

Combined-Cycle Gas Steam Turbine Power Plants

3rd Edition

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PennWell®

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Contents

1. Introduction	1
2. The Electricity Market	5
3. Economics	11
4. Thermodynamic Principles of the Combined-Cycle Plant	35
5. Combined-Cycle Concepts	45
6. Applications of Combined Cycles	135
7. Components	165
8. Control and Automation	211
9. Operating and Part Load Behavior	225
10. Environmental Consideration	261
11. Developmental Trends	277
12. Integrated Gasification Combined Cycle	287
13. Carbon Dioxide Capture and Storage	321
14. Typical Combined-Cycle Plants	349
15. Conclusion	389
Appendix A: Conversions Table	393
Appendix B: Calculation of the Operating Performance of Combined-Cycle Installations	403
Appendix C: Symbols Used	405
Bibliography	407
Index	413
About the Authors	474

1 Introduction

This third edition of the book *Combined-Cycle Gas & Steam Turbine Power Plants* has been updated and extended to give an accurate picture of today's state of this interesting technology for power generation. It also includes chapters on actual themes such as CO₂ capture and storage, as well as integrated gasification combined-cycle plants (IGCC). These topics have gained a lot of attention as part of the discussion around global warming as potential solutions to this issue.

In substance, the book gives a comprehensive overview about the combined-cycle power plant from a thermodynamic, technical, and economical point of view. It is intended to provide material for lectures and provide an excellent understanding of the potential of this technology. Thanks to practical examples, it offers a real help for professional work. It is equally well suited for students interested in power generation.

The book strives to answer to the following two questions:

- What is a combined-cycle power plant?
- Why are combined-cycle plants among the leading technologies for large power plants?

Combined cycle can be defined as a combination of two thermal cycles in one plant. When two cycles are combined, the efficiency that can be achieved is higher than that of one cycle alone. Thermal cycles with

the same or with different working media can be combined; however, a combination of cycles with different working media is more interesting because their advantages can complement one another. Normally, when two cycles are combined, the cycle operating at the higher temperature level is called the *topping cycle*. The waste heat it produces is then used in a second process that operates at a lower temperature level, and is therefore called the *bottoming cycle*.

Careful selection of the working media means that an overall process can be created, which makes optimum thermodynamic use of the heat in the upper range of temperatures and returns waste heat to the environment at the lowest temperature level possible. Normally the topping and bottoming cycles are coupled in a heat exchanger.

The combination used today for commercial power generation is that of a gas topping cycle with a water/steam bottoming cycle. Figure 1–1 shows a simplified flow diagram for such a cycle, in which the exhaust heat of a simple cycle gas turbine is used to generate steam that will be expanded in a steam turbine.

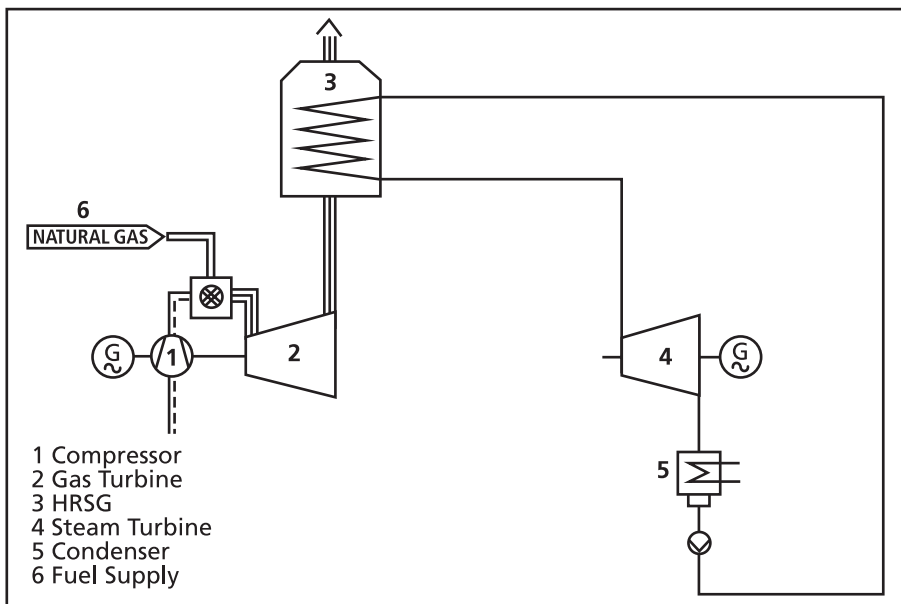


Figure 1–1 Simplified flow diagram of a combined cycle

Replacement of the water/steam in this type of cycle with organic fluids or ammonia has been suggested in the literature because of potential advantages over water in the low exhaust-gas temperature range. As gas turbine exhaust temperatures are increased in line with gas turbine development, however, these advantages become insignificant compared to the high development costs and the potential hazard to the environment through problems such as ammonia leakage. These cycles do not appear likely ever to replace the steam process in a combined-cycle power plant.

The subject of this book is mainly the combination of an open cycle gas turbine with a water/steam cycle. This combination—commonly known as the combined cycle—has several advantages:

- Air is a straightforward medium that can be used in modern gas turbines at high turbine inlet temperature levels (above 1200°C/2192°F), providing the optimum prerequisites for a good topping cycle.
- Steam/water is inexpensive, widely available, non-hazardous, and suitable for medium and low temperature ranges, being ideal for the bottoming cycle.

The initial breakthrough of these cycles onto the commercial power generation market was possible due to the development of the gas turbine. Only since the late 1970s have gas turbine inlet temperatures—and hence, exhaust-gas temperatures—been sufficiently high to allow the design of high-efficiency combined cycles. This breakthrough was made easier because the components of the plant were not new, having already been proven in power plant applications with simple cycle gas turbines and steam turbine processes. This helped to keep development costs low. The result was a power plant with high efficiency, low installation cost, and a fast delivery time.

Today, state of the art combined-cycle plants can reach net efficiency based on the lower heating value (LHV) of the fuel above 58%, and it is only a question of a few years until 60% will be overcome.

2 The Electricity Market

Basic Requirements

The fundamental difference between electricity and most other commodities is that electricity cannot be stored in a practical manner on a large scale. Storing electricity directly is very expensive and can be done only for small quantities (e.g., car batteries). Indirect storage through water or compressed air is more suitable for large-scale applications, but it is very much dependent on topography or geology and is, in many cases, not economical. Actually only storage of water pumped into lakes during off-peak time to be used during peak hours has reached market acceptance.

For this reason, electricity must be produced when the customers need it. It has to be transported by means of extensive transmission and distribution systems, which help to stabilize and equalize the load in the system. Nevertheless, large fluctuations in demand during the day require quick reactions from generation plants to maintain the balance between demand and production.

Fulfillment of this task has been the main focus of the industry from the beginning. A reliable supply of electricity, efficiently delivered, was and is the major priority.

In the last few years a new priority has been set by a global trend to deregulate the electric power market. Deregulation means an opening to competition of what has been a generally closed and protected

industry. Private investors have begun to install their own power plants and supply power to the grid. This has created a major focus shift: generators now have to compete to sell their product.

In the past—in a regulated environment—generators sold the power they generated on a cost-plus basis. Production costs were of lower priority than the requirement of grid stability and reliability. Generators focused on reliability through extensive specifications and sufficient system redundancy. Costs could usually be transferred to the end-use customers. The new competitive situation has altered key success factors for the electricity generators.

Today, overall production cost is a key to their success. They must offer electricity at the lowest cost, yet still meet the requirement of flexible adjustment between demand and supply. This cost factor is of major importance for the merchant plants. These are plants built by investors who accept the full market risk, expecting that their assets will be cost competitive, and that they will get a good return on their investment during the lifetime of the plant. Risk and its mitigation carry a much higher weight in this environment.

On one side of this equation, higher competitive risks must be taken to survive in the new markets. On the other side, risks such as cost and schedule overruns have to be minimized. These factors lead us to new behaviors, one of which is that electricity generators are buying plants for fixed, lump-sum prices with guaranteed throughput times. Constructing new plants—with long lead times and high capital costs—are considered to be risky from the private investor's point of view.

Combined-cycle power plants benefit from this change. They have low investment costs and short construction times when compared to large coal-fired stations, and even more so when compared to nuclear plants.

The other benefits of combined cycles are high efficiency and low environmental impact. Worldwide, levels of emissions of all kinds must meet stringent regulations acceptable to the public. It is, therefore, important for power producers to invest in plants with an inherently low level of emissions. Risk mitigation and public acceptance are paramount. Clean plants are easier to permit, build, and operate. Combined-cycle plants—especially those fired with natural gas—are

a good choice with their low emissions. (See chapter 10 for more information about this point.)

Throughout this book, we will review different combined-cycle applications, including cogeneration applications; however, more than 80% of the market concerns plants for pure power generation. We will examine how such plants affect and are affected by all of the variables discussed so far in this chapter.

The data cited in table 2–1 are valid for 2007. All figures are rounded to gigawatts (GW). The striking points in this table are:

- Steam turbine plant (STP) for coal-fired application represent about 30% of the total installed capacity worldwide.
- Combined-cycle plants have a total capacity of more than 800 GW, and are approaching a share of 20% compared to less than 5% ten years ago.

Table 2–1 Installed capacity 2007

		GW		
Steam turbines	Coal fired plant	1400		
	Gas fired plant	300		
	Oil fired plant	330		
	Other steam plant	70 ¹⁾		
	Nuclear power plant	395		
	Steam turbine for combined cycle	220		
		Total	2,715	
Gas turbines	GT in simple cycle	310		
	GT in combined-cycle	610		
		Total	920	
Hydro	Hydro power plant	780		
	Hydro pump storage plant	140		
		Total	920	
Diesel		Total	150	
New renewables ²⁾		Total	105	
		Total	4,810	

Source GS4, BUD 2008-03-07 and IRINA 2008-03-10

1) geothermal, solar, wind, wood and waste 2) coal, gas and oil fired plants

Table 2–2 shows a geographical breakdown of the worldwide electrical energy generated by technology by **terawatt hour (TWh)**.

Table 2–2 World net electricity generation by type 2004 (TWh)

	Americas	Asia & Oceania	Europe	Africa	Total
Thermal Plants ¹⁾	3,397.0	5,318.2	1,811.7	407.7	10,934.6
Hydro Plants	1,214.1	917.5	541.7	86.0	2,759.3
Nuclear Plants	902.0	717.7	980.9	14.3	2,615.9
Other ²⁾	140.1	58.6	140.8	2.0	341.5
Total	5,653.2	7,012.0	3,475.1	510.0	16,650.3

Source: Energy Information Administration: International Energy Annual, October 2007

1) coal, gas and oil fired plants 2) geothermal, solar, wind, wood and waste

The salient points are:

- Two-thirds of the electricity produced worldwide is generated by thermal plants. Coal alone represents more than 40% of the total.
- Combined cycles today generate more than 10%.
- Nuclear and hydro plants are each generating about 15% of the total.
- Other renewables account for about 2%.

Table 2–3 shows the breakdown of the thermal power plants by fuel consumed in 2007.

Table 2–3 World fuel consumption for fossil fuelled power plants

Worldwide consumption (TTOE ¹)		
Oil	283,419	9.6 %
Gas	815,693	27.5 %
Coal	1,862,587	62.9 %
Total	2,961,699	100.0 %

Source: 2005 Energy Balances for World, 2007

1) Thousand Tons of Oil Equivalent

Figure 2–1 shows the evolution of the installed new power plant capacity over the year. Striking are the peaks in 2000 and 2007. The first one is due to a boom in USA where substantial new capacity of combined-cycle plants were installed. The second peak in 2007 is due to large new construction in emerging countries such as China or India as well as in oil producing countries, and also the replacement of old power plants in Europe.

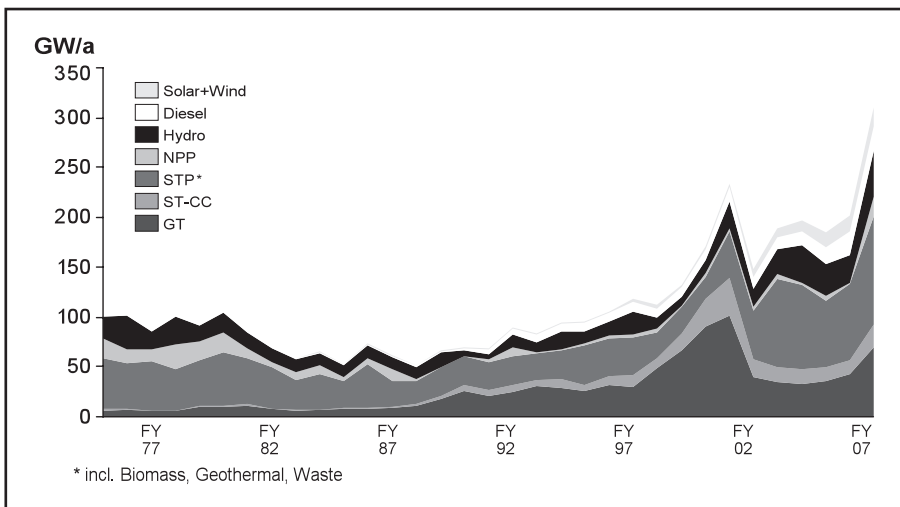


Figure 2–1 Market development since 1975 of new power plants sold per year

The Availability of Fuel

Critical for the future of combined-cycle power plants is the long-term availability of natural gas at reasonable price conditions.

Table 2–4 gives indication about the availability of reserves compared to production today for the main fossil fuels coal, gas and oil.

Taking into account all proven gas reserves, it can be calculated that at the present rate of production the global natural gas reserves will

last for approximately 60 years. This estimation shows that natural gas should still be available in the foreseeable future.

Table 2–4 Fossil fuels: proved reserves vs. yearly consumption worldwide in 2007

Fuel	Proved Reserves	Yearly Consumption	Ratio of Proved Reserves to Production
Natural gas	$170 \times 10^{12} \text{ Nm}^3$	$3,020 \times 10^9 \text{ Nm}^3/\text{a}$	56 a
Oil	$1,300 \times 10^9 \text{ barrels}$	$31 \times 10^9 \text{ barrels/a}$	42 a
Coal	$910 \times 10^9 \text{ ton}$	$6.2 \times 10^9 \text{ ton/a}$	147 a

Source: EIA and BP

3 Economics

In today's mostly liberalized electricity market, the economics of a power plant are evaluated as any other industrial investment.

Typically, an investor sets a target return on an investment; for instance, return on equity (ROE) or internal rate of return (IRR). Similarly to any other investment, this expected return depends on the risk profile of the project. The ROE depends on following factors:

- Market price of the electricity
- Production cost of electricity that includes
 - Interest and amortization cost
 - Fuel cost
 - Operation and maintenance cost

Next to the fuel cost, the **market price of electricity** is usually the biggest unknown factor. The market price in a liberalized economy can fluctuate widely.

Private investors, or independent power producers (IPP), generally try to find electricity off-takers who are buying the power at defined rates under long-term sales contracts (LTSC). Ideally the rate is coupled with the price of fuel, thus removing the most important investment risks.

Thanks to this project structure the investment risk is low, enabling the investor to get access to good credit conditions from financing institutions.

The risks remaining are mostly related to the plant itself: cost, efficiency and reliability.

The extreme case of this type of project is the **Tolling Agreement**. The investor has an agreement with a client, who supplies him with the fuel and takes off the electricity to convert the fuel into electricity at an agreed-upon fee, based on expected efficiency and availability.

The normal power plant project is, however, fully exposed to the market, which leads to a much higher risk profile. In this case, it is important to make a thorough market analysis in order to develop a solid market scenario. This analysis should include sensitivity analyses and worst-case scenarios.

In fossil fuel plants, both electricity and fuel markets have to be analyzed; however, both markets are coupled to a certain extent. For example, if in a given market most of the players are using gas as fuel, the price of electricity will go up if the gas prices are going up and vice versa.

The **interest and amortization cost** has a direct impact on the production cost of electricity produced. The main factors are:

- Debt-to-equity ratio
- Terms of the debt, such as interest and amortization

The debt-to-equity ratio depends mostly on the risk profile of the project. An IPP project with little market risk can be highly leveraged, which means a high percentage of the project financing can be provided in the form of debt. In good projects up to 80% of the capital required can be borrowed, and the equity share is only 20%.

In risky projects, a much lower leverage is possible. Typically, up to 50% of the capital required has to be provided as equity. The normal utility project falls in this category.

Normally, these merchant projects are realized by large utilities with a strong balance sheet. Financing is done based on the balance sheet of the investor (on-balance sheet financing). In this case, the borrower has the balance sheet of the owner of the power plant (the utility) as a guarantee.

On the other side, IPP projects are financed with project financing schemes. Here, a special purpose company (SPC) is created that owns only this power plant. Consequently, the borrower has only the power plants as a guarantee. It is important to have well-structured contracts for construction, operation and maintenance, power sales, and fuel supply to keep the risks at an acceptable level for the lender.

Generally, this type of project is realized by a general contractor under a fixed, lump-sum turnkey contract, also known as an engineering procurement and construction contract (EPC).

Another important factor is the type of power the plant is producing: The value of electricity varies over the year and over the day. Typically, peak power has a much higher value than off-peak power. Consequently, a plant designed to run at base load, such as a nuclear power plant or a coal fired power plant, will in average sell the power at a lower price than a power plant designed to run at peak time.

Typically, one distinguishes among following type of plants.

- Base load (>5000 hrs/a)
- Intermediate load (2000 to 5000 hrs/a)
- Peak load operation (<2000 hrs/a)

Plants with low capital cost but costly fuels are more suitable for peak load operation than those with high capital cost and cheap fuel, which are better suited for base load operation.

Plant cost and flexibility are important factors when defining for which kind of application a given power plant is suitable.

The terms of the financing depend on the market situation and the risk of the selected financing structure. Usually, a project financing structure leads to higher interest rates than a balance sheet financing, provided the borrower has a solid balance sheet.

Production Cost of Electricity

The production costs of electricity are generally expressed in US\$/MWh or US cts/kWh, depending on following parameters:

- Capital cost of the project
- Fuel cost
- Operation and maintenance cost

The **capital cost** per unit of electricity for a given power plant depends on following elements:

- Investment costs
- Financing structure (discussed earlier in this chapter)
- Interest rate and return on equity
- Load factor of the plant (or equivalent utilization time)

The **investment costs** are the sum of following positions:

- Power plant contract price(s)
- Interest during construction (depending upon the construction time)
- Owner's cost for the realization of the project (project manager, owner's engineer, land cost, etc.)

The **financing structure** is defined by the debt-to-equity ratio of the financing and the return on equity is the return expected by the investors on his capital. Both are linked to the risks of the project.

The load factor results from the type of application the plant is intended for: base, intermediate or peak load operation, and the availability and reliability of the power station.

Fuel costs per unit of electricity are proportional to the specific price of the fuel, and inversely proportional to the average electrical efficiency

of the installation. This average electrical efficiency must not be mixed up with the electrical efficiency at rated load. It is defined as follows:

$$\bar{\eta} = \eta \cdot \eta_{\text{Oper}} \quad (3-1)$$

where:

η is the electrical net efficiency at rated load. (This is the % of the fuel that is converted into electricity at rated load and new and clean condition)

η_{Oper} is the operating efficiency, which takes into account the following losses:

- start-up and shutdown losses
- higher fuel consumption for part load operation
- aging and fouling of the plant

Operation and maintenance costs consist of fixed costs of operation, maintenance and administration (staff, insurance, etc.), and the variable costs of operation and maintenance (cost of repair, consumables, spare parts, etc.).

The cost of electricity is calculated by adding the capital cost, fuel cost, and operation and maintenance costs.

Present value is generally the basis for economic comparisons among different types of plants. The various costs for a power station are incurred at different times, but for financial calculations are corrected to a single reference time. This is generally the date on which commercial operation starts. These converted amounts are referred to as present value.

The cost of electricity (US\$/MWh) is calculated as:

$$Y_{\text{EL}} = \frac{\text{TCR} \cdot \Psi}{P \cdot T_{\text{eq}}} + \frac{Y_{\text{F}}}{\bar{\eta}} + \frac{U_{\text{fix}}}{P \cdot T_{\text{eq}}} + u_{\text{var}} \quad (3-2)$$

where:

TCR total capital requirement to be written off (current value of all expenses during planning, procurement,

construction, and commissioning, such as the price of the plant, interest during construction, etc.) (US\$)

Ψ annuity factor: $\Psi = \frac{q-1}{1-q^{-n}} [1/a]$

P rated power output in MW

T_{eq} equivalent utilization time at rated power output, in hours per annum (h/a)

Y_F price of fuel (US\$/MWh thermal = $3.412 \times \text{US\$/MBTU}$)

$\bar{\eta}$ average plant net efficiency

U_{fix} fixed cost of operation, maintenance, and administration

u_{var} variable cost of operation, maintenance, and repair (US\$/MWh)

q $1+z$

z average discount rate in percent per annum (%/a)

n amortization in years

The equivalent utilization time at rated output is the electrical energy generated by a plant in a period of time divided by the electrical energy, which could be produced by the plant running the whole year at rated output (this is a measure for the load factor). This definition enables corrections to be made for the effects of part-load operation, so that different projects can be analyzed on a comparable basis.

For fuel, operation, and maintenance costs, no escalation rates have been applied to calculate the cost of electricity.

For modeling purposes, the cost of electricity can be calculated by using the equivalent utilization time and the fuel price as variables. In a deregulated power generation market, power stations do not sell their electricity based on an average cost of electricity basis, but rather on the basis of demand and supply. Therefore, it is important to understand that the previously referenced equation contains fixed and variable costs in order to avoid producing electricity in times when the sale price is lower than the variable costs.

Fixed costs are:

- Interest and depreciation on capital
- The fixed costs of operation, maintenance and administration (e.g., staff)

Variable costs are:

- The fuel cost
- The variable costs of operation, maintenance and repair (e.g., spare parts)

If the market price of the electricity is falling below the variable cost, the power plant should be shut down because no contribution to the fixed cost can be generated.

Competitive standing of combined-cycle power plants

On the following pages, the combined-cycle power plant is compared with other thermal plants. The comparison evaluates the following types of power stations:

- Steam turbine plants
- Gas turbine plants
- Nuclear plants
- Combined-cycle plants
- Biomass power plants

The range of output ratings under consideration is between 30 and 1250 MW. Combined cycles with a smaller output can, of course, be built, but they are less interesting for pure power generation because the specific costs increase very fast at lower rated output. Applications for heat and power production (e.g., district heating or industrial processes) can make sense.

Comparison of the power plant prices

Table 3–1 shows the specific price of the various types of power plants considered. These prices are valid for a power plant built under a turnkey contract with a fixed, lump-sum price. They are based on prices valid in 2007 assuming progress payment, without interest during construction and other owner’s costs.

Table 3–1 Specific price of various power plants in US\$/kW

Type of Plant		Specific Price (US\$/kW)
Combined Cycle Power Plant	(800 MW)	550–650
Combined Cycle Power Plant	(60 MW)	700–800
Gas Turbine Plant	(250 MW)	300–400
Gas Turbine Plant	(60 MW)	500–600
Steam Power Plant (coal)	(800 MW)	1200–1400
Steam Power Plant (coal)	(60 MW ¹)	1000–1200
Nuclear Power Plant	(1250 MW)	2000–3000
Biomass Power Plant	(30 MW)	2000–2500

1) Simple non-reheat plant

Remark: These prices are valid for 2007. They do not include owner’s cost and interest during construction.

The data shown indicate trends, so appropriate caution must be taken in applying the data because many factors affect the price of a power plant. These factors include market conditions, site-related factors, types of cooling and related structures, emission limits, labor rates, commercial risks, legal regulations, and so forth.

Table 3–1 also shows the low investment costs required for the gas turbine, which have contributed significantly to its wide-spread acceptance, especially in oil producing countries. Taken together with its simplicity and short start-up time to full load, the gas turbine is an attractive peak-load machine.

Steam power plants are more expensive than combined-cycle power plants. A coal-fired plant, for example, costs approximately two times more than a combined-cycle plant with the same output.

The costs of a nuclear power station are much higher and can vary from one country to the other due to the variety of safety requirements, permitting processes, and other local requirements. In addition, this type of plant has practically disappeared from the European and American markets because of heavy political opposition. A revival seems likely, but it is difficult at this time to give solid information on the cost of nuclear plants.

Figure 3–1 shows the breakdown of the cost of a 400 MW combined-cycle plant according to the different systems. Notice that the gas turbine makes up only one-third of the cost, but delivers two-thirds of the output. If we add the related auxiliary systems and civil work, the cost of the gas turbine island would make up approximately 50% of the total combined-cycle plant cost.

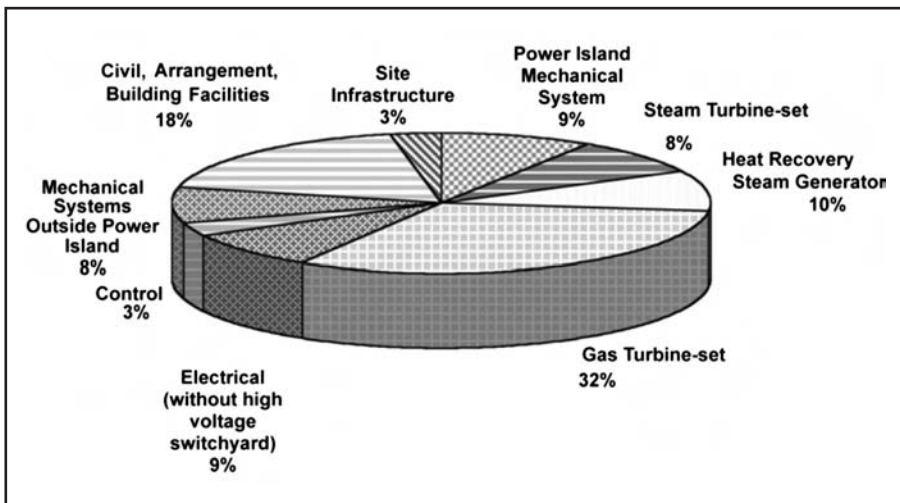


Figure 3–1 Breakdown of the capital requirement for combined-cycle power plant

Comparison of efficiency and fuel costs

At today's fossil fuel prices, efficiency is the most important cost factor for installations operated at intermediate or base load. If an expensive fuel is used, such as natural gas or liquefied natural gas (LNG), the

efficiency is crucial. For that reason high efficiency is a prerequisite for an economical fossil-fired plant. The net efficiency of a power station is defined as:

$$\eta = \frac{P_{\text{net}}}{\dot{Q}} \quad (3-3)$$

where:

P_{net} Power output at the high voltage terminals of the step-up transformer in MW. This number considers the power consumption of all plant auxiliaries

$\dot{Q}$ Heat input to the power station in MJ/s measured at the plant boundary (i.e., mass flow of the fuel multiplied by the lower heating value of the fuel (LHV))

Fuel that contains hydrogen produces water vapor when burned. If the combustion products are cooled to the point where this entire vapor is condensed, the maximum possible heat is extracted defining the higher heating value (HHV) of the fuel. In practice, this latent heat cannot be used in power plants, and the heat extracted is the lower heating value (LHV). The considerations in the following chapters are always based on the LHV of the fuel.

Table 3–2 indicates the electrical efficiency (new and clean) at nominal output of the different types of power plant. Steam turbine power plants have been subdivided into coal-fired and nuclear plants. The values for combined-cycle plants are valid for plants without supplementary firing. Table 3–2 points out the thermodynamic superiority of the combined-cycle plant. This was made possible, to a large extent, by the improvement of gas turbine technology, which already achieves efficiency of 38% to 40% in simple cycle. Only a few years ago, the efficiency of a newly installed coal-fired steam power plant was at these levels, but with much higher investment cost and complexity!

Coal-fired power plants are competitive for base load application in places where cheap coal can be easily transported.

Table 3–2 Net efficiency of various power plants

Type of Plant		Net Efficiency LHV (%)
Combined Cycle Power Plant	(800 MW)	55–59
Combined Cycle Power Plant	(60 MW)	50–54
Gas Turbine Plant	(250 MW)	38–40
Gas Turbine Plant	(60 MW)	35–42
Steam Power Plant (coal)	(800 MW)	(42–) 47
Steam Power Plant (coal)	(60 MW)	30–35
Nuclear Power Plant	(1250 MW)	35
Biomass Power Plant	(30 MW)	28–32

Based on LHV of the fuel at ISO ambient conditions and equipment new and clean

Nuclear power plants have a low efficiency because the light water reactor technology used today allows the generation of steam only with low temperature and pressure. Typically, it is saturated or slightly superheated steam with a pressure of approximately 60 bar (860 psig).

In the future, new technology using, for instance, high temperature reactors will lead to higher efficiency.

Because of its low efficiency, nuclear power needs large cooling towers that are approximately three times as large as for a combined-cycle plant with the same output.

The right selection of the fuel and the corresponding type of power plant is critical for the long-term economical operation of the plant.

The following considerations play a role for the selection of the type of fuel used in a given power plant project:

- Long-term availability of the fuel at a competitive price
- Alternative for the primary fuel as backup
- Risk of supply shortages due to political interference
- Environmental considerations that favor a clean fuel, such as natural gas
- Independence from a single fuel source
- Strategic reasons to use a domestic fuel
- Financing requirements (e.g., uninterruptible fuel supply)

Table 3–3 lists the fuels that can be burned in different types of power plants today.

Table 3–3 Fuel flexibility of various power plants

Fuel	Gas Turbine	Combined Cycle	Steam Power Plant	Biomass Power Plant
Natural gas / LNG	Yes	Yes	Yes ¹	No
Distillate (oil #2)	Yes	Yes	Yes ¹	No
Crude oil / heavy fuel	Yes ²	Yes ²	Yes	No
Coal	No	No ⁴	Yes	No
Refuse	No	No	Yes	Yes
Biomass	No	No	Yes	Yes
Coal gas, low calorific gas	Yes ³	Yes ³	Yes	No
Nuclear fuel	No	No	Yes	No

1) Due to the lower efficiency of steam power plant using this kind of fuel leads to high production costs 2) Heavy oil or crude oil can be burned in some older types of gas turbine 3) These fuels can be used in gas turbines. Modifications to the gas turbine are necessary for fuels with a low heating value, see also chapter 6 4) Can be used in an IGCC plant, see also chapter 12
 LNG stands for liquefied natural gas.

Some gas turbines can burn heavy oil or crude oil. Industrial (heavy-duty) gas turbines are more suitable for this type of fuel than those derived from jet technology (aero derivatives). Large combustion chambers are better capable of burning heavy fuels than those with several smaller burners/combustion chambers because the latter are more sensitive to changes in flame length, radiation, and so on. An additional requirement for burning heavy oil or crude oil in a gas turbine is the correct treatment of the fuel, generally by means of cleaning the fuel and/or dosing it with additives. These steps make it possible to remove or inhibit elements that cause high temperature corrosion, such as vanadium and sodium.

It must, however, be mentioned that with today’s environmental regulations, the use of heavy (residual) oil in gas turbines is practically impossible in industrialized countries.

Modern gas turbines with high firing temperatures are not designed for heavy fuel operation, but only for natural gas and distillate oil.

As can be seen from tables 3–1 and 3–2, combined-cycle plants are low in capital cost but burn an expensive fuel when compared to coal-fired steam power plants and nuclear plants. The biggest contribution in the cost of the electricity generated is, therefore, the one for the fuel costs. Long-term supply agreements or partnerships with fuel companies, or equity investments by fuel suppliers into the plant, are possible ways to cope with the risk of wide swings in the fuel prices.

Fuel flexibility is higher for a steam turbine power plant than a combined-cycle power plant. But combined-cycle plants are superior to steam power plants for power generation when gas or diesel oil is used, due to higher efficiency and lower investment costs.

As seen in figure 3–2, the prices of fossil fuels are highly volatile, especially gas and oil. This volatility must be duly considered during the economic evaluation of a project and, as already mentioned, it is important to understand how far the market prices of electricity are coupled with the prices of the fuel selected.

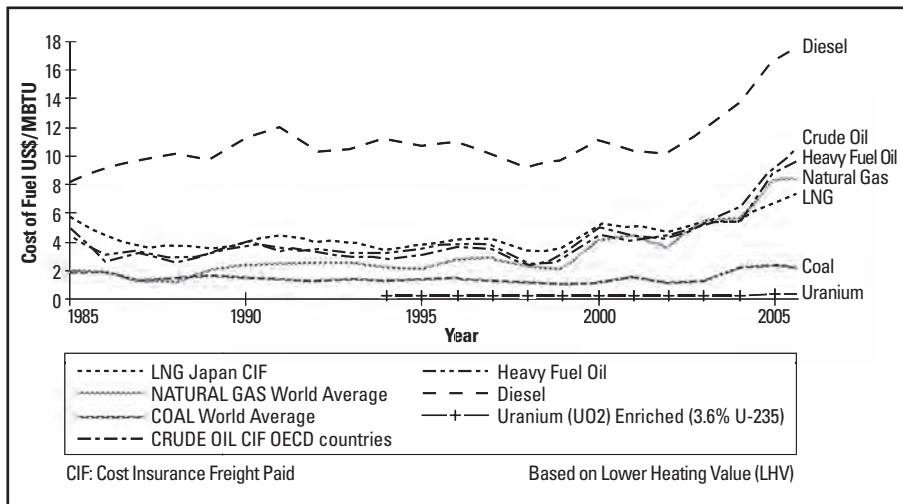


Figure 3–2 The cost of fuels over the years

Uranium, the fuel for nuclear power stations, and coal, the main fuel for steam power plants, have comparatively low costs. Combined-cycle power plants mainly use natural gas that is more expensive and generally more volatile than coal. Since 2004 all the prices of all fuels have increased substantially. Despite this apparent disadvantage, combined-cycle power plants are very popular with investors because they:

- Efficiently convert fuel into electricity
- Have a low impact on the environment
- Have high operation flexibility (startup time, high-load gradient)
- Have a short construction time
- Are easier to permit than other types of plants

Comparison of operation and maintenance costs

At current levels of fuel and capital cost, operation and maintenance costs affect the economy of a power plant only in a limited manner. They account for less than a tenth of the cost of electricity in a combined-cycle plant. Table 3–4 shows the variable operation and maintenance cost of the different power plants (cost depending upon the operation regime of the plant). Variable costs for a combined-cycle plant are lower than for gas turbine plants because these costs are driven by the high spare part cost of the gas turbine, which can be distributed over a larger output in the combined-cycle plant.

Table 3–4 Variable operating and maintenance costs for various power plants of different sizes

Type of Plant		Cost in US\$/MWh
Combined Cycle Power Plant	(800 MW)	2–3
Combined Cycle Power Plant	(60 MW)	3–4
Gas Turbine Power Plant	(250 MW)	3–4
Gas Turbine Power Plant	(60 MW)	4–5
Steam Power Plant (coal)	(800 MW)	2.5–3.5
Nuclear Power Plant	(1250 MW)	2.0
Biomass Power Plant	(30 MW)	5–8

Table 3–5 shows the fixed operation and maintenance costs of the different power plants (cost independent of the operating regime). These are mostly personnel and insurance costs.

Table 3–5 Fixed operation and maintenance costs for various power plants of different sizes

Type of Plant		Cost in Mio. US\$/a
Combined Cycle Power Plant	(800 MW)	6–8
Combined Cycle Power Plant	(60 MW)	3–4
Gas Turbine Power Plant	(250 MW)	2–2.5
Gas Turbine Power Plant	(60 MW)	1–1.5
Steam Power Plant (coal)	(800 MW)	12–15
Nuclear Power Plant	(1250 MW)	40–60
Biomass Power Plant	(30 MW)	3–4

Remark: These costs include the insurance cost.

Nuclear power stations and conventional steam turbine power plants require a much larger staff than for other kinds of plants, which explains the high fixed costs of these types of power plant.

Comparison of availability and reliability

The availability of a power plant is defined as:

$$AF = \frac{PH - SOH - FOH}{PH} \quad (3-4)$$

where:

PH hours of the period considered (normally one year)
which amounts to 8,760h

SOH scheduled outage hours for planned maintenance

FOH forced outage hours for unplanned outages and repairs

The reliability of a power plant is defined as:

$$RF = \frac{PH - FOH}{PH} \quad (3-5)$$

So reliability is the percentage of the time between planned overhauls where the plant is generating or is ready to generate electricity, whereas the availability is the percentage of total time where power could be produced.

Availability and reliability have a big impact on plant economy. When a unit is down, power must be generated in another power station or purchased from another producer. In each case, replacement power is generally more expensive. The power station's fixed costs are incurred whether the plant is running or not.

In deregulated markets, reliability is crucial. At peak tariff hours, a major portion of the income is generated and the plant must be reliable. Scheduled outages can be planned for off-peak periods when tariffs are close to or even below variable costs. Then only a small income loss results from the planned outages.

Typical average figures for the availability and reliability of well designed and maintained plants are indicated in table 3–6.

Table 3–6 Availability and reliability of various power plants

Type of Plant	Availability	Reliability
Combined cycle plant	90–94%	95–98%
Gas turbine plant (gas fired)	90–95%	97–99%
Steam turbine plant (coal fired)	88–92%	94–98%
Nuclear power plant	88–92%	94–98%

These figures are valid for plants operated at base load. They would be lower for peak- or intermediate-load operation because frequent startups and shutdowns reduce the lifetime of critical components and increase the scheduled maintenance and forced outage rates.

The major factors determining plant availability and reliability are:

- Design of the major components
- Engineering of the plant as whole, especially of the interfaces between the systems

- Mode of operation (whether base-, intermediate-, or peak-load duty)
- Type of fuel
- Qualifications and skill of the operating and maintenance staff
- Adherence to manufacturer's operating and maintenance instructions (preventive maintenance)

A high availability has a positive impact on the cost of electricity because it allows an operator to run a power plant with a higher utilization time per year and, therefore, achieve a higher income.

Comparison of construction time

The time required for construction affects the economics of a project. The longer it takes, the larger the capital employed because interest, insurance, and taxes during the construction period add to the price of the plant. It also creates an additional risk for the investor because the market forecast has to look more in the future than for a plant with a shorter realization time.

Table 3–7 shows the time required to build the various types of power plants.

A gas turbine in a simple-cycle application can be installed within the shortest time frame because of its standardized design. Gas turbines, therefore, help secure power generation in fast-growing economies. Additional time is needed for the completion of a combined-cycle plant. Combined-cycle plants can be installed in two phases, with the gas turbine running first in simple-cycle mode, and then in combined-cycle mode as the steam cycle becomes available. With this procedure, two-thirds of the power is available in the time required for a gas turbine installation. However, an outage of two to three months is needed to convert the gas turbine power plant from simple cycle to combined cycle.

Table 3–7 Construction times for various power plants

Type of Plant	Time in Months ³
Combined cycle power plant ²	20–30
Gas turbine plant ²	12–24
Steam power plant (coal)	40–50
Nuclear power plant	60–80 ¹
Biomass power plant	22–26

1) Very difficult to give good figures due to the lack of actual experience at least in Europe and USA 2) Depends very much on the market situation 3) From notice to proceed to commercial operation

Besides the actual construction time of a power plant, the time elapsing between start of the project until financial closing and its predictability are also important factors to be considered in the development of a project.

The time for the development of a project varies widely from one project to the other due to local factors. However, it is a fact that gas-fired combined-cycle power plants are easier to develop than coal-fired plants. The main reason is the easier permitting and lower risk of opposition against this kind of project.

Comparison of the cost of electricity

Based on the data presented in table 3–8 and equation (3–2), the cost of electricity has been calculated for various parameters of the following plants.

- Combined-cycle power plant (800 MW)
- Gas turbine plant (250 MW)
- Steam-power plant (coal) (800 MW)
- Nuclear-power plant (1250 MW)

Table 3–8 Inputs for economical comparison

Type of Plant	Description	Investment cost ¹ in US\$/kW	Average efficiency (LHV) in %	Fuel price ² in US\$/MBtu	Depreciation ¹ time in [a]
Combined cycle power plant	(800 MW) 2 × GT 1 × ST	750	56.5	8.0	25
Gas turbine plant	(250 MW) 1 × GT	413	37.5	8.0	25
Steam power plant coal	(800 MW) 1 × ST	1716	44	3.5	25
Nuclear power plant	(1250 MW) 1 × ST	3500	34.5	0.5	40

1) Including interest during construction and owner's cost 2) Fuel price based on LHV

Figure 3–3 shows the cost of electricity for the various plants running in base load (8000 hrs/a). The costs are broken down in fuel, operation and maintenance (O&M), and capital cost. The gas-fired plants, combined-cycle plants, and gas turbine plants show high fuel cost and comparatively low capital cost. On the other side, we have the nuclear power plant with its high capital cost but very low fuel cost. In between, we have the coal-fired plant.

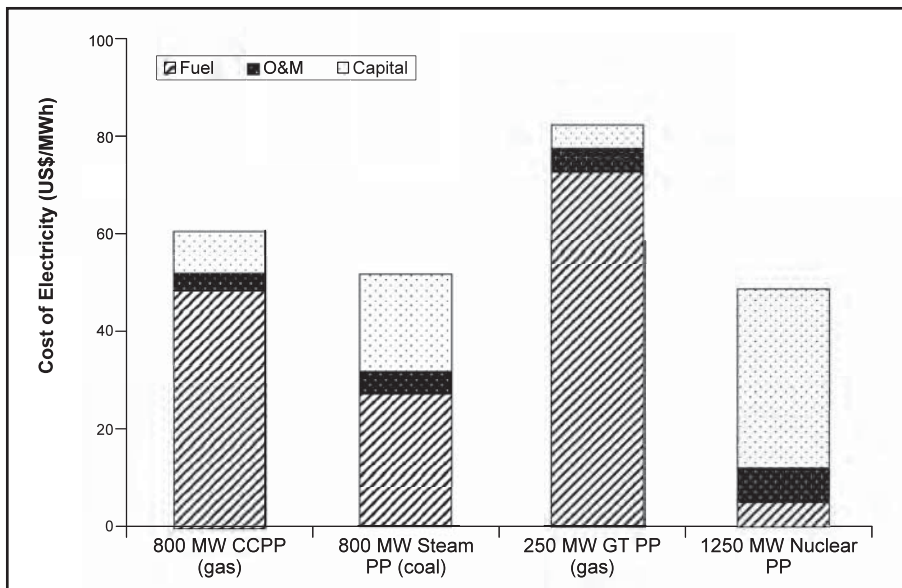
**Figure 3–3** Comparison of cost of electricity for base load operation

Figure 3–4 shows the same comparison but for an equivalent utilization time of only 4000 hrs/a. Here the combined-cycle power plants (CCPPs) show the lowest cost of electricity and the nuclear power plant the highest. Out of these figures, the obvious conclusion is that CCPPs are best for intermediate load, and coal-fired or nuclear power plants are well suited for base load application.

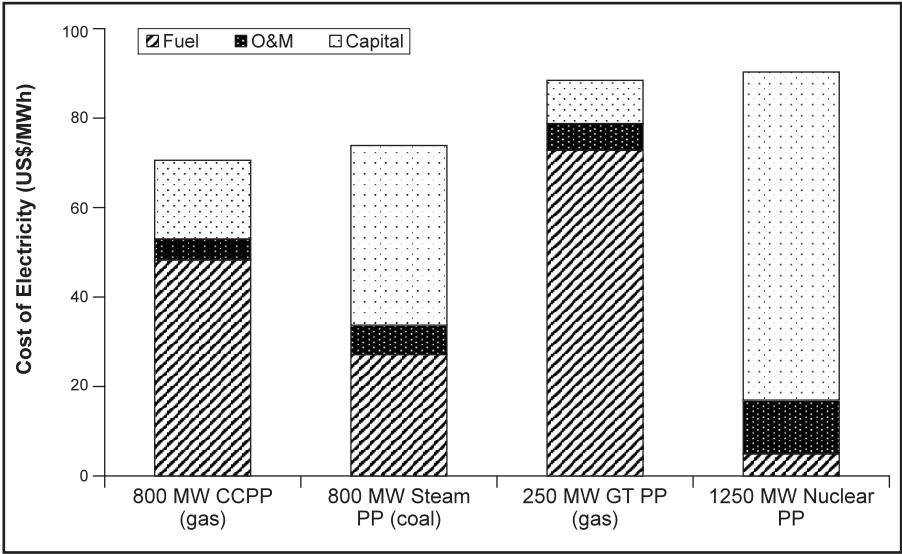


Figure 3–4 Comparison of cost of electricity for intermediate load

Figure 3–5 shows the influence of the charge of fuel cost on the cost of electricity. The CCPP and the gas turbine have the highest sensitivity, and the nuclear power plant has the lowest.

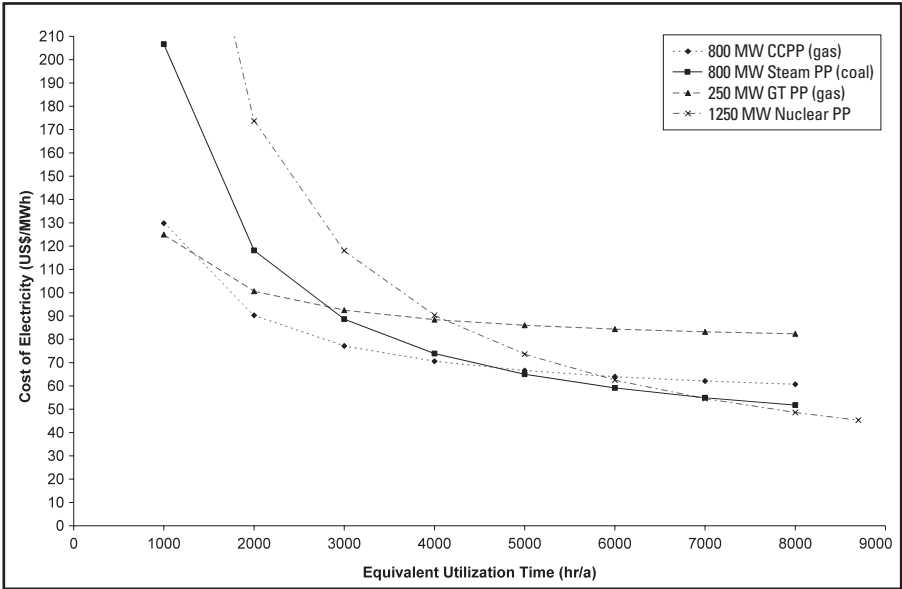


Figure 3-6 Influence of the equivalent utilization time on the cost of electricity

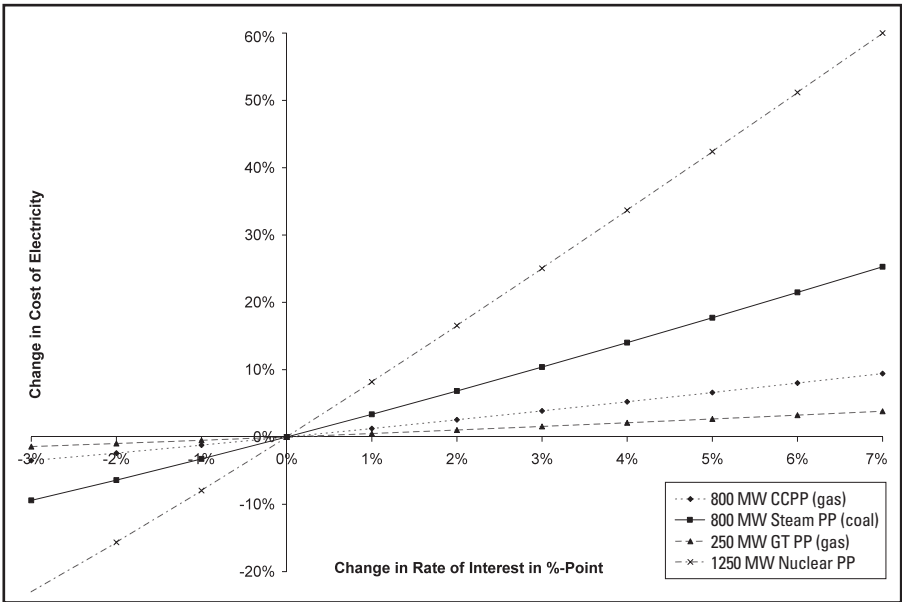


Figure 3-7 Influence of interest rate on the cost of electricity

The CO₂ emissions of the plant are having a more direct impact on the economics of a power plant due to the effort to globally limit these kinds of emissions. If we compare the various plants under this aspect, we get the following picture:

The combined-cycle plant emits about 40% of the CO₂ of a coal-fired plant.

Nuclear power plants or biomass plants are obviously still better because they produce nearly no additional CO₂ (see chapter 13).

No cost for CO₂ emissions were considered in the previous calculations.

Efficiency improvements

As noted, combined-cycle power plants are generally fired with natural gas, which is more expensive than uranium or coal. Consequently, the efficiency of the conversion of fuel into electricity is very important. The question of how much can additionally be invested in a combined-cycle plant to gain additional electrical efficiency is answered as follows.

The maximum additional capital that can be invested to increase the efficiency by 1% is given by the limit at which the cost of electricity remains constant.

Using equation (3–2) to determine the cost of electricity, the following calculations can be done.

$$\begin{aligned}
 Y_{EL} &= \frac{TCR_1 \cdot \Psi}{P_1 \cdot T_{eq}} + \frac{Y_F}{\eta_1} + \frac{U_{FIX1}}{P_1 \cdot T_{eq}} + u_{var1} \\
 &= \frac{TCR_2 \cdot \Psi}{1.01 \cdot P_1 \cdot T_{eq}} + \frac{Y_F}{1.01 \cdot \eta_1} + \frac{U_{FIX2}}{1.01 \cdot P_1 \cdot T_{eq}} + u_{var2}
 \end{aligned} \tag{3–6}$$

Index 1 is used for the plant that is used as the reference

Index 2 is used for the plant with a 1% improved efficiency and 1% improved output

Because operation and maintenance costs are approximately only a tenth of the cost of electricity and remain for both plants, the previous equation can be simplified to:

$$\frac{TCR_1 \cdot \Psi}{P_1 \cdot T_{eq}} + \frac{Y_F}{\bar{\eta}_1} \approx \frac{TCR_2 \cdot \Psi}{1.01 \cdot P_1 \cdot T_{eq}} + \frac{Y_F}{1.01 \cdot \bar{\eta}_1} \quad (3-7)$$

which can be solved for TCR_2 .

$$TCR_2 \approx TCR_1 \cdot 1.01 + \left(\frac{Y_F}{\bar{\eta}_1} - \frac{Y_F}{1.01 \cdot \bar{\eta}_1} \right) \frac{1.01 \cdot P_1 \cdot T_{eq}}{\Psi} \quad (3-8)$$

with:

$$\begin{aligned} TCR_1 &= 600 \text{ million U.S. \$} \\ Y_F &= 8.0 \text{ U.S. \$ /MBTU} \\ \bar{\eta}_1 &= 56.5\% \\ \Psi &= 9.37 \text{ (25 years/8\%)} \\ P_1 &= 800 \text{ MW} \\ T_{eq} &= 5000 \text{ hrs/a} \\ TCR_2 &= 626.6 \text{ million US\$} = 1.044 \times TCR_1 \end{aligned}$$

In this example, for a 1% efficiency increase, 4.4% more capital can be invested. It can be seen from previous equation that this percentage will increase if the fuel price is higher or a lower discount rate is used. A smaller percentage results for fewer operating hours per year or a shorter depreciation period.

These considerations are essential in optimizing a combined-cycle plant for a given application.

4 Thermodynamic Principles of the Combined-Cycle Plant

Basic Considerations

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

$$\eta_c = \frac{T_E - T_A}{T_E} \quad (4-1)$$

where:

η_c = Carnot efficiency [%]

T_E = Temperature of the energy supplied [K]

T_A = Temperature of the environment [K]

Naturally, the efficiencies of real processes are lower because there are losses involved. A distinction is drawn between energetic and exergetic losses. Energetic losses are mainly heat losses, and are thus energy that is lost from the process. Exergetic losses are internal losses caused by irreversible processes in accordance with the second law of thermodynamics.

