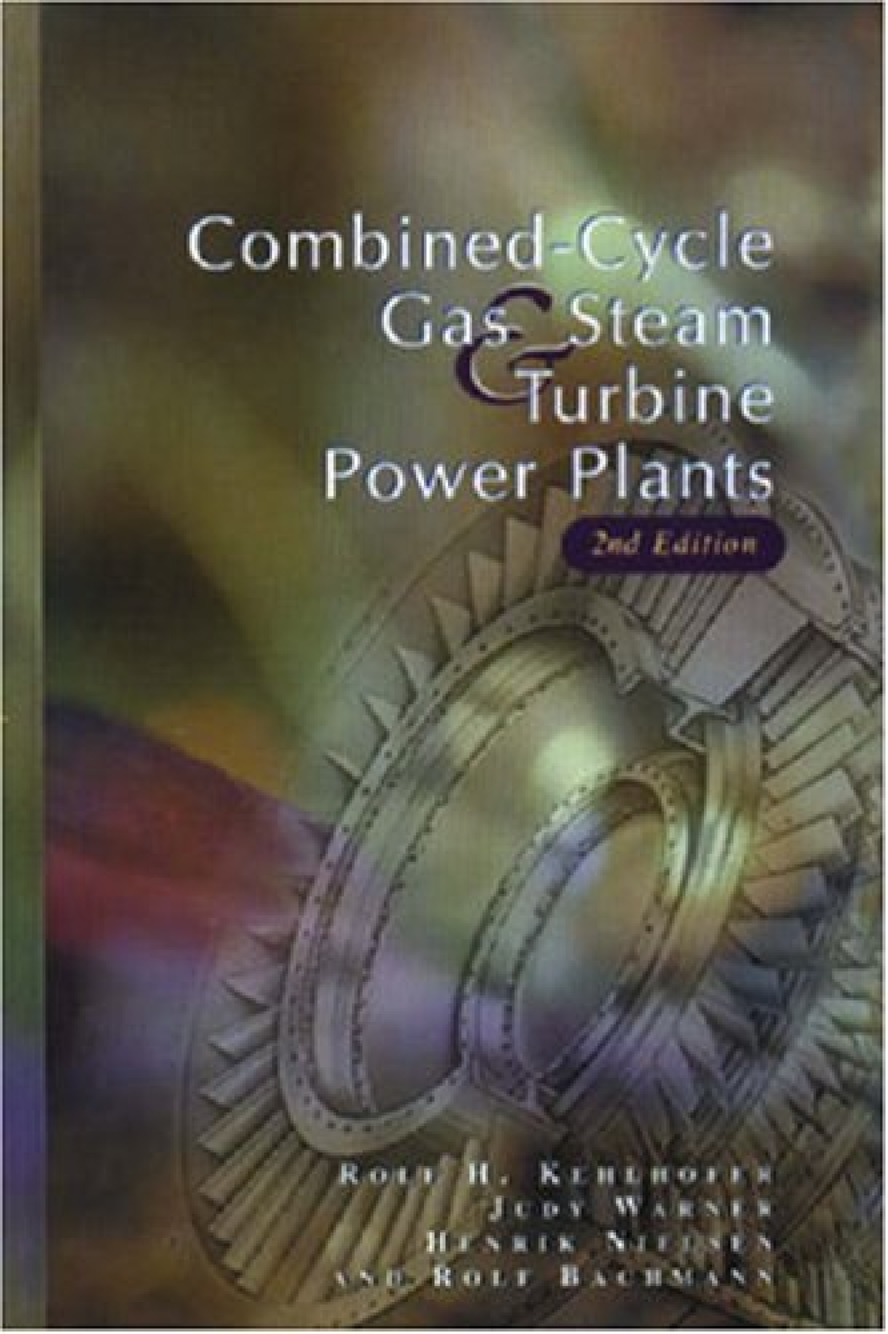
Combined-Cycle
Gas & Steam
Turbine
Power Plants
2nd Edition
Rolf H. Kehlhofer
Judy Warner
Henrik Nielsen
and Rolf Bachmann

COMBINED-CYCLE GAS & STEAM TURBINE POWER PLANTS

Rolf Kehlhofer

PennWell

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CONTENTS

Chapter 1

INTRODUCTION

The literature has often suggested combining two or more thermal cycles within a single power plant. In all cases, the intention was to increase efficiency over that of single cycles. Thermal processes can be combined in this way whether they operate with the same or with differing working media. However, a combination of cycles with different working media is more interesting because their advantages can complement one another.

Normally the cycles can be classed as a "topping" and a "bottoming" cycle. The first cycle, to which most of the heat is supplied, is called the "topping cycle." The waste heat it produces is then utilized in a second process which operates at a lower temperature level and is therefore referred to as a "bottoming cycle."

Careful selection of the working media makes it possible to create an overall process that makes optimum thermodynamic use of the heat in the upper range of temperatures and returns waste heat to the environment at as low a temperature level as possible. Normally the "topping" and "bottoming" cycles are coupled in a heat exchanger.

Up to the present time, only one combined cycle has found wide acceptance: the combination gas turbine/steam turbine power plant. So far, plants of this type have burned generally fossil fuels (principally liquid fuels or gases.)

Fig. 1 is a simplified flow diagram for an installation of this type, in which an open-cycle gas turbine is followed by a steam process. The heat given off by the gas turbine is used to generate steam.

Other combinations are also possible, e.g., a mercury vapor process or replacing the water with organic fluids or ammonia.

The mercury vapor process is no longer of interest today since even conventional steam power plants achieve higher efficiencies. Organic fluids or ammonia have certain advantages over water in the low temperature range, such as reduced volume flows, no wetness. However, the disadvantages, i.e., development costs, environmental impact, etc., appear great enough to prevent their ever replacing the steam process in a combined-cycle power plant. The discussion that follows deals mainly with the combination of an open-cycle gas turbine with a water/steam cycle. Certain special applications using closed-cycle gas turbines will also be dealt with briefly.

Why has the combination gas turbine/steam turbine power plant, unlike other combined-cycle power plants, managed to find wide acceptance? Two main reasons can be given:

- It is made up of components that have already proven themselves in power plants with a single cycle. Development costs are therefore low.
- Air is a relatively non-problematic and inexpensive medium that can be used in modern gas turbines at an elevated temperature level (above 1000 °C). That provides the optimum prerequisites for a good "topping cycle."

The steam process uses water, which is likewise inexpensive and widely available, but better suited for the medium and low temperature ranges. The waste heat from a modern gas turbine has a temperature level advantageous for a good steam process.

It therefore is quite reasonable to use the steam process for the "bottoming cycle." That such combination gas turbine/steam turbine power plants were not more widely used even earlier has clearly been due to the historical development of the gas turbine. Only in recent years have gas turbines attained inlet temperatures that make it possible to design a very high-efficiency cycle. Today, however, the installed power capacity of combined-cycle gas turbine/steam turbine power plants worldwide world totals more than 30,000 MW.

Figure 1-1

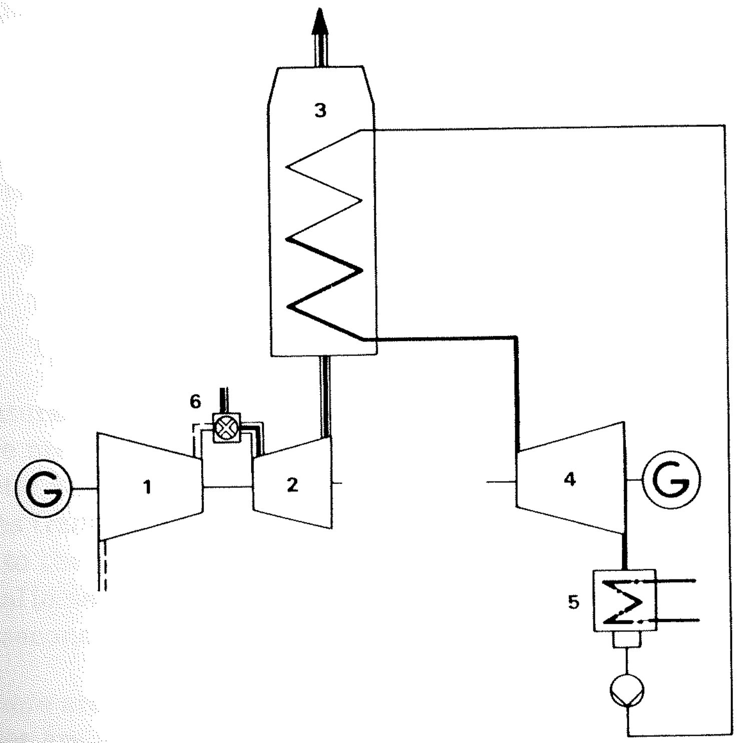

Fig. 1-1: Simplified flow diagram of a combination gas turbine/steam turbine power plant

1. Compressor
2. Gas turbine
3. Steam generator
4. Steam turbine
5. Condenser
6. Fuel supply

Chapter 2

THERMODYNAMIC PRINCIPLES OF THE COMBINED-CYCLE PLANT

2.1 Basic Considerations

The Carnot efficiency is the maximum efficiency of an ideal thermal process:

$$\eta_C = \frac{T_W - T_K}{T_W} \tag{1}$$

Here,

η_C = Carnot efficiency
T_W = Temperature of the energy supplied
T_K = Temperature of the environment

Naturally, the efficiencies of real processes are lower since there are losses involved. A distinction is drawn between energetic and exergetic losses. Energetic losses are mainly heat losses (radiation and convection), and are thus energy that is lost to the process. Exergetic losses, on the other hand, are internal losses caused by irreverisible processes in accordance with the second law of thermodynamics [1].

There are two major reasons why the efficiencies of real processes are lower than the Carnot efficiency:

First, the temperature differential in the heat being supplied to the cycle is very great. In a conventional steam power plant, for example, the maximum steam temperature is only about

810K (980°F), while the combustion temperature in the boiler is approx. 2000 K. Then, too, the temperature of the waste heat from the process is higher than the ambient temperature. Both heat exchange processes cause losses.

The best way to improve the process efficiency is to reduce these losses, which can be accomplished by raising the maximum temperature in the cycle, or by releasing the waste heat at as low a temperature as possible.

The interest in combined-cycles arises particularly from these two considerations. By its nature, no single cycle can make both improvements to an equal extent. It thus seems reasonable to combine two cycles: one with high process temperatures, and the other with a good cold end.

In an open-cycle gas turbine, the process temperatures attainable are very high because its energy is supplied directly to the cycle without heat exchangers. The exhaust heat temperature, however, is also quite high. In the steam cycle, the maximum process temperature is not very high, but the exhaust heat is returned to the environment on the cold end at a very low temperature.

Combining a gas turbine and a steam turbine thus offers the best possible basis for a high-efficiency thermal process (Table 2-1).

The last line in the table shows the ''Carnot efficiencies'' of the various processes, i.e., the efficiencies that would be attainable if the processes took place without internal exergetic losses. Although that naturally is not the case, this figure can be used as an indicator of the quality of a thermal process. The value shown makes clear just how interesting the combined-cycle power plant is when compared to the single-cycle processes. Even a sophisticated installation such as a reheat steam turbine power plant has a theroretical Carnot efficiency 10 to 15 points lower

than that of a combined-cycle plant. On the other hand, the exergetic losses in the combined cycle are higher because the temperature differential for exchanging heat between the exhausts from the gas turbine and the water/steam cycle is relatively great. It is thus clear why the differences between the actual efficiencies attained by a combined-cycle power plant and the other processes are not quite that large.

As shown by Fig. 2-1, which compares the temperature/entropy diagrams of the four processes, the combined cycle best utilizes the temperature differential in the heat supplied, even though there is an additional exergetic loss between the gas and the steam processes.

Table 2-1: Thermodynamic Comparison of Gas Turbine, Steam Turbine, and Combined-Cycle Power Plants

	Gas Turbine	Steam Power Plant with Reheat	Steam Power Plant without Reheat	Combined-Cycle Power Plant
Average temperature of the heat supplied, in K (in °F)	950 - 1000 (1250 - 1340)	640 - 700 (690 - 800)	550 - 630 (530 - 675)	950 - 1000 (1250 - 1340)
Average temperature of exhaust heat, in K (in °F)	500 - 550 (440 - 530)	320 - 350 (115 - 170)	320 - 350 (115 - 170)	320 - 350 (115 - 170)
Carnot efficiency, in %	42 - 47	45 - 54	37 - 50	63 - 68

Figure 2-1

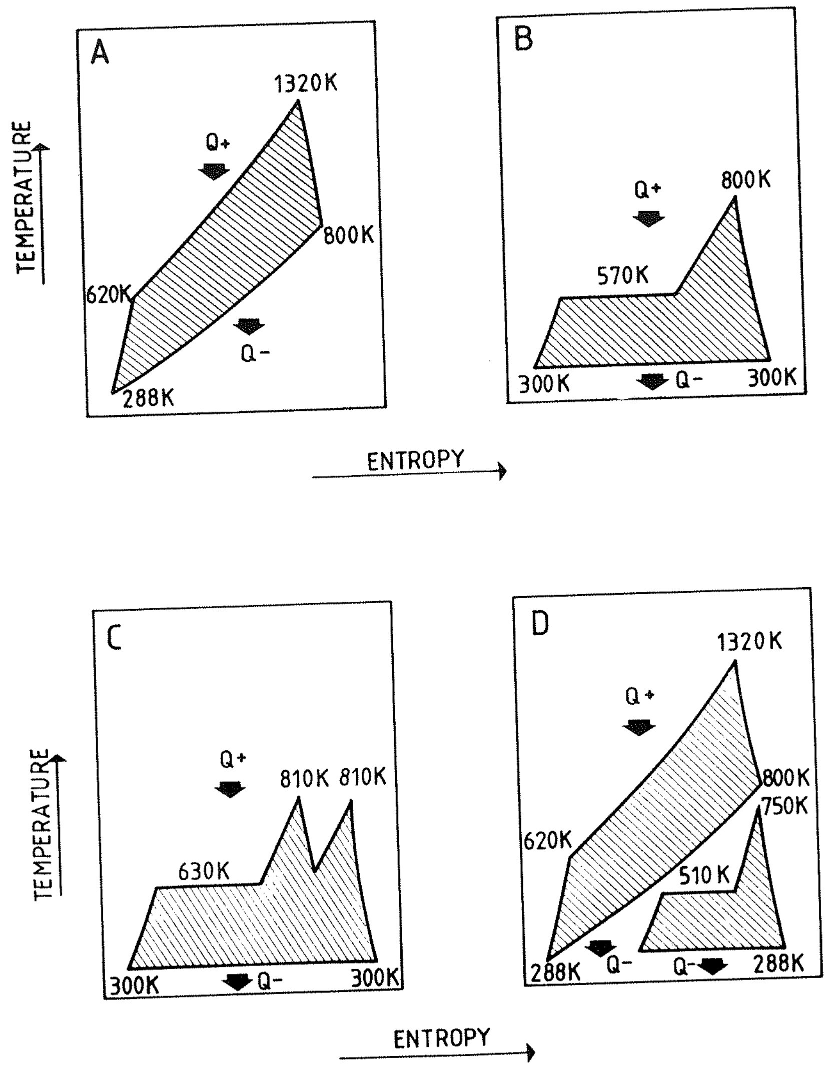

Fig. 2-1: Temperature/ Entropy Diagrams

A. Gas turbine
B. Steam turbine without reheat
C. Reheat steam turbine
D. Combined-cycle gas turbine/steam turbine power plant

2.2 Thermal Efficiency of the Combined-Cycle Plant

It was assumed in Section 2.1 that fuel energy is being supplied only in the gas turbine. There are, however, also combined-cycle installations with additional firing in the steam generator, i.e., in which a portion of the heat is supplied directly to the steam process.

Accordingly, the general definition of the thermal efficiency of a combined-cycle plant is:

$$\eta_K = \frac{P_{GT} + P_{ST}}{\dot{Q}_{GT} + \dot{Q}_{SF}} \tag{2}$$

If there is no supplementary firing in the waste heat boiler (heat supplied $\dot{Q}_{SF} = 0$), this formula simplifies into:

$$\eta_K = \frac{P_{GT} + P_{ST}}{\dot{Q}_{GT}} \tag{3}$$

In the general case, the efficiencies of the single cycles can be defined as follows:

- for the gas turbine process:

$$\eta_{GT} = \frac{P_{GT}}{\dot{Q}_{GT}} \tag{4}$$

- for the steam turbine process:

$$\eta_{ST} = \frac{P_{ST}}{\dot{Q}_{SF} + \dot{Q}_{Exh}} \tag{5}$$

$$\dot{Q}_{Exh} \cong \dot{Q}_{GT}(1 - \eta_{GT}) \tag{6}$$

Combining these two equations yields:

$$\eta_{ST} = \frac{P_{ST}}{\dot{Q}_{SF} + \dot{Q}_{GT}(1 - \eta_{GT})} \tag{7}$$

2.2.1 The Effect of Additional Firing in the Waste Heat Boiler on Overall Efficiency

Substituting Equations (4) and (7) into Equation (2), one obtains:

$$\eta_K = \frac{\eta_{GT}\,\dot{Q}_{GT} + \eta_{ST}\,(\dot{Q}_{SF} + \dot{Q}_{GT}\,[1 - \eta_{GT}])}{\dot{Q}_{GT} + \dot{Q}_{SF}} \tag{8}$$

Additional firing in the waste heat boiler improves the overall efficiency of the combined-cycle installation whenever:

$$\frac{\partial\,\eta_K}{\partial\,\dot{Q}_{SF}} > 0 \tag{9}$$

Differentiation of Equation (8) produces the inequality:

$$\frac{\partial\,\eta_K}{\partial\,\dot{Q}_{SF}} = \frac{1}{(\dot{Q}_{GT} + \dot{Q}_{SF})^2}\left\{ \eta_{GT}\cdot\dot{Q}_{GT}\left(\frac{\partial\,\eta_{ST}}{\partial\,\dot{Q}_{SF}}\,\dot{Q}_{SF} + \eta_{ST}\right)\right.$$

$$\cdot\,(\dot{Q}_{GT} + \dot{Q}_{SF}) - \eta_{ST}\,\dot{Q}_{SF} + \left[\frac{\partial\,\eta_{ST}}{\partial\,\dot{Q}_{SF}}\,\dot{Q}_{GT}\,(1 - \eta_{GT})\right]\cdot \tag{10}$$

$$\left.\cdot\,(\dot{Q}_{GT} + \dot{Q}_{SF}) - \eta_{ST}\,\dot{Q}_{GT}\,(1 - \eta_{GT})\right\} > 0$$

This yields:

$$\frac{\partial\,\eta_{ST}}{\partial\,\dot{Q}_{SF}}\,[\dot{Q}_{SF} + \dot{Q}_{GT}\,(1 - \eta_{GT})] + \eta_{ST} > \frac{\eta_{GT}\,\dot{Q}_{GT} + \eta_{ST}\,[\dot{Q}_{SF} + \dot{Q}_{GT}\,(1 - \eta_{GT})]}{\dot{Q}_{GT} + \dot{Q}_{SF}} \tag{11}$$

Since the second term of the inequality is equal to K, the inequality reduces to:

$$\frac{\partial\,\eta_{ST}}{\partial\,\dot{Q}_{SF}}\,[\dot{Q}_{SF} + \dot{Q}_{GT}\,(1 - \eta_{GT})] > \eta_K - \eta_{ST} \tag{12}$$

The term $[\dot{Q}_{SF} + \dot{Q}_{GT}\,(1 - \eta_{GT})]$ is none other than the heat input to the steam cycle. The formula thus becomes:

$$\frac{\partial\,\eta_{ST}}{\partial\,\dot{Q}_{SF}}\cdot\frac{P_{ST}}{\eta_{ST}} > \eta_K - \eta_{ST} \tag{13}$$

Equation (13) means that increasing the additional firing improves the efficiency of the combined-cycle plant only if it improves the efficiency of the steam process. The greater the difference is between the efficiencies of the combined-cycle and the steam process, and the lower the temperature is of the heat input to the steam process, the more effective that improvement will be. For that reason, additional firing is becoming less and less interesting: the efficiency of the combined-cycle installation increases far more rapidly than that of the steam process, continually increasing the difference $(\eta_K - \eta_{ST})$. In view of the considerations in Section 2.1, it is generally better to burn the fuel in a modern gas turbine, because the heat is supplied to the process at a temperature level higher than that in the steam process.

The problems involved in combined-cycle installations with additional firing are discussed in more detail in Section 3.2 below.

2.2.2 Efficiency of Combined-Cycle Plants without Additional Firing in the Waste Heat Boiler

Without additional firing, Equation (8) can be written as follows: (14)

$$\eta_K = \frac{\eta_{GT} \cdot \dot{Q}_{GT} + \eta_{ST} \cdot \dot{Q}_{GT}\,(1 - \eta_{GT})}{\dot{Q}_{GT}} = \eta_{GT} + \eta_{ST}\,(1 - \eta_{GT})$$

Differentiation makes it possible to estimate the effect that a change in efficiency of the gas turbine has on overall efficiency:

$$\frac{\partial\,\eta_K}{\partial\,\eta_{GT}} = 1 + \frac{\partial\,\eta_{ST}}{\partial\,\eta_{GT}}\,(1 - \eta_{GT}) - \eta_{ST} \tag{15}$$

Increasing the gas turbine efficiency improves the overall efficiency only if:

$$\frac{\partial\,\eta_K}{\partial\,\eta_{GT}} > 0 \tag{16}$$

From Equation (15) one obtains:

$$-\frac{\partial\eta_{ST}}{\partial\eta_{GT}} < \frac{1 - \eta_{ST}}{1 - \eta_{ST}} \tag{17}$$

Improving the gas turbine efficiency is helpful only if it does not cause too great a drop in the efficiency of the steam process.

Table 2-2 shows the maximum allowable reduction— $\frac{\partial\eta_{ST}}{\partial\eta_{GT}}$ as a function of the gas turbine efficiency.

This table indicates that the higher the efficiency of the gas turbine, the greater may be the reduction in efficiency of the steam process. The proportion of the overall output being provided by the gas turbine increases, reducing the effect of lower efficiency in the steam cycle. But a gas turbine with a maximum efficiency still does not provide an optimum combined-cycle plant. For example— with a constant turbine inlet temperature— a gas turbine with a very high pressure ratio attains a higher efficiency that a machine with a moderate pressure ratio. However, the efficiency of the combined-cycle plant with the second machine is significantly better because the steam turbine that follows operates far more efficiently with the higher exhaust gas temperature and produces a greater output.

Table 2-2: Allowable Reduction in Steam Process Efficiency as a Function of Gas Turbine Efficiency (Steam process efficiency = 0.25)

η_{GT}	0.2	0.3	0.4
$-\frac{\partial\eta_{ST}}{\partial\eta_{GT}}$	0.94	1.07	1.25

Fig. 2-2a shows the efficiency of the gas turbine alone as a function of the turbine inlet and exhaust temperatures. The maximum efficiency is reached when the exhaust gas temperatures are quite low. (A low exhaust temperature means a high pressure ratio.)

Fig. 2-2b shows the overall efficiency of the combined-cycle in the same way. Compared to Fig. 2-2a, the optimum point has shifted toward higher exhaust temperatures from the gas turbine. Due to economical considerations, present-day gas turbines are generally optimized with respect not to efficiency but to maximum power density. Fortunately, this optimum coincides fairly accurately with the optimum efficiency of the combined-cycle plant. As a result— most of today's gas turbines are optimally suited for combined-cycle installations.

Gas turbines of a more complicated design, i.e., with intermediate cooling in the compressor or recuperator, are less suitable for combined cycles. They normally have low exhaust gas temperatures, so that the efficiency of the steam turbine can only be low. We shall not discuss a reheat gas turbine here since this type of machine has disappeared from the market due to its complexity.

In summary, it may be said that:

The gas turbine with the highest efficiency does not necessarily produce the best overall efficiency of the combined-cycle plant. The turbine inlet temperature is a far more important factor.

Similar considerations also apply with regard to the efficiency of the steam cycle. These, however,are less important because the gas turbine is generally the "standard machine." The exhaust heat available for the steam process is thus a given, and the problem lies only in its maximum conversion into mechanical energy (refer on this point to Section 2.3.)

Figure 2-2

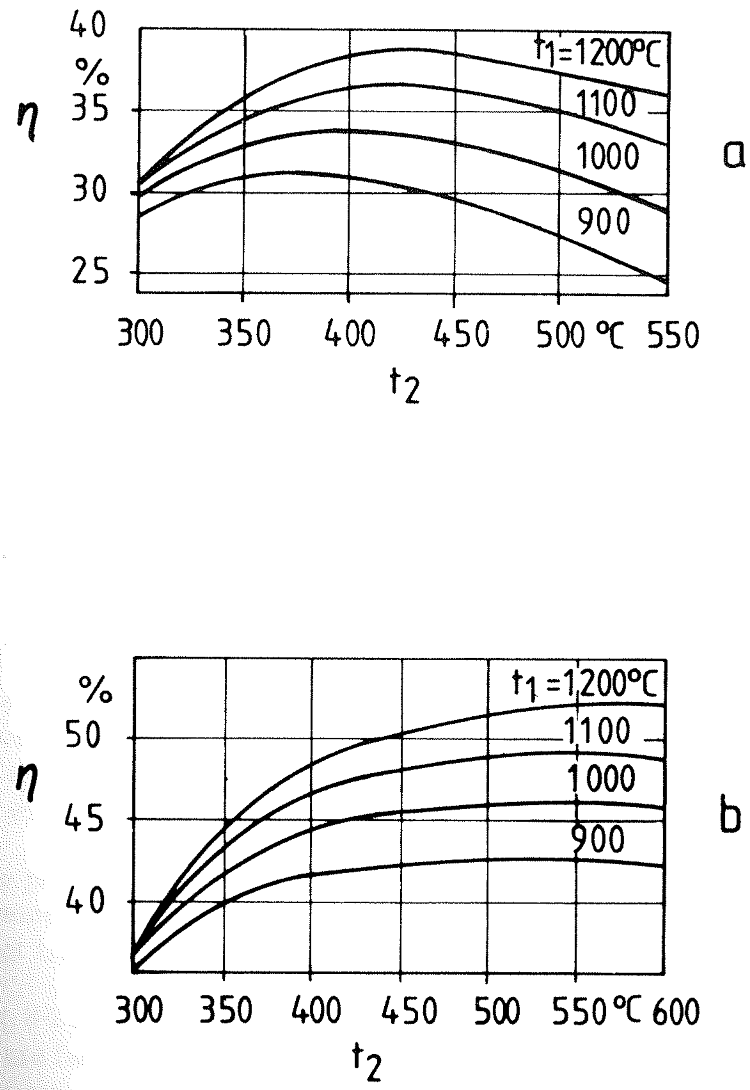

Fig. 2-2: Thermal Efficiency of Gas Turbines in Combined-Cycle Plants as a Function of the Turbine Inlet and Exhaust Gas Temperatures

a) Gas turbine alone
b) Combined-cycle plant
t_1 Gas turbine inlet
t_2 Gas turbine exhaust efficiency

Chapter 3

SYSTEM LAYOUTS

The main problem in laying out a combined-cycle plant is making optimum use of the exhaust heat from the gas turbine in the waste heat boiler. This heat transfer between the ''topping'' and the ''bottoming'' cycle entails losses (see Section 2). Heat utilization is therefore not optimum, either energetically or exergetically, and is limited by three factors:

- The physical properties of the water and exhaust gases cause exergetic and energetic losses.
- The heat exchanger cannot be infinitely large.
- The low temperature corrosion that can occur at the end of the heat exchanger limits how far the exhaust gases can be cooled.

It is mainly the first of these considerations that limits thermodynamically optimum utilization of the thermal energy. Fig. 3-1 shows the changes in temperature that would occur in an ''ideal'' heat exchanger of infinite size, operating without exergy loss.

The product, mass flow times specific heat capacity, must be the same in both media at any given point in order to make such a heat transfer possible.

Fig. 3-2 shows the temperature changes in a waste heat boiler that are far removed from this ''ideal heat exchange.'' Because water evaporates at a constant temperature, a boiler can never be an ''ideal heat exchanger.'' Even with an infinitely large heat transfer surface, the exergetic losses can never be equal to zero.

In addition to this physical limitation, there is also a chemical-limitation on energetic use of the exhaust gases imposed by low temperature corrosion. This corrosion, caused by sulphur, occurs whenever the exhaust gases are cooled below a certain temperature, the sulphuric acid dewpoint.

In a waste heat boiler, the heat transfer on the flue gas side is not as good as on the steam or water side. For that reason, the surface temperature of the pipes on the flue gas side is approximately the same as the water or steam temperature. If these pipes are to be protected against an attack of low temperature corrosion, the feedwater temperature must remain approximately as high as the acid dewpoint. Thus, a high stack temperature for the flue gases does no good if the temperature of the feedwater is too low (refer also to Section 5.2). Low temperature corrosion can occur even when burning fuels containing no sulphur if the temperature drops below the water dewpoint.

3.1 Combined-Cycle Plants without Additional Firing

In combined-cycle plants without additional firing, all the fuel is burned in the gas turbine. The steam turbine then utilizes the exhaust heat from the gas turbine, with no additional source of thermal energy. This type of combined-cycle plant is already in widespread use because it is simple and inexpensive and high efficiencies can be attained with modern gas turbines.

The number of systems possible for the steam process in such combined-cycle plants is quite large because attempts have been made to improve the quality of the heat exchange between the flue gas and the water or steam by using complex systems. This has led to systems that utilize the exhaust heat well both exergetically and energetically.

Combined-cycle plants without additional firing often are made up of several gas turbines and waste heat boilers that supply steam to a single steam turbine. In the following, we generally speak only of one gas turbine and one waste heat boiler, but all layouts can also be adapted for several gas turbines. Because the simplest system is typical of all, it has been discussed more in detail, and the other possibilities have then been derived from it.

3.1.1 Single-Pressure System

The simplest arrangement for a combined-cycle plant is a single-pressure system (Fig. 3-3) without special equipment added. This consists of one or more gas turbines with a single-pressure waste heat boiler, a condensing steam turbine, a water- or air-cooled condenser, and a single-stage feedwater preheater in the deaerator. The steam for the deaerator is tapped from the steam turbine.

The waste heat boiler consists of three parts:

- the feedwater preheater (economizer), which is heated by the flue gases;
- the evaporator, and
- the superheater.

The evaporator used can either have forced circulation (as shown) or natural circulation.

Example of Single-Pressure System

Fig. 3-4 shows the heat balance in a typical single-pressure combined-cycle plant having a 70 MW gas turbine. The exhaust gases are used to generator approx. 35 kg/s (277,200 lb/hr) steam at 34 bar (480 psig) and 475 °C (887 °F). That steam then drives a steam turbine with an output of 35 MW. Because of the good

Figure 3-1

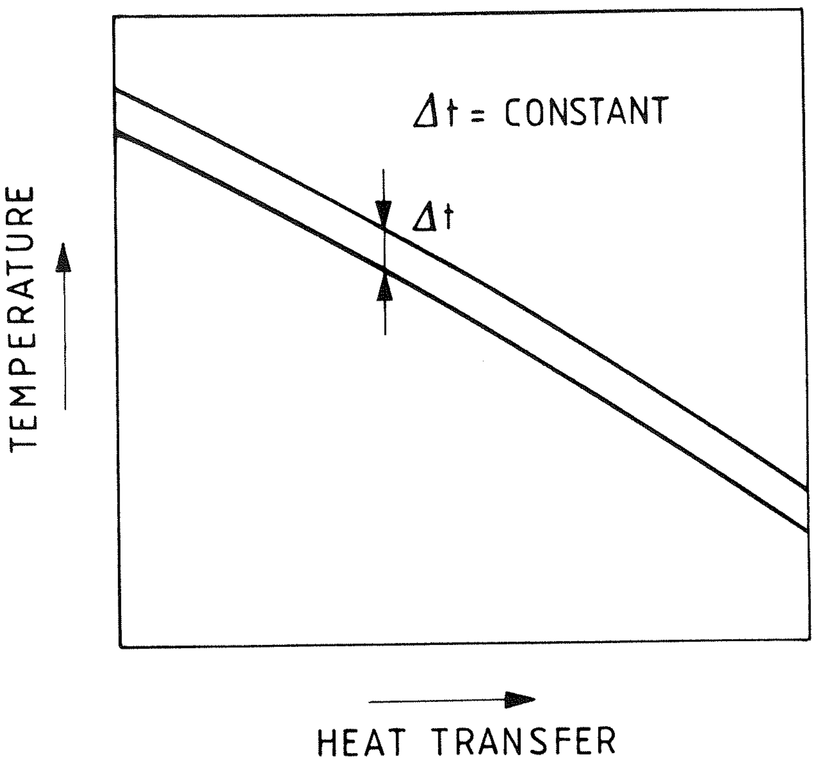

Fig. 3-1: Temperature/Heat Diagram: Ideal Heat Exchange

Figure 3-2

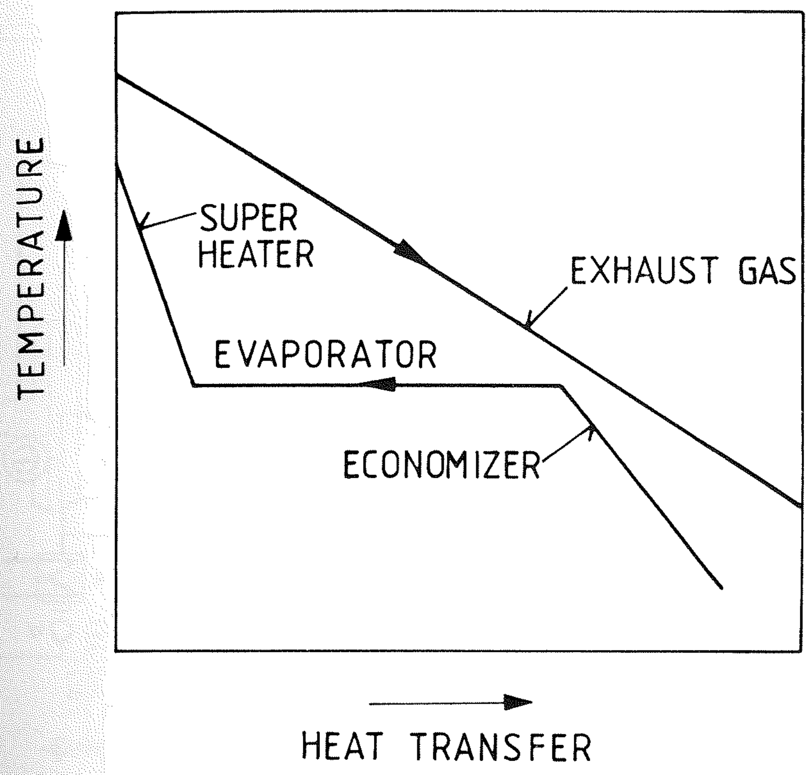

Fig. 3-2: Temperature/Heat Diagram: Heat Exchange in a Waste Heat Boiler

Figure 3-3

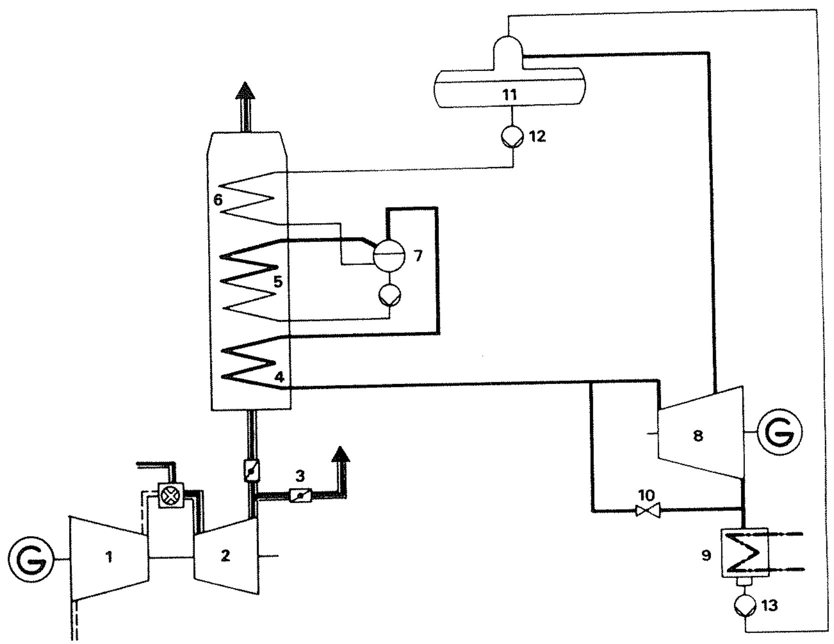

Fig. 3-3: Flow diagram of the single-pressure system

1	Compressor	6	Economizer	11	Feedwater tank/ deaerator
2	Gas turbine	7	Boiler drum	12	Feedwater pump
3	Bypass stack	8	Steam turbine	13	Condensate pump
4	Superheater	9	Condenser		
5	Evaporator	10	Steam bypass		

Figure 3-4

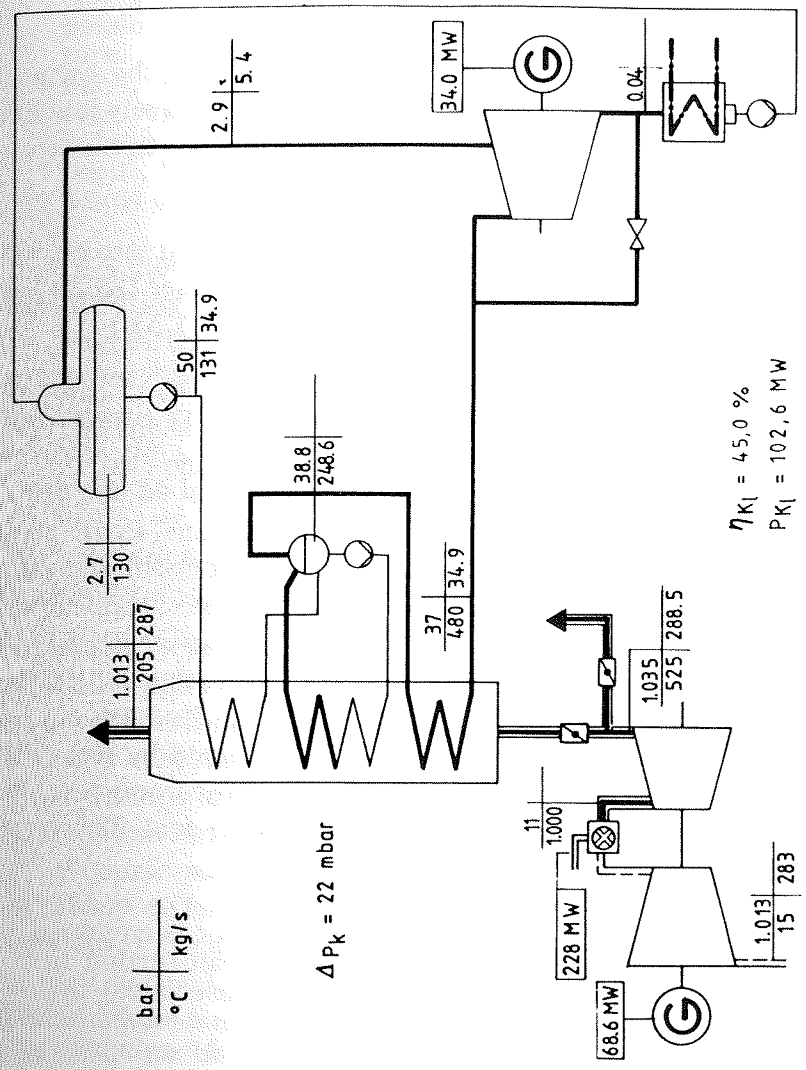

Fig. 3-4: Heat balance in the single-pressure system (example)

river-water cooling system, pressure in the condenser is 0.04 bar (0.58 psia), resulting in a gross efficiency of the installation of 45% (Table 3-1, page 30).

Noteworthy is the poor energetic utilization of the exhaust heat from the gas turbine. Together with the relatively low live steam data, this produces a fairly modest efficiency in the steam process. Fig. 3-5 shows the energy flow.

45% of the thermal energy supplied is converted into electrical energy. The rest is removed in the condenser (28.3%) or through the stack (25.2%) or is lost elsewhere (1.5%).

Fig. 3-6 shows the exergy flow of the same plant. The heat that has to be removed in the condenser is only about half that of a conventional steam power plant of the same size.

One significant difference between a conventional steam plant and the steam process in a combined-cycle plant lies in the boiler feedwater preheating. A conventional steam plant attains a better efficiency if the temperature of the feed-water is brought to a high level by means of multi-stage preheating. In a combined-cycle power plant, however, the boiler feedwater must be as cold as possible, with the limit determined by low temperature corrosion: the temperature of the water must not be significantly below the dewpoint for sulphuric acid. There are two reasons for this difference:

- Normally, a conventional steam generator is equipped with a regenerative air preheater that can further utilize the energy remaining in the flue gases after the economizer. There is nothing like that in a waste heat boiler, so that the energy remaining in the exhaust gases after the economizer is lost.

- As shown in Fig. 3-7, the smallest temperature difference between the water and the exhaust gases in the economizer is on the warmer end of the heat exchanger. That means: the amount of steam production possible does not depend on the feedwater temperature. In a conventional steam generator, on the other hand, the smallest temperature difference is on the other end of the economizer because the water flow is far larger in proportion to the flue gas flow. As a result, the amount of steam production possible depends on the feedwater temperature.

Fig. 3-8 shows two examples of conventional steam generators with differing feed-water temperatures. It is obvious that with the same difference in temperature at the end of the economizer, the heat available for evaporation and superheating is significantly greater where the feedwater temperature is higher. Thus, the amount of live steam produced by a conventional boiler can be increased by raising the feedwater temperature.

The Influence of Ambient Conditions on Output and Efficiency

We will discuss here only the effect that different ambient conditions have on the design point for the installation. How an already dimensioned combined-cycle plant behaves will be discussed in Section 7, Operating and Part-Load Behavior. Those considerations are valid, however, only for the steam turbine since the gas turbine remains the same in all cases. Gas turbines are, of course, standardized, i.e., one given machine is used even for widely different ambient conditions. This can be justified ecoomically because a gas turbine that has been optimized for an air temperature of 15°C (59°F) does not look significantly different from one that has been designed for, say, 40°C (104°F). The costs for developing a new machine would thus not be justifiable.

Figure 3-5

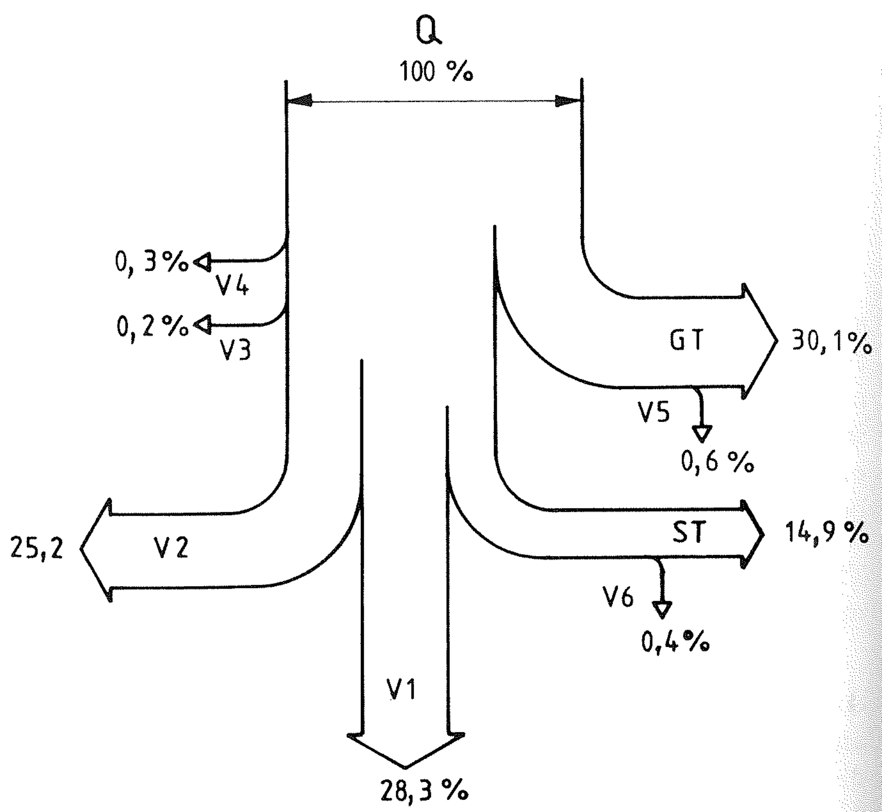

Fig. 3-5: Energy Flow Diagram for the Single-Pressure Combined-Cycle Plant

Q	Energy input
V1	Loss in condenser
V2	Loss in stack
V3	Loss due to radiation in waste heat boiler
V4	Loss in flue gas bypass
V5	Loss in generator and radiation, gas turbine
V6	Loss in generator and radiation, steam turbine
GT	Electricity produced in the gas turbine
ST	Electricity produced in the steam turbine

Figure 3-6

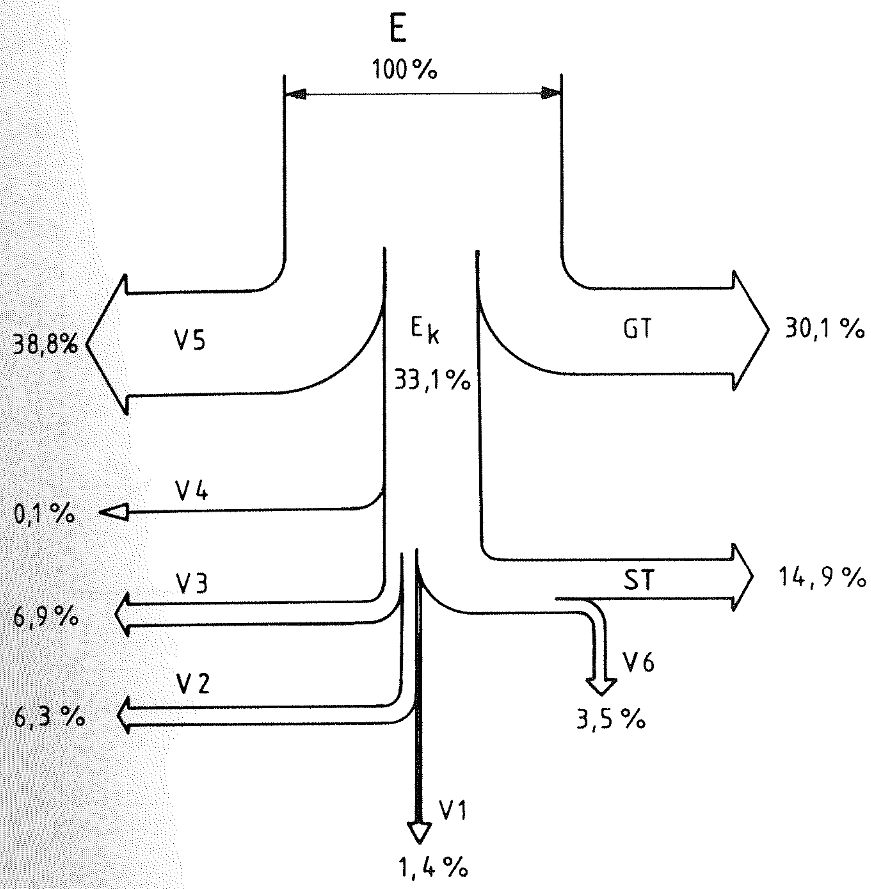

Fig. 3-6: Exergy Flow Diagram of the Single-Pressure Combined-Cycle Plant

E	Exergy input
V1	Loss in condenser
V2	Loss in stack
V3	Loss in waste heat boiler
V4	Loss in flue gas bypass
V5	Losses in gas turbine
V6	Losses in steam turbine
GT	Electricity produced in the gas turbine
ST	Electricity produced in the steam turbine
E_k	Exergy supplied to the waste heat boiler

Figure 3-7

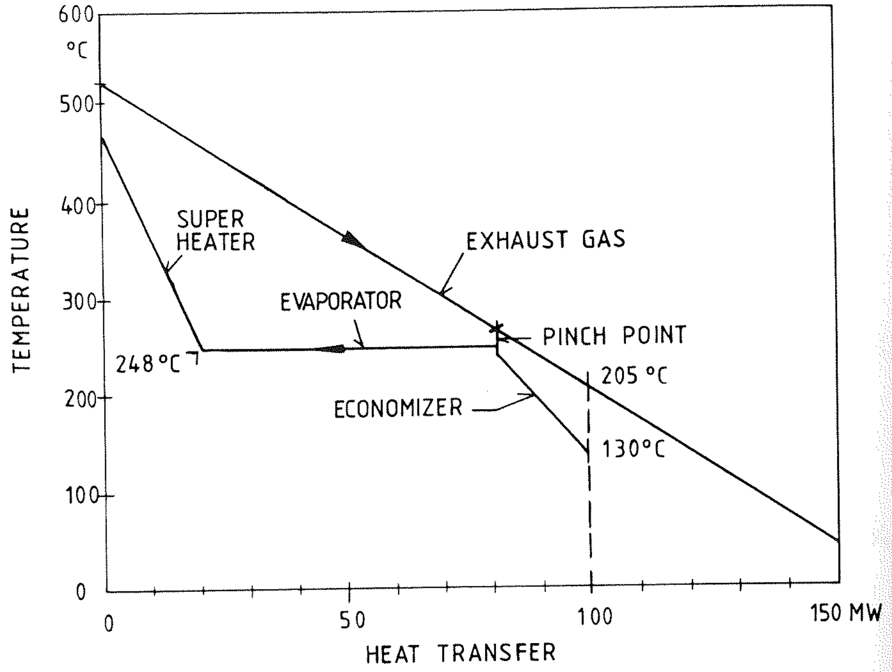

Fig. 3-7: Temperature/Heat Diagram of the Single-Pressure Boiler with a Pinch point of 15°C (27°F)

Figure 3-8

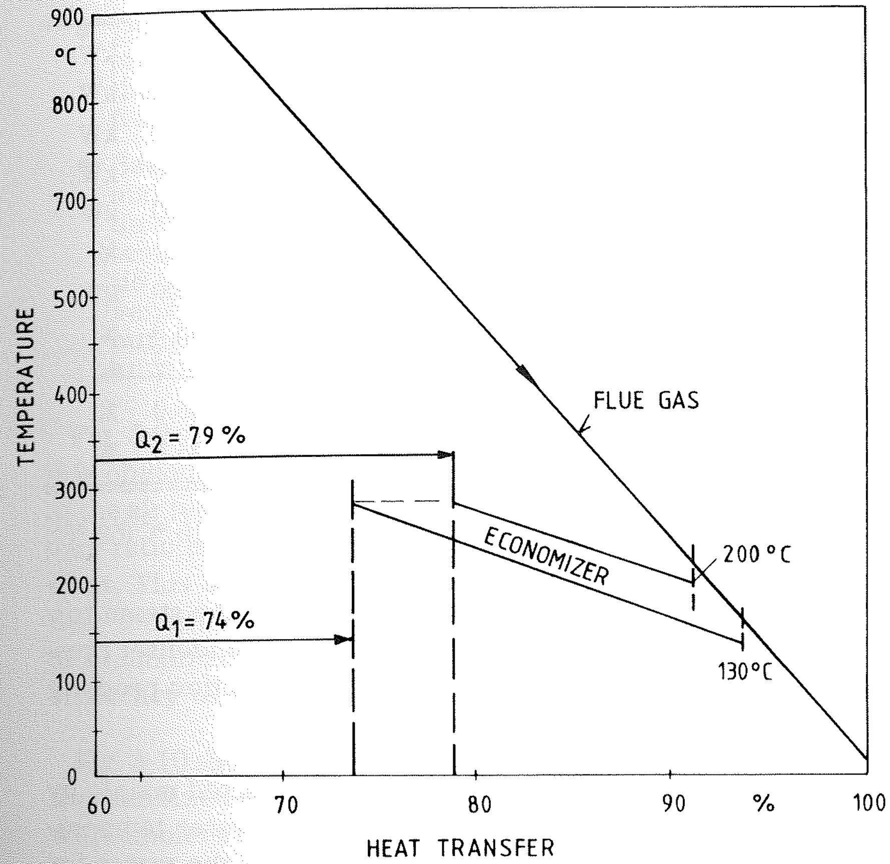

Fig. 3-8: Temperature/Heat Diagram of a Conventional Steam Generator Thermal Energy Available for Evaporation and Superheater
Example 1: 74% of heat supplied (Q_1)
Example 2: 79% of heat supplied (Q_2)
Live steam pressure = 63 bar
Live steam temperature = 485 °C

Table 3-1: Main Technical Data of the Single-Pressure Combined-Cycle Plant

Gas turbine output	68 600	kW
Steam turbine output	34 000	kW
Station service power required	1 100	kW
Net power output of plant	101 500	kW
Thermal energy supplied (Diesel fuel)	228 000	kW
Efficiency of gas turbine	30.1	%
Heat contained in exhaust gases	157 000	kW
Utilization rate for waste heat energy*	63.3	%
Efficiency of the steam process	21.7	%
Gross efficiency of the plant	45.0	%
Net efficiency of the plant	44.5	%

* 100% utilization if the exhaust gases are cooled down to 15 °C (59 °F).

The situation is different on the steam end of the steam turbine. The exhaust steam section designed for a condenser pressure of, say, 0.2 bar (2.9 psia) can no longer function properly if the pressure is only 0.04 bar (0.58 psia).

The design of the combined-cycle plant is affected mainly by the air temperature, air pressure, and cooling water temperature. The relative humidity is important only if the water for cooling the condenser is recooled in a wet cooling tower.

Air Temperature

There are three reasons why the air temperature has a large influence on the power output and efficiency of an open-cycle gas turbine:

- Increasing the air temperature reduces the density of the air, and thereby reduces the air mass flow drawn in.
- The power consumed by the compressor increases in proportion to the intake temperature (in K), without there being a corresponding increase in the output from the turbine.
- Because the absorption capacity of the turbine remains constant, the pressure before the turbine is reduced, since the mass flow decreases as the air temperature rises. This again reduces the pressure ratio within the turbine. The same principle applies inversely, of course, to the compressor, but because its output is less than than of the turbine, the total balance is negative.

Fig. 3-9 shows this change in a temperature/entropy diagram. It is obvious that the exhaust gas temperature becomes higher as the air temperature increases. This is because the turbine pressure ratio is reduced while the inlet temperature remains constant. This behavior of the exhaust gas temperature explains why the effect that the air temperature has on the efficiency of a combined-cycle plant differs from that which it has on the efficiency of the gas turbine alone.

Fig. 3-10 shows the relative efficiencies of the gas turbine and the combined-cycle plant as a function of the air temperature, with ambient conditions remaining otherwise unchanged. As it shows, an increase in the air temperature even has a slightly positive effect on the efficiency of the combined-cycle plant, since the increased temperature in the gas turbine exhaust raises the efficiency of the steam process (Fig. 3-11) enough to more than compensate for the reduced efficiency of the gas turbine unit.

This behavior is not surprising when one remembers the Carnot efficiency [Equation (1)]. The rise in the final temperature

Figure 3-9

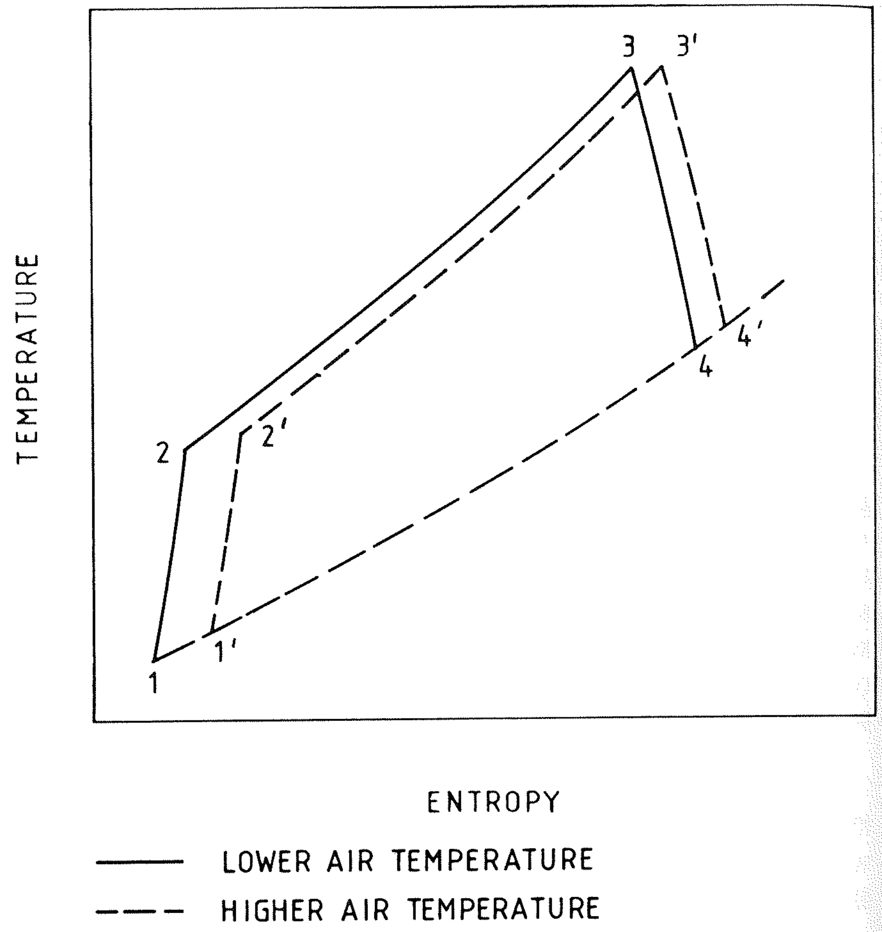

Fig. 3-9: Temperature/Entropy Diagram for a Gas Turbine at Two Different Air Temperatures

1-2 = Compressor
2-3 = Combustion chamber
3-4 = Turbine

Figure 3-10

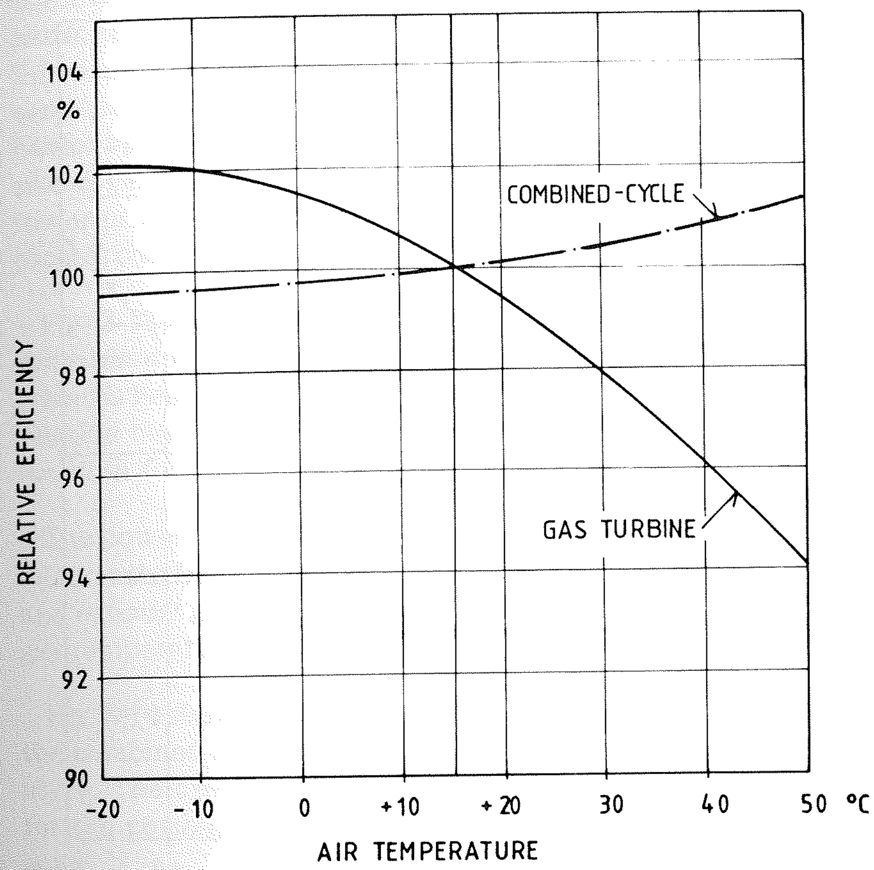

Fig. 3-10 Relative Efficiency of Gas Turbines and Combined-Cycle Plants as a Function of the Air Temperature
Cooling water temperature 20°C (68°F)

Figure 3-11

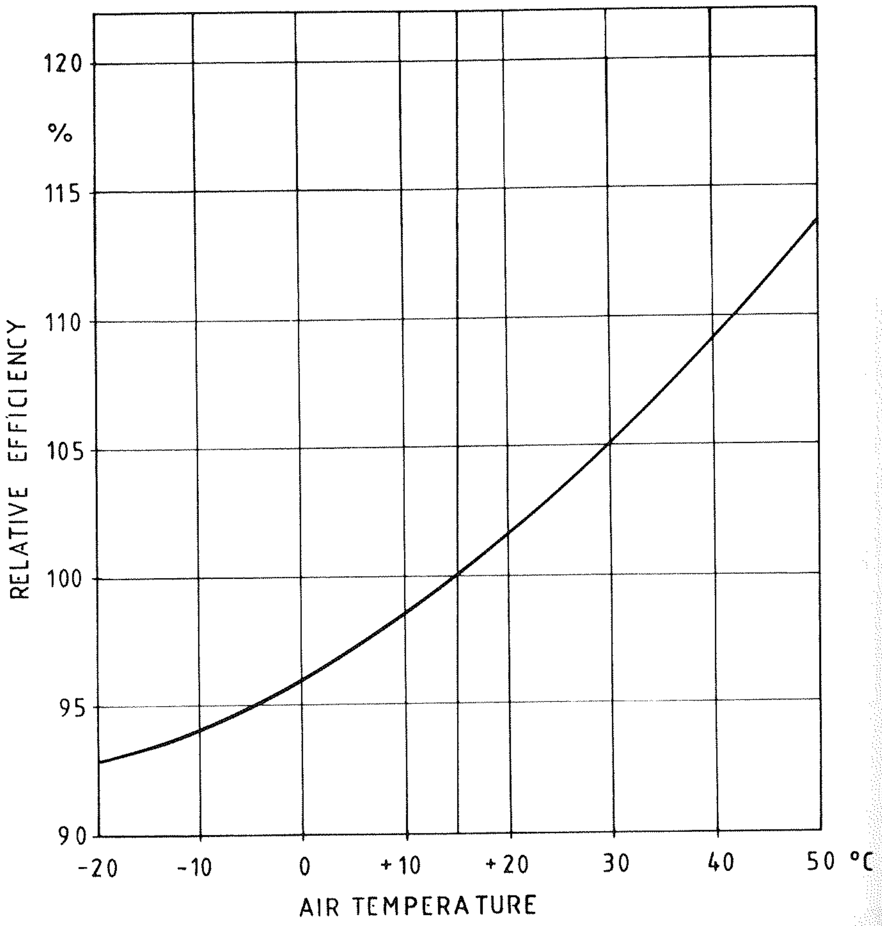

Fig. 3-11: Relative Efficiency of the Steam Process in Combined-Cycle Plants as a Function of the Air Temperature
Cooling water temperature 20°C (68°F)

of compression causes a slight increase in the average temperature of the heat supplied T_W as well. Because most of the exhaust heat is carried off in the condenser, the cold temperature T_W changes only insignificantly. The overall efficiency of the combined-cycle plant is thus more likely to increase. This behavior applies only if the temperature of the water cooling the steam turbine condenser remains unchanged. With a cooling tower or an air-cooled condenser, the efficiency of the steam process changes because the condenser pressure is now different.

Fig. 3-12 shows how the overall efficiency of the combined-cycle plant changes with the air temperature when the cooling water is being recooled in a wet cooling tower with a constant relative humidity in the air of 60%. Fig. 3-13 shows the same function for the case with direct air-cooled condensation.

The power output from the combined-cycle plant reacts quite differently from its efficiency. Here the reduced flows of air and exhaust gases play a more important role than the exhaust gas temperature.

Fig. 3-14 shows how the power outputs of the gas turbine and the combined-cycle plant change depending on the air temperature. The drop-off at higher temperatures is less pronounced for the combined-cycle plant than for the gas turbine alone.

Air Pressure and Site Elevation

Gas turbines are normally designed for an air pressure of 1.013 bar (14.7 psia), which corresponds approximately to the average pressure prevailing at sea level. A different site elevations results in a different average air pressure (Fig. 3-15).

The effect of the air pressure on the efficiency of a gas turbine is equal to zero if the temperatures remain unchanged. On the other hand, the power output changes with the air mass flow taken in, which varies in proportion to the intake pressure and thereby also affects the flow of exhaust gas. The exhaust heat available for the steam process likewise varies in proportion to the air pressure. If one assumes that no change takes place in the efficiency of the steam process, which corresponds quite well to the real situation, this then causes a similar variation in the power output from the steam turbine.

Because the power outputs of the gas turbine and the steam turbine vary in proportion to the air pressure, the total power output of the combined-cycle plant varies correspondingly. The efficiency of the plant remains constant, however, since both the thermal energy supplied and the air flow are varying in proportion to the air pressure.

Cooling media for the Condenser

To condense the steam, a cooling medium must be used to carry off the waste heat from the condenser. Generally this is water, which has a high specific thermal capacity and good heat transfer properties. Where water is in short supply, cooling can be done in air in a wet cooling tower; where no water is available, an air-cooled condenser or a dry cooling tower are necessary.

The temperature of the cooling medium affects the efficiency of the thermal process. The lower that temperature is, the higher the efficiency that can be attained [refer to Equation (1)]. A lower temperature makes possible a lower pressure in the condenser, producing a greater useful enthalpy drop in the steam turbine.

Fig. 3-16 contains typical approximate values for condenser pressure as a function of the design temperature for the cooling medium. There are three different cases:

Figure 3-12

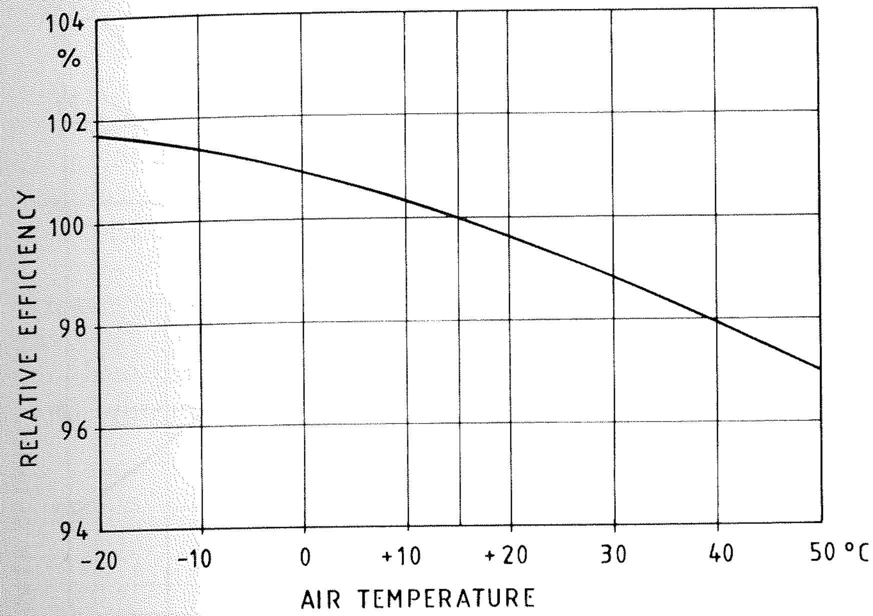

Fig. 3-12: Effect of Air Temperature on the Efficiency of Combined-Cycle Plants with a Wet Cooling Tower
Relative Humidity of Air 60%

Figure 3-13

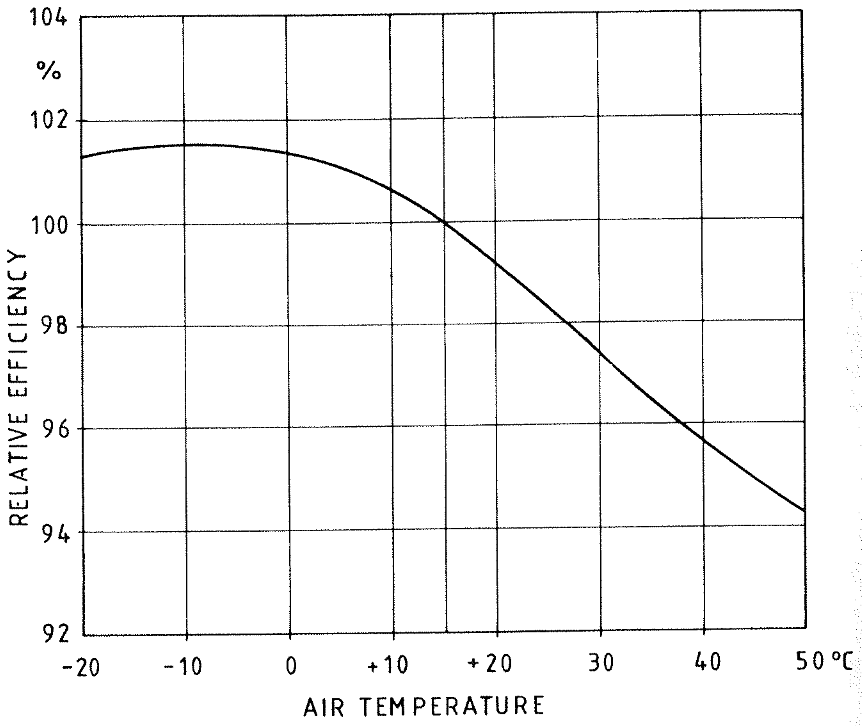

Fig. 3-13: Effect of the Air Temperature on the Efficiency of Combined-Cycle Plants with Direct Air-Cooled Condensation

Figure 3-14

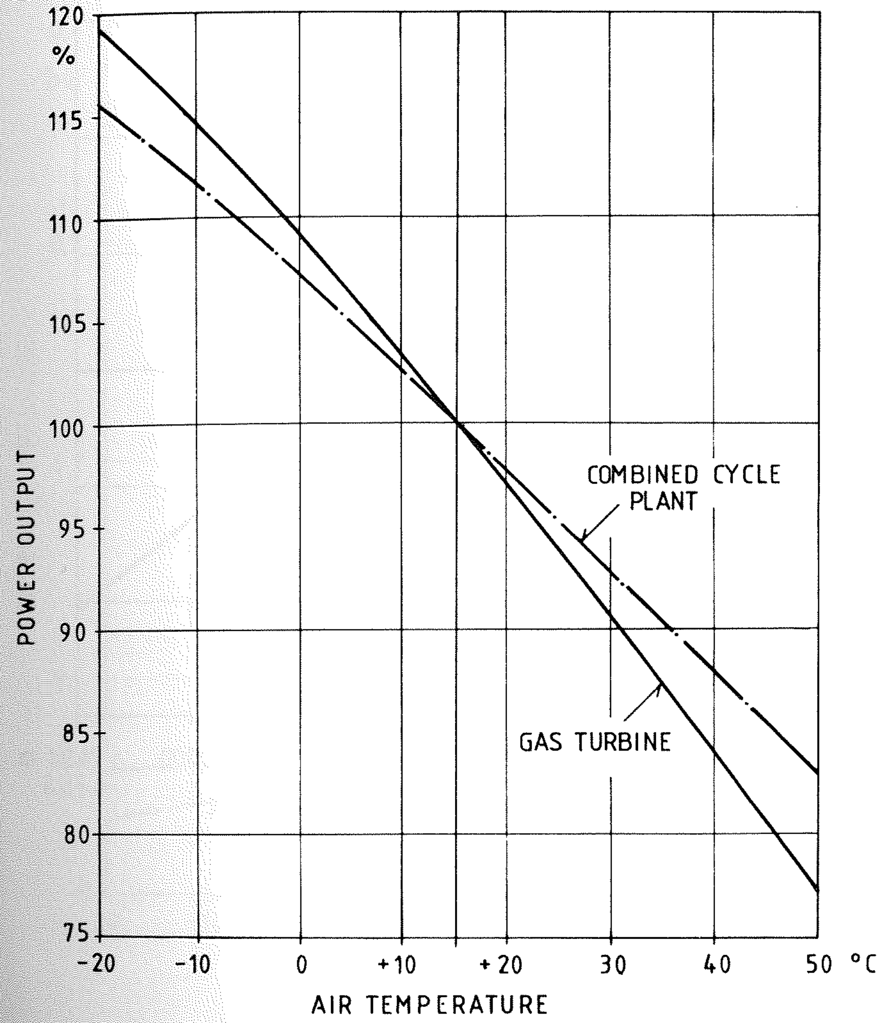

Fig. 3-14: Relative Power Output and Efficiency of Gas Turbines and Combined-Cycle Plants as Functions of Air Temperature
Cooling water temperature 20°C (68°F)

Figure 3-15

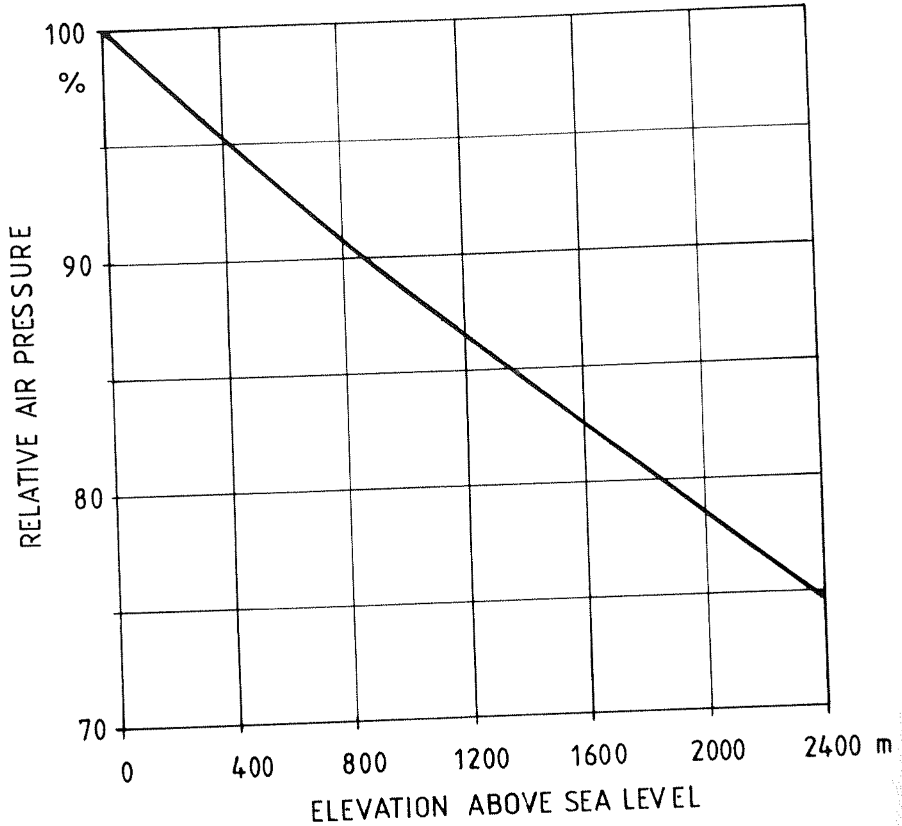

Fig. 3-15: Standard air pressure as a function of elevation
100% = 1.013 bar (14.7 psia)

Figure 3-16

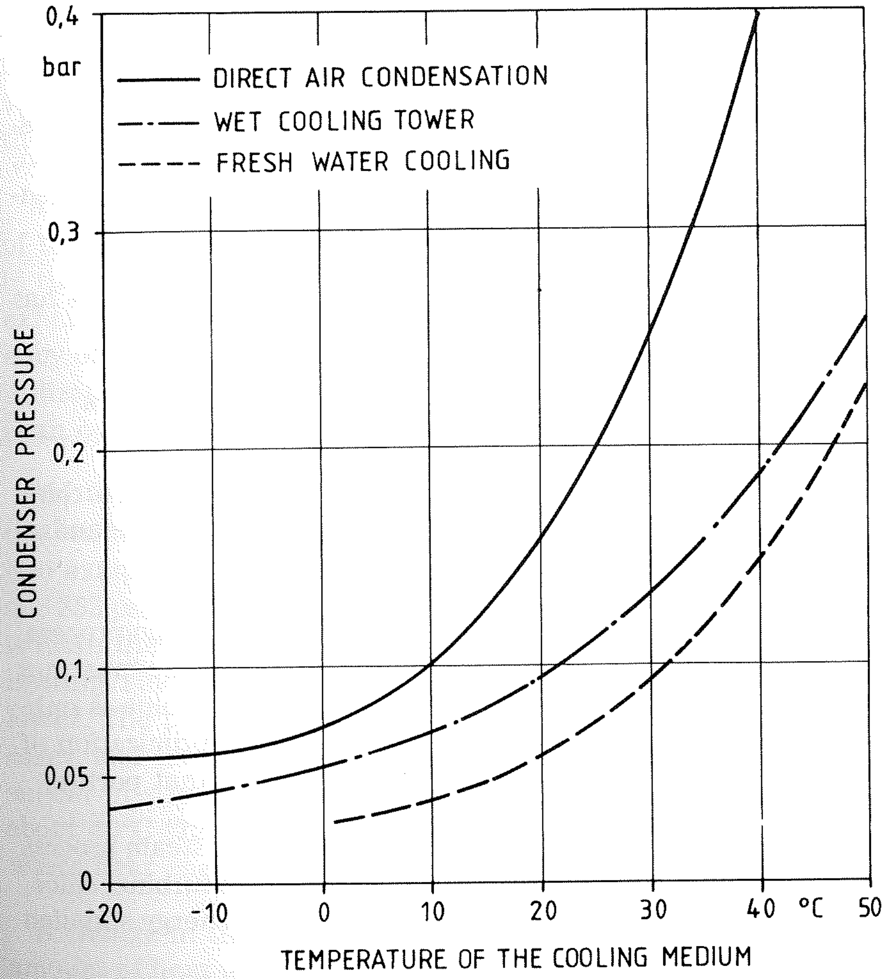

Fig. 3-16: Approximate Values for Selecting the Condenser Pressure for Fresh Water Cooling, Wet Cooling Tower, and Direct Air Condensation (Air for cooling tower and direct air-cooled condensation)

Source: ABB Asea Brown Boveri
Relative humidity of air = 60%

- direct water cooling
- water cooling, with water recooled in a wet cooling tower
- direct air cooling

The greatest vacuums are attained with direct water cooling, the least with direct condensation with air. In the comparison, it must also be borne in mind that the water temperature is generally lower than that of the air.

For the wet cooling tower, a relative air humidity of 60% has been assumed.

The Effect of the Most Important Design Parameters on Power Output and Efficiency

When dimensioning a combined-cycle plant, the gas turbine design is generally a given, since the gas turbine is a standardized machine.

The free parameters for the design involve the steam process, and it is mainly these that are discussed below. One must not forget, however, that the output of the steam turbine is only approx. 30 to 40% of the total power output. Optimization of the steam process can therefore only influence that portion.

Another important point: The efficiency of the steam process is always proportional to the output of the steam turbine, since, in a plant without additional firing, the thermal energy supplied to the steam process is a given.

Live Steam Data

The selection of the live steam data for a combined-cycle plant with a single-pressure system is a compromise between optimum energetic and optimum exergetic utilization of the exhaust heat from the gas turbine. The main determining factor is the live steam pressure selected.

Live Steam Pressure

In a combined-cycle plant, a high live steam pressure does not necessarily mean a high efficiency. Fig. 3-17 shows how the efficiency of the steam process depends on the live steam pressure. It is striking that the best efficiency is attained even while the live steam pressure is quite low.

A higher pressure does indeed bring an increased efficiency of the water/steam cycle due to the greater enthalpy gradient in the turbine. The rate of waste heat energy utilization in the exhaust gases, however, drops off sharply. The overall efficiency of the steam process is the product of the rate of energy utilization and the efficiency of the water/steam cycle. There is an optimum at approx. 30 bar (435 psia).

Fig. 3-18 explains the increased rate of energy utilization in the waste heat boiler: the temperature/heat diagrams are for two examples with live steam pressures of 15 and 60 bar (203 and 855 psig) respectively. At the lower live steam pressure, there is more thermal energy available for evaporation and superheating, since the evaporation temperature is correspondingly lower. The pinch point of the evaporator is the same in both cases, and the surface area of the heat exchanger is therefore similar in size. As a result, the stack temperature at 15 bar is about 40°C (72°F) lower than at 60 bar, which means that the waste heat energy is being better utilized.

A change in the live steam pressure also greatly affects the amount of heat to be removed in the condenser (Fig. 3-19). The power output increases when pressures are lower since a greater amount of heat is being removed from the exhaust gases and converted into electricity at a lower efficiency.

Economical considerations can thus make it advisable to raise the live steam pressure above the thermodynamic optimum. This can provide the following advantages:

- a reduction in the exhaust steam flow, or, if the size of the steam turbine remains unchanged, smaller exhaust losses.
- a smaller condenser
- a reduction of the cooling water requirement

Especially in the case of power plants with expensive air-cooled condensers, this can mean considerably lower costs.

Live steam flows greater than that in the example shown shift the optimum toward higher live steam pressures, since the volume flows also are larger. The live steam pressure selected is thus less important for larger steam turbines than for smaller installations. For that reason, it is advantageous in larger combined-cycle plants with several gas turbines to select a live steam pressure that is above the optimum. The reduced volume flow that results makes it possible to employ piping and valves with smaller dimensions. The trend is the opposite when the flow of live steam is reduced. The optimum live steam pressure also depends on the total amount of live steam: increasing the amount improves efficiency in the high pressure section of the steam turbine. With a larger volume flow, longer blades are required in the first row, which reduces the edging losses.

Live Steam Temperature

In contrast to the live steam pressure, raising the live steam temperature always brings with it a slight increase in efficiency (Fig. 3-20). There are two reasons for this improvement with increased superheating:

- improved thermodynamics of the cycle,
- increased steam turbine efficiency due to reduced wetness in the low pressure section.

That means an improved efficiency of the water/steam cycle that more than compensates for the slight drop in the rate of waste heat energy utilization. Moreover, for the steam turbine, increasing the live steam temperature means less erosion in the final stages (because of the reduced water content in the steam).

The temperature of the gas turbine exhaust gas provides the upper limit for the live steam temperature. However, a sufficient difference in temperature is necessary between the exhaust gas and the live steam in order to limit the size of the superheater. Moreover, too high a live steam temperature can also cause a disproportionate increase in plant costs since a great amount of expensive material is required for the piping, the superheater, and the steam turbine. In most cases, however, the exhaust gas temperature sets the limit for the live steam temperature level.

Feedwater Preheating

In order to attain a good rate of waste heat energy utilization, the temperature of the feedwater should be kept as low as possible. The thermodynamic quality of the water/steam cycle remains largely unaffected (Fig. 3-21).

In the example given in Section 3.1.1, preheating has been reduced to a single stage: the feedwater tank/deaerator. A multistage preheating would improve the efficiency but it has not been considered here because the solutions shown in Sections 3.1.1 and 3.1.2 are clearly better. Dividing preheating into several stages does not improve the rate of energy utilization in this simple single-pressure system, which is the greatest disadvantage of this type of system. Even with minimum feedwater temperatures, the stack temperature remains at approx. 200 °C (392 °F). Some of this lost energy can be recovered by improving the system layout (refer to Sections 3.2 and 3.3).

