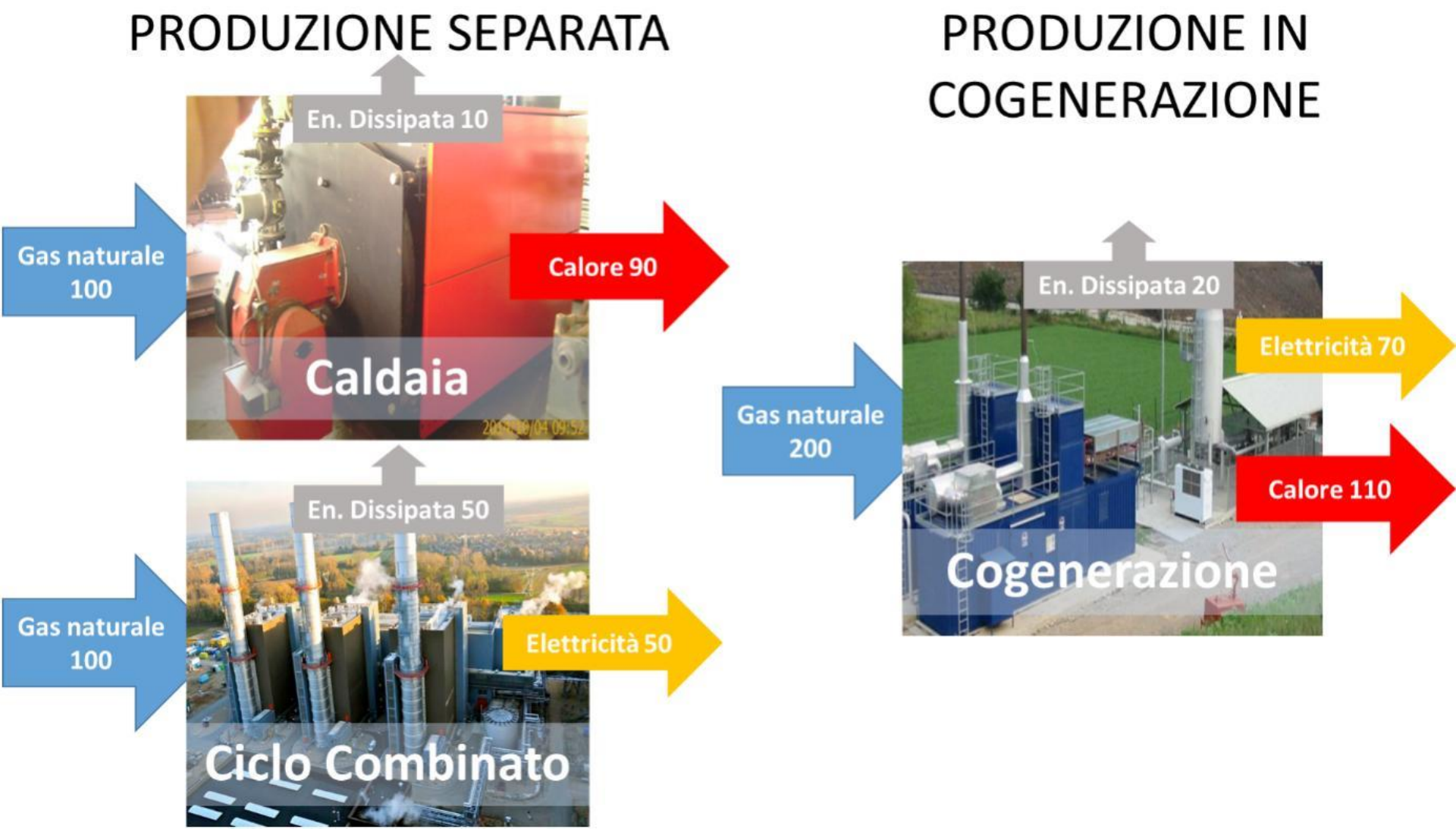
$$(745-187)\times\frac{45,000}{0.8\times 860,000}=36.5\ \text{MW}$$

Then the energy saving is 6.2 MW, that is 15% with a first principle efficiency of about 80%.
The amount of saved fuel can be estimated at 533 kg/h burning oil.

Other cogeneration systems allow well higher savings.

This is the reason why a cogeneration plant is sometimes assimilated to a renewable energy source.

Comparison with modern technologies



A first problem arises from the efficiency evaluation of cogeneration systems as they are characterized by a mechanical/electric efficiency and by a thermal efficiency.
The only evaluation of the first principle efficiency is misleading as a high efficiency might be obtained with a very high thermal efficiency, whereas such a plant is surely worse than a plant with a large electric production even if with a lower first principle efficiency.
A first parameter was introduced in Italy by an act (CIP 6/92). This was the IEN (Index of ENergy):

$$IEN = \frac{E_e}{E_c} + \frac{E_t}{0.9 \times E_c} - a$$

$$a = \left(\frac{1}{0.51} - 1\right)\left(0.51 - \frac{E_e}{E_c}\right)$$

Developing the previous equations the *IEN* can be expressed as:

$$IEN = \frac{\eta_e}{0.51} + \frac{\eta_t}{0.9} - 0.49$$

E_e= yearly produced electricity by the plant net of energy requested by the ancillary services;
E_t=yearly useful thermal energy produced by the plant;
E_c=yearly requested energy by the plant as traditional fossil fuels.

In the Act CIP 6/92 the plant was assimilated to a renewable energy system if:

$$IEN \geq 0.51$$

Which is equivalent to:

$$\frac{E_e}{0.51} + \frac{E_t}{0.9} - E_c > 0$$

When *IEN* exceeds 0.51, the best separate technologies (η_e=0.51 electric efficiency of a combined cycle at the time of the Act and η_t=0.90 for the boiler) are more energy expensive than the cogeneration plant

The plant proposed at the beginning of this lecture did not fit with these requirements because of the difficulty to obtain these results on a yearly basis.

To respect the limits of the Act, a high electric efficiency is required together with an appreciable fraction of thermal energy. A higher fraction of thermal energy would allow to respect the limits in an easier way but with poor economic results, as the Act recognized benefits only to the electric production.

Other indexes were developed with the same purpose of evaluating the efficiency of a cogeneration system. One is the IRE (Indice di Risparmio Energetico – Index of energy saving). It is defined considering the fuel request E_{cc} to obtain electric energy E_e and thermal energy E_t assuming conventional efficiencies for the two separate technologies η_{ec} e η_{tc}:

$$E_{cc} = \frac{E_e}{\eta_{ec}} + \frac{E_t}{\eta_{tc}}$$

Instead a cogeneration system requires:

$$E_c = \frac{E_e}{\eta_e} = \frac{E_t}{\eta_t}$$

The IRE is defined as:

$$IRE = \frac{E_{cc} - E_c}{E_{cc}} = 1 - \frac{E_c}{\frac{E_e}{\eta_{ec}} + \frac{E_t}{\eta_{tc}}} = 1 - \frac{1}{\frac{\eta_e}{\eta_{ec}} + \frac{\eta_t}{\eta_{tc}}}$$

The physical meaning is the % saving with respect to the best two separate technologies.

A further condition was imposed: $LT = \frac{E_t}{E_t + E_e} \geq 0.15$

An index different both from IEN and IRE was given by an Act (261/90) that allows a special fiscal treatment for natural gas consumptions in the electricity production.

A conventional natural gas amount is exempted from paying consumption tax. The amount is obtained multiplying the produced electricity kWh (measured by a fiscal counter) by a coefficient that nowadays is 0,250 m^3/kWh.

The tax to be paid is given multiplying the tax value by a factor:

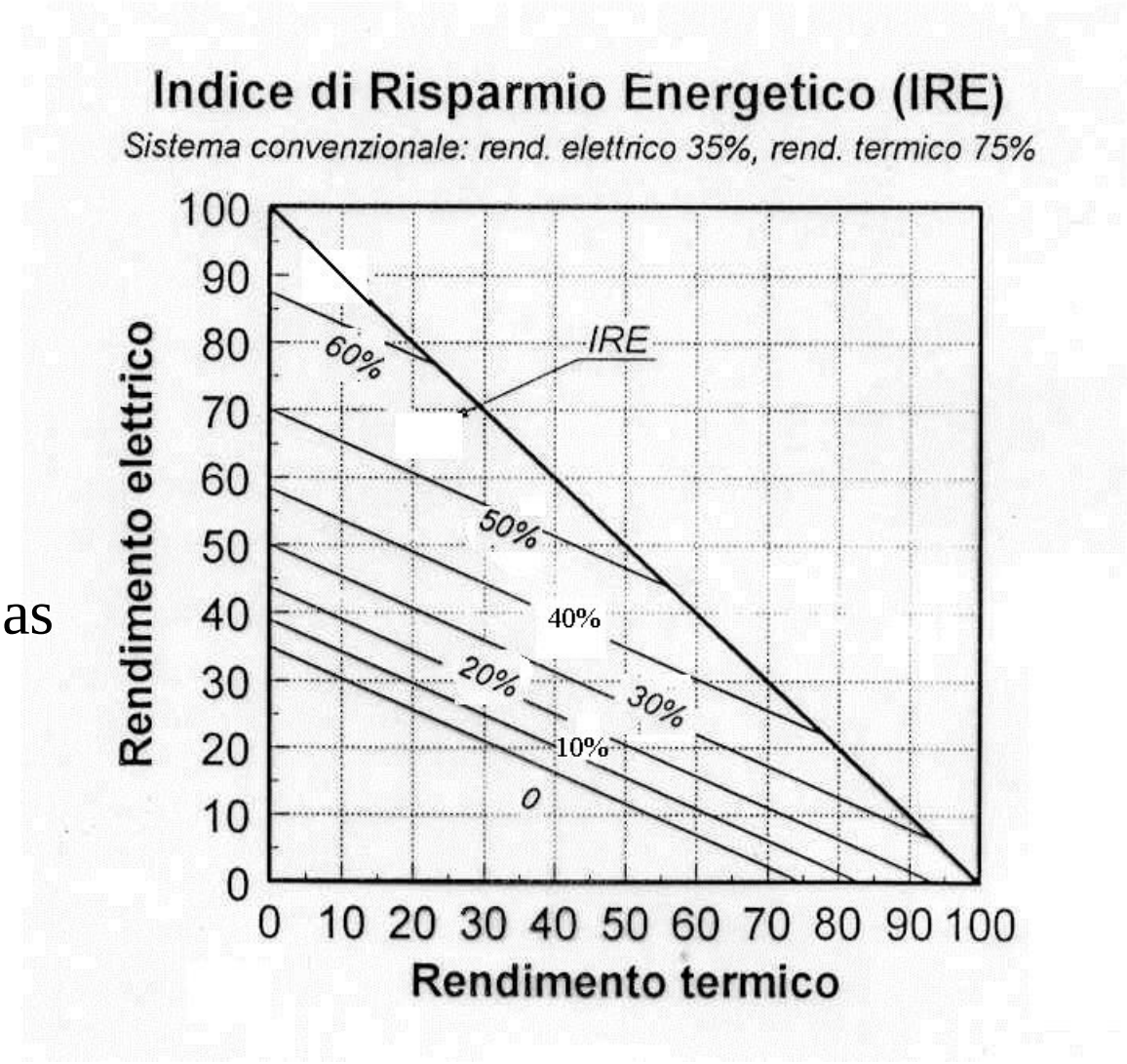
$$I_m = 1 - 0.250 \times CalorificValue \times \eta_m \times \eta_g =$$

$$1 - 0.250 \times \frac{8250}{860} \times \eta_m \times \eta_g = 1 - \frac{\eta_m \times \eta_g}{0.417}$$

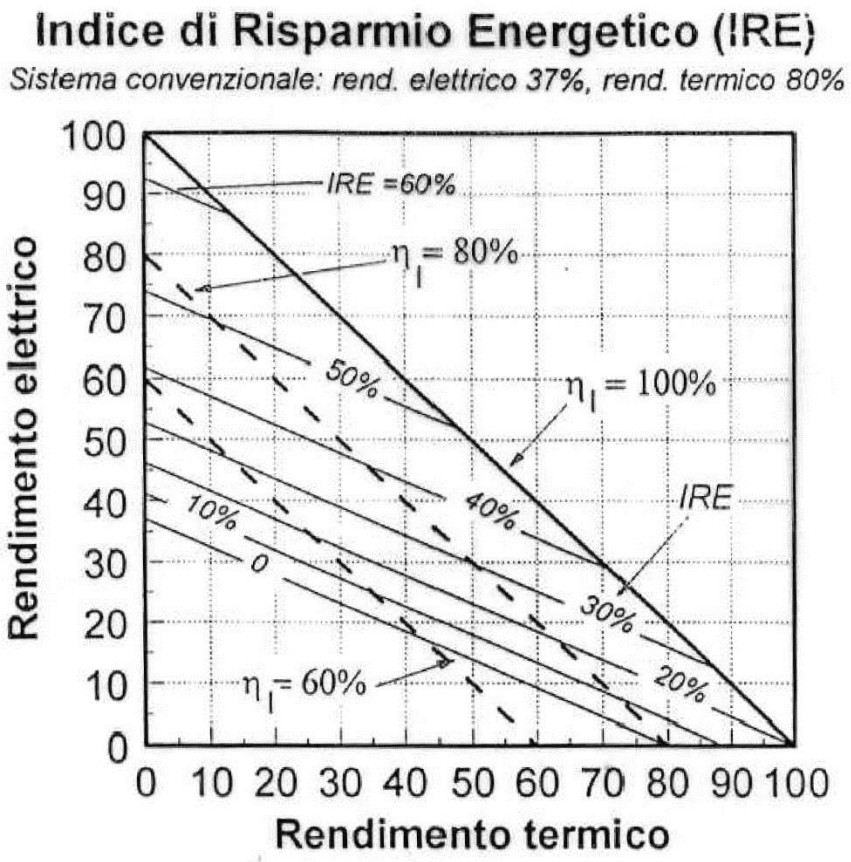
η_m = Mechanical efficiency;
η_g = Cycle efficiency.

In the above situation, no tax is due for an efficiency of about 42%.

A first evaluation can be proposed regarding a situation with an underdeveloped energy system with an electric efficiency of 0.35 and a thermal efficiency of 0.75 (yearly values).
The figure allows an immediate IRE evaluation as for every point within the triangular area gives the % saving with respect to the conventional situation just considered.

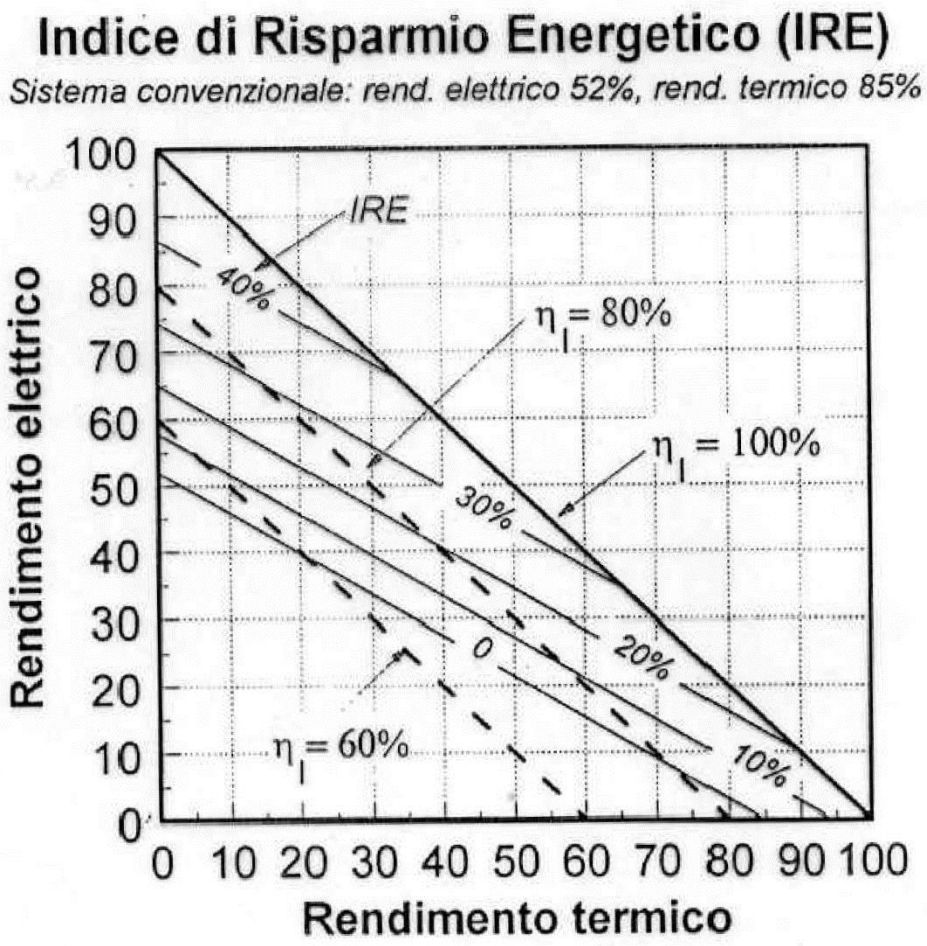


The following figure represents a situation with higher conventional technology efficiency, that is 37% for the electricity and 80% for thermal energy.
The constant IRE curves are represented together with the first principle efficiency curves (60-80-100%).



The technological evolution considers the electricity produced by combined cycles (electric efficiency 52%) and thermal efficiency 86%.

Of course the evaluation of a same cogeneration system is strictly dependent on the assumed conventional efficiencies.



The factors that suggest a possible advantage of a cogeneration system are the following:

GENERALLY

• high electric and thermal power;
• the most constant consumption of electricity and heat, or variable in a similar ratio;
• assurance of an electric and thermal request in the future;
• low cost.

ELECTRICITY DEMAND

• high unitary cost of electricity;
• high value attributed to the elimination of electric black out;
• low cost of connecting to the electric grid.

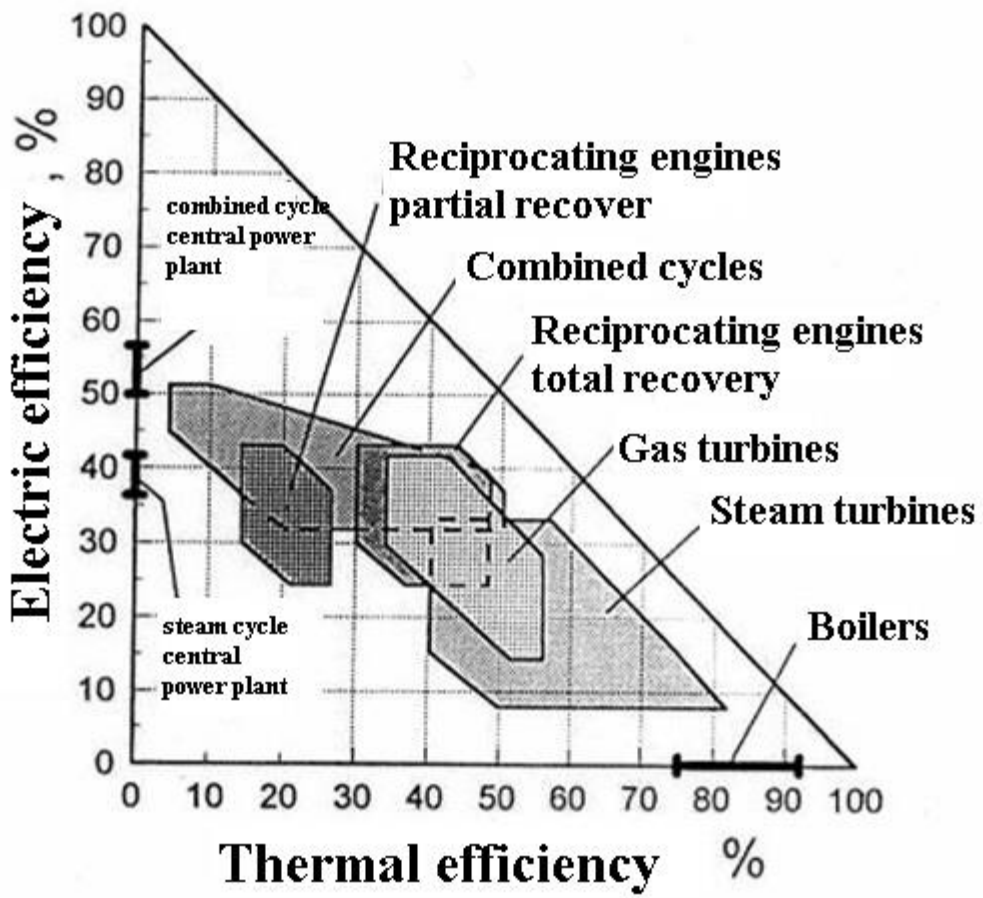
THERMAL DEMAND

• high unitary cost of thermal energy (district heating for example);
• low temperature requested for the hot fluid (higher heat recovery possibilities);
• limited extension of the thermal distribution grid (lower investment cost for the grid, thermal losses and pumping costs.

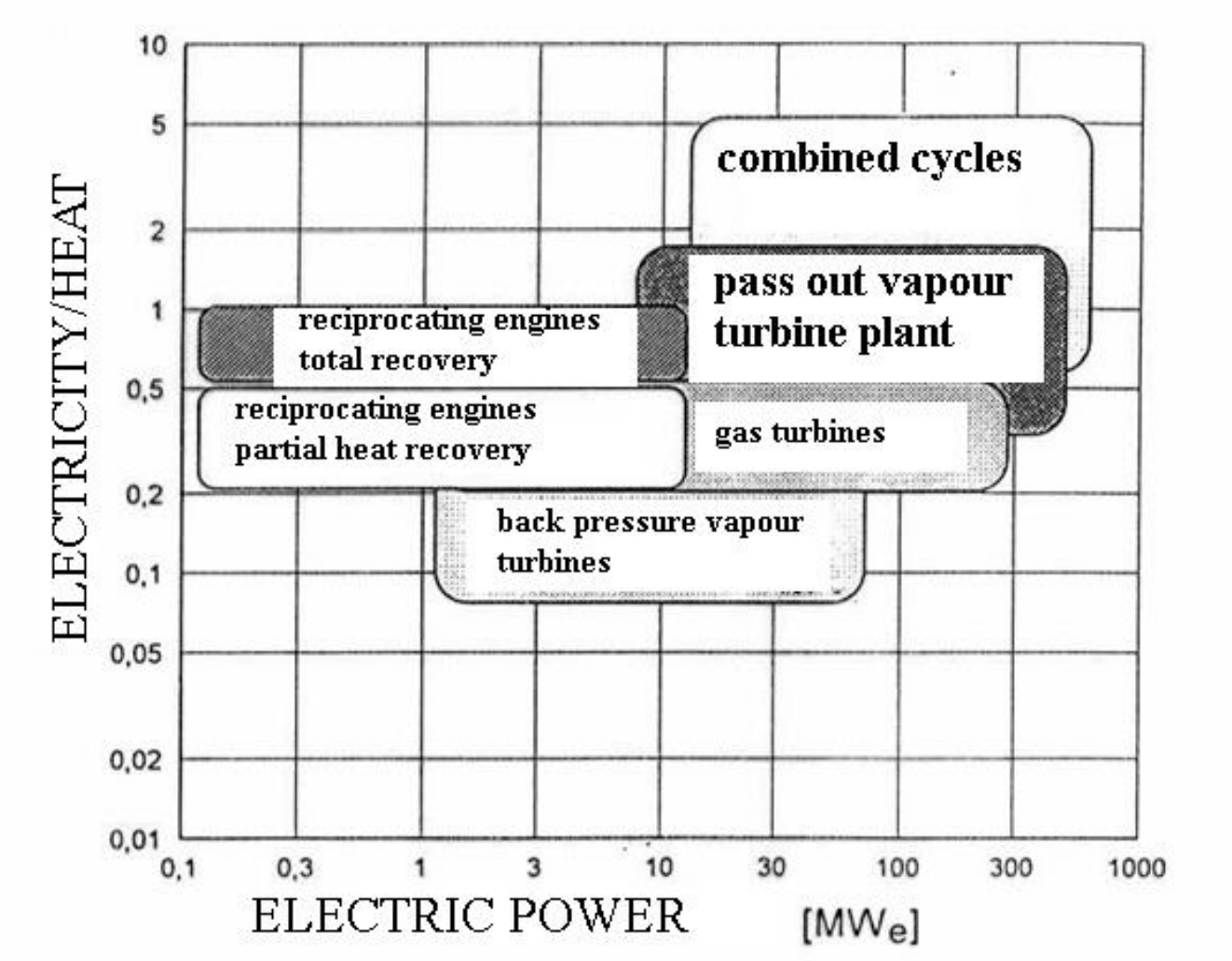
COOLING DEMAND

• Complementarity with electric demand (compression chillers) or thermal (absorption chillers) as to reduce the seasonal variation of the demand.

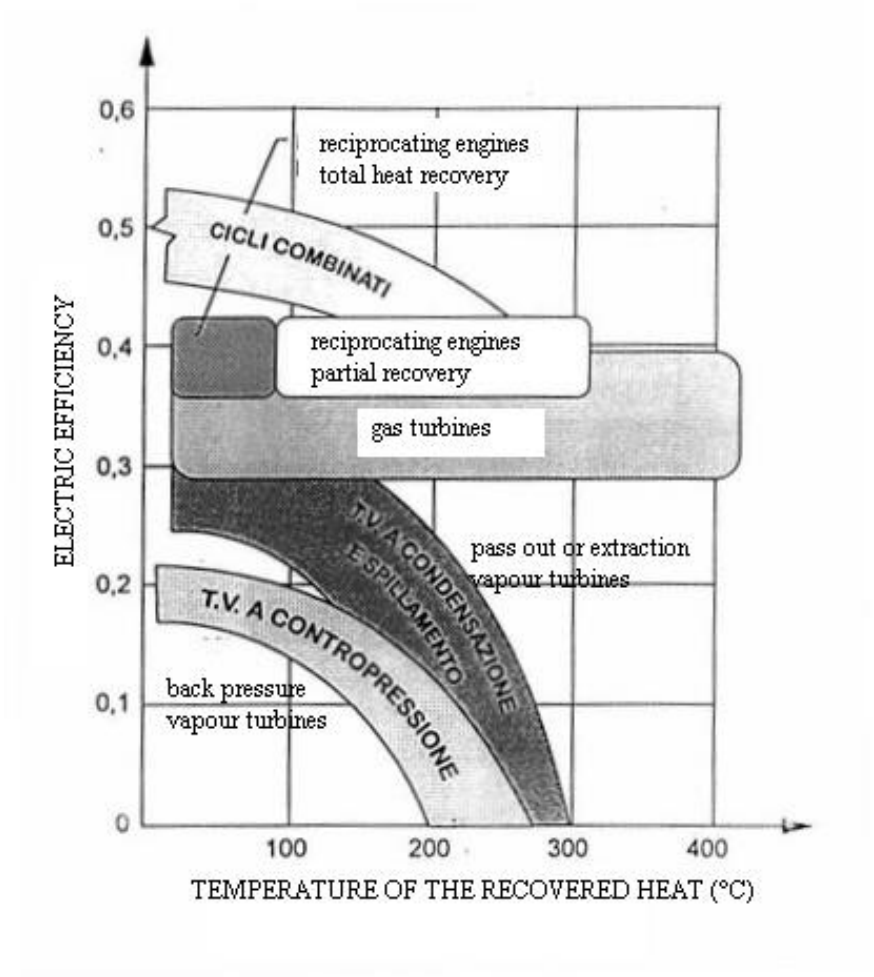
The technologies to be employed together with the possible efficiencies are represented in the IRE triangle in the figure:



However the available systems are characterized by different electric/thermal ratios and by different capacity ranges.



A final important element to be considered is the influence of the temperature of the cogenerated heat on the electric efficiency that can be relevant for some technologies and not for others:



Industrial cogeneration is conventionally classified into 4 electric power ranges:

- microcogeneration from 10 to 100 kW;
- low range power from 100 kW to 2 MW;
- medium range from 2 to 25 MW;
- high power range over 25 MW.

As regards microcogeneration, the only commercial technology is reciprocating internal combustion (i.c.) engines with some emerging technologies such as microturbines, Stirling engine and fuel cells.

In the low range, there is competition between reciprocating engines and gas turbines, where the former prevails for lower powers and the latter for higher powers.
In the medium range the competition is between gas turbines and combined cycles whereas in the high-power range, only combined cycles and vapor power plants.

In order to choose the technology in a right way, it is of paramount importance to know the trend of thermal (split according to the temperature) and electricity demand.

A typical index is the ratio between the thermal and total energy demand. When the thermal heat requirement is null I_t=0 whereas I_t=1 for electric requirement null:

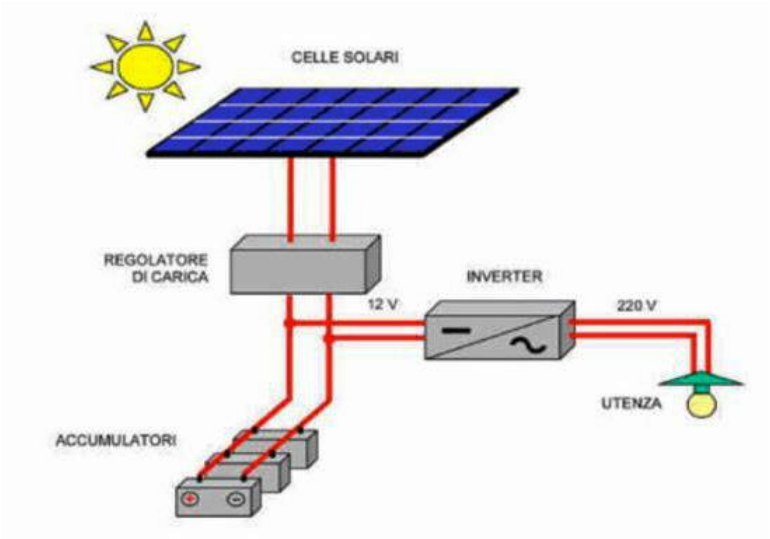
$$I_t = \frac{E_t}{E_t + E_e}$$

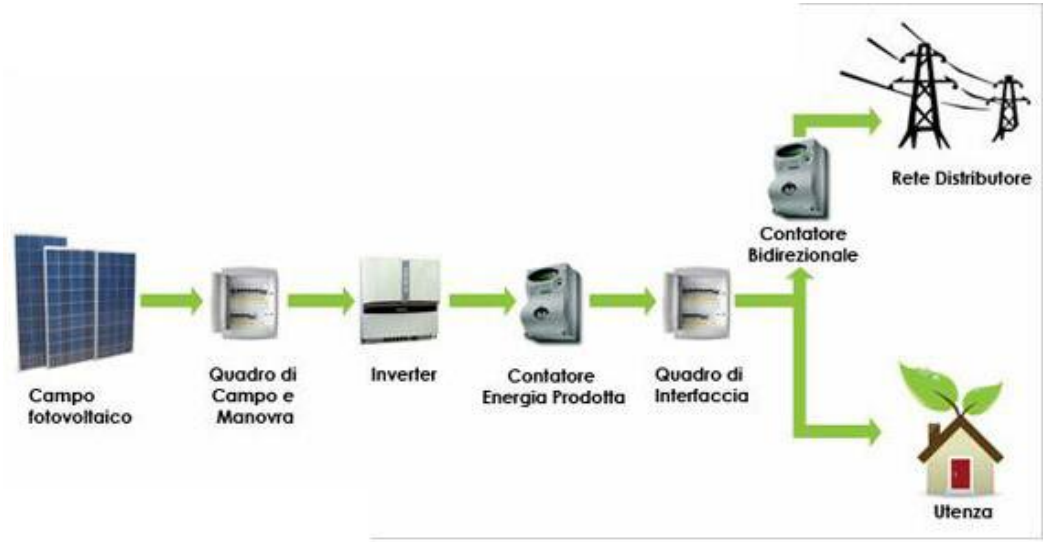
Industrial activities can be conventionally divided into three groups:

- low thermal index I_t<0.2 (mechanical and carpentry industry)
- medium thermal index 0.2<I_t<0.5 (textile, electronics)
- high thermal index I_t>0.5 textile, paper, tannery, chemical, building materials, etc.

Cogeneration is usually profitable when the thermal index is at a medium value and it is almost surely profitable for a high thermal index.

As regards the relation between the cogeneration system and the grid, the operation can be in island or in parallel (much more frequent).
For the parallel connection, different situation can exist for example with only taking the integration energy or transfer to the grid of the electricity surplus or taking all the requirement in case of out of order.





The plant can be sized according to:

- the thermal demand, taking from the grid the required electricity or selling to the grid the potential electricity surplus;
- the electricity demand eventually not exploiting completely the thermal recovery;
- either supplying the missing heat part with a traditional boiler.

RECIPROCATING I.C. ENGINES COGENERATION

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Reciprocating i.c. engines usually operate with natural gas or diesel oil.
Fuel oil is employed only for high capacity engines.

Engine efficiency can range from 25% to more than 40% for higher values for high capacity engines.
Heat recovery is at two different thermal levels.
One temperature level is about 80-90°C and it is due mainly to cylinder jacket cooling. The other level is at a well higher temperature and it is due to exhaust gases cooling.

A possible presentation is to consider the different characteristics of the engines as the capacity increases.

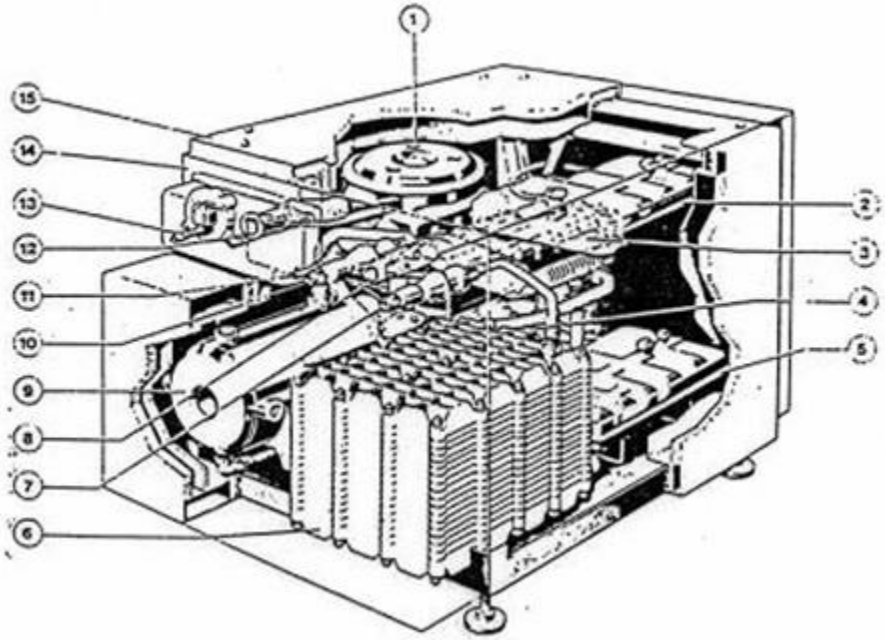
A historical position is occupied by TOTEM (*TOTal Energy Module*) in the low capacity range.
It was one of the first realization in the world of small cogeneration systems, produced by the car manufacturer FIAT at the end of the 70's. It was based on a largely produced engine type 903 rods and rocker motor. It was the motor of some small cars like model 127 and A112. Then the engine derives directly from automotive.

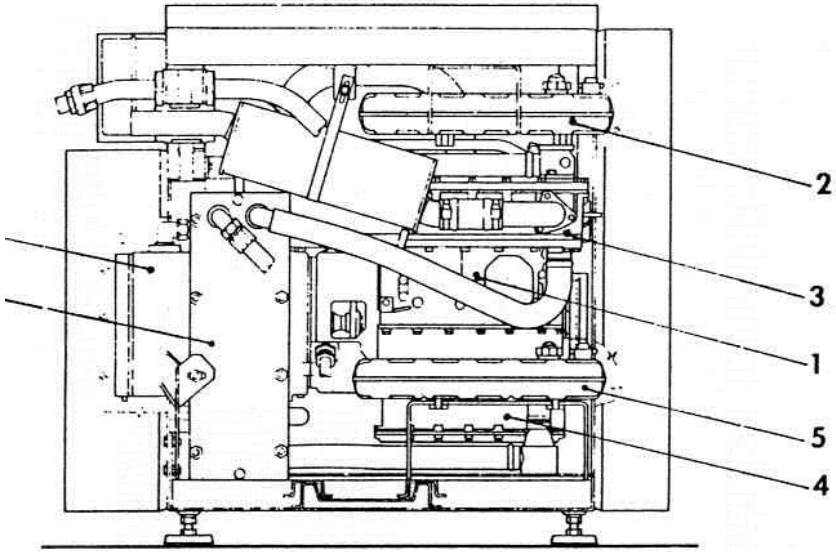
The risk is a limited life of the motor.

It operated at a rotation speed of 3000 rpm producing an electric power of 15 kW.

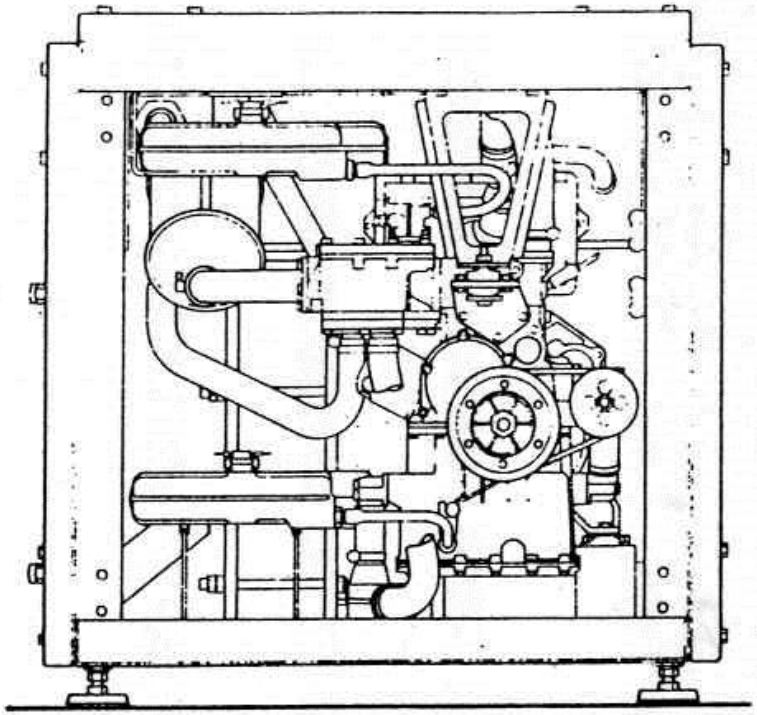
TOTEM could use natural gas, LGP, biogas.
The electric efficiency was 27% and thermal efficiency 70%.

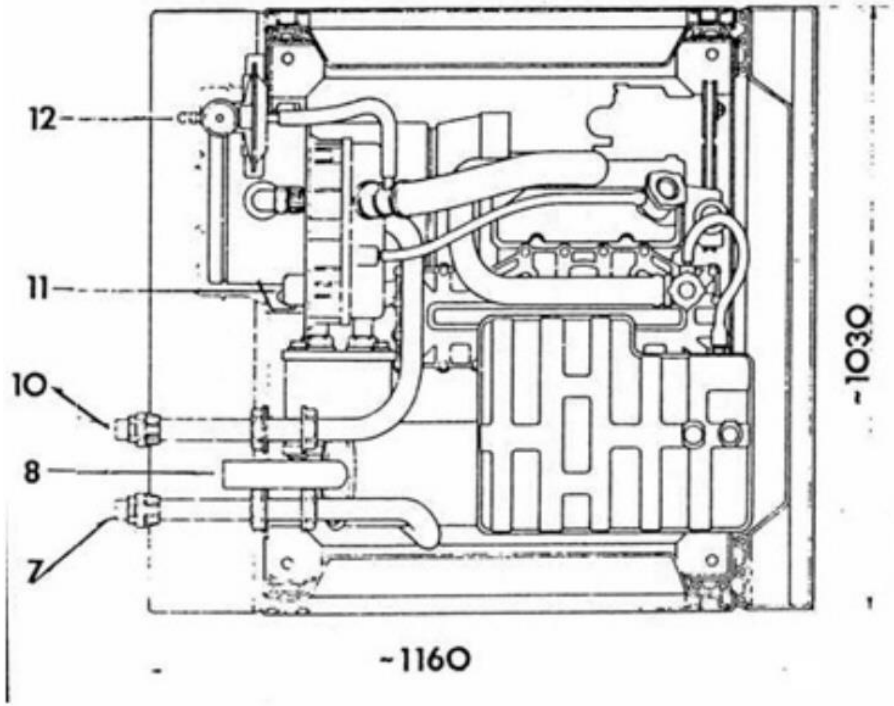
Funzionamento		• In PARALLELO alla rete elettrica • Potenza	• In PARALLELO con funzione di EMERGENZA anti black-out o ad isola in INDIPENDENTE • Potenza costante o rispettivamente variabile
Dimensioni		**Modulo di cogeneraz.**	**Mod. di cogeneraz.** / **Gruppo di eccitaz.**
LarghezzaxLunghezzaxAltezza	mm	903x1130x1000	903x1130x1000 / 800x370x350
Peso approssimativo	kg	490	520 / 130
Prestazioni			
Potenza in ingresso	kW (MJ/h)	56,2 (202,35)	15 (54) ÷ 56,2 (202,35)
Potenza elettrica nominale	kW	15 (cosφ = 0,86)	0 ÷ 14,5
Pot. term. in uscita	kW (kcal/h)	39 (33500)	13 (11180) ÷ 39 (33500)
Rendimento alla potenza max	%	93 ÷ 97	90 ÷ 97
Consumi			
Metano (CH_4 35,5 MJ/m³)	m³/h	5,7	2,0 ÷ 5,7
G.P.L. (50,5 MJ/kg)	kg/h	4,0	1,5 ÷ 4,0
Biogas (60% CH_4 22,6 MJ/ m³)	m³/h	(13,8 kW max) 8,4	(13,3 kW max) 3,0 ÷ 8,4
Altri combustibili		Consumi da valutare in base al potere calorifico	Consumi da valutare in base al potere calorifico
Circuito idraulico secondario		**Dati alla potenza nominale**	**Dati alla potenza massima**
Temperatura max acqua in ingresso	°C	70	70
Temperatura max acqua in uscita	°C	86	86
Portata nominale acqua	l/h	2000	2000
Generatore elettrico		**Asincrono trifase con eccitazione da rete**	**Asincrono trifase con autoeccitazione**
Tensioni trifasi di rete	V/Hz	380/50 208/60 415/50	380/50 208/60 415/50
Avviamento		Da generatore elettrico stella/triangolo	Motorino d'avviamento C.C.
Collegamento con l'utenza		Triangolo	Stella con neutro accessibile
Condizioni operative			
Temperatura ambiente	°C	0 ÷ 40	0 ÷ 40
Umidità relativa	%	0 ÷ 95	0 ÷ 95





1 – 127 gas powered motor
2 – water tank capacity 9 kg
3 – heat exchanger exhaust gas-water
4 – heat exchanger lubricating oil-water
5 – lubricating oil tank capacity 8 kg
6 – heat exchanger water - water
7 – outlet hot water
8 – exhaust gas discharge
9 – Electric generator (15 KW - 380 V Trifase - 50 Hz)
10 – Water inlet
11 – Air inlet
12 – Gas supply





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Weight C.A. 560 kg.