The process efficiency can be improved by raising the maximum temperature in the cycle, releasing the waste heat at a lower temperature, or improving the process to minimize the internal exergetic losses.

The interest in combined cycles arises particularly from these 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 a simple-cycle gas turbine, attainable process temperatures are high as energy is supplied directly to the cycle without heat exchange. The exhaust heat temperature, however, is also quite high. In the steam cycle, the maximum process temperature is much lower than in the gas turbine process, but the exhaust heat is returned to the environment at a low temperature. As illustrated in table 4–1, combining a gas turbine and a steam turbine thus offers the best possible basis for a high efficiency thermal process.

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 is not the case in reality, 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 processes with only one cycle. Even a sophisticated supercritical conventional reheat steam turbine power plant has a Carnot efficiency around 20% lower than that of a good combined-cycle plant.

For combined-cycle power plants actual plant efficiencies are around 75% of the Carnot efficiency, whereas for conventional steam power plants this figure is around 80%. The differences between the actual efficiencies attained with a combined-cycle power plant and the other processes are, therefore, not quite as large as illustrated in table 4–1. The relatively larger drop in the combined-cycle efficiency is caused by higher internal energy losses due to the temperature differential for exchanging heat between the gas turbine exhaust and the water/steam cycle.

Table 4–1 Thermodynamic comparison of gas turbine, steam turbine and combined-cycle processes

	GT	ST	CC
Average temperature of heat supplied, K (°C)	1000–1350 (727–1078)	640–700 (368–428)	1000–1350 (727–1078)
Average temperature of exhaust heat, K (°C)	550–600 (278–328)	300–350 (28–78)	300–350 (28–78)
Carnot efficiency, %	45–55	45–57	65–78

GT = Gas Turbine Power Plant, ST = Steam Turbine Power Plant, CC = Combined Cycle Power Plant

As shown in figure 4–1, which compares the temperature/entropy diagrams of the processes, the combined cycle best utilizes the temperature differential in the heat supplied despite an additional exergetic loss between the gas and the steam process.

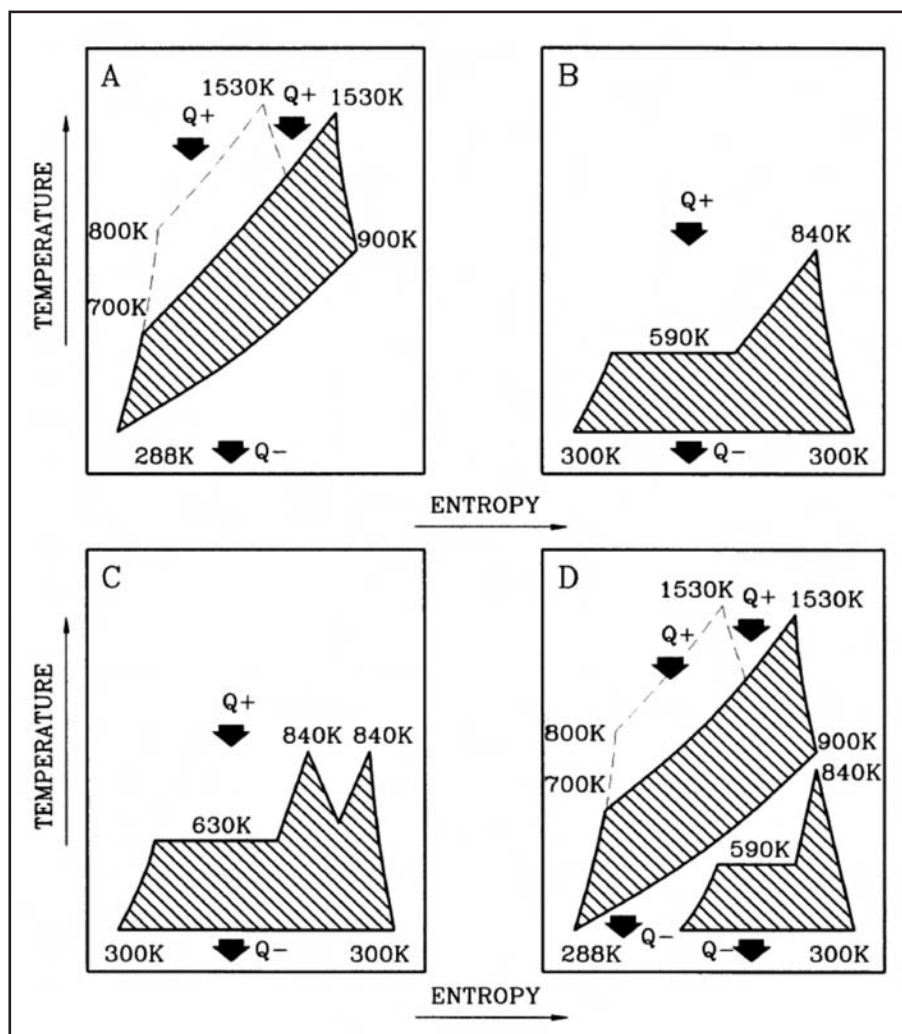


Figure 4–1 Temperature/entropy diagrams for various cycles

Definitions of gas turbine inlet temperature (TIT) are shown in figure 4–2.

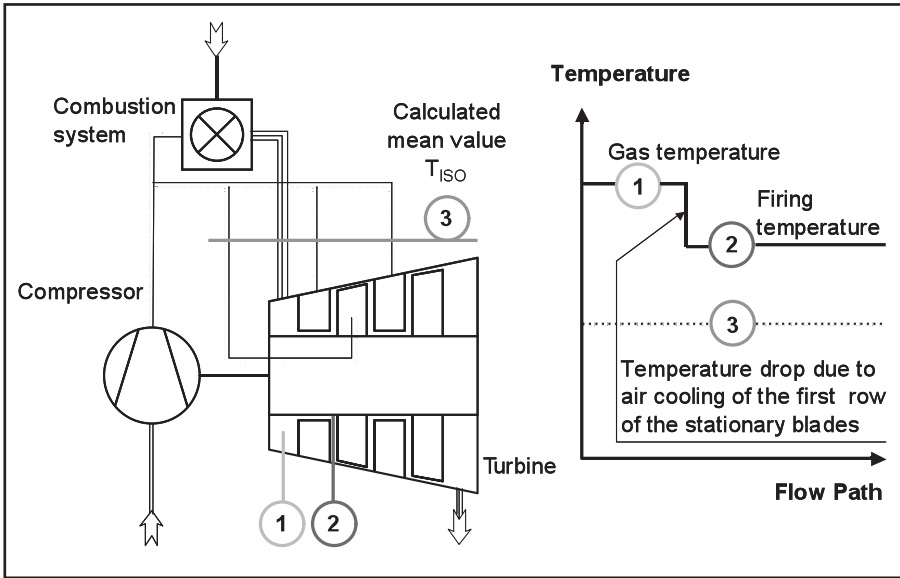


Figure 4–2 Gas turbine inlet temperature (TIT) definitions

All TIT values in this book are according to T_{ISO} (calculated mean value).

Efficiency of the Combined-Cycle Plant

It has been assumed until now that fuel energy is being supplied to the cycle only in the gas turbine. There are also combined-cycle installations with additional firing in the heat recovery steam generator (HRSG), in which a portion of the heat is supplied directly to the steam process.

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

$$\eta_{CC} = \frac{P_{GT} + P_{ST}}{\dot{Q}_{GT} + \dot{Q}_{SF}} \quad (4-2)$$

where:

P_{GT} Gas turbine output

P_{ST} Steam turbine output

$\dot{Q}_{GT}$ Gas turbine fuel consumption

$\dot{Q}_{SF}$ Additional/supplementary firing fuel consumption

This equation shows the gross efficiency of the combined cycle because no station service power consumption and electrical losses, also called auxiliary consumption (P_{Aux}), have been deducted. If station auxiliary consumption is considered the net efficiency of the combined cycle is given by:

$$\eta_{CC,net} = \frac{P_{GT} + P_{ST} - P_{Aux}}{\dot{Q}_{GT} + \dot{Q}_{SF}} \quad (4-3)$$

In general, the efficiencies of the simple cycle gas and steam turbine processes can be defined in a similar manner:

$$\eta_{GT} = \frac{P_{GT}}{\dot{Q}_{GT}} \quad (4-4)$$

$$\eta_{ST} = \frac{P_{ST}}{\dot{Q}_{GT,Exh} + \dot{Q}_{SF}} \quad (4-5)$$

where:

$$\dot{Q}_{GT,Exh} \cong \dot{Q}_{GT} (1 - \eta_{GT}) \quad (4-6)$$

Combining these two equations yields:

$$\eta_{ST} = \frac{P_{ST}}{\dot{Q}_{GT} (1 - \eta_{GT}) + \dot{Q}_{SF}} \quad (4-7)$$

This equation expresses the steam process efficiency of the combined cycle.

If there is no supplementary firing in the HRSG, equations (4-2) to (4-7) can be simplified by eliminating $\dot{Q}_{SF}$, ($\dot{Q}_{SF} = 0$). In view of earlier

considerations, it is generally better to burn the fuel directly in a modern gas turbine rather than in the HRSG because the temperature level at which heat is supplied to the process is higher (GT versus ST process in table 4–1). For that reason, the utilization of supplementary firing is decreasing.

The factors involved in combined-cycle installations with supplementary firing are discussed in more detailed in chapter 5.

Efficiency of combined cycles without supplementary firing in the HRSG

The most common and straightforward type of combined cycle is one in which fuel is supplied in the gas turbine combustion chamber without additional heat supplied in the HRSG. By substituting equations (4–4) and (4–7) into equation (4–2):

$$\begin{aligned}\eta_{CC} &= \frac{\eta_{GT} \dot{Q}_{GT} + \eta_{ST} \dot{Q}_{GT} (1 - \eta_{GT})}{\dot{Q}_{GT}} \\ &= \eta_{GT} + \eta_{ST} (1 - \eta_{GT})\end{aligned}\quad (4-8)$$

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

$$\frac{\partial \eta_{CC}}{\partial \eta_{GT}} = 1 + \frac{\partial \eta_{ST}}{\partial \eta_{GT}} (1 - \eta_{GT}) - \eta_{ST} \quad (4-9)$$

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

$$\frac{\partial \eta_{CC}}{\partial \eta_{GT}} > 0 \quad (4-10)$$

From equation (4–9):

$$-\frac{\partial \eta_{ST}}{\partial \eta_{GT}} < \frac{1 - \eta_{ST}}{1 - \eta_{GT}} \quad (4-11)$$

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 4–2 indicates that when the efficiency of the gas turbine is raised, the reduction in efficiency of the steam process may be greater. For example, steam process efficiency can be reduced from 30% to 27.8% (30/1.08) in case the gas turbine efficiency is raised from 30% to 35% to keep the overall combined-cycle efficiency.

Table 4–2 Allowable Reduction in Steam Process Efficiency as a Function of Gas Turbine Efficiency ($\eta_{ST} = 0.30$)

η_{GT}	0.3	0.35	0.4
$-\frac{\partial \eta_{ST}}{\partial \eta_{ST}}$	1.0	1.08	1.17

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 maximum efficiency still does not always provide an optimum combined-cycle plant. For example, for a gas turbine with single stage combustion at constant turbine inlet temperature, a very high-pressure ratio attains a higher efficiency than a moderate pressure ratio. However, the efficiency of the combined-cycle plant with the second machine is normally better because the steam turbine operates far more efficiently with the higher exhaust gas temperature and produces a greater output.

Figure 4–3 shows that in a simple-cycle (GT) power plant, the efficiency is driven by the pressure ratio. However, in case of a combined-cycle plant, the efficiency is driven by the gas turbine inlet temperature (TIT).

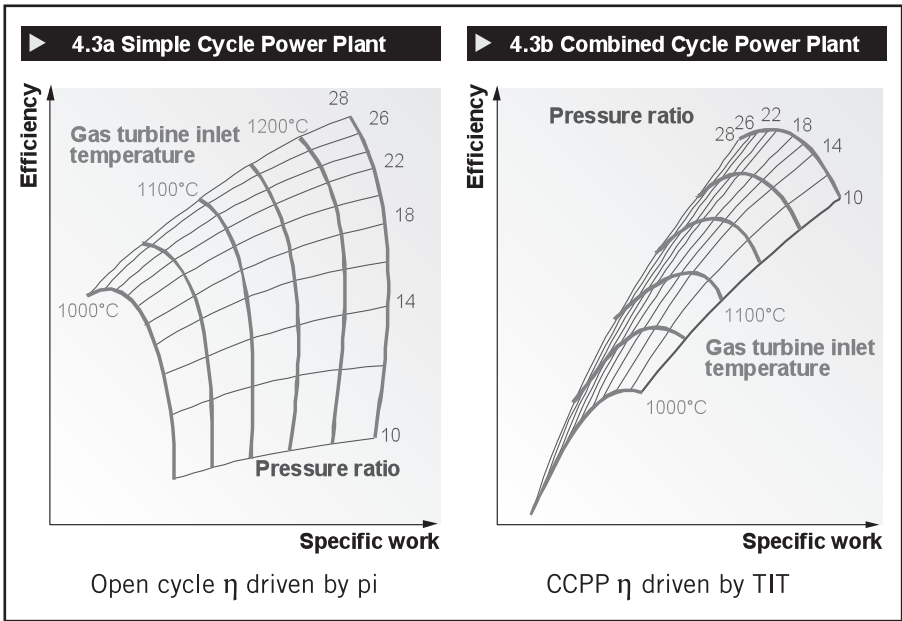


Figure 4-3 The efficiency of a simple-cycle GT and a combined-cycle plant as function of the gas turbine inlet temperature and pressure ratio

Figure 4-4a demonstrates the efficiency of the simple-cycle gas turbine with single-stage combustion as a function of the turbine inlet and exhaust gas temperatures. The maximum efficiency is reached when the exhaust gas temperatures are quite low. In this case a low exhaust temperature is equivalent to a high-pressure ratio.

Figure 4-4b shows the overall efficiency of the combined cycle based on the same gas turbine. Compared to figure 4-4a, the optimum point has shifted toward higher exhaust temperatures from the gas turbine, which again indicates an over-proportional improvement of the water/steam cycle when compared to the loss in gas turbine efficiency. For economic reasons, current gas turbines are generally optimized with respect to maximum power density (output per unit air flow) rather than efficiency. Often, 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.

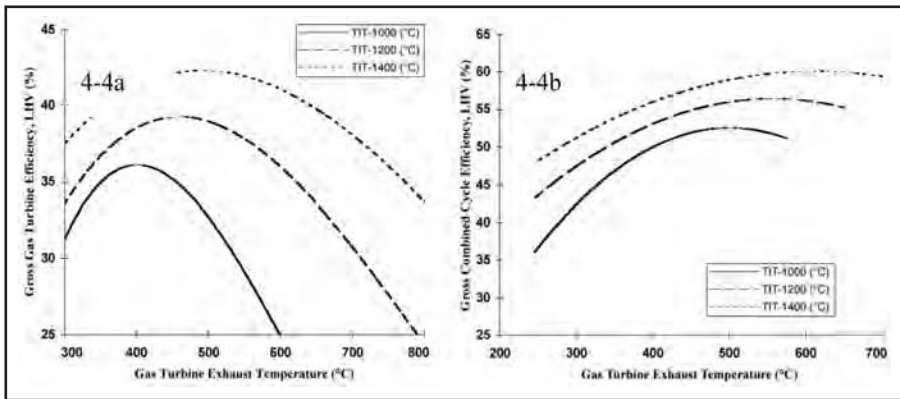


Figure 4-4a and 4-4b Efficiency of a simple-cycle GT with single stage combustion and combined cycle as a function of turbine inlet temperature (TIT) and turbine exhaust temperature

Gas turbines of a more complicated design (i.e., with intermediate cooling in the compressor or recuperator) are less suitable for combined-cycle plants. They normally have a high simple cycle efficiency combined with a low exhaust gas temperature, so that the efficiency of the water/steam cycle is accordingly lower.

The effects of high-pressure ratio and low combined-cycle efficiency can be decoupled if the gas turbine is designed with sequential combustion (air, upon leaving the compressor is passed through the first combustion chamber and expands in the first turbine stage before final combustion and expansion). Gas turbines with sequential combustion have practically the same simple-cycle efficiencies as single-combustion gas turbines at the same overall pressure ratio.

For comparison the same curves shown in figure 4-4a and 4-4b are shown for gas turbines with sequential combustion in figure 4-5a and 4-5b. Figure 4-5a shows almost the same simple cycle gas turbine efficiency level for the same gas turbine inlet temperatures (TIT). However, the exhaust gas temperatures are substantially higher, clearly improving the combined-cycle efficiency levels of figure 4-5b when compared to figure 4-4b. As for figure 4-4a, a low exhaust gas temperature is equivalent to a high pressure ratio of the sequential part (low pressure part) of the gas turbine. For consistency, the pressure ratio of the high-pressure turbine is kept constant at a ratio of 1.7:1.

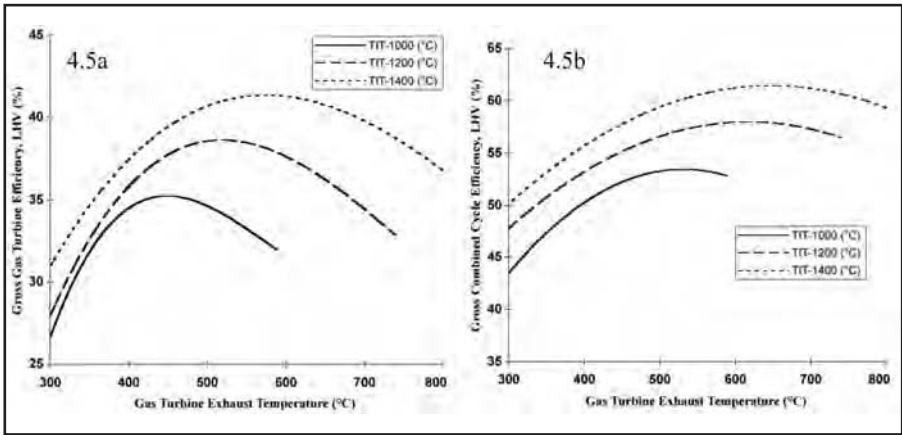


Figure 4-5a and 4-5b Efficiency of a simple-cycle GT with sequential combustion and combined cycle as a function of turbine inlet temperature (TIT) and turbine exhaust temperature

The main advantage of a sequential-fired gas turbine is that the drawback of single-combustion gas turbines (i.e., pressure ratio and exhaust gas temperature) is eliminated through the reheat process. This gives the ideal base for improved combined-cycle efficiencies, which also fits well to the Carnot comparison.

In summary, it may be said that the optimum gas turbine for simple cycle and the optimum gas turbine for combined-cycle plants are not the same. The gas turbine with the highest efficiency does not necessarily produce the best overall efficiency of the combined-cycle plant.

The type of gas turbine (i.e., gas turbine concept) and turbine inlet temperature are important factors. Similar considerations also apply with regard to the efficiency of the steam cycle. The gas turbine is generally a standard machine, and must, therefore, be optimized by the manufactures for its main application (i.e., combined cycle or simple cycle). The exhaust heat available for the steam process is thus given, and the challenge lies mainly in a cost effective conversion into a mechanical energy.

5 Combined-Cycle Concepts

Selection of the Combined-Cycle Power Plant Concept

This section shows what factors and process steps must be considered when choosing a combined-cycle power plant concept (see figure 5–1). There are three main steps in this process:

1. Analyze the customer requirements
2. Assess the site related factors/influence of the site conditions
3. Determine the plant concept solution

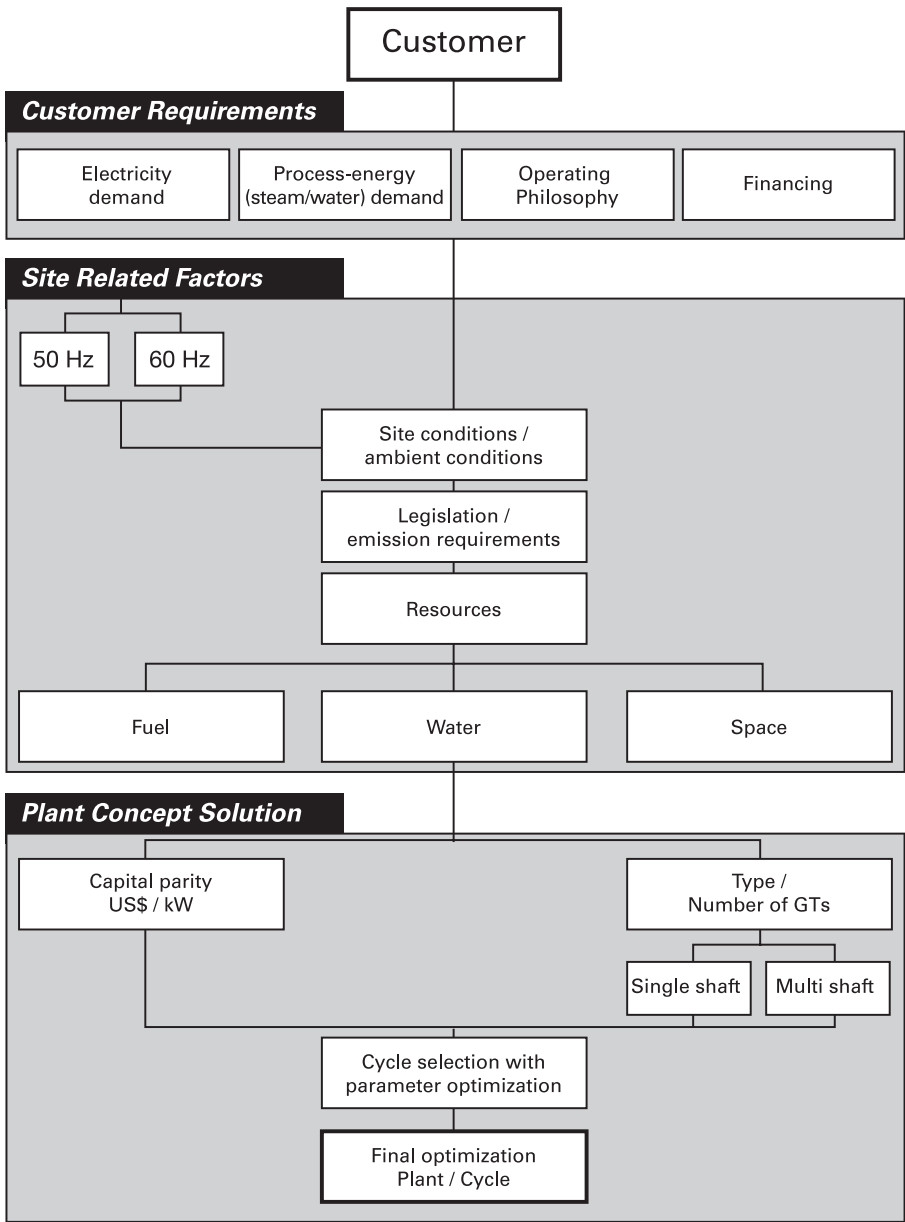


Figure 5–1 Selection of a combined-cycle power plant concept

For large combined-cycle power plants (>200 MW) with the main goal being to produce electricity in an efficient manner, the following described selections and optimization steps are normally done for the “Reference plant.”

Compared to the beginning period of combined-cycle power plants, most of the Engineering, Procurement and Construction (EPC) companies, having their own gas turbine (product), applied a standardization approach based on actual market conditions and or needs. The development from tailor-made plants to standardized plants is shown in figure 5–2.

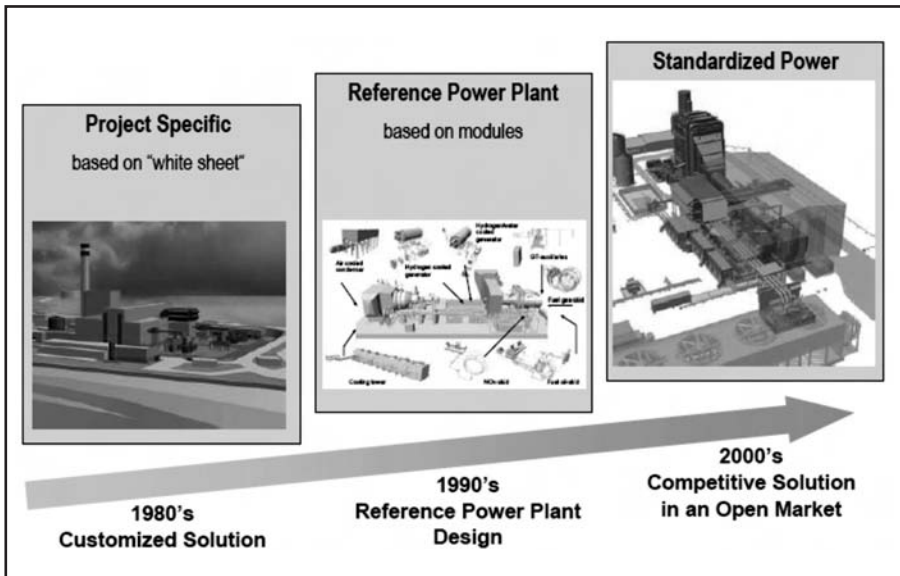


Figure 5–2 Evolutionary change in combined-cycle design philosophy

For the power island, the idea of this approach (with some exchangeable modules such as two or three options for the cold end of the steam turbine with condensation unit) is that the pre-engineered solution is fully reusable, and has the following benefits:

- Time (shorter delivery time)
- Quality (robust design)
- Risk (exchangeable components in case of problems)
- Cost (benefits by learning- and volume effect)

These benefits outweigh the small loss of efficiency compared to a fully customized designed power plant (see figure 5–3).

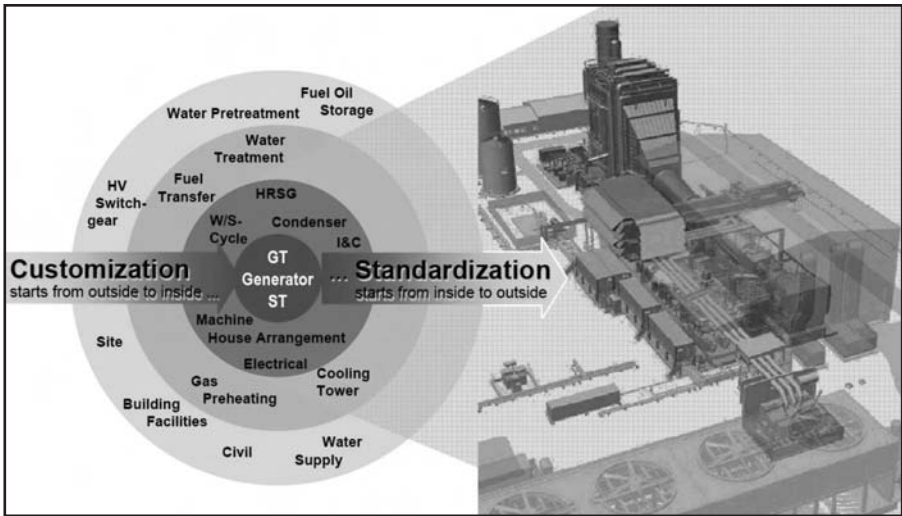


Figure 5–3 Standardization approach

With reference to a potential reuse, the one-to-one configuration is the best choice for the standardization. See also the comparison between multi-shaft and single shaft (table 5–3) later in this chapter.

Analysis of Customer Requirements

The customer (market) requirements define what a plant must be able to achieve in terms of performance and operational capability. The most significant requirement is the electricity power demand. When identified, it is important to determine any limits above or below the nominal power rating, which may be imposed, for example, by grid limitations or the need to meet an internal industry power demand. The best fit of turbines corresponding to this range must be found, taking into consideration the base load point at which the gas turbines will be operating.

If there is a steam-process demand it is necessary to look at the temperature, pressure, and mass flow required at the supply limit. The fluctuations allowed under these conditions and the demand variation over time should be known to design the plant for all possible operating conditions. If there is a return of steam or condensate from the industrial process, its condition and quality need to be considered in the cycle design as well (see chapter 6).

Combined-cycle power plants are intended for various uses such as base load, cycling, or frequency control. To account for such factors at the design stage, it is important to have information about expected operating philosophy, operating hours as well as number of starts per year, load scheme, grid requirements, and so on.

In case of a liberalized electricity market, financing becomes more and more important. This is especially true in the case of project financing. To be worthy of credit, any new power plant must be able to compete with other power plants on the open market on the basis of \$/MWh generated, not just for the first year but throughout the pay-back period of the plant and preferably even longer.

Site-Related Factors

Site-related factors are specific to the intended location of the combined-cycle plant and are usually out of control of the power plant developer. Those that affect the choice of the cycle must be considered in the cycle selection process. One fundamental factor, which is usually country dependent, is whether the local electricity grid is rated at 50 or 60 Hz. This affects the selection of the gas turbine type because larger gas turbines are designed for specific frequencies. Smaller gas turbines are usually geared and, with an adequate gearbox, can operate with both of these frequencies.

Ambient conditions

Ambient conditions affect the performance of the cycle. This section looks at possible influences of these conditions regarding changes of the design point. How an already dimensioned combined-cycle plant behaves with different ambient conditions will be discussed in chapter 9.

The gas turbine is a standardized machine, and can be used for widely different ambient conditions. This can be justified economically 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 air temperatures such as 40°C (104°F). In such a case the costs for developing a new machine would not be justified. Manufacturers quote gas turbine performances at ISO ambient conditions of 15°C (59°F), 1.013 bar (14.7 psia), and 60% relative humidity. The gas turbine will perform differently at other ambient conditions, and this will have an effect on the water/steam cycle process.

Unlike the gas turbine, the steam turbine is usually designed for a specific application. The exhaust steam section design, for example, depends on the condenser pressure at the design point, such as the exhaust section that would be chosen for a condenser design pressure of 0.2 bar (5.9”Hg). It can no longer function optimally if the operation pressure is only 0.045 bar (1.3”Hg). Also, blade path design in a steam turbine depends on the live-steam pressure, which is not the same for all cycles.

Nevertheless, as already explained, this detailed optimization and fine-tuning is done in most of the cases for the reference plant design and not anymore on a case-by-case basis, or for each and every project.

Air temperature and pressure are the most crucial ambient conditions. Relative humidity has only a minor influence, but becomes more important if the cooling concept is equipped with a wet cell (evaporative) cooling tower.

Ambient air temperature

There are three reasons why air temperature has a significant influence on the power output and efficiency of the gas turbine:

- Increasing the ambient temperature reduces the density of the air and consequently reduces the air mass flow into the compressor as constant volume engine. This is the main reason for changes in the gas turbine power output.
- The specific power consumed by the compressor increases proportionally to the air intake temperature (in K) without a corresponding increase in the output from the turbine part.
- As the air temperature rises and the mass flow decreases, the pressure ratio within the turbine is reduced. Due to the swallowing capacity of the turbine section and the reduced mass flow, the pressure at the turbine inlet of the GT is reduced. This leads to a lower pressure ratio within the turbine, and applies inversely, of course, to the compressor; however, because its output is less than that of the turbine, the total balance is negative.

Figure 5–4 shows the gas turbine characteristics at two different ambient pressures in a temperature/entropy diagram. The exhaust gas temperature rises as the air temperature increases because the turbine pressure ratio is reduced, although the gas turbine inlet temperature (TIT) remains constant. The result is a decrease in the gas turbine efficiency and output while the ambient temperature rises. However, the effect on the performance of the combined cycle is somewhat less because a higher exhaust gas temperature improves the performance (power output) of the steam cycle.

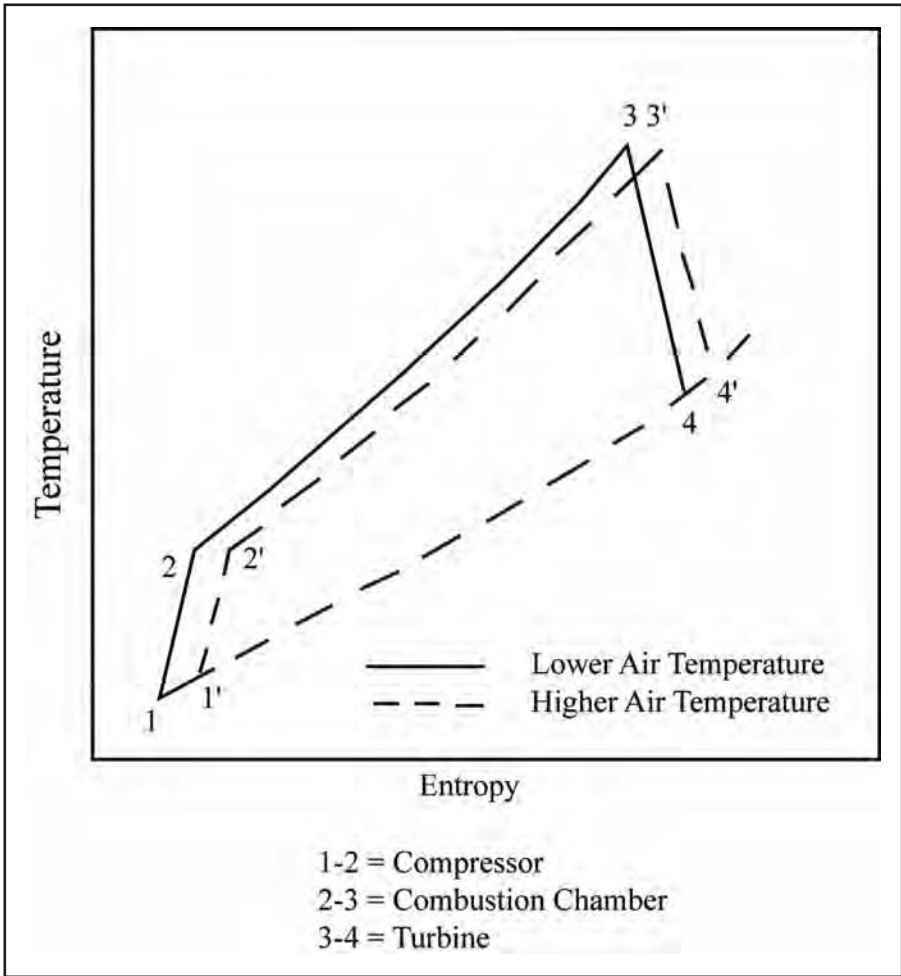


Figure 5-4 Entropy/temperature diagram for a gas turbine process at two different ambient air temperatures

Figure 5-5 shows the relative efficiency of the gas turbine, steam process, and the combined-cycle plant due to the air temperature; other ambient conditions as well as the condenser pressure remain unchanged. An increase in the air temperature has a slightly positive effect on the efficiency of the combined-cycle plant. Because the increased temperature in the gas turbine exhaust enhances the efficiency of the steam process, it more than compensates for the reduced efficiency of the gas turbine unit.

This behavior is not surprising considering the Carnot efficiency (equation 4-1). The increase in the compressor outlet temperature causes a slight increase in the average temperature of the heat supplied (T_E). Most of the exhaust heat is dissipated in the condenser; therefore, the cold temperature (T_A) does not change because the condenser pressure is kept constant. The overall efficiency of the combined-cycle plant will increase.

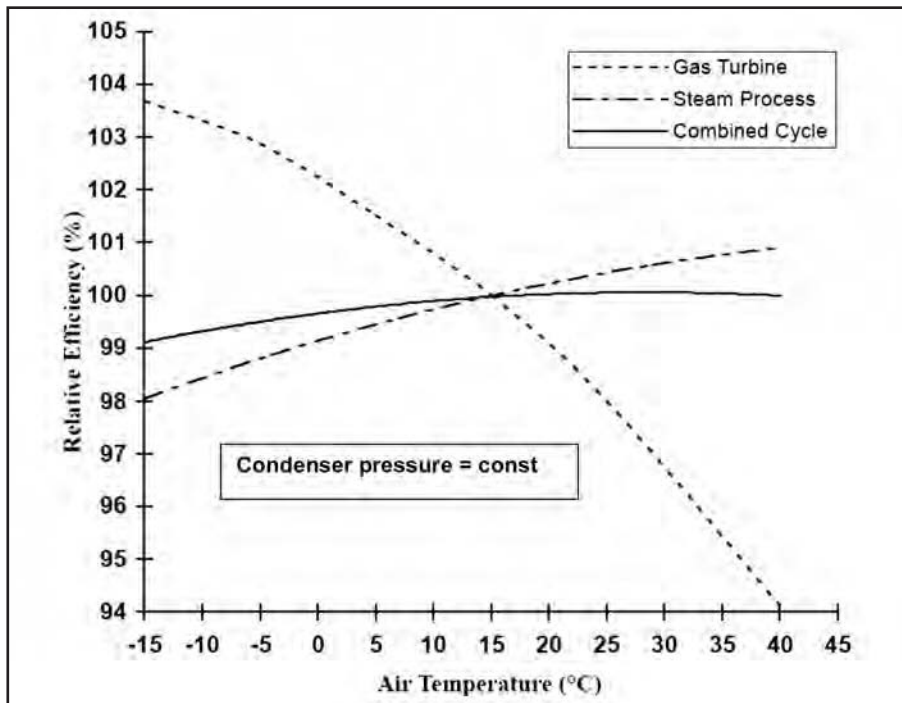


Figure 5-5 Relative efficiency of gas turbine, steam process, and combined cycle as function of the air temperature

Figure 5-6 shows how the power output of the gas turbine, steam turbine, and combined cycle decrease with an increase in the air temperature. The effect is less pronounced for the combined cycle than for the gas turbine alone. The power output of the combined cycle is affected differently from the efficiency because change in mass flow of air and exhaust gases are more dominant than the exhaust gas temperature.

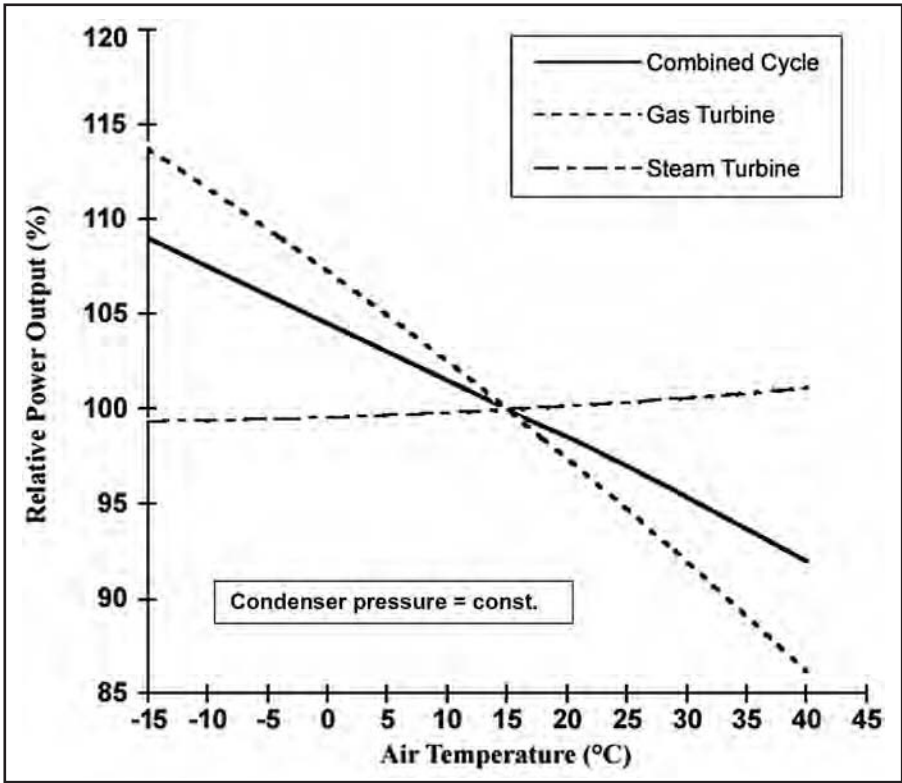


Figure 5–6 Relative power output of a gas turbine, steam turbine, and combined cycle as function of the air temperature

Ambient air pressure

Gas turbine performance is normally quoted at an air pressure of 1.013 bar (14.7 psia)—ISO condition, which corresponds approximately to the average pressure prevailing at sea level. A different site elevation and daily weather variations result in a different air pressure.

Figure 5–7 shows the relationship between site elevation and ambient air pressure, and its influence on relative power output of the gas turbine, steam turbine, and combined cycle. The air pressure has no effect on the efficiency if the ambient temperature is constant, even though the ambient air pressure has an influence on the air density similar to the one of the air temperature. At a lower ambient pressure, the backpressure of

the gas turbine is correspondingly lower, not considering inlet and outlet pressure drops. This leaves no influence on the gas turbine process with the exception of the reduced mass flow. Assuming that no change takes place in the efficiency of the steam process (which corresponds well to the real situation), there is the same variation in the power output of the steam turbine and hence the combined cycle.

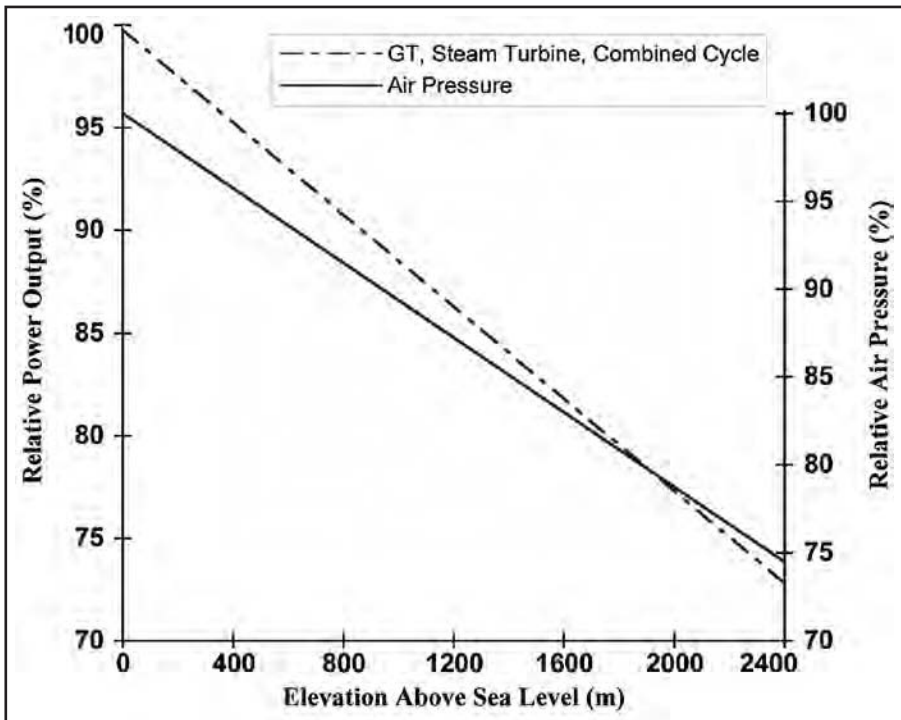


Figure 5-7 Relative power output of gas turbine, steam turbine, and combined cycle and relative air pressure versus elevation above sea level

Because the power outputs of the gas turbine and the steam turbine vary in proportion to the air pressure, total power output of the combined-cycle plant varies in proportion. The fact that the gas turbine inlet and outlet pressure drops were held constant for the calculations of figure 5-7 accounts for the slight difference in the relative power output compared to the air pressure. The efficiency of the plant remains constant because both the thermal energy supplied as well as the airflow vary in proportion to the air pressure.

Relative humidity

Figure 5–8 shows that the gas turbine and combined-cycle output will increase if the relative humidity of the ambient air increases, with other conditions remaining constant. This is because at higher levels of humidity there will be a higher water content in the working medium of the gas cycle, resulting in a better gas turbine enthalpy drop and more exhaust gas energy entering the heat recovery steam generator (HRSG).

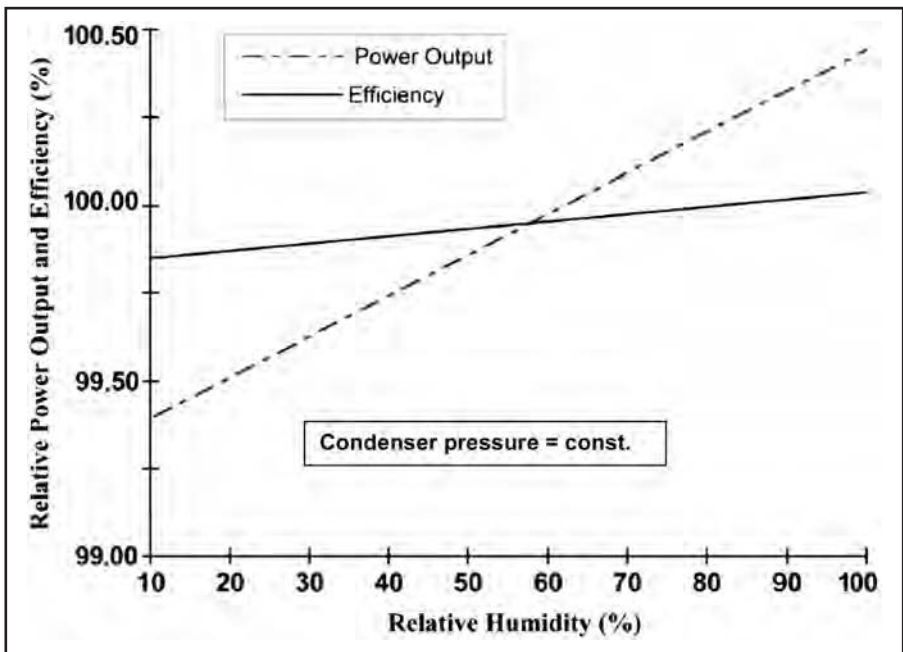


Figure 5–8 Relative power output and efficiency of gas turbine and combined cycle as function of relative humidity

There is a further influence for plants with wet cooling towers where the relative humidity directly influences the condenser vacuum, and hence the steam turbine exhaust temperature. A lower humidity results in a better vacuum. This is discussed in the “Resources” section later in this chapter.

Legislation

Many environmental considerations affect design and construction of all types of power stations. The choice of a combined-cycle concept is mainly influenced by legislation. This legislation limits emission levels, especially that of nitrous oxides (NO_x). Current dry low NO_x gas turbine combustors can often achieve the required levels with gaseous fuel, but special measures may have to be taken if oil fuels are fired.

One option to reduce the formation of NO_x during combustion is to lower the temperature of the flame because the speed of the reaction producing NO_x is noticeably faster at very high temperatures. The common approach for gaseous fuels is to use premix burners to lower the flame temperature, mixing air at the outlet of the compressor with the fuel before it is ignited in a vortex breakdown zone. Injecting water or steam into the combustor can produce the desired temperature reduction; however, it will affect the output and efficiency of the gas turbine. This approach is, therefore, mainly used for fuel oil where premix is problematic due to the risk of pre-ignition. The amount of possible NO_x reduction depends on the water or steam to fuel ratio as described in chapter 10.

For gas turbines in simple cycle with no HRSG (and, therefore, no available steam source), water is injected for NO_x control, but with a negative effect on efficiency.

The problem with steam injection is finding steam at a suitable pressure level in the steam cycle. Generally the high-pressure (HP) live-steam pressure is too high and the intermediate-pressure (IP), in case of GT's with a high compressor ratio and low-pressure (LP) steam pressure is too low. For large industrial gas turbines the pressure level required for steam injection is between 25 and 50 bar (348 and 710 psig), depending on the type of gas turbine as well as the load. Using pressure-reduced HP steam is usually the simplest and the least expensive solution, but is exergetically undesirable.

For this reason, a base-load plant with steam injection into the GT will usually have a steam extraction in the steam turbine at the appropriate pressure level, with a backup from the HP live-steam line

for those operating points at where the extraction pressure is too low. Steam extraction is, however, more problematic with steam turbines in combined cycle blocks 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. Thus it becomes necessary to switch over to live steam, which negatively affects the overall efficiency.

Power output is increased by both water and steam injection due to the resulting increase in the mass flow through the gas turbine. Water injection has a greater effect because steam turbine output is decreased with steam injection. Water or steam injection is also a means of power augmentation (see also chapters 9 and 6). Efficiency of the combined-cycle plant is decreased in both cases; however, less so by steam than by water, because steam brings more internal energy to the combustor. In the case of water, valuable fuel has to be used to provide the relatively low temperature energy for the vaporization. For the same reason hotter water partly reduces the loss in efficiency.

Figure 5–9 shows the effect that injecting water or steam has on the output and efficiency of a dual-pressure cycle. The steam is extracted from the steam turbine, the water from the makeup water line at 15°C (59°F) or at the outlet of the economizer at 150°C (302°F). The amount of water or steam is provided by the water or steam to fuel ratio, which is determined by the necessary reduction level of NO_x (or, in very few special cases, by the amount of power augmentation required).

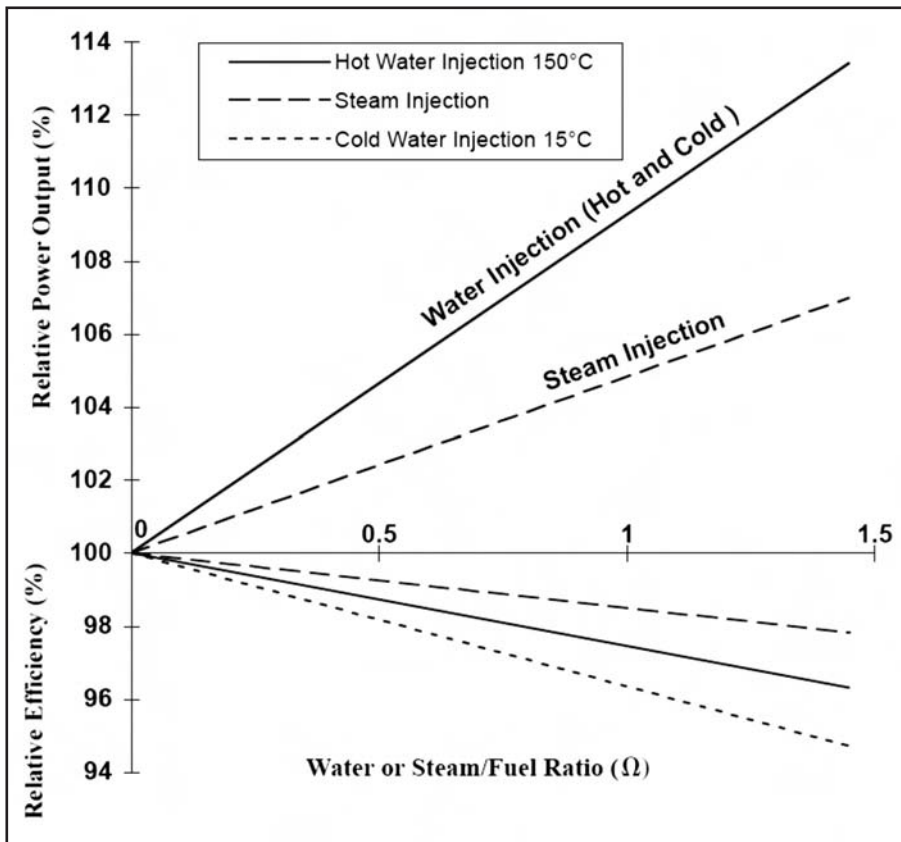


Figure 5-9 Effect of water and steam injection on relative combined-cycle power output and efficiency versus water or steam/fuel ratio (with TIT = constant)

Apart from NO_x , other emissions are often restricted, such as CO , particulates, and so forth. Those emissions influence the component design rather than the cycle performance, however, and so do not influence the selection of the concept.