Figure 3-17

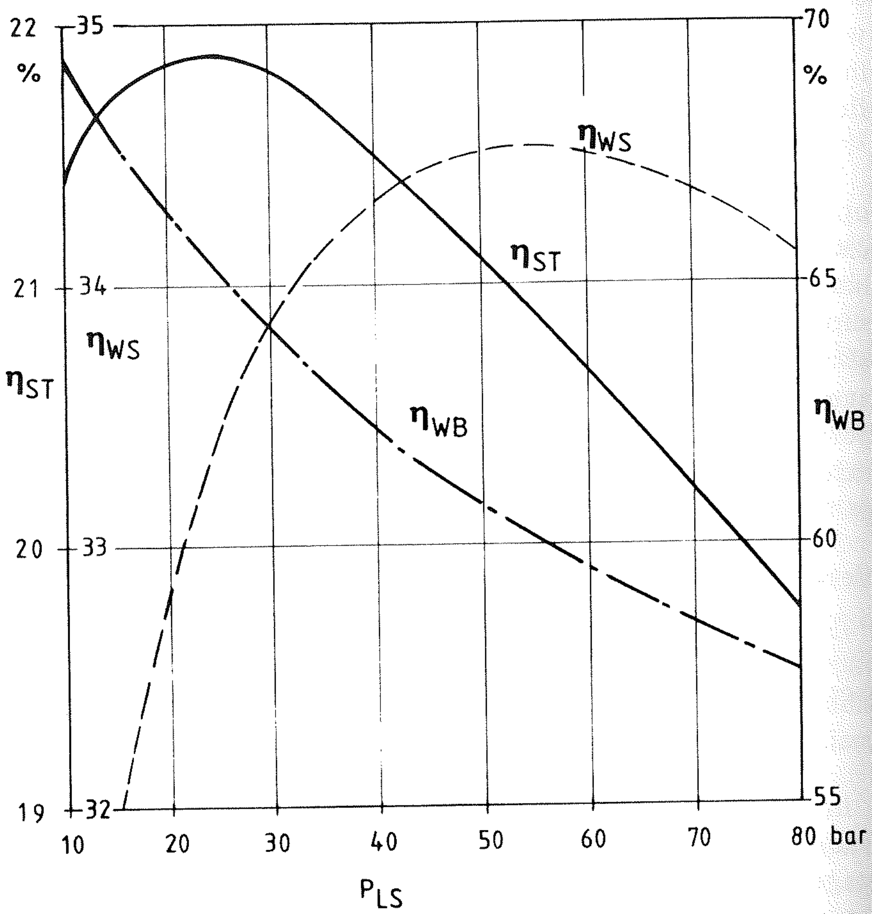

Fig. 3-17: Effect of the Live Steam Pressure on the Efficiency of the Steam Process and the Rate of Waste Heat Energy Utilization

η_{WS}	Efficiency of the water/steam cycle
η_{ST}	Efficiency of the steam process
η_{WB}	Rate of Waste Heat Energy Utilization
P_{LS}	Live steam pressure at the turbine inlet

Figure 3-18

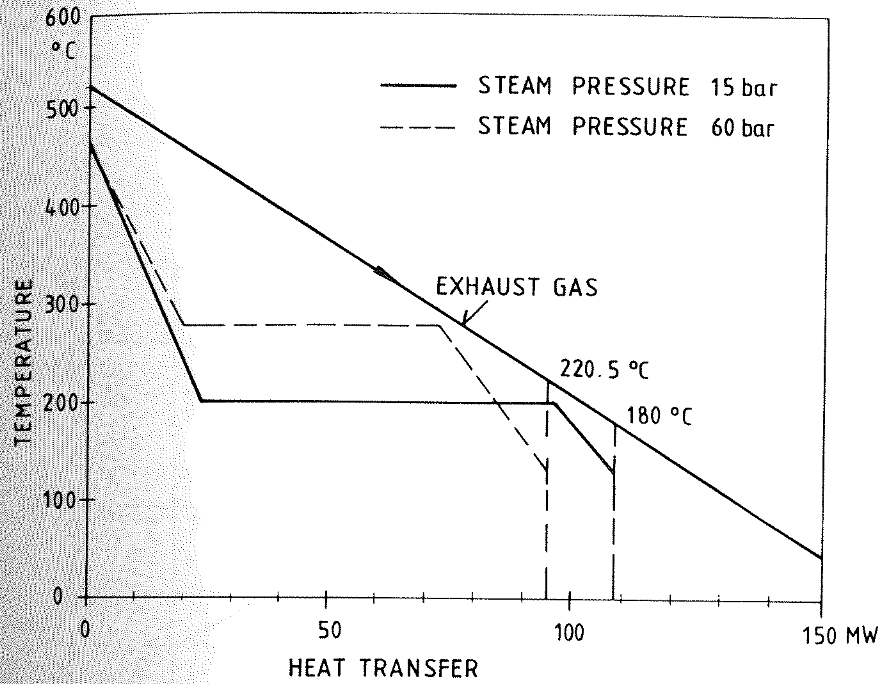

Fig. 3-18: Temperature/Heat Diagram of a Single-Pressure Boiler with Live Steam Pressures of 15 and 60 bar (203 and 855 psig) Respectively

Figure 3-19

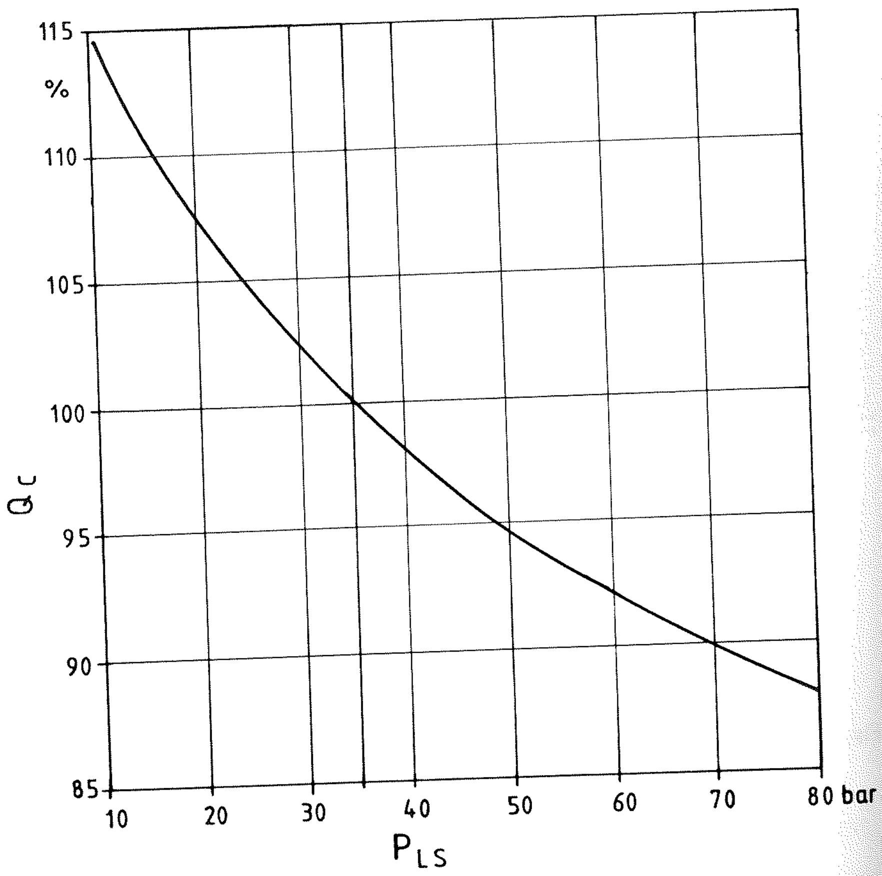

Fig. 3-19: Effect of the Live Steam Pressure on the Waste Heat from a Condenser

	Live steam temperature	475°C (875°F)
	Condenser pressure	0.04 bar (0.58 psia)
P_{os}	Live steam pressure	
QC	Waste heat from condenser	

Figure 3-20

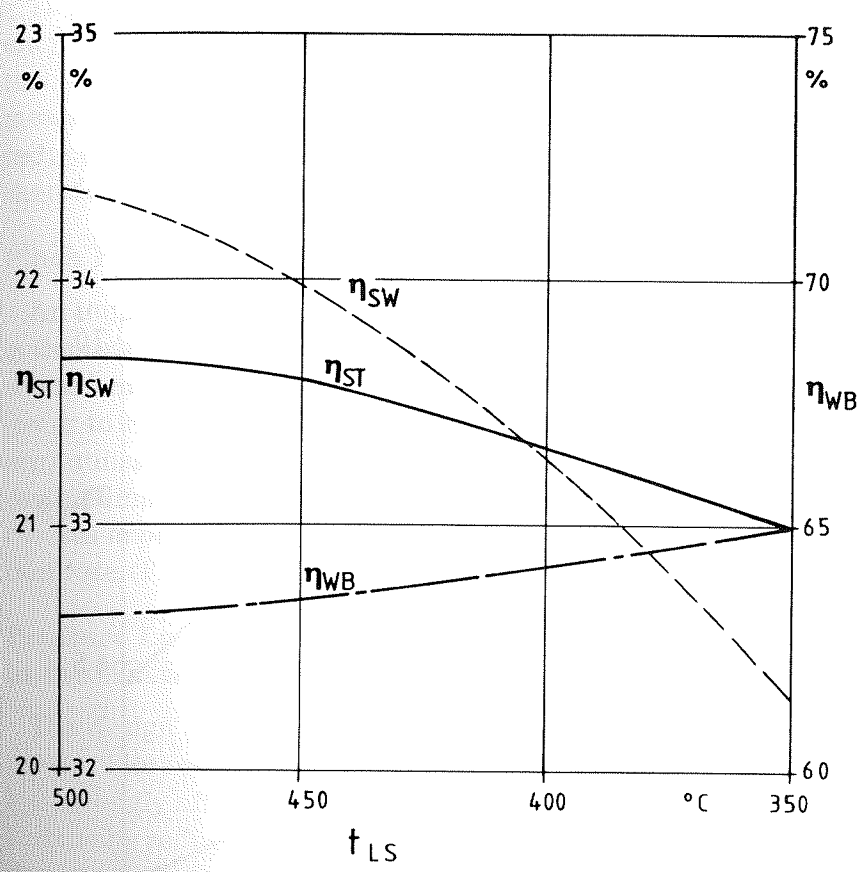

Fig. 3-20: Effect of the Live Steam Temperature on the Efficiency of the Steam Process and Rate of Waste Heat Energy Utilization

T_{LS} Live Steam Temperature at the Turbine Inlet
[Other terms as in Fig. 3-17]

Figure 3-21

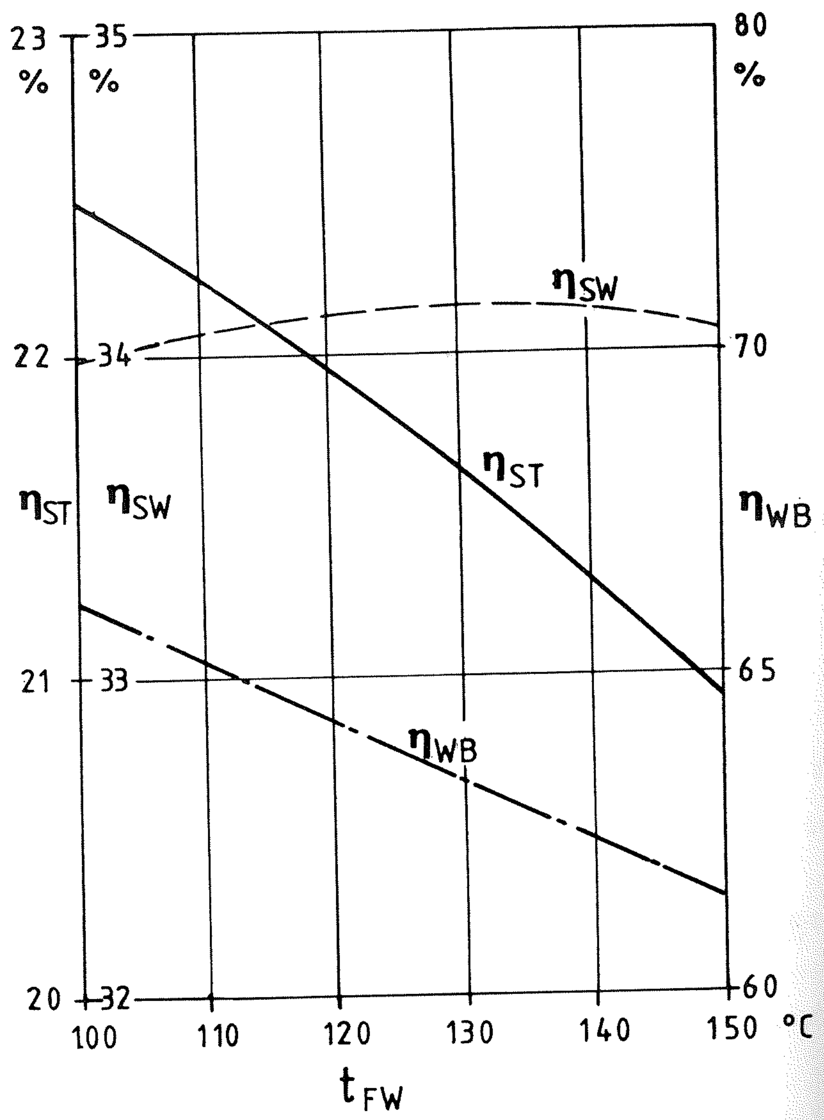

Fig. 3-21: Effect of the Feedwater Temperature t_{fw} on the Efficiency of the Steam Process and Rate of Waste Heat Energy Utilization

t_{fw} = Feedwater Temperature
[Other terms as in Fig. 3-17]

Condenser Pressure

The condenser pressure has a major influence on the efficiency of the steam process because the enthalpy drop in the steam turbine changes sharply (Fig. 3-22). An increase in the pressure means an decrease in power output. However, plant costs are reduced, due to the lower volume flow of exhaust steam and therefore the reduced size of the steam turbine and condenser.

Pinch Point of the Waste Heat Boiler

An important parameter in the optimization of a steam cycle is the temperature difference rating (the ''pinch point'') of the waste heat boiler, which affects the amount of steam generated (refer to Fig. 3-7). By reducing the pinch point, the rate of energy utilization in the waste heat boiler can be influenced within certain limits. However, the surface of the heat exchanger increases exponentially, which quickly sets a limit for the utilization rate (Fig. 3-23).

Pressure Loss on the Flue-Gas Side in the Waste Heat Boiler

The design of the waste heat boiler should be such that the pressure loss on its flue-gas side remains as low as possible. This loss strongly affects the power output and efficiency of the gas turbine by reducing the pressure ratio in the turbine. In present-day gas turbines, this loss is approximately 0.8% for each 1% pressure loss. Some of the lost output is recovered in the steam cycle, but the maximum rate of recovery is 35%.

Gas Turbine Exhaust Temperature

The temperature level of the gas turbine exhaust gas is important for the efficiency of the steam cycle. If the turbine inlet temperatures remain constant, a gas turbine with a higher exhaust gas temperature and a poorer overall efficiency produces the better combined cycle, assuming identical compressor and turbine efficiencies (refer also to Fig. 2-2).

When the gas turbine exhaust temperature is lowered, both the thermodynamic quality of the steam process and the energy utilization rate of the waste heat boiler deteriorate (Fig. 3-24).

When evaluating the suitability of a gas turbine for a combined-cycle process then, consideration must be given not only to its efficiency but also to its exhaust gas temperature.

3.1.2 Single-Pressure System with a Preheating Loop in the Waste Heat Boiler

The major disadvantage of the single-pressure system (Section 3.1.1) is its relatively poor rate at which it utilizes waste heat energy. The easiest improvement is to use an additional heat exchanger at the end of the waste heat boiler to recover additional heat for preheating the feedwater. This preheating loop must be designed so that temperatures do not drop below the acid dewpoint. It is therefore not possible to send the condensate directly into the boiler.

There are two ways to solve this problem: with water or with steam. Fig. 3-25 shows the version using water, in which a pump is used to bring a large amount of water to a high pressure level. There must be more water than condensate: too great a temperature rise due to the heat transfer would cause temperatures to drop below the dewpoint. After being warmed up in the preheating loop, the water flows into a flash tank that produces the steam required for the deaerator. The remainder is returned directly to the feedwater tank. The main disadvantage of this layout is the great amount of power required to drive the circulating pump, since the water must be pressurized to approx. 20 bar (290 psi).

Fig. 3-26 shows a version in which a low pressure evaporator generates saturated steam for the deaerator. In this case, the power required to drive the pump is quite small, approximately

Figure 3-22

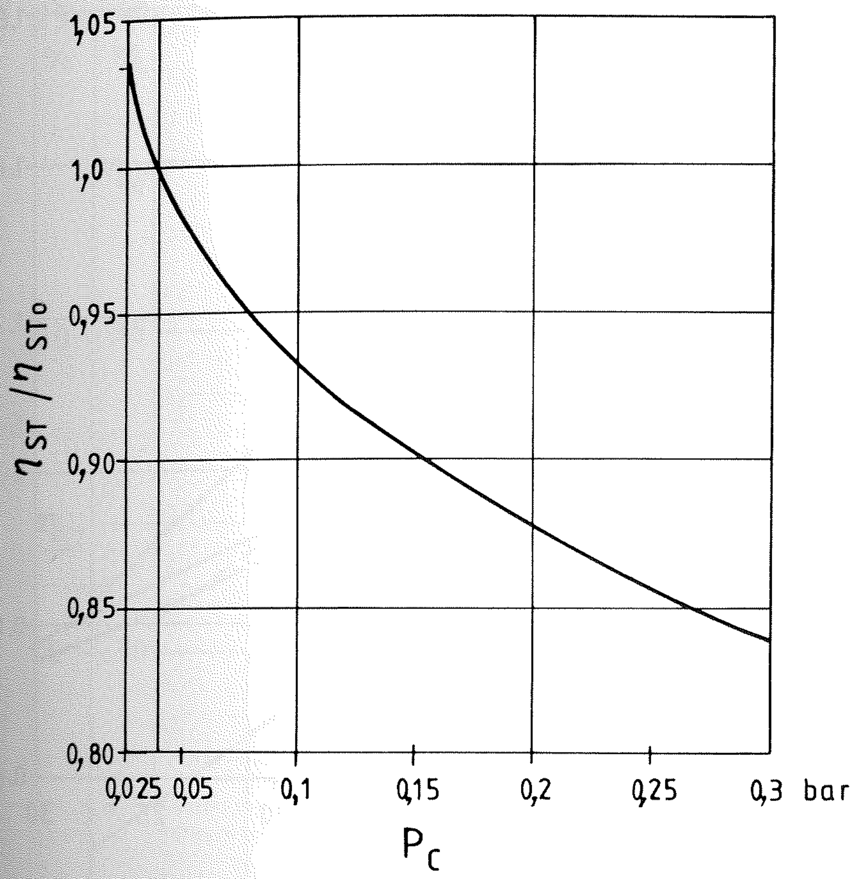

Fig. 3-22: Effect of Condenser Pressure on the Efficiency of the Steam Process

P_C Condenser pressure

η_{ST}/η_{STo} Relative efficiency of the steam process

Figure 3-23

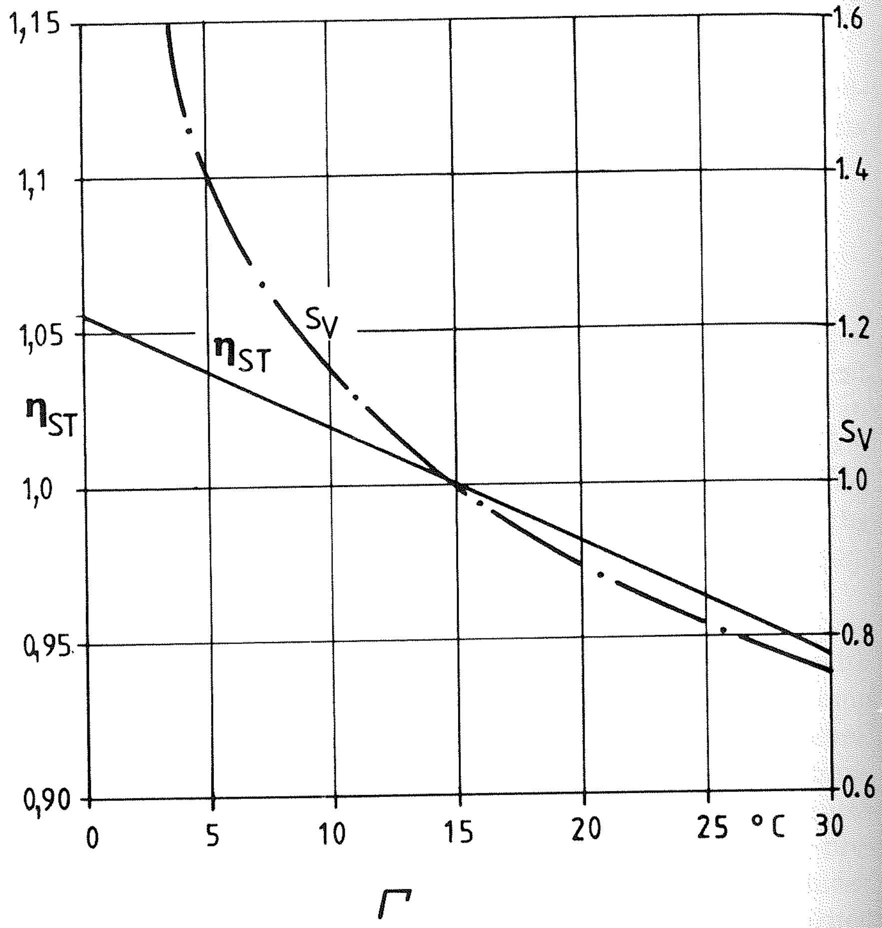

Fig. 3-23: Effect of the Pinch Point of the Waste Heat Boiler on the Efficiency of the Steam Process and the Heat Transfer Surface in the Evaporator

η_{ST}	Efficiency of the Steam Process
S_E	Heat Transfer
T	Pinch Point

Figure 3-24

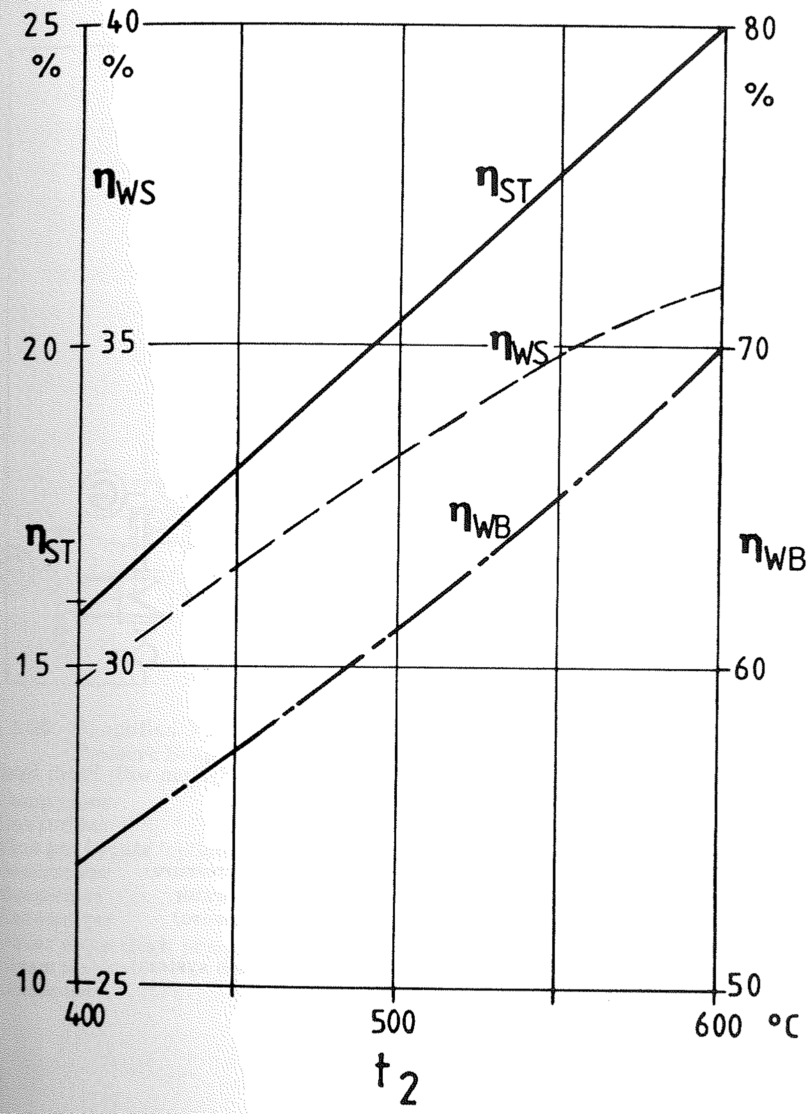

Fig. 3-24: Efficiency of the Steam Process and Rate of Waste Heat Energy Utilization as Functions of the Gas Turbine Exhaust Gas Temperature

η_{WS}	Efficiency of the water/steam cycle
η_{ST}	Efficiency of the steam process
η_{WB}	Rate of Waste Heat Energy Utilization
t_2	Gas Turbine Exhaust Gas Temperature

Figure 3-25

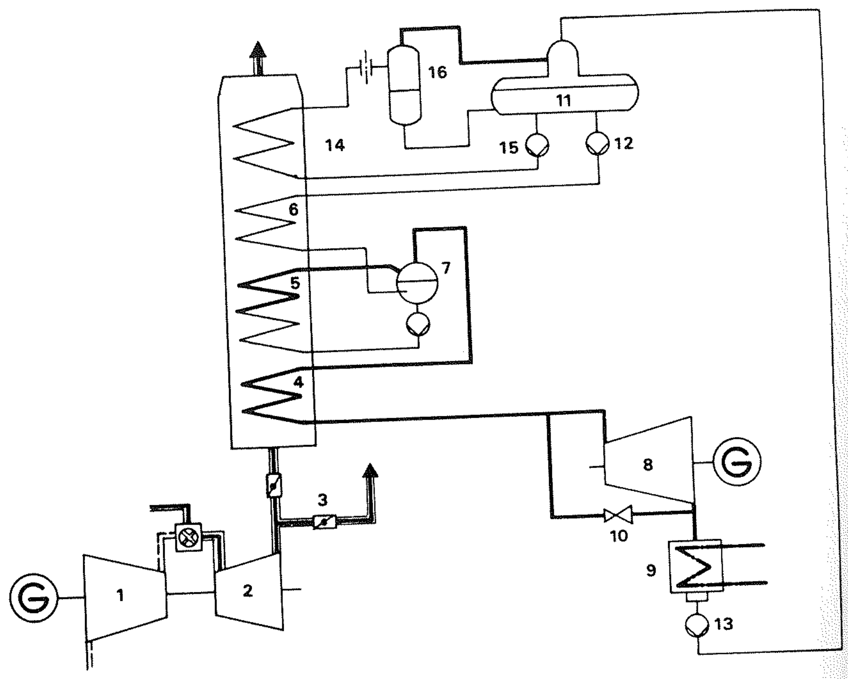

Fig. 3-25: Simplified Flow Diagram of the Single-Pressure System with Flash System as a Preheating Loop

1 Compressor
2 Gas turbine
3 Flue gas bypass (optional)
4 Superheater
5 Evaporator
6 Economizer
7 Boiler drum (high pressure)
8 Steam turbine
9 Condenser
10 Steam bypass (high pressure)
11 Feedwater tank, deaerator
12 Feed pump (high pressure)
13 Condensate pump
14 Preheating loop (flash system)
15 Booster pumps
16 Flash tank

Figure 3-26

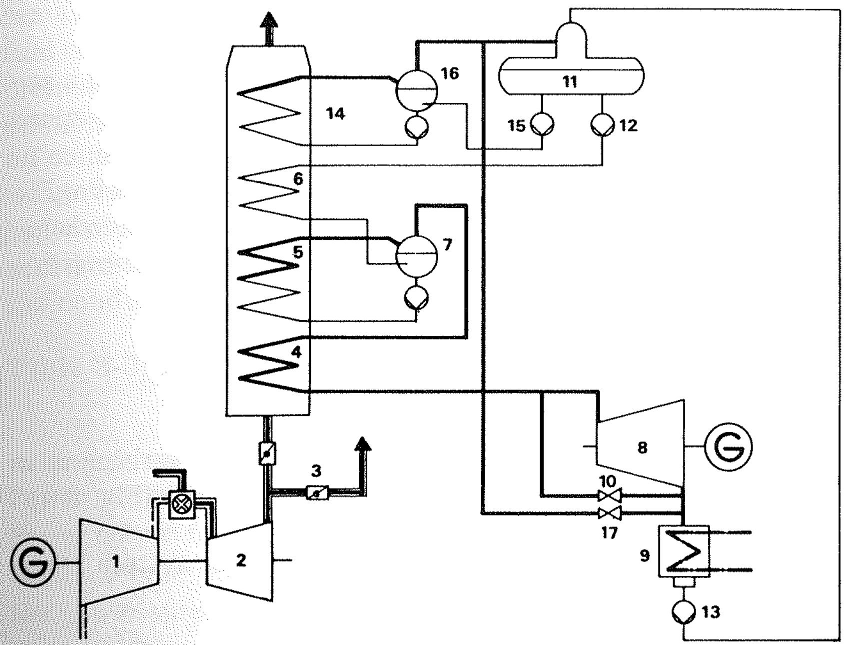

Fig. 3-26: Simplified Flow Diagram of the Single-Pressure System with a Low Pressure Evaporator as a Preheating Loop

1 Compressor
2 Gas turbine
3 Flue gas bypass (optional)
4 Superheater
5 Evaporator
6 Economizer
7 Boiler drum (high pressure)
8 Steam turbine
9 Condenser
10 Steam bypass (high pressure)
11 Feedwater tank, deaerator
12 Feed pump (high pressure)
13 Condensate pump
14 Preheating loop (low pressure evaporator)
15 Feed pump (low pressure)
16 Boiler drum (low pressure) (optional)
17 Steam bypass (low pressure)

10% of that required for the version using water. The evaporator itself can be of the natural circulation or the forced circulation type.

In this second design, it is sometimes possible to avoid a separate low pressure drum. The feedwater tank then functions as a low-pressure drum, resulting in a simple system since no feed pumps or drum level controls are required. However, because of the two-phase flow, special care must be taken when designing the piping and the introduction of the water/steam mixture into the feed-water tank.

Example of a Single-Pressure Combined-Cycle Plant with a Preheating Loop

This is shown in Fig. 3-27, using the same gas turbine as in the example for the simple single-pressure system (Fig. 3-4).

Table 3-3 lists the main technical data of this system when equipped with a low pressure evaporator.

Compared to the simple single-pressure system, it attains a significantly higher steam turbine output, improving overall efficiency by 2.5%. This is because in this case no steam is tapped from the turbine. As a result, the entire live steam flow can expand to the condenser pressure. But the larger volume flow of exhaust steam produced is a certain disadvantage since the dimensions of the steam turbine exhaust and the condenser must be larger.

The increase in the amount of heat to be removed from the condenser is more than proportional to the increase in power output. The energy utilization rate of the waste heat boiler rises by about 15% while the power output from the steam turbine increases only by 8%, since the additional exhaust heat recovered is at a low temperature level. The rate for converting it into mechanical energy (exergy) is therefore modest. The increase in

plant costs, however, is low compared to the improvement in efficiency. This type of system attains a high efficiency, but still remains uncomplicated and accordingly low in cost. Even if the fuel contains very high levels of sulphur, the feedwater can be preheated to sufficiently high a temperature without any reduction in efficiency worth mentioning.

Fig. 3-28 shows the temperature/heat diagram for the waste heat boiler. The exhaust gases are cooled by approximately an additional 50 °C (90 °F) in the preheating loop in order to warm the condensate to 130 °C (266 °F).

Table 3-3: Main Technical Data of the Single-Pressure Combined-Cycle Plant with a Preheating Loop

Power output from the gas turbine	68 000	kW
Power output from the steam turbine	36 800	kW
Station service power required	1 200	kW
Net power output from the plant	104 000	kW
Heat supplied	228 000	kW
Efficiency of the gas turbine	30.0	%
Heat contained in the exhaust gases	157 200	kW
Rate of Waste Heat Energy Utilization*	72.5	%
Efficiency of the steam process	23.4	%
Gross efficiency of the plant	46.1	%
Net efficiency of the plant	45.6	%

* 100% utilization if the exhaust gases are cooled down to 15 °C (59 °F)

Effect of Environmental Conditions on Power Output and Efficiency

Ambient conditions affect the combined-cycle plant with a preheating loop in approximately the same way as the simple single-pressure system (See Section 3.1.2). We will therefore no treat separately the various parameters that depend on the environment.

Fig. 3-29 shows the effects that the temperatures of the air and the cooling water have on the power output and efficiency of the plant as a whole. It is obvious that a rise in air temperature causes a reduction in power output and a slight improvement in overall efficiency. On the other hand, a high temperature for the cooling water affects both parameters negatively.

Effect of the Most Important Design Parameters on Power Output and Efficiency

The effect of most parameters is similar to that for the simple single-pressure system (See Section 3.1.1).

Live Steam Data

The effects of live steam pressure and live steam temperature on the efficiency of the steam cycle are practically the same as for a simple single-pressure system. The optimum live steam pressure is at approximately the same level. Slight shifts toward higher pressure can result due to a larger exhaust steam volume flow.

However, installing a preheating loop in the waste heat boiler imposes a limit on the minimum live steam pressure. As can be seen from Fig. 3-30, the flue gas temperature after the economizer drops when the live steam pressure falls. Because the minimum temperature of the water in the boiler is determined by the sulphuric acid dewpoint, the amount of useful heat in the preheating loop is reduced correspondingly.

If a high feedwater temperature is required, the live steam pressure selected must not be too low. Otherwise a portion of the preheating would have to be done in a low pressure preheater

Figure 3-27

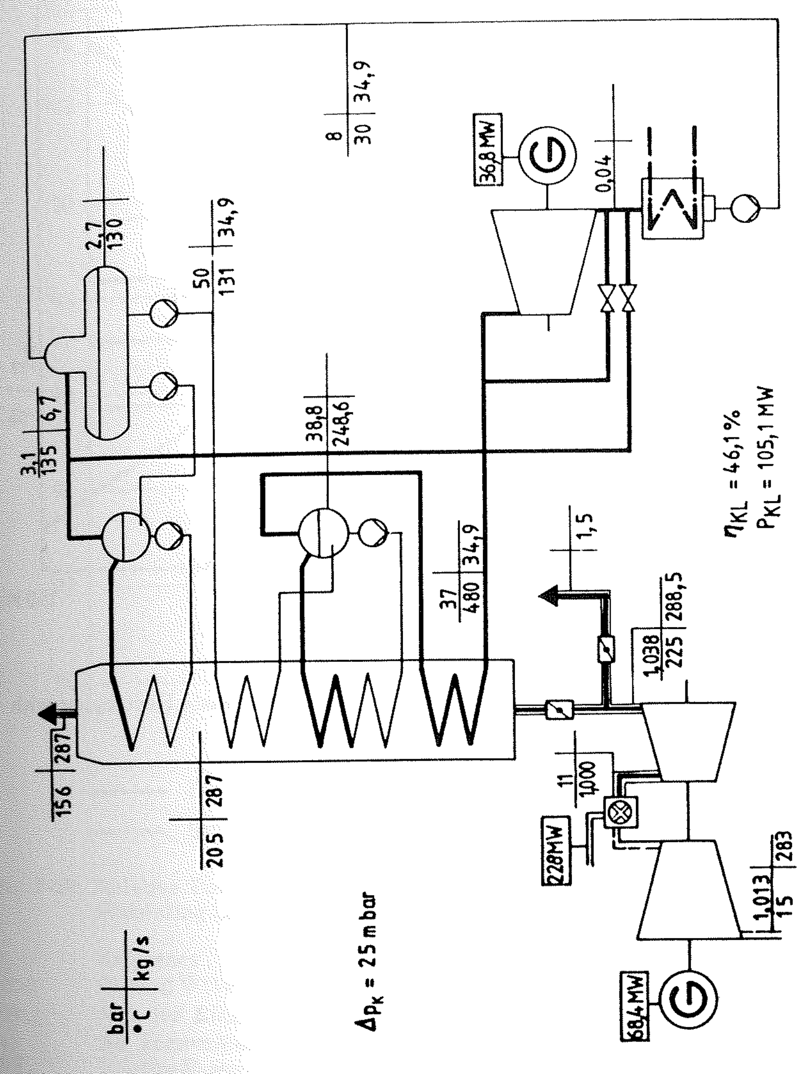

Fig. 3-27: Heat Balance of the Single-Pressure System with a Preheating Loop

Figure 3-28

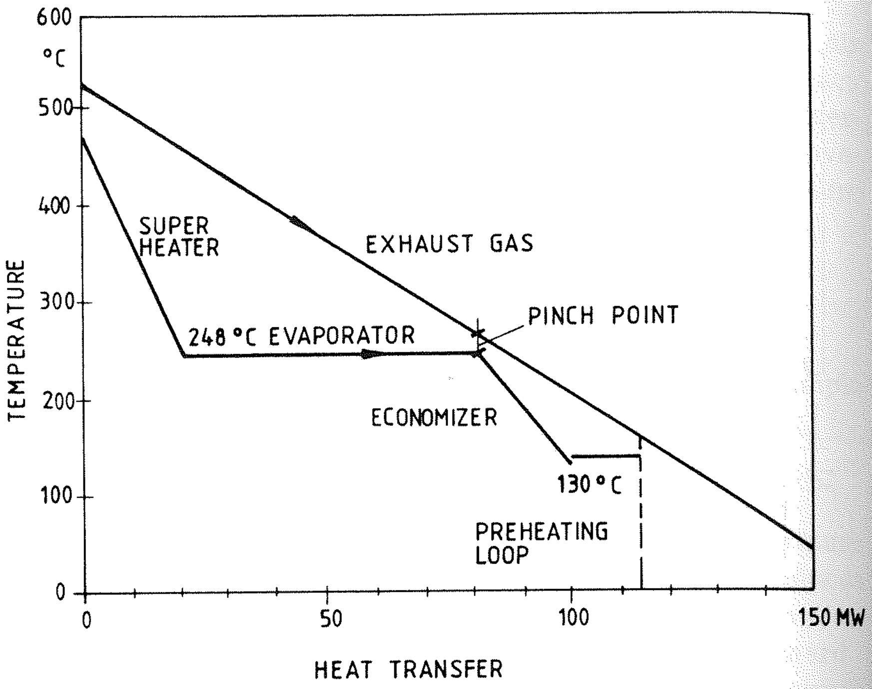

Fig. 3-28: Temperature/Heat Diagram of a Single-Pressure Waste Heat Boiler with a Preheating Loop

Pinch Point = 15°C (27°F)

Figure 3-29

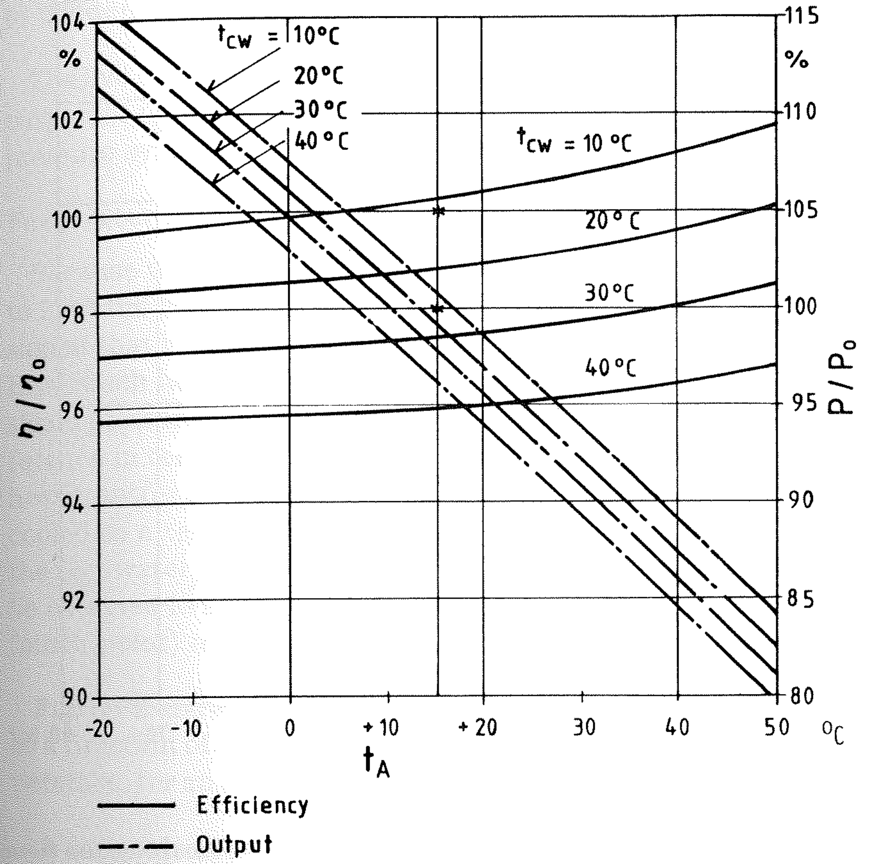

Fig. 3-29: Relative Power Output and Efficiency of Combined-Cycle Plants with a Preheating Loop, as Functions of the Air and Cooling Water Temperatures

*	Reference
t_{CW}	Cooling Water Temperature
R/η_o	Relative Efficiency
t_L	Air Temperature
P/P_o	Relative Power Output

Figure 3-30

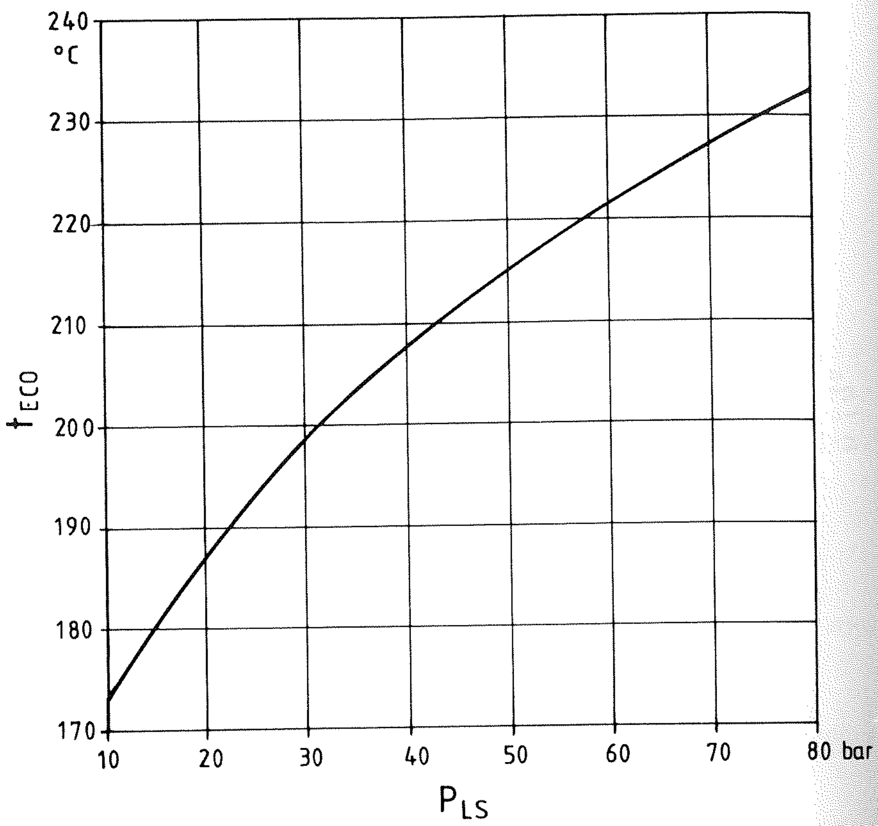

Fig. 3-30: Flue Gas Temperature after the Economizer as a Function of the Live Steam Pressure

t_{Eco}	Flue gas temperature after economizer	
P_{LS}	Live steam pressure	
	Gas turbine exhaust gas flow	288.5kg/s
	Gas turbine exhaust gas temperature	525°C
	Live steam temperature	480°C

heated with extraction steam, thereby reducing the efficiency of the steam cycle because a portion of the steam is not expanded to condenser pressure.

The live steam temperature affects the efficiency (or the power output) of the steam process in the same way as in a simple single-pressure system.

Feedwater Preheating

Because this system uses the waste heat in the exhaust gases to heat the condensate, the preheating loop must be so dimensioned that it can supply the heat required for the condensate. How much heat is available for preheating depends on the live steam pressure and the feedwater temperature. Because the difference in temperature must at least be at a certain level if the heat transfer is to take place, the exhaust gases can at best be cooled to a temperature from 10 to 30 °C (18 to 54 °F) above the feedwater temperature. Fig. 3-31 shows how much heat can be obtained in a preheating loop with a temperature difference (pinch point) of 15 °C (27 °F).

Fig. 3-32 shows the heat required to preheat the feedwater as a function of the condenser pressure and the feedwater temperature, for an average live steam pressure of 30 bar (420 psig).

A comparison of Fig. 3-31 and 3-32 shows that problems are possible if all the pre-heating is done in the preheating loop when the condenser pressure is very low and a high feedwater temperature is called for. These problems occur whenever the fuel contains very high levels of sulphur, which raises the acid dewpoint. For deep vacuums in the condenser, then, it is often necessary to use a low pressure preheater heated with extraction steam to reduce the amount of heat needed in the feedwater tank/deaerator. This defuses the problem of the heat output required for the preheating loop in the waste heat boiler.

A low pressure preheater has a negative effect on the steam process efficiency because less heat is recovered from the exhaust gases. However, the reduced wetness and exit losses in the turbine to a large extent compensate for that negative effect.

Condenser Pressure

The effect of the condenser pressure on the efficiency of the steam process is similar to that in the simple single-pressure system, but the change in efficiency is somewhat more pronounced because the exhaust steam flow is about 10 to 15% greater.

Pinch Point of the Waste Heat Boiler

The effect that the pinch point of the waste heat boiler has on the efficiency of the steam process is similar to that in a simple single-pressure boiler (cf. Section 3.1.1). However, a reduction of the pinch point affects not only the surface of the evaporator and the economizer but also that of the preheating loop. There are two reasons for this:

- The flue gas temperature after the economizer falls, reducing the amount of heat available for the preheating loop.
- The heat required for feedwater heating increases since a greater flow of feedwater is needed for the increased steam production. The preheating loop has to take up more energy.

Other Parameters

We will not investigate the effects of the other design parameters here because they differ only insignificantly from those in a simple single-pressure system.

3.1.3 Two-Pressure System

A single-pressure system with a preheating loop provides better waste heat utilization than a simple single-pressure system. Nevertheless, that utilization is neither energetically nor exergetically optimum. In many cases, the low pressure evaporator could, at no great expense, produce more steam than required to preheat the feedwater and that excess steam could be converted into mechanical energy if it were admitted into the turbine at some suitable point. To do this, the steam turbine must have two steam admissions: one for high pressure, and another for low pressure steam (two-pressure turbine).

Fig. 3-33 shows a system of this type, further equipped with two low pressure pre-heaters. This not only provides better utilization of the waste heat as mentioned above, but also makes better thermodynamic use of the low pressure steam. A larger proportion of the low pressure steam can flow into the turbine through the low pressure preheater, while the feedwater is being pre-heated in the first section using low quality steam.

Before the low pressure steam reaches the turbine, it can be slightly super-heated. The thermodynamic advantage of doing this, however, is minimal because the pressure drop between the steam turbine and the drum is increased. This reduces the amount of steam generated because the saturation temperature in the low pressure evaporator is raised. If the water separation in the drum is effective enough, the saturated steam can be sent directly into the turbine.

When using low-sulphur or sulphur-free fuels, further improvement to this system becomes possible. When the dewpoint is low enough, the exhaust gases can preheat a more or less significant portion of the feedwater in a low temperature economizer. Fig. 3-34 shows an example burning sulphur-free natural gas. The feedwater here is preheated far enough in a deaerator so that its temperature is above the water dewpoint of the exhaust gases (approx. 50 °C) (122 °F). Because this temperature is so low, the deaeration takes place in this case under a vacuum. Following the feedwater tank/dearator, all the feedwater is heated in a

Figure 3-31

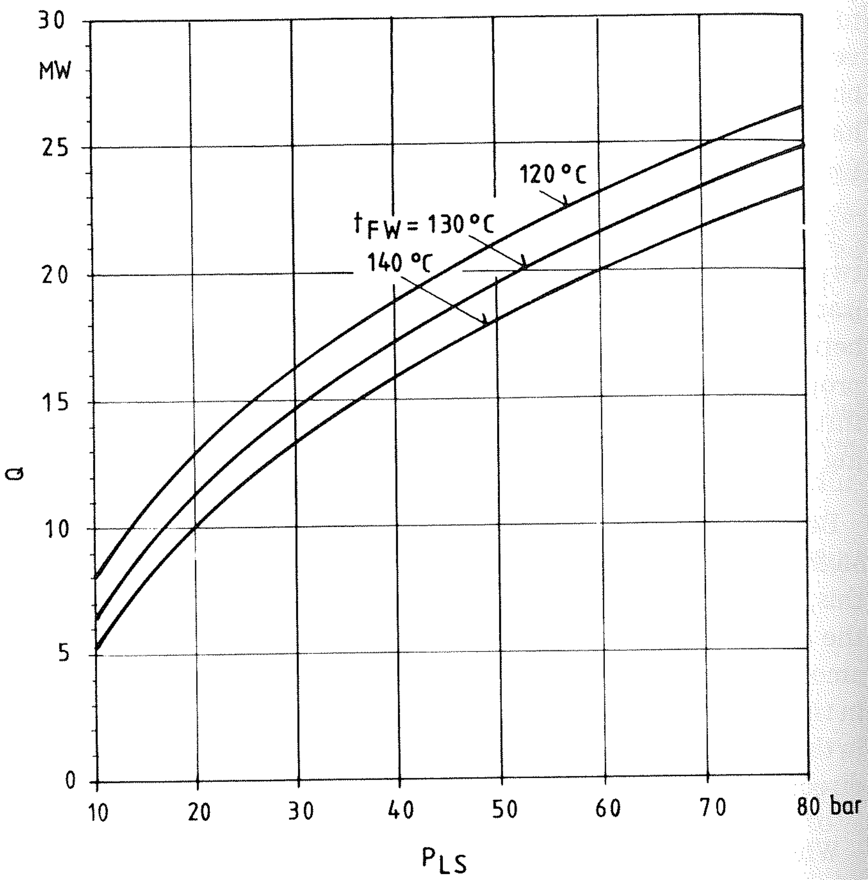

Fig. 3-31: Effect of Live Steam Pressure and Feedwater Temperature on the Usable Heat in the Preheating Loop

Q	Heat output	
t_{FW}	Feedwater temperature	
	Gas turbine exhaust flow	288.5 kg/s
	Gas turbine exhaust gas temperature	525°C
	Live steam temperature	480°C

Figure 3-32

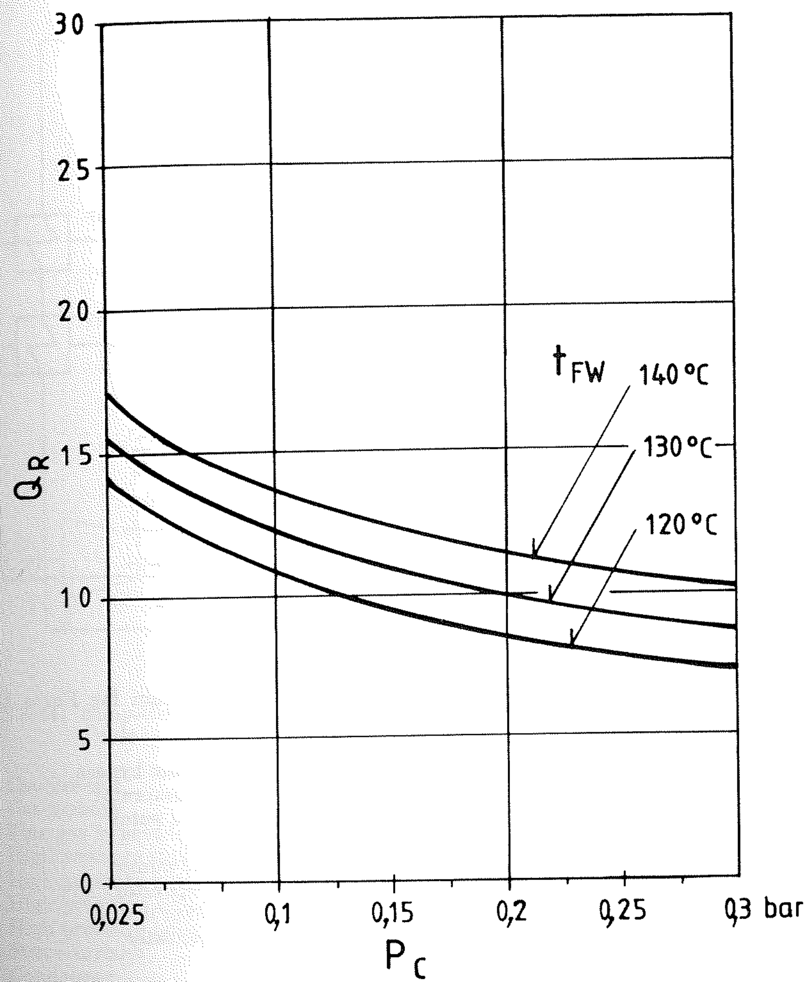

Fig. 3-32: Heat required to Preheat the Feedwater, as a Function of the Condenser Pressure and Feedwater Temperature

Q_R	Heat required	
t_{FW}	Feedwater temperature	
PC	Condenser Pressure	
	Gas turbine exhaust gas flow	288.5 kg/s
	Gas turbine exhaust gas temperature	525°C
	Live steam temperature	480°C
	Live steam flow	34.9 kg/s

Figure 3-33

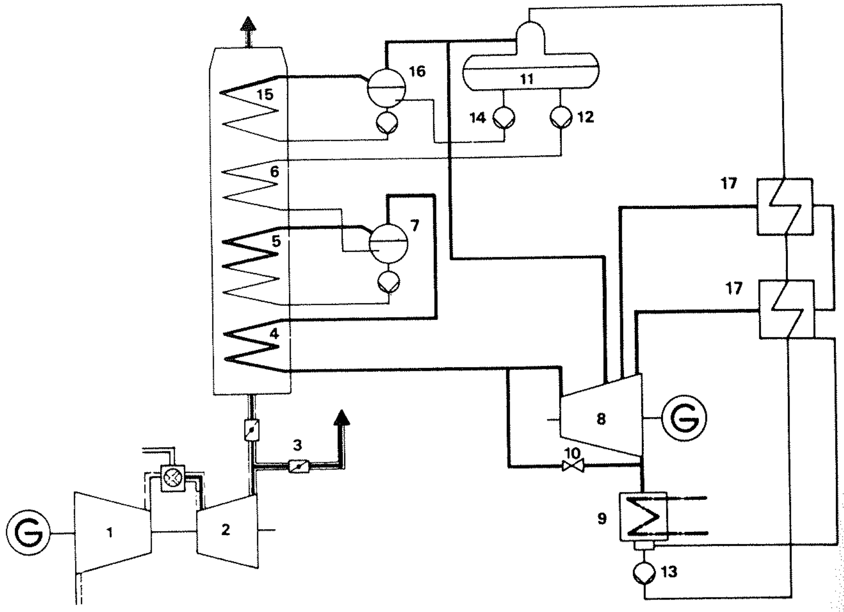

Fig. 3-33: Simplified Flow Diagram for a Two-Pressure System for Fuels that Contain Sulphur

1 Compressor
2 Gas turbine
3 Flue gas bypass (optional)
4 High pressure superheater
5 High pressure evaporator
6 High pressure economizer
7 High pressure boiler drum
8 Steam turbine
9 Condenser
10 High pressure steam bypass
11 Feedwater tank/deaerator
12 High pressure feed pump
13 Condensate pump
14 Low pressure feed pump
15 Low pressure evaporator
16 Low pressure boiler drum
17 Low pressure preheater

Figure 3-34

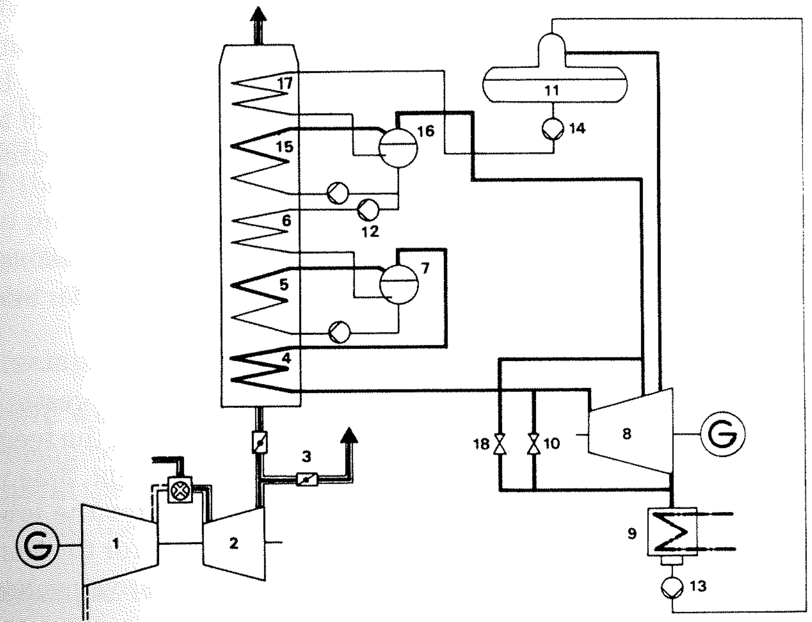

Fig. 3-34: Simplified Flow Diagram for a Two-Pressure System for Sulphur-Free Fuels

1 Compressor
2 Gas turbine
3 Flue gas bypass (optional)
4 High pressure superheater
5 High pressure evaporator
6 High pressure economizer
7 High pressure boiler drum
8 Steam turbine
9 Condenser
10 High pressure steam bypass
11 Feedwater tank/deaerator
12 High pressure feed pump
13 Condensate pump
14 Low pressure feed pump
15 Low pressure evaporator
16 Low pressure boiler drum
17 Low pressure economizer
18 Low pressure steam bypass

low pressure economizer to approximately the saturation temperature of the low pressure steam. It is then admitted to the low pressure drum. Next, a high pressure feedwater pump circulates the feedwater for the high pressure evaporator from the low pressure drum into the high pressure steam generator. In this case, too, it is possible to supply the low pressure steam to the turbine either as saturated steam or as slightly superheated steam.

In addition to this system, there are further variants possible. Most of these are not as good thermodynamically, but offer certain operational advantages.

One example is shown in Fig. 3-35, where the high pressure and low pressure feed-water are separated directly after the feedwater tank. The low pressure economizer shown in Fig. 3-34 is therefore divided into a low pressure economizer for the low pressure feedwater and a high pressure economizer for the first step in preheating the high pressure feedwater. This system has the following advantages:

- better availability, since the high pressure portion can remain in operation even if either the low pressure pump or the circulating pump fails
- fewer problems with steaming out in the low pressure economizer during part-load operation.

On the other hand, a slight reduction of about 5% in low pressure steam generation must be accepted in most cases.

Another possibility that operates without vacuum deaeration is shown in Fig. 3-36. The deaerator here operates at a slight overpressure. To do this, it requires extraction steam of a better quality than that in a system with vacuum deaeration. To keep flows within reasonable limits, the condensate is preheated by the feed-water in a water-to-water heat exchanger. This means

that most of the feedwater preheating is still being accomplished using exhaust gas heat. The boiler feed-water temperature must not drop below the water dewpoint (if the fuel is sulphur-free) or the acid dewpoint (if it contains sulphur).

The disadvantage here is the reduced efficiency resulting from withdrawing higher quality steam from the turbine. Moreover, if the condensation pressure is low, it may become necessary to provide another low pressure pre-heater heated with extraction steam in order to reduce wetness at the end of the turbine. That would reduce the power output slightly further.

Examples of Two-Pressure Combined-Cycle Plants

We will discuss here examples of two typical two-pressure combined-cycle plants, both based on the same gas turbine as that in the single-pressure systems. The first is designed for burning oil, the second for burning sulphur-free natural gas.

Example 1: Two-pressure system for fuels containing sulphur

Fig. 3-37 shows the main technical data for this unit. The major difference from the single-pressure system with a preheating loop lies in the 3-stage feedwater preheating. Two low-pressure preheaters heated with extraction steam reduce the amount of steam required for the deaearator, which supplies quite a large amount of excess steam to the low pressure steam turbine where it produces additional mechanical energy. The live steam pressure has been raised to 60 bar (870 psia) in order to improve the efficiency of the steam process. Unlike the single-pressure systems, this system is not significantly affected by the poor rate of heat utilization in the high pressure portion of the waste heat boiler because the heat that is not utilized is recovered in the low pressure portion. Table 3-3 (page 83) shows the main technical data of this plant.

Figure 3-35

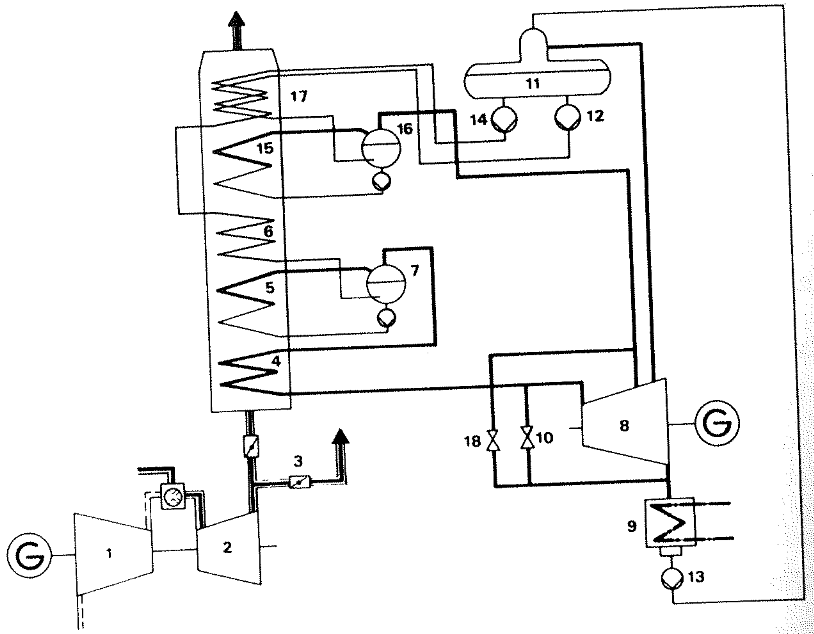

Fig. 3-35: Two-Pressure System with Separate Economizer in the Low Temperature Range

1 Compressor
2 Gas turbine
3 Flue gas bypass (optional)
4 High pressure superheater
5 High pressure evaporator
6 High pressure economizer
7 High pressure boiler drum
8 Steam turbine
9 Condenser
10 High pressure steam bypass
11 Feedwater tank/deaerator
12 High pressure feed pump
13 Condensate pump
14 Low pressure feed pump
15 Low pressure evaporator
16 Low pressure boiler drum
17 Low pressure economizer
18 Low pressure steam bypass

Figure 3-36

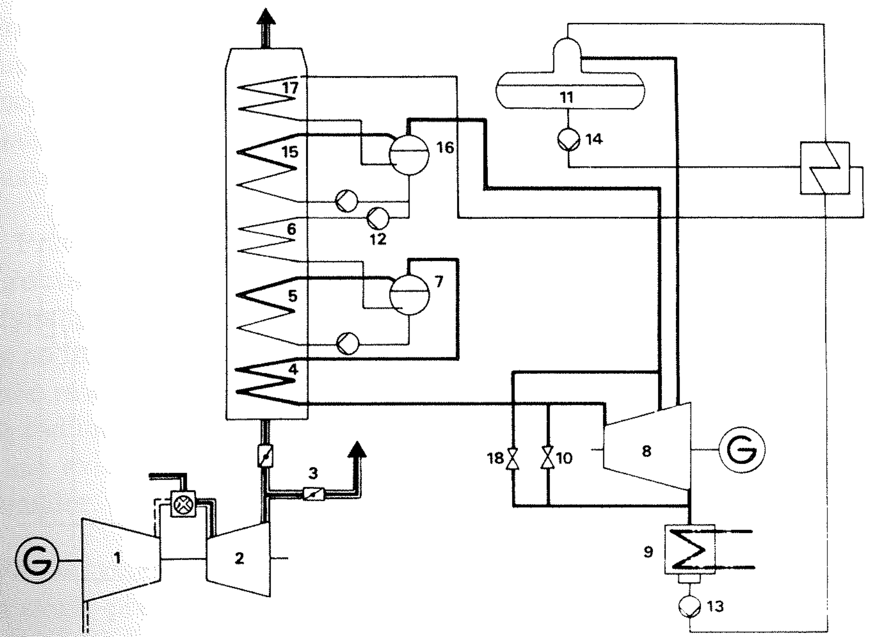

Fig. 3-36: Two-Pressure System with Feedwater Used to Preheat Condensate

1 Compressor
2 Gas turbine
3 Flue gas bypass (optional)
4 High pressure superheater
5 High pressure evaporator
6 High pressure economizer
7 High pressure boiler drum
8 Steam turbine
9 Condenser
10 High pressure steam bypass
11 Feedwater tank/deaerator
12 High pressure feed pump
13 Condensate pump
14 Low pressure feed pump
15 Low pressure evaporator
16 Low pressure boiler drum
17 Low pressure economizer
18 Low pressure steam bypass
19 Feedwater preheater

Figure 3-37

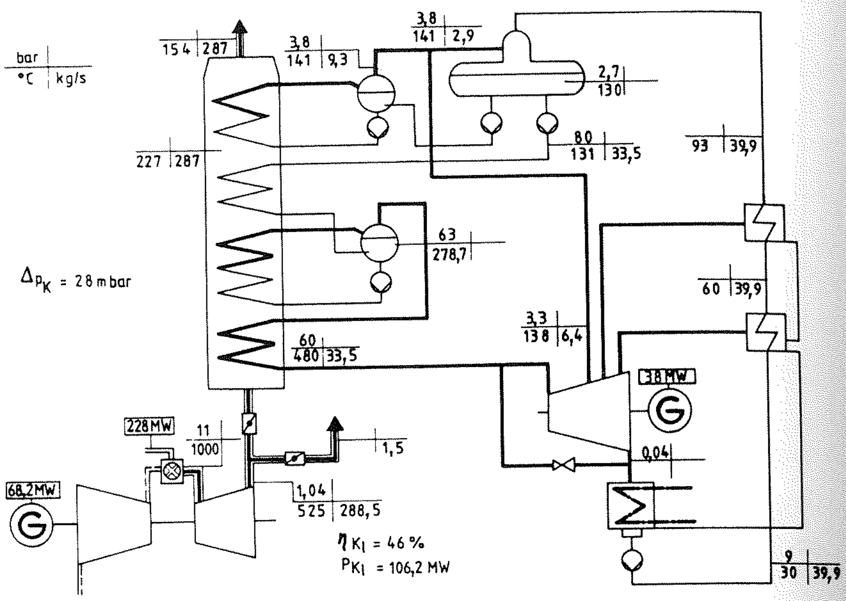

Fig. 3-37: Heat Balance of the Two-Pressure System for Fuels Containing Sulphur

Because the system has been improved thermodynamically, the power output from the steam turbine is more than 11% greater than that from the simple single-pressure system and more than 3% greater than that in the system with a pre-heating loop. The increase in station service power required only partially detracts from this gain: net efficiency rises to approximately 46%.

Practically the entire increase in efficiency over a single-pressure system with a preheating loop is provided by the thermodynamic improvements made to the water/steam cycle. There is no improvement worth mentioning in the waste heat utilization rate. The amount of heat to be carried off in the condensor is reduced somewhat so that the dimensions of the steam turbine exhaust and the condenser may perhaps be somewhat smaller.

This system is significantly more complex than a system with a single-pressure turbine and is of interest only if the gain in power output promises sufficient economic gain.

Example 2: Two-pressure system for sulphur-free fuels

If the fuel contains no sulphur, the efficiency of the steam process can be raised by increasing the rate of waste heat utilization in the steam generator. Fig. 3-38 shows the heat balance for a typical plant burning sulphur-free natural gas. The low pressure economizer at the end of the waste heat steam generator makes it possible to cool the exhaust gases down to practically 100 °C (212 °F). Because the feedwater temperature is then only about 60 °C (140 °F), the two low pressure preheaters in Example 1 are no longer needed. Otherwise, the design is the same. Steam production is somewhat greater however because the exhaust gases from a natural gas firing contain more heat. Table 3-4 (page 84) shows the main technical data of this plant.

Figure 3-38

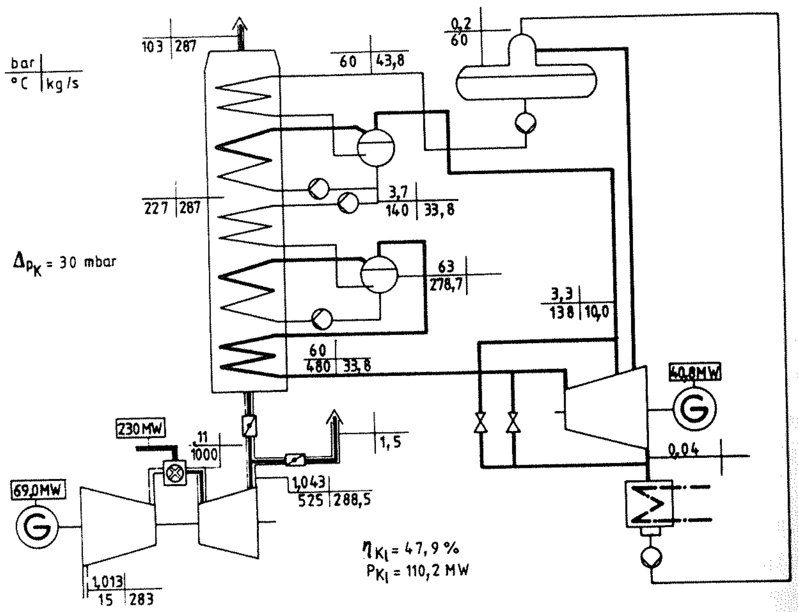

Fig. 3-38: Heat Balance of the Two-Pressure System for Sulphur-Free Fuels

There are two reasons why the efficiency is better than that in Example 1:

- Burning natural gas increases efficiency (approx. 40% of the improvement)
- Better utilization of the waste heat (approx. 60% of the improvement)

Because its temperature is low, the additional heat absorbed cannot efficiently be converted into work. A large portion of it is carried off again in the condenser. The exhaust steam flow from the turbine and the cooling water system are approximately 20% larger than in Example 1, increasing the costs for the plant. However, the improvement in efficiency is great enough so that the increase in investment costs is in most cases worthwhile if sulphur-free gas is being burned.

Fig. 3-39 shows the heat flow diagram for Example 2. Compared to the simple single-pressure system (See Fig. 3-5), the sharp reduction in stack losses (V2) and the significantly greater condenser losses (V1) are striking.

Fig. 3-40 is the temperature/heat diagram of the steam generator. As it shows, approximately 70% of the heat exchange takes place in the high pressure portion and approx. 30% in the low pressure portion. That corresponds approximately to the ratios of high pressure and low pressure steam generation respectively.

Effect of Ambient Conditions on Power Output and Efficiency

Fig. 3-41 shows the effect of the two most important ambient conditions, air and cooling water temperatures, for the plant with a low-pressure economizer. As in the case of single-pressure systems, an increase in the air temperature affects overall efficiency of the plant positively. In this case, the gradient is even

Figure 3-39

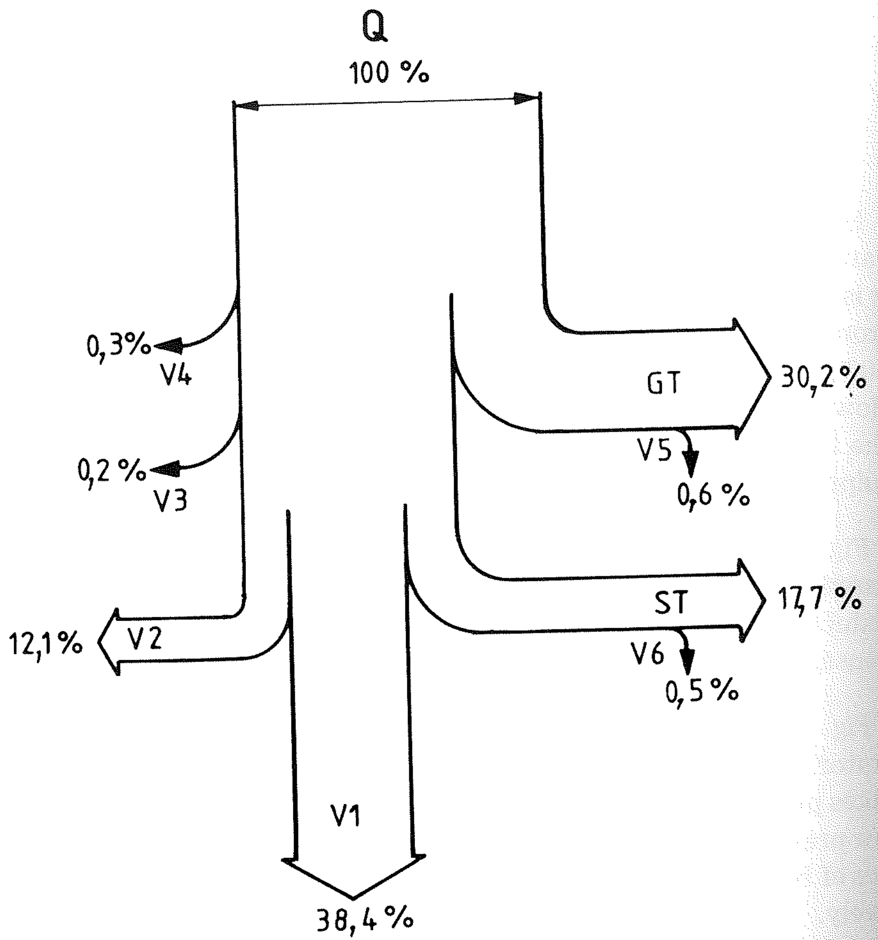

Fig. 3-39: Energy Flow Diagram for the Two-Pressure Combined-Cycle Plant with a Low Pressure Economizer

Q	Energy input
V1	Condenser Loss
V2	Stack Loss
V3	Loss due to Radiation in the Waste Heat Boiler
V4	Loss in the Flue Gas Bypass
V5	Loss due to the Gas Turbine Generator and Radiation
V6	Loss due to the Steam Turbine Generator and Radiation
GT	Electricity Produced in the Gas Turbine
ST	Electricity Produced in the Steam Turbine

Figure 3-40

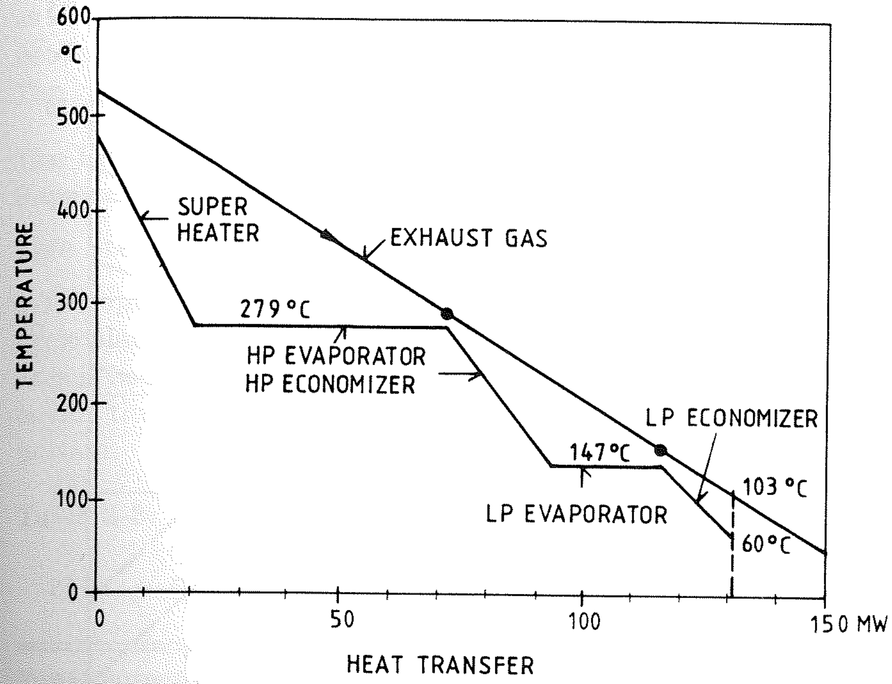

Fig. 3-40: Temperature/Heat Diagram of a Two-Pressure Waste Heat Boiler with a Low Pressure Economizer

Figure 3-41

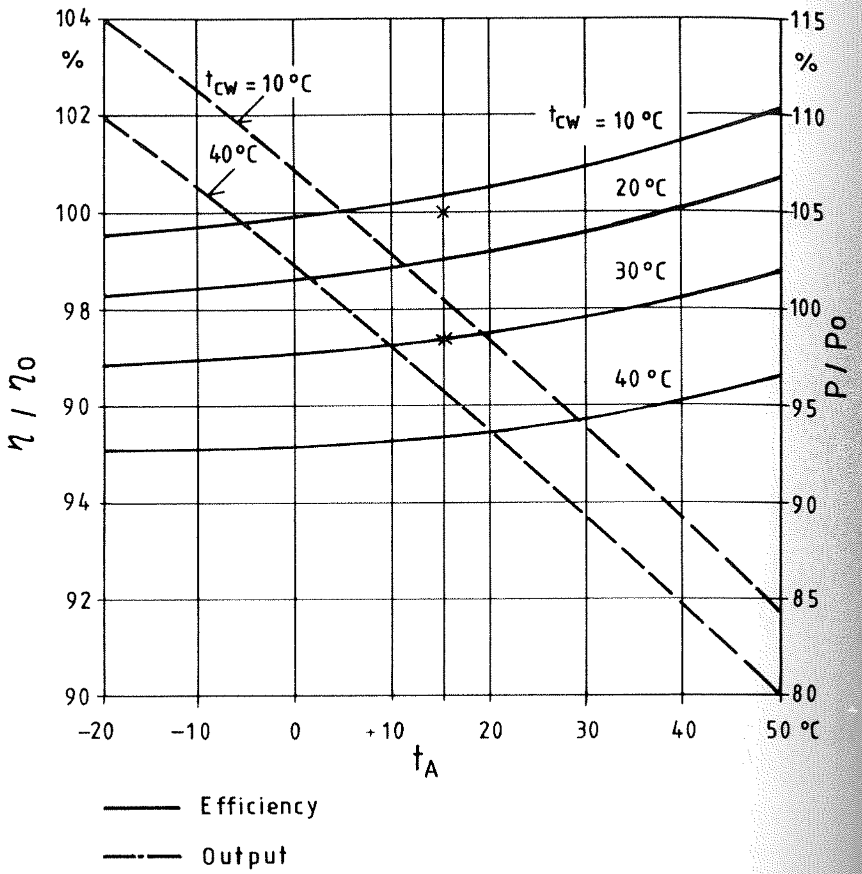

Fig. 3-41: Relative Power Output and Efficiency of Two-Pressure Combined-Cycle Plants as Functions of the Air and Cooling Water Temperatures

*	Reference
η/η_o	Relative Efficiency
t_A	Air Temperature
P/P_o	Relative Power Output
t_{CW}	Cooling Water Temperature

higher because the improvement is produced by the steam process, whose share in the overall power output is greater. Again, as in the case of single-pressure systems, the air pressure does not have any effect on the efficiency of the combined-cycle plant. The change in power output, however, is proportional.

The Effect of the Most Important Design Parameters on Power Output and Efficiency

Live Steam Data

The optimum live steam data for single-pressure combined-cycle plants are not the same as those for a two-pressure plant. This is because of the low pressure steam generator which recovers the waste heat that is not utilized in the high pressure portion.

Table 3-3: Main Technical Data of the Two-Pressure Combined-Cycle Plant for Fuels Containing Sulphur

Power output of the gas turbine	68 200	kW
Power output of the steam turbine	38 000	kW
Station service power required	1 320	kW
Net power output of the plant	104 900	kW
Heat supplied	228 000	kW
Efficiency of the gas turbine	29.9	%
Heat contained in the exhaust gases	157 400	kW
Rate of waste heat energy utilization*	73.0	%
Efficiency of the steam process	24.1	%
Gross efficiency of the plant	46.6	%
Net efficiency of the plant	46.0	%

* 100% Utilization if the exhaust gases are cooled down to 15 °C (59 °F)

Table 3-4: Main Technical Data of the Two-Pressure Combined-Cycle Plant for Sulphur-Free Fuels

Power output of the gas turbine	69 400	kW
Power output of the steam turbine	40 800	kW
Station service power reired	1 200	kW
Net power output of the plant	109 000	kW
Heat supplied	230 000	kW
Efficiency of the gas turbine	30.2	%
Heat contained in the exhaust gases	159 300	kW
Rate of waste heat energy utilization rate*	82.4	%
Efficiency of the steam process	25.6	%
Gross efficiency of the plant	47.9	%
Net efficiency of the plant	47.4	%

* 100% Utilization if the exhaust gases are cooled down to 15 °C (59 °F)

Live Steam Pressures

Two aspects enter into consideration when selecting the high pressure and low pressure live steam pressures:

- The high pressure steam pressure must be relatively high to attain good exergetic utilization of the waste heat.
- The low pressure steam pressure must be low to attain good energetic utilization of the waste heat.

Fig. 3-42 shows the efficiency (or the power output) of the steam process for the second example as a function of the high pressure live steam pressure. Other parameters remain unchanged. Fig. 3-43 shows the same data as a function of the low pressure live steam pressure.