The characteristic indexes for this engine are:

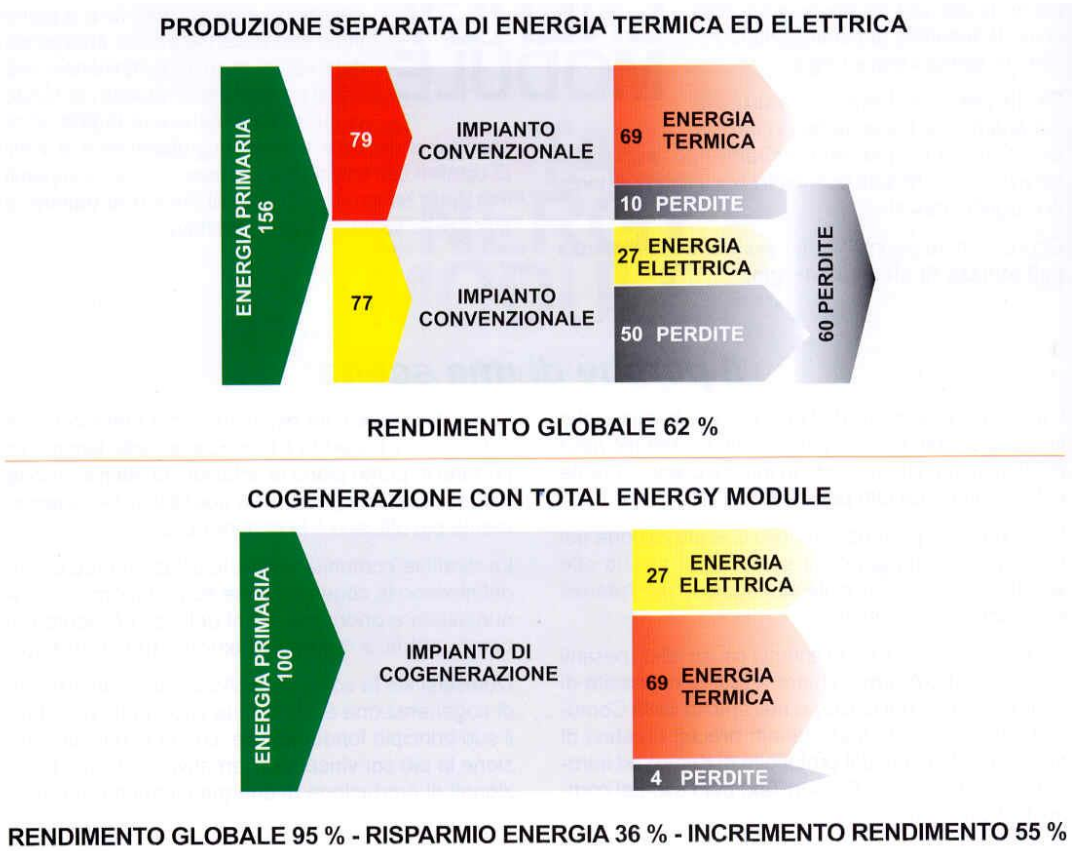
Ratio between fuel energy and electric energy:

3.64 kWh/kWh

Thermal to Electric ratio:

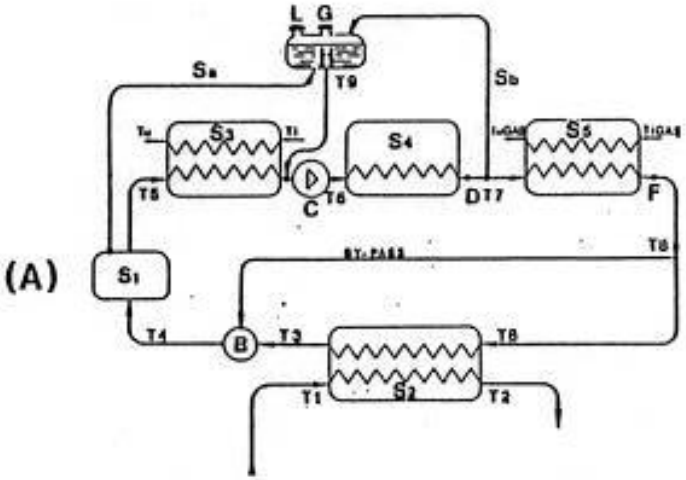
$$\frac{T}{E} = 2.6$$

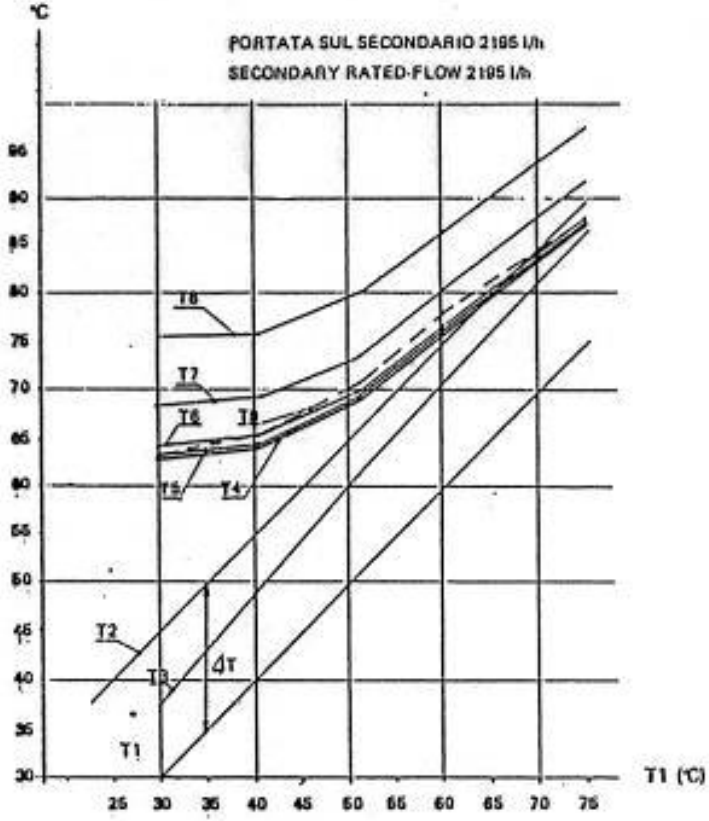
Specific fuel consumption: 0.38 m^3/kWh



The system was designed to heat up 2 m^3/h water from 70 to 86°C.

When heat is not required, an air-cooled radiator is obviously provided.





Some TOTEM models were recently proposed on the market

MODELLO		TOTEM 10	TOTEM 20	TOTEM 25
POTENZA @ dati rilevati a potenza elettrica nominale e con acqua ingresso 70°C se non diversamente specificato; metano 20 mbar, dati riferiti a pci=10,2 kWh/Nm³; aria 25°C e 101,3 kPa				
Potenza elettrica nominale	kW	10,0	20,0	25,0
Autoconsumi	kW	0,195	0,205	0,205
Intervallo modulazione elettrica	kW	≥ 5	≥ 7,5	
Potenza termica nominale	kW	21,6 (25,2*)	41,9 (48,5*)	50,2 (57,6*)
Rendimento elettrico netto	%	29,6	31,2	32,5
Rendimento totale	%	93,6 (104,3*)	96,5 (106,8*)	97,8 (107,4*)
Combustibile		metano		
Consumo combustibile	Nm^3/h	3,31	6,28	7,54
Potenza in ingresso	kW	33,7	64,1	76,9
DIMENSIONI E PESI				
h x p x l (con pannelli montati - versione standard)	cm	128 x 78 x 181		
Peso (a pieno carico)	kg	720	780	

* acqua in ingresso a 35 °C

In the low capacity range, other models can be considered with the characteristics listed in the table, but without particular differences with TOTEM

Tabella 2: Alcuni moduli di microcogenerazione

		Intelligen *Alpha-550*	GRI-KOHLER *RCS*	Sachs	T.E.M. *Totem 7*	Intelligen *Model 2010*	Kubota *GP 15E*	T.E.M. *Totem 15*
Potenza	P_{in}(kW)	23,1	22,13	20	29	26 36,6 46,8	66,5	56,2
	P_{el}(kW)	4,8	5	5,5	7	8 10 12	15	15
	P_{ter}(kW)	11,4	13,4	12,5	19,9	14,9 22,2 29,3	35,4	39
Rendimenti	η_{el}(%)	21	23	27	24,1	31 27 26	25	26,7
	η_{ter}(%)	49	60	62	68,6	57 61 63	53,2	69,4
	η_{Tot}(%)	92,8* 71,5**	83	90	92,7	88*	78,2	96,1
Peso e dimensioni	(kg)	-	-	510	400	463	1070	490
	Lunghezza (mm)	-	1323	1060	1130	1473	1840	1130
	Altezza (mm)	-	813	1000	1000	914	1630	1000
	Profondità (mm)	-	749	720	903	762	710	903
Motore	Marca	Lister-Petter Diesel	Yanmar Diesel	Sachs	Fiat	Ford	-	Fiat
	N° cilindri	2	3	1	4	4	-	4
	Cilindrata (cm^3)	903	658	583	903	999 1118 1297	-	903
Generatore		Asincrono (monof.)	Sincrono (monof.)	Asincrono (monof.)	Asincrono (trif.)	Asincrono (monof.)	-	Asincrono (trif.)
Rumore dB(A)		68,4	65	52	-	64	70	-

* Elettrico+Termico+Aria Calda ** Elettrico+Termico

In the low capacity range, some engines were designed just for cogeneration modules. The main characteristic is a low rotation speed and a design for a useful life of at least 30,000 hours.

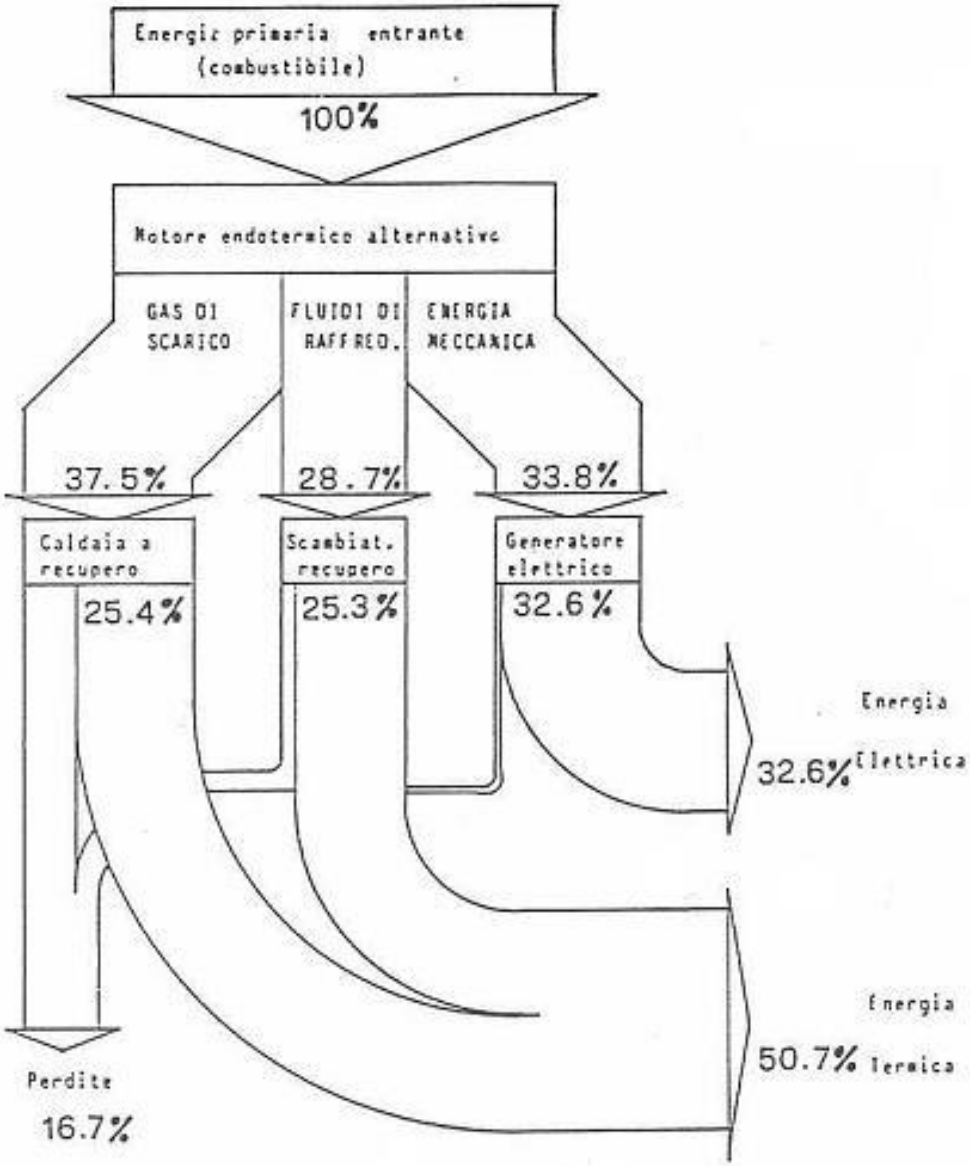
Model	Power introduced kW	Electrical power kW	Thermal power kW	Electrical yield %	Total yield %	Fuel consumption mc/h	Length mm	Width mm	Height mm
BB 35 AM	118	35	71	29,7	89,9	12,3	2810	1000	2000
BB 35 SM	118	35	71	29,7	89,8	12,3	2810	1000	2000
BB 60 AM	200	60	120	30	90	20,9	3310	1000	2100
BB 60 SM	200	60	120	30	90	20,9	3310	1000	2100
BB 80 AM	242	80	138	33,1	90,1	25,2	3690	1100	1650
BB 90 AM	290	90	170	31	89,7	30,2	3310	1100	2400
BB 90 SM	290	90	170	31	89,7	30,2	3310	1100	2400
BB 103 SM	289	103	138	35,6	83,4	30,1	3690	1100	1560
BB 124 SM	363	124	203	34,2	90,1	37,8	3850	1300	1850
BB 125 AM	376	125	210	33,2	89,1	39,2	3310	1200	2400
BB 125 SM	376	125	210	33,2	89,1	39,2	3310	1200	2400
BB 140 SM	392	140	207	35,7	88,5	40,9	3200	1400	2200

Electric efficiency can reach 32%.
For example, the engine that gives 90 kW electricity is a 6 cylinders aspirated motor with 11 liters displacement and a 3200 kg weight.

Unità di cogenerazione sincrone alimentate a gas naturale (P.C.I. = 34,53 MJ/Sm^3)

Modello	U M	BB 20 S	BB 30 S	BB 50 S	BB 60 S	BB 90 SS
Potenza assorbita	kW	75	108	174	200	282
Potenza elettrica	kW	20	30	50	60	90
Potenza termica	kW	50	69	107	120	170
Rendimento totale	%	93,3	91,7	90,2	90,0	92,2
Motore		Teledyne CES	VALMET	VALMET	VALMET	SCANIA
Tipo		TM 27	420 G	634 G	634 G	11 G
Pot. netta 1500 g/1' (ISO 3046)	kW	22	33	54	65	97
Sistema di alimentazione		Aspirato	Aspirato	Aspirato	Aspirato	Aspirato
Cilindrata	litri	2,700	4,400	7,400	7,400	11,020
N° cilindri		4	4	6	6	6
Rapporto di compressione		8,2 : 1	9,5 : 1	9,5 : 1	9,5 : 1	11 : 1
Consumo gas naturale	Sm^3/h	7,8	11,3	18,1	20,8	29,4
Consumo olio lubrificazione	g/kWh	0,5	0,3	0,3	0,3	0,6
Alternatore		STAMFORD	STAMFORD	STAMFORD	STAMFORD	STAMFORD
Tipo		UIC 224 D	UIC224 D	UIC 274 D	UIC 274 D	UIC 274 E
Potenza	kW	20	30	50	60	90
Tensione	V	400	400	400	400	400
Corrente nominale	A	36	54	90	108	162
Fattore di potenza		0,8 - 1,0	0,8 - 1,0	0,8 - 1,0	0,8 - 1,0	0,8 - 1,0
Frequenza	Hz	50	50	50	50	50
Rendimento	%	89	89,3	92,4	91,9	92,6
Classe di isolamento		H	H	H	H	H
Grado di protezione		IP 21	IP 21	IP 21	IP 21	IP 21
Recupero termico		Alfa - Laval Bowman	Alfa Laval Bowman	Alfa Laval Bowman	Alfa Laval Bowman	Alfa Laval Bowman
Max temp. mandata/ritorno	°C	82/70	85/72	90/70	90/70	90/70
Portata acqua	m^3/h	3,6	4,6	4,6	5,2	7,3
Rumorosità a 1 metro	dB(A)	< 70	< 70	< 70	< 70	< 70
Lunghezza	m	2,780	2,780	3,210	3,210	3,690
Larghezza	m	0,900	0,900	0,900	0,900	1,100
Altezza	m	1,250	1,250	1,460	1,460	1,560
Massa	kg	1150	1540	2200	2200	3200

Here is the flowchart of a model engine *BiBloc* 90A (motor *Valmet*)



Lubricating oil consumption is evaluated from 0.3 to 0.6 g/kWh whereas the lubricating oil has to be changed every 400 hours on averge.

As the sump capacity is 22 liters, the lubricating oil use is from 32 to 42 liters every 400 hours for the 90 kW engine, that is a hourly oil cost of 80 g. This means about 1 g/kWh for a cost of about 1 c€/kWh.

The characteristic indexes for this engine are:

Ratio between fuel energy and electric energy:

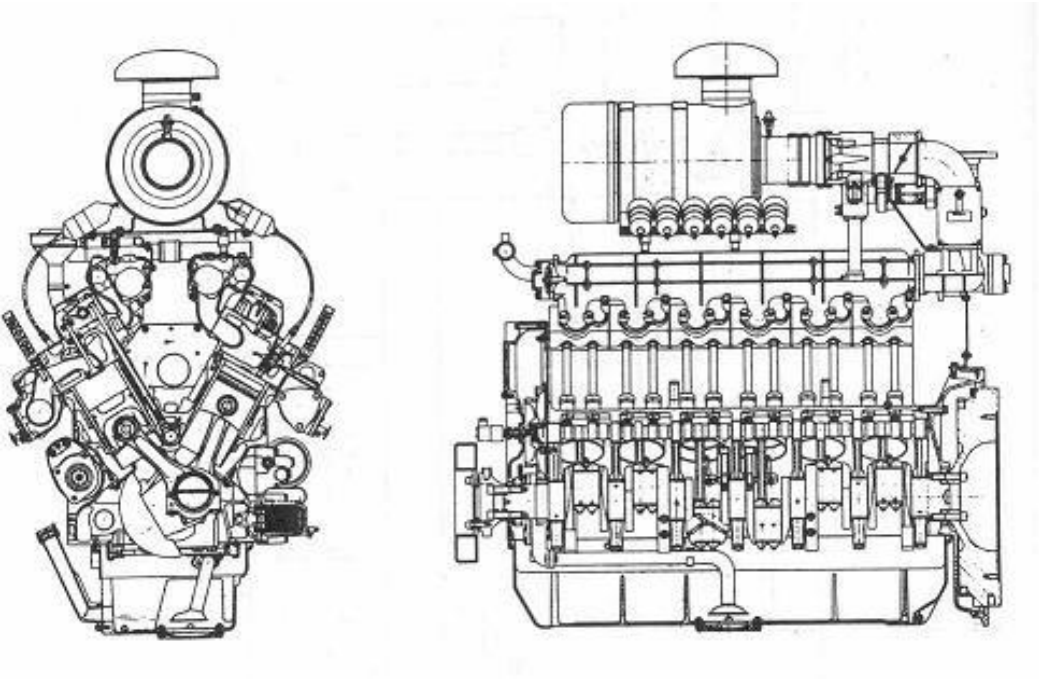
3.13 kWh/kWh

Thermal to Electric ratio:

$$\frac{T}{E} = 1.9$$

Specific fuel consumption: 0.327 m^3/kWh

A typical product in the medium capacity range are the ***Jenbacher*** engines

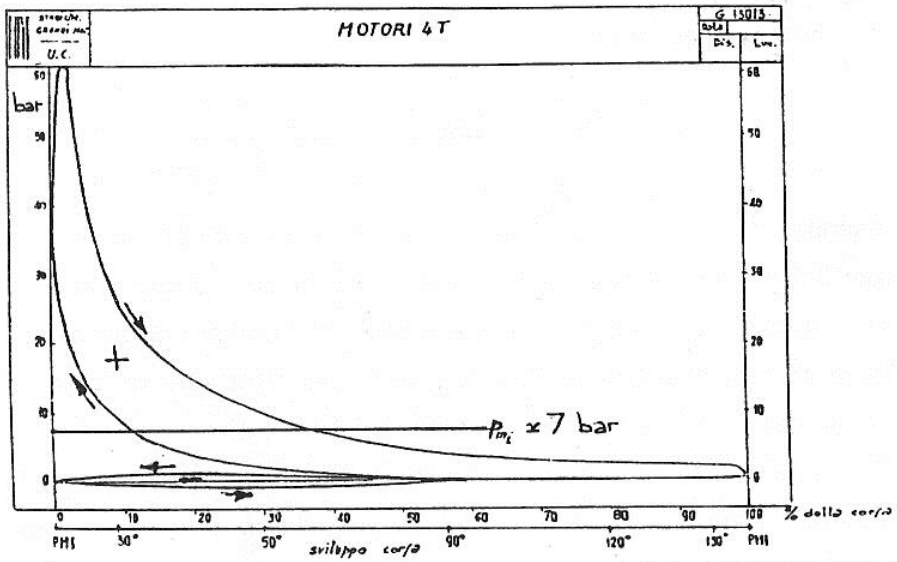


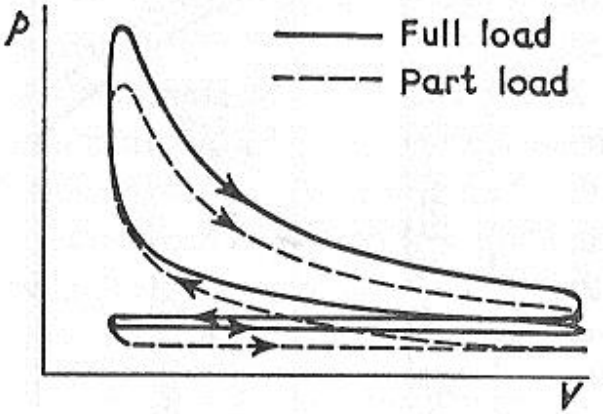
A typical engine is the JMS 212 model, whose electric power is 382 kW, mechanical efficiency 38% (electric efficiency 36.2%).

Some data regarding this engine are an average lubricating oil consumption of 0.12 kg/h and a threefold heat recovery: 260 kW from water and lubricating oil cooling, 230 kW from exhaust gases and 55 kW from the turbocompressor *intercooler* .

The engine can operate at 75% of nominal electric power producing 286 kW electricity, whereas the heat recovery is 430 kW with the following three contributions 190, 213 e 23 kW.

At 50%, 190 kW electricity is produced, and the heat recovery is 313 kW (160,153, 0). The average effective pressure passes from 12.74 bar at nominal operation to 9.55 at 75% and 6.37 at 50%. Correspondently, the efficiency goes from 36% to 34% and to 31%.





The characteristic indexes for this engine are:

Ratio between fuel energy and electric energy:

2.76 kWh/kWh

Thermal to Electric ratio:

$$\frac{T}{E} = 1.4$$

Specific fuel consumption: 0.288 m^3/kWh

At 75% the values are the following:

Ratio between fuel energy and electric energy:

2.94 kWh/kWh

Thermal to Electric ratio:

$$\frac{T}{E} = 1.5$$

Specific fuel consumption: 0.307 m^3/kWh

At 50% the values are:

Ratio between fuel energy and electric energy:

3.21 kWh/kWh

Thermal to Electric ratio:

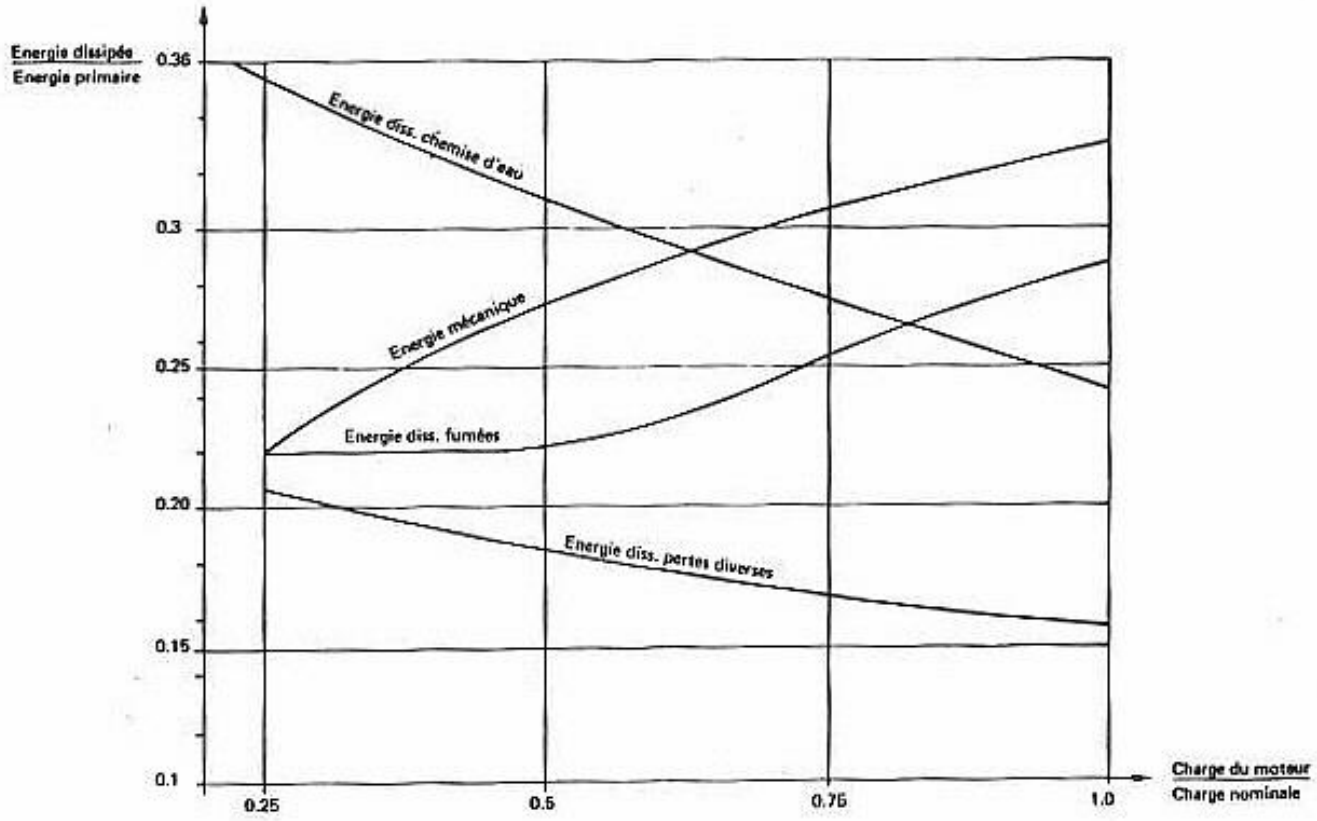
$$\frac{T}{E} = 1.65$$

Specific fuel consumption: 0.336 m^3/kWh

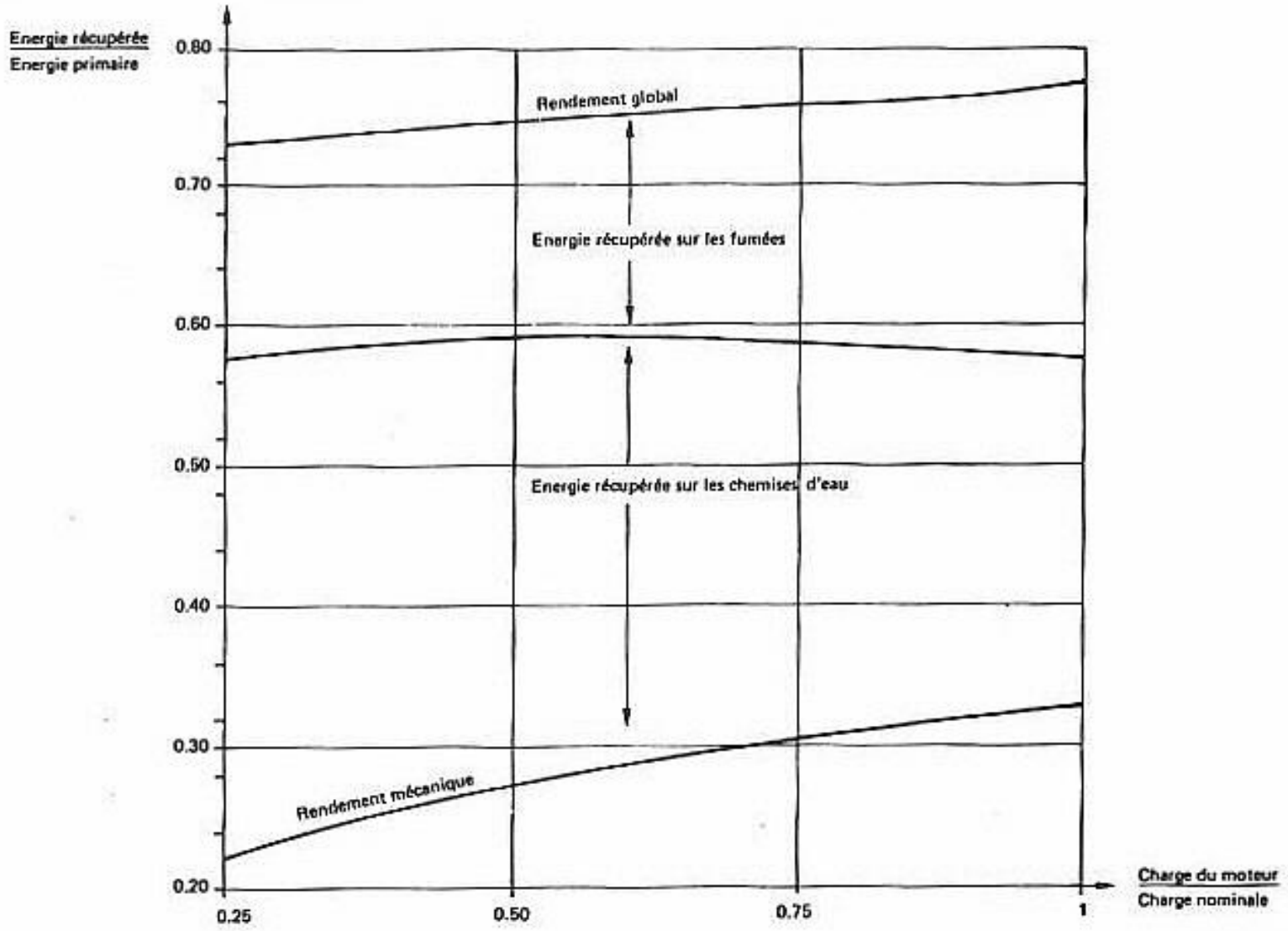
The weight of the whole operating system is 8000 kg.

The average lubricating oil consumption is 0.12 kg/h, that is 0.3 g/kWh.

Typical trends of the different parameters at partial loads are represented.
The electric efficiency reduces at the reducing of the load whereas the fraction of heat recovery of the cooling water increases, and the fraction of the exhaust gases reduces.



The various heat recoveries at partial loads are here represented

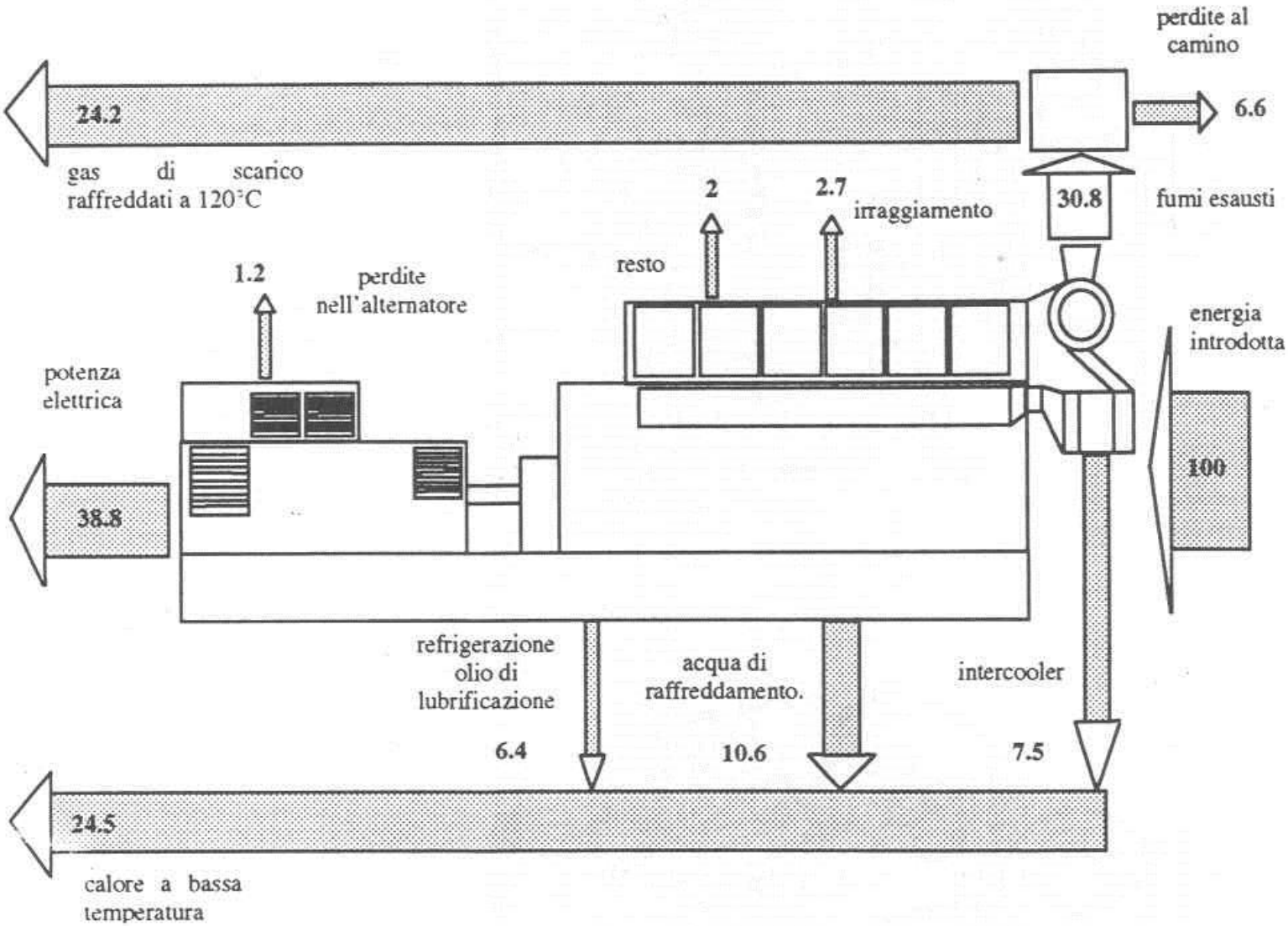


Technological steam can be produced in the medium and high capacity cogenerators in addition to hot water at the usual temperatures of 85-90°C.

PRODUZIONE ACQUA CALDA 70/85°C E VAPORE

Modello	potenza elettrica	potenza introdotta	portata		rendimento		
			vapore	acq.calda	elettrico	termico	totale
			kg/h	mch	%	%	%
Vapore 6 bar							
JMS 212	382	1054	343	15.0	36.2	46.7	83.0
JMS 312	495	1273	420	16.4	38.9	44.7	83.5
JMS 316	660	1681	554	21.7	39.2	44.7	83.9
JMS 320	826	2101	684	27.7	39.3	44.9	84.2
JMS 612	1067	2803	811	39.6	38.1	44.1	82.2
JMS 616	1420	3666	1021	51.8	38.7	43.4	82.1
Vapore 10 bar							
JMS 212	382	1054	321	15.0	36.2	45.5	81.7
JMS 312	495	1273	394	16.4	38.9	43.4	82.3
JMS 316	660	1681	519	21.7	39.2	43.4	82.6
JMS 320	826	2101	642	27.7	39.3	43.6	82.9
JMS 612	1067	2803	750	39.6	38.1	42.8	80.9
JMS 616	1420	3666	942	51.8	38.7	42.0	80.7
Vapore 16 bar							
JMS 212	382	1054	299	15.0	36.2	44.1	80.3
JMS 312	495	1273	368	16.4	38.9	42.1	81.0
JMS 316	660	1681	484	21.7	39.2	42.1	81.3
JMS 320	826	2101	599	27.7	39.3	42.3	81.6
JMS 612	1067	2803	689	39.6	38.1	41.3	79.4
JMS 616	1420	3666	862	51.8	38.7	40.6	79.3

A possible flowchart of such engines is here represented



When higher capacity engines are considered, with an electric power higher than 1 MW, an increase in the efficiency and a reduction of rotation speed can be observed.
A limit regarding all the reciprocating engines is the mean linear speed of the piston.
Some years ago, this speed could not exceed 8/10 m/s.
Over time, better lubricating oils and materials have allowed a constant increase of that limit.

Nowadays, a motorcycle engine easily arrives at a linear piston speed double than the previous limit.

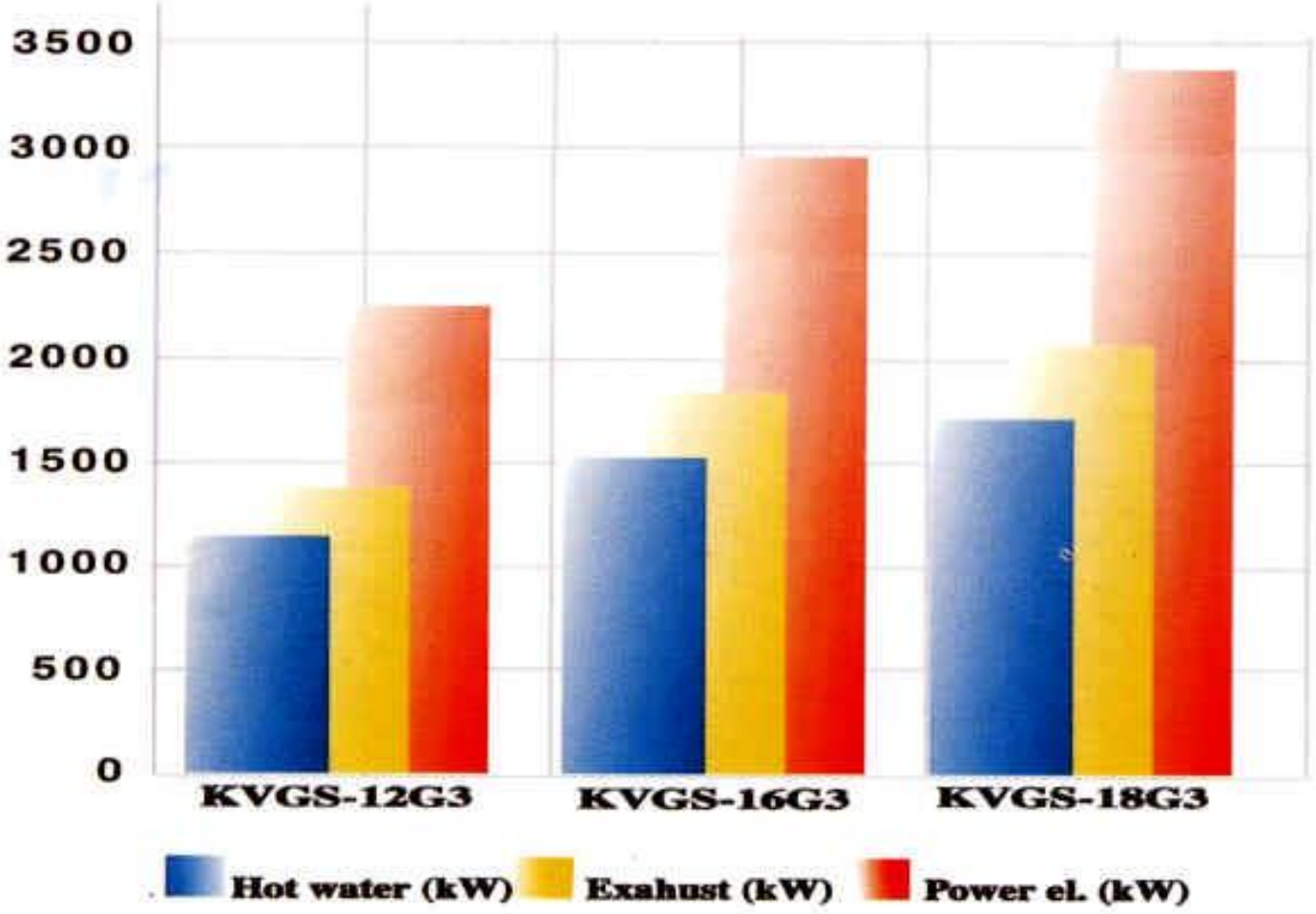
Example of my motorcycle
Suzuki RF 600 bore 65 mm
stroke 45.2 mm
Max torque at 9500 rpm
Max power at11500 rpm

$$v(m/s) = \frac{0.0452 \times 10,0000 \times 2}{60} = 17.3$$

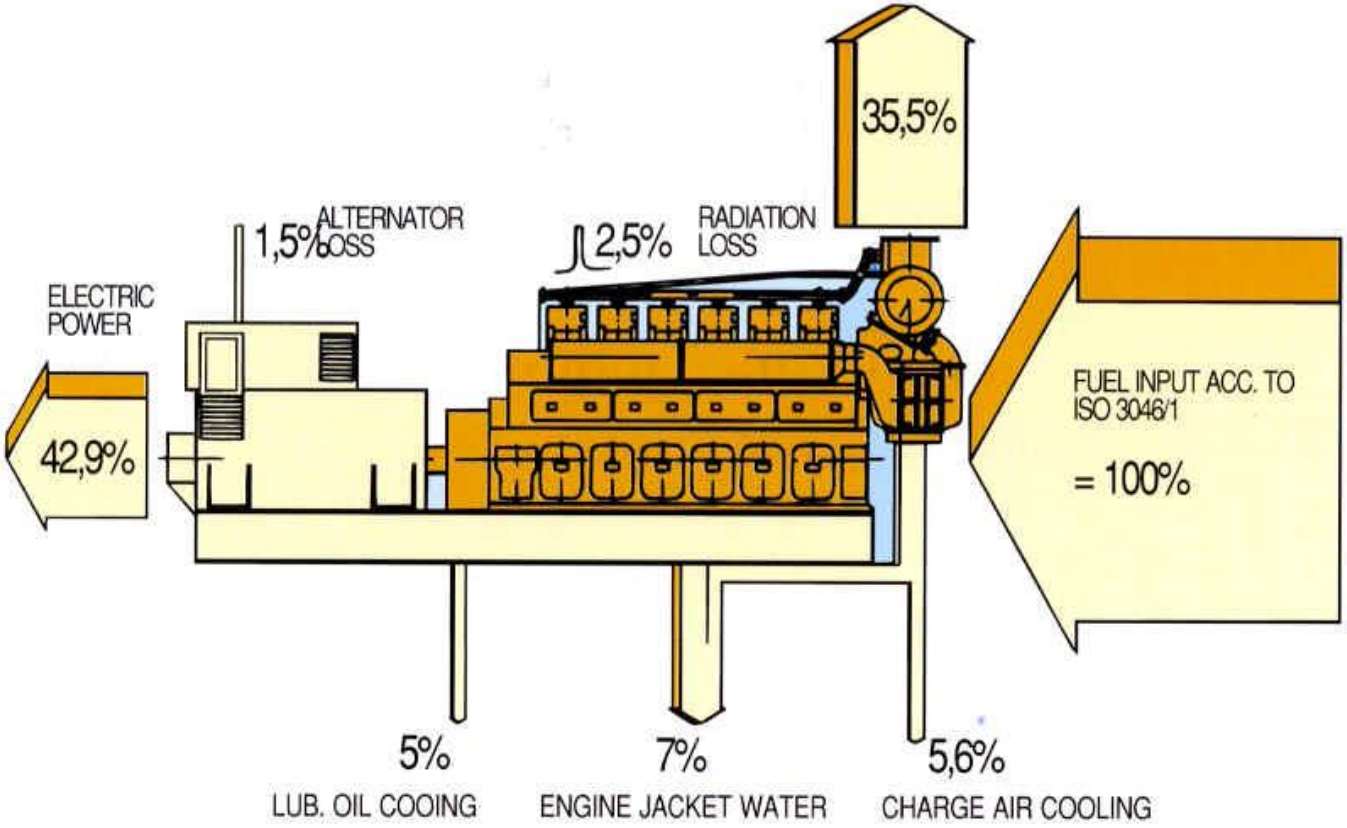
However the increase in the high capacity cogeneration engines is limited to favor a longer life.

The efficiency is strongly increased, especially for the motors supplied with burning oil due to a very high injection pressure of the fuel.

In the high capacity engines, the thermal fraction reduces in favor of the electric: a typical electric efficiency can be higher than 40%, whereas the thermal efficiency can lower : less than 50%.