Another restriction could be a limitation of the cooling water temperature in case of fresh-water cooling (with sea water or river water) either by a limitation of a maximum allowable temperature rise, or with a limitation of the maximum allowed cooling water outlet temperature or restrictions regarding visible plumes in case of a wet cooling tower. This is discussed in the following section.

Resources

Available resources are different at every site. Cooling media and fuel are the main resources affecting the selection of the cycle concept. Although space is also a resource, it affects the cycle only when space is restricted by, for example, the need to use existing buildings or part of an existing cycle. This may lower the capital cost of the plant, but could also limit the possibilities for performance improvements by, for example, restricting the size of the HRSG or influencing the choice of shaft configuration.

Cooling media

To condense the steam, a cooling medium must be used to carry off the waste heat from the condenser.

There are three options for the main cooling system:

- Fresh-water cooling (with sea or river water)
- Evaporation cooling (wet cell cooling tower)
- Dry cooling system (in case of a direct system with an air-cooled condenser [ACC])

Cooling with water has a clear advantage because water has a high specific thermal capacity and good heat transfer properties. Therefore, where water is available in required quantities, cooling can be done in a direct system (fresh water from a river or the sea) or in a wet cooling tower. The cooling water quantities in case of fresh water cooling are approximately 60 to 100 times the exhaust steam flow (in some special cases with very limited temperature rise even higher) and for wet cooling systems as cooling tower makeup water around 1 to 1.4 times the exhaust steam flow or approximately a minimum amount of 0.3 kg/s (2400 lb/h) per MW installed combined-cycle capacity (without supplementary firing).

To ease the permitting process where water is not available, a direct air-cooling system with an air-cooled condenser (or a dry cooling tower as indirect system) is used. The disadvantages of this method include higher costs and higher exhaust steam (condenser) pressure, mainly in the off-design points with high ambient temperatures (therefore less power output of the steam turbine) leading to lower plant efficiency. In combined-cycle plants with a direct air-cooling system (ACC) the ambient air temperature directly influences the ST output and consequently the efficiency, which is not (or to a less extent) the case for the other cooling system (see table 5–1).

Table 5–1 Comparison of combined cycle performance data for different cooling systems and ambient temperatures

	Design: ambient temperature = 6°C (43°F)		Off-design: Summer ambient temp = 30°C (88°F)	
	P MW	Efficiency η %	P MW	η %
Cooling tower	411.7 100%	57.5 100%	363.5 100%	56.1 100%
ACC	408.1 99.1%	57.0 99.1%	345.4 95%	53.3 95%

In special cases where only a limited amount of water is available, and an evaporation system could be used with restrictions regarding visible plumes, a combination of a dry and wet system with hybrid cooler cells (consisting of a dry and a wet section) could be the optimal solution. Design parameters/performance data are similar to the wet cell cooling tower, but would result in slightly higher costs.

The temperature of the cooling medium has a major effect on the efficiency of the thermal process. The lower the temperature the higher the efficiency that can be attained, because the pressure in the condenser is lower, producing a greater useful enthalpy drop in the steam turbine and hence an increase in steam turbine output and in plant efficiency. This is illustrated in figure 5–10.

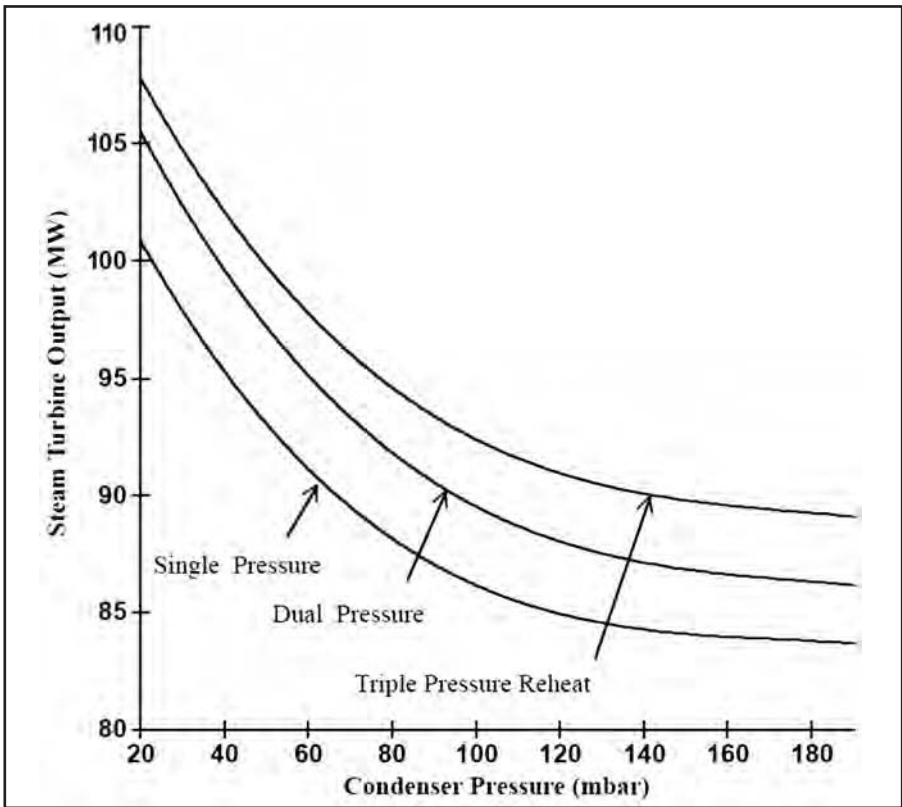


Figure 5–10 Effect of condenser pressure on steam turbine output

The trend is similar for single-, dual-, and triple-pressure cycles (discussed later in this chapter). However, the effect is less significant above approximately 100 mbar (3.0”Hg) condenser pressure as the relative change in the pressure decreases and there is less impact on the steam turbine enthalpy drop. Plant costs are reduced if the pressure is higher because of the lower volume flow of exhaust steam resulting in a smaller steam turbine. This impact of a change in the condenser vacuum is increased with a dual- or even triple-pressure than with a single-pressure cycle because the exhaust steam flow is larger.

Figure 5–11 shows typical condenser design pressure values as function of the temperature of the cooling medium for direct water cooling (fresh water), cooling with a wet cooling tower, and direct air cooling. These values are calculated based on fuel gas price levels (2007)

in Europe and the United States. The best vacuums are attained with direct water cooling; the worst ones are with direct condensation with air, mainly with ambient air temperatures higher than 20 to 25°C (68 to 77°F). In comparison with ambient air temperature, the corresponding water temperature is generally lower. The curves for direct cooling and cooling tower cooling are calculated with cooling water temperature rises of 9K and 11K, respectively.

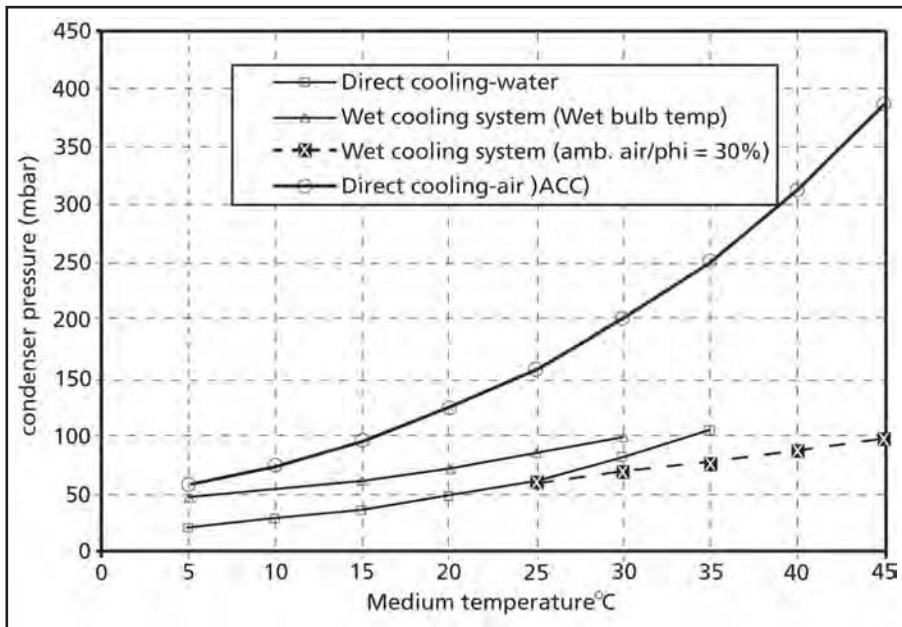


Figure 5-11 Temperature of cooling medium versus condenser pressure for different types of cooling systems

In case of wet cell cooling tower, the wet bulb temperature is determined as temperature of the cooling medium. It is a function of the dry bulb temperature and the relative humidity, and can be read from an enthalpy-moisture content diagram for air (Mollier diagram). Out of the statistical values the maximum is not going beyond about 30°C (86°F). Therefore, in case of higher ambient air temperatures in the range of about 30 to 45°C, a relative humidity (ϕ) of 30% was selected in figure 5-11 for comparison reasons.

The influence on the steam turbine output of the various cooling systems can be derived out of figure 5–10 in combination with figure 5–11.

Fuel

This section explains how the sulfur content in the fuel influences the cycle concept because of the feedwater temperature. The fuel type and composition also have a direct influence on the gas turbine performance and the emissions produced. Power plants often have a main fuel and a backup fuel.

The lower heating value (LHV) of the fuel is important because it defines the mass flow of fuel, which must be supplied to the gas turbine. The lower the LHV, the higher the mass flow of fuel required to provide a certain chemical heat input, normally resulting in a higher power output and efficiency. This is why low British thermal unit (BTU) gases can result in high-power outputs if they are supplied at the pressure required by the gas turbine.

The fuel composition is equally important in influencing the performance of the cycle because it determines the enthalpy of the gas entering the gas turbine and, hence, the available enthalpy drop and gas turbine output as well as the amount of steam generated in the heat recovery steam generator (HRSG). This is the reason why the influence of the fuel on performance cannot be given as a function of LHV only.

One way of improving the efficiency of the cycle is to raise the apparent LHV ($\text{LHV} + \text{sensible heat}$) of the fuel by preheating it with hot water from the economizer of the HRSG. This improves heat utilization in the HRSG because additional water is heated in the economizer. The fuel consumption is correspondingly reduced because each unit of fuel contains more sensible heat, so a lower fuel mass flow is required. Figure 5–12 shows schematically a hot water extraction from the economizer (5) of a dual-pressure HRSG. The cooled water (4) returns to the feedwater tank. Preheating natural gas fuel from 15°C (59°F) to 150°C (302°F) can improve the plant efficiency by approximately 0.7% (relative). Today, depending on the type of gas turbine, fuel gas

preheating can go up to 250°C (482°F); if hot water is not available, it can also be done economically with steam.

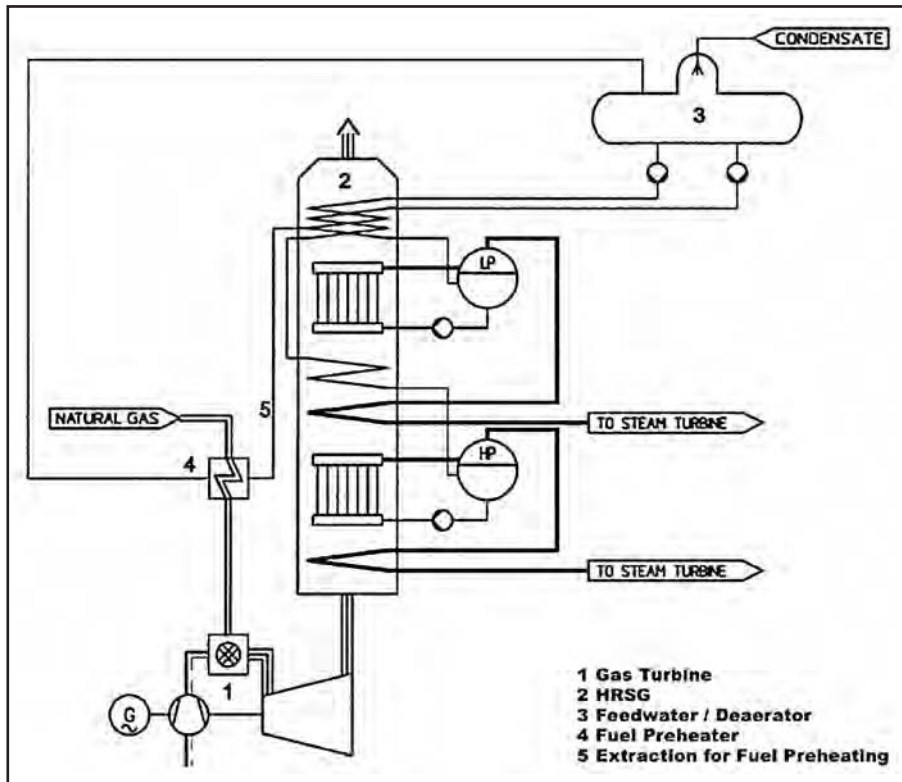


Figure 5–12 Flow diagram to show fuel preheating

Another factor, which may be important for gas-fired plants, is the available gas supply pressure. The gas turbine requires a certain pressure, depending on the design of the burners in the combustor and the gas turbine pressure ratio. Sometimes a gas compressor is required to increase supply pressure, which will also increase the temperature of the fuel in proportion to the compression ratio. The benefit of gas preheater in such cases is reduced, perhaps by even rendering it uneconomical because the efficiency improvement is too slight to justify the additional investment in the water/gas heat exchanger, HRSG surface, and piping. (See chapter 7 for more information about fuels for gas turbines.)

Table 5–2 lists the fuels that can be fired in a gas turbine.

Table 5–2 Possible fuels for combined cycle applications

a) Standard Fuels	b) Special Liquid fuels	c) Special Gas Fuels
Natural gas	Methanol/Kerosene/Naphtha	Synthetic gas
Diesel oil (oil # 2)	Crude oil	Blast furnace gas
	Heavy oil, residuals	Coal gas with medium or low heating value
	Oil shale	

Note: The use of categories b) and c) is limited, because their burning depends on the exact chemical analysis and the type of gas turbine involved. Generally, industrial gas turbines with large combustors are better able to handle these fuels than modern gas turbines with high turbine inlet temperatures.

Determining the Plant Concept Solution

When plant requirements and the site data are known, a concept for the cycle can be determined. In doing so, both technical and economic aspects must be taken into account.

Selection of the gas turbine

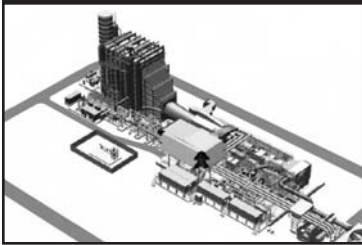
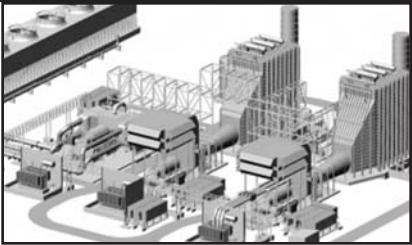
The first stage of the cycle selection is to determine the size and number of gas turbines needed to meet a certain power output and eventually a certain process steam production.

For a plant with a given gas turbine, supplying process steam will decrease the power output because steam is removed from the steam cycle rather than being used for power generation. As a result, a larger gas turbine could be selected, rather than if process steam were not required. A gas turbine of a lower rating could be chosen for a given application, with peak power produced by an additional steam turbine output through supplementary firing (or with other power augmentation methods). With regard to the steam turbine, it is of no importance if the required exhaust gas temperature is attained directly from the gas turbine exhaust or by means of supplementary firing in the HRSG. However, from an efficiency

point of view, preference would be to a fuel feeding directly in the gas turbine at a higher exergetic level than in the HRSG duct burner.

If more than one gas turbine is needed, a choice must be made between a multi-shaft configuration or several single shaft blocks. Several manufacturers have developed single shaft power plants (power islands) for their gas turbines. These standardized units provide advantages related to fast installation times, availability, and space requirements. Table 5–3 shows the comparison between a 2–1 multi-shaft combined cycle and two single-shaft units.

Table 5–3 Comparison multi-shaft versus single shaft (2 single shaft units versus a multi shaft 2-1 unit)

	
Components	Single shaft: Less generators required Less HV-switchyards required Less electrical equipment required One compact lube oil system
Civil	Multi shaft: No synchronous clutch required One large versus 2 smaller ST Less auxiliaries (pumps etc) required
Costs	Single shaft: Smaller footprint required Multi shaft: Higher flexibility in plant layout
Performance	Single shaft: Lower specific costs of plant if standardized Lower cost for grid connection
Performance	Single shaft: At full load both solutions have almost the same performance level in case of larger plants
Flexibility	Multi shaft: Faster start-up time from half load operation to full load
Availability	Single shaft: Higher availability (less complexity – clear maintenance concept)

Not all gas turbines with the same inlet airflow have the same performance. It depends on the turbine inlet temperature (TIT), the design

concept (i.e., one or two stages of combustion), and the gas turbine exhaust gas temperature (which is a function of the pressure ratio). Because this has a marked influence on the performance of the combined cycle, it is worth looking at in more detail.

When the gas turbine exhaust temperature is lowered, both the thermodynamic quality of the steam process (seen by the steam turbine output) and the HRSG efficiency decrease as shown in figure 5–13.

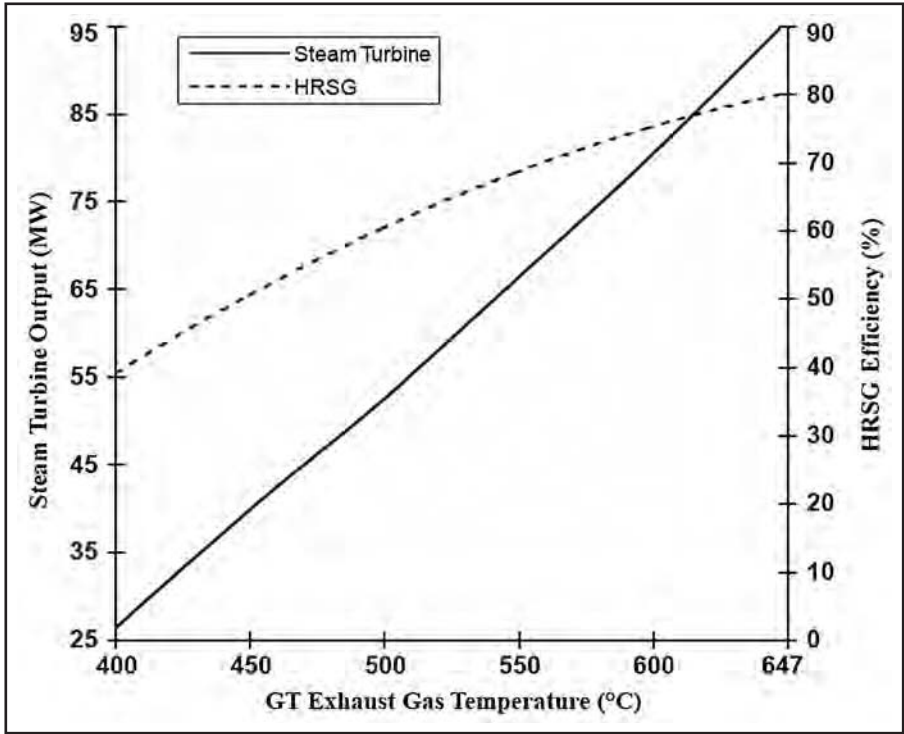


Figure 5–13 Steam turbine output and HRSG efficiency versus gas turbine exhaust temperature for a single-pressure cycle

The effect is more pronounced with the single pressure than with a dual or even triple-pressure reheat cycle because the energy utilization rate falls off faster.

Figure 5–14 shows the ratio between the output of the dual-pressure and the single-pressure cycles as a function of the gas turbine exhaust gas temperatures.

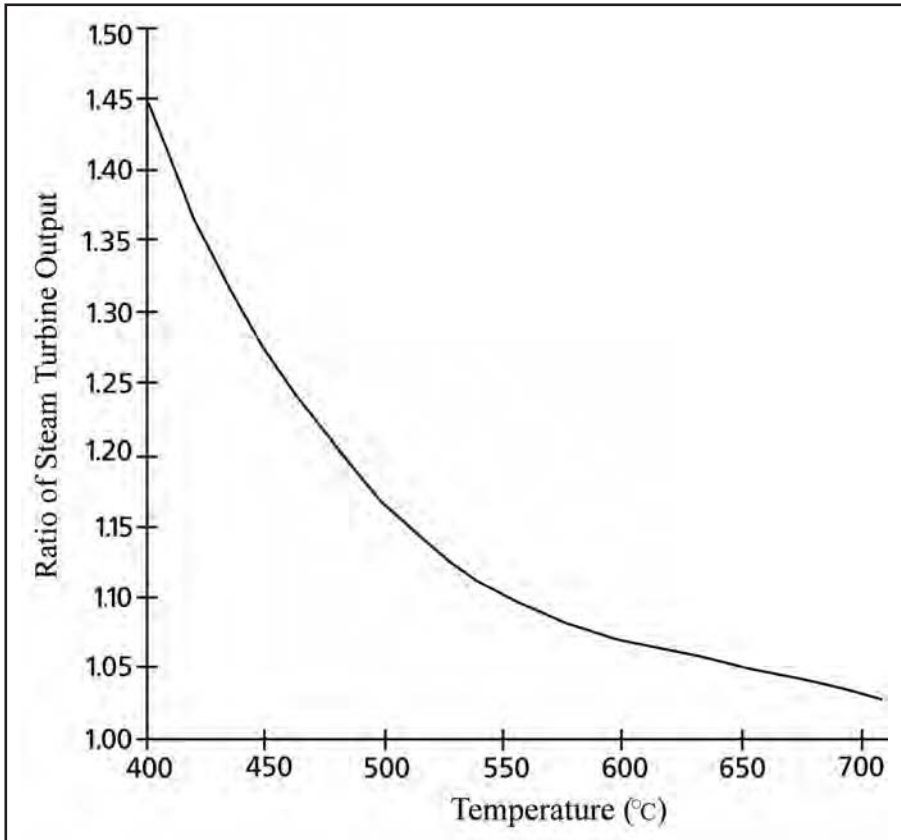


Figure 5–14 Ratio of steam turbine output of a dual-pressure compared to a single-pressure cycle as a function of the gas turbine exhaust temperature

With increasing exhaust gas temperature, the advantage of a dual-pressure versus a single-pressure cycle diminishes more and more, and at a certain temperature the ratio equals practically to 1. The theoretical temperature limit is a function of the chosen steam parameters (temperatures and pressures). For normal parameters this limit can be reached at around 750°C. For highly advanced and individually optimized cycles, this limit can theoretically slide up to

around 900°C or higher. Because cycles with (high) supplementary firing are mostly derived from normal cycles (e.g., single or dual-pressure cycles), the lower limit will apply here. Therefore, cycles with (high) supplementary firing are mostly single-pressure cycle.

Combined Cycle—Water/ Steam Cycle Concepts

The main challenge in designing a combined-cycle plant with a given gas turbine is how to transfer the gas turbine exhaust heat to the water/steam cycle to achieve an optimum steam turbine output. The focus is on the HRSG in which the heat transfer between the gas cycle and the water/steam cycle takes place.

Figure 5–15 shows the energy exchange that would take place in an idealized heat exchanger, in which the product (mass flow times specific heat capacity, or the energy transferred per unit temperature) must be the same in both media at any given point so that there are minimum exergetic losses in a given heat exchanger. There must be a temperature difference between the two media to allow the energy transfer. Because the temperature difference tends towards zero, the heat transfer surface of the heat exchanger tends towards infinity and the exergy losses towards zero.

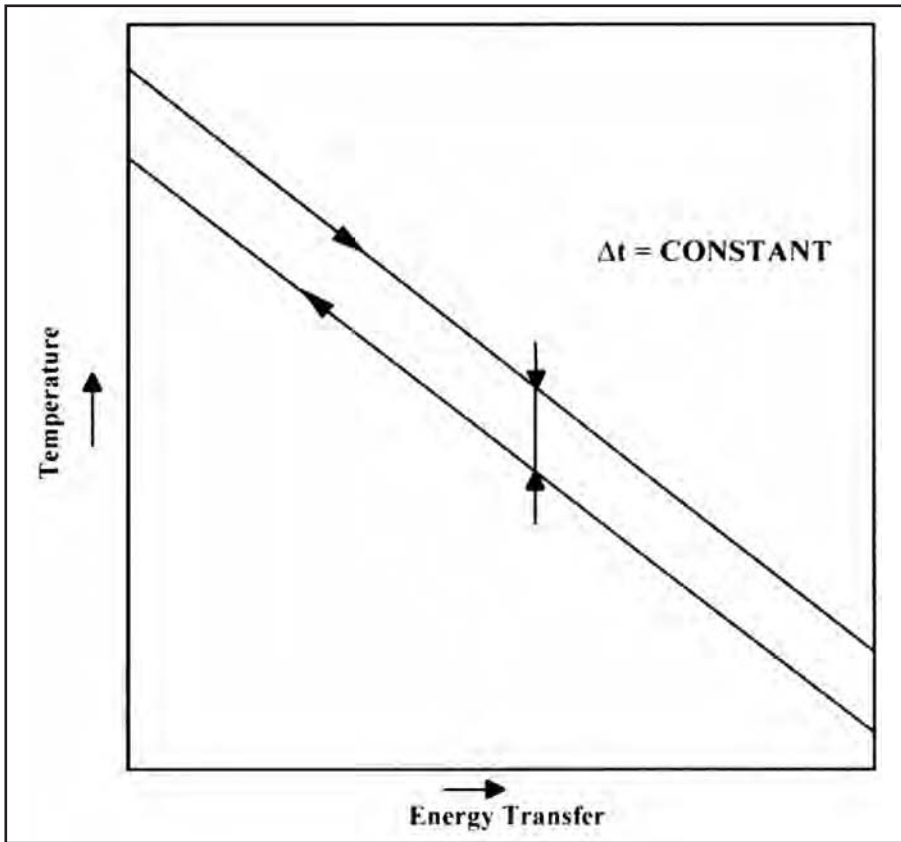


Figure 5–15 Energy/temperature diagram for an idealized heat exchanger

The heat transfer in a HRSG entails losses associated with three main factors:

- The physical properties of water, steam, and exhaust gases do not match the higher exergetic losses than the ones of an idealized heat exchanger as shown in figure 5–15.
- The heat transfer surface cannot be infinitely large.
- The temperature of the feedwater must be high enough to prevent forming corrosive acids in the exhaust gas, where it comes in contact with the cold tubes in the HRSG. This minimum feedwater temperature limits the energy utilization in the HRSG (temperature to which the exhaust gas can be cooled).

The extent of loss minimization (and maximum heat utilization) depends on the concept and the main parameters of the cycle. In a more complex cycle the heat will generally be used more efficiently, improving the performance, but also increasing the cost. In practice a compromise between performance and cost must always be made.

Basic Combined-Cycle—Water/Steam Cycle Concepts

In this section, the most common water/steam cycle concepts are presented and explained, starting with the simplest and leading to more complex cycles. A heat balance for each of the main cycle concepts is given, based on ISO conditions (ambient temperature 15°C (59°F); ambient pressure 1.013 bar, (14.7 psia); relative humidity 60%; condenser vacuum 0.045 bar (1.3”Hg) with a gas turbine with sequential combustion, rated at 178 MW and a steam turbine with water cooled condenser. The gas turbine is equipped with cooling air coolers that generate additional steam for the water/steam cycle and boost the steam turbine output. Because these features are the same for all of the heat balances, a clear comparison can be made among them, showing how the cycle concept influences the heat utilization.

Additionally, an analysis is done for each concept to show how the main cycle parameters, such as live-steam temperatures and pressures, and HRSG design parameters, influence the performance of the cycle. When designing a combined cycle with a given gas turbine, the free parameters are those of the steam cycle. Therefore, the influence of the various parameters is analyzed with respect to the steam turbine output. This is because in a plant without additional firing, the thermal energy supplied to the steam process is given by the gas turbine exhaust gas and the efficiency of the steam process is always proportional to the steam turbine output. The steam turbine does, however, account for only about 30 to 40% of the total combined-cycle power output; therefore, optimization of the steam process can only influence that portion.

Although the cycles shown have only one gas turbine, the concepts and results are also valid for cycles with several gas turbines, HRSG's, and other types of industrial gas turbines.

Single-pressure cycle

The simplest type of combined cycle is a basic single-pressure cycle, so called because the HRSG generates steam for the steam turbine at only one pressure level. A typical flow diagram is shown in figure 5–16 with a gas turbine exhausting into a single HRSG.

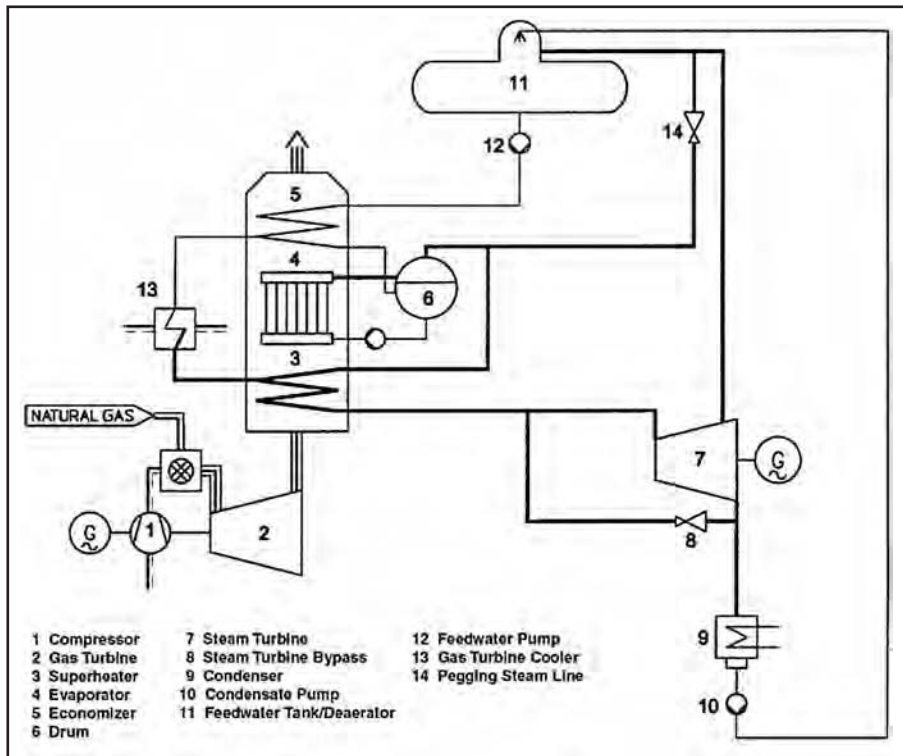


Figure 5–16 Flow diagram of a single-pressure cycle

The steam turbine (7) has a steam turbine bypass (8) into the condenser, which is used to accommodate the steam if for any reason it cannot be admitted to the steam turbine, for example

during startup, in case of a steam turbine trip, or because the steam turbine is out of operation. After the condenser (9), a condensate pump (10) is used to pump the condensate back to the feedwater tank/deaerator (11). The feedwater pump (12) returns the feedwater to the HRSG. Heating steam for the deaerator is extracted from the steam turbine with a pegging steam supply from the HRSG drum, in case the pressure at the steam turbine bleed point becomes too low at off-design conditions.

The HRSG consists of three heat exchanger sections:

- The economizer (5)
- The evaporator (4)
- The superheater (3)

In the economizer, the feedwater is heated up to a temperature close to its saturation point. In the evaporator, feedwater is evaporated at constant temperature and pressure. The water and saturated steam are separated in the drum (6) and the steam is fed to the superheater where it is superheated to the desired live-steam temperature.

Figure 5–17 shows the temperature/energy diagram for the single-pressure HRSG. The heat exchange in these three different sections is distinctly recognizable. It differs from an idealized heat exchanger as shown in figure 5–15 mainly because the water evaporates at constant temperature. The area between the gas and water/steam temperature lines illustrates the exergy loss between the exhaust gas and the water/steam cycle. Even with an infinitely large heat transfer surface, this exergy loss can neither be equal to zero nor can the heat exchange process in a boiler ever be ideal.

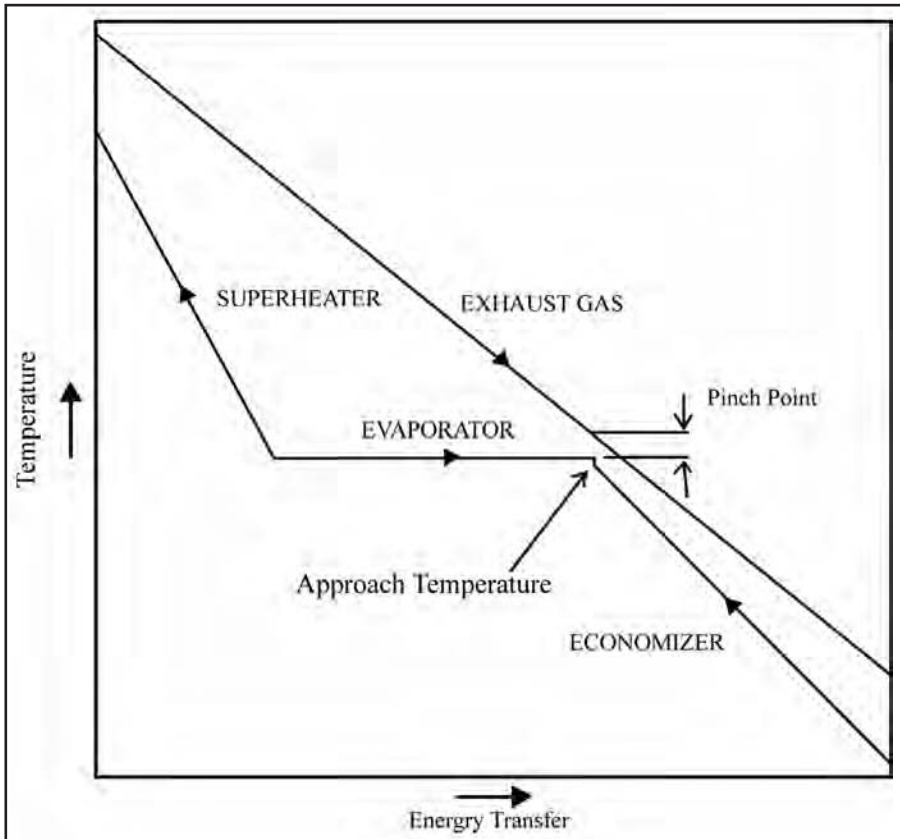


Figure 5–17 Energy/temperature diagram of a single-pressure HRSG

Two important parameters defining the HRSG are marked on the diagram in figure 5–17. The approach temperature is the difference between the saturation temperature in the drum and the water temperature at the economizer outlet. This difference, typically 5 to 12K (9 to 22°F), helps to avoid evaporation in the economizer at off-design conditions.

The pinch point temperature is the difference between the evaporator temperature on the water/steam side and the outlet temperature on the exhaust gas side. This is important when defining the heating surface of the HRSG. The lower the pinch point, the more heating surface is required, and the more steam is generated. Pinch points are typically between 8 and 15K (14 to 27°F), depending on the economic parameters of the plant.

Example of a single-pressure cycle

Figure 5–18 shows the heat balance for the single-pressure cycle where 73.3 kg/s (579'400 lb/hr) steam at 105 bar (1508 psig) and 568°C (1054°F) is generated. A small loss of temperature and pressure is taking place in the live-steam line, after which the steam is expanded in a steam turbine with an output of 94.8 MW. The resulting gross electrical efficiency of this cycle is 57.7%.

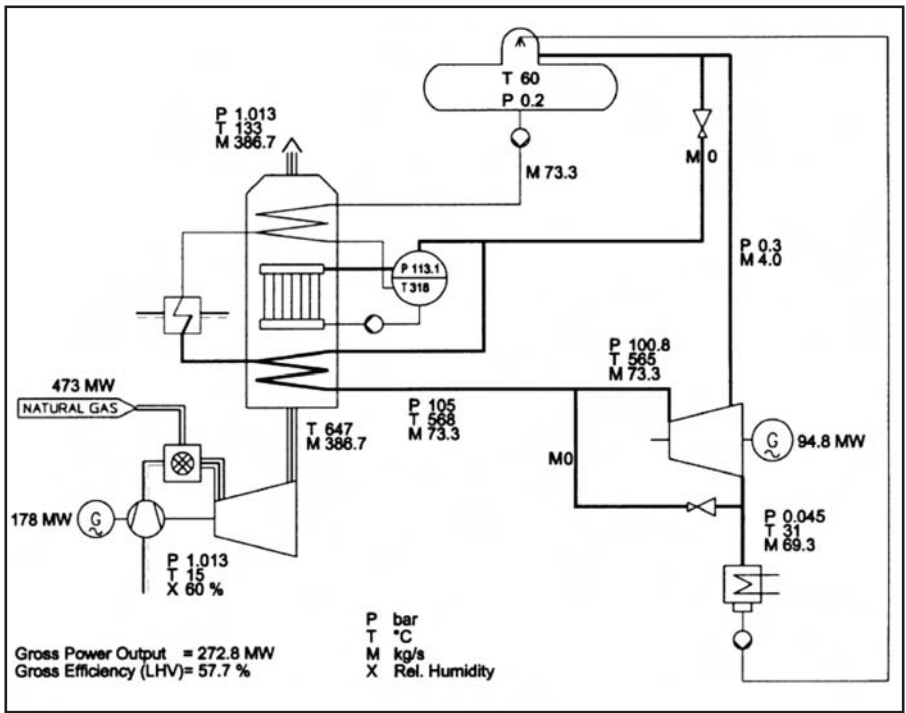


Figure 5–18 Heat balance for a single-pressure cycle

Energetic utilization of the exhaust heat is relatively low, considering that the feedwater temperature is 60°C (140°F) as illustrated by the stack temperature of 133°C (271°F), as figure 5–19 shows in an energy flow diagram; 11.4% of the fuel energy supplied is lost through the boiler stack, another 29.9% is discharged in the condenser, and about 1% is equipment losses.

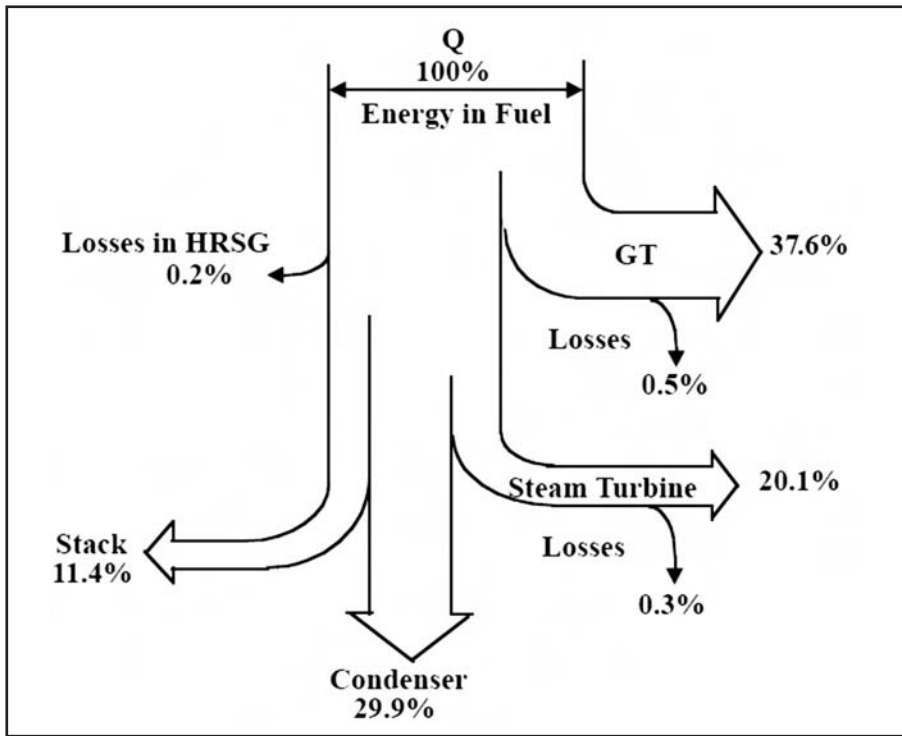


Figure 5–19 Energy flow diagram for the single-pressure combined-cycle power plant

Main design parameters of the single-pressure cycle

Live-steam pressure. In a combined-cycle plant, a high live-steam pressure does not necessarily mean a high efficiency of the cycle. Figure 5–20 shows how the steam turbine output varies with live-steam pressure for a single-pressure cycle. Expanding the steam at a higher live-steam pressure will give a higher steam turbine output due to the greater enthalpy drop in the steam turbine. However, because of the higher evaporation temperature in the HRSG, less steam will be generated, resulting in a higher exhaust gas temperature and, therefore, in a lower HRSG efficiency. An optimum is normally found between these two influences. In the pressure range shown in fig. 5–20, the optimum live-steam pressure is actually higher than 110 bar. However, at 110 bar the moisture content in the steam exhausted into the condenser is already 16%, which is actually quite high for the last

stage of the steam turbine and is, therefore, the limiting factor for the selection of the live-steam pressure.

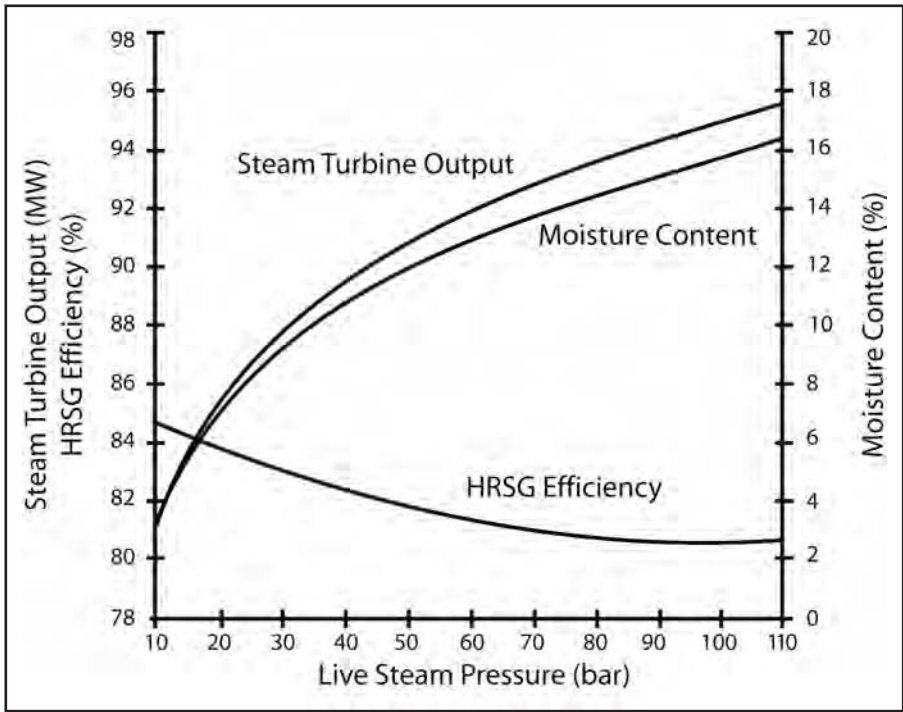


Figure 5-20 Effect of live-steam pressure on steam turbine output for a single-pressure cycle (including steam turbine exhaust moisture content and HRSG efficiency)

The impact of the moisture content on the polytropic steam turbine efficiency is shown in the appendices (equation A-29).

It is interesting that the high HRSG efficiency does not correspond to the high steam turbine output. At a lower live-steam pressure there is a lower stack temperature and the energy in the exhaust gas is well utilized. However, the steam turbine output at this live-steam pressure is lower because of higher exergy losses in the HRSG. The reverse happens at higher dry air cooling section. It shows that exergy is more dominant than energy in determining the steam turbine output.

This effect is illustrated in figure 5–21, the energy/temperature diagram for two different single-pressure cycles with live-steam pressures of 40 and 105 bar (566 and 1508 psig). At the lower live-steam pressure, more thermal energy is available for evaporation and superheating because the evaporation temperature is correspondingly lower. The pinch point of the evaporator is the same in both cases. As a result, the stack temperature at 40 bar is about 11°C (20°F) lower than at 105 bar, which means that more waste heat energy is being utilized.

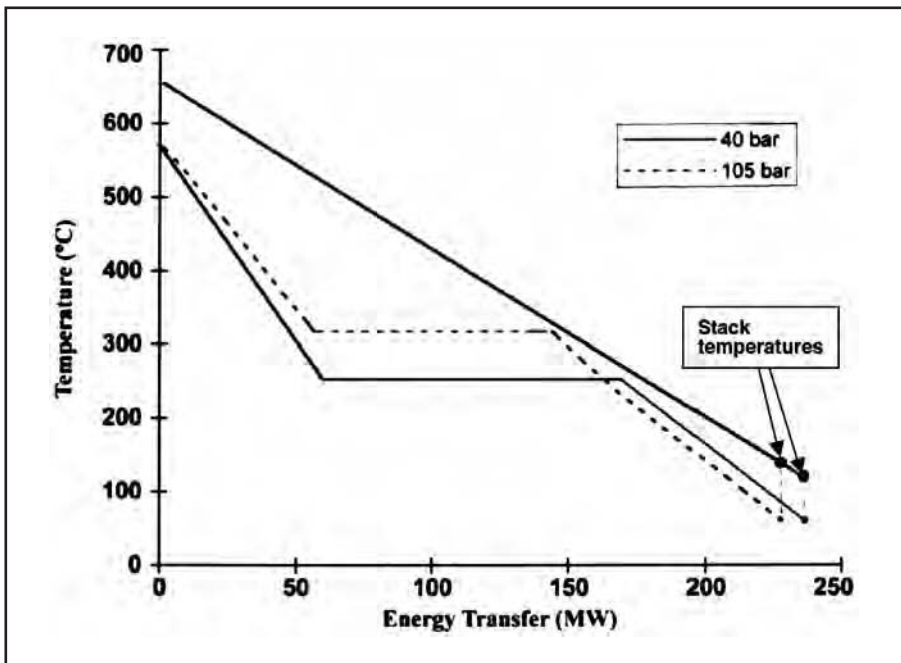


Figure 5–21 Energy/temperature diagram of a single-pressure HRSG with live-steam pressure of 40 and 105 bar (566 and 1508 psig)

A negative aspect of a higher live-steam pressure in a single-pressure cycle is the increase in the moisture content at the end of the steam turbine. Too much moisture increases the risk of erosion, taking place in the last stages of the turbine. A limit is set at about 16%. The moisture content depends also very much on the condenser pressure: the higher the pressure, the lower the moisture content.

A change in the live-steam pressure affects the amount of heat to be removed in the condenser as shown in figure 5–22 (because of change in the steam mass flow).

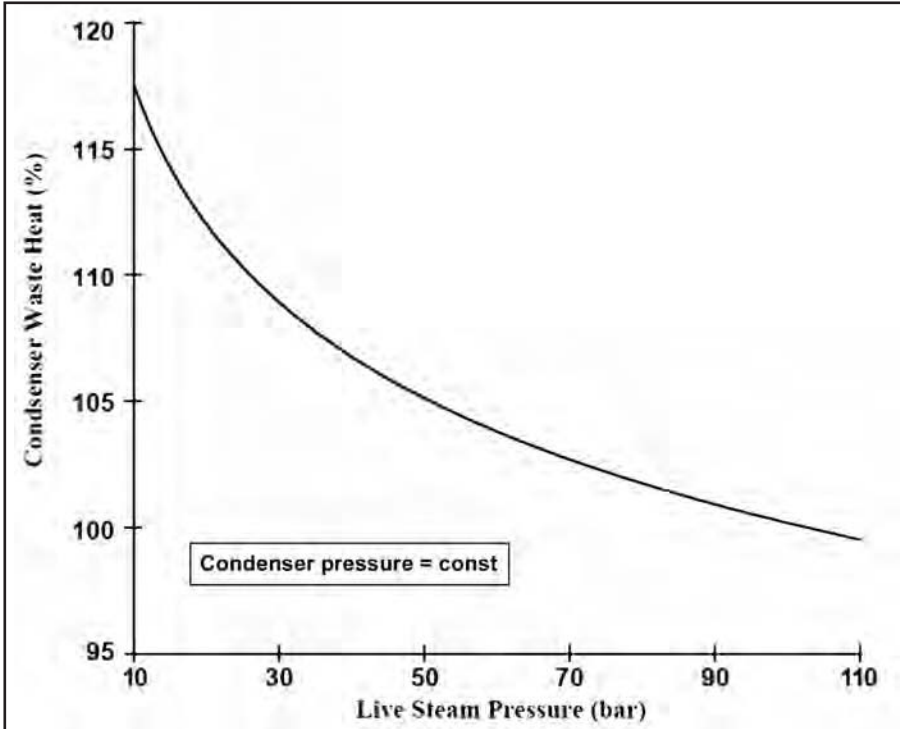


Figure 5–22 Effect of live-steam pressure on condenser waste heat at constant condenser pressure

A higher live-steam pressure creates the following economical advantages:

- A smaller exhaust section in the steam turbine
- A smaller volume flow leading to smaller live-steam piping and valve dimensions
- A reduction of the cooling water requirement
- Smaller cooling water equipment (condenser, pumps, cooling tower, etc.)

This can lead to considerably lower costs, especially for power plants with expensive air-cooled condensers.

The total amount of live steam can also influence the optimum live-steam pressure because it has an influence on the steam turbine efficiency. Increasing the amount of live steam will result in a larger volume flow at the turbine inlet, leading to longer blades in the first row of the turbine, which reduces the secondary blading losses. It follows that the optimum live-steam pressure also depends on the type of gas turbine used because different exhaust gas conditions will result in different live-steam flows for a given live-steam pressure.

In the example, a pressure of 105 bar (1508 psig) is chosen because it is the highest pressure possible without excessive moisture content in the steam turbine exhaust.

Live-steam temperature. For the chosen live-steam pressure as shown in figure 5–23, raising the live-steam temperature causes a slight decrease in steam turbine output. This is the result of two opposite effects. Increasing the live-steam temperature (same as when live-steam pressure is increased) results in a greater enthalpy drop in the steam turbine; however, at the same time, additional superheating removes energy, which would otherwise be used for steam production, resulting in a lower steam flow as well as a corresponding loss in the steam turbine output and an increase in the stack temperature. The latter is the dominant effect in this case.

The live-steam temperature cannot be reduced below a certain limit for a given live-steam pressure because of the resulting increase in the moisture content in the steam turbine. To use a lower live-steam temperature, the pressure would have to be reduced, which, would result in a more negative effect on the steam turbine output than raising the live-steam temperature.

In the example with 105 bar (1508 psig) live-steam pressure, a temperature of 568°C (1054°F) has been chosen, which is close to the upper limit for current materials used for these plants.