Figure 3-42

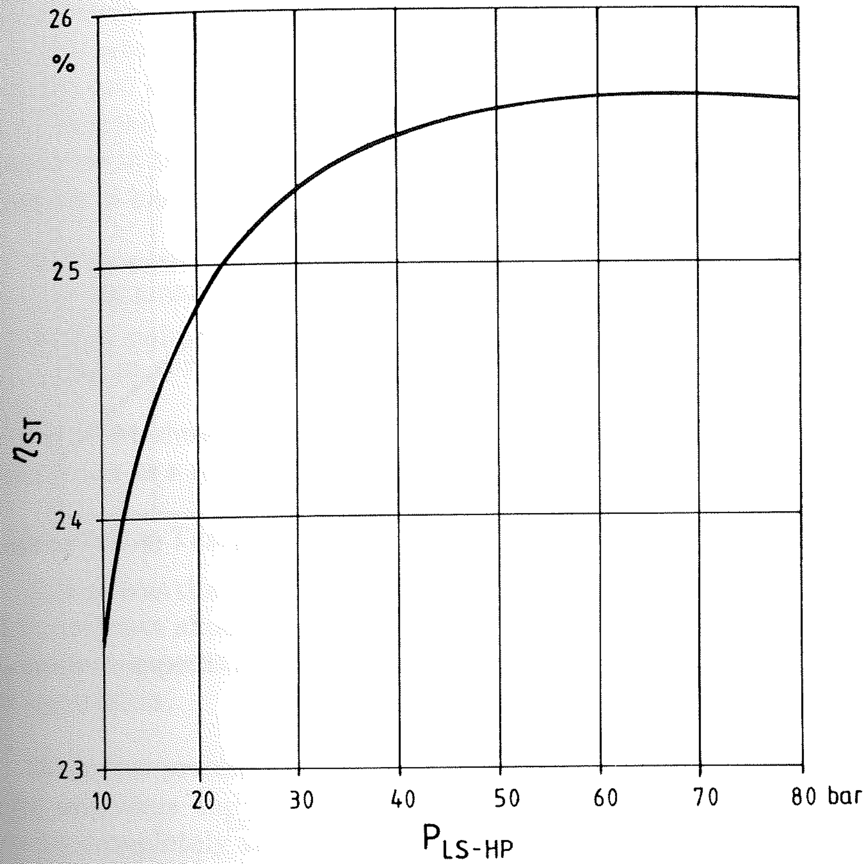

Fig. 3-42: Effect of the High Pressure Steam Pressure on the Efficiency of the Steam Process

η_{ST}	Efficiency of steam process	
P_{LS-HP}	High pressure live steam pressure before turbine	
	High temperature live steam temperature	475°C
	Low pressure live steam pressure	3.3 bar
	Gas turbine exhaust gas temperature	525°C

Figure 3-43

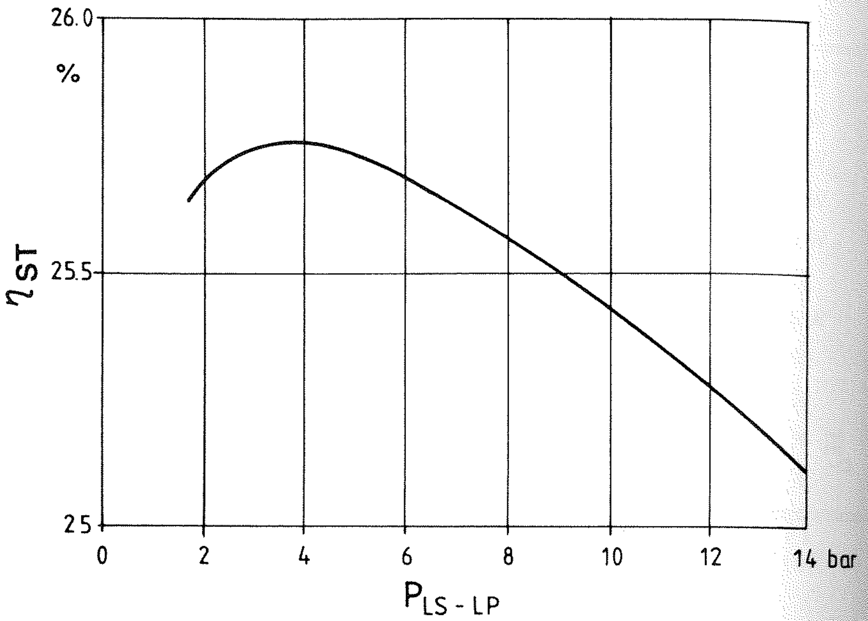

Fig. 3-43: Effect of the Low Pressure Steam Pressure on the Efficiency of the Steam Process

ηST	Efficiency of steam process	
$P_{LS\text{-}LP}$	Low pressure live steam pressure before turbine	
	High temperature live steam temperature	475°C
	High pressure live steam pressure	57 bar
	Low pressure live steam temperature	200°C

The paths of the two curves explains the contrary functions of the high pressure and the low pressure evaporators. The purpose of the first is to generate high quality steam, that of the second is to utilize the remaining waste heat as fully as possible, which can be accomplished only if the pressure in the evaporator is relatively low. However, there are two reasons why the pressure in the low pressure evaporator should not drop below approx. 3 bar (44 psia):

- The enthalpy drop available in the turbine becomes very small, and
- The volume flow of steam becomes very large, resulting in correspondingly large duct cross-sections.

Fig. 3-44 shows the rate of waste heat energy utilization in the boiler as a function of the low pressure live steam pressure.

Live Steam Temperatures

Just as was the case for single-pressure systems, the live steam temperature should here, too, be as high as possible, without however approaching too closely the gas turbine exhaust gas temperature.

In the low pressure evaporator, to be sure, a higher superheating improves the efficiency slightly (Figure 3-45). However, doing completely without a super-heater provides the advantage of reducing the pressure drop between the evaporator and the steam turbine. All in all, the lack of superheating is compensated for almost completely.

In selecting the low pressure live steam temperature, the difference in temperature between the high pressure steam after expansion and the low pressure steam at the mixing point in the turbine must be taken into account. If the difference is too great, that causes unnecessary thermal stresses within the machine. But a high low pressure steam temperature presents the

Figure 3-44

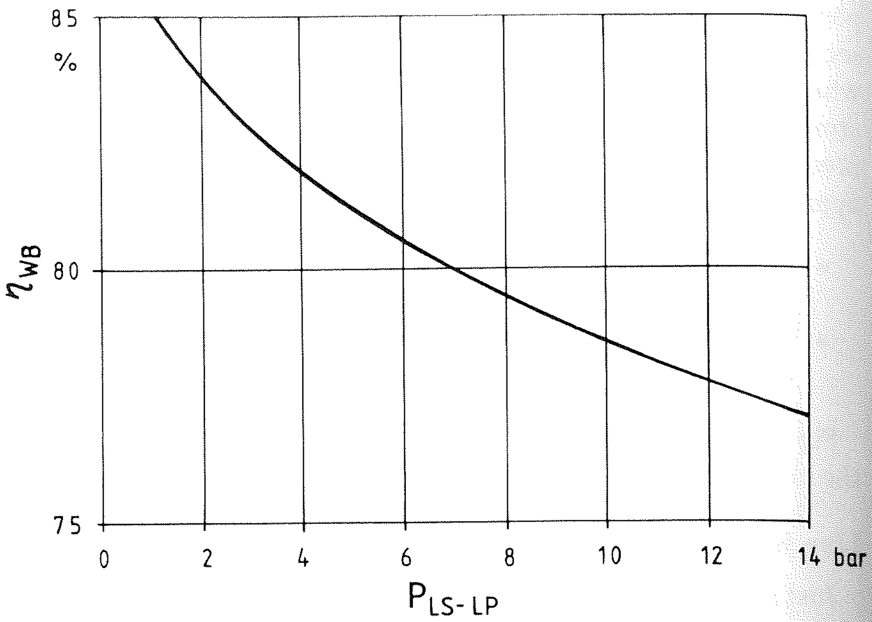

Fig. 3-44: Effect of the Low Pressure Steam Pressure on the Rate of Waste Heat Energy Utilization

ηWB	Rate of waste heat energy utilization	
$P_{LS\text{-}LP}$	Low pressure live steam pressure before turbine	
	High temperature live steam temperature	475°C
	High pressure live steam pressure	57 bar
	Low pressure live steam temperature	200°C

Figure 3-45

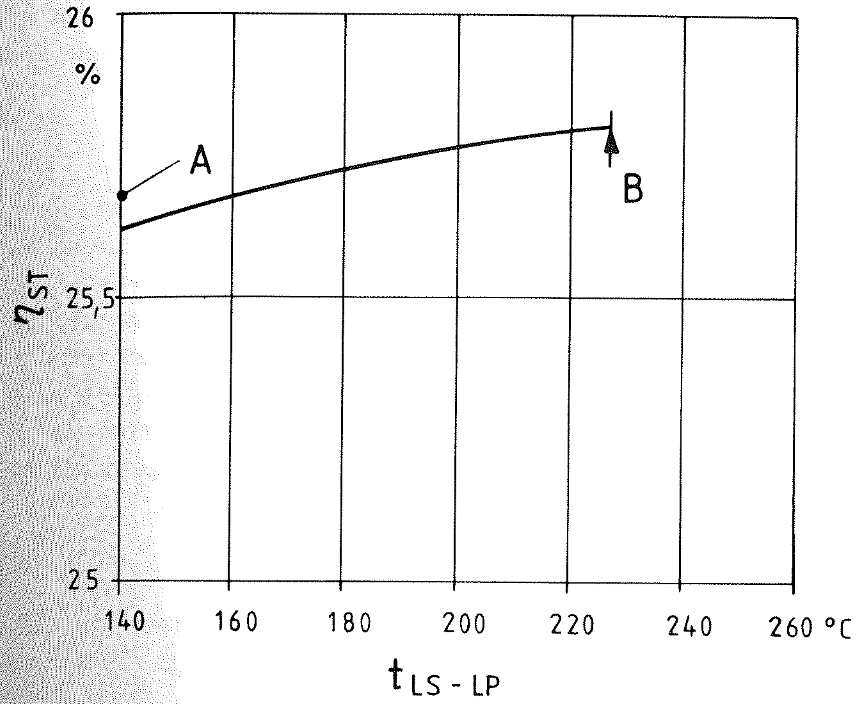

Fig. 3-45: Effect of the Low Pressure Steam Temperature on the Efficiency of the Steam Process

ηST	Efficiency of steam process	
$P_{LS\text{-}LP}$	Low pressure live steam temperature	
	High temperature live steam temperature	475°C
	High pressure live steam pressure	57 bar
	Gas turbine exhaust gas temperature	200°C
A	with a low pressure superheater	
B	Limit, T on low pressure superheater = 0°C	

advantage of a kind of a low "reheating," reducing the risk of erosion due to wetness in the turbine. This consideration can be the reason for installing a low pressure superheater, particularly if the pressure of the high pressure live steam is high and that in the condenser is low.

Feedwater Preheating

As in the case of the simple single-pressure system, the feedwater temperature greatly affects the efficiency of the steam process since it directly influences the rate of the waste heat utilization in the boiler. If it is necessary, in order to prevent low temperature corrosion, to have a high feedwater temperature, multi-stage preheating should be provided (1 to 2 low pressure preheaters and 1 deaerator). Fig. 3-46 shows how the feedwater temperature and the number of pre-heaters affect the efficiency of the steam process in such a case.

Condenser Pressure

Fig. 3-47 shows how the condenser pressure affects the efficiency or the power output of the steam process in the second example.

A deterioration in the condenser vacuum has a greater effect here than with single-pressure systems because the exhaust steam flow is greater. In the first example, where feedwater temperatures are higher, the effect is approximately the same as for the system with a preheating loop, since the exhaust steam flows are similar in both cases.

Pinch Point of the Waste Heat Boiler

The pinch point of the high pressure evaporator is less important here than with a single-pressure system because the heat that is not utilized is recovered in the low pressure evaporator. The loss in power output is due only to the difference in exergy

Figure 3-46

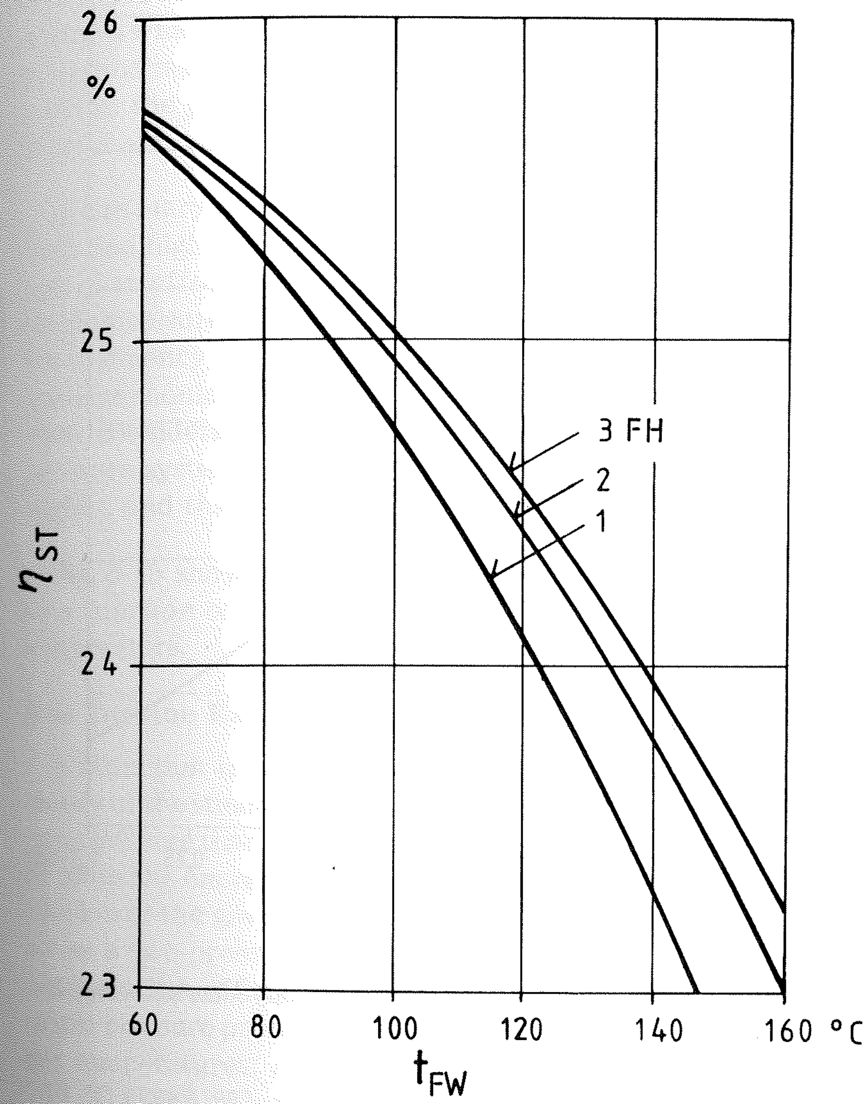

Fig. 3-46: Effect of the Feedwater Temperature and the Number of Low Pressure Preheaters on the Efficiency of the Steam Process

FH	Number of stages for feedwater preheating
ηST	Efficiency of steam process
t_{FW}	Feedwater temperature

Figure 3-47

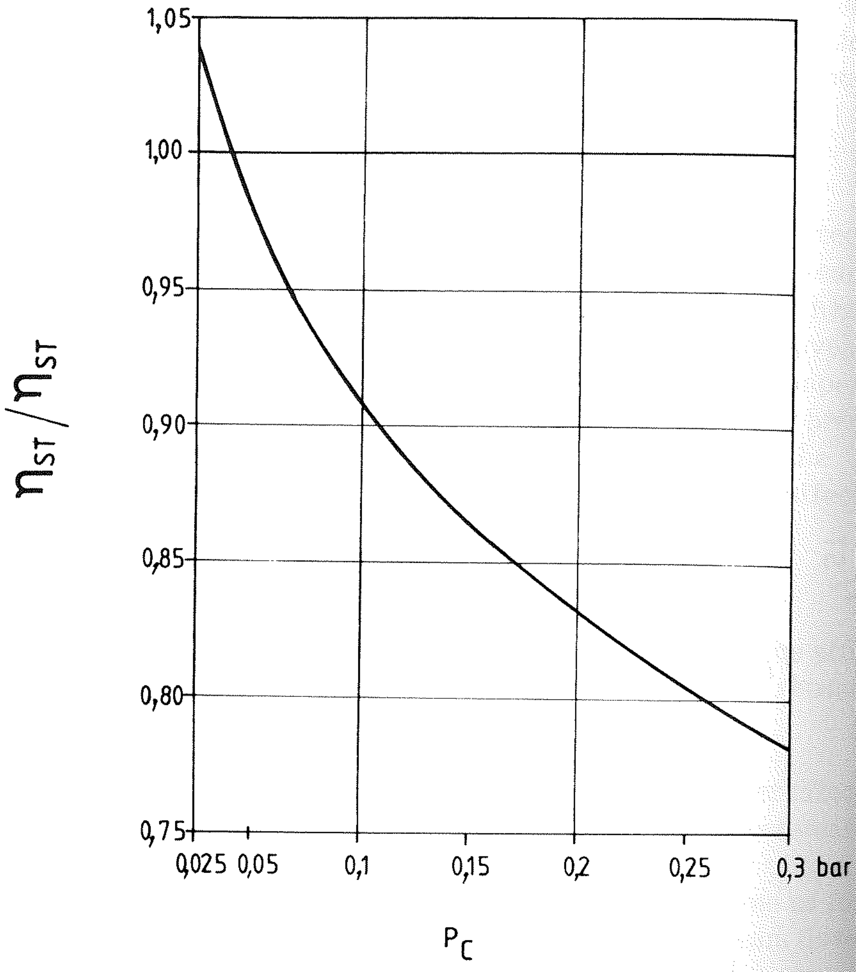

Fig. 3-47: Effect of the Condenser Pressure on the Efficiency of the Steam Process

	Feedwater temperature	60°C
ηST/ηSTo	Relative efficiency of the steam process	
PC	Condenser pressure	

between the high pressure and the low pressure steam portions. For that reason, it is about half as great as with single-pressure systems where the useful pressure drop in the low pressure steam is approximately half that of the high pressure steam. For that reason, the pinch point selected for the low pressure portion should also not be too low.

In summary: With a two-pressure system, the pinch points of both the high pressure and the low pressure evaporators have less of an effect on the efficiency of the steam proces than with single-pressure systems. If equal economic value is attached to the efficiency, then, the pinch points selected for two-pressure systems should be larger than those for single-pressure systems. That consideration is purely academic, however, since two-pressure systems are selected only where efficiency is valued highly, and that in turn means low pinch points.

Fig. 3-48 shows the relative efficiency of the steam process as a function of the pinch points of the high and low pressure evaporators.

Gas Turbine Exhaust Gas Temperature

A reduction in the exhaust gas temperature lowers the efficiency of the steam process. This reduction, however, is less pronounced here than with the single-pressure system (Cf., Fig. 3-24) because the energy utilization rate does not drop off as quickly. The lower the gas turbine exhaust gas temperature is, the more sense a two-pressure system makes. Fig. 3-49 shows the ratio between the efficiencies of the two-pressure and the simple single-pressure processes as a function of the gas turbine exhaust gas temperatures. At a theoretical exhaust gas temperature of 750 °C (1382 °F), this ratio is pracically equal to 1. This fact is put to use in systems that have supplementary firing (refer to Section 2).

Figure 3-48

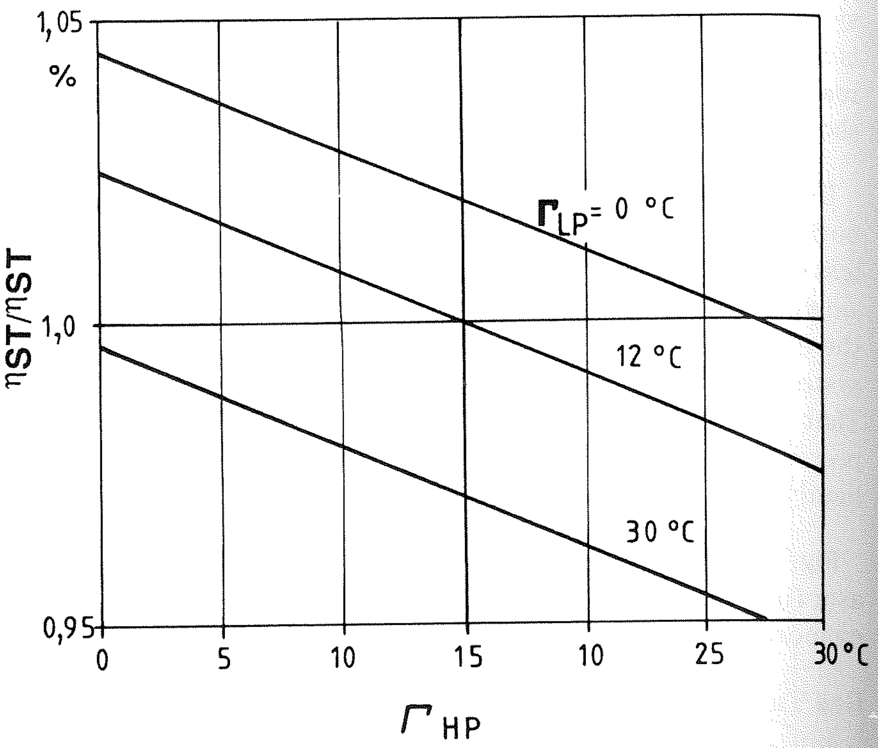

Fig. 3-48: Effect of the Pinch Points of the High Pressure and Low Pressure Evaporators on the Efficiency of the Steam Process

τHP Pinch point of the high pressure evaporator
τLP Pinch point of the low pressure evaporator
ηST/ηSTo Relative efficiency of the steam process

Figure 3-49

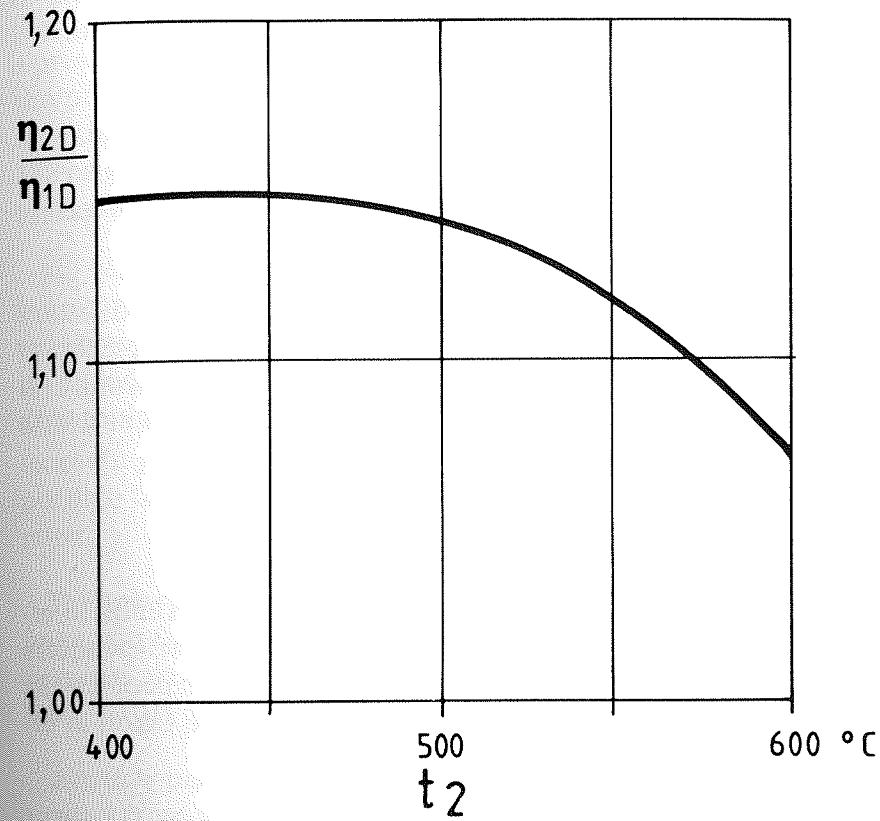

Fig. 3-49: Ratio of Efficiency of the Two-Pressure System to that of the Simple Single-Pressure Process, as a Function of the Gas Turbine Exhaust Gas Temperature

t2 Gas turbine exhaust gas temperature
η2D Efficiency of the two-pressure system
η1D Efficiency of the single-pressure system

3.1.4 Special Systems

In addition to the four systems discussed in Section 3.1.1 to 3.1.3, there are also others that can at times prove useful. We will cite two examples:

- A system with steam or water injection into the gas turbine to reduce nitrogen oxide emissions (NO_X)
- A system using a single waste heat boiler for two gas turbines

The system with steam or water injection is taking on increased importance as environmental protection regulations become ever more stringent; the system using a single waste heat boiler for two gas turbines is of interest mainly for smaller machines with unit power ratings of between 5 and 30 MW.

System with Steam or Water Injection into the Gas Turbine

Environmental protection laws such as those currently in effect in the USA, Japan, and in most European countries require that the NO_X levels in the exhaust be very low. With present-day gas turbine combustors, special measures must be taken in order to maintain these levels.

One way to reduce the formation of NO_X during combustion is by lowering the temperature of the flame, since the speed of the reaction producing NO_X is noticeably rapid only at very high temperatures. Injecting water or steam into the combustor can produce the temperature reduction desired (refer to Section 9.1).

For gas turbines alone, with no waste heat boiler, it is easier to inject water but efficiency is lower than with steam injection.

The problem with steam injection is finding steam at the suitable pressure level. Live steam is generally either at too high (high pressure live steam) or too low (low pressure live steam) a pressure. Depending on the gas turbine and the load involved, the pressure level required for steam injection is, at least for large industrial gas turbines, between 15 and 25 bar (203 and 350 psig). Using reduced high pressure steam is the simplest and the least expensive solution but is exergetically undesirable.

Fig. 3-50 shows an improved solution which employs a three-pressure boiler with a standard high pressure portion, a medium-pressure evaporator for generating the injection steam, and a low pressure portion for preheating the feedwater. The arrangement is relatively complicated. It can be simplified by extracting steam from the turbine (Fig. 3-51), which makes it possible to use the standard systems without additional equipment.

The answer as to which of these two systems is the better must be sought from one case to the next. It is certain that the three-pressure arrangement attains a slightly higher efficiency at full load, but it costs more.

Another possible disadvantage of a solution employing steam tapped from the turbine might be part-load operation in installations with several gas turbines. Unless all the gas turbines are in operation, the pressure at the extraction point decreases so far that it is in most cases inadequate. It thus becomes necessary to switch over to live steam, which again negatively affects efficiency. The three-pressure system is better in this regard. With that system, if the amount of injection steam generated is insufficient, only enough live steam need be used to cover the actual shortage itself.

Table 3-5 shows a comparison between the single-pressure system with a pre-heater loop without steam injection into the gas turbine and the same system with injection of extracted steam

Figure 3-50

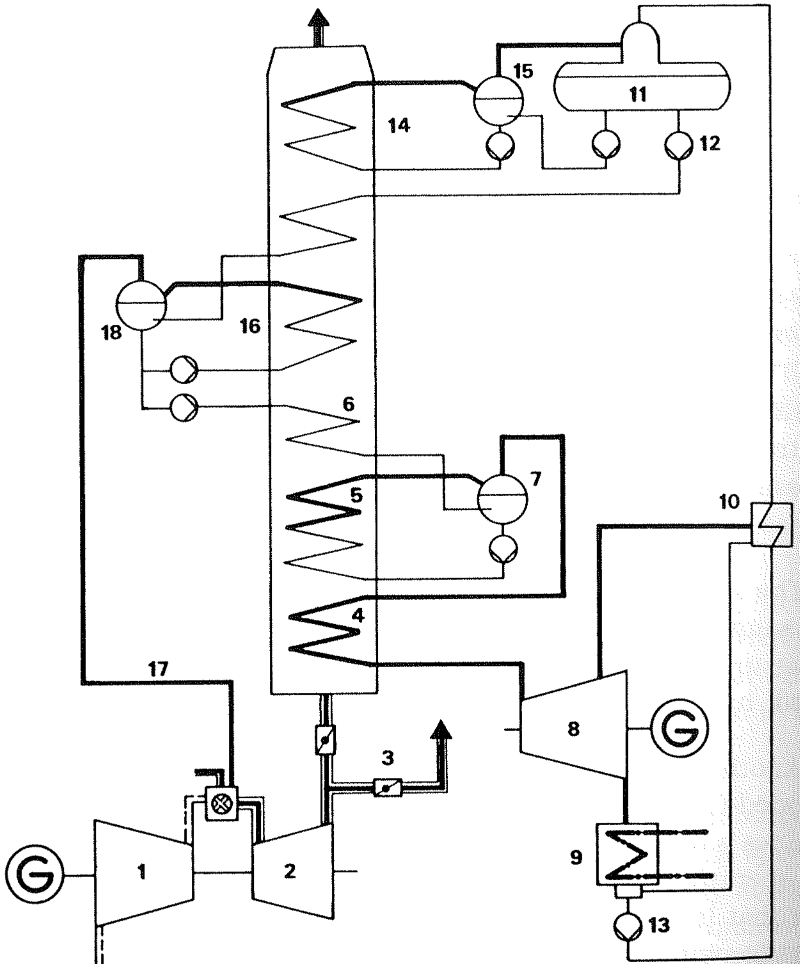

Fig. 3-50: System with 3-pressure boiler for Steam Injection into the Gas Turbine

1 Compressor
2 Gas turbine
3 Flue gas bypass (optional)
4 High pressure superheater
5 High pressure evaporator
6 High pressure economizer
7 High pressure boiler drum
8 Steam turbine
9 Condenser
10 Low pressure preheater
11 Feedwater tank/deaerator
12 High pressure feedwater pump
13 Condensate pump
14 Low pressure evaporator
15 Low pressure drum
16 Medium-pressure evaporator
17 Steam injection line
18 Medium- pressure drum

Figure 3-51

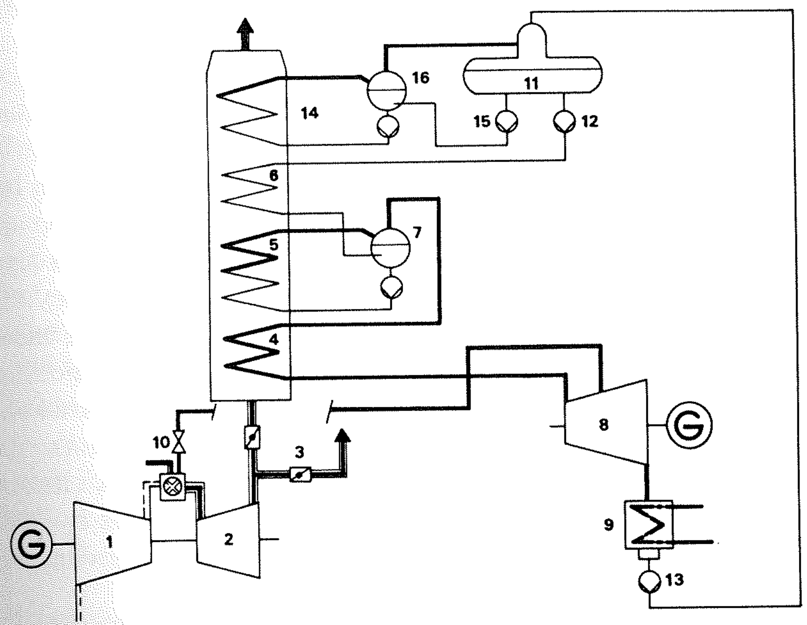

Fig. 3-51: Arrangement with an extraction turbine for steam injection into the gas turbine

1 Compressor
2 Gas turbine
3 Flue gas bypass (optional equipment)
4 High pressure superheater
5 High pressure evaporator
6 High pressure economizer
7 High pressure boiler drum
8 Steam turbine
9 Condenser
10 Steam injection
11 Feedwater tank/deaerator
12 High pressure feedwater pump
13 Condensate pump
14 Low pressure evaporator
15 Low pressure drum
16 Medium-pressure evaporator
17 Steam injection line
18 Medium pressure drum

or with water injection. This comparison, of course, does not in itself possess any absolute validity because the amount of steam injected varies with specifications and type of gas turbine. It does, however, show trends.

The comparison shows that the system with steam injection has a slightly higher overall power output and less waste heat to be dissipated in the condenser than the dry system. This last fact indicates that there is less waste heat from the turbine and the condenser, which also makes the installation less expensive. Dis-advantages, on the other hand, are its lower efficiency and the greater amount of additional water required. In some cases, then, the solution with steam injection can be more economical that the standard solution. The prerequisite for this, however, is having available a low cost source of additional water. The system with water injection has the lowest efficiency but its output is approx. 7% greater. Water consumption is less by 30% because the water has a better cooling effect than steam.

Table 3-5: Comparison of the single-pressure system with preheater loop with and without steam or water injection, Fuel Oil #2

Injection in the gas turbine	Water	Steam	Dry	
NO_x emission	75	75	250	vppm (15%O_2,dry)
Gas turbine output	73 800	76 000	68 400	kW
Steam turbine output	38 600	31 200	36 800	kW
Heat input	255 700	239 500	228 000	kW
Station service power	1 300	1 200	1 200	kW
Net power output	111 100	106 500	104 000	kW
Net efficiency	43.5	44.7	45.6	%
Waste heat in condenser	78 900	61 800	76 100	kW
Additional water required	7.0	10.0	0.4	kg/s

The sharp drop in efficiency, approx. 2% with steam and almost 5% with water injection explains why all gas turbine manufacturers are working on the development of dry low NO_x alternatives as presented in Section 10.3 in order to attain low NO_x emission levels without requiring water or steam injection.

It is also possible to conceive of a solution that no longer has any steam turbine at all: All of the steam is directed into the gas turbine [69]. Fig. 3-52 shows one such system which could be of interest as a peaking unit in countries where water is plentiful. It is simple and attains an efficiency higher than that of the gas turbine alone. However, if the steam flow injected is equal to more than approx. 2 - 4% of the air mass flow, major modifications must be made to the gas turbine, principally modifications to the compressor. This system is therefore only of a very limited interest. Because its efficiency is lower and its water consumption far greater than with the normal combined cycle, the range for its economical application is quite limited.

These types of highly sophisticated systems are being marketed at present under such names as STIG (Steam-Injected Gas Turbine), Turbostig, etc. [45], [46]. They all suffer from the disadvantage that water consumption is high, and that their efficiency is nevertheless not as high as in normal combined-cycle plants. They frequently require specially designed gas turbines, which is seriously limiting their acceptance. STIG systems are suitable solutions for smaller cogeneration plants with aeroderived gas turbines. Fig. 3-53 shows one such system, with one waste heat boiler for two gas turbines.

This system is designed to simplify and reduce the costs of the steam process whenever the combined-cycle plant includes several gas turbines. It is of interest mainly with smaller gas turbines since a real saving is possible with this system whenever only one steam generator is provided to serve two gas turbines.

The cost reduction is less with larger machines. The advantages of this solution (Fig. 3-54) are:

- savings for the evaporator
- simpler steam circuit
- better efficiency when operating on only one gas turbine.

On the other hand, its disadvantage is that every gas turbine absolutely must be equipped with a flue gas bypass for start-up and shut-down. In plants where such a bypass is not otherwise required, a cost increase results which cancels out to a great extent the saving effected in connection with the boiler.

The reduction in availability can be considered modest since the unfired waste heat boiler is a reliable component.

In summary, it can be stated that this arrangement is interesting only for combined-cycle plants employing small gas turbines that one intends to equip with a flue gas bypass anyway. It is primarily of interest with jet gas turbines having a radial exhaust gas channel, since these can be built without highly expensive flue gas ducts. The boiler can be placed between two gas turbines.

3.2 Supplementary Fired Combined-Cycle Plants

In an open-cycle gas turbine, only 25-35% of the oxygen contained in the air is used for combustion. The remainder can be used for an supplementary firing in the steam generator, which renders the combined-cycle process even more versatile with regard to design, operation, and choice of fuel.

Earlier combined-cycle installations generally had supplementary firing. The fact that that is frequently no longer the case today can be attributed to progress in the development of the

Figure 3-52

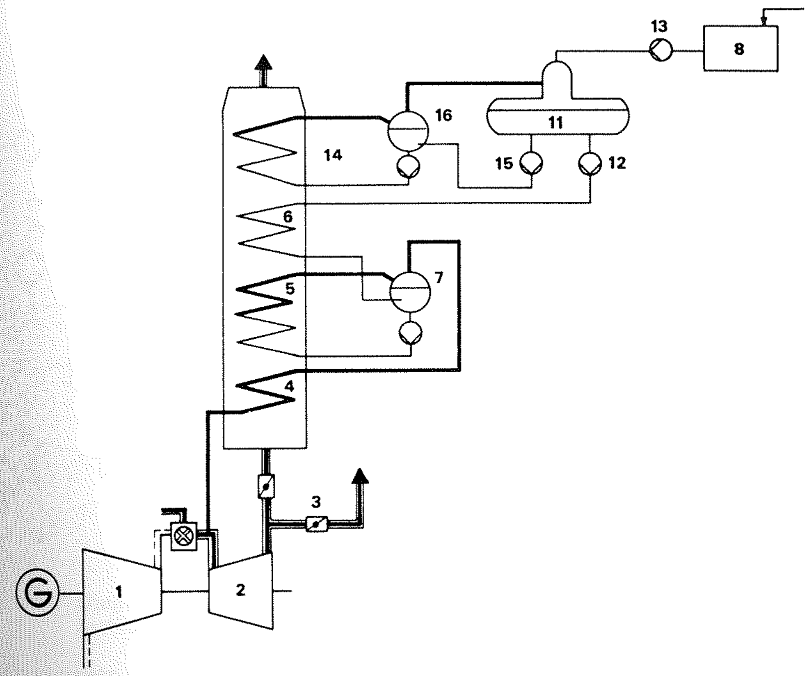

Fig. 3-52: Arrangement without a steam turbine, with 100% steam injection in the gas turbine

1 Compressor
2 Gas turbine
3 Flue gas bypass (optional equipment)
4 High pressure superheater
5 High pressure evaporator
6 High pressure economizer
7 High pressure boiler drum
8 Tank for additional water
9 Condenser
10 Steam injection
11 Feedwater tank/deaerator
12 High pressure feedwater pump
13 Condensate pump
14 Low pressure evaporator
15 Low pressure feedwater pump
16 Low pressure drum

Figure 3-53

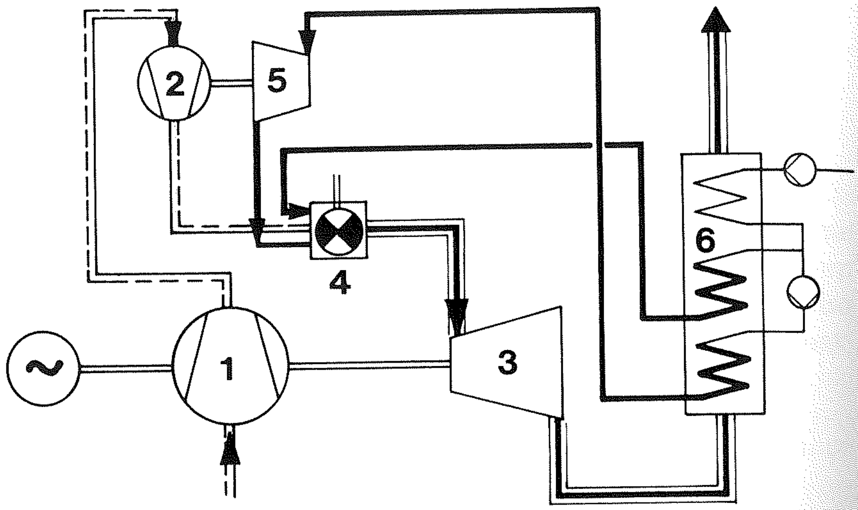

Fig. 3-53: STIG Cycle

1 LP - compressor
2 HP Compressor
3 Gas Turbine
4 Combustor
5 Steam Turbine
6 Waste Heat boiler

Figure 3-54

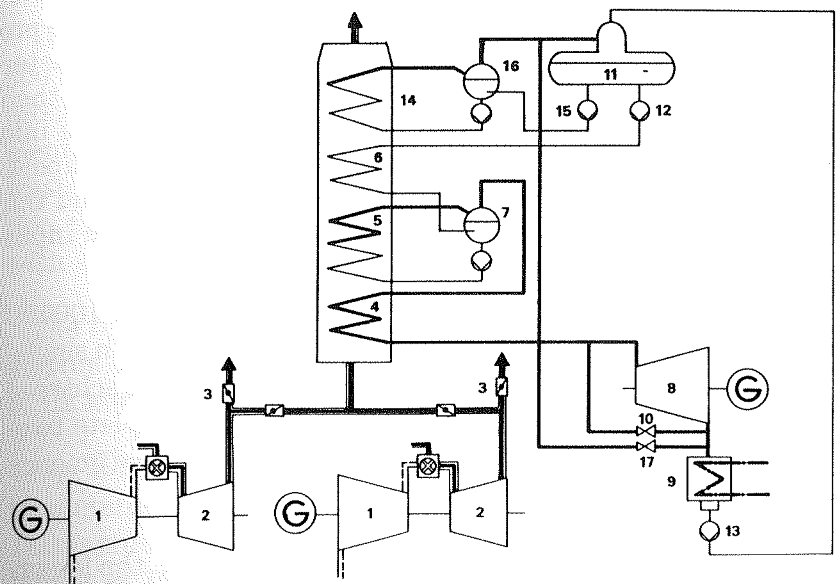

Fig. 3-54: Diagram of the principle of a combined-cycle installation with two gas turbines and one waste heat boiler

1 Compressor
2 Gas turbine
3 Flue gas bypass
4 Superheater
5 Evaporator
6 Economizer
7 High pressure boiler drum
8 Steam turbine
9 Condenser
10 High pressure steam bypass
11 Feedwater tank, deaerator
12 High pressure feed pump
13 Condensate pump
14 Preheater loop (low pressure evaporator)
15 Low pressure feed pump
16 Low pressure boiler drum (optional equipment)
17 Low pressure steam bypass

gas turbine. Thermodynamic interest in supplementary firing decreases as the gas turbine inlet temperature rises (Section 2.2). Fig. 3-55 (p. 109) shows the efficiency of the combined-cycle process using the gas turbine inlet temperature as a parameter. The curves are valid for single-pressure steam circuits without supplementary firing. Older gas turbines had low turbine inlet temperatures. With these machines, an increase in temperature to 750°C (1382°F) improves overall efficiency. Beyond that point, supplementary firing brings increases only in the input.

In gas turbines with inlet temperatures in excess of 1000 °C (1832 °F), the gain is negligible even in the lower range. In single-pressure processes, only a slight gain in efficiency can be achieved with a supplementary firing to 750 °C (1382 °F). Complete two-pressure processes, however, attain their maximum efficiency when utilizing the waste heat alone.

As gas turbine inlet temperatures keep increasing, the importance of supplementary firing will diminish even further. Nevertheless, the increased operating and fuel flexibilities of the combined-cycle with supplementary firing may be an advantage in special cases. Particularly in installations used for cogeneration of heat and power, this arrangement makes it possible to control the electrical and thermal outputs separately (refer to Section 4).

Combined-cycle installations with supplementary firing fall into one of two categories:

- units with limited supplementary firing, which are similar to units without supplementary firing;
- units with maximum supplementary firing, in which most of the oxygen contained in the gas turbine exhausts is utilized. This type of power plant is based on the conventional steam process.

3.2.1 Combined-Cycle Plants with Limited Supplementary Firing

The supplementary firing heats the exhausts gas to at most 800 to 900 °C (1472 to 1672 °F). The arrangement of the steam process used is similar to that of installations without supplementary firing. Up to temperatures of 750 °C (1382 °F), simple waste heat boilers can be used, without cooling of the combustion chambers. Beyond that point, a cooling similar to that used with a conventional steam generator is necessary.

The fuels used are oil or gas. With a simple waste heat boiler and uncooled combustion chambers, gas is the best fuel because of its low radiation and ease of ignition.

Fig. 3-55 shows that the efficiency attains a maximum at a temperature (after the supplementary firing) of 750 °C (1382 °F). This is true because the heat exchange in the economizer is optimum at 750 °C since the curves for flue gas and water temperatures run parallel. The exchange of heat can therefore take place with a minimum loss of exergy. Fig. 3-56 (p. 110) shows the temperature/heat diagrams for temperatures of 500° (932° F), 750° (1382°F), and 1000°C (1832°F) after supplementary firing. At 500°C, the temperature curves in the economizer are convergent, with the minimum difference in temperature on the evaporator end. This pattern is the same as that for a waste heat boiler without supplementary firing (refer to Sect. 1). At 1000°C, on the other hand, the minimum difference in temperature–considered from the water end–is at the inlet to the economizer. This pattern corresponds to hat of a conventional steam generator.

750 °C (1382 °F) after the supplementary firing is the limit case with a constant difference in temperature along the entire economizer. This means that the exhausts can practically be cooled down to the feedwater temperature, thereby eliminating the need for low pressure evaporators (Sections 3.1.2 and 3.1.3).

Unlike conventional power plants, the feedwater temperature here depends solely on the sulphuric acid dewpoint (Section 5.2). Thermodynamic improvement by multi-stage preheating to higher temperatures serves no purpose. This pattern corresponds to that of combined-cycle plants without a supplementary fired waste heat boiler.

Example of a Combined-Cycle Unit with Limited Supplementary Firing

Fig. 3-57 shows the heat balance of a typical combined-cycle plant with supplementary firing to 750 °C (1382 °F). Once again, the gas turbine is the same 70 MW machine as that used for the examples without supplementary firing. The intended fuel is sulphur-free natural gas, which produces results optimum with regard to efficiency. Natural gas has the further advantage that it can be burned easily in a waste heat boiler without a cooled combustion chamber. With oil, even that is more problematic (Section 5.2). The basic arrangement for this installation is the same as that for the purely one-pressure system in 3.1.1. Because the fuel contains no sulphur, the feedwater temperature can be reduced to 60 °C (140 °F). Deaeration therefore takes place under a vacuum.

Fig. 3-58 shows the corresponding Temperature/Heat diagram. One can see the optimum temperature pattern in the economizer, resulting in a low stack temperature. This is the reason why a low pressure evaporator would bring not further improved utilization of the waste heat energy at full load. At part loads, however, or when the supplementary firing is switched off, the stack temperature rises. For installations that are frequently operated at part loads, it can thus make economic sense to select an arrangement with a preheater loop. The same consideration also applies to plants with a temperature after supplementary firing lower than 750 °C (1382 °F) at their design point.

Figure 3-55

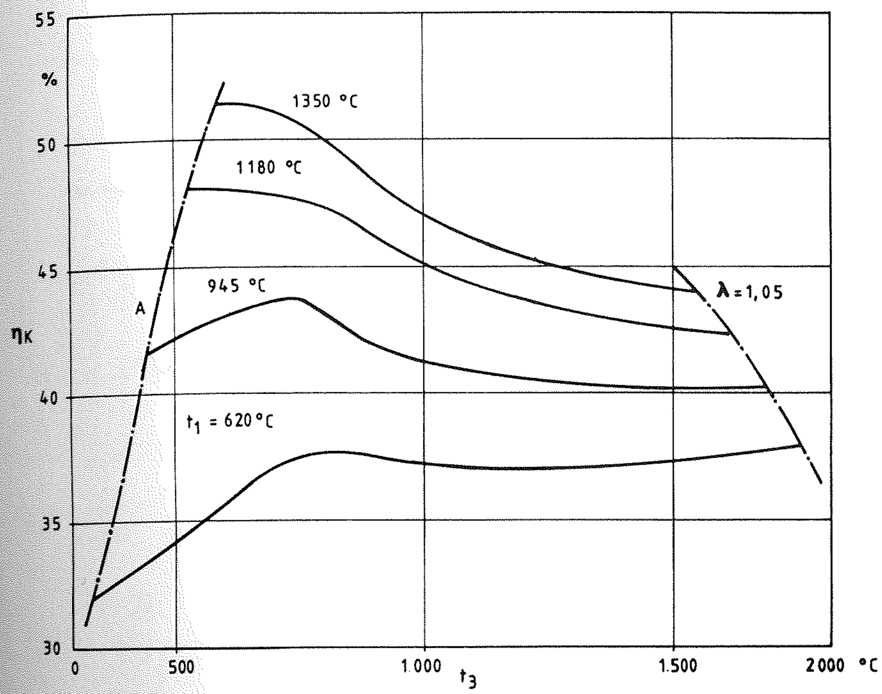

Fig. 3-55: Efficiency of the refired combined-cycle installation as a function of the flue gas temperature after supplementary firing and gas turbine inlet temperature

η_K	Net efficiency of the combined-cycle plant
λ	Excess air ratio of the combustor
t_1	Gas turbine inlet temperature
t_3	Flue gas temperature after supplementary firing

Figure 3-56

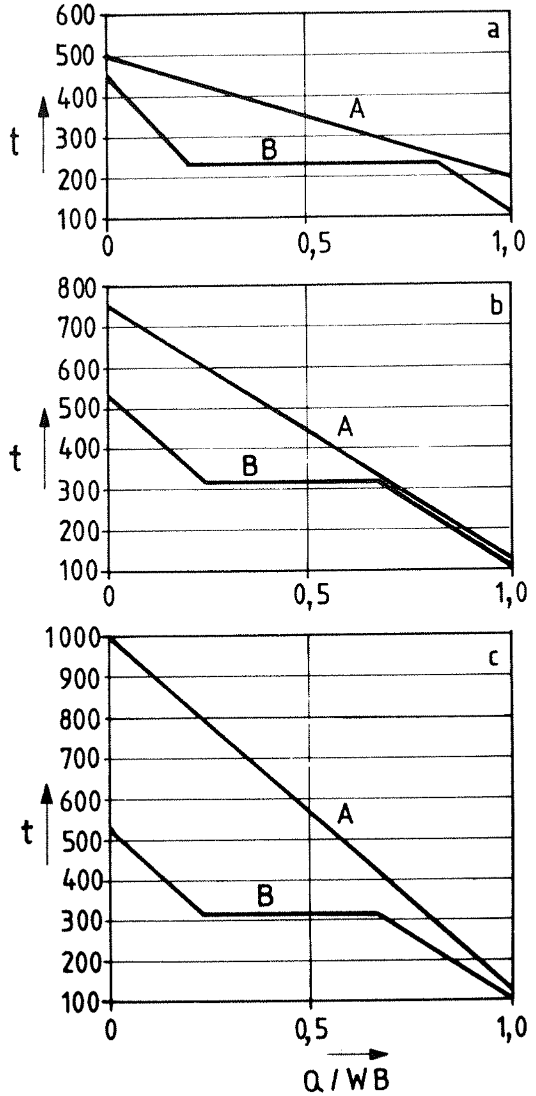

Fig. 3-56: Temperature/Heat diagram for 500° (932°F), 750° (1382°F) (b), and 1000°C (1832°F) (c) after supplementary firing

t	Temperature in °C
Q/Q_k	Heat exchanged
A	Flue gas
B	Water/Steam

Figure 3-57

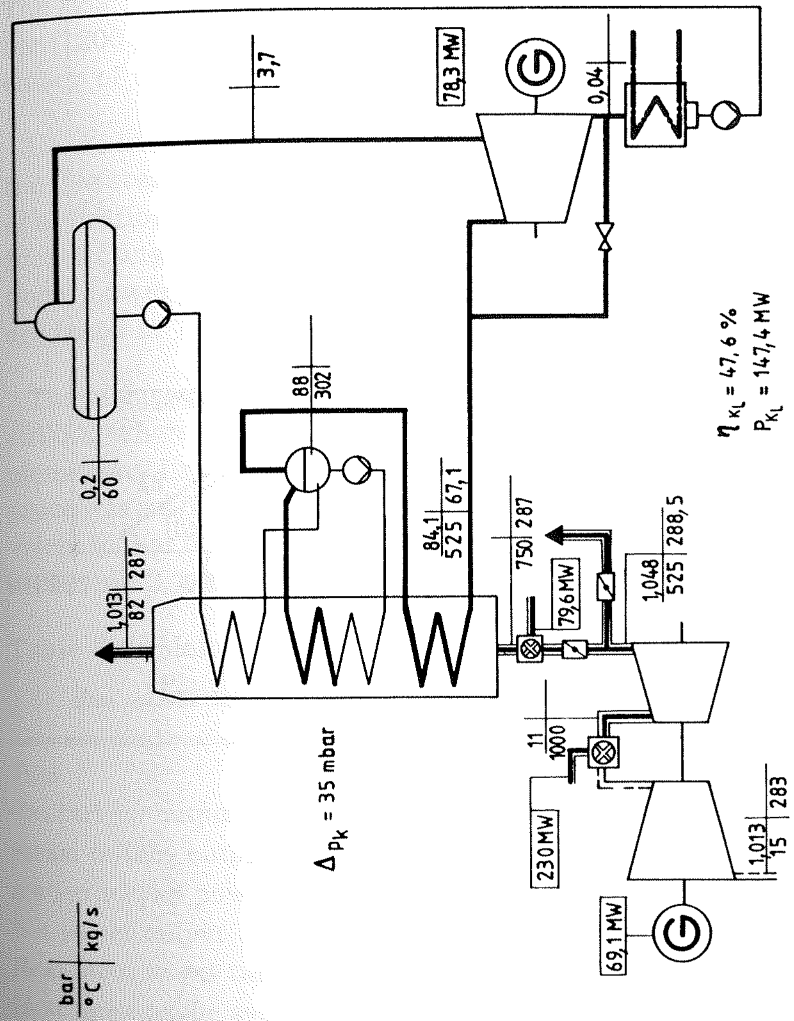

Fig. 3-57: Heat balance of a combined-cycle plant with limited supplementary firing

Figure 3-58

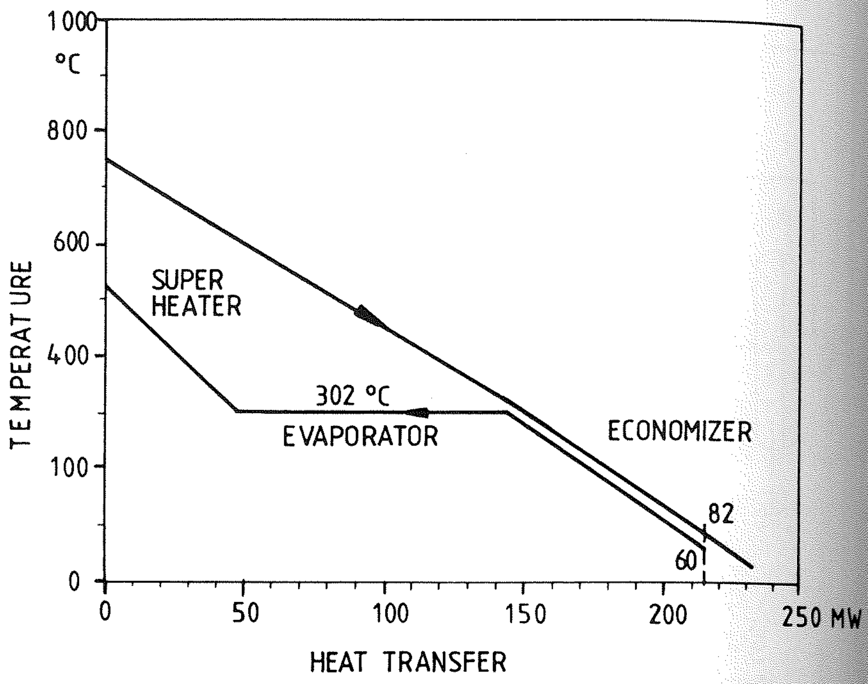

Fig. 3-58: Temperature/Heat Diagram of a Combined-Cycle Installation with Limited Supplementary Firing

The significant difference in comparison to combined-cycle plants without supplementary firing lies in the live steam data. The relatively high flue gas temperature brings these up to 84 bar (1200 psig) and 525 °C (977 °F), thereby improving the efficiency of the steam process.

Unlike in combined-cycle plants without supplementary firing, the live steam data have practically no effect here on the rate of utilization of the waste heat energy. Accordingly, the criteria for selecting these data can be similar to those used for conventional power plants. The main technical data of this example are listed in Table 3-6 below.

The high steam turbine output and condenser waste heat are striking when compared to combined-cycle plants without supplementary firing. The increased efficiency of the steam turbine process is sufficient to attain an overall efficiency almost equivalent to that of the two-pressure combined-cycle plant discussed in Section 3.1.3.

Table 3-6: Main Technical Data of the Combined-Cycle Plant with Limited Supplementary Firing

Fuel	Natural gas	
Gas turbine output	69 100	kW
Steam turbine output	78 300	kW
Station service power	2 100	kW
Net power output	145 300	kW
Heat input to gas turbine (LHV)	230 000	kW
Heat input to the supplementary firing unit (LHV)	79 600	kW
Efficiency of the gas turbine	30.0	%
Heat contained in exhausts	159 300	kW
Efficiency of the steam process	32.9	%
Gross efficiency of the plant	47.6	%
Net efficiency of the plant	46.9	%

The Influence of Ambient Conditions on Power Output and Efficiency

If the temperature after supplementary firing is constant, the effects of air pressure and air temperature are similar to those in installations without supplementary firing. These two design parameters have a very pronounced influence on the power output, but the efficiency remains to a large extent unaffected.

Because the steam turbine is providing a greater portion of the total power output, the temperature of the condenser cooling medium has a stronger effect on the overall power output and the overall efficiency. Its effect on the steam process alone is similar to that in plants without supplementary firing (Fig. 3-22). However, because of the higher live steam data, the loss of enthalpy drop is slightly reduced.

The Influence of the Most Important Design Parameters on Power Output and Efficiency

Flue gas Temperature after Supplementary Firing

The temperature after the supplementary firing is the most important design parameter because it strongly influences the power output and the design of the plant.

Fig. 3-59 shows how relative power output and efficiency depend on the temperature after supplementary firing. The bottom limit, 525 °C (977 °F), represents utilization of the gas turbine waste heat alone. Two different systems have been shown on the diagrams:

- a single-pressure system (Fig. 3-57)
- a two-pressure system (Fig. 3-34)

The basis for comparison is the single-pressure system without supplementary firing. Calculations assume use of a sulphur-free natural gas. When burning oil, the paths of the curves for the single-pressure system would not be significantly changed, but there would be less difference between the single and two-pressure systems (Section 3.6).

One should note the decreasing difference between the efficiencies of single and two-pressure systems as the flue gas temperature after supplementary firing increases. The curve in Fig. 3-49 confirms that the two-pressure process provides no advantages over the single-pressure process at temperatures above 750 °C (1382 °F). Here, too, the improvement at low flue gas temperatures is greater with the two-pressure process. It makes no difference at all for the steam turbine whether this flue gas temperature is attained directly from the gas turbine or by means of supplementary firing. The results indicated in Section 3.1.3 are therefore also valid for installations with supplementary firing.

Fig. 3-59 shows another reason why the machine behaves in this manner. Here, the rate of energy utilization in the single-pressure process continues to rise as the temperature after the supplementary firing increases, up to 750 °C (1382 °F). This is due to the decreasing stack temperature and the increasing temperature differential between the steam generator inlet and outlet. Because the rate of thermal energy utilization increases as the temperature rises, the improvement possible with a two-pressure system becomes continually smaller.

Live Steam Data

The live-steam data for a combined-cycle plant with supplementary firing to approx. 750 °C (1382 °F) are comparable to those of a conventional steam turbine plant with the same power output from the steam turbine. The live steam temperatures have been raised in parallel to the pressure, according to values customary for steam power plants. As is the case in

Figure 3-59

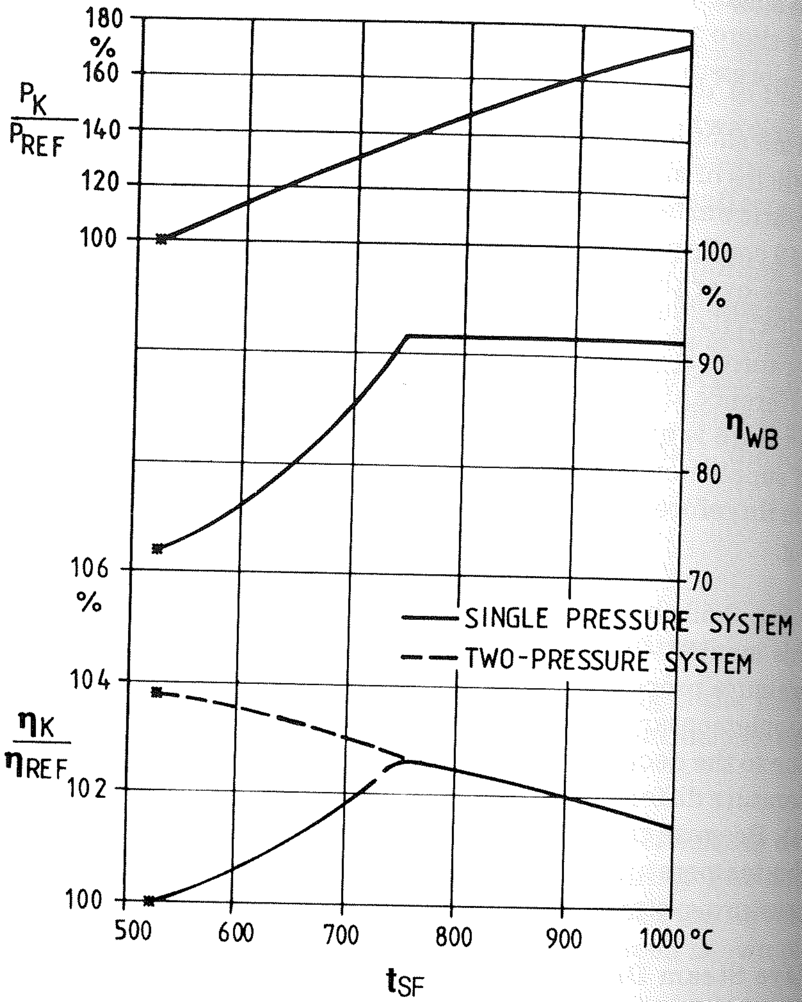

Fig. 3-59: Effect of the Temperature after Supplementary Firing on Power Output and Efficiency of the Combined-Cycle Plant and on the Rate of Waste Energy Utilization

P_K/P_{REF}	Relative power output of the combined-cycle plant
η_K/η_{REF}	Relative efficiency of the combined-cycle plant
η_{WB}	Rate of utilization of waste heat energy
t_{SF}	Flue gas temperature after supplementary firing
*	Reference = Plant without supplementary firing (525°C/977°F)

conventional steam plants, efficiency increases steadily with pressure. The rate of waste heat utilization at 750°C (1382°F) after supplementary firing is practically unaffected by the live steam pressure. The reduction in exergy losses at higher live steam data therefore directly affects overall efficiency of the power plant.

Feedwater Preheating

As in combined-cycle plants without supplementary firing, the temperature of the feedwater greatly influences the efficiency of the installation as a whole. Because it directly affects the rate of waste heat utilization in the steam generator, the temperature selected should be as low as possible. If the risk of low temperature corrosion makes it necessary to preheat to a higher temperature, that preheating should take place in several stages.

For this reason, when burning oil, a two to three-stage preheating system should be employed to reach the required 110 to 140 °C (230 to 284 °F). The efficiency of a power station is, however, affected by the higher feedwater temperature.

When the design temperature for the flue gas (after supplementary firing) is less than 750 °C (1382 °F), a system with a preheating loop can be a reasonable way to make optimum use of the heat (Section 3.1.2). The more often the unit is run at part load, the truer that becomes.

Condenser Pressure

In supplementary fired combined-cycle plants, the increased portion of the total power output being provided by the steam turbine means that condenser pressure is of more importance than in combined-cycle plants which merely utilize the waste heat, particularly in machines where the live steam data are low. With higher data, the enthalpy drop in the turbine is greater and for that reason, the relative effect of a change in condenser pressure is less pronounced. More detailed information on the effect it has on the power output of the steam turbine can be obtained from [1]. It should be noted, however, that combined-cycle plants have a greater exhaust steam flow than do conventional steam power plants with the same steam turbine output, since there is less preheating of the feed-water. For that reason, the condenser pressure exerts a greater influence on the power output from the steam turbine.

3.2.2 Combined-Cycle Plants with Maximum Supplementary Firing

The basic idea for combined-cycle plants without or with limited supplementary firing in the waste heat boiler was to make the best possible use of the gas turbine's waste heat. In plants with maximum supplementary firing, the point of departure was not the gas turbine but the conventional steam process. The idea was to provide a prior gas turbine in order to improve the efficiency of the large conventional power station. In this case, the gas turbine is— stated with some exaggeration— only "an improved air blower with an air heater built in." It also happens to provide electricity.

This approach is reflected in the ratio of the outputs of the steam and the gas turbines. Depending on the amount of excess air, this is between 4 and 10, as compared to 0.5 in combined-cycle plants without supplementary firing.

The steam process is consequently almost the same as that in a conventional steam power plant. In most cases, the plant is a unit with steam reheating and multi-stage regenerative feed-water heating.

The number of possible systems available is large, since practically all known steam processes can be equipped with a gas

turbine. The ratio between the outputs of the steam turbine and the gas turbine can be varied at will, either by increasing the air for combustion in the steam generator using an additional fresh air fan, or by using only part of the oxygen remaining in the gas turbine exhaust gas. The example shown in Fig. 3-60 consists of a gas turbine and a steam turbine with reheat. In a power plant of this type, the regenerative air pre-heater usually found in conventional steam generators is superfluous because hot gas turbine flue gas is being used as the combustion air. The stack temperature would therefore be very high if waste heat from the boiler were not being used for feedwater heating. The ratio of feedwater flow to flue gas flow is much greater than in a waste heat boiler. Due to considerations of exergy only a portion of the feedwater should be directed through the economizer that is being heated with flue gas. The rest flows as normal through the steam-heated high pressure preheaters. The heat transfer in this part-flow economizer is therefore ideal because the curves for water and steam temperatures run parallel to one another (Fig. 3-61).

In plants burning sulphur-free natural gas, the energy is the flue gas can be put to even better use by using a second part-flow economizer in the low pressure portion of the feed water preheater. This arrangement is advantageous, however, only if the dewpoint of the exhausts is very low (Fig. 3-62).

Combined cycle installations with maximum supplementary firing are generally equipped with a fresh air fan which makes it possible to operate the steam process even when the gas turbine is not in operation. This increases the availability of the unit, to be sure— but the efficiency in operation of the fan is only moderate. Since this is only used as an emergency operating mode, that is is no great importance here.

Figure 3-60

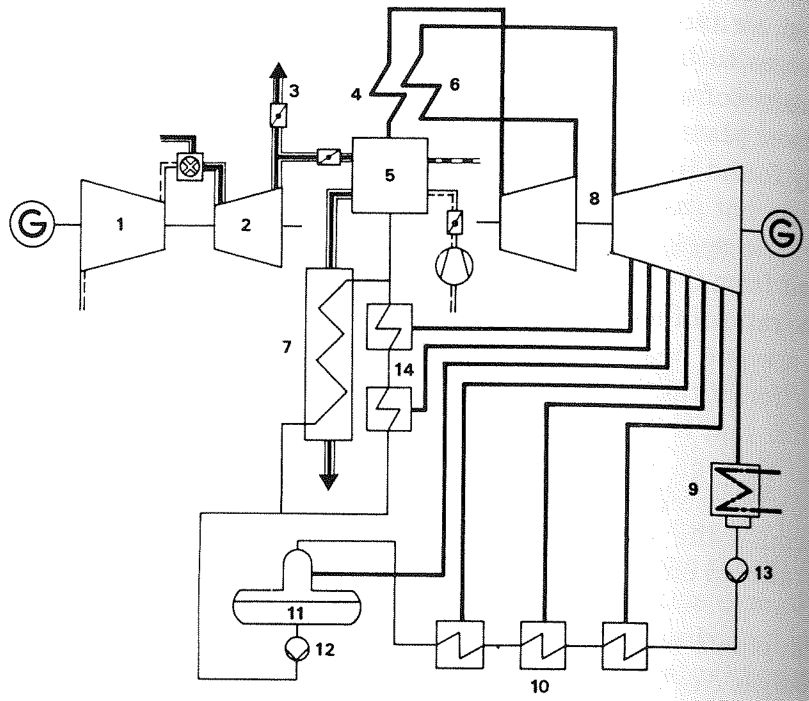

Fig. 3-60 Arrangement of a Combined-Cycle Plant with Maximum Supplementary Firing

1	Compressor	8	Steam turbine
2	Gas Turbine	9	Condenser
3	Flue gas bypass	10	Low pressure economizer
4	Superheater	11	Feedwater tank/deaerator
5	Steam generator	12	Feed pumps
6	Intermediate superheater	13	Condensate pumps
7	Part-flow preheater	14	High Pressure preheater

Figure 3-61

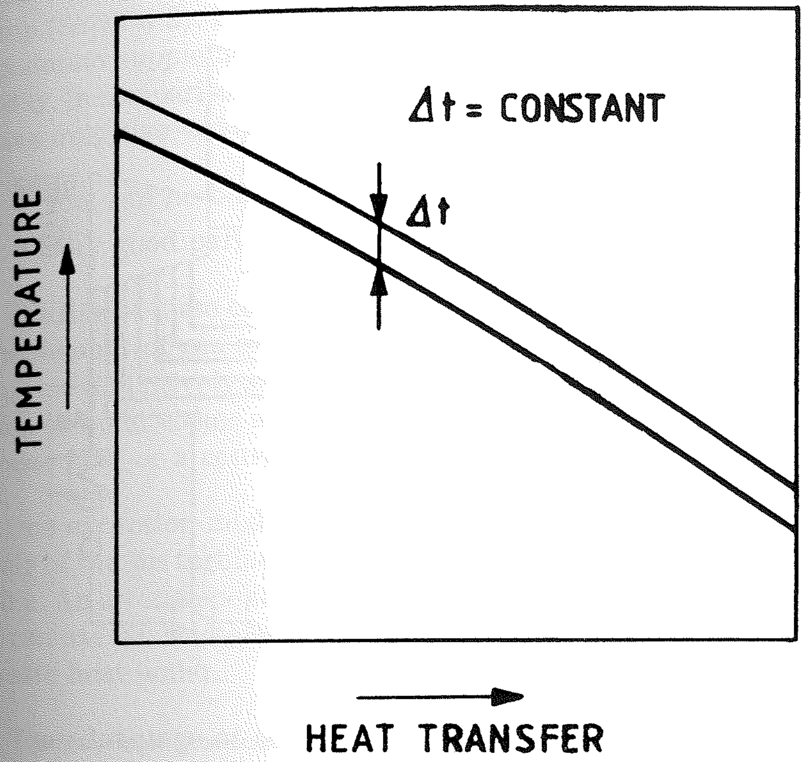

Figure 3-62

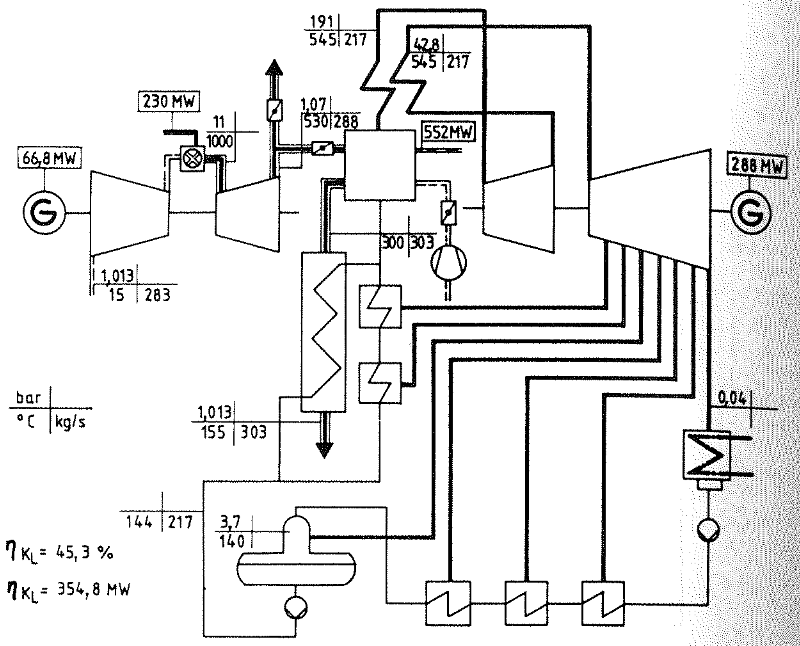

Fig. 3-62; Example of a Combined-Cycle Plant with Maximum Supplementary Firing (coal)

Using a large supplementary firing in combined cycle systems with high gas turbine inlet temperatures causes efficiency to drop (Section 2). For that reason, combined-cycle plants with maximum supplementary firing are only of slight importance today. In comparison to simple combined-cycle installations they have only two advantages:

- coal can be burned in the steam generator
- very good part-load efficiency.

The first of these considerations is a true advantage only where the amount of gas or oil available at a low price from the gas turbine is relatively slight. In other cases, it is generally a better idea to build either a combined-cycle plant utilizing only the waste heat or a conventional coal-burning power station.

One exception might be a case calling for peaking operation only of the gas turbine and base load operation of the steam turbine. An arrangement of this type possesses the advantage that it can use the waste heat from the gas turbine without a special waste heat boiler.

The advantage of good part-load behavior is also valid only to a limited extent because combined-cycle plants without supplementary firing also attain high part-load efficiencies if they include several gas turbines (refer to Section 7).

The disadvantages of units with maximum supplementary firing when compared to simple units utilizing merely the heat contained in the exhaust gas are considerable:

- poorer efficiency
- higher investment costs required
- more complex installations, more difficult to operate and maintain, especially if the steam generator is coal-fired

For these reasons, this type of plant will not be discussed in further detail here. However, the example below can provide a basis for comparison with the other arrangements.

Example of a Combined-Cycle Plant with Maximum Supplementary Firing

This example is also based on the same 70 MW gas turbine. Its power output is lower here because there is a greater pressure drop in the exhaust gases in the steam generator. The gas turbine burns natural gas, the steam generator coal.

Due to maximum possible utilization of the oxygen in the exhausts (excess air co-efficient = 1.3), however, a high overall plant output of approx. 340 MW is attainable. The steam process has a simple reheating, a seven-stage regenerative feedwater preheater, and a high pressure part-flow economizer. The main technical data for it are shown in Table 3-7.

Table 3-7: Main Technical Data of the Combined-Cycle Plant with Maximum Supplementary Firing

Gas turbine output	66 800	kW
Steam turbine output	228 000	kW
Station service power (not including feed pumps)*	6 000	kW
Net power output	348 800	kW
Heat input to gas turbine (natural gas) (LHV)	230 000	kW
Heat input to the supplementary firing unit (coal)(LHV)	551 500	kW
Efficiency of the gas turbine	29.0	%
Heat contained in gas turbine exhaust gases	161 900	kW
Efficiency of the steam process	40.4	%
Gross efficiency of the plant	45.3	%
Net efficiency of the plant	44.6	%

* Steam turbine used to drive the feed pump

With a gas-fired steam generator, a net efficiency of approx. 45.5% can be attained by adding a low pressure part-flow economizer to the steam cycle. Because the power output from the steam turbine is so high, the heat flow to be dissipated in the condenser is practically as great as that in a conventional steam power plant. This therefore makes the selection of a site more difficult than with a simple combined-cycle plant, far more water is required.

The Effect of Ambient Conditions and of Design Parameters

Just as in combined-cycle plants without supplementary firing, the air temperature strongly affects the overall power output, at least in those cases where the oxygen in the exhausts is being fully utilized without employing an additional fresh air fan.

The temperature of the cooling medium in the condenser is more important than in units without supplementary firing because the steam turbine is larger. However, the greater enthalpy drop available in the steam turbine partially compensates for this.

The criteria that apply to selection of the steam plant design are the same as those for conventional steam power plants. Due to the size of the units, turbines with reheat systems are generally selected.

3.3 Conversion of Conventional Steam Power Plants

The conversion of older steam power plants into combined-cycle units— also known as "repowering"— is one interesting way to make it possible to continue using at least parts of older steam power plants that have become uneconomical. In this procedure, the boilers are normally replaced with gas turbines and waste heat boilers. Suitable for such an action are steam turbine groups in older power stations which have relatively low live steam data quite well adaptable for combined-cycle installations. These 20-to-25-year old steam turbines generally still have a considerable service life expectancy left, but their boilers are often ready for scrapping.

Fig. 3-63 and 3-64 show the example of a conventional steam turbine plant, before and after such a conversion. From the original installation, it was possible to reuse the following components:

- building
- steam turbine and generator
- condenser
- main cooling system
- main transformer for the steam turbine
- the high voltage equipment

The following parts were dismantled:

- boiler
- piping and fittings
- feedwater preheaters
- condensate pumps
- feedwater pumps
- control equipment

This list can vary from one case to another. Generally, however— as in the example— it makes sense only to retain those components which can be continued in use with no great expense. All parts with only a slight value remaining should be scrapped, for such elements frequently cause unforeseeable extra costs and negatively affect the availability of the repowered installation. The feedwater pre-heating system is a special case because it must, for a combined-cycle plant, be different in de-

Figure 3-63

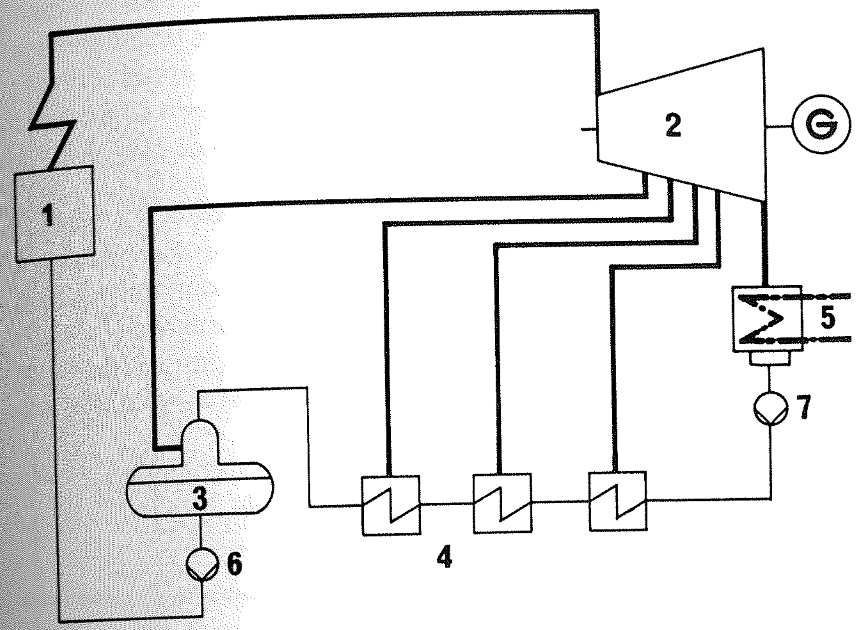

Fig. 3-63: Steam Power Plant before Conversion to a Combined-Cycle Plant

1 Steam generator
2 Steam turbine
3 Feedwater tank/deaerator
4 Low pressure preheater
5 Condenser
6 Feed pump
7 Condensate pumps

Figure 3-64

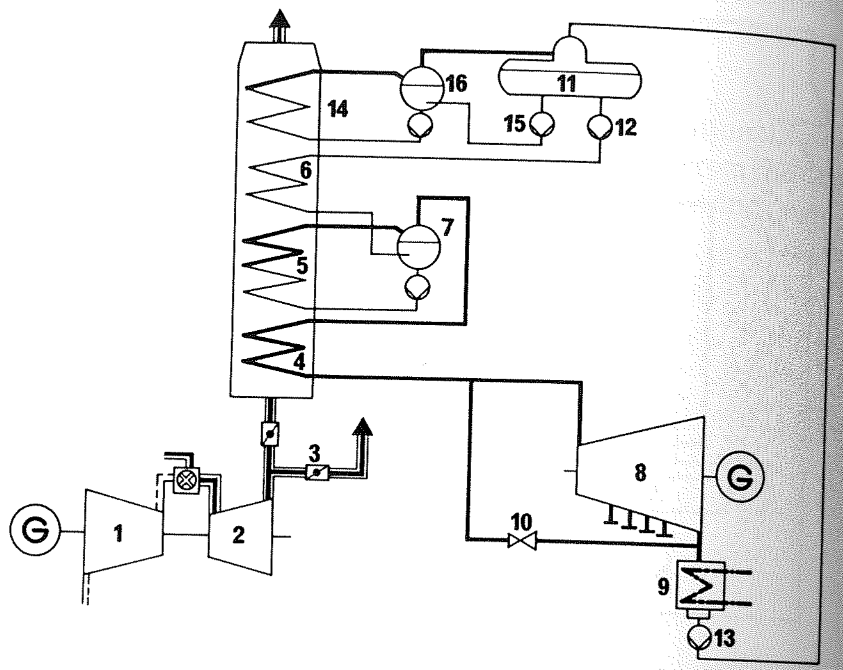

Fig. 3-64: Combined-Cycle Plant with Existing Steam Turbine

1 Gas Turbine
2 Compressor
3 Flue gas bypass
4 Superheater
5 Evaporator
6 Economizer
7 Drum
8 Steam Turbine
9 Condenser
10 Steam bypass
11 Feedwater tank/deaerator
12 HP Feedwater pump
13
14 LP evaporator
15 LP Feedwater pump
16 LP drum

sign from that for a steam power plant because of thermodynamic considerations. Thus, in most cases, no extraction points are needed any more on the steam turbine, which increases the power output from that turbine. However, the generator and the transformer must be able to handle this additional power output, and in certain cases that can lead to restrictions.

Table 3-8 shows the significant gain in efficiency that can be attained with re-powering. The power output of the plant as a whole has actually been tripled. A supplementary firing could reduce the overall power output because the gas turbine exhausts would then only be covering a portion of the heat demand of the steam generator.

Table 3-8: Comparison of a steam power plant before and after repowering to a combined-cycle power plant

	before repowering	after repowering	
Net output from power plant	30 300	102 200	kW
Power output from steam turbine	32 000	35 000	kW
Power output from gas turbine		68 400	kW
Station service power	1 700	1 200	kW
Heat input	107 000	228 000	kW
Net efficiency	28.3	44.8	%

One advantage of supplementary firing in repowering is the possibility of closer adaptation to the live steam data of the original design. Because the design of the gas turbine has been standardized, there are fixed limits on the amount of steam that can be produced, and these do not necessarily lie close to the design values for the steam turbine.

Another approach to repowering can be used with more modern steam power plants equipped with reheat steam turbines. To improve the efficiency of such a unit, the fresh air fan can be replaced in supplying the oxygen needed for combustion by installing a new gas turbine before the existing steam generator. Here, the existing steam boiler continues in use, but must be adapted to its new operating mode. The modifications required are due mainly to the much higher temperature of the gas turbine exhausts (approx. 500 °C/ 932 °F as compared to 300 - 350 °C / 572 - 662 °F for the fresh air after an air preheater). The major parts that must be modified are the:

- burners
- fresh air ducts
- perhaps the reheater

A waste heat recovery system must be installed after the steam generator to handle most of the condensate and feedwater heating. The complete plant appears similar to those shown in Fig. 3-60 and 3-62.

The major problems that arise with this type of repowering are in connection with:

a) space availability for installation of the gas turbine and the waste heat recovery system

b) adaptation of the boiler and the overall operating concept to the new mode of operation

For steam power plants that burn gas or oil, however, this is one very interesting possibility for raising efficiency by more than 10% and power output by 20 to 30% at relatively low investment costs. With coal-burning units, there is less potential economic gain because the conversion itself is more complex and there is less improvement in efficiency.

3.4 Combined-Cycle Installations with Closed-Cycle Gas Turbines [63 to 68]

Gas turbines may operate either in an open-cycle or a closed-cycle process. The closed-cycle process is, however, much less frequently built than the open-cycle. Fig. 3-65 shows the simplest arrangement, consisting of compressor, heater, turbine, and cooler. Theoretically, the medium can be any gas, but almost all installations that have been built employ air. Helium or nitrogen are other possible media, with helium in particular of possible interest for nuclear power stations.

The efficiency of the gas turbine can be raised by employing more complicated systems, e.g., a recuperator, compressor with intermediate colling, or the like. However, as was the case with open-cycle gas turbines, the simpler arrangements are generally the most suitable ones.

The great advantage of the closed-cycle gas turbine is the great freedom it offers in selection of the fuel. In addition to oil or gas, coal or nuclear fuel can also be used. Its main disadvantage is that the heat is supplied to the process via a heat exchanger, which limits the turbine inlet temperature to levels lower than those in open-cycle gas turbines. According to [64], efficiencies of more than 45% should be attainable. Fig. 3-66 shows the way in which the efficiency of the combined-cycle process depends on the compressor pressure ratio and the gas turbine inlet temperature.

One interesting potential application involves nuclear power plants with helium-cooled reactors (high temperature reactors or fast breeding reactors). In these plants, the gas would be in direct contact with the reactor. The maximum process temperature is limited because the highest process temperatures attainable in the reactor are today at 950 °C (1742 °F).