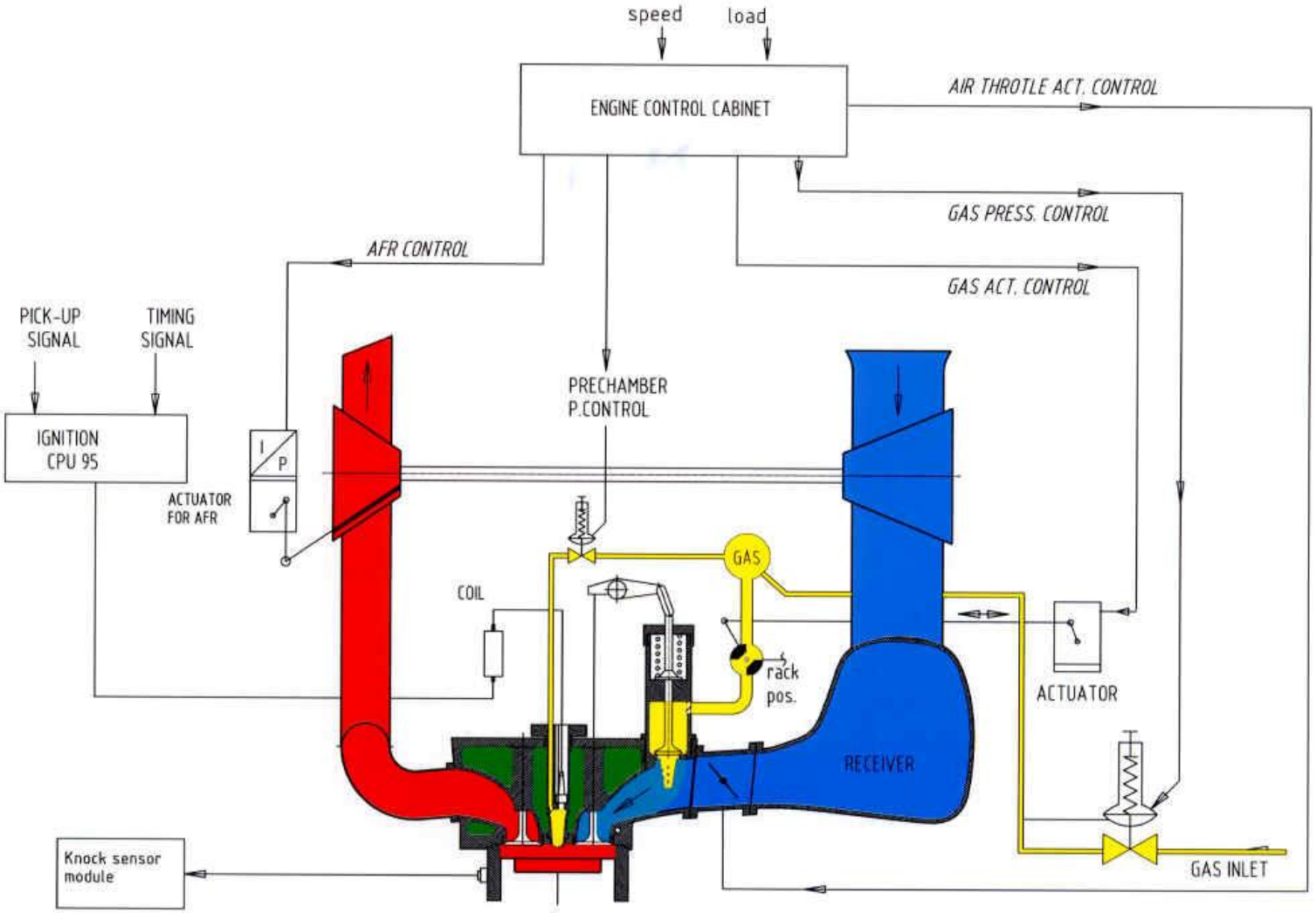


A typical flow chart for a 2-3 MW capacity engine is here represented

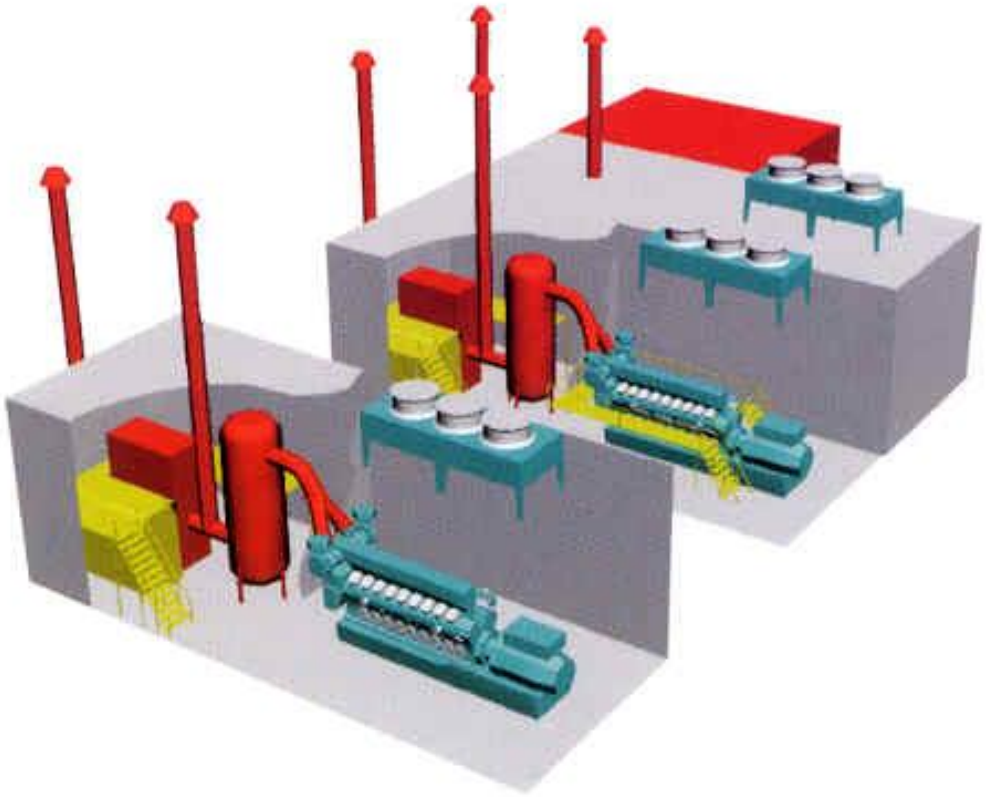


The diagram shows a typical Sankey diagram for gas engine type KV-G3, running on natural gas.

The fuel supply system and the control system are generally very sophisticated



These engines are often employed in cogeneration central plants with an installed power obtained by multiple high capacity engines exceeding sometimes 20 MW.



The weight of this engine can exceed 20 t.

The characteristic indexes for this engine are:

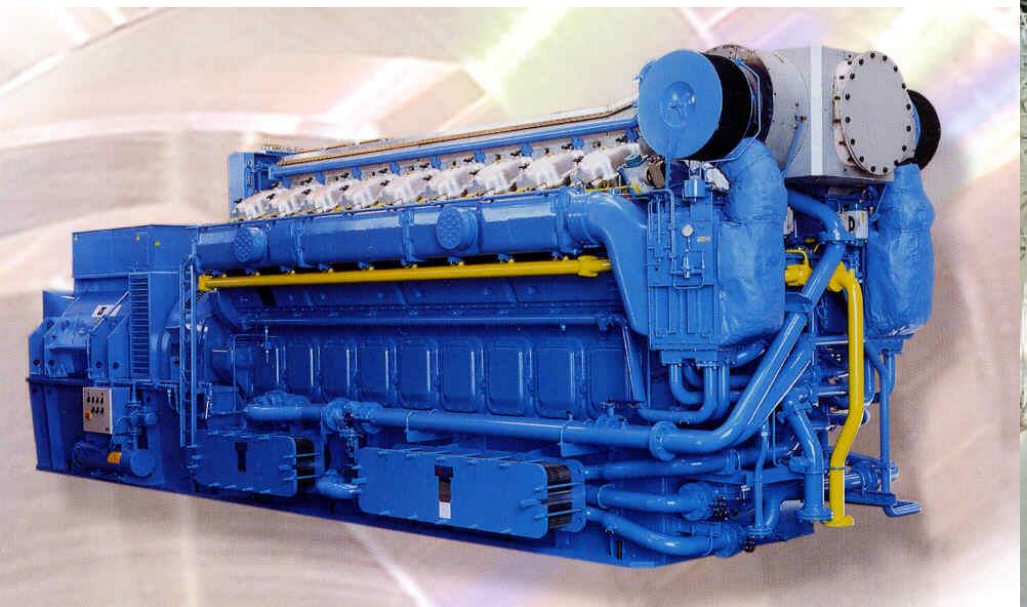
Ratio between fuel energy and electric energy:

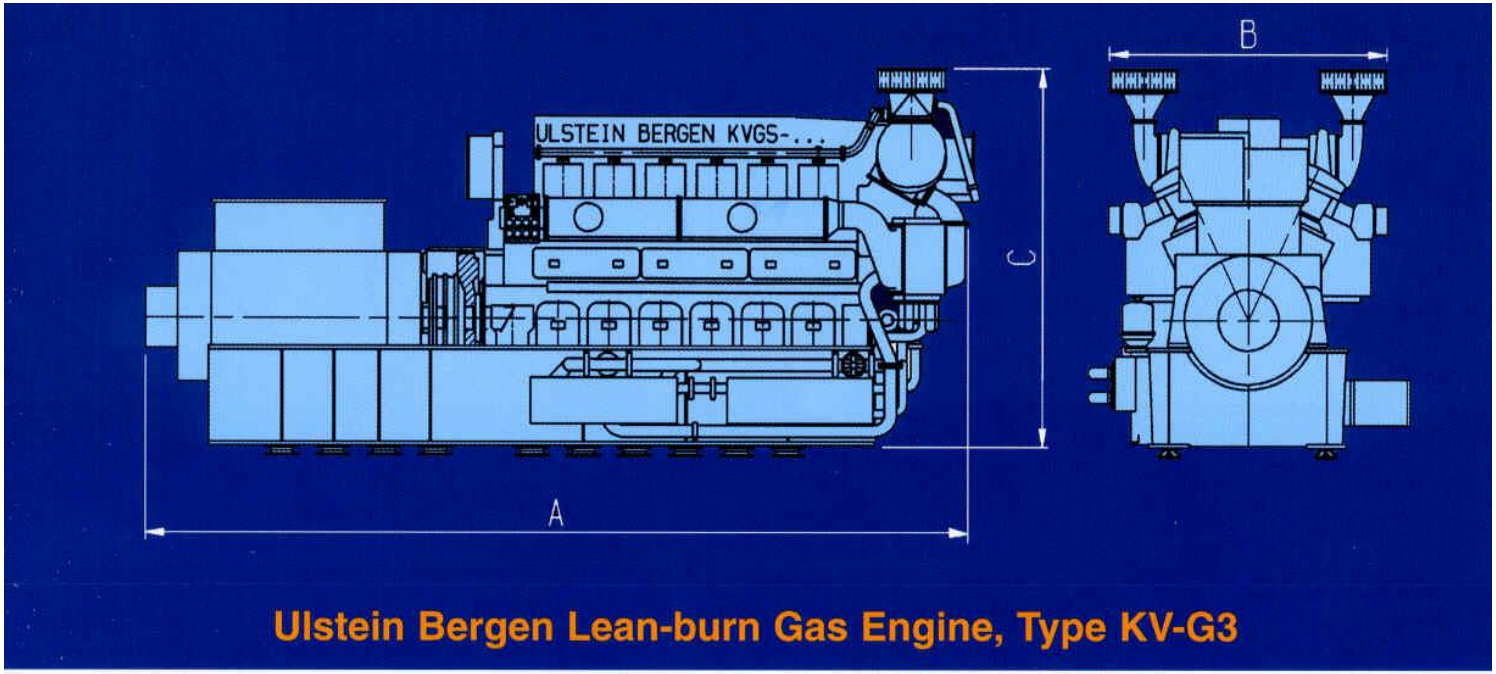
2.3-2.6 kWh/kWh

Thermal to Electric ratio:

$$\frac{T}{E} = 1.2$$

Specific fuel consumption: 0.244 m^3/kWh





Ulstein Bergen Lean-burn Gas Engine, Type KV-G3

Performance data

Engine Type	A	B	C	Weight (dry) [kg]
KVGS-12G3	7825	3050	3250	37050
KVGS-16G3	8870	3150	3250	48300
KVGS-18G3	9350	3150	3250	55800

All dimensions are in mm.

Exhaust gas raw emissions (mg/nm3)

NOx	CO	NMHC
500	850	270

Also able to meet requirement of NOx = 250.

Engine Type		KVGS-12G3	KVGS-16G3	KVGS-18G3
Number of cylinders		12	16	18
Bore/Stroke	mm	250/300	250/300	250/300
Engine speed	min^{-1}	1000	1000	1000
Electrical output, cosø = 0.8	kW	2220	2960	3330
Fuel gas consumption	kW	5295	7010	7875

The specific fuel consumption can reach even 0.217 m^3/kWh
The displacement is about 1800 liters with a weight that can exceed 300 tons.

Power plant – Main data

Power plant gross electrical output

Number of units	9R32	12V32	16V32	18V32
	MW_e	MW_e	MW_e	MW_e
1	3.2	4.3	5.8	6.5
2	6.5	8.7	11.6	13.0
3	9.7	13.0	17.3	19.5
4	12.9	17.4	23.1	26.0
5	16.2	21.7	28.9	32.6
6		26.0	34.7	39.1
7			40.5	45.6
8			46.2	52.1
9			52.0	58.6
10			57.8	65.1

At ISO reference conditions

Power plant available gross hot water output
Heavy fuel oil, Light fuel oil

Number of units	9R32	12V32	16V32	18V32
	MW_e	MW_e	MW_e	MW_e
1	2.9	3.7	5.0	5.7
2	5.9	7.4	10.0	11.5
3	8.8	11.0	15.1	17.2
4	11.7	14.7	20.1	22.9
5	14.7	18.4	25.1	28.7
6		22.1	30.1	34.4
7			35.1	40.1
8			40.2	45.8
9			45.2	51.6
10			50.2	57.3

Hot water generation up to 120°C

Power plant available gross hot water output
Natural gas

Number of units	9R32	12V32	16V32	18V32
	MW_e	MW_e	MW_e	MW_e
1	3.5	4.4	6.0	6.8
2	6.9	8.8	12.0	13.6
3	10.4	13.2	18.0	20.4
4	13.8	17.6	24.0	27.2
5	17.3	22.0	30.0	34.0
6		26.4	36.0	40.8
7			42.0	47.6
8			48.0	54.5
9			54.0	61.3
10			60.0	68.1

Power plant gross steam output

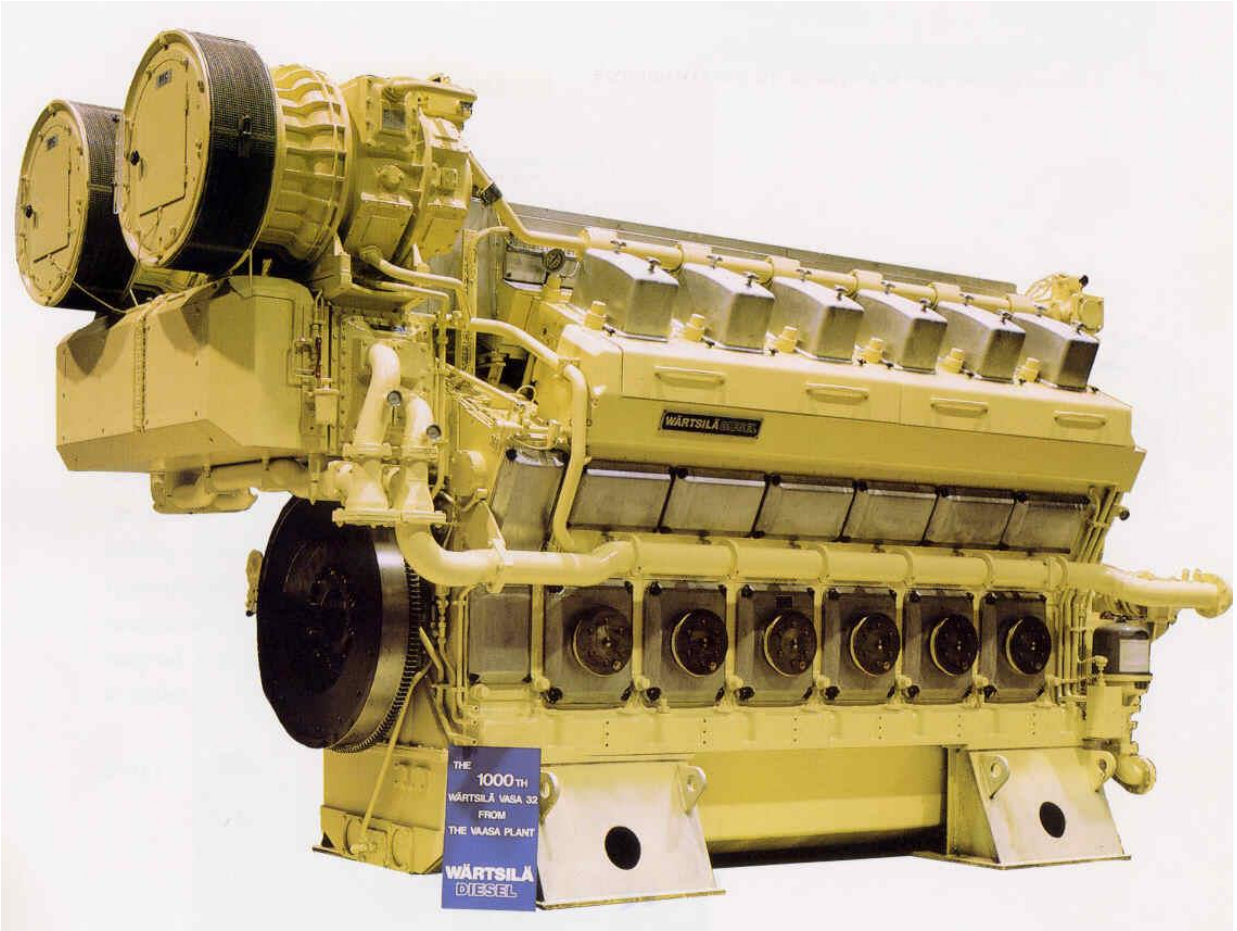
Number of units	Engine type 9R32		12V32		16V32		18V32	
	t/h	MW_{th}	t/h	MW_{th}	t/h	MW_{th}	t/h	MW_{th}
1	1.6	1.1	2.0	1.3	2.8	1.9	3.2	2.2
2	3.2	2.2	4.0	2.6	5.6	3.7	6.4	4.3
3	4.8	3.2	5.9	3.9	8.4	5.6	9.7	6.5
4	6.4	4.3	7.9	5.2	11.2	7.4	12.9	8.7
5	8.1	5.4	9.9	6.5	14.0	9.3	16.1	10.8
6			11.9	7.8	16.8	11.2	19.3	13.0
7					19.7	13.0	22.6	15.1
8					22.5	14.9	25.8	17.3
9					25.3	16.8	29.0	19.5
10					28.1	18.6	32.2	21.6

Saturated steam at 8 bar(a) and outputs at ISO reference conditions.

Prime mover – Main data

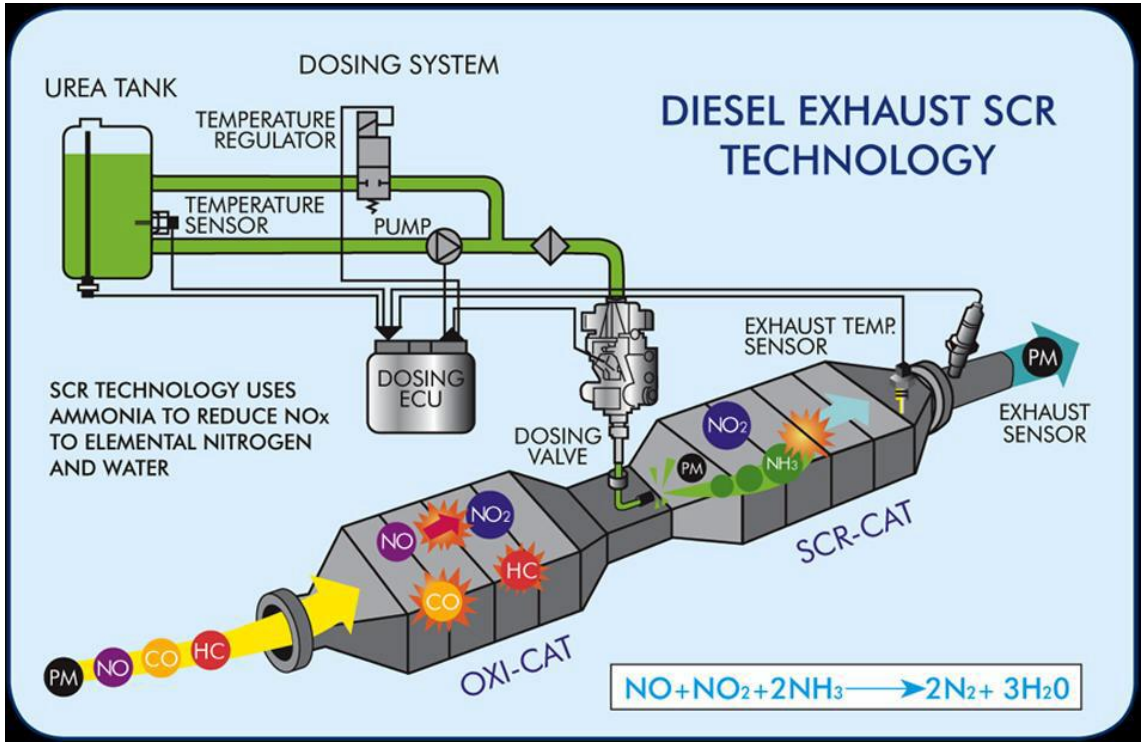
Cylinder bore	320 mm
Piston stroke	350 mm
Speed	720 - 750 rpm
Mean effective pressure	21.9, 21.3 bar
Piston speed	8.4-8.75 m/s
FUEL SPECIFICATION	
Fuel oil	730 cSt/50°C 7200 sRI/100°F
ISO 8217, ISO-F-RMH 55	
Natural gas	

The engines can be dual fueled, either only gas fueled with injection pressure till 350 bar. The system operates according to a diesel cycle.
In the dual fueled motors, at least 5% diesel oil is supplied to ignite the gas in the combustion chamber.

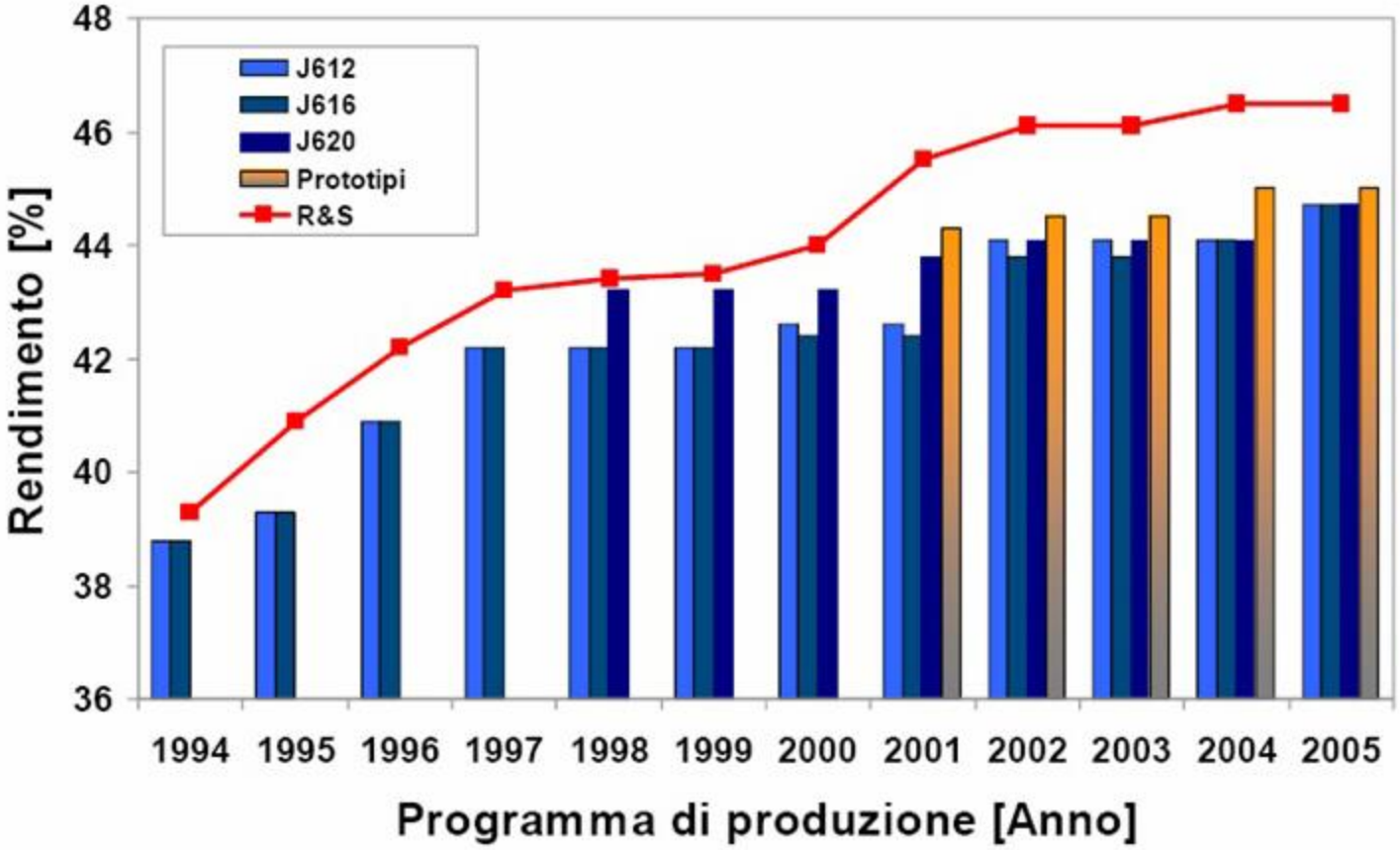


The unitary cost of these cogeneration groups can be evalutated between 600 and 1000 € for electric kW with strong scale effects.

A relevant problem is the possible atmospheric pollution as the NO_x content can be higher than 500 mg/m^3.
The high capacity engines are equipped with a device called *Selective Catalytic Reduction*, where ammonia is injected in the exhaust gases that are discharged through a catalyst producing H_2O and N_2. This way, the NO_x content can be lowered till 40-50 ppm.

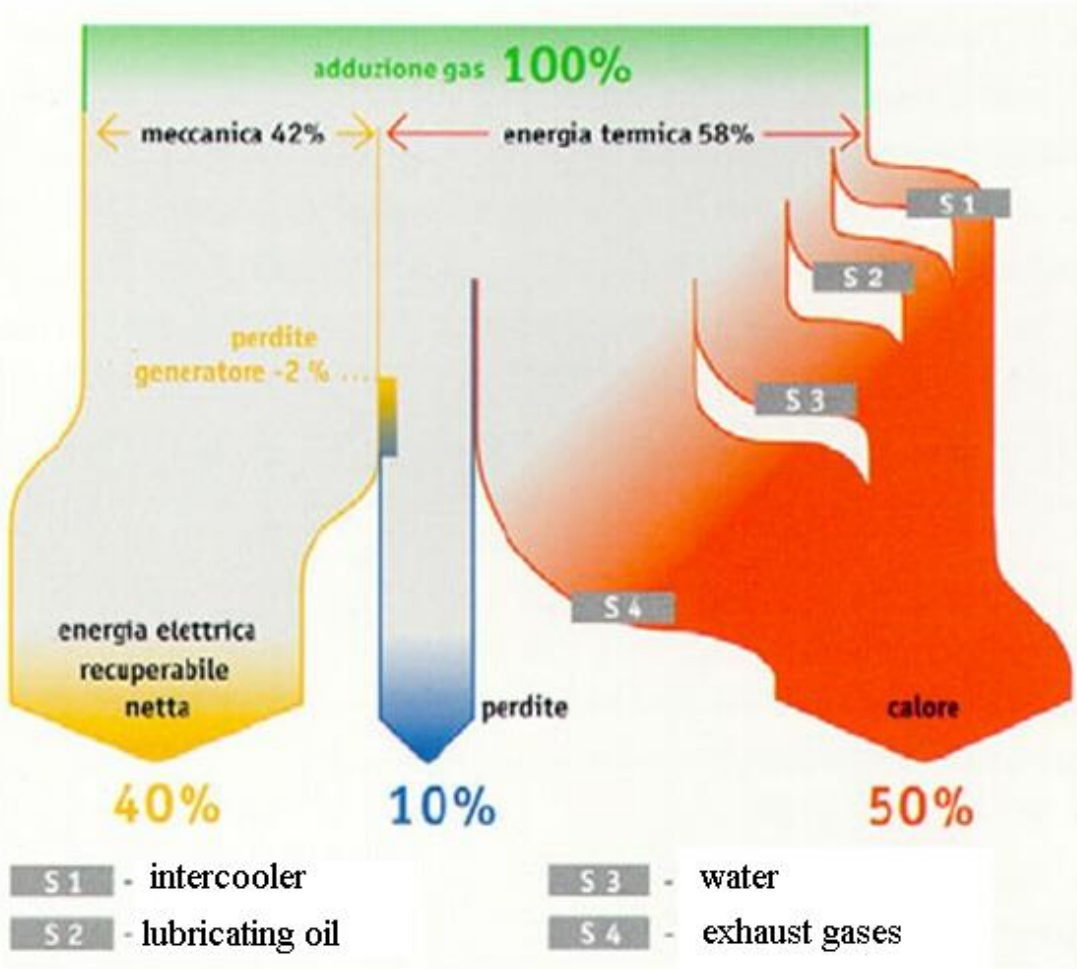


Development of mechanical efficiency



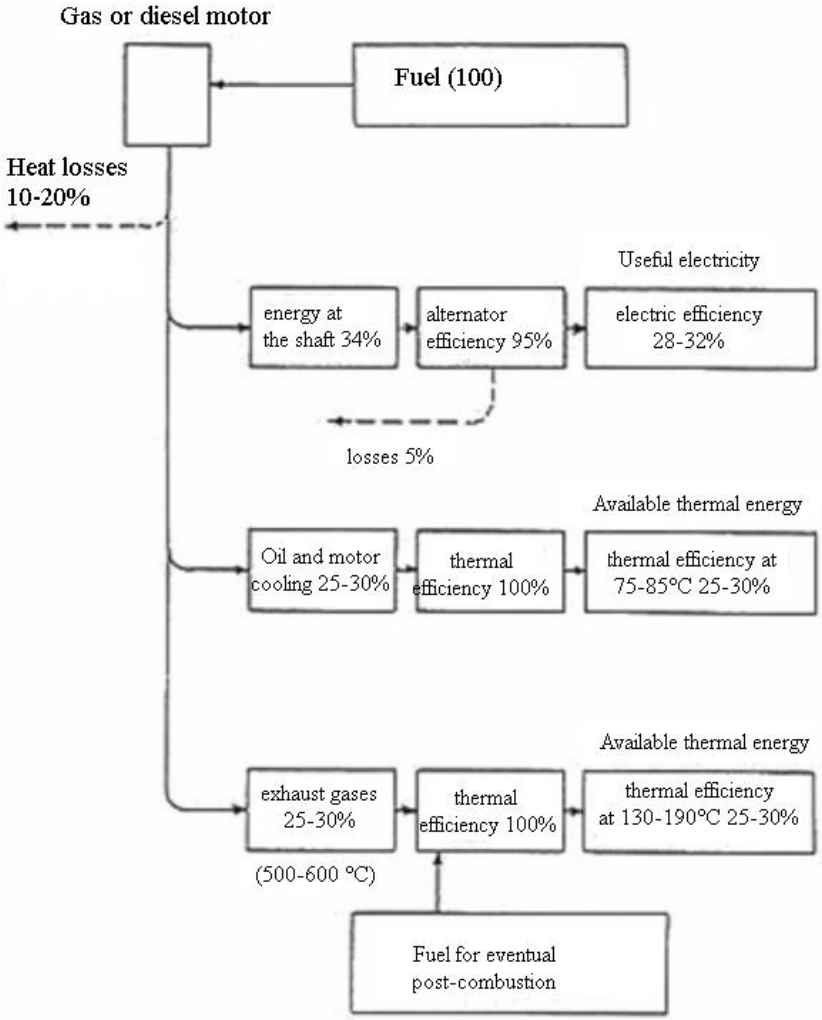
GE Jenbacher

FLOW CHART IN COGENERATION



L'intercooler è uno scambiatore di calore che raffredda l'aria compressa di alimentazione dei motori dei veicoli dotati di turbocompressore.

Summary scheme regarding cogeneration with i.c. reciprocating engines. The values are for low-medium capacity engines.



The possible post-combustion, while it allows a higher flexibility in the thermal/electric ratio, requires an addition in the combustion air, as the excess air is low even if the lean combustion and the value is not far from the stoichiometric value 14.5:1.

The situation is different for the Diesel cycle that operates with an air excess variable with the engine speed with ratios that can be 20-25:1.

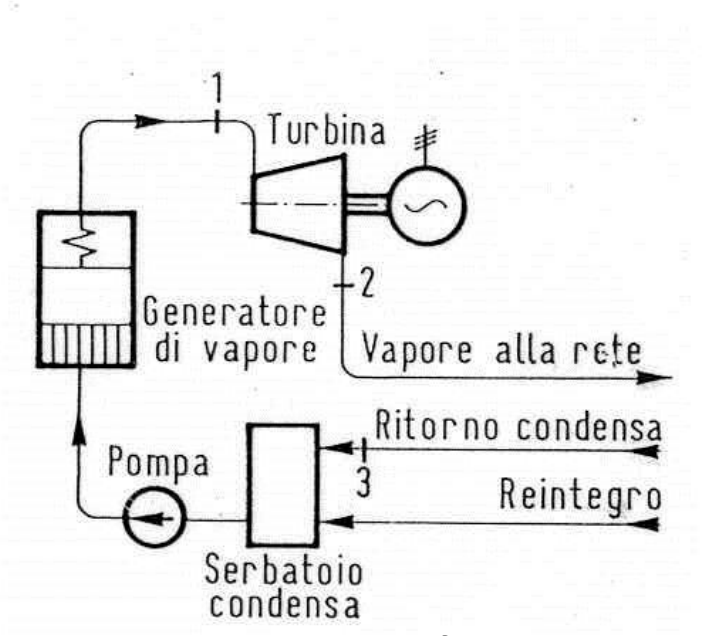
COGENERATION
WITH VAPOUR POWER CYCLES

COGENERATION WITH VAPOUR POWER PLANTS

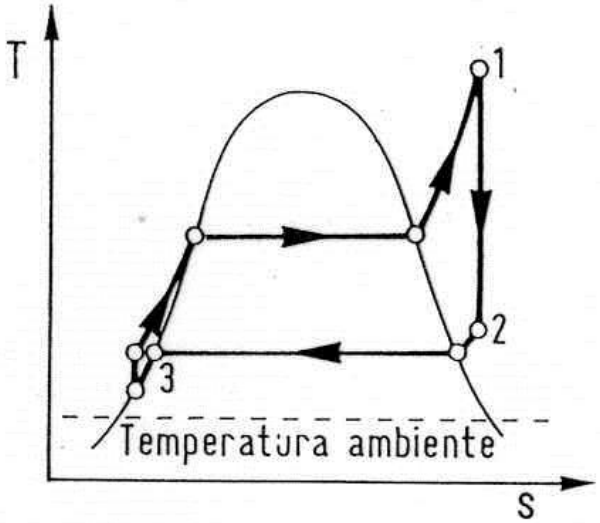
They were the most widespread plants until a few decades ago. Nowadays it is difficult to find such a plant with capacities lower than 5 MW_e.

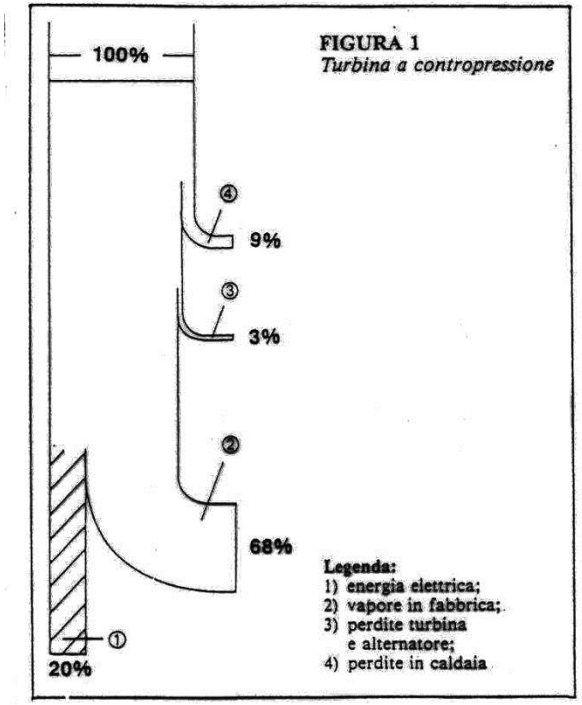
They are rather expensive and bulky plants with the favourable possibility of using almost every kind of fuels, for example municipal solid waste.

The simplest scheme is the back-pressure plant:

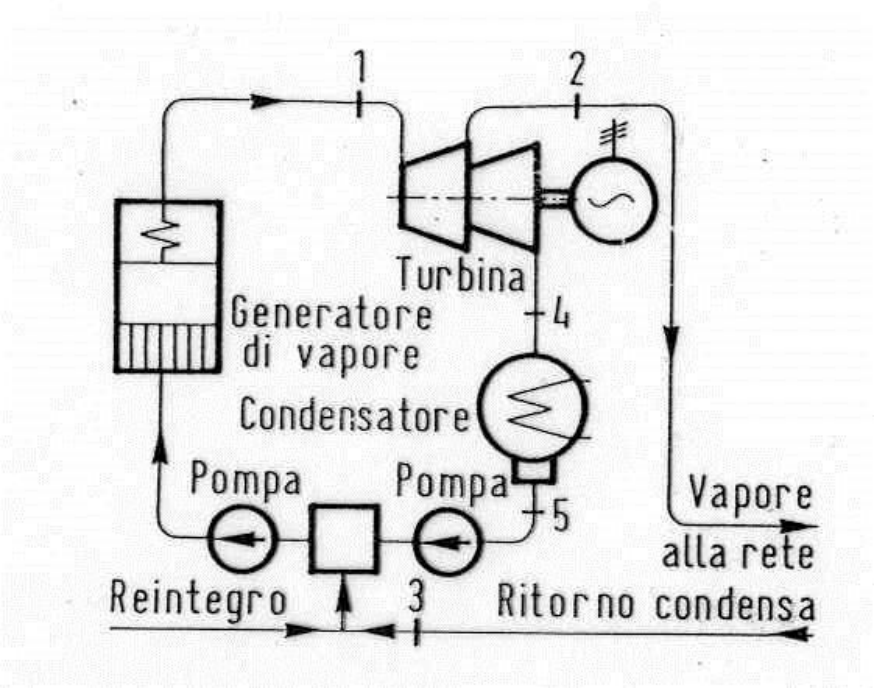


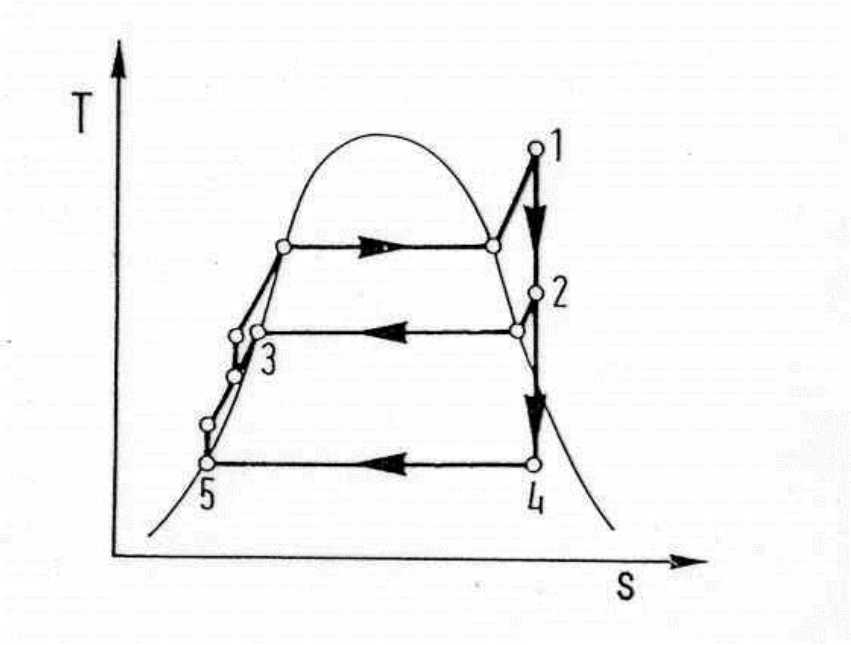
The back-pressure plant has a rather low efficiency but the advantage of eliminating the last turbine stages that are the least efficient and challenging in manufacturing.
The main defect is in the fixed ratio of thermal and electric energy so that it cannot well fit to the demand.
The low efficiency is due to the rather high back pressure (8-10 bar).





Higher efficiency and flexibility can be obtained with the pass out or extraction turbine plant:





To reach high electric efficiency it is necessary:

• to reduce the pass-out fractions, that strongly influence the efficiency;

• to increase the plant capacity to some tens of MW.

In the vapour turbine, the losses due to vapour leaks and friction are proportionally higher the lower the capacity. Besides, these turbines are strongly penalized at partial load.
The optimal performance is in fact obtained for a given angle of attack of the fluid on the blading. If the efficiency at full load is 40%, it might lower at 36% for ¾ partial load and at 30% for ½.
The steam generators are challenging for temperatures and pressure with particular maintenance with low quality fuels.
The condensers are heavy and bulky and require large water amounts for cooling.

Typical efficiencies can reach 35% for industrial cogeneration at high temperatures and pressure, lowering to 25% for lower pressure frequent in industrial applications of 40-60 bar.

A possible example of such a plant is relative to a real plant in a paper mill.
A paper mill presents a high electricity demand to operate the equipment and a lot of thermal energy to prepare the pulp and to heat the drying cylinders.

The cogeneration plant gives 11 MW_e and 49.5 MW_t. The thermal energy is supplied as 10 t/h of technological steam at 13 ate (from pass out) and 60 t/h from back pressure at 3.5 ate. The former is needed to produce the pulp and the latter to keep the temperature of the cylinders where the pulp sheet is dried in many subsequent steps.

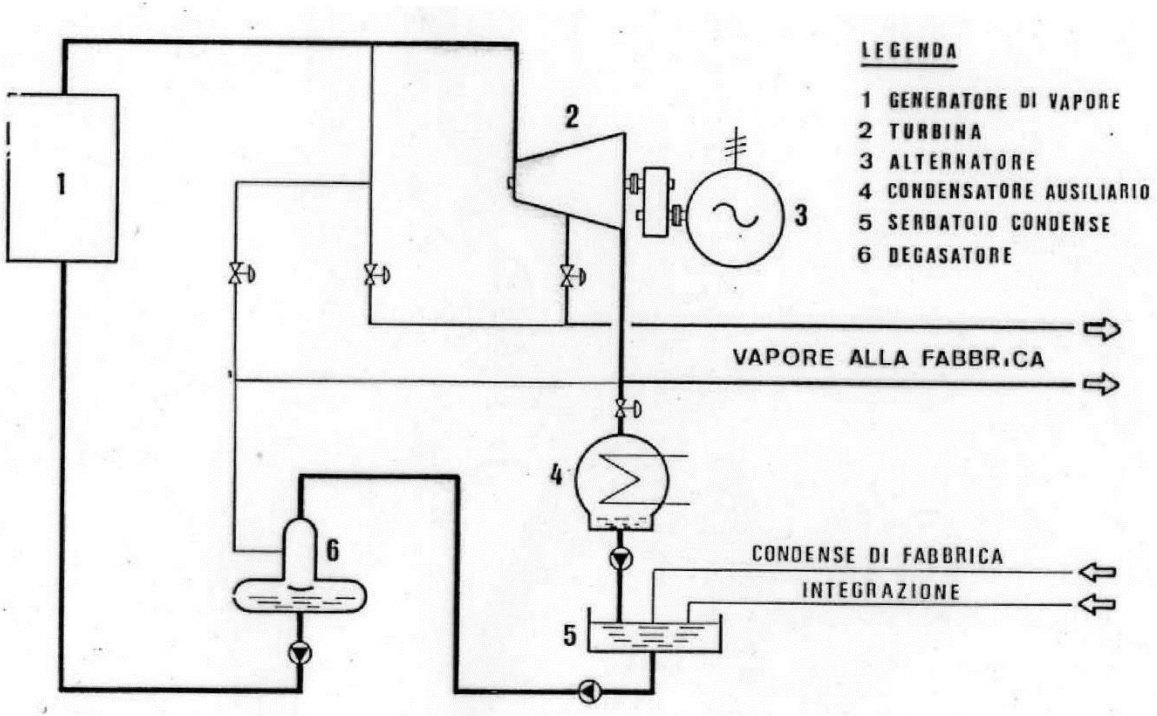


Fig.1 - IMPIANTO DI COGENERAZIONE: SCHEMA TERMICO SEMPLIFICATO

The steam generator consists of a vertical water pressurized tubes operating at natural circulation with two cylindrical bodies and overheating. The maximum vapour production is 98 t/h at a pressure of 84 ate and at a temperature of 475°C.

The maximum efficiency vapour production is 60 t/h.

The feed water temperature is 147°C, whereas the gas exhaust temperature out of the *Ljungstroem* wheel is 161°C. The efficiency of the generator at nominal capacity is 92.5%.