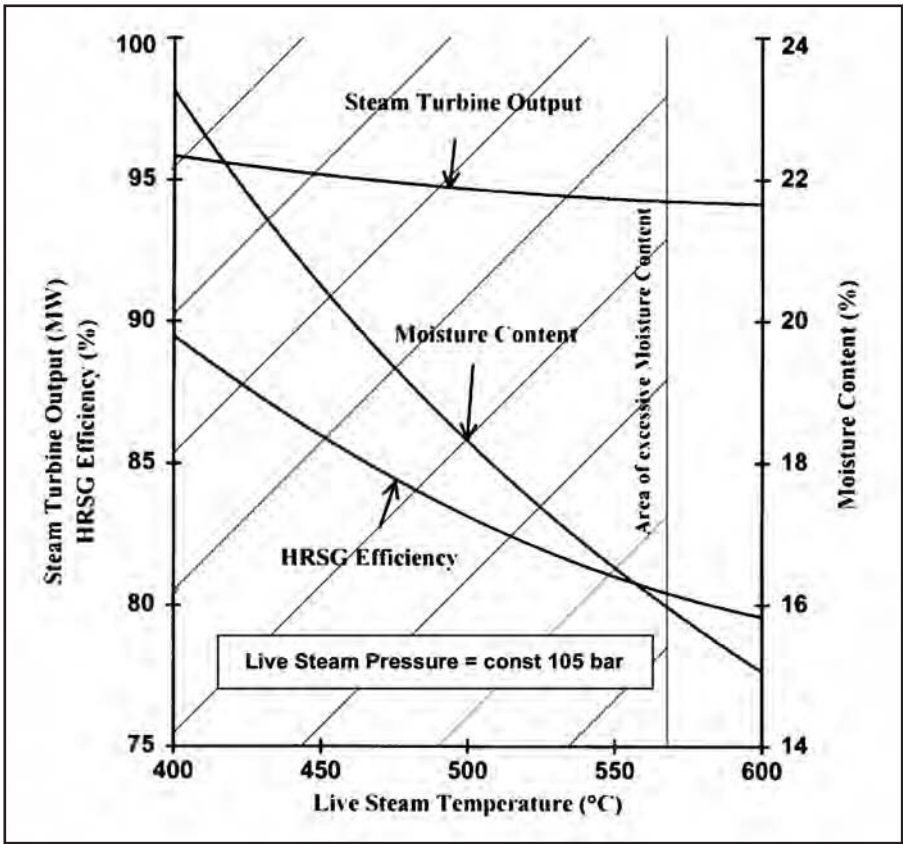


Figure 5–23 Effect of live-steam temperature on steam turbine output for a single-pressure cycle with 105 bar (1508 psig) live-steam pressure (including HRSG efficiency and steam turbine exhaust moisture content)

For gas turbines with lower exhaust gas temperatures than the one in the example, a lower live-steam pressure level would have to be chosen for the reasons previously discussed.

Design Parameters of the HRSG

Pinch point and approach temperature

An important parameter in the optimization of a steam cycle is the pinch point of the HRSG (defined in figure 5–18), which directly affects the amount of steam generated. Figure 5–24 shows how by reducing the pinch point (with other parameters kept constant) steam turbine output can be increased. This is due to a better rate of heat utilization in the HRSG. However, the surface of the heat exchanger—and, hence, the cost of the HRSG – increases exponentially as the pinch point tends towards zero. In figure 5–24, a reference pinch point of 12K (21.6°F) is used.

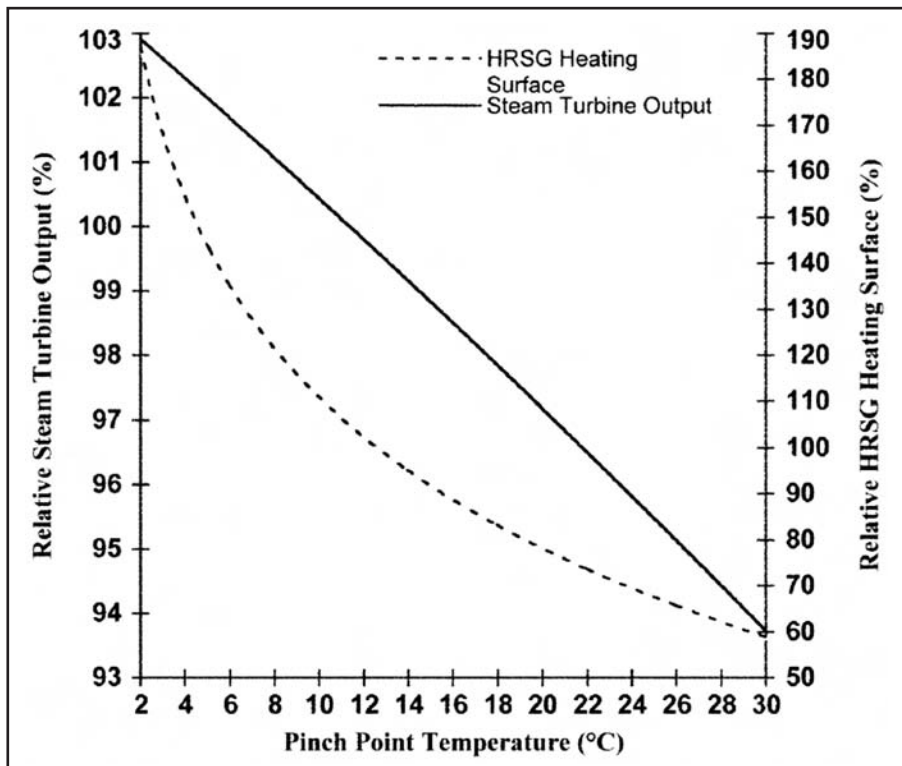


Figure 5–24 Effect of pinch point on relative steam turbine power output and relative HRSG heating surface

Similarly, a smaller approach temperature (defined in figure 5–18) leads to a better heat utilization but increased HRSG surface. For a drum-type HRSG, a lower limit is set on the approach temperature by the need to minimize steaming in the economizers for off-design points. Our examples use 5K (9°R) for the approach temperature.

It is the sum of the pinch point and approach temperatures that determine the steam production in the HRSG for a given live-steam pressure and temperature. This means that a HRSG with 10K (18°F) pinch point and 5K (9°F) approach temperature would have the same steam production as one with 5K (9°F) pinch point and 10K (18°F) approach temperature. However, the HRSG surface will not be the same in each case (due to a different logarithmic temperature difference in heat exchange) and so the optimum must be found, bearing in mind the limitation of the steaming in the economizers and the impact of the HRSG surface.

Pressure drop on the water/steam side

A further influence on the energy available for evaporation and superheating is the steam side pressure drop in the superheater. A higher pressure drop for a given live-steam pressure means that the evaporation takes place at a correspondingly higher pressure and temperature level, where less energy is available for steam production. Other pressure losses, such as those in the economizers, do not influence the steam production, but will have an effect on the power consumed by the feedwater pumps.

Pressure loss on the exhaust gas side

The design of the HRSG should be such that the pressure loss on the exhaust gas side (or backpressure of gas turbine) remains as low as possible. This loss strongly affects the power output and efficiency of the gas turbine because an increase in the backpressure will reduce the enthalpy drop in the gas turbine. Some of the lost output is, however, recovered in the water/steam cycle due to an increased gas turbine exhaust gas temperature. This effect is shown in figure 5–25.

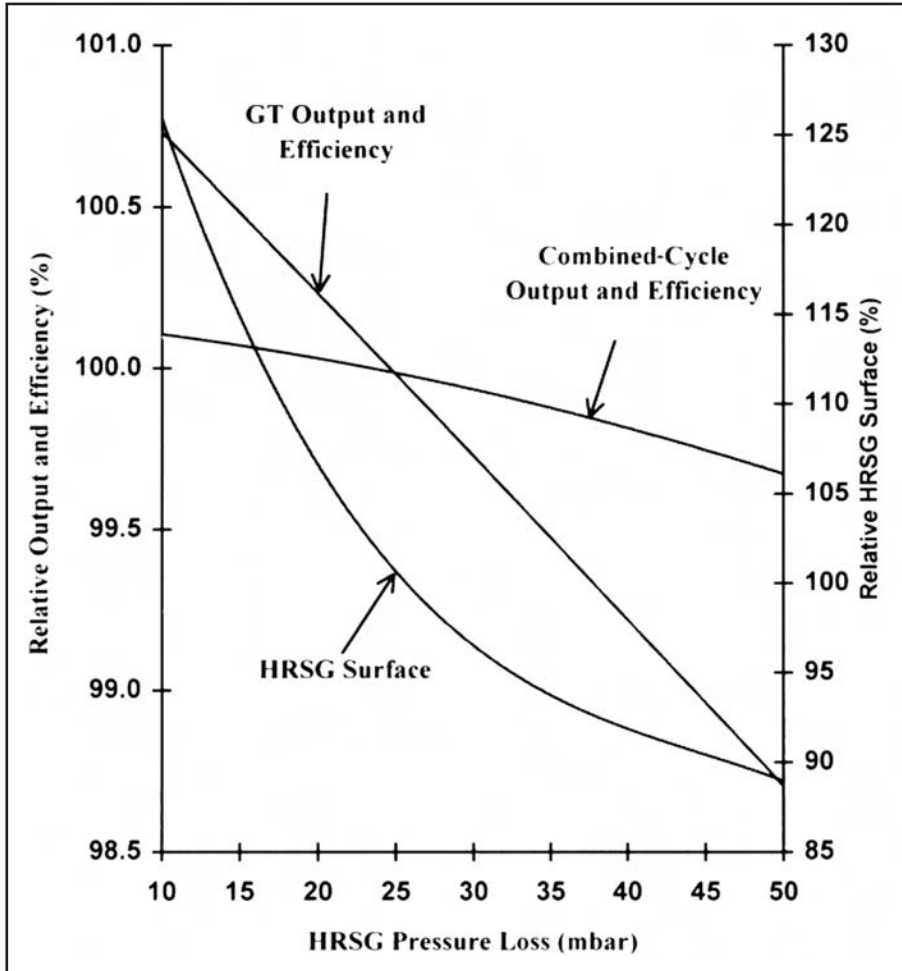


Figure 5-25 Influence of HRSG backpressure on combined-cycle output and efficiency, GT output and efficiency, and HRSG surface

Lower pressure losses on the flue gas side, however, have a negative impact on the heat transfer due to lower gas velocities around the tube bundle leading to higher HRSG surface. Typical HRSG exhaust gas pressure losses are between 25 to 30 mbar (9.9 to 11.8" WC). In the example, 25 mbar is used.

Feedwater preheating

Between a combined-cycle power plant and a conventional steam power plant, there are significant differences in the design/concept of the feedwater preheating system.

Combined-cycle plant. Based on equation (4–2) it is clearly visible that with a given gas turbine (power output and fuel input is fixed) the efficiency of the plant will be maximized if the steam turbine output is maximized. This is the case if no steam will be extracted from the steam turbine for feedwater preheating.

Conventional Steam Plant. With a regenerative preheating of the feedwater, an improved Rankine cycle (closer to the ideal Carnot) will be achieved. This means that with a higher feedwater temperature, less fuel input into the boiler is needed.

To attain a good rate of exhaust gas heat utilization in a combined cycle, the temperature of the feedwater should be kept as low as possible. Figure 5–26 demonstrates that for cycles with only one stage of preheating using a steam turbine, extraction output and efficiency fall sharply as the feedwater temperature is increased. This is because the exhaust gases can, ideally, only be cooled down to a temperature of about 10 to 15K (18 to 27°F) above the feedwater temperature. The higher the feedwater temperature, the hotter the HRSG stack temperature, the more energy is wasted.

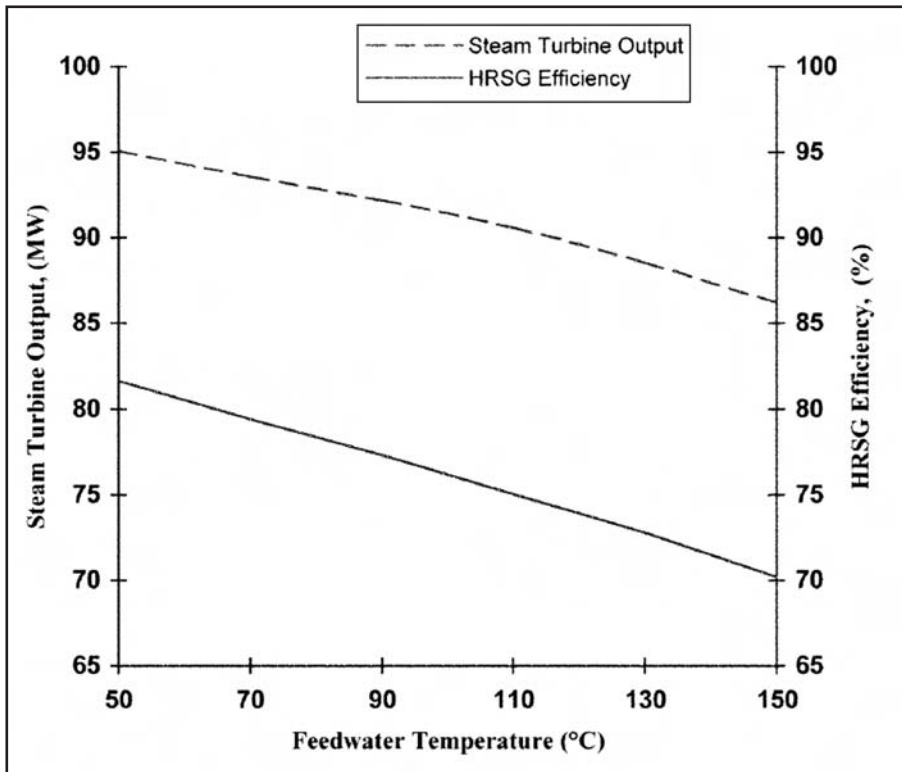


Figure 5–26 Effect of feedwater temperature on steam turbine output and HRSG efficiency for cycles with one stage of preheating

This is one significant difference between a conventional steam plant with a high feedwater temperature and the steam process in a combined-cycle plant. A conventional steam plant attains better efficiency if the temperature of the feedwater is brought to a high level by means of multistage preheating. There are two reasons for this difference.

A conventional steam generator is usually equipped with a regenerative air preheater that can further utilize the energy remaining in the flue gases after the economizer. This is not the case in a HRSG, where the energy remaining in the exhaust gases after the economizer is lost. In principle, air coming into the gas turbine could be similarly preheated, further lowering the stack temperature. However, this would lower the plant output significantly due to the decrease in air density at

the compressor inlet and the fact that the gas turbines are volumetric machines, so that the inlet air volume flow to the gas turbine is always constant, however the mass flow decreases.

Also, as shown in figure 5–27, the smallest temperature difference between the water and exhaust gases in the economizer of an HRSG is on the warmer end of the heat exchanger.

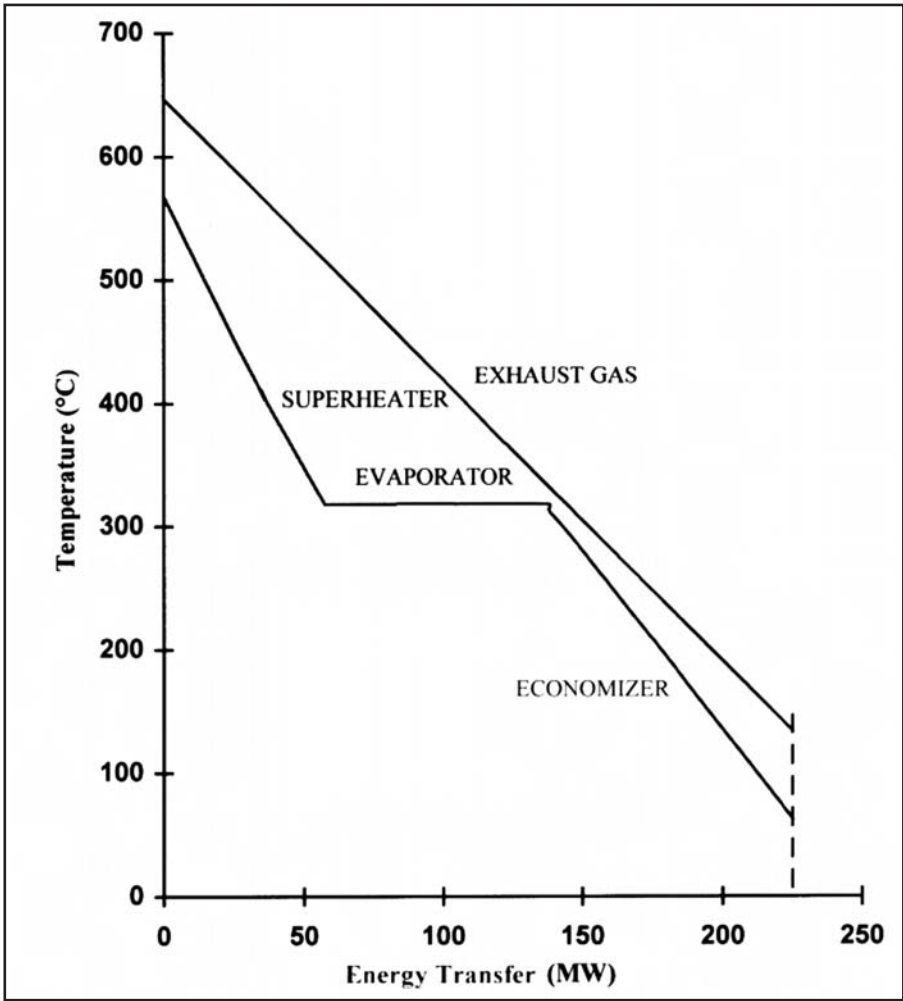


Figure 5–27 Energy/temperature diagram for a single-pressure HRSG

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 cold end of the economizer because the water flow is far larger in proportion to the flue gas flow. As a result, in conventional boilers, the amount of steam production possible depends on the feedwater temperature.

Figure 5–28 shows two examples of conventional steam generators with differing feedwater 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. Therefore, raising the feedwater temperature can increase the amount of live steam produced by a conventional boiler. Further, by preheating the boiler feedwater, the water/steam cycle efficiency is increased because more steam is fully utilized and less energy is lost in the condenser.

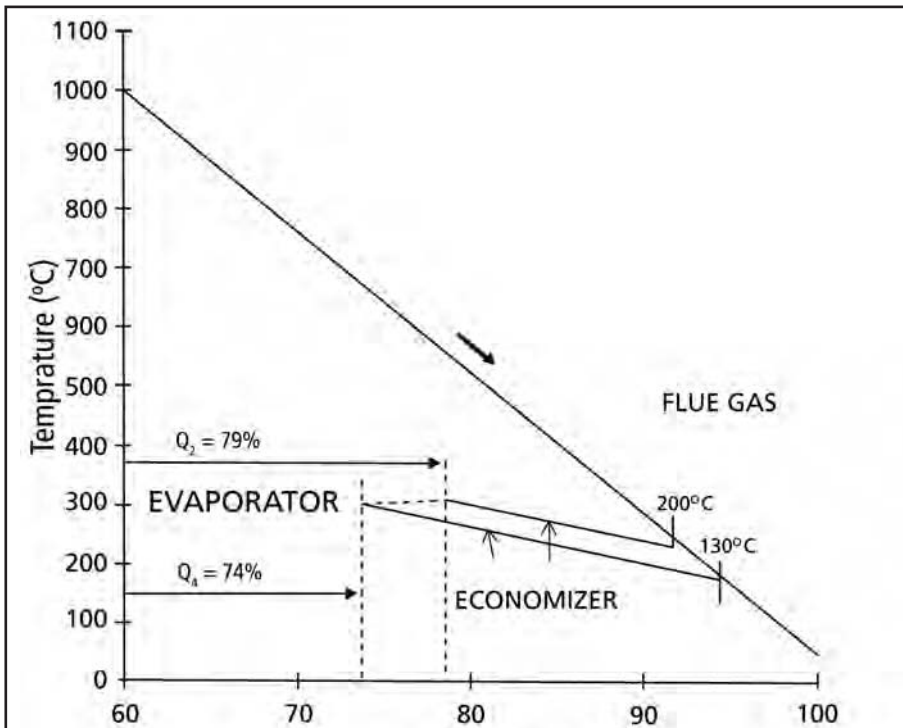


Figure 5–28 Energy/temperature diagram for a conventional boiler

6 Applications of Combined Cycles

The combined cycles described in chapter 5 are purely for electricity generation. In this chapter we explore how the thermodynamic advantages of combining gas and steam cycles can apply for other purposes—the production of thermal energy, or the repowering of existing steam power plants by integrating a gas cycle into the existing steam cycle. Some other, less conventional concepts are also discussed, although most of these have limited commercial usage.

Cogeneration

Cogeneration means the simultaneous production of electrical and thermal energy in the same power plant. The thermal energy is usually in the form of steam or hot water. The types of cogeneration plants discussed in this chapter fall into three main categories:

- Industrial power stations supplying heat to an industrial process
- District-heating power plants
- Power plants coupled to seawater desalination plants

The thermodynamic superiority of a combined-cycle plant over a conventional steam power plant is even more pronounced in cogeneration plants than in plants used to generate electricity alone. As

shown in chapter 4, table 4–1, the temperature difference between the heat supplied to the cycle and the waste heat rejected to the environment is much bigger in a combined-cycle plant than in a steam power plant. This difference increases in relative terms if useful process heat is produced. In this case, the useful temperature drop gets reduced by the same amount, leading to an even higher ratio between combined-cycle plants and steam turbine plants.

A cogeneration plant may also have supplementary firing in the heat recovery steam generator (HRSG). This offers greater design and operating flexibility because the steam production can be controlled, independently of the electrical power output, by regulating the fuel input to the HRSG. The power output is controlled by the gas turbine. The cycle efficiency will, however, be normally lower if supplementary firing is used, not always making this an economically desirable solution.

Thermal energy in the form of steam can be extracted from the HRSG, the live-steam or reheat steam lines, or from an extraction in the steam turbine. The optimum extraction point depends on the required steam pressure, temperature, and quantity over the load range. These parameters are very important in defining the type of cycle that should be used. The advantages and disadvantages of various solutions are discussed in the following sections.

Industrial power stations

Wherever both electrical power and process steam are needed, it is thermodynamically and usually also economically better to produce both products in a single plant. The number of possible concepts is large because each plant is, to a certain extent, unique. The following examples have only one process steam supply. Often, steam supplies at different pressure levels are required, but the basic considerations remain unchanged. There are three basic steam extraction concepts that relate to the steam turbine:

- Cycle with back pressure steam turbine
- Cycle with extraction/condensing steam turbine
- Cycle with no steam turbine

The last concept is not really a combined-cycle power plant, but rather a gas turbine plant with HRSG.

A simplified flow diagram for a cycle with a back pressure steam turbine is shown in figure 6–1. The steam turbine (7) is designed so that the exhaust pressure matches the process requirement (3). A bypass (5) around the steam turbine is designed to reduce the live steam to the required pressure and temperature level for the process in case the steam turbine is out of operation and the process must still be supplied. In this cycle all of the steam generated in the HRSG goes into the process, except for a small quantity used for feedwater tank heating and deaeration.

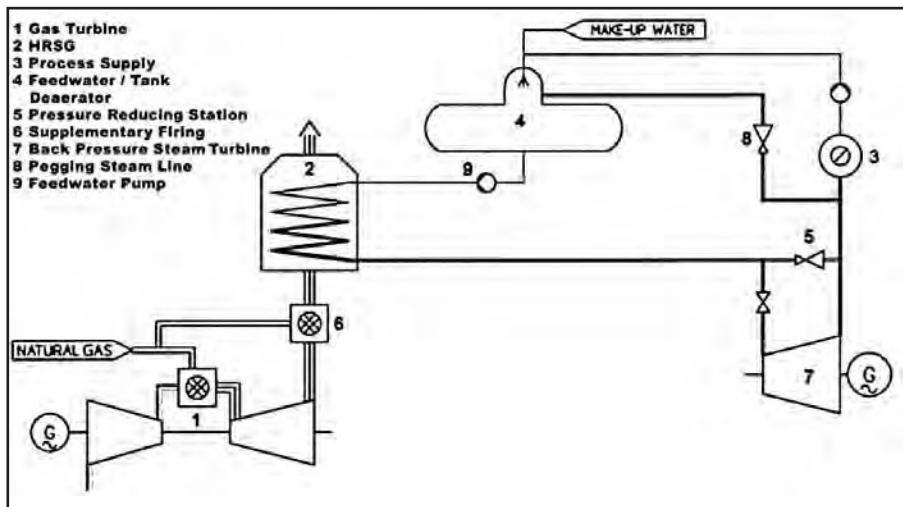


Figure 6–1 Simplified flow diagram of a cogeneration cycle with a back pressure turbine

For such a cycle, all or some of the steam supplied to the industrial process may be returned as condensate. Any loss in mass flow must be replaced with makeup water that usually enters the cycle at the deaerator because it is fully oxygenated. If condensate is returned from an industrial process, the quality may be poor and it may have to be treated before re-entering the water/steam cycle to achieve a suitable quality. If the return condensate is at a high temperature, this may

A cycle with no steam turbine is shown in figure 6–3. This is appropriate if a large quantity of process steam is required at a high pressure and temperature, leaving little pressure drops for a steam turbine expansion. In this example, there is supplementary firing to achieve sufficient process steam mass flow. The steam goes directly from the HRSG to the industrial process (5) and the HRSG is designed to generate steam at the process conditions. The pressure may be controlled by a pressure reducing station (4), but it is quite common for the main pressure control to take place within the plant to which the process steam is being exported.

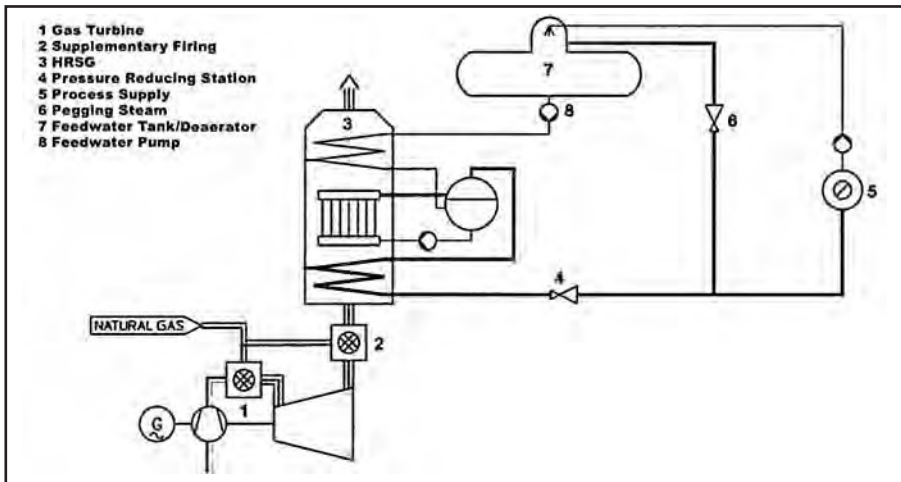


Figure 6–3 Flow diagram of a cogeneration cycle with no steam turbine

Evaluation of a cogeneration cycle

The performance of a cogeneration cycle is defined not only by the efficiency, but also by parameters such as fuel utilization and power coefficient. These parameters take into account the thermal as well as the electrical output.

Fuel utilization is a measure of how much of the fuel supplied is usefully used in the plant. It is equal to the sum of electrical output and thermal output divided by the fuel input.

The power coefficient (also called the alpha-value) is defined as the ratio between the electrical and the thermal output. Combined cycles tend to have high power coefficients; they are, therefore, more likely to be used for cogeneration applications with a relatively high power demand. This is because the electrical output of the gas turbine—about two-thirds of the total plant output—cannot be converted into thermal energy. In a conventional steam power plant, all of the energy produced could be exported to the process, giving a possible power coefficient of zero.

Figure 6–4 is a heat balance for a cycle with a back-pressure turbine using the 178 MW gas turbine and a single-pressure cycle with supplementary firing to 750°C (1382°F). The exhaust pressure of the steam turbine (and, hence, the process pressure) is 3 bar (29 psig). Condensate returns from the process at 50°C (122°F). A small amount of steam is taken from the steam turbine exhaust for feedwater heating to achieve a feedwater temperature of 60°C (140°F).

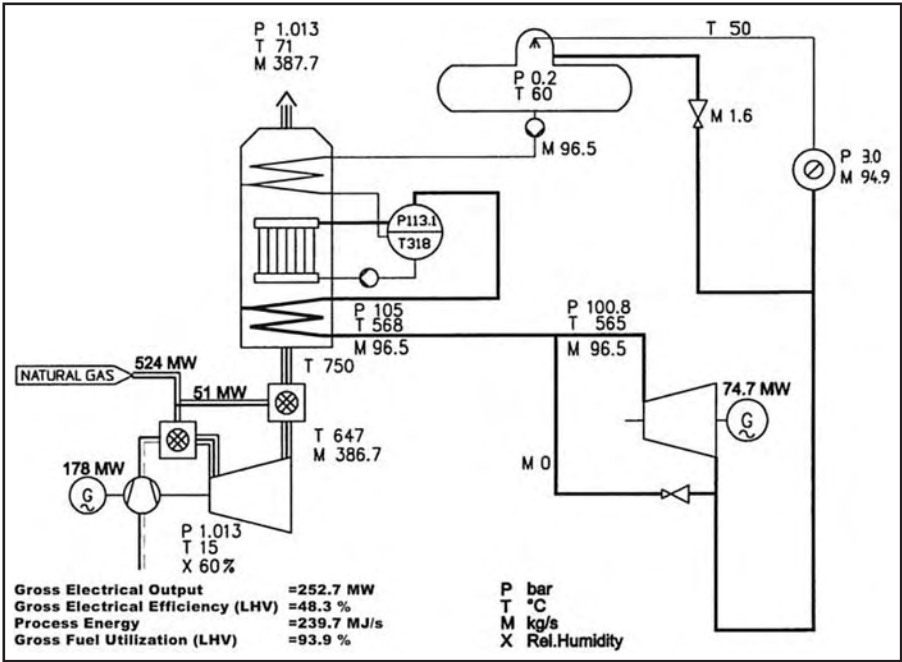


Figure 6–4 Heat balance for a single-pressure cogeneration cycle with supplementary firing

The cycle is directly comparable to figure 5–60 (single-pressure cycle with supplementary firing for electricity generation only), except that the cold condenser of figure 5–60 is replaced by the industrial process operating at a higher pressure level. Performance of the HRSG is the same in both cycles with identical stack temperatures and steam flows. Due to the high turbine back pressure, the output in the cogeneration cycle is 50.8 MW less than in the condensing cycle. The electrical efficiency decreases from 58% to 48.3%. However, 239.7 MJ/s of thermal energy is exported to the process in this case. This results in a fuel utilization of 93.9%. The remaining 6.1% of the energy is lost through the stack (5.5%), in radiation losses, and in generator and mechanical losses in the steam turbine and gas turbine. There is no condenser loss because all of the steam is used in the process. The power coefficient is 105%.

The power coefficient of a plant is mostly affected by three parameters:

- Amount of fuel supplied directly to the boiler (if any)
- Size of the condensing portion of an extraction/condensing turbine
- Pressure level of the process steam

Figure 6–5 shows the effect of process steam pressure on the relative power output and power coefficient for a combined cycle with a back-pressure steam turbine. The relative power output is the ratio between the electrical output of the considered cogeneration plant and the output of a condensing plant based on the same gas turbine. The reference point at 100% relative power output is equivalent to a condensing cycle. The higher the process steam pressure, the less electricity is produced because there is a smaller enthalpy drop in the steam turbine. In theory, the power co-efficient approaches 170% for back pressures equal to those of condensing applications. As the pressure increases the power coefficient falls exponentially to the limit set by the gas turbine output because the steam turbine electrical output is falling and the thermal output rising at the same time.

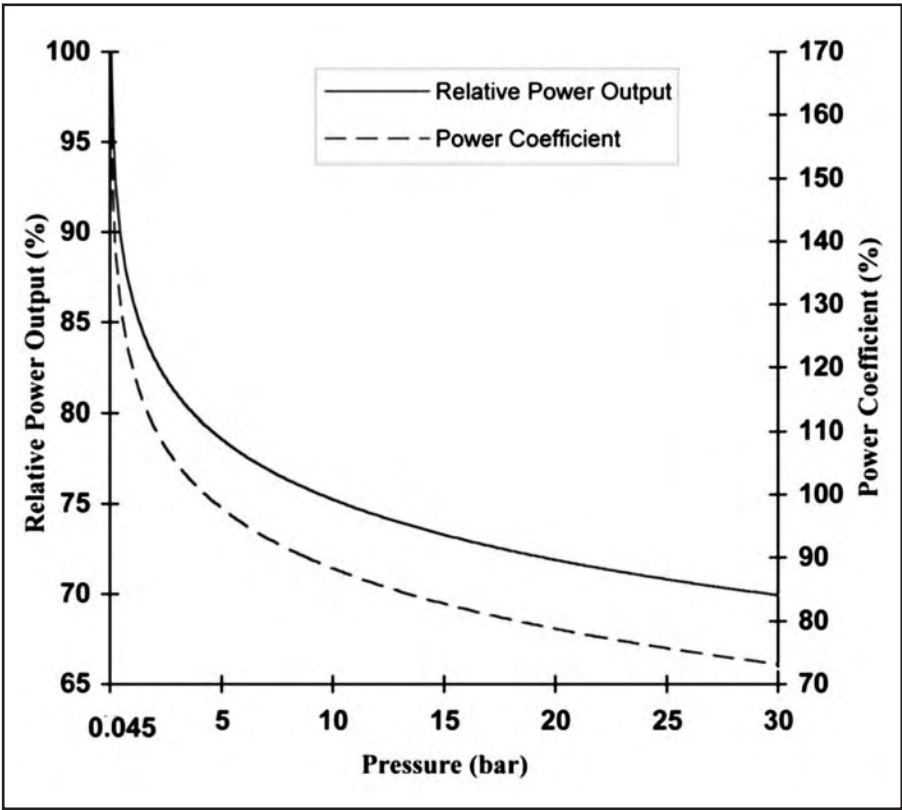


Figure 6–5 Effect of process steam pressure on relative combined-cycle power output and power coefficient for a single-pressure cycle with 750°C supplementary firing

As the power coefficient increases, the electrical efficiency approaches asymptotically that of a fully condensing combined-cycle plant (figure 6–6). The fuel utilization remains nearly constant because in a plant with no condenser, almost all of the steam entering the steam turbine is usefully used. With a variation of the process steam pressure, there is a shift between electrical and thermal output. The sum of both remains nearly constant except for some losses in the generator and transformer.

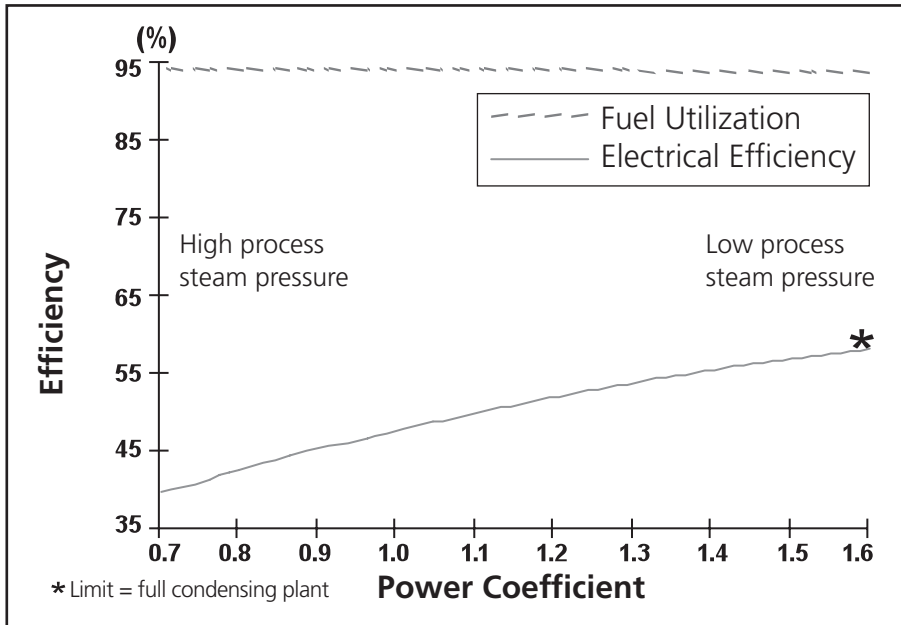


Figure 6-6 Effect of power coefficient on electrical efficiency and fuel utilization for a single-pressure cycle with 750°C supplementary firing

Where supplementary firing is installed, additional steam generation contributes to the electrical output of the steam turbine only and has no impact on the gas turbine output. Thermal output, therefore, increases at a much greater rate than electrical output with increasing supplementary firing temperature resulting in a decrease in the power coefficient.

If higher power coefficients are required, an extraction/condensing steam turbine offers greater design and operating flexibility. The condensing portion of the turbine enables the electrical power output to be increased, but only at the cost of thermal output and fuel utilization. If designed properly, such a cycle can operate at any point in the range from full condensing to full back-pressure mode. For back-pressure mode, cooling steam may be required for the low-pressure part of the steam turbine.

The highest fuel utilization can be obtained with a cycle with a back-pressure turbine but the flexibility is poor because the steam production can only be reduced by decreasing the gas turbine load. This measure would, however, have a negative impact on the electrical efficiency.

Main design parameters

The behavior and the influence of the design parameters of a combined-cycle power plant for cogeneration application is similar to the plants described in chapter 5 designed for pure power generation.

The biggest difference is because with increased process-steam pressure it may be better to increase the life steam pressure to keep a reasonable high enthalpy drop in the steam turbine. This is especially true when back-pressure steam turbines with a high back pressure are used.

However, this leads to poorer heat utilization in the HRSG in the case of a single-pressure process. To avoid these losses, a multipressure cycle can be used where the low-pressure steam can directly be fed into the process steam system.

Figure 6–7 shows a cycle using an unfired dual-pressure HRSG (2) and a back-pressure turbine (3). The LP steam is fed directly into the process steam system (4). The process steam supply is backed up with a pressure reducing station (6) from the HP live-steam line. It may be necessary to regulate the temperature of the process steam by injecting feedwater in the process steam line.

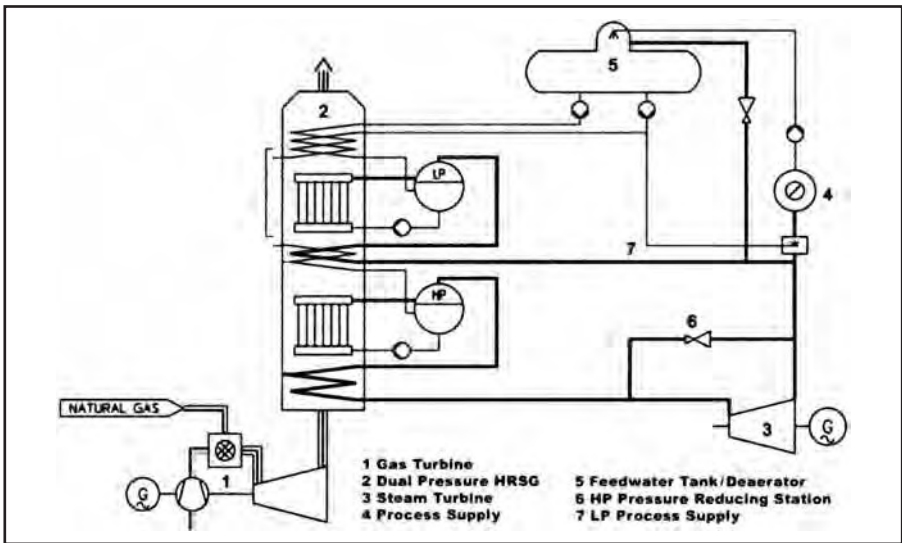


Figure 6–7 Flow diagram of a cogeneration cycle with a dual-pressure HRSG

District Heating Power Plants

A district heating system uses hot water or steam to supply heat for house heating. An efficient way to produce this hot water or steam is by extracting steam from a power plant at the required temperature level. A combined-cycle power plant is well suited for this type of cogeneration of power and heat.

District heating water is required at temperature levels much lower than those used for processes in industrial power plants. Usual forwarding temperatures are between 80 and 120°C (176 and 248°F) with return temperatures of 40 to 70°C (104 to 158°F).

Steam extracted from the LP part of the steam turbine is used to heat this water. For exergetic reasons in larger installations it is better to divide the heating into two or even three pressure stages. Figure 6–8 shows the temperatures of hot water and steam required in a one-stage and a three-stage process. Where a single stage is used, all steam must be extracted at the higher pressure level. This means there is less steam expanding in the steam turbine, so the steam turbine power output is lower than the one with a three-stage system. If only a small temperature increase in the district heating water is required, a single stage of heating may be good enough.

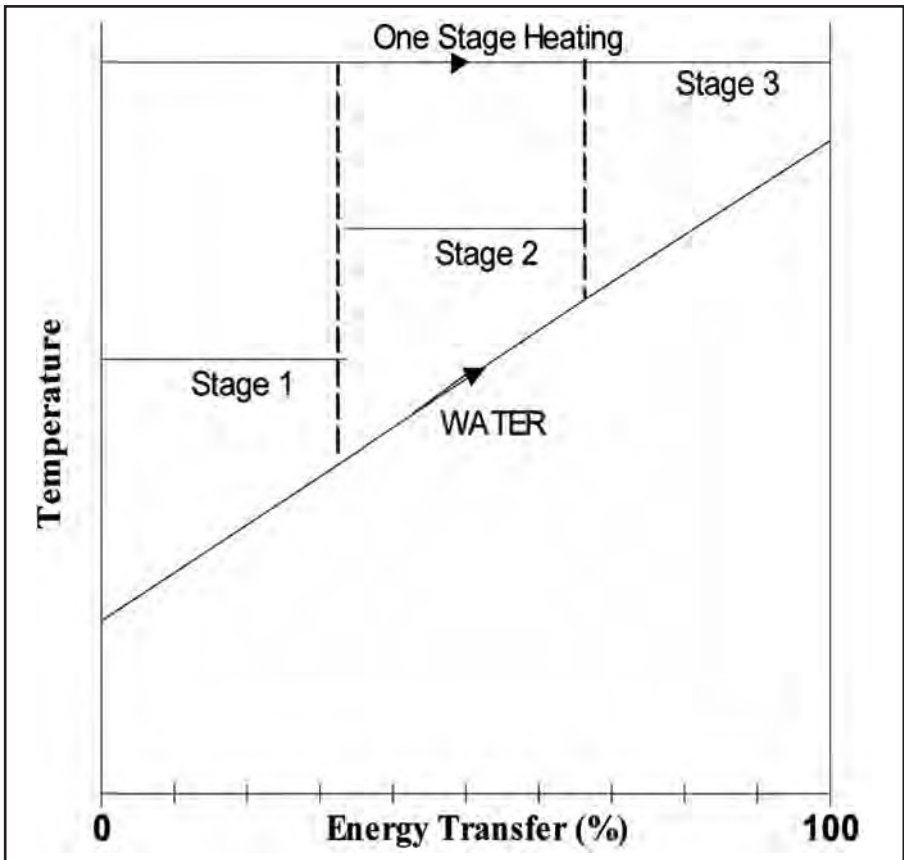


Figure 6-8 Comparison of one-stage and three-stage heating of district heating water

The main design criteria for district heating power plants are, therefore, the heat output and the heating water temperatures, according to the ambient temperature. The strong positive influence that low air temperature has on power output of the combined-cycle plant is an advantage in this case, as maximum heating output is demanded, when ambient temperatures are lowest. The design temperatures of the district heating water represent a compromise between maximization of electrical output and low costs for transportation of the heat.

The cycle must have a high degree of operating flexibility, but must not become too complicated or too expensive. In particular, district heating power plants should not be designed for extreme demand conditions because this would make them too expensive and bring into

question the economical feasibility. Only about half of the peak heat demand will be covered by the cogeneration plant.

In district heating power plants, independent control of electricity and heat production is not always required. These power plants are usually integrated into large grids, where other power stations are available for adjusting the total electrical power output to meet the demand. Usually the only output parameter that must be controlled in the district heating plant is the thermal output and the water temperature.

Figure 6–9 shows the flow diagram and heat balance for a typical example of combined-cycle plant using an extraction/back-pressure turbine with heating in two stages and a dual-pressure cycle (with the same HRSG as figure 5–33). The first stage of district heating is done using the steam turbine exhaust steam. The exhaust steam, at a pressure of 0.38 bar (11.2"Hg), goes into the first district heating condenser where it is used to raise the temperature of the district heating water from 50 to 70°C (122 to 158°F). An extraction from the steam turbine at 0.78 bar (23"Hg) is used for the second stage of district heating, bringing water temperature up to 90°C (194°F).

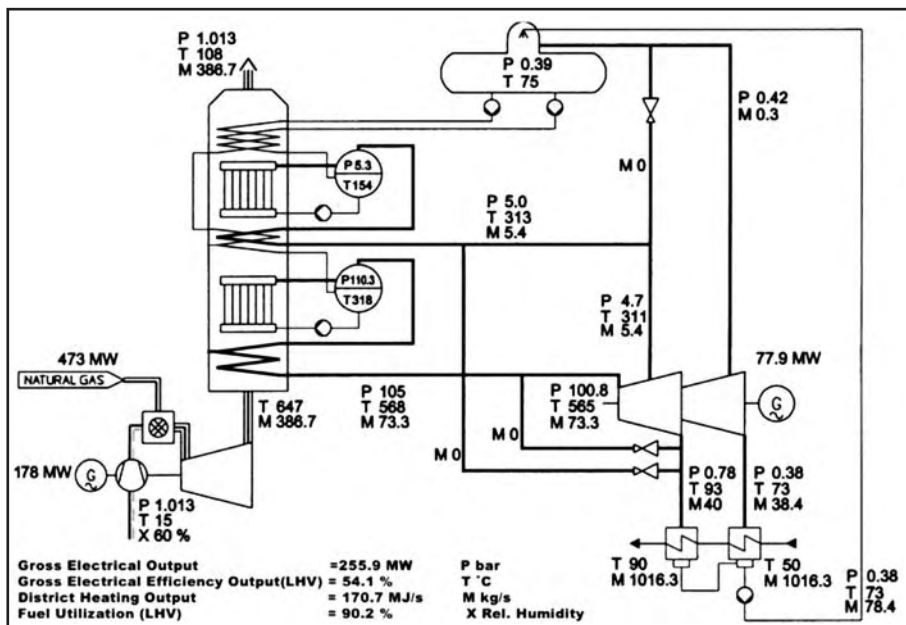


Figure 6–9 Heat balance for a cycle with two stages of district heating

Approximately half of the steam mass flow is used in each district heating condenser, and at the design point it is optimal to have the same heat load in each stage, as shown in this case. Condensate from the second-stage condenser cascades into the first-stage condenser and is returned to the feedwater tank.

Compared to the condensing cycle of figure 5–34, steam turbine output has fallen by 21.1 MW due to the steam extraction and the higher back pressure. The electrical efficiency falls from 58.6% to 54.1%, but the fuel utilization is reaching 90.2%.

Figure 6–10 shows a cycle using an extraction/condensing steam turbine. This time, each stage of district heating uses a steam turbine extraction, and there is a cold condenser at the exhaust of the steam turbine into which the steam not used for district heating is condensed after having been expanded in the low pressure steam turbine. This type of cycle can be used if there is a need to increase the flexibility of operation between district heating and electricity generation or in case a higher electrical output is required.

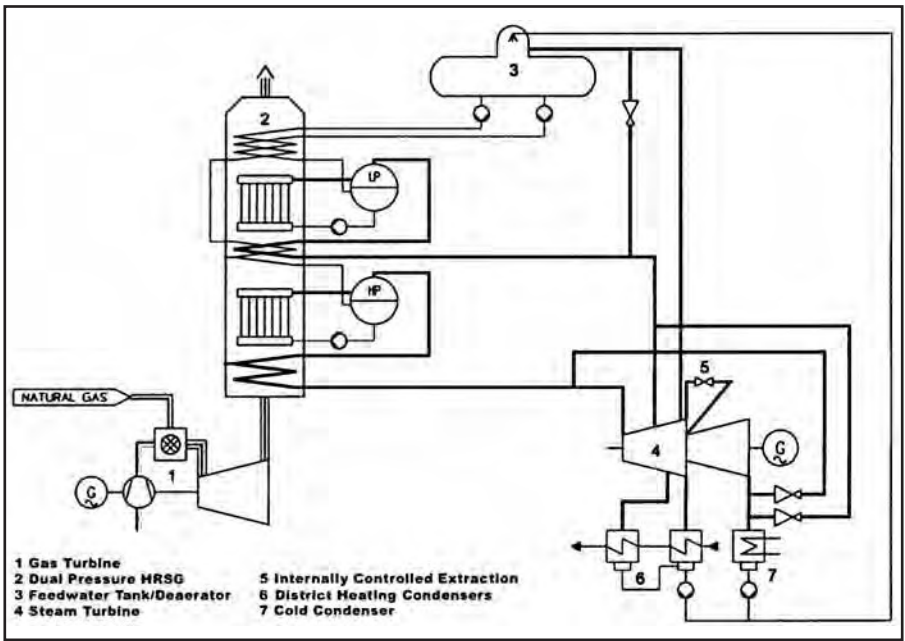


Figure 6–10 Flow diagram of a district heating/condensing cycle

Cycles coupled with seawater desalination units

In desalination plants, sweet water is produced out of salt water (usually seawater). There are several possible processes to desalt seawater, but often the process involves distillation, most of them use heat to evaporate part of the seawater. By condensation of the steam produced sweet water is obtained. A combined-cycle plant is a good source of heat for this evaporation process.

Combining a power station with a seawater desalination unit is particularly suited to combined-cycle plants because such power plants are generally built in oil- and natural gas-rich countries, where ideal fuels for combined cycles are readily available at a reasonable cost.

Larger desalination plants are usually designed to use the multistage multistage process. Seawater is heated in a row of cells with increasing pressure with the final stage of heating (at around 100°C [212°F]) taking place in a heater supplied with steam from the combined cycle. Heated water returning through the cells heats incoming seawater in a counter flow heat exchange process. On entry to each cell the heated water undergoes a sudden pressure drop, causing instant evaporation of a part of the water, known as *flashing*. A part of the steam produced then condenses and is collected while the remainder recondenses, heating the cold water entering the cell before it moves on to the next stage. The heating steam is usually returned to the water/steam cycle in the form of condensate that—provided the quality is in order—can be admitted directly into the feedwater tank.

The maximum temperature of the water being heated is limited to prevent excessive formation of calcium carbonate (CaCO_3) deposits. This limit is between 90°C (194°F) and 120°C (248°F) depending upon the type of additives being used. Corresponding heating-steam pressures are between 1 and 2.5 bar (0 and 22 psig), which are ideal for a combined-cycle plant because a back-pressure steam turbine with a high enthalpy drop can be used, ensuring a high electrical output.

The electrical power output and the flow of process steam must usually be controlled independently of one another, so it is appropriate to install supplementary firing or to use an extraction-condensing steam turbine.

Figure 6–11 shows a typical flow diagram based on a single-pressure, supplementary-fired HRSG, a back-pressure turbine that feeds a multiflash desalination system.

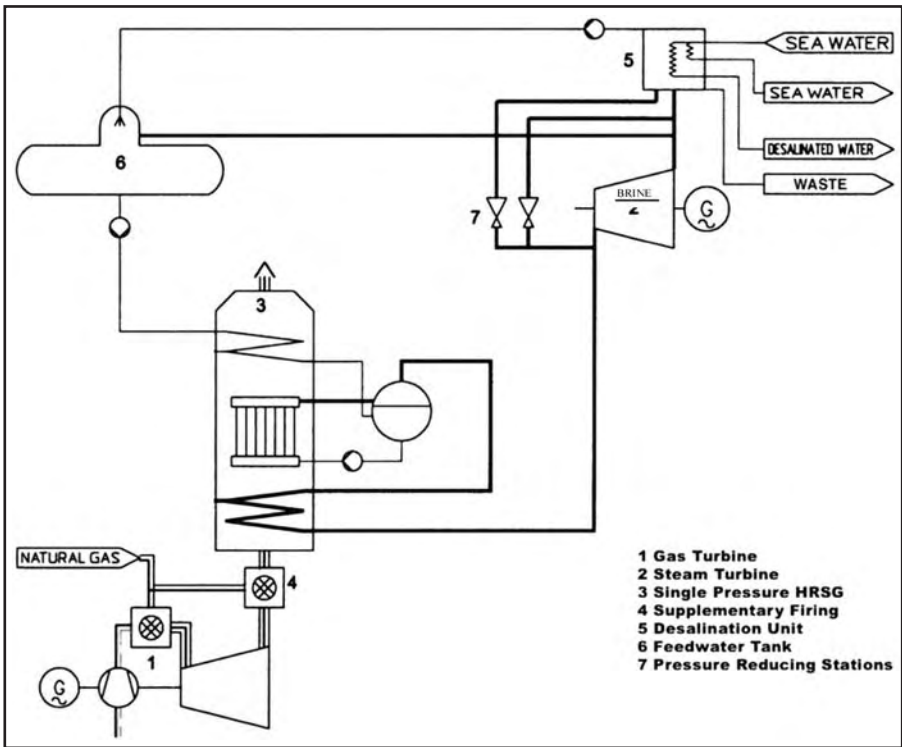


Figure 6–11 Flow diagram of a cycle coupled with a seawater desalination plant

Heating steam is directly exhausted from a back pressure turbine (2) to the desalination unit (5), where it is used to heat the seawater. The steam condenses in the desalination process and is returned to the feedwater tank (6). If the steam turbine is out of operation, supply of the desalination steam is provided through a pressure-reducing station (7). The high quantities of process steam at a low pressure require very large diameter pipes to the desalination unit.