Figure 3-65

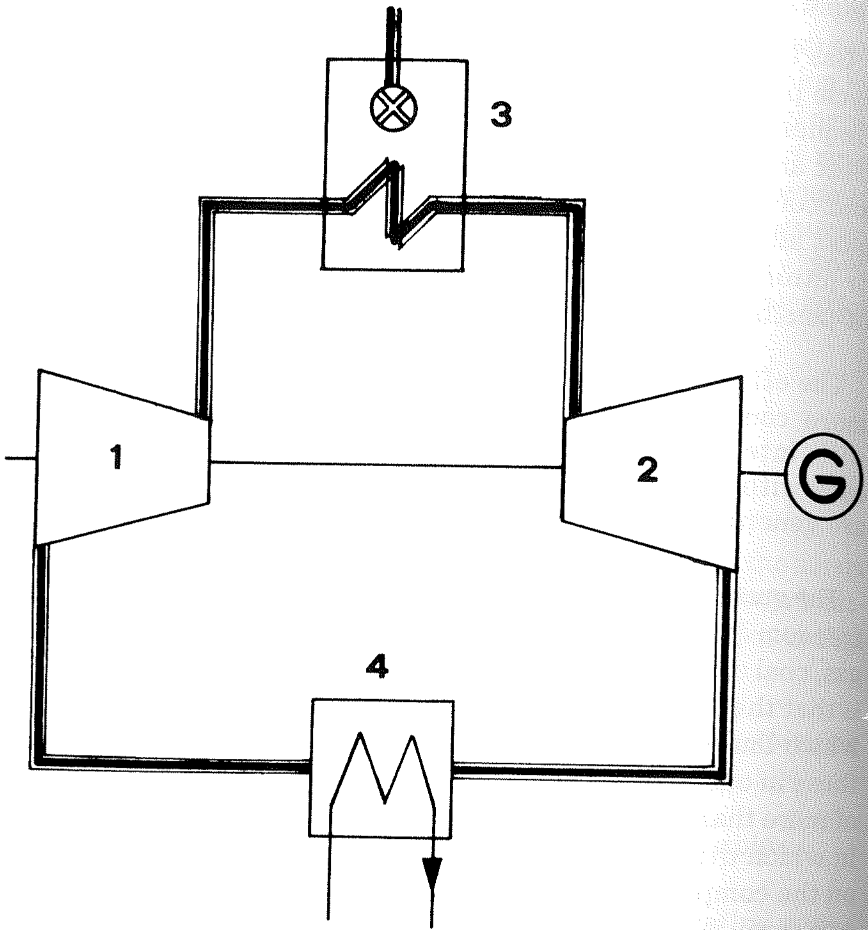

Fig. 3-65: The Closed-Cycle Gas Turbine

1 Compressor
2 Gas turbine
3 Heater
4 Cooler

Figure 3-66

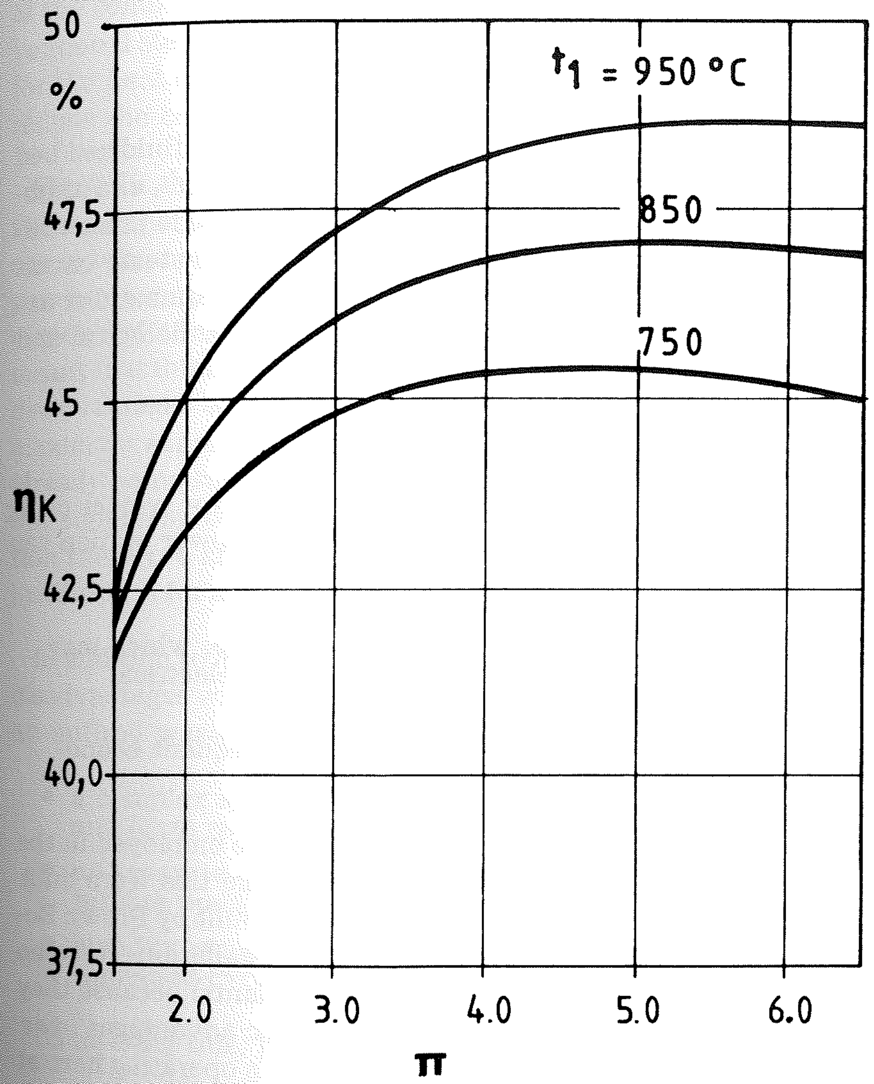

Fig. 3-66: Dependence of the Efficiency of a Combined-Cycle Plant with a Closed-Cycle Gas Turbine on the Compressor Pressure Ratio and the Inlet Temperature to the Gas Turbine (as in [64])

t_1 Gas turbine inlet temperature
π Compressor pressure ratio
η_K Combined-cycle plant efficiency

The use of combined-cycle plants with closed-cycle gas turbines can be considered for the following applications:

- burning coal in a fluidized bed
- high temperature reactors.

Fig. 3-67 shows one possible arrangement with fluidized bed combustion, a closed-cycle air or helium gas turbine, and a subsequent reheating steam process. A combined-cycle plant with a high temperature reactor, helium turbine, and steam circuit can be seen in Fig. 3-68. In neither case can a commercial use be expected in the near future because the technological and economic hurdles are too great.

3.5 Pressurized Steam Generators

As one final combination we should mention a system using a pressurized steam generator. This type of power station can fall into one of two categories:

- installations with a simple charging group, which may or may not provide (a small amount of) electrical power
- installations with gas turbines and subsequent economizers

Fig. 3-69 shows the diagram of the principle employed in the first of these arrangements. Power plants like this were built quite early: the more than 100 Velox boilers built by Brown Boveri operate in this way. However, such installations have no real future (at least for oil or gas-burning plants) because they cannot provide any genuine thermodynamic advantages over conventional steam plants. The gas turbine is operating here at a very low temperature level because it is using the exhaust gas from the steam generator as a working medium. Thus, no use is being made of the high temperature potential offered by the machine, and its power output is for that reason practically null. Typical efficiencies of such units are in the range of 40 - 42%.

Its most important advantage is the small, compact steam generator. Because of the higher pressure and the greater speed of the combustion gas, the heat transfer is better than in a conventional steam generator. Consequently, the surface required for the heat exchange is smaller, and that theoretically should reduce the costs of the installation.

Fig. 3-70 shows an installation of the second type. Here, unlike with the group employing a simple charging group, advantage is taken of the high gas inlet temperature. The steam generator simply replaces the gas turbine combustor. Net efficiencies of 43 to 46% are within the range of possibility. That means that this arrangement should also have no thermodynamic disadvantages when compared to combined-cycle plants with maximum supplementary firing. But even these values are already being surpassed by simple combined-cycle plants without supplementary firing which today exceed 50%. The question therefore arises as to whether there will be any economic justification for an oil or gas-fired plant of this type in the future.

In addition to its lower efficiency, it has the following further disadvantages when compared to the combined-cycle plant with an unfired waste heat boiler:

- a complex system requiring a special steam generator and large recuperator
- no separate operation of the gas turbine and steam turbine possible
- high investment cost compared to the simple combined-cycle plant

Countering this, it has the following advantages:

- small steam generator— though this advantage is partially cancelled out by the subsequent economizer required
- quick control over the steam process possible because of the small storage capacity of the steam generator.

Figure 3-67

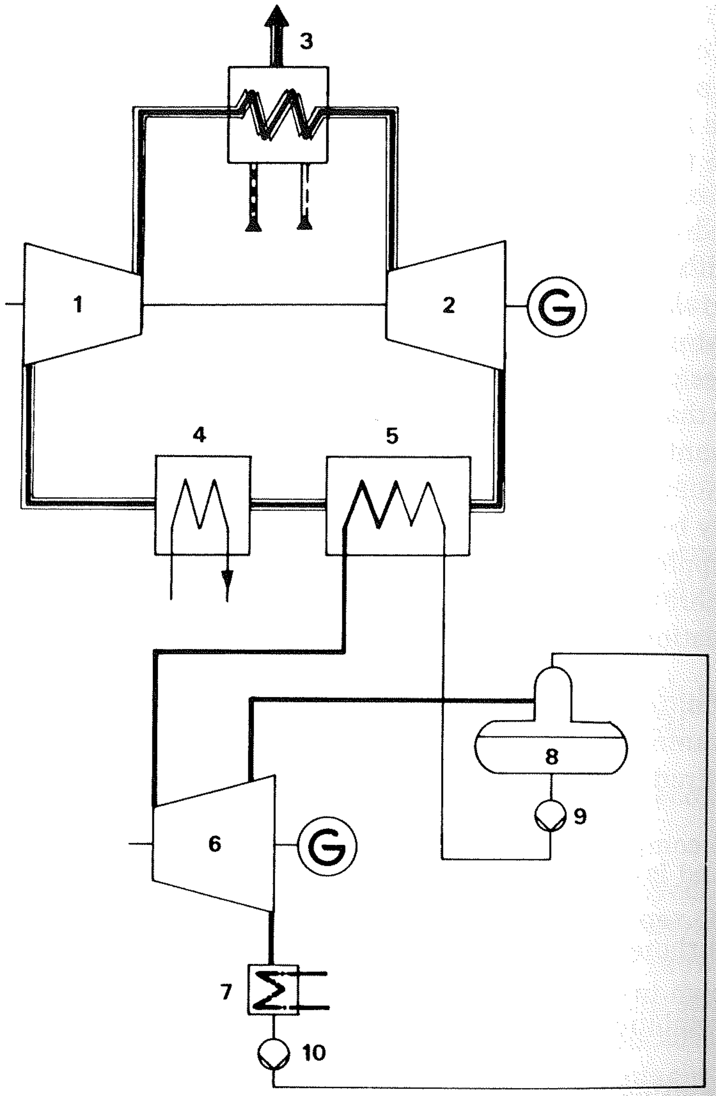

Fig. 3-67: Combined-Cycle Plant with Closed-Cycle Gas Turbine and Fluidized Bed Combustion

1 Compressor
2 Gas turbine
3 Heater
4 Cooler
5 Waste heat
6 Steam turbine
7 Condenser
8 Feedwater tank/deaerator
9 Feed pumps
10 Condensate pumps

Figure 3-68

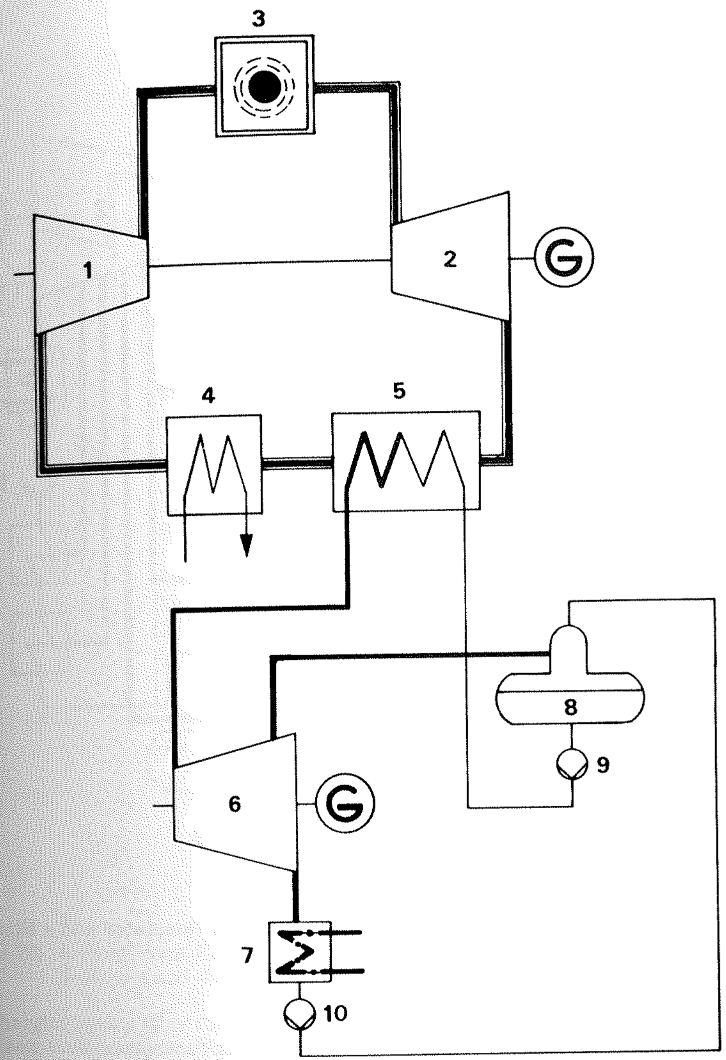

Fig. 3-68: High Temperature Reactor with Helium Gas Turbine Followed by a Steam Process

1 Compressor
2 Gas turbine
3 Reactor
4 Cooler
5 Waste heat boiler
6 Steam turbine
7 Condenser
8 Feedwater tank/deaerator
9 Feed pumps
10 Condensate pumps

Figure 3-69

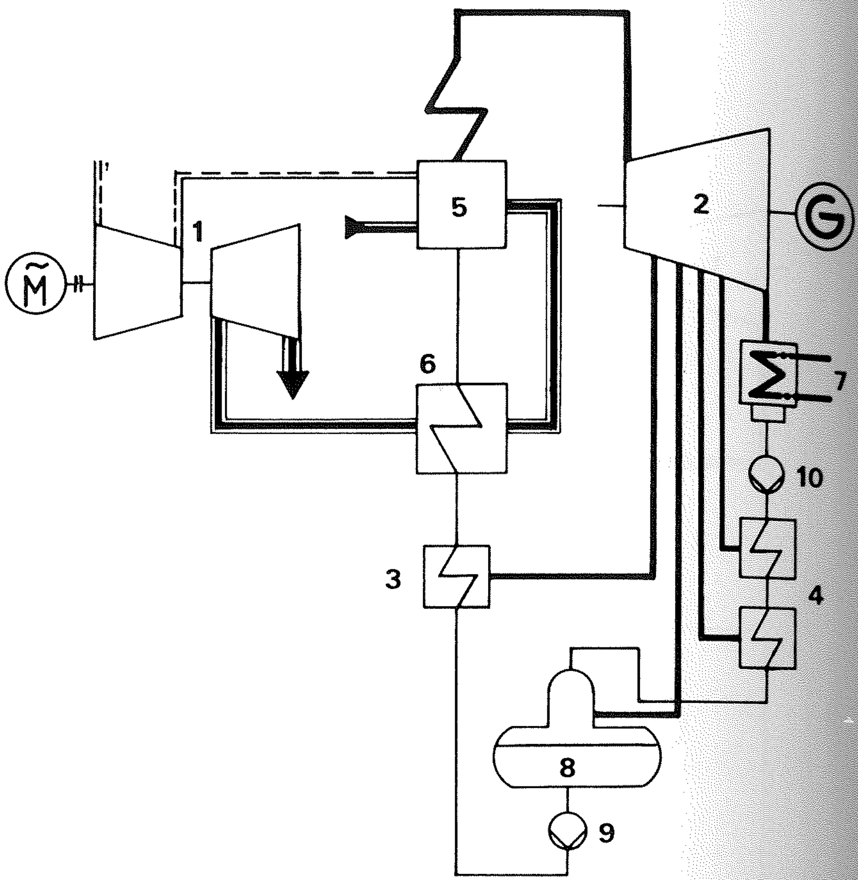

Fig. 3-69: Arrangement Employing a Pressurized Steam Generator and a Simple Charging Group

1 Charging group (Gas turbine)
2 Steam turbine
3 High pressure preheater
4 Low pressure preheater
5 Steam generator
6 Economizer
7 Condenser
8 Feedwater tank/deaerator
9 Feed pumps
10 Condenser

Figure 3-70

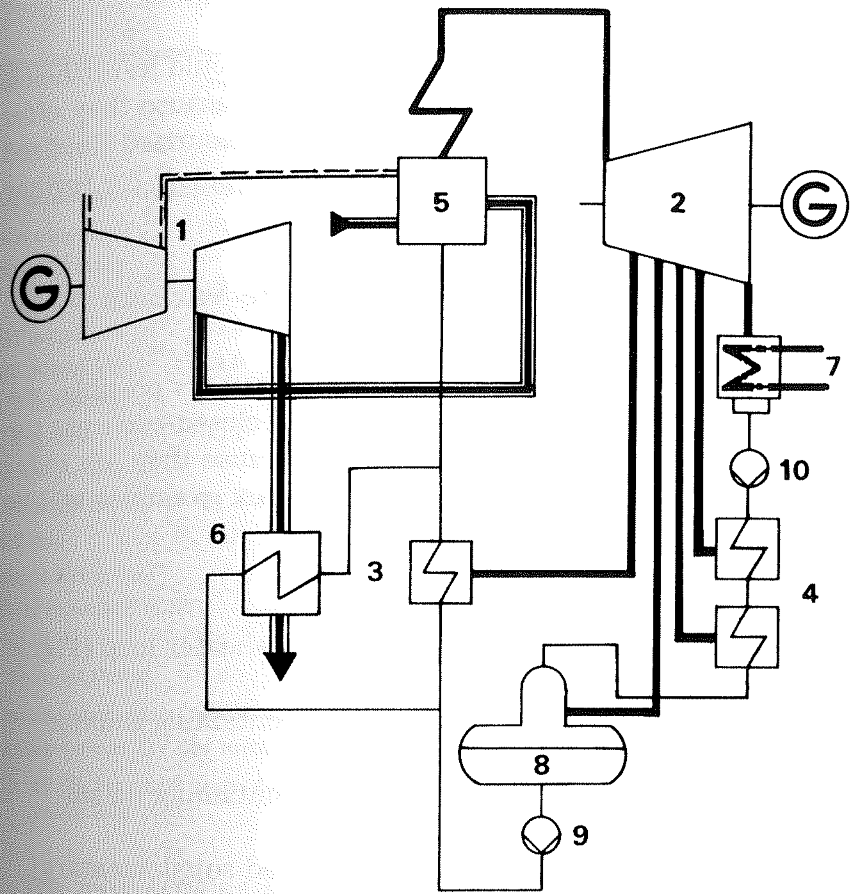

Fig. 3-70: Arrangement Employing a Pressurized Steam Generator with Gas Turbines and Subsequent Economizer

1 Charging group (Gas turbine)
2 Steam turbine
3 High pressure preheater
4 Low pressure preheater
5 Steam generator
6 Economizer
7 Condenser
8 Feedwater tank/deaerator
9 Feed pumps
10 Condenser

The balance is more likely to be on the negative side for an oil or gas-fired plant.

Plants with pressurized steam generators could nevertheless still become very interesting in the future because they offer a possibility of burning coal cleanly in a pressurized fluidized bed combustor (PFBC). Refer to Section 10 below for further information about such plants.

3.6 Summary and Evaluation of the Various Arrangements Possible

In our evaluation of the various arrangements possible, we will omit from consideration systems with closed-cycle gas turbines or pressurized steam generators because they are today of academic interest only. The following six examples will be compared below:

- the single-pressure sytem (Fig. 3-4)
- the single-pressure system with a preheater loop (Fig. 3-27)
- the two-pressure system for fuels containing sulphur (Fig. 3-37)
- the two-pressure system for fuels containing no sulphur (Fig. 3-38)
- the combined-cycle plant with limited supplementary firing (Fig. 3-57)
- the combined-cycle plant with maximum supplementary firing (Fig. 3-60)

All arrangements are based on the same gas turbine, rated at approx. 70 MW and are as a result directly comparable (Table 3-9).

Table 3-9: Comparison of Power Output and Efficiency of the Various Systems

	Single-Pressure	Single-Pressure, Preheater Loop	2-Pressure Fuel with Sulphur	2-Pressure Fuel with no Sulphur	Limited Aux. Firing	Maximum Aux. Firing
Net output,kW	101.5	104.5	104.9	109.0	145.3	348.8
Net efficiency, % (LHV)	44.5	45.6	46.0	47.4	46.9	44.6[1])
GT output,kW	68.6	68.4	68.2	69.4	69.1	66.8
ST output,kW	34.0	36.8	38.0	40.8	78.3	288.0
GT efficiency, % (LHV)	30.1	30.0	29.9	30.2	30.0	29.0
Steam process ef ficiency,%	21.7	23.4	24.1	25.6	32.9	40.4
GT fuel	Oil	Oil	Oil	Gas	Gas	Gas
Suppl.Firing fuel	---	---	---	---	Gas	Coal
Heat input,MW (LHV)						
a) GT	228.0	228.0	228.0	230.0	230.0	230.0
b) Suppl.Firing	0	0	0	0	79.6	551.5

Abbreviations: GT Gas turbine ST Steam turbine Suppl. Supplementary

1) 45.5% for an oil or gas-fired steam generator with a low pressure part-flow econo mizer (Fig. 3-62)

The most striking point is the high net efficiency of the gas-fired two-pressure system. The unit with limited supplementary firing is not, however, far behind: its efficiency is less only by a factor of approx. 1%. Its power output, however, is about 40% greater because the steam turbine produces about twice as much power. This arrangement can be of interest whenever a higher power rating— unattainable with utilization of the waste heat alone— is required. Moreover, the specific investment costs required are more likely to be lower than those for the two-pressure system. But this type of plant is more complex and it will become less and less attractive in the future as gas turbine inlet temperatures continue to increase.

The net efficiency of the combined-cycle with maximum supplementary firing is poor, but it can cover 70% of its fuel requirements with coal, which is frequently an advantage.

Efficiency is only one of the important criteria when deciding on the selection of a power plant. A second is the price, but it is difficult in a Handbook of this type to provide exact data here. Only comparative prices are possible. The basis for comparison is in all cases the simple single-pressure system and these relative prices are valid as specific prices for installations with a comparable power rating (Table 3-10).

Table 3-10: Comparison of Specific Prices for the Various Systems, in %

	Single-Pressure	Single-Pressure, Preheater Loop	2-Pressure Fuel with Sulphur	2-Pressure Fuel with no Sulphur	Limited Suppl. Firing	Maximum Suppl. Firing (Gas)	Maximum Suppl. Firing (Coal)
Relative price	100*	101-103	105-108	106-110	103-110	130-150	200-300

* Basis for comparison

The higher specific price for the unit with a comparable power rating when employing maximum supplementary firing is due to the fact that a steam plant is more expensive than a gas turbine, which means that the relative price for installations with a proportionally large steam component will be less favorable. This is especially true for a plant with a coal-burning boiler, which today usually requires installation of a scrubber to remove the sulphur from the exhaust gases.

A third important consideration is the amount of cooling water required.

Table 3-11: Comparison of the Amount of Cooling Water Required for the Various Arrangements

	Single-Pressure	Single-Pressure, Preheater Loop	2-Pressure Fuel with Sulphur	2-Pressure Fuel with no Sulphur	Limited Suppl. Firing	Maximum Suppl. Firing (Coal)
Condenser waste heat,MW	64.3	76.1	74.6	88.4	133.4	670
Cooling water required,kg/s*	1530	1810	1780	2105	3180	15940
Net power output, MW	101.5	104.0	104.9	109.0	145.3	348.8
Specific cooling water req'd* kg/s/MW	15.1	17.4	17.0	19.3	21.9	46.5

* Increase in temperature of the cooling water: 10°C (18°F)

One major advantage of the single-pressure system by itself is its lower specific cooling water requirement. The other combined-cycle plants without supplementary firing need between 12 and 18% more water per installed MW because in these improved circuits, additional low exergy heat is being supplied to the steam process from the exhaust gas. Because this energy is not easily convertable into mechanical energy, it must in large part be dissipated once again in the condenser.

With regard to the amount of cooling water required, the combined-cycle plant with maximum supplementary firing turns out especially poor. It requires proportionally three times as much cooling water as do plants without supplementary firing. But we are dealing here more with a reheating steam plant than with a true combined-cycle installation, and consequently the amount of cooling water required is large.

Flexibility in fuel selection also plays an important role. Present-day gas turbines can only handle liquid or gas fuels (Table 14).

Table 3-12: Possible Gas Turbine Fuels

a) Standard Fuels	Natural gas Diesel oil
b) Special liquid fuels	Methanol Crude Heavy oil, residuals Oil shale oil
c) Special gas fuels	Synthetic gas Blast furnace gas Coal gas, with medium or low calorific value

Note: The use of categories b) and c) is limited, because whether they may be burned depends on their exact chemical analysis and the type of gas turbine involved. Generally, industrial gas turbines with large combustors are better able to handle these fuels.

Selection of the steam process depends on the gas turbine fuel and possibly on the fuel used for supplementary firing. Table 3-13 provides guidelines in this regard.

Table 3-13: Fuels for Gas Turbines and Supplementary Firing

	Single-Pressure	Single-Pressure, Preheater Loop	2-Pressure Fuel with Sulphur	2-Pressure Fuel with no Sulphur	Limited Suppl. Firing	Maximum Suppl. Firing
Gas turbine fuel	Diesel oil, gas, etc.	Diesel oil, gas, etc.	Diesel oil, gas, etc.	No-Sulphur fuel	Diesel oil, gas,	Diesel oil, gas, etc.
Suppl.firing fuel					Gas, Diesel oil	Coal, Gas Heavy oil

The two-pressure system with a low pressure economizer is suitable for use only with fuel with a very low or no sulphur level. For other fuels, it loses its advantage due to the higher feedwater temperature required to avoid low temperature corrosion.

The power ouput and efficiency of the system with a preheater loop remain practically unaffected by changes in the feedwater temperature. For that reason, it is equally suitable for all gas turbine fuels. For no-sulphur fuels, however, its efficiency is lower than that of the two-pressure system. In such cases, it should be selected only where efficiency is not the critical factor.

The advantage of the unit with maximum supplementary firing is that it is capable of burning solid fuels as well. This additional freedom should not be overlooked when considering its disadvantages in other respects, especially its high investment costs and technical complexity.

The installations without supplementary firing are of greater importance today. Their advantages include:

- high efficiency
- a simple steam process
- low investment costs
- quick installation
- simple operation and maintenance.

Particular trump cards of such plants are their simplicity and their low live steam data (Table 3-15). These plants attain very high availability ratings and provide easy operation and maintenance.

Table 3-15: Comparison of Steam Process Data for the Various Systems Discussed in Chapter 3

	Single-Pressure	Single-Pressure, Preheater Loop	2-Pressure Fuel with Sulphur	2-Pressure Fuel with no Sulphur	Limited Aux. Firing	Maximum Aux. Firing
Reheat	no	no	no	no	no/yes	yes
Live steam pressure,bar	37	37	60	60	84	190
Live steam temperature, °C	480	480	480	480	525	545
Feedwater temperature, °C	130	130	130	60	60	255
No. of feed-water pre-heaters	1	1	3	1	1	7

Chapter 4

COMBINED-CYCLE PLANTS FOR COGENERATION

The thermodynamic advantage of a combined-cycle applies not only for use in a power plant that produces power alone, but also for those which provide heat or process steam as well. Fig. 4-1 shows the flow diagram of such an installation with a backpressure turbine.

The thermodynamic superiority of the combined-cycle plant over a conventional power plant is even more pronounced in cogeneration plants than it is in plants used only to generate electricity. The drop between the average heat input to the process and that of the exhausts is greater in a combined-cycle plant than in a steam power plant.

If both types of power plants have to supply heat at the same temperature level, the loss in temperature drop is the same in both cases but the relative loss in combined-cycles is smaller because the total drop available is larger (Fig. 4-2).

The following combined-cycle installations can be considered for cogeneration plants:

- a combined-cycle plant with a backpressure turbine (Fig. 4-1)
- a combined-cycle plant with an extraction/condensing turbine (Fig. 4-3)
- a gas turbine with a waste heat boiler (Fig. 4-4)

Though the gas turbine with a waste heat boiler is no longer a genuine combined-cycle plant (it operates without a steam turbine), it can be viewed as a limit case. All installations can be equippped with supplementary firing, which might even be of considerable advantage for the cogeneration process because it offers a much greater design and operating flexibility than available with waste heat utilization alone. The production of steam or thermal energy can be controlled independently of the electrical power output because the gas turbine assumes control of the power output and the auxiliary firing handles control of the steam or heat generation. Cogeneration plants can be classified into three categories as follows:

- industrial power stations to supply process steam to industrial plants
- thermal power stations to supply thermal energy to district heating systems
- power plants coupled to seawater desalination plants

4.1 Industrial Power Stations

Wherever both electrical power and process steam are needed, it is thermodynamically and generally also economically better to produce both products in a single plant.

The number of possible solutions is large because each plant is a "special case." As an example, we will cite here a process with a single pressure level for the process steam. Often the cases involved are more complicated but the basic considerations remain unchanged. One important core parameter for cogeneration plants is the power coefficient, the ratio between the electrical power and the thermal energy produced [61].

One characteristic of the combined-cycle plant is its high minimum value for this power coefficient. It is therefore more likely to be suitable for processes where a relatively great amount of power must be generated.

Figure 4-1

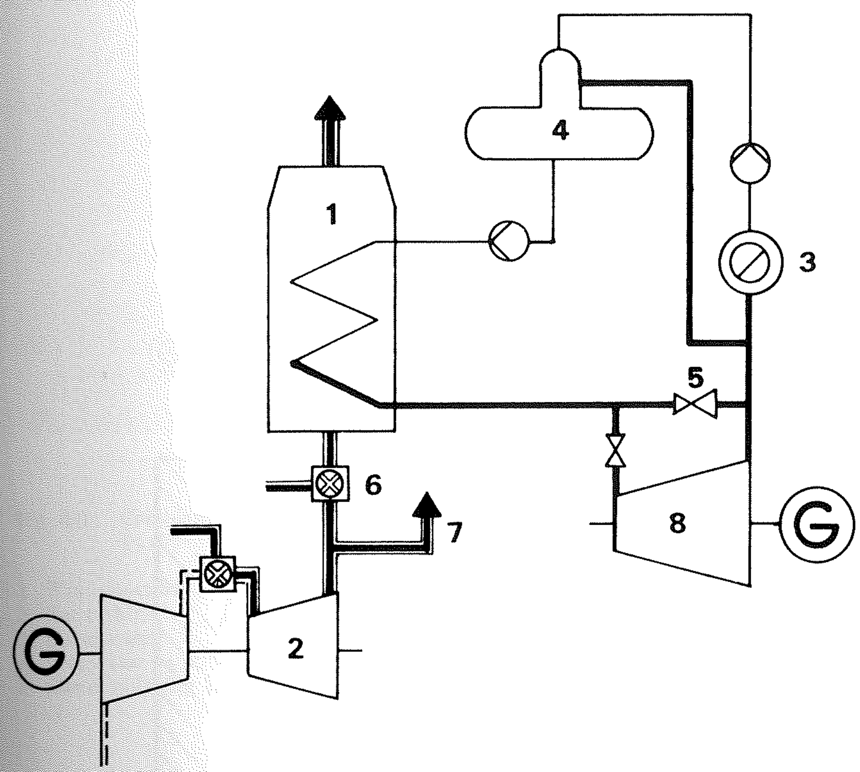

Fig. 4-1: Diagram of the Principle of a Combined-Cycle Plant used for Cogeneration

1	Waste heat boiler	5	Steam reducing station
2	Gas turbine	6	Supplementary firing
3	Steam user	7	Flue gas bypass
4	Feedwater tank/deaerator	8	Backpressure steam turbine

Figure 4-2

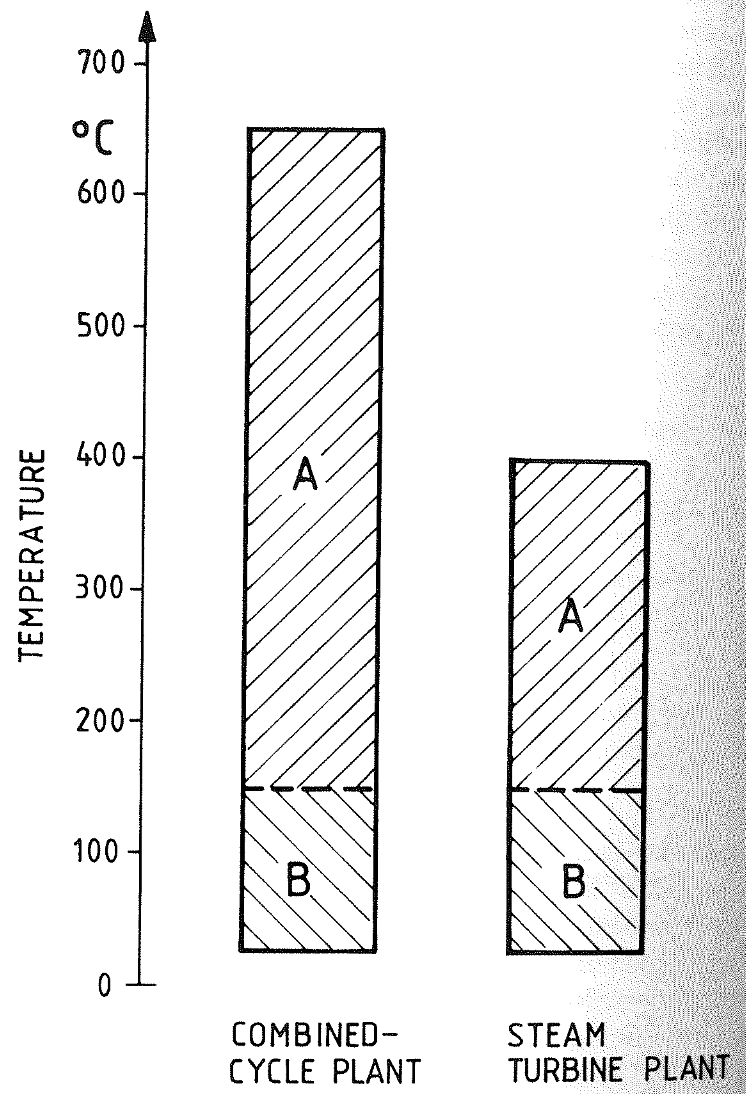

Fig. 4-2: Comparison of the Temperature Drop of Combined-Cycle Plants and Steam Turbine Power Plants

A Temperature drop available in a backpressure steam turbine plant
B Temperature drop available in the industrial process
A+B Temperature drop available in a condensing steam turbine power plant

Figure 4-4

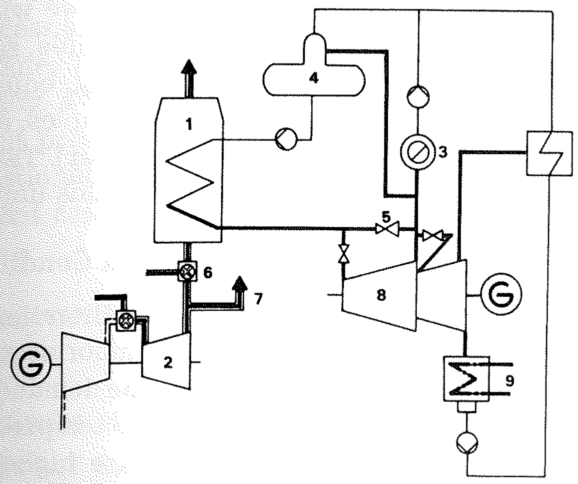

Fig. 4-3: Combined-Cycle Power Plant with Extraction/ Condensing Steam Turbine

1 Waste heat boiler
2 Gas turbine
3 Steam user
4 Feedwater tank/deaerator
5 Steam reducing station
6 Supplementary firing
7 Flue gas bypass
8 Backpressure steam turbine
9 Condenser

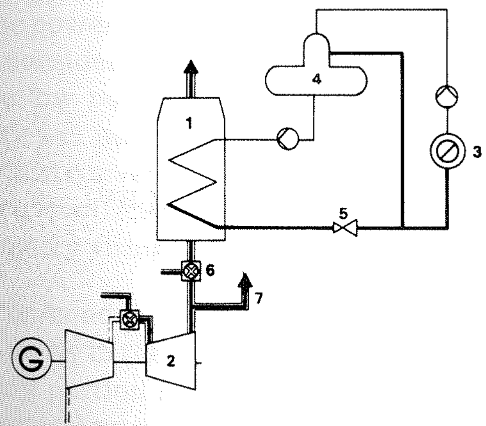

Fig. 4-4: Gas Turbine with Waste Heat Boiler

1 Waste heat boiler
2 Gas turbine
3 Steam user
4 Feedwater tank/deaerator
5 Steam reducing station
6 Supplementary firing
7 Flue gas bypass

Example of a Combined-Cycle Industrial Power Plant

Just as we did for power generation alone, we will base our further considerations here on one given example, specifically: a gas-burning plant with supplementary firing, equipped with the same 70 MW gas turbine as our other examples, and a backpressure turbine (absolute backpressure 3.5 bar). Table 4-1 and Fig. 4-5 contain the main technical data for this unit.

The Effect of the Most Important Design Conditions

As with combined-cycle plants used for generation of power alone, the air temperature is of particular importance here, too, for power output. In industrial processes, the demand for power generally does not depend on the ambient temperature. As a result, one is often compelled to select the highest ambient temperature for design purposes.

Table 4-1: Main Technical Data of the Combined-Cycle Industrial Power Plant

Fuel	Natural gas	
Gas turbine power output	69 100	kW
Backpressure steam turbine power output	44 700	kW
Station service power	1 400	kW
Net power output of the plant	112 400	kW
Heat input to the gas turbine (LHV)	230 000	kW
Heat input to the supplementary firing (LHV)	79 600	kW
Process steam flow	65.3	kg/s
Process steam pressure	3.5	bar
Thermal energy of process steam	152 000	kW
Rate of fuel utilization	85.4	%
Power coefficient	0.74	
Electrical yield	36.8	%
Efficiency of power production	79.9	%

The pressure level of the process steam and the power coefficient also are of importance for design, because the pressure of the process steam directly affects the enthalpy drop in the steam turbine. The higher the pressure, the less electricity is produced. Fig. 4-6 shows the effect on the overall output of the plant. If this pressure is very high, the use a steam turbine becomes questionable since its pressure differential is then too small. In that case, the steam process reduces to a waste heat boiler.

The power coefficient of the plant is affected mainly by three parameters:

- the amount of fuel supplied directly to the boiler
- the size of the condensing portion of an extraction/condensing turbine
- the pressure level of the process steam

Supplementary firing makes it possible to lower the power coefficient. However, this capability is limited because lowering it too much would reduce the thermodynamic advantages of the combined-cycle plant. Combined-cycle plants should be employed only where the power coefficient is high. That feature must not, however, be considered as a disadvantage since for industries which require low power coefficients it is often a better idea to obtain the power needed from the connected grid and to generate the steam in conventional steam generators. Fig. 4-7 shows how the power coefficient depends on the temperature after the supplementary firing.

An extraction/condensing turbine offers greater design and operating flexibility in the direction of higher power coefficients. The condensing portion of the turbine makes it possible to increase the electrical power produced at the cost of process steam generation. However, this procedure works out unfavorably on

Figure 4-5

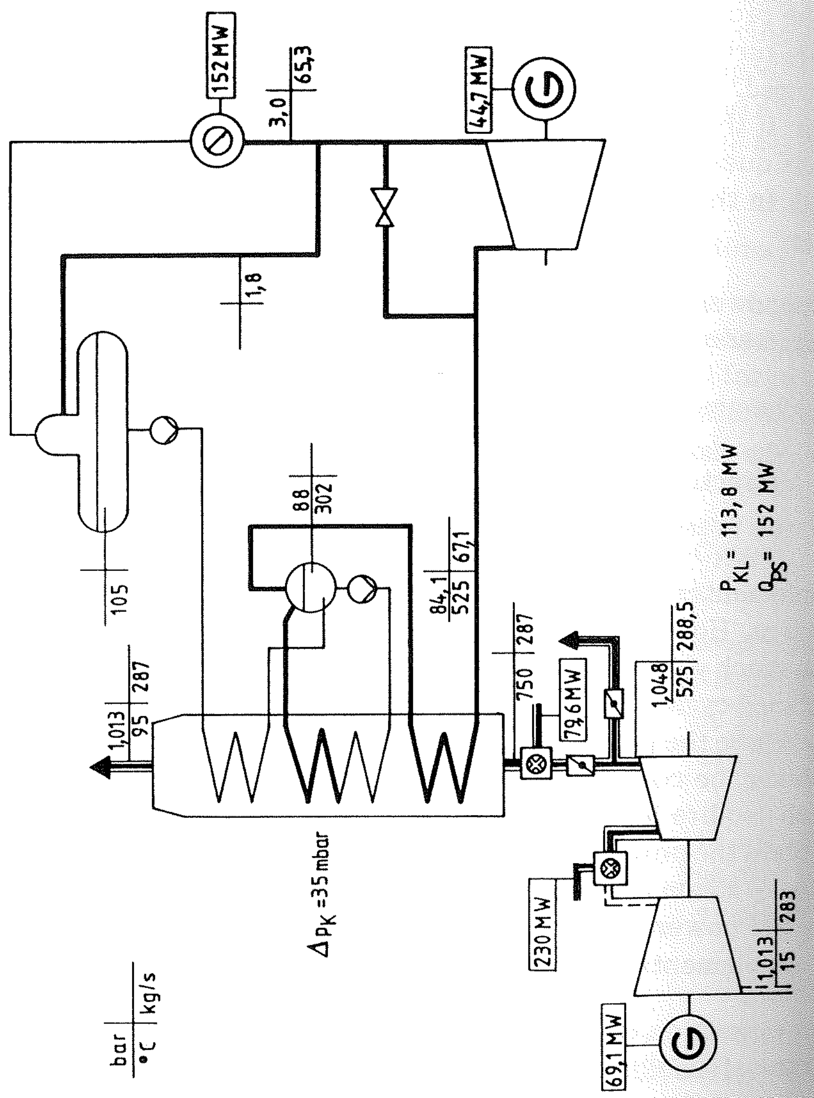

Fig. 4-5: Example of a Combined-Cycle Industrial Power Plant

Figure 4-6

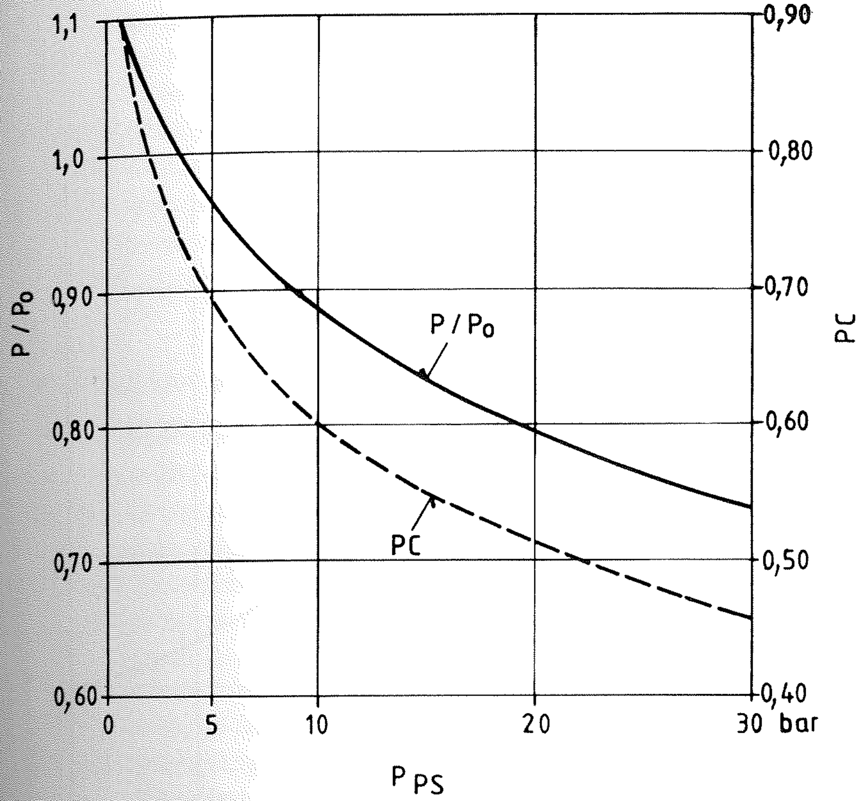

Fig. 4-6: Effect of Process Steam Pressure on the Electrical Power Output of the Combined-Cycle Plant

P/P_0	Relative power output
PC	Power coefficient
P_{PS}	Process steam pressure

Figure 4-7

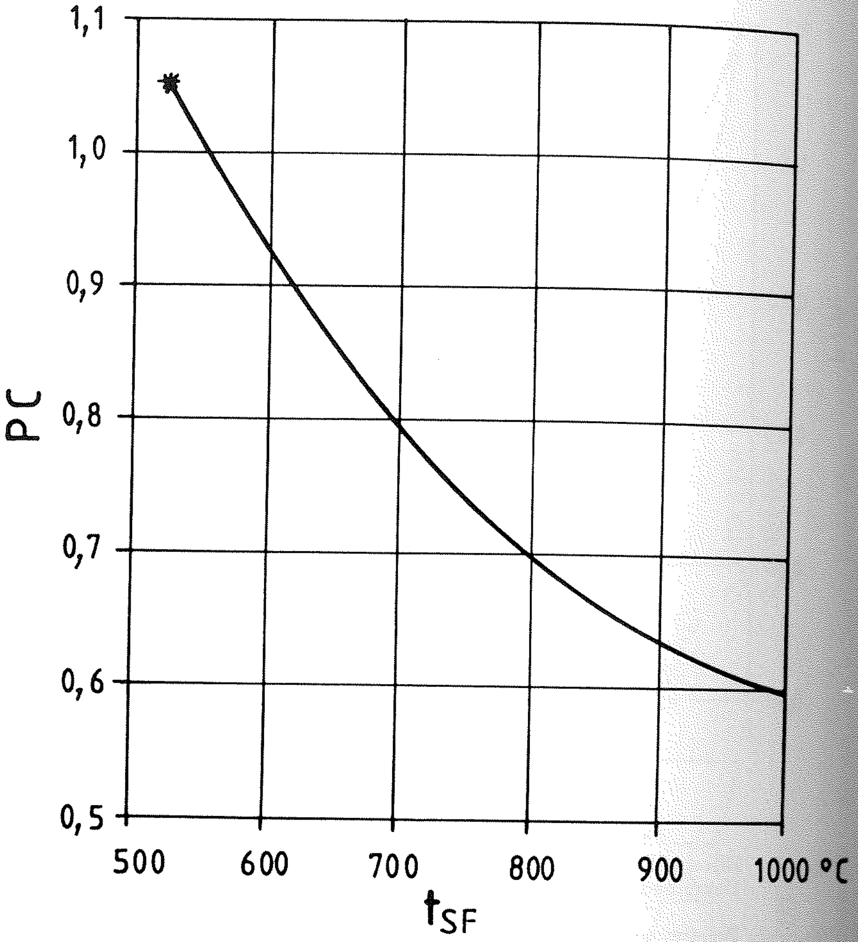

Fig. 4-7: Effect of the Temperature after the Supplementary Firing on the Power Coefficient (*Case without Supplementary Firing)

PC Power Coefficient
t_{SF} Flue gas temperature after supplementary firing

the overall efficiency. As shown in Fig. 4-8, when the power coefficient increases, this indicator approaches asymptotically the efficiency of the combined-cycle plant used to generate power alone. That means that the backpressure turbine represents the optimum case for cogeneration. As soon as the power coefficient required changes, one moves away from that optimum. To the left of the maximum is the case with mixed process steam production: one portion of the process steam is being produced in the backpressure turbine and the remainder in a steam generator. Here, too, the efficiency of power production drops off rapidly.

Effect of the Most Important Parameters

When selecting the design data, it is necessary to distinguish between a fired and an unfired waste heat boiler. With supplementary firing, one should select live steam data similar to those for conventional steam power plants. Just as for plants used to generate electricity, the feedwater temperature must be as low as possible so as to attain a good utilization of the heat in the waste heat boiler.

Without supplementary firing, the live steam data should be selected according to criteria similar to those indicated in Section 3.1. The live steam pressure should, however, be higher to assure a reasonably high enthalpy drop between the live steam and the process steam. This is especially true if a relatively high pressure level is required for the process steam. Poorer heat utilization results, and this loss must be regained in a low pressure evaporator. Normally such low pressure systems are used only for the generating of process steam. Generally it is not worthwhile to introduce the low pressure steam into the turbine because of the slight pressure differential available between the low pressure and the process steam.

Fig. 4-9 shows a system employing an unfired two-pressure waste heat boiler and a backpressure turbine. The low pressure steam is fed directly into the process steam system. The low pressure economizer shown is of interest only when burning sulphur-free fuel. In oil-burning plants, one should employ a system similar to that in Section 3.1.2 (single-pressure system with a preheater loop).

When designing a cogeneration power plant, the rule is always that for given economic conditions its design must always be simpler and less expensive than that for a power plant which generates power alone. Thus, for example, the efficiency of the turbine can be a bit poorer because the losses in the machine are recovered energetically in the form of additional process heat.

4.2 District Heating Power Plants

Generally, district heating power plants produce heating water at temperatures lower than those in industrial power plants. Usual levels are between 80 and 150 °C (176 and 302 °F). Because steam is used to heat the water, it can, for exergetic reasons, be helpful in larger installations, to break down the heating into two or even three stages. Fig. 4-10 shows the temperatures of hot water and steam in a 1-stage and a 3-stage process. Where only one stage is used, all the steam must be extracted at the higher pressure level, which means that the power output attainable is less than that with a 3-stage system.

Other design criteria for district heating power plants are their heat output and the water temperatures, which depend on ambient temperatures. The system used must therefore possess great operating flexibility, but must not become too complicated or too expensive. In particular, no district heating power plant should ever be designed for extreme conditions: its economy of operation would be questionable.

Figure 4-8

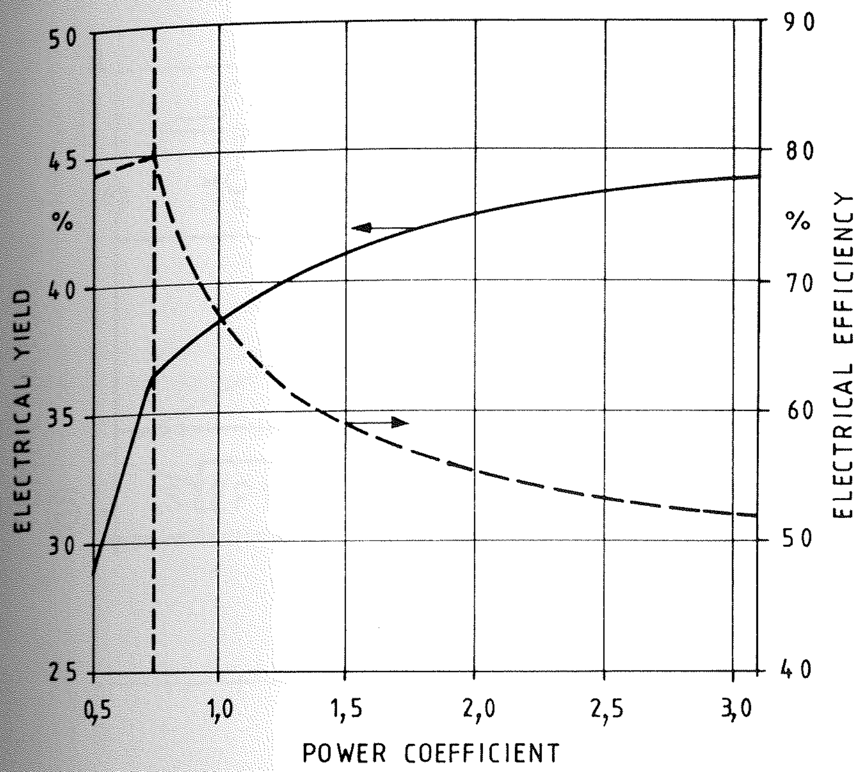

Fig. 4-8: Effect of the Power Coefficient on the Electrical Efficiency and Electrical Yield

Figure 4-9

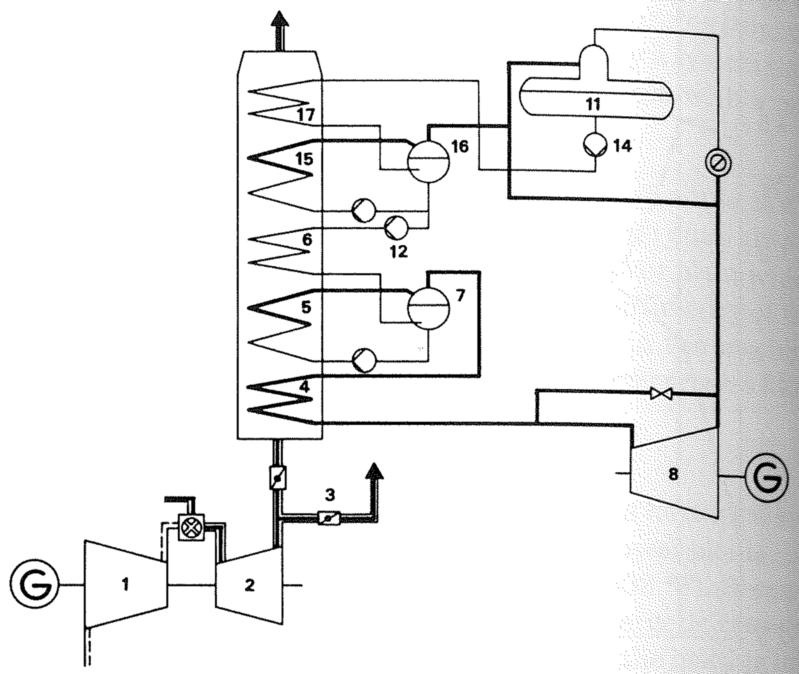

Fig. 4-9: Combined-Cycle Industrial Power Plant with Two-Pressure Waste Heat Boiler

1 Waste heat boiler
2 Gas turbine
3 Steam user
4 Feedwater tank/deaerator
5 Pressure reducing station
6 Supplementary
7 Flue gas bypass
8 Backpressure steam turbine

Figure 4-10

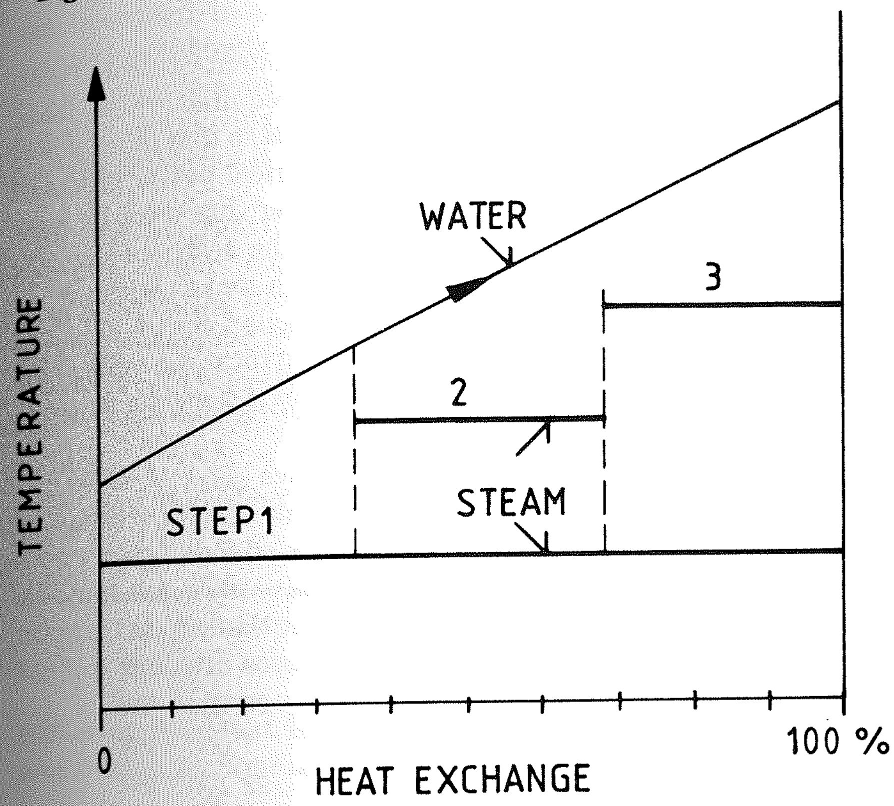

Fig. 4-10: Comparison of 1-stage and 3-stage heating of the hot water

Example of a Combined-Cycle Power Plant for District Heating

In district heating power plants, independent control of electricty and heat production is generally not required. These power plants are usually integrated into large grids so that other power stations can take over adjusting the electrical power produced to the demand. The only output parameter that must be regulated is the heat output. For that reason, the design of the combined-cycle plant should be simple, which means without supplementary firing in the waste heat boiler. Fig. 4-17 shows the arrangement and heat balance of a typical example using an extraction/backpressure turbine. Table 4-2 shows its main technical data.

Table 4-2: Main Technical Data of the Combined-Cycle Heating Power Plant

Fuel	Natural gas		
Gas turbine output	69 400	kW	
Steam turbine output	23 700	kW	
Station service power	1 000	kW	
Net power output	92 100	kW	
District heating water return temperature	70	°C	(158 °F)
District heating water supply temperature	110	°C	(230 °F)
Heating output	109 500	kW	
Heat input (LHV)	230 000	kW	
Rate of fuel utilization (LHV)	87.6	%	
Power coefficient	0.84		
Electrical yield (LHV)	40.05	%	

The system depicted is based on a single-pressure steam process. In order to increase the rate of utilization of the waste heat energy, it is followed by a heating circuit heated by the flue gas. In the example shown, this circuit is built as a closed system. A system employing the district heating water directly would be conceivable but it would mean greater problems in operation.

As a variant from the example shown, the steam turbine could be an extraction/ condensing turbine, which would provide a greater operating flexibility. Whether or not such an added investment would be worthwhile depends mainly on the value attached to the additional electricity which could be produced. Fig. 4-12 shows a system using an extraction/condensing turbine.

In certain cases, even a complete two-pressure system could be conceivable. The prerequisites for this however would be that a very high value be attached to the electrical power produced and that the district heating water supply temperature be low in order that there would be a reasonable enthalpy drop between the low pressure steam and the heating condenser(s).

Effect of the Most Important Ambient Conditions and Design Parameters

In combined-cycle plants for district heating, the strong effect that the air temperature has on the power output is more likely to be a positive factor since maximum output is demanded when temperatures are lowest.

As was true in industrial power plants, the temperature level of the heat being supplied affects the power output of the steam turbine. For that reason, the temperatures selected for the district heating water should be as low as possible. The design temperatures represent a compromise between maximum electrical output and low costs for transportation of the heat.

As in combined-cycle industrial power plants, the live steam pressure should also be higher than in power plants for power generation alone. Levels of between 40 and 70 bar (565 and 1000 psig) are typical for optimum design of installations without supplementary firing.

4.3 Power Plants Coupled with Seawater Desalinization Units

Combination of a power station with a seawater desalinization unit is one especially interesting application for cogeneration. Combined-cycle plants are outstandingly well suited to this purpose because such power plants are generally built in oil-rich countries and ideal fuels for combined-cycle installations are therefore easily available at a reasonable cost.

Larger seawater desalinization plants are always designed to use the multi-stage ''multiflash'' process. One significant technical requirement imposed by the process of such equipment is that the maximum temperature of the water being heated be limited. The reason for this is the way in which the water is treated to prevent $CaCO_3$ deposits. Generally the method involves treatment with polysulphate or sulphuric acid.

With polysulphate, the maximum temperature to which the water can be heated is 90 °C (194 °F); with sulphuric acid, 120 °C (248 °F). The resulting heating steam pressures are between 1 and 2.5 bar (14 and 36 psia), which provide ideal conditions for a combined cycle plant because the usable enthalpy drop in the back-pressure steam turbine is high, ensuring a large electrical output.

Example

In combined-cycle plants used with seawater desalinization units, the electrical power output and the flow of process steam must be controlled independently of one another. A supplementary firing is therefore recommended. Fig. 4-13 shows the principle and the heat balance of such a power station.

The desalinization process is based on a ''Multiflash'' system in which the seawater is treated with polysulphate. The heating steam pressure is 1.2 bar (approx. 17 psia) and the specific heat consumption 250 kJ/kg (108 Btu/lb) distillate. This corresponds approximately to a 20-stage desalinization plant. The main technical data are shown in Table 4-3.

An installation of this type becomes very interesting if the value attached to the electrical yield is high. Even if the process heat is viewed as a pure loss, the electrical efficiency reaches approximately 40%. This high a value is attained in a conventional steam power plant at best when only electricity is being produced. The combined-cycle plant is less suitable if the ratio between fresh water and electrical power must be high. The power coefficient would then be low. In such a case, either the additional steam demanded for the desalinization unit must be supplied from an auxiliary boiler, or a different type of power plant must be chosen. Higher power coefficients can be attained using extraction/condensing steam turbines. However, the significant advantage of requiring only a negligible amount of cooling water in the steam process is lost and the unit becomes more complicated and expensive.

Table 4-3: Main Technical Data for the Combined-Cycle Plant Coupled to a Seawater Desalinization Unit

Air temperature	30	°C
Fuel	Natural gas	
Gas turbine output	2 x 69 100	kW
Steam turbine output	109 600	kW
Station service power	2 700	kW
New power output of the plant	245 100	kW
Process steam flow	130.6	kg/s
Distilled water flow	1130	kg/s
Process steam pressure	1.2	bar
Heat input to the gas turbine (LHV)	2 x 230 000	kW
Heat input to the supplementary firing (LHV)	2 x 79 600	kW
Rate of fuel utilization	85.3	%
Power coefficient	0.886	
Electrical yield	39.6	%

Figure 4-11

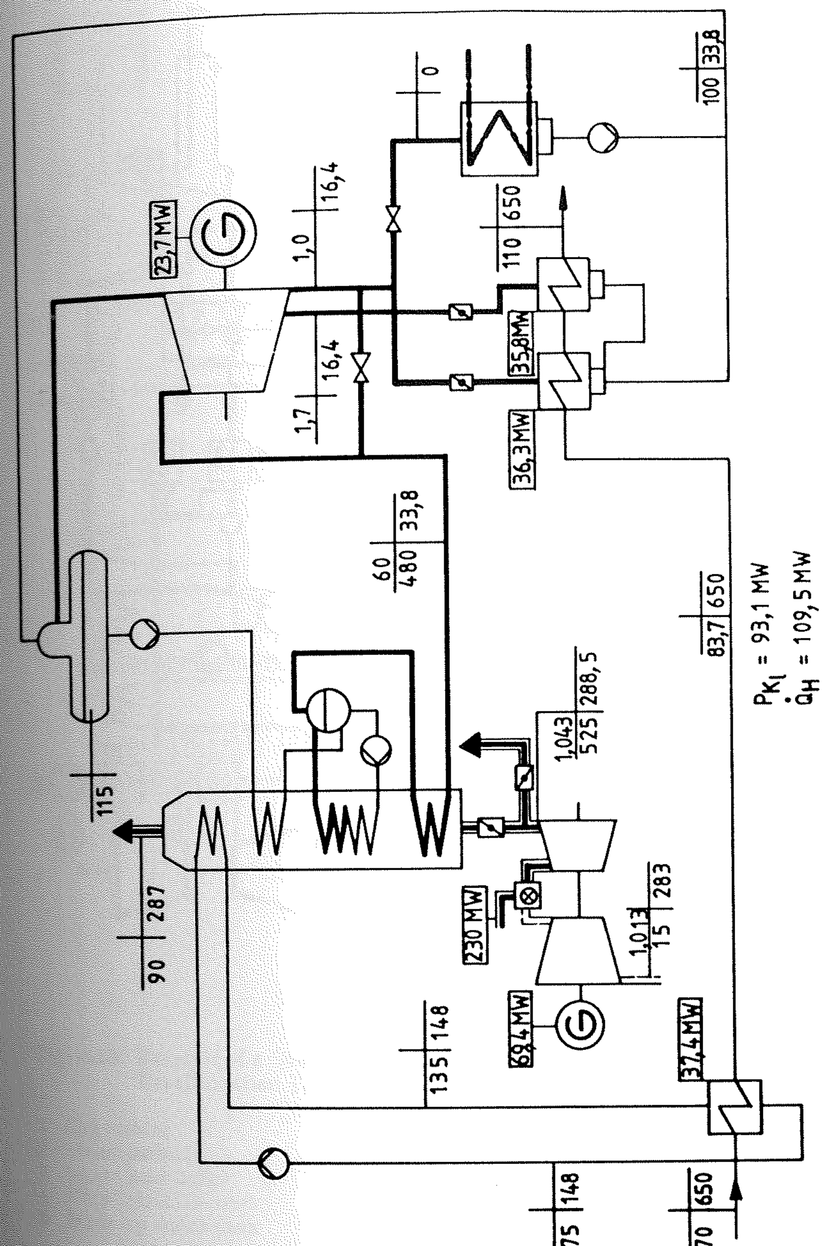

Fig. 4-11: Example of a Combined-cycle Heating Power Plant with Back-pressure Turbine

Figure 4-12

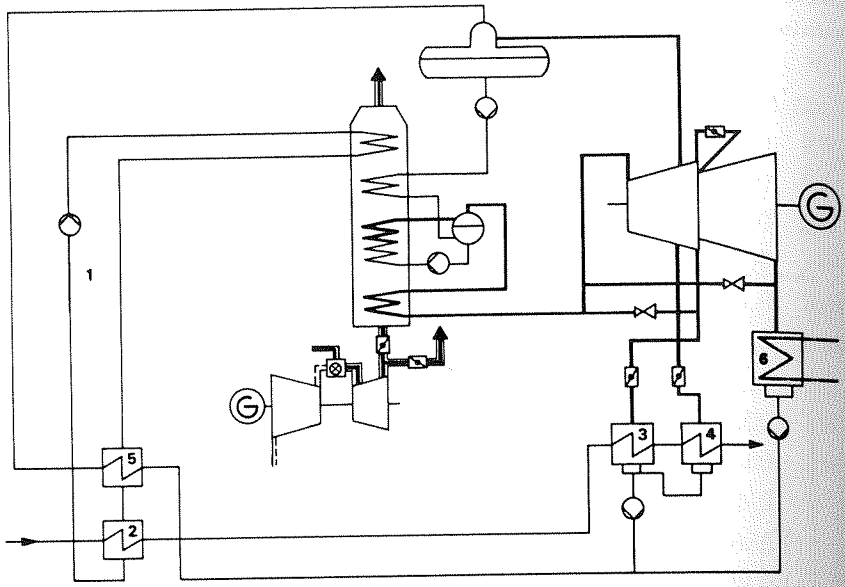

Fig. 4-12: Combined-Cycle Heating Power Plant with Extraction/Condensing Turbine

1 Heating circuit
2 Heaters
3-4 Heating condensers
5 Preheaters
6 Condenser

Figure 4-13

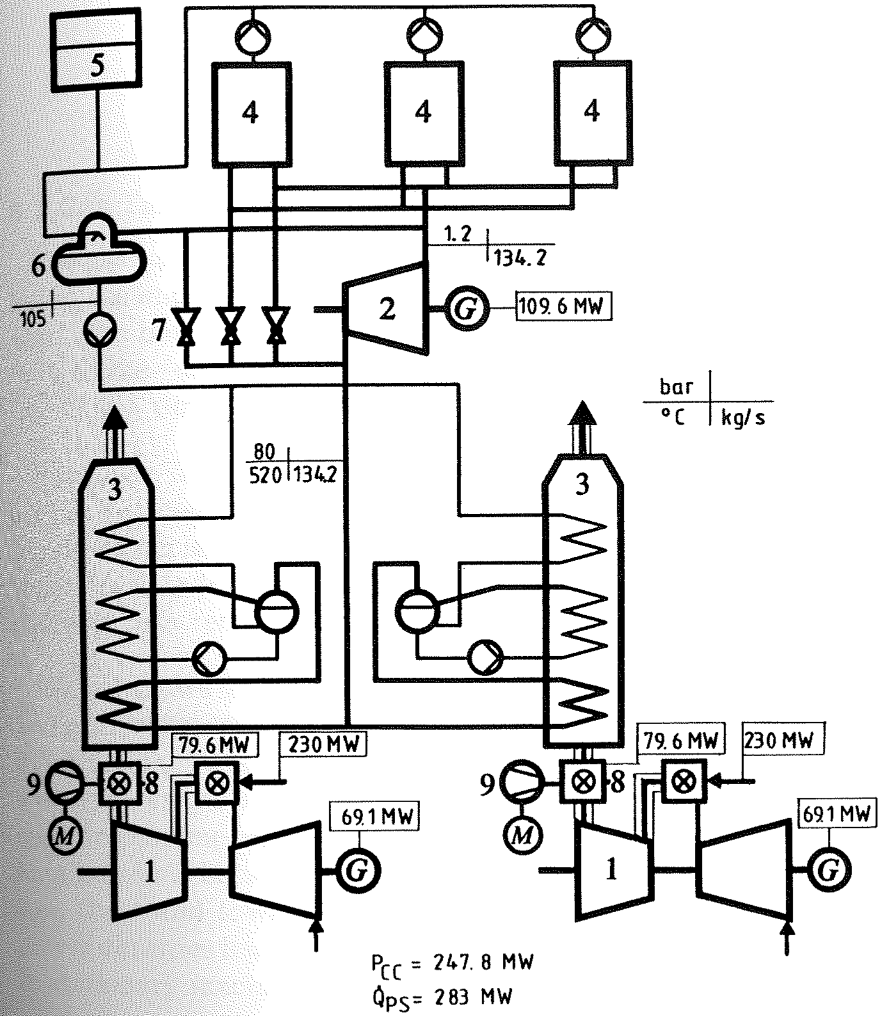

Fig. 4-13: Example of a Combined-Cycle Power Plant Coupled with a Seawater Desalinization Unit

1 Gas turbine
2 Steam turbine
3 Waste heat boiler
4 Desalinization unit
5 Make-up water tank
6 Feedwater tank
7 Pressure reducing stations
8 Supplementary firing
9 Fresh air fan

Chapter 5

COMPONENTS

5.1 Gas Turbine

The gas turbine in the most important component in the combined gas and steam turbine power plant. The combined-cycle plant has been able to become a competitive thermal process only as the result of the rapid development in the direction of higher gas turbine inlet temperatures.

Parallel to this development in the turbine, there has also been an improvement in the compressor. Today the compressor can handle much larger mass flows and higher pressure ratios, making it possible to attain considerably higher power outputs and thereby both reduce costs and improve efficiency.

Fig. 5-1 shows the historical development of maximum air flows and gas turbine inlet temperatures. Inlet temperatures are higher in jet turbines than in industrial gas turbines. In a jet turbine, weight plays the dominant role: procurement and maintenance costs are less important than with stationary gas turbines, where long intervals between overhauls are demanded. For that reason, the trend toward higher inlet temperatures and greater power densities has progressed more rapidly in jet turbines than in stationary machines.

Stationary gas turbines can be classified into one of three categories:

- industrial gas turbines derived from steam turbine technology

- industrial gas turbines derived originally from jet technology
- the aero-derivative turbine, consisting of a jet engine followed by a power turbine

The last of these is normally a two-shaft turbine with a variable speed for the compressor and the driving turbine. This is an advantage with regard to part-load efficiency since the amount of air taken in is reduced due to the lower speed. One disadvantage when operating with a generator, however, is that there is no compressor braking the power turbine during load shedding. Two-shaft turbines are usually used for compressor or pump drives, where the operating speed of the power turbine is also variable. On the other hand, turbines of the first two types are practially always built as single-shaft machines when used to drive a generator with an output greater than 15 to 20 MW. Fig. 5-2 to 5-4 show typical modern gas turbines of each of the three main types.

One important fact is that the machines have become standardized. As a result, they can be built as stock units, making possible shorter lead-times and lower prices. Because of this standardization, there are only a few types of machines available on the market and it is never possible to buy an exact specified power capacity. However, the advantages brought by standardization outweigh this consideration.

Table 5-1 shows the characteristic data of modern gas turbines used for combined-cycle installations.

Table 5-1: Main Technical Data of the Most Common Gas Turbines Available on the Market

Power output (ISO conditions)	1 - 150	MW
Efficiency (ISO conditions)	28 - 35	%
Pressure ratio	10 - 18	
Gas turbine inlet temperature[1)]	950 - 1150	°C
Exhaust temperature	480 - 570	°C
Exhaust flow	30 - 500	kg/s

1) According to ISO definition. Actual hot gas temperatures are 100 to 200 °C (180 to 360 °F) higher, depending on the amount of cooling air required for the turbine.

The main problem with gas turbines lies in how fast their development has been progressing. They are developed primarily for use as a gas turbine alone. But because fuel costs are also rising quickly, the attempt is being made to make a corresponding improvement in gas turbine efficiency and to reduce the specific investment costs required for them. As a result, turbine inlet temperatures have risen very quickly, which has not in the past always had a positive effect on availability.

At the present time, this situation has changed and gas turbines have attained a very high reliability. They can therefore be installed in large base-load or medium-load combined-cycle plants and provide the same high availability as conventional steam turbine power plants.

For this type of applications, fouling in the compressor and the turbine is of greater concern. Compressor fouling occurs because the gas turbine operates in an open cycle, drawing in air which can never be cleaned completely. Turbine fouling becomes a problem only if such "dirty" fuels as crude or residual oils are used.

Figure 5-1

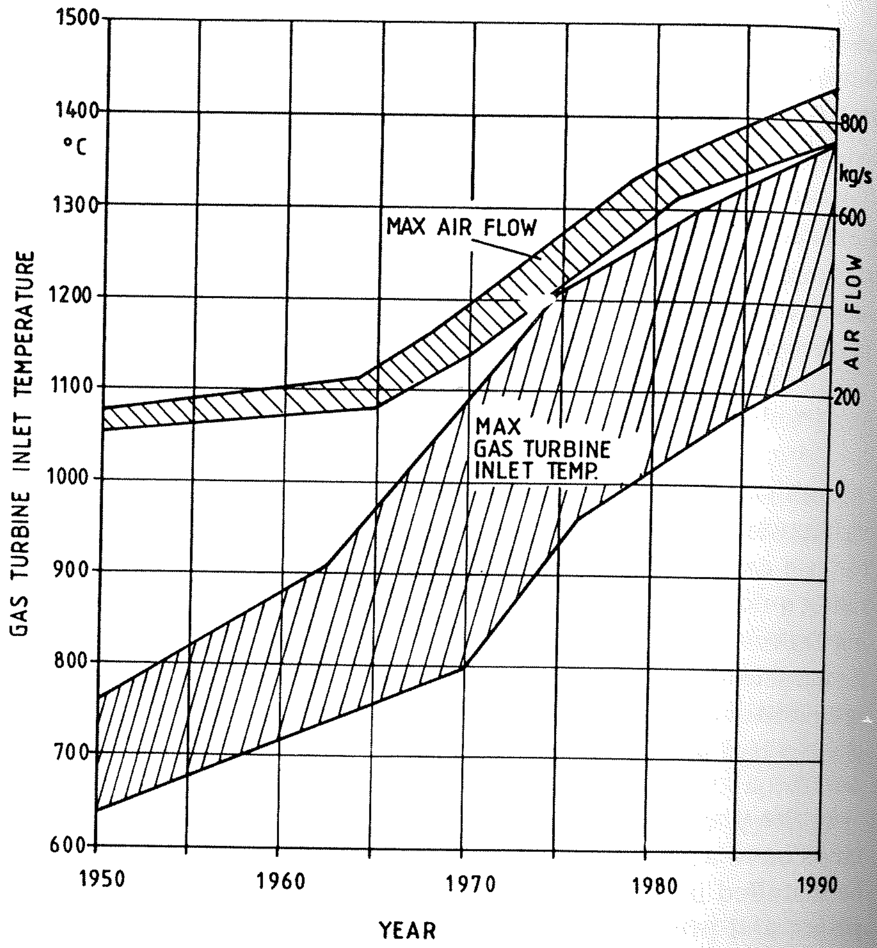

Fig. 5-1: Historical Trends in Gas Turbine Inlet Temperatures and Fir Flows over the Years

Note: The gas turbine inlet temperature indicated is the actual hot gas temperature before the first stage of the turbine. The inlet temperature as defined by ISO is 150 to 250°C lower in modern gas turbines. The upper limits are temperatures reached in jet engines.

Figure 5-2

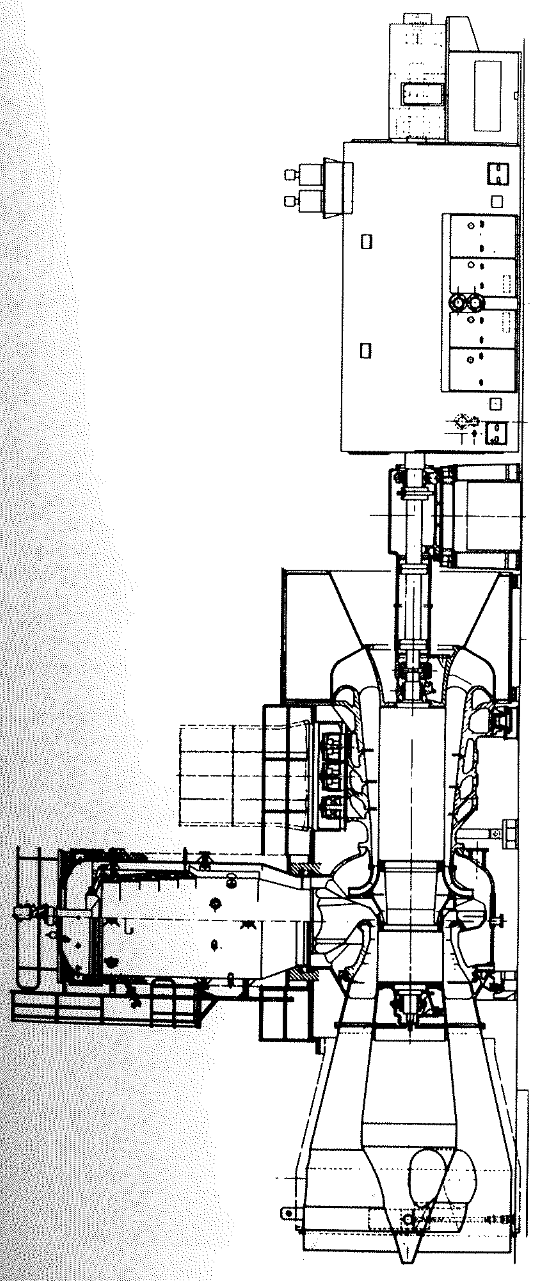

Fig. 5-2: A typical Industrial Gas Turbine

Figure 5-3

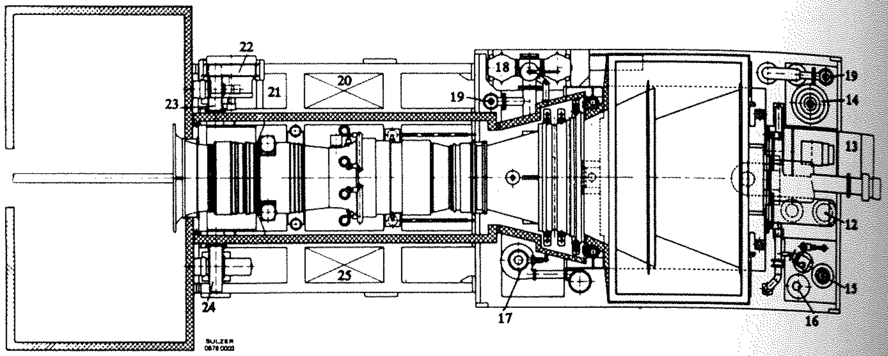

Fig. 5-3: Diagram of an Aero-Derivative Turbine

1 Intake housing
2 Starting shaft
4 Acoustic hood
5 Base frame for gas generator
6 Transition housing connecting gas generator and power turbine
7 Power turbine
8 Diffuser and exhaust housing
9 Auxiliaries and ancillaries
10 Power clutch
11 Base frame for power turbine and driven machines
12 Mechanically driven lube oil pump for power turbine and driven machines
13 Machine-mounted AC generator (optional equipment
14 Electrically powered lube oil pump for power turbine and driven machines
15 DC powered lube oil pump for cooling the power turbine bearings
16 Oil intake pipe, with strainer
17 Oil vapor extraction fan, driven by an oil motor
18 Switch-over twin lube oil filter
19 Constant pressure valve for lube oil
20 Lube and hydraulic oil system for gas generator
21 Lube oil tank for gas generator
22 air/oil heat exchanger for gas generator
23 Cooling air fan, driven by hydromotor
24 Cooling air fan, driven by electro-motor
25 Fuel control system

Figure 5-4

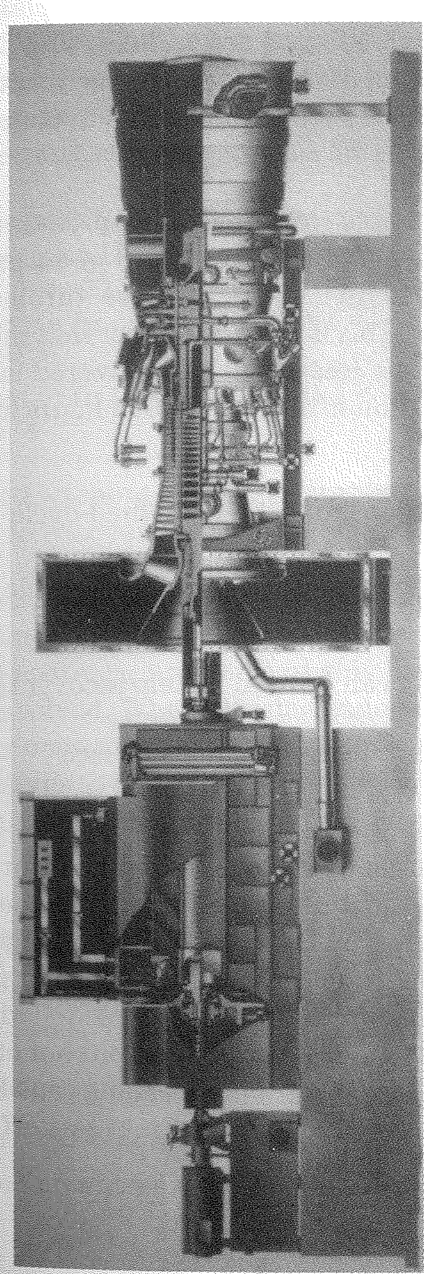

Fig. 5-4: Example of an Industrial Gas Turbine Based on Jet Technology

Compressor fouling is countered by installing an air filtration system that is suited to the environment at the plant site. The filters most frequently employed are self-cleaning pulse filters or 2-stage filters. The first of these is particularly suitable in dry climates. It is helpful in a base-load plant to oversize the filter in order to limit the rate of fouling.

It is, however, impossible to keep the compressor completely clean. The fouling that results causes losses in output and efficiency that are greater in single-cycle gas turbines than in combined-cycle plants. This is due to the fact that in combined-cycle plants, some of the losses can be recovered in the steam cycle. Typical degradation after 1000 - 2000 hours in operation on a clean fuel:

- Reduction in output from the combined-cycle plant
  3 to 6%
- Reduction in efficiency of the combined-cycle plant
  2 to 3 %

Two types of cleaning can be used to help recover these losses:

a) a "dry" cleaning, using nutshells or rice

b) washing

The second of these methods is more suitable for large modern gas turbines because it is more efficient and does less damage to protective coatings on the compressor blading. However, it requires shutting down and cooling the engine it is preferrable to wash at low speed— typically ignition speed— with the machine cold. The machine must therefore be put out of operation for approx. 24 hours. The compressor washing can also be done with the gas turbine at full load, but this is less efficient that low-speed washing.

Fouling in the turbine is due mainly to the ash contained in dirty fuels and in the additives used to inhibit high temperature corrosion. Turbine fouling is unavoidable, but it can be held within limits by selecting the correct type of additives. It is less of a problem in peaking or medium-load operation because of the self-cleaning effect produced by start-up and shut-down.

