8.3 INTERNAL COMBUSTION ENGINES

Efficiencies of internal combustion engines are quite variable depending on type and size: 15 to 22% for small gas turbines (micro-GT), 35 - 40% for large modern gas turbines, 25 to 30% for small gas engines, and 35-45% for large diesel and gas engines. Moreover, the efficiency of reciprocating engines varies little with the rotation speed, while that of gas turbines, which operate at nearly constant air flow depends strongly on the load.

Heat is rejected either in exhaust gases or by cooling water, according to a distribution that varies widely depending on the engine type, as shown in Tables 4.1 and 8.1 (you should refer to Chapters 2 and 3 for more details on the operation of internal combustion engines).

TABLE 8.1

Power discharged by kW of useful power

	cooling water (T ≈ 80 - 100 °C)	exhaust (T ≈ 400 - 500 °C)
small gas engine	1.00	1.33
diesel	0.56	1.22
gas turbine		1.8 à 3.5

Moreover, the strong excess air (400%) used in gas turbines means that their exhaust gases contain a lot of oxygen (16 to 18% O_2). It is therefore possible to exploit these gases (usually very clean, especially if the fuel used is natural gas), continuing the combustion processes in furnaces or boilers, or even using them directly as drying fluid.

Given these characteristics, it is obvious that possible configurations are very diverse in terms of internal combustion engine cogeneration.

8.3.1 RECIPROCATING ENGINES

The simplest and most common solution is the production of either hot water at a temperature of 100 °C, or superheated steam at 110-120 °C, as auxiliary to a classical boiler (Figure 8.3.1). Depending on its purity, water can be directly heated in the engine, or must pass through a low temperature heat exchanger. It then recovers the exhaust gas heat in a heat exchanger placed in series. Hundreds of such units of all capacities (a few kW to several MW) are installed worldwide.

A second solution is to cool the engine by an air flow which, in series, provides a convective cooling, and then passes through a recovery heat exchanger on oil, the supercharger intercooler if it exists, the classic radiator cooling system, and finally an air/flue gas exhaust. Hot air is then used for drying, its enthalpy being, if necessary, raised by a supplementary burner.

The engine can also be used for air conditioning, driving a compressor directly connected on the shaft, to obtain industrial cooling or chilled water, heat recovered being used for purposes of either heating or cooling in an absorption machine. An alternator can at times be coupled to the engine instead of the compressor, which allows, according to the tariffs of electricity and refrigeration needs, to modulate the production.

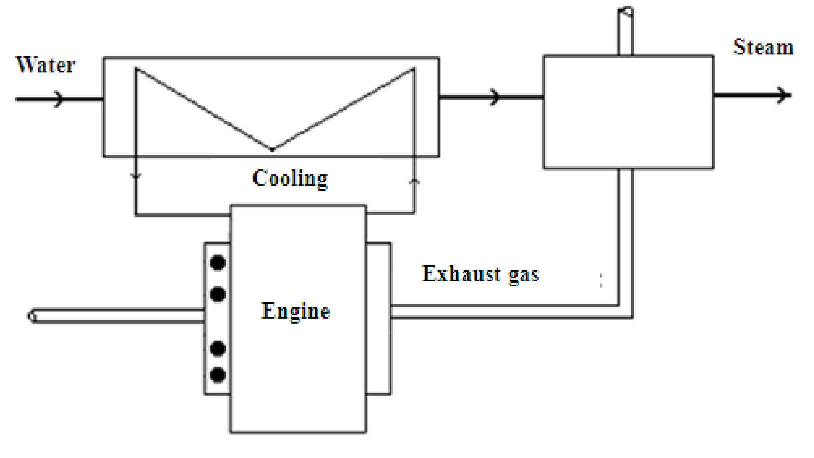

Figure 8.3.1: Internal combustion engine cogeneration unit

The reciprocating engine may finally directly drive a heat pump compressor. The overall efficiency of the system can be very high, given the coefficients of performance of heat pumps.

The heat-power ratio CF is quite low, between 0.5 and 1.5. Overall efficiency is generally very good, above 70%. Mechanical efficiency is usually very high, between 30 and 35% for small gas engines, and up to 45% for large diesel and gas engines. Specific equivalent consumption C_E is of the order of 1.6 to 2.

8.3.2 GAS TURBINES

In gas turbines, the total residual heat is found in the exhaust. The performance of the cogeneration system is directly related to the recovery of these gases.

One solution is to cool exhaust gases in an air-fumes heat exchanger which can heat the air which is then used for many applications. If the turbine is stopped, an auxiliary boiler ensures the supply of heat to meet the plant needs. One generally uses several cascading recovery exchangers on the exhaust, so it can be cooled as much as possible and air is available at different temperatures for various uses.

Figure 8.3.2: Cogeneration gas turbine, Documentation SECC

Another solution, widely used today, especially to replace an existing boiler, is to install a heat recovery steam generator (HRSG) at the output of the gas turbine. The problem of optimizing the recovery exchangers is very similar to the one we have discussed in section

7.3: the best configuration must indeed both cool at best the GT exhaust and provide heat at the highest temperature level possible as needed.