With a combined cycle, electrical efficiencies of more than 50% can be reached for such desalination applications. A conventional steam power plant or gas turbine would not reach this level of efficiency.

A combined-cycle plant is less suitable if the ratio between fresh water produced and electrical power must be high, such as a low power coefficient. In that case, either the additional steam demanded for the desalination unit must be supplied from an auxiliary boiler, or a different type of power plant must be chosen.

Repowering

In this part various applications of the principle of combined cycle to repower existing steam or gas power plants are described.

Converting existing steam power plants into combined-cycle plants is known as repowering. It is ideal for plants in which the steam turbine, after many years of operation, still has considerable service life expectancy, but the boilers are ready for replacement. The boilers are normally replaced or supplemented with gas turbines and HRSGs. Steam turbine units in older power stations generally have relatively low live-steam data and can easily be adapted for use in a combined cycle. Repowering to a combined cycle can improve the efficiency of an existing plant to a level relatively close to that of new combined-cycle plants. Some plants are repowered purely to benefit from this efficiency increase, even though they are far from the end of their design life.

Conversion of conventional steam power plants

Figure 6–12 is a simplified flow diagram for a typical conventional steam turbine plant before repowering. The conventional boiler (1) generates steam for the nonreheat steam turbine (2). There are four extractions from the steam turbine for feedwater preheating—three to LP preheaters (4) and one to the feedwater tank (3). Condensing in this case is done using a water-cooled condenser (5). HP preheaters and reheat are not shown in this example, but their presence would not affect the principle of the repowering concept.

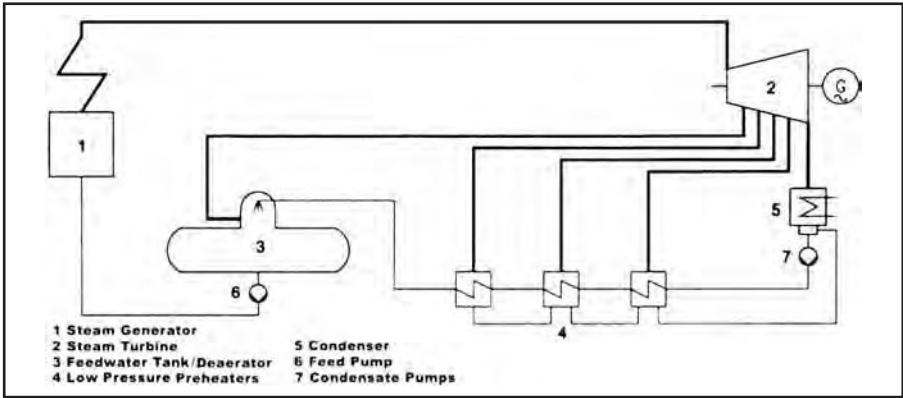


Figure 6–12 Flow diagram of a conventional nonreheat steam power plant

Figure 6–13 shows the same cycle after repowering with an oil-fired gas turbine and single-pressure HRSG. A preheating loop is installed in the HRSG to supply steam for deaeration (6). This means that the existing heaters are not needed, and the steam turbine extractions must be blocked off. An additional feedwater pump (15) must be installed for the preheating loop.

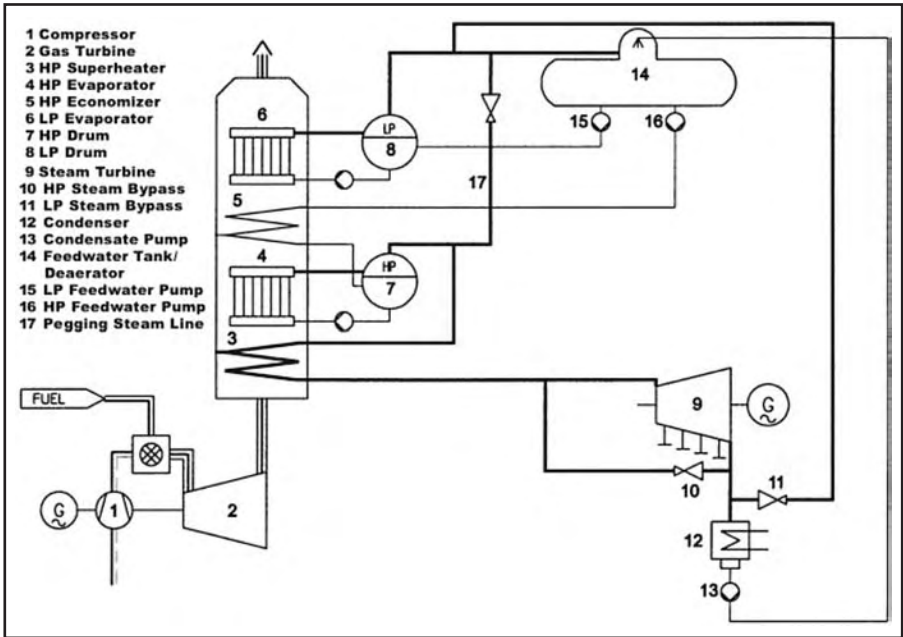


Figure 6–13 Flow diagram of a combined cycle plant using an existing steam turbine

For natural gas fuel with low sulfur content, a dual-pressure cycle would have been chosen leading to a better utilization of the exhaust gas.

If a concept using several pressure levels or even reheat is used, the question arises, where to feed the steam in the existing steam turbine. LP steam can often be admitted into the crossover pipe between the HP and LP parts of the turbine. If the steam turbine is too small, it may not have an internal crossover pipe. In such a case, one of the existing feed heating extractions could be adapted for use as an admission point.

When repowering, a decision must be made about which equipment will be retained from the existing steam plant. This will vary from case to case and depends on technical, economical, and reliability criteria. Typically, the following systems will be reused:

- Building and foundations
- Steam turbine and generator
- Condenser (maybe with new tubes)
- Main cooling system
- Main transformer for the steam turbine
- The high-voltage equipment

These larger components would be expensive to replace. It may, however, be preferable to replace most of the smaller components of the steam cycle because this can be done at relatively low cost. Retaining them may create operational problems and have a negative effect on the availability of the repowered installation. These items include:

- Condensate pumps
- Feedwater pumps
- Control equipment
- Piping and fittings
- Valves

To achieve efficiencies close to those of green field combined cycles it is important to have a good fit between the size of the gas turbine(s) and

the steam turbine and that the steam turbine has live-steam parameters close to the optimum values of a new plant. If the steam turbine is too large and the gas turbine is too small, the HRSG will not be able to produce enough steam to build up sufficient steam pressure in the steam turbine to achieve good thermodynamic cycle data.

In case the steam turbine has lower live-steam data than required for an optimum cycle, the efficiency will also suffer.

Gas turbine with a conventional boiler

Modern steam power plants with reheat steam turbines can be repowered using a concept called the *fully fired combined-cycle plant* to improve the efficiency.

Conventional boilers use fresh-air fans to supply the air for combustion. Combustion air can also be provided by the exhaust gas of a gas turbine, if it is installed near the existing steam generator. The boiler must, however, be adapted to this new operating mode. Because the temperature of gas turbine exhaust is much higher than air after an air preheater—typically up to 550 to 650°C, (1022 to 1202°F) versus 300 to 350°C (572 to 662°F)—modifications of the burners, and maybe of some heat transfer sections of the boiler, are required. The high temperature of GT exhaust gas requires a quite sophisticated ducting system between the GT and the boiler.

Figure 6–14 is the flow diagram for such a cycle. The gas turbine exhaust gas flows first through the conventional boiler (4) where the remaining oxygen is used for the combustion and then through a waste-heat recovery system (5 and 8) used for most of the feedwater preheating. The rest of the preheating is done using the existing steam turbine preheaters (8 and 9). The fresh air fan (14) is retained for use in case the gas turbine is out of operation. The bypass stack in the gas turbine exhaust (3) provides extra operating flexibility and allows the gas turbine to operate in single-cycle mode if the water-/steam-cycle or main boiler is out of operation.

Such repowering is highly sophisticated and costly. The economical value of this solution is often questionable.

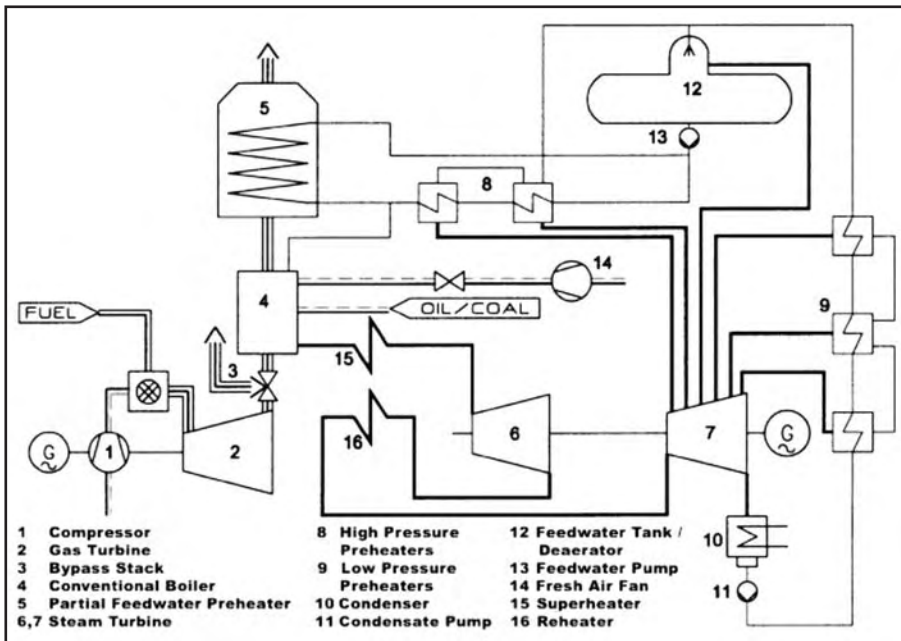


Figure 6–14 Flow diagram of a gas turbine combined with a conventional steam cycle (fully fired combined-cycle plant)

Obviously a key factor when considering such a conversion is the availability of space close to the boiler so that the gas turbine and the heat recovery system can be installed at the end of the boiler.

For steam power plants that burn gas or oil, the efficiency can be raised thanks to this conversion by more than 10% (relative) and power output by 20 to 30%. With coal-burning units, there is less potential gain because the conversion itself is even more complex and there is less improvement in efficiency.

Parallel-fired combined-cycle plant

Figure 6–15 shows the parallel-fired combined-cycle plant, a repowered cycle designed to have a very good efficiency combined with high operating flexibility.

A special aspect of this concept is that it is primarily intended for conventional coal fired plants where the conventional boiler is to remain

unmodified in operation. The gas turbine (1) and HRSG (2 and 3) are installed in parallel to the conventional boiler to provide a second source of HP live steam for the steam turbine. Also here, the exhaust heat at the cold end of the HRSG is used for feedwater preheating. The HP section of the HRSG is designed to generate HP steam to be used in the steam turbine in parallel to the steam provided by the conventional boiler. The existing steam and feedwater heaters are reused but only partially fed with water, reducing the steam flow extracted from the turbine and increasing the electrical output. The remaining feedwater is heated up in the HRSG.

In Figure 6–15, total live-steam flow from the HRSG and conventional boiler expands in the steam turbine (5) and the total amount of cold reheat steam is fed to the reheat section of the conventional boiler where it is reheated before expanding through the rest of the steam turbine. The reheat temperature should be kept at the original level to maintain maximum efficiency. If the additional flow from the HRSG is relatively high compared to that of the conventional plant, measures must be taken to maintain the reheat temperature (e.g., exhaust gas recirculation, tilting burners, etc.).

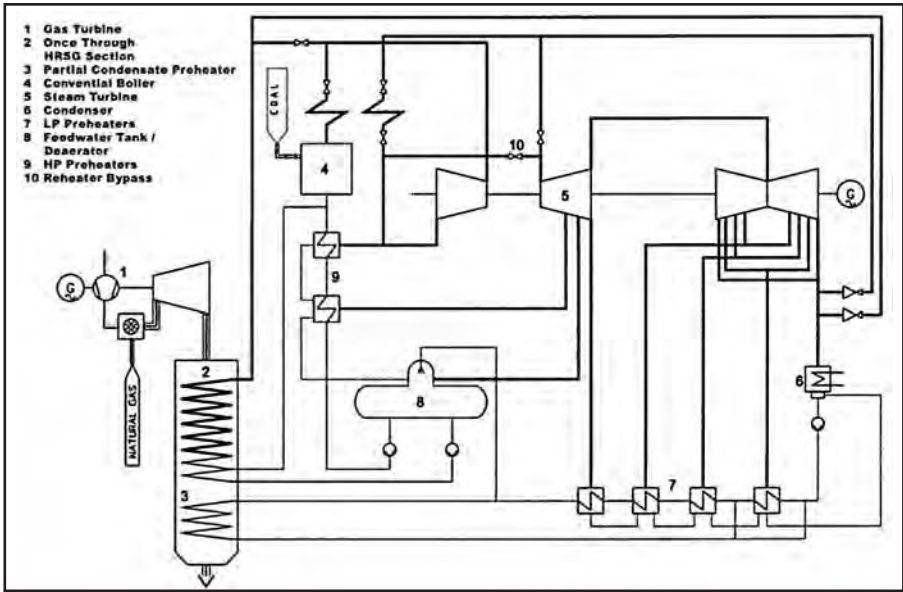


Figure 6–15 Flow diagram of a parallel-fired combined-cycle plant

There are three main operating modes for this concept:

- Original mode without the gas turbine and HRSG in operation
- Parallel mode, where the conventional cycle, gas turbine, and HRSG are in operation at the same time
- Pure combined-cycle mode, where the gas turbine, HRSG, and steam turbine are in operation without the conventional boiler. If the original cycle has reheat, a reheat bypass is necessary for this mode of operation.

Table 6–1 shows some performance data for a converted 500 MW conventional steam turbine power plant with the 178 MW gas turbine.

Table 6–1 Conversion of a 500 MW steam turbine power plant into a parallel-fired combined cycle plant

		Original mode	Parallel mode	Combined Cycle mode
Fuel Input (LHV)	MW	1,312.9	1,496.6	473.0
Steam Turbine Output	MW	502.3	495.4	87.0
Gas Turbine Output	MW	0	178	178
Total Gross Output	MW	502.3	673.4	265.0
Gross Efficiency (LHV)	%	38.26	45.0	56.0
Auxiliary Consumption	MW	32.7	30.5	4.9
Total Net Output	MW	469.6	642.9	260.1
Net Efficiency (LHV)	%	35.77	42.96	55.00
Net Heat Rate (LHV)	kJ/kWh	10,065	8,380	6,545
Net Heat Rate (LHV)	Btu/kWh	9,540	7,943	6,204
Net Marginal Efficiency of Gas (LHV)	%	—	58.90	55.00

Highest overall efficiencies are obtained with pure combined-cycle mode and the highest output with hybrid mode. However, in parallel mode the net marginal gas efficiency (the contribution of the gas-fired in the gas turbine to the total efficiency) is higher than in combined-cycle mode. This means that plant fired in parallel is a very efficient way of burning natural gas, making this an interesting alternative to separate plants. Combinations of the previously mentioned modes are also possible resulting in high operating flexibility and a wide operating regime. The conventional boiler can be fired with coal, oil, or gas.

The choice of operating mode at any moment will depend on the market prices for electricity and fuels. As the relative prices of gas and coal and the demand for electrical output vary, the operator can decide which mode can be run most economically at any one time.

It must, however, also be mentioned here that this solution is complicated and will not gain a large market acceptance.

Conversion of an Existing Gas Turbine Power Plant into Combined Cycle

The conversion of an existing simple-cycle gas turbine plant into combined cycle is quite easy compared to the conversion of a steam power plant. Downstream of existing gas turbines a steam cycle can be added, having the same characteristics as the steam cycle of an optimized new combined-cycle plant. The major issue is the availability of space close to the gas turbines to install the HRSG without long ducts for interconnection. The other factors to determine whether such a conversion makes economical sense are age and status of the gas turbine.

If the gas turbines are too old or in bad condition it does not make much sense to install a complete new steam part, which has much higher specific cost than the gas turbines (approximately twice the cost of a gas turbine plant).

In addition with older gas turbines, the resulting combined-cycle plants will only have a low efficiency compared to a completely new plant.

Special Applications

There are several variations in the way a gas cycle can be combined with a steam cycle, which can largely deviate from the type of combined cycles discussed so far. Some of these cycles have proven to be of some interest; others may gain some acceptance in the future; and some will never be realized. As yet, few of these have achieved any commercial status because they are either complex or costly or use components, which are not commercially available. Some of the most important of these applications will be discussed here.

Applications with alternative fuels

Where no natural gas is available, there is increasing interest in looking for ways to use available alternative fuels such as coal, heavy oil, and crude oils. Often these fuels have high sulfur content or are unsuitable for firing in gas turbines, so they must be treated before they can be used. Generally steam or heating is needed in the fuel treatment process, which can be provided by the combined-cycle plant and results in some very interesting but complex integrated processes. One of these processes has gained some level of commercial acceptance and is described in chapter 12.

Pressurized fluidized-bed combustion (PFBC). A typical cycle of this type is shown in figure 6–16. The gas turbine has an intercooler (4) between a low-pressure (1) and a high-pressure (2) compressor. At the outlet of the compressor, air is fed to the combustion vessel (5) with a pressure of 12 to 16 bar (160 to 218 psig) and forced through the fluidized bed where the combustion of the coal takes place. Coal is mixed with sorbent before entering the combustion process to bind chemically the sulfur contents in the coal. To obtain a high degree of desulfurization, the bed is maintained at a temperature of 850°C (1560°F). Before leaving the combustion vessel, the exhaust gas is led through cyclones (8) to clean it before it is expanded in the gas turbine (3). The exhaust heat of the gas turbine is recovered in an economizer (10), which is used for the final stages of feedwater preheating (17). The steam turbine is a single-pressure turbine with reheat (11).

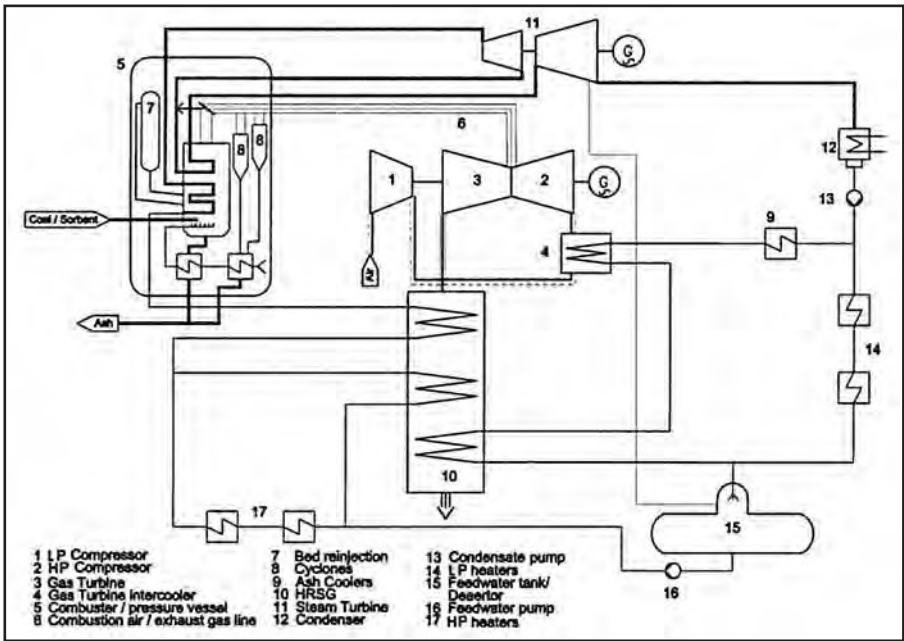


Figure 6–16 Flow diagram of a PFBC process

Net efficiencies are clearly above those of conventional power plants burning coal, although well below the level of pure combined-cycle plants because of the relatively low gas turbine inlet temperature (i.e., 850°C [1562°F]).

Another advantage of this cycle is that the fluidized bed with its steam generator is very compact. It takes much less space than a conventional boiler, although this is partly counteracted by the large economizer and the gas turbine. The small storage capacity of the steam generator means that the steam process reacts very quickly to changes in demand from the load control system.

However, the design is complex and requires a specially designed gas turbine very different from today’s standardized industrial gas turbines. This means that the investment cost are very high compared to the normal gas-fired combined-cycle plant. Consequently, there is little chance that this system will ever gain market acceptance. The only major supplier having promoted this technology is ABB, who has left this business altogether.

Steam injection into the gas turbine (STIG) cycles

We have already seen how injection of steam into the gas turbine can be used to reduce NO_x emissions. This decreases the efficiency of the combined cycle, but increases the power output significantly. Cycles in which steam is injected into the gas turbine are known as steam-injected gas turbine (STIG) cycles.

Figure 6–17 shows a STIG cycle in which all the generated steam is directed into the gas turbine (4), apart from a small amount used for feedwater preheating and deaeration. Because steam is required only in the gas turbine at one pressure level, a single-pressure HRSG is used (2). The quantity of makeup water to the cycle is very high because all of the generated steam is lost to the atmosphere in the process. This type of cycle is suitable for use as a peaking unit in countries where water is plentiful. It is simple and cheap, attaining a high specific output and a higher efficiency than the gas turbine alone, although not as high as that of a combined cycle. However, if the injected steam flow is equal to more than approximately 2% to 4% of the air-mass flow, major modifications must be made to the gas turbine. This limits the commercial viability of the concept because gas

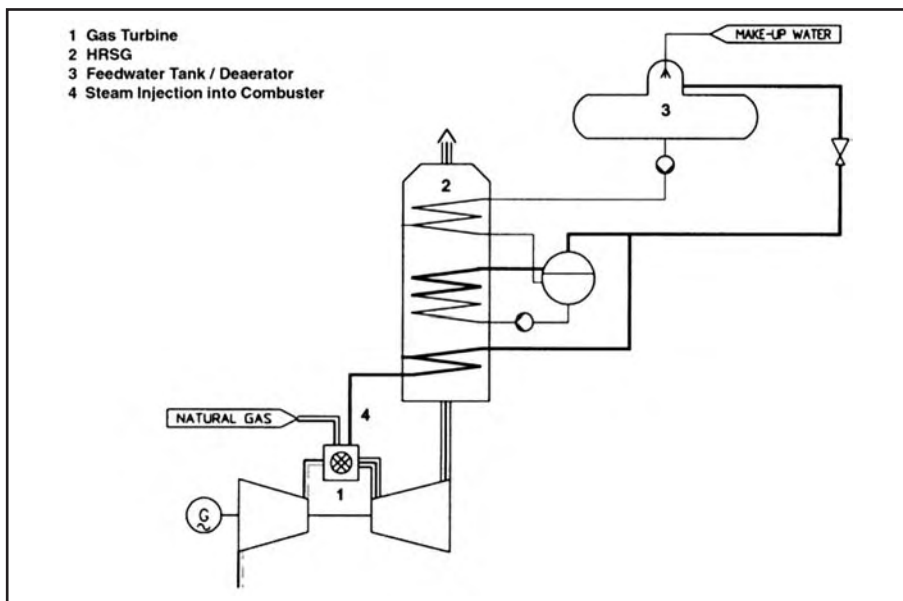


Figure 6–17 Flow diagram of a STIG cycle

turbines are standardized and are only modified for special applications when a substantial economic benefit can be expected.

Another, more efficient variation on this type of cycle is the Turbo STIG cycle, shown in figure 6–18. It is suitable for smaller plants with aero-derivative gas turbines. A steam turbine (4) is installed on the same shaft as the gas turbine (1) and shares the gas turbine generator. The single-pressure HRSG (2) generates steam for the steam turbine, expanding it to a level suitable for reheating in the HRSG and injecting (6) into the gas turbine. Thanks to the steam turbine, the power output and efficiency of this cycle is higher than the STIG cycle, but still does not reach the level of a normal combined cycle. This, and the fact that the water consumption is so high, is part of the reason for the limited commercial acceptance of the STIG cycles or Turbo STIG.

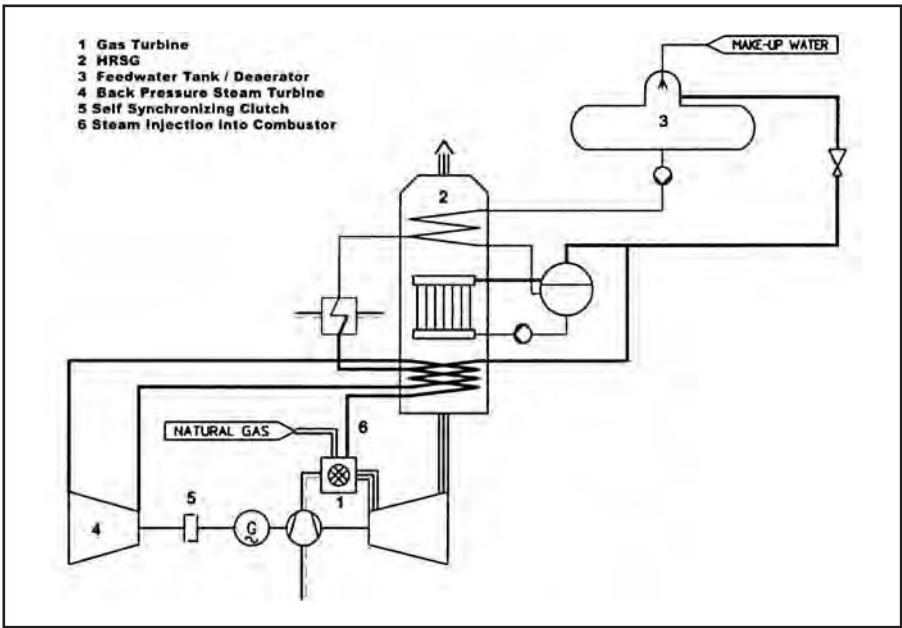


Figure 6–18 Flow diagram of a turbo STIG cycle

Humid air turbine (HAT) cycle

The principle here is to increase gas turbine output by using humid air as the working fluid. A flow diagram is shown in figure 6–19. Heat from compressor cooling (3 and 4) and the gas turbine exhaust is used to generate steam, which is being mixed with the compressed air. The result is an airflow with up to 25% steam content. This air is then heated, again using gas turbine exhaust heat, before entering the gas turbine.

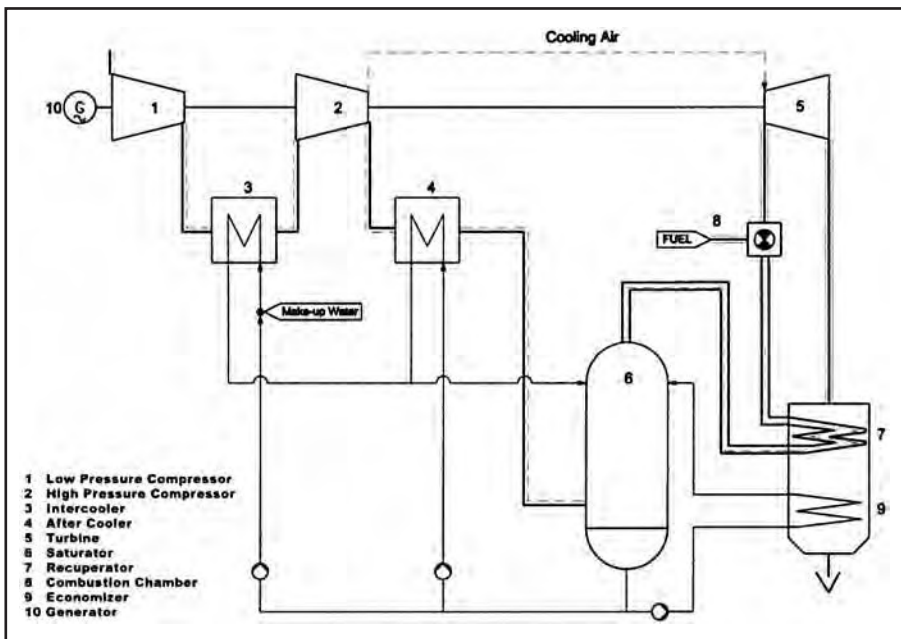


Figure 6–19 Flow diagram of a HAT cycle

Efficiencies reached with this cycle are not as high as those of normal combined cycles, but the HAT cycle can reach higher specific output levels. The cycle is also suited for use with coal gasification plants where available low-grade energy could be used to generate the required steam.

Difficulties in the HAT cycle lie in the operational complexity and the high water consumption that would cause problems where water is scarce. Also, new gas turbine development is needed due to the changed flow pattern in both the compressor and turbine of the gas turbine. This will probably be the main obstacle in commercializing this cycle.

Cycles with alternative working media

Another way of improving the performance of a combined-cycle plant is using fluids other than pure water/steam in the bottoming cycle. The idea is that by using a mixture of fluids, such as water mixed with ammonia, the evaporation in the HRSG will no longer take place at only one temperature but over a range of temperatures. This serves to reduce the exergy loss between exhaust gas and working fluid, thereby increasing the efficiency.

The cycles operate at only one pressure and are theoretically more suitable for gas turbines with relatively low exhaust temperatures, such as aero derivative machines. Condensation takes place at more than one temperature, which complicates the condensing system. Another disadvantage is that this cycle relates to the toxicity or aggressiveness of the fluids used. Leakages could lead to environmental pollution or health problems.

As gas turbines develop towards higher inlet and exhaust gas temperatures, the advantages of these alternative working media for combined-cycle plants will vanish.

7 Components

Gas Turbine

The gas turbine is the key component of a combined-cycle plant, generating approximately two-thirds of the total output.

The gas turbine process is simple: Ambient air is filtered, compressed to a pressure of 14 to 30 bar (190 to 420 psig), and used to burn the fuel, producing a hot gas with a temperature generally higher than 1,000°C (1832°F). This gas is expanded in a turbine that drives the compressor and generator. The expanded hot gas leaves the turbine at ambient pressure and at a temperature between 450 to 650°C (842 to 1202°F) depending on the gas turbine efficiency, pressure ratio, and turbine inlet temperature.

The combined-cycle plant has become a competitive thermal power plant only as a result of the rapid development in gas turbine engineering, which is still ongoing. Gas turbine development generally involves increasing gas turbine inlet temperatures (e.g., by improving cooling technologies) and increasing compressor air flows.

Increasing gas turbine inlet temperatures produce a higher useful enthalpy drop and, therefore, increase the efficiency and output of the gas turbine and the combined-cycle plant. This can usually be achieved using the same compressor, and even though an additional investment for materials may be necessary, the specific cost of the plant is reduced. Because fuel costs and capital costs are the main drivers for the cost of electricity generation, the gas turbine inlet temperature

should be increased to the full potential of the material to improve the competitiveness of the product.

Parallel to the development of the turbine, there has also been some development in the compressor as well as substantial development in combustion systems. Today's compressors can handle much larger air mass flows and higher pressure ratios, resulting in higher power outputs, reduced specific costs, and improved efficiency.

Combustion technology plays a leading role in modern gas turbine technology. Rapidly growing demand for increased turbine inlet temperatures and tightened emission regulations represent a great challenge. Conflicting demands must be brought into harmony. The combustion system has to produce a turbine inlet temperature as high as possible for performance reasons, but maintain a flame temperature as low as possible due to emissions limits (NO_x production increases rapidly with flame temperature).

Stoichiometric combustion in diffusion burners results in high flame temperatures; therefore, the use of premix burners for lean combustion with significantly lower combustion temperatures compared to diffusion burners is common practice. Various design configurations exist for premix burners, but all are based on the targets of producing a uniform fuel/air mixture, operation with reliable flame stability (safeguarding against extinguishing and flame flashback), achieving uniform temperature distribution at turbine inlet, and, last but not least, ensuring complete combustion of fuel.

Conflicting requirements have to be overcome; these include aspects such as maximum air for the combustion process but also sufficient cooling air for hot gas path parts as gas temperatures increase. This results in smaller combustor designs with reduced surface area that requires less cooling air. One positive aspect that results from smaller combustors is the decrease in nitrogen oxides formation. NO_x formation directly depends on the residence time of hot flue gases in the combustor. On the other hand, complete burnout of fuel must still be ensured in these small combustors.

Such highly developed compact and highly stressed premix combustion systems tend to combustion-driven oscillations. Pressure fluctuations in the combustor driven by a thermo-acoustic resonance system causes pressure fluctuations at the burner outlet, which result in oscillating

premixed fuel/air mass flow and additionally fluctuations in composition (fuel to air ratio) caused by differences in the acoustic properties of the air and fuel system. The resulting heat release fluctuations can create a self-induced feedback loop. Excessively increasing amplitudes in combination with the harmonic frequencies of the combustion system can cause combustion disturbances (flame flashback or extinction), mechanical influences, and pronounced casing vibrations resulting in wearing and fretting damage, and so on. In case high combustion dynamics occur during operation, it is good practice to use a protection system, which takes action on engine parameters to decrease the dynamics.

Additionally, competitiveness is increased with new gas turbine concepts, such as sequential combustion that appeared on the market in the 1940s and was re-introduced in the 1990s. The gas turbine has two combustion stages, with an intermediate turbine section, and a higher exhaust temperature. This means higher combined-cycle efficiencies can be achieved without raising the firing temperatures.

Figure 7–1 depicts the historical development of maximum air flows and gas turbine inlet temperatures.

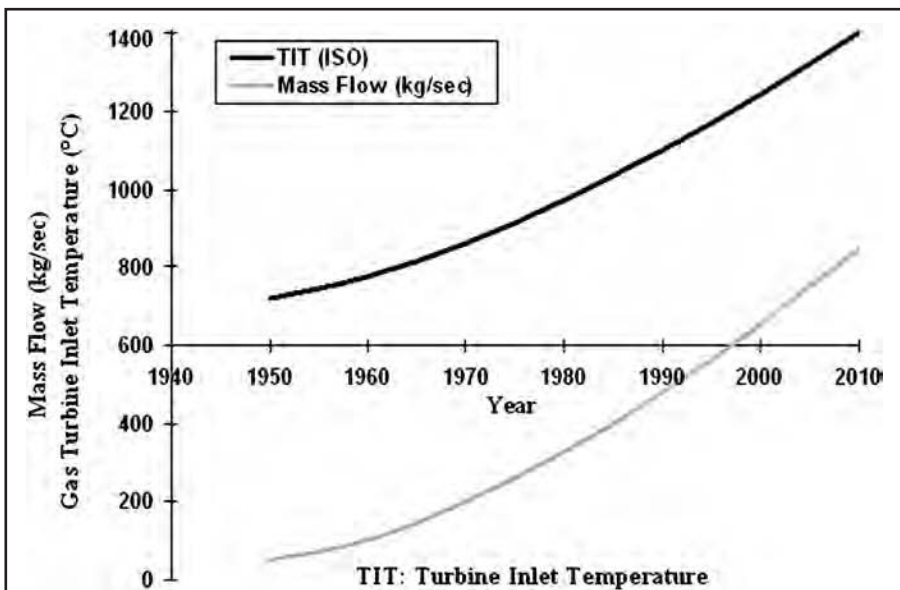


Figure 7–1 Historical trends in gas turbine inlet temperatures and compressor air flows

Two categories of turbines

Power generation gas turbines can be classified into two categories:

- The *aeroderivative gas turbine*, comprising a jet turbine modified for industrial duty and frequently incorporating a separate power turbine
- Heavy-duty industrial gas turbines, originally derived from steam turbine or jet technology

The *aeroderivative gas turbine* is normally a two- or three-shaft turbine with a variable-speed compressor and drive turbine. This is an advantage for part-load efficiency because airflow is reduced at lower speeds.

Inlet temperatures in jet engines are higher than in industrial gas turbines. In a jet turbine, weight plays the dominant role, and thus product and maintenance costs are less important than with industrial gas turbines, where long intervals between inspections are demanded. For that reason, and because of the smaller dimensions of the hot gas path, the trend toward higher inlet temperatures and greater power densities has progressed more rapidly in jet turbines than in stationary machines.

Aeroderivatives generally offer higher efficiencies than their frame counterparts as a result of aero technology. Furthermore, they are smaller and lighter for a given power output, and can be started rapidly because of their inherent low inertia. Because these turbines are derived from jet engines, they retain many of the features designed to allow rapid “on the wing” maintenance of aero engines. The small physical size of the aeroderivative enables it to be removed from its nacelle and replaced within a day.

A disadvantage when the aeroderivative is operating with a generator is that there is no compressor braking the power turbine in the event of load shedding. Two-shaft turbines are usually used for compressor or pump drives, where the operating speed of the power turbine is also variable. Due to the maximum size of aircraft, aeroderivatives are limited to approximately 50 MW electrical output.

Heavy-duty industrial gas turbines are practically always designed as single-shaft machines when used to drive generators with outputs

greater than approximately 30 MW. Due to the increased turbine inlet temperatures and compressor air-mass flows, today's gas turbines for power generation can achieve electrical outputs of up to 340 MW. Figures 7-2 and 7-3 depict typical modern gas turbines. Figure 7-2 shows an aeroderivative unit, and figure 7-3 a heavy-duty industrial gas turbine designed for 340 MW output.

As gas turbines have become standardized, they have begun to be manufactured on the basis of sales forecasts rather than orders received. This results in a series of frame sizes for each gas turbine manufacturer, and, hence, shorter installation times, lower costs, and lower prices. Major developments in heavy-duty industrial gas turbines have been achieved in the last decade. A major impact has been made by the introduction to the market of gas turbines with sequential combustion.

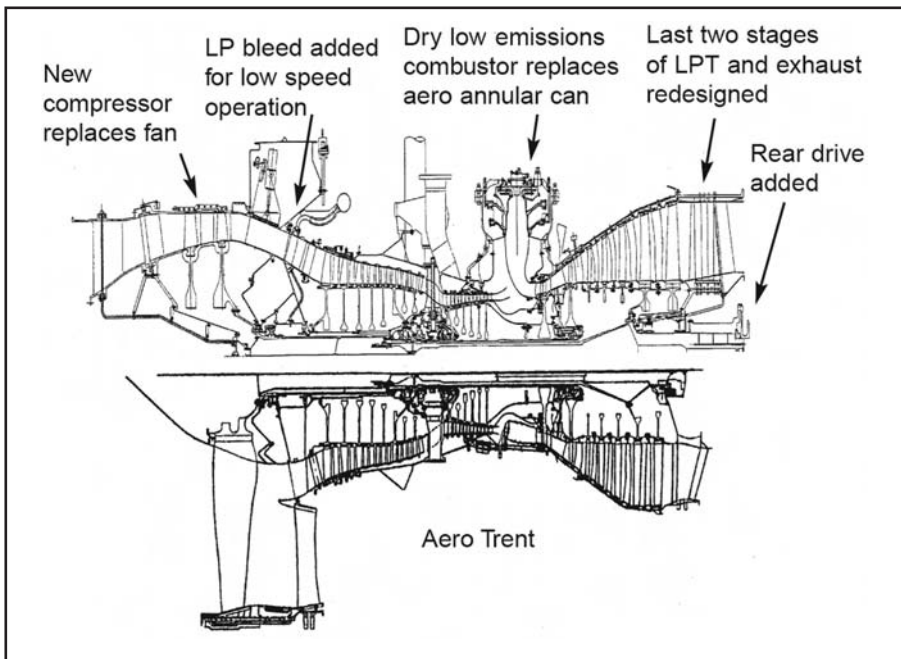


Figure 7-2 Industrial Trent derived from the aero Trent 800.

Source: Rolls-Royce plc. Registered trademark Rolls-Royce plc.

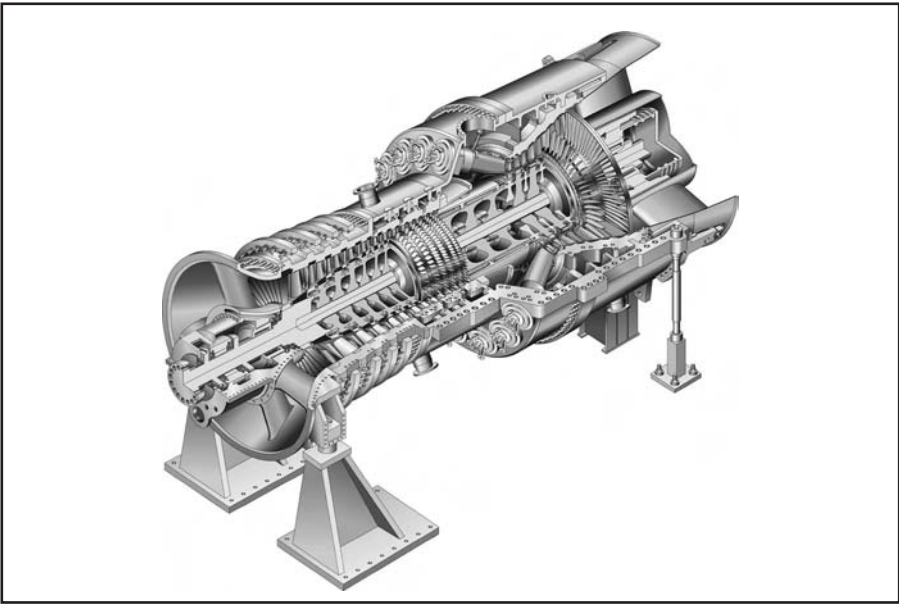


Figure 7–3 Heavy-duty industrial gas turbine

Gas turbines with sequential combustion

In a gas turbine with sequential combustion, compressed air enters the first combustion chamber located downstream of the compressor outlet. Here, fuel is combusted, raising the gas temperature to the inlet temperature for the first turbine. The hot gas expands as it passes through this turbine stage, generating power before entering the second combustion chamber, where additional fuel is combusted to reach the inlet temperature for the second turbine section, where the hot gas is expanded to atmospheric pressure.

With the same gas turbine inlet temperatures as in a gas turbine with a single combustion stage, a higher efficiency can be achieved with the same emissions levels.

Figure 7–4 shows a gas turbine with sequential combustion. The compressor is designed for a pressure ratio of 30, followed by the high-pressure combustion chamber, the high-pressure turbine, low-pressure combustion chamber, and low-pressure turbine. A compact arrangement fits all this equipment into one casing on a single-shaft.

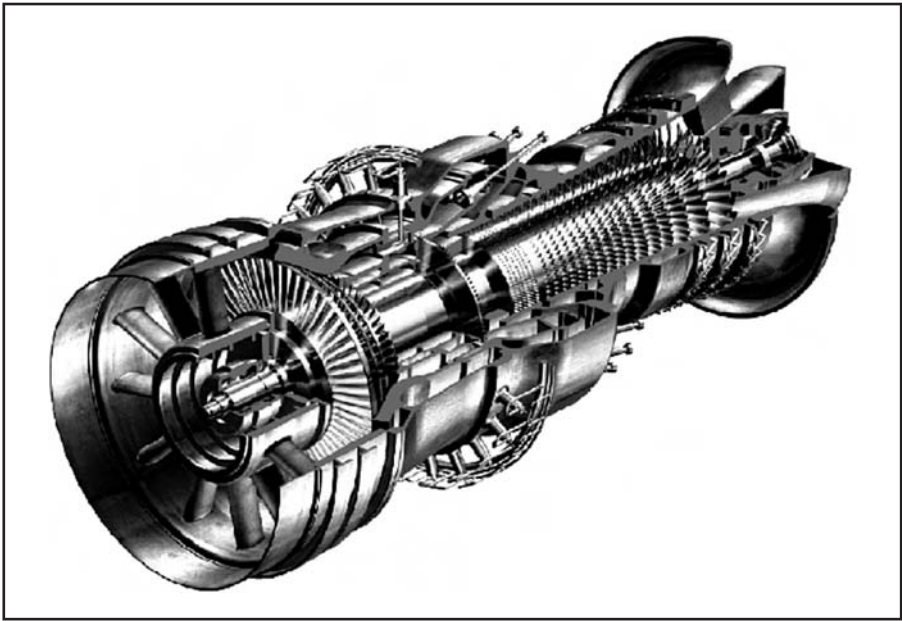


Figure 7–4 Gas turbine with sequential combustion

Table 7–1 lists the main characteristic data of modern gas turbines designed for power generation.

Table 7–1 Main characteristic data of modern gas turbines for power generation	
Power Output (ISO condition)	up to 340 MW
Efficiency (ISO condition)	34–40%
Gas turbine inlet temperature (ISO 2314)	1,100–1,350°C (2,012–2,462°F)
Exhaust gas temperature	450–650°C (842–1,202°F)
Exhaust gas flow	50–820 kg/s (397,000–6,500,000lb/hr)

In the past, gas turbines were mainly developed for use in simple-cycle operation and the majority of gas turbine installations were of this type. These gas turbines were often used as peaking and stand-by machines. The first combined-cycle power plants were based on the same gas turbines, but operated in intermediate and base load applications.

The main application for the gas turbine is in a combined-cycle configuration; this has had a corresponding impact on gas turbine development and equipment. The main market breakthrough of the gas turbine in combined-cycle operation took place in the mid-1980s and the technology is now mature and widely applied. Improving gas turbine performance lowers the cost of electricity, increases the competitiveness of the product, and thus motivates manufacturers to invest in new designs and upgrades.

Because new models of given gas turbines are often sold before the first unit running has accumulated many operating hours, attention must be paid to corresponding risk mitigation measures.

Inspections

Gas turbines achieve a very high level of reliability through regular and properly administered inspection and maintenance activities. In a combined-cycle plant, the inspections for the other equipment can usually be completed in the window defined by the gas turbine inspection schedule.

Degradation

The output of a gas turbine is subject to degradation that has two main causes:

- *Fouling*, which is recoverable. During operation, attention must be paid to compressor and turbine fouling.
- *Aging*, which is non-recoverable unless parts are replaced.

Compressor fouling occurs because the gas turbine operates in an open cycle. The compressor ingests air that cannot be completely cleaned. Compressor fouling is reduced by an air filtration system that is suited to the environmental conditions at the plant site. The filters most frequently used are two-stage filters or self-cleaning pulse filters. The latter of these is only suitable for use in dry climates.

For economical operation of a power plant with gas turbines, proper design and size of the air filter system, together with well balanced maintenance intervals for compressor cleaning (on-line and off-line) and

filter replacement, are essential. Pressure drops in the air intake system result in performance losses. One mbar intake pressure drop causes:

- Approximately 0.1% output loss
- Approximately 0.03% (approx. 0.013% point) efficiency loss

The design and size of the filter structure dictate the basic pressure drop of the intake system. The design pressure drop of a new and clean filter bank is normally on the order of 2 mbar. The approach velocity of air upstream of the filter surface is approximately 3 m/s.

It is, however, impossible to keep the compressor completely clean. The fouling that results causes losses in output and efficiency.

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

- On-line washing
- Off-line washing

On-line washing is straightforward but because the temperature increases through the compressor, the cleaning solution evaporates and cleaning is limited to the first compressor rows.

Off-line washing is better suited to large, modern gas turbines because it is more effective. However, it requires shutting down and cooling the engine. Washing at low speed is preferred (e.g., ignition speed with the machine cold). The machine is, therefore, out of operation for approximately 24 hours, chiefly for cooling down prior to and for drying the engine after washing. This type of washing is best done before or after an inspection. The amount of cleaning solution needed is on the order of 40 to 200 liters per washing cycle.

Keeping degradation as low as possible requires an optimized maintenance program that includes a well-balanced combination of following measures:

- On-line compressor cleaning
- Off-line compressor cleaning
- Air intake filter replacement (partial and complete replacement dependent on distribution of dirt deposits)

A typical recommendation for replacement of air filter elements is a total pressure drop of 7 to 8 mbar over all filter elements.

After compressor cleaning in a gas-fired machine, any remaining degradation is due to aging. The symptoms of aging after 8000 hours in operation on a clean fuel are a:

- Reduction in combined-cycle plant output of about 0.8 to 1.5%
- Reduction in combined-cycle plant efficiency of about 0.5 to 0.8%

The rate of aging decreases with time between major overhauls.

Turbine fouling is caused principally by the ash formed by combustion of heavy fuels and in the additives used to inhibit high-temperature corrosion. It is unavoidable with heavy fuels, but can be limited by selecting the proper type of additives. Heavy- and crude-oil-fired gas turbines can also be equipped with a turbine washing system. Turbine fouling is less of a problem in peaking or intermediate-load operation because of the self-cleaning effect of startup and shutdown.

Typical degradation (after compressor washing) in combined-cycle plants after 8,000 hours of operation on heavy or crude oil are a:

- Reduction in combined-cycle plant output of 4 to 5.5%
- Reduction in combined-cycle plant efficiency of 1.5 to 1.9%

The rate of degradation also decreases with time. Figure 7–5 shows the correlation between degradation and maintenance measures with a simplified graph.

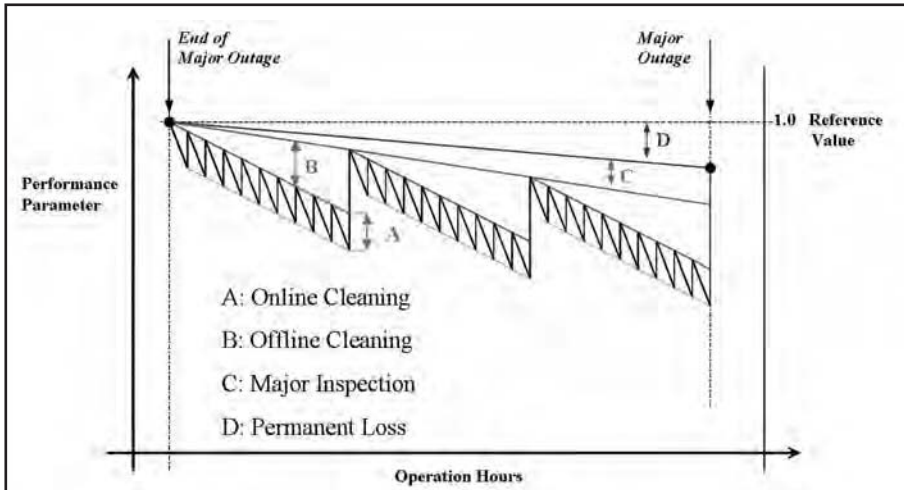


Figure 7-5 Types of losses contributing to overall performance degradation
(source: ASME Paper 2003-GT-38184)

In the past, corrosion problems were one of the major causes of gas turbine failures. Because of the use of better blading materials and coatings, problems of this type have practically been eliminated. Whenever heavy fuels are used, particularly those containing vanadium or sodium, it is necessary to use additives or treat the fuel to prevent high-temperature corrosion. The additives commonly used are based on magnesium, chromium, or silicon.

Fuels for gas turbines

More than three-quarters of installed gas turbine capacity use natural gas and light distillate fuel whereas fuel oil is most often used only as a backup fuel. These clean fuels enable the highest performance with state-of-the-art gas turbines, allow clean combustion with the lowest emissions levels, are easy to handle, are compatible with approved standard combustors and auxiliary equipment, provide the simplest mode of operation, are synonymous with high plant availability, and require the lowest maintenance effort.

