CHAPTER 9

COGENERATION PLANTS

9.1 INTRODUCTION

In conventional utility power plants, electric energy is generally produced with an overall efficiency, that is, the ratio between useful output and input power as fuel, of less than 34-38% because of the large quantity of heat discharged into the atmosphere through cooling towers, lakes or rivers, without recovery.
Differently, in cogeneration plants, either useful heat and mechanical or electric power are generated from fuel or power is produced by recovering low-level heat from processes. An overall efficiency ranging from 55% to 85% is obtainable, depending on the type of cogeneration plant.

Cogeneration is an effective method of primary energy conservation; from this point of view, cogeneration can be applied whenever it is economically justified. The correct use of this system implies a balance between electric-power requirements and process-heat requirements in quantity and in quality. If this balance does not exist, electric power must be exchanged with utility and additional heat must be produced by boiler plants. The discharge of unnecessary heat should be avoided; otherwise, the system works with lower efficiency like a conventional utility plant.

Before the widespread diffusion of utilities to produce and distribute electric energy to end users, cogeneration plants were very common in industry. Afterwards, because of local regulations and tariffications, these plants have been unevenly located and used, the choice depending on the cost of primary energy, mainly coal and oil, and electric energy tariffs.

9.2 FORMS OF COGENERATION

Cogeneration plants can be grouped basically into two types referred to as topping cycles or bottoming cycles.

The topping cycles produce power, mechanical or electric, before delivering thermal energy to the end users. Typical examples are the backpressure or non-condensing steam turbine cycle, the gas turbine and combined cycles and the diesel engine cycle where exhausts are utilized as heat for end-user needs.

The bottoming cycle recovers thermal energy, which would normally be discarded, to produce process steam and electricity. In this cycle, first thermal energy is used for the process, then the exhaust energy is used to produce mechanical or electric power at the bottom of the cycle. This cycle is most attractive where there is a large quantity of thermal energy at a temperature of 623 K (350°C; 662°F) or greater associated with exothermic reactions, as in many chemical processes and in rotary kilns and furnaces. Recovery of the steam to be used in a steam turbine is the commonest bottoming cycle; for lower temperatures other cycles, like the saturated Rankine cycle with organic fluids, are sometimes used.

The electric generator may be synchronous or asynchronous; the choice between them depends on the working mode: if the system is independent of the utility grid the synchronous generator must be used; if the system is interconnected with the utility grid, either type can conveniently be used. The generator voltage can be at low or medium level depending on the size and on the layout of the internal distribution network.

9.3 THE BACKPRESSURE OR NON-CONDENSING STEAM TURBINE CYCLE

Fig. 9.1 shows typical energy flows in a utility plant designed to generate electric power alone and in a cogeneration plant with backpressure steam turbine exhausting steam headers to the plant process.

The amount of power which can be produced by expanding steam in a prime mover is limited by the Available Energy (AE) between the inlet and outlet of the

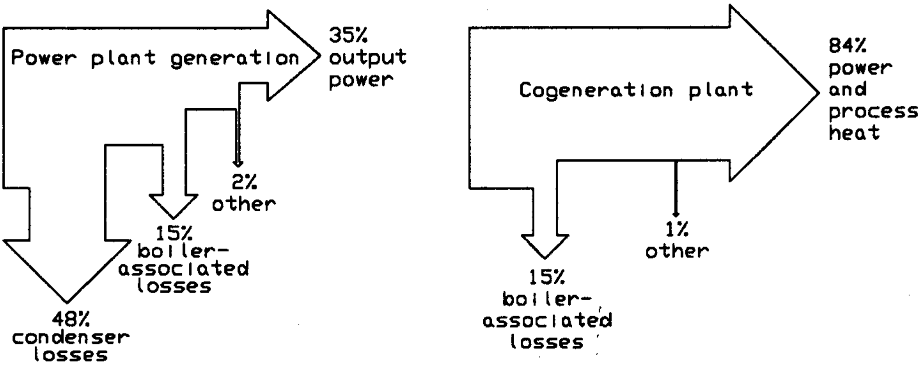


Fig. 9.1 Energy balance in traditional steam turbine power plants and in cogeneration plants

steam turbine. This energy is the enthalpy difference between the inlet superheated steam, at high pressure and temperature, and the outlet steam at lower pressure along an ideal isoentropic expansion. The Mollier diagram or equivalent steam tables (see Chapter 6.4) can conveniently be used for this purpose (see Fig. 9.2). Alternatively, Theoretical Steam Rate tables such as those published by ASME can be used; these report the Theoretical Steam flow Rate (TSR) required to generate 1 kWh in a 100% efficiency expansion process (see Table 9.1). TSR is the ratio between the energy content of 1 kWh (3600 kJ/kWh) and the Available Energy AE (kJ/kg); it represents the amount of steam theoretically needed to produce 1 kWh:

$$TSR(kg/kWh) = 3600\,(kJ/kWh) / AE\,(kJ/kg)$$

where

3600 kJ/kWh = conversion factor (see Table 2.4)

AE = $h_{in}-h_{out}$ along an isoentropic expansion

h_{out} = enthalpy of the steam at outlet condition

h_{in} = enthalpy of the steam at inlet condition

Typical values of Theoretical Steam flow Rate (TSR) are 7-8 kg of steam/kWh with a pressure drop from 4 MPa (40 bar; 590 psi) to 0.4 MPa (4 bar; 59 psi).

