17

Combined Cycle Power Plants

Abstract: The excellent efficiencies reached today by combined cycle power plants (above 60% LHV), are the result of integration into a single production unit of two complementary technologies in terms of temperature levels: gas turbines, which operate at high temperature (in an aero-derivative turbine gases typically enter at 1300°C in the expansion turbine, and come out at around 500°C), and steam plants, which operate at lower temperatures (between 450°C and 30°C in this case).

In section 12.1.5.1 we saw that regeneration can significantly increase the performance of the Brayton cycle, but the percentage of energy recovered is even lower than the temperature and pressure levels of this cycle are higher. In modern gas turbines, regeneration is rarely possible or economically worthwhile. Another way to enhance the residual enthalpy of the exhaust gases is to use them as a heat source for a second cycle of production of mechanical energy. **Combined cycles** correspond to this new generation of thermal power plants.

Keywords: compression, expansion, combustion, combined cycle, HRSG, afterburner.

The principle of a combined cycle is to operate in cascade one or more gas turbines, followed by a steam power plant whose heat source is the cold source of gas turbines (Figure 17.1.1). Under these conditions, the gas turbine exhaust gas is recovered in a recovery boiler that produces steam that is then expanded in a condensing turbine. The combined cycle thus obtained is a particularly successful marriage in the search for improved thermal performance: with currently available machines,

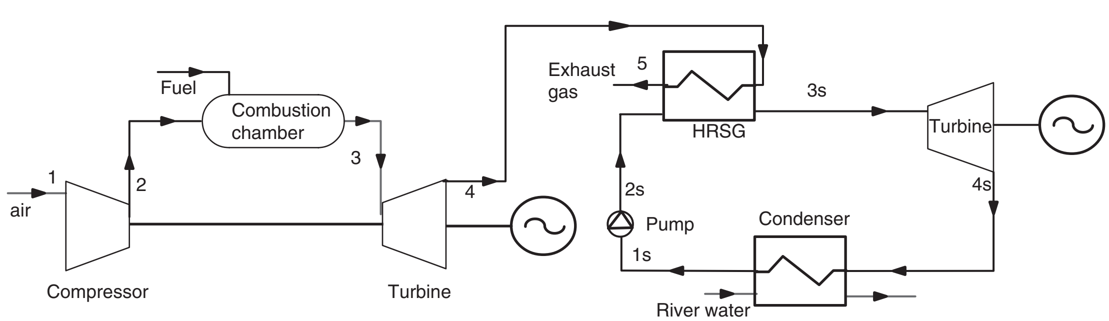

FIGURE 17.1.1
Sketch of a combined cycle

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efficiencies exceed 55% and are higher than those we can hope, even in the medium term, of the most advanced future steam plants.

As we shall see next section, in a simple combined cycle of the type described below, the gas turbine provides two-thirds of the total capacity. The steam turbine, fueled by superheated steam conditions of 85 to 100 bar and 510−540°C, provides the remaining third.

17.1 COMBINED CYCLE WITHOUT AFTERBURNER

The simplest combined cycle (that is to say, without afterburner) is shown in Figure 17.1.1: as the temperature of the gas turbine exhaust gas can exceed 550°C, the maximum temperature level reached in a steam cycle, it is quite possible to recover the enthalpy available at the output of a gas turbine to heat a steam cycle.

With some simplifying assumptions, it is possible to construct an entropy chart allowing, for a set of suitable scales, to superimpose the two thermodynamic cycles (Figure 17.1.2). In this diagram, where the work done is proportional to the area of the cycle, the gas turbine provides more power than the steam engine (two-thirds of the total in practice).

We can sometimes improve the cycle efficiency by using the various changes discussed during the presentation of the steam cycle: reheat and extractions.

However, as discussed below, the problem of steam cycle optimization differs substantially from that of large steam power plants, due to the pinch that appears in the heat recovery steam generator (HRSG).