Typical degradation in combined-cycle plants after 1000 - 2000 hours in operation on heavy or crude oil:

- Reduction in output from the combined-cycle plant
  5 to 10 %
- Reduction in efficiency of the combined-cycle plant
  3 to 5 %

Low-speed washing of the cold turbine and compressor makes it possible to recover from 50 to 80% of these losses.

In the past, corrosion problems were one of the major causes of gas turbine failure. Because of the use of better blading materials and coatings, problems of this nature have today practically been solved. Whenever uncleaned fuels are being burned, particularly ones containing vanadium or sodium, it is necessary to use additives to prevent high temperature corrosion. The additives commonly used are based on magnesium, chromium, or silicon.

5.2 Steam Generator

The waste heat boilers are the links connecting the gas and the steam processes. There are three main types:

- waste heat boilers without supplementary firing
- waste heat boilers with supplementary firing
- steam generators with maximum supplementary firing

As can be seen from Section 3, the first of these types is the most interesting and the remarks that follow therefore concentrate mainly on it.

5.2.1 Waste Heat Boilers without Supplementary Firing

A waste heat boiler without supplementary firing is practically a heat exchanger. However, the requirements imposed for operation in a combined-cycle installation pose special problems that are often underestimated. In particular, provision must be made to accommodate the short start-up time of the gas turbine. Even so, the waste heat boiler is a simple component with high reliability and availability.

Waste heat boilers without supplementary firing can be built according to two principles:

- steam generators with forced circulation (Lamont type),
- steam generators with natural circulation

Either type of waste heat boiler can be used in a combined-cycle plant. A forced circulation boiler has advantages, however, that render it especially suitable for combined-cycle applications:

- minimum space requirements arising from its vertical design
- fast, easy start-up
- suitability for designs with a low pinch point
- less sensitivity to steaming out in the economizer

The main advantage of a natural circulation boiler is that no circulation pumps are needed.

The optimum waste heat boiler must fulfill the following–sometimes contradictory–conditions:

- the rate of waste heat utilization must be high (high efficiency);
- pressure losses on the flue gas side must be low in order to pre vent losses in power output and efficiency of the gas turbine
- low temperature corrosion must be prevented
- the pressure gradient permissible during start-up must be large

It is particularly difficult to meet the first two of these conditions at the same time. Because of the low temperature, the heat transfer takes place— transfer by means of radiation is negligible— almost completely by convection. Since the differences in temperature between the exhaust gas and the water or steam must be small in order to attain a good rate of waste heat utilization, the surfaces required for the heat exchange are large. This would mean large pressure losses unless the speed of the flue gas were low, which would again increase the size of the heat exchange surface. However, this problem can be solved satisfactorily by using small-diameter finned tubes. Another result of the small tube diameter is the small amount of water in the evaporator. This means that the thermal capacity will be small and favors quick changes in load.

The waste heat boilers being built today have very low pinch points and small pressure drops on the flue gas side. Values of 8 to 10 K (15 to 18 °F) at pressure losses of 25 to 30 mbar (10 to 12 in WG) are attainable.

Fig. 5-5 shows a typical modern forced-circulation waste heat boiler. Its main features are:

- a vertical arrangement, with the heat exchangers suspended in the steel structure

- circulating pumps to assure constant circulation within the evaporator
- drum hung directly on the steam generator
- finned tubes used for all heat exchange surfaces

Low Temperature Corrosion

When designing waste heat boilers, one must take care to prevent or at least to restrict low temperature corrosion. To do this, all surfaces in contact with the flue gas must be at a temperature above or only slightly below the sulphuric acid dewpoint. When burning a sulphur-free fuel, the limit is determined by the water dewpoint.

Because the heat transfer on the flue gas side is less than that on the water side, by a factor of approximately 100, it can be assumed that the temperature of the metal in the tubes is the same as the water or steam temperature. The water temperature should therefore nowhere be below the dewpoint, even when the flue gas temperature is very high. It is permissible, however, to drop slightly below the dewpoint for sulphuric acid, because the rate of corrosion in that range remains slow. Fig. 5-6 [taken from Ref. 97] shows the relationship between the loss due to corrosion and the temperature and concentration of sulphuric acid in the flue gas. Points worth noting are, on the one hand, the maximum between 100 and 130 °C (212 and 286 °F) and, on the other, the rapid increase when the temperature drops below the water dewpoint. The acid dewpoint depends upon the following factors:

- the amount of sulphur contained in the fuel
- the excess air in the combustion
- the rate of conversion X of SO_2 into SO_3
- the amount of water contained in the exhaust gas

Figure 5-5

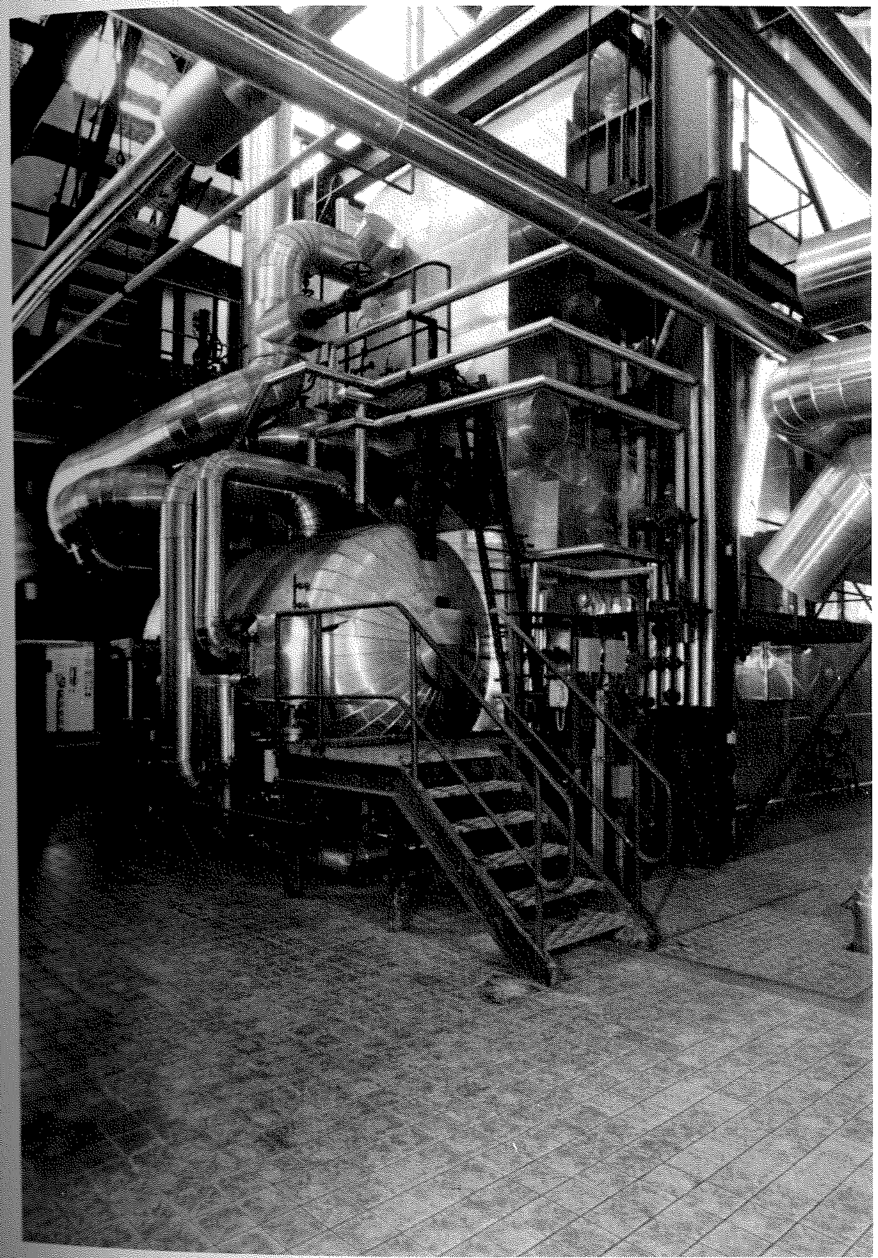

Fig. 5-5: Example of a Waste Heat Boiler without Supplementary Firing

Figure 5-6

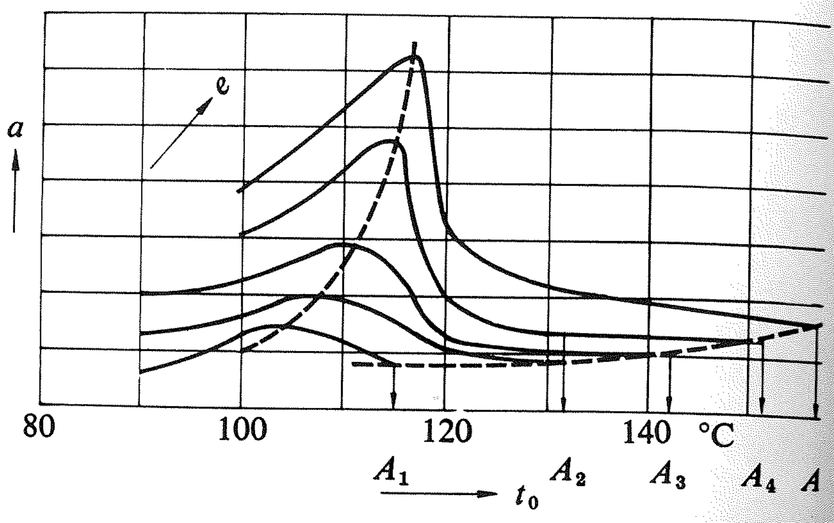

Fig. 5-6: Material Loss due to Corrosion, as a Function of Temperature and Concentration of Sulphuric Acid

A_1 - A_5	Acid dewpoints
e	Concentration of sulphuric acid
a	Material loss

The rate of conversion X from SO_2 to SO_3 depends on the type of gas turbine. Normal values are between 1 and 8%. Fig. 5-7 [taken from Ref. 97] shows how the acid dewpoint depends on the sulphur content of the fuel, the rate of conversion X, and the excess air coefficient.

Since the corrosion attack is not rapid if the temperature only drops slightly below the dewpoint, and the surface temperature of the tubes is also a few degrees higher than the water temperature, the feedwater temperature with fuels that contain sulphur can be 5 to 10 °C (9 to 18 °F) below the theoretical acid dewpoint.

Even with sulphur-free fuels, the feedwater temperature must not under any circumstances be below the water dewpoint because of the rapid increase in the rate of corrosion as soon as the tube temperature falls below that limit.

Theoretically, suitable precautions (selection of materials, additives) can make it possible to operate heat exchangers at temperatures below the acid dewpoint. However, the effectiveness and above all the economics of such measures are questionable because the additional heat obtained is low in value due to its low exergy.

Table 5-2 provides guidelines for feedwater temperatures.

Table 5-2: Selection of Feedwater Temperatures

Oil, more than 2% sulphur	140 - 145	°C
Oil, less than 1% sulphur	110 - 130	°C
Natural gas, no sulphur	50 - 60	°C

Figure 5-7

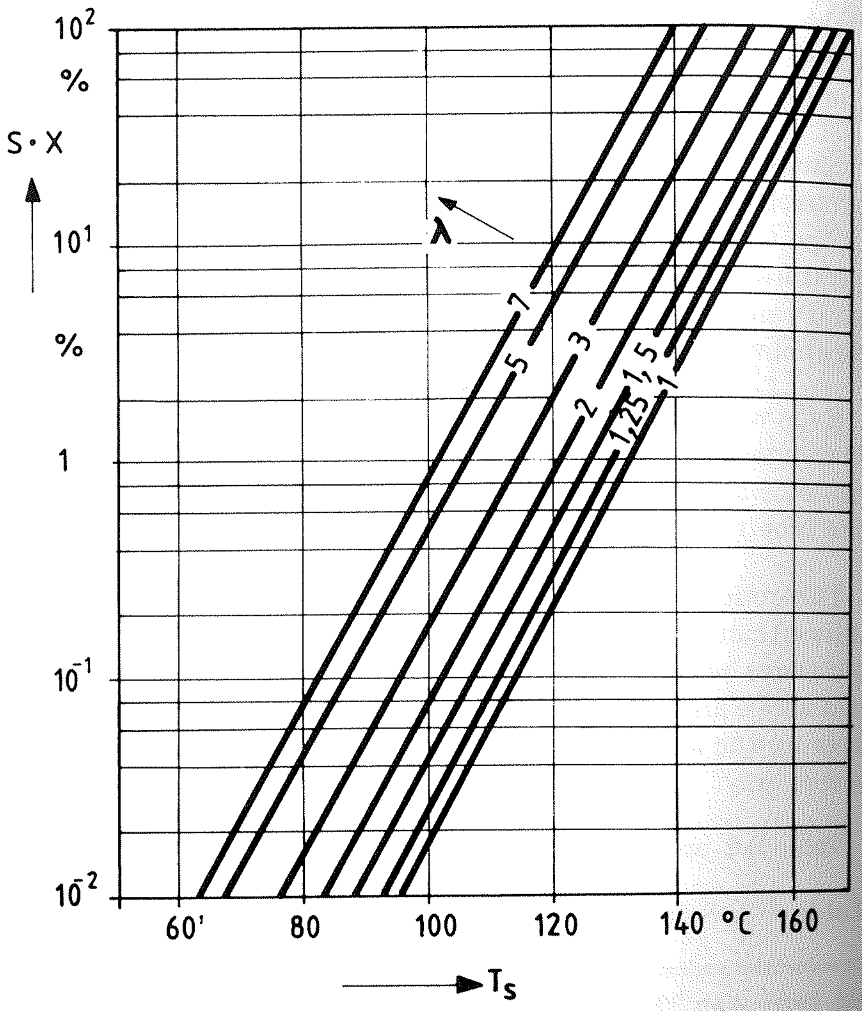

Fig. 5-7: Acid Dewpoint as a Function of the Sulphur Content of the Fuel, the Rate of Conversion from SO_2 to SO_3, and the excess air coefficient λ

T_S	Acid dew points
S	Sulphur content of fuel
X	Rate of conversion from SO_2 to SO_3
λ	Excess air ratio

Optimum design of a Waste Heat Boiler

When designing a waste heat boiler, one should strive for an optimum between cost and gain. The cost depends mainly on the heat exchange surface installed. The indicator generally used is the pinch point of the evaporator (the minimum difference in temperature between the water and the flue gas). As can be seen from Fig. 3-22, the area of the evaporator increases exponentially as the temperature differential decreases, while the increase in steam generation is only linear. For that reason, the pinch point selected is the critical factor determining the heating surface. In installations where a high value is attached to efficiency, the pinch point is 10 to 15 K (18 to 27 °F); where efficiency is valued lower, it can increase to 15 to 20 K (27 to 36 °F).

The heat exchange surface constitutes 40 to 50% of the total cost of a boiler, while the other 50 to 60% remain practically unaffected by the size of the surface. In an extreme case, too large a heat exchange surface can cause pressure losses on the flue gas side to be such that the resulting reduction in power output and efficiency of the gas turbine is greater than the power output provided by the steam turbine. A pressure drop of 10 mbar (4 in. WG) reduces the power output and efficiency by approx. 0.8%. Only a portion of this loss can be recovered in the steam process.

Problems in Operation [105]

The main problem affecting design of the waste heat boiler is the quick start-up of the gas turbine. The rapid expansions that occur during a start-up can be accommodated taking suitable design precautions such as suspension of the tube bundles, etc.

The main limit on the rate of loading arises from the drum. To make a quick start possible, the walls of the drum should be as thin as possible, which can be done provided the design live steam pressure is low.

Optimum steam pressures for installations without supplementary firing are low, 30 to 70 bar (420 to 1000 psig), which is an advantage for quick start-up.

Another operating problem is the volumetric change within the evaporator during start-up. The large differences in specific volume between water and steam at low and medium pressures cause large amounts of water to be expelled from the evaporator at the start of the evaporation process. The drum must be able to take up most of this water since otherwise a greater amount of water would be lost through the emergency drain of the drum during each start-up. The gross volume of the drum should therefore— depending on starting time and acceptable water loss— be 1.5 to 2.5 times as great as the volume of evaporator steam in normal operation. This problem of volumetric change can be held within bounds by employing recirculation so that at least no steaming out occurs in the economizer. To improve part-load efficiency and behavior of the combined-cycle plant, it is possible to operate the boiler at variable pressures:

The system is generally operated at a lower pressure when the steam turbine load is lower than at full load. This can be accomplished by employing sliding pressure operation (see Chap. 7). For example, for half-load for the power station as a whole, the waste heat boiler in a system with 2 gas turbines and 1 steam turbine can be operated with only one of the gas turbines running at full load. The live steam pressure in sliding pressure operation is only half as great as that at full load, which causes the volume flows in the evaporator, the superheater, and in the live steam duct to double. (The boiler in operation is still at full load while the connected gas turbine is at full load.) One consequence of this reduction in live steam pressure at part load is that some of the water already

steams out in the economizer. In order to keep this within limits, the economizer is generally so dimensioned that the feedwater at the outlet is slightly under-cooled at full load. This difference between the feedwater temperature and its saturation temperature is known as the "approach temperature." Because it causes a reduction in the amount of steam generated, it should be kept as small as possible, typically 5 to 10°C (9 to 18°F).

5.2.2 Waste Heat Boiler with Limited Supplementary Firing

The principle of operation of a waste heat boiler with limited supplementary firing is the same as that for the unfired boiler. There are various designs available for the firing itself. Units that do not exceed a gas temperature of approx. 750 °C (1382 °F) after the supplementary firing can be built with simple duct burners without requiring cooled combustion chamber walls (Fig. 5-8).

This system is particularly well suited to burning natural gas, with which there is no problem in attaining a uniform temperature distribution after the burners and the radiation to the walls of the combustion chamber is low. For that reason, most of the units of this construction burn natural gas. There are systems available for burning oil but because they involve major problems, it is generally a better idea to look for a different solution whenever a fairly large volume of oil must be burned. A cooled combustion chamber with oil burners such as that used in conventional steam generators is one good method of providing supplementary firing. Fig. 5-9 shows how it is constructed. The evaporator consists of a natural circulation portion, used to cool the combustion chamber, and a forced circulation portion. In order to avoid unnecessary throttling of the flue gas flow, it may be advisable to supply the burners directly with cooling air from a separate air fan.

Figure 5-8

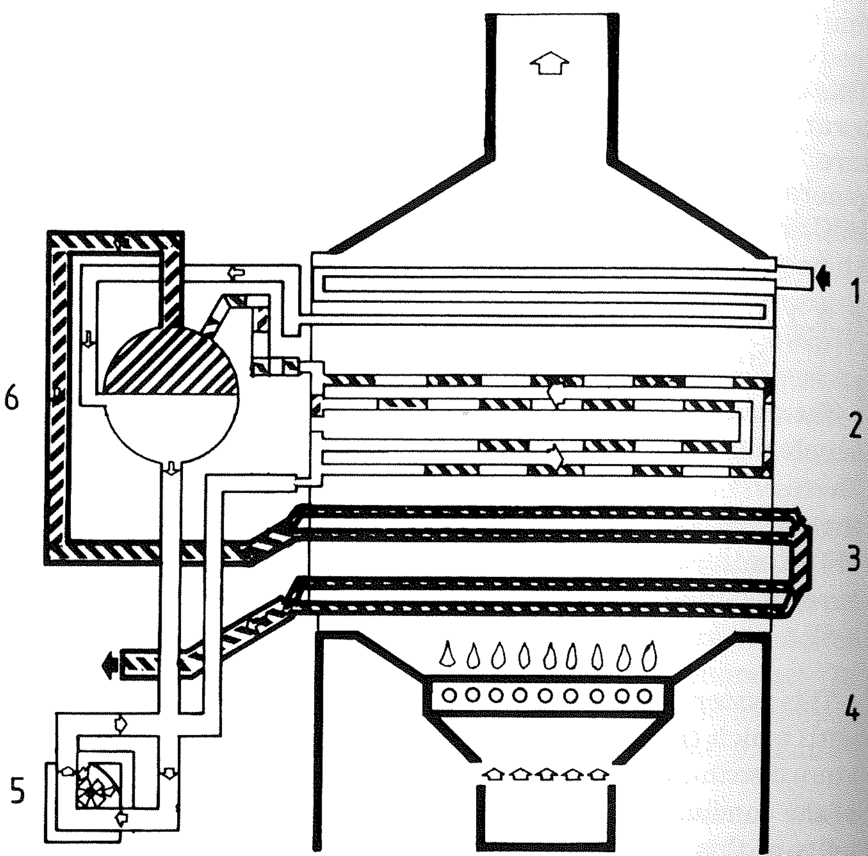

Fig. 5-8: Principle of a Waste Heat Boiler with Limited Supplementary Firing and Uncooled Combustion Chamber

1 Economizer
2 Evaporator
3 Superheater
4 Supplementary firing
5 Circulating pump
6 Drum

Figure 5-9

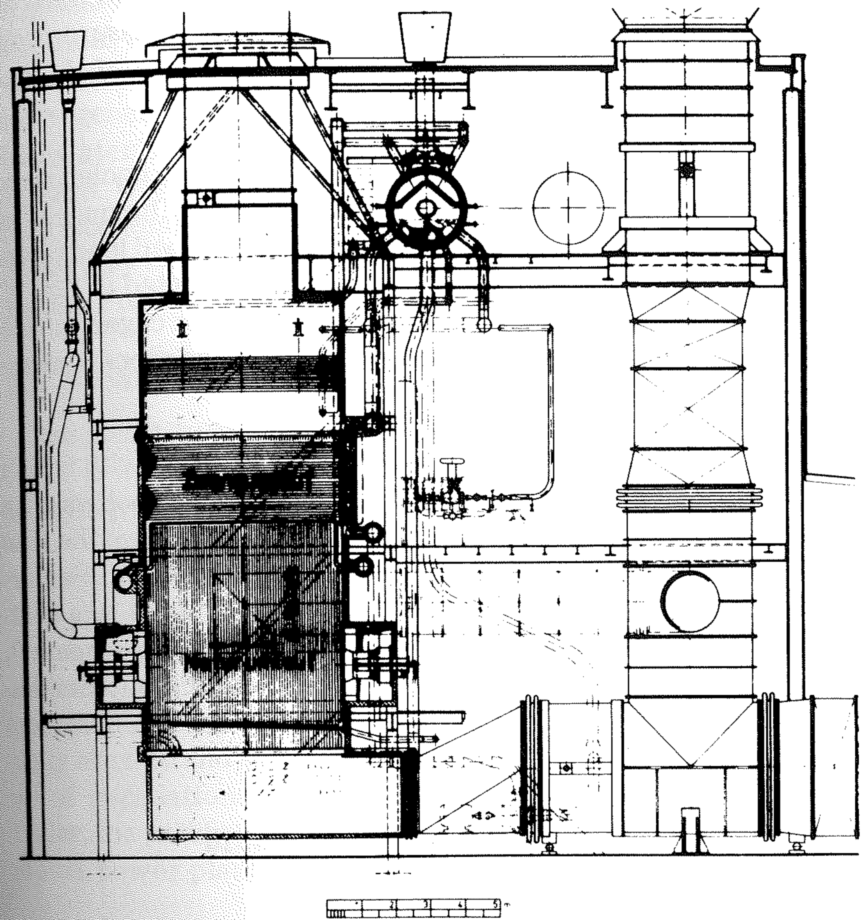

Fig. 5-9: Waste Heat Boiler with Limited Supplementary Firing and Cooled Combustion Chamber

Zwangsumlauf = Evaporator with forced circulation
Naturumlauf = Evaporator with natural circulation (Combustion chamber)

If the burners are supplied directly with flue gas, the main flow of the flue gas must be throttled down by approx. 20 to 30 mbar (8 to 12 in. WG), corresponding to the pressure drop through the burner. This reduces the power output of the gas turbine by approx. 1.5%.

The great advantage of supplementary firing with cooled combustion chambers lies in its operating flexibility. The output of the supplementary firing can be varied within a broad load range and the maximum temperature is no longer restricted to 750 °C (1382 °F). This system thus is particularly ideal for cogeneration whenever a broad control range is required for the process steam flow at various gas turbine loads.

Fig. 5-10 shows another system with 2-stage firing with which the output of the supplementary firing of waste heat boilers with uncooled combustion chambers can be increased. Here the flue gas is heated after the gas turbine in a first stage to a temperature not to exceed 750 °C (1382 °F). This is followed by a cooling in a first heat exchanger (e.g., evaporator or superheater). The exhaust gases can then be reheated in a second supplementary firing before they flow through the final section of the waste heat boiler.

5.2.3 Steam Generator with Maximum Supplementary Firing

With this type of steam generator, the exhaust gases from the gas turbine are used primarily as oxygen carriers. The heat content of the exhausts is small in comparison to the output of the firing in the boiler. It is therefore no longer correct to speak of a waste heat boiler.

The design of a steam generator of this type is practically identical to that of a conventional boiler, except that there is no regenerative air preheater. The gas turbine exhausts are already at a temperature of 480 to 550 °C (896 to 1012 °F), which renders a further heating superfluous. Therefore, in order to make it possible to cool the exhaust gas after the steam generator to a normal temperature, an additional economizer is provided which takes over a portion of the feedwater preheating from the regenerative preheating. The best arrangement divides the feedwater between the economizer and the high pressure feed heaters. When the fuel is gas, an additional low pressure partflow economizer improves efficiency. The fuel burned in the boiler may be oil, gas, or pulverized coal. Fig. 5-11 shows an example. For more detailed information, see Ref. [102].

5.3 Steam Turbine

The steam turbine used for a modern combined-cycle installation is a simple machine with relatively low live steam data. It must possess the following main characteristics:

- high efficiency
- short start-up times

Short start-up times are of particular importance because the combined-cycle plants are often used as medium-load units with daily start-up and shut-downs. These features are required above all of installations without supplementary firing. With a fired boiler, the problems that arise are similar to those in conventional steam power plants.

Fig. 5-12 shows a steam turbine used for a combined-cycle installation without supplementary firing. This is a single-cylinder 95 MW turbine with double exhaust section. Because the turbine is run only in sliding pressure operation, no control stage has been provided. There are likewise no extraction points because all preheating of the feedwater takes place in the waste heat boiler. When designing such a steam turbine, one must remember that the live steam temperature is lower during partload operation. It order to maintain an approximately constant

Figure 5-10

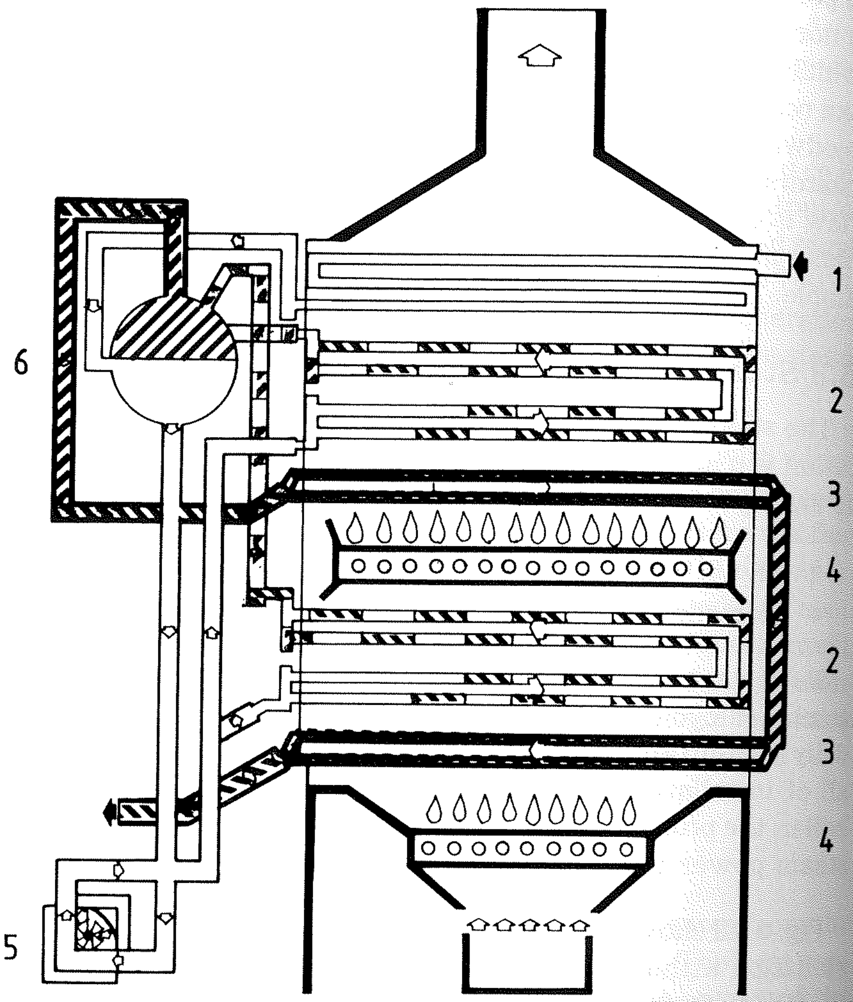

Fig. 5-10: Principle of the Waste Heat Boiler with 2-Stage Supplementary Firing

1 Economizer
2 Evaporator
3 Superheater
4 Supplementary Firing
5 Circulating pump
6 Drum

Figure 5-11

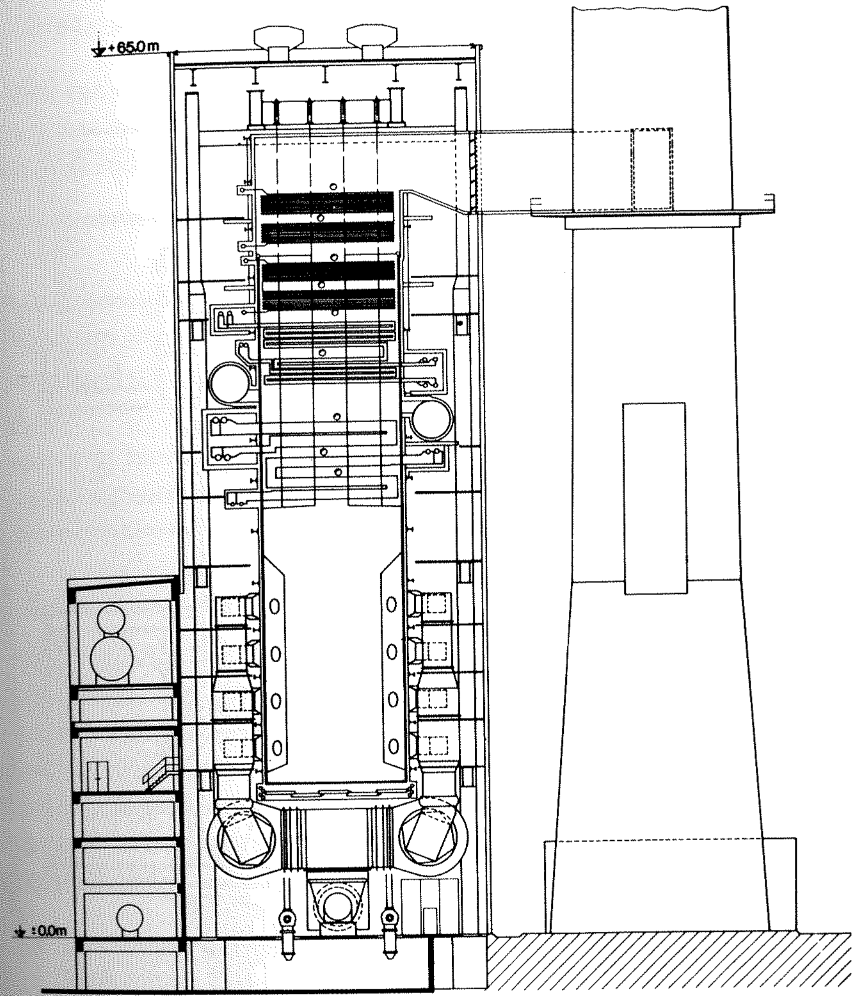

Fig. 5-11: Typical Steam Generator with Maximum Supplementary Firing

Figure 5-12

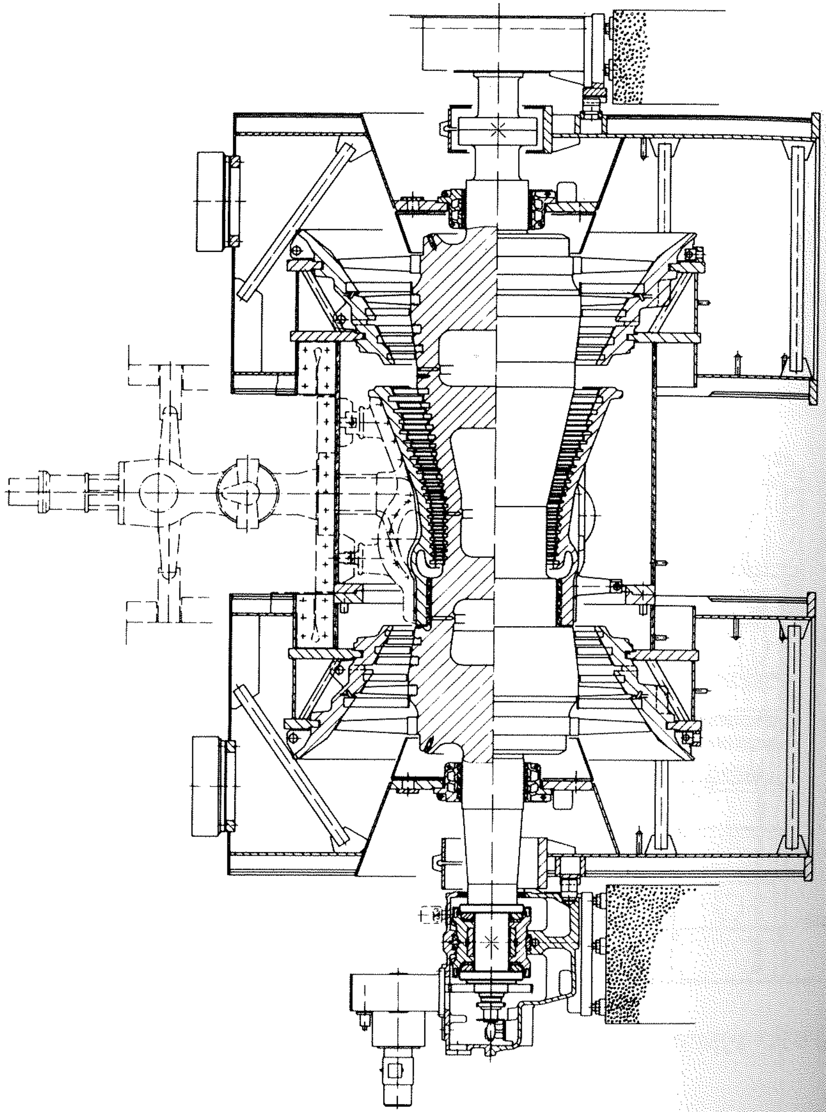

Fig. 5-12: Typical Steam Turbine used in a Combined-Cycle Plant

wetness at the end of the turbine, it is best to reduce the live steam pressure parallel to the load. Sliding pressure operation is ideally suited to this purpose.

The major features of a steam turbine for a combined-cycle plant are compared below with those of a turbine for a conventional steam plant:

	Combined-Cycle	Conventional Steam Plant
Live steam pressure, bar	30 - 80	140 - 250
Live steam temperature, °C	450 - 520	520 - 540
Reheat	No	Yes
Number of bleed points	0 - 1	6 - 8
Specific exhaust steam flow, kg/MWs	1	0.6 - 0.7

5.4 Generators

The majority of the gas or steam turbines used in combined-cycle plants are coupled directly to turbogenerators (2-pole generators). For units with ratings less than 20 MW, however, high speed turbines are used that require a reduction gear. In that case, 4-pole generators are more economical. There would be three types of turbogenerators that could be considered for combined-cycle plants:

- air-cooled generators with an open-circuit cooling system
- air-cooled generators with an closed-circuit cooling system
- hydrogen-cooled generators

Generators with open-circuit air cooling are best with regard to costs and cooling requirements, but problems can arise with regard to fouling and noise. Generator with closed-circuit air cooling are being built today for capacities up to appox. 200 MVA. These machines are reasonable in cost and problem-free in operation. The full-load efficiency of modern air-cooled turbogenerators is quite high. Hydrogen-cooled generators do, in fact, attain even higher efficiencies (particularly at part loads) than air-cooled machines. However, they require additional auxiliaries and monitoring equipment, are more complicated in design, and, as a result, more expensive.

Fig. 5-13 shows a generator with closed-circuit air cooling. Water is used to cool the air and can, in turn, be cooled back down with air. These machines are well-suited for use with gas or with steam turbines with power capacities of up to 200 MVA. For higher outputs, or if a very high part-load efficiency is required, hydrogen cooling must be selected. [100]

5.5 Electrical Equipment

The single-line diagram of a combined-cycle power plant is similar to that of a conventional power station containing several turbines. Fig. 5-14 shows an example in which the combined-cycle plant consists of two gas turbines and one steam turbine.

The power required for station service can be taken either from each of the gas turbines or the steam turbine separately, or the gas turbines can, by themselves, also supply the power for the auxiliaries of the steam portion. In combined-cycle plants with several gas turbines, each group of two gas turbines can be connected jointly to a three-winding transformer, producing savings, particularly for the high voltage switchgear. Combined-cycle plants with an output less than approx. 100 MW generally do not have a medium-voltage. All their auxiliaries are equipped with low voltage motors.

5.6 Control Equipment

The control equipment is the nervous system of a power plant. Its tasks are the control and protection of the installation, and

Figure 5-13

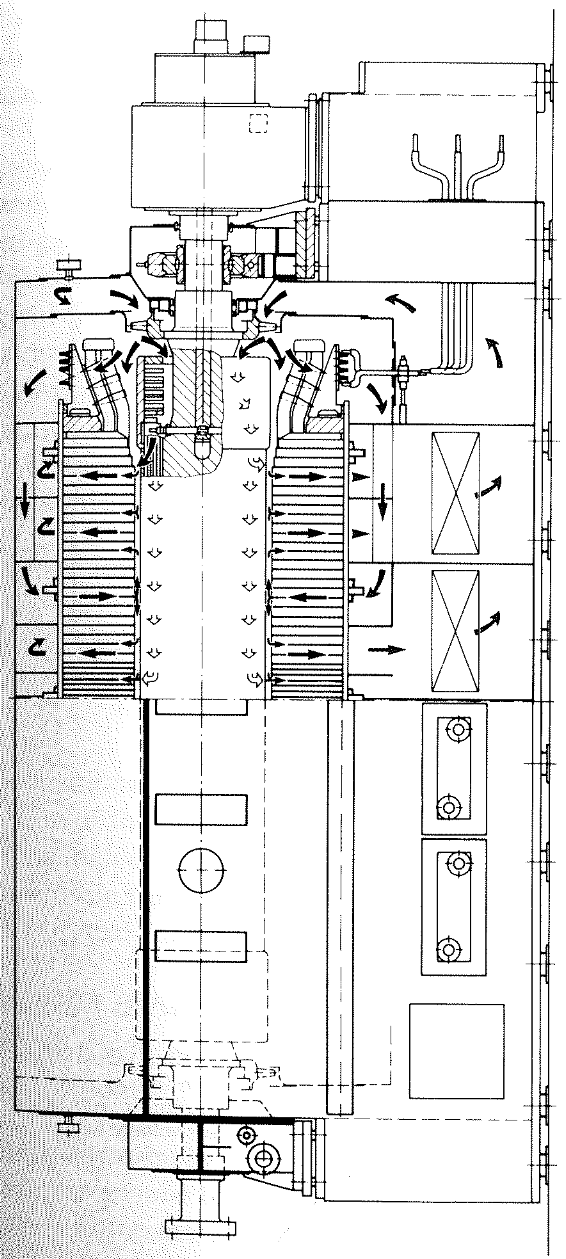

Fig. 5-13: Typical Air-Cooled Turbogenerator with Closed Cooling System

Figure 5-14

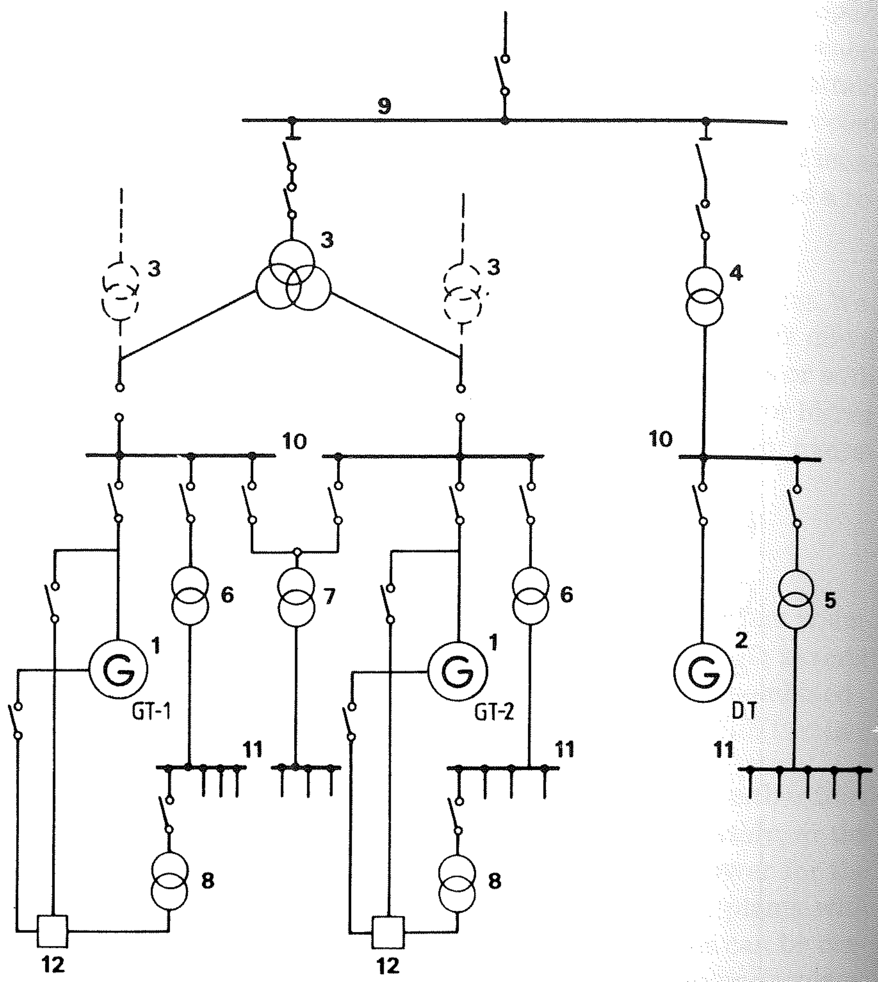

Fig. 5-14: Typical Single-Line Diagram for a Combined-Cycle Installation

1 Gas turbine generator
2 Steam turbine generator
3 or 3' Gas turbine block transformer(s)
4 Steam turbine block transformer
5, 6, 7 Station service transformers
8 Starting transformers
9 High voltage bus
10 Medium voltage bus
11 Low voltage bus
12 Static starting device

data acquisition. It must provide assurance of safe and reliable operation. Because the standard gas turbine is supplied as a fully automated machine, the steam process should likewise be correspondingly automated so as to achieve a certain degree of standardization in operation of the plant as a whole, thereby reducing the risk of operator error. For this reason, the control and automation systems of a combined-cycle plant form a relatively complex system even though the process itself is fairly simple. Modern combined-cycle plants generally have electronic control systems. A hierarchic and decentralized structure for the open and closed-loop control systems is best adapted to the logic of the whole process. It simplifies planning, makes it possible to commission the plant in stages, and raises the availability of the power station. Using the open-loop controls as an example, Fig. 5-15 shows the structure of an hierarchic system of this type. A highly automated open-loop control system encompasses three hierarchic levels:

On the drive level, all individual drives are controlled and monitored. The safety devices in the switchgear act directly on the switches and the relays. These signals are sent both to the logical control circuits of the drive level and to the higher hierarchic levels.

On the functional group level, the individual drives for one complete portion of the process are gathered together into functional groups. The logical control circuit on this level encompass interlocks, automatic throw switches and preselection of drives. Examples of typical functional groups are:

- feedwater pumps
- cooling water pumps
- lube oil pumps

The block level includes the logical control circuits that link the functional groups to one another. These include, for example, the fully automatic starting equipment for gas and steam turbines or overriding logical controls that coordinate the operation of the gas turbines, boilers, and steam turbine in fully automated installations.

As in conventional plants, process computers are today playing an important role in combined-cycle installations, since they can supply operators with valuable information needed to assure optimum operation and maintenance. Some of the tasks taken over by process computers here are sequencing events, keeping long-term records and statistics, optimization of the heat rate and of the intervals between cleanings and overhauls, etc. [101]

5.7 Other Components

In addition to the major components mentioned above, the combined-cycle power plant also includes much more apparatus and many other systems similar to those in conventional steam power stations. For example:

- condenser
- cooling system
- feedwater tank / deaerator
- feed pumps
- condensate pumps
- piping and fittings
- condenser ejector system
- water treatment plant
- compressed air supply
- flue gas bypass
- steam turbine bypass

Three different cooling systems can be considered for combined-cycle plants:

- direct cooling using river or seawater
- indirect cooling with a wet cooling tower
- direct air cooling in an air condenser

For the last of these three, no cooling water is required. Indirect air cooling requires additional water to replace evaporation and drain losses. The amount needed is approximately equal to that of the condensate. The first variant with direct water cooling requires water in an amount approximately 40 to 50 times greater than that of the condensate.

If the combined-cycle plants are equipped with flue gas bypasses for single-cycle operation of the gas turbine(s), the design and the controls for the flue gas damper(s) play a special role. This component meet the following requirements:

- It must provide a tight seal both to the boiler and to the bypass stack.
- During start-up or shut-down, it must take over the function of the firing in a conventional boiler.
- Its availability must be high. If it malfunctions, the gas turbine and the boiler go out of service.
- If two separate dampers are provided for the bypass stack and the boiler, they must be interlocked with one another so they can never both be closed simultaneously.

Fig. 5-16 shows a flue gas damper with only one flap valve. This can be used to shut off either the boiler or the bypass stack. The advantage is that both paths can never be closed at the same time.

Figure 5-15

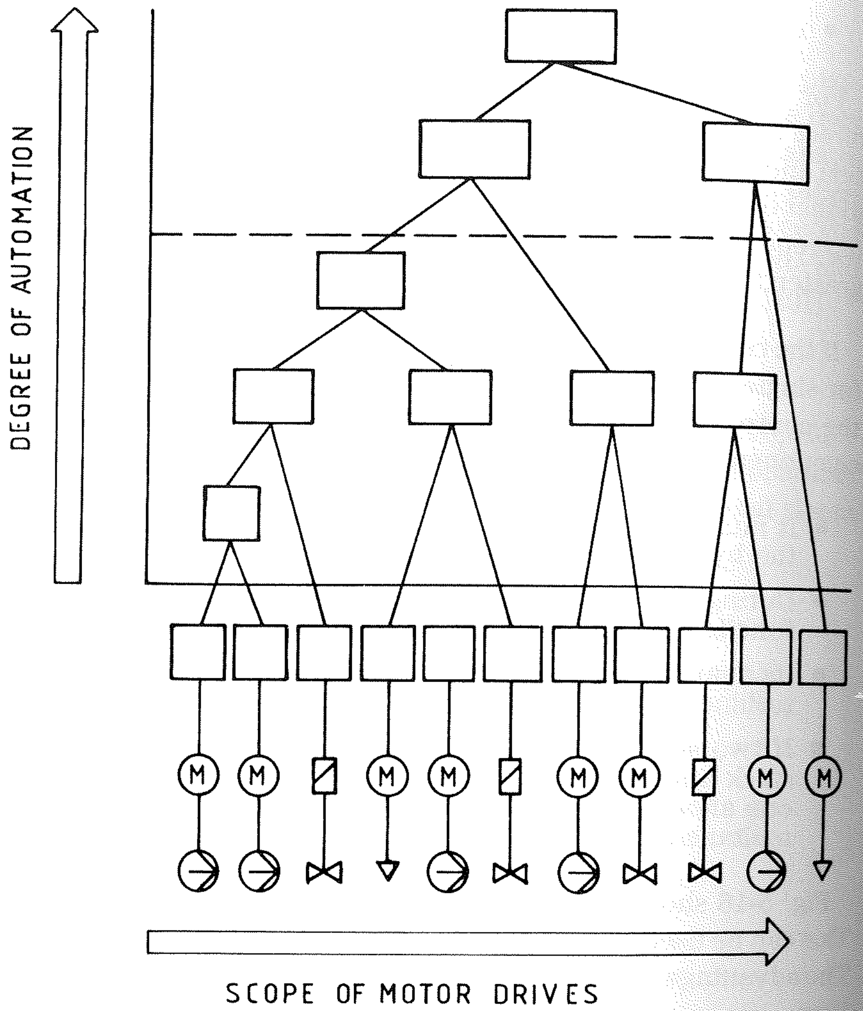

Fig. 5-15: Hierarchic Structure of the Control and Automation Equipment

Figure 5-16

Fig. 5-16: Typical Flue Gas Damper

Chapter 6

CONTROL AND AUTOMATION

Every power plant must cover the demand of an electrical grid either in part or completely. The equipment discussed below is used in modern combined-cycle installations not only regulate to their power output but also to assure the safe start-up and shut-down of the plants and their proper dynamic behavior.

6.1 The Concept of Closed-Loop Control and Operation

The closed-loop controls for a power station can be grouped into

- the main control circuit which adjusts the output of the plant to demand;
- the secondary control circuits which maintain the important process parameters within permissible limits. Controls for levels, temperatures, or pressure are likewise included among these.

Applied logically, this leads to the hierarchic structure of the entire control system already mentioned in Section 5.6.

6.1.1 Frequency/Load Control

In combined-cycle plants consisting of several gas turbines and one steam turbine, there are several possible way to adjust the output of the power plant:

- only the power output of the gas turbines is adjusted
- the outputs of the steam and the gas turbines are adjusted.

The first method is generally applied for plants without supplementary firing because varying the steam turbine load produces only a temporary effect. Over the longer range, the output from the steam turbine automatically adjusts to the amount of heat being supplied by the gas turbine. It is, however, sometimes suggested that a load or frequency control of the steam turbine be provided in plants without supplementary firing to handle sudden increases or decreases in load. Additional complications and the poorer efficiencies at full and part loads, and expecially the fact that the gas turbine(s) generate(s) approx. two-thirds of the total power output, argue generally for a solution without a control for the steam turbine power output.

Fig. 6-1 shows the design of the closed-loop load controls where the output of the steam turbine is not controlled. The entire steam cycle is operating with purely sliding pressure; the steam turbine inlet valves are fully open. This is the mode of operation best suited for high part-load efficiencies and low wetness at the steam turbine exhaust during low-load operation. The higher-level control is not absolutely necessary because the power output of the plant can be adjusted by changing the setpoints of the individual gas turbine controls.

The output of the gas turbine is controlled by changing the amount of fuel supplied. In the upper load range, it can sometimes also be adjusted by varying the amount of intake air. Variable guide vanes in the first stage of the compressor are usually used to accomplish this. With a system of this type, the turbine inlet temperature remains constant between 100% and approx. 80% load. Below that level, the temperature must be reduced in order to protect the last turbine stage from too high temperatures.

If supplementary firing has been provided, it can be a good idea to equip the steam turbine with a load control. The steam process then operates in a manner similar to that of a conventional steam plant.

Figure 6-1

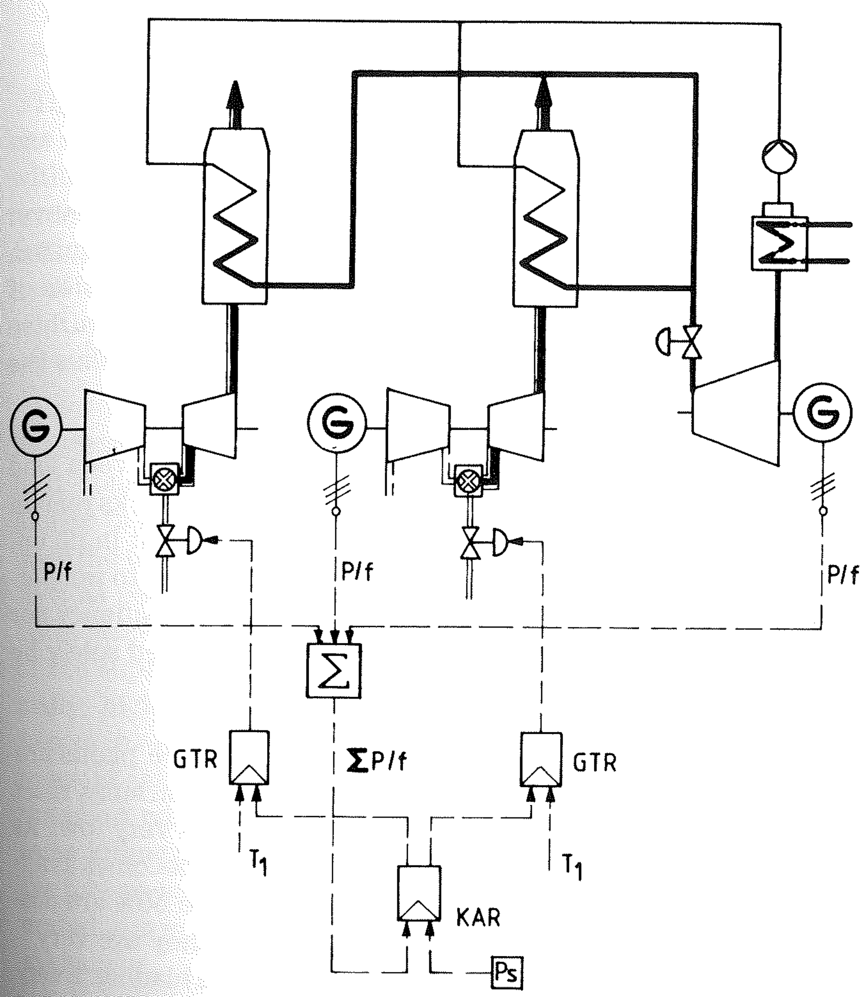

Fig. 6-1: Principle of Load Control System with no Steam Turbine Load Control
GTR Gas turbine load controls

p/f	Load/frequency
T_1	Limit for gas turbine inlet temperature
P_S	Load setpoint
KAR	Overall load control

Fig. 6-2 shows one possible design for the closed-loop controls:

- The turbine inlet valve regulates the power output of the steam turbine.
- The amount of steam generated is varied to fit demand by adjusting the supplementary firing in order to maintain a constant live steam pressure.

A control system of this type can be of advantage whenever fairly large jumps and changes in power plant load are required. The steam turbine is then capable of taking over a portion of the load surges. This affects the service life of the gas turbine positively because it reduces the changes in load and thus the changes in temperature in that turbine.

Up to this point, no distinction has been made between load and frequency control systems. In principle, the remarks remain valid for both. However, one must bear in mind that in its upper load range, the gas turbine can be switched over to temperature control. In that case, neither fluctuations in frequency nor load have any influence on the plant: it is controlled solely by the turbine inlet temperature.

In conclusion, it can be said that combined-cycle plants are very well suited to rapid load changes. The gas turbines react extremely quickly because their time constant is very low. As soon as the fuel valve opens, more added power becomes available on the shaft. Gas turbine load jumps of up to 50% are possible, but they cannot be recommended since they are very detrimental for the life expectancy of the turbine blading. Every change in load produces thermal stresses in the gas turbine because of the change in turbine inlet temperature.

6.1.2 Secondary Closed Control Loops

Fig. 6-3 shows the essential closed control loops that are required in a combined-cycle process without supplementary fir-

ing to maintain safe operating conditions. These will be described briefly below:

a. Drum level control:

This is normally a three-element control system which forms one signal from the feedwater and live steam flows and the level within the drum. This signal is used to position the feedwater control valve. Low pressure drums are often equipped only with a single-element control that uses only the level within the drum itself as a signal.

b. Live steam temperature control:

Because of the low flue gas temperatures, no control of the live steam temperature is absolutely necessary with unfired waste heat boilers. If such a control is provided, it serves more as a limiter than as an actual control. Its purpose is generally to reduce the temperature peaks under extreme operating conditions such as peaking operation. For that reason, the cooling often takes place after the superheater and not between two portions of the superheater as in a conventional steam generator. Normally, high pressure feedwater is injected into the live steam to cool the steam down to the required temperature. However, to avoid salt deposits in the turbine, it is necessary to operate the plant with fully demineralized water. This is impossible in some industrial plants, which means that the temperature of the live steam must be controlled either by mixing it with saturated steam extracted after the drum or by means of a heat exchanger.

No real temperature control extending over a broad load range is possible in purely waste-heat boiler operation because the turbine exhaust gas temperatures drop off rapidly during part-load operation of the gas turbine. In plants with supplementary firing, the relationships are more like those of a conventional steam generator. Because elevated gas temperatures are possible in

Figure 6-2

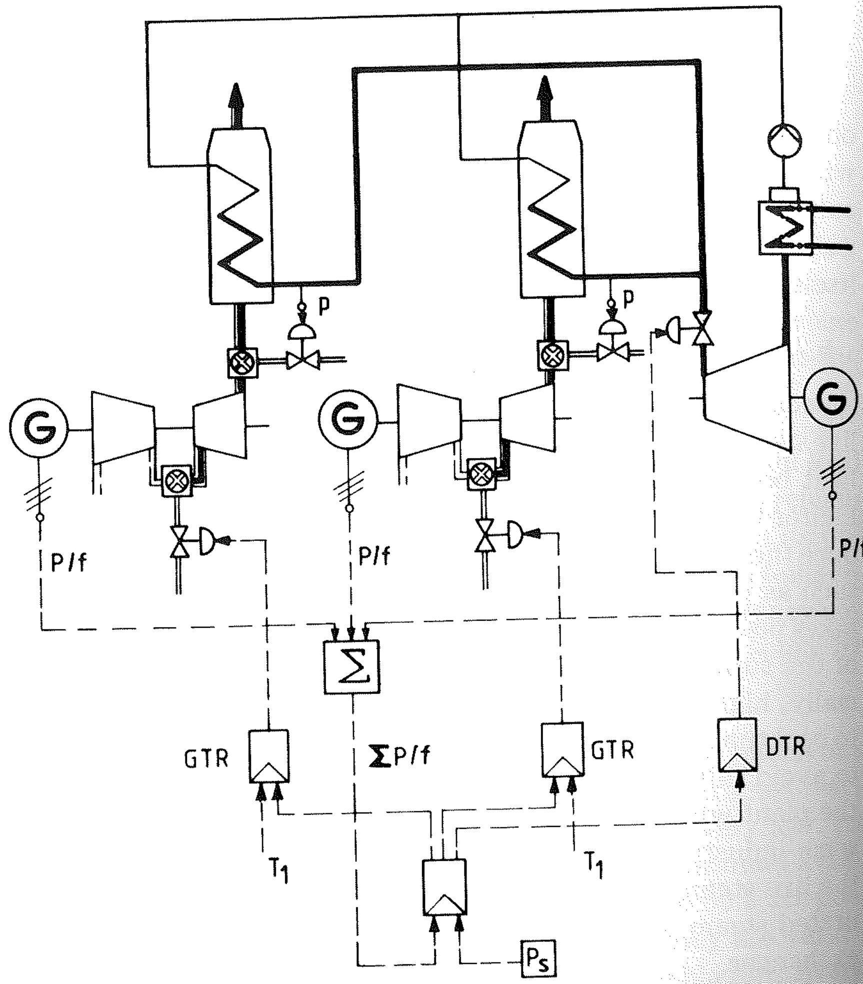

Fig. 6-2: Principle of Load Control System with Steam Turbine Load Control

p	Live steam pressure controls
GTR	Gas turbine load control
DTR	Steam turbine load control
p/f	Load/frequency
T_1	Limit for gas turbine inlet temperature
P_S	Load setpoint
KAR	Overall load control

Figure 6-3

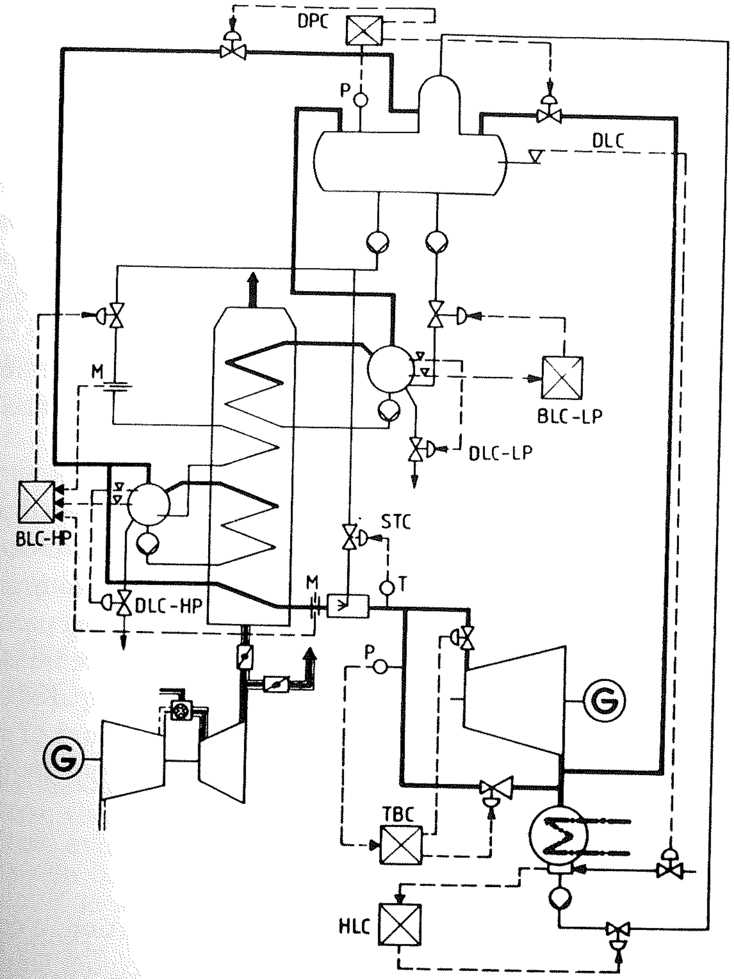

Fig. 6-3: Closed Control Loops in a Combined-Cycle Plant without Supplementary Firing

BLC-LP	Low pressure drum level control system
BLC-HP	High pressure drum level control system
DLC-LP	Low pressure emergency control drain
DLC-HP	High pressure emergency control drain
STC	Steam temperature control
DPC	Pressure control, feedwater tank
DLC	Level control, feedwater tank
HLC	Hotwell level control
TBC	Steam turbine and bypass control system

these cases, it is important that the temperatures of the steam and the superheater tubes be maintained within safe limits. To do this, the superheater is divided into two or three sections with water injected between them.

c. Feedwater temperature:

In order to keep low temperature corrosion within acceptable bounds, the feedwater temperature must not, even in the lower load range, drop significantly below the acid or water dewpoint. On the other hand, this temperature must be as low as possible in order to assure good utilization of the heat available. It is therefore recommended that the feedwater temperature be held at a more or less constant level corresponding approximately to the acid dewpoint.

Whenever steam is being extracted from a turbine bleed point, the deaerator must be heated at part loads by using reduced live steam (back-up steam). In plants with a preheater loop in the waste heat boiler, the opposite problem occurs during part-load operation of the gas turbines: more low pressure steam is generated than required. This excess energy must be dissipated either by increasing the pressure, which reduces the amount of steam generated, or by directing the excess low pressure steam to the condenser.

d. Live steam pressure:

If the steam turbine is in sliding pressure operation, a continuous control for the the live steam pressure becomes superfluous. However, such a control is necessary for non-steady-state conditions such as start-up, shut-down, or malfunction. For fixed pressure operation, on the other hand, the control must be engaged at all times.

How this control is accomplished depends on the equipment of the power plant. In principle one or several of the following components can be used to control the live steam pressure:

- steam turbine control valves
- steam turbine bypass
- starting valve
- flue gas bypass

Practically all modern combined-cycle plants are equipped with a steam turbine bypass, which can provide several advantages:

- flexible operation during start-up, shut-down, turbine trip, or quick changes in load
- shorter start-up times
- environmental acceptability (since less steam escapes).

Combined-cycle plants with repowered conventional steam turbines often have a starting duct instead of the steam turbine bypass. During start-up, this directs the steam either across the roof and to the atmosphere, or into a starting condenser.

The flue gas bypass is frequently omitted due to economical considerations, specifically for the following reasons:

- A flue gas bypass often leaks, and a certain amount of leakage must be accepted. In addition to that flow loss, there is a further loss due to radiation. The total loss, even with new dampers, may amount to between 0.5 and 2% of the energy contained in the exhaust gases.
- Flue gas dampers are sensitive equipment. They must be very well designed to assure safe and reliable operation. A cheap damper can, over the course of time, develop leaks.
- A good flue gas bypass is fairly expensive.

On the other hand, there are also many advantages in having a plant with a well-designed flue gas bypass:

- greater flexibility in operation
- increased availability, since the gas turbine can continue in operation even when the boiler is unavailable
- reduced risk of explosion within the boiler because the gas turbine can be lit off in bypass operation

Table 6-1 shows which live steam pressure control systems units are used for the various modes of operation of a combined-cycle plant with an unfired heat recovery boiler.

Table 6-1: Various Control Systems Used to Regulate Live Steam Pressure

Type of Operation	Steam Bypass	Steam Turbine Control Valves	Flue Gas Bypass*
Start-up	+	+	+
Shut-down	+	+	+
Normal operation (sliding pressure)	–	–	–
Normal operation (fixed pressure)	–	+	–
Operation at low load	–	+	–
Steam turbine switch-off or trip	+	–	+
Waste heat boiler switch-off	+	–	+
Gas turbine trip	+	–	+

* if provided; + in operation, – not in operation

Combined-cycle plants with supplementary firing are generally operated with a fixed live steam pressure. During normal operation, pressure is held constant by varying the fuel flow to the boiler. The steam turbine takes over control of the pressure only in such special cases as start-up, shut-down, supplementary firing shut-down, etc.

e. Level in feedwater tank and hotwell

These are controlled by regulating the flows of condensate and cycle make-up water. What level is assigned to which unit is not particularly important: the selection depends on the size of the two tanks.

f. Other control loops

Other control loops, such as those for lube oil and control oil pressure, etc. will not be discussed here.

6.1.3 Start-Up and Shut-Down of the Combined-Cycle Plant

Combined-cycle power plants are usually started up and shut down to a large extent automatically. It must therefore be possible to operate the controls for units to be activated during start-up and shut-down from the central control room. Whether the commands are to be issued to the individual drives or drive groups by the operating staff or from a higher-level automatic starting program must be decided on a case-by-case basis. In base-load installations that operate with only a few starts, full automation of the steam process is not an absolute necessity. The starting procedure differs depending on whether or not the plant has a flue gas bypass.

a. Plants with a flue gas bypass

If a flue gas bypass is provided, the gas turbine can be started, synchronized, and loaded practically independently of the steam process. As soon as the gas turbine is on line, the steam process can be started up. However, it is also possible to switch on the steam process while the gas turbine is in load operation. The flue gas dampers are used to adjust the heat supply to the requirements of the waste heat boiler. It may be necessary to place a restriction on the flue gas temperature during a cold start of the waste heat boiler if the superheater cannot handle the full flue gas temperature without being cooled. The steam turbine can be started up as soon as the steam data are high enough, which generally means that the steam must have reached about 40 to 60% nominal pressure and be superheated by at least 50 to 80 °C (90 to 144 °F).

The bypass stack is used for light-off of the gas turbine. If the flue gas damper closes 100% tight, no purging of the waste heat boiler is required prior to start-up since no fuel can get into the boiler even if fuel system malfunctions should occur. If the flue gas damper is not 100% tight, it is best to purge the boiler before starting the gas turbine, particularly after a malfunction or an overhaul.

Until the steam turbine takes over the full steam flow, the excess steam flows across the steam turbine bypass or the starting valve. If supplementary firing has been provided, it must not be lit until the gas turbine is at full load and the steam turbine has taken over the entire steam flow. With the supplementary firing on, the steam process can then be further loaded. The plant is shut down by shedding load from the gas turbines (or first, from the supplementary firing). Once the flue gas temperature has reached a prescribed minimum level, the steam turbine is shut down. The boiler and the gas turbine are then further unloaded and shut off. The bypass damper must be open and the boiler damper closed before shutting down the gas turbine.

b. Plants without a flue gas bypass

Here attention must be paid to the steam process when loading the gas turbine since the entire flow of flue gas passes through the boiler. Particularly during a cold start of the waste heat boiler, the gas turbine must not be loaded at full speed or the rate of temperature change in the boiler drum would exceed the maximum level permissible. A second minor problem arises in the light-off of the gas turbine, during which a high peak occurs in the flue gas temperature.

In order to prevent explosions, it can be advisable, especially with oil-fired gas turbines, to purge the boiler before lighting the gas turbine. This is done by operating the gas turbine at ignition speed for a few minutes, without igniting it. Otherwise the start-up and shut-down of combined-cycle plants with and without flue gas bypasses are very similar.

6.2 Dynamic Behavior

The dynamic behavior of modern combined-cycle plants is characterized by their short start-up times and quick load change capability.

Above all, the gas turbine can be started and loaded quickly. Because its reaction time is also short, it is capable of following quick changes and surges in load. Generally, load changes are effectuated only by adjusting the turbine inlet temperature. As a result, every major change in load reduces the life expectancy of the gas turbines more than would be the case for a steam turbine. If there are variable guide vanes at the compressor inlet, they reduce the thermal stresses in the upper load range because the load is regulated with a constant turbine inlet temperature. However, one factor negatively affecting expected service life is that this temperature is, on an average, higher than that in a system employing only throttling of the fuel supply.

The steam process with an unfired waste heat boiler has low live steam data. It is therefore quite capable of following the short start-up times of the gas turbine. However, the steam turbine and the waste heat boiler must be designed to withstand these stresses.

Modern combined-cycle plants with ratings between 100 and 500 MW can be started within the following times:

- warm start (after 8 to 14 hr. at standstill): 20 to 50 minutes
- cold start : 60 to 120 minutes

Because the gas turbines are already at full load after 10 to 30 minutes, 2/3 of the power output is already available after that time— even in a cold start.

Fig. 6-4 shows the example of a 120 MW combined-cycle plant which can be brought to full load in an little as 26 minutes after 14 hours at standstill, and that with a normal start of the gas turbine.

Because the time constant of a combined-cycle plant is quite low, the plant is well suited to handling quick changes in load. When used in grids that often call for larger changes in load, however, special precautions must be taken because the steam turbine cannot make any sudden jumps in load during sliding pressure operation. The entire surge must therefore be accommodated by the gas turbine, and that can, over the long range, mean severe stressing.

Operating the steam turbine at a fixed pressure improves the situation somewhat. Equipping the waste heat boiler with supplementary firing to increase the portion of the surge that can be taken up by the steam turbine is another possibility. However, since it increases the share of the share portion, which reacts more slowly, it does not make the plant more flexible.

Figure 6-4

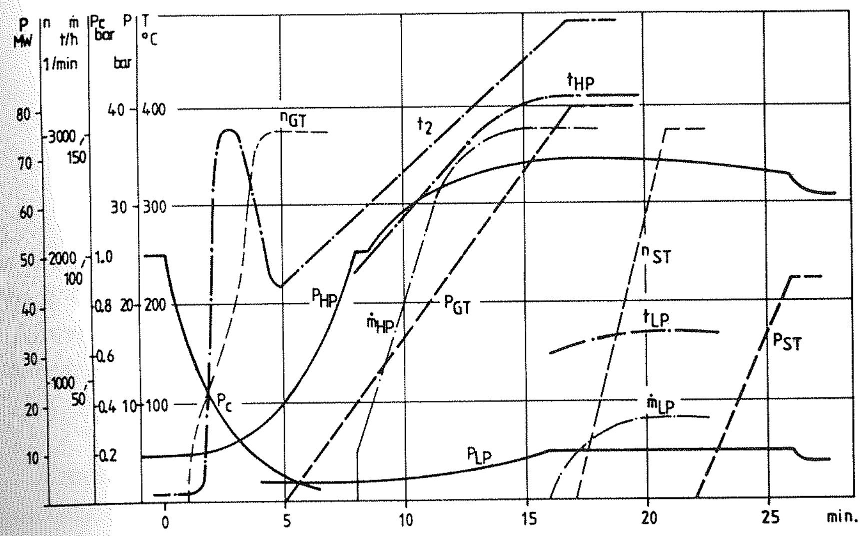

Fig. 6-4: Warm Start of a 120 MW Combined-Cycle Plant

P_{GT}	Gas turbine output
n_{GT}	Gas turbine output
t_2	Gas turbine exhaust temperature
P_{ST}	Steam turbine output
n_{ST}	Gas turbine speed
P_{HP}	High pressure live steam pressure
t_{HP}	High pressure live steam temperature
m_{HP}	High pressure live steam flow
P_{NP}	Low pressure live steam pressure
t_{NP}	Low pressure live steam temperature
m_{NP}	Low pressure live steam flow
P_C	Condenser pressure

Chapter 7

OPERATING AND PART-LOAD BEHAVIOR

The way in which a power plant responds to changes in its outside conditions (ambient conditions, part load) is of great importance both for its economy and its safe operation. It is therefore important to have precise knowledge of both the steady-state and the dynamic operating behavior of the plant.

Theoretical calculation of the dynamic behavior is costly and difficult. For that reason, one frequently limits himself to operating experience in other similar plants or to estimates. A more exact calculation of the behavior would certainly be advantageous but it is usually omitted due to considerations of time and cost. In all cases, however, the calculations of steady-state operating and part-load behavior should be completed.

7.1 Basis Used for Calculations

The calculation of the steady-state part-load and operating behavior of the steam portion in a combined-cycle plant differs significantly from that for a conventional steam plant. The differences involve mainly the boiler and the operating mode of the plant. In a waste heat boiler, the heat is transfered mainly by means of convection, while in a conventional boiler it takes place due to radiation.

The steam turbine of a combined-cycle plant functions most economically using the sliding pressure - sliding temperature process, i.e., it is run "uncontrolled." The steam data are determined only by the exhaust flow and exhaust temperature of the gas turbine and by the swallowing capacity of the steam turbine. In contrast, a conventional plant is generally operated at a fixed pressure, i.e., the live steam pressure and temperature remain constant. That simplifies calculations because the steam pressure and the steam temperature are known in advance. The steam turbine and the boiler can therefore be considered independent of one another. Calculations for the gas turbine are no problem since one is dealing with standard machines for which correction curves are available to account for changes in ambient conditions and for part-load operation.

Statement of the Problem

Calculating the operating behavior of an installation directly from the geometry of that unit would be very time-consuming. The process can, however, be simplified by referring all values to the thermodynamic data at the design point. If that design point is known, general equations (the Law of Cones, heat transfer law, etc.) can be used to reduce the calculation problem to a reasonable number of equations, without the necessity of considering the dimensions of the unit itself. A brief study of the details of the method for calculation described in Ref. [103] can be found in the Appendix to this Chapter.

7.2 Part-Load Behavior

A careful economic evaluation must also give consideration to the part-load behavior of a power plant. Power plants must also have as high an efficiency as possible at part-loads. In modern combined-cycle plants without supplementary firing, the efficiency of the plant as a whole depends mainly on the efficiency of the gas turbine.

Fig. 7-1 shows the efficiency curves for a gas turbine and a combined-cycle plant, based on the full-load efficiency of the gas turbine. The similarity of the paths of these two curves is

striking, but the efficiency of the combined cycle is approximately 50% higher at every point than that of the gas turbine. In other words: the ratio between the outputs of the gas turbine and the steam turbine remains approximately constant across the entire range of loads. Fig. 7-2 shows the curves for power outputs and live steam data.

These calculations have been based on purely sliding pressure operation above 50% load. Below that point, the live steam pressure is held constant by means of the steam turbine inlet valves. One striking point shown in Fig. 7-1 is the quick deterioration in efficiency at part loads. This is due to load control of the gas turbine (by changing the turbine inlet temperature). At part-load with single-shaft machines, the flow of intake air remains practically constant. When the load is reduced, the turbine inlet temperature drops. This also lowers the average temperature of the heat being supplied and thus the efficiency. Improvements could be attained by:

- reducing the amount of air taken in during part-load operation
- installing several gas turbines

The air flow can be reduced by:

- installing variable inlet guide vanes in the compressor
- preheating the intake air
- using two-shaft machines

Two-shaft machines are available only for low ratings: they suffer from the disadvantages cited in Section 5.1. Compressors with variable inlet guide vanes are offered for use in combined-cycle plants by several suppliers of gas turbines. Using this method, the power output can be rduced to approx. 80 to 85% without a change in the turbine inlet temperature. Below that level, the inlet temperature must be reduced to avoid overheating in the last turbine stage.

Figure 7-1

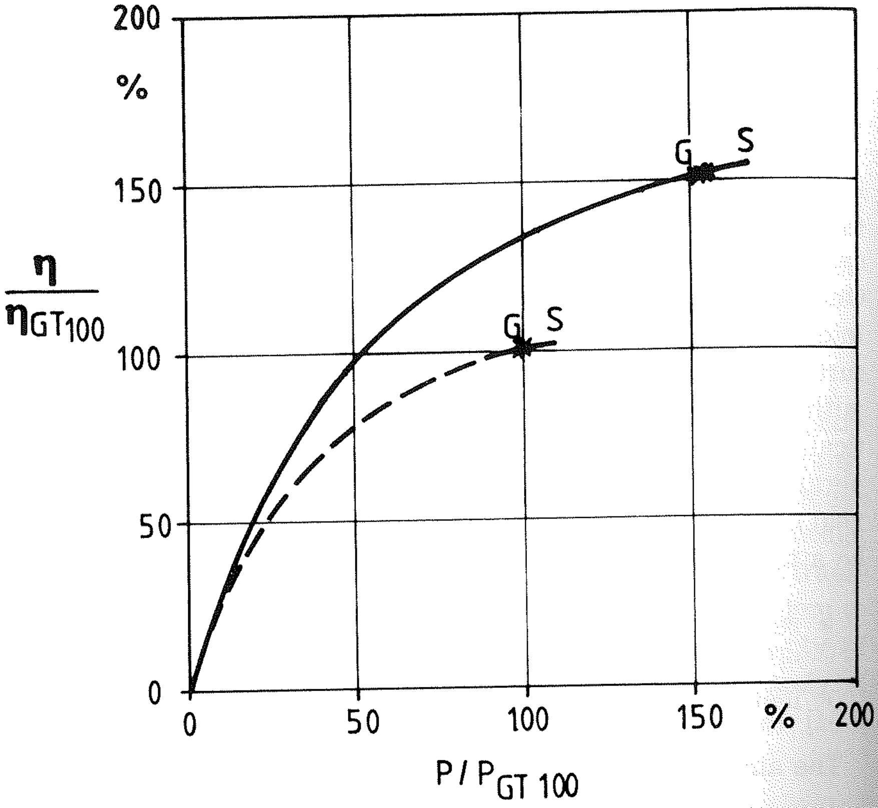

Fig. 7-1: Part-load Efficiency of the Gas Turbine and the Combined-Cycle Plant

——	Combined-cycle plant
- - - -	Gas turbine
G	Base load
S	Peak load
P/P_{GT100}	Relative power output
η/η_{GT100}	Relative efficiency
*	Reference 100% load of gas turbine

Figure 7-2

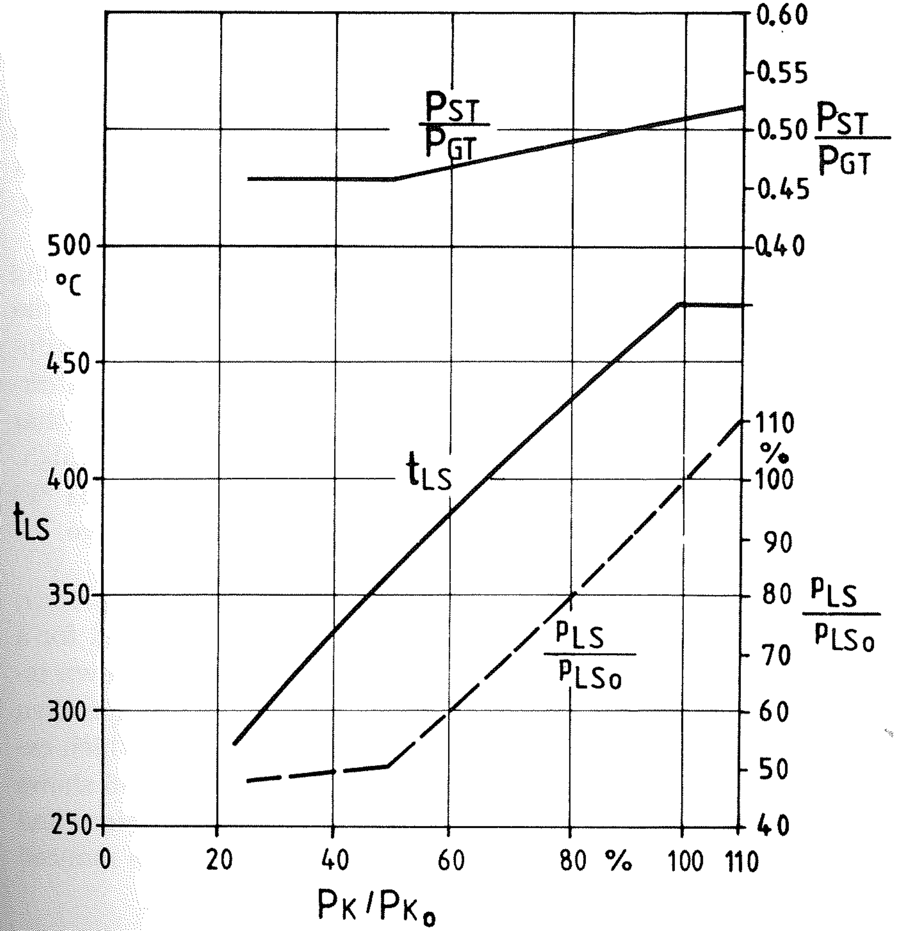

Fig. 7-2: Power Outputs and Live Steam Data of a Combined-Cycle Plant at Part Load

P_K/P_{KO}	Relative power output of the combined-cycle plant
P_{ST}/P_{GT}	Ratio of steam turbine output to gas turbine output
P_{LS}/P_{LSO}	Relative live steam pressure
t_{LS}	Live steam temperature

Fig. 7-3 shows the part-load efficiency of a combined-cycle plant with variable inlet guide vanes. There is a perceptible improvement over the entire load range as compared to turbines employing temperature control alone. The advantage of preheating the intake air is that there are no particular difficulties in doing it. This increases the efficiency of a combined-cycle plant (refer to Section 7.3), but, of course, a source of heat is required.

Fig. 7-4 shows an example where the air is preheated using low pressure steam from a waste heat boiler. Fig. 7-5 shows the efficiency of a plant employing intake air preheating. In the upper load range, the efficiency here is even better than with variable inlet guide vanes, but this system has the disadvantage that its effectiveness drops off when the ambient temerature is high. The air can only be heated to approx. 50 - 55 °C (122 - 131 °F) without exceeding the limit imposed by compressor surge.

To make an even greater improvement in part-load efficiency, a design employing several gas turbines should be used for a combined-cycle plant. This makes it possible to shut down individual gas turbines at part loads. The other gas turbines then run at a higher load and a higher inlet temperature, which exerts a positive effect on the overall efficiency. Fig. 7-6 shows the part-load efficiency of a plant with four gas turbines and one steam turbine. The load of the plant as a whole is reduced as follows:

- down to 75%, there is a parallel reduction in load on all four gas turbines
- at 75% one gas turbine is shut down
- down to 50%, there is a parallel reduction in load on the three remaining gas turbines
- at 50%, a second gas turbine is shut down
- etc.