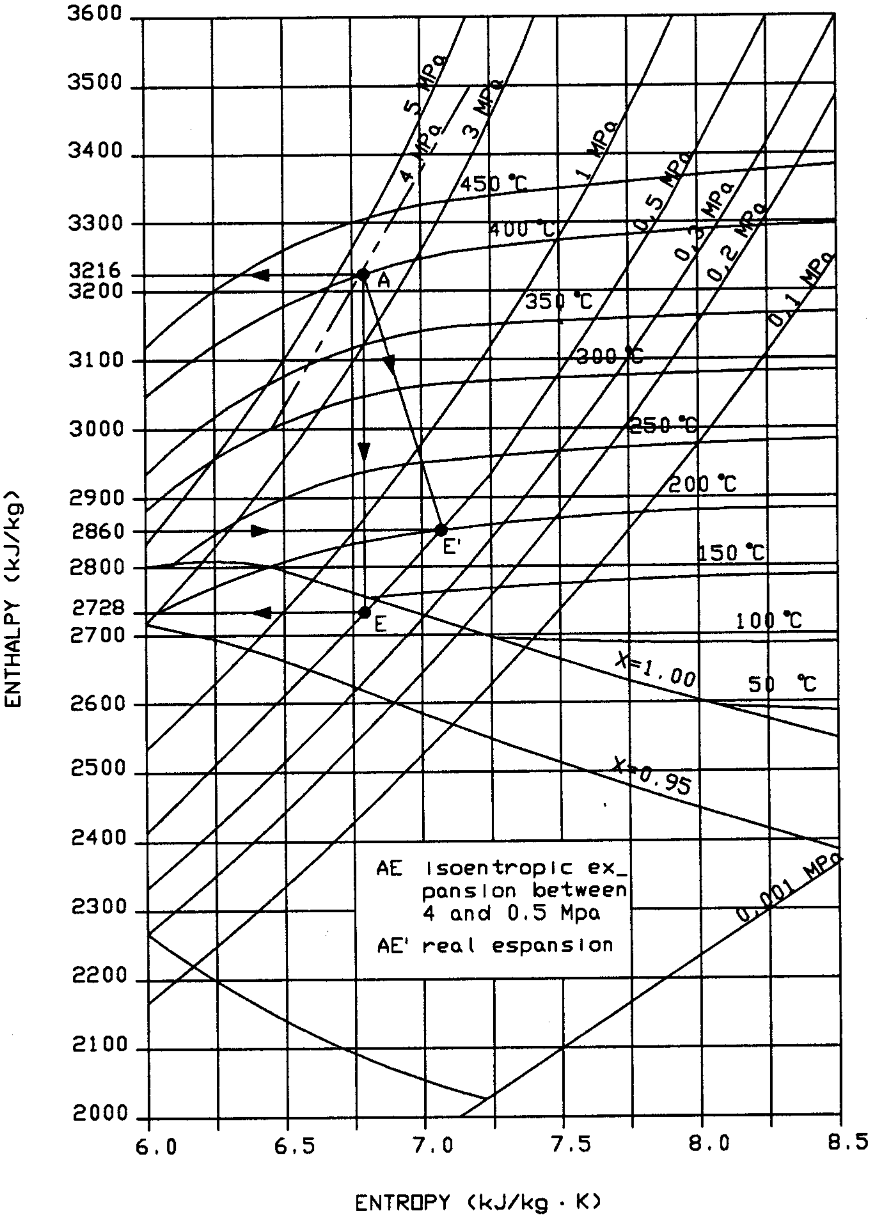


Fig. 9.2 Theoretical Steam Rate and Available Energy representation by the Mollier diagram

Table 9.1 Theoretical Steam Rate

EXHAUST STEAM PRESSURE	MPa	0.034	0.172	0.345	0.690	1.034	1.379	0.000333
	psi	5	25	50	100	150	200	0.048
	bar	0.345	1.724	3.448	6.895	10.343	13.790	0.0033
CASE 1								
(Input 1.034 MPa)								
TSR VALUES	lb/kWh	21.7	31.1	46.0				10.88
	kg/kWh	9.8	14.1	20.9				4.9
AE VALUES	kJ/kg	365.74	255.19	172.53				729.475
CASE 2								
(Input 1.724 MPa)								
TSR VALUES	lb/kWh	16.6	21.7	28.2	45.2	76.5		9.34
	kg/kWh	7.5	9.8	12.8	20.5	34.7		4.2
AE VALUES	kJ/kg	478.11	365.74	281.44	175.59	103.74		849.752
CASE 3								
(Input 2.758 MPa)								
TSR VALUES	lb/kWh	13.0	16.0	19.4	26.5	35.4	48.2	8.04
	kg/kWh	5.9	7.3	8.8	12.0	16.1	21.9	3.6
AE VALUES	kJ/kg	610.51	496.04	409.10	299.49	224.20	164.66	987.150
CASE 4								
(Input 4.137 MPa)								
TSR VALUES	lb/kWh	11.1	13.2	15.4	19.4	23.8	29	7.25
	kg/kWh	5.0	6.0	7.0	8.8	10.8	13.2	3.3
AE VALUES	kJ/kg	715.01	601.26	515.36	409.10	333.47	273.67	1094.715
CASE 5								
(Input 5.861 MPa)								
TSR VALUES	lb/kWh	9.8	11.5	13.1	15.9	18.6	21.5	6.72
	kg/kWh	4.4	5.2	5.9	7.2	8.4	9.8	3.0
AE VALUES	kJ/kg	809.86	690.14	605.85	499.16	426.70	369.14	1181.054
CASE 6								
(Input 8.619 MPa)								
TSR VALUES	lb/kWh	8.8	10.1	11.3	13.3	15.1	16.8	6.26
	kg/kWh	4.0	4.6	5.1	6.0	6.8	7.6	2.8
AE VALUES	kJ/kg	901.89	785.81	702.36	596.74	525.60	472.42	1267.840
CASE 7								
(Input 9.998 MPa)								
TSR VALUES	lb/kWh	8.4	9.5	10.5	12.2	13.8	15.2	6.01
	kg/kWh	3.8	4.3	4.8	5.5	6.3	6.9	2.7
AE VALUES	kJ/kg	944.84	835.44	755.87	650.54	575.12	522.15	1320.579

Initial steam condition							
	CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7
MPa	1.034	1.724	2.758	4.137	5.861	8.619	9.998
psi	150	250	400	600	850	1250	1450
bar	10.3	17.2	27.6	41.4	58.6	86.2	100.0
°C	186	260	343	399	441	482	510
°F	366	500	650	750	825	900	950
kJ/kg	2781	2935	3105	3209	3281	3346	3399
Btu/lb	1196	1262	1335	1380	1411	1438	1461

The previous TSR value can conveniently be converted into an Actual Steam Rate (ASR) by introducing the efficiency of the turbine which takes into account the shift from the isoentropic expansion and the efficiency of the electric generator. Then:

$$ASR(kg/kWh) = TSR/\eta = (3600/AE) \cdot 1/(\eta_T \cdot \eta_G)$$

where

η_T = turbine efficiency

η_G = electric generator efficiency from the shaft to the electric output section.

The electric output is then the ratio between the Actual Steam Flow (ASF) through the turbine as required by the process and the Actual Steam Rate (ASR):

$$P_e(kW) = ASF\ (kg/h)/ASR\ (kg/kWh)$$

Typical values are 10000 kg/h of steam to produce 1000 kW of electric power with a pressure drop of 4 MPa (40 bar; 590 psi).