17.1.1 Overall performance

The enthalpy exchange in a combined cycle can be summarized by the diagram in Figure 17.1.3.

- the gas turbine receives heat Q_g from the hot source. It provides on the one hand a useful work τ_g, and secondly a heat $(Q_v + Q_p)$. The first term is the heat supplied to the steam cycle, the second losses;
- the steam cycle produces useful work τ_v, and the condenser rejects heat Q_c.

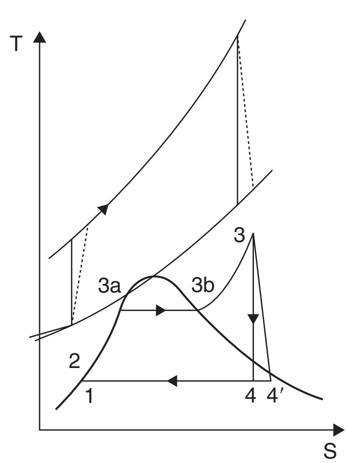

FIGURE 17.1.2
Combined cycle in the entropy chart

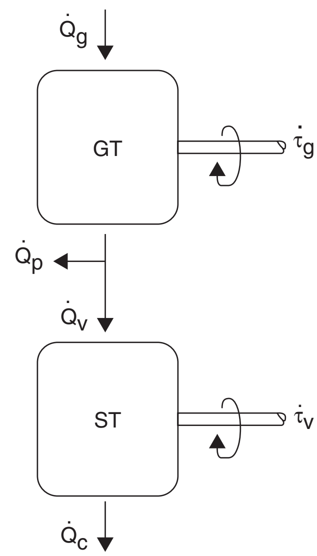

FIGURE 17.1.3
Block diagram of a combined cycle

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Let us call η_g the gas turbine efficiency, η_v that of the steam cycle, η_{cc} that of the combined cycle, and ε the HRSG effectiveness, that is to say the ratio of Q_v to $Q_p + Q_v$:

$$\varepsilon = \frac{Q_v}{Q_p + Q_v} = \frac{Q_g}{Q_p + Q_v}\frac{Q_v}{Q_g} = \frac{1}{1-\eta_g}\frac{Q_v}{Q_g}$$

$$\eta_{cc} = \frac{\tau_g + \tau_v}{Q_g} = \eta_g + \eta_v\frac{Q_v}{Q_g} = \eta_g + \varepsilon(1-\eta_g)\eta_v$$

$$\eta_{cc} = \eta_g + \varepsilon(1 - \eta_g)\eta_v \tag{17.1.1}$$

The combined cycle efficiency is equal to the sum of that of the gas turbine and the product of its complement to 1 by the HRSG effectiveness and the steam cycle efficiency.

For example, with $\eta_g = 0.29$, $\eta_v = 0.32$, $\varepsilon = 0.83$, we obtain $\eta_{cc} = 0.48$.

17.1.2 Reduced efficiency and power

The reduced power introduced section 12.1.4.2 can also be expressed here as:

$$W_0 = \eta_{cc}\frac{\theta - r^{\beta_c}}{\theta - 1} \tag{17.1.2}$$

Assuming at first approximation that the steam cycle efficiency varies linearly with gas turbine exhaust temperature, we obtain in terms of overall efficiency and power the abacus shown in Figure 17.1.4. We can recognize the lower left part corresponding to the gas turbine alone (Figure 12.1.17). The contribution of the steam cycle is particularly significant: 50 to 60% more capacity and efficiency gains of 30–50% depending on temperature and compression ratio.