The remaining quarter uses unconventional fuels, such as special fuels like low BTU natural gas, off-gases from iron making processes, synthesis

gases (syngases) derived from various source fuels, but also heavy fuel oil, residual oil, crude oil, naphtha or gas condensate fuels, and so on. Combusting these fuels in gas turbines requires special efforts for the burner design, for the gas turbine compressor/turbine design, for fuel storage and supply systems, for fuel treatment and/or conditioning, for safety aspects and for emissions control. Table 7–2 gives an overview of critical fuel properties and their consequences for gas turbines.

Table 7–2 Critical fuel properties

Critical Fuel Properties	Fuels	Effect
Low Viscosity (reduced lubricity)	Naphtha, Kerosene, Condensates	Effect on fuel supply system (e.g. pump design)
High Viscosity	Heavy Oils Heavy Residues	Effect on fuel supply system (e.g. pump design, trace heating) Requires start-up, shut down and flushing systems Effect on fuel atomization (fuel heating)
Low density (high volume flow)	Naphtha, Condensates	Limits for fuel supply system and burner nozzles
Low flash point Low boiling point (high vapour pressure)	Naphtha, Kerosene, Liquified Petroleum Gases (LPG) High Speed Diesel (HSD) Condensates	Increased explosion protection effort (inkl. start-up and shut-down fuel system) Increased ventilation effort
Low ignition point (auto ignition)	Naphtha, Condensates	Effect on premix capability Increased explosion protection effort Increased ventilation effort
Contaminants (high temperature corrosion, ash deposits)	Contaminated fuel oils, Crude oils, Heavy Oils Heavy residues	Effect on blading and hot gas path. Counter-measures: - Temperature reduction (reduced performance) - Fuel treatment (washing/inhibitor dosing)
Low heating value (high volume flow)	Process and synthesis gases (low calorific gases), Low BTU natural gas	Effect on layout of compressor, burner nozzles, fuel supply system
High heating value (low volume flow)	LPG, gaseous and liquid	Effect on layout of burners, fuel supply system Limited start-up and part-load capability
High H ₂ content	Process and synthesis gases (low calorific gases)	High flame velocity, Effect on premix capability more explosive than natural gas
High dew point	Gases with high boiling components	Droplets causes erosion and non constant heat flow

Gaseous fuels

For comparison of gaseous fuels the heating value is not the only criterion. The Wobbe index (or Wobbe number) is used for classification of fuels. The Wobbe index is a key parameter for the heat capacity of a gaseous fuel or, in other words, it is an indicator for the interchangeability of gases. Gases with same Wobbe index produce an identical burner heat load at the same burner pressure and can therefore be used in the same burner.

The Wobbe index for lower heating value is defined as:

$$W_i = \frac{Q}{\sqrt{\frac{\rho}{\rho_d}}} \left[\text{J/m}^3, \text{kWh/m}^3 \text{ or } \text{Btu/ft}^3 \right] \quad (7.1)$$

where W_i = Wobbe index, Q = heat input based on lower heating value, ρ = density of fuel gas at standard conditions, ρ_d = density of air at standard conditions.

Gases with same Wobbe Index or within a range of ± 2 to $\pm 5\%$ for premix burners ($\pm 15\%$ for diffusion burners) can be used in the same burner. The Wobbe index of a gaseous fuel can be adjusted by diluting it with inert or lean gases (e.g., steam, nitrogen) or improved by adding rich gases (e.g., evaporated LNG).

Some fuels are usually known as liquids, but will normally be vaporized for combustion in gas turbine combustors. Propane, butane, and liquefied petroleum gas (LPG) are typical fuels of this kind.

Natural gas in pipeline quality is the typical standard fuel suitable for use in standard fuel supply systems and combustors. Critical properties such as Wobbe index, heating value, composition, and the compounds with usually restricted concentrations such as hydrogen, higher hydrocarbons ($>C_3$), hydrogen sulfide, and mercaptans are set and maintained within defined and acceptable limits.

Depending on pipeline pressure and the pressure ratio of the gas turbine, two different cases are possible:

- The pipeline pressure is too low and compression is necessary.
In this case it is of essential importance to avoid any carryover

of liquids (e.g., lubricants) from compressor equipment into the fuel gas stream.

- The pipeline pressure is too high and pressure reduction is necessary. The resulting temperature drop involves a risk of condensation of gaseous constituents. Separation of liquid phases and heating of the gas may be necessary.

In both of these cases, the control system for regulating the pressure of fuel gas supplied to the gas turbine system requires particular attention. Rapid flow changes are frequent, especially during gas turbine startup and shutdown, or, as an extreme case, on gas turbine trip. Typical gas pressure controllers are not able to handle such rapid changes with a fast response. Gas turbine controllers require a defined, constant fuel gas pressure upstream of the load control valve. Pressure fluctuations result in load fluctuations. To moderate such pressure fluctuations especially in multi-unit configurations, a certain buffer volume in the fuel gas supply system between primary pressure controller and gas turbine load controller should be provided.

The fuel gas temperature upstream of burners has to be kept in a certain safety margin above the dew points of water and H_2S to avoid formation of droplets in the fuel gas stream and ingress into the burners as well as to prevent corrosion.

Liquefied natural gas (LNG) has to be vaporized to a gas. Produced by a liquefaction process, LNG is a very clean and rich fuel gas, from which all higher hydrocarbons (C_6+), inert components (N_2 and CO_2) and most impurities have been removed. Pure evaporated LNG has a higher Wobbe index than pipeline natural gas. Nevertheless, variations in fuel characteristics are encountered with LNG from various sources. Sudden changes in fuel characteristic must be avoided (e.g., between different shipments). In particular the advanced premix burners demand tighter control of the variability in fuel constituents.

Low BTU natural gas contains a higher percentage of nitrogen and carbon dioxide as well as impurities such as hydrogen sulfide and mercaptans. For calorific values below an engine-specific typical minimum value (in the range of approx. 35 to 45 MJ/kg), dry low- NO_x premix operation is excluded and for even lower values, modified burners and gas turbine compressors are required. Blending with other

high-calorific value gases may be necessary to achieve levels necessary for ignition, flammability and Wobbe index as well as for composition of a constant, defined quality.

Off gases from iron and steel industries, such as top gas, blast furnace gas, and converter gas (COREX gas and coke oven gas are more similar to syngases and are listed there) range in the very low to medium BTU classes and have a high carbon monoxide content. They require careful cleaning that may involve removal of many inadmissible contaminants. Blending with high calorific gases as well as storage systems for intermittent processes (e.g., converter gas) are required to maintain a constant gas composition.

Syngases are mainly gasifier gases derived from various source fuels such as coal, biomass and refinery residuals, and so on. COREX gas and coke oven gas from iron production are included among the syngases due to their high hydrogen contents.

Typical characteristics of syngases are low calorific value, with high contents of hydrogen and carbon monoxide; therefore they require a special fuel supply system, modified burners as well as an adapted gas turbine compressor design. Furthermore, increased explosion protection measures and provisions to safeguard against toxic carbon monoxide must be implemented.

Modified burners are necessary for hydrogen rich fuels and increased mass flow to compensate for low calorific value. The higher flame temperature when firing hydrogen produces more NO_x than combustion of natural gas. Other concerns are related to the higher flame velocity of hydrogen and thus the risk of flashback in dry low- NO_x premix burners. Use of diffusion burners and NO_x control by diluting the syngas with inert gases (e.g., nitrogen) is commonplace.

Compressor modification is required by the low calorific value of these gases and the resulting higher turbine mass flows and therefore increased pressure ratio in the gas turbine.

Vaporized liquids such as propane, butane, or mixtures of the two (i.e., LPG) are low-boiling hydrocarbons that have to be vaporized for use in gas turbines. These fuels belong to the very high calorific value class. They are normally clean, but require a modified burner design and safety measures pertaining to explosion hazards.

When these fuels are used, it is essential that no liquid phase occurs in the fuel supply system that would result in disastrous load fluctuations of the gas turbine. The fuel system needs effective measures to prevent condensation of gas inside the piping system that occurs especially after shutdown and cool down of the system.

Table 7–3 lists typical compositions of fuel gases for gas turbines.

Table 7–3 Typical composition of fuel gases for gas turbines

		Natural gas Standard	Natural gas Low-BTU	Gasification coal	Gasification residue	Gasification biomass	Blastfurnace gas	Converter gas	COREX gas
H ₂	[%vol.]		0.2	12.3	27.2	11.2	2.5	1.0	16.1
CO	[%vol.]			24.8	31.8	20.2	22.8	69.2	43.0
CO ₂	[%vol.]	0.2	0.5	0.8	4.9	12.0	21.2	14.6	36.5
N ₂	[%vol.]	3.0	62.3	42.0	0.5	44.6	53.5	14.9	2.8
CH ₄	[%vol.]	93.0	36.0		0.3	5.6			1.6
Ar	[%vol.]			0.6	0.7				
H ₂ O	[%vol.]			19.1	34.6				
O ₂	[%vol.]			0.4					
CnHm	[%vol.]	3.8	1.0			0.2		0.3	
Density	[kg/m ³]	0.77	1.06	1.03	0.82	1.10	1.42	1.26	1.32
LHV	[MJ/kg]	47.0	12.8	4.3	8.6	5.3	2.3	6.7	5.9

Liquid fuels

Light distillates, according to defined standards as ASTM D2880 (Distillate No. 2-GT) or EN DIN 51603 (*Heizöl EL*), are products with fixed, reproducible properties that can be used with standard fuel supply systems, combustors, and hot gas path parts. Essential properties such as viscosity, density, ash, flashpoint, vapor pressure, admissible trace metal contamination levels (mainly sodium, potassium, calcium, vanadium, and lead), and so on are specified so that such fuels can be combusted without further measures and without any impact on normal performance levels, maintenance intervals, duration, and effort as well as the service life of parts.

Ash-forming fuels, such as heavy oils, residual oils, and crude oils, are generally contaminated with corrosive impurities (trace metals, mainly

sodium, potassium and vanadium). These fuels require treatment before combustion. This includes heating to lower the viscosity for pumping, washing, and finally for atomization by the burner nozzles. Depending on the type and amount of contaminants present, turbine inlet temperature must be lowered (loss of performance), overhaul intervals are shortened (loss of availability), and hot gas path part replacement costs increase. The burner system is normally identical to that for light distillate.

Sodium, potassium, and other water-soluble contaminants have to be washed out by forming an oil-and-water mixture, and removed by centrifuging or electrostatic separation.

Oil-soluble contaminants, mainly vanadium but also lead, must be controlled with suitable inhibitors. Additionally, the gas turbine inlet temperature has to be lowered to keep it below the melting point of compounds formed from these contaminants to prevent molten corrosive products from adhering to the surface of hot gas path parts.

Maximum acceptable limits for these contaminants need careful monitoring to prevent the hot gas path parts from high-temperature corrosion, but also burner nozzles and the entire fuel system from wear, coking, and other forms of damage.

Unavoidable ash formation and the resulting deposits on blades and the entire gas flow path lowers performance in addition to the lower performance already caused by the reduced turbine inlet temperature. The requisite periodic off-line turbine washing after a minimum cool-down time of hot parts degrades the availability of the power plant; this is compounded by increased maintenance efforts associated with shorter intervals for overhauls and longer outages. More frequent replacement of hot gas path parts compared to plants operated on light distillate fuel has to be expected.

Heavy fuels normally cannot be ignited for gas turbine startup; therefore a startup and shutdown fuel—usually light distillate—is needed with its own storage, forwarding system, and fuel changeover equipment. This system must be able to flush the fuel system after gas turbine trip during operation on heavy fuel oil if flushing the pipes was not possible by normal fuel changeover. Thermal insulation of all fuel equipment as well as trace heating for piping sections that are not flushable also have to be considered.

Additional special features are valid for these ash-forming heavy fuels if they contain high volatile fractions. This applies especially for crude oil, but also sometimes for heavy oils or residual oils if they are blended. Necessary measures are described under low boiling and high volatile liquid fuels.

Low boiling and high volatile liquid fuels such as kerosene, naphtha (as a refinery distillation product), and gas condensates (as natural gas liquids) are mostly free from impurities (contaminants typically enter the fuel due to negligence during transport, storage and handling). Naphtha and gas condensates have a high calorific value (>42 MJ/kg) but a low density, low viscosity, and a comparatively high vapor pressure (max. 0.9 bar at 38°C compared to 0.003 bar of light distillate).

Such fuels have nearly similar operating conditions as light distillates in terms of combustion. However, safety requirements for storage, forwarding, and injection systems as well as for the environs are very extensive compared to light distillate or natural gas. Leak-tight systems, advanced ventilation, additional enclosures, leakage detection, and, in some cases, explosion-proof equipment are mandatory.

Fuels with high vapor pressures (e.g., naphtha) require a safe startup and shutdown fuel (normally light distillate) to prevent explosive vapors from entering the combustion chamber and ultimately the entire gas turbine.

The low viscosity and lubrication properties of these fuels require the use of centrifugal pumps instead of positive displacement pumps. Depending on the type of mechanical parts employed in the fuel distribution system, lubricating additives may also be necessary.

Inerting and flushing systems with safe media may be needed for components with open liquid surfaces (e.g., leakage collecting tanks) and for maintenance activities. If vapors are heavier than air, sewer systems as well as pipe and cable trenches have to be protected against vapor ingress.

These special features are also valid for liquid fuels with high volatile fractions (especially crude oil and lighter fractions blended heavy oils or residual oils).

Methanol is comparable to naphtha in terms of its basic requirements for fuel supply system and safety aspects (additionally, the toxicity of methanol has to be considered).

A significant difference to such other liquid fuels is methanol's low calorific value of approx. 20 MJ/kg, which is only about half that of light fuel oil or naphtha. Consequently, specific burners are necessary for methanol firing as well as an enlarged capacity fuel supply system.

Heat Recovery Steam Generator

The heat recovery steam generator (HRSG) is the link between the gas turbine and the steam turbine process. Three main configurations are widespread:

- HRSG without supplementary firing
- HRSG with supplementary firing
- Steam generators with maximum supplementary firing

As elucidated in chapter 5, HRSGs without supplementary firing are most common in combined-cycle plants.

The function of the HRSG is to convert the thermal energy in the gas turbine exhaust into steam. After heating in the economizer, water enters the drum at slightly subcooled conditions. From the drum, it is circulated to the evaporator and returns to the drum as a water/steam mixture where water and steam are separated. Saturated steam leaves the drum and is forwarded to the superheater where it is exposed to the maximum heat exchange temperature of the hottest exhaust leaving the gas turbine. The heat exchange in an HRSG can take place on up to three pressure levels, depending on the desired amount of energy and exergy to be recovered. Today, steam generation at two or three pressure levels is most commonly used.

HRSG without supplementary firing

Construction. A HRSG without supplementary firing is essentially an entirely convective heat exchanger. The requirements imposed by operation of the combined-cycle power plant are often underestimated. In particular, provision must be made to accommodate the short startup

time of the gas turbine and the requirements imposed by quick load changes. The HRSG must be designed for high reliability and availability. HRSGs can be built in two basic configurations, based on the direction of gas turbine exhaust flow through the boiler.

Vertical HRSG. In the past, vertical HRSGs were most often known as forced-circulation HRSGs because of the use of circulating pumps to provide positive circulation of boiler water through the evaporator sections. In this type of boiler, the heat transfer tubes are horizontal, suspended from uncooled tube supports located in the gas path. Vertical HRSGs can also be designed with evaporators that function without the use of circulating pumps. However, whereas natural-circulation in horizontal boilers starts automatically, it must be ensured in vertical boilers by specific design measures.

Figure 7–6 shows a forced-circulation HRSG. The exhaust-gas flow is vertical, with horizontal tube bundles suspended in the steel structure. Circulating pumps ensure constant circulation within the evaporator. The structural steel frame of the steam generator supports the drums.

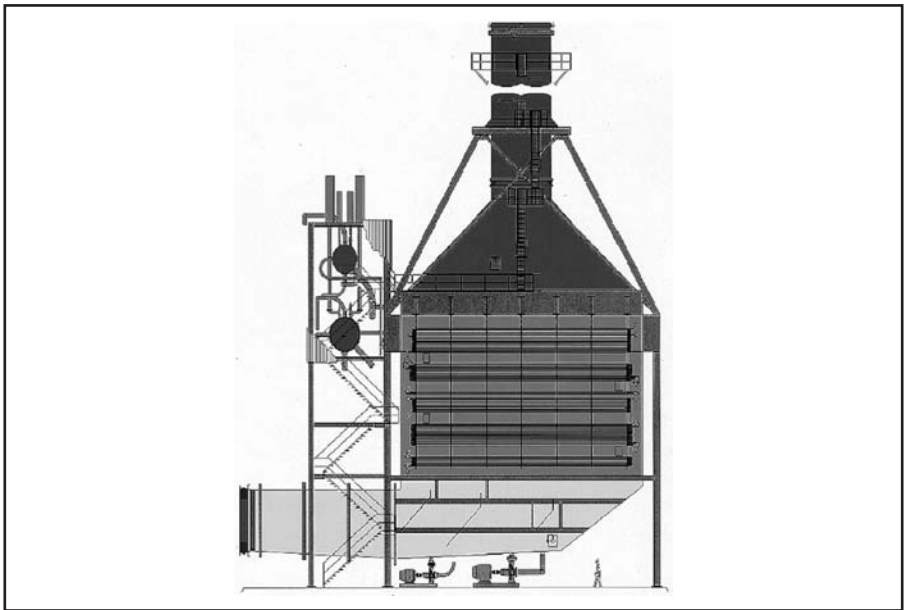


Figure 7–6 Forced circulation heat recovery steam generator

Horizontal HRSG. The horizontal HRSG is typically also known as a natural-circulation HRSG because circulation through the evaporator takes place entirely by gravity, based on the density difference of water and boiling water mixtures. In this type of boiler, the heat transfer tubes are vertical, and essentially self-supporting.

Figure 7–7 shows a natural-circulation HRSG. The exhaust-gas flow is horizontal. The steel structure is more compact than on a unit with vertical gas flow.

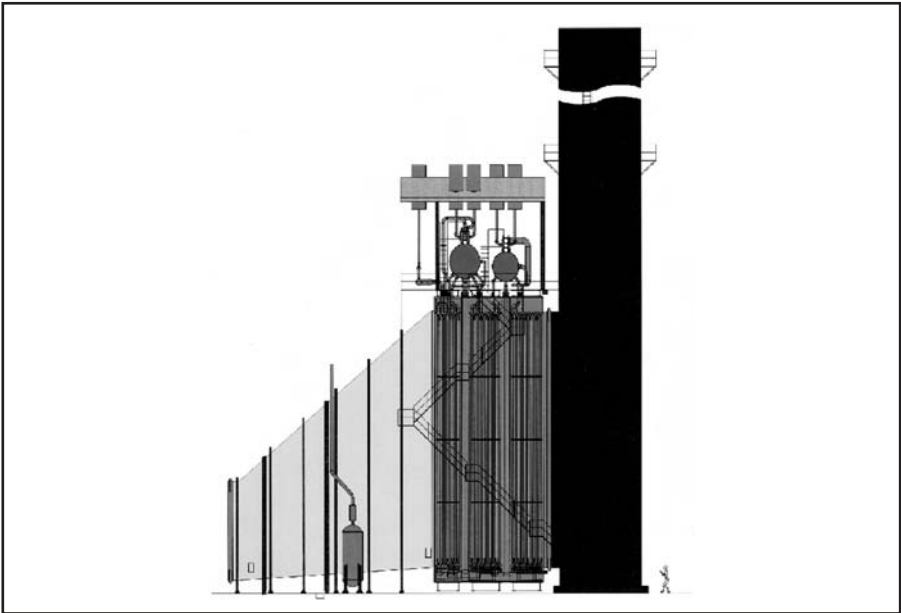


Figure 7–7 Natural circulation heat recovery steam generator

Design comparison. Either type—vertical or horizontal—can be used in a combined-cycle plant.

In the past, vertical HRSGs had several advantages that made them especially well suited to combined-cycle applications:

- Minimum footprint requirement arising from the vertical design

- Smaller boiler volumes because of the use of smaller diameter tubes
- Less sensitivity to steam blockage in economizers during startup

The main advantage of the horizontal HRSG is that no circulation pumps whatsoever are needed—an important consideration for applications with design pressures above 100 bar (1430 psig), where pumps must be designed and operated with special care. Additionally, there is an advantage with vertical tubes in the evaporator because the tubes with the highest heat absorption in the evaporator have the most vigorous circulation, and tube dry-out cannot occur in vertical tubes.

Current design techniques for natural-circulation boilers have overcome the disadvantages relative to vertical boilers. Space requirements and startup times are identical, water volumes in evaporators have been reduced by the use of smaller diameter tubes, and steam blockage is better handled in a modern, natural-circulation boiler. The same pinch points can be achieved in the high- and intermediate-pressure evaporator. Differences occur in steam performance only in large HRSGs with a tight low-pressure pinch point.

The presence of both technologies on the market indicates that both meet customer expectations and the preference for one or the other is more a historical or regional preference. Today most of the plants are equipped with horizontal natural-circulation HRSGs.

Once-through HRSG. The HRSGs previously described use a steam drum for water/steam separation and water retention. Combined-cycle plants are often operated in cycling duties with frequent load changes and start-stop cycles. HRSGs employing a drum with design pressures of 100 bar (1430 psig) and beyond impose restrictions on this mode of operation.

In a once-through steam generator, the economizer, evaporator, and superheater are basically one tube—water enters at one end and steam leaves at the other end, eliminating the drum and the circulation pumps. This design has advantages at higher steam pressures because

the drum does not impose limits during startups and load changes. Both horizontal and vertical HRSGs can be built with the once-through circulation principle. The principle of once-through evaporation (BENSON® Once-Through design) compared to a drum-type boiler design is illustrated in figure 7–8.

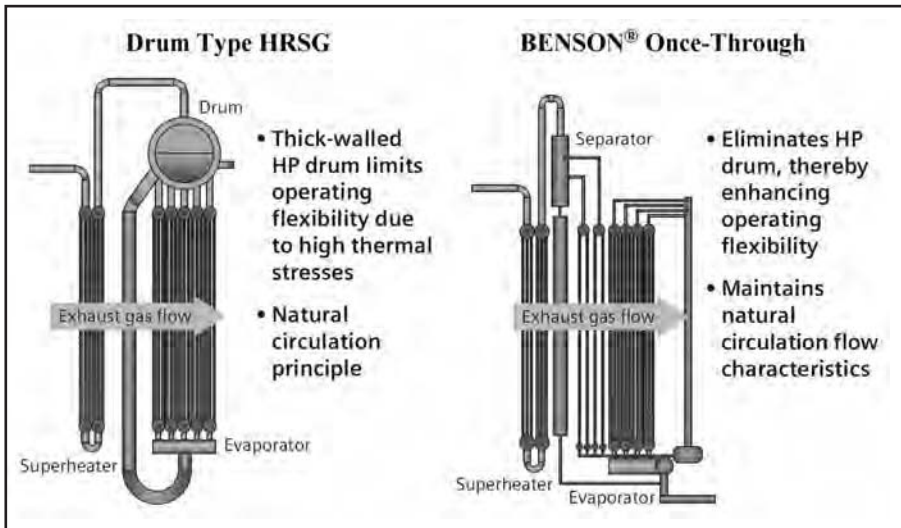


Figure 7–8 Principle of drum type and once-through evaporation

Advanced HRSGs for maximum efficiency are of a three-pressure reheat configuration with the high-pressure section in a once-through design, and medium- and low-pressure sections as drum-type evaporators. Figure 7–9 shows a three-pressure reheat once-through HRSG with horizontal gas flow. The high pressure section is of the once-through type, while the low and intermediate pressure sections are of conventional, natural-circulation design with LP and IP drums located on top of the boiler.

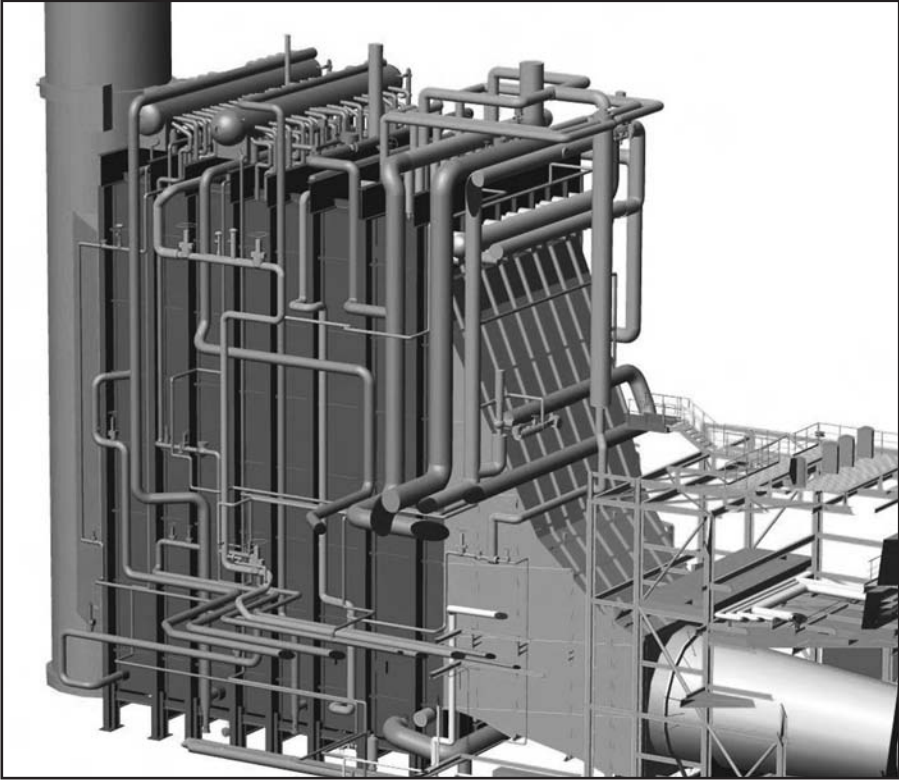


Figure 7-9 Three-pressure reheat once-through HRSG with drum type LP and IP sections

Casing of the HRSG. Two main variants are used for the casing of the HRSG:

- Cold-casing design with internal insulation
- Hot-casing design with external insulation

The cold-casing design is widely used with natural-circulation steam generators and has advantages at high exhaust temperatures. The casing withstands the exhaust pressure forces and the internal insulation keeps it at a low temperature. The casing construction also imposes no limits on startup time to allow for thermal expansion of the casing.

The hot-casing design is often used with the vertical HRSGs and has advantages at lower exhaust temperatures. When fuels with high

sulfur content are fired, a hot casing at the cold end of the HRSG can limit corrosion.

With modern gas turbines running at high exhaust temperatures, a hot-casing design requires high-alloy materials in the hot end of the boiler. Today, some vertical HRSGs also use a cold-casing design.

Finned Tubing. Heat transfer in the HRSG is mainly by convection. Heat transfer on the water side is much better than on the exhaust side, and so finned tubes are employed on the exhaust-gas side to increase the heat transfer surface.

Normal fin density for a HRSG downstream of a gas turbine (firing natural gas or No. 2 oil) is 5 to 7 fins per inch (200 to 280 fins per meter) with a foreseeable development potential of up to 340 fins per meter. If the gas turbine uses heavier fuels, the fin density is reduced to 3 to 4 fins per inch (120 to 160 fins per meter) to better handle deposits.

The optimum HRSG must fulfill the following, sometimes contradictory, conditions:

- The rate of heat recovery must be high (high efficiency)
- Pressure drop on the exhaust side of the gas turbine must be low to limit losses in power output and efficiency of the gas turbine
- The permissible pressure gradient during startup must be steep
- Low-temperature corrosion must be prevented

It is particularly difficult to meet the first two conditions at the same time. Because of the relatively low temperature, the heat transfer takes place mainly by means of convection. The differences in temperature between the exhaust and the water (or steam) must be small to obtain a good rate of heat recovery, which requires large surfaces. This means a large pressure drop in the exhaust unless the velocity of the gas is kept low, which would result in a lower heat transfer coefficient and, therefore, further increase the surfaces required. Using small-diameter finned tubes helps to solve this problem.

Another benefit of small-tube diameters is the small amount of water in the evaporator, resulting in a lower thermal constant, which favors quick load changes.

HRSGs being built today have low pinch points and small pressure drops on the exhaust side. Pinch point values of 6 to 15K (11 to 27°R) at pressure drops of 25 to 35 mbar (10" to 14" W.C.) are attainable even with a triple-pressure reheat boiler.

Low-temperature corrosion. When designing HRSGs, care must be taken to prevent or restrict low-temperature corrosion. To accomplish this, all surfaces that come into contact with the exhaust must be at a temperature above or slightly below the sulfuric acid dew point. When burning a sulfur-free fuel in the gas turbine, the limit is determined by the water dew point.

Suitable precautions enable operation of heat exchangers at temperatures below the acid or water dew points. This can be done by selecting appropriate materials, or by adding corrosion allowances to the design of affected tubes. Because there are only a few tubes in this temperature range, this strategy can be beneficial despite the low exergy of the heat gained.

Optimum design of a HRSG. Designing a HRSG involves optimizing between cost and benefit. The main cost driver is the heat exchange surface installed. The indicator generally used is the pinch point in the evaporator. The surface of the evaporator increases exponentially as the pinch point decreases, whereas the increase in steam generation is only linear. For that reason, the pinch point selected is a critical factor in determining the heat transfer surface. In installations where efficiency is highly valued, the pinch point is 8 to 15K (14 to 27°R); where efficiency is lower, the pinch point can be higher, 15 to 25K (27 to 45°R).

Operating experience. One challenge affecting the design of the HRSG is the quick startup capability of the gas turbine, especially for the cold and warm plant conditions. The rapid thermal expansion that occurs during a startup can be accommodated through suitable design measures such as suspension of the tube bundles, drum design, tube-to-header connections, and so on.

The main constraint for the loading gradient is often dictated by the drum. To enable as quick a start as possible, the walls of the drum should be as thin as possible, which can be achieved, provided the design live-steam pressure is sufficiently low. Modern gas turbines have higher exhaust temperatures than older models; therefore, higher live-steam pressures, especially with reheat steam cycles, are now more attractive and this results in longer startup times.

Once-through HRSGs eliminate the thick high-pressure drum and, therefore, afford the desired high thermal flexibility.

Another point of concern is the volumetric change in the evaporator during startup. The large differences in specific volume between steam and water at low and intermediate pressures cause large amounts of water to be expelled from the evaporator at the onset of the evaporation process. If the drum cannot accommodate most of this water, a large amount of water would be lost through the emergency drain of the drum during every startup, or an undesired emergency trip of the unit would be required to prevent water carryover into the steam system.

To improve part-load efficiency and behavior of the combined-cycle plant, the boiler is operated in sliding-pressure mode. That means the system is generally operated at a lower pressure when the steam turbine is not at full load. This can be accomplished by keeping the steam turbine control valves fully open. For example, in a system with two gas turbines and two HRSGs feeding a common steam turbine, half-load of the whole power station can be accomplished with only one of the gas turbines running at full load. In sliding-pressure mode, the live-steam pressure is at 50% of the pressure at full load. The steam volumes in the evaporator, superheater, and live-steam lines of the HRSG in operation are doubled.

During off-design conditions, economizers can start to generate steam, which can block tubes and reduce the performance of the HRSG. To keep this within limits, the economizer is dimensioned so that the feedwater at the outlet is slightly subcooled at full load. This difference between the saturation temperature and the water temperature at the economizer outlet 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 2 to 12K [4 to 22°R]). Proper routing of the economizer outlet tubes to the drum also prevents blockage if there is steaming in the economizer.

Another way of preventing steaming in the economizer is to install the feedwater control valve downstream of the economizer; the economizer is thus kept at a higher pressure and steaming is prevented.

HRSG with limited supplementary firing

Despite the fact that the majority of the HRSGs are unfired, there are occasional applications where a limited amount of supplementary firing is required.

The operating principles of a HRSG with limited supplementary firing are the same as those for the unfired boiler. There are various designs available for the firing system itself. Units that do not exceed a gas temperature of approximately 780°C (1436°F) downstream of the supplementary firing can be built with simple duct burners. This limit can be extended but requires modifications to the design of the HRSG.

Figure 7–10 shows a HRSG with supplementary firing. This system is particularly well suited to combusting natural gas, which attains uniform temperature distribution downstream of the burners. Radiation to the walls of the combustion chamber is relatively low. For that reason, the majority of the HRSGs of this type combust natural gas. There are systems available for oil, but the burner equipment is more complex and costly to install, operate and maintain.

The importance of supplementary firing in HRSGs used for power generation alone is diminishing. This is mainly caused by two facts:

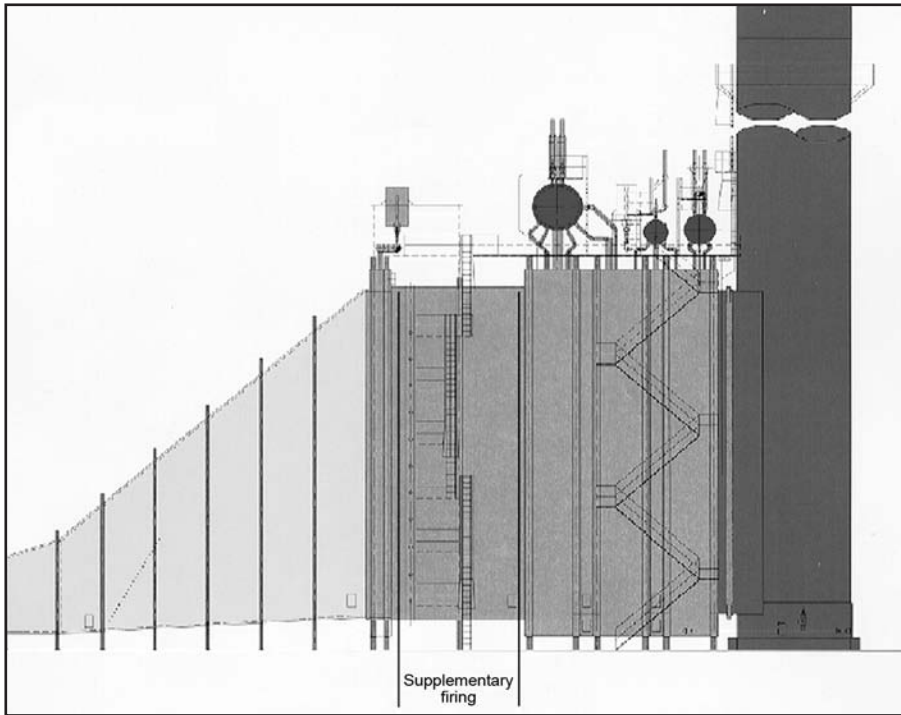


Figure 7-10 Supplementary fired heat recovery steam generator

- Modern gas turbines have exhaust temperatures closer to the maximum allowable HRSG inlet gas temperature, thereby reducing the effect of any supplementary firing.
- Supplementary firing downstream of a modern gas turbine results in an efficiency decrease for the combined cycle. With older gas turbine models, the efficiency was more or less constant.

Supplementary firing is most often applied in combined-cycle cogeneration plants where the amounts of process steam must be varied independently of the electric power generated. In this case, supplementary firing is used to control the amount of process steam generated. Additional applications include combusting gases, which are not suitable for firing in the gas turbine (e.g., pressure is too low or heat content unsuitable), and reaching a higher power output, but with the expensive penalty of a lower power generation efficiency.

A larger amount of fuel could be fired if the combustion chamber of the duct burner has cooled walls. Circulating water from the evaporator provides cooling in such configurations. Despite the higher operating flexibility, the additional cost and complexity very seldom justify this solution.

Arranging supplementary firing in various heat exchange tube bundles can also increase HRSG output. In this application, the exhaust is cooled downstream of the gas turbine by a heat exchanger such as an evaporator or superheater. The exhaust can then be reheated in a duct burner before passing through the remaining portion of the HRSG. This solution is rare and reserved for niche applications that have difficulty competing with an unfired HRSG or other solutions.

Steam generator with maximum supplementary firing

The maximum firing rate is set by the oxygen content of the gas turbine exhaust. With this type of steam generator, the exhaust from the gas turbine is used primarily as an oxygen carrier. The heat content of the gas turbine exhaust is low compared with the heat input of firing in the boiler. It is, therefore, no longer correct to speak of a HRSG.

The design of a steam generator of this type is practically identical to that of a conventional boiler with a furnace, except that there is no regenerative air preheater. The gas turbine exhaust has a temperature of 450 to 650°C (842 to 1202°F), rendering a regenerative heater unnecessary. To cool the exhaust to a sufficiently low temperature downstream of the steam generator, an additional economizer is provided, which takes over a portion of the feedwater preheating from the regenerative preheating of the steam turbine. The best arrangement splits the feedwater between the economizer and the high-pressure feedheaters. When the fuel is gas, an additional low-pressure partial flow economizer improves efficiency. The fuel combusted in the boiler may be oil, gas, or pulverized coal.

This application can be used to increase the output from an existing conventional steam turbine plant using a gas turbine and its exhaust energy. Due to the high integration of both cycles (e.g., the burners of the existing boiler must be retrofitted to make them suitable for the

exhaust application), the complexity and the number of interfaces are considerable (see chapter 6).

Steam Turbine Technology

The most important requirements for a modern combined-cycle steam turbine are:

- High efficiency
- Short startup times
- Short installation times
- Floor-mounted configuration

In the past, combined-cycle steam turbines were applications of industrial steam turbines or derivatives from conventional steam turbine plants. The main differences between conventional steam turbines and combined-cycle steam turbines are:

- Higher power output
- Higher live-steam temperatures and pressures
- Higher number of extractions for feedwater heating

Today new combined cycles frequently are equipped with reheat steam turbines.

The main differences between combined-cycle steam turbines and conventional steam turbines are:

- Fewer or even no bleed-points as opposed to 6 to 8 for feed heating
- Floor-mounted configuration
- Shorter startup times
- Lower power output
- Lower live-steam pressures 100 to 170 bar (1430 to 2460 psig) as opposed to 160 to 300 bar (2310 to 4340 psig)

With the different requirement profile and the large volume of the combined-cycle plant orders, an optimized design for this application is justified.

Characteristics of combined-cycle steam turbines

Steam turbines used for combined-cycle installations are simple machines. In the past they used relatively low live-steam parameters; with increasing gas turbine exhaust temperatures, optimal steam pressures also increased. Live-steam temperatures have now reached the level of those at conventional steam power plants.

Combined-cycle plants frequently generate steam at more than one pressure level. Due to multiple inlets, the steam mass flow in the steam turbine increases from the inlet towards the exhaust (the bleed for the partial feedwater preheating involves only a small reduction). In a conventional steam turbine, the steam-mass flow is reduced to roughly 60% of the inlet flow at the exhaust.

Short startup times are of particular importance because combined-cycle plants are often used as medium-load units with daily or weekly startups and shutdowns.

Table 7–4 illustrates how in recent years, the combined-cycle plant has profoundly changed gas turbine technology, resulting in a major impact on steam turbine requirements:

Table 7–4 Change of boundary conditions for steam turbines in combined-cycle plants

	Traditional	Modern
Gas turbine exhaust temperature	450–550°C (842–1,022°F)	550–650°C (1,022–1,202°F)
Live steam temperatures	420–520°C (788–968°F)	520–600°C (968–1,112°F)
Number of steam pressure levels	1 or 2	2 or 3
Live steam pressure	30–100 bar (420–1,430 psig)	100–170 bar (1,430–2,460 psig)
Reheat steam cycle	no	yes
Number of gas turbines per steam turbine	1 to 5	1 or 2

Multishaft and Single-Shaft Plants

In a multishaft combined-cycle plant, there are generally several gas turbines with HRSGs generating steam for a single-steam turbine. The steam and gas turbines use separate shafts, generators, step-up transformers, and so on. By combining the steam production of all the HRSGs, a larger steam volume enters the steam turbine, which generally raises the steam turbine efficiency.

Modern gas turbines achieve higher output with high exhaust temperatures. With the large gas turbines on the market, one steam turbine per gas turbine or one steam turbine for two gas turbines is common.

If one steam turbine per gas turbine is installed, the single-shaft application is the most common solution—gas turbine and steam turbine driving the same generator.

A plant with two gas turbines can be built either in a two-gas-turbine-on-one-steam-turbine configuration (multishaft) or as a plant with two gas turbines, each in a single-shaft configuration.

There are two concepts for single-shaft plants. In the first, the generator is situated between the gas turbine and the steam turbine, each turbine driving one end of the generator, the steam turbine is coupled to the generator rigidly or with a clutch. In the second, the generator is at one end of the line of shafting, driven by both turbines from the same end. The steam turbine, therefore, is rigidly coupled to the gas turbine on one end and to the generator on the other. Figure 7–11 illustrates various single-shaft arrangements.

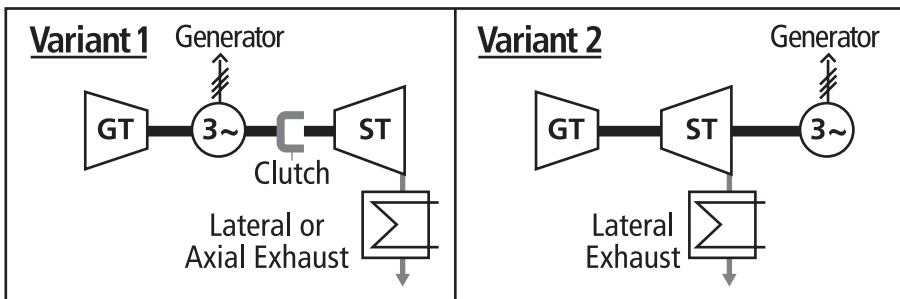


Figure 7–11 Different single-shaft combined-cycle configurations

The type of turbine-generator configuration (multishaft or single-shaft, with or without clutch) has a significant impact on components used in the respective combined cycle.

Single-shaft configurations with the generator situated between the two turbines also require a connecting flange on the generator exciter end that is designed for handling the steam turbine torque.

Steam turbines arranged between gas turbine and generator in single-shaft units also need a connecting flange on the high-pressure turbine end and the rotor must be able to transfer the torque of the gas turbine. Steam turbines located at the end of single-shaft power trains are mostly connected to the generator on the high-pressure turbine end and have an axial steam outlet at the end.

Clutch

A combined-cycle single-shaft configuration with the generator between the two turbines enables installation of a clutch between steam turbine and generator. The clutch is normally of the synchronous self-shifting type. This means the clutch engages in that moment when the steam turbine speed tries to overrun the rigidly coupled gas-turbine generator and disengages if the torque transmitted from the steam turbine to the generator becomes zero.

The clutch allows startup and operation of gas turbine without driving the steam turbine. This results in a lower starting power requirement and eliminates certain safety measures for the steam turbine (e.g., cooling steam or sealing steam).

Furthermore, it provides design opportunities for accommodating axial thermal expansion. The clutch itself compensates a portion of axial displacements, and the two thrust bearings allow selective distribution of the remaining axial expansion (reducing tip clearance losses). In addition, the clutch allows more operational flexibility such as gas turbine simple-cycle operation or early maintenance activities on gas turbine during steam turbine cool down.

On the other hand, the clutch is an additional component with a potential impact on availability. Additionally, the generator located at the end of the line of shafting has advantages during

generator overhaul and the rare necessity of rotor removal. However, single-shaft units without a clutch definitely need auxiliary steam supply to cool the steam turbine during startup. This is not necessary in units with a clutch.

Figure 7–12 shows a steam turbine for a single-shaft unit in a triple-pressure reheat plant. The steam turbine output is 142 MW with live-steam parameters of 170 bar (2460 psig) and 565°C (1049°F).

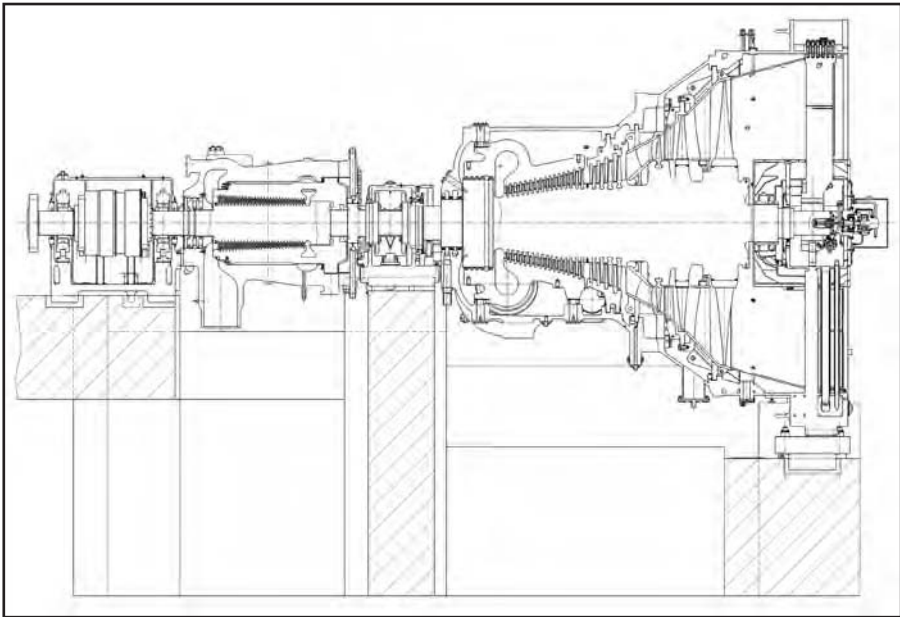


Figure 7–12 Cross section of a 142 MW reheat steam turbine with a separate HP turbine and a combined IP/LP turbine with axial exhaust

Live-steam pressure

Early gas turbines had lower power outputs and with their lower exhaust temperatures, the optimum pressure for the steam cycle was low. This was beneficial for the steam turbine for two reasons: low pressures yield larger steam volumes that can be more efficiently expanded in the steam turbine, and with a non-reheat cycle, a low

live-steam pressure limits the moisture content in the steam turbine exhaust and, therefore, erosion.

Modern gas turbines with their high exhaust temperatures result in combined cycles with high live-steam pressures in the steam cycle.

A geared high-pressure steam turbine enables high-efficiency expansion with low live-steam volumes by using a gearbox to increase the speed of the high-pressure turbine, increasing blade length, reducing secondary blade losses, and resulting in an improved overall efficiency.

Figure 7–13 depicts a steam turbine with a geared high-pressure turbine that takes advantage of high steam pressure/low steam volume in a combined-cycle plant. It also shows the compact dimensions of the geared high-pressure turbine. The additional investment for the gearbox is also justified by the increased thermal flexibility of the high-pressure turbine, which is important for the cycling duties of the combined-cycle plant.

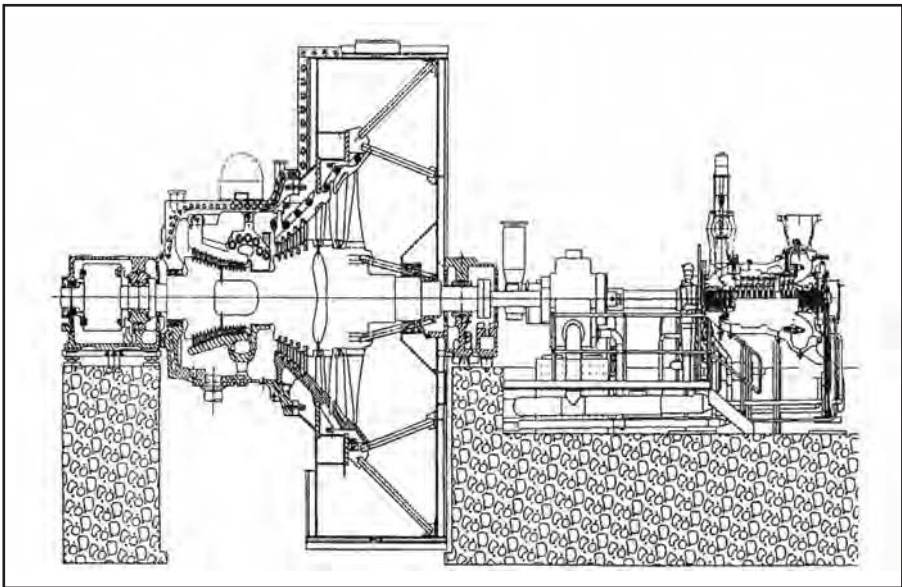


Figure 7–13 Cross section of a two-casing steam turbine with geared HP turbine

Generators

The majority of gas turbines and steam turbines are directly coupled to two-pole generators. For units with ratings below 40 MW, four-pole generators that run at half speed are more economical. It is advantageous that turbines in this output range usually already have a gearbox, therefore, using a four-pole generator only requires adaptation of the gearbox reduction ratio.

Four types of generators are used in combined-cycle plants:

- Air-cooled generators with open-circuit air cooling
- Air-cooled generators with a closed-circuit air (totally enclosed water to air-cooled, TEWAC)
- Hydrogen-cooled generators
- Water-cooled generators in single-shaft plants

Generators with *open-circuit air-cooling* are low in cost and have no need for additional cooling; however, problems with fouling, corrosive atmospheres, and noise can arise. Generators with *closed-circuit air-cooling* are built for capacities up to 480 MVA. These machines are reasonable in cost and provide excellent reliability. The full-load efficiency of modern air-cooled generators is above 98%. *Hydrogen-* and *water-cooled generators* attain efficiencies of approximately 99% at full load, making their efficiency performance superior, particularly at part-load, to air-cooled machines. The power density of water-cooled generators is greater, but their physical dimensions are smaller. However, water-cooled machines require additional auxiliaries and monitoring equipment, are more complex in design, and, as a result, more expensive than air-cooled machines. In general, generators achieve high reliability; air-cooled machines offer somewhat higher reliability because of their simplicity.

Figure 7–14 shows a generator with closed-circuit air cooling. Water is used to cool the air. These machines are well suited for use in single-shaft combined-cycle plants where the gas turbine and the steam turbine drive the same generator. Nowadays, air-cooled generators of up to 480 MVA have been built and tested due to the overall benefits of air cooling. For higher outputs, hydrogen or water-cooling must be selected.