If ASR is established, depending on the turbine cycle and operating characteristics, the relationship between the inlet and outlet steam enthalpy is as follows:

$$h_{out} = h_{in} - AE \cdot \eta_T = h_{in} - 3600/(\eta_G \cdot ASR) \qquad (kJ/kg)$$

Depending on the size of the turbine, on the quantity and quality of process steam demand and on other operating factors, several options are available (see Fig. 9.3): straight non-condensing turbine, single or multiple extraction noncondensing turbine.

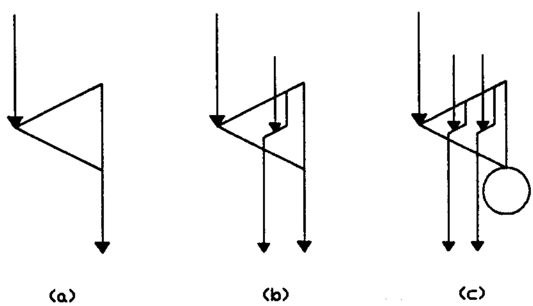
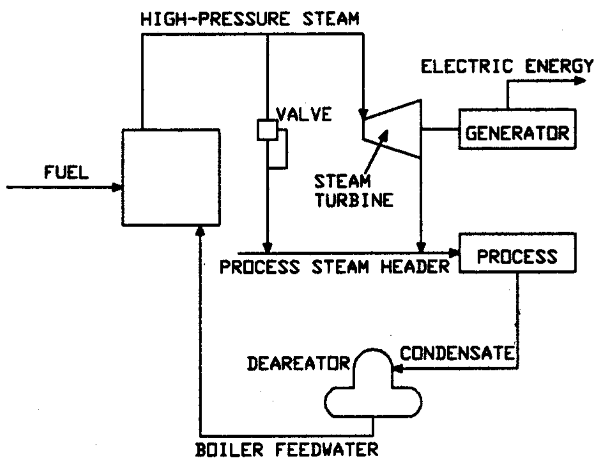


Fig. 9.3 Steam turbine types for cogeneration systems: (a) straight noncondensing (b) single extraction noncondensing (c) double extraction condensing

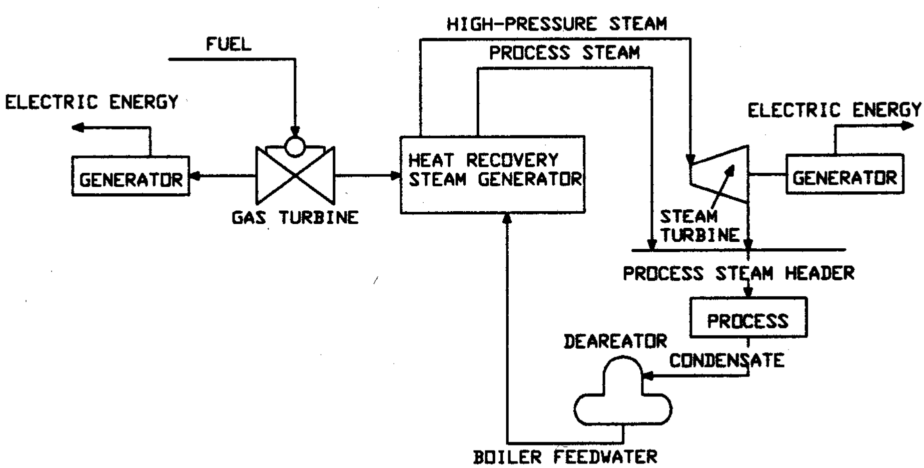
Fig. 9.4 illustrates a typical industrial steam turbine cycle. It consists of a high-pressure boiler, generally 4-10 MPa (40-100 bar; 590-1470 psi) generating superheated steam for admission to a backpressure or noncondensing steam turbine. The steam turbine drives either an electric generator or other equipment such as compressors, pumps, etc.. The majority of the steam energy content remains in the outlet steam which will be utilized in the process; the energy required for mechanical power and related losses is delivered between the inlet and outlet of the turbine.

The main factors governing the optimal exploitation of a steam turbine cycle based on a fixed exhaust pressure and a constant net heat to process can be summarized as follows:

— the overall efficiency of a steam turbine plant is influenced by the inlet volume flow, inlet/outlet pressure ratio, geometry associated with turbine staging, throttling losses, mechanical coupling and electric generator losses.
— Table 9.2 shows typical values of turbine and overall efficiency and ASR coefficient for a range of unit sizes of backpressure and condensing turbines. Small single-valve, single-stage units have low efficiency, less than 50%. The multivalve, multistage units may reach an efficiency of up to 80% in large power plants; in consequence, the greater the efficiency the greater the electric power generated with the same steam flow. A proper sizing of the prime mover to meet the process requirements is the first step in the optimization of a cogeneration plant;
— the overall efficiency of a steam turbine diminishes with the output power rate, so there is always a minimum useful value below which operation is not economically worthwhile. Technical characteristics given by manufacturers must be consulted;
— an increase in the initial steam pressure and/or steam temperature will increase the amount of electric energy generated because of the increase in the enthalpy inlet-outlet difference (see Table 9.1). Since an increase of the inlet steam temperatures generally results in an increase of the temperature of the steam supplied to the process, a top limit exists to prevent outlet steam being too highly superheated, as this would necessitate a desuperheating process before delivery to the end users. In such a case, desuperheating water must be added downstream of the turbine outlet; in consequence, the steam flow through the turbine must generally be reduced by the quantity of water added so that the electric power generated is not increased by using higher inlet steam conditions;



Topping Cycle



Combined Cycle

Fig. 9.4 Plant combined cycle cogeneration steam system

— an increase of the energy available for power generation is also possible if the outlet pressure is reduced with a given set of initial steam conditions. The outlet pressure must be the lowest value compatible with the end-user needs;
— an increase in the power generated, with the same fuel consumption in the boiler plant, is possible if feedwater is heated by using steam extracted from turbine stages or exhausted from the process. The level of the power increase depends on the number of heaters and on the temperatures;
— to avoid damage like the erosion of the turbine blades by liquid droplets, inlet steam is always superheated. The outlet steam has generally a quality index (see quality index in Chapter 6.4) of not less than 90%.