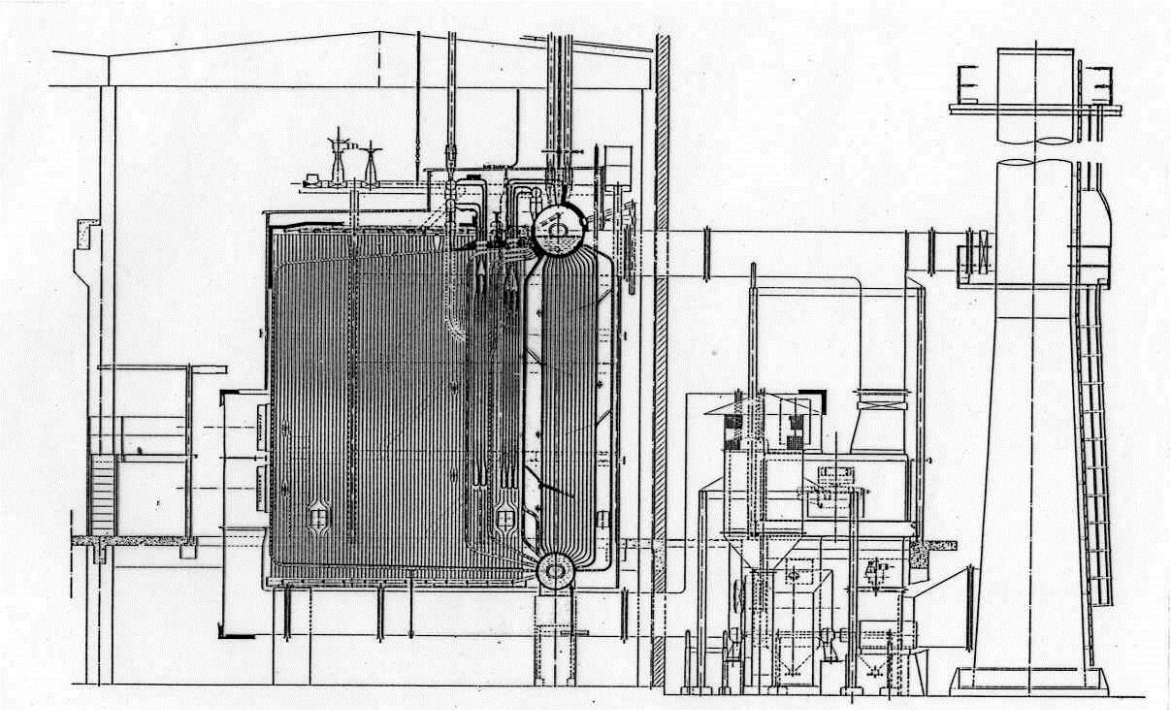
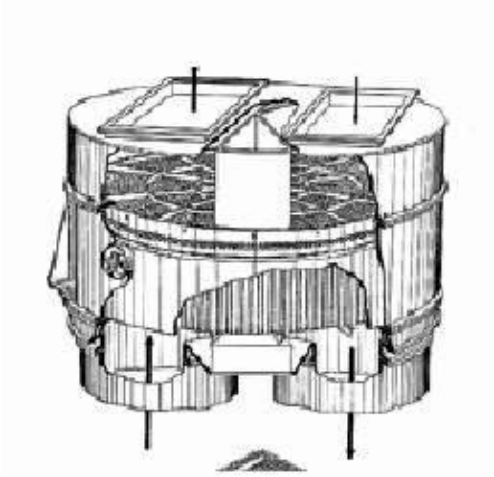


Fig. 2 - IMPIANTO DI COGENERAZIONE: SEZIONE GENERATORE DI VAPORE

The fuels can be natural gas or burning oil separately or in mixed combustion.



The turbine is of action and reaction type, backpressure, single cylinder with a rotational speed of 6,000 rpm (alternator at 1500 rpm), steam pressure at the inlet 80 ate at 470 °C.
Steam at the inlet 70 t/h, 10 t/h passed out and 60 t/h discharged.

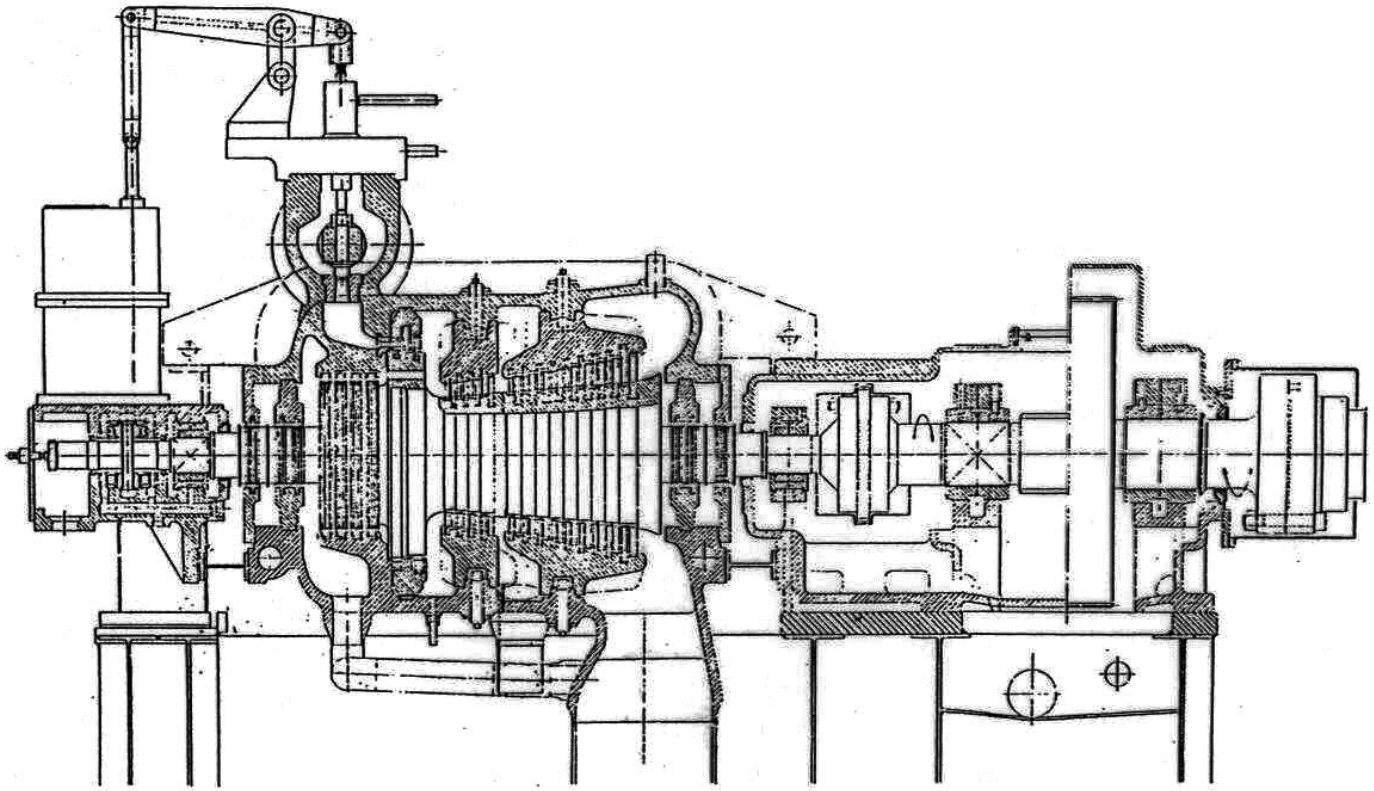
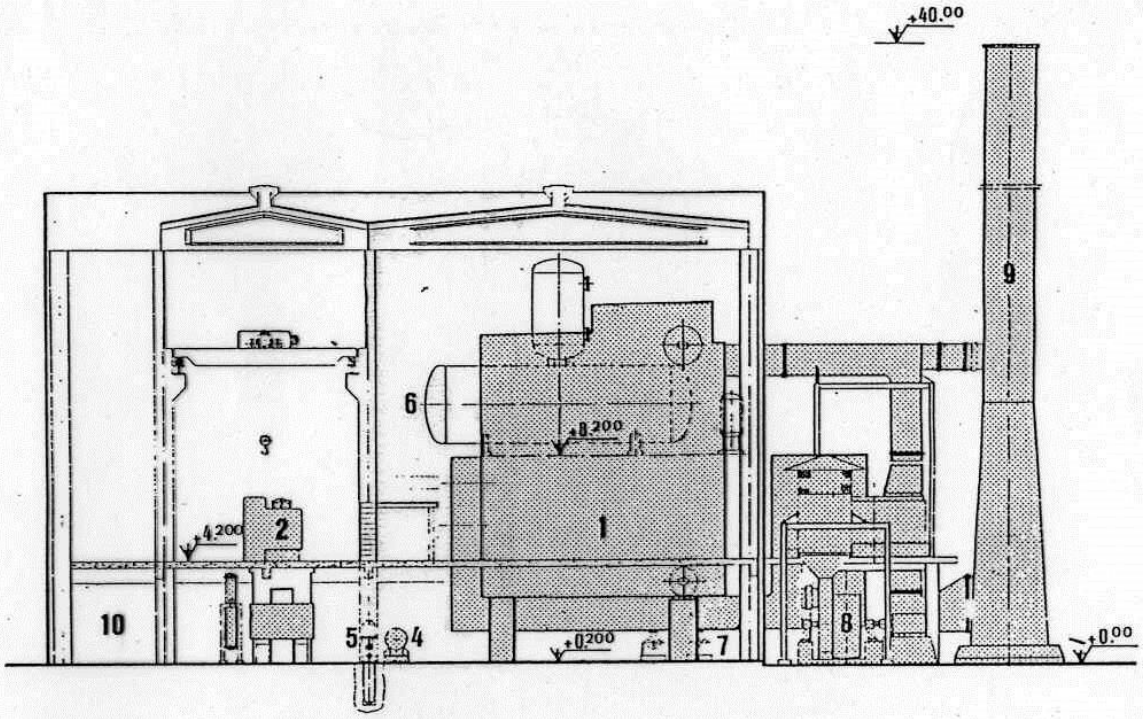


Fig. 3 – IMPIANTO DI COGENERAZIONE : SEZIONE TURBINA

The performances in nominal operation are the following:
10.2 MW_e with 10 t/h of technological steam at 13 ate and 195°C and 60 t/h at 3.5 ate (147°C).
The hourly fuel consumption is 5,750 kg/h Bunker C (a burning oil).



LEGENDA

1 GENERATORE DI VAPORE
2 TURBINA
3 ALTERNATORE
4 CONDENSATORE AUSILIARIO
5 POMPE CONDENSATO
6 DEGASATORE
7 POMPE ALIMENTO
8 VENTILATORE
9 CAMINO
10 AREA STAZIONI DI RIDUZIONE

0 2 4 6 8 10 metri

Fig. 4b – IMPIANTO DI COGENERAZIONE: DISPOSIZIONE - SEZIONE

The electrical efficiency is 16% and thermal efficiency is given by:

$$\frac{(667-147)\times 10,000+(656-147)\times 60,000}{5750\times 9,700}=0.64$$

Thermal/electric ratio is 5, I_t=0.8. IEN = 0.53

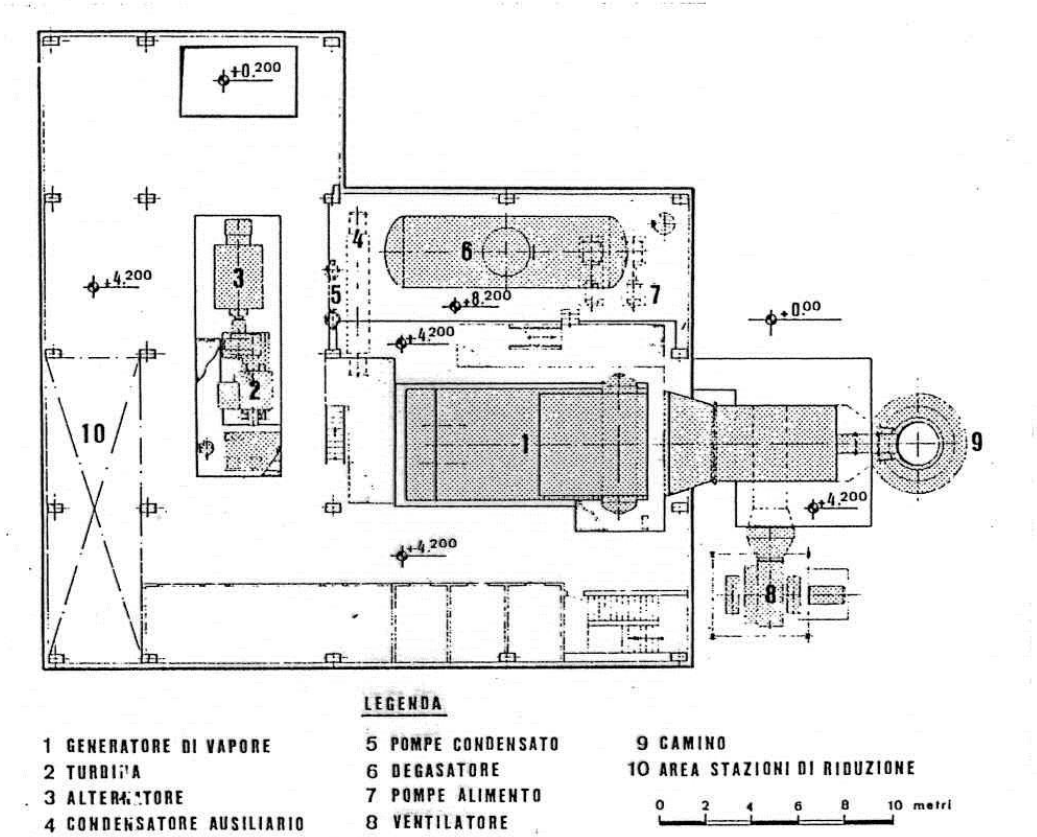
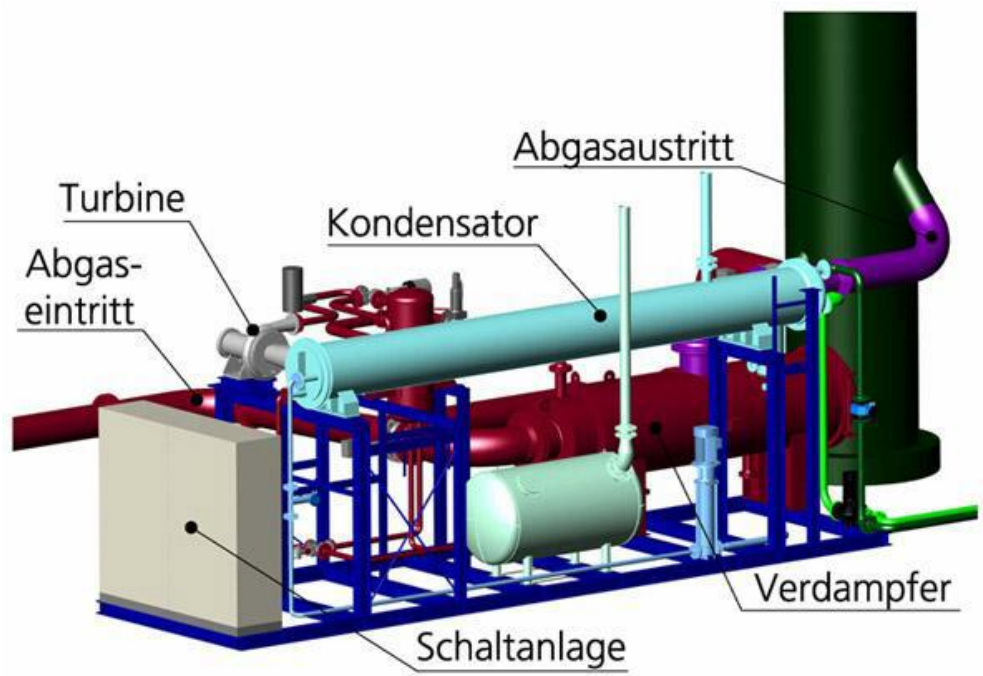


Fig. 4a – IMPIANTO DI COGENERAZIONE: DISPOSIZIONE - PIANTA

The cogeneration vapour power turbines present the following characteristics:

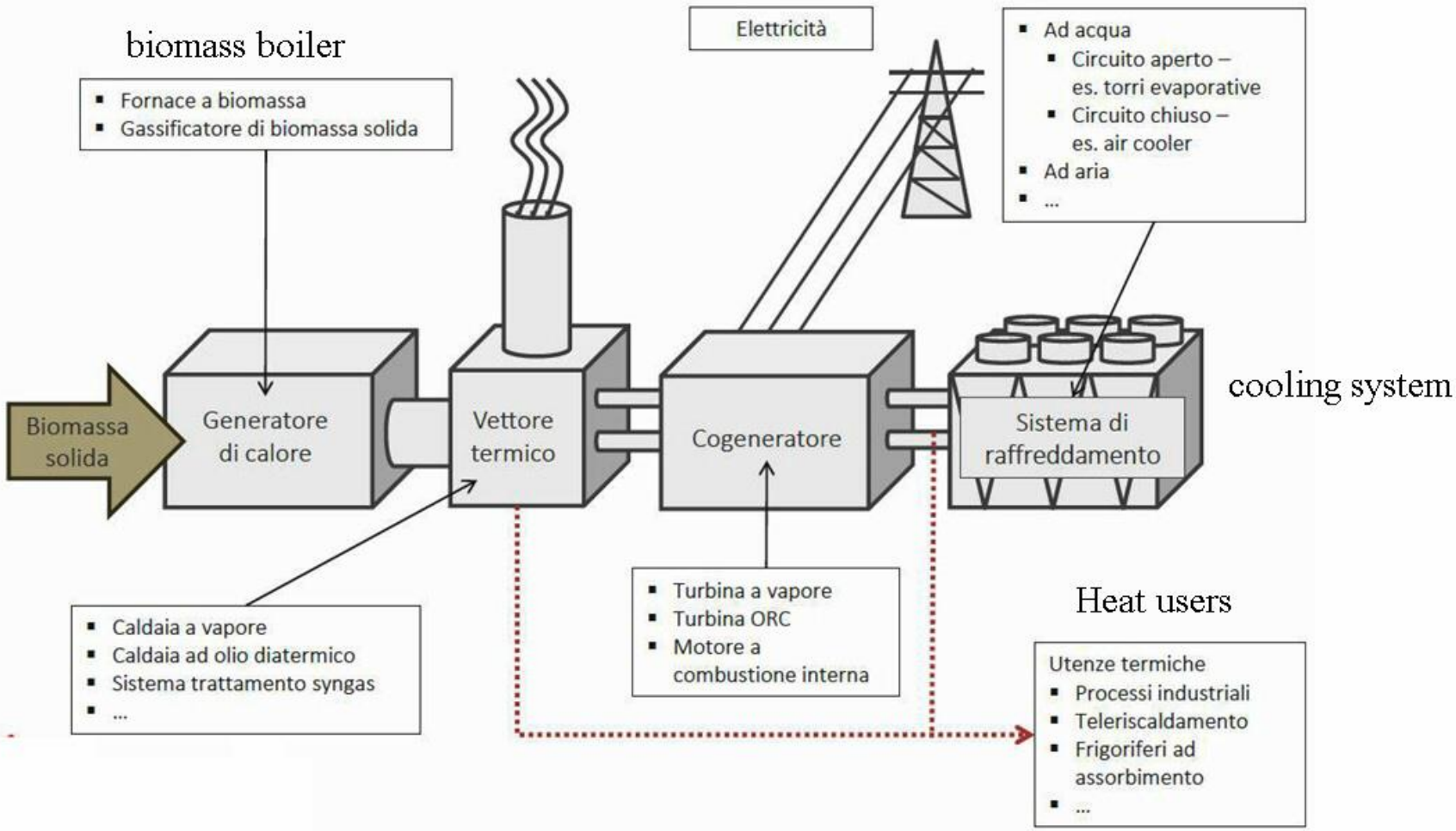
- thermal/electric ratio from 4-10;
- capacity higher than 500 kW;
- first principle efficiency 85%;
- specific investment cost 1.5-2.0 k€/kW;
- working cost 5-8 c€/kWh;
- they can use every fuels;
- low electric efficiency;
- fixed thermal/electric ratio for back pressure;
- low first principle efficiency for pass-out turbines.

COGENERATION WITH PLANTS ORC (Organic Rankine Cycle)(*)



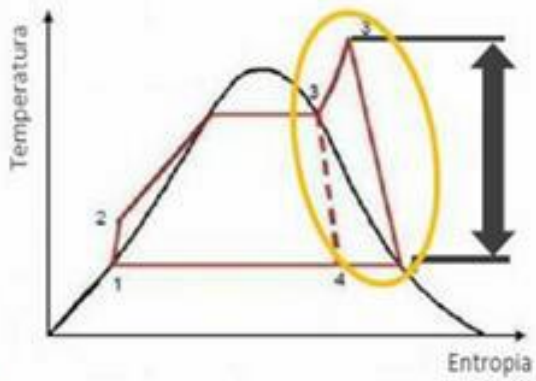
(*) Some slides come from the presentation "Prospettive future per la cogenerazione alimentata da biomassa solida" of Francesco Campana, Sales Engineer of TURBODEN SpA and from the presentation "Microturbine e ORC" of Alessandro Dorigati, CEO Progeco Srl

Cogeneration with solid biomass

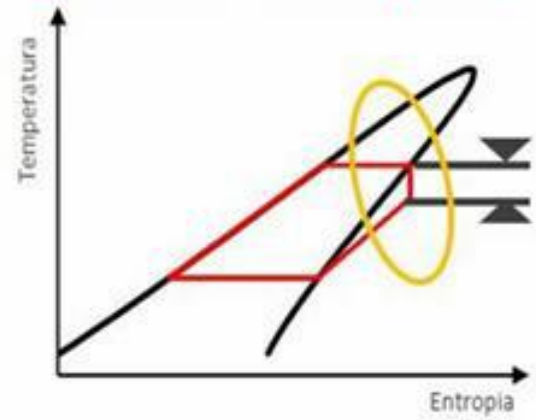


Comparison between steam Rankine cycle and ORC

Rankine steam cycle

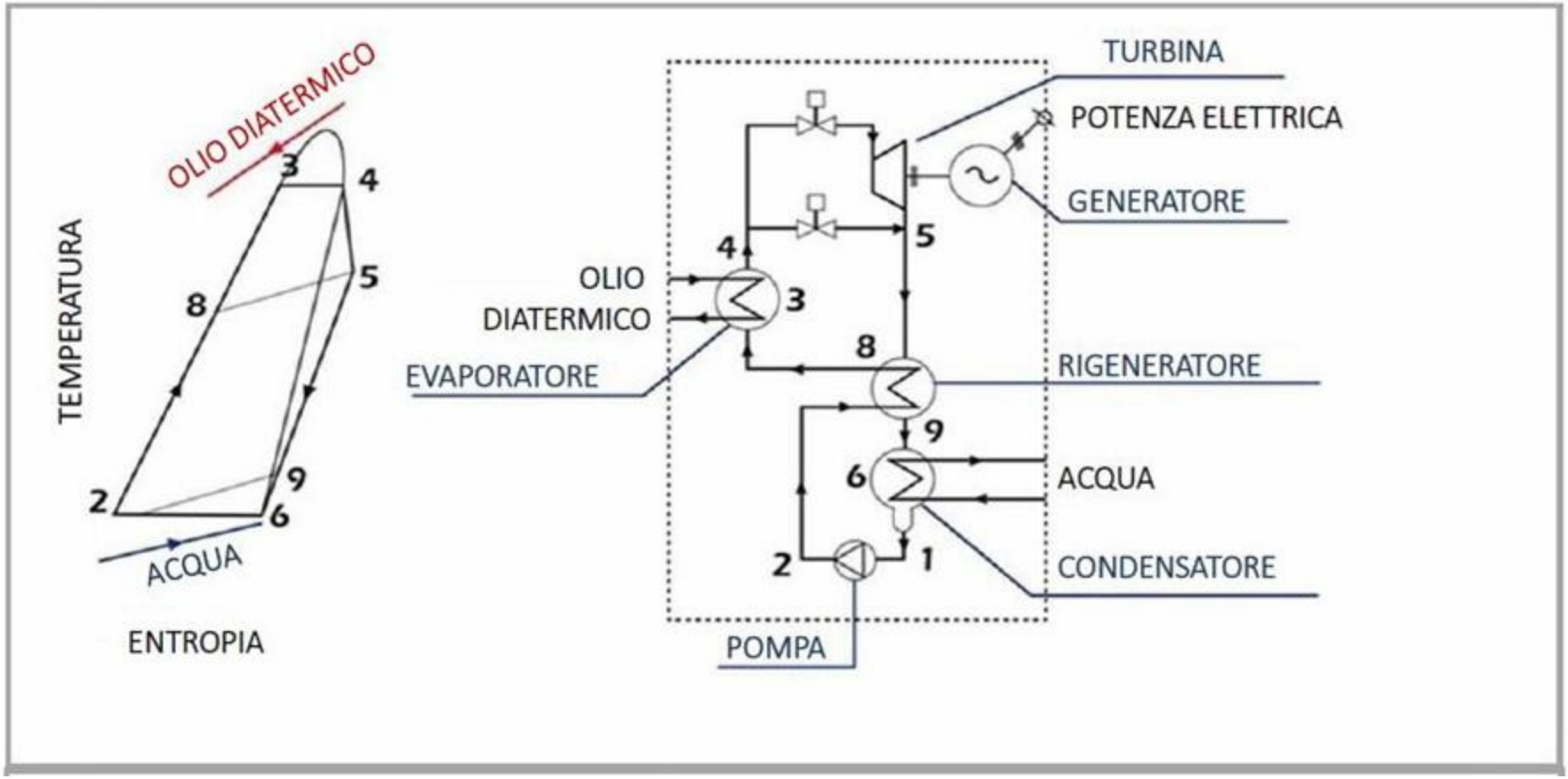


Organic Rankine Cycle (ORC)



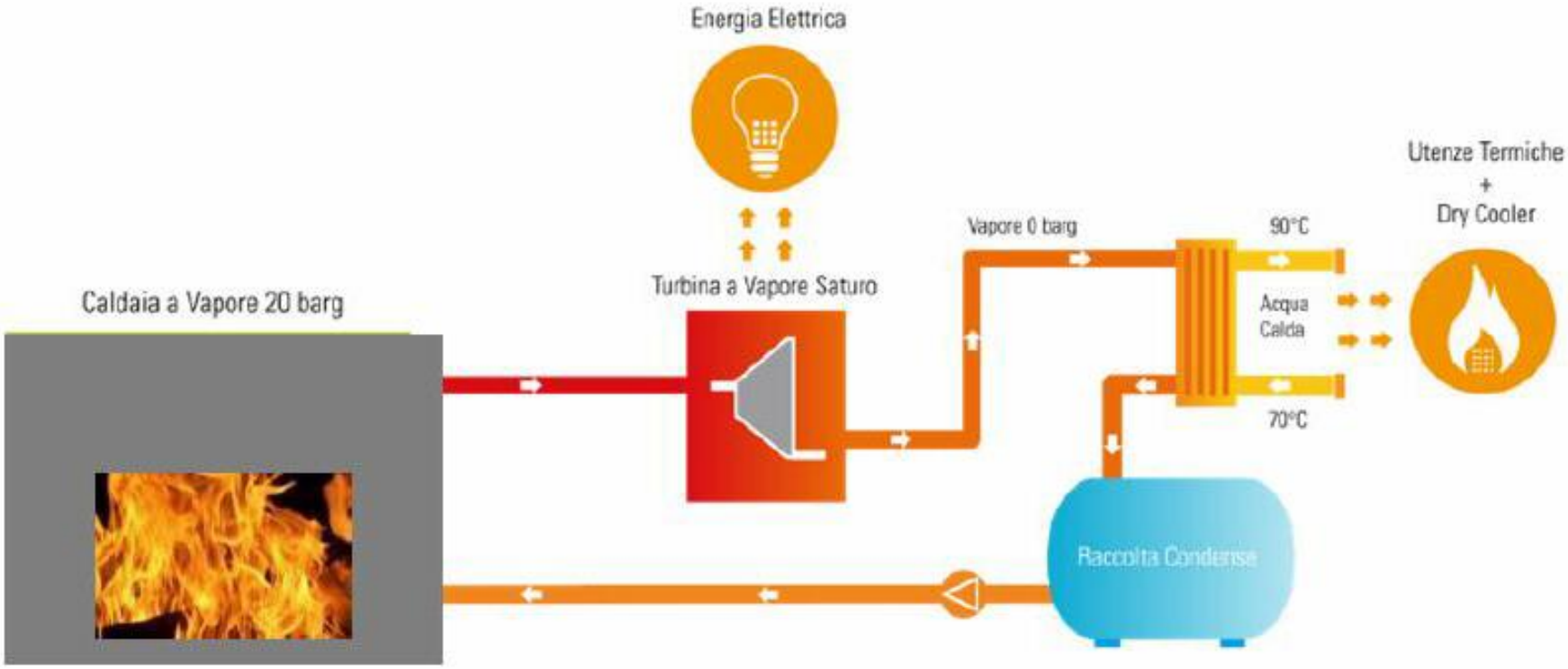
	Rankine steam cycle	Organic Rankine Cycle (ORC)
Caratteristiche termodinamiche	• high enthalpy drop; • overheating needed • risk of blade erosion	• low enthalpy drop; • overheating not necessary; • supercritical pressure avoided; • no risk of blad erosion;
Costi di gestione e manutenzione	• the water must be treated; • licensed person in charge needed • high pressures and temperatures	• the fluid does not produce corrosion • low operational and maintenance cost; • completely automatica
Altre caratteristiche	• suitable only for capacity higher than 19 MWe • low flexibility and poor performance at partial load	• high flexibility at partial loads; • high overall efficiency; • possibility of exploiting low temperature sources (90°C)

ORC CYCLE

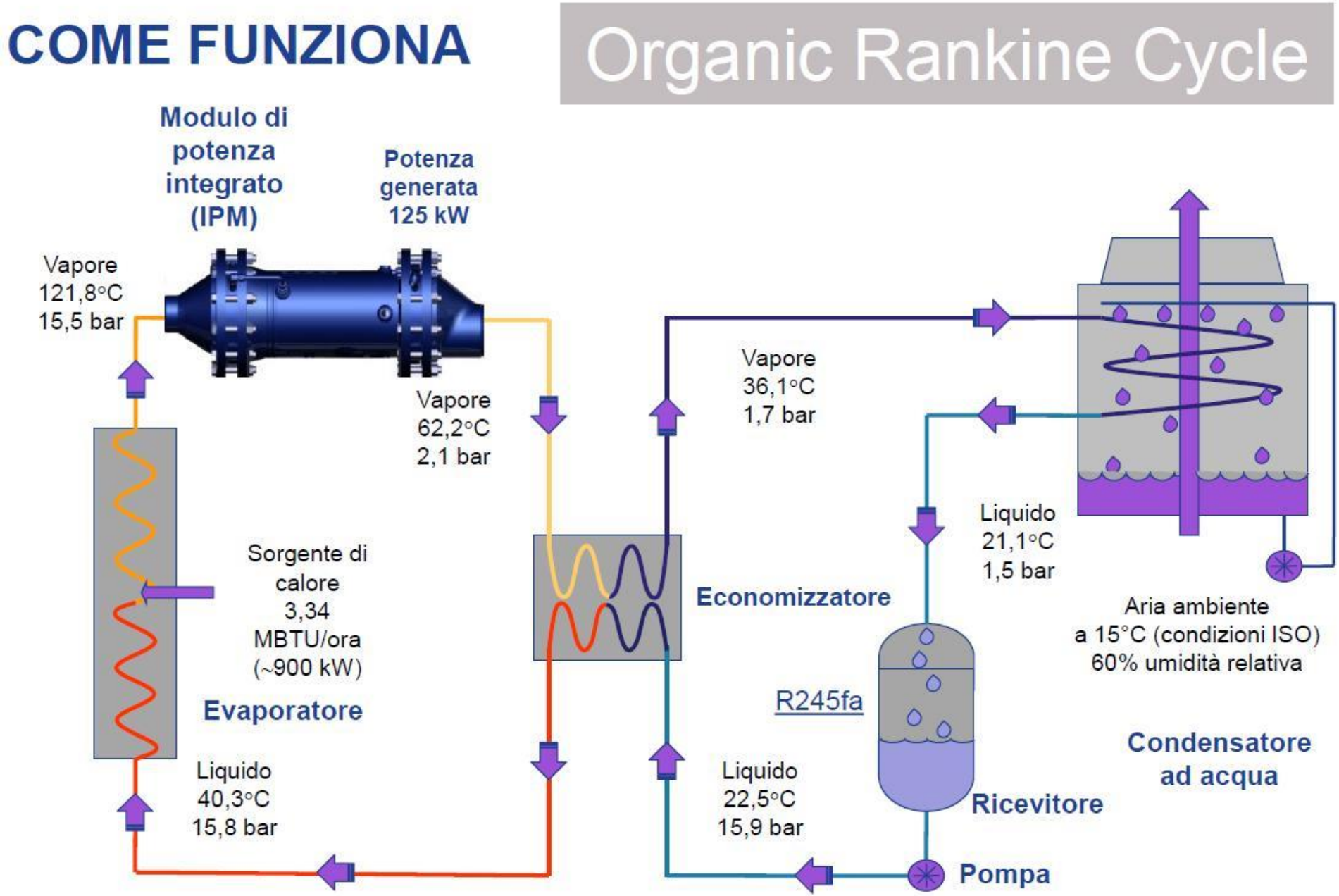


The evaporator uses diathermic oil at high temperature to preheat and vaporize a suitable organic fluid (8-3-4). The organic vapour expands in the turbine (4-5) which is directly connected to the electric generator by an elastic joint. The vapour passes through the regenerator (5-9) so that the organic fluid is preheated (2-8). The organic vapour is then condensed in a condenser cooled by cooling water (9-6-1). The organic liquid is finally pumped to the regenerator (1-2) and from here to the evaporator, closing the cycle.

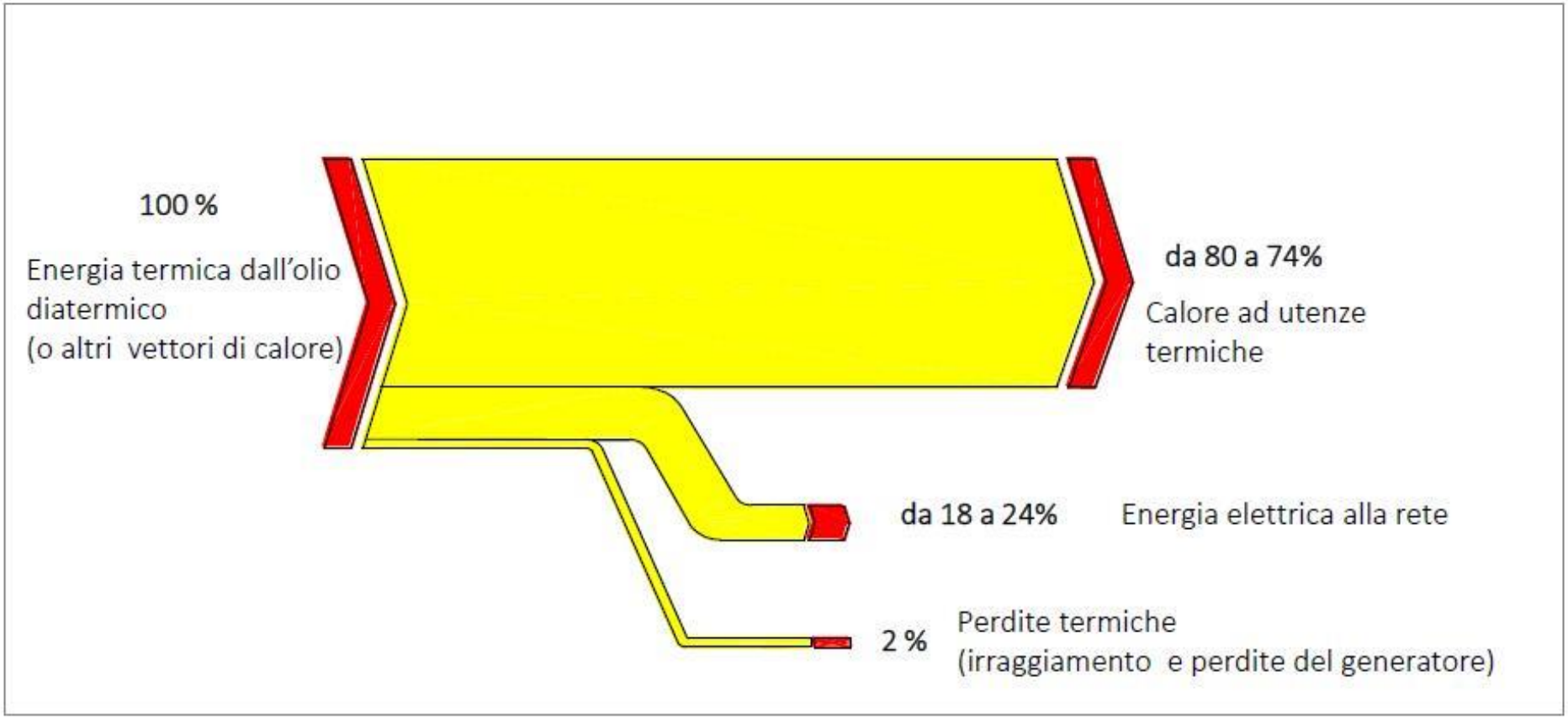
ORC cogeneration systems operate at much lower pressures than the traditional water vapour power plants, and with capacities that can be as low as one hundred kW.



They can be supplied by wasted heat (heat recovery)

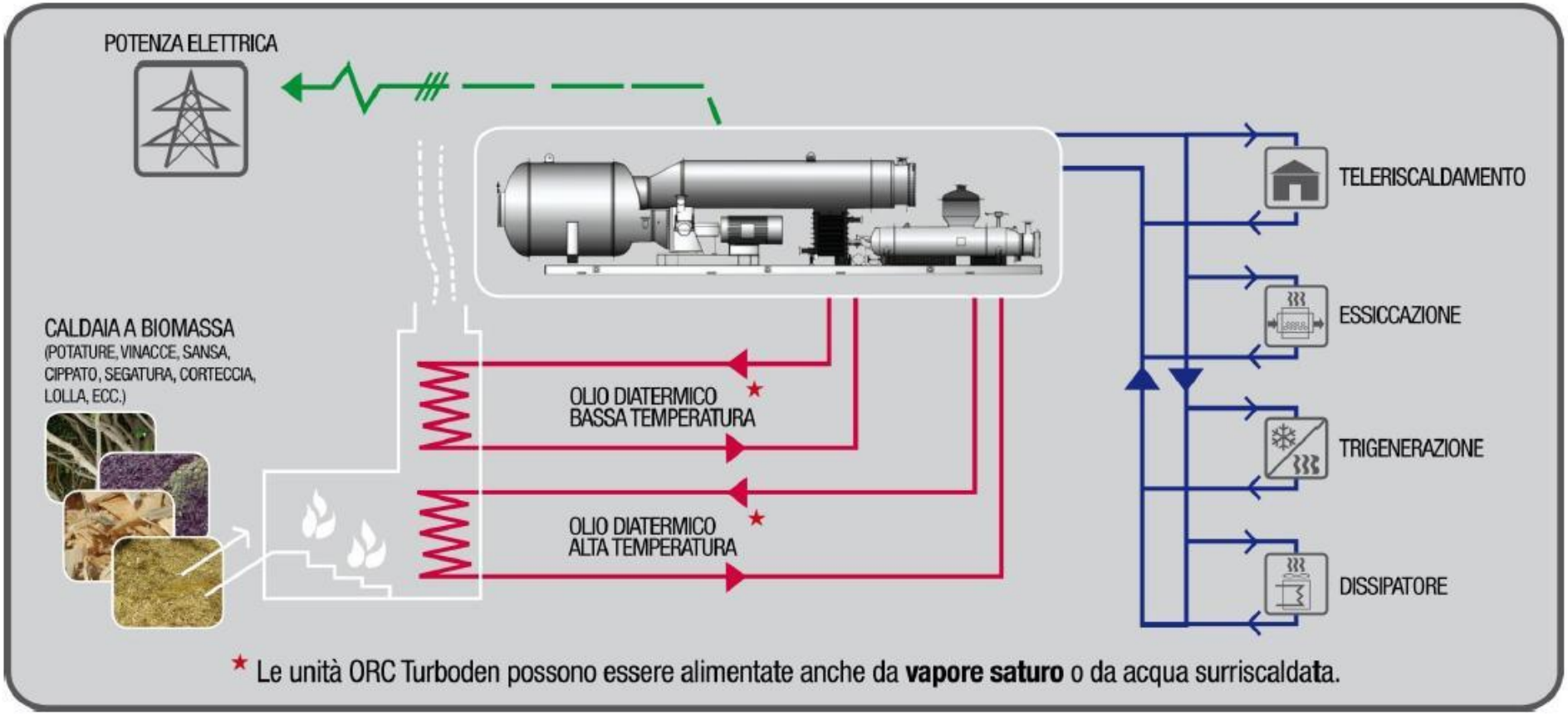


Performance of an ORC plant



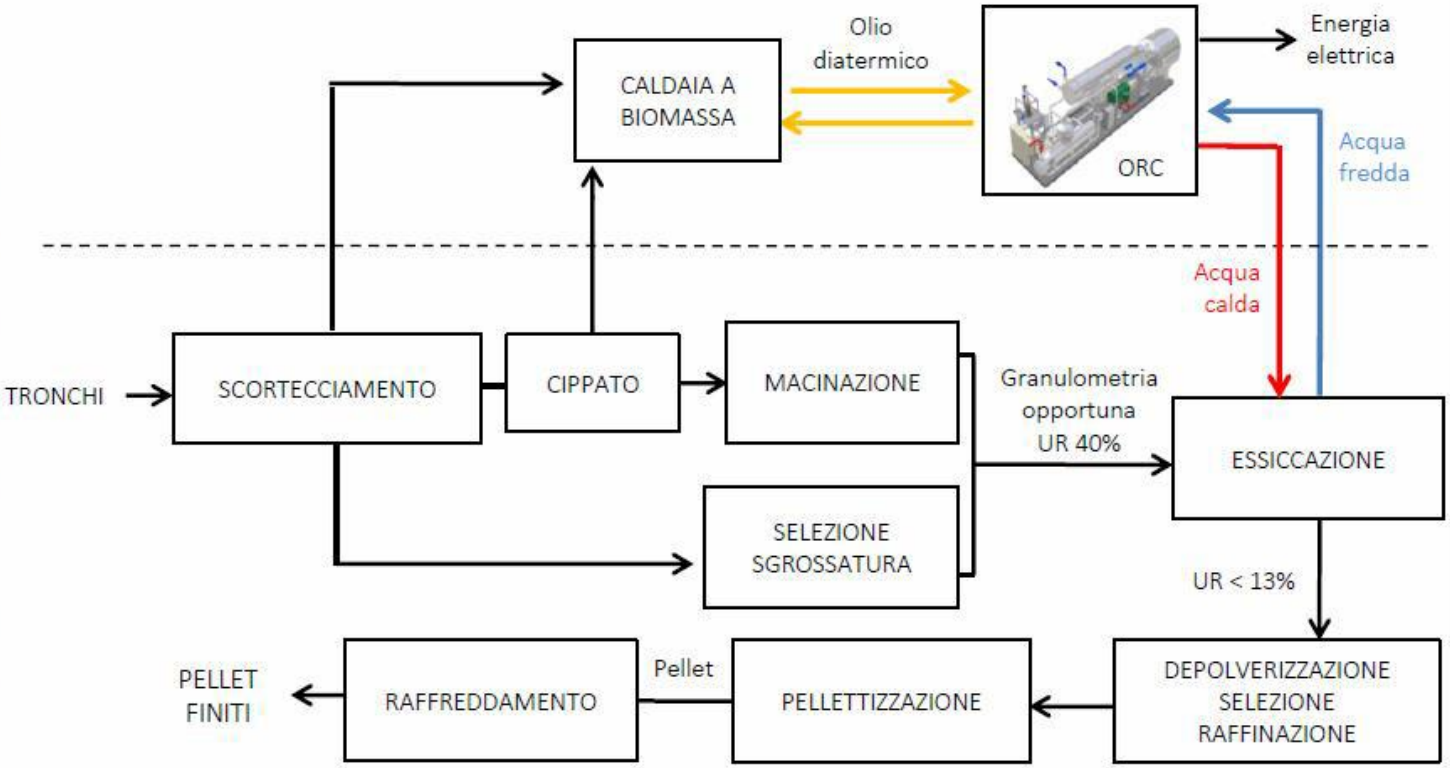
- Efficienza elettrica lorda: circa 24%
- Efficienza energetica totale: 98%

Example of a cogeneration ORC plant using biomass as fuel



Esempio di impianto di cogenerazione ORC

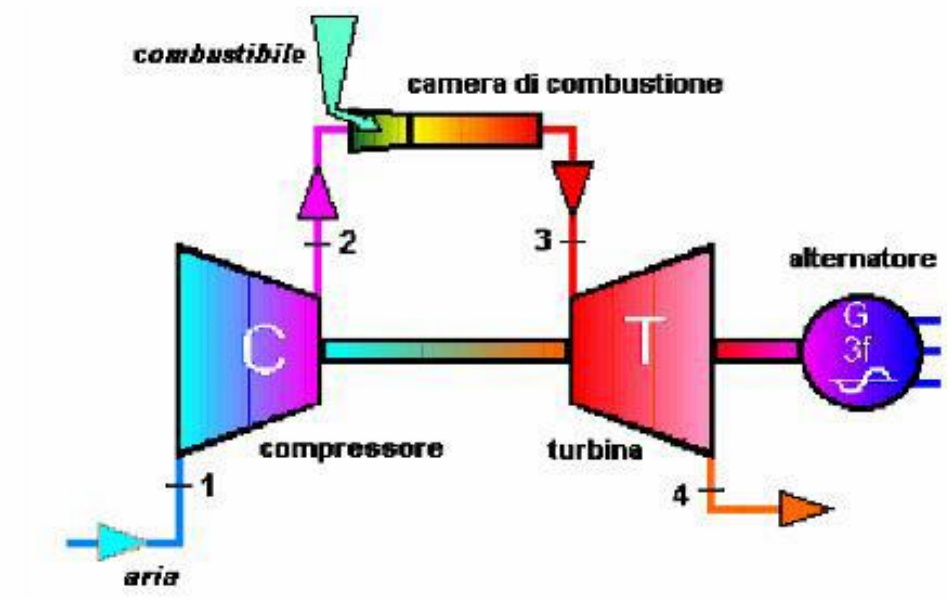
Produzione di pellet

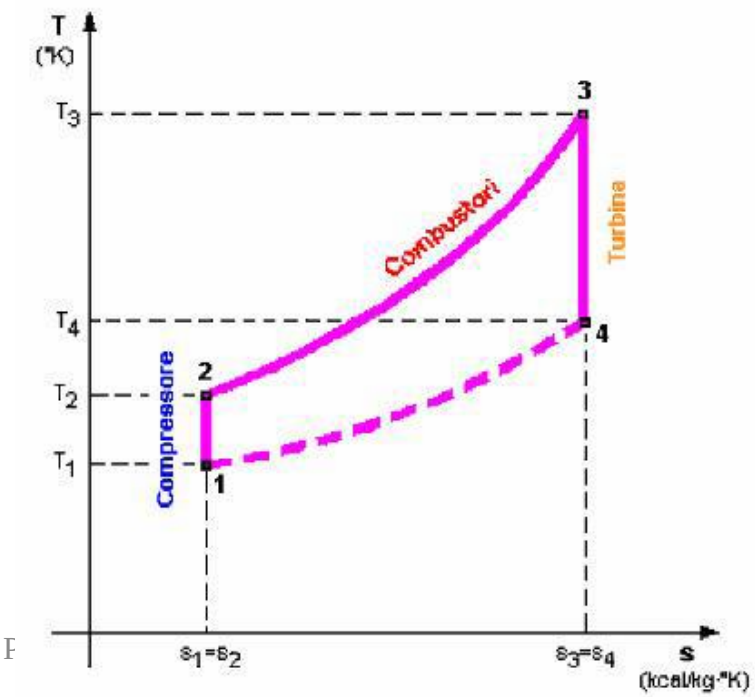


COGENERATION BY GAS TURBINES

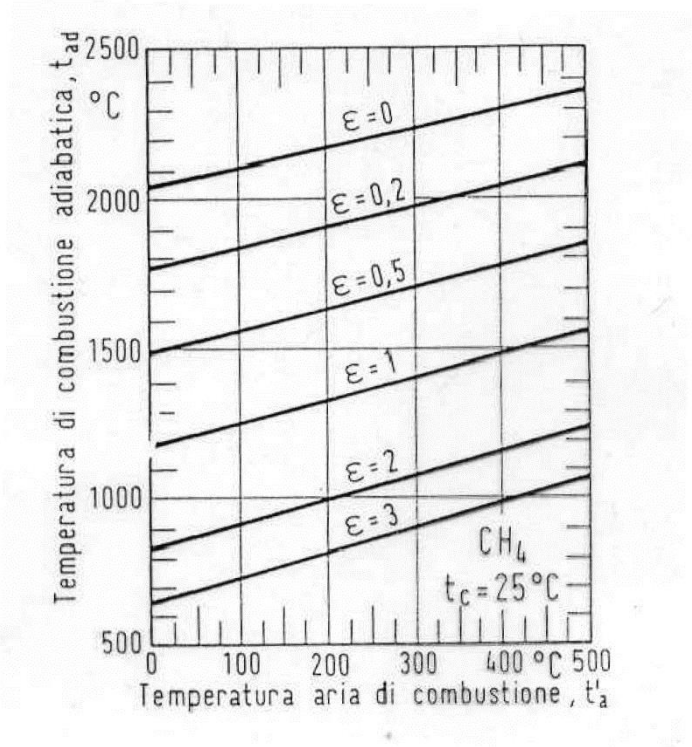
GAS TURBINE COGENERATION

Gas turbines are the equipment that received the greatest development in the last years.
Capacities can range from few hundreds of kW to 200 MW or more.
For small capacities, the efficiency can be about 20-25% and the efficiency at partial load is strongly penalized.