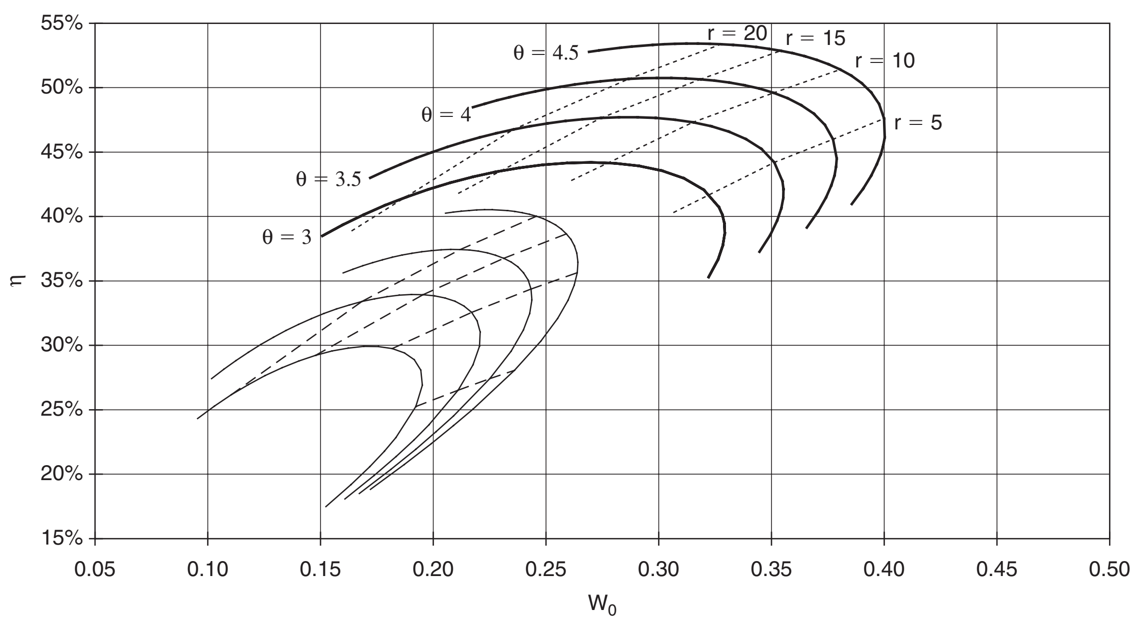

FIGURE 17.1.4
Efficiency and useful work of a combined cycle

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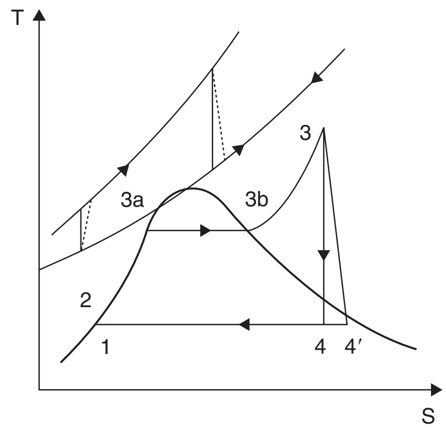

FIGURE 17.2.1
Combined cycle with afterburner

17.2 COMBINED CYCLE WITH AFTERBURNER

It is also possible to perform an afterburning of gas turbine exhaust, to have more power in the steam cycle, and especially to better control the combined cycle. This is called combined cycle with afterburner (Figure 17.2.1). The efficiency drops slightly because of course the heat generated by the afterburner is not valued in the GT. In this case, the total power is divided about equally between each machine.

This type of cycle is mainly appropriate when converting conventional steam plants in combined cycle plants, by adding a topping gas cycle. The conventional plant having its boiler, afterburning can be achieved without needing large investment.

17.3 COMBINED CYCLE OPTIMIZATION

Thermodynamic analysis of combined cycle without afterburner showed that overall performance is given by equation (17.1.1): $\eta_{cc} = \eta_g + \varepsilon(1 - \eta_g)\eta_v$

This expression shows that it is as important to optimize the steam cycle as the recovery steam generator, and thus its effectiveness ε. Difficulties arise because the problem is highly constrained and there may be conflicts between these two objectives.