Table 9.2 Overall efficiency and ASR coefficient for backpressure and condensing turbines

Type of Unit	Size MW	AE kJ/kg	range η_T %		range $\eta_T \cdot \eta_G$ %		range ASR lb/kWh		range ASR kg/kWh	
BACKPRESSURE Single- valve/ single-stage	0.1-1	515.36	40	50	38	48	40.5	32.4	18.4	14.7
Multivalve/ multistage	1-5	515.36	65	75	62	71	24.9	21.6	11.3	9.8
	5-25	515.36	75	80	71	76	21.6	20.3	9.8	9.2
CONDENSING Single- valve/ single-stage	0.1-1	1320.579	40	50	38	48	15.8	12.7	7.2	5.7
Multivalve/ multistage	3-20	1320.579	70	76	67	72	9.0	8.3	4.1	3.8
	2-50	1320.579	76	80	72	76	8.3	7.9	3.8	3.6

Notes
Typical operating conditions for medium power backpressure turbine
See CASE 4 in Table 9.1 with output pressure 0.34 MPa, 3.4 bar, 50 psi
Input steam 4.137 MPa, 41.37 bar, 600 psi

Typical operating conditions for condensing turbines
See CASE 7 in Table 9.1 with atmospheric output pressure

Typical values for industrial applications with an electric power ranging between 500 kW and 5000 kW are: inlet pressure 4-10 MPa, outlet pressure 0.3-1 MPa, Actual Steam Rate 10-15 kg of steam/kWh, oil consumption in the boiler plant 0.7-1 kg of oil/kWh.

An example of a steam turbine cycle is given in Chapter 9.7.

9.4 THE GAS TURBINE CYCLE

The gas turbine as prime mover associated with a heat recovery boiler or with direct use of the exhausts in the process is another highly efficient topping cycle. It is available for both mechanical and electric generator drive in a wide range of sizes from a few hundreds to hundreds of thousands of kW. One can distinguish between industrial turbines, in the range from 1 MW to 200 MW, and aero-derivative turbines in the range from 2 MW to 40 MW. The revolution speed is generally 3000-3600 r/min for power higher than 60 MW and 6000-12000 r/min for lower power.

Gas turbine power plants may operate on either an open or a closed cycle, generally the Brayton cycle, as shown in Fig. 9.5. In the open cycle, which is the commoner, atmospheric air is continuously drawn into the compressor; then, air at high pressure enters a combustion chamber where it is mixed with fuel and combustion occurs resulting in combustion exhaust at high temperature. The combustion products expand through the turbine and they are discharged into the environment. Part of the mechanical power (typically 60%) is generally used to drive the compressor and the remainder to generate electricity or to drive other loads.

Fig. 9.6 shows typical energy flows for a gas turbine plant without heat recovery and for a cogeneration plant with exhaust heat recovery (50%) for steam generation or for direct recovery in the process.
The maximum mechanical power available at the shaft does not generally exceed 20-25% of the turbine input power as fuel in small and medium-sized units (1-5 MW). The amount of the recoverable heat depends on the bottom temperature level required by the end user and in consequence on the stack-gas temperature.
Typical input power values, in the case of natural gas input, are 545 Sm^3/h (5228 kW) to produce 1000 kW of electric power and 2500-3700 kW of recoverable heat. Specific consumptions, Sm^3 of natural gas per kWh, are lower for bigger gas turbine units.

The overall efficiency of the system, that is, the ratio of total output to input power, is roughly 70-80%, depending on the bottom level of the thermal energy user which determines the temperature of the stack exhausts. Higher values are not generally possible, because of the great amount of the air flow and so of the exhausts. Gasoil instead of natural gas can also be used with appropriate burners.

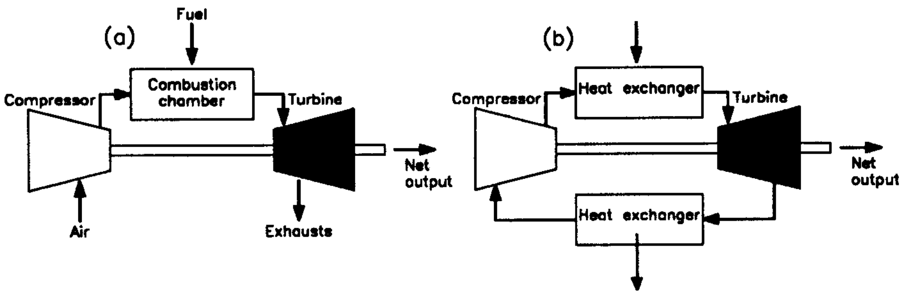


Fig. 9.5 Simple gas turbine: a) open to the atmosphere; b) closed

Combined cycles, where exhausts with or without supplementary burners produce steam and then electric energy in a steam turbine cycle, can also be used for sizes higher than 20 MW of total electric output. These cycles allow much higher power-producing capability per unit of steam than the backpressure system or the gas turbine by themselves.

Combined cycles with gas turbine and condensing or backpressure turbines are widely used with the ratio of output electric power to input power equal to 46-52%.

The principal data to take into account for selecting gas turbines are:

— unit fuel consumption/output power. Typical values range between 4.5 kW to 5.5 kW input (0.47-0.57 Sm^3/h of natural gas) per 1 kW of output shaft power. Gasoil can be used instead of natural gas;

— exhaust flow and temperature values, on which the selection of the recovery system can be based;

— the bottom level of temperature required from the process end users which limits the temperature at which the exhaust can be cooled;

— ambient temperature on which depend both the density and so the mass of air flowing through the compressor, which is a nearly constant volume flow-rate machine, and the bottom temperature of the cycle;

— atmospheric pressure which varies according to the altitude. In consequence, the density and so the mass of air flowing through the compressor changes;

— pressure drops upstream of the compressor and turbine system and downstream of the turbine that may influence the gas turbine useful output;

— the inlet air compression ratio and temperature, to which efficiency is closely related. Air compression ratios of 5 to 7 are quite common; higher ratios can