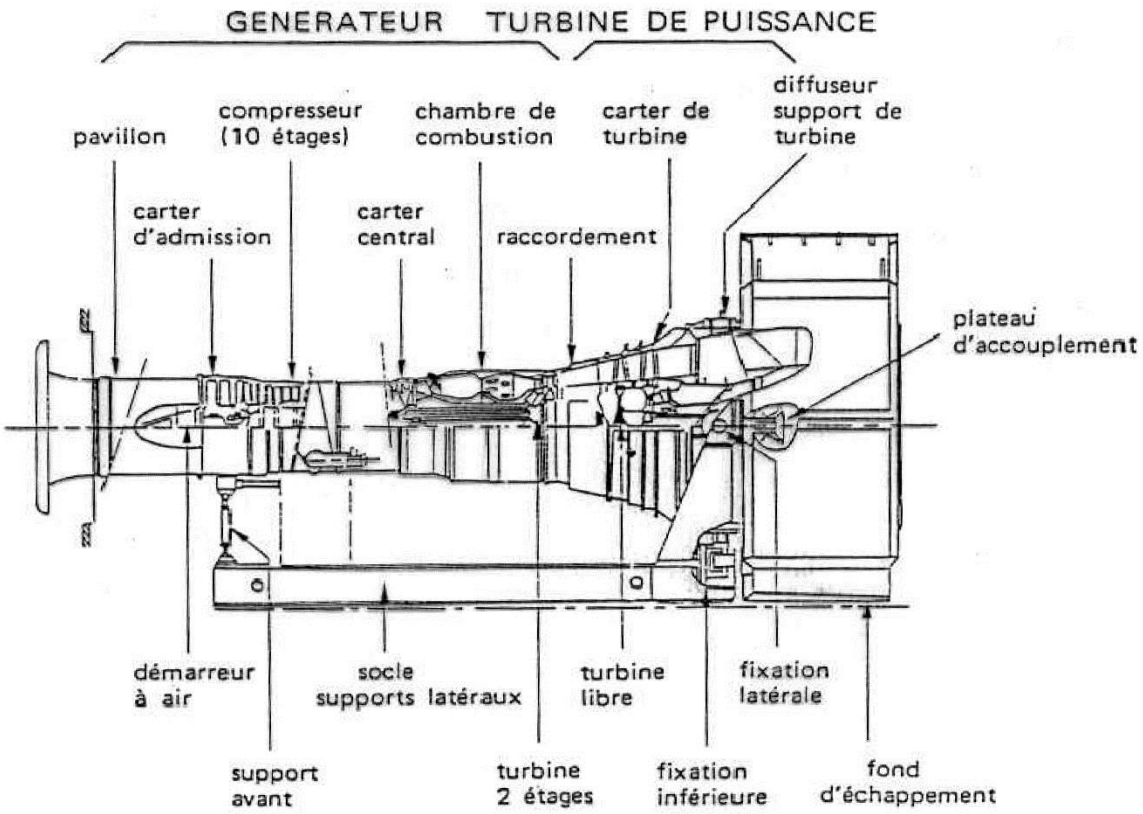
Gas turbines operate with an air excess that can be 2-300%. This facilitates a possible post combustion.



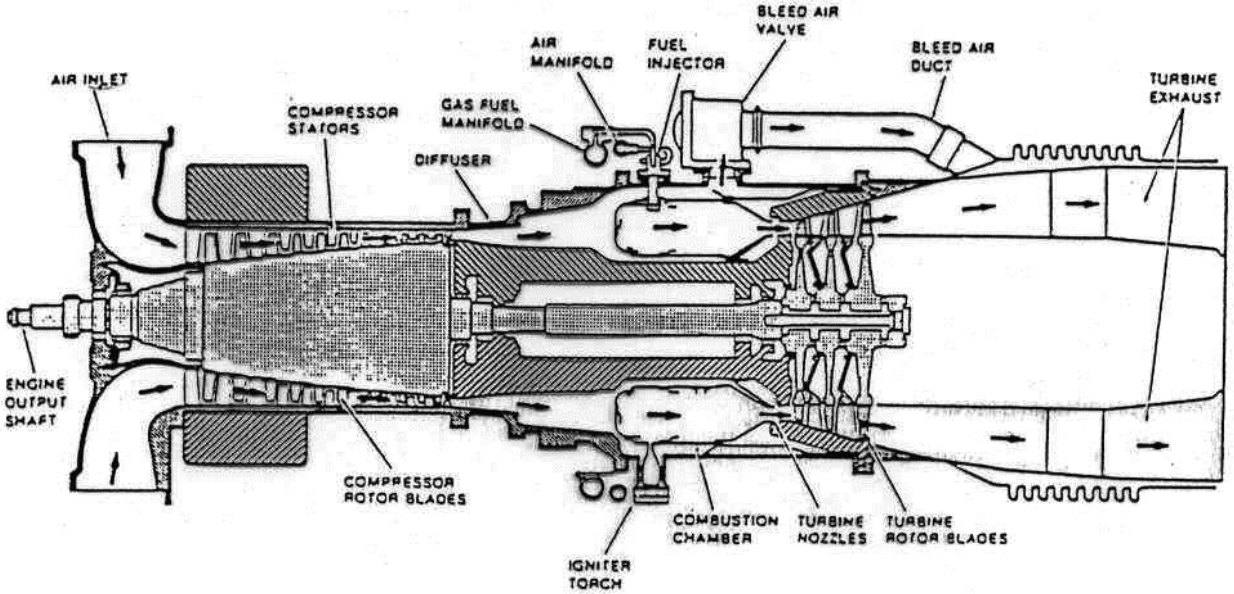
The NOx emissions can be reduced by injecting water or steam into the combustion chamber.

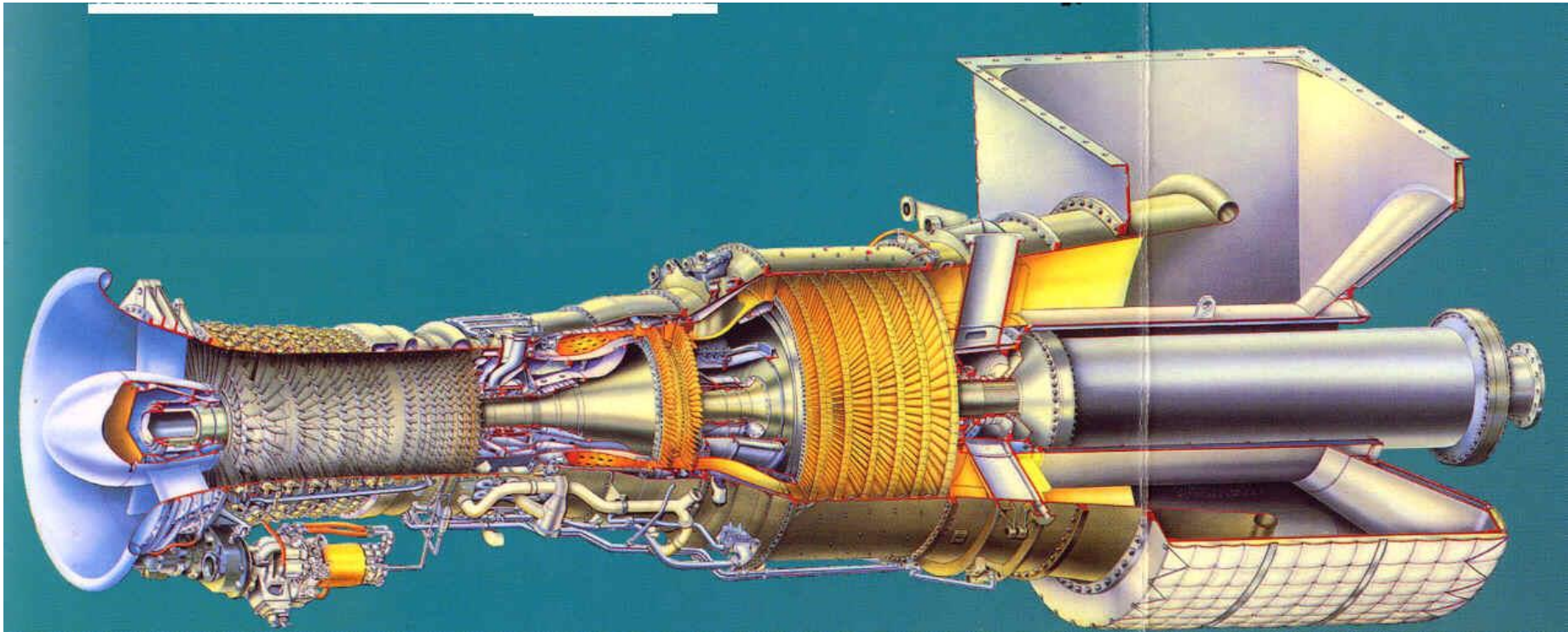
Gas turbines are light, compact and require few auxiliaries. They are very sensible to the fuel quality, and the maintenance costs are bound to the fuel quality.
The turbine is composed of an axial multistage compressor (they can be about 10). Then an annular combustion chamber follows. The burners can be even 20 with a two-flame ignition.

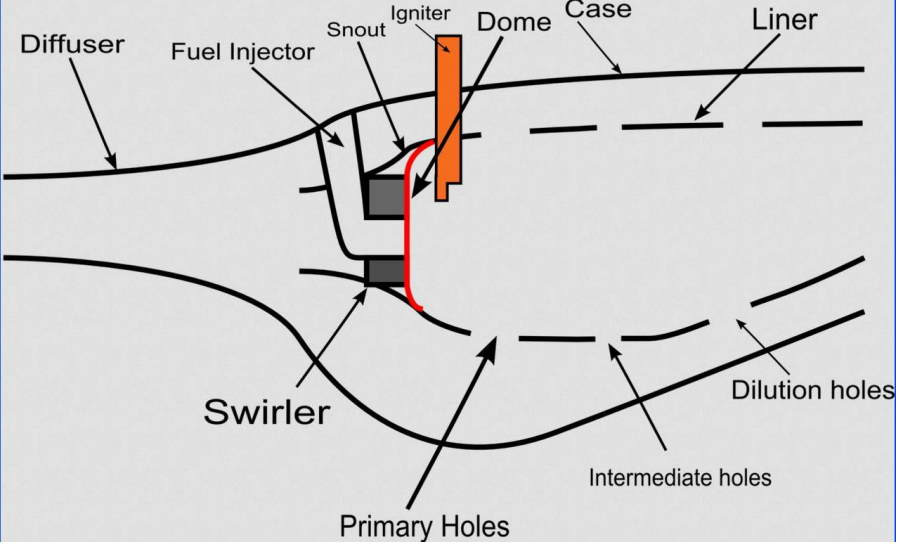
A first stage of the turbine often drives the compressor, while the subsequent provides useful work (double shaft in line)



The start is given by an electric motor or by a compressed air turbine or by a hydraulic system.





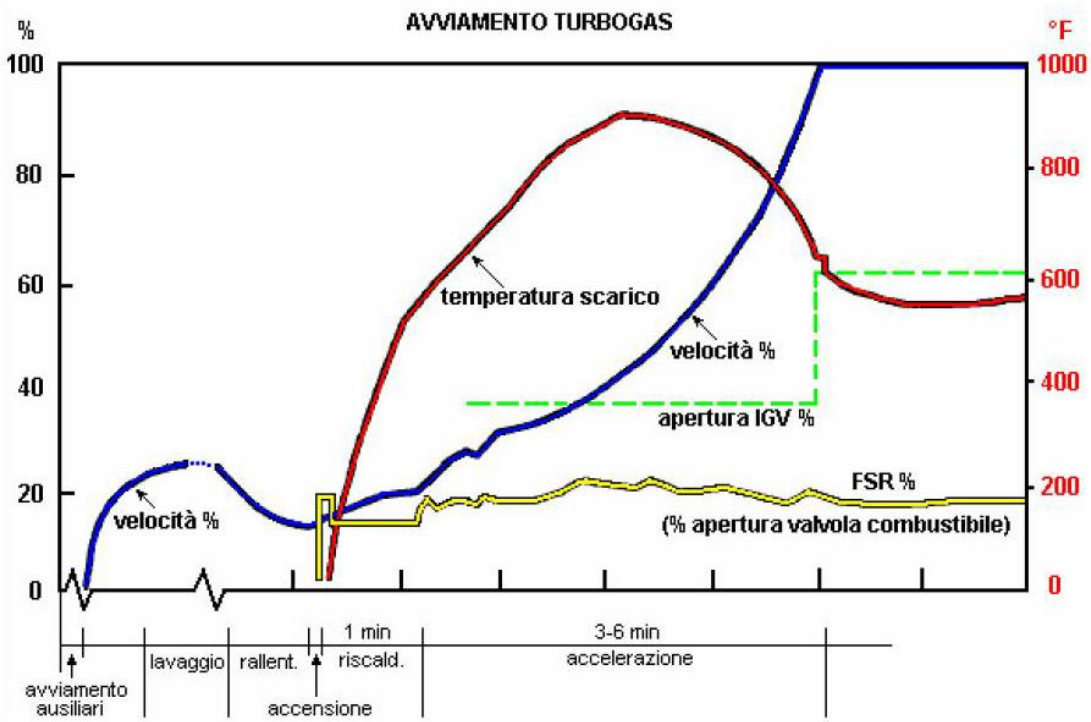


In the starting phase, thanks to the starter, the turbine accelerates till a combustion chamber washing speed (700 rpm) in 2 minutes.

(IGV=Inlet Guide Vane)

FSR, Fuel Stroke Reference is the reference value that determines the placement of the gas control valves, and in the end turbine power output

At the end of the washing, the turbine moves at the speed of about 400 rpm and the ignition can take place.



The turbine technology regards turbines for industrial use (*heavy duty*):

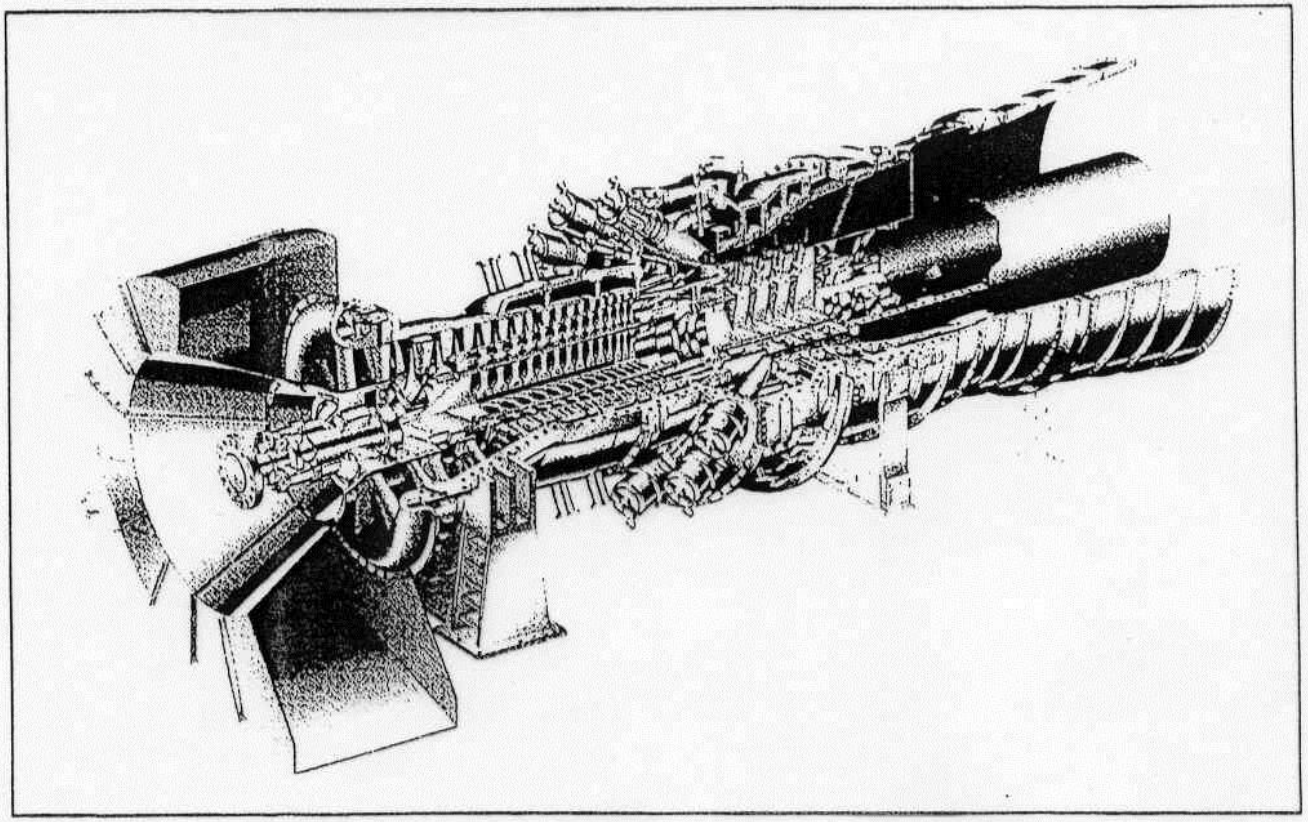


Fig. 5.6: *Spaccato di turbina a gas industriale "heavy-duty" di grande potenza. A partire dal lato sinistro del disegno (parte inferiore di questa pagina) sono individuabili la presa d'aria, il compressore, il combustore, la turbina ed infine il diffusore conico attraverso il quale vengono espulsi i gas di scarico.*

and aeroderivative turbines

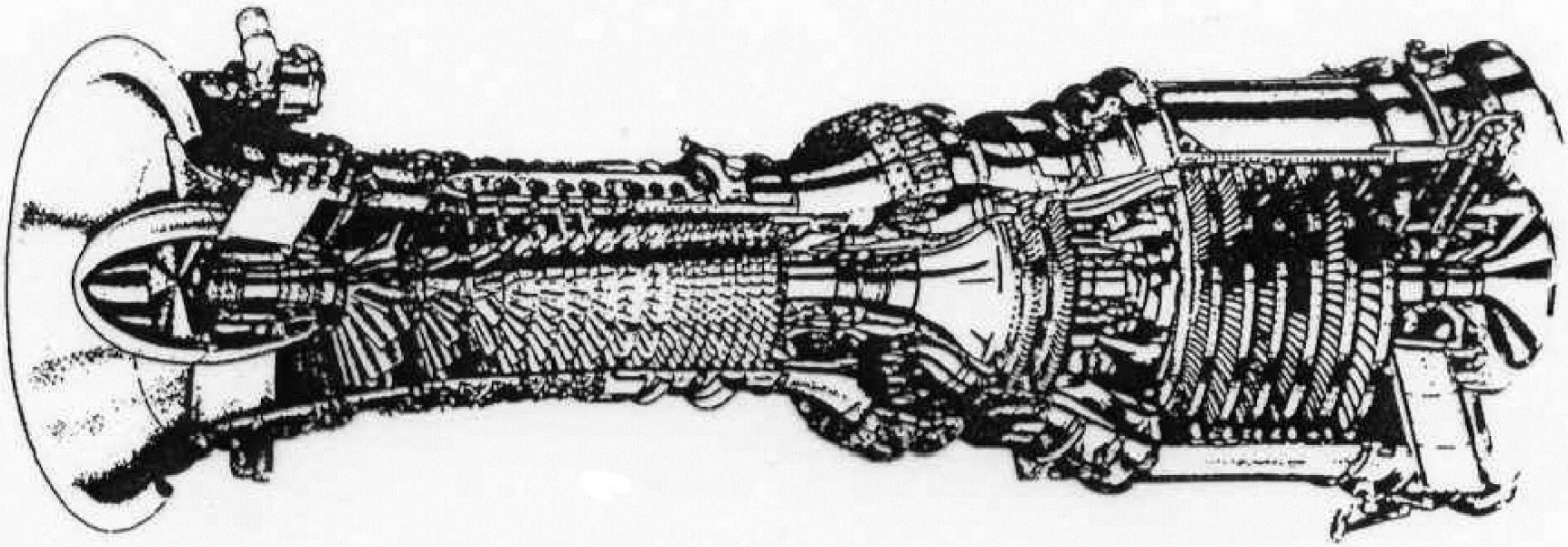
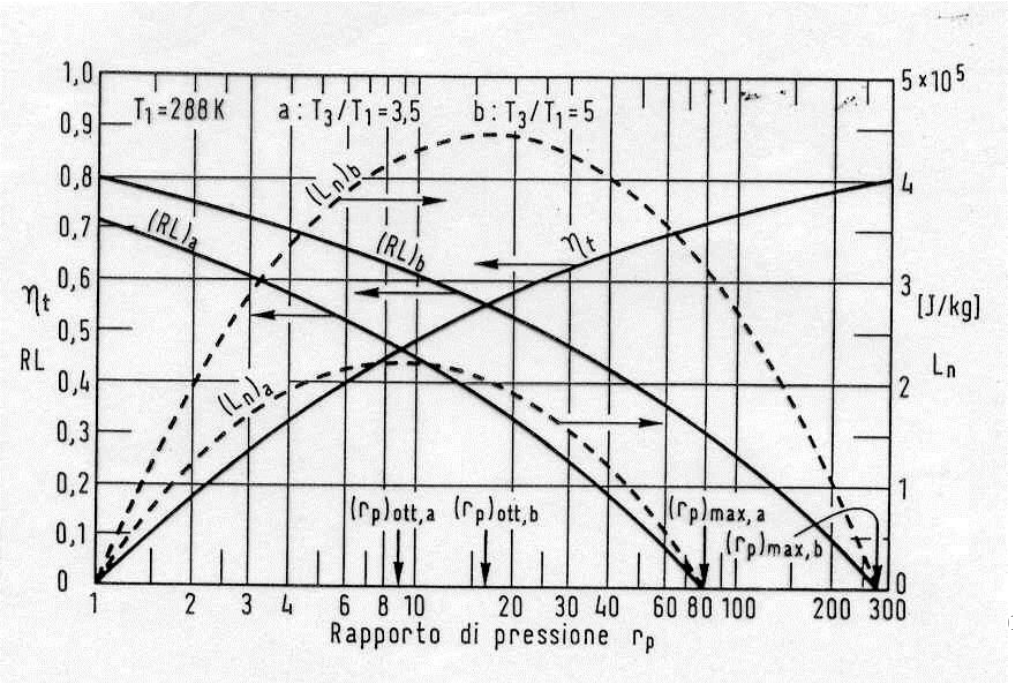


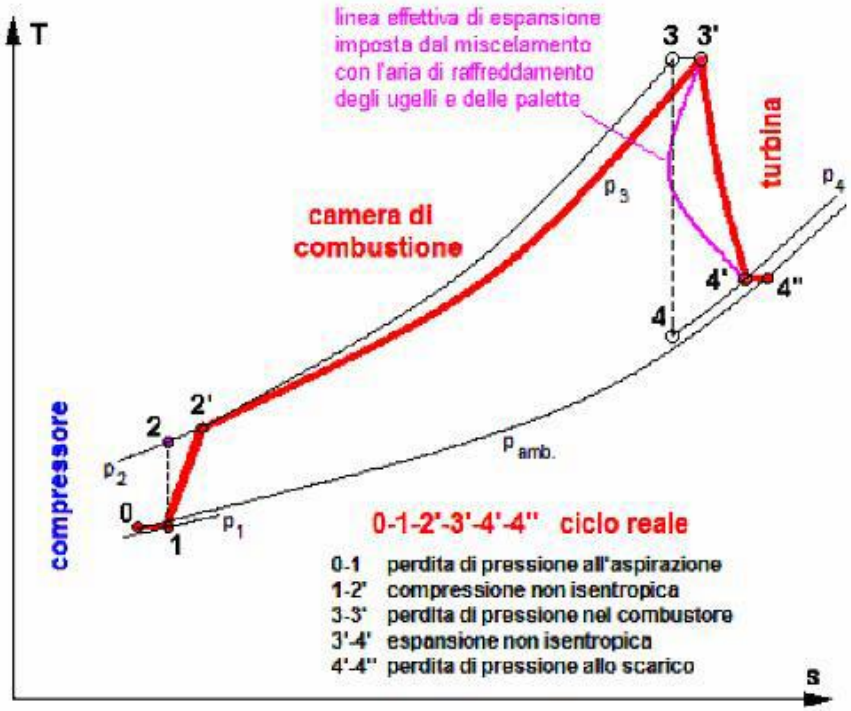
Fig. 5.7: *Spaccato di turbina a gas di derivazione aeronautica. A partire dal lato sinistro del disegno sono individuabili la campana per la presa dell'aria, il compressore, il combustore, la turbina di alta pressione ed infine la turbina di bassa pressione.*

As it is well known, the *Brayton-Joule* cycle efficiency increases when the pressure ratio goes up, but there is a limit on the inlet temperature into the turbine that must be acceptable to the blades material.

Given the bearable maximum temperature, the choice of the pressure ratio tends to favour the maximum net specific work. A high value of this work allows to reduce the size of the turbine: this is of great importance for the aircraft engines.

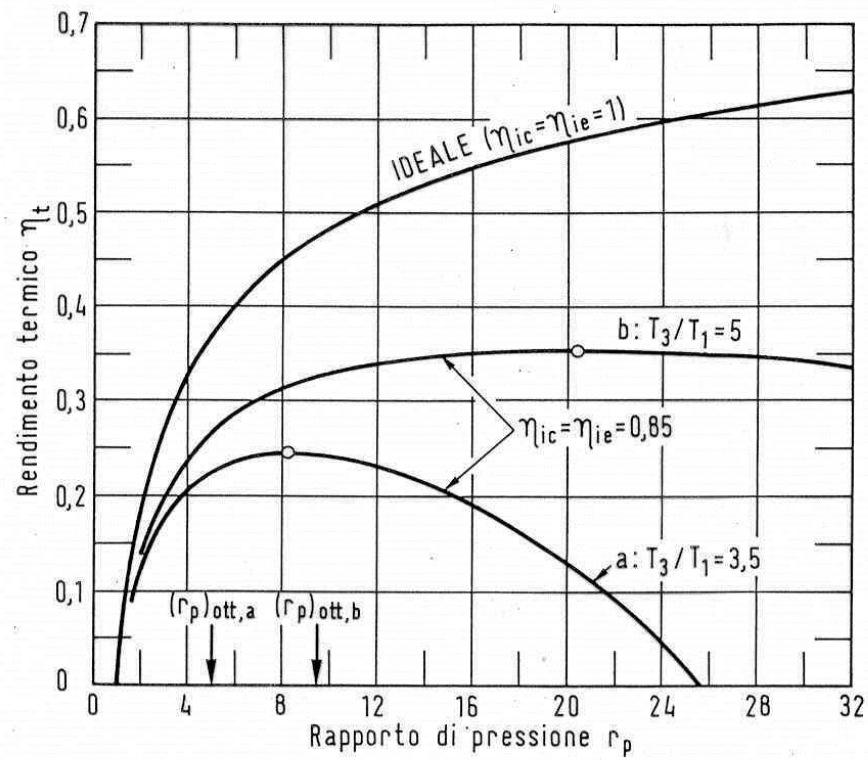


The choice of the materials and the manufacturing process allows higher inlet temperature into the turbine in the aeroderivative turbines. This increases the efficiency and the specific work. However, the cost is higher and the life shorter.



$$\eta_t = 1 - \frac{1}{r_{pk}^{\frac{k-1}{k}}} = 1 - \frac{T_1}{T_2}$$

$$(r_p)_{\max} = \lim_{T_2 \to T_3} r_p = \lim_{T_2 \to T_3} \frac{p_2}{p_1} = \lim_{T_2 \to T_3} \left(\frac{T_2}{T_1}\right)^{\frac{k}{k-1}} = \left(\frac{T_3}{T_1}\right)^{\frac{k}{k-1}}$$



The graph supplies the efficiency as a function of the compression ratio in an ideal situation and for 0.85 isentropic efficiency in compression and expansion. Pressure ratios of maximum specific net value are given in real operations.

Current situations can be summarized by the following table taken by a dated technical paper with the shrewdness of considering future as new and new as conventional:

Table 1. Performance data for the studied gas turbine types. Data represent reasonable values valid for ISO conditions, i.e. ambient air of 1.013 bara, 15°C and relative humidity of 60%

GT label	η_{EL} (%)	T_{TO} (°C)	f_X (kg/kWh fuel)	T_{T1} (°C)	Compression ratio
NI	30	550	4.1	1,250	13
CI	25	495	5.2	900	7
CI1	26	450	5.4	—	9
CI2	32	375	6.2	850	12
FA	40	450	4.3	1,220	30
NA	35	485	4.3	1,220	21
CA	28	560	4.2	1,035	10
CA1	28	440	5.4	875	9

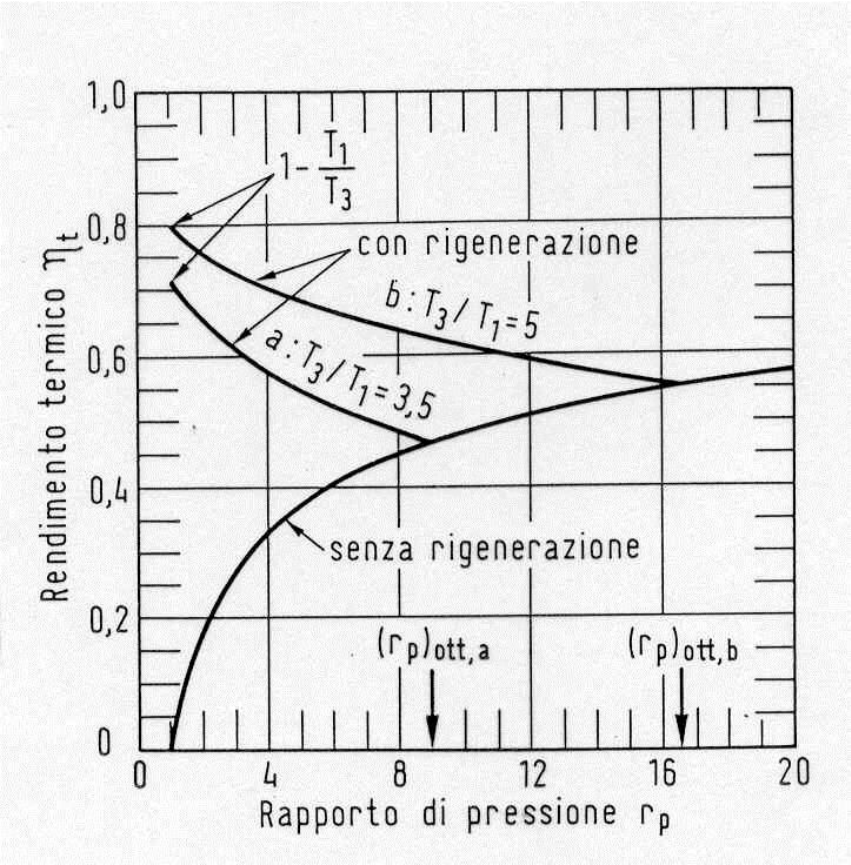
NI=New Industrial;
CI= Conventional Industrial;
CI1/2=Conventional Worse Alternative;
FA = Future Aeroderivative;
NA = New Aeroderivative;
CA = Conventional Aeroderivative;
CA1=Conventional Worse Alternative
f_x = specific flow (exhaust gas) to fuel ratio

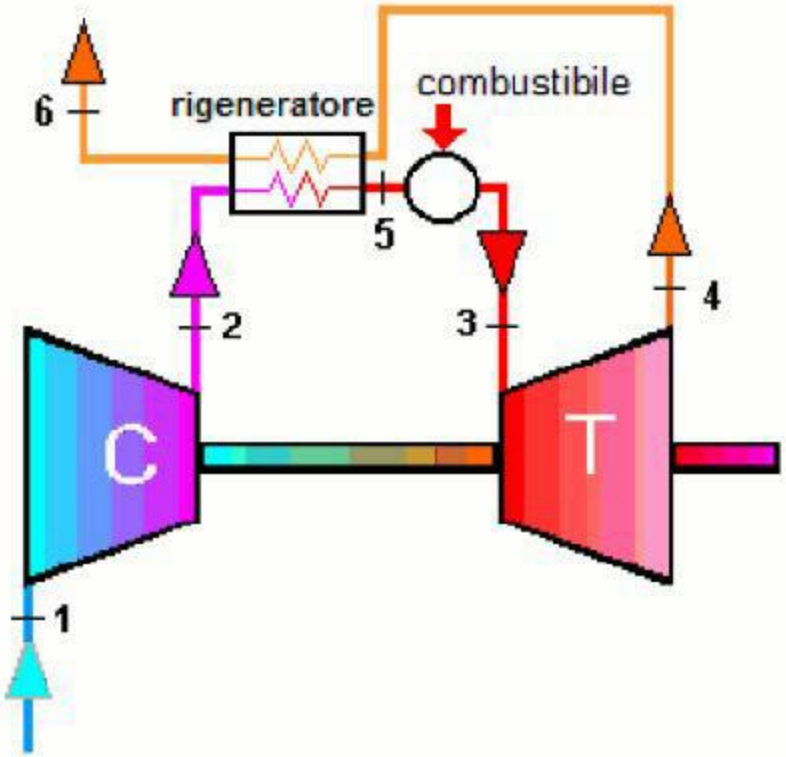
Taken from Stroemberg et al: Consequences of Recent Gas Turbine Developments for Industrial CHP Applications - Heat Recovery Systems Vol. 13 pp. 219-231, 1993.

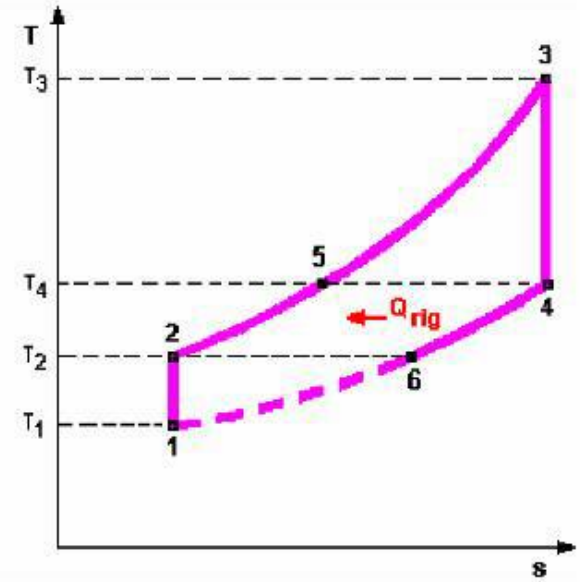
Data taken from a catalogue of ANSALDO Energia

THERMODYNAMIC DATA					
Model		V64.3A	V94.2	V94.3A	V94.2K(3)
Frequency	S-1	50/60(1)	50	50	50
Turbine inlet temperature according to ISO 2314	°C	1190	1060	1230	1060
Power output at terminals (ISO-Base load)(2)	MW	68	159	258	170
Pressure ratio	–	16.6	11.0	16.6	11
Exhaust mass flow	kg/s	192	510	634	510
Exhaust temperature	°C	589	539	568	539
Efficiency at terminals (ISO-Base load)(2)	%	34.8(1)	34.5	38.4	36.8
NOx Emissions	ppm	25	25	25	25

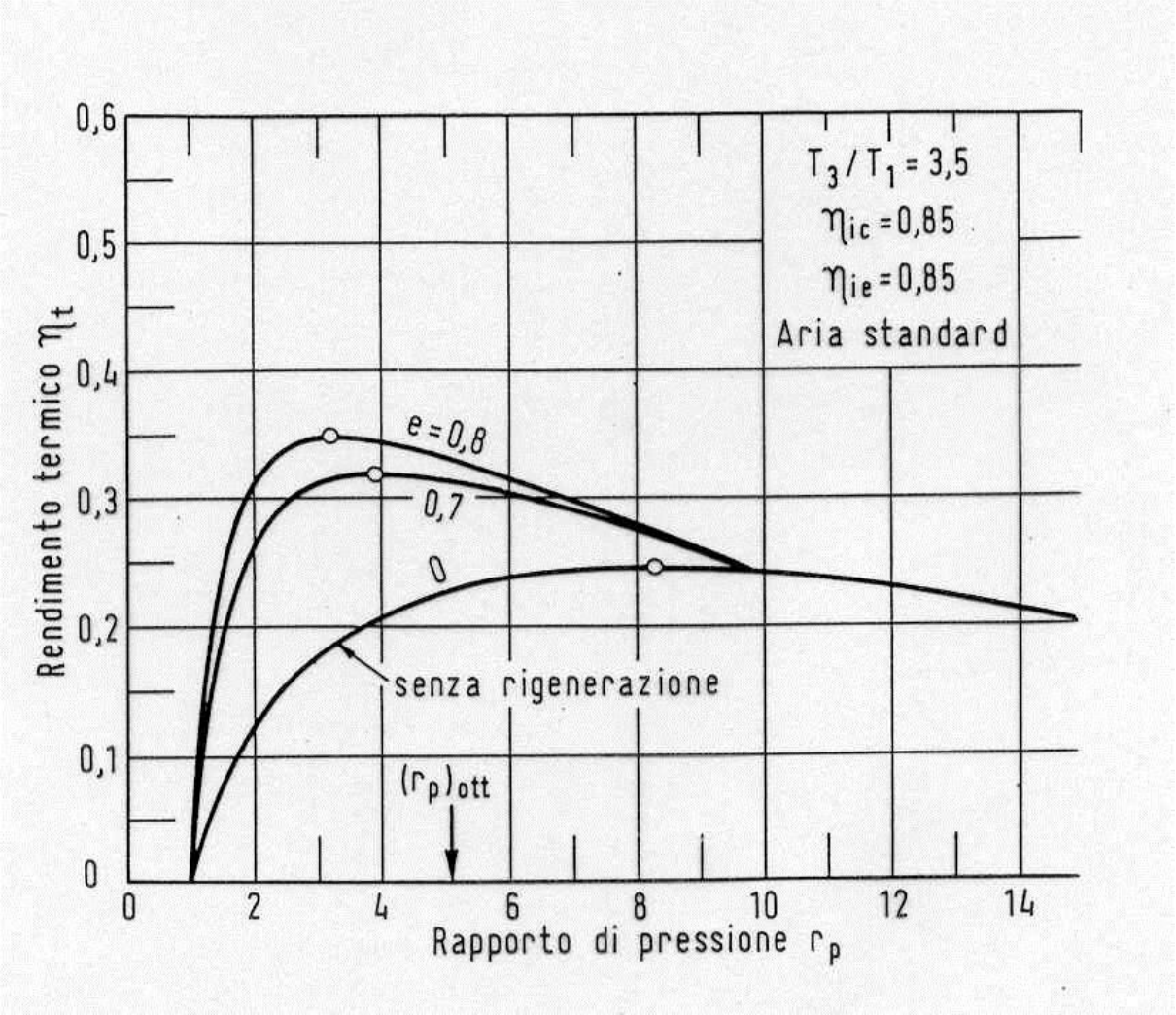
The high pressure and temperatures values that can be reached nowadays have put aside the turbine technology that operates with regeneration.



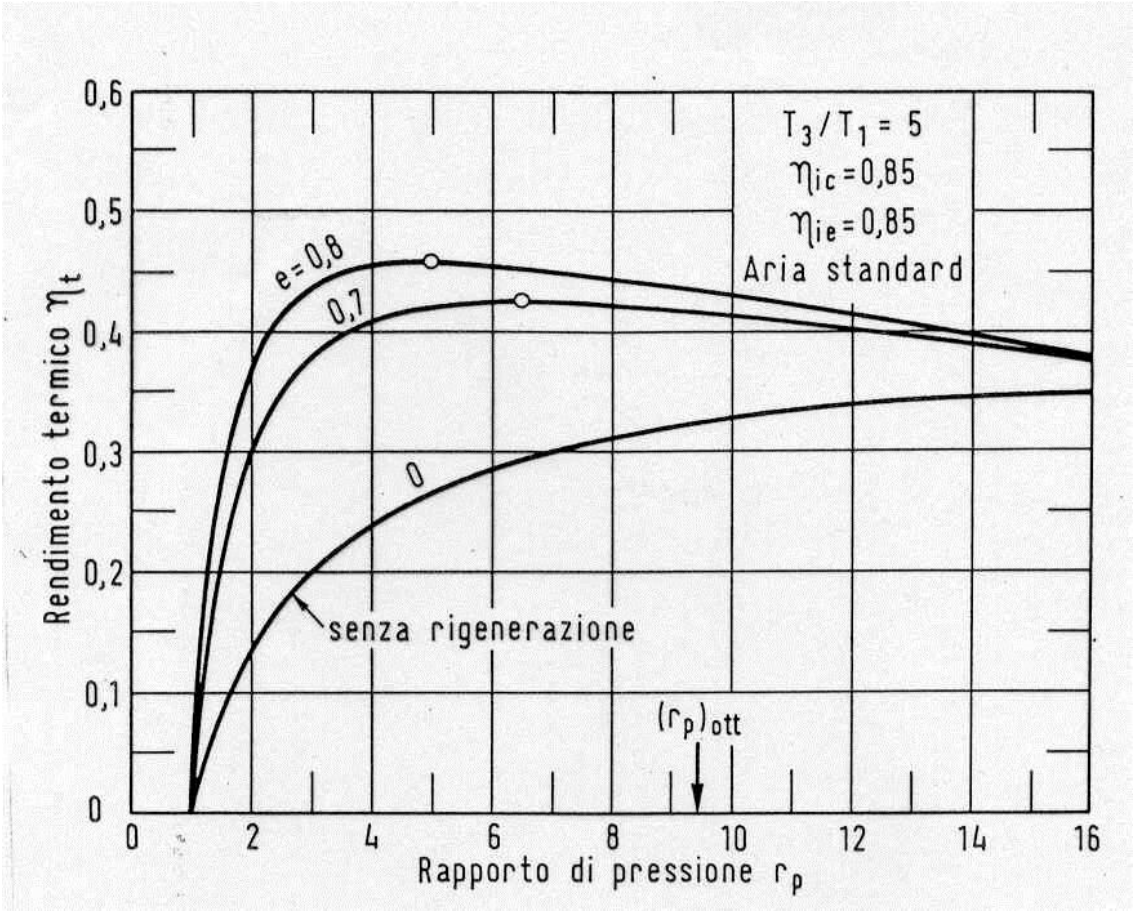




Indeed, the regenerative cycle is expensive and bulky and it can be justified only when the inlet turbine into the turbine are not too high.