We have already given some indication in section 15.6.2 on the problem of optimizing the HRSG. In particular, we saw that the gas discharge temperature in the atmosphere must be high enough to avoid condensation. But as we also seek to reduce them as much as possible to recover the enthalpy available, we see that in general it is unnecessary to perform high extraction on this type of steam plant, gains remaining low.

The optimization of such a combined cycle is based on the reduction of its internal irreversibilities, which can be grouped into three broad categories: mechanical irreversibilities, that take place in the compressor and turbines, combustion irreversibilities, and purely thermal irreversibilities, related to temperature differences in the heat exchangers.

Much has already been done to limit the mechanical irreversibilities, and reducing combustion irreversibility is directly related to the maximum temperature of the fumes, which itself depends on the strength of the combustion chamber materials and above all of the initial expansion stages in the gas turbine (stator and rotor).

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So we focus in what follows only on the reduction of thermal irreversibilities, i.e. on the optimization of plants whose turbine outlet temperature is fixed. These irreversibilities result from differences in temperature between the hot and cold parts of the cycle.

In cogeneration plants (CHP) studied in Chapter 18, problems arise in a similar manner, especially if steam needs at medium and high pressure are important.

In a combined cycle plant, the vein of hot gases exiting the gas turbine must be cooled by water of the steam recovery cycle. In a single pressure cycle, water enters the heat exchanger in the liquid state at about 30°C after being compressed by the feedwater pumps downstream of the condenser. It is heated at the boiling temperature corresponding to its pressure (economizer), then vaporized at constant temperature and superheated before being expanded in the steam turbine. The diagram in Figure 15.6.7 shows the heat transfer within the heat exchanger between hot gases and water. The associated enthalpy diagram shows that if we set for technical reasons a pinch minimum value (temperature difference between both fluids) between points 6 and 9 on the one hand, and between points 4 and 11 on the other hand, heat exchanges take actually place with much larger differences in nearly all of the heat exchanger. This stems from the need to vaporize water, which induces a very important "plateau" at a constant temperature.

The example in Figure 17.3.1 corresponds to such a combined cycle. Hot gases exit the gas turbine at 559°C and maximum pressure of the steam cycle is equal to 120 bar. In these circumstances it is impossible to cool gases below 169°C, which represents a significant loss.

WORKED EXAMPLE

Modeling of a combined cycle in Thermoptim

This example shows how to model with Thermoptim a combined cycle and fit the steam cycle flow rate knowing the one which flows through the gas turbine. It is presented in the Diapason session S41En (http://www.thermoptim.org/sections/enseignement/cours-en-ligne/seances-diapason/session-s41en-single). **[CRC_pa_4]**

The combined cycle exergy balance can be built as explained in section 10.2.3 of Part 2, resulting in Table 17.3.1.

The exergy efficiency reaches 45%, the main irreversibilities being still located in the combustion chamber (56%). Other losses are spread fairly evenly among the various components and exhaust. In descending order, they take place in the air compressor, gas turbine, steam turbine, economizer, evaporator, superheater, exhaust and condenser.

Irreversibilities in turbomachines represent 22% of the total. Manufacturers continue in their efforts to reduce them, and very significant progress has already been made in recent decades. Therefore opportunities for improvement are becoming fewer.

Losses in the HRSG and those in the exhaust are linked, as we have already seen. They represent 12.7% and 29% of losses outside the combustion chamber. Their reduction is therefore an important issue. The ideal heat exchange corresponds to the case where the curve of gas cooling and that water of heating would be parallel. The heat exchanger would then operate in counter-flow and irreversibility would be minimal. This is not feasible with water, and the single pressure cycle has strong internal irreversibilities.

To improve the cycle performance, we use multiple steam cycles at different pressure levels (two, three or even four). Figure 17.3.2 shows the value of using multi-level pressure: with some simplifying assumptions and a choice of scales, we can superimpose on an entropy chart gas turbine and steam power plant cycles. In all three cases, the grey surface represents work provided for the same heat input in the gas turbine. The rectangle in the dashed line is the Carnot cycle.