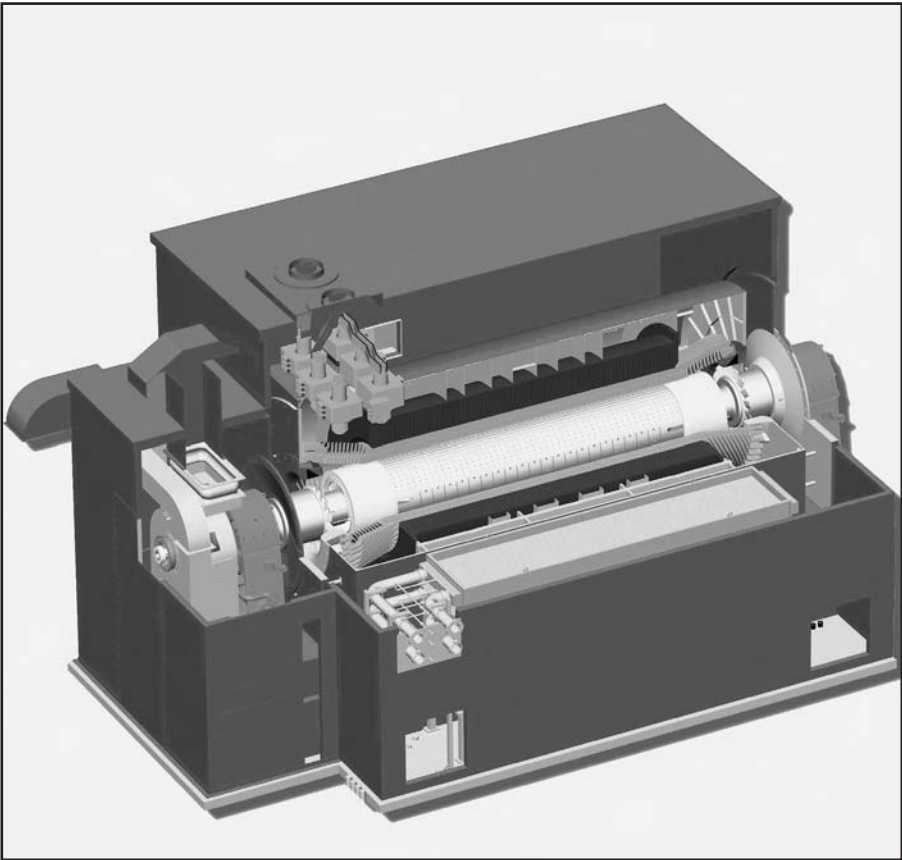


Figure 7–14 Cutaway drawing of an air-cooled generator for use in combined-cycle power plants

Electrical Equipment

A single-line diagram for a combined-cycle power plant with a single shaft is shown in figure 7–15. It is similar to a single-line diagram for other types of power stations. Key features in this diagram are:

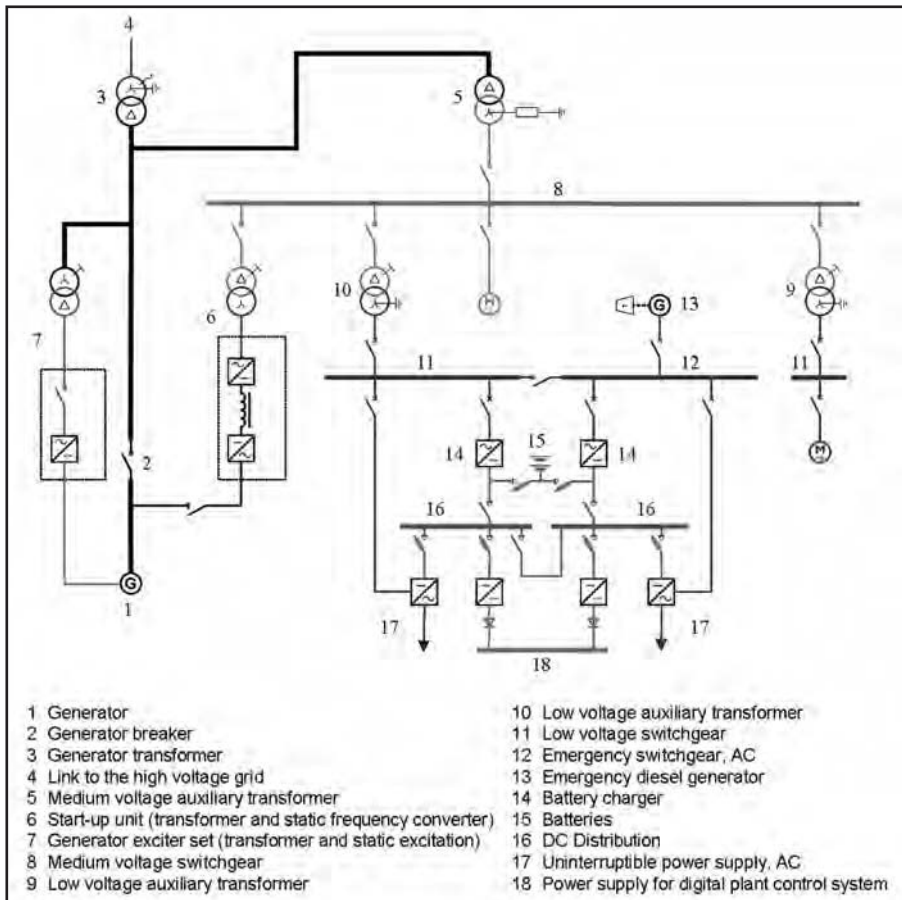


Figure 7–15 Single-line diagram

- One generator for the gas turbine and steam turbine (only one set of leads to the high voltage switchyard is needed)
- Auxiliaries fed from the generator output, which eliminates a separate feeder from the grid and allows the plant to run at full speed no load during a grid blackout, ready for reconnection
- Auxiliary transformers to feed the auxiliaries and to feed the static frequency converter to start the gas turbine using the generator as a motor
- Medium voltage bus for large motors
- Uninterruptible power supply for important loads

Cooling System

Three fundamentally different cooling configurations can be implemented in combined-cycle plants:

- Direct air cooling in an air cooled condenser (alternatively indirect air cooling)
- Evaporative cooling with a wet or hybrid cooling tower
- Once-through water cooling using river water or seawater

For *direct air cooling*, no cooling water is required, but the output and efficiency of the plant are reduced due to higher vacuum levels (figure 7–16). This is used in regions where no water is available, or as a means to minimize the impact on the infrastructure and environment, thereby facilitating permitting.

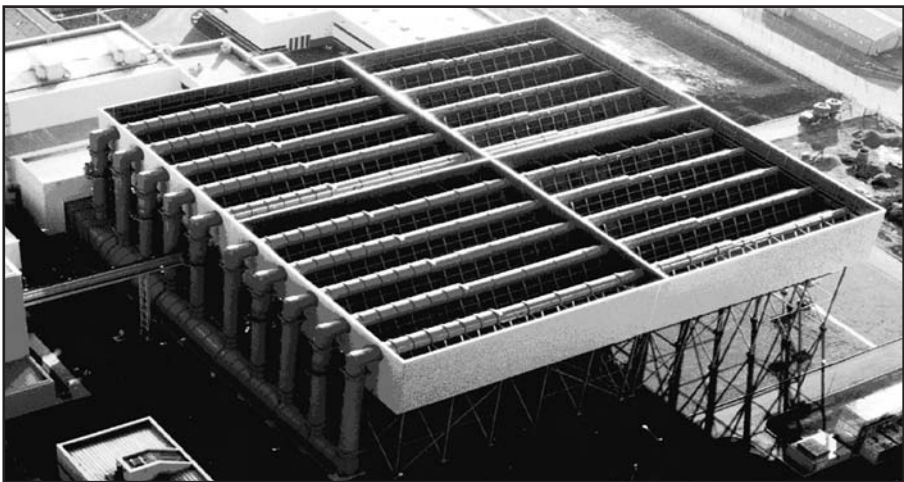


Figure 7–16 Typical arrangement of an air-cooled condenser

Alternatively, an indirect air cooling system could be used that employs dry cooling towers. Indirect cooling systems are more flexible in terms of arrangement, but usually also more expensive.

Evaporative cooling systems require water to replace evaporation and blowdown losses. This amount of makeup water depends on the

exhaust steam of the steam turbine and amounts to approximately 0.3 kg/s (2400 lb/h) per MW installed plant capacity for a combined-cycle plant without supplementary firing.

An evaporative cooling system for combined-cycle plants is primarily of the mechanical draft type (wet-cell cooling tower as shown in figure 7–17). This is based on the fact that only half of the cooling capacity is necessary for combined-cycle plants compared to conventional steam power plants with an identical installed capacity. As the specific cooling demand is relatively small in combined-cycle plants, the required auxiliary power consumption for mechanical draft evaporative cooling systems is also low related to the gross plant capacity. On the other hand, the specific investment for natural draft cooling towers is very high, even for small cooling demands; therefore, mechanical draft cooling towers are most often the more economical option for combined-cycle plants (auxiliary power consumption versus investment).

Mechanical draft evaporative cooling systems for combined-cycle plants require approx. 0.5 to 0.7% of the gross plant capacity as auxiliary power consumption (cooling water pump and cooling fan power), whereas the demand for natural draft systems is less than 0.5%, which is comparable to direct air cooling with approx. 0.4 to 0.5% (for large ACC).



Figure 7–17 Typical arrangement of a wet-cell cooling tower

The once-through water-cooling variant has a water requirement that is on the order of 40 to 60 times greater than for evaporative cooling systems. After serving as heat sink, this water is returned to the water source (e.g., river, sea, or cooling pond) from which it was taken.

Normally, the decision on the type of cooling system used depends on the available supply of cooling water because the most economic cooling system is often a once-through water-cooled condenser. In a small number of cases, boundary conditions such as plume abatement or limitations prevailing at the site are determining factors when selecting the type of cooling system.

For example, a typical plume abatement concept for evaporative cooling systems is a hybrid cooling tower (figure 7–18). In hybrid cooling towers, 10 to 15% of the waste heat is rejected in a dry air cooling section, and the remaining waste heat in a conventional mechanical draft evaporative cooling system. The hot dry air downstream of the dry section reduces the humidity in the evaporative section and thus prevents formation of a visible plume.

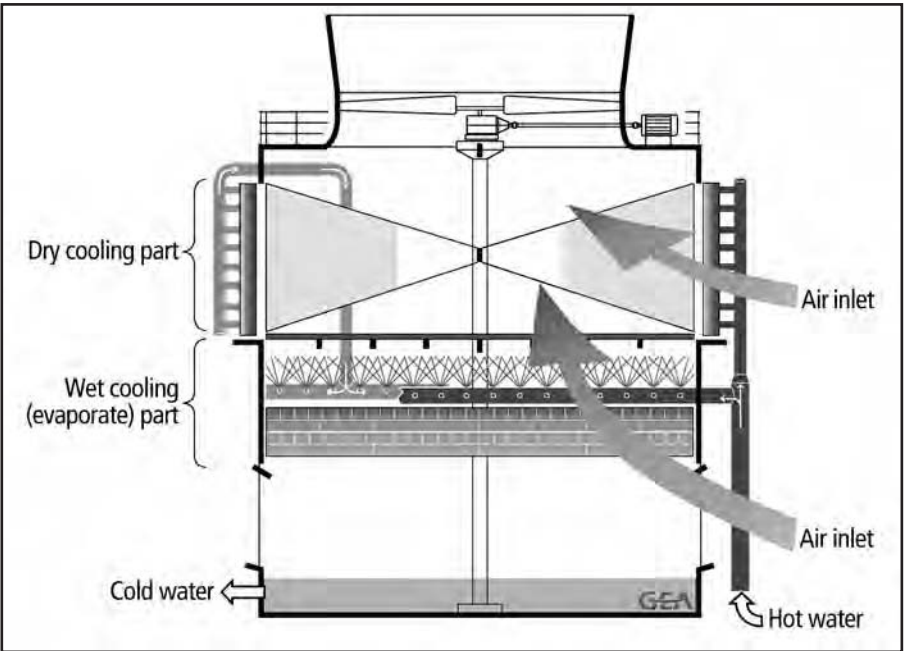


Figure 7–18 Principle of hybrid cooling tower

In a combined-cycle plant, however, the steam turbine generates only one-third of the installed electrical capacity (the gas turbine generates the other two-thirds), and requires only half the cooling capacity of a conventional steam power plant with an equal installed capacity. Hence the cooling system for combined-cycle plants has less impact on power output and efficiency when compared to the cooling system in conventional steam power plants.

Bypass Stack

Combined-cycle plants are sometimes equipped with exhaust bypass dampers and bypass stacks to permit for simple-cycle operation of the gas turbine(s). With modern gas turbines, this feature is becoming less and less important.

The reasons for this are:

- The design is expensive with gas turbine exhaust temperatures of 600°C and above.
- If the damper fails, the entire plant must be shut down.
- An exhaust damper has leakage losses with resultant losses in output and efficiency.
- The associated costs are significant.
- All these aspects become even more pronounced with advanced large gas turbines and the resulting larger dimensions, increased leakages, as well as higher exhaust temperatures.

An exhaust bypass stack (with or without damper) is a prerequisite for phased installation of a combined-cycle plant—one in which the gas turbine runs in simple-cycle operation before the steam cycle is connected. This allows the plant operator to get two-thirds of the final plant capacity on line sooner and thus create revenue at an earlier time. This can be economically attractive for plants with large capital payments. However, connection of the steam cycle requires

interruption of normal gas turbine operation because the gas turbines are needed for commissioning of the steam cycle.

In deregulated markets, none of this is justified—the investment for the bypass stack, the expensive power generation during gas turbine simple-cycle operation, and the business interruption to connect the steam cycle—especially because a steam cycle can be installed in less than two years.

An exception might be feasible for locations with very low fuel costs and for countries with urgent need of new electricity supply. In such cases, phased construction could be economical also for advanced, large gas turbines. Nevertheless even for such cases a flue gas damper seems not to be recommendable due to previously mentioned inherent disadvantages. Use of simple blanking plates should be considered for changeover from simple-cycle operation to combined-cycle operation after completion of the water-steam cycle. For maintenance reasons, changeover to bypass operation will be possible within a reasonable time.

Instrumentation and Control

The complexity of a combined-cycle plant with its rapid interactions, especially during fast startup, requires a modern, advanced control system. Several power plant automation tasks such as turbine control for the gas turbine and steam turbine, boiler control, balance of plant (BOP), and integration of secondary equipment have to be performed in an optimized manner to maintain full control over the entire plant for all operating procedures involving transient processes and including startup and shutdown of the overall plant. Refer to chapter 8 for more information on this subject.

Other Components

In addition to the major equipment described in this chapter, the combined-cycle plant includes other equipment and systems. These include, for example:

- Fuel supply systems including a gas compressor, if required
- Steam turbine condenser
- Feedwater tank/deaerator
- Feedwater pumps
- Condensate pumps
- Piping and fittings
- Evacuation system
- Water treatment plant
- Compressed air supply
- Steam turbine bypass
- Civil engineering
- Heating, ventilation, and air conditioning
- Auxiliary boiler, if required

These are similar regardless of whether used in combined-cycle plants or in other types of power plants and, therefore, are not described here in detail.

8

Control and Automation

Power plants are normally operated to meet a demand dictated by the electrical grid, and where there is one, by the process steam system. The control systems discussed in this chapter are used to fulfill these requirements, to ensure correct control in transient modes such as startup and shutdown, and to ensure the safety of the plant under all operating conditions.

Control System

The control system is the brain of the power plant. Its tasks are the supervision, control, and protection of the plant. It enables safe and reliable operation.

The gas turbine is supplied with a standardized control system, providing a fully automated machine operation. The water/steam process is correspondingly automated to achieve a certain degree of uniform operation of the plant as whole, thereby reducing the risk of human error.

For this reason, the control and automation systems of a combined-cycle plant form a relatively complex system, even though the thermal process is fairly simple. Fully electronic control systems are applied in modern combined-cycle plants.

Main features of the control system for combined-cycle power plants are:

- Truly distributed architecture
- Complete range of functions for process control
- Communication capability due to several bus levels
- Compliance with standard communication protocols
- Openness for third-party applications
- On-line programmability with easy creation/editing of the programs

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 and raises the availability of the plant.

A highly automated control system encompasses the following hierarchic levels (see figure 8–1):

- **Drive level:** All individual drives are monitored and controlled. Drive protection and interlockings are provided by the control system. Signal exchange with higher hierarchical levels is also provided.
- **Functional group level:** The individual drives for one complete portion of the process are assembled into functional groups. The logical control circuit on this level encompasses interlocks, automatic switching, and preselection of drives. Examples of typical function groups are feedwater pumps, cooling water pumps, and lube oil system.
- **The machine level:** Includes the logical control circuits that link the function groups to each other. These include, for example, the fully automatic starting equipment for the gas turbine or the steam turbine.
- **The unit level:** This includes the controls that coordinate the operation of the whole combined-cycle power plant.

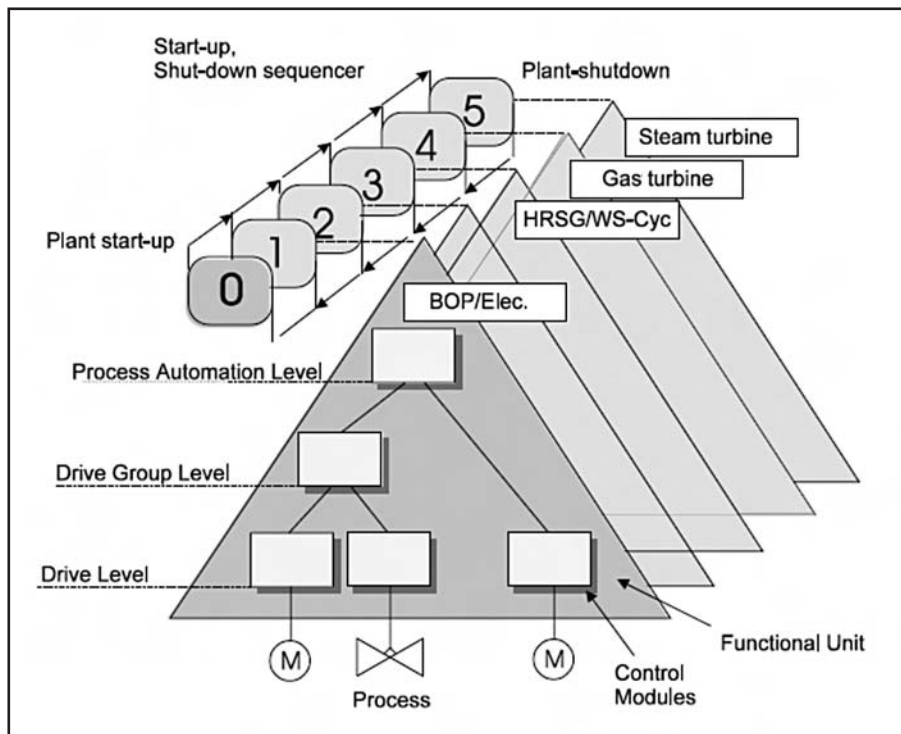


Figure 8–1 Hierarchic levels of automation

For a base-load power plant, a lower degree of automation could be selected because the frequent starting and stopping of an intermediate or peak load duty plant requires a higher degree of automation.

Process computers provide sequencing events, keep long term records and statistics, provide management information of the economy of the plant, optimize the heat rate and operation of the plant, and advise on the intervals between cleanings, inspections, and other maintenance work.

Figure 8-2 shows a standard layout for a modern combined-cycle power plant control room.



Figure 8–2 Standard layout for a modern combined cycle power plant control room

One type of control in a combined-cycle plant is closed loop control. This is when a controller receives actual measured data as an input, and uses it to correct a signal to a control device, with the aim of reaching the set point of a given parameter. The important closed control loops for a combined cycle fall into two main groups:

- The main plant load control loop
- The secondary control loops, which maintain the important process parameters, such as levels, temperatures, and pressures within permissible limits

A third group of closed control loops are component specific, such as minimum flow control for a pump or lube oil pressure control, and are intended primarily for the safe working of individual items of equipment. Because they do not directly affect the control of the cycle as a whole, they are not discussed here.

Applied logically, this leads to the hierarchical structure of the entire control system already mentioned.

Load Control and Frequency Response

The electrical output of a combined-cycle plant without supplementary firing is controlled by means of the gas turbine only. The steam turbine will always follow the gas turbine by generating power with whatever steam is available from the heat recovery steam generator (HRSG).

The gas turbine output is controlled by a combination of variable inlet guide vane (VIGV) control, and gas turbine inlet temperature (TIT) control. The TIT is controlled by a combination of the fuel flow admitted to the combustor and the VIGV setting. Modern gas turbines are equipped with up to three rows of VIGVs allowing a high gas turbine exhaust gas temperature down to approximately 40% GT load. Below that level, the turbine inlet temperature is further reduced because the airflow cannot be further reduced.

After a gas turbine load change, the steam turbine load will adjust automatically with a few minutes delay dependent on the response time of the HRSG. It is, however, sometimes suggested that independent load/frequency control of the steam turbine should be provided for sudden increases or decreases in load. Such a system would require the steam turbine to be operated with continuous throttle control, resulting in much poorer efficiencies at full and part loads and additional complications. Because the gas turbine generates approximately two-thirds of the total power output, a solution without control for the steam turbine power output is generally preferred. This is also supported by the fact that modern gas turbines react extremely quickly to frequency variations, and can usually compensate for the delay in the steam turbine response with falling frequencies.

Figure 8–3 illustrates the concept of closed loop load/frequency control without separate steam turbine control for a plant with two gas turbines and one steam turbine. An overall plant set point, P_s , is given to the overall plant load control system, KAR, which determines how the load should be distributed between the gas turbines. It receives a load/frequency signal LFC from each of the three generators, which is used to determine whether it is necessary to make any correction to the load. The load of the individual gas turbines is controlled by setting the position of the VIGVs and the TIT control, which varies the air and fuel flow to the gas turbine. Because the TIT cannot be directly measured, readings are taken of the turbine pressure ratio and exhaust gas temperature, from which the TIT is calculated.

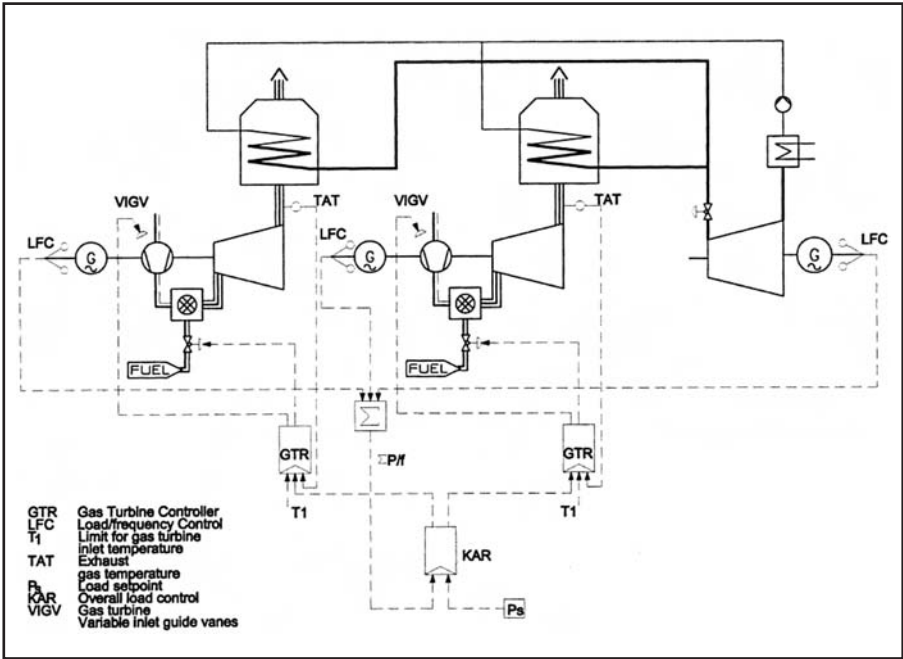


Figure 8–3 Principle diagram for a combined-cycle load control system

The entire steam cycle is operated in sliding pressure mode with fully open steam turbine valves down to approximately 50% live-steam pressure. This is the mode of operation best suited for high part-load efficiencies. More sophisticated control is not absolutely necessary

because the power output of the plant can be adjusted by changing the set points of the individual gas turbine controls.

If supplementary firing is provided, it may be beneficial to provide independent load control for the steam turbine. The steam process then operates in a manner similar to that of a conventional steam plant, where the amount of steam generated is varied to fit demand by adjusting the supplementary firing fuel input.

Up to this point, no distinction has been made between load and frequency control. In principle, the remarks remain valid for both. However, another very important aspect of the load/frequency control is the capability of a plant to react to rapid fluctuations in frequency that may occur in the electrical grid. This is known as frequency response, and must normally be done in a matter of seconds, whereas loading takes place over several minutes.

To sustain stable operation of a plant, a grid frequency dead band of typically ± 0.1 Hz is introduced within which the plant will not respond. Outside this dead band a droop setting is followed. The standard gas turbine droop setting is 5%, which means that a grid frequency drop of 5% would cause a 100% load increase. The droop characteristic setting is defined during the planning phase, and is typically in the range of 3 to 8%.

For plant configurations without steam turbine load control, the steam turbine will not be able to support falling frequencies within the 10 to 15 seconds normally required by grid codes, so the total response will have to come from the gas turbine. However, for increasing frequencies both the gas turbine and steam turbine will be able to support the grid. In such a mode the steam turbine valves will just close, allowing less steam to expand through the steam turbine.

Figure 8–4 shows a typical droop characteristic for a combined-cycle power plant with a 5% droop setting. If the frequency drops by 1%, the gas turbine load will initially jump 20% and the steam turbine load 0%. This result corresponds to a combined-cycle response of approximately 13%. On the other hand if the frequency increases by 1% the combined-cycle response of 20% is achieved by load changes of 20% in both the

gas turbine and steam turbine, illustrated in the diagram by equivalent gas turbine load change of 27%.

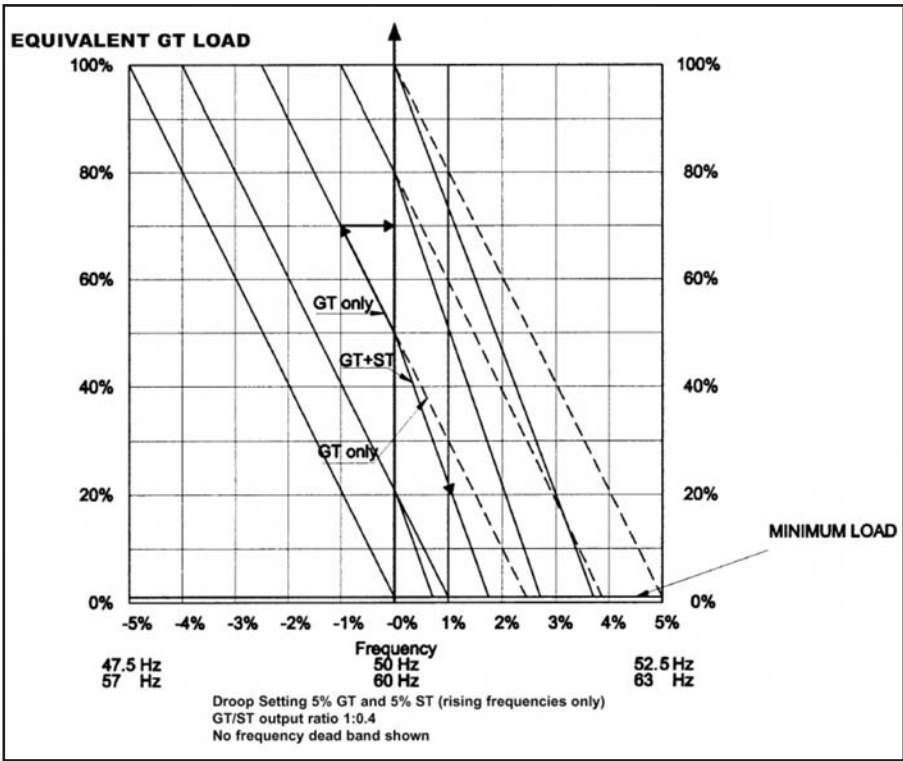


Figure 8–4 Typical combined-cycle droop characteristic of a GT load controller

To perform a plant load jump while the frequency is falling, it is essential that the gas turbine is operating below the maximum output level. Any operation, even in frequency support mode, above this level maximum is not possible. For frequency support gas turbines are typically operated between 50 and 95% load.

In conclusion, it can be said that combined-cycle plants are very well suited to rapid load changes. 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 35% are possible, but they are not recommended because they are detrimental to the life expectancy of the turbine blading.

Secondary Closed Control Loops

Figure 8–5 shows the essential closed control loops that are required to maintain safe operating conditions in a dual-pressure combined cycle. The diagram shows the input signals into each controller and the command signals to the control valves. These control loops, which are typical for all combined-cycle concepts, are described as follows.

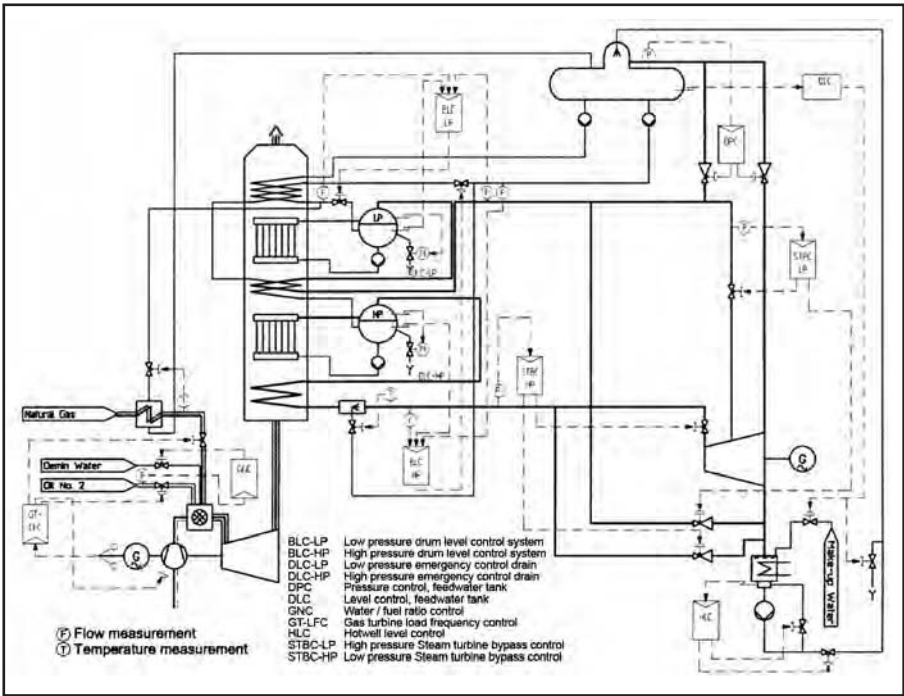


Figure 8–5 Closed control loops in a combined-cycle plant

Drum level control, BLC-HP, and BLC-LP

This is normally a three-element control system, as shown in figure 8–5 which forms one signal from the feedwater, the live-steam flows, and the drum level. This signal is then used to position the feedwater control valve.

Live-steam temperature control, STC

The live-steam temperature control loop in drum-type HRSGs actually limits rather than controls the live-steam temperature. Its purpose is to reduce the temperature peaks during off-design operating conditions, such as hot ambient temperatures, part load, and peak load. For that reason, attemperation 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 it down to the required temperature.

No real temperature control extending over a broad load range is possible in purely HRSG operation because the turbine exhaust gas temperatures drop off with cold ambient temperatures and extreme part loads of the gas turbine. In plants with supplementary firing, the relationships are more like those of a conventional steam generator. Because higher flue gas temperatures within the HRSG are possible in 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 could be divided into two sections with the attemperator installed in between them.

If the HRSG is of the once-through type, the live-steam temperature is generally controlled by the feedwater flow to the HRSG. With such an arrangement, however, attemperation is still required at extreme part load conditions and during startup.

Feedwater temperature, DPC

To avoid low temperature corrosion in the cold part of the HRSG, the feed water temperature should not, even in the lower load range, drop significantly below the acid or water dew point. On the other hand, this temperature must be as low as possible to ensure good utilization of the heat available, as discussed in chapter 5 “Combined-cycle Concepts”. It is, therefore, recommended that the feedwater temperature be held at a constant level corresponding approximately to the acid dew point.

For the control loop shown on the diagram in figure 8–5, deaerator heating is done primarily with a steam turbine extraction, but if the pressure in this extraction is not sufficient at any load point then pegging steam from a live-steam source (LP in figure 8–5) is used. In plants with a low-pressure preheater loop in the HRSG, the opposite problem could occur during part load operation of the gas turbines, where more low-pressure steam could be generated than required. This excess energy must be dissipated either by increasing the pressure in the feedwater tank/deaerator, which reduces the amount of steam generated, or by directing the excess low-pressure steam to the condenser, which again slightly increases the vacuum and, therefore, marginally lowers the steam turbine output.

For plants without a feedwater tank, the minimum feedwater temperature is normally controlled by hot water recirculation from the low-pressure economizer outlet into the HRSG feedwater inlet (see figure 5–57). To compensate the pressure losses in the economizer, a small booster pump is foreseen. The amount of water is determined by the desired feedwater temperature. For high sulfur fuels (normally backup fuels), the economizer is bypassed and the water enters directly into the low-pressure drum.

Live-steam pressure, STBC-HP and STBC-LP

Usually the steam turbine is in sliding pressure operation down to approximately 50%, so continuous control for the live-steam pressure is needless in this range. Below this the pressure is kept constant by closing the steam turbine valves. Control is, however, necessary for nonsteady-state conditions such as startup, shutdown, or malfunction.

How this control is accomplished depends on the plant equipment. The live-steam pressure is controlled by the steam turbine control valves and/or steam turbine bypass control valves.

A 100% steam bypass provides the following advantages:

- Flexible operation during startup, shutdown, turbine trip, or quick changes in load
- Shorter startup times

- Environmental acceptability (because no steam is vented to the atmosphere)
- No actuating of the HRSG safety valves in case of a steam turbine trip

Table 8–1 shows when main steam turbine valves and steam bypass valves are used for various modes of operation of a combined-cycle plant with an unfired HRSG.

Table 8–1 Operation of steam turbine control loops in a single shaft combined cycle plant

Type of operation	Steam by-pass	ST control valves
Start-up	+	+
Shut-down	+	+
Normal sliding pressure operation	–	–
Fixed pressure operation below 50%	–	+
Steam turbine switch off or trip	+	–
HRSG switch-off	+	–
Gas turbine trip	+	–
+ in operation, – not in operation		

Level in feedwater tank and condenser hotwell, DLC and HLC

These levels ensure that there is sufficient head for the feedwater and condensate pumps and the necessary water buffer in the cycle condenser. Hotwell level is controlled by adjusting the main condensate flow control valve, which will open for increasing and close for decreasing hotwell levels. If the level in the feedwater tank is too high, the drain valve in the main condensate line is opened to prevent the level rising further. If the feedwater tank level is too low, makeup water is admitted to the cycle, usually via the condenser, which in turn raises the hotwell level, and thereby the feedwater tank level.

Supplementary firing

If the plant has supplementary firing, this will usually be controlled independently to meet a required output or process steam demand by regulating the amount of fuel admitted to the burners. This is often used to control variations in process steam flows, which cannot be achieved by the combined-cycle GT load control alone. If demand falls, the supplementary firing load is normally reduced first before the gas turbine load is decreased.

Process energy

If there is a process steam extraction, it is usually controlled independently of the main process control loops as previously described. Sometimes the control may even be done externally to the combined cycle, in the plant that is receiving the process. Usually, however, the pressure must be regulated, either internally or externally to the steam turbine, or using a combination of both. Sometimes the temperature must be controlled with attemperation.

Power augmentation

The various possible solutions are described and explained in chapter 9. The control of this system is done manually from the control room, and is normally not part of the plant load controller.

9 Operating and Part-Load Behavior

The way in which a power plant responds to changes in its outside conditions (ambient conditions, power demand by the grid) is of great importance for the overall plant economy as well as for a safe and reliable operation.

In a liberalized open market, gas turbines as well as combined-cycle power plants have the unique potential to react quickly and with flexibility to changes in grid and market requirements, and can, therefore, be used as *trading tool*. It is thus important to have the following:

- A well-designed power plant with the required flexibility (e.g., short startup time, high-loading gradients, possibilities for frequency support, good part-load behavior, additional system for power augmentation, etc.)
- Good knowledge (tool) to forecast accurate potential load jumps, or calculate cost of electricity at certain part-load conditions.

Combined-Cycle Off-Design Behavior

As opposed to the plant design, in which all the components of the water/steam cycle are defined to meet certain design criteria, the performance of a plant under off-design conditions will depend

on the behavior of these fixed components with changes in the operating environment.

It is important to have precise knowledge of both the steady state and the dynamic operating behavior of the plant. Theoretical calculations of the dynamic behavior are costly and difficult. For that reason, to predict this behavior, operating experience from other similar plants or estimates (sometimes a simulator) are frequently used in combination with steady-state plant calculations.

A more exact calculation of the behavior would certainly be advantageous, but it is usually omitted due to consideration of time and cost. However, the calculations of the *steady-state operating and part-load behavior* have to be done.

Calculation basis

The calculation of the steady-state operating and part-load behavior of the steam part in a combined-cycle plant differs significantly from that of a conventional steam plant. The difference is related mainly to the boiler and the operating mode of the plant. In a heat recovery steam generator (HRSG), the heat is transferred mainly by means of convection, whereas in a conventional boiler it takes place due to radiation.

The steam turbine of a combined-cycle power plant functions most economically using sliding pressure, as shown in figure 9–1. (Sliding temperature process; that is, runs uncontrolled.)

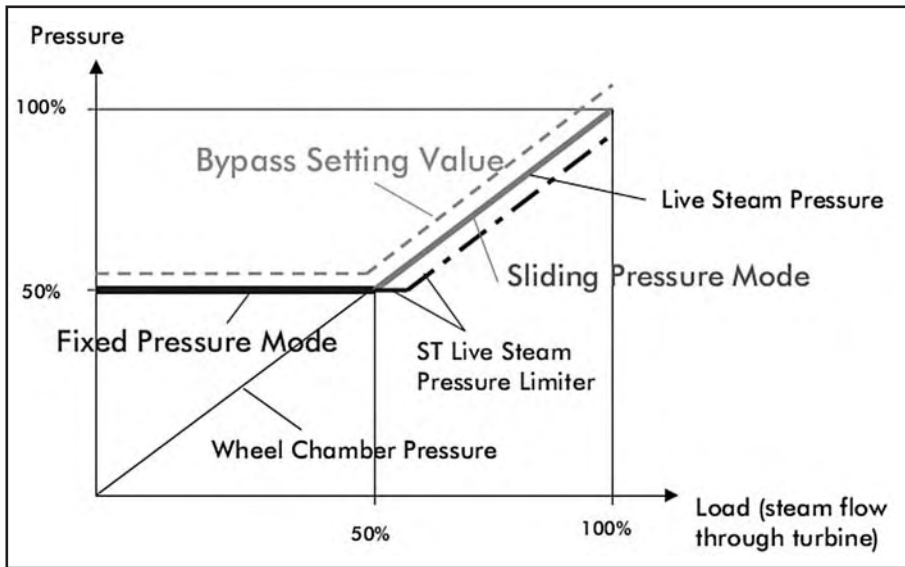


Figure 9-1 Sliding pressure diagram

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 steam plant is generally operated at a fixed pressure; that is, live-steam pressure and temperature remain constant. Calculations are thus simplified because steam pressure and steam temperature are known in advance, and the steam turbine and the boiler can, therefore, be considered independent of each other. Calculations of the gas turbines are not a problem because gas turbines are standardized machines for which correction curves (or calculation tools) are available to account for changes in ambient conditions as well as for part-load operation.

Solution approach

Calculating the operating behavior of an installation, considering the geometry of this plant, would be very time-consuming. The process can, however, be simplified by referring all values to the thermodynamic data at the design point. If the 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 this calculation method can be found in appendix A.

Combined-Cycle Off-Design Corrections

The influence of a variation of the operating conditions is generally different from the behavior shown in chapter 5. Therefore, the variations of the design conditions generally lead to a different design of the components of the water/steam cycle (exceptions are standardized plants). In the case of off-design operation, the actual geometry of the components, such as HRSG, steam turbine, cooling system, remains unchanged.

Operating behavior will differ from plant to plant, depending on the actual design point. In the following paragraphs, some typical variations are shown. These parameters should normally be considered when correcting the combined-cycle performance from one set of operating conditions to another (e.g., for combined-cycle performance testing):

- Plant load
- Ambient air temperature
- Ambient air pressure
- Ambient relative humidity
- Cooling water temperature (only in case of fresh-water cooling)
- Frequency
- Power factor and voltage of the generators
- Process energy extraction (only for cogeneration application)
- Fuel type and quality

Effect of ambient air temperature

At the design point, ambient air temperature has a large influence on the power output of the gas turbine and the combined-cycle plant (see chapter 5). Operating behavior of the plant in off-design conditions with changes in ambient air temperature is very similar to the behavior for different design points.

In particular, efficiency increases slightly if the air temperature increases (under the assumption that the vacuum within the condenser remains constant, as shown in figure 5–6). However, this effect is marginal and mainly affects the pinch point in the HRSG and condenser. These changes in performance are so slight and hardly detectable on the curve compared to the design influence discussed in chapter 5.

In case of combined-cycle plants with direct air cooling (e.g., with air-cooled condenser) or a wet cooling system, the ambient temperature will also have an influence on the condenser pressure and, consequently, on the steam turbine output and combined-cycle efficiency. Such a correction curve is plant specific (indicative values can be taken from figure 5–11 and figure 9–2.)

Effect of ambient pressure

The main factor influencing the ambient pressure is the site elevation, which is purely a design issue, as described in chapter 5. For a given site the power plant may see daily weather variations, which are the only causes of change in the ambient pressure. These corrections are basically the same as described in chapter 5. The correction will only affect plant output, and the efficiency will remain practically constant.

Effect of ambient relative humidity

The effect of changes in the ambient relative humidity is also similar to the one described in chapter 5. An increase in relative humidity increases the enthalpy of the working media of the cycle as well as the density of the air into the gas turbine. Compared to the design point it enhances the energy to the HRSG, causing slightly higher energy transfer through the HRSG sections. This leads to a marginal increase in

the pinch point of the HRSG to transfer this additional energy, resulting in a slight negative tendency for off-design calculations.

In the case of combined-cycle plants with a wet cooling system the relative humidity will also have a larger influence on the condenser cooling water inlet temperature and, consequently, on the condenser pressure, steam turbine output, and combined-cycle efficiency. Such a correction curve is plant specific (indicative values can be taken from figure 5–11 and figure 9–2.)

Cooling water temperature

The cooling water temperature has a major impact on the efficiency of the thermal cycle, directly affects the condenser pressure, and, consequently, affects the enthalpy drop in the steam turbine and the steam turbine output. At a given steam turbine exhaust area (selected at the design point) a change in the cooling water temperature affects with the corresponding condenser pressure the volume flow of the steam turbine exhaust, the steam velocity in the steam turbine exhaust part, and, consequently, the exhaust losses. Deviating from the design point the exhaust steam volume flow will be different as well as the exhaust losses. Generally exhaust losses are increased if the cooling temperature (condenser pressure) falls, so the benefit due to a better condenser vacuum is reduced. If the cooling water temperature is higher, the condenser pressure increases, thereby reducing the steam turbine output. The operating behavior is thus quite different from the one to arise if the size of the turbine was in all cases adapted to the temperature of the cooling water (identical starting point in the exhaust losses curve), as it is the case in chapter 5.

Figure 9–2 shows the effect of the condenser vacuum (steam turbine backpressure) on the relative efficiency of the combined-cycle plant for a typical direct water cooling (and wet cooling tower with reduced steam turbine exhaust area) application. It is a plant-specific curve, based on a given steam turbine with a fixed exhaust area (designed for a certain vacuum). The curve is, therefore, only generally applicable for plants with identical design conditions. To determine the condenser backpressure of different cooling media temperature, figure 5–11 is valid as long as the plant runs at full load.

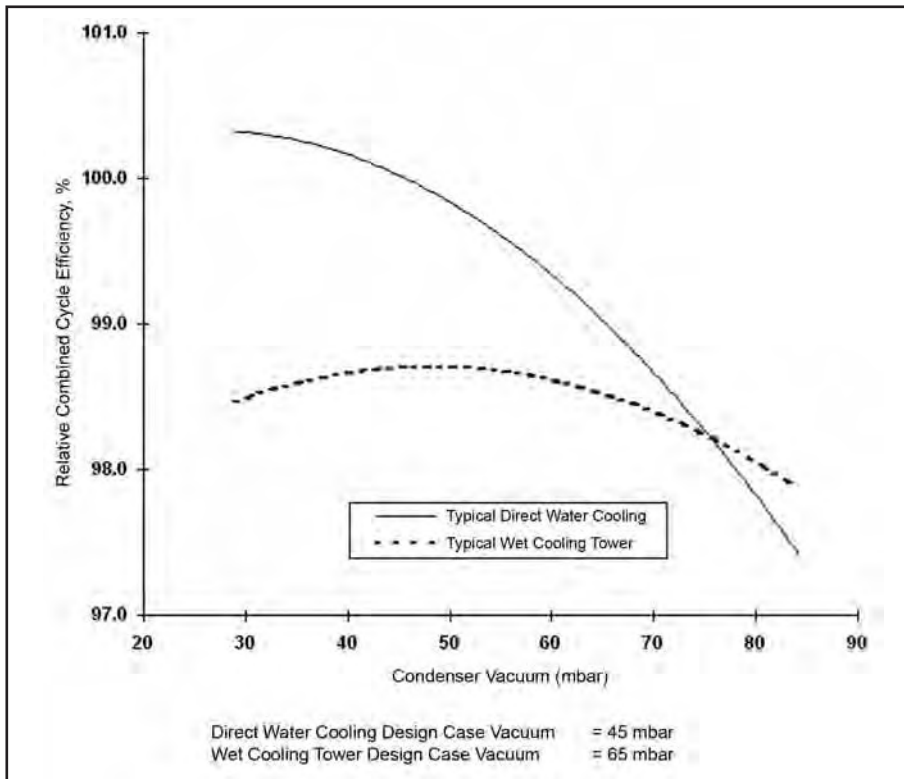


Figure 9-2 Effect of condenser vacuum on combined-cycle efficiency

For part-load conditions, less steam is produced in the HRSG, consequently reducing the amount of steam leaving the steam turbine exhaust into the condenser. If the cooling water flow is maintained for full as well as part loads, the vacuum is reduced even further for part loads due to the lower heating of the cooling media and the smaller pinch point in the condenser.

Electrical corrections

Frequency. The grid frequency has a major impact on plant behavior because it determines generator speed and, consequently, gas turbine speed. The gas turbine compressor speed defines the airflow entering the gas turbine, which is significant for the plant performance. Gas turbines are normally designed to operate at nominal firing

temperatures for frequencies from 47.5 to 52.5 Hz for a 50 Hz grid, and 57 to 63 for a 60 Hz grid. The same design criteria are valid for steam turbines.

Figure 9–3 shows a typical variation of combined-cycle output and efficiency for frequency variations. The output decreases for lower frequencies and the efficiency stays within a narrow range of the nominal frequency point.

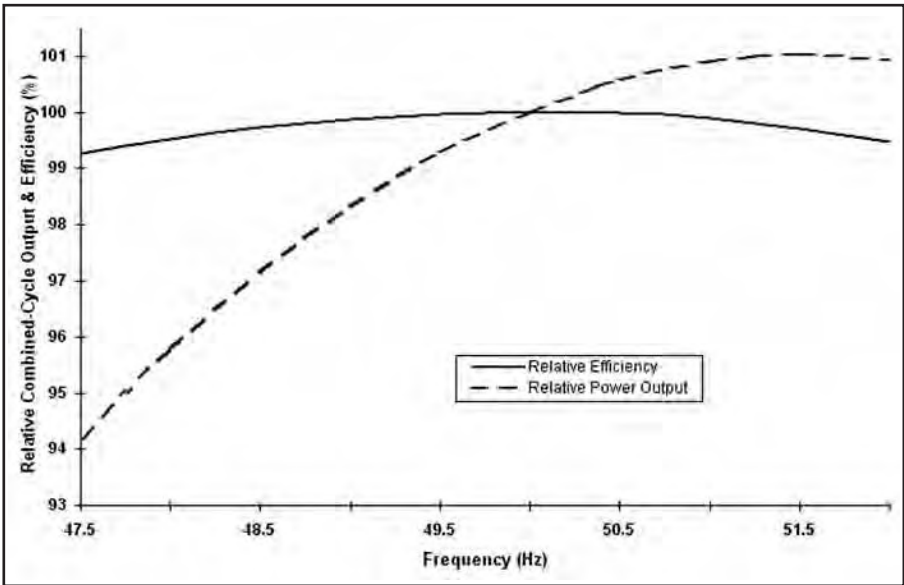


Figure 9–3 Effect of frequency on relative combined-cycle output and efficiency for full-load operation

Power factor. The plant power factors are dictated by the grid and influence the maximum generator capability as well as the generator efficiency, which in turn affects the output at the generator terminals. A normal power factor is in the range of 0.85 to 0.95. At the nominal point of the generator, a change in power factor from 0.8 to 1 would improve the generator efficiency by 0.3 to 0.4%. For lower loads of the generator the difference tends asymptotically towards zero.

Process energy. Plants with process heat extractions (cogeneration plants) are often highly customized, and universally valid off-design behavior cannot be given. The extracted process energy has a major effect on plant performance, making a plant-specific curve necessary (or calculation with an off-design heat-balance simulation program).

Fuel type and quality

The main off-design influence of the fuel on cycle performance occurs when a backup fuel is fired (e.g., oil instead of natural gas). The reason for this lies in the fuel composition and possibly the need for water or steam injection to meet local emission requirements.

Variations in the composition for the same type of fuel also influence the plant performance because a different fuel composition results in a different chemical composition after combustion.

Fuel components will also determine the lower heating value (LHV) of the fuel. If the LHV decreases, the fuel mass flow (heat input divided by LHV) increases to provide the same heat input to the gas turbine. This again results in increased flow through the turbine part of the gas turbine, which has a positive impact on output (provided the fuel has not first to be compressed in the plant).

If the chemical impact of the combustion products drags the performance in the opposite direction to the LHV influence, however, the total influence could be different. The normal tendency is, though, positive. It is, therefore, not possible to show an impact on a variety of fuels. These influences should be treated on a project specific basis, where the actual fuel compositions are known. A typical correction for gas fuel is shown in figure 9–4.

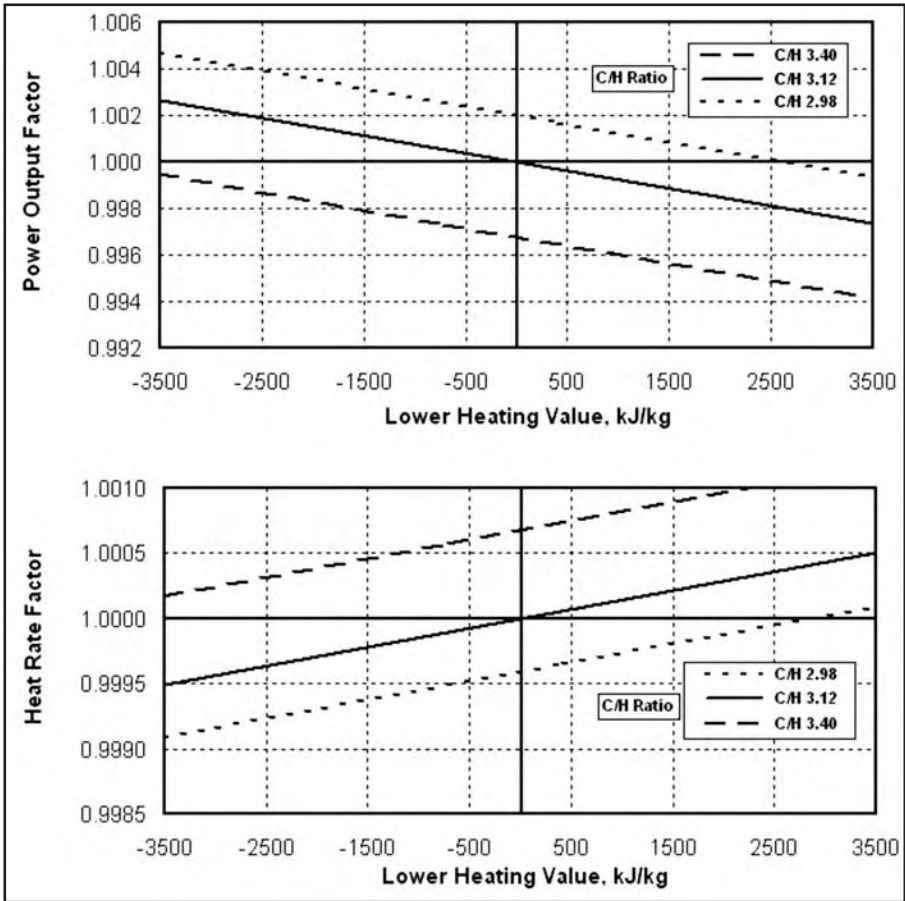


Figure 9-4 Effect of fuel composition and lower heating value on combined-cycle output and efficiency (base-load, gas operation with wet cooling tower)

Part-Load Behavior

In combined-cycle plants without supplementary firing, efficiency depends mainly on the gas turbine efficiency, the load of the gas turbine(s), exhaust gas temperature, and plant size. So far, all of the corrections discussed have been related to a plant performance with gas turbines running at full load (with nominal firing temperature).