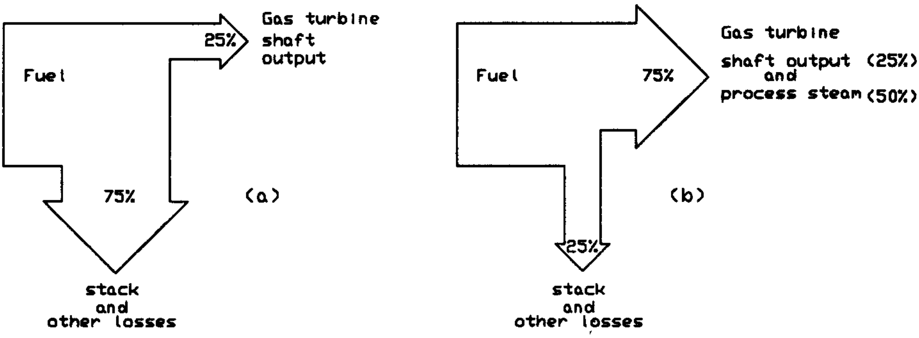


Fig. 9.6 Energy balance (a) in a gas turbine and (b) in a cogeneration plant with a gas turbine (exhausts are recovered to produce steam)

be used to improve efficiency. Note that the influence of air compressor inlet temperature and of altitude is considerable since they affect the weight of air (and oxygen) available for combustion. Standard conditions for land-based systems are generally 288 K (15.6°C; 60°F) intake at 0.1 MPa pressure (1 atm; 1 bar; 14.2 psi). A reduction in ambient air density, due to either altitude or high temperature, results in a derating of the turbine power output. Typical deratings are as follows: 10% per 1000 m of increase of altitude; 2% per 977 Pa (100 mm H20; 3.9 inch H20) due to pressure drops downstream and upstream of the turbine; 7-10% per 10 K or °C (18°F) of increases in ambient temperature;

— the mass of air is considerably higher than the theoretical value needed for combustion (excess air equal to 300-400%) since it is imposed by the operating constraints of the air compressor-turbine system;
— if natural gas is used, the pressure of the gas which is roughly 1.5 the air pressure at the inlet of the turbine, must be at least 1-1.2 MPa. In fact, typical values of air pressure are 0.6-1.5 MPa for industrial turbines and 1.5-3 MPa for aero-derivative turbines. A gas compressor may be required to ensure these conditions.

With steam production, heat-recovering steam generators are installed downstream of the turbine exhaust outlet. The exhaust temperature is about 723-823 K (450-550°C; 842-1022°F), the lower for aero-derivative turbines, with an excess air of 400%. The exhaust is 15-18% oxygen.

According to the classification made in Chapter 6, the generators may be of the unfired, supplementary fired or the fired type.

Unfired generators use the exhaust of the gas turbine unit and they are convective heat exchangers. The exhaust gas temperature is roughly 773-823 K (500-550°C; 1000°F).

Supplementary fired generators use a supplementary burner located downstream of the gas turbine duct to raise the temperature of the exhaust to a maximum of 1088-1143 K (815-870°C; 1500-1600°F). Superheated steam at high pressure, suitable for steam turbines in combined cycle plants, is produced.

Fired generators are similar to steam boilers. Gas turbine exhausts are used to preheat the combustion air. Outlet air can be added if necessary to meet the steam demand. Turbine exhausts are bypassed if they are in excess.

The energy recovery from the exhausts is closely related to the saturation temperature of the steam required by the process, which affects the minimum temperature value at which the exhaust can be cooled through the unfired boiler. Lower values can be obtained if an economizer is installed to pre-heat feedwater (see Chapter 6.9).

A preliminary evaluation of the amount of steam that can be generated is as follows:

$$m_{steam} = m_g \cdot c_p \cdot (t_g - t_s) \cdot \eta/(h_s - h_i)$$

where

m_{steam} = steam flow-rate produced by the recovery system in kg/s

m_g = exhaust gas flow-rate in kg/s

c_p = specific heat of the exhaust; typical value is 1 kJ/kg · K (0.24 Btu/lb · °F)

t_g = temperature of the exhaust gas

t_s = saturation temperature of the steam

h_s = enthalpy of the superheated steam or of the saturated steam

h_i = enthalpy of the saturated liquid in the steam drum

η = effectiveness (typical value 0.95)

In the case of supplementary burners, the same relationship can be used by varying the inlet gas temperature until the desired steam flow is reached.

Typical values of operating parameters for several gas turbines are shown in Table 9.3. Note that the output power can be regulated by varying the input fuel or by throttling the air input in the compressor.

Additional equipment to start the compressor, such as electric motor, reciprocating engine motor or compressed air tanks, are always required.

An example of a gas turbine cycle is given in Chapter 9.7.

Table 9.3 Technical parameters for standard gas turbines

Size Unit	Specific consumption	Input fuel as natural gas			Input air flow			Heat recovery	Output electric energy	Total efficiency	Electric efficiency
MW	kJ/kWh	10^6kJ/h (c)	10^6Btu/h	Sm^3/h	$10^3 Sm^3$/h	10^3kg/h	10^3lb/h	10^6kJ/h (a)	10^6kJ/h (b)	% (a+b)/(c)	% (b)/(c)
0.61	18515	11.3	10.7	329	14.7	19.0	41.9	5.7	2.2	69.9	19.4
1	18916	18.9	17.9	551	24.6	31.8	70.1	9.5	3.6	69.5	19.0
5	14995	75.0	71.1	2184	97.7	126.0	277.8	37.8	18.0	74.4	24.0
10	12410	124.1	117.6	3615	161.7	208.6	459.9	62.6	36.0	79.4	29.0
25	12200	305.0	289.1	8886	397.4	512.7	1130.3	153.8	90.0	79.9	29.5
40	12000	480.0	455.0	13984	625.5	806.8	1778.8	242.1	144.0	80.4	30.0
100	12000	1200.0	1137.4	34960	1563.7	2017.1	4447.0	605.1	360.0	80.4	30.0

Notes
— air density in standard conditions 1.29 kg/Sm^3
— combustion air 45 Sm^3 per unity of Sm^3 of natural gas
— air pressure ratio in the range 5-7
— exhaust temperature 723 K, 450°C, 842°F; stack exhaust temperature 423 K, 150°C, 302°F
temperature drop available for recovery 300 K, 300°C, 540°F

9.5 THE DIESEL ENGINE CYCLE

Reciprocating engine types, principally the spark-ignited gas engine for natural gas or the Diesel engine for liquid fuel, are widely applied in cogeneration systems to drive electric generators and mechanical loads such as compressors and pumps. Alternating machines can be designed to run on either gas or gasoil.

Diesel engines are available in a wide range of power, from several to thousands kW at different operating speeds ranging from 100 r/min to 1800 r/min according to the size and technical characteristics of the system.

Fig. 9.7 shows a typical energy flow in a cogeneration plant with a diesel engine: shaft output (35%), steam from exhaust (20%), hot water from cooling (30%), losses (15%).

Exhaust gas, at a temperature in the range 623-723 K (350-450°C; 662-842°F), permits steam generation at a saturation pressure of 0.3-1 MPa, suitable for industrial applications.