Figures 8.3.2 and 8.3.3 show the two main components of a cogeneration plant for district heating network of this kind built by the Société d'Exploitation des Centrales de Chauffe (SECC). The turbine is a Solar Mars 1000, 10.4 MWe, 15 m long, 3m high and 3m wide. Its total mass (turbine, gearbox, alternator) is about 100 t. The 14.5 MW water tube and single pressure level recovery boiler, brand Bono, has a height of 6.5 m and a width of 5.5 m, a mass of 45 t. In the foreground, the photo shows the bypass chimney that allows for a bypass of the boiler. The vaporization drums are also clearly visible.

Figure 8.3.3: Cogeneration HRSG, Documentation SECC

As we have seen, we can also use an afterburner, which can raise the enthalpy of fumes in pursuing the combustion of residual oxygen in a boiler or furnace. This method avoids the investment of various smoke-air exchangers needed in the previous case. The boiler and burner are of a somewhat unusual type given the level of exhaust gas temperature and composition. It can however be used only if the fuel is desulfurized. Given the cost of light distillates and kerosene, it is generally adapted to natural gas.

The afterburner may also be used in combination with a recovery exchanger, and thus allows better control of the thermal power depending on the needs. The gas turbine flow is nearly constant so that its efficiency is the best possible (cf. example in section 8.5.2).

When the gas turbine used is a micro-turbine rated less than 100 kW, it is enough to heat water, either upstream of an existing boiler, or for hot water uses. The recovery exchanger is then simpler and less expensive than a HRSG. Example presented in section 8.5.1 corresponds to this configuration.

Another very efficient scheme is to directly use the exhaust gases as hot fluid in a dryer. As in the case of the reciprocating engine, the turbine itself is placed in the airflow, so that all losses can be recovered, leading to an overall efficiency close to 1. In addition, the output pressure of the exhaust gas is sufficient to avoid any fan. The hot gas temperatures (400 - 500 °C) being compatible with many industrial requirements, applications of this method are numerous.

If necessary, the two principles above can be combined. Finally, the gas turbine can, like the reciprocating engine, be used to directly drive a compressor or heat pump.

In many cases it may be advantageous to choose a configuration combining a gas turbine and one or more steam turbines to convert into electricity a portion of the heat recovered from the GT exhaust. It is thus possible to design cogeneration combined-cycles, with many variations possible, for example using a back-pressure steam turbine. The performance of these facilities is generally excellent, and they lend themselves well to the rehabilitation of existing cogeneration units, the GT replacing the old boiler (see Example in section 8.5.2).

Heat-power ratio CF is slightly higher than for reciprocating engines, between 0.8 and 1.8. Overall performance is generally very good, above 80%. Mechanical efficiency varies with the size of the gas turbine between 25 and 30% for small and up to 40% for very large ones. Specific equivalent consumption C_E is between 1.4 and 1.7.

8.4 CRITERIA FOR SELECTION

In general, CHP leads to a better use of primary energy than is allowed for separate production of heat and mechanical power. However, the decision-maker rarely views the problem in terms of primary energy saving: he must justify his choices based on the micro-economic context in which they operate.

Before deciding to use a cogeneration facility, it is necessary to make an extensive study of energy and thermal needs and their evolution over time. Indeed, the corresponding investments are generally high, and, to amortize them, the facility must operate at its optimum economic point as long as possible when the energy price is justified, and if possible at about 80 - 90% of the maximum engine capacity, range leading to the best technical performance.

The profitability calculation depends fundamentally on operating conditions and their evolution over time, each facility representing a special case that requires detailed study. It is particularly important to properly ensure that the statutory criteria authorizing the resale of electricity to utilities will be respected.

If security considerations require the use of an independent mechanical power production unit, it is almost certain that its use in cogeneration will be profitable, the overhead being limited to expenses for heat recovery, often well below those corresponding to the engine and alternator.

Moreover, the technical solution choice depends on many factors. For purposes of comparison, Table 8.2 provides approximate values of the different performance indicators for key possible technologies. The first two rows correspond to what is now called micro-cogeneration, based on Stirling engines or small capacity micro turbines. The configuration GT + ST is representative of hybrid solutions between cogeneration and combined cycle, an example of which is given below (8.5.2).

TABLE 8.2 COMPARISON OF DIFFERENT TECHNOLOGIES

	capacity	η_g	η_m	**CF**	C_E
Stirling	0.5 - 100 kW	> 70 %	15 to 30 %	1.2 to 7	1.8 to 2.6
micro-GT	25 - 75 kW	> 80 %	25 to 32 %	1.5 to 2.2	1.5 to 1.7
engines	0.05 - 50 MW	> 70 %	25 - 45 %	0.5 to 1.8	1.6 to 2
GT	5 - 200 MW	> 80 %	35 - 40 %	0.8 to 1.3	1.4 to 1.6
ST	0.5 - 200 MW	> 80 %	6 to 22 %	3 to 12	1.6 to 3
GT + ST	20 - 200 MW	> 80 %	> 40 %	0.8 to 1.2	1 to 1.4