Figure 7-3

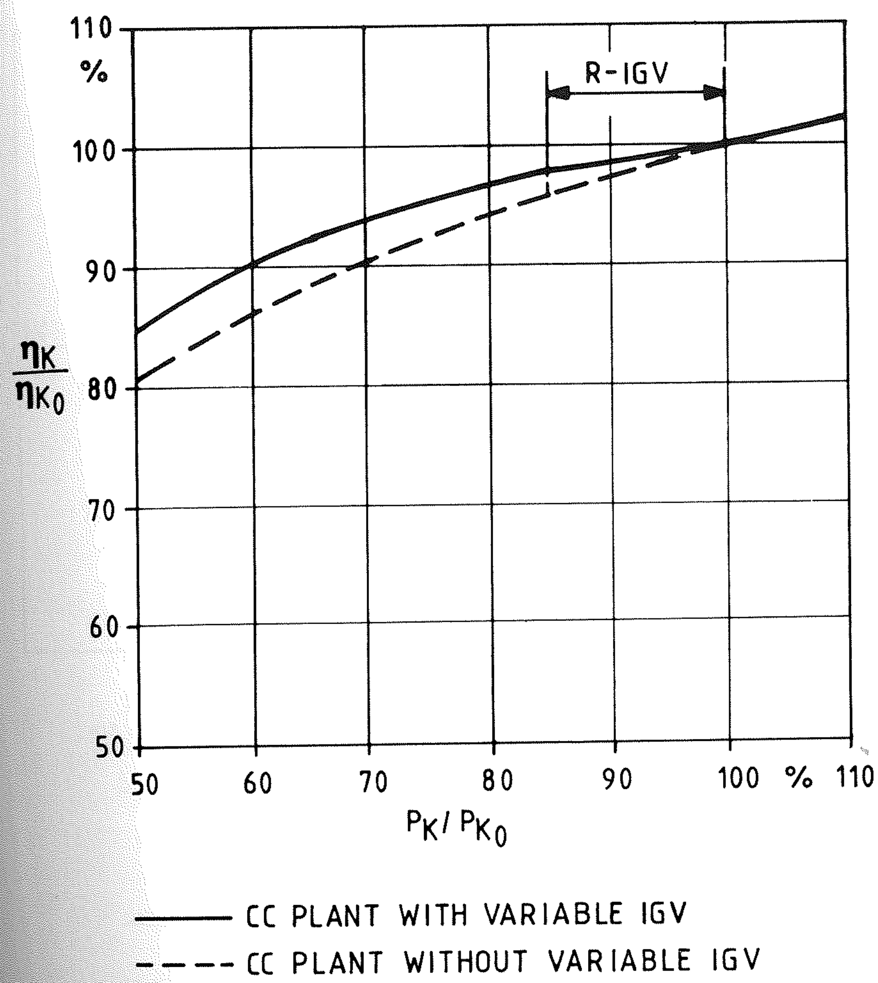

Fig. 7-3: Part-Load Efficiency of a Combined-Cycle Plant with Variable Inlet Guide Vanes in the Compressor

η_K/η_{KO}	Relative efficiency of the combined-cycle plant
P_K/P_{KO}	Relative power output of the combined-cycle plant
IGV	Variable inlet guide vales
R-IGV	Range of the variable inlet guide vane control

Figure 7-4

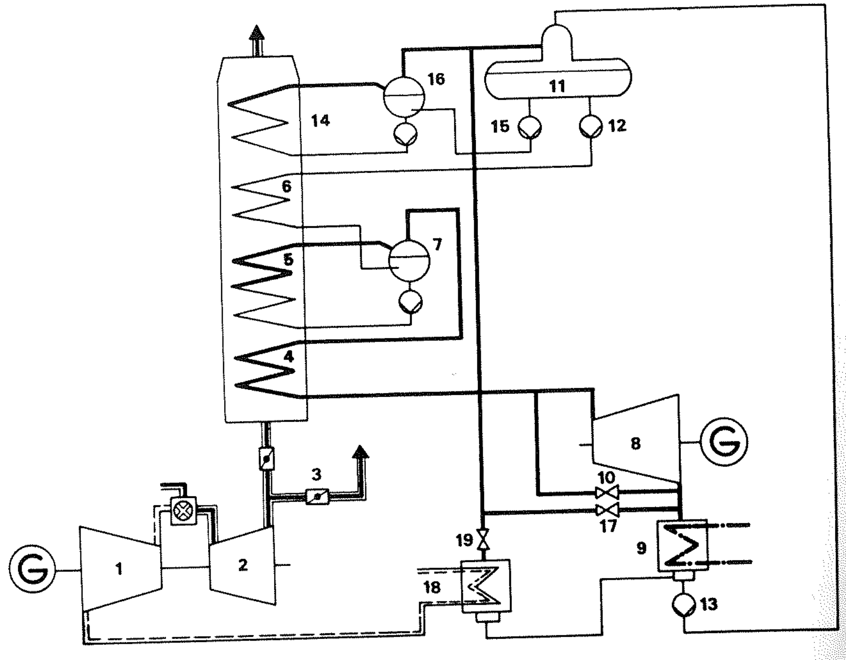

Fig. 7-4: Combined-cycle Plant with Preheating of Gas Turbine Inlet Air

1-17:	refer to Fig. 29
18	preheater
19	Control valve for air preheater

Figure 7-5

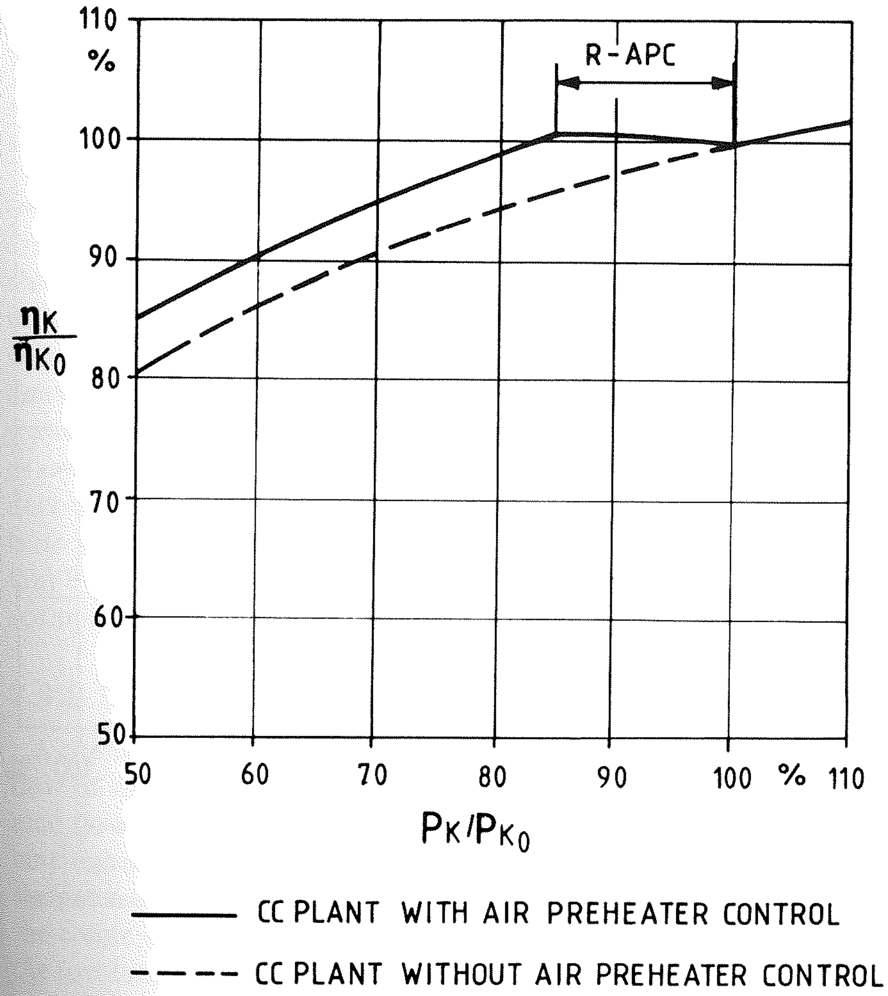

Fig. 7-5: Part-Load Efficiency of a Combined-Cycle Plant with Preheating of Gas Turbine Inlet Air

η_K/η_{KO}	Relative efficiency of the combined-cycle plant
P_K/P_{KO}	Relative power output of the combined-cycle plant
R-APC	Range of the gas turbine air preheater control

Figure 7-6

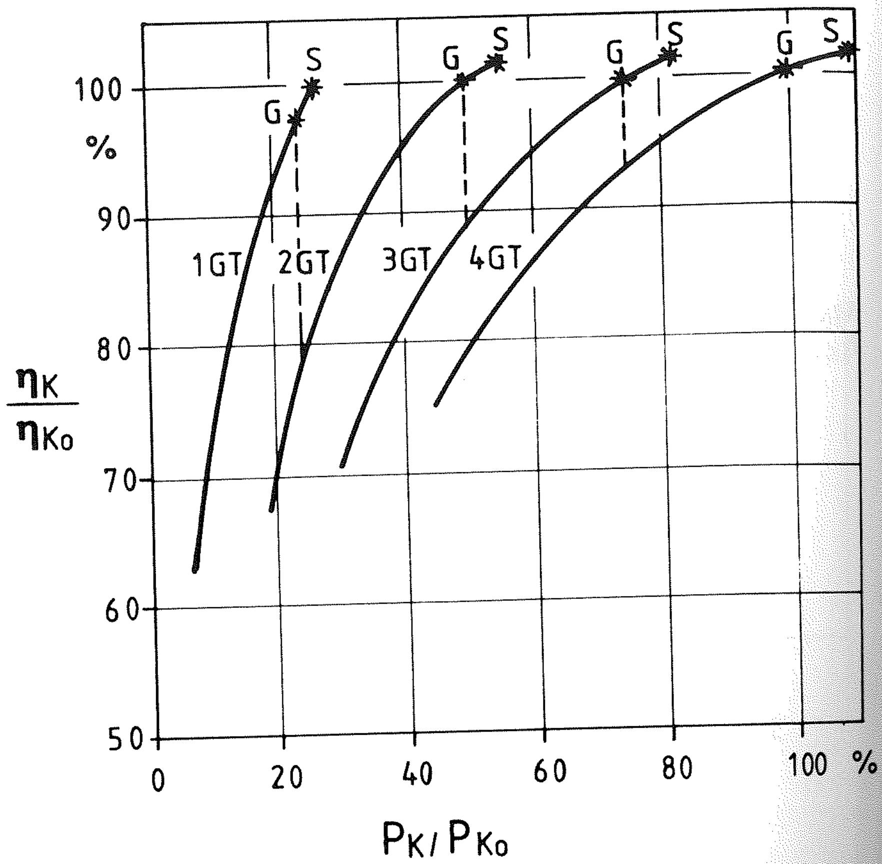

Fig. 7-6: Part-load Efficiency of a Combined-Cycle Plant with Four Gas Turbines

P_K/P_{KO}	Relative power output of the combined-cycle plant
η_K/η_{KO}	Relative efficiency of the combined-cycle plant
1...4 GT	Number of gas turbines in operation
G	Base load
S	Peak load

With this mode of operation, the efficiency at 75%, 50%, and 25% load is practically as high as that at full load.

7.3 The Effect of Ambient Temperatures

7.3.1 Effect of the Air Temperature

At the design point, the air temperature exerts a great influence over the power output of the gas turbine and the combined-cycle plant. The behavior of the plant designed responds to changes in the air temperature in a way similar to that described in Section 2. In particular, efficiency increases slightly here, too, if the air temperature rises while the vacuum within the condensor remains constant (refer to Fig. 3-9). When, instead of river water with a constant temperature, air is used either directly or indirectly to condense the steam, the efficiency of the cycle decreases as the temperature rises. In that case, the behavior is no longer comparable to that indicated by the curves in Section 2, since the steam turbine designed for this case has been optimized for a given condenser pressure.

7.3.2 Cooling Water Temperature

A change in the cooling water temperature affects the volume flow of exhaust steam. Quite quickly, the exhaust steam volume flow no longer is correctly adapted to the size of the turbine exhaust. This either increases exit losses when the temperature falls or, if the temperatures are higher, increases the condenser pressure, thereby reducing the enthalpy drop in the turbine. The operating behavior is thus quite different from that described in Section 2, in which the size of the turbine is in all cases adapted to the temperature of the cooling water. Fig. 7-7 shows the effect of the cooling water temperature on the efficiency and the power output of a steam process.

Figure 7-7

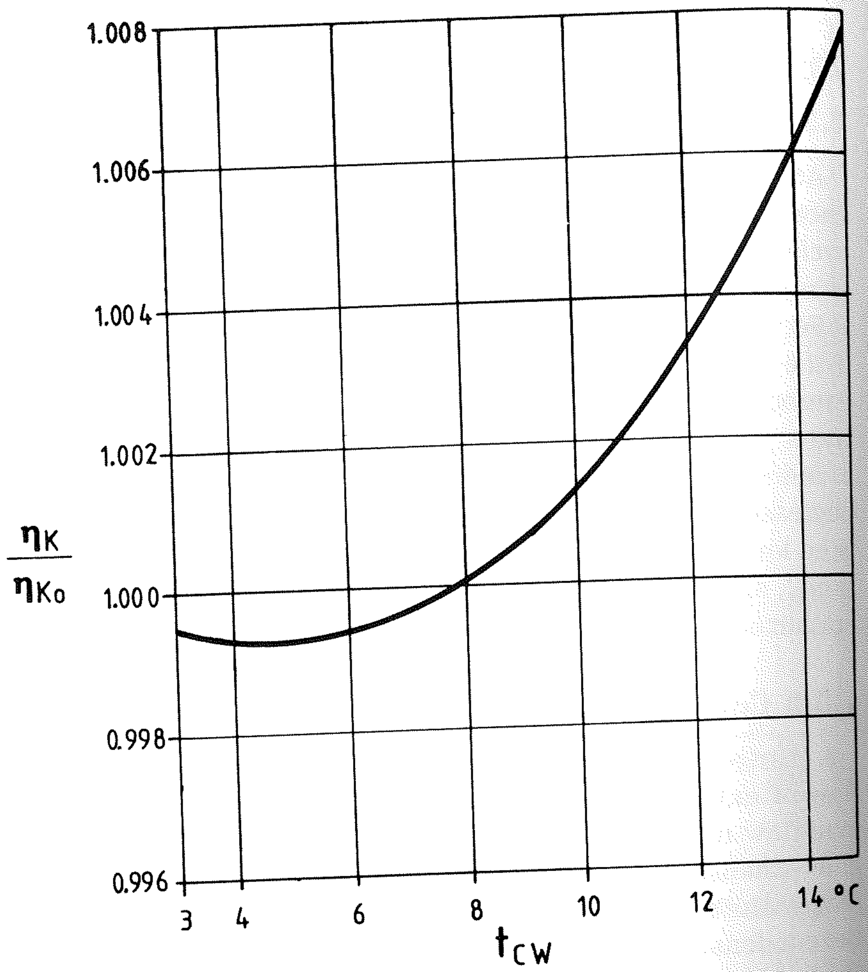

Fig. 7-7: Effect of Cooling Water Temperature on the Efficiency of a Combined-cycle Plant (Operating Performance)

tCW Cooling water temperature

7.4 Aceptance Tests and Commissioning

7.4.1 Acceptance Tests

Acceptance tests for a combined-cycle plant are always a special problem because the gas turbine, waste heat boiler, and steam turbine all interact upon one another. It is therefore best to award contracts for combined-cycle plants to a single general contractor who will assume responsibility for supplying the entire plant on a turnkey or a semi-turnkey basis and who guarantees the power output and efficiency of the plant as whole. For combined-cycle plants, it is easier in any event to measure the values guaranteed for overall plant performance than those for each major component individually. The amount of waste heat supplied to the waste heat boiler by the gas turbine, in particular, cannot be measured accurately. When overall values are guaranteed, the fuel flow, electrical output, and ambient conditions of the power plant must be measured. These are quantities which can be determined with relative exactness.

Correction Curves

The technical values guaranteed for a power plant or one of its major components are valid only if all ambient or design conditions are as defined. Suitable correction curves must be used to allow for any deviations. Thus, for example, the effect that the air temperature has on the power output and efficiency of a gas turbine must be measured and — if need be— correction must be made.

For conventional power stations and for gas turbines, the methods used for corrections are described in the standards (e.g., ASME, ISO, DIN, etc.). No standards have yet been established for combined-cycle plants. We will therefore indicate below one possible solution to this problem that has proven useful.

The basic rule is that a limit must be defined around the plant within which all components and systems supplied by a given contractor are included, and only those components and systems. All others for which he is not responsible are excluded. The guaranteed values must clearly define the ambient or design conditions for which they are valid, i.e., the conditions as measured at the limit defining his responsibility. In the case of a combined-cycle plant used for power generation only and supplied on a turnkey basis, such values and the marginal conditions to be defined might be as follows:

a. Guaranteed values

- Overall power output of the combined-cycle plant
- efficiency

b. Design or Ambient Conditions for the Guarantees

- Air temperature
- Air pressure
- Relative humidity
- Cooling water temperature, and flow (if cooling water pumps are supplied by another party)
- Frequency
- Power factor of the generators
- Voltage

c. Comparison between Measured and Guaranteed Values

Two different methods can be used for doing this. In the first, the values measured are compared with those guaranteed as follows:

- Correction of the guaranteed values to the ambient conditions at the time of measurement
- Comparison of the data measured with these corrected guarantees

In this way, one takes into account the fact that the model used for calculations is, strictly speaking, valid only for the theoretical (guaranteed) installation, not for the installation as it actually has been built. This method is especially suitable if computer programs are used for making the corrections, since they can best calculate the behavior of the theoretical plant.

The standards, however, recommend a second, reverse procedure. The values measured are corrected to the design conditions of the guarantee, since in many cases, guarantees must be given for several load points. In a contract, it is best to make allowance for this fact by using a guaranteed weighted average value.

The same marginal conditions are generally used for all load points but one can scarcely assume that the ambient conditions that prevail will actually remain unchanged while all load points are being measured. The theoretically correct procedure thus leads both mathematically and contractually to unsolvable problems in calculating the average value measured. On the other hand, when the procedure indicated in the standards is used, all measured points are corrected to warranty conditions. There is thus no problem in comparing guarantees with measurements.

The theoretical arguments favoring the first of these procedures can be countered by recognizing that the ambient conditions must affect the theoretical and the actual plants in similar ways. The procedure in the standards is therefore not incorrect.

Deviations in ambient conditions must remain within bounds during the measurements. This is another reason why the correction curves are valid both for the theoretical and for the actual plant, and will not cause significant error.

Correction of the Measured Power Output of a Combined-Cycle Plant

It has proven to be a good idea to correct the power outputs of the gas turbine and the steam turbine separately. For the gas turbine, the usual correction curves are used to take into account the effects produced by air temperature, air pressure, rotational speed, etc.

The power output measured for the steam turbine is corrected using curves that show the indirect effects of air temperature, air pressure, and gas turbine speed on the steam process and the direct effect of the cooling water temperature. To calculate these curves, it is best to use a computer model which simulates the steam process as a whole (refer to Section 7.1). Changes in the data of the ambient air produce changes in the gas turbine exhaust data and these latter affect the power output of the steam turbine.

The advantage of this procedure is that it can, with certain adaptations, be used even if the gas turbine is put into operation at a somewhat earlier date than the steam turbine. According to the standards, both the gas turbine and the steam turbine must be measured as new machines. That necessarily means that a certain time interval will separate the gas turbine and steam turbine measurements.

To demonstrate that the guarantees have been met, the guaranteed power output is compared with the power output measured and corrected.

The power output, measured and corrected, is defined as follows:

$$P_{k\text{-Corr}} = P_{ST\text{-Corr}} + P_{GT\text{-Corr}} \tag{18}$$

Correction of the Measured Efficiency of a Combined-Cycle Plant

The guaranteed efficiency of a combined-cycle plant without supplementary firing may be written in the form:

$$\eta_{K\text{-GAR}} = \frac{P_{ST} + \Sigma P_{GTi}}{\Sigma \dot{Q}_{GTi}} = \frac{P_K}{\Sigma \dot{Q}_{GTi}} \tag{19}$$

To compare the measurements, the heat flow supplied must be corrected. The measured and corrected efficiency is then obtained from the equation:

$$\eta_{K\text{-CORR}} = \frac{P_{K\text{-CORR}}}{\Sigma \dot{Q}_{GT\text{-CORRi}}} \tag{20}$$

It is a good idea to present the correction curves for gas turbines in such a way that the heat input can be corrected directly without detouring via the efficiency of the gas turbine. This procedure can also be used for combined-cycle plants with supplementary firing.

7.4.2 Commissioning

Here it is important whether or not the waste heat boiler is equipped with a flue gas bypass. If so, the gas turbine can be started independent of the steam process. Moreover, an earlier commissioning of the gas turbine is quite conceivable, since standardized gas turbines have a shorter lead time than steam turbines, which are designed and built on a case-by-case basis.

The commissioning of the steam tubine can be similiar to that in a conventional steam turbine plant, with the flue gas dampers here replacing the boiler firing. If there is no flue gas bypass, the gas turbine and the waste heat boiler must be put into operation parallel to one another. The gas turbine cannot start until the boiler has been made ready for it and, inversely, the boiler cannot start until there is flue gas available for it.

Otherwise the commissioning of the gas turbine and the steam processes is similar to that of conventional installations. Special attention must be paid to coordinating the operating mode of the waste heat boiler, the gas, and the steam turbines. This is even more particularly the case in fully automated plants.

Chapter 8

COMPARISON OF THE COMBINED-CYCLE PLANT WITH OTHER THERMAL POWER STATIONS

Sections 2 and 3 show that combined-cycle plants are thermodynamically attractive. Thermodynamics does, indeed, play an important role in selection of the type of power plant, but it is not the only criterion for decision. Such other factors as price, environmental impact, fuel availability, etc. must also be taken into account.

In the following, we will compare the combined-cycle plant with other thermal processes in order to determine those cases in which it is actually of interest.

The main competition comes at present from steam and gas turbine power plants and that situation will not change in the near future. For smaller to medium power outputs, however, a diesel power plant can also be a genuine alternative. The high efficiency of modern diesel engines is comparable to that of a combined-cycle plant with the same power rating. Up to a power output of 20 to 30 MW, it is thoroughly conceivable that diesel power plant might be the optimum choice. On the other hand, for greater power capacities, the diesel plant loses its attractiveness because investment and maintenance costs are higher than those for a combined-cycle plant, without compensating for that fact by providing greater fuel flexibility. The diesel engine is also less desirable from an environmental point of view because it is more difficult to attain low emission levels with it, particularly for NO_x and unburned hydrocarbons.

The comparison below is restricted to the following types of power stations:

- steam turbine plants
- gas turbine plants
- combined-cycle plants

The main range of ratings under consideration is between 30 and 500 MW.

Combined-cycle plants with a smaller power output can, of course, be built, but they are less interesting for power generation alone because their relative costs increase as the power rating decreases. They are best used mainly for industrial or district heating power stations, but even for these applications, their minimum economical size is approx. 10 MW.

8.1 Economy

Every industrial plant strives to keep production costs as low as possible, and power plants are no exception in this regard. Political factors and environmental protection legislation impose certain limits on this goal, but economy of operation remains the most important criterion when selecting what type of power plant to build.

The costs for producing electricity in a power plant vary include three types of costs:

- capital costs
- fuel costs
- operation and administrative costs

The first of these depends on the price and the amortization rate for the plant, on interest, or on the desired yield on capital

investmants (annuity factor) and on the load factor of the plant, which, in turn, depends on the load, the operating time desired, and plant availability (refer to 8.1.4). Capital costs are also influenced by the interest incurred while the plant is being built.

The specific fuel costs are inversely proportional to the average efficiency of the installation. This average efficiency must not, however, be confused with the thermal efficiency at rated load . It is defined as follows:

$$\bar{\eta} = \eta \cdot \eta_{\text{Oper}} \tag{21}$$

η_{Oper} is operating efficiency, which takes into account the following losses:

- start-up and shut-down losses
- higher fuel consumption for part-load operation
- miscellaneous other heat and energy losses, e.g., due to fouling, aging, operator error, etc.

Operating and admistrative costs include:

- constant costs of operation and administration (staff costs, insurance, etc.)
- variable costs of operation and repair (maintenance, replacement parts, etc.)

The basis used for comparisons of economy is generally a present value comparison. The various costs for a power station come due at different times. For that reason, for financial calculations, they are corrected to a single reference time, which is generally the date of starting into commercial operation. These converted amounts are referred to as ''present value''.

The simplified formula used to figure the present value for all expenses is:

$$PV = TCR + \frac{T_{Nj} \cdot Y_F}{\bar{\eta} \cdot \psi} \cdot P + \frac{U}{\psi} \cdot \qquad (22)$$

in which:

TCR Total capital requirement to be written off (Present value of all expenditures during the period of construction and commissioning, such as the price of the plant, construction interest, etc.); in monetary units

T_{Nj} Equivalent utilization time at rated power output, in hr per annum; T_{Nj} = energy generated during the year, divided by the rated output

Y_F price of fuel, in monetary units per kW hr.

$\bar{\eta}$ Average plant efficiency

ψ Annuity factor in 1/a; $\psi = \dfrac{q-1}{1-q^{-n}}$

q 1 + Z

Z interest rate

n Amortization period in years

P Rated power output in kW

U Operating and adminstrative costs, including taxes and insurance, in monetary units per annum

The power production costs can be derived from the present value using the following formula:

$$Y_{EL} = \frac{PV}{T_{Nj} \cdot P \cdot \psi} \cdot \qquad (23)$$

Below we will discuss and compare the most important factors affecting the economy of a power plant for the various types of station.

8.1.1 Comparison of efficiency

At today's fuel prices, the thermal efficiency is the crucial factor for installations operated at medium and base load. For that reason, high efficiency is a prerequisite for having an economical plant.

Fig. 8-1 shows how the thermal efficiency at rated load for the types of power plant under consideration depends on the power output. Steam turbine plants have been further broken down into plants with and without reheating.

Double reheating steam plants have not been considered because they are seldom constructed. Among the combined-cycle plants, only those without or with only limited supplementary firing are shown.

The chart makes clear the thermodynamic superiority of the combined cycle. Surpassed by far are the gas turbines which, even with a high turbine inlet temperature of approx. 1100 °C (2012 °F), only attain an efficiency of 30% to 35%.

8.1.2 Comparison of Price

After efficiency, price is the most important criterion for selection. Fig. 8-2 shows how the specific investment costs for the various types of power plant depend on their power output. These costs are valid for a turnkey installation including machine transformer, but not as workshop, offices, staff facilities, and the like. They have been based on 1988 price levels and progress payments, and do not include interest during construction. The data shown merely indicate trends: appropriate caution must be taken in applying them, since very many factors affect the price of a power plant: erection site, commercial risks, the political situation, impediments to construction, legal regulations, etc.

Figure 8-1

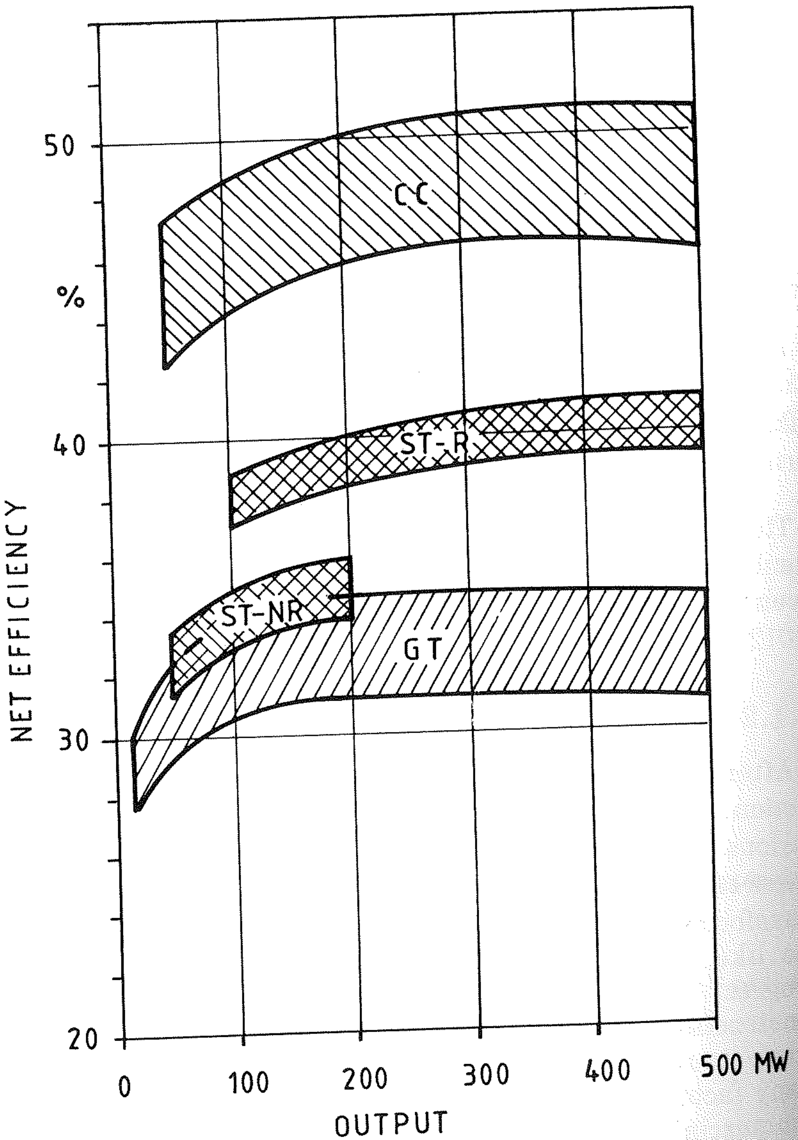

Fig. 8-1: Comparison of the Thermal Efficiencies of Various Types of Power Plant

CC Combined-cycle plant
ST-R Reheat steam turbine plant
ST-NR Non-reheat steam turbine plant

Figure 8-2

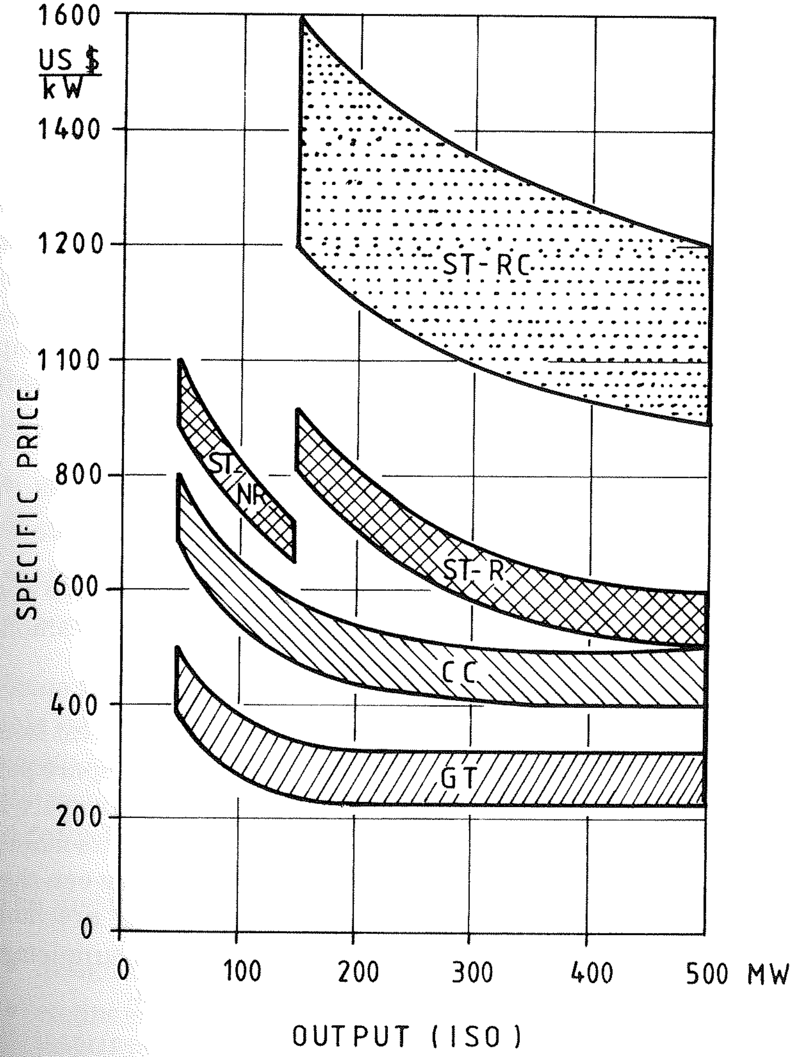

Fig. 8-2: Comparison of the Investment Costs Required for Various Types of Power Plants (Price basis: 1988)

CC Combined-cycle plant
ST-RC Reheat steam turbine plant, coal-fired
ST-R Reheat steam turbine plant, oil or gas-fired
ST-NR Non-reheat steam turbine plant, oil or gas-fired
GT Gas turbine power plant

This diagram clearly shows the low investment costs required for the gas turbine, which have contributed significantly to its wide-spread acceptance.

Steam power stations are significantly more expensive than a combined-cycle power plant. A coal-burning plant, for example, costs two to three times as much as a combined-cycle plant with the same power output. Modern combined-cycle power plants are therefore simpler and less expensive than steam power units.

Nuclear power plants have not been included in this comparison because the investment costs required for them are far too dependent upon political and other local considerations.

8.1.3 Comparison of Operating and Administrative Costs

At today's fuel costs, the operating and administrative costs affect the economy of a power plant only slightly. They amount only to 5 to 10% as much as the fuel costs.

Because of the simplicity of the gas turbine, it is lowest in operating and maintenance costs even if it requires more spare parts than a steam turbine alone. Little staff and maintenance are required. A steam power plant requires more staff and its maintenance costs are higher. Combined-cycle plants fall between these two extremes: units without supplementary firing are more like a gas turbine plant, and those with maximum supplementary firing more like a steam turbine power plant.

8.1.4 Comparison of Availability

The availability of a power plant greatly affects its economy. Whenever a unit is down, the electricity must either be generated in another power station or— if there is no reserve available— purchased from another electric utility. In both cases, the replacement energy is more expensive than that generated by the plant itself, since capital costs are incurred whether the plant is running or not.

No values for availability can be stated that will be valid for all cases since such factors as preventive maintenance and operating mode make a huge difference. However, according to well-known statistics, all the plants under consideration have similar availabilities when used under the same operating conditions. [406]

Typical figures for the time availability of well-designed and maintained plants are as follows:

• gas turbine plants (gas-fired)	88 - 95 %
• steam turbine plants (oil or gas-fired)	85 - 90 %
• steam turbine plants (coal-burning)	80 - 85 %
• combined-cycle plants (gas-fired)	85 - 90 %

These figures are valid for plants operated at base load; they would be lower for peaking or medium-load machines because frequent start-ups and shut-downs greatly reduce life expectancy of the machine and therefore increase the scheduled maintenance and forced outage rates.

The major factors determining plant availability are:

- design of the major components
- engineering of the plant as a whole, especially of the interfaces between systems
- mode of operation (base, medium, or peak load)
- type of fuel
- qualifications and skill of the operating and maintenance crews

8.1.5 Comparison of Construction Time

The time required for construction affects the economy of a unit. The longer it takes, the larger the capital rqeuirement to be written off, since construction interest, price increases for materials, insurance, and taxes during the construction period add to the price of the plant.

Fig. 8-3 shows the amount of time required to build the various types of power plants. The gas turbine, because of its standardized design, can be built with the shortest lead time, which has encouraged its widespread acceptance. More time is needed up to the completion of a combined-cycle plant. One can, however, commission the gas turbines prior to the steam process, so that from 60 to 70% of the power output is available after the same time as would be required for a gas turbine power plant. This is a great advantage over a conventional steam power plant, which can deliver power only after two to four years.

8.1.6 Comparison of Economy

The diagrams below show the effects of the most important parameters on the economy of a power plant:

1. Fig. 8-4 to 8-6: Dependence of Cost of Power Generation on Fuel Prices, for 50, 200, and 500 MW plants
2. Fig. 8-7 to 8-9: Dependence of Cost of Power Generation on the Equivalent Utilization Time, for 50, 200, and 500 MW plants
3. Fig. 8-10: Dependence of Cost of Power Generation on the Annuity Factor, for a 200 MW plant.

A combined-cycle power plant has less fuel flexibility than a steam power plant. The question can therefore be asked whether an oil or gas burning combined-cycle plant is more economical

Figure 8-3

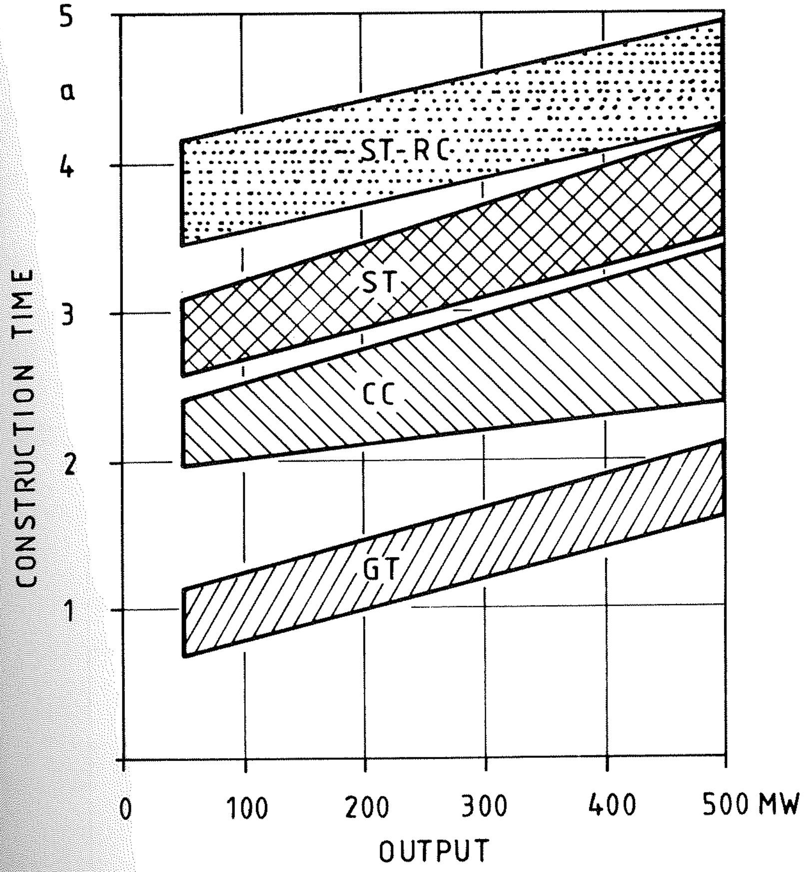

Fig. 8-3: Comparison of the Construction Time for Various Types of Power Plants

CC	Combined-cycle plant
ST-RC	Reheat steam turbine plant, coal-fired
ST	Steam turbine plant, oil or gas-fired
GT	Gas turbine power plant

Figure 8-4

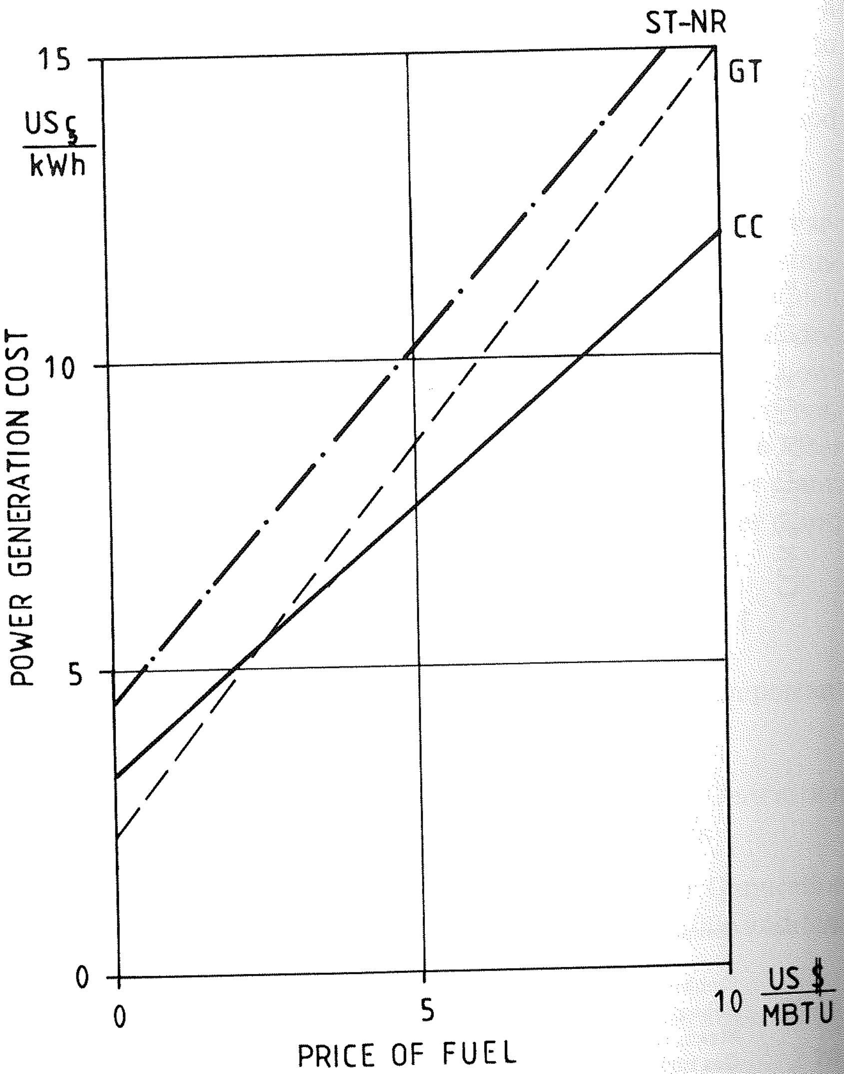

Fig. 8-4: Dependence of Power Generation Costs on Price of Fuel

CC	Combined-cycle plant
GT	Gas turbine plant
ST-NR	Non-reheat steam turbine plant

Power rating = 50 MW
Annuity factor = 11%
Equivalent utilization time H = 4000 hr per annum

Figure 8-5

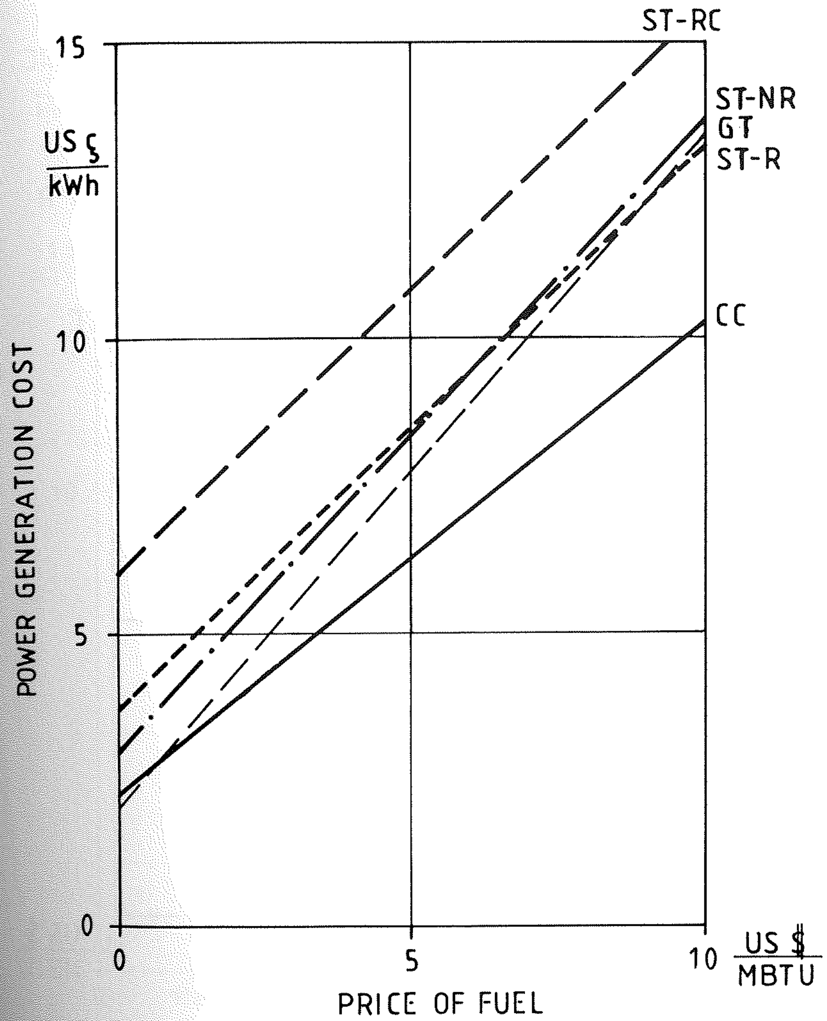

Fig. 8-5: Dependence of Power Generation Costs on Price of Fuel

CC	Combined-cycle plant
GT	Gas turbine plant
ST	Reheat steam turbine plant (oil or gas)
ST-RC	Reheat steam turbine plant (coal)

Power rating = 200 MW
Annuity factor = 11%
Equivalent utilization time H = 4000 hr per annum

Figure 8-6

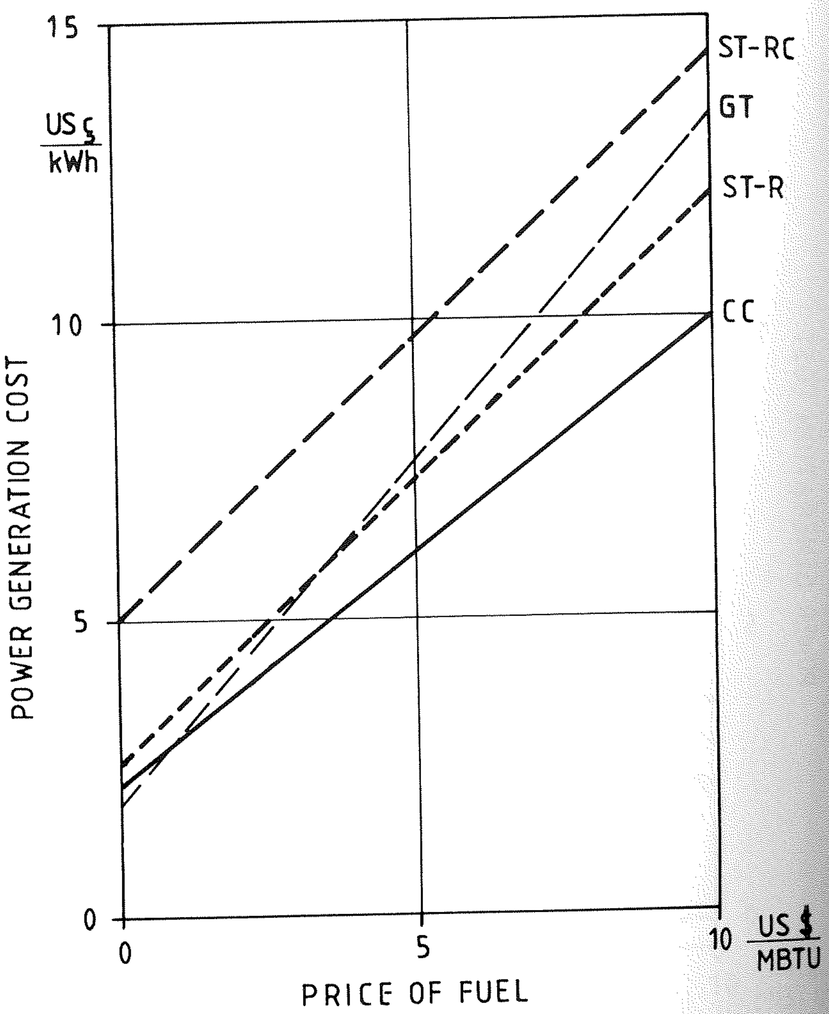

Fig. 8-6: Dependence of Power Generation Costs on the Price of Fuel

CC	Combined-cycle plant
GT	Gas turbine plant
ST-R	Reheat steam turbine plant (oil or gas)
ST-RC	Reheat steam turbine plant (coal)
Power rating	= 500 MW
Annuity factor	= 11%
Equivalent utilization time H	= 4000 hr per annum

Figure 8-7

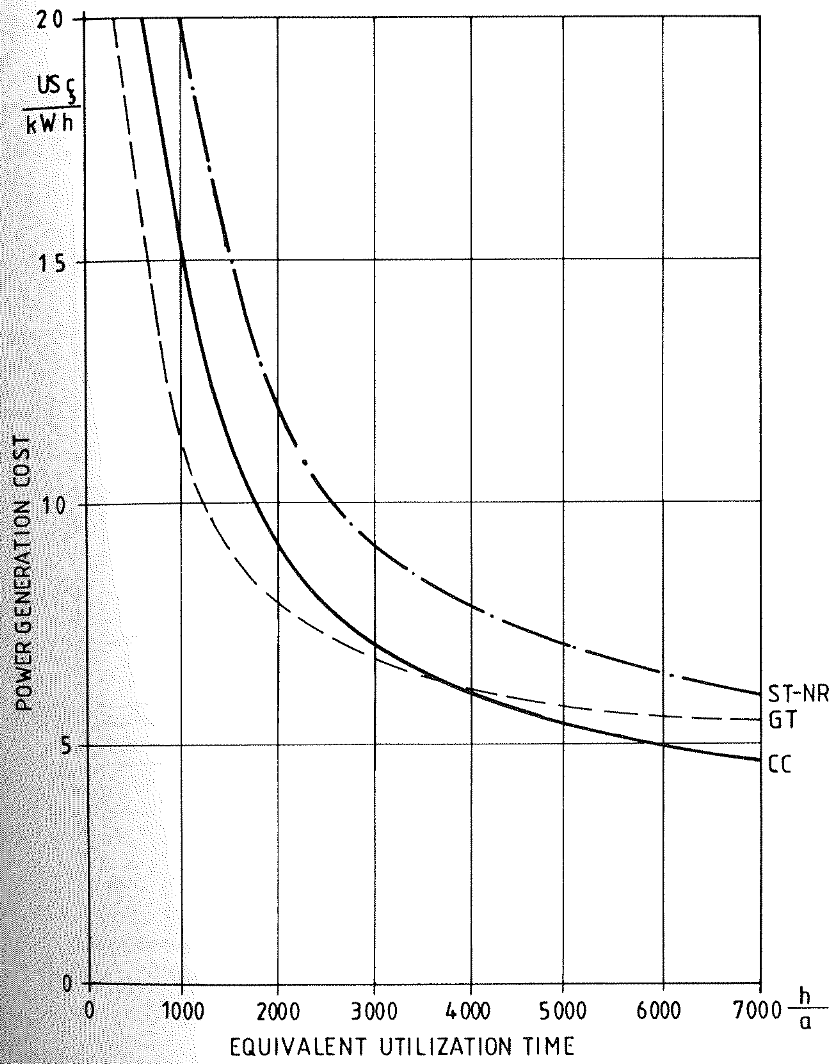

Fig. 8-7: Dependence of Power Generation Costs on Equivalent Utilization Time

CC	Combined-cycle plant
GT	Gas turbine plant
ST-NR	Non-reheat steam turbine plant
Power rating	= 50 MW
Fuel Price	= US $3/$10^6$ Btu (LHV)
Annuity factor	= 11%

Figure 8-8

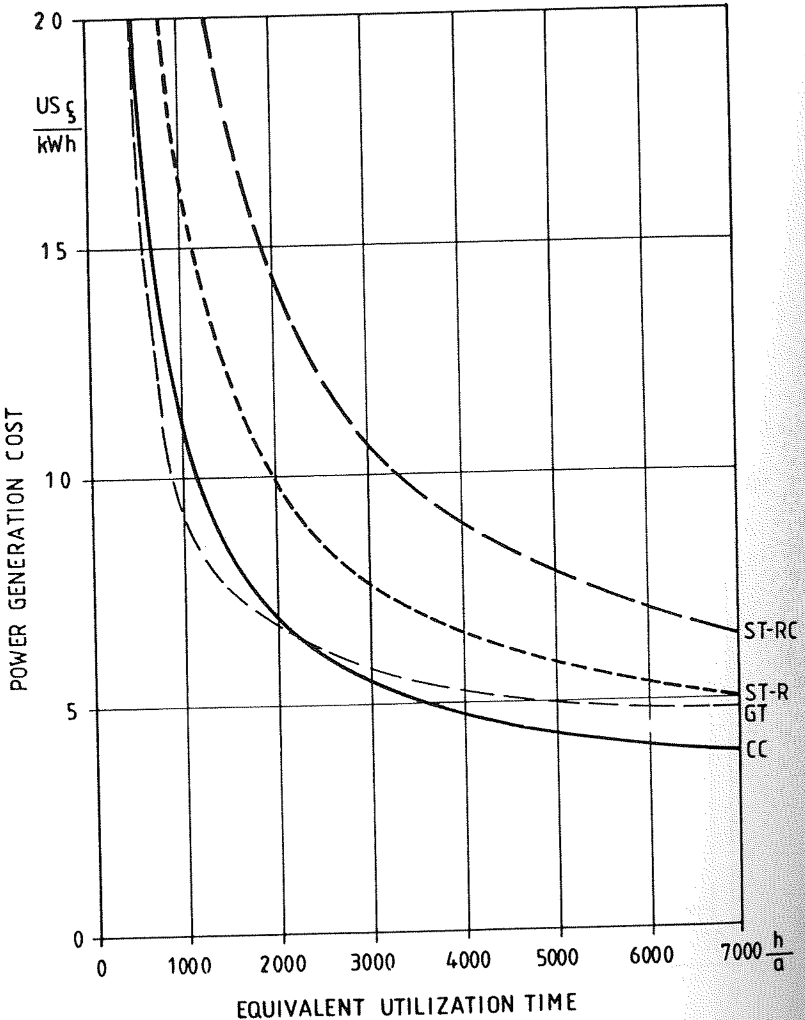

Fig. 8-8: Dependence of Power Generation Costs on Equivalent Utilization Time

CC	Combined-cycle plant
GT	Gas turbine plant
ST-R	Reheat steam turbine plant (oil or gas)
ST-RC	Reheat steam turbine plant (coal)
Power rating	= 200 MW
Fuel Price	= US $3/$10^6$ Btu (LHV)
Annuity factor	= 11%

Figure 8-9

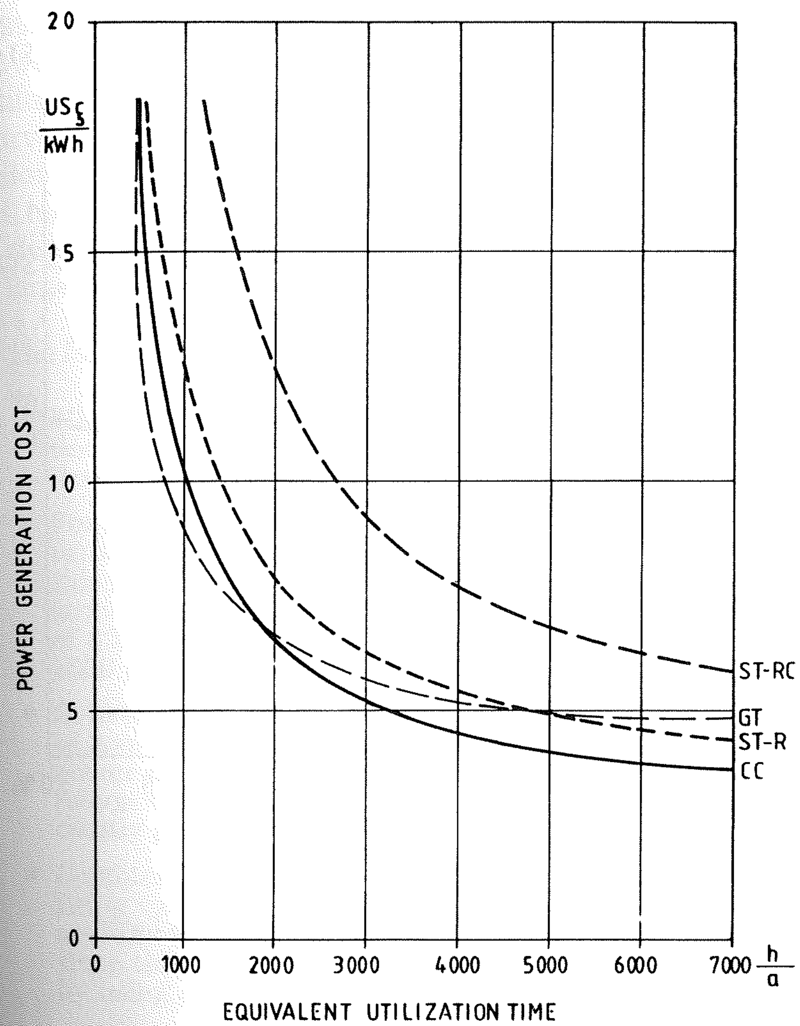

Fig. 8-9: Dependence of Power Generation Costs on the Price of Fuel

CC	Combined-cycle plant
GT	Gas turbine plant
ST-R	Reheat steam turbine plant (oil or gas)
ST-RC	Reheat steam turbine plant (coal)
Power rating	= 500 MW
Fuel Price	= US $3/$10^6$ Btu (LHV)
Annuity factor	= 11%

Figure 8-10

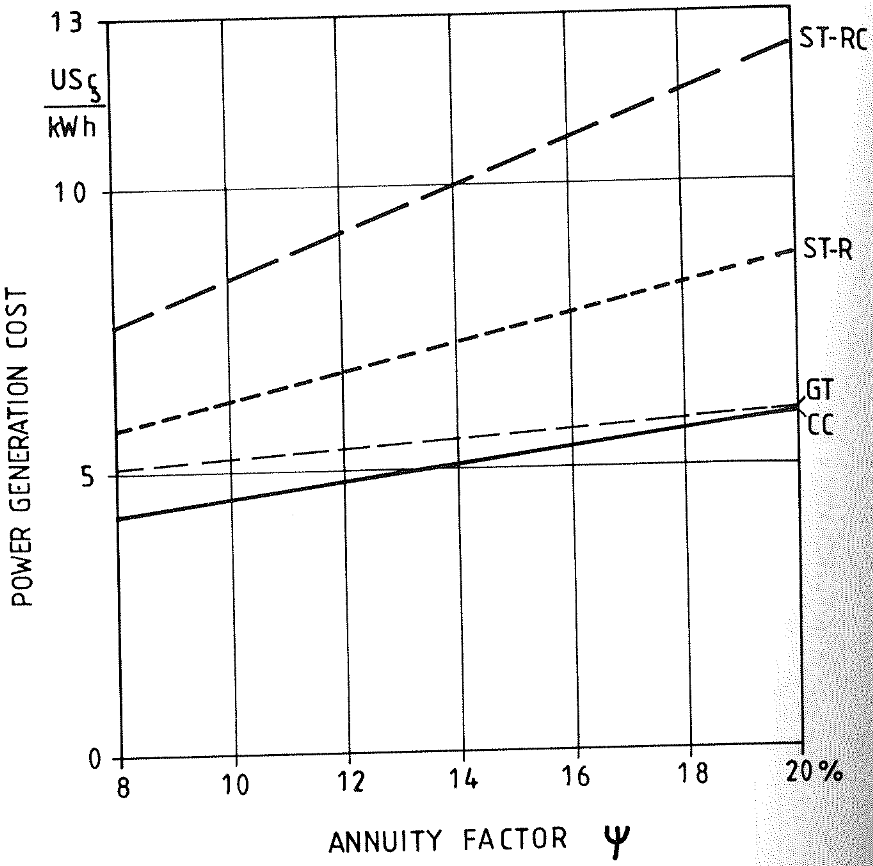

Fig. 8-10: Dependence of Power Generation Costs on the Annuity Factor

CC	Combined-cycle plant	
GT	Gas turbine plant	
ST-R	Reheat steam turbine plant (oil or gas)	
ST-RC	Reheat steam turbine plant (coal)	
Power rating		= 200 MW
Fuel Price		= US $3/$10^6$ Btu (LHV)
Equivalent utilization time H		= 4000 hr per annum

than a steam power station that burns a less expensive fuel, e.g., coal. Fig. 8-11 shows the difference in the price of fuel between a 200 MW combined-cycle plant and a coal-burning reheat steam power plant of the same size, as a function of the equivalent utilization time. If the actual fuel price differential is above the curve, the coal-fired unit produces electricity cheaper; if it is below the curve, the combined-cycle is better. It must be stated, however, that it is generally easier to obtain a permit to build a clean gas-fired plant or a combined-cycle plant than to build a large steam turbine plant, particularly one that burns coal.

The following conclusions can be drawn from these diagrams:

- Whenever oil or gas is being burned in a power station, the combined-cycle plant is more economical than the steam power plant.
- For short utilization periods (a peaking plant), the gas turbine is most economical. At today's fuel prices, this limit is at about 1000 to 2000 hr per annum for large plants. Even when burning crude oil at a price of $20 (US) per barrel, the limit is around 1000 to 1500 hr per annum.
- Conventional steam power plants are suitable for use as coal-burning base load or possibly medium-load power plants. However, the price differential between the coal and the gas turbine fuel must be sufficiently great (approx. $3-6 US per GJ or MBtu).
- Combined-cycle plants with maximum supplementary firing can be of interest if there is only a small amount of gas or oil available at favorable terms, since the rest of the fuel requirement can be covered by using coal for the supplementary firing.

8.2 Fuels

The selection of the fuel and the corresponding type of power plant is determined not only by short-term economic considerations but also in accordance with political criteria and assumptions about long-term developments in the prices for the various possible fuels. In this regard, the following aspects can become important in selecting the type of power station to be built:

- long-term availability of the fuel at a reasonable cost
- risk of a supply shortage due to political interference, such as war, boycott, etc.
- political opposition to nuclear power plants
- environmental protection

The result of all these factors may well be that the fuel selected is may be one other than that which appears best at the time of plant construction.

The long-term source of fuel can be taken into account only to a limited extent, by including in the calculations of economic costs an estimated price increase for the fuel during the expected service life of the unit. However such estimates should be handled with care. In any case, the greater the fuel flexibility of the plant chosen, the less the risk from possible increases in fuel prices.

Table 8-1 lists the fuels that can be burned in the various power plants today.

The fuel flexibility of combined-cycle plants is less than that of steam power plants. Some gas turbines can burn heavy oil or crude, provided the machine is designed to do so. Industrial gas turbines are more suitable than those derived from jet technology. It is easier to burn special fuels in gas turbines with large combustors than in those with several smaller combustors or an annular combustor, since the latter are more sensitive to changes in flame length, radiation, etc.

A second requirement for burning heavy oil or crude in a gas turbine is the correct treatment of the fuel, generally by means of washing and dosing with additives. These steps make it possible to remove or inhibit elements that cause high temperature corrosion, such as vanadium, sodium, etc. The specific problems involved with the burning of coal in a combined-cycle plant are dealt with in more detail in Section 9.2.

Table 8-1: Various Types of Power plants and the Fuels They can Burn

	Gas Turbine	Combined-cycle without sup. firing	Combined-Cycle with sup. firing	Steam power plant
Natural gas	Yes	Yes	Yes	Yes
Diesel oil	Yes	Yes	Yes	Yes
Crude	Yes 1)	Yes 1)	Yes 2)	Yes
Heavy oil	Yes 1)	Yes 1)	Yes 2)	Yes
Brown coal	No	No	No	Yes
Bituminous coal	No	No	Yes 3)	Yes
Refuse	No	No	No	Yes
Agricultural waste	No	No	No	Yes
Blast furnace gas	Yes 4)	Yes 4)	Yes 4)	Yes
Chemical industry gas	Yes 4)	Yes 4)	Yes 4)	Yes
Coal gas	Yes 4)	Yes 4)	Yes 4)	Yes

1) Heavy oil or crude cannot be burned in every gas turbine. Generally a fuel treatment unit is required.

2) Note 1) applies to the gas turbine. In the supplementary firing, however, heavy oil can be burned just as well as in a conventional steam generator.

3) To be used only in the supplementary firing.

4) These fuels can generally be used as gas turbine fuels. However, adaptations on the machine are required if the heat value is low.

Figure 8-11

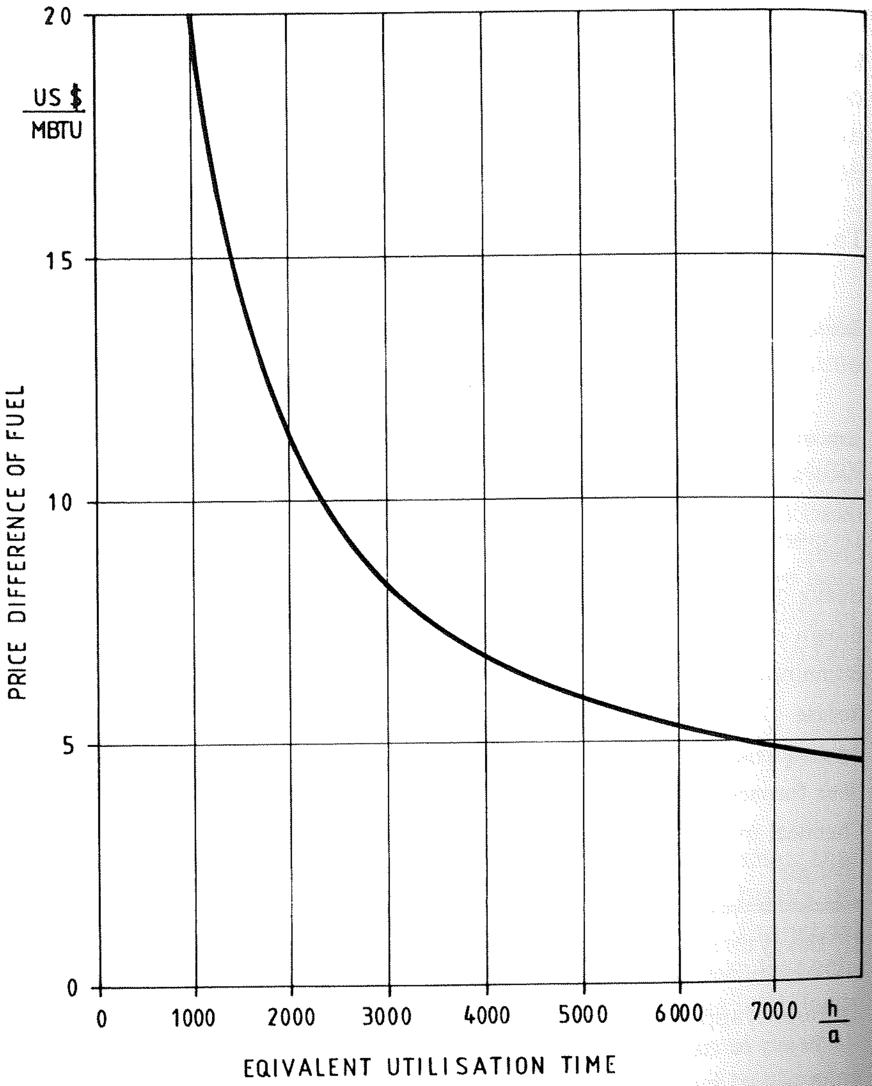

Fig. 8-11: Permissible Difference Between the Price of Fuel for a Combined-Cycle Plant and That for a Steam Power Plant

Annuity Factor = 11%

Chapter 9

ENVIRONMENTAL CONSIDERATIONS

The impact of any power plant upon its environment must be kept as low as possible, but striking a reasonable relationship between the cost and the results obtained should serve as a guideline, even for those responsible for making and enforcing the regulations. The following emissions from a power station directly affect the environment:

- products of combustion (exhausts and ash)
- waste heat
- noise

The exhausts can include the following components: H_2O, N_2, O_2, NO, NO_2, CO_2, CO, C_nH_n (unburned hydrocarbons, UHC), SO_2, SO_3, dust, fly ash, heavy metals, chlorides, etc.

The first three of these are harmless; the others can impact negatively upon the environment. Their concentraion in the exhaust gas depends upon the composition of the fuel and the type of installation in question. However, a high efficiency always works out positively since the proportion of emissions per unit of electrical energy produced drops off.

A combined-cycle plant is beneficial because of its high efficiency, and the high excess air coefficients customary in gas turbines produce a practically complete combustion, i.e., a very low concentration of unburned elements such as CO or unburned hydrocarbons. The large air flows have the further advantage of strongly diluting the pollutants.

For these reasons, a combined-cycle plant is quite well suited for use in heavily populated areas. Particularly when the fuel burned is natural gas, the only toxic emissions contained in the exhausts are NO and NO_2. The NO_x ($NO + NO_2$) level is the most important environmental problem with gas turbines because NO_x generates nitric acid (H_2NO_3) in the atmosphere, and this, together with sulphuric acid (H_2SO_4) is one of the factors responsible for acid rain.

9.1 Reduction of No_x Emissions

NO_x is produced in large quantities only at very high temperature levels. The NO_x concentrations at the equilibrium shown in Fig. 9-1 as a function of the air temperature are attained only after a very long time. The situation in a gas turbine combustor is quite different, first, because a combustion actually takes place and, second, because residence times at high temperatures are fairly limited. The major factors affecting NO_x production in the combustor are thus:

- the excess air ratio of the combustion (λ)
- the temperature of the air after the compressor, which, in turn, depends on the pressure ratio,
- the duration of the combustion

As can be seen from Fig. 9-1, NO_x is formed only when the temperatures are high, such as those found in the flame of the combustor. The temperature of this flame depends on the excess air ratio λ; as shown in Fig. 9-2, it is highest in the case of stoichiometric combustion ($\lambda = 1$).

Fig. 9-3 also shows how concentrations of NO_x depend on the excess air ratio (λ) and the compressor pressure ratio. It is evident that a peak is reached with a factor λ of approx. 1.2. Below that level, the flame temperature is higher but there is less oxygen available to form NO_x, since most of it is used for the combustion. Above that level, NO_x decreases because an overabundance of air within the flame lowers the flame temperature.

Very low excess air ratios are beneficial from the point of view of NO_x formation but are very detrimental to efficiency and cause the production of large amounts of CO and unburned hydrocarbons (UHC).

Normally, gas turbine combustors operate with an excess air ratio of approx. 1 at full load, ensuring a good, stable combustion over the entire load range. Obviously, NO_x emissions will be very high unless special precautions are taken. Typically, NO_x levels in the exhaust gases after mixing with the cooling and secondary air are in the range from 120 to 300 vppm.

The simplest way to reduce the NO_x concentration is to cool the flame, which can easily be accomplished by injecting water or steam into it. Fig. 9-4 shows the reduction factors for NO_x emission which can be attained as a function of the amount of water or steam injected, indicated by the coefficient Ω , the ratio between the flows of water or steam and fuel. At a ratio $\Omega = 1$, the typical reduction factor is approx. 5 with water, and approx. 3 with steam. Steam is less efficient than water because the evaporation takes place outside the flame.

With this method, it is possible to attain NO_x levels in the dry exhaust gases from a gas-fired gas turbine or a combined-cycle plant as low as 40 vppm. In some cases, even 25 vppm is attainable.

Steam or water injection is a simple way to reduce emissions, but it does entail the following disadvantages:

Figure 9-1

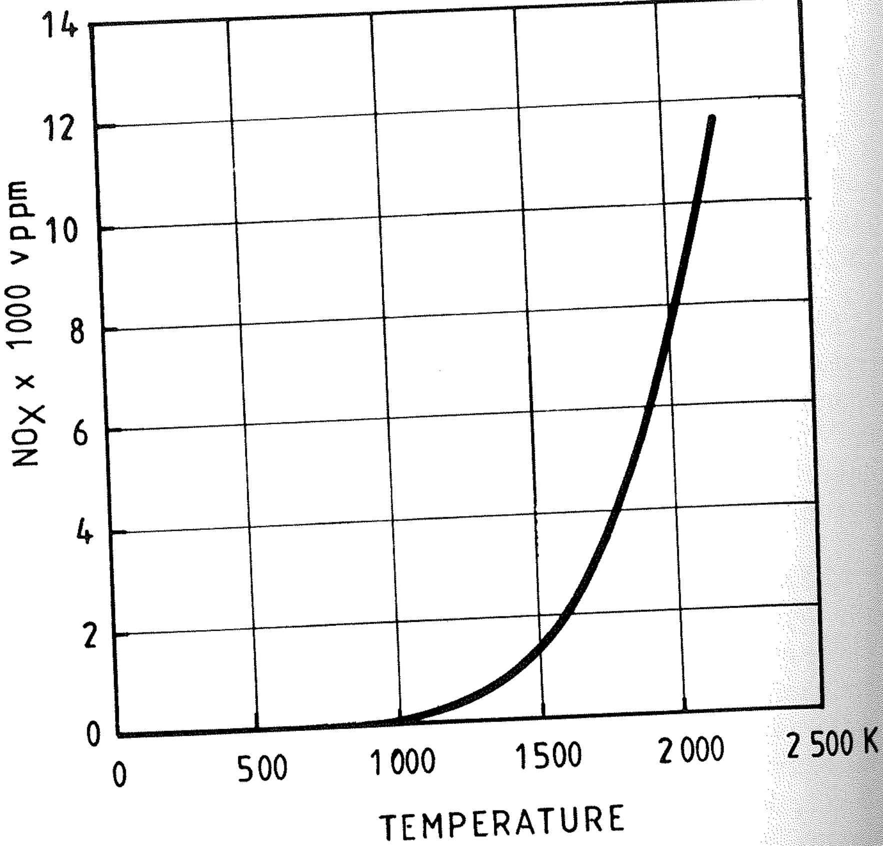

Fig. 9-1: NOx Equilibrium as a Function of Air Temperature

Figure 9-2

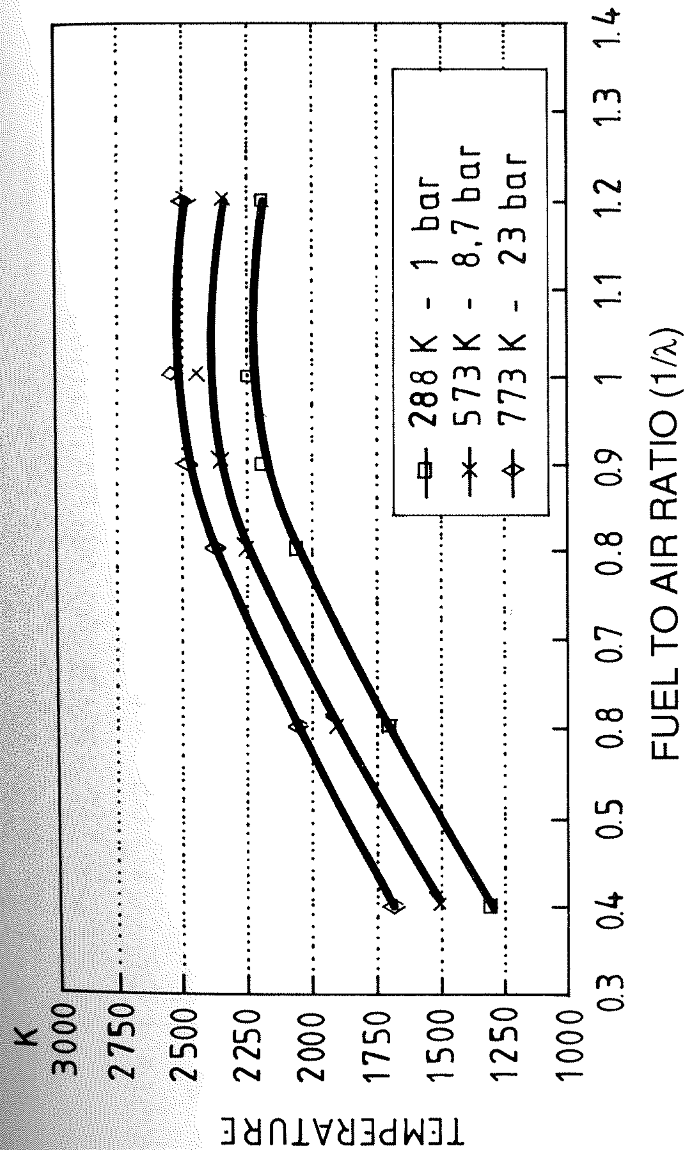

Fig. 9-2: Flame Temperature as a Function of the Excess Air Ratio

Figure 9-3

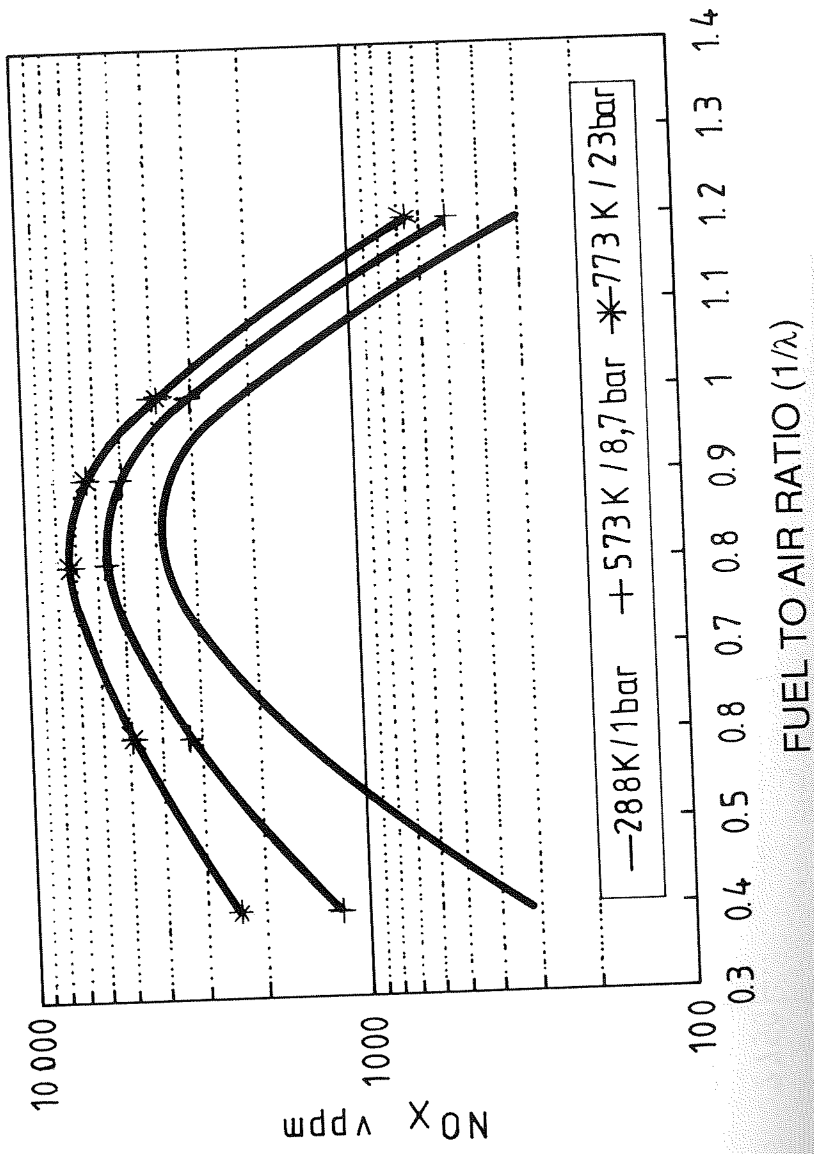

Fig. 9-3: NOx Concentration as a Function of Excess Air Ratio and Compressor Pressure Ratio

Figure 9-4

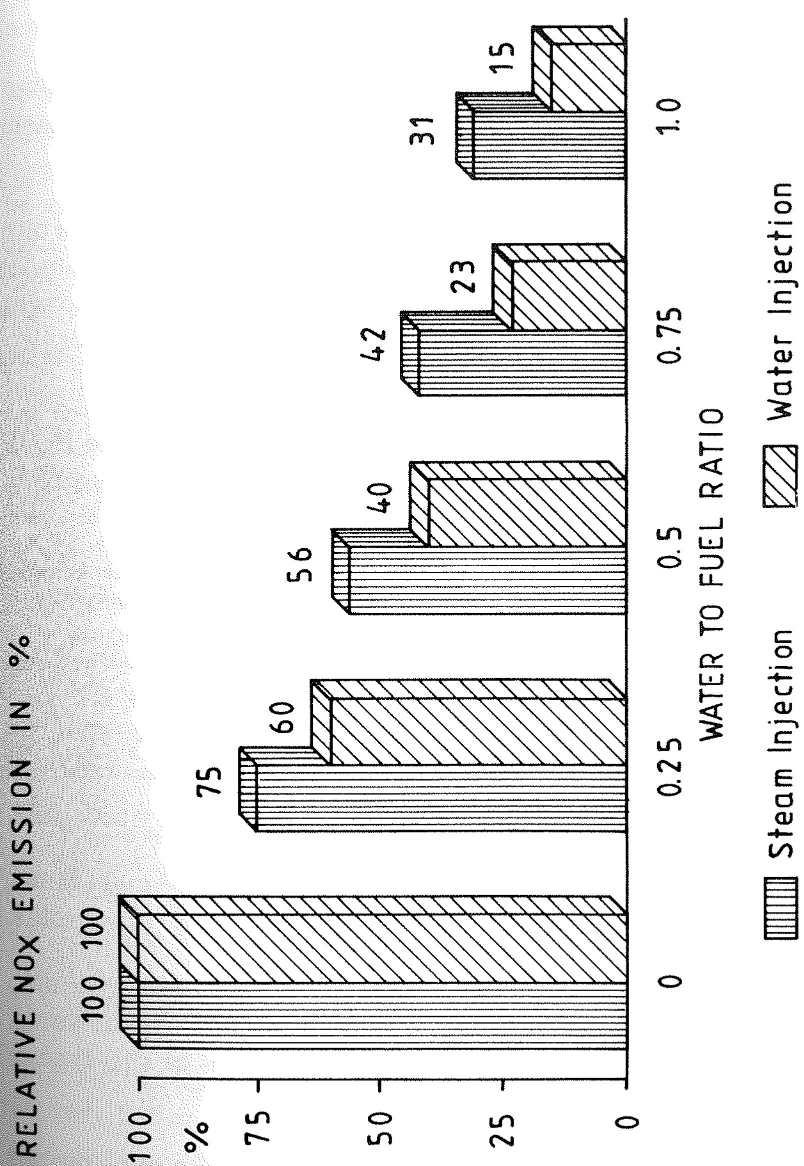

Fig. 9-4: NOx Reduction Factor as a Function of the Water or Steam to Fuel Ratio

- Large amounts of demineralized water are required.
- The efficiency of the combined-cycle plant is lower, particularly if water injection is used.

The fact that the output capability of the same plant is higher, especially with water injection, only partly compensates for these disadvantages.

Fig. 9-5 shows how steam and water injection affect the output and efficiency of a combined-cycle plant, as a function of the ratio of water to fuel, Ω. With a ratio $\Omega = 1$, the following changes in output and efficiency from those in a normal dry cycle without injection may be considered as typical:

Table 9-1: Output and Efficiency of a Combined-Cycle Plant with Water or Steam Injection, as Compared to the Same Plant without Injection

	Change in Efficiency, %	Change in Output, %
Water injection, $\Omega = 1$	– 4.8 %	+ 4.7 %
Steam injection, $\Omega = 1$	– 1.9 %	+ 1.1 %

In all cases the fuel is natural gas and a two-pressure cycle is being used.

With steam injection, the cycle is similar to the example shown in Fig. 3-50, but there is a dual steam admission into the turbine.

A 3-pressure cycle as shown in Fig. 3-49 would reduce the drop in efficiency caused by the steam injection but the gain would, in most cases, be offset by the additional costs of this type of plant (refer also to Section 3.1.4).

These disadvantages of steam or water injection have pushed all builders of gas turbines toward development of a new type of combustor with which the NO_x levels attainable are similar

to those with injection systems. Section 10.3 provides further, more detailed information about these combustors, referred to as dry low NO_x combustors.