Jacket and piston water cooling as well as lubricant cooling water can provide hot water at an average temperature of 343-353 K (70-80°C; 158-176°F) which can be used for space-heating or industrial low-temperature end users. Superheated water can also be produced with special design of the diesel engine and recovery system.

The maximum mechanical power available at the shaft generally ranges from 28 to 40% of the diesel engine input power as fuel. The higher values refer to turbocharged machines. In consequence, the quantity of heat rejection equals 72-60% of the fuel input. The amount of the recoverable heat depends on the bottom temperature level required by the end user in the form of hot water.

Cooling equipment must also be provided if users of hot water are not in operation. Exhausts will be discharged into the atmosphere.

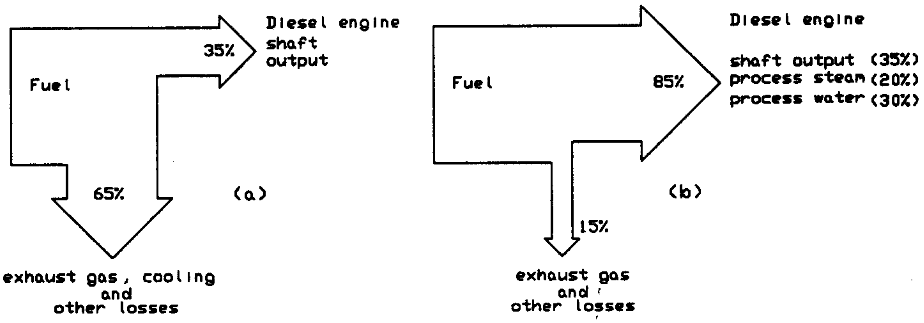


Fig. 9.7 Energy balance (a) in a diesel engine and (b) in a cogeneration plant with a diesel engine (exhausts are recovered to produce steam)

Typical input-power values, in the case of natural gas input, are 345 Sm^3/h (3300 kW) to produce 1000 kW of electric power and 1750 kW of recoverable heat. The overall efficiency of the system, that is, the ratio between output and input power, is roughly 83%. The ratio of output electric power to input power is roughly 30%.

Typical values of operating parameters for several diesel-engines fed by natural gas are shown in Table 9.4. As a general rule, this type of prime mover is attractive if the process requires large quantities of low-level heat recovered from the jacket water and lubricant oil cooling systems which may reach 50% of the heat rejection. An example of a diesel engine cycle is reported in Chapter 9.7.

9.6 DETERMINING THE FEASIBILITY OF COGENERATION

Cogeneration feasibility is based on economic and technical factors which have to be correlated to complete a valid evaluation. Major factors for consideration are:

— the ratio between the electricity demand and the fuel for the factory defined as daily, monthly and yearly ratio. This ratio must be consistent with the ratio between electric output power and heat recovery for the cogeneration system chosen;

Table 9.4 Technical parameters for diesel engines

Size Unit	Specific consumption	Input fuel as natural gas			Input air flow			Heat recovery	Output electric energy	Total efficiency	Electric efficiency
MW	kJ/kWh	10^6kJ/h (c)	10^6Btu/h	Sm^3/h	10^3Sm^3/h	10^3kg/h	10^3lb/h	10^6kJ/h (a)	10^6kJ/h (b)	% (a+b)/(c)	% (b)/(c)
0.25	12000	3.0	2.8	87	1.3	1.7	3.7	1.5	0.9	78.83	30.00
1	11500	11.5	10.9	335	5.0	6.5	14.3	5.6	3.6	80.14	31.30
5	11500	57.5	54.5	1675	25.1	32.4	71.5	28.1	18.0	80.14	31.30
10	11250	112.5	106.6	3277	49.2	63.4	139.8	54.9	36.0	80.83	32.00
25	11250	281.3	266.6	8194	122.9	158.5	349.5	137.3	90.0	80.83	32.00
40	11250	450.0	426.5	13110	196.6	253.7	559.3	219.7	144.0	80.83	32.00
100	11250	1125.0	1066.4	32775	491.6	634.2	1398.2	549.4	360.0	80.83	32.00

Notes
— air density in standard condition 1.29 kg/Sm^3
— combustion air 15 Sm^3 per unity of Sm^3 of natural gas

— the profile of thermal demand, including temperature levels of end user requirements and typical fluctuations of the demand (daily, monthly, yearly). Temperature levels must be consistent with the level of heat rejected from the cogeneration system;
— the profile of electric demand and typical fluctuations as for the thermal profile. Thermal and electric profiles must be correlated with each other;
— purchased fuel and electricity costs, present and projected future costs;
— working hours per year and per the total life of the plant;
— plant system sized for the present factory needs and for the future;
— capital cost of the cogeneration plant and operating cost during the life of the plant;
— environmental issues.

Many cogeneration approaches can be followed in order to make a choice among system types and sizes. In order to ensure, however, the highest efficiency of the system, the recovery of the rejected heat must be effective in any operating condition of the cogeneration plant. Additional boiler plants will satisfy the end-user requirements, if these are higher than the recovery heat. Depending on the industrial processes, this constraint can be more or less important in determining the size of the plant.

A first approach is to design a system which is capable of meeting thermal load requirements, regardless of the electric demand. It is connected to the utility grid and sells excess or buys additional electricity depending on the factory's thermal and electric profile and on the operating conditions.

A second approach is to design a system which is capable of meeting either peak or base electric load requirements, regardless of the thermal demand which nevertheless must be greater than the heat rejected. It is connected to the utility grid and sells excess or buys additional electricity depending on the sizing and on the operating conditions.

A third approach is to design a system independent of the utility grid. It requires overcapacity or redundant equipment to ensure reliability which is guaranteed by the utility in the first two approaches. These systems have traditionally been oversized to meet peak electric demand, with supplementary equipment to satisfy the thermal demand if necessary.

9.7 PRACTICAL EXAMPLES

Three examples of cogeneration plant are given below. The saving in primary energy in comparison with thermal utility plants is also shown.

> **Notice that cogeneration plants save primary energy, not energy entering the factory. They can effect an energy-cost saving only if the balance between electric energy and thermal energy costs is favorable for the factory.**

Example 1
Cogeneration plant with steam turbine

Table 9.5 lists data and operating parameters of a steam turbine in two typical working conditions (input-output pressure drop equal to 3.793 MPa and 8.963 MPa). Specific consumptions attributed to the production of electric energy are calculated as kJ/kWh or kg_{oil}/kWh entering the boiler plant.