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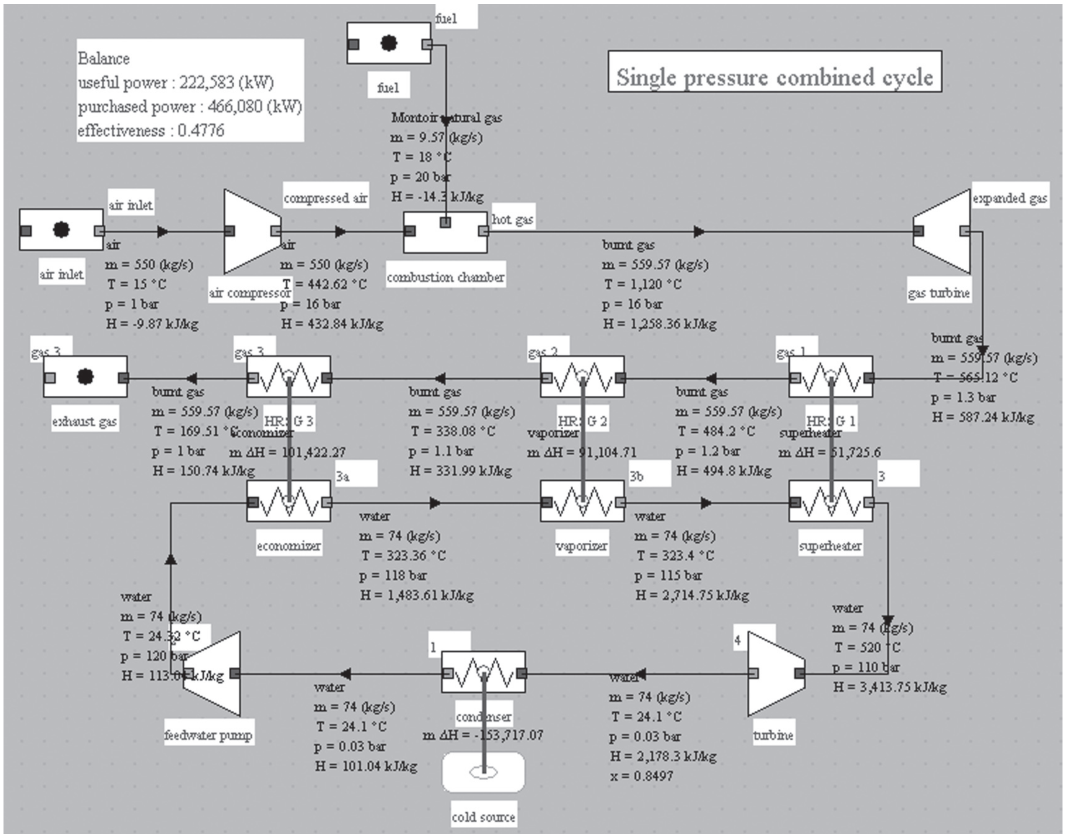

FIGURE 17.3.1

Synoptic view of a single pressure combined cycle

TABLE 17.3.1

EXERGY BALANCE OF SINGLE PRESSURE COMBINED CYCLE

Exergy balance process	$T_0 = 288.15\,K = 15°C$ τ (MW)	Q (MW)	$m\Delta h$ (MW)	T_k (K)	Δx_q (MW)	$m\Delta x_h$ (MW)	Δx_{hi} (MW)	% loss
HRSG 3		−101	−101			−50	15	5.96%
Economizer			101	101			35	
HRSG 2		−91	−91			−57	10	3.80%
Vaporizer		91	91			47		
HRSG 1		−52	−52			−37	8	2.93%
Superheater		52	52			29		
Feedw. pump	1		1			1		
Steam turbine	−91		−91			−107	16	6.09%
Condenser		−154	−154	288	0	−5	5	1.83%
Air compr.	243		243			221	22	8.66%
Gas turb.	−376		−376			−394	19	7.36%
Comb. ch.			466		**466**	323	143	55.75%
Asp/exhaust						−20	20	7.63%
Cycle	−223	312	90			−12	**257**	100.0%
Sigma(x_q+)			**466**					
Sigma(tau+)			**0.00**					
Exergy efficiency			45%					