Some local regulations, e.g., those in California or Tokyo, require NO_x emission levels much lower than 40 vppm. In these cases, it is generally necessary to install a reduction system in the exhaust system. These systems, known as "Selective Catalytic Reduction" (SCR) systems, inject ammonia (NH_3) into the exhaust gases before a catalyst and can thereby remove approx. 90% of the NO_x from them. The chemical reactions involved are as follows:

$$4\,NO + 4NH_3 + O_2 \rightarrow 4\,N_2 + 6\,H_2O$$

$$6\,NO_2 + 8NH_3 \rightarrow 7\,N_2 + 12\,H_2O$$

Technically these are well-proven systems, but they entail the following disadvantages:

- Investment costs are high (equal to about 20% of the cost of the gas turbine).
- Replacement costs are high (the life expectancy of the catalyst is between 4 and 8 years).
- The catalyst must be installed in the center of the high pressure evaporator of the waste heat boiler, since this reaction functions properly only at temperatures between 300 and 400 °C (572 and 752 °F).
- The use of ammonia is necessary.
- Efficiency is slightly lower because of the increased pressure loss in the waste heat boiler.

Fig. 9-6 shows a typical SCR system for installation in a heat recovery boiler.

Using these systems in conjunction with steam or water injection, it is today technically possible to attain an NO_x level in the exhaust gases from a combined-cycle plant of less than 10

Figure 9-5

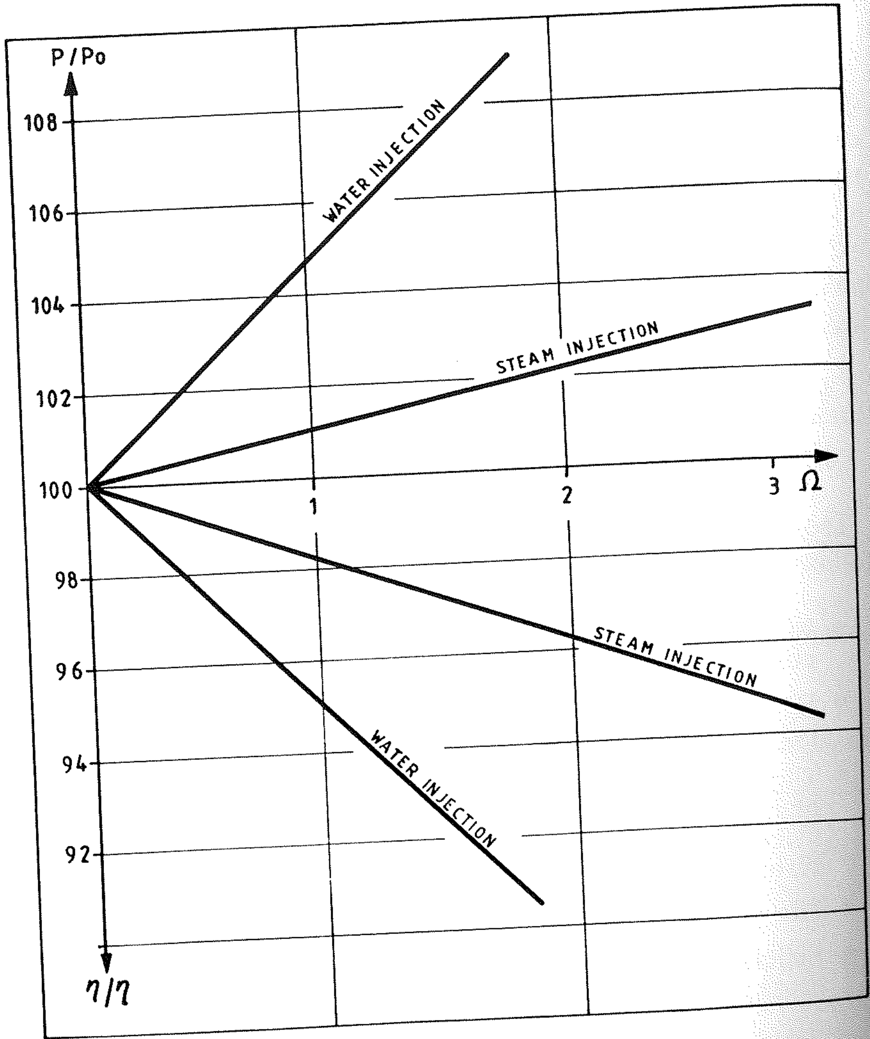

Fig. 9-5: Output and Efficiency of the Combined-cycle Plant as a Function of Water or Steam Injection

Figure 9-6

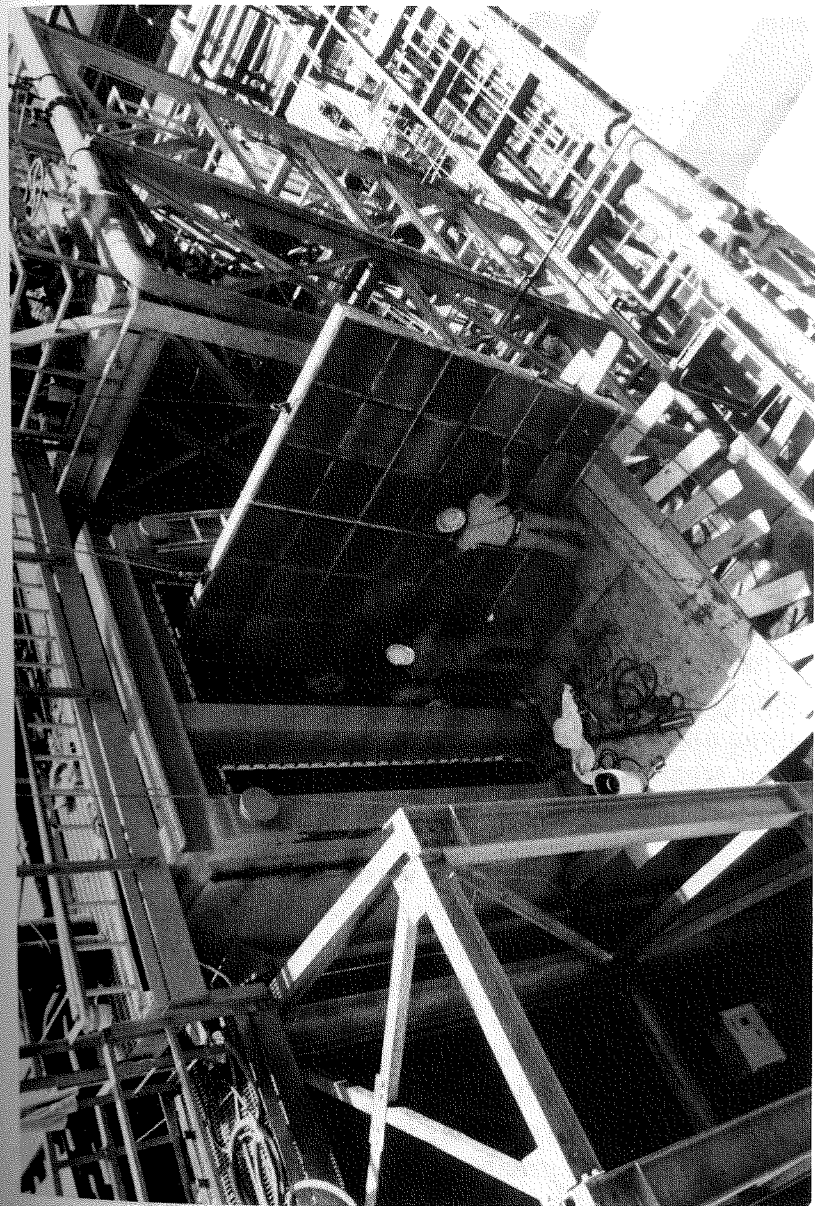

Fig. 9-6: Selective Catalytic Reduction (SCR) System

vppm. Catalytic reduction can also be used in conventional steam plants, but the minimum level of NO_x attainable is higher, and— at least in a coal-burning plant— much higher.

9.2 SO_x Emissions

The concentrations of SO_2 and SO_3 produced depend mainly on the quality of the fuel. Because gas turbines generally use clean fuels, this is less of a problem in combined-cycle plants than in coal-burning power stations. The latter plants can, however, be equipped with scrubbers which reduce sulphur emissions by approximately 90% by converting the SO_2 into plaster of Paris. A similar system could in theory be installed after a combined-cycle plant as well, but it would be much too expensive due to the high excess air. The use of a fuel with a low sulphur content is therefore a less costly solution.

9.3 Waste Heat Rejection

Another environmental problem is the waste heat that every power station supplies to the environment. Here, too, the high efficiency of the combined-cycle plant is an advantage: from any given amount of primary energy, a greater amount of electricity or steam is being produced, which reduces the amount of waste heat remaining.

In addition to the quantity of waste heat, however, the form in which that heat is given off the the environment is also important. The effect is less if the power plant heats the air instead of giving off its heat to a river or the sea.

One disadvantage of a steam power plant is that its waste heat can best be dissipated with water. It can, of course, also be given off to the air, but the expense involved (cooling tower or air-cooled condenser) is greater. A combined-cycle plant has the advantage over a steam plant in that it requires only half as much cooling water.

Table 9-2 shows the amounts of waste heat that must be dissipated, as percentages of the primary energy input. Cooling towers have not been taken into consideration, i.e., the condenser is considered as being cooled with river or sea water.

A gas turbine requires practically no cooling water, which has contributed greatly to its widespread acceptance in water-poor countries.

Table 9-2: Comparison of the Heat to be Dissipated as Percentages of the Energy input

Cooling medium	Gas Turbine	Combined-Cycle Plant	Steam Power Plant without reheat	Steam Power Plant with reheat
Air 1)	68-75	27-14	10-14	10-15
Water	0	28-38	52-56	44-52

1) mainly stack losses

9.4 Noise Immissions

A final environmental problem is noise, but this can be solved using the acoustic insulation available today . The costs are approximately the same for all types of power plant installations.

Chapter 10

DEVELOPMENTAL TRENDS

Main trends in development are in three directions:

- toward increased efficiency and power output of the gas turbine
- toward the use of coal in combined-cycle plants
- toward attaining lower NO_X emission levels without water or steam injection (dry low NO_X combustors)

The first of these trends is a continuation of the development that has led to the break-through of the combined-cycle plant in the past few years.

The attempt to use fuels other than oil or natural gas in combined-cycle plants or gas turbines is not new. Development today is concentrating mainly on the utilization of coal in plants known as **I**ntegrated **C**oal **G**asification **C**ombined-**C**ycle (IGCC) or **P**ressurized **F**luidized **B**ed **C**ombustor (PFBC) plants.

10.1 New Gas Turbines

The positive effect of a high gas turbine inlet temperature on the efficiency of a gas turbine or a combined-cycle plant has already been discussed in Section 2 (refer to Fig. 2-2). It seems reasonable to look for further improvement mainly in the direction of higher gas inlet temperatures, which has become possible through the development of new materials and improved cooling systems. Research projects here are concerned mainly with improved air cooling of the hot gas path of the gas turbine. More advanced cooling technologies, for example, employing water or steam cooling systems, do not yet appear close to a stage of development where they could be used in a real gas turbine.

For these reasons, work at present is concentrating on the development of new materials, e.g., γ' superalloys, oxide dispersion strength (ODS) alloys, directionally solidified (DS) blades, and of more efficient air-cooling systems employing

- improved film cooling
- impingement cooling

The use of ceramic materials in gas turbines, mainly for blading, still appears far from ready for commercial application because of the very low reliability that must be expected with such blades. Ceramics are presently being used only for certain parts in the hot gas path of a few gas turbines.

Parallel to this, improvements are also being made to the compressor. The advantages offered by the higher gas temperature cannot by fully exploited only if the pressure ratio of the machine is increased to an appropriate level. High unit ratings are also being attained by increasing the air flow through the compressor. With modern blading, compressors are able to handle volume flows that seemed utopian just a few years ago. With the use of transonic stages, it is already possible to attain equivalent outputs and pressure ratios with many fewer compressor stages.

With these improvements, combined-cycle efficiencies of more than 50% are already being attained. Gas turbines with unit power capacities of more than 200 MW for 50 Hz applications and 150 MW for 60 Hz applications mean lower costs, so that combined-cycle power plants will become even more interesting for large power stations. It can be expected that gas-fired combined-cycle plants will, within the next 10 years, reach efficiencies of 52 to 55% or more (LHV).

10.2 Coal Gasification, Fluidized Bed Combustion

This development is directed at making it possible to use coal directly or indirectly in combined-cycle installations. Two paths have been suggested and seem close to commercial realization:

- autothermal coal gasification
- pressurized fluidized-bed combustion

10.2.1 A Combined Cycle Plant with Coal Gasification

The gasification of coal is a very old technology. Before natural gas was introduced on the market, coal gasification was used to produce fuel gas for distribution in urban areas. It is also used quite frequently in the chemical and petrochemical industries to produce raw materials for chemical processes.

Autothermal coal gasification is based on partial combustion of the coal in order to generate the heat required for the gasification process itself, in which the coal is converted, generally together with water, into H_2, CO, CO_2, CH_4, H_2S, and H_2O. The main reactions involved are:

$$\begin{array}{lllll} C + H_2O & \rightarrow & CO & + & H_2 \\ C + CO_2 & \rightarrow & 2\,CO & & \\ C + 2H_2 & \rightarrow & CH_4 & & \\ CO + 3H_2 & \rightarrow & CH_4 & + & H_2O \end{array}$$

Because the overall reaction is endothermic, the coal must be partially burnt. The oxygen required for combustion to take place is supplied to the gasifier by injecting either air or oxygen into it. Gasification processes can therefore be classified either as:

1) air-blown gasifiers, which produce a gas with a low calorific value, typically 5000 to 6000 kJ/kg (100 to 120 Btu/scf)

2) oxygen-blown gasifiers, which produce a gas with a medum calorific value, typically 15,000 kJ/kg (or 330 Btu/scf), approx. one-third the calorific value of natural gas.

Either type of gasifier can be employed in combined-cycle applications; the advantages and disadvantages of each will be discussed further below.

Table 10-1 shows the composition of typical gas obtained in the two different gasification processes.

Table 10-1: Typical Composition of Gas, in % by volume

	Natural Gas (Reference)	Coal Gasification Oxygen-blown	Air-blown
CO	–	60	25
H_2O	–	30	18
CO_2	–	3	7
N_2	10	2	49
CH_4	90	5	1
Relative heating value	1 (Reference)	1/3	1/9
Adiabatic flame temperature	2100 °C (3812 °F)	2300 °C (4172 °F)	1900 °C (3452 °F)

In addition to this general classification based upon the medium used for oxidation, a distinction must also be made based on the flow pattern in the gasifier itself. There are three main types:

1) fixed bed gasifiers with counter-current flow
2) fluidized bed gasifiers with bubbling or circulating bed
3) entrained bed gasifiers with co-current flow

Fig. 10-1 shows the working principle of these various types. The gasifiers which are closest to large-scale commercial application for combined-cycles are of either the fixed-bed type (British Gas-Lurgi slagging gasifier) or the entrained-bed type (Texaco, Shell, Dow, Prenflow), both oxygen-blown.

Gasifiers can also be classified according to their operating pressure, i.e., whether they operate at atmospheric pressure or at an overpressure. Only the second of these types is of interest for combined-cycle plant applications because the gas pressure at the inlet to the gas turbine must be at least 20 bar (290 psia). For that reason, gasifiers for combined-cycle plants typically operate at 20 to 30 bar (275 to 420 psig). Table 10-2 shows the classification of the major gasification processes.

Coal gasification is particularly interesting from the point of view of emission control. It is far easier to obtain a very clean fuel by cleaning fuel gas than by cleaning the combustion gas from a coal-fired boiler. The main reasons for this are:

1) The volume flow of the coal gas to be treated is less than 1% that of the exhaust gas from the boiler.

2) It is chemically much easier to remove H_2S than SO_2, which makes it economically feasible to remove more than 99% of the sulphur, as compared to only 90% in the scrubber of a coal-fired boiler.

3) Most other pollutants, i.e., heavy metals, chlorides, etc., are also removed in the gasification process.

4) The major byproduct of the desulphurization process with coal gasification, except in a fluidized bed gasifier, is elemental sulphur, which is easy to transport and to sell. With other desulphurization processes, 5 to 10 times as much waste product is produced, since the sulphur is integrated into molecules of products like gypsum.

Table 10-2

MAJOR COAL GASIFICATION SYSTEMS

	British Gas/ Lurgi	Texaco	Shell	Prenflo	DOW	KRW	HT Winkler
Oxidant	O_2	O_2	O_2	O_2	O_2	O_2/air	O_2/air
Coal gas heating value	MBtu	MBtu	MBtu	MBtu	MBtu	MBtu/LBtu	MBtu/LBtu
Type of gasifier	Fixed bed	Entrained bed	Entrained bed	Entrained bed	Entrained bed	Fluidized bed	Fluidized bed
Flow pattern	Counter-current flow	Cocurrent flow	Cocurrent flow	Cocurrent flow	Cocurrent flow	Bubbling or circulating flow	Bubbling or circulating flow
Raw gas temperature [°C]	500 - 600	1400 - 1700	1400 - 1700	1400 - 1700	1400 - 1700	850 - 900	850 - 900
Gasifier pressure	Compatible with gas turbine requirements						
Raw gas cooling: - syngas cooler - quench	No Yes	Yes Yes	Yes Yes	Yes Yes	Yes Yes	Yes (Yes) 2)	Yes (Yes) 2)

Notes: 1) The syngas cooler is also referred to as a gasification heat recovery generator.
2) With fluidized bed gasifiers, no quenching is required if hot gas clean-up is provided.

Figure 10-1

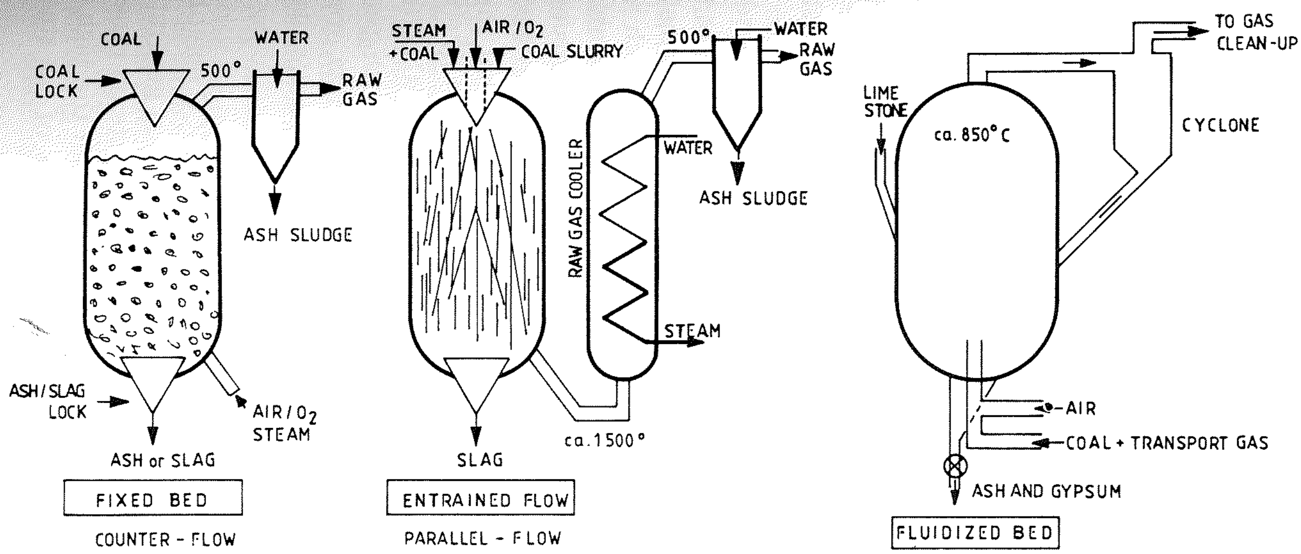

Fig. 10-1: Design Principles of the Main Types of Gasifiers

5) Most oxygen-blown gasifiers produce unleachable slag. Waste disposal is far easier than with the ash produced by coal-fired boilers.

For all these reasons, it is clear that coal gasification may well be very interesting for power generation in the future: it is the most promising method to produce a truly clean coal-burning plant.

Coal gasification can be either a system integrated into a combined-cycle plant, or a non-integrated system, with the gasification and combined-cycle plants quite distant from one another. Only the first of these two possibilities appears really interesting at present, since integration, with its good utilization of the waste heat generated in the gasification process, very greatly improves the overall efficiency of the entire system.

One major consideration when designing an integrated gasification combined-cycle plant (IGCC) is the temperature at which the gas is to be cleaned. All successful industrial systems being used today work at near ambient temperatures, which means that the coal gas must be cooled down after it leaves the gasifiers. If the gas was very hot, it is important that this energy be recovered efficiently. One of the main differences between fixed-bed gasifiers and entrained-bed gasifiers is that the gas temperature of the former is typically approx. 500 to 600 °C (900 - 1100 °F) and that of the latter approx. 1500 °C (2700 °F). This is why entrained-bed gasifiers are equipped with gas coolers to generate either saturated or superheated high pressure steam. This steam is then used in the combined-cycle plant to generate more power.

The example shown in Fig. 10-2 is a typical IGCC plant based on a Shell entrained bed gasifier unit. The steam generated in the gasifiers and that from the heat-recovery steam generator of the combined-cycle plant both flow through the high pressure section of the steam turbine. About 40% of the steam is

generated in the gasification plant, which demonstrates clearly the importance of integrating the gasification process into the combined-cycle plant.

The gas in the coal bed of a fixed-bed gasifier is cooled to a much lower temperature because of the countercurrent flow of the gas and the coal. Heat recovery after the gasifier is therefore far less important, and such plants therefore usually generate only a very limited amount of low pressure steam. Fig. 10-3 shows an example of a coal-fired combined-cycle plant using British Gas-Lurgi slagging gasifiers. It can be seen quite clearly that the degree of integration is far less important here than with the Shell gasifier.

The systems used to clean the gas are quite similar in both cases. One difference is that the gas produced in a fixed-bed gasifier contains tars and phenol which must be separated out and recycled to the gasifier. This is not true with an entrained-bed gasifier because of the much higher process temperature which cracks the heavy hydrocarbons.

Table 10-3 provides the main technical data of the IGCC plant shown in Fig. 10-2, using a 150 MW ABB gas turbine, with an efficiency of 34% under ISO conditions.

Table 10-3: Main Technical Data of IGCC Plants

- Type of gasifier	entrained bed	
- Coal	Illinois No. 6	
- Calorific value of the coal gas	12 500 kJ/kg (290 Btu/scf)	
- Desulphurization rate	99 %	
- NO_x emissions	◄ 135 mg/GJ (75 vppm)	
- Coal input (LHV)	577	MW
- Gas turbine output	173	MW
- Steam turbine output	110	MW
- Total consumption for auxiliaries	33	MW
- Net output	250	MW
- Net efficiency (LHV)	43.3	%

These figures show the efficiency that can be expected from an IGCC plant employing commercially proven components in 1988. To reduce the NO_X emissions to the low level indicated for these two plants, the fuel gas is mixed with water vapor. The costs for such a plant are in approximately the same range as those for modern coal-fired units with scrubbers and de-NO_X systems.

Two fairly large IGCC plants already built demonstrate that this is a proven technology. The first of these is the 100 MW Coolwater plant in California, a demonstration plant that has been operating since 1984 with a Texaco gasifier. Although this plant has an efficiency of only 31.3%, the main purpose in building it was not to achieve the highest efficiency possible, but to show that the integration of coal gasification and a combined-cycle plant does actually work. The second of these is at the Dow chemical plant in Plaquemine, LA, with a total output of approx. 160 MW.

One very interesting feature of IGCC plants is their potential for phased or staged construction, i.e., an IGCC plant can be completed in two or three steps:

Step 1: the gas turbine portion
Step 2: conversion to a combined-cycle plant
Step 3: coal gasification

The advantage of this method for building an IGCC plant is, of course, the relatively small capital requirement for the first two steps. Larger investments do not have to be made until Step 3, when economical and operation considerations require conversion of the gas-fired plant into a coal-fired unit.

It can be expected that the efficiencies of IGCC plants in the year 2000 will be between 46 and 48% (LHV) due to the following improvements:

- gas turbines with higher inlet temperatures
- dry low NO_X combustors
- hot gas clean-up

The introduction of cleaning systems operating at 400 to 500 °C (752 to 932 °F) would improve the outlook for air-blown gasifiers. At present, this type of gasifier is at a disadvantage because of the larger exergetic and energetic losses that occur during cooling of the gas prior to the gas treatment. The gas flow is approximately three times as great as in the case of oxygen-blown gasifiers. As soon as the fuel gas only needs to be cooled down to 400 to 500 °C, that disadvantage will be greatly reduced. Air-blown gasification systems obviously have the advantage that no air separation plant is needed.

Still another field for possible application of coal gasification could be the cogeneration of power and chemical raw materials in the same plant (e.g., power and methanol).

10.2.2 Pressurized Fluidized Bed Combustion Systems (PFBC)

The idea behind pressurized fluidized bed combustion systems (PFBC) is quite different from that on which the IGCC is based. In coal gasification, one will want to burn coal cleanly, utilizing the advantages of the gas-fired combined-cycle plant. Essentially, this means using gas turbines with a high turbine inlet temperature. In contrast to this, PFBC takes its point of departure from conventional steam power plants.

As everyone knows, subsequent flue gas desulphurization and denitrification is very involved and expensive. For that reason, in fluidized beds, both atmospheric and pressurized, desulphurization is accomplished by reaction of the sulphur with limestone during the combustion. Since the temperature must not exceed approx. 900 °C (1652 °F), relatively little NO_X is produced as well.

Figure 10-2

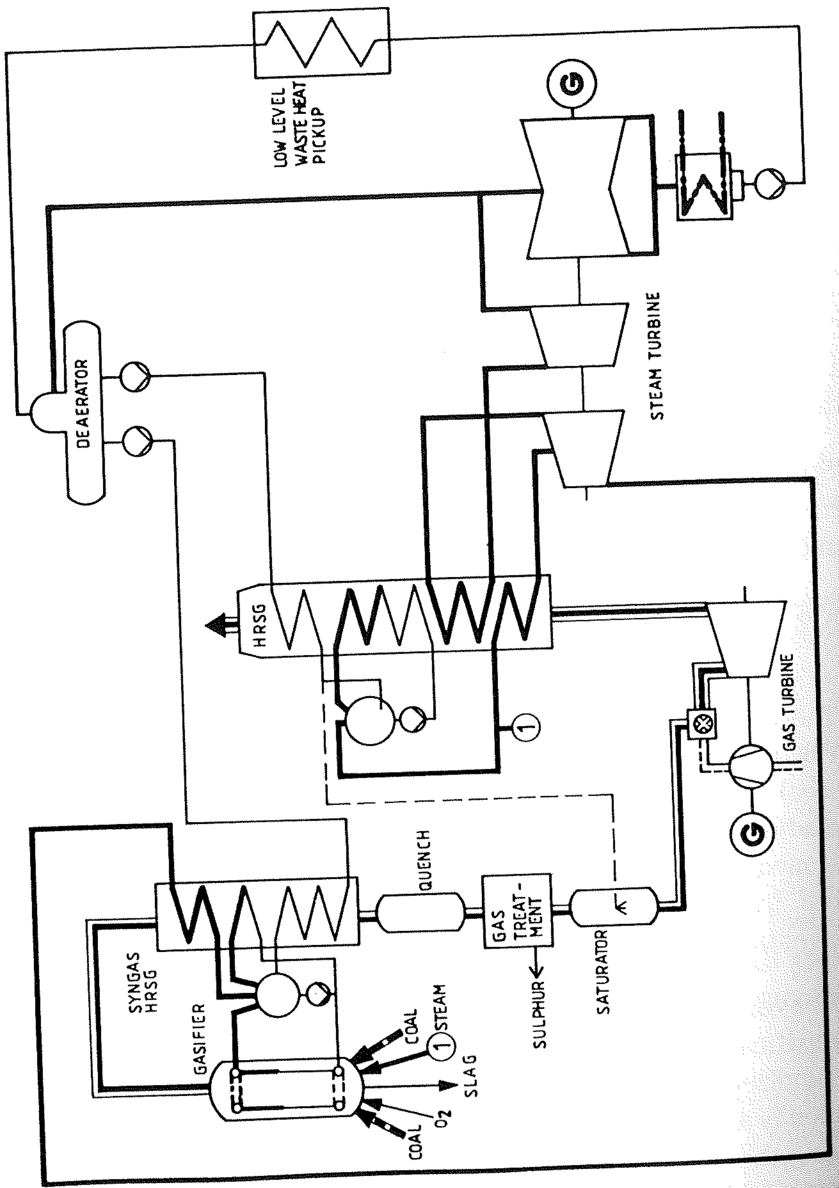

Fig. 10-2: IGCC Plant Based on an Entrained Bed Gasifier (Shell)

Figure 10-3

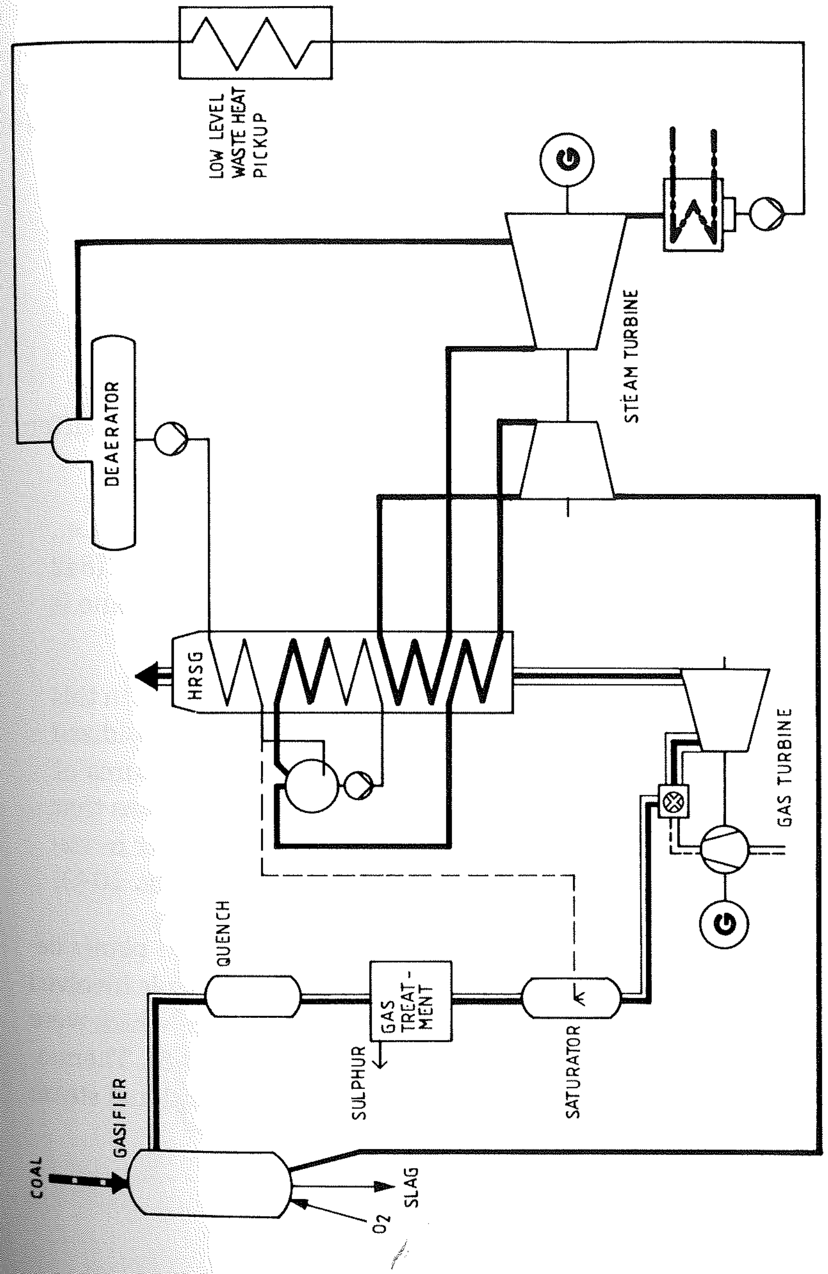

Fig. 10-3: ISCC Plant Based on a Fixed Bed Gasifier (BGC-Lurgi)

This process is presently being used in many boilers with fluidized beds that operate at atmospheric pressure. Because the maximum power density permissible is quite low, the dimensions of fluidized beds of this type become very large, even for low steam generation outputs. This quickly gave birth to the idea of operating the entire process at an overpressure, which provides the following advantages:

- The dimensions of the fluidized bed and the steam generator are reduced approximately proportional to the pressure.
- The efficiency of the overall plant can be raised by using a gas turbine as the charging unit.

At present there are two different types of PFBC plant, differing in the inlet temperature to the gas turbine:

1) PFBC plants with a low temperature gas turbine, i.e., the turbine inlet temperature is approx. 450 - 500 °C (842 - 932 °F), or just high enough so that the gas turbine is capable of driving a compressor (Fig. 10-4).

2) PFBC plants with a high temperature gas turbine. In this case, the turbine is located directly after the fluidized bed and is driven by exhaust gas with temperatures of approx. 800 to 850 °C (1572 - 1662 °F). This means that the turbine produces a greater amount of power in excess of that needed to drive the compressor (Fig. 10-5).

Plants of the first type are not true combined-cycle plants because the gas turbine is only a charging unit. The principle involved is exactly the same as in power plants of 50 years ago which were equipped with BBC Velox boilers (refer to Section 3.5). Thermodynamically, such a plant is equivalent to a conventional steam turbine power plant.

Figure 10-4

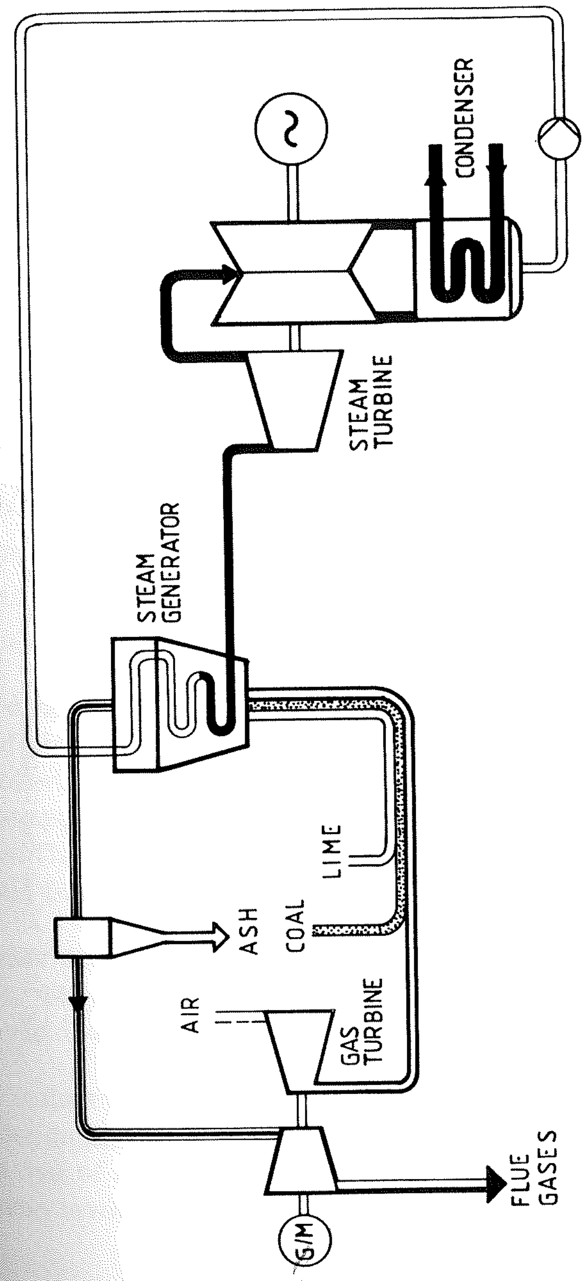

Fig. 10-4: Pressurized Fluidized Bed Plant with a Low Temperature Turbine

Figure 10-5

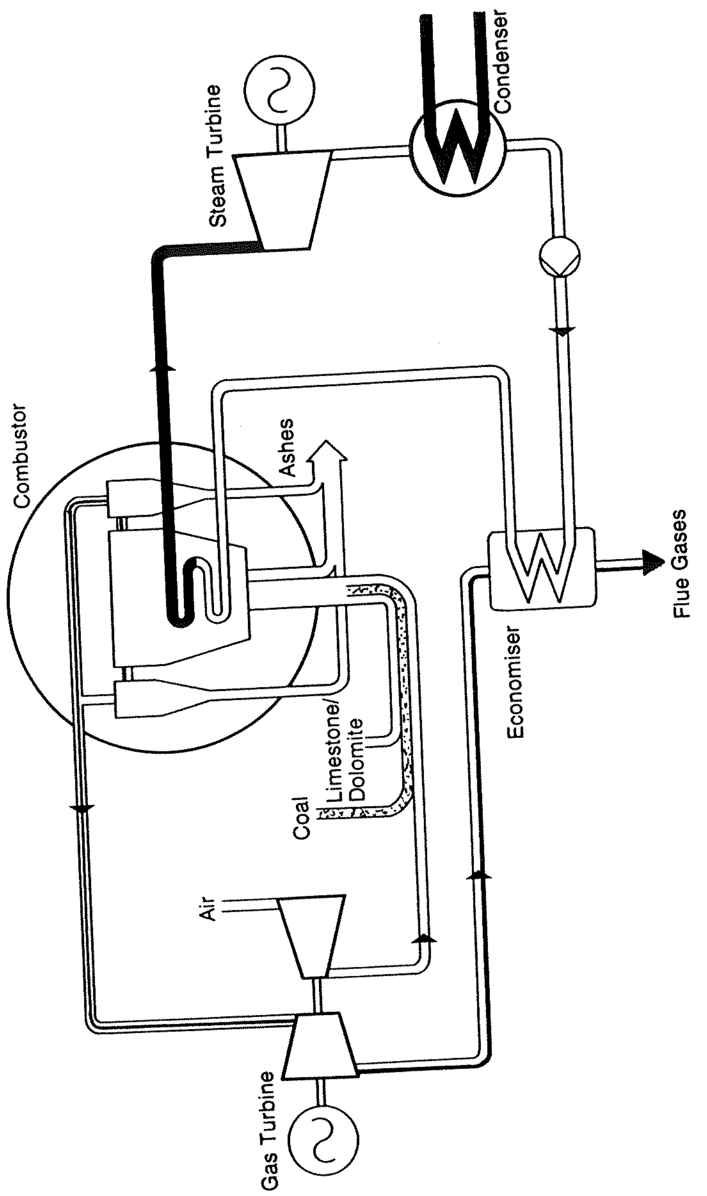

Fig. 10-5: Pressurized Fluidized Bed Plant with a High Temperature Turbine

The principal advantage of this type of plant may be the relative ease with which dust may be removed from the flue gas, which, at temperatures of 450 to 500 °C (842 - 932 °F), is not likely to be too problematic.

On the other hand, PFBC plants with a high temperature gas turbine attain efficiencies of 43 to 44%, i.e., significantly above those of the steam power plant. The major technical problem with this type of plant is obtaining effective filtration of the flue gas at a temperature level of 850 to 900 °C (1572 to 1662 °F).

More information about this will become available in the early 90's when operating experience with the first PFBC plants from Asea (the present Asea Brown Boveri) has been gathered and analyzed. The main technical data of this type of plant are as follows:

• net power output	80 MW
• gas turbine output	17 MW
• steam turbine output	67 MW
• net efficiency	42.5 %

The diagram in Fig. 10-5 shows the principle of operation of this type of plant; Fig. 10-6 shows details of the pressurized fluidized bed boiler. The flue gas is cleaned in 2-stage cyclones, since no high temperature filters are yet available for commercial applications. This means that the flue gas must then be cleaned in electrostatic filters after the economizer.

Further development of such plants may attain efficiencies of approx. 44 to 45% by using larger units and steam cycles with improved live steam data.

The main advantages of the PFBC plant are assuredly its compact design and its potential efficiency, which is from 5 to 10% above that of a conventional steam power plant. Disadvantages, however, involve the necessity of flue gas scrubbing at high temperatures and the fact that the gas turbine is of a special design which cannot fully utilize the high temperature potential available with modern gas turbines because of restrictions on the temperature within the fluidized bed. Then too— unlike in plants with a low temperature gas turbine, the economizer in a plant with a high temperature gas turbine operates at atmospheric pressure because it is situated after the gas turbine. This affects the compactness of the plant.

Suggestions for using the full temperature potential of the gas turbine do, of course, exist, but whether or not they can be realized commercially is an open question, since they make the plant more complicated and problematic. These are based on partially gasifying the coal or directing it through a pyrolysis before supplying it to the fluidized bed. The gas produced thereby is burned following the fluidized bed to bring the gas turbine inlet temperature to the levels usual for normal turbines. Overall efficiencies of 48 to 50% should be attainable with such systems.

How the PFBC process might make a breakthrough on the market in competition to IGCC and conventional steam power plants is a question that remains to be answered. If success is attained in the first four large-scale plants currently under construction, the prospects for the process could well be very good.

In addition to these two types of installations with gas turbines, there are also suggestions for using a hot air turbine. This system has been shown in Section 3.4 with a closed-cycle gas turbine. Similar systems employing an open air-process have also been suggested.

In both cases, the air is used as the medium to cool the fluidized bed. It thereafter expands in the turbine before being used as combustion air in the open process.

The advantage of these arrangements is that the turbine is operating on a clean medium. Their disadvantage, however, is the large surface area required for the atmospheric fluidized bed. A further problem lies in selection of the material for the fluidized bed cooling tubes, which are not as efficiently cooled. The heat transfer coefficient of air is lower than that of water or steam.

10.3 Dry Low NO_X Combustors

As was shown in Section 9.1, the methods commonly used today to reduce NO_X emissions are indeed effective, but they entail two disadvantages:

- high water consumption
- a reduction in the efficiency of the combined-cycle plant

Particularly in the case of base-load installations, this impacts negatively on plant economy. Moreover, the high rate of water consumption is not unproblematic from an environmental point of view. For these reasons, various development projects are currently in progress, directed toward reducing NO_X emissions by changes in the combustion technology. Theoretically, there are two paths that such development can take:

a) oxygen-lean combustion ($\lambda < 1$)

b) oxygen-rich combustion ($\lambda > 1$)

Fig. 9-3 shows the dependence of the NO_X concentrations on the excess air ratio. Only a small amount of NO_X can form in an oxygen-lean combustion, despite the high flame temperature, because there is scarcely any oxygen available for that to happen. However, in order to attain a complete combustion, there must be a second, follow-up combustion stage in which there is almost no NO_X formed due to the lower temperature. This approach is being used in modern steam generators and is referred to as "staged combustion."

For gas turbines, however, because of the high overall excess air ratio (λ = 3 to 3.5), a different and more effective procedure is being applied: combustion with excess air. This functions on the same principle as reduction by means of steam or water injection: the large amount of excess air effectively cools the flame. The procedure is subject to limits due to considerations of flame stability. With excess air ratios λ of approx. 3, the combustion becomes very poor and the flame is completely extinguished.

This represents no problem while the gas turbine is at full load, because there is not enough air available for the burner anyway for the excess air ratio to exceed 2.0 significantly. The remaining air is required to cool the hot parts and the turbine blading. It is, however, a major problem at part loads.

For the combustion actually to take place at the desired excess air ratio, the air and fuel must be mixed homogenously with one another. For that reason, the burners used with this procedure are referred to as "premix" burners. Fig. 10-7 shows a typical example. The air and the natural gas are premixed in a tube and are burned downstream from the swirl basket, which is located at the end of the premixing tube and is used to hold the flame. The basic problems involved with this type of burner can also be seen immediately from this example:

- It is very difficult to burn liquid fuel, since mixing becomes problematic.
- There is a risk of back-flash, i.e., that the combustion will take place even in the premixing zone.

This sytem operates very well with natural gas, provided that the air temperature is not too high, which means that the pressure ratio of the gas turbine must be less than approx. 15 to 16.

With gases with a higher flame propagation speed, e.g., hydrogen (H_2) or carbon monoxide (CO), this procedure can scarcely

Figure 10-6

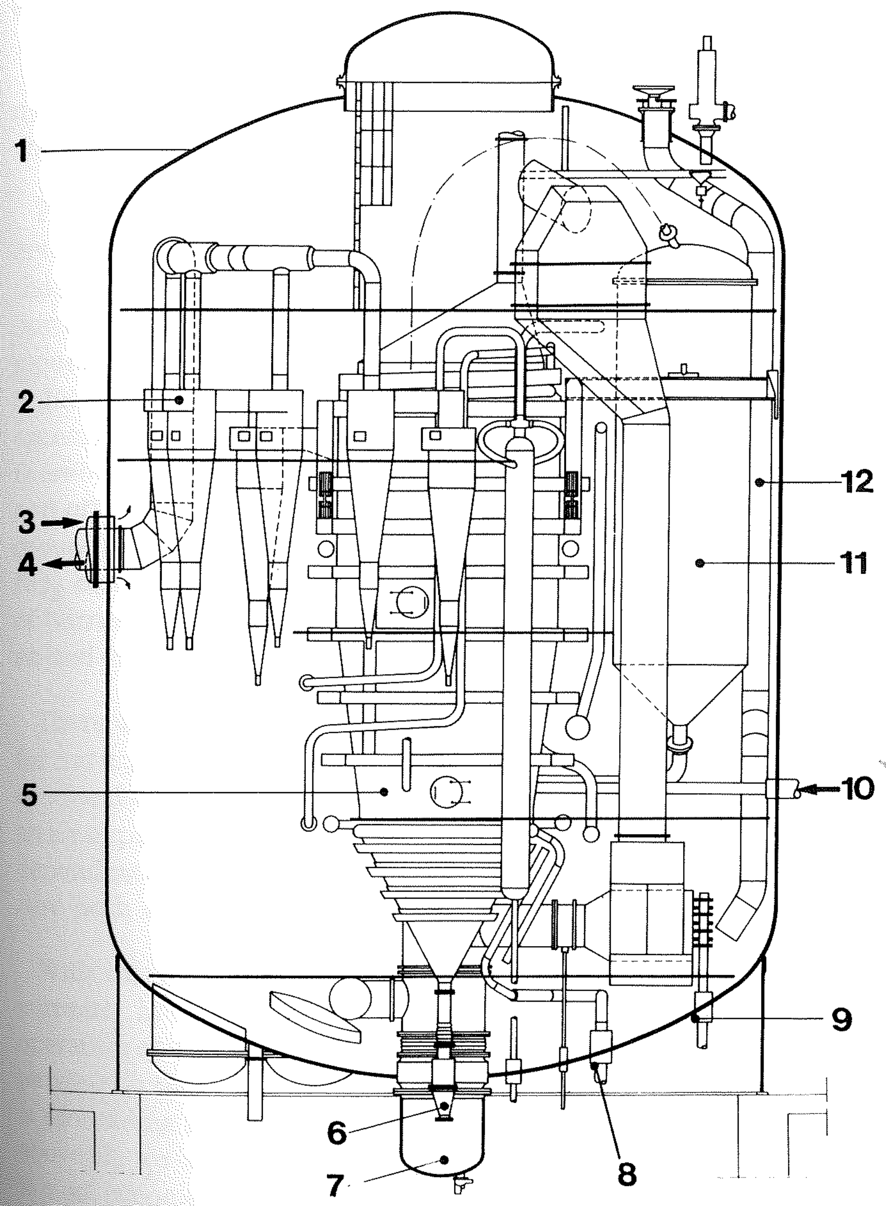

Fig. 10-6: ABB Pressurized Fluidized Bed Boiler

Figure 10-7

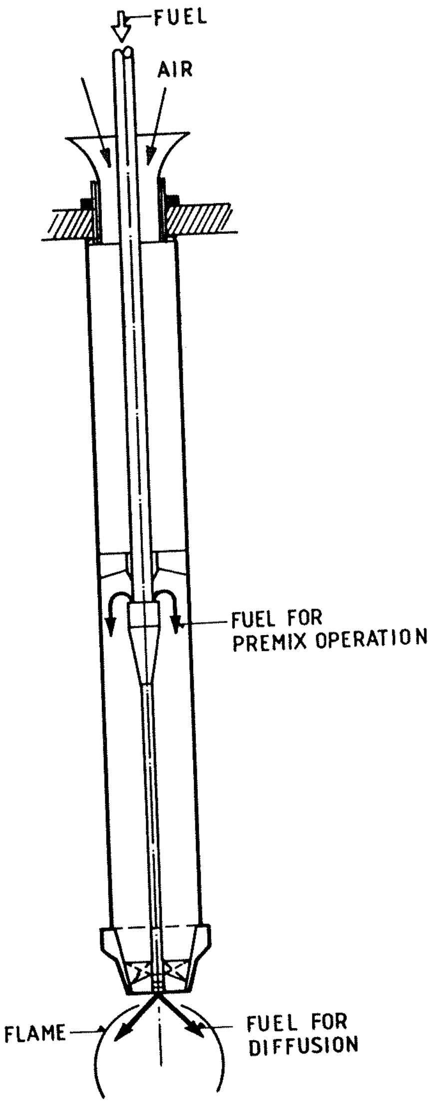

Fig. 10-7: Premix Burner

be employed due to that factor. This type of burner has scarcely ever worked well with liquid fuels.

There are a few gas turbines from various manufacturers in commercial operation that have dry low NO_x combustors of this type, operating on gas only.

Fig. 10-8 shows a combustor of this type equipped with 36 burners similar to that in Fig. 10-7; Fig. 10-8 shows a photograph of the burners viewed from below, with the swirl baskets of the 36 burners that are used to contain the flames. This combustor is mounted on a 70 MW gas turbine. 36 burners were selected in order to solve the problem of part-load operation. To prevent the excess air ratio in the burners from becoming too high at part loads, thereby extinguishing the flame, the burners can be switched on and off in groups. Fig. 10-9 shows the loads at which the various groups are switched on or off, and the corresponding changes in NO_x concentrations in the exhausts. In all, there are five groups of burners in this combustor: Fig. 10-10 shows how they are connected on the fuel end and controlled with a single control valve.

There are other solutions to the problems at part load, such as, for example, a more or less progressive switch-over from premix combustion to normal diffusion combustion, or the installation of an air bypass on the burner which allows more or less air to escape depending on the load involved. This also makes it possible to assure both a high excess air ratio and a stable flame.

With this combustion technology, there are large gas turbines currently in operation with NO_x emissions ranging, in gas-fired operation at full load, between 25 and 75 vppm at 15% O_2, depending on the design. Further development of this type of burner is proceeding in the direction of even lower emission levels and, above all, toward greater fuel flexibility. Of particular importance, it must become possible in the future to burn liquid fuels and coal gases that are rich in H_2 or CO.

So-called "surface burners" represent another path toward NO_X reduction in which use is made of a different parameter to achieve low NO_X emission levels, viz. time. The placing of several very small flames over a large surface area makes the residence time within the flame very short, leaving little time for NO_X to form. The major problem in this case is keeping the residence time as short as possible while still maintaining a good combustion that does not produce large amounts of carbon monoxide (CO) and unburned hydrocarbons (UHC).

Fig. 10-11 is a typical annular combustor that operates on this principle, used as equipment for a gas-fired 45 MW turbine. One can see the swirl baskets used to hold the small flames. In all, there are 200 swirl baskets installed on the ring. Theoretically, this combustor could also be fired on oil. NO_X emission levels in the exhausts at full load are in the range from approx. 50 to 80 vppm.

Figure 10-8

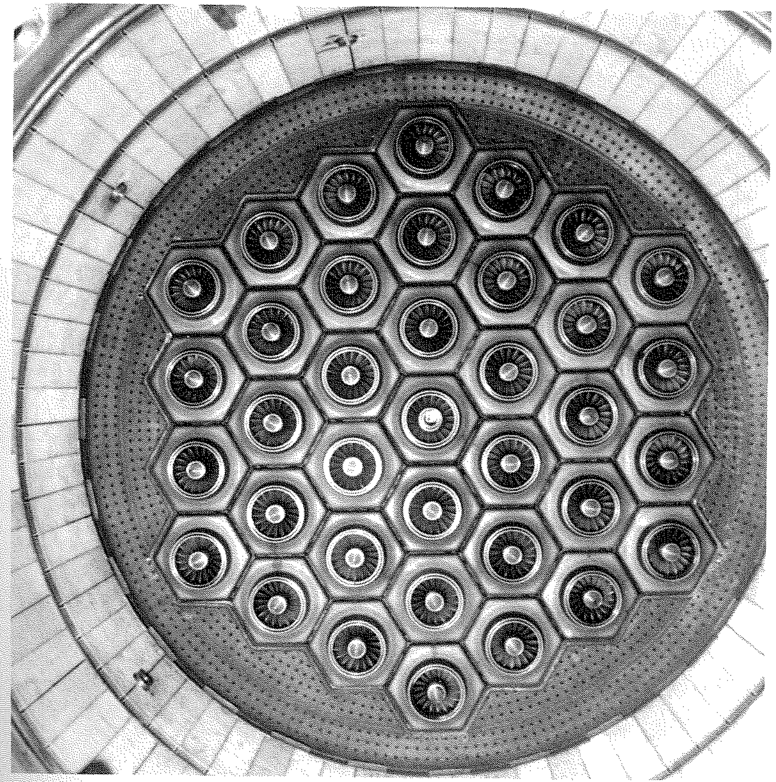

Fig. 10-8: Dry Low NOx Combustor: Principle

Figure 10-9

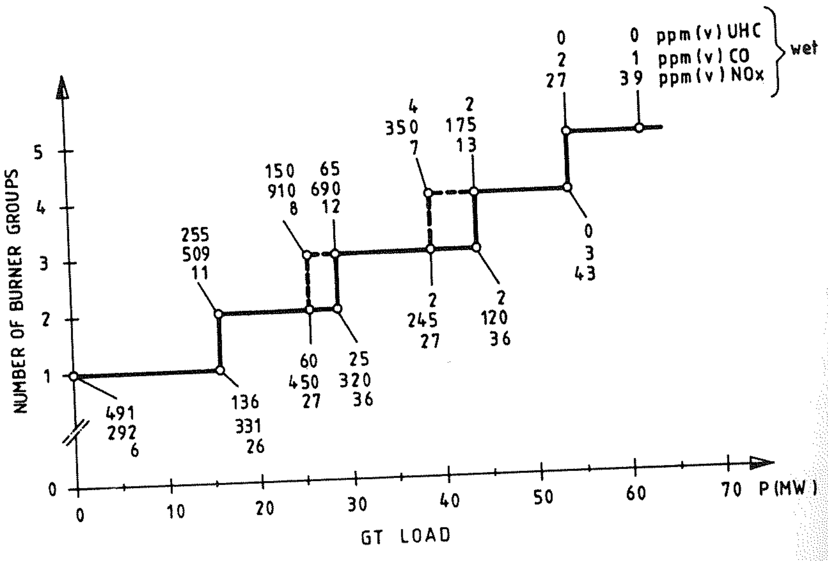

Fig. 10-9: Dry Low NOx Combustor: Dependence of the NOx Concentration on Gas Turbine Load

Figure 10-10

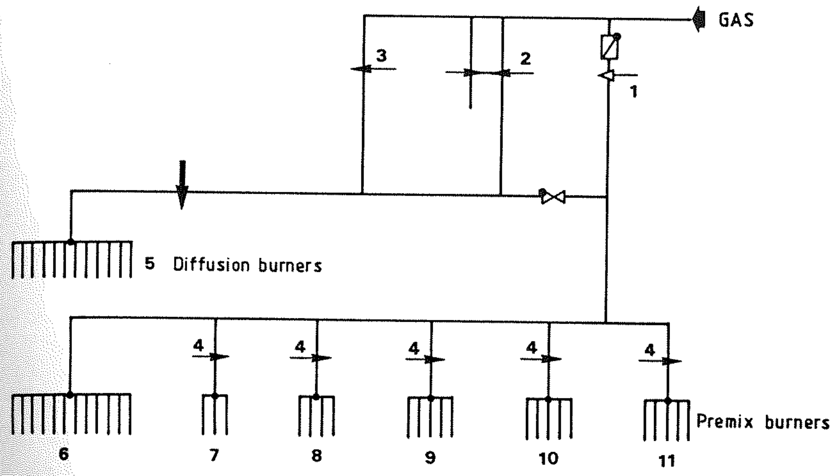

Fig. 10-10: Dry Low NOx Combustor: Principle of Fuel Flow Control and Grouping of Burners

Figure 10-11

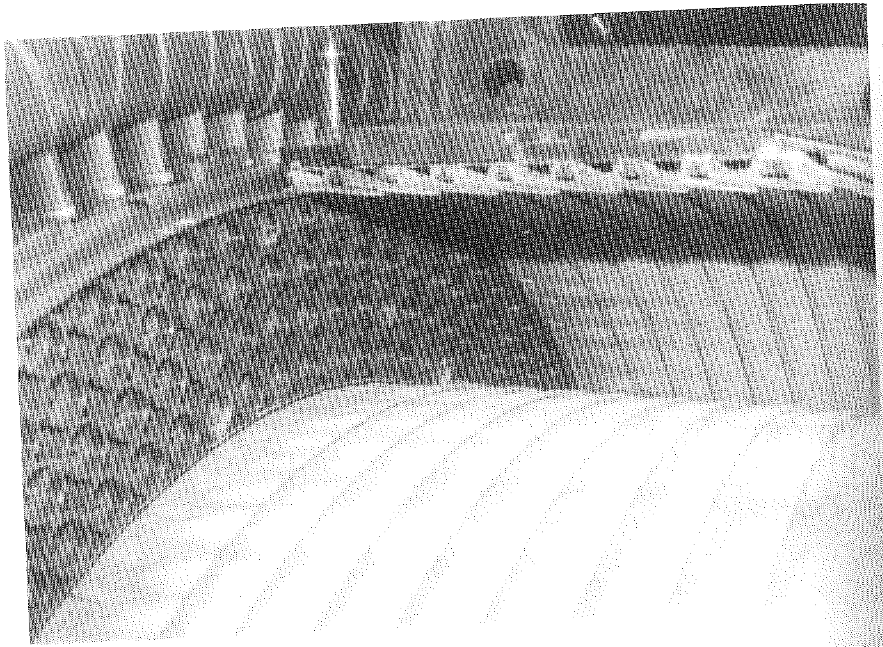

Fig. 10-11: Annular Combustor with Swirl Basket Burners

Chapter 11

SOME TYPICAL COMBINED-CYCLE PLANTS ALREADY BUILT

11.1 Combined-Cycle Plants without Supplementary Firing

11.1.1 The 120 MW Combined-Cycle Plant of the NEWAG at Korneuburg (Austria)

This gas-burning combined-cycle plant began operation in 1980. Fig. 11-1 shows the principle on which it was built. It consists of an 80 MW gas turbine and a 45 MW 2-pressure steam turbine. Because very good use is made of the waste heat from the gas turbine, a net efficiency of 46.6% is attained in base-load operation. Table 11-1 shows the main technical data.

The unit was constructed within an existing machine building to replace an older combined-cycle plant. Fig. 11-2 shows the layout, Fig. 11-3 the machine building with the gas and steam turbines.

This combined-cycle plant is being used to cover medium loads and is started and shut-down daily. It therefore has fully automatic controls which, after 14 hours at standstill, bring the plant up to full load again within approx. 26 minutes.

Because operation of the gas turbine alone scarcely enters into consideration, the plant was built without a flue gas bypass. However, the waste heat boiler is designed to permit operation of the gas turbine at full load even while the boiler itself is dry.

Figure 11-1

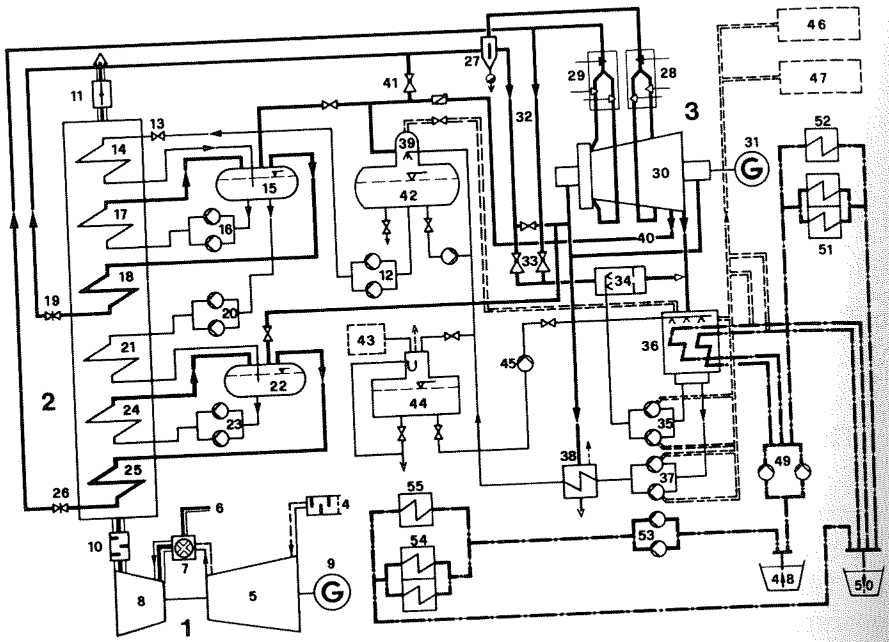

Fig. 11-1: Design Principle used for the 120 MW Combined-Cycle Plant at Korneuburg

1 Gas turboset
2 Waste heat boiler
3 Steam turboset
4 Air intake filter
5 Turbocompressor
6 Fuel supply line
7 Combustor
8 Gas turbine
9 Gas turbine generator
10 Exhaust sound damper
11 Stack rain damper
12 Low pressure feedwater pumps
13 Low pressure feedwater control valve
14 Low pressure economizer
15 Low pressure drum
16 Low pressure circulating pump
17 Low pressure evaporator
18 Low pressure superheater
19 Low pressure steam line
20 High pressure feedwater pumps
21 High pressure resuperator
22 High pressure drum
23 High pressure circulating pump
24 High pressure evaporator
25 High pressure superheater
26 High pressure steam line
27 Low pressure water separator
28 Low pressure trip valve
29 High pressure trip valve
30 Steam turbine
31 Steam turbine generator
32 High pressure and low pressure bypass
33 Bypass control valves
34 Injection steam heater
35 Injection water pump
36 Condenser
37 Condensate pumps
38 Vapor condenser
39 Deaerator/mixing preheater
40 Extraction steam line
41 Support steam line
42 Feedwater tank
43 Boiler water treatment
44 Boiler water tank
45 Replacement water pump
46 Starting vacuum pump
47 Operating vacuum ejectors
48 River water intake for Cooling
49 Main cooling water pumps
50 River water return
51 Air cooler, steam turbine generator
52 Steam turbine oil cooler
53 Gas turbine cooling water pumps
54 Air cooler, gas turbine generator
55 Gas turbine oil cooler

Figure 11-2

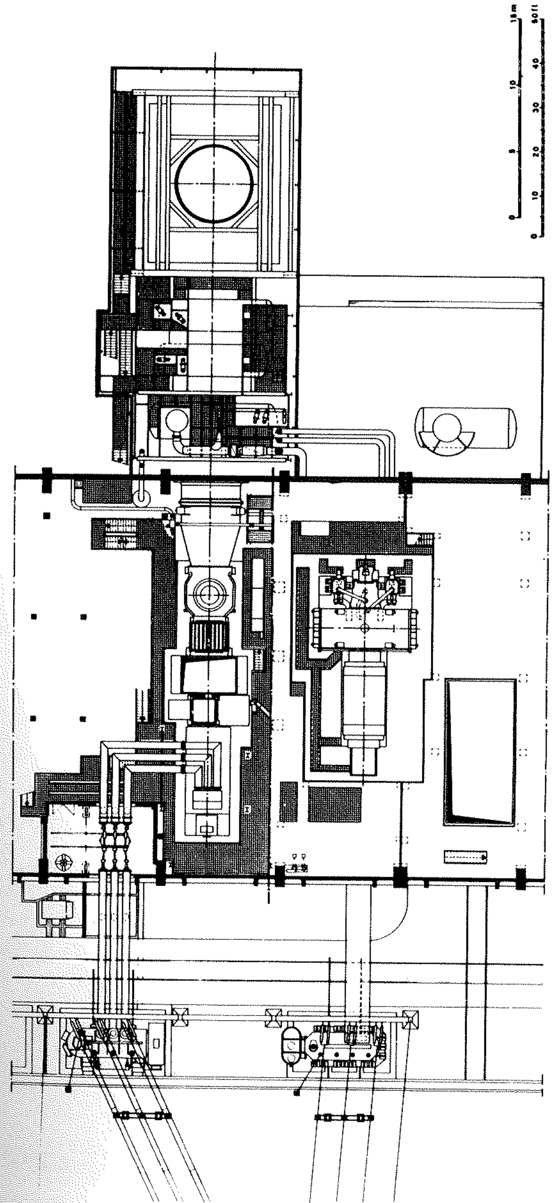

Fig. 11-2: Layout of the 120 MW Combined-Cycle Plant at Korneuburg

Figure 11-3

Fig. 11-3: Machine room of the 120 MW Combined-cycle Plant at Korneuburg

Table 11-1: Main Technical Data of the 120 MW Combined-Cycle Plant at Korneuburg

Fuel	Natural gas	
Air temperature	10	°C
Cooling water temperature	8	°C
Gas turbine power output at base load	81.1	MW
Steam turbine power output	48.7	MW
Station service power	1.0	MW
Net power output	128.8	MW
Net efficiency	46.6	%
Condenser pressure	0.036	bar
Stack temperature	95	°C
Low pressure feedwater temperature	58	°C
Gas turbine exhaust gas flow	363	kg/s
Gas turbine exhaust gas temperature	491	°C
Gas turbine inlet temperature	945	°C
High pressure live steam data	33	bar
	438	°C
High pressure live steam flow	43	kg/s
Low pressure live steam data	4.4	bar
	182	°C
Low pressure live steam flow	7.9	kg/s

11.1.2 The Tunghsiao Combined-Cycle Plant of the TPC, Taiwan [26]

The Tunghsiao combined-cycle plant consists of three blocks, each with a power capacity of approx. 300 MW. The unit described in detail here is the third of these blocks, but blocks one and two are similar in construction. The entire plant has been in operation since 1982. Fig. 11-4 shows the principle of the structure of the installation, which consists of three 60 MW gas turbines and a 95 MW single-pressure steam turbine and is capable of burning either residual or diesel fuel. Economic consideratons lend special interest to the burning of residual. The oil is washed and dosed with additives for combustion so as to prevent high temperature corrosion. Table 11-2 contains the main technical data.

The gas turbines and the waste heat boilers are built outdoors, the gas turbines in a light enclosure (Fig. 11-5). On the other hand, the steam turbine is accommodated in a conventional machine building.

Table 11-2: Main Technical Data for the Tunghsiao Combined-Cycle Plant, Unit 3

Fuel		Residual oil	
Air temperature		32.2	°C
Cooling water temperature		29.4	°C
Gas turbine power output at base load	3 x	60.1	MW
Steam turbine power output		95.3	MW
Station service power		3.9	MW
Net power output of the plant		271.7	MW
Net efficiency of the plant (LHV)		44.4	%
Condenser pressure		0.09	bar
Stack temperature		171	°C
Feedwater temperature		145	°C
Gas turbine exhaust gas flow		262	kg/s
Gas turbine exhaust gas temperature		523	°C
Gas turbine inlet temperature		1000	°C
Live steam flow	3 x	32.2	kg/s
Live steam data (at base load)		52	bar
		474	°C
Low pressure steam flow	3 x	6.8	kg/s

Figure 11-4

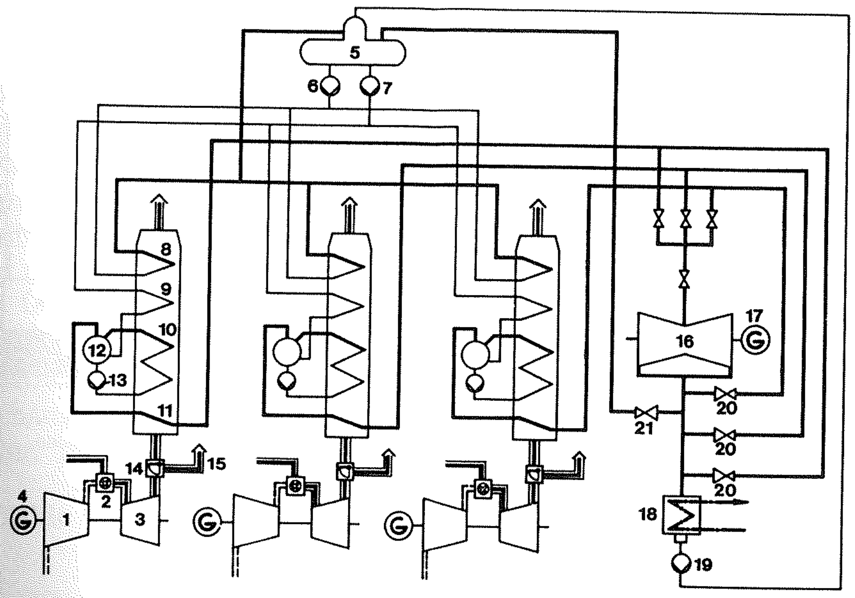

Fig. 11-4: Design Principle of the Tunghsiao Combined-Cycle Plant (Unit 3)

1	Compressor	11	Superheater
2	Combustor	12	High pressure drum
3	Gas turbine	13	High pressure circulating pump
4	Generator	14	Bypass flap valves
5	Feedwater tank/deaerator	15	Bypass stack
6	Low pressure feed pumps	16	Steam turbine
7	High pressure feed pumps	17	Generator
8	Low pressure evaporator	18	Condenser
9	Economizer	19	Condensate pumps
10	High pressure evaporator	20	Steam bypasses

Figure 11-5

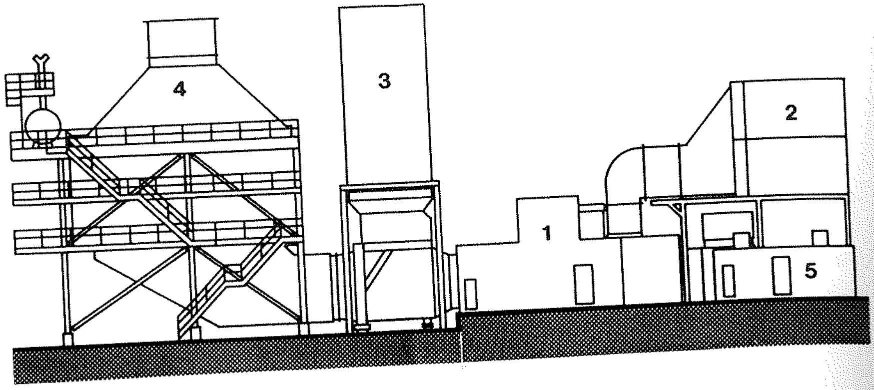

Fig. 11-5: Layout of the Gas Turbines and Waste Heat Boilers at the Tunghsiao Combined-Cycle Power Plant (Unit 3)

1 Gas turbine
2 Control block
3 Bypass
4 Waste heat boiler

11.1.3 The 1200 MW Trakya Combined-Cycle Plant TEK, Turkey [111]

The Trakya combined-cycle plant of this Turkish utility comprises four 300 MW units, each having two gas turbines and one steam turbine. The gas turbines are rated at 95 MW under site conditions. The steam turbine, with a single casing and two low pressure exhausts, is rated at 110 MW, Fig. 11-6 shows the cycle of the plant, which is a typical dual pressure cycle.

One interesting feature of this plant is that the condenser is cooled with a dry cooling tower built on the Hungarian Heller principle. The condenser is therefore not a surface heat exchanger: the steam is condensed by mixing it with cooling water. Table 11-3 shows the main technical data.

Table 11-3: Main Technical Data for one 300 MW unit of the 1200 MW Trakya Combined-Cycle Plant

		Base load	Peak load	
Air temperature	:	15	15	°C
Fuel	:	natural gas		
Gas turbine output	:	2 x 97.5	2 x 104.5	MW
Steam turbine output	:	105	118	MW
Auxiliary power	:	3.6	3.7	MW
Net total power output	:	296.4	323.3	MW
Net efficiency (LHV)	:	49	50	%
High pressure steam flow	:	86.5	93	kg/sec
High pressure steam temperature	:	477	498	°C
High pressure steam pressure	:	49.4	53.9	bar
Low pressure steam flow	:	20.5	20	kg/sec
Low pressure steam temperature	:	197	197	°C
Low pressure steam pressure	:	4.9	5.2	bar
Stack temperature	:	98	99	°C
Feedwater temperature	:	51	51	°C

11.1.4 The 1090 MW Combined-Cycle Power Plant at Higashi-Nigata, Japan

The 1090 MW Higashi-Nigata combined-cycle plant owned by Tohuku Electric comprises two blocks, each with three 133 MW gas turbines and one 196 MW steam turbine. The plant is fired on vaporized liquid natural gas (LNG) and is operated as a base-load unit.

The plant has a dual pressure steam cycle and its condenser is cooled directly with sea water.

NO_X emissions are less than 15 vppm. In order to attain this low a level, the gas turbines have been equipped with dry low NO_X combustors which reduce the NO_X concentration to approx. 75 vppm. The emissions are then further reduced in the selective catalytic reduction (SCR) system installed in the heat recovery steam generator. Table 11-4 shows the main technical data of this plant.

Table 11-4: Main Technical Data of One Unit of the 1090 MW Higashi-NIgata Combined Cycle Power Plant

Fuel	:	LNG	
Air temperature	:	– 1	°C
Cooling water temperature	:	+ 18	°C
Gas turbine power output	:	133	MW
Steam turbine power output	:	195.5	MW
Station service power	:	19.9	MW
Power output of the plant	:	1090	MW
Efficiency of the plant (LHV)	:	48.5	%

High pressure live steam flow	:	150	kg/sec
High pressure live steam data	:	64.7	bar
		500	°C
Low pressure live steam flow	:	44.1	kg/sec
Low pressure live steam data	:	5.9	bar
		saturated	
Condenser	:	0.043	bar
Stack temperature	:	109	°C

Fig. 11-7 shows the general layout of the power plant and Fig. 11-8 a view of it.

11.2 Combined-Cycle Plants with Supplementary Firing

11.2.1 Lausward Power Station of the Stadtwerke Duesseldorf AG, Germany

This 420 MW combined-cycle block consists of a 60 MW gas turbine, a 70 MW gas turbine, and a 300 MW reheat steam turbine. It started into operation between 1974 and 1976. The gas turbines burn natural gas or light heating oil, and the steam generators natural gas, heavy, or light heating oil, individually or in any mixture of fuels desired.

Fig. 11-9 and 11-10 show the structural design of the plant, which can be started up and shut-down fully automatically. Each gas turbine has a 100% capacity flue gas bypass, making single-cycle operation possible. At any given time, one of the two gas turbines takes over combined-cycle operation along with the steam generator while the other is available for solo operation in the upper medium-load range or as a standby unit. The exhaust gas heat can be supplied to a district heating system heat exchanger. But it is also possible to operate the steam process by itself since the steam generator is equipped with a fresh air fan. Table 11-5 shows the most important data and Fig. 11-11a and b the general layout.

Figure 11-6

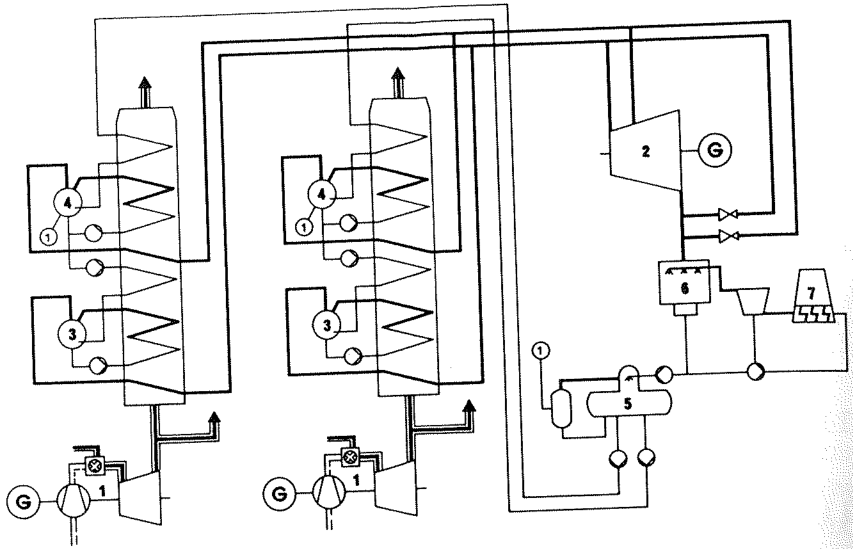

Fig. 11-6: Design Principle of the Trakya Combined-Cycle Plant (1 Unit)

1 Gas turbine
2 Steam turbine
3 HP drum
4 LP drum
5 Deaerator/feedwater tank
6 Mixing Condenser
7 Cooling tower

Figure 11-7

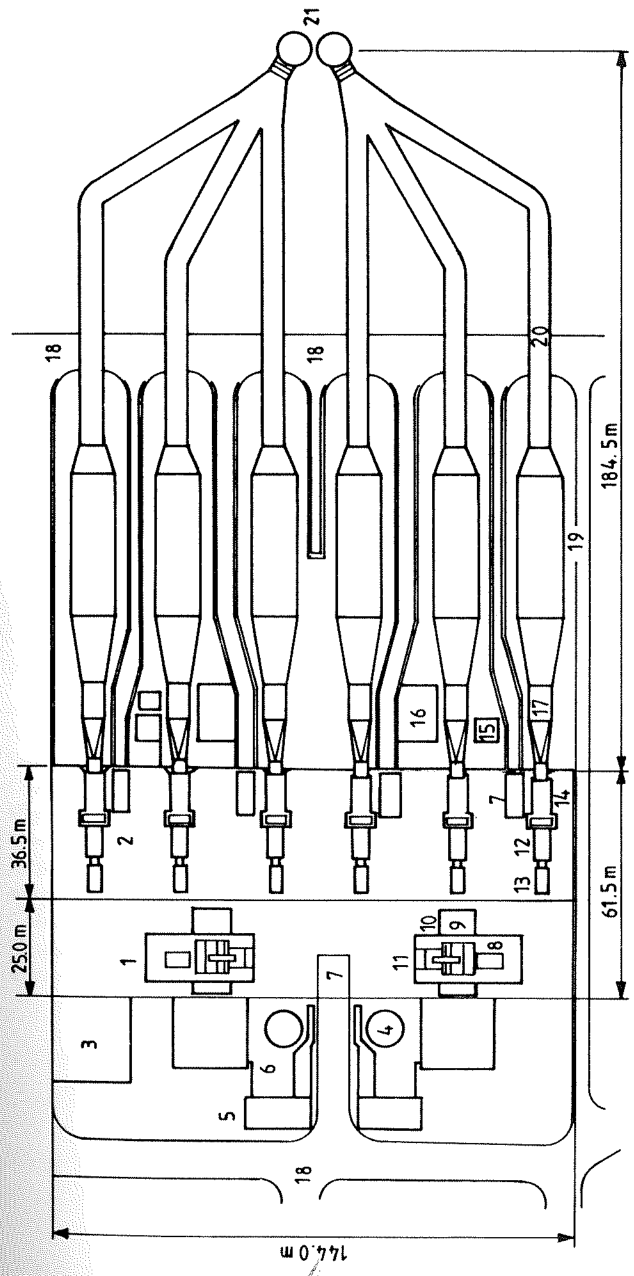

Fig. 11-7: Layout of the 1090 MW Higashi-Nigata Combined-Cycle Plant

Figure 11-8

Fig. 11-8: View of the Higashi-Nigata Combined-Cycle Plant

Figure 11-9

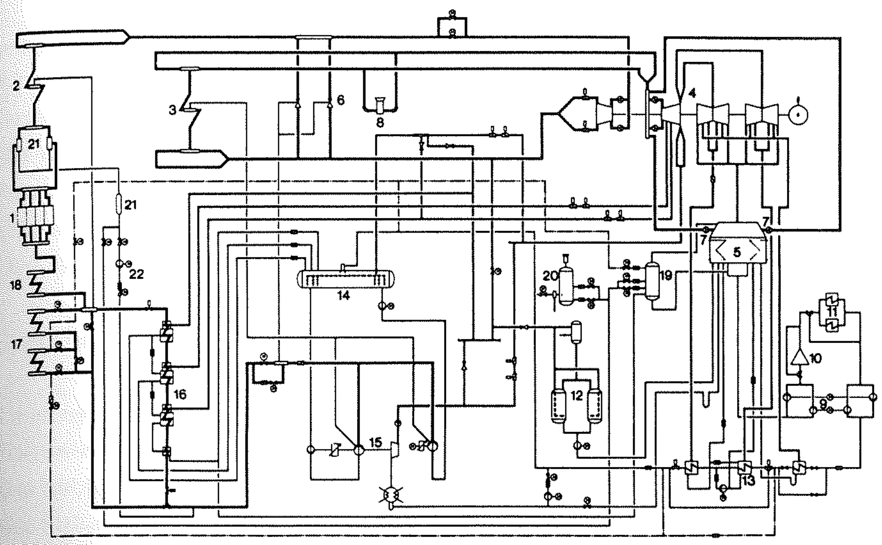

Fig. 11-9: Piping Diagram for the 420 MW Combined-Cycle Power Plant at Lausward

1 Steam generator
2 High pressure superheater
3 Reheater
4 Steam turbogroup
5 Condenser
6 High pressure reducing station
7 Low pressure reducing station
8 Reheater safety valves and sound damper
9 Condensate pumps
10 Condensate cleaning unit
11 Generator cooler
12 Condensate tank with replacement water deaerator
13 Low pressure feedwater heater
14 Feedwater tank with spray deaerator
15 Feedwater pumps
16 High pressure feedwater heater
17 Part-flow economizers
18 Full-flow economizer
19 Flash-box
20 Start-up flash-box
21 Boiler start-up flash system
22 Boiler circulating pump

Figure 11-10

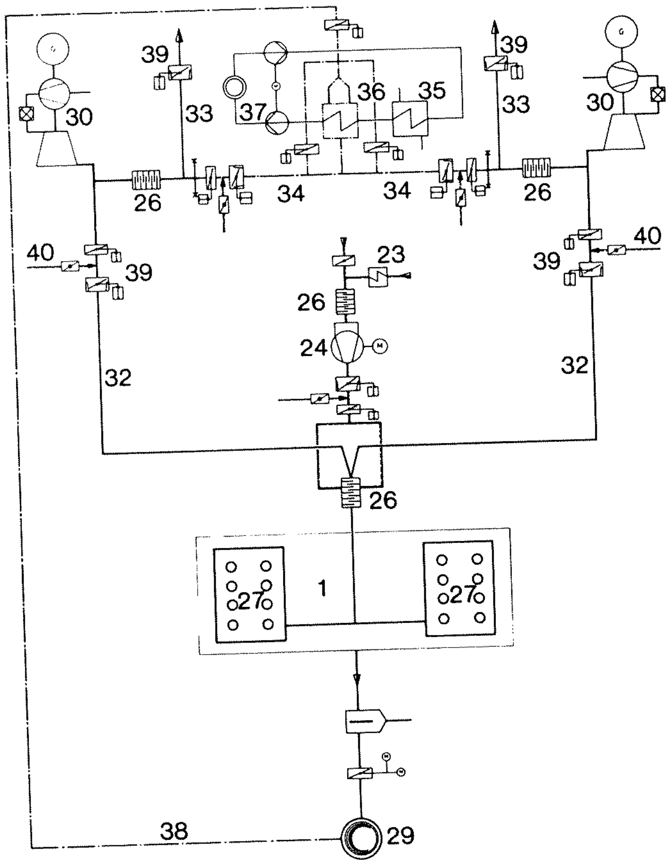

Fig. 11-10: Flue Gas/Air Flow Diagram of the 420 MW Combined-Cycle Plant at Lausward

- 23 Steam air preheater
- 24 Fresh air ventilation fan
- 25 Fresh air duct to steam generator
- 26 Silencer
- 27 Burners
- 28 Flue gas duct
- 29 Stack
- 30 Gas turbines 1 and 2
- 32 Gas turbine exhaust duct to steam generator
- 33 Gas turbine exhaust stack for single-cycle operation
- 34 Gas turbine exhaust duct to exhaust gas heat exchanger (to be installed later)
- 35 Steam heated heat exchanger for district heating
- 36 Gas turbine exhaust gas heat exchanger for district heating
- 37 District heating system
- 38 Flue gas duct to main stack (to be installed later)
- 39 Hydraulically operated dampers
- 40 Sealing air supply

Table 11-5: Main Technical Data of the Combined-Cycle Plant at Lausward

Air temperature	5	°C
Gas turbine power output (bypass operation)	74/83	MW
Gas turbine power output (combined-cycle operation)	62/70	MW
Steam turbine power output	300	MW
Live steam flow	236	kg/s
Live steam data	188	bar
	540	°C
Steam data at outlet from reheater	40.6	°C
	540	°C
Condenser pressure	0.028	bar
Gas turbine exhaust temperature (combined-cycle operation)	415/452	°C
Gas turbine exhaust gas flow	378	kg/s
Gas turbine inlet temperature	890/935	°C
Net power output when burning oil (LHV)	34.9	MW
Net efficiency, in best point, burning oil	43.8	%

11.2.2 The 750 MW Power Plant Gersteinwerk, Block K, VEW, Germany

This 750 MW combined-cycle plant consists of a gas-burning turbine followed by a coal-burning forced-circulation steam generator and a reheat steam turbine. Two fresh air fans make it possible, with restrictions on load, to operate the steam process by itself, without the gas turbine. One of the fresh air fans also supplies air during combined-cycle operation to the steam generator burner, since the oxygen level in the gas turbine exhausts is not sufficient for full steam generation.