Primary energy saving in comparison with standard thermal utility plants and variation of the factory energy consumption (additional fuel consumption and reduction of kWh from utilities) are also shown.

Economic evaluations can also be made by introducing the cost of electric energy and fuel. For a preliminary evaluation, the average cost of electric energy purchased from utilities can be introduced (see Table 20.3); for a more detailed analysis, it is necessary to calculate the purchased energy and the consequent cost corresponding to the new demand profile of the plant for utilities-energy. As a general rule, the new cost per unit of purchased electric energy will be higher than the current cost because of a lower exploitation of the power supplied under the utility contract (as happens if the cogeneration plant covers the base of the electric demand and the utility supply the additional demand). Local regulations concerning the selling of energy to the utilities and its purchase from them in emergency must also be considered.

Maintenance costs, too, must be taken into account. They can be introduced as a fixed cost per unit of kWh produced.

Example 2
Cogeneration plant with gas turbine

Table 9.6 lists data and operating parameters of a gas turbine in typical working conditions. Specific consumptions attributed to the production of electric energy are calculated as kJ/kWh or Sm^3/kWh of natural gas entering the gas turbine.

Primary energy saving in comparison with standard thermal utility plants and variation of the factory energy consumption (additional fuel consumption and reduction of kWh from utilities) are also shown.

The recoverable heat varies according to the end-user requirements: hot air for drying, steam and hot water with or without additional burners.

Economic evaluations can also be made by introducing the cost of electric energy and fuel. For a preliminary evaluation, the average cost of electric energy purchased from utilities can be introduced (see Table 20.3); for a more detailed analysis, it is necessary to calculate the purchased energy and the consequent cost corresponding to the new demand profile of the plant for utilities-energy. As a general rule, the new cost per unit of purchased electric energy will be higher than the current cost because of a lower exploitation of the power supplied under the utility contract (as happens if the cogeneration plant covers the base of the electric demand and the utility supply the additional demand). Local regulations concerning the selling of energy to the utility and its purchase from them in emergency must also be considered.

Maintenance costs, too, must be taken into account. They can be introduced as a fixed cost per unit of kWh produced.

Example 3
Cogeneration plant with diesel engine

Table 9.7 lists data and operating parameters of a diesel engine in typical working conditions. Specific consumptions attributed to the production of electric energy are calculated as kJ/kWh or kg_{gasoil}/kWh or Sm^3/kWh of natural gas entering the engine.

Primary energy saving in comparison with standard thermal utility plants and variation of the factory energy consumption (additional fuel consumption and reduction of kWh from utilities) are also shown.

The recoverable heat varies according to the end-user requirements: hot air for drying, steam and hot water at different temperatures to which the possibility of a complete exploitation of the rejected heat is correlated.

Economic evaluations can also be made by introducing the cost of electric energy and fuel. For a preliminary evaluation, the average cost of electric energy purchased from utilities can be introduced (see Table 20.3); for a more detailed analysis, it is necessary to calculate the purchased energy and the consequent cost corresponding to the new demand profile of the plant for utilities-energy. As a general rule, the new cost per unit of purchased electric energy will be higher than the current cost because of a lower exploitation of the power supplied under the utility contract (as happens if the cogeneration plant covers the base of the electric demand and the utility supply the additional demand). Local regulations concerning the selling of energy to the utility and its purchase from them in emergency must also be considered.

Maintenance costs, too, must be taken into account. They can be introduced as a fixed cost per unit of kWh produced.

Table 9.5 Examples of steam turbine cycles (reference 1 t of steam as input)

	INPUT (1 t of steam)			
a	pressure	MPa	4.137	9.997
b	temperature	°C	399	510
b'		°F	750	950
c	enthalpy	MJ	3209	3399
	OUTPUT (1 t of steam)			
d	pressure	MPa	0.344	1.034
e	theoretical enthalpy	MJ	2693.64	2824
$f = c - e$	available energy AE	MJ	515.36	575
g	expansion efficiency	%	75	75
$h = f \cdot g/100$	useful drop	MJ	386.52	431.25
$i = c - h$	actual output enthalpy	MJ	2822.48	2967.75
l	temperature	°C	180	220
l'		°F	356	428
m	average condensate enthalpy	MJ	440	440
$n = i - m$	useful heat for process	MJ	2382.48	2527.75
	BOILER			
$o = c$	output steam enthalpy	MJ	3209	3399
$p = m$	average condensate enthalpy	MJ	440	440
$q = o - p$	necessary heat	MJ	2769	2959
r	boiler efficiency (NHV as reference)	%	89	90
$s = q \cdot 100/r$	fuel input	MJ	3111.235	3287.777
$t = s/41.86$	oil	kg_{oil}	74.431	78.654
$t' = s/34.325$	natural gas	Sm^3	90.219	95.339
	ELECTRIC GENERATOR			
u	efficiency	%	96	94
$v = h \cdot u/100$	output electric energy	MJ	371.059	405.375
$v' = v \cdot 1000/3600$		kWh	103.073	112.605
$w = 1000/v'$	**SPECIFIC CONSUMPTION**	kgvap/kWh	9.701	8.880

Table 9.5 (continued)

	PLANT			
A	useful electric power	kW	1000	1000
$B = w \cdot A/1000$	steam flow	t/h	9.701	8.880
$C = s \cdot B$	input fuel	MJ/h	30184.73	29197.33
$D = t \cdot B$	oil	kg_{oil}/h	722.122	698.5
$E = t' \cdot B$	natural gas	Sm^3/h	875.300	846.667
$F = n \cdot B$	useful heat for process	MJ/h	23114.45	22447.85
$G = A \cdot 3600/1000$	output electric power	MJ/h	3600	3600
$H = F + G$	total output	MJ/h	26714.41	26047.81
$I = H/C$	plant efficiency	%	88.5	89.21
$L = (C - F \cdot 100/r)/A$	equiv.specific consumption	MJ/kWh	4.213	4.255
		Btu/kWh	3933	4033
		kcal/kWh	1006.554	1016.548
$L' = L/41.86$		kgoil/kWh	0.101	0.101
	PRIMARY ENERGY SAVING			
$M = F/A$	useful heat/electric power	MJ/kWh	23.114	22.447
$N = 9.63 - L$ (*)	primary energy saving	MJ/kWh	5.414	5.372
		Btu/kWh	5132	5092
		kcal/kWh	1293.445	1283.451
$O = N \cdot A/41860$		TOE/h	0.129	0.128
P	operating hours	h/year	6000	6000
$Q = O \cdot P$	primary energy saving	TOE/year	776.067	770.07
	FACTORY ENERGY CONSUMPTION			
$R = L' \cdot P \cdot A/1000$	factory additional oil	TOE/year	603.932	609.929
$S = A \cdot P/1000$	reduction kWh from utilities	MWh/year	6000	6000