When the pressure ratio increases, the possible advantage of a regenerative cycle decreases, especially since regenerative cycles prevents the use of the gas exhaust enthalpy for cogeneration.



The evolution of materials and special alloys has allowed to increment remarkably the turbine inlet temperature.

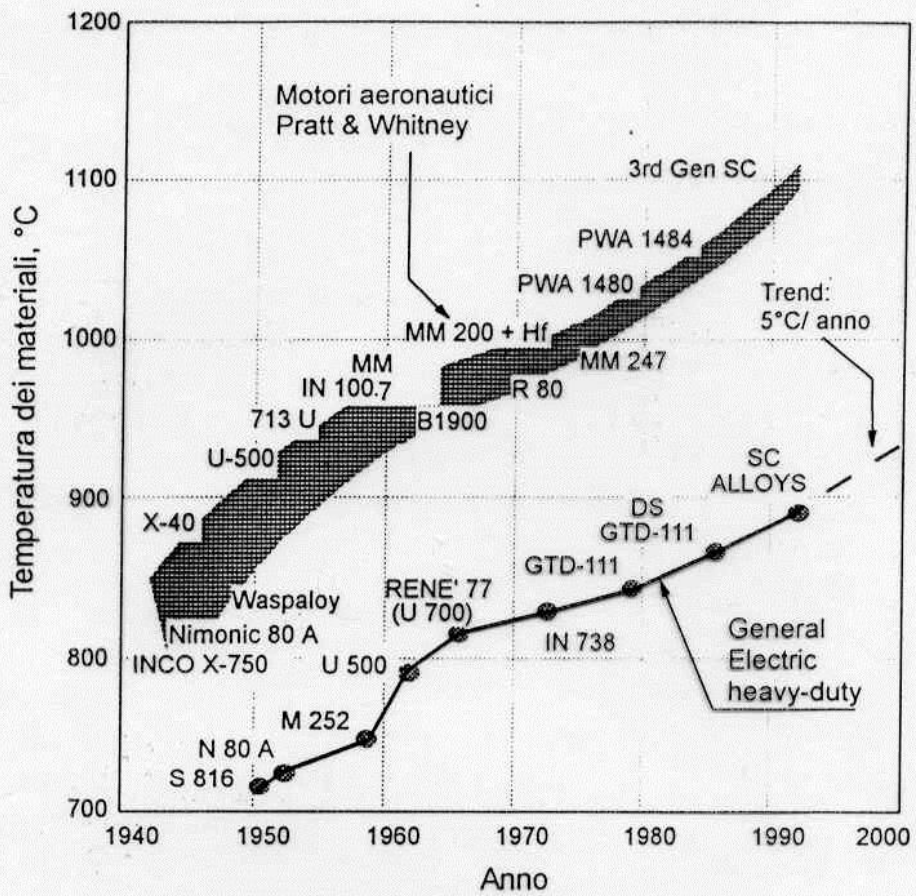


Fig. 5.2: *Evoluzione della temperatura operativa dei materiali avanzati impiegati nelle turbine a gas aeronautiche ed industriali (o heavy-duty).*

Besides the use of special materials, a fundamental factor in the turbine inlet temperature increase has been a cooling system of the blades that allows a metal temperature lower than 4-500°C regarding the temperature of the gas that reaches the blades.

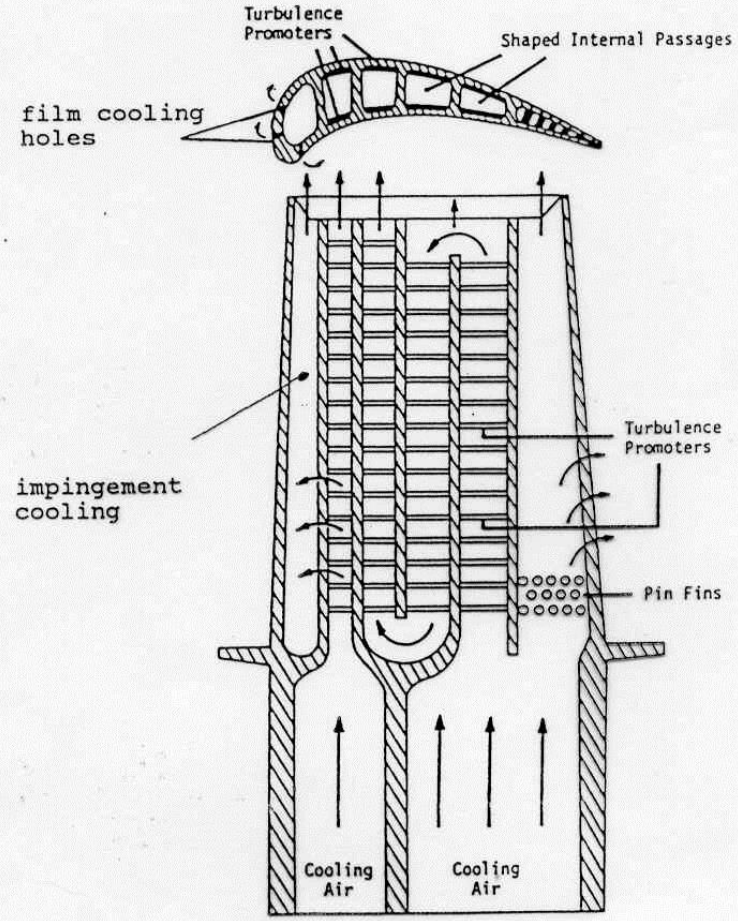


Fig. 5.4: *Struttura della pala rotorica di una moderna turbina a gas. L'aria relativamente fredda prelevata dal compressore (vedi Fig. 5.3) percorre una serie di canali interni ricavati all'interno della pala e viene quindi espulsa attraverso una serie di fori alla superficie oppure alla cima della pala. Grazie all'aria che circola all'interno della pala, la temperatura del metallo si mantiene a valori molto inferiori a quelli dei prodotti di combustione che fluiscono all'esterno: nel primo stadio delle turbine di ultima generazione i gas possono raggiungere i 1300°C. mentre il metallo rimane al di sotto di 800-850°C.*

The efficiency usually increases when increasing the turbine power. Moreover, a constant efficiency increase can be observed over the years due to the constant increase of the pressure ratio.

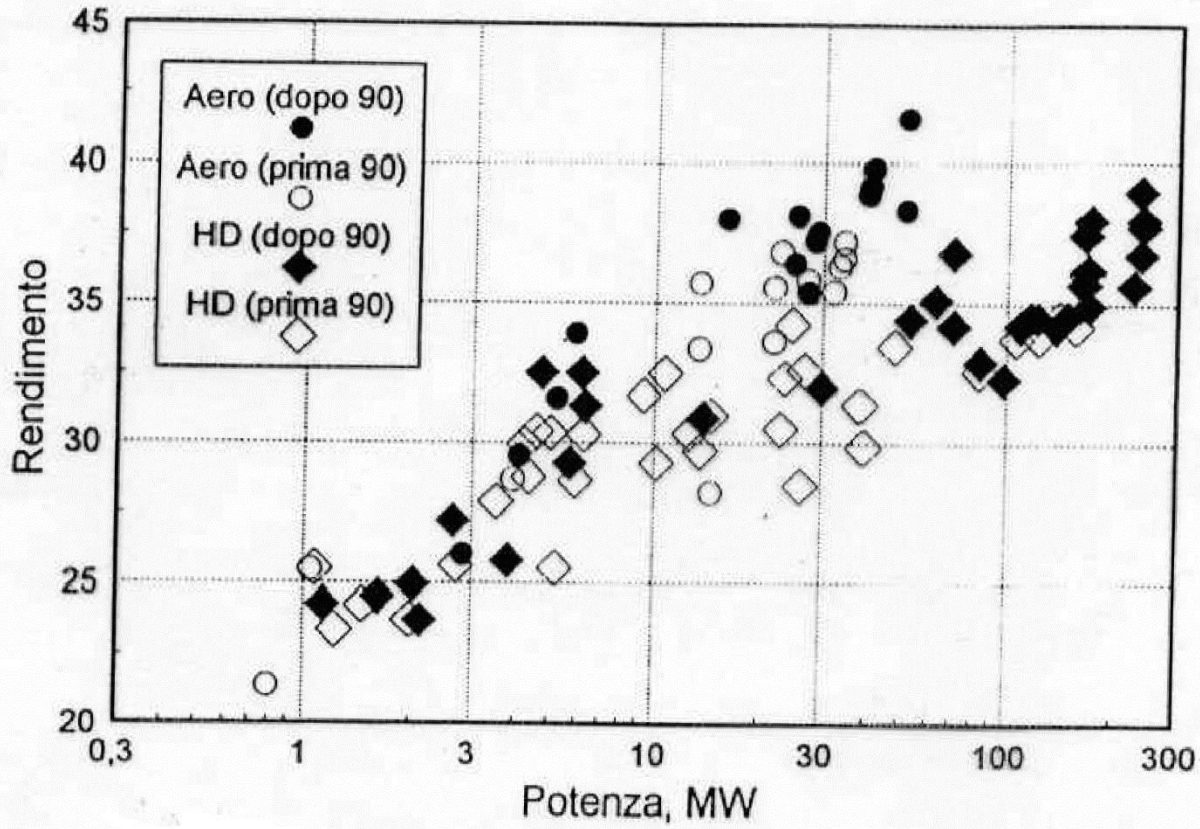


Fig. 5.10: *Rendimento in condizioni ISO delle turbine a gas oggi sul mercato. La figura evidenzia il significativo effetto scala già illustrato in Fig. 2.5, in virtù del quale il rendimento aumenta all'aumentare della potenza.*

Nowadays, the turbine inlet temperature can exceed 1400°C in the aeroderivative turbines

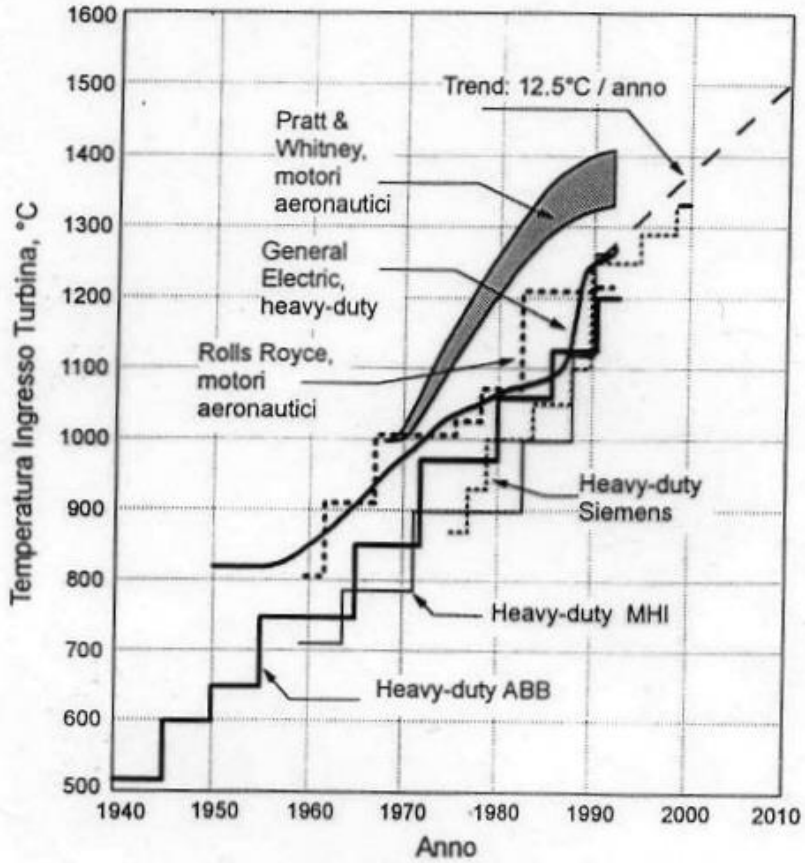
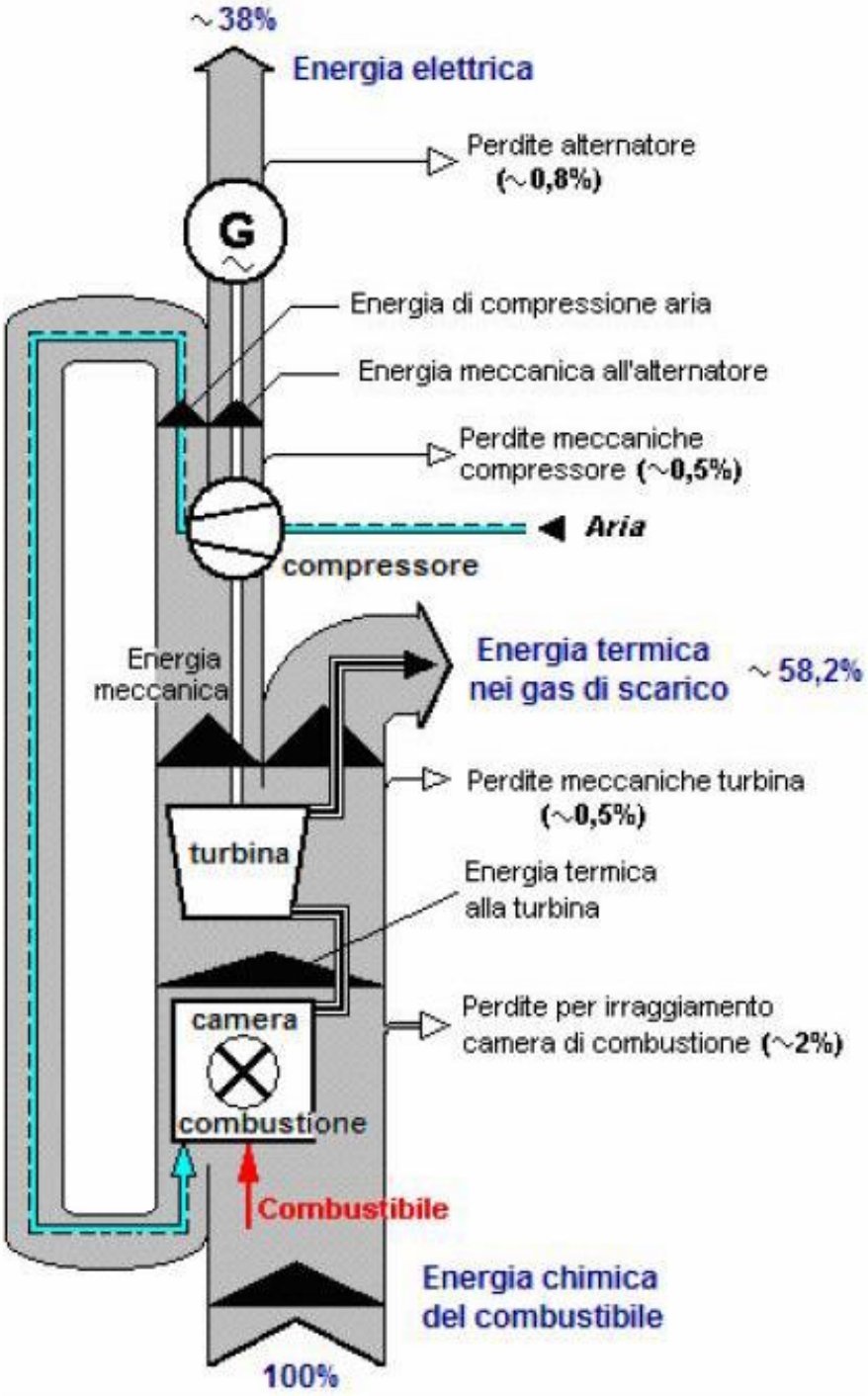


Fig. 5.1: *Evoluzione della temperatura di ingresso in turbina per turbine a gas aeronautiche e industriali (dette anche "heavy-duty"). La temperatura riportata sull'asse delle ordinate (TIT = Temperatura di Ingresso in Turbina) é la temperatura totale all'ingressodel primo rotore della turbina; a causa dei flussi di raffreddamento (vedi Fig. 5.3) essa é in generale diversa dalla temperatura all'uscita del combustore.*

The most recent gas turbines exceed a 38% electric efficiency with a heat recovery of 58% on the gas exhaust.



Let us compare a back-pressure vapour turbine with a gas turbine at the increase of the technological steam produced in the heat recovery. When this pressure increases the back-pressure turbine efficiency lowers and the thermal fraction increases.

Instead, the electric efficiency is not modified for the gas turbine, whereas the thermal fraction reduces as the heat recovery temperature is a limit for the gas exhaust cooling.

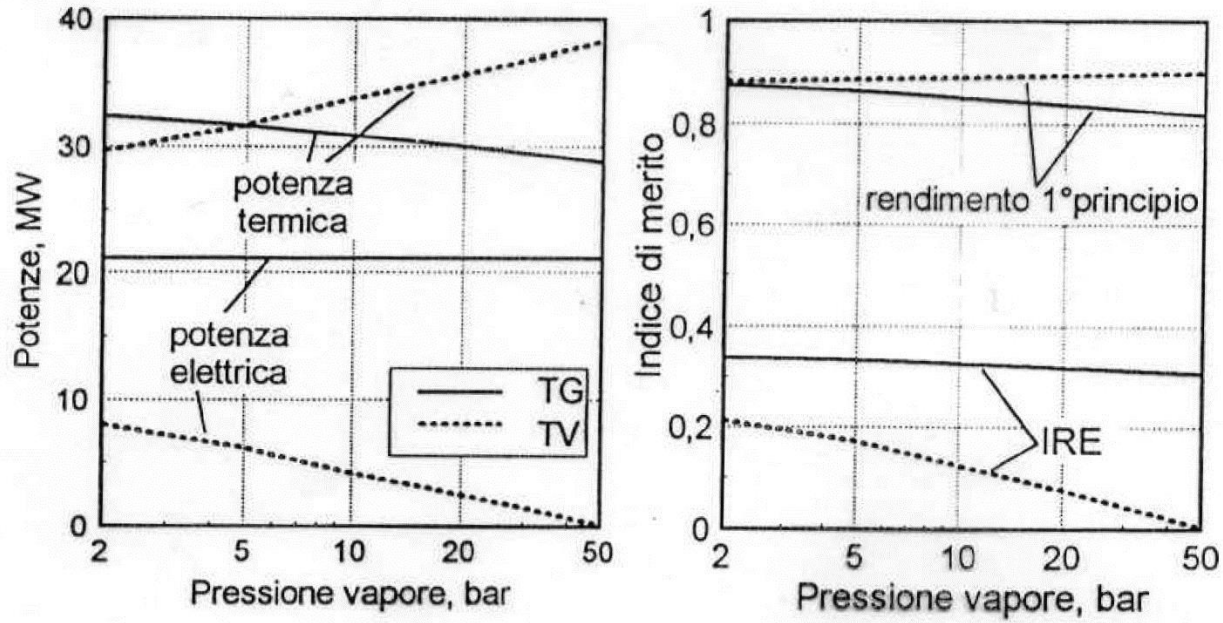
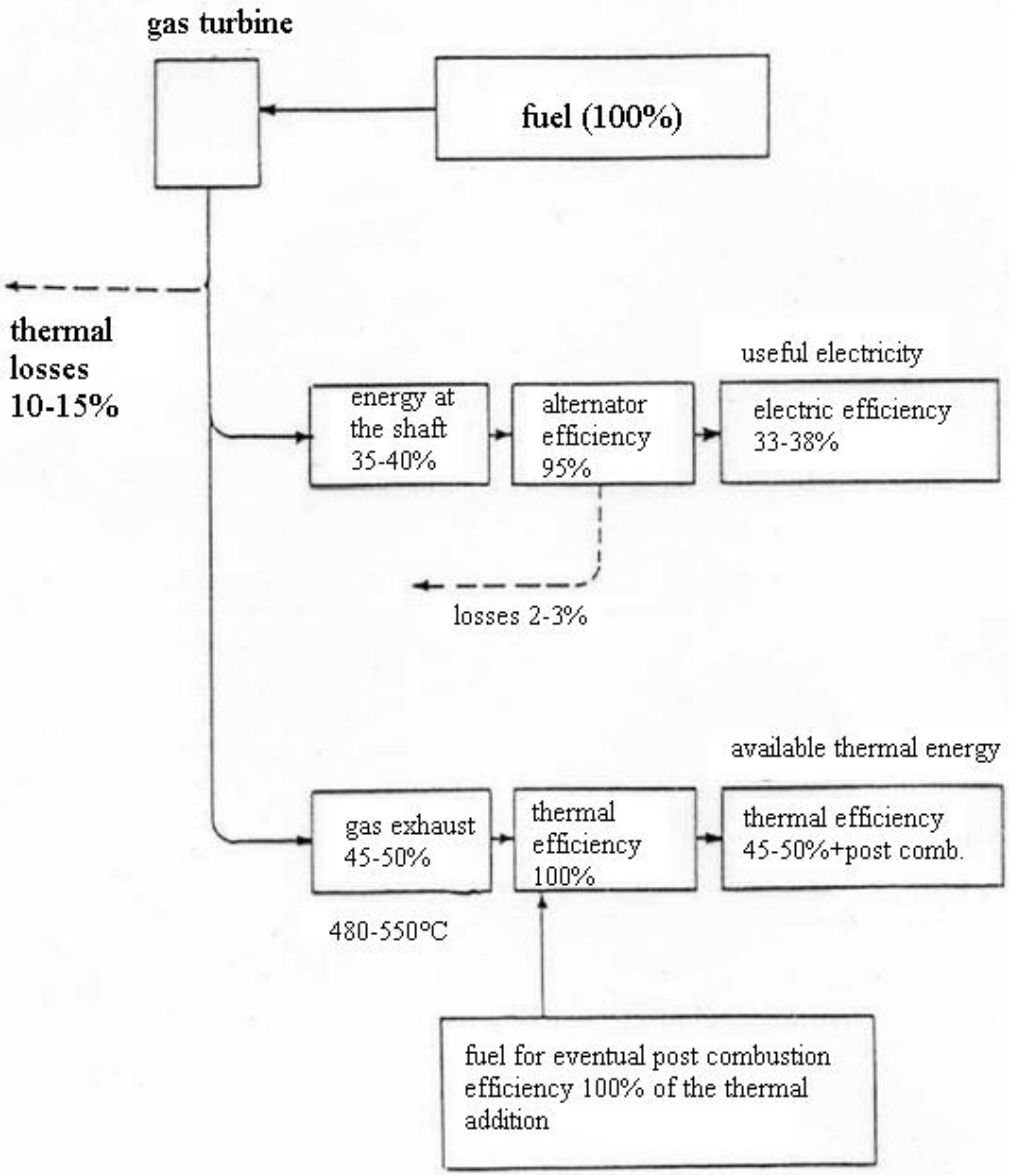
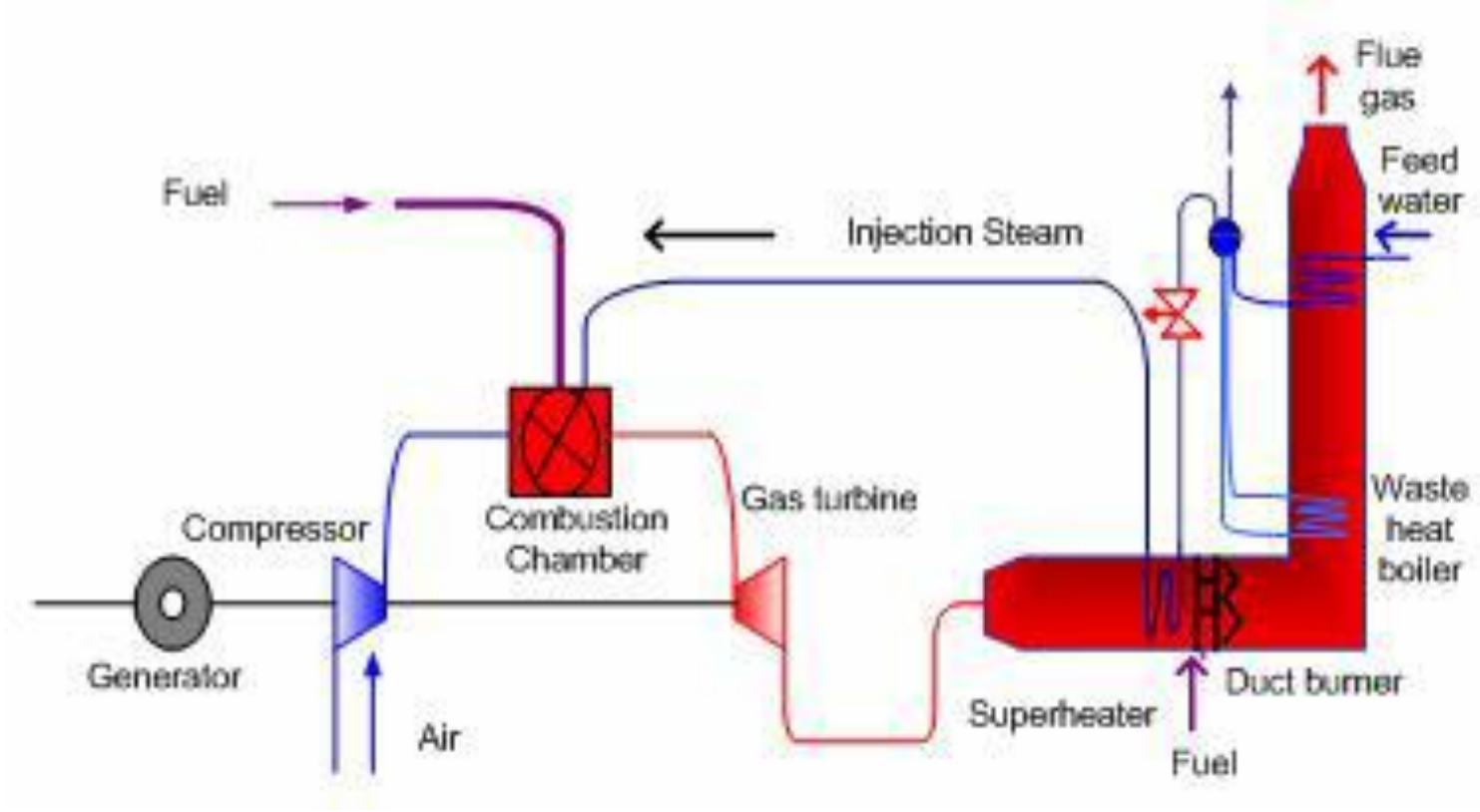


Fig. 5.20: *Potenze utili, rendimento di primo principio (eq. 1.2) e indice di risparmio energetico (IRE, vedi eq. 1.4) di due impianti cogenerativi al variare della pressione a cui è prodotto il vapore da inviare all'utenza termica. Le linee continue si riferiscono ad un impianto basato su una turbina a gas General Electric LM2500 con recupero semplice ("TG"); le linee tratteggiate ad un impianto a vapore a contropressione ("TV") con condizioni del vapore vivo 60 bar, 480°C e rendimento del generatore di vapore 90%.*

This scheme represents the general behavior of a gas turbine of small to medium capacity when the gas is exhausted at 180-200°C. The thermal/electric ratio is about 2-3 for smaller capacity whereas it can reach 1-2 for the highest capacities.
The thermal index is between 0.5 and 0.7. An possible post combustion can increase this index till even 4-5.

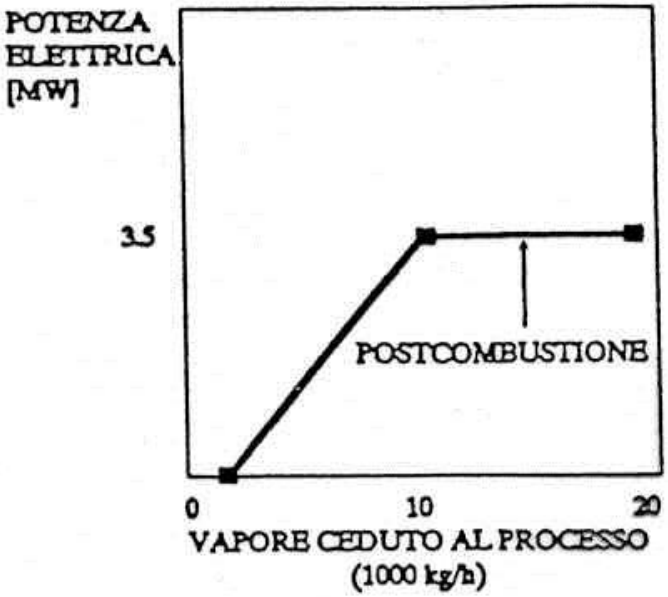


A modification of the gas turbine cycle that offers an interesting flexibility is the so-called *Cheng* cycle, where a fraction or the whole recoverable heat produces water vapor to be expanded in the turbine together with the combustion products.

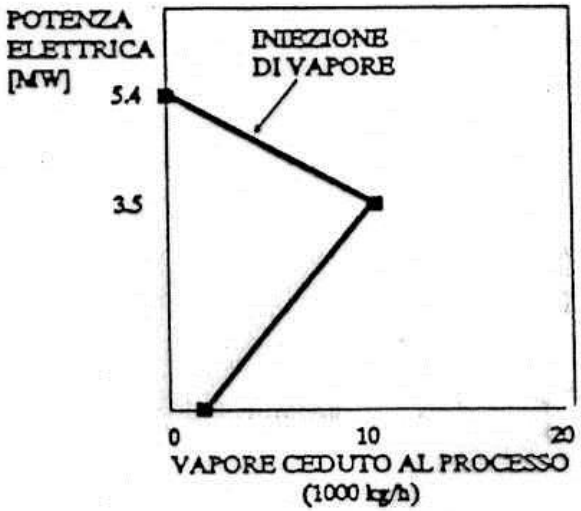


Whereas in the simple cycle the electric power is fixed together with the thermal power, unless a post combustion is provided, many possibilities are available in the *Cheng* cycle.

A fraction of the water vapor produced with the heat recovery can be sent to the turbine with a sensible increase in the electric power.
The figure represents a *Cheng* cycle that has a nominal production of 3.5 MW electricity and 10 tons of technological steam. With the steam injection into the turbine, the electric power can reach 5.4 MW without cogeneration.

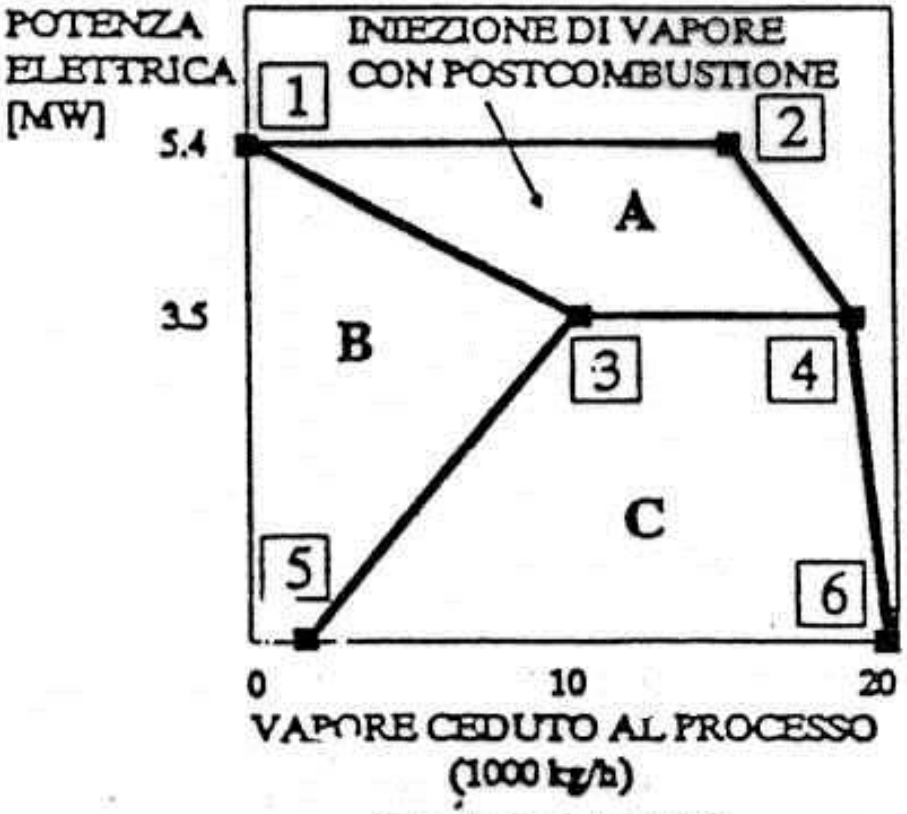


Ciclo semplice



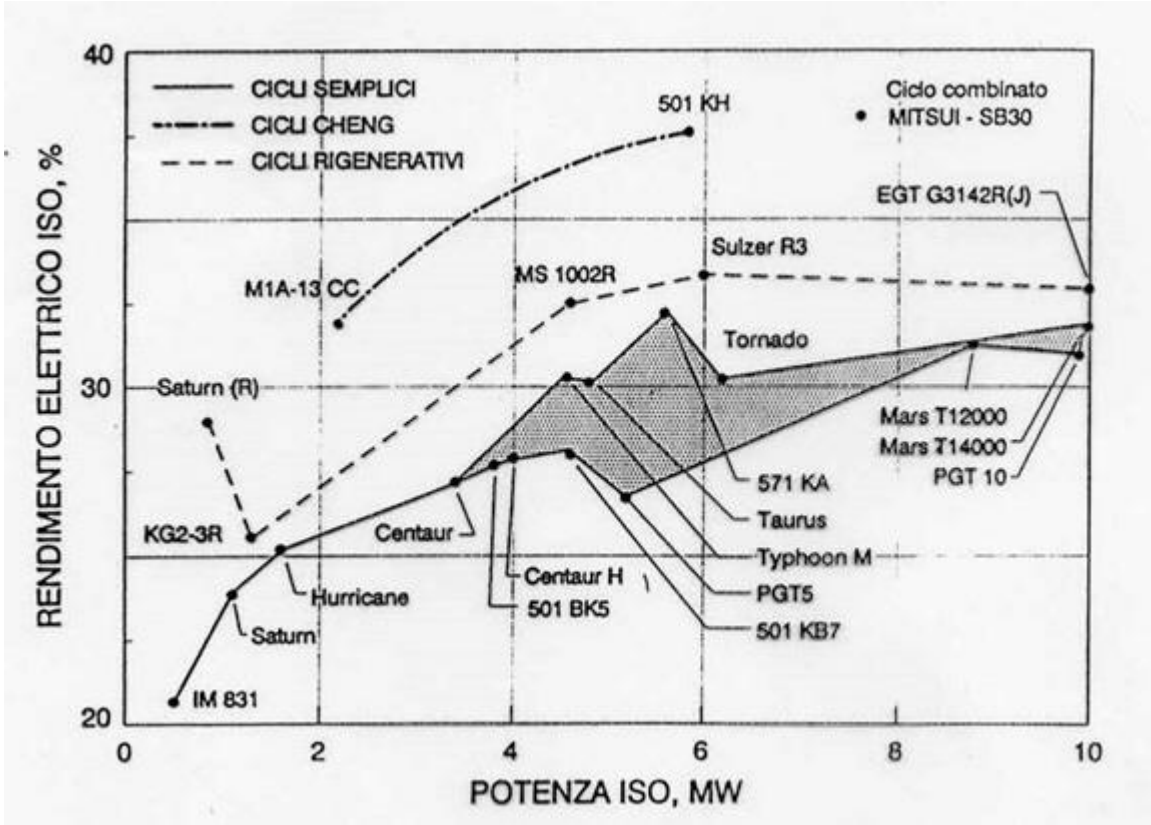
CICLO CHENG

The *Cheng* cycle allows many degrees of freedom, particularly if post combustion is also considered.
This can be greatly appreciated in the seasonal applications, for example in district heating.
For high heating demand in winter, the cycle operates in complete cogeneration without vapor injection, whereas in summer the operation could be only electric with the highest available electric power.



CICLO CHENG
condizioni operative nominali con iniezione di vapore e postcombustione

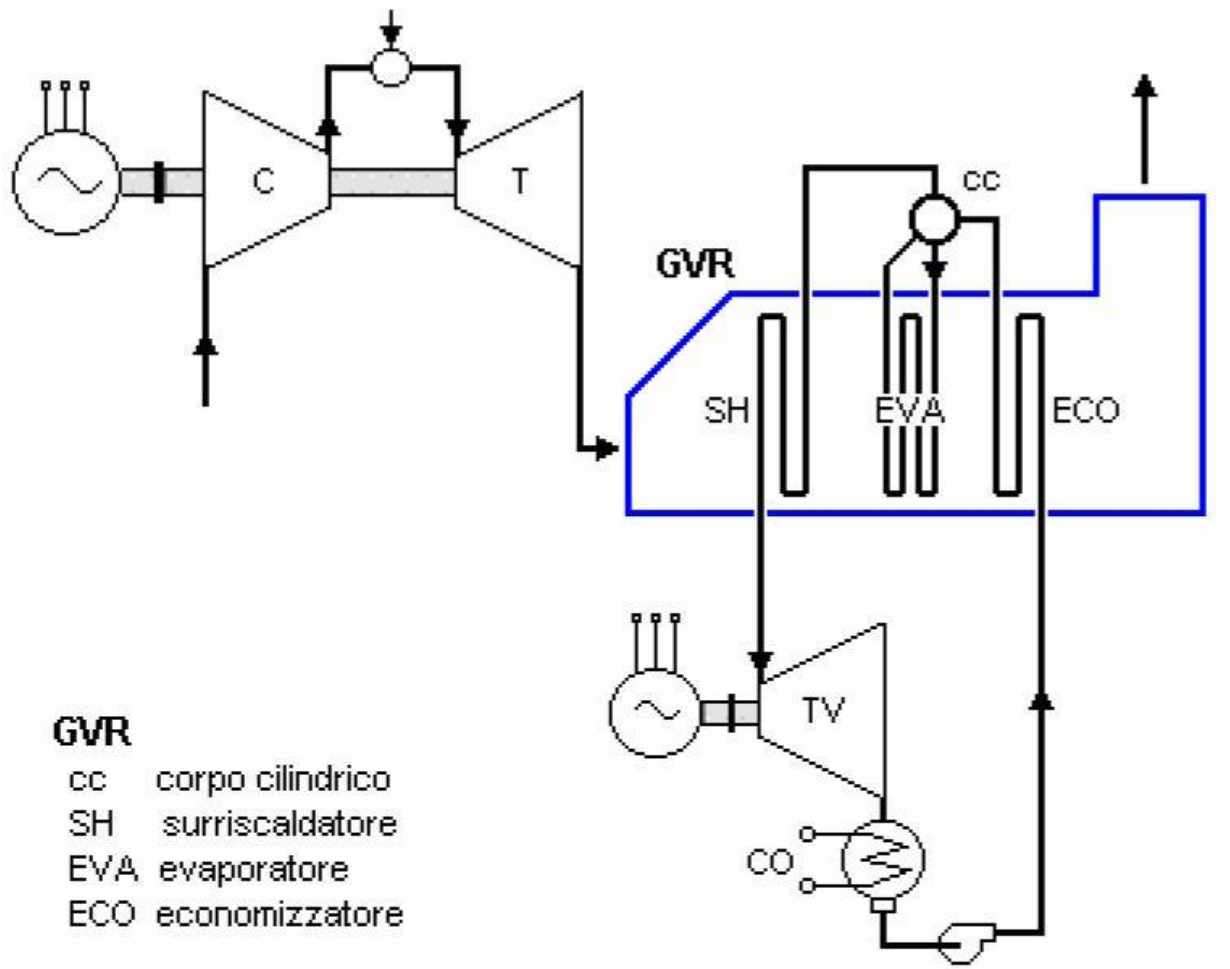
An existing plant operating with *Cheng* cycle gives 3.5 MW_e with 10 t/h of technological steam (η=0.73, η_e=0.27, η_t=0.46), or it can use the 10 t/h in combustion chamber, producing 5.4 MW_e without process heat available and an efficiency η_e=0.42.



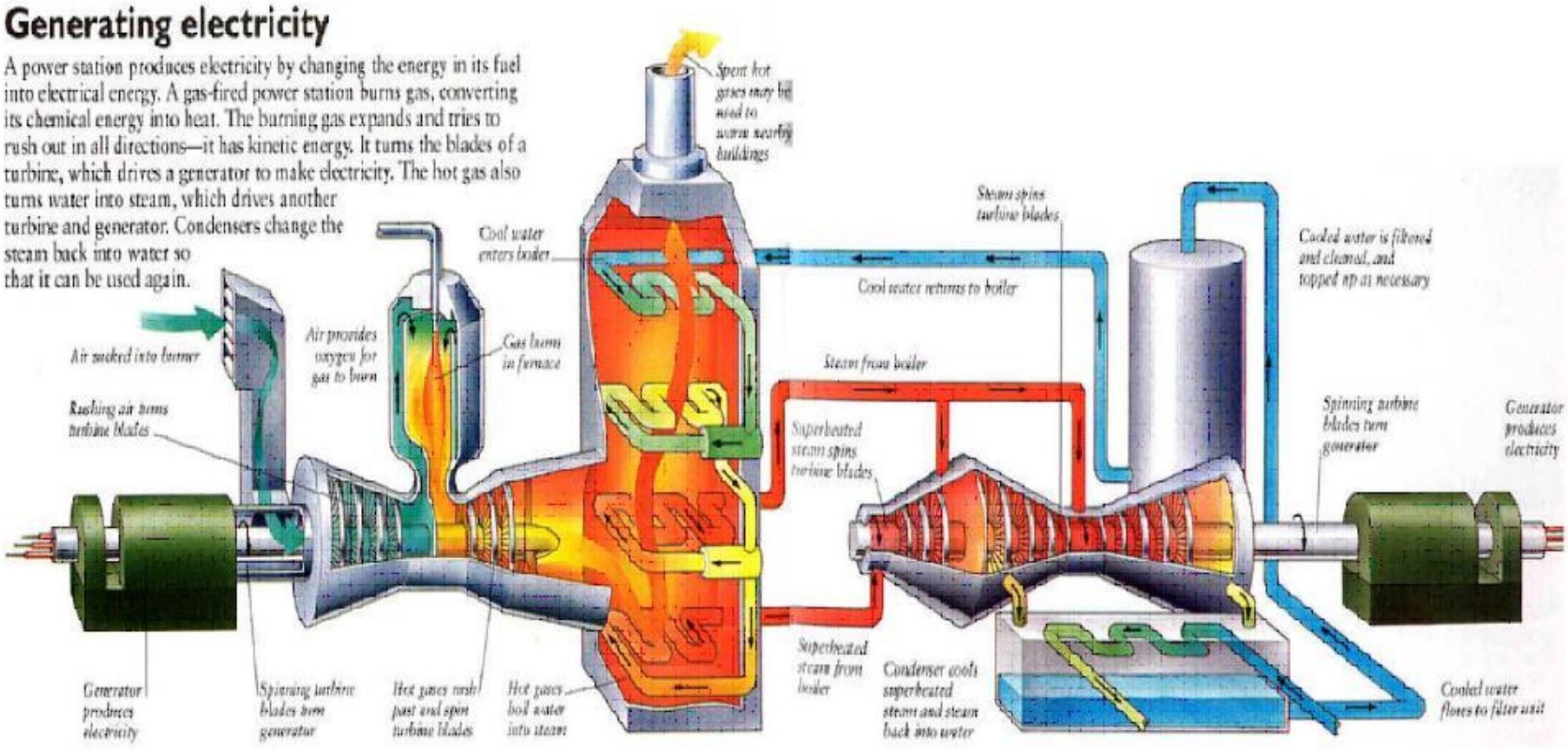
Efficiency of different models of low capacity gas turbines

COGENERATION IN PLANT OPERATING WITH COMBINED CYCLE

In the combined cycle, the heat recovery from the gas exhaust drives a water vapor turbine supplying heat to a vapor generator



If the main target of the plant is the production of electricity at very high efficiency, the combined cycle is the best choice.