75% of the flue gas is sent to a desulphurization unit after it leaves the steam generator. The uncleaned portion is mixed with the cleaned to attain a stack temperature of approx. 70 °C. The power plant is cooled using a natural circulation wet cooling tower.

Fig. 11-12 shows the principle of design and Fig. 11-13 the layout. Table 11-6 contains the most important technical data.

Table 11-6: Main Technical Data (Combined-Cycle Operation, of the Power Plant Gersteinwerk, Block K

Fuel for gas turbine	Natural gas	
Fuel for steam generator	Bituminous coal	
Air temperature	10	°C
Relative humidity	80	%
Gas turbine power output	112.5	MW
Steam turbine power output	656	MW
Net power plant output	711	MW
Net efficiency of the installation (LHV)	40.85	%
Gas turbine exhaust gas flow	503.4	kg/s
Gas turbine exhaust gas temperature	481	°C
Gas turbine inlet temperature	950	°C
Live steam flow	495.5	kg/s
Live steam data before turbine	186	bar
	530	°C
Live steam data after reheater, before turbine	40	bar
	530	°C
Condenser pressure	0.066	bar
Auxiliary fan supply capacity	233	kg/s

This plant was designed to allow later conversion to a partial coal gasification system, with the gas produced being used in the gas turbine and the char in the boiler.

Figure 11-11a

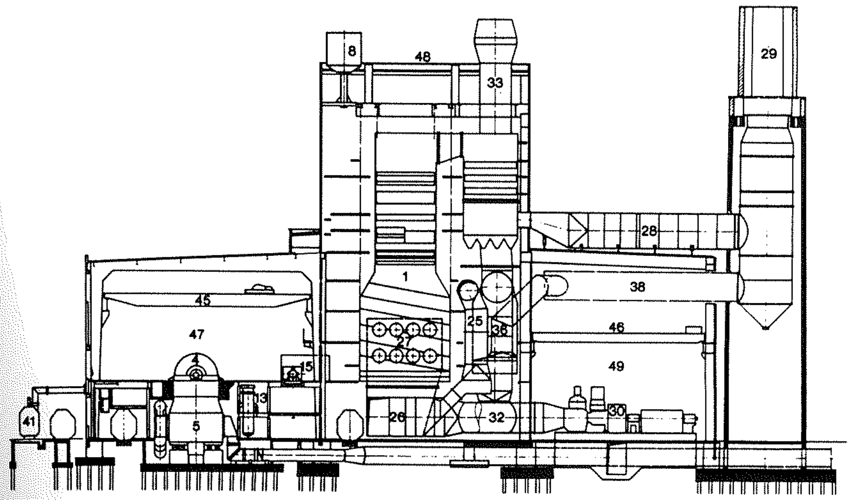

Fig. 11-11a: Layout of the 420 MW Combined-Cycle Plant at Lausward

1 Steam generator
2 High pressure superheater
3 Reheater
4 Steam turbogroup
5 Condenser
6 High pressure reducing station
7 Low pressure reducing station
8 Reheater safety valves and silencer
9 Condensate pumps
10 Condensate cleaning unit
11 Generator cooler
12 Condensate tank with replacement water deaerator
13 Low pressure feedwater heater
14 Feedwater tank with spray deaerator
15 Feedwater pumps
16 High pressure feedwater heater
17 Part-flow economizers 1 and 2
18 Full-flow economizer
22 Boiler circulating pump
23 Steam air preheater
24 Fresh air ventilation fan
25 Fresh air duct to steam generator
26 Muffler
27 Air enclosure
28 Flue gas duct
29 Stack
30 Gas turbines 1 and 2
31 Air intake shaft for gas turbine
32 Gas turbine exhaust duct to steam generator
33 Gas turbine exhaust stack for single-cycle operation
34 Gas turbine exhaust duct to exhaust gas heat exchanger (to be installed later)
35 Steam heated heat exchanger for district heating
36 Gas turbine exhaust gas heat exchanger for district heating (to be installed later)
39 Hydraulically operated dampers
40 Sealing air supply
41 Steam turbogroup transformer
42 Gas turbogroup transformers
43 Station service transformers
44 Control room
45 Machine hall crane
46 Gas turbine machine hall crane
47 Machine hall
48 Boiler room with water section and electrical section
49 Gas turbine machine hall
50 Reducing station for natural gas

Figure 11-11b

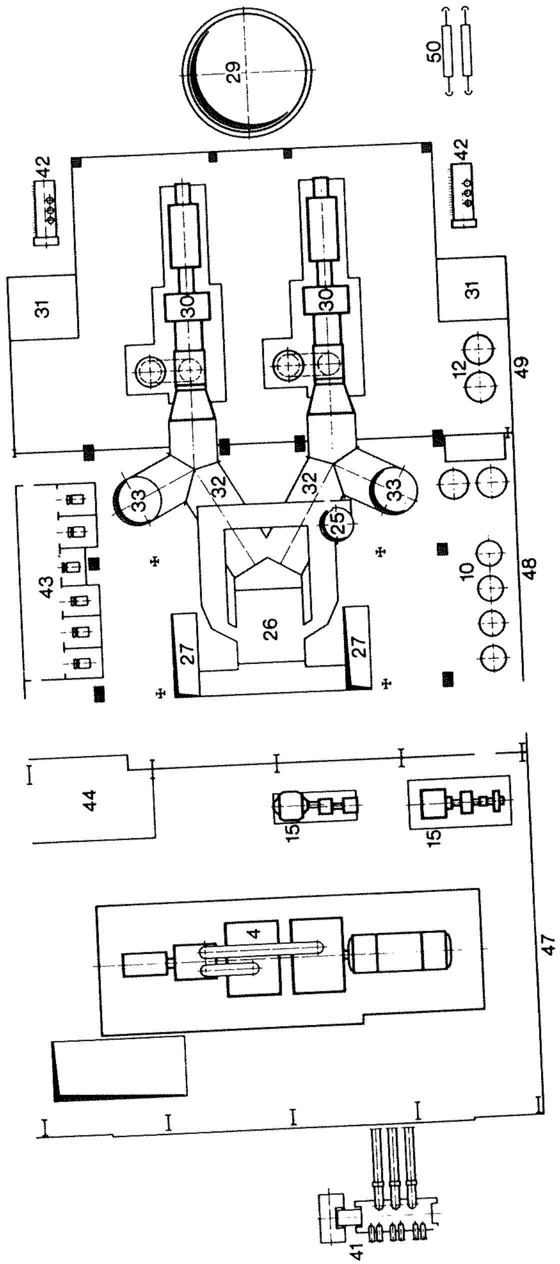

Fig. 11-11b: Layout of the 420 MW Combined-Cycle Plant at Lausward

Figure 11-12

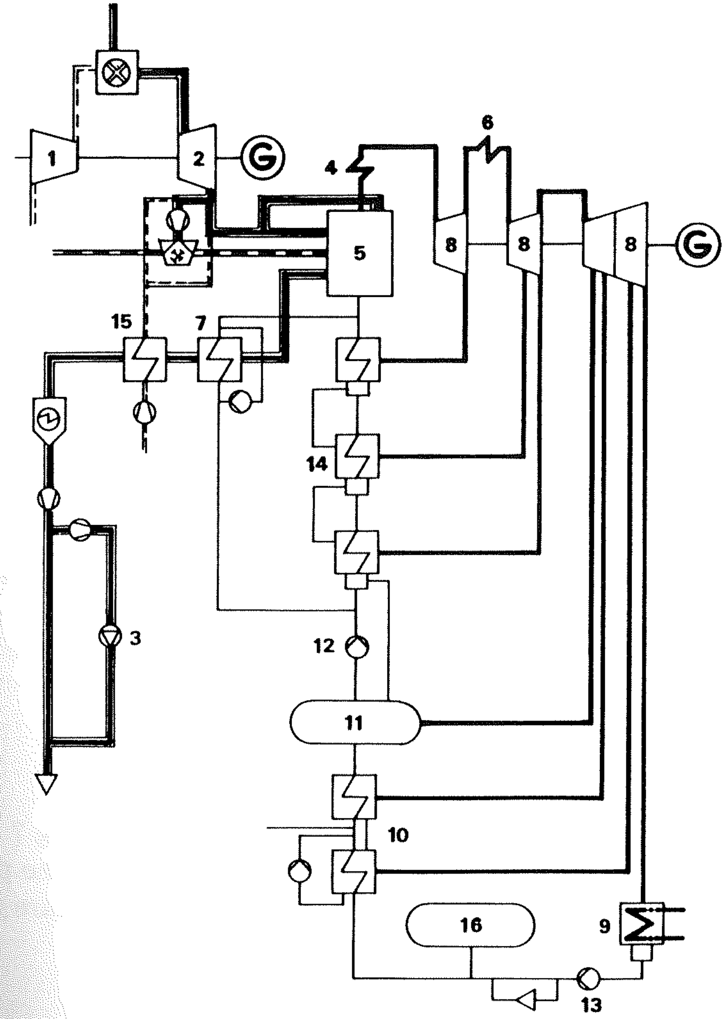

Fig. 11-12: Design Principle of the 75- MW Unit, Block K, at the Gersteinwerk Power Plant

1	Compressor	7	Part-flow economizer	12	Feed pumps
2	Gas turbine	8	Steam turbine	13	Condensate pumps
3	Desulphurization unit	9	Condenser	14	High pressure feed-water heater
4	Superheater	10	Low pressure feed-water heater	15	Air preheater
5	Steam generator	11	Feedwater tank/ deaerator	16	Condensate tank
6	Reheater				

11.2.3 Hemweg Unit 7, EB Amsterdam, the Netherlands (Repowering) [112]

This plant is a very interesting example of the repowering of an existing steam turbine plant. Unlike the example discussed in Section 3.3, the boiler here has be reused in the new regime. Hemweg 7 formerly was a modern 500 MW steam turbine plant with reheat, fired either with natural gas or heavy oil. To increase the efficiency of this plant, which uses relatively expensive fuels, it has been converted into a combined-cycle plant. The resulting plant is similar to that with maximum supplementary firing in the boiler as presented in Section 3.2.2, but with one major difference: Neither the boiler, nor the steam turbine, nor the feedwater heating system was originally designed for operation with a gas turbine instead of an air blower and regenerative air preheater. Modifications of both the boiler and the feedwater heating system were therefore necessary. The following changes had to be made on the boiler to allow it to operate on hot gas turbine exhausts instead of preheated air:

- new ducts to the burners
- new wind box with gas burners
- additional new low and high pressure economizers

Fig. 11-14 shows the thermal diagram of the Hemweg 7 combined cycle after repowering, Fig. 11-15 the layout of the plant. The main technical data before and after the conversion are given in Table 11-7.

Table 11-7: Main Technical Data of the Hemweg 7 Power Plant

		Before Repowering	After Repowering
Fuel for gas turbine	-	-	natural gas
Fuel for steam generator	-	oil/gas	oil/gas
Air temperature	°C	15	15
Gas turbine power output	MW	-	134.9
Steam turbine power output	MW	500	464
Station service power	MW	13.5	8.7
Net efficiency of the plant (LHV)	%	41.3	45.9
Gas turbine inlet temperature	°C	-	1070
Gas turbine exhaust temperature	°C	-	534
Gas turbine exhaust flow	kg/s	-	506
Live steam flow	kg/s	414.9	344.4
Live steam data before steam turbine	bar	177.5	161
	°C	535	535
Condenser pressure	bar	0.035	0.036

11.3 Combined-Cycle Plants for Cogeneration

11.3.1 Pegus Unit 12, Utrecht, the Netherlands

The Pegus Unit 12 combined-cycle plant in the Netherlands is one of the most modern combined-cycle plants. It has been conceived as a cogeneration plant suplying heat to three different district heating systems and power to the grid. The most interesting feature of this plant is that despite its being a cogeneration plant, it is capable of supplying power alone during the summer, at the extremely high efficiency of nearly 52%. Fig. 11-16 shows the principle of design of the plant, which includes one 150 MW gas-burning industrial gas turbine equipped with a dry low NO_x combustor that produces less than 50 vppm of NO_x without the injection of water or steam. Fig. 11-17 shows a model of the 150 MW gas turbine.

The energy in the exhaust gas is recovered in a sophisticated 3-pressure boiler, with the steam from the high pressure section being reheated. The intermediate pressure steam is used in the steam turbine. The low pressure steam is produced by flashing high pressure water and is used for feedwater heating in the deaerator, with the excess being piped to the steam turbine. The hot water from the flash system is either used for the district heating systems or, in summer, to heat condensate.

The steam turbine has three casings. The steam expanded in the high pressure turbine is reheated in the boiler. The intermediate pressure turbine has a second admission for the intermediate pressure steam and three extractions for supplying heating steam to four district heating condensers. The steam that is not extracted is fed to the low pressure turbine, which is designed to operate at optimum efficiency even when no steam is being extracted.

Figure 11-13

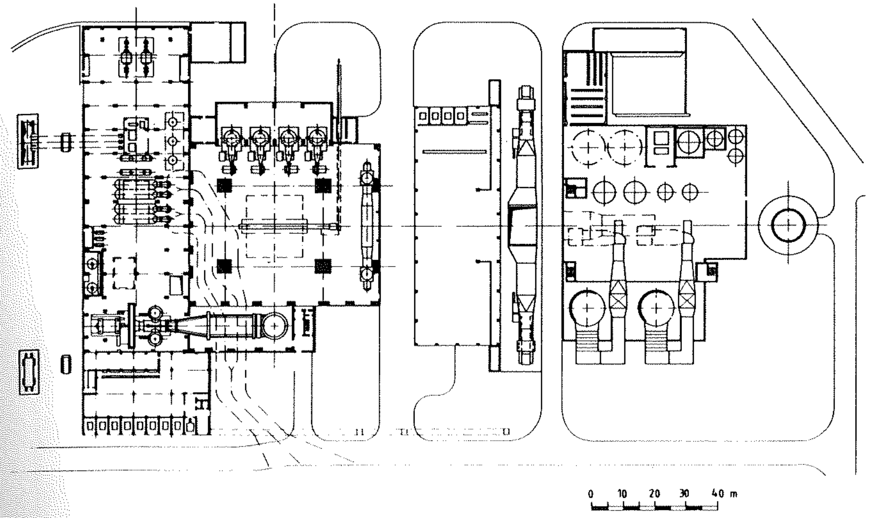

Fig. 11-13: Layout of the 75- MW Unit, Block K at the Gersteinwerk Power Plant

1	Steam generator	7	E-filter	13	Gas turbogroup transformer
2	Steam turbogroup	8	Desulphurization unit	14	Steam turbogroup
3	Condenser	9	Intake duct	15	10 kV switchgear
4	Gas turbogroup	10	Fresh air ventilation	16	Low voltage switchgear
5	Coal crushers	11	Gypsum storage		
6	Feedwater pumps	12	Stack		

Figure 11-14

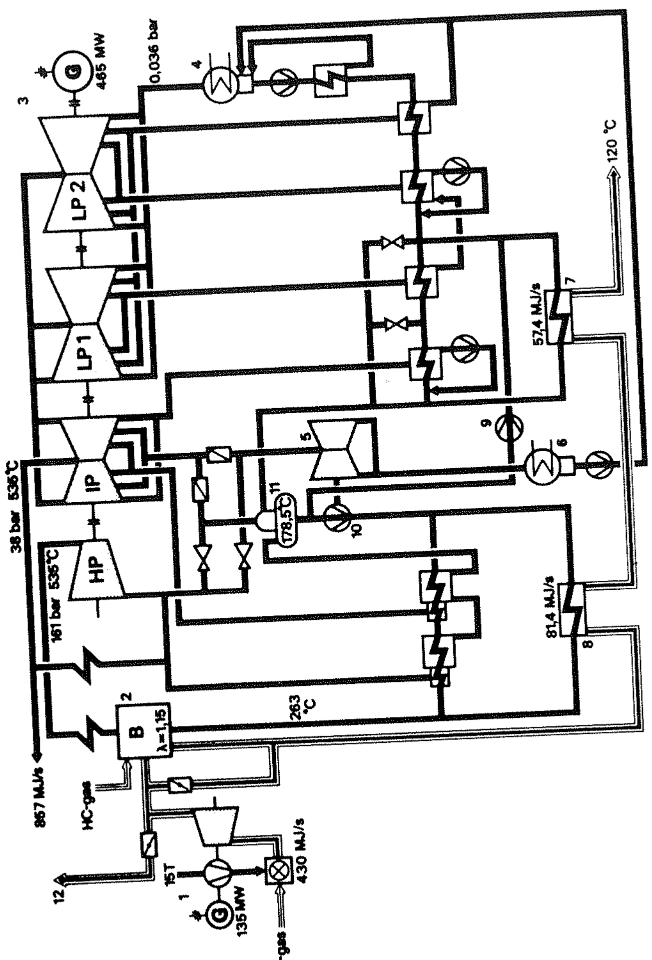

Fig. 11-14: Design Principle of the Hemweg Unit 7 Combined-cycle Plant

1 Gas turbine unit
2 Steam generator
3 Steam turbine unit
4 Condenser
5 Boiler feed pump turbine
6 Auxiliary condenser (BFPT)
7 LP-Eco
8 HP-Eco
9 Circulating pump (during oil firing or low load operation only)
10 Boiler feed pump
11 Deaerator
12 Bypass stack

Figure 11-15

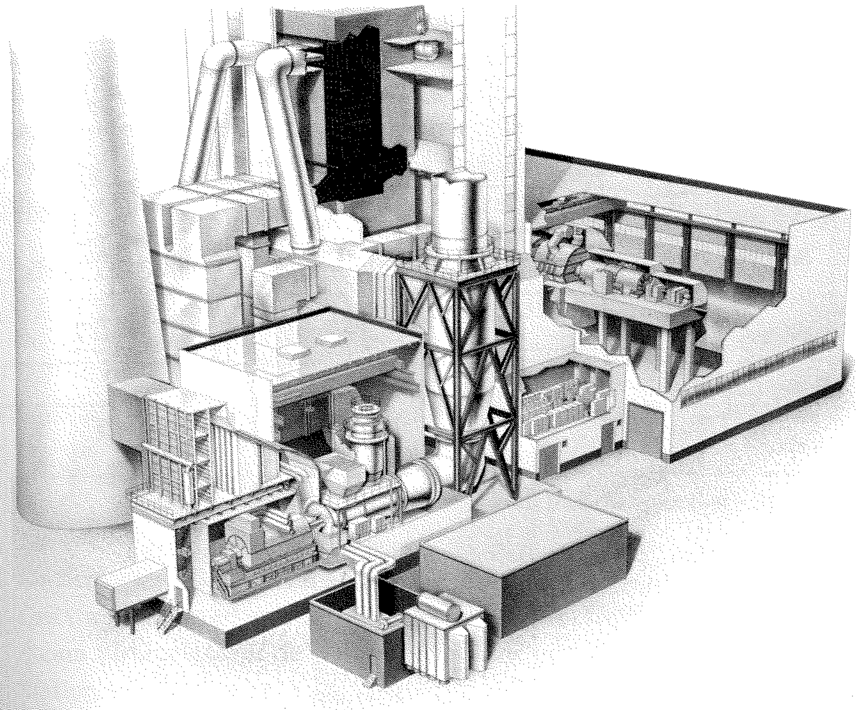

Fig. 11-15: Layout of the Hemweg Unit 7 Combined-Cycle Plant

Figure 11-16

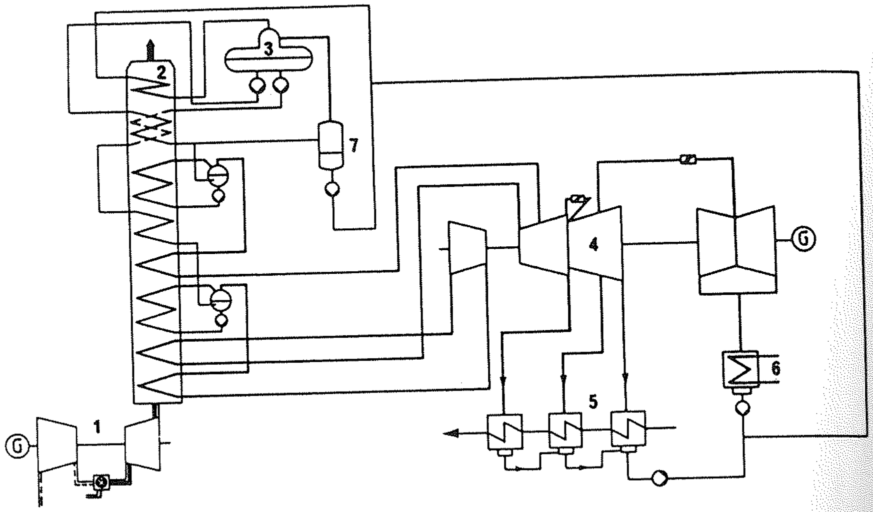

Fig. 11-16: Design Principle of the Pegus Unit 12 Combined-Cycle Plant

1 Gas turbine
2 Waste heat boiler
3 Deaerator/feedwater tank
4 Steam turbine
5 Heating condensers
6 Condenser
7 Flash tank

Table 11-8 lists the main technical data of this plant.

Fig. 11-18 shows the dry low NO_X combustor for the gas turbine. This has 54 burners which operate on the premixing principle, i.e., air and natural gas are mixed together before they enter the combustor. Groups of these burners must be switched off at part load in order to maintain an excess air ratio low enough to ensure good combustion. Fig. 11-19 shows the layout of the power plant, which is entirely an indoor installation.

Table 11-8: Main Technical Data of the Pegus Unit 12 Combined-Cycle Cogeneration Plant

		Maximum Heating Output	Maximum Electrical Output
Fuel	-	natural gas	natural gas
Air temperature	°C	0	15
Cooling water temperature	°C	4	20
Gas turbine power output	MW	153.5	141.4
Steam turbine power output	MW	59.7	84.6
Station service power	MW	2.1	2.7
Total net electrical output	MW	211.1	223.3
Net electrical yield (LHV)	%	45.65	51.74
Total heat output	MW	180.6	-
Incoming temperature, heating water	°C	65	-
Outgoing temperature, heating water	°C	105	-
Rate of fuel utilization (LHV)	%	84.7	51.74
Gas turbine exhaust temperature	°C	520.1	524.6
Gas turbine exhaust flow	kg/s	529.1	505.4
Gas turbine inlet temperature	°C	1070	1070
Live steam flow	kg/s	54.8	53.5
Live steam pressure	bar	69.5	68
Live steam temperature	°C	490	490

11.3.2 The Combined-Cycle Gas and Steam Turbine Cogeneration Plant, Unit 10, at PEGUS, in Utrecht, the Netherlands (Repowering) [50]

This combined-cycle plant started operation in late 1978. It is equipped with two 30 MW gas turbines and one 40 MW extraction/condensing turbine. One important feature is the steam turbine, which was built as long ago as 1959 as a two-cylinder condensing turbine. During the repowering, an extraction was installed between the high pressure and the low pressure casings. This makes it possible to operate either as a heating plant alone or as a straight condensing plant (Fig. 11-20). The exhausts from both the two gas turbines are supplied to two single-pressure waste heat boilers. At the end of the boilers, there is a heating loop which recovers the heat that remains downstream from the steam generator. This is used partly for district heating and partly for feedwater preheating. Because the fuel burned is sulphur-free gas, the exhausts can be cooled to 100 °C (212 °F). For the main technical data, refer to Table 11-9.

Table 11-9: Main Technical Data of the Combined-Cycle Cogeneration Plant, Unit 10, at PEGUS, Utrecht

		Maximum Heating Output	Maximum Electrical Output
Fuel		Natural gas	Natural gas
Air temperature	°C	0	0
Gas turbine power output	MW	2 x 30.8	2 x 30.8
Steam turbine power output	MW	26	38
Total electrical power output	MW	87.6	99.6
Electrical yield (LHV)	%	35.9	40.9
Heat output of heating condenser	MW	79.8	0
Heat output of heat exchanger	MW	31.8	18.6
Total heat output	MW	111.6	18.6
Incoming temperature, heating water	°C	65	65
Outgoing temperature, heating water	°C	95	95
Rate of fuel utilization (LHV)	%	81.8	41.5
Gas turbine exhaust temperature	°C	519	519
Gas turbine exhaust flow	kg/s	2 x 158	2 x 158
Gas turbine inlet temperature	°C	945	945
Live steam flow	kg/s	2 x 19	2 x 19
Live steam data	bar/°C	38/645	38/465
Stack temperature	°C	100	100

Figure 11-17

Fig. 11-17: Model of the 150 MW ABB Gas turbine, Type 13 E

Figure 11-18

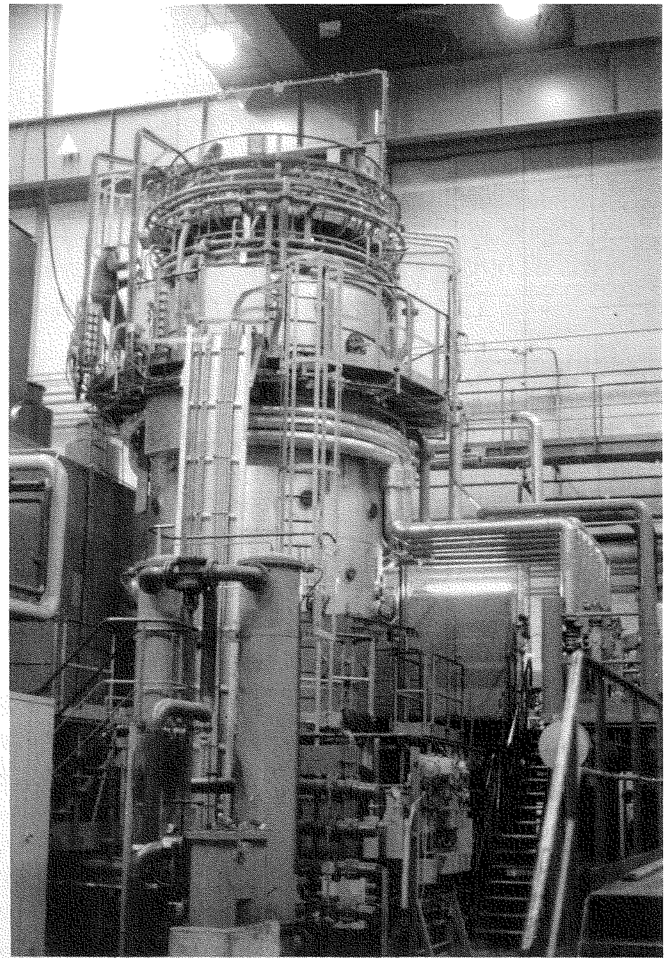

Fig. 11-18: Dry Low NOx Combustor

Figure 11-19

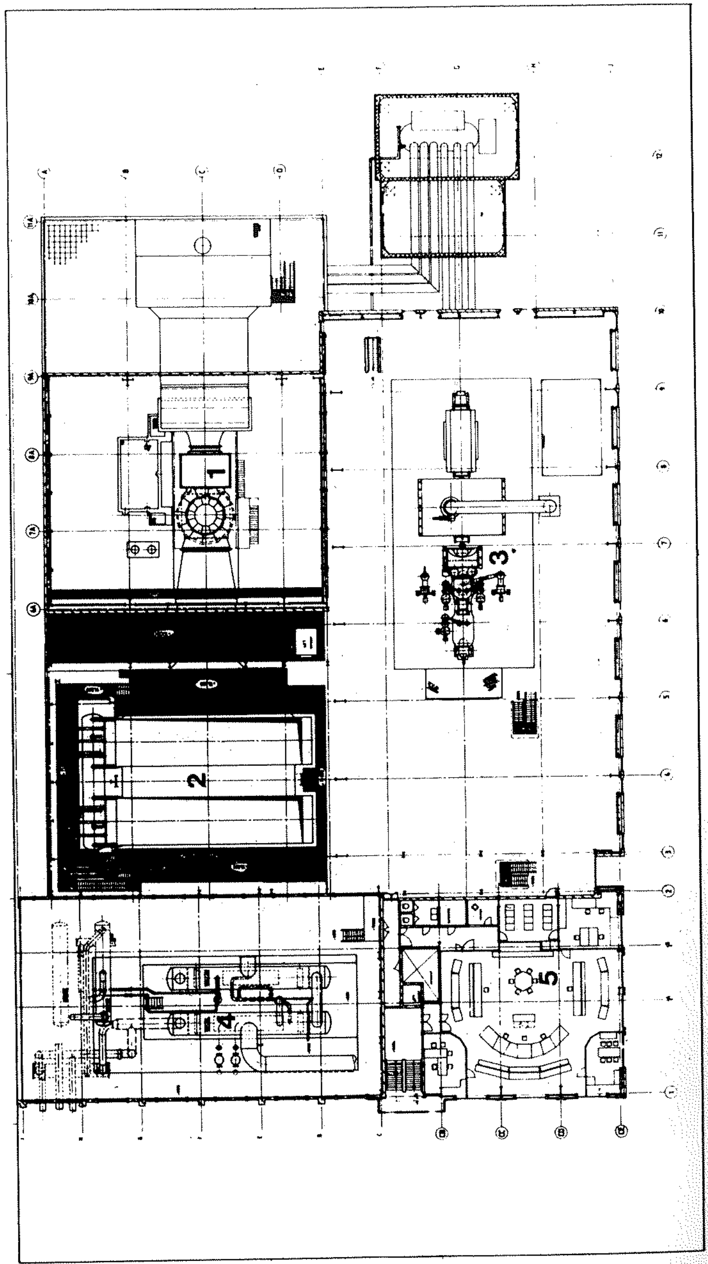

Fig. 11-19: Layout of the Pegus Unit 12 Combined-Cycle Plant

A comparison of the data for maximum heating output with those for maximum electrical output shows the great flexibility of this installation. In winter, it provides heat and electricity at an energetic utilization rate of 82% and in summer it produces electricity at an afficiency of approx. 41%.

Fig. 11-21 shows the layout of the power plant. The gas turbines and the waste heat boilers are installed in the converted boiler house of the former steam plant. Fig. 11-22 shows one of the two gas turbines, Fig. 11-23 the repowered steam turbine with the new extraction and cross-over pipes.

11.3.3 The Combined-Cycle Cogeneration Plant H 4/5 of Elektromark AG at Hagen, Germany

This plant is another interesting example of the cogeneration of heat and electricity, The 220 MW combined-cycleplant started into operation in 1981. It consists of two gas-burning 75 MW gas turbines, two two-pressure waste heat boilers, and one extraction/condensing turbine (Fig. 11-24). It supplies process steam at 12.8 and 4.5 bar (170 and 50 psig) to a paper mill and district heating water at a max. temperature of 110 °C (230 °F) to a large printing plant nearby. Provision has been made for expansion to provide additional district heating in the future.

The waste heat boilers each have a high pressure and a low pressure steam generator with a economizer and a superheater. The low pressure steam is fed either into the process steam system or into the steam turbine. Table 11-10 shows the data at the design point of the installation. The overall layout of the entire plant is again that shown in Fig. 11-25. Fig. 11-26 shows a general view of the power station.

Figure 11-20

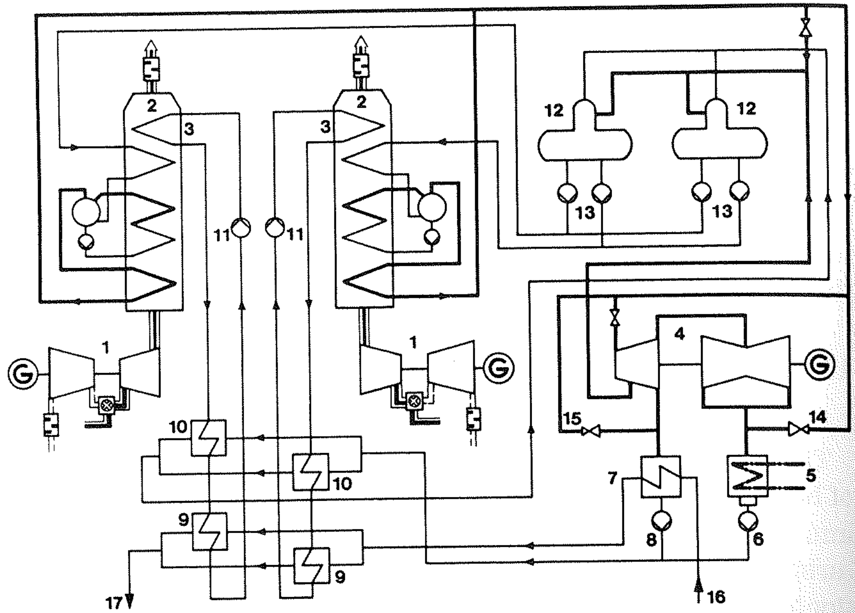

Fig. 11-20: Design Principle of the 100 MW Combined-Cycle Cogeneration Plant, Unit 10, at PEGUS, Utrecht

1 Gas turbine
2 Waste heat boiler
3 Heating loop
4 Steam turbine
5 Steam turbine condenser
6 Condensate pumps
7 Heating Condenser
8 Hot condensate pumps
9 Heater
10 Preheater
11 Recirculating pumps
12 Feedwater tank/deaerator
13 Feed pumps
14 Steam bypass
15 Steam bypass to heating condenser
16,17 District heating water system

Figure 11-21

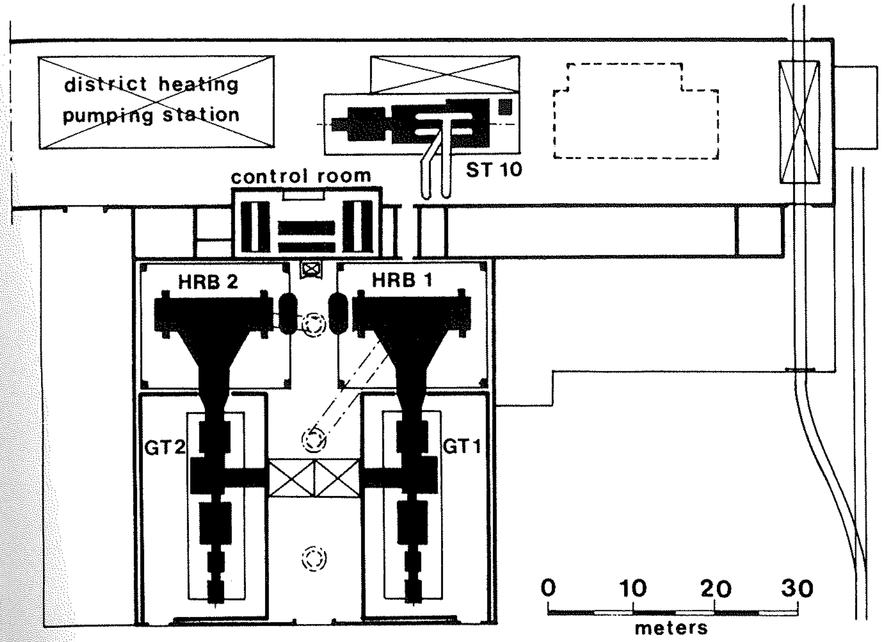

Fig. 11-21: Layout of the 100 MW Combined-Cycle Cogeneration Plant, Unit 10, at PEGUS, Utrecht

Figure 11-22

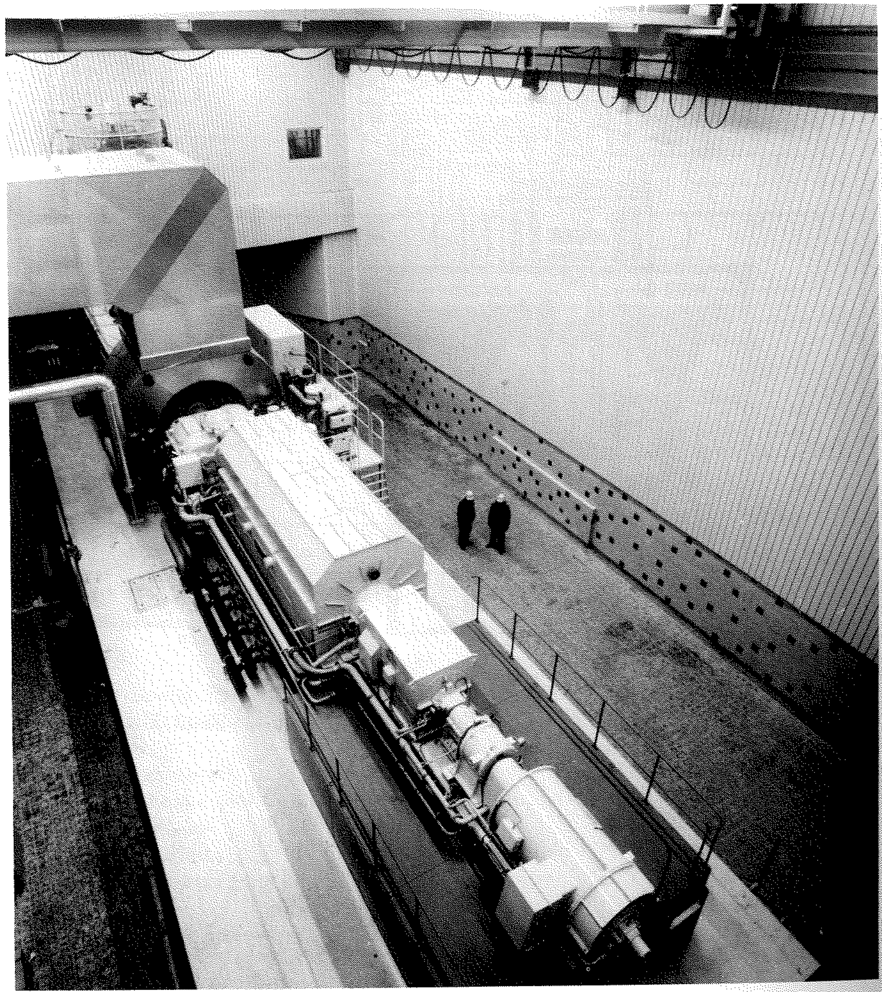

Fig. 11-22: Gas turbine for the 100 MW Combined-Cycle Cogeneration Plant, Unit 10, at PEGUS, Utrecht

Figure 11-23

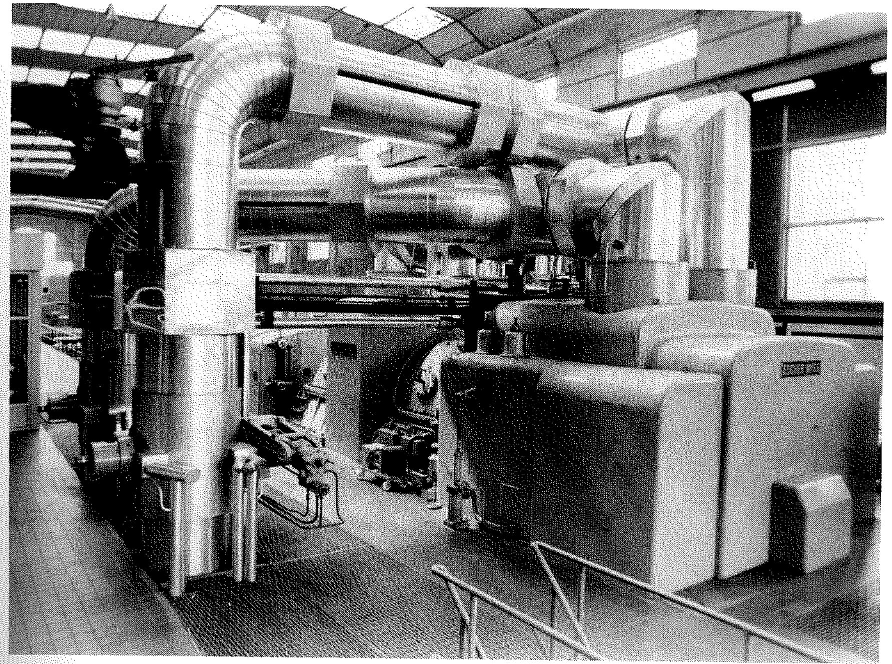

Fig. 11-23: Steam turbine for the 100 MW Combined-Cycle Cogeneration Plant at PEGUS, Utrecht

Figure 11-24

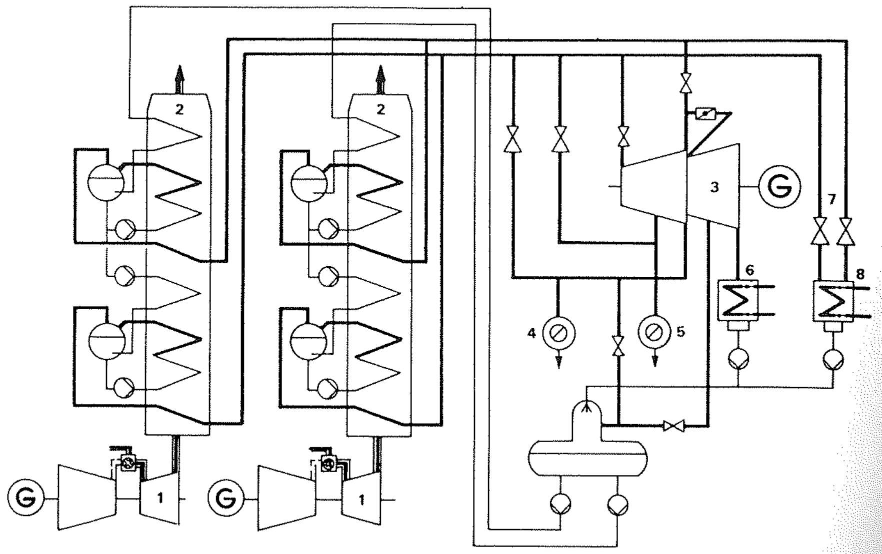

Fig. 11-24: Design Principle of the 220 MW Combined-Cycle Cogeneration Plant at Hagen

1	Gas turbines	6	Condenser
2	Waste heat boilers	7	Reducing station
3	Steam turbine	8	Auxiliary condenser
4, 5	Steam users		

Figure 11-25

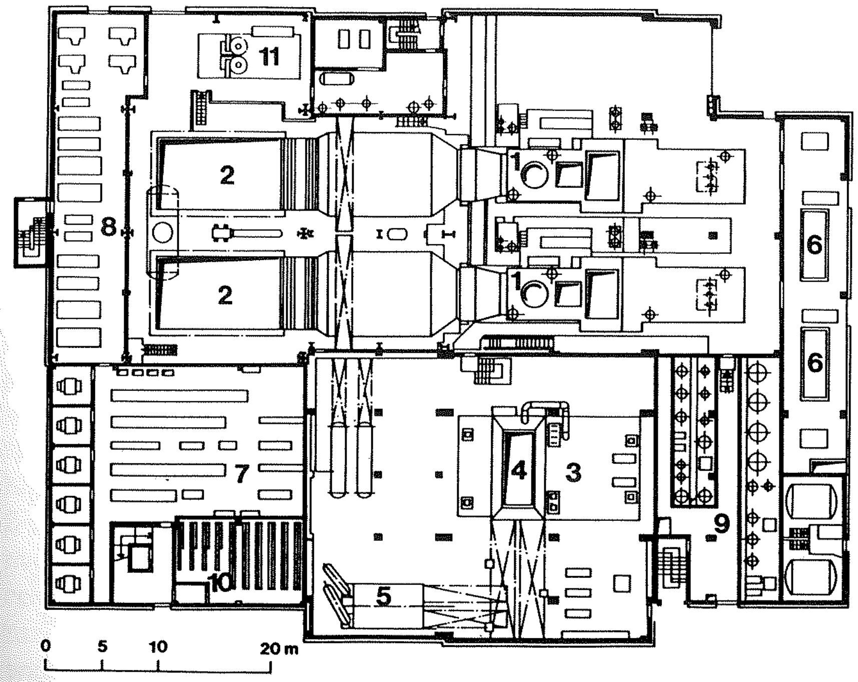

Fig. 11-25: Layout of the 220 MW Combined-Cycle Cogeneration Plant at Hagen

1	Gas turbines	5	Auxiliary condenser	9	Water treatment unit
2	Waste heat boilers	6	Main transformers	10	Batteries
3	Steam turbine	7	Low voltage switchgear	11	Auxiliary boiler
4	Condenser	8	Feedwater and boiler circulating pumps		

Table 11-10: Main Technical Data of the Combined-Cycle Cogeneration Plant at Hagen

Fuel	Natural gas	
Air temperature	15	°C
Gas turbine power output	2 x 76.6	MW
Steam turbine power output	72.1	MW
Total electrical power output	225.3	MW
Electrical yield (LHV)	42.2	%
Process steam flow (12.8 bar)	2.8	kg/s
Process steam flow (4.5 bar)	34.7	kg/s
Process heat	93.7	MW
Rate of energy utilization (LHV)	59.8	%
Gas turbine exhaust temperature	500	°C
Gas turbine exhaust flow	2 x 365	kg/s
Gas turbine inlet temperature	945	°C
High pressure live steam flow	2 x 40.3	kg/s
High pressure live steam pressure	40.8	bar
High pressure live steam temperature	470	°C
Low pressure live steam flow	2 x 10.4	kg/s
Low pressure live steam pressure	5.5	bar
Low pressure live steam temperature	210	°C
Low pressure feedwater temperature	55	°C
Stack temperature	110	°C

11.3.4 The 100 MW Combined-Cycle Plant AES-Placerita, California, USA [113]

The 100 MW combined-cycle plant built at Placerita Canyon near Los Angeles is a typical example of a US cogeneration plant built under the PURPA law (Public Utility Regulatory Policies Act), which was passed with the intention of encouraging the cogeneration of heat and power. The plant supplies 100 MW of electrical power to the grid and generates 31.5 kg/s (250,000 lb/hr) of wet steam (85% quality) at 70 bar (1000 psig). This process steam is used for enhanced oil recovery, i.e., it is injected into the ground to increase the production from the oil wells located around the power plant.

Fig. 11-27 shows the thermal diagram of the combined cycle. The plant has two gas-fired gas turbines rated at 43 MW under site conditions. Under normal operating conditions, steam is injected into the combustor to reduce NO_X emissions. In case of emergency, water injection is also possible. The exhaust from one of the gas turbines is directed to a heat recovery boiler which generates the 85% quality process steam (steam injection boiler), while the flue gas from the other is used to generate superheated steam in a normal dual pressure boiler (steam electric boiler). Both high pressure and low pressure steam are expanded in the 2-pressure condensing steam turbine. An interconnecting duct makes it possible to control the process steam flow. If less process steam is required, some of the flue gas from the first gas turbine is ducted over to the steam electric boiler. The cooling water is cooled back down in a forced-draft wet cooling tower.

The extremely low emission levels from this power plant are certainly its most interesting feature: The concentration of NO_X in the dry exhaust had to be held below 7 vppm at 15% O2. The injection of steam or water into the combustor is not by itself sufficient to attain this low a level. Selective catalytic reduction (SCR) systems had to be installed in both boilers. These systems inject ammonia to convert the NO_X into nitrogen and water. The chemical reactions have already been indicated in Section 9.1. Selective catalytic reduction works properly and with good conversion efficiency only within a temperature window of about 350 °C (662 °F). If the temperature drops below 250 to 300 °C (482 to 572 °F), the reaction is too slow. Above 400 to 450 °C (752 to 842 °F) the ammonia is converted into NO_X. The SCR units therefore must be installed in the evaporators of the heat recovery boilers, thereby dividing them into two sections.

A catalytic converter for CO has also been installed in each boiler to reduce by oxidation the emission of carbon monoxide from less than 10 vppm to approx. 1 vppm after the turbine. The basic reaction here is

$$2\,CO + O_2 \longrightarrow 2\,CO_2$$

Expensive acoustical precautions were taken to reduce noise levels at 240 m (800 ft) to less than 39 dbA, including the use of a cooling tower with low speed fans, special building walls, large silencers on the gas turbines, etc.

Table 11-11 shows the levels of the most important pollutants after the gas turbines and after the boilers.

Table 11-11: Emission levels from the AES Placerita Plant

	After the gas turbines	After the boilers
NO_x (dry, 15% O_2)	38	7
CO	< 10	< 2
UHC (Particulates)	< 2	< 1
Noise	39 dbA at 800 ft	

Table 11-12 summarizes the main technical data for this plant; Fig. 11-28 shows its layout.

Table 11-12: Main Technical Data of the AES Placerita Combined-Cycle Plant

Fuel		natural gas
Air temperature	°C	24
Relative humidity	%	26.7
Gas turbine power output	MW	2 x 43.8
Steam turbine power output	MW	15.0
Station service power	MW	2.6
Net power output	MW	100
Process steam flow	kg/s	31.5
Process steam conditions	bar	70 bar / 85% quality
High pressure:		
Live steam flow	kg/s	24.4
Live steam pressure	bar	41.4
Live steam temperature	°C	475
Low pressure:		
Live steam flow	kg/s	4.9
Live steam conditions	bar	5.3 bar / saturated
Condenser pressure	bar	0.082
Stack temperature	°C	115 / 123
Gas turbine inlet temperature	°C	1085
Gas turbine exhaust temperature	°C	535
Gas turbine exhaust flow	kg/s	2 x 167

Figure 11-26

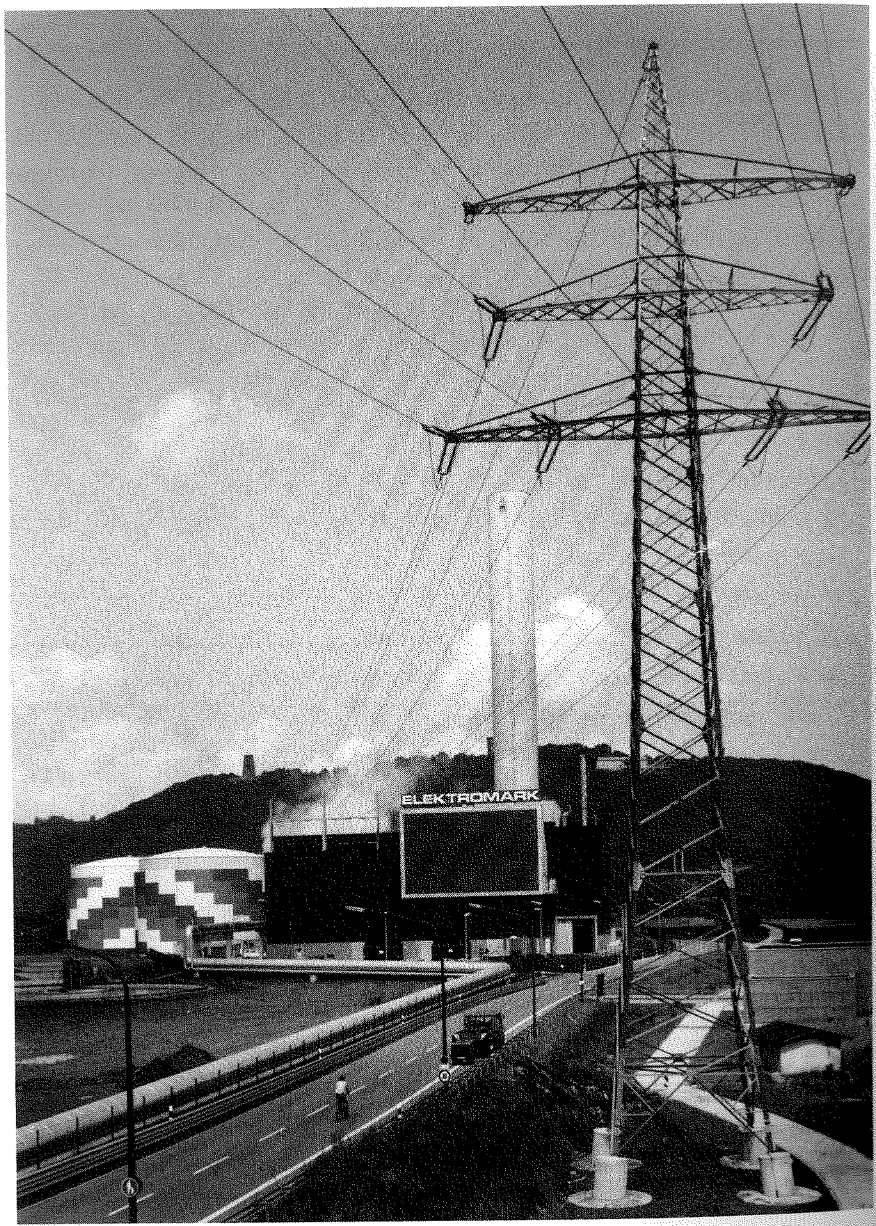

Fig. 11-26: View of the 220 MW Combined-Cycle Cogeneration Plant at Hagen

Figure 11-27

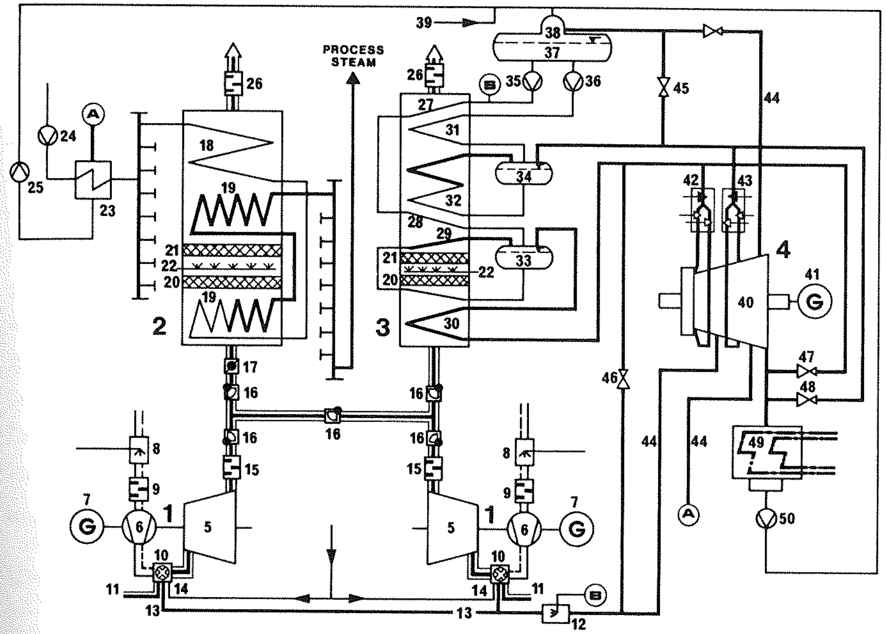

Fig. 11-27: Design Principle of the AES Placerita Combined-Cycle Plant

1 Gas turboset
2 SI-boiler
3 SE-boiler
4 Steam turboset
5 Gas turbine (GT)
6 Turbocompressor
7 GT turbogenerator
8 Evaporation cooler
9 Air intake silencer
10 Combustion chamber
11 Fuel feed
12 Steam injection de-superheater
13 Steam injection
14 Water injection
15 Exhaust silencer
16 Flapgate isolator
17 Modulating damper
18 Si-boiler economizer
19 SI-boiler steamer
20 CO catalytic converter
21 SCR catalytic converter
22 Ammonia injection
23 SI-boiler feedwater pre-heater
24 SE-boiler feedwater pumps
25 Condensate return pump
26 Stack silencer
27 SE-boiler HP economizer (Part 2)
28 SE-boiler economizer (Part 1)
29 SE-boiler HP evaporator
30 SE-boiler HP superheater
31 SE-boiler LP economizer
32 SE-boiler LP evaporator
33 SE-boiler HP drum
34 SE-boiler LP drum
35 SE-boiler HP feed-water pumps
36 SE-boiler LP feed-water pumps
37 SE-boiler feedwater tank
38 Deaerator/feedheater
39 Replacement water
40 Steam turbine (ST)
41 ST turbogenerator
42 HP stop and control valve
43 LP stop and control valve
44 Bleed steam line
45 Back-up steam bypass to deaerator
46 Back-up steam bypass to steam injection
47 HP steam bypass sect
48 LP steam bypass section
49 Steam condenser
50 Condensate pumps

Figure 11-28

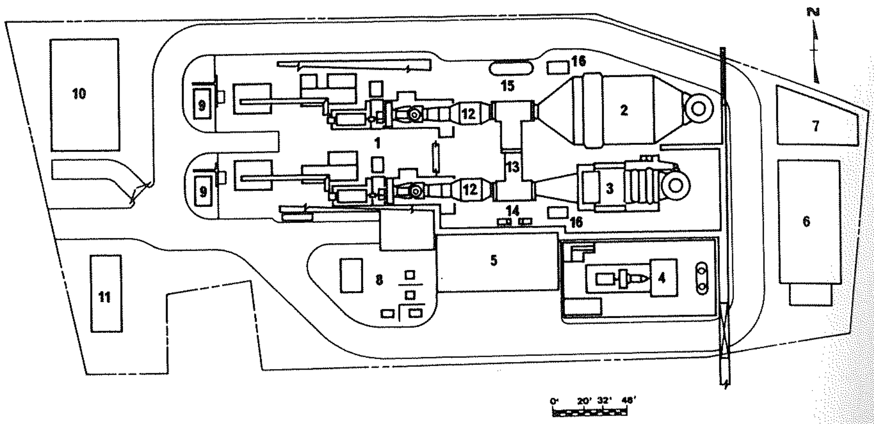

Fig. 11-28: Layout of the AES Placerita Combined-Cycle Plant

1 Gas turboset building
2 SI-boiler
3 SE-boiler
4 Steam turboset building
5 Control building
6 Cooling tower area
7 Chemical treatment area
8 ST transformer and auxiliary transformer area
9 GT transformers
10 Substations
11 Warehouse and workshop
12 Exhaust gas silencers
13 Cross-duct for exhaust
14 NOx water injection pumps
15 Ammonia storage tank
16 Environmental control module

Chapter 12

CONCLUSIONS

The thermodynamic advantages of the combined-cycle plant over simple gas or steam processes and its great potential for development will favor its increased use in the future. Systems employing waste heat utilization alone will stand in the foreground in the future, since only they can fully exploit the high temperature potential of the gas turbine. For that reason, installations with maximum supplementary firing will further lose in importance.

The combined-cycle plant without supplementary firing has the following advantages:

- high efficiency
  Efficiencies of more than 50% can be attained.
- low investment costs
  Because 2/3 of the output is produced in a gas turbine and only 1/3 in a simple steam process, the investment costs required are approx. 30% less than those for a conventional steam power plant.
- small amount of water required
  The amount of cooling water required is only about 40 to 50% as much as for a steam plant.
- great operating flexibility
  The simple steam process makes it possible to start-up and shut-down the plants quickly, which also affects efficiency in a positive direction (reduced start-up losses)
- phased installation
  Because the gas turbines can go into operation much sooner than the steam process, expansion in stages is

possible. This makes it possible to adjust to the growth in demand for energy in a grid. In a third step, coal gasification can be installed if there is too sharp an increase in the price of gas or oil.

- simplicity of operation
  A combined-cycle plant without supplementary firing is significantly simpler to run than a conventional steam plant. Moreover, because combined-cycle plants are generally operated fully automatically, they are also especially suitable for use where operating staff is less experienced.
- low environmental impact
  Gas-burning combined-cycle plants in particular are ideally suitable for use in heavily populated regions because of their high efficiency and their low emission levels for pollutants. In particular, the very low nitrogen oxide levels of clean combined-cycle plants will be one of their most attractive features. Furthermore, gas-fired combined-cycle plants produce per kWh only 40% of the CO_2 produced by a coal-fired plant.
- advantages for cogeneration of heat and electricity
  The good thermodynamic properties of the combined-cycle are highly desirable here. Electrical yields of more than 40% are quite common in heating or industrial power plants with a backpressure turbine.

The limited fuel flexibility of the combined-cycle plant is its greatest disadvantage for use in countries where oil and gas are in short supply. However, combined-cycle plants with coal gasification or PFBC plants could in the future become an attractive alternative to conventional coal-fired steam power plants with flue gas scrubbing. Net efficiencies of more than 45% can certainly be attained, which would permit the environmentally sound and economical use of coal in a combined-cycle plant.

CONVERSIONS

Conversion of the main units used

Multiply	by	to obtain
bar	14.504	psi
Btu	1.055	kJ
Kelvin	1.8	°Rankine
ft	0.30480	m
gal (US)	3.7854	l
inch	2.54	cm
kJ	0.94781	Btu
kg	2.20046	lb
l	0.26417	US gal
m	3.2808	ft
m	1.0936	yd
psi	0.068948	bar

Conversion formulas

To convert	into	Formula:
°C	°F	(9/5)°C + 32
°F	°C	5/9 (°F - 32)
Kelvin	°C	K + 273

Comment:

The efficiencies cited in this book are in all cases based on the Lower Heating Value (LHV).

SYMBOLS USED

η_{WB}	Rate of heat utilization in waste-heat boiler
η_C	Carnot efficiency
C	Velocity
d_1	Outside diameter of a tube or pipe
d_2	Inside diameter of a tube or pipe
f	Frequency
h	Enthalpy
Δh	Difference in enthalpy
η_{IS}	Isentropic efficiency
k	Heat flow coefficient
$\dot{m}$	Mass flow
n	Rotational speed
Nu	Nusselt number
P	Power output
p	Pressure
PC	Power coefficient
Δp	Difference in pressure
Δp_k	Pressure loss on flue gas end in waste-heat boiler
$\eta_{pol.}$	Polytropic efficiency
$\eta_{pol.tr.}$	Polytropic efficiency for dry steam
Pr	Prandtl number
$\dot{Q}$	Heat flow
Q	Heat flow, amount of heat
Re	Reynolds number
S	Surface area
T	Temperature in K

t	Temperature in °C
Δt	Difference in temperature
V	Loss
ν	Specific volume
x	Steam content of the wet steam
α	Heat transfer coefficient
η	Efficiency
η	Dynamic viscosity
Γ	Pinch point of the evaporator
λ	Excess air coefficient
λ	Heat conductance
$\bar{\mu}$	Average absorption capacity of the turbine
ρ	Density

INDICES USED

A	Air
A	Outlet from a heat exchanger or feedwater heater
CW	Cooling water
E	Inlet to a heat exchanger or feedwater heater
Eco	Economizer
Exh	Exhaust gases
G	Flue gas
GT	Gas turbine
H	Heater
HP	High pressure
K	Combined-cycle plant / Waste-heat boiler
kl	At the generator terminals
LP	Low pressure
LS	Live steam
MP	Medium pressure
PS	Process steam
S	Steam
SF	Supplementary firing
ST	Steam turbine, steam process
Sup	Supplied
WB	Waste-heat boiler
WS	Water/steam circuit
0	Design point
1	Gas turbine inlet
2	Gas turbine outlet
3	Flue gas temperature after supplementary firing
α	Stage group of a turbine: inlet
ω	Stage group of a turbine: outlet

APPENDIX I

CALCULATION OF THE OPERATING PERFORMANCE OF COMBINED-CYCLE INSTALLATIONS
(Refer to Section 7.1)

1. Equations for the heat exchangers

The equations of energy, impulse, and continuity are used to calculate the steady-state behavior of economizers. The continuity equation comes down in the steady state to:

$$\Sigma \dot{m} = 0. \tag{24}$$

The impulse equation can be simplified into:

$$\Delta p = \quad (\text{geometry}) \tag{25}$$

However, because the pressure losses both in the economizer and in the evaporator has a negligible influence on the energy equations, the assumption

$$\Delta p = 0. \tag{26}$$

is valid. In this case, the pressures along the heat exchanger remain constant, on both the gas and water sides. The energy equation for a small section dx of a heat exchanger, which can be treated approximately as a tube, can be written as follows:

$$\mathrm{d}\dot{Q} = k \cdot \Delta t \cdot \pi \cdot d \cdot \mathrm{d}x. \tag{27}$$

If it is assumed that the heat transfer coefficient k remains constant over the entire length of the heat exchanger (economizer or evaporator), Equation 27 becomes:

$$\dot{Q} = k \cdot S \int_0^L \Delta l\,(x). \tag{28}$$

In the general case, the expression $\int_0^L \Delta l\,(x)$ cannot be integrated. The heat exchanger must therefore be dealt with in the small element.

In the special cases of a heat exchanger with counter or parallel flow, however, integration is possible assuming that the specific heat capacities of both media along the heat exchanger remain constant.

The result of the integration is the logarithmic average value for the difference in temperature, which can be written in the form:

$$\int_0^L \Delta l\,(x) = \frac{\Delta t_E - \Delta t_A}{\ln\left(\frac{\Delta t_E}{\Delta t_A}\right)} = \Delta t_m. \tag{29}$$

This average value can also be used for a recuperator or an evaporator. The heat exchangers do not, in fact, operate in accordance with an ideal counterflow principle, but the errors remain negligible.

Substituting Equation (29) into Equation (28) yields:

$$\dot{Q} = k \cdot S \cdot \Delta t_m. \tag{30}$$

From Equation (24), the amount of heat exchanged can be expressed as follows:

$$\dot{Q} = \dot{m}_S \cdot \Delta h_S = \dot{m}_G \cdot \Delta h_G. \tag{31}$$

At the design point, Equations (30) and (31) become:

$$\dot{Q}_0 = k_0 \cdot S \cdot \Delta t_{m0}. \tag{32}$$

$$\dot{Q}_0 = \dot{m}_{S0} \cdot \Delta h_{S0} = \dot{m}_{G0} \cdot \Delta h_{G0}. \tag{33}$$

Dividing Equation (30) by Equation (32) and Equation (31) by (33) yields the formulas:

$$\frac{\dot{Q}}{\dot{Q}_0} = \frac{k \cdot \Delta t_m}{k_0 \cdot \Delta t_{m0}} \tag{34}$$

$$\frac{\dot{Q}}{\dot{Q}_0} = \frac{\dot{m}_G \cdot \Delta h_G}{\dot{m}_{G0} \cdot \Delta h_{G0}}. \tag{35}$$

Subtracting Equation (35) from (34) produces:

$$\frac{\Delta t_m}{\Delta t_{m0}} = \frac{k_0 \cdot \dot{m}_G \cdot \Delta h_G}{k \cdot \dot{m}_{G0} \cdot \Delta h_{G0}}. \tag{36}$$

This is the non-dimensional, global equation of heat transfer for the heat exchanger. If, in addition, Equation (31) is taken into consideration and the heat flow coefficient k is known, a system of equations is obtained that defines the heat exchanger.

2. Finding the heat flow coefficient

This can be calculated using the following equation:

$$k = \frac{1}{\frac{1}{\alpha_G} + \frac{d_1}{2\,\lambda} \ln \frac{d_1}{d_2} + \frac{d_1}{d_2 \cdot \alpha_S}}. \tag{37}$$

However, the relative values k/k_0 appear in the heat transfer equation. From this:

$$K = \frac{k}{k_0} = \frac{\frac{1}{\alpha_{G0}} + \frac{d_1}{d_2 \cdot \alpha_{S0}}}{\frac{1}{\alpha_G} + \frac{d_1}{d_2 \cdot \alpha_S}}. \tag{38}$$

The heat transfer coefficients on the gas end of the economizer and the evaporator (α_G) are from 0.1 to 0.01 times as large as those on the steam end (α_S). Moreover, both values always shift in the same direction (+ +, − −)

For these reasons, the following relationship may be used:

$$K = \frac{k}{k_0} = \frac{\alpha_G}{\alpha_{G0}}. \tag{39}$$

The α-value on the gas end can be calculated as follows using the Nusselt number:

$$Nu_G = C \cdot Re^m \cdot Pr^n = \frac{\alpha_G \cdot d_1}{\lambda_G}. \tag{40}$$

Here, C, m, and n are constants that depend mainly upon the geometry involved. From this, the following expression is obtained:

$$\alpha_G = C' \cdot \lambda_G \cdot Re^m \cdot Pr^n. \tag{41}$$

If this is substituted into Equation (39), the geometric constant C' disappears:

$$K = \frac{\lambda_G \cdot Re^m \cdot Pr^n}{\lambda_{G0} \cdot Re_0^m \cdot Pr_0^n}. \tag{42}$$

For gases, the Prandtl number is almost exactly a constant. Therefore:

$$K = \frac{\lambda_G}{\lambda_{G0}} \cdot \left(\frac{Re}{Re_0}\right)^m. \tag{43}$$

For the Reynolds number, the following expression applies:

$$Re = \frac{c_G \cdot \rho_G \cdot d_1}{\eta_G}. \tag{44}$$

By substituting m_G/S for $c_G\ Q_G$, one obtains:

$$Re = \frac{\dot{m}_G \cdot d_1}{\eta_G \cdot S}. \tag{45}$$

Then, substituting this expression into Equation (43), the geometric parameters disappear:

$$K = \frac{\lambda_G}{\lambda_{G0}} \left(\frac{\dot{m}_G \cdot \eta_{G0}}{\dot{m}_{G0} \cdot \eta_G}\right)^m. \tag{46}$$

If the mass flow is constant, all that remains is:

$$K = \frac{\lambda_G}{\lambda_{G0}} \cdot \left(\frac{\eta_{G0}}{\eta_G}\right)^m. \tag{47}$$

For m, one can use 0.57 for pipes that are offset from and 0.62 for pipes that are lined up with one another.

The value of the expression $\frac{\lambda_G}{\lambda_{G0}} \cdot \left(\frac{\eta_{G0}}{\eta_G}\right)^m$,

does not vary greatly and depends practically only on the properties of the gas. It can be replaced with the following approximation:

$$\frac{\lambda_G}{\lambda_{G0}} \cdot \left(\frac{\eta_{G0}}{\eta_G}\right)^m = 1 - (\bar{t}_0 - \bar{t}) \cdot 5 \cdot 10^{-4} \tag{48}$$

(in SI-Units)

$\bar{t}_0$ and are $\bar{t}$ the average gas temperatures along the heat exchanger in the design and operating point. This produces for the relative value of K :

$$K = \left(\frac{\dot{m}_G}{\dot{m}_{G0}}\right)^m \cdot [1 - (\bar{t}_0 - \bar{t})] \cdot 5 \cdot 10^{-4}. \tag{49}$$

In this equation, only m depends to a slight extent on the geometry of the boiler.

It is more complicated to calculate an exact value for K in the case of a superheater because the heat transfer on the steam end is poorer than that in the evaporator.

When all of these equations have been obtained for all parts of the boiler, the waste-heat boiler has been defined mathematically. Similar equations can also be formulated for calculating the condenser.

When calculating the economizer and the evaporator of a drum boiler, the problem arises that the state of the feedwater at the inlet to the drum is not clearly defined. For that reason, two different cases must be considered:

- The feedwater is supplied with steam at its entry into the drum (partial steam out in the economizer)
- The feedwater is undercooled at its exit from the economizer and must be heated to saturation temperature in the drum.

3. The Steam Turbine

Most steam turbines in combined-cycle plants operate in sliding pressure operation and generally have no control stage with nozzle groups. This simplifies calculations, because simulation of the control stage and the inlet valves is fairly complicated.

A portion of a steam turbine with no extraction is defined by one equation for its absorption capacity and one for its efficiency. The absorption capacity is approximately using the Law of Cones. In general, according to Ref. [1]: (50)

$$\frac{\dot{m}_S}{\dot{m}_{SO}} = \frac{\overline{M} \cdot p_\alpha}{\overline{M}_0 \cdot p_{\alpha 0}} \cdot \sqrt{\frac{p_{\alpha 0} \cdot v_{\alpha 0}}{p_\alpha \cdot v_\alpha}} \cdot \sqrt{\frac{1 - \left(\frac{p_\omega}{p_\alpha}\right)^{\frac{n+1}{n}}}{1 - \left(\frac{p_{\omega 0}}{p_{\alpha 0}}\right)^{\frac{n+1}{n}}}}$$

In steam turbines, the pressure ratio is always very small. This makes it possible to replace the quadratic expression with 1. The ratio of the absorption capacities is likewise close to 1.

What remains is then:

$$\frac{\dot{m}_S}{\dot{m}_{SO}} = \sqrt{\frac{p_\alpha \cdot v_{\alpha 0}}{p_{\alpha 0} \cdot v_\alpha}} \quad (51)$$

At a constant rotational speed, the efficiency of a stage depends only upon the enthalpy drop involved. In part-load operation, however, no relatively great change occurs in that gradient except in the last stages. Because this means that the greatest portion of the machine is operating at a constant efficiency, it can be assumed that the polytropic efficiency remains constant. The turbine efficiency is calculated in the same way as for the design point.

The following formulas are used to calculate efficiency:

For parts of the turbine operating in the superheated zone:

$$\eta_{pol.\,tr.} = \textbf{constant} \quad (52)$$

For parts in the saturated steam:

$$\eta_{pol} = \eta_{pol.\,tr.} - \frac{(1 - x_\alpha) + (1 - x_\alpha)}{2}. \quad (53)$$

The polytropic efficiency selected should be such that the design power output is once again actually attained in the design point.

The following equation is used to determine the adiabatic efficiency:

$$\eta_{is} = \frac{1 - \left(\frac{p_\omega}{p_\alpha}\right)^{\frac{x-1}{x}\eta_{pol}}}{1 - \left(\frac{p_\omega}{p_\alpha}\right)^{\frac{x-1}{x}}}. \quad (54)$$

These equations make it possible to establish the expansion line of the steam turbine. The power output of the steam turbine can be determined from this by allowing for dummy piston, exhaust, generator, and mechanical losses. The dummy piston losses in single-flow reaction turbines are approx. proportional to the live steam pressure, and are typically between 400 to 600 kW. Mechanical losses range from 150 to 250 kW and exhaust losses at full load are generally in the range of 20 to 35 kJ/kg steam.

4. Solving the System of Equations

Taken together, all the equations in the waste-heat boiler, the steam turbine, etc. produce a system which can only be solved by iteration.

The following values are known:

- thermodynamic data in the design point
- the marginal conditions for the particular operation to be calculated (exhaust data for the gas turbine, cooling water data, etc.)
- operating mode of the feedwater tank (sliding or fixed pressure)
- Gas and Steam Tables

The following information must be found:

- behavior of the steam circuit

Fig. Appendix-1 shows the method used for solution. One starts with the superheater, inputting into the computer a first estimate for live steam temperature and pressure. Using the Law of Cones and the energy equation, one can then calculate the live steam flow and the gas temperature following the superheater. Next, from the heat transfer equation, a new value for live steam temperature can be determined. This is then used

for further iteration. The procedure is repeated until all three equations have been fulfilled.

The energy and heat transfer equations for the economizer and the evaporator can be used to determine a second approximation for live steam pressure.

If the feedwater tank is in sliding pressure operation, a first estimate for feedwater temperature is also necessary.

The new value obtained for live steam pressure is then used to continue calculation of the superheater and the turbine until all equations for the boiler and the Law of Cones agree. The next step is to calculate the preheating of the feedwater. This is used—if the pressure in the feedwater tank varies— to find a new approximation for feedwater temperature. The boiler is then recalculated, using this new value. Finally, the condenser pressure and extraction flow are determined in another iteration. Then, from this information, one can determine the power output of the steam turbine.

Figure Appendix-1

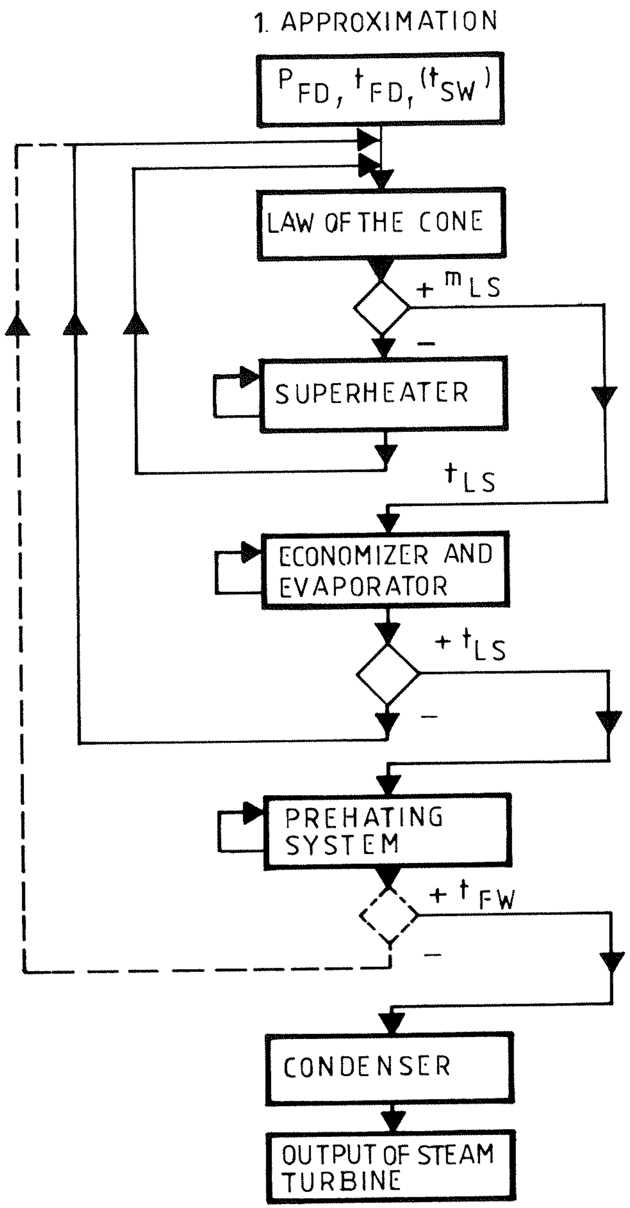

Fig. Appendix 1: Calculation of Operating and Part-Load Behavior: Method for Solving the System of Equations

DEFINITION OF TERMS AND SYMBOLS

The selection of terms and symbols below has been based on national and international definitions and expanded— where it appeared necessary— with additional terms and symbols. The selection was adapted to the special technical requirements of this book.

Term	Symbol	Unit	Definition
Annual service hours (Annual operating time)	$T_{ann.}$	hr/yr	Number of hours per year (operating time per year) during which a unit (or a group of units) was or is to be operated, continuously or with interruptions
Annual utilization time for nominal output (utilization time, utilization hours)	T_{Nj}	hr/yr	The annual utilization time for nominal output is obtained by dividing the operating output during the operating period by the nominal power output.
Approach temperature	ST	K	Undercooling of the feedwater at the inlet to the boiler drum (difference between actual temperature of feedwater and saturation)
Availability factor in terms of nominal work (Energy)	k_{AN}	-	The quotient obtained by dividing the work available $P_V \cdot T_V$ by the nominal work $k_{AN} = \frac{P_V \cdot T_V}{A_N}$

Term	Symbol	Unit	Definition
Availability (time)	η_T		The availability (time) of a power station or power plant unit is obtained by dividing the availability (the sum of operating time plus time at readiness) by the nominal time:
"Bottoming cycle"			A thermal process that operates in the lower range of temperature, following after a high temperature process
Efficiency of power generation	η_{EL}		In cogeneration plants, the efficiency of power generation is obtained by dividing the electrical power output by the amount of additional fuel supplied. This additional fuel is required because electricity is being produced in addition to the process heat.
Efficiency of the steam process	η_{ST}		The efficiency of the steam process is obtained by dividing the electrical power output of the steam turbine by the heat supplied to the steam process. In a process with waste heat utilization alone, it can be found using the formula:
Efficiency of the steam/ water cycle	η_{SW}		The efficiency of the steam/ water cycle is obtained by dividing the electrical output from the steam turbine generator by the amount of heat supplied to the water or steam in the boiler.

Term	Symbol	Unit	Definition
Power coefficient	PC or σ	kWs/kJ kWs/kJ	The flow coefficient of a plant cogenerating heat and electricity is obtained by dividing the net electrical power generated in a given time span by the usable heat generated in that same time span, both limited to the limit of the plant
Exergy	E(e)	kJ/kg	The maximum technological work obtainable from a system in accordance with the Second Law of Thermodynamics if the system is brought reversibly into equilibrium with its environment.
Exhaust loss of the steam turbine	k_W	kJ/kg	Non-recoverable losses due to kinetic energy in the exhaust steam of the turbine
Forced outage rate (FOR)	P_T		The forced outage rate of power stations, power plant blocks, or their components is obtained by dividing their outage time due to malfunction by the sum of the operating time plus outage time due to malfunction:
Generator output	P_{GEN}	kW (kVA)	The generator output of a power station or a power plant block is the power available at the generator terminals. The generator output is the gross output. $P_{GEN} = P_{GROSS}$

Term	Symbol	Unit	Definition
Heat output	$\dot{Q}_H$	kW	The heat output to cover non-block-connected heat demand, e.g., heat supplied to a district heating system
ISO conditions			Standard environmental conditions per ISO: Total air temperature 15 °C Total air pressure 1.013 bar Relative humidity 60 %
Pinch point of a waste heat boiler	Γ	K	The minimum difference in temperature between the exhaust gas and the water or steam is a waste boiler.
Rate of fuel utilization	η_n	kJ/kJ	The rate of fuel utilization in a plant cogenerating heat and electricity is equal to the quotient obtained by dividing the sum of net electrical power and usable heat generated in a given time span by the energetic equivalent of the fuel supplied in the same time span.
Rate of waste heat utilization	η_{WB}	-	The quotient obtained by dividing the heat supplied to the water or steam by the waste heat available to the waste heat boiler
Rate of work utilization	η_{AE}	-	The rate of work utilization of a production capacity in a given time span is the quotient of the production in that time span divided by the work which the same unit could have produced

Term	Symbol	Unit	Definition
			with the full production capacity continually in operation. The two work measurements must be of the same type, gross or net.
Reliability (time)	z_T		The reliability (time) is obtained by dividing the operating time by the sum of operating time plus outage time due to malfunctions:
Station service power	$P_{EIG(EL)}$	kW	The station service power of a power station or power plant block is the amount of power required to drive all motor-driven block auxiliaries and ancillaries (power consumption of the motors), plus the electrical losses in station service transformers and electrical transmission losses within the power station.
Thermal efficiency	η_{kl}		The thermal efficiency of a power station or a power plant block generating electricity alone is obtained by dividing the electrical power output by the amount of energy supplied.
"Topping cycle"			A thermal process operating in the upper range of temperatures, followed by a low temperature process.
Wetness or moisture losses			The energy losses due to wetness in the wet steam section of the turbine

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