Note
(*) reference specific consumption in utility power plant 9630 kJ/kWh

Table 9.6 Examples of gas turbine cycles (reference 100 MJ as input or $94.7 \cdot 10^3$ Btu)

	INPUT (100 MJ as input fuel)		
a	natural gas	MJ	100.00
$b = a \cdot 1000/34325$		Sm^3	2.91
$c = b \cdot 0.75$		kg	2.18
	OUTPUT		
d	shaft energy	MJ	21.00
$e = d \cdot 1000/3600$		kWh	5.83
f	exhaust gas mass	kg	141.6
$g = f/1.29$	exhaust gas volume	Sm^3	109.43
h	exhaust gas temperature	°C	500.00
h'		°F	932
i	exhaust gas specific heat	$kJ/kg \cdot K$	1.08
$l = f \cdot h \cdot i/1000$	exhaust gas enthalpy	MJ	76.23
m	exhaust circuit losses	MJ	3.81
n	typical recoverable exhaust	MJ	50.69
$o = l - m$	max recoverable from exhaust	MJ	72.42
	ELECTRIC GENERATOR		
p	efficiency	%	94.00
$q = d \cdot p/100$	output electric energy	MJ	19.74
$r = q \cdot 1000/3600$		kWh	5.48
	SPECIFIC CONSUMPTION		
$s = a/r$	Input fuel	MJ/kWh	18.24
$t = b/r$		Sm^3/kWh	0.53

Table 9.6 (continued)

	PLANT		
A	useful electric power	kW	1000.00
B = A · s	input fuel	MJ/h	18237.08
C = A · t	natural gas input	Sm^3/h	531.3
D = A · f/r	exhaust flow-rate	kg/h	25744.05
E = A · n/r	typical recoverable heat	MJ/h	9244.38
F = A · 3600/1000	useful electric power	MJ/h	3600.00
G = E+F	useful output	MJ/h	12844.37
H = G/B	plant efficiency	%	0.70
I = (B-E/0.85)/A	specific consumption	MJ/kWh	7.36
		Btu/kWh	6977.58
		kcal/kWh	1758.56
	PRIMARY ENERGY SAVING		
L = (E/0.85)/A	recoverable heat/electric power	MJ/kWh	10.86
M = 9.63 – I (*)	primary energy saving	MJ/kWh	2.27
N = M · A/41860		TOE/h	0.054
O	operating hours	h/year	6000.00
P = N · O	primary energy saving	TOE/year	325.17
	FACTORY ENERGY CONSUMPTION		
Q = (B – E/0.85) · · O/34.325	factory additional fuel	Sm^3/year	1286760
Q' = Q · 34.325/41860		TOE/year	1061.58
R = A · O/1000	reduction kWh from utilities	MWh/year	6000

Notes
equivalent boiler efficiency 85% in step I (Net Heating Value as reference)
(*) reference specific consumption in utility power plant 9630 kJ/kWh
c_p air at 500°C, 932°F equal to 1.08 kJ/kg · K
natural gas density 0.75 kg/Sm^3
air density at 500°C, 932°F equal to 0.45 kg/Sm^3

Table 9.7 Examples of diesel engine cogeneration systems (reference 100 MJ as input or 94.7 · 10^3 Btu)

	INPUT (100 MJ as input fuel)		
a	natural gas	MJ	100.00
b = a · 1000/34325		Sm^3	2.91
c = b · 0.75		kg	2.18
	OUTPUT		
d	shaft energy	MJ	35.10
e = d · 1000/3600		kWh	9.75
f	exhaust gas mass	kg	50.00
g = f/1.29	exhaust gas volume	Sm^3	38.76
h	exhaust gas temperature	°C	475.00
h'		°F	887
i	exhaust gas specific heat	kJ/kg · K	1.08
l = f · h · i/1000	exhaust gas enthalpy	MJ	25.65
m	exhaust circuit losses	MJ	1.28
n	typical recoverable exhaust	MJ	19.83
o = l − m	max recoverable from exhaust	MJ	24.37
p	heat recovery from cooling equipment	MJ	29.75
	ELECTRIC GENERATOR		
q	efficiency	%	94.00
r = d · q/100	output electric energy	MJ	32.99
s = r · 1000/3600		kWh	9.17
	SPECIFIC CONSUMPTION		
t = a/s	Input fuel	MJ/kWh	10.91
u = b/s		Sm^3/kWh	0.32

Table 9.7 (continued)

	PLANT		
A	useful electric power	kW	1000.00
$B = A \cdot t$	input fuel	MJ/h	10911.07
$C = A \cdot u$	natural gas input	Sm^3/h	317.88
$D = A \cdot f/s$	exhaust flow-rate	kg/h	5455.54
$E = A \cdot (n + p)/s$	typical recoverable heat	MJ/h	5410.26
$F = A \cdot 3600/1000$	useful electric power	MJ/h	3600.00
$G = E+F$	useful output	MJ/h	9010.26
$H = G/B$	plant efficiency	%	0.83
$I = (B - E/0.85)/A$	specific consumption	MJ/kWh	4.55
		Btu/kWh	4309.06
		kcal/kWh	1086.02
	PRIMARY ENERGY SAVING		
$L = (E/0.85)/ A$	recoverable heat/electric power	MJ/kWh	6.37
$M = 9.63 - I$ (*)	primary energy saving	MJ/kWh	5.08
$N = M \cdot A/41860$		TOE/h	0.121
O	operating hours	h/year	6000.00
$P = N \cdot O$	primary energy saving	TOE/year	728.28
	FACTORY ENERGY CONSUMPTION		
$Q = (B - E/0.85) \cdot$ $\cdot O/34.325$	factory additional fuel	Sm^3/year	794763
$Q' = Q \cdot 34.325/41860$		TOE/year	651.70
$R = A \cdot O$	reduction kWh from utilities	MWh/year	6000

Notes
equivalent boiler efficiency 85% in step I (Net Heating Value as reference)
(*) reference specific consumption in utility power plant 9630 kJ/kWh
c_p air at 500°C, 932°F equal to 1.08 kJ/kg · K
natural gas density 0.75 kg/Sm^3
air density at 500°C, 932°F equal to 0.45 kg/Sm^3