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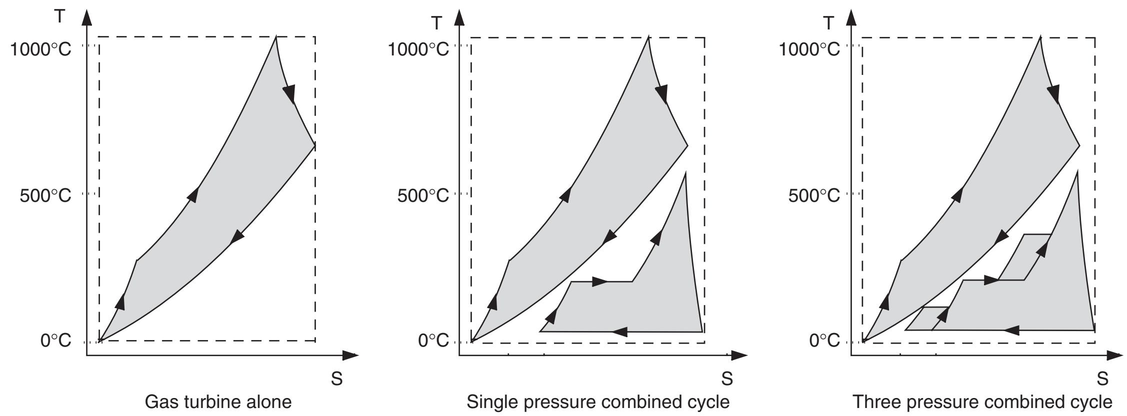