In a combined cycle, the gas exhaust cooling can develop water vapor that is then overheated in a *Hirn* cycle.
The pressure in this cycle is strictly connected to the so-called *pinch point*, that is to the minimum acceptable temperature between the cooled gas exhaust and the vapor.

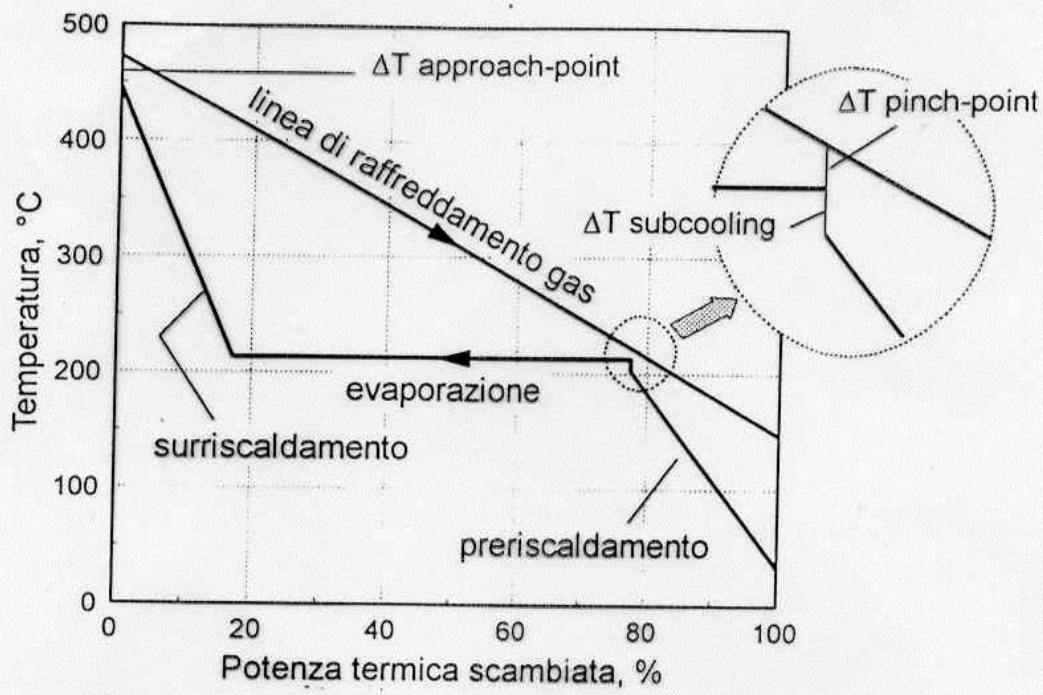
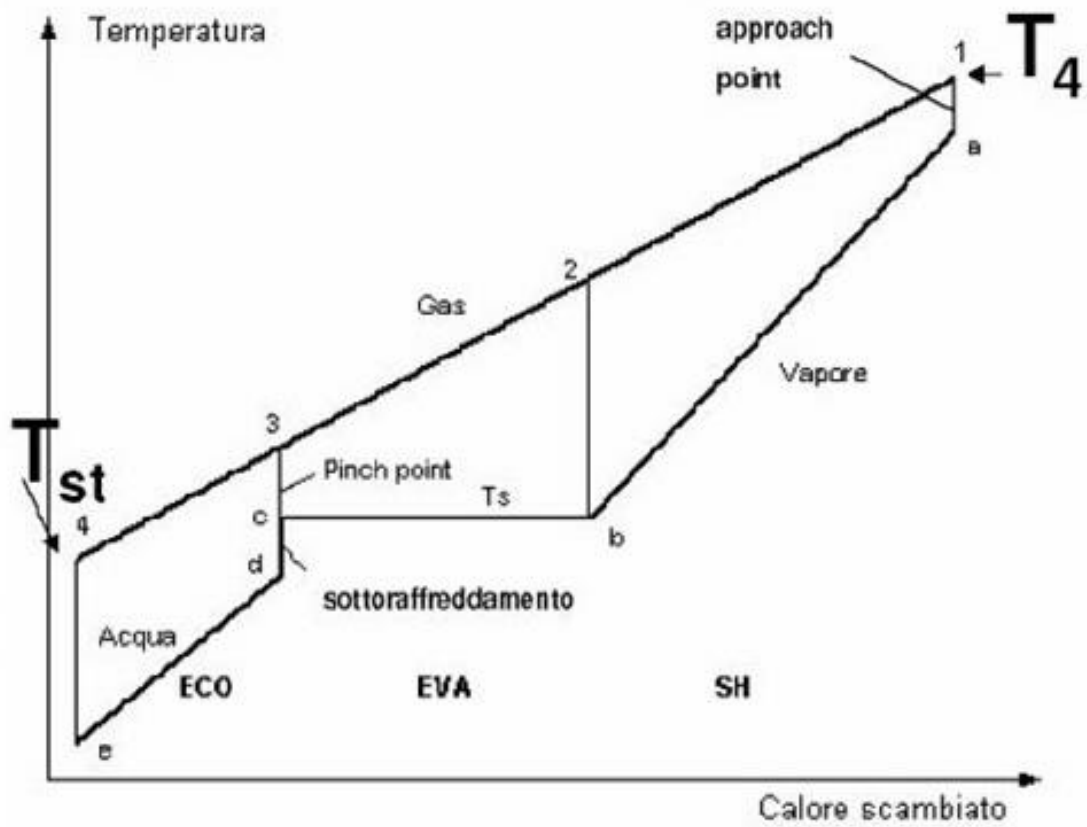


Fig. 5.19: *Diagramma temperatura-calore scambiato della caldaia a recupero di una turbina a gas.*

- The phase change liquid-vapor introduces in some parts of the boiler strong temperature differences between water and gas exhaust;
- When the water saturation temperature is reached, we find the minimum temperature difference between the gas exhaust and the water (Pinch Point); another characteristic point is the temperature difference between gas exhaust and vapor, called Approach DT_{AP};
- A small supercooling of about 5-15°C is provided to eliminate the risk of early evaporation in the tubes of the economizer.



$$\eta_{HRSG} = Q_3 / Q_2 = \frac{T_4 - T_{st}}{T_4 - T_1}$$

The cycle can be conceived with a single pressure level for the vapor looking for an optimum choice for the pressure, either at dual pressure levels with better efficiency and higher complexity.

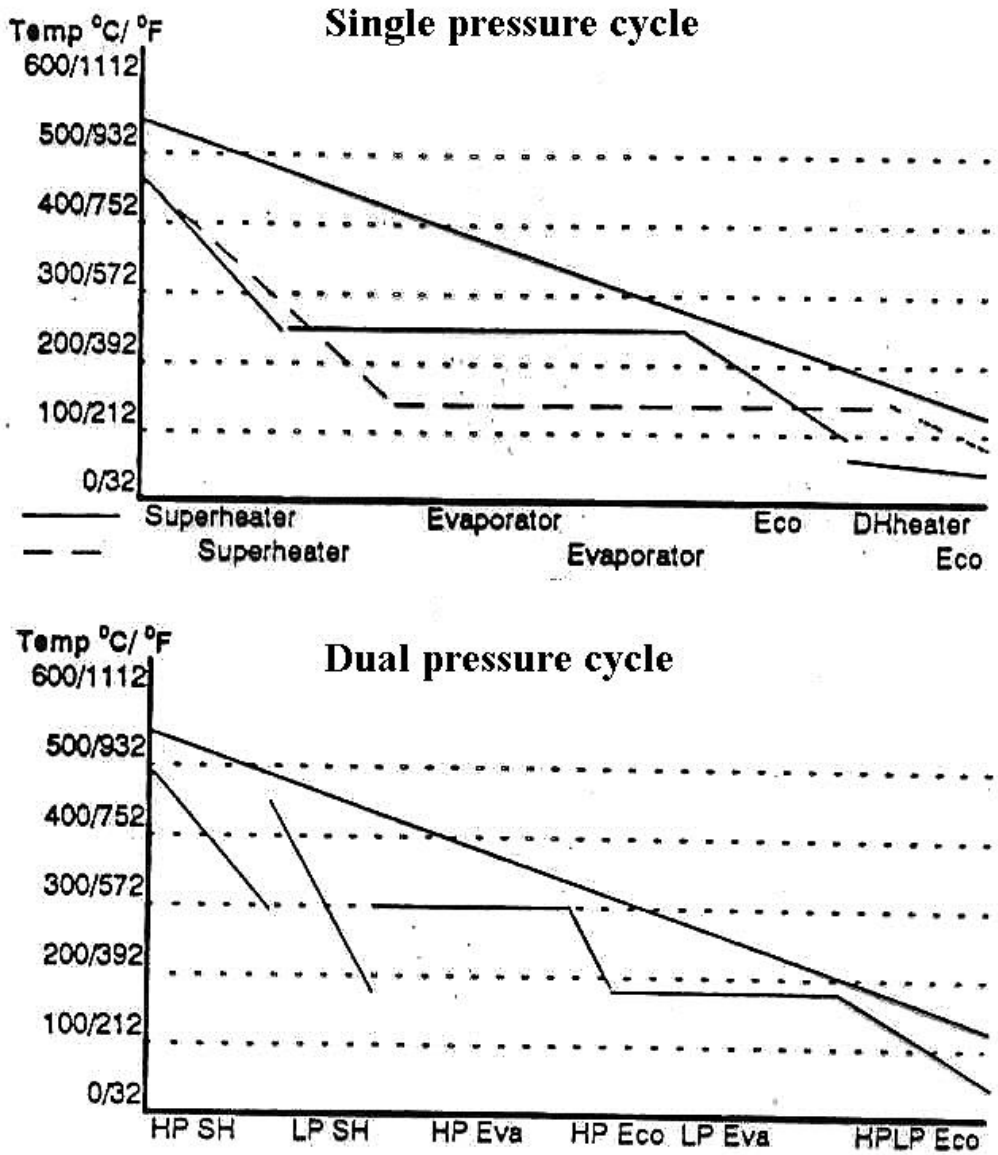
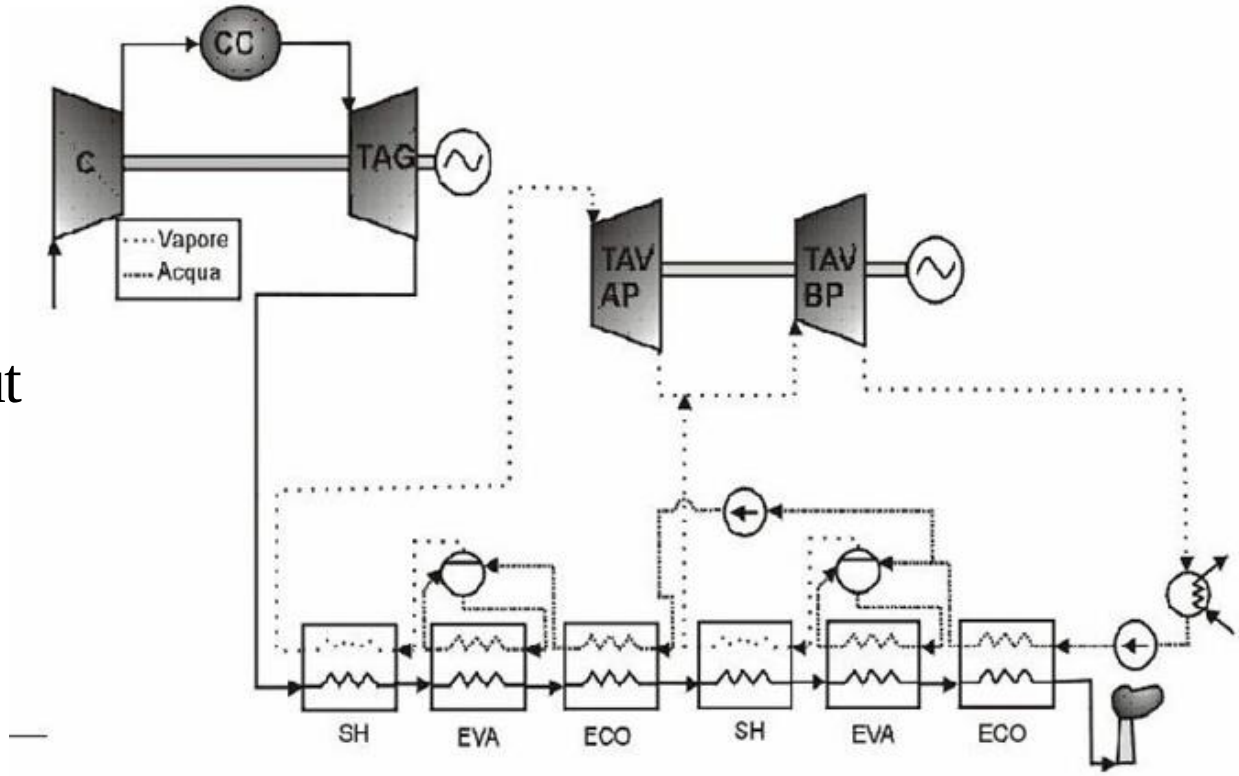


Figure 4 Temperature diagram of boiler in combined cycle.

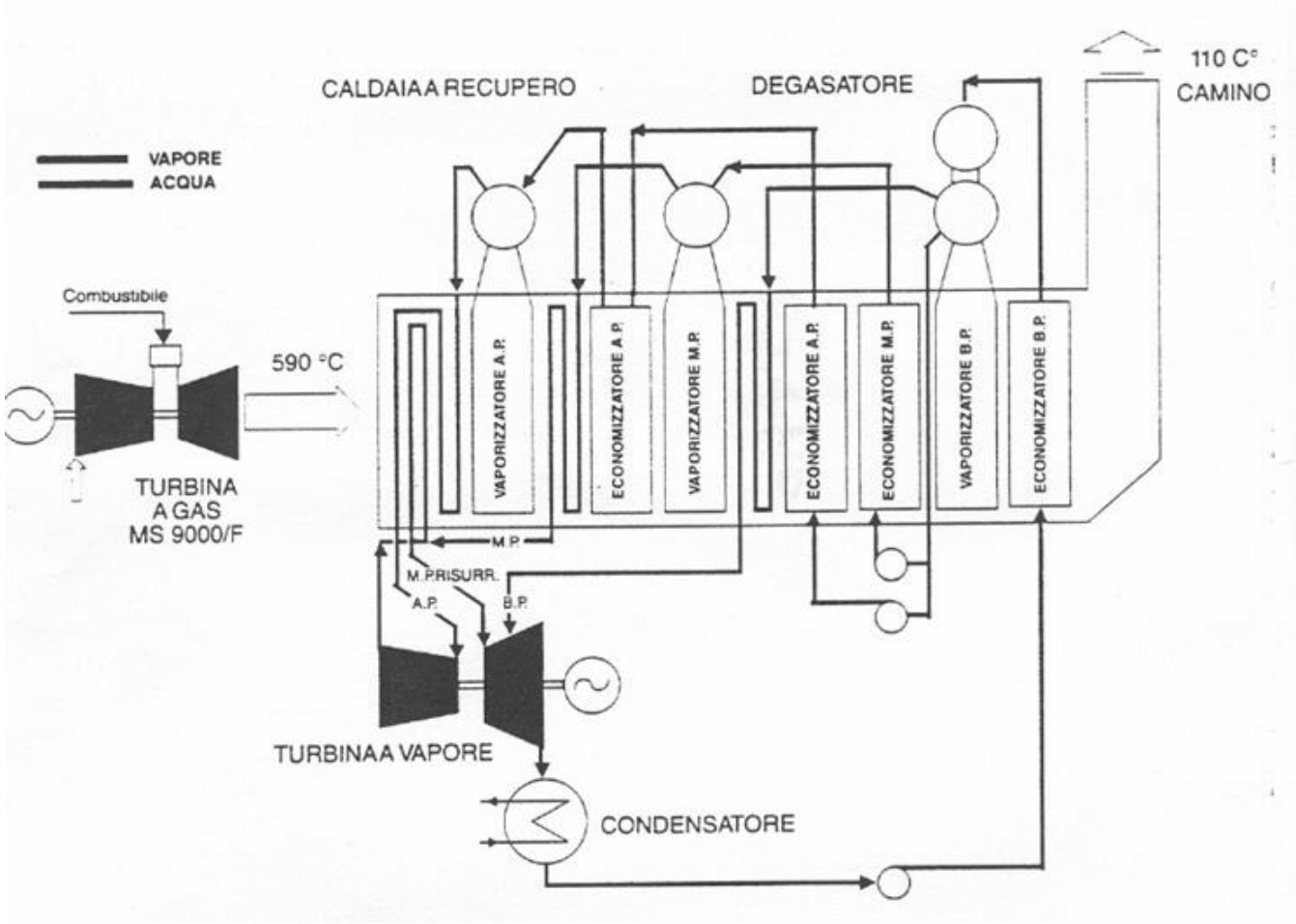
• Cycle efficiency can be improved operating at a dual pressure level.

• The water enters the recovery boiler at two distinct pressure levels, one for the low-pressure circuit (5-10 bar) and the other for the high-pressure circuit (70-100 bar). The produced vapor is sent to distinct sections of the turbine. To the low-pressure turbine arrives also the vapor flow from the high-pressure turbine.

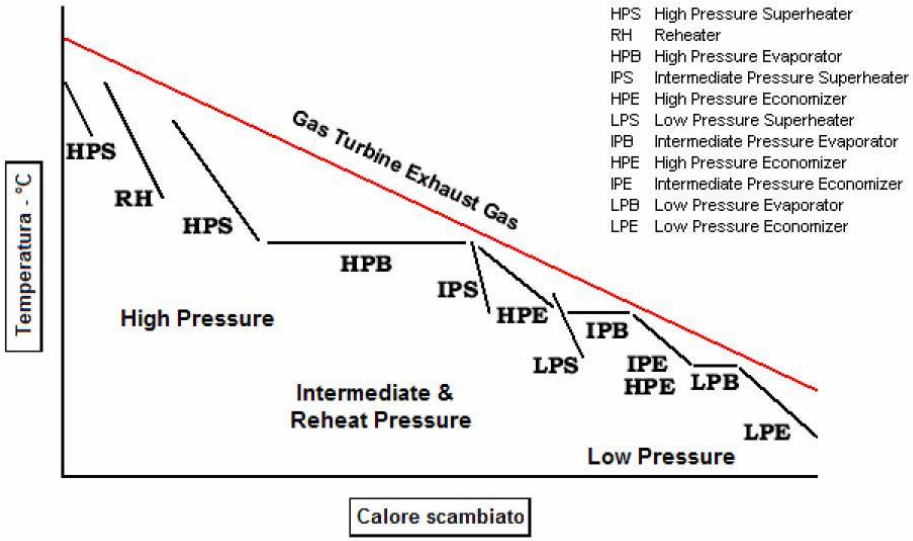
The plant is more expensive as the recovery boiler is more complex, but higher efficiency is possible as the heat recovery is now better.

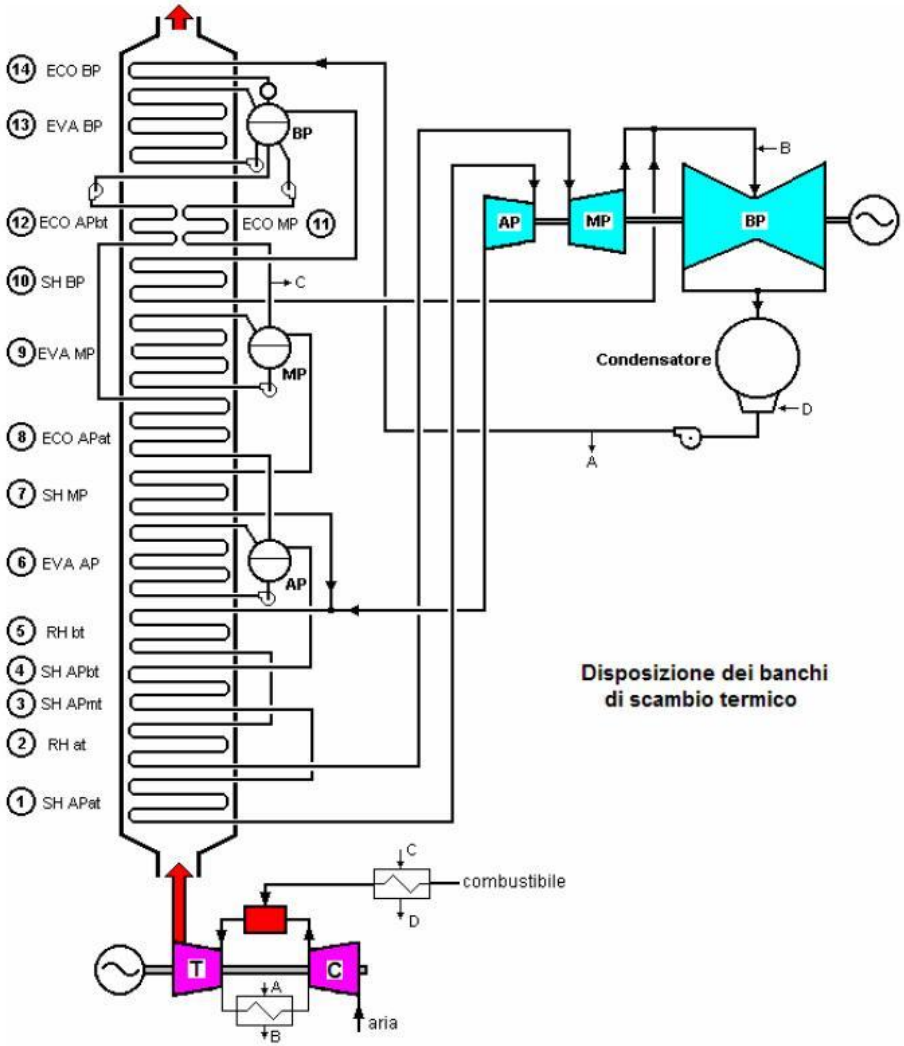


Example of a cycle at three pressure levels



A three-pressure level cycle is here represented.
It allows a very high electric efficiency that can even exceed 58%

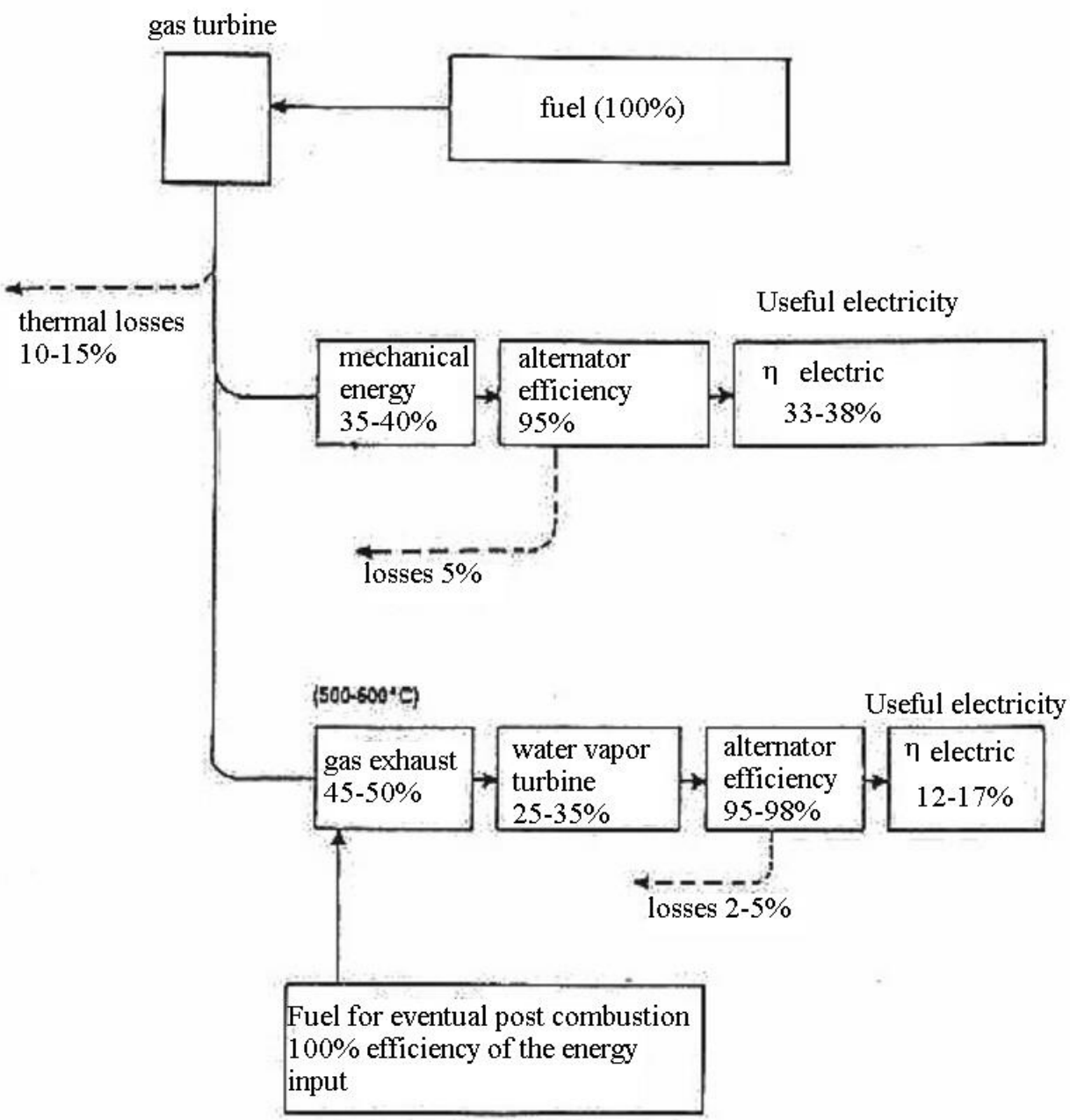




Data regarding combined cycle systems manufactured by Ansaldo Energia and a picture of a high capacity vapor power turbine.

COBRA PLANT PERFORMANCE				
Configuration	Gas Turbine (MW)	Steam Turbine (MW)	Plant Output (MW)	Plant Efficiency (%)
Cobra 1.94.3A	250	130	375	57.5
Cobra 2.94.3A	500	260	750	57.5
Cobra 1.64.3A	67	37	102	54.0
Cobra 2.64.3A	134	77	207	55.9
Cobra 1.94.2	154	89	238	52.3
Cobra 2.94.2	308	178	478	52.5

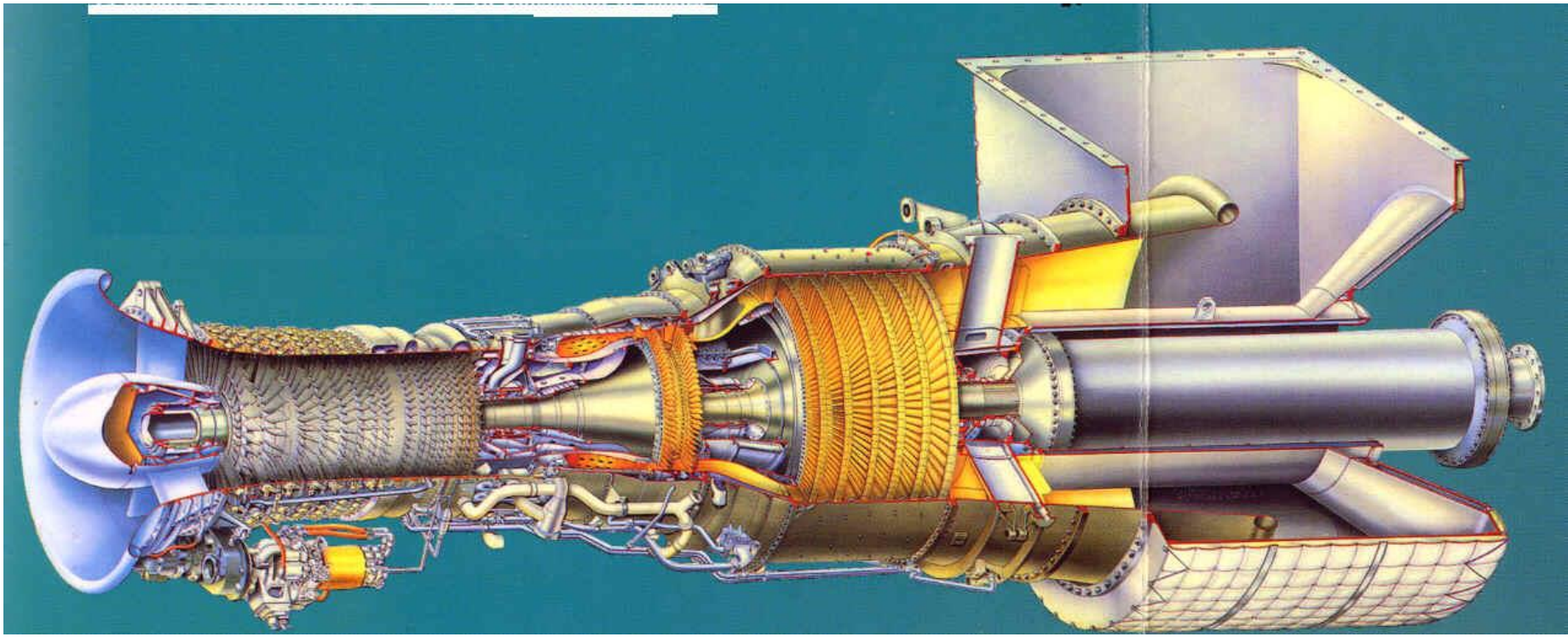
The scheme represents the possible behavior of a combined cycle system of medium capacity.



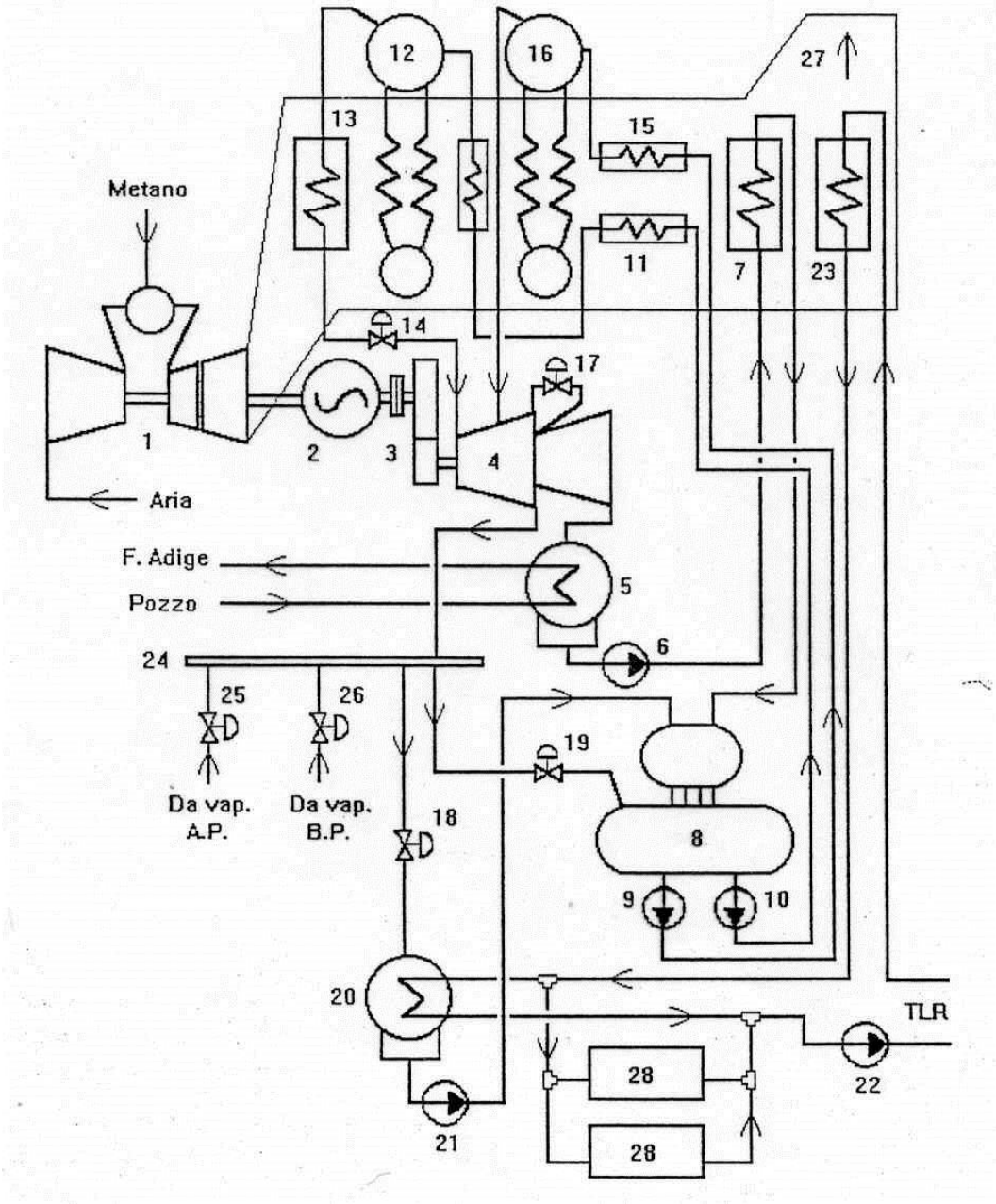
The cogeneration power station of Borgo Trento (AGSM Verona, Italy) started in 1994 and operates according a combined cycle. The heart of the system is an aeroderivative turbine realized by Fiat Avio on a license of General Electric (this model of turbine equipped the aircraft DC 10).

The mechanical power of the turbine is 21 MW. The turbine is dual shaft. A first stage of the turbine drives the compressor at a speed of 10,000 rpm. The second stage gives useful work and rotates at a speed of 3,000 rpm.

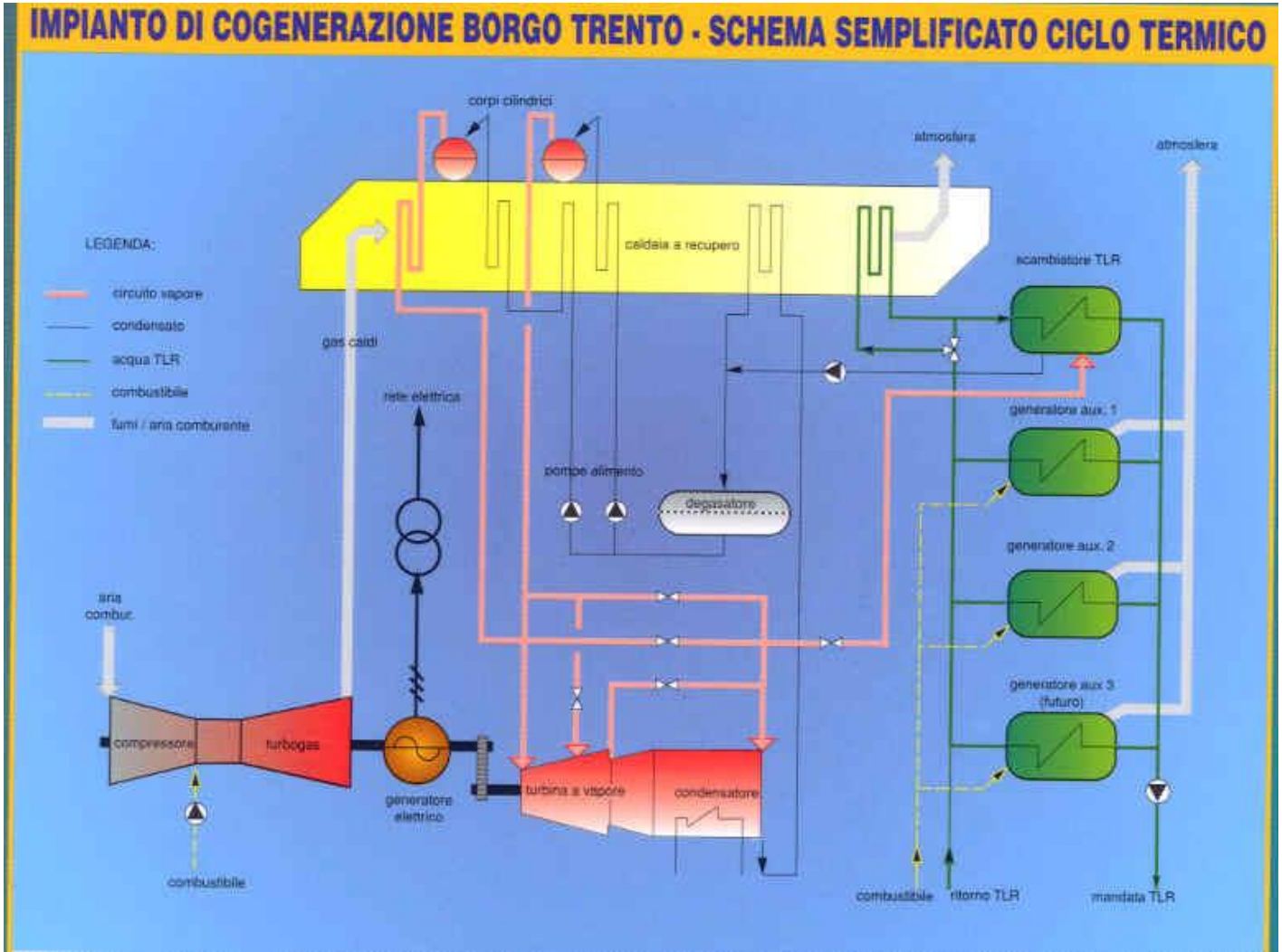
A water injection system is provided to control the NO_x. This system caused a lot of problems in operation.



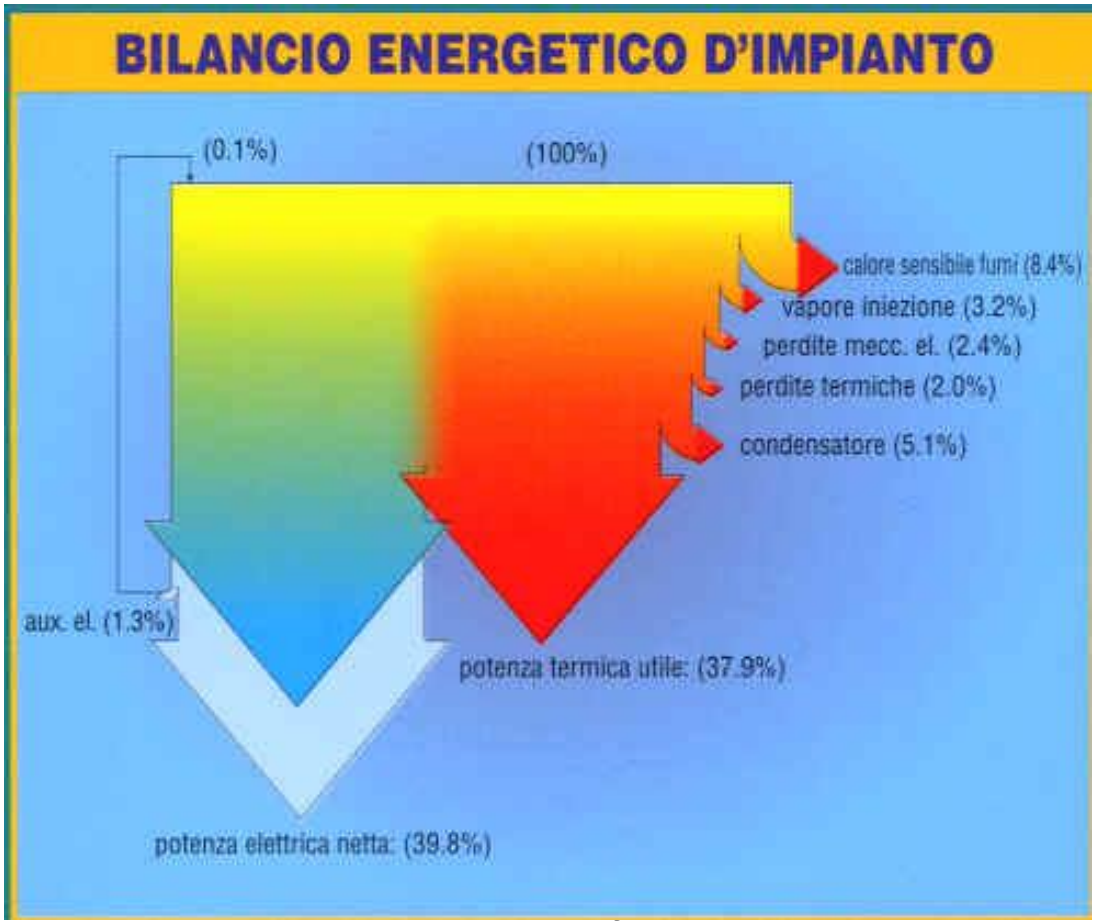
The scheme represents the most important parts of the system, comprising the district heating plant that operates at pressurized water at a temperature of 120°C with a return at 60°C.
When the demand is at its maximum, the plant supplies more than 700 m^3/h of hot water.



At the outlet of the turbine, the exhaust at 500°C supplies heat to the recovery boiler that produces vapor at two pressure levels.
The extraction turbine gives 9 MWe when operating without pass out (5,600 rpm).



Operating in cogeneration, the electric power is 25 MW with a thermal power for district heating of 24 MW. When the vapor turbine does not work, the thermal power can exceed 30 MW.



Then the plant can work according to a nominal operation or a max electricity production disposition or a maximum thermal production disposition.

PRESTAZIONI DELL'IMPIANTO (dati di collaudo)

	NE	NC
Potenza elettrica ai morsetti macchina (MW)	28,6	24,8
Potenza termica per teleriscaldamento (MW)		23,6
Rendimento elettrico (%)	45,9	39,8
Rendimento termico (%)		37,9
Indice energetico (CIP 6/92)	47,5	71,5

Nota:
NE = assetto di massima produzione elettrica
NC = configurazione di cogenerazione nominale

DATI DI IMPIANTO

Producibilità elettrica (MWh/anno)	200.000
Temperatura acqua tele mandata/ritorno (°C)	120/60
Potenza termica massima impianto	64.000

In maximum thermal production disposition η_e=0.34 and η_t=0.48.

The main plant data are here listed. When in winter, the whole heat recovery from the gas exhaust is not enough, auxiliary traditional gas boiler are installed with a nominal power of 34 MW_t.

CARATTERISTICHE MACCHINARI

Turbogas	
Costruttore	Fiat Avio - General Electric
Tipo	LM 2500
Potenza elettrica	21.000 kW
Numero stadi compressore	16
Numero stadi turbina a.p.	2
Numero stadi turbina b.p.	6
Turbina a vapore	
Costruttore	Ansaldo
Potenza elettrica	8.650 kW
Velocità di rotazione	5.600 giri/min.
Generatore elettrico	
Costruttore	Ansaldo
Potenza	35.000 kVA
Velocità di rotazione	3.000 giri/min.
Tensione	11.000 V
Caldaia a recupero	
Costruttore	Rhuts
Produzione di vapore	37.500 kg/h
Portata fumi	210.000 Smc/h
Temperatura fumi in ingresso	530 °C
Temperatura vapore a.p.	450 °C
Temperatura fumi in uscita	90 °C
Altezza camino	46 m
Caldaie ausiliarie	
Costruttori	IVAR - ICI
Numero generatori	3
Tipo	a tubi di fumo
Potenza complessiva	34.000 kW

The graph illustrates how combined cycles have largely overcome the efficiency even of the most advanced vapor power cycles.