FIGURE 17.3.2
Comparison of work provided by a gas turbine and combined cycles

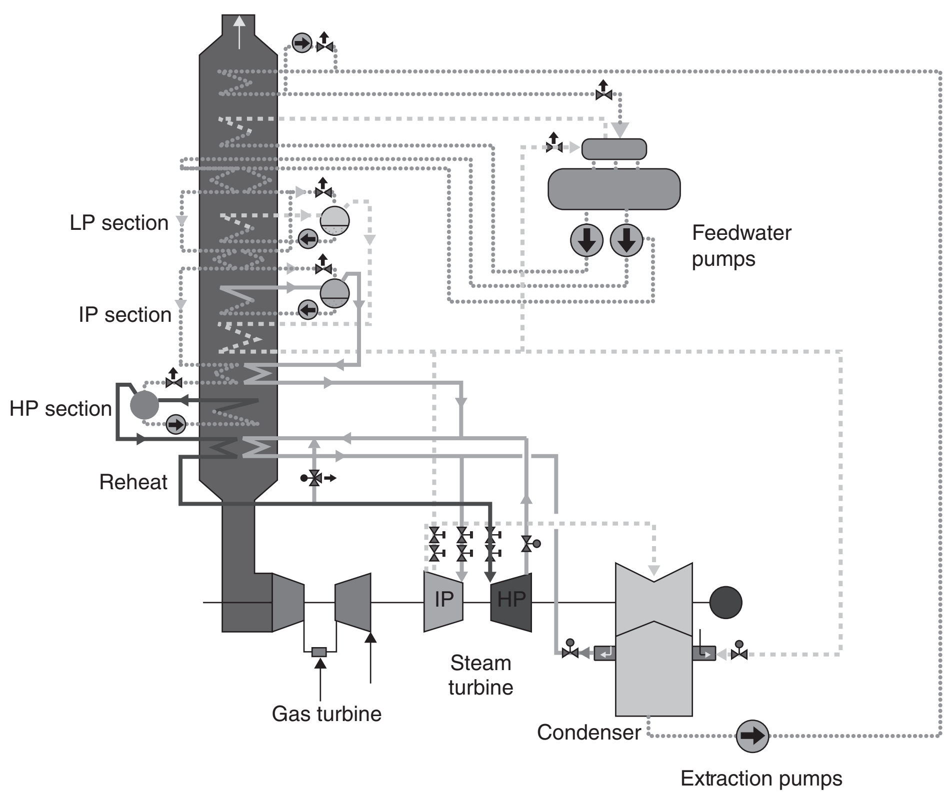

FIGURE 17.3.3
Three pressure HRSG, Documentation Alstom Power

The left figure corresponds to a single gas turbine, the middle to a single pressure combined cycle, and the right to a three pressure combined cycle. The gain provided by the increased number of pressure levels is very clear, but designing the HRSG is a complex optimization problem and quite new, which does not arise in conventional boilers.

The optimization of such cycles is a complex problem, because to get the better cooling of the hot gas stream, there are many degrees of freedom on the pressure levels, on the corresponding flow rates, and on placement of heat exchangers (in series or in parallel). Figure 17.3.3 gives an example

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FIGURE 17.3.4
Supercritical mono-tubular HRSG, Documentation Alstom Power

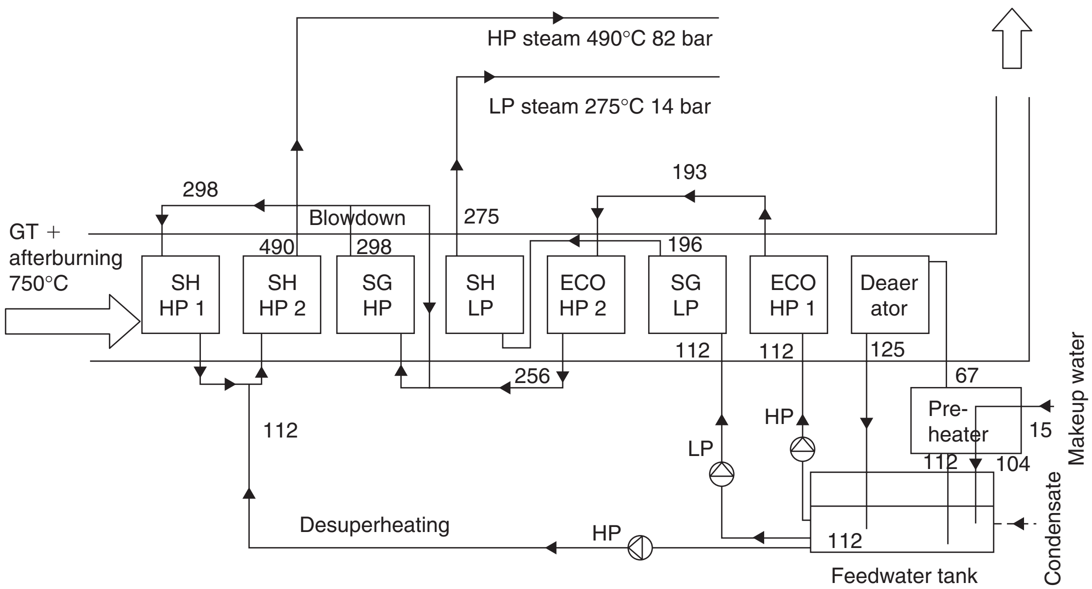

FIGURE 17.3.5
Exchanger arrangement of a dual pressure HRSG

of an industrial three pressure HRSG, with the whole combined cycle. Figure 17.3.4 shows the look of a supercritical mono-tubular HRSG and Figure 17.3.5 shows the exchanger arrangement of a dual pressure HRSG for a non-optimized cogeneration unit.