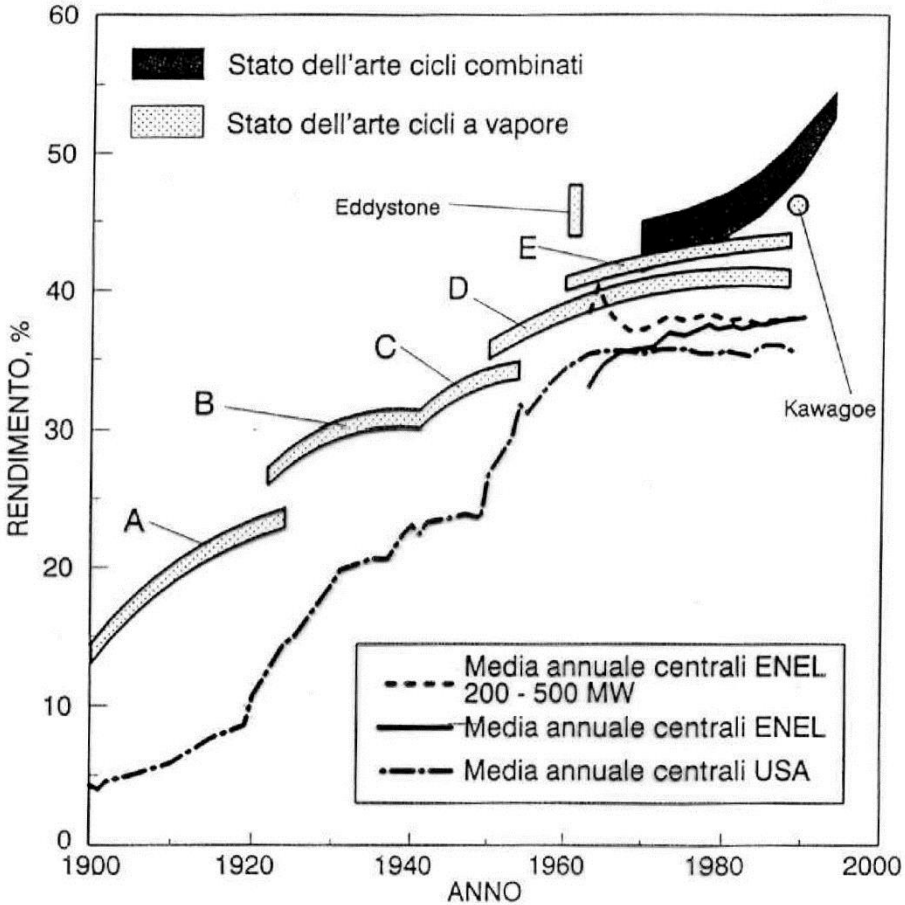


Fig. 6.1: *Evoluzione del rendimento dei cicli a vapore e dei cicli combinati a partire dall'inizio del secolo. Le lettere A-E indicano i vari stadi evolutivi della tecnologia a vapore: A=cicli saturi; B=cicli rigenerativi; C=cicli rigenerativi con surriscaldamento; D=cicli con risurriscaldamento; E=cicli ipercritici con doppio risurriscaldamento. Il rendimento medio annuale degli impianti commerciali è inferiore allo stato dell' arte, poichè riflette l' obsolescenza degli impianti e il funzionamento a carico parziale. La minore efficienza delle centrali USA rispetto a quelle ENEL è in parte da attribuirsi a un uso più massiccio delle torri evaporative. Le prestazioni dei cicli a vapore più spinti sono praticamente ferme da oltre trent'anni, come testimonia il confronto tra la centrale USA di Eddystone e quella giapponese di Kawagoe. Entrambi questi impianti sono ipercritici con doppio risurriscaldamento, una soluzione che nonostante gli alti rendimenti è tuttora poco diffusa per problemi di costo, affidabilità e resistenza dei materiali. La centrale di Eddystone è indicata da una fascia poichè il rendimento previsto di quasi il 50% in realtà non fu mai raggiunto, ed i dati sugli effettivi rendimenti netti di esercizio sono di difficile reperimento.*

This is a summary table that compares the different cogeneration technologies (the more stars, the better the technology)

	Motori		Turbine a gas	Turbine a vapore		Cicli Combinati
	rec. totale	rec. solo gas		contro-pressione	spilla-mento	
electric efficiency	****		**	**	***	****
thermal efficiency	***	**	***	****	***	***
investment cost	****		****	***	*	**
high temperature heat	**	***	****	*	**	**
time of building	****		****	**	*	**
necessary area	***		****	**	*	***
gas emissions	**		***	***		****
noise/vibration	*		**	***		**
flexibility	***		**	*	****	****
reliability	**		****	***		****
maitenance and personnel	*		****	***	**	***
useful life	**		***	****		***
starting time	****		****	**	*	***
water requirement	***		****	*		**
partial load behavior	***		**	**	***	****
sensitivity to ambient conditions	***		*	****	***	***

Tab. 2.1: *Confronto qualitativo tra le principali caratteristiche di impianti di cogenerazione alimentati a gas naturale. Maggiore il numero di asterischi, migliori le prospettive per la cogenerazione.*

MICRO COGENERATION

Micro-cogeneration considers capacities from few kW to about 1 hundred kW.
It can be applied to small buildings such as single-family homes or small hotels.

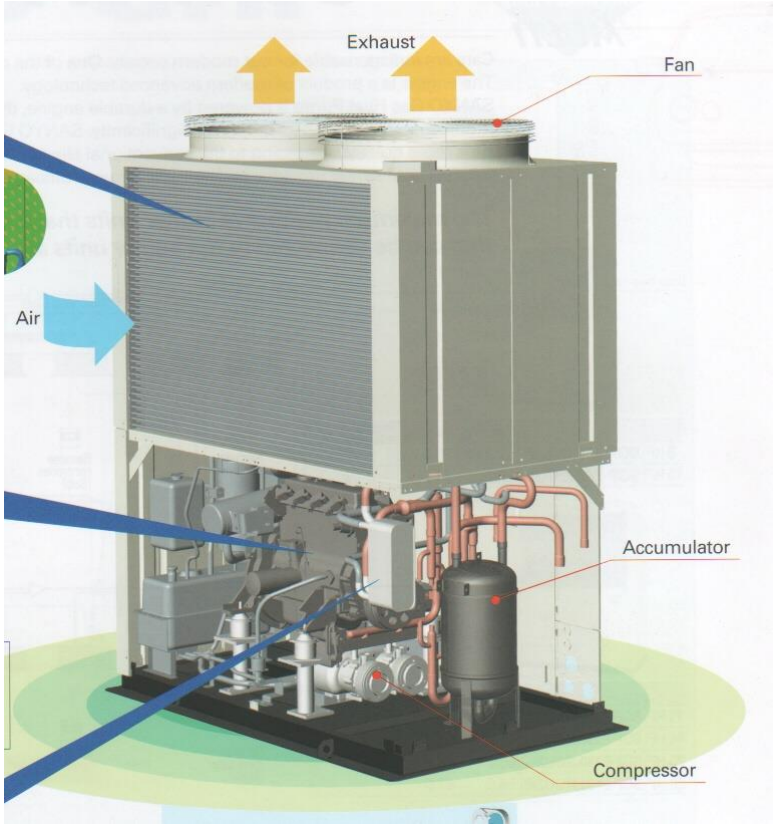
The most widespread technological system in micro cogeneration is actually the reciprocating engine operating in Otto or Diesel cycle with the following typical performance:

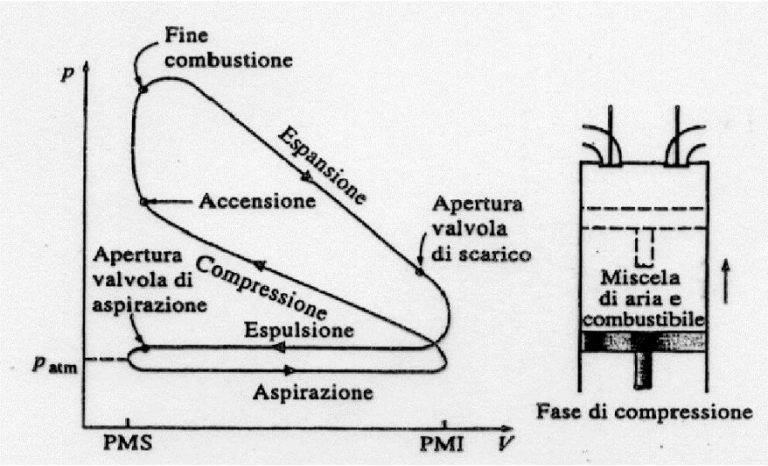
- electric power 3-15 kW;
- η_e minimum 20%;
- η_t minimum 60 al 75%;
- ordinary maintenance every 3-4000 h;
- complete revision after 30,000 h;
- noise level 50-60 dB(A) at 1 m.

These systems allow an effective energy saving only if the available thermal energy is completely utilized.

A Japanese manufacturer realized a heat pump driven by a small capacity reciprocating engine with three or four cylinders, displacement less than 1 liter, mechanical power 7-8 kW and claimed maintenance intervales of 10,000 hours (lubricating oil change comprised!). The heat pump thermal capacity was about 30 kW.

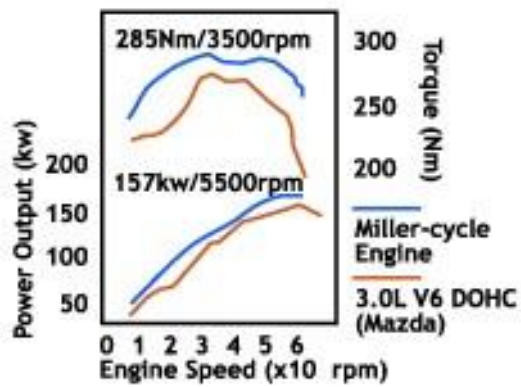
A further novelty was the operation according *Miller* cycle. *Miller* cycle is operated on supercharged motors and it is similar to *Otto* cycle, except for the suction phase that is completed at the Lower Dead Point (LDP) for the *Otto* cycle whereas it goes on after LDP in *Miller*.



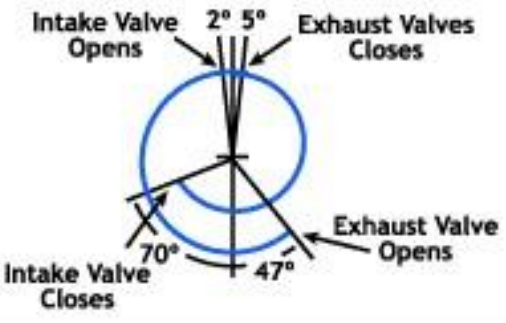


Then the real compression phase is delayed as the suction valve is still open so that a part of the mixture charge goes back. The charge reduction (and the reduction of volumetric efficiency) is limited by the supercharge due to a volumetric compressor. In return, the compression work of the piston is reduced, particularly severe when the piston rises from the LDP. The specific power reduction is modest, whereas motor operations are more regular with less pollution, particularly of NO_x and unburned.

The combustion is improved also because the charge is at a higher temperature than a charge provided by a turbo compressor with intercooler. The work in the phase of compression is lower of about 10-15% with an efficiency increment of some percentage.

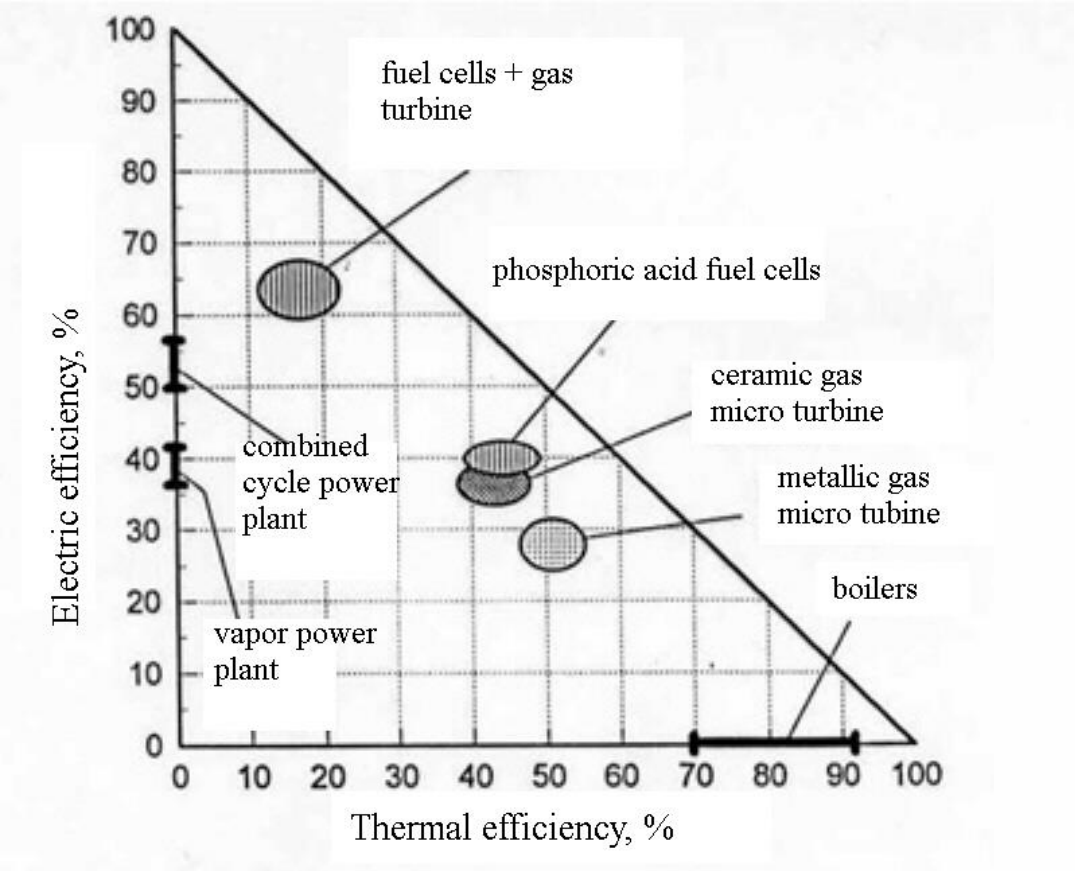


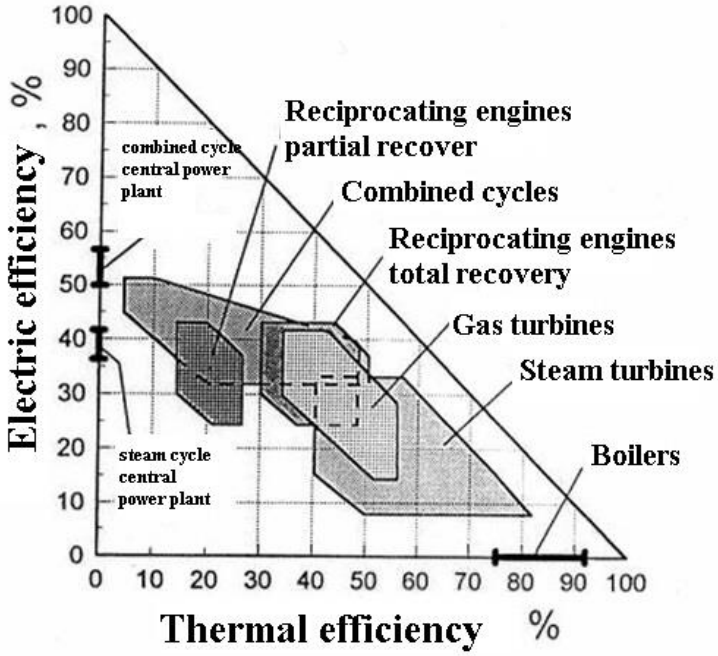
Engine Output, Power Performance



Valve Timing

Today's research on micro cogeneration involves micro turbines, fuel cells and *Stirling* cycle engines. The figure illustrates the position of these machines on the thermal/electric efficiency triangle.
The smaller figure reminds the comparison previously presented of the conventional cogeneration systems.
Of the listed technologies, only micro turbine have an important commercial diffusion.





GAS MICRO TURBINES

The most widespread capacity of a gas turbine is higher than 1 MW.
Micro turbines instead usually have capacities lower than 100 kW. They are quite innovative machines. In fact such a low capacity imposes a completely new architecture for the machine.
Whereas lower capacity traditional gas turbines are conceived at a scale down of higher capacity equipment, (they work in a simple cycle with axial compressor and turbine), micro turbines work on a regenerative cycle with radial compressor and turbine.

The fundamental components are:

- compressor;
- regenerator;
- burner;
- turbine;
- static frequency converter.

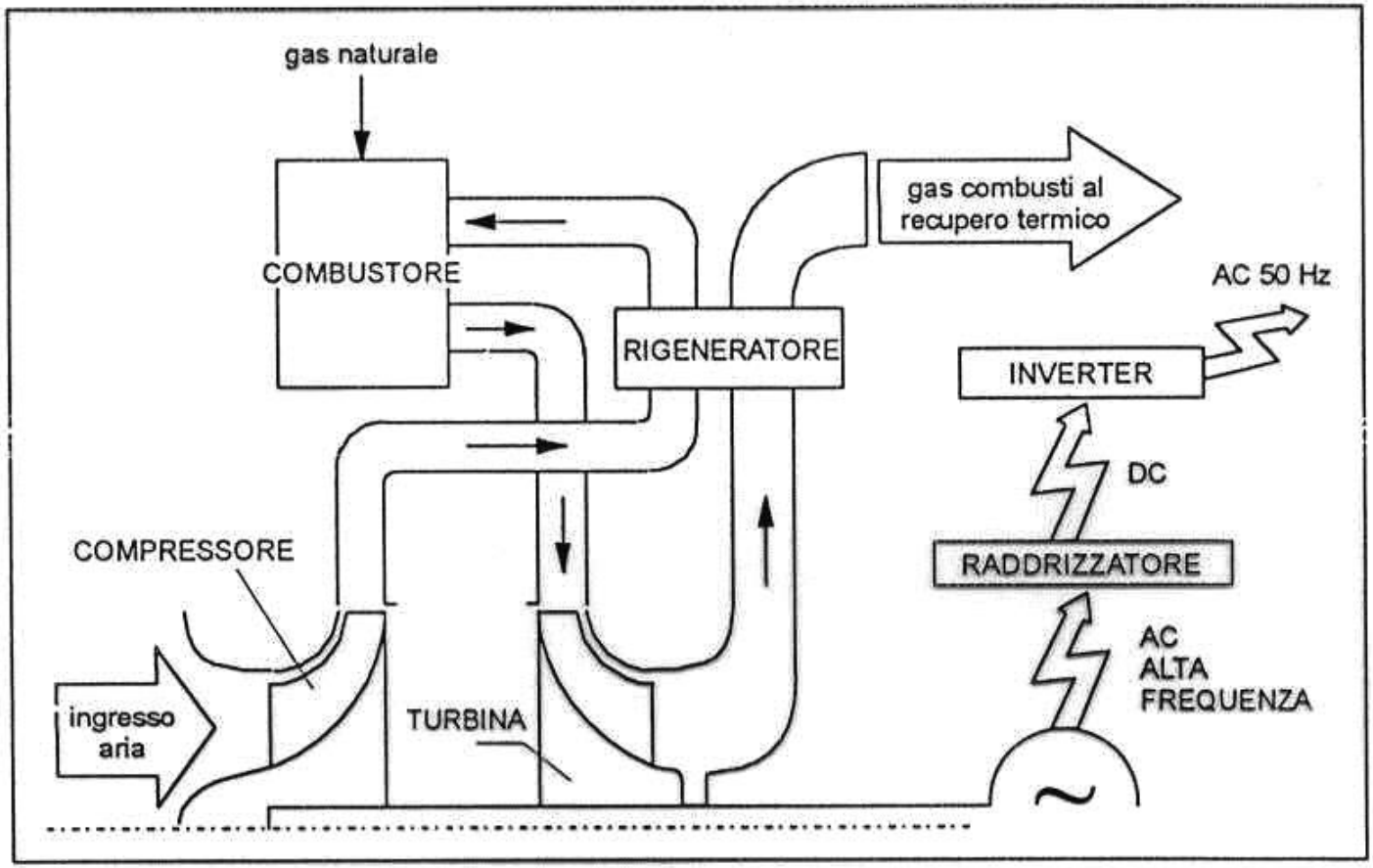


Fig.2.1: *Schema concettuale dell'architettura generalmente adottata per le microturbine a gas, che ne mostra i componenti fondamentali*

Turbocompressor

Two main features:

• they operate at a very high rotational speed around 100,000 rpm;

• both compressor and turbine are radial.

The very high rotational speed is due to the very low gas exhaust flow rate (due of course to the small capacity).

The specific work of a gas turbine (*Brayton-Joule* cycle) is around 150-250 kJ/kg.

To develop one hundred kW, flow rate of 0.5 kg/s is enough.

Small sizes correspond to small flow rate, as the velocity inside the turbine is imposed by the pressure drop and it is independent from the size.

Then the peripherical velocity does not change very much with the turbine size.
This velocity is ωr so that for small r values the angular velocity ω must be very high.

A more precise justification derives from the consideration of the specific rotational speed N_s , an adimensional group conceived in the similitude theory of turbomachinery:

$$N_s = \omega \frac{\sqrt{V}}{\Delta h_{is}^{3/4}}$$

V is the volumetric flow rate and Δh_{is} the enthalpy drop.
The best efficiency values are obtained for N_s between 0.3 and 0.6.

As the denominator of the ratio is not so different for conventional and micro turbines, if a turbine has 100 MW power capacity, a suitable rotational speed can be 3,000 rpm. Instead, if the turbine has 100 kW power capacity (1000 times lower), a possible rotational speed can be evaluated in:

$$3.000\sqrt{1000} = 100.000 \text{ rpm}$$

Moreover, in order to obtain the maximum enthalpy drop from a single stage, the peripherical velocity u should be at the highest values allowed by the material resistance. The stress on the material is proportional to u^2. For this reason, the rotational speed is inversely proportional to the diameter of a single stage for a given material.

A 100 MW turbine can have a 2 m diameter. Then a 100 kW turbine realized as a simple scale down should have a diameter of:

$$2000 \times \frac{3,000}{100,000} = 60 \text{ mm}$$

Such a small diameter cannot be proposed. Consequently, instead of multistage axial compressors, a single stage radial compressor is adopted with a pressure ratio from 4 to 6. This pressure ratio is low with respect to the actual turbine technology, nonetheless it can realize equally efficient cycles.

These turbocompressors look like the system employed in the supercharged i.c. engines.

The regenerator

In a simple cycle, the gas exhaust temperature is high for low pressure ratio, whereas the inlet temperature into the combustion chamber is low (that is after the compressor).

To give an idea, with ρ=4 the gas exhaust is at 710°C, while the compressed air enter the combustion chamber at 184°C.

The correspondent efficiency is then very low (16%).

Therefore, micro turbines operate according to a regenerative cycle.

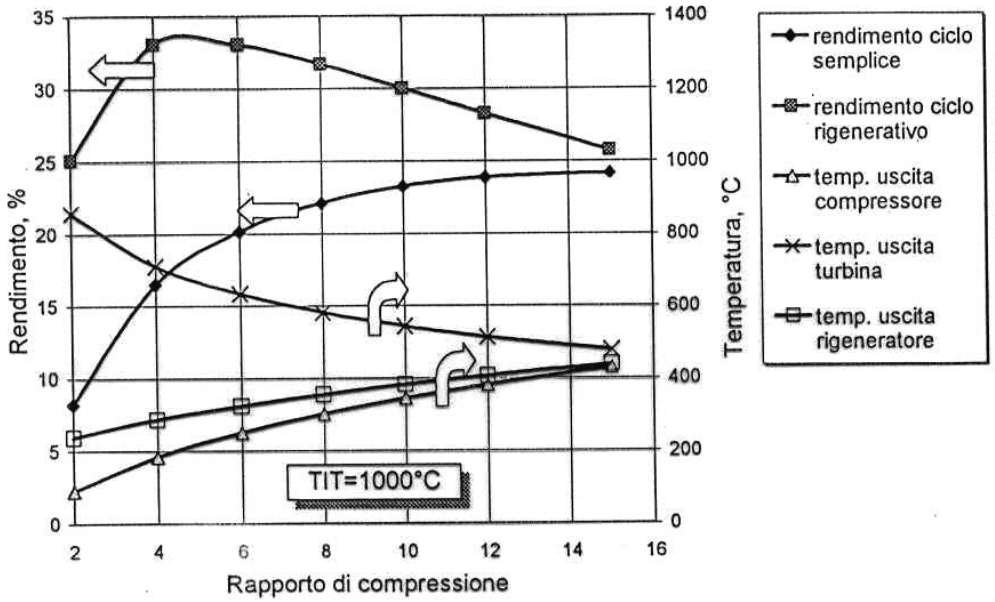


Fig.2.2: *Confronto tra cicli semplici e rigenerativi per piccole turbine con temperatura massima di 1000°C.*

Two different regenerator technologies are used:

- a heat exchanging surface (air to air without contact);
- a rotary matrix heat exchanger.

The burner

It is similar to the burner used in the traditional gas turbines. The small size gives rise to a cooler flame due to the radiation heat exchange with the chamber walls. Consequently, the NO_x are reduced, and they can remain within10-15 ppm.

Frequency conversion system

To produce a.c. electricity at 50 Hz when the shaft rotates at 100,000 rpm, a mechanical gearbox reducer would be necessary with very high costs and power absorption (besides noise and reliability).

Instead, electricity at high frequency is produced (1500 Hz) that is converted statically at 50 Hz by a current rectifier that produces c.c. electricity finally converted to the required frequency by an inverter.

This system has the great advantage of maintaining the final frequency constant independently from the rotational speed of the shaft. Then the shaft rotational speed can vary at partial loads with advantages in an easy control and operational flexibility.

Performance analysis

Main data regard pressure ratio, turbine inlet temperature, isentropic efficiency in compression and expansion, overall efficiency η.
Main data of different machines are listed below:

Caso	A	B	C	D	E	F	G	H
Potenza, kW	50	50	100	200	300	140	300	300
η %	15,4	26,2	40	30	42	30	42	42
T_{max} (°C)	827	1010	1350	1200	1350	1200	1350	1350
ρ	4	3,3	4,96	7,3	7,3	5,9	8	4,5
η_{ad} compr.	0,80	0,77	0,78	0,79	0,815	0,78	0,82	0,84
η_{ad} exp	0,84	0,88	0,855	0,83	0,875	0,822	0,855	0,838

Some evaluations are proposed here as a function of the pressure ratio for different turbine inlet temperatures.
Temperatures as high as 900-1000°C are obtained with Ni alloys (for example Inconel). Higher values can be reached with ceramic materials.
An optimum value of the pressure ratio ρ exists: lower values require a too high charge on the regenerator, increasing the losses. Higher values give intrinsically lower efficiencies (see slide 96).

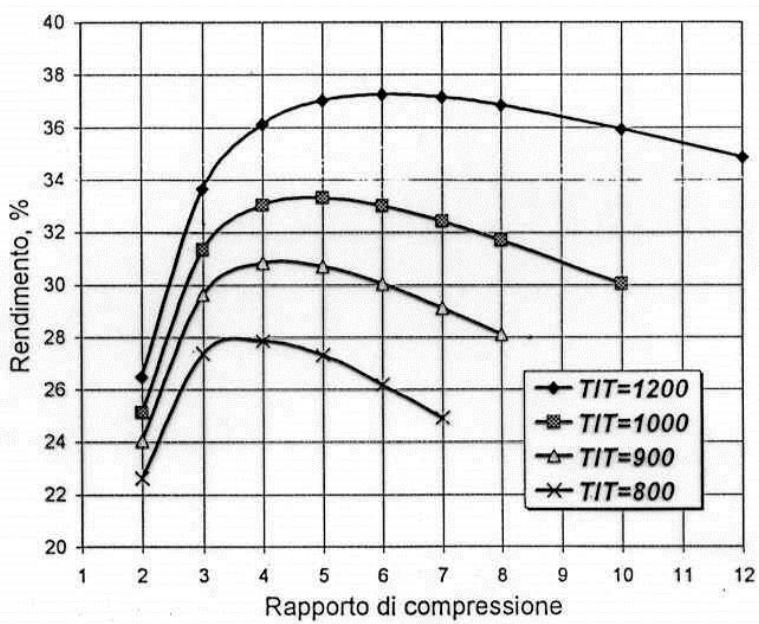


Fig.2.3: *Rendimento di cicli rigenerativi di turbina a gas, calcolati cono le assunzioni dell'ultima colonna di tab.2.1. TIT [°C] è la massima temperatura del ciclo.*

Obviously, the specific work increases with the turbine inlet temperature and with the pressure ratio.

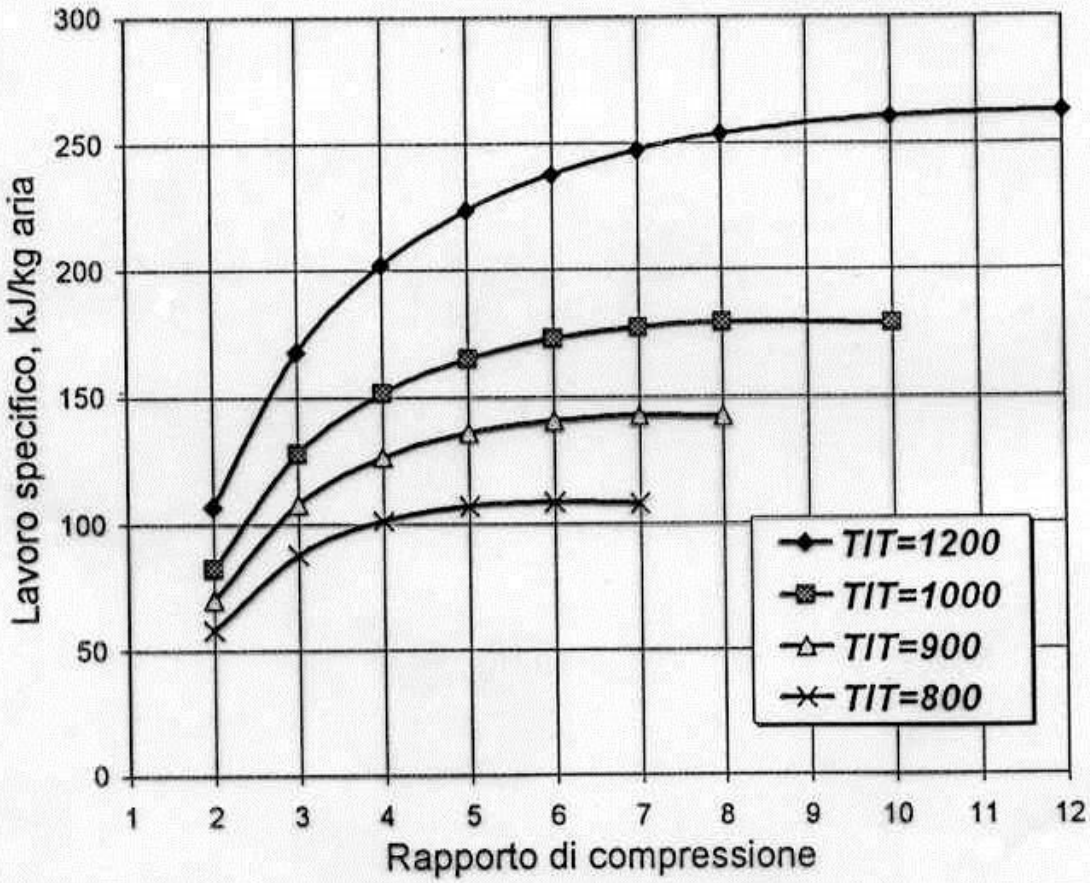


Fig.2.4: *Lavoro specifico di cicli rigenerativi di turbina a gas, calcolati secondo le assunzioni dell'ultima colonna di tab.2.1. TIT in °C.*

Consider now how pressure ratio and turbine inlet temperature influence the electric/thermal ratio of the system.

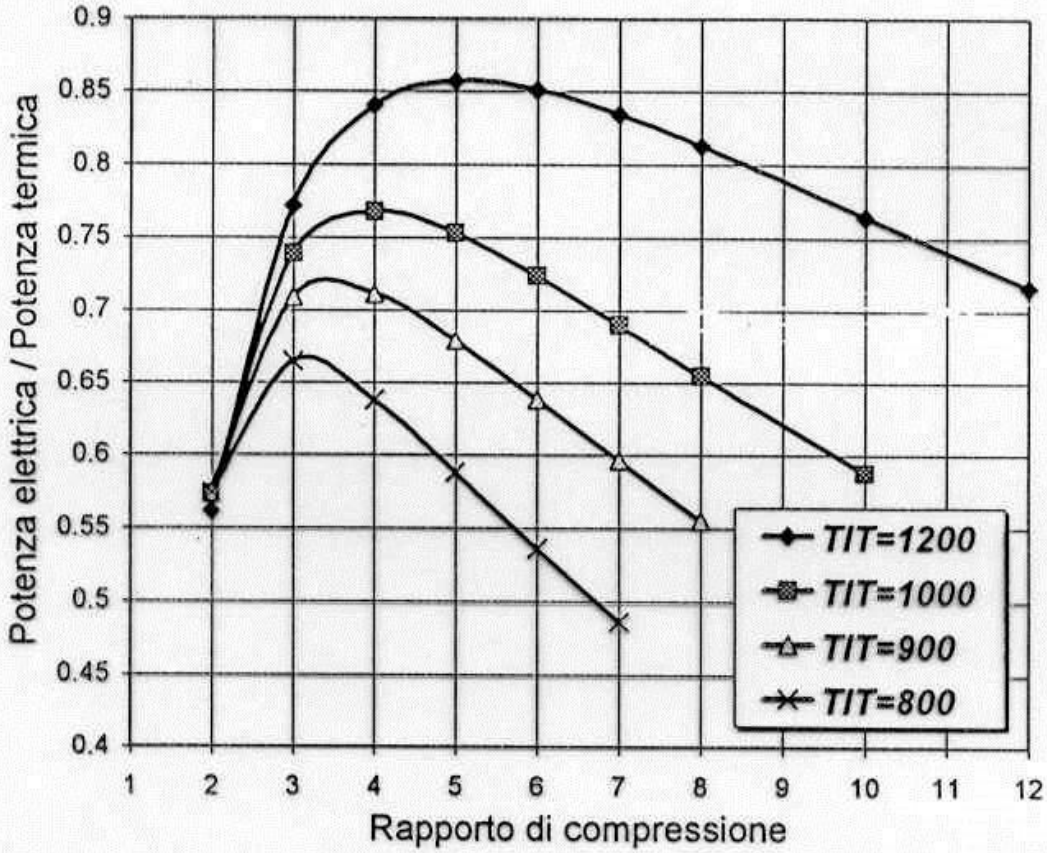


Fig.2.5: *Rapporto tra potenza elettrica e potenza termica ottenibile mediante recupero dai gas di scarico (temperatura finale dei gas: 100°C) per cicli rigenerativi di turbina a gas, calcolati secondo le assunzioni dell'ultima colonna di tab.2.1. TIT in °C.*

Partial load operation can be obtained varying the rotational speed with low penalties

For example, an existing turbine operates with ρ=3.8; t_{in}=900°C producing 55.2 kW at 85,000 rpm. At 72,000 rpm, it gives 50% of the electric power. The efficiency passes from 30% to 26.4%.

As the turbine inlet temperature is not changed, the cycle thermodynamics is not varied. The power reduction is obtained by a flow rate reduction.
This implies a low pressure drop through the regenerator, increasing the expansion ratio in turbine with respect to the compression ratio in the compressor.
Moreover, as the thermal change in the regenerator is lowered, higher thermal efficiency can be reached.

A scheme of a real micro turbine is here represented with the values of temperature, pressure and mass flow rate in the various parts of the system.

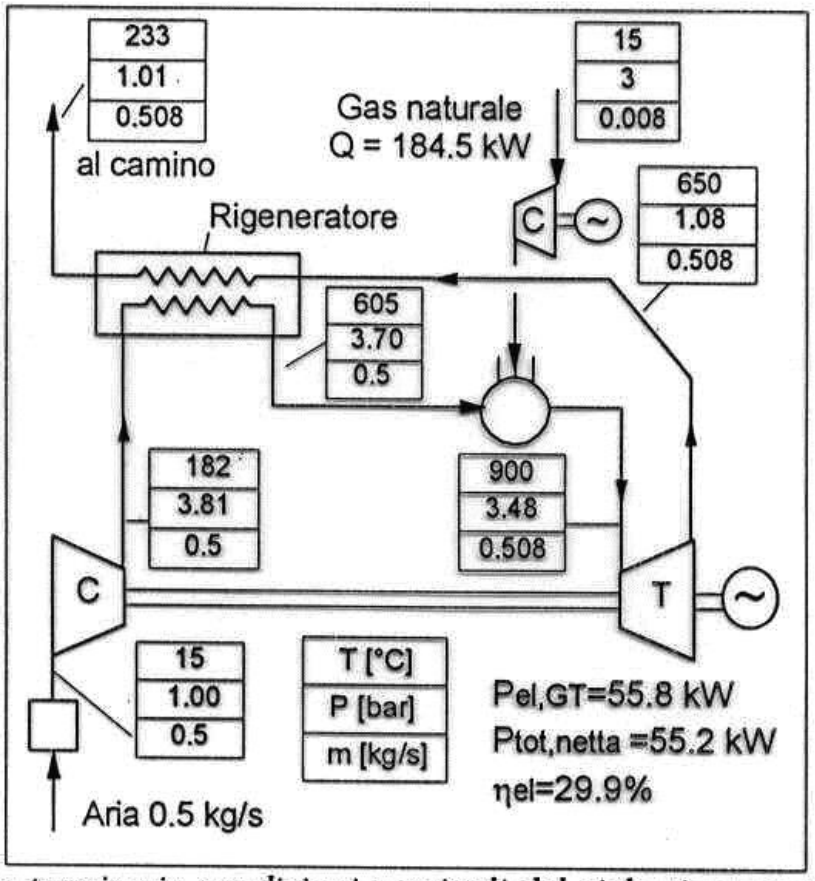
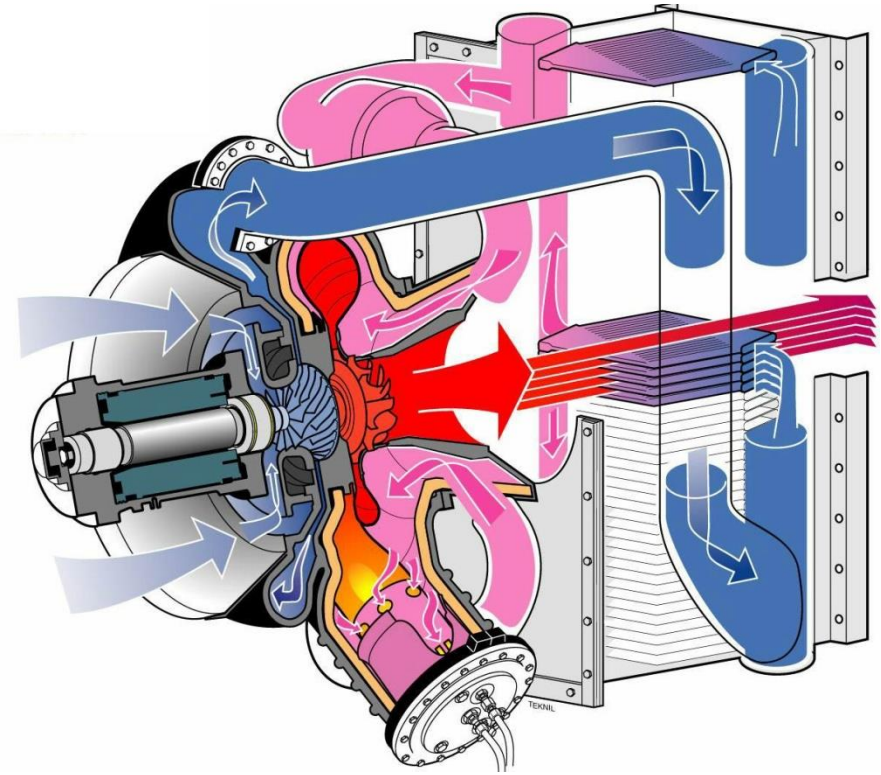
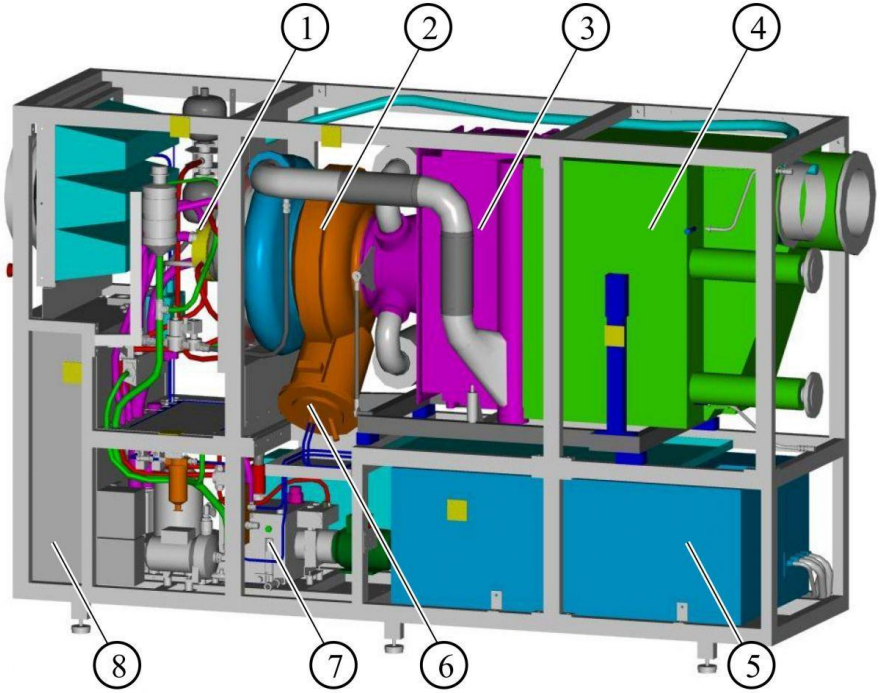


Fig.2.10: *Bilancio termico in condizioni nominali del ciclo rigenerativo di microturbina considerato per l'analisi in fuori-progetto.*





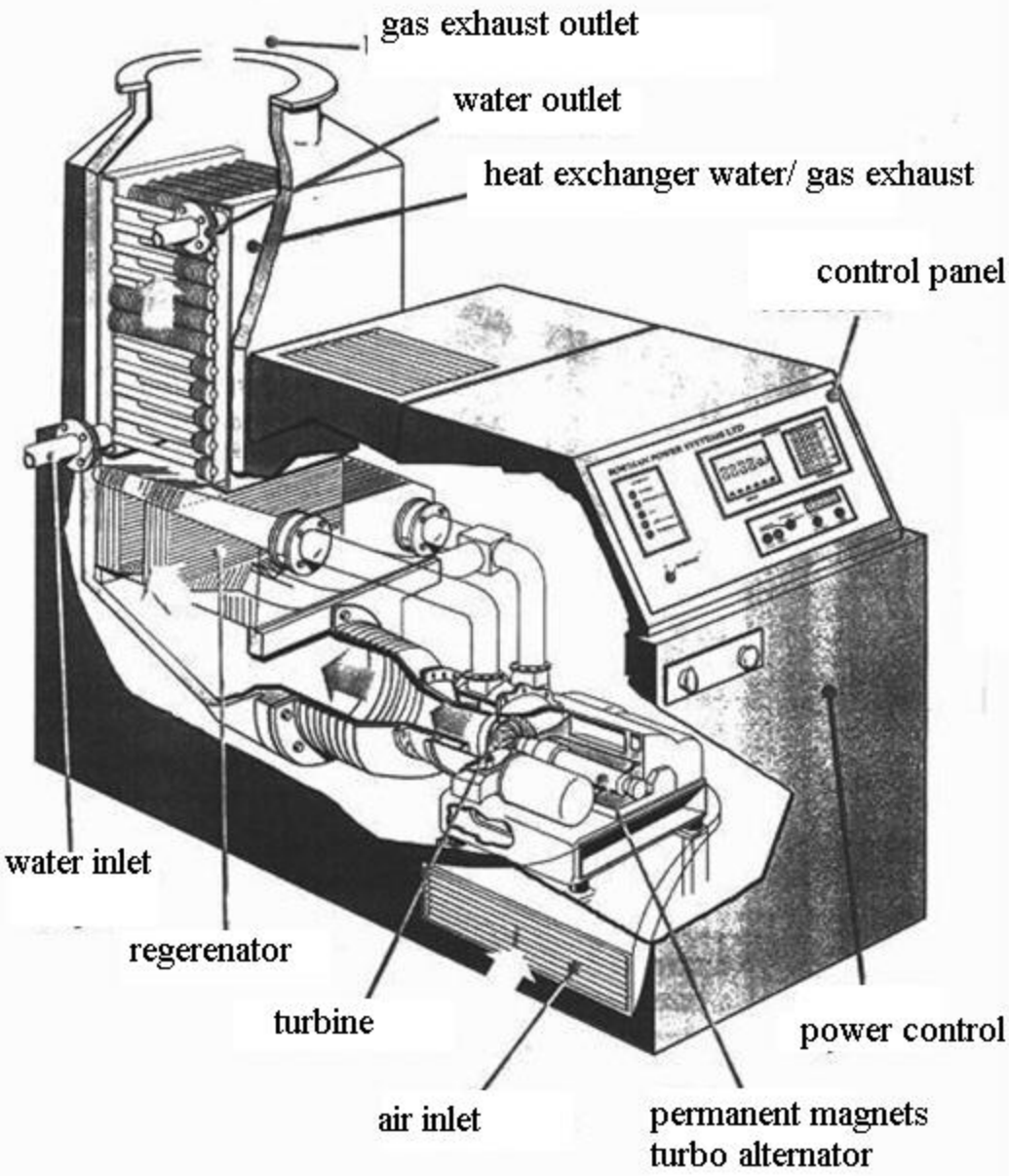
1. Electrical Generator
2. Gas Turbine Engine
3. Recuperator
4. Heat Exchanger
5. Power Electronics
6. Combustion Chamber
7. Auxiliary Systems
8. Control System

Scheme of the Turbec T100 Microturbine 100 kW model, efficiency 33% , thermal production (hot water) 155 kW at 70-90 °C

Costruttore	Potenza (kW)	Rendimento (LHV)	Tmax (°C)	β	Portata aria (kg/s)	n° giri / minuto
Allied Signal Aerospace Co.	48.4	30%	1010	3.3	0.44	75000
Capstone	24	28%	900	3.25	n.d.	96000
Elliott Turbomachinery Co.	45	30%	1010	4	0.41	116000
Elliott Turbomachinery Co.	80	30%	1010	4	0.835	68000

The figure represents the complete system with all the various parts.

The section of the turbine and electric generator is a small part of the whole system.



Rena **Fig.2.15**: *Sezione del package cogenerativo completo proposto dalla Bowman e sviluppato sulla base della turbina a gas Elliott Energy da 45 kW.*