This optimization problem is quite new. It did not arise in old power plants, in which very large irreversibilities occurred for technical and economic reasons related to the thermal resistance of steel boilers and sulfur content of fumes. There is no proven method to solve it.

The optimization method presented in Chapter 22, known as Systems Integration, is the extension to the case of power generation or cogeneration plants, of the pinch method developed in the context of chemical engineering to optimize the configuration of very large exchanger networks (such as those of a refinery).

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The example presented section 12.6 shows in particular how this method can be implemented to size a dual pressure combined cycle.

17.4 GAS TURBINE AND COMBINED CYCLES VARIATIONS

Other cycles, more complex to study, have been proposed and are the subject of various studies and experiments. They will be presented in Chapter 24 of Part 4. Two of them are particularly interesting and can yield high efficiencies, because the heat exchange with the exhaust gases is made with very low irreversibility: the humid air cycle and the Kalina cycle.

In the humid air cycle, the air leaving the compressor is humidified in a saturator with water heated by exhaust gases. It leaves the saturator cooled and is then preheated in a regenerator before being directed to the combustion chamber and then into the turbine. This cycle looks a little like a steam injection cycle, but its performance is much better.

The Kalina cycle is a combined cycle variation where water is replaced by a water-ammonia mix which vaporizes and condenses with a temperature glide.

Furthermore, many variations are possible in combined cycles: for example it is possible in the gas turbine, to make an intermediate cooling during compression, or a reheat. More complex cycles are under investigation. Their detailed presentation is beyond the scope of this book: we limit ourselves to some thermodynamic considerations on a key combined cycle component: the recovery steam generator. For other developments, the reader may refer to the thesis of H. Abdullah (1988) referenced at the end of Chapter 12.

Finally, we will study in the next chapter cogeneration plants, for simultaneous production of mechanical power and heat, among which will appear some variants of the combined cycles considered above.

17.5 DIESEL COMBINED CYCLE

We discussed in section 13.9.2.1 the rise of temperatures in diesel engines. This has achieved the following gains, particularly suitable for use in combined cycle:

- mechanical efficiency increased from 45% to 47%;
- enthalpy of exhaust gas increased from 27% to 32%;
- cooling losses reduced from 24% to 16%.

Today, a less than 100 MW diesel combined cycle based on medium-speed diesel engine reaches an overall efficiency of 55%, which makes it competitive with gas turbines in this capacity range.

17.6 CONCLUSIONS AND OUTLOOK

The very high efficiencies provided by combined cycles explain the enthusiasm for these plants, and the rapid development of their market: one of the largest such plant in operation, Futtsu in Japan, has an installed capacity of 2,000 MW.

Among the many benefits of combined cycles, we can mention:

- as for gas turbines, combined cycles are normally designed standardized and modular, so that different components are factory built and assembled quickly on site, while a steam power plant must be calculated and built case by case;
- thus, combined cycles have almost no effect of scale. It is not necessary to construct a single unit of large capacity: we can start with small units, and add others as demand grows;

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- due to the excellent efficiencies that are achieved and also the use of fuels with low sulfur and nitrogen, the environmental impact of these technologies is much lower than that of their competitors: CO_2 emissions are only equal to 40% of those of coal steam plants, and they require three times less water cooling;
- in the same way, the combined cycle footprint is close to 80 m^2/MW, against about 200 m^2/MW for a steam power plant. It is therefore easier to locate them close to consumption areas.

One major limitation is that gas turbines require the use of clean fuel (expensive), such as natural gas or light distillates, which excludes the use of heavy fuel oil or coal, traditional basic fuels for power plants. However, development of coal gasification would allow this energy source to fuel these efficient combined cycles.

In the coming years, efficiencies should rise from 50 to 60%, and prices fall further, thereby increasing the economic competitiveness of these machines.

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FURTHER READING

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