Figure 9–5 shows the part-load efficiency of a combined-cycle plant and the associated gas turbine, each relative to the 100% load case. At higher loads the part-load efficiency is acceptable down to an approximately 50% load due to the following reasons.

The gas turbine used is equipped with one or even more rows of compressor variable inlet guide vanes, keeping the efficiency of the combined-cycle plant almost constant down to approx. 80 to 85% load. This is because a high exhaust gas temperature can be maintained as the air mass flow is reduced. Below that level, the inlet temperature (TIT) must be reduced, leading to an increasingly fast reduction of efficiencies.

Also, the steam turbine is calculated with sliding pressure mode down to about 50% load, also providing good utilization of the exhaust gas in this range. Below that point, the live-steam pressure is held constant by means of the steam turbine inlet valves, resulting in throttling losses and increasing stack losses.

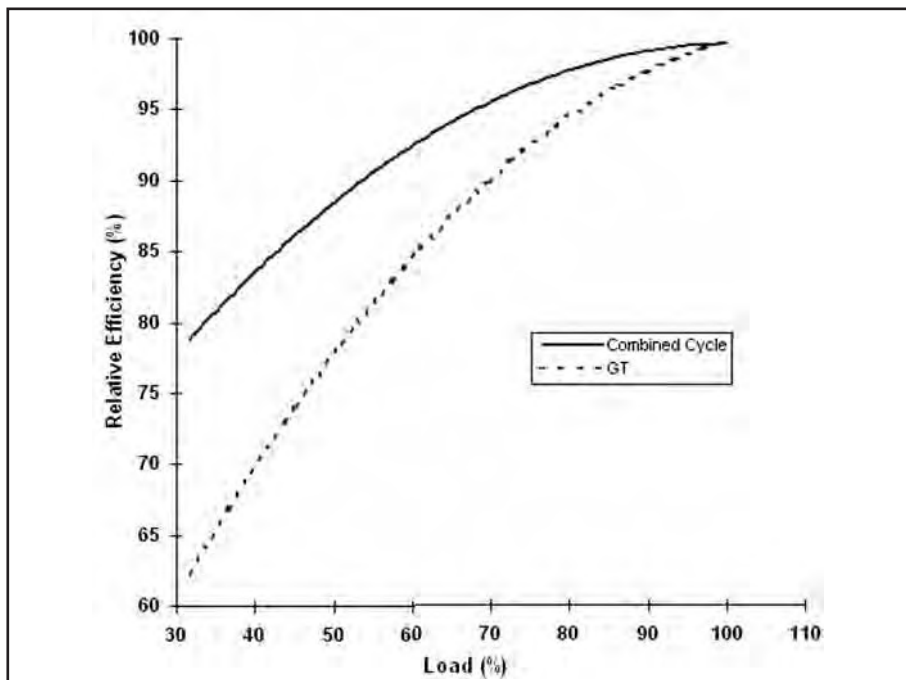


Figure 9–5 Part-load efficiency of gas turbine and combined cycle

Reheat gas turbines have a different (better) part-load behavior mainly at lower loads.

For full load operation the gas turbine accounts for two-thirds of the power output, and the steam turbine for one-third. Figure 9–6 shows how the ratio of steam turbine to gas turbine power output (PST/PGT) shifts towards more steam turbine output at part loads.

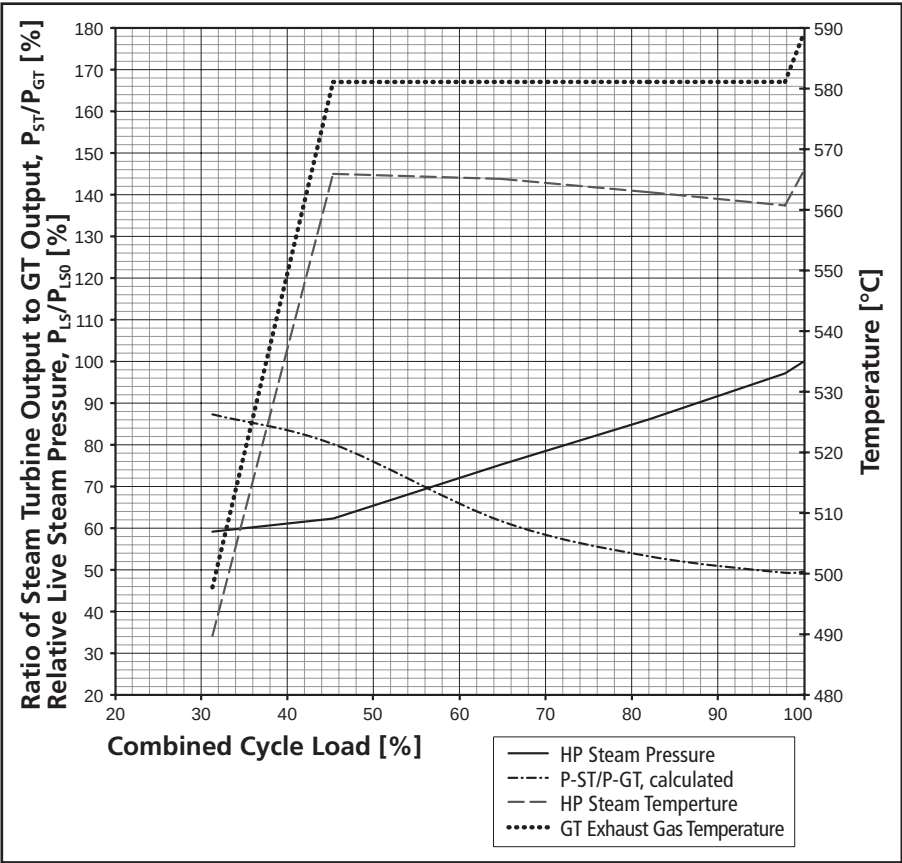


Figure 9–6 Ratio of steam turbine and gas turbine output and live-steam data of a combined-cycle plant at part load

Additionally, the live-steam temperature and relative live-steam pressure of the water/steam cycle are shown. The live-steam temperature is kept constant by means of attemperation at the HRSG superheater

outlet. The reason for the flat temperature profile lies in the variable inlet guide vane control, which allows the gas turbine to operate at a lower flow with the nominal TIT. The live-steam pressure drops down to 50% of the full-load live-steam pressure and is then controlled by the steam turbine valves.

There are several site-specific possibilities for further part load efficiency improvements, such as:

- Several gas turbines in the plant configuration
- Air preheating for sites with cold ambient air temperatures

A combined-cycle plant with several gas turbines is operated differently at part load to keep the efficiency at highest possible level. For 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.

With this mode of operation, the efficiency at 75%, 50%, and 25% load is slightly lower than at full load. If, however, four independent single-shaft combined-cycle blocks are selected, the part-load efficiencies are as shown in figure 9–7. In this case, the full-load efficiency would be achieved at the points 100, 75, 50, and 25% load because at these points the individual steam turbines are also running at full load.

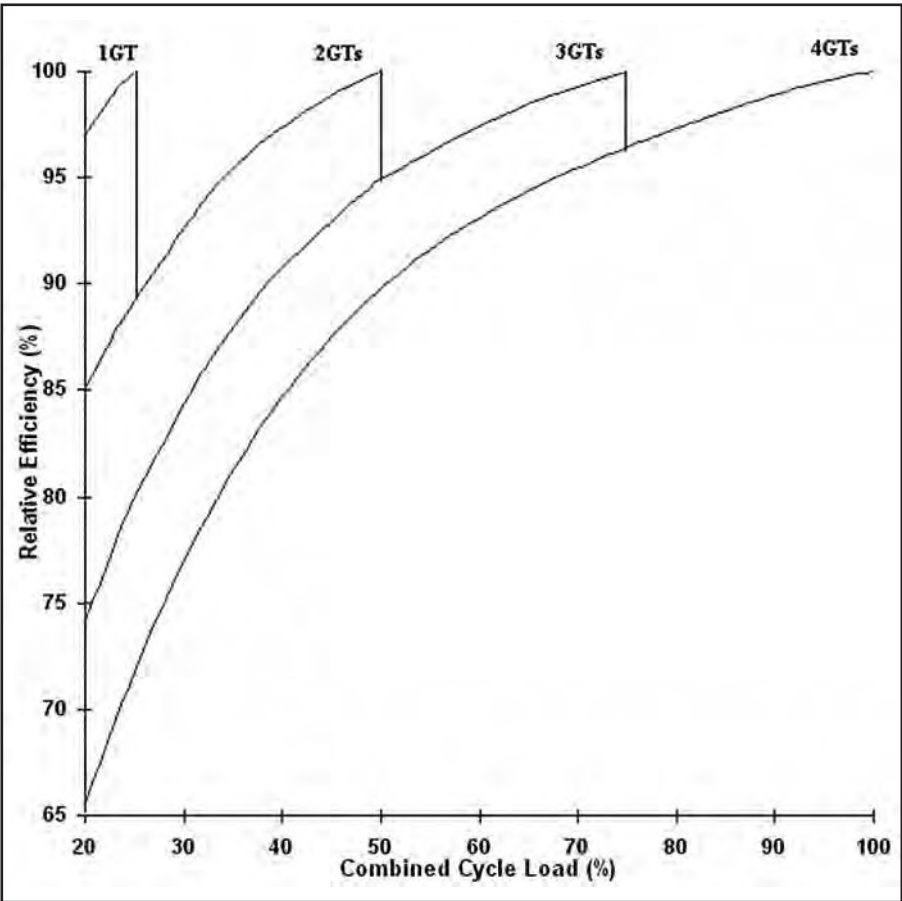


Figure 9-7 Part-load efficiency of combined-cycle plant with four single-shaft blocks

Combined-Cycle Testing Procedures

Plant performance guarantees are given when a power plant is sold. These guarantees will apply to a set of performance parameters and ambient conditions that cannot normally be recreated for the performance test. Therefore, to demonstrate that the guaranteed values have been met, plant performance must be measured under actual site

conditions, and the results must be corrected with a series of corrections for the parameters described previously. These corrected values can then be compared with the guaranteed values.

In a combined-cycle plant, the gas turbine, HRSG, and steam turbine all interact with each other. If the contract for the plant goes to a single general contractor on a turnkey basis or to a single contractor for the power-island (including all main equipment such as gas turbine, boiler, steam turbine, generator, condenser, etc.), the power output and efficiency of the plant as a whole can be guaranteed. For combined-cycle plants, it is much easier and linked with fewer uncertainties to measure the values for overall plant performance instead of each major component individually. The amount of exhaust heat supplied to the HRSG (by the gas turbine in particular) cannot be measured accurately. When overall plant values are guaranteed, the fuel flow, lower heating value (with fuel samples), electrical output, and ambient conditions of the power plant must be measured. These are quantities that can be determined with relative accuracy. Nevertheless, a certain overall measurement uncertainty will remain; in case of a gas-fired combined-cycle power plant, about $\pm 0.5\%$ in overall power output and $\pm 0.8\%$ in net efficiency.

Thereafter, a correction factor is determined for each parameter, quantifying its influence on actual performance because each parameter is not at design/guarantee value. These correction factors are multiplied to give values, which can be directly compared with the guarantees.

The power output of the gas turbine and the steam turbine are often corrected 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, and so on.
- The power output measured for the steam turbine is corrected by using curves that show the indirect effects of air temperature, air pressure, and gas turbine speed on the steam process as well as the direct effect of the cooling water temperature (or air temperature and humidity).

To calculate these curves, it is best to use a computer model that simulates the steam process as a whole (as previously described in this chapter). Changes in ambient air data produce changes in the gas turbine exhaust data, and these affect the power output of the steam turbine.

The advantage of this procedure is that it can, with certain modifications, be used even if the gas turbine is put into operation at a somewhat earlier date than the steam turbine (phased construction). The method is, however, rather complicated and requires quite a few corrections to cover the interactions between the gas turbine(s) and steam turbine. Therefore, especially for single-shaft combined cycles and for plants without phased construction, there is a trend towards overall combined-cycle corrections.

For steam turbine plants and gas turbines and combined-cycle power plants the methods used for corrections are described in international standards such as ASME PTC 22 and 46 or ISO 2314.

Correction with a heat balance computer model: In cases of more complex combined-cycle plants (e.g., with district heating or process steam extractions), the correction procedures with correction curve for each and every parameter would be too complex. Therefore, it is easier and more accurate to make the correction from test results to guarantee condition (or vice versa) with a heat balance calculation program. Principles of correction are shown in figure 9–8. The correction from the measured condition back to guarantee condition (3 → 4) has an advantage in case of a weighted guarantee, resulting out of different individual guarantee points. Of course, the calculation model must be tested and jointly agreed by all parties prior to testing.

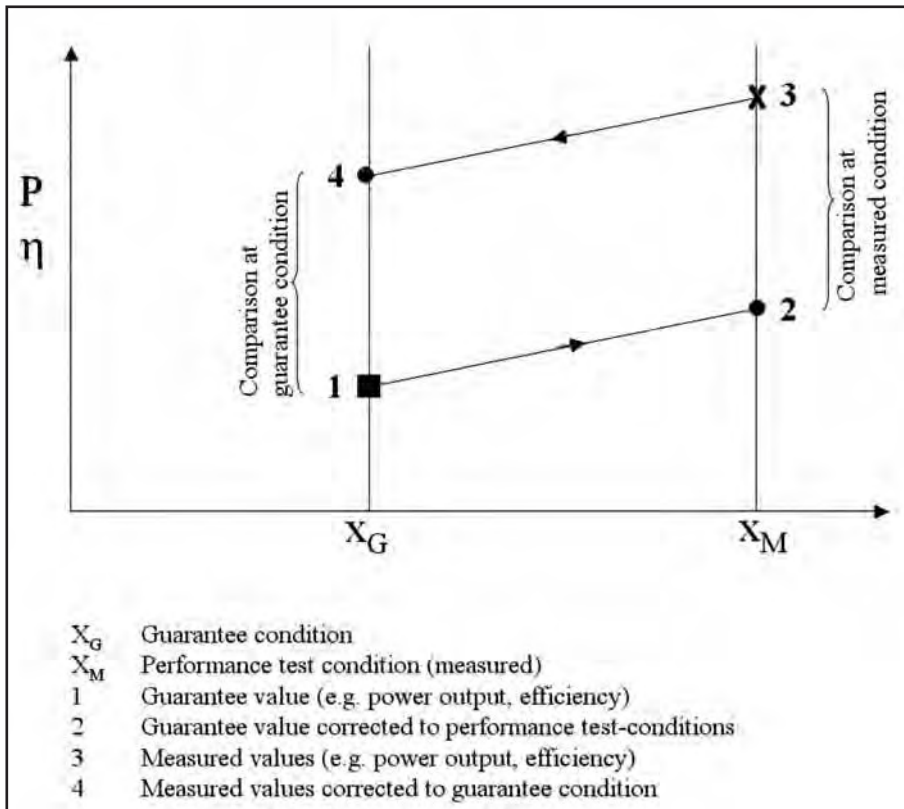


Figure 9-8 Performance guarantee comparison

Combined-Cycle Power Plant Degradation

Performance predictions or even performance guarantees are required for long-term economical evaluation to be given after a certain number of operating hours or years of operation. After commissioning at the time of the plant performance test, the plant is normally said in “new and clean condition”. For operating hours beyond this point, the power plant performance value will slightly decrease as function of time due to degradation, which is mainly caused by the gas turbine

and partially recuperated in the steam process. Typical values are given in chapter 7. Plant performance degrades from new and clean values because of many factors, including GT compressor fouling, increased leakages, and airfoil surface finish changes. Plant operation can also have a significant impact on performance. Off-line and on-line compressor washes, inlet filter replacements, frequency of start/stops, plant chemistry, readjustment of instruments, and controller setting values can be major factors.

Figure 9–9 indicates a typical degradation on power output and efficiency for a large combined-cycle plant with natural gas firing after GT compressor off-line washing and with a state-of-the-art maintenance and operation of the entire plant. The last one can be achieved by an operation and maintenance contract with the plant (GT) supplier, with clearly defined guarantees for degradation as well as for availability.

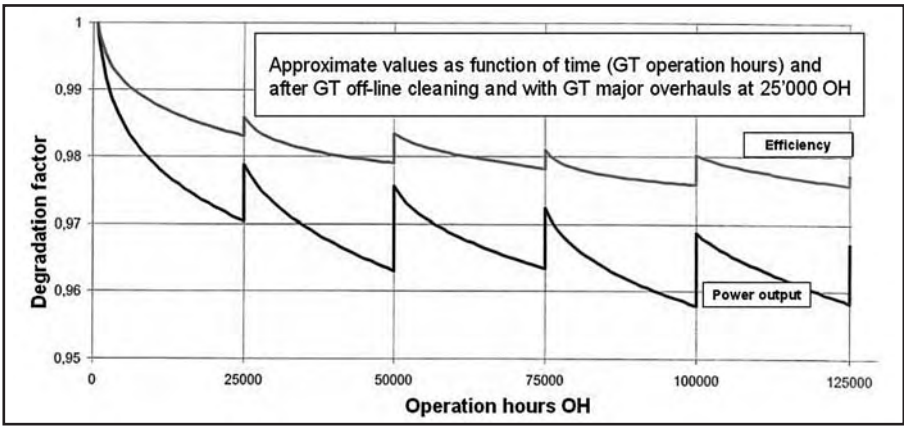


Figure 9–9 Expected non-recoverable combined-cycle power plant degradation of power output and efficiency with GT operating on clean fuels

As already explained in chapter 7, the degradation factor is a function of various elements. A few of them can be optimized (e.g., compressor washing, exchange of GT air intake filter, etc.) and kept as low as possible. To visualize the performance degradation over the plant operation time, a performance monitoring system in the control room can record specific values, and correct these data to reference conditions (see figure 9–10).

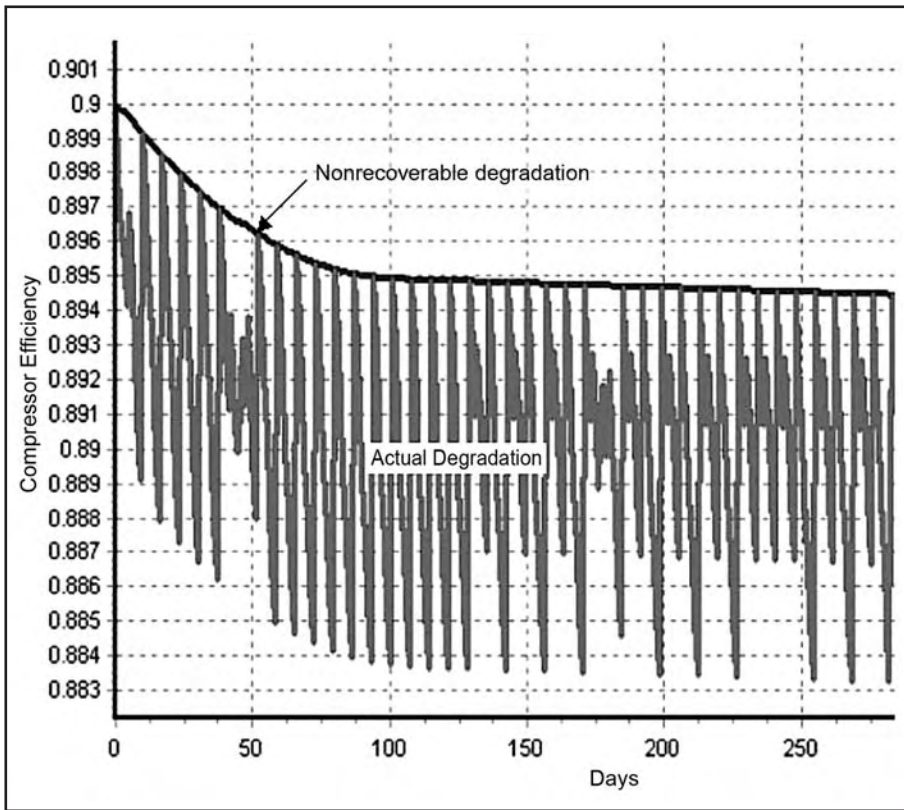


Figure 9–10 Gas turbine compressor efficiency as function of days operation (or operation hours)

Figure 9–10 clearly shows that the real average power output and efficiency at guaranteed condition over a defined interval (e.g., between PAC and first GT overhaul) is the base for an economical calculation, is not easy to forecast, and will, in most of the cases, not be guaranteed.

For a well-maintained and operated combined-cycle power plant the real figures are about 3 to 4% for the average power output, and about 2 to 3% below the values for efficiency according the curve shown in figure 9–9.

Plant performance monitoring

Such an on-line power plant management software tool can assist the local operation and maintenance team in maximizing plant performance (minimize permanent degradation losses) and consequently owners' operating income. The system's primary duty is to correct and compare the current plant performance values with the base line plant (mostly new and clean, or a predefined degradation curve) at guaranteed conditions, with a correction procedure as described under combined-cycle off-design corrections. By means of such a tool, the operator gets a clear and early indication about deviation to target performance, and he can react with improvement measures (e.g., off-line, on-line washing of compressor, GT air intake filter replacement, calibrate instruments etc.).

Power Output Enhancement- Power Augmentation

After the selection of the combined-cycle power plant concept has been done, other factors are addressed. These factors include:

- Is there a need for peak power production with premium paid for the resulting power?
- Does peak power demand occur on hot days (summer peaking) only?
- Is there a need to compensate the power reduction continuously during summer period?
- Is frequency support required?

All these questions must be checked individually. The goal of power augmentation basically is to increase the overall net power output of a combined-cycle power plant for a limited time (i.e., during midday peak with high ambient air temperature) with possibly some concessions regarding efficiency.

There are different possible solutions for power augmentation applications, such as:

- Supplementary firing in boiler
- Steam/water injection into the gas turbine
- GT peak load firing
- GT air inlet cooling

GT air inlet cooling is mainly for larger combined-cycle power plants a realistic economical option and use the effect, that with lowering of the air inlet temperature the power output will be increased (see correction curve power output as function of the air temperature in chapter 5).

The control of this system is usually done manually from the control room, but today, mainly in cases where these systems are used for load jumps, they are integrated in the overall plant load controller.

GT air inlet cooling

Because gas turbines are constant-volume-flow engines, they are very sensitive to changing ambient temperature and pressure. This reduction of plant performance at high ambient temperature is particularly unfortunate because of the retail price for electricity that usually rises rapidly with higher ambient temperatures. The resulting mismatch of demand and supply does have a negative impact on the profitability of combined-cycle power plants. Thus, a variety of options have been developed to particularly account for this problem, such as:

- Evaporative cooling
- Fogging
- Evaporation compressor cooling (over/high fogging)
- Chiller

An overview of the different air inlet cooling systems is given in table 9–1.

Table 9–1 Overview of GT air inlet cooling systems

	(Demin) water consumpt.	Power increase	Air humidity limitation	Net efficiency reduction	Spec. Investment cost
Evap. Cooling	X	X	XXX	X	X
Fogging	X	X	XXX	X	X
High-/over fogging	XX	XX	X	X	XX
Chiller	N.A.	XX	N.A.	XX	XXX

X low XX mid XXX high

Evaporative cooling. The aim of this system is to cool down the air entering the gas turbine by evaporation of water, which also increases the mass flow. The evaporative cooler increases humidity close to saturation. The best result in power increase can be achieved if the air inlet temperature is high and humidity is low.

An overview of cooling potential at different locations is shown in table 9–2.

Table 9–2 Max ambient temperature and corresponding cooling potentials at different locations

Location	Germany	Gulf-region (seashore)	Saudi-Arabia (mainland)	Tropical Region	South USA
Max air temperature of °C	38–40	46	55	36–38	44
Cooling Potential K	14–16	14–15	26–28	10–12	18

The amount of evaporated water depends on ambient air temperature and humidity and is, in case of ambient temperature of 40°C and 50% humidity with a typical evaporative cooler efficiency of 85%, about 0.4% of the gas turbine inlet air mass flow.

The corresponding relative power increase of a combined cycle is shown in figure 9–11.

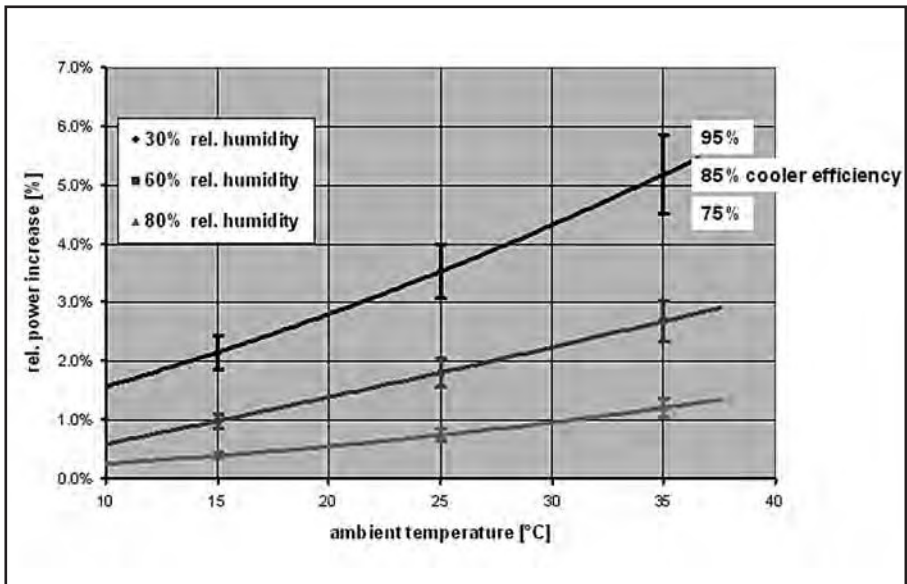


Figure 9–11 Relative power increase of a combined-cycle power plant as function of ambient temperature and humidity

For rough estimations the following could be applied: 1°C temperature decrease corresponds to a combine-cycle power increase of about +0.4 to 0.5%; the overall efficiency remains more or less the same.

An evaporative cooler solution does only make sense at locations with humidity below 70 to 80%. At air temperature levels lower than 10°C the cooling is usually not anymore in operation.

The system consists of an evaporative cooler (cellulose or fibreglass) and a droplet separator producing an additional pressure drop of about 1.5 to 3 mbar in the air intake system.

Fogging system. The principles are similar to the evaporative cooler. That means, the injected water (demineralized water) downstream of the air intake filter (see figure 9–12) cool down the inlet air with an improved cooling efficiency (up to 95% compared to about 85%). The injected water (small water droplets) evaporates and the air is cooled down close to saturation. In contrast to evaporative coolers, fogging systems have a negligible pressure drop and are ideal for retrofitting.

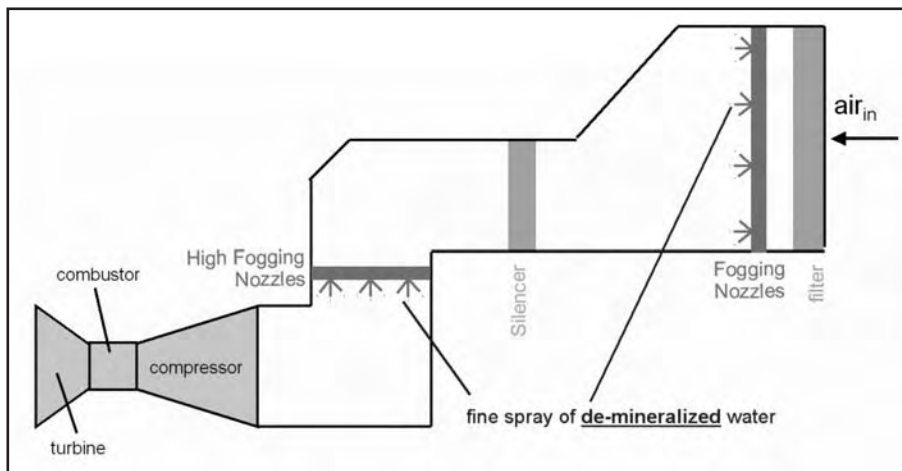


Figure 9–12 Typical arrangement of a fogging system in the GT air intake

High-fogging system. To further increase power augmentation, an additional nozzle rack is installed near the gas turbine compressor intake. These systems have different names such as high fogging, wet compression, over spray, or over-fogging system, and are usually in operation together with a fogging or evaporation cooling system.

The high-fogging system sprays small water droplets ($<50\mu\text{m}$) through nozzles into the air stream. These droplets evaporate mainly inside the compressor as the air is heated up during compression. Two effects mainly increase the power of the gas turbine:

- The air is intercooled in the compressor, which reduces the compression work and the compressor exit temperature.
- The mass flow through the turbine is increased.

The increase of power with fogging or evaporative cooling depends on ambient conditions. The high-fogging power increase is nearly independent of ambient humidity and temperature.

The total water mass flow capacity of a high-fogging system is gas turbine-specific and currently about 1.2% of the air intake mass flow at ISO condition, which results in a gas turbine power output increase of about 8 to 10%; combined cycle, about +5 to 7%. At high ambient temperature the total amount of water for air inlet cooling can go up

to 2% of the air intake mass flow. The overall combined-cycle plant efficiency will be reduced by about -1.5% in case of a spray water amount of 2%.

To avoid erosion corrosion problems within the first part of the GT compressor, a well-designed and tested water spray system is mandatory to secure a uniform and fine water atomization into the GT air intake system. Additionally the first few rows in the GT compressor may need some modification for this kind of operation.

Chiller. Refrigerated air cooling is one of the most effective ways to increase the power of a gas turbine during high ambient temperatures. Chillers, unlike evaporative coolers, are not limited by the ambient temperature, and do not need additional water, which can be an important factor at dry area locations during the summer season.

The gas turbine inlet air can be cooled down to a fixed value ($>10^{\circ}\text{C}$), which can be lower than the wet bulb temperature.

As shown in figure 9–13 the power consumption for the chiller unit increases drastically for cooling beyond the line when the air saturation line is reached (amount of work Δh_2 to cool down air from 23°C to 10°C is much higher than Δh_1 for cooling from 35°C down to 23°C).

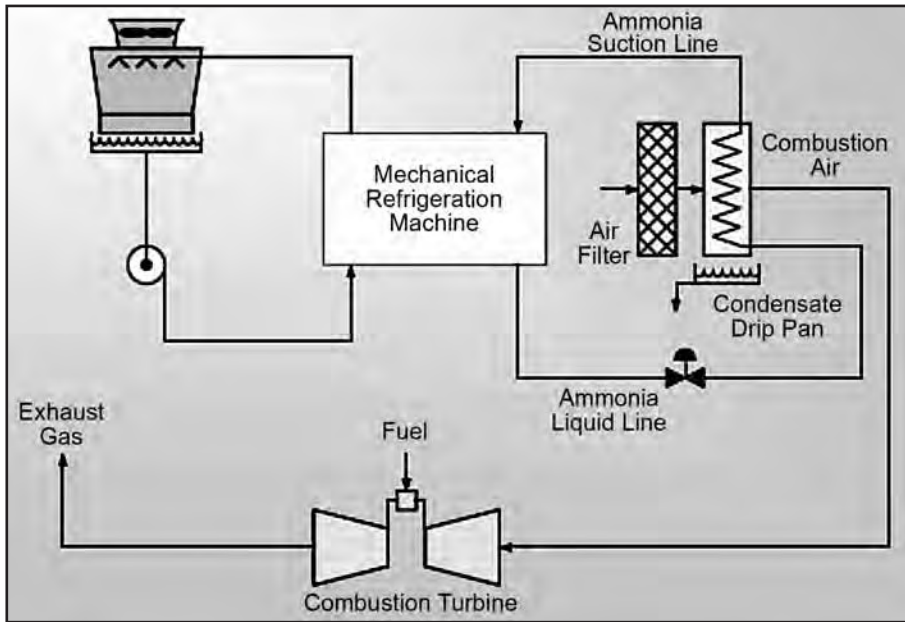


Figure 9–14 Typical diagram of a chiller system

The cooling of air can be provided either continuously or by using a thermal storage. Continuous cooling makes sense in cases where power enhancement is required for more than six to eight hours a day. A system with thermal storage is the better solution in all other cases. With such a thermal energy storage (TES) system, a cold reserve (in form of ice) can be built during the nonpeak hours and then be used for power increase during the few peak hours.

A typical example of a power improvement in a combined-cycle power plant with a chiller (without a TES design) is shown in table 9–3.

Table 9–3 Typical example of combined cycle power improvement with chiller

		Without chiller	With chiller
Design ambient temperature	°C	35	35
Design wet bulb temperature	°C	26	26
Design ambient RH	%	50	50
GT inlet air temperature	°C	35	10
Total net power output w/o chiller	MW	356	401
Parasitic power from chiller	MW		–13
Total net power output	MW	356	388
Net power increase	MW		32
% change	%		9.0%

The parasitic loads are in the range of about 20 to 30% of the power gain. In the example shown in table 9–3, the load is almost 30% because of the quite low cooling air temperature.

Summary

All systems, except the solution with chiller, need additional high-quality water (demineralized water) and are limited by the amount of moisture present in the air. Therefore, the application of these solutions is limited for sites with an average relative humidity of about 70 to 80%. The control of the different systems is done as follows:

- Evaporative cooler: No control of the evaporated water (only on-off possible)
- Fogging system: Air inlet temperature at GT compressor inlet can be controlled by the amount of injected water (with certain limits)
- Chiller system: Inlet air is cooled down to a fixed set point by a closed loop control (minimum temperature 10 to 13°C)

The high- or over-fogging (wet compression) solution has the largest potential for an increased power output by 10% (up to 20%). With this

system special attention has to be given to a well-designed spray system (optimized droplet spectrum), including an adequate integration into the plant control system, as well as to the GT compressor (first 5 to 6 stages), to avoid extensive erosion/corrosion problems.

The specific costs of these additional systems are quite different. Table 9–4 gives a rough estimation, but as previously explained, mainly for the chiller system, various figures can heavily influence these costs.

Table 9–4 Specific investment costs for additional output of air inlet cooling system

		Spec. Investment Cost
Fogging System	US\$/kW	25–50
Evaporative Cooling System	US\$/kW	40–70
High Fogging System	US\$/kW	120–160
Chilling System (w/o TES)	US\$/kW	250–350

Startup and Shutdown of a Combined-Cycle Power Plant

Combined-cycle power plants are usually operated automatically. Therefore, during startup and shutdown it must be possible to activate equipment from the central control room. Whether the commands are to be issued to the individual drives, drive groups by the operating staff, from a higher level automatic starting program must be decided on a case-by-case basis. In base-load installations operating with few starts, full automation of the steam process is not always necessary.

The dynamic behavior of modern combined-cycle plants is characterized by the short startup time and quick load change capability. Above all, the gas turbine can be started and loaded quickly. Because its reaction time is short, it is capable of following quick changes and surges in load.

Modern combined-cycle plants in the 80 to 400 MW range can be started within the following times:

- Hot start after 8 hours standstill: 40 to 60 minutes
- Warm start after 48 hours standstill: 80 to 120 minutes
- Cold start after 120 hours standstill: 120 to 170 minutes

Larger figures correspond to 250 to 400 MW combined-cycle plants (one-to-one configuration)

Gas turbine startup is basically independent of its standstill time, allowing two-thirds of the combined-cycle power to be available within 30 to 40 minutes after activating the startup sequence. The startup time for the steam turbine depends on the time required to heat up parts of the machine (i.e., rotor) without exceeding thermal stress limits imposed by the material. Therefore, the stand-still time is very significant because it determines the temperature of these parts when the startup sequence is initiated. This temperature is the base for the allowable loading gradient of the steam turbine and the HRSG.

To increase the flexibility during plant startup a stress controller for the steam turbine (TSC) can allow different startup modes for the ST. It can be preselected by the operator, following grid requirements. Thermal stress and, as a result, lifetime consumption of the ST is cumulated, monitored, and indicate the remaining lifetime of the ST.

The operator has the choice between FAST, NORMAL and ECONOMIC loading modes. The FAST startup mode (maximum 800 ST starts) reduces the ST startup time, whereas the ECONOMIC mode (maximum 8,000 ST starts) saves ST lifetime consumption. The NORMAL loading is equal to standard cycling duties (maximum 2200 starts during ST life).

Starting procedure

To prevent explosion of any unburned hydrocarbons left in the system from earlier operation, it may be advisable—eventually mandatory by NFPA boiler code, especially when firing oil—to purge the HRSG before igniting the gas turbine. This is done by running the gas turbine at ignition speed (approximately 20 to 30% of nominal speed) with the generator as a motor, or with similar starting equipment, to blow air through the HRSG. The purge time (approximately five minutes)

depends on the volume behind the gas turbine (e.g., bypass stack or HRSG), which has to be exchanged by up to a factor of five with clean air before gas turbine ignition can take place.

Plant startup begins after purging with the ignition of the gas turbine, running up to nominal speed, synchronized, and loading to the desired load.

An overview of expected values for unit startup times is given in table 9–5.

Table 9–5 Expected start-up times for a 400 MW combined cycle plant

Start-up condition	Expected values for unit start-up times*	ST operating mode
Cold start-up (plant shutdown for 120 h)	155 min.	FAST
Warm start-up (plant shutdown for 48h, over weekend)	105 min.	NORMAL
Hot start-up (plant shutdown for 8h, over night)	47 min. (31 min.)	ECONOMIC (fast)

*Plant start-up time begins with GT ignition and is completed with GT at IGV base load position and HP and IP/HRH bypass station closed.

The plant start-up time does not consider any pre-start preparations like filling of the water / steam cycle and HRSG, cold purging of the HRSG or waiting time for sufficient steam chemistry. Further it is assumed, that the plant was shut down without any disturbances and the steam turbine follows the regular cool down behavior (i.e. no forced cooling).

All systems of the plant have to run smoothly without upsets. The plant start-up time is achieved due to the high level of automation.

The operating regime is based on the optimized use of the turbine stress controller. NORMAL operation mode of the ST is determined by the design load regime as requested by the customer. The possible operating regime for FAST/ECONOMIC ST-operation mode used during start-up are resulting in higher respectively lower stress during ST start-up. With the selection of the start-up mode different start-up times can be achieved.

During gas turbine startup, depending on the actual startup condition (hot, warm, or cold), steam is generated more or less rapidly in the HRSG. In general, appropriate steam properties for a steam turbine startup are reached at approximately 50 to 60% gas turbine load. This means 40 to 60% of nominal pressure and a sufficient degree of superheat (e.g., around 50 K). Before starting the steam turbine, the gland steam system must be in operation and the condenser evacuated. Until the steam turbine takes over the available steam flow, the excess steam flows through the steam turbine bypass. If supplementary firing

is installed, it should not be brought into operation until the gas turbine is at full load, respectively the steam turbine bypasses are closed and the steam turbine can accommodate the additional steam flow.

Figures 9–15, 9–16, and 9–17 show the three different startup sequences for a 250 to 400 MW class combined-cycle plant (in a one-GT-to-one-ST configuration), which can be brought to full load in as little as 40 to 50 minutes after eight hours standstill.

To ensure an early as possible GT load where the exhaust gas emissions are within acceptable limits and on the other side to produce with the exhaust gas energy in the HRSG sufficient steam (pressure and temperature) a shorter or longer holding point at around 40% GT load is set in all cases. A further loading of the GT up to full load or set target load is normally done at the point where the steam bypasses are closed and the steam turbine is able to admit all produced steam.

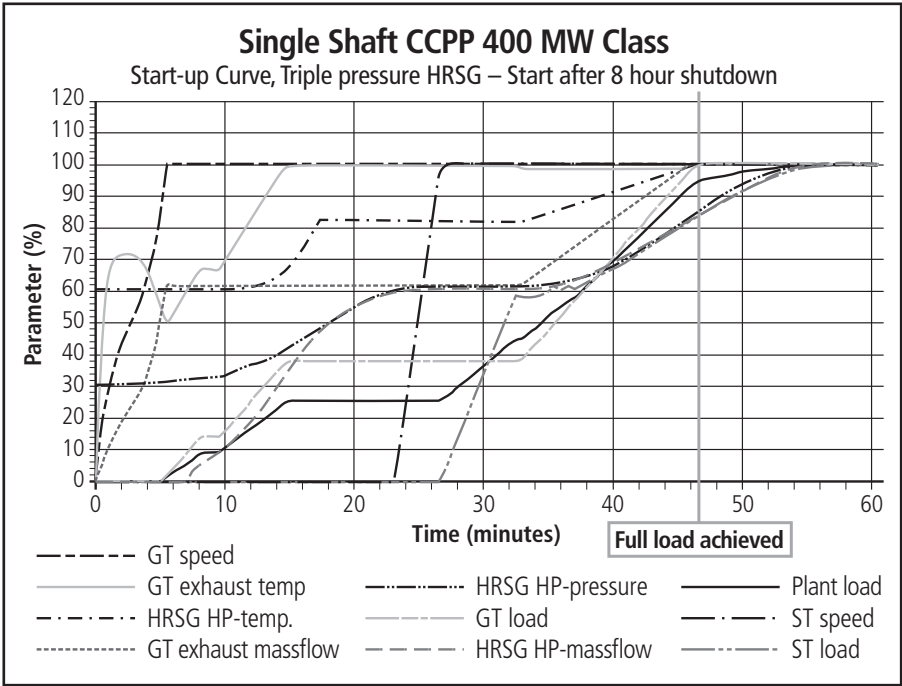


Figure 9–15 Startup curve for a 250 to 400 MW class combined cycle after eight hours standstill

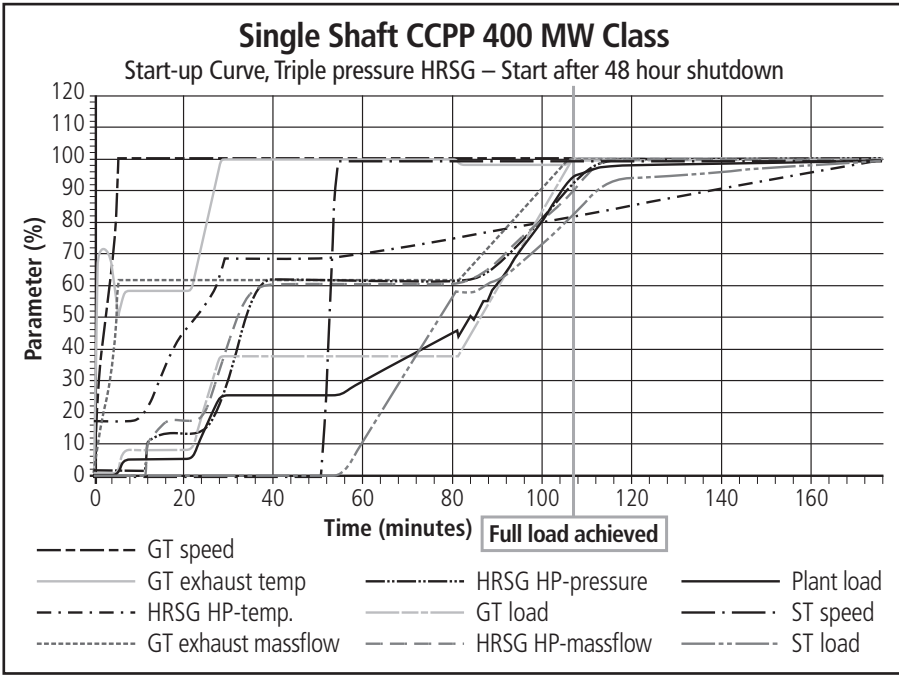


Figure 9-16 Startup curve for a 250 to 400 MW class combined cycle after 48 hours standstill

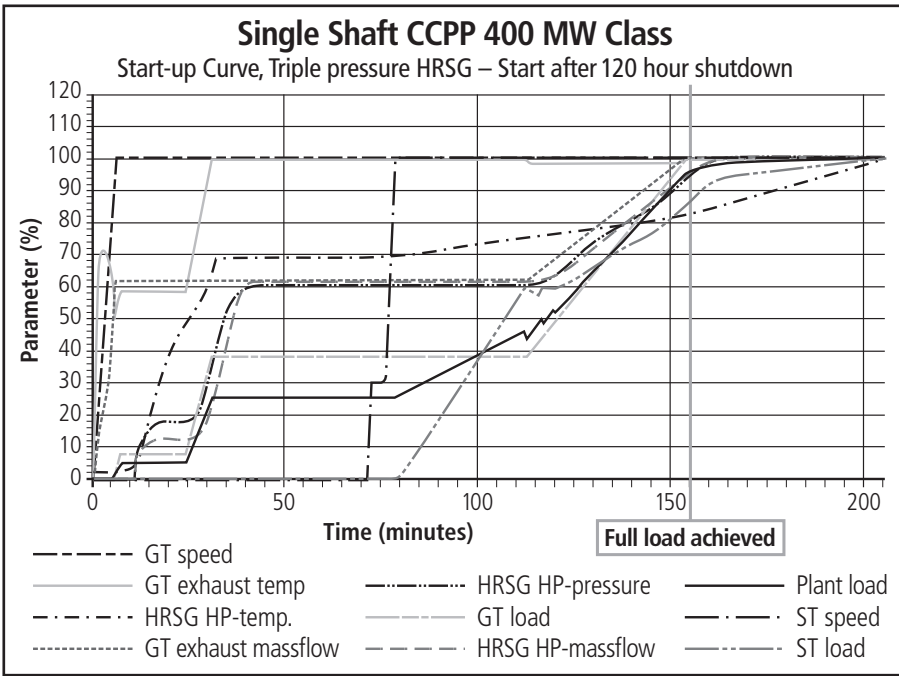


Figure 9-17 Startup curve for a 250 to 400 MW class combined cycle after 120 hours standstill

Plant shutdown

During unit shutdown, gas turbine and consequently steam turbine load is reduced according to the allowable transients to the inlet guide vane (IGV) point at nearly constant GT outlet temperature and approximately 60% of the exhaust gas/steam mass flow. At this point the GT load is kept constant and at full-steam temperature the shutdown program of the ST is started. The excess steam is dumped into the condenser via the respective bypass stations. At a preset load the ST trip is initiated to not cool down the ST unnecessarily.

After ST disconnection, GT load is reduced temperature controlled according to HRSG requirements.

To reduce the startup time after an overnight outage, closing of the HRSG outlet with a damper is recommended. Breaking of the condenser vacuum and shutting down of the gland (seal) steam system is recommended only in case of major plant outages. Thus the ingress of O₂ and CO₂ into the water/steam cycle is prevented, and the waiting time for the required steam purity is reduced at the next startup.

A typical shutdown diagram is shown in figure 9–18.

Fuel Changeover

Despite of the fact that the majority of the gas turbines are fueled by natural gas during normal operation, many of them are equipped with facilities to change over to fuel oil operation. Because of its suitability for storage, fuel oil is a good substitute in case of a sudden breakdown of the natural gas supply. The changeover delay has to be as short as possible, with a changeover avoiding shutdown and restart of the gas turbine to prevent an unplanned interruption of the power generation. The shift back to natural gas operation is usually less critical in terms of time, and a plant shutdown before changing back to natural gas is, therefore, acceptable.

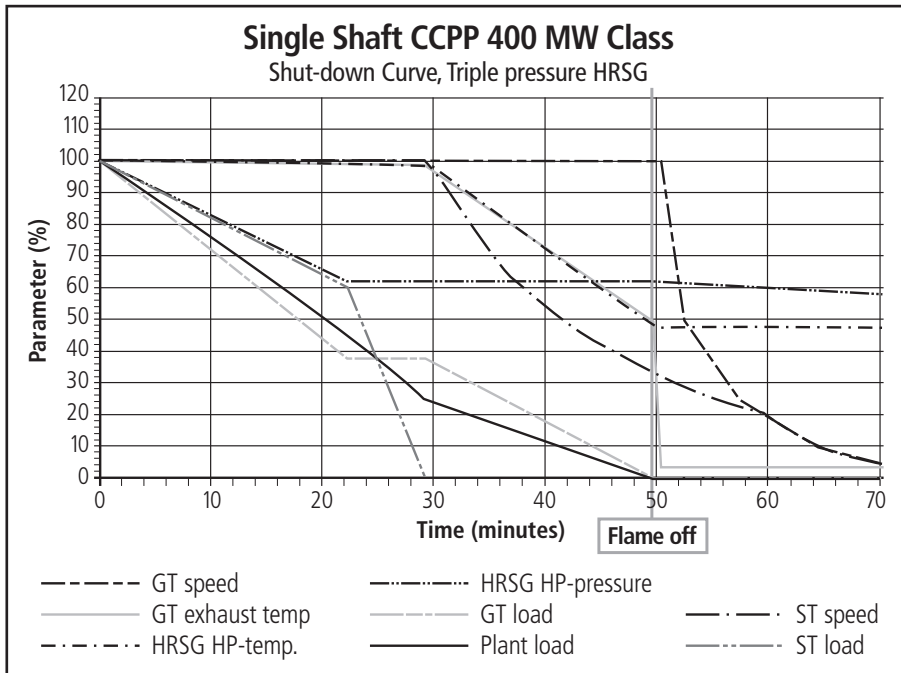


Figure 9-18 Combined-cycle shutdown curve

The overall time of the changeover is determined by the lead times of the fuel oil supply system, the gas turbine fuel oil system, and the changeover operation itself. The latter includes the following steps, and is shown in figure 9-19:

- Step1: Load reduction from full-load operation (natural gas premix mode)
- Step 2: Change over to fuel oil diffusion mode
- Step 3: Switch over to fuel oil premix mode, followed by a load increase back to full load

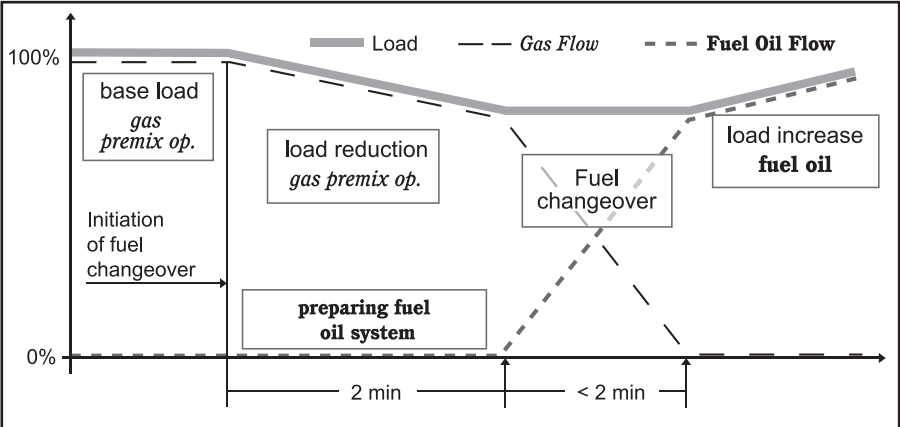


Figure 9-19 Switch over diagram from gas to oil operation

The transition from gas to oil fire in Step 2 can be done in the easiest way during the diffusion regime because no pilot burners are in service. Special attention should be paid to the maintenance of minimum flow rates of gas and oil to avoid flame pulsation, or even backfiring. Because of the enhanced NO_x emissions, the diffusion regime should be as short as possible.

The reverse sequence from oil to gas fuel operation needs water flushing of the closed oil burners to avoid clogging. The flow rate of the water is optimized between a maximum cleaning capability and a minimum impact on the stability of the gas flame.

10 Environmental Considerations

The impact any power plant has on its environs must be minimized. Legislation in different countries establishes rules and laws that have to be fulfilled. Quite often, emission limits are based on the best available emissions control technology. Exhaust emissions to the environment are mainly controlled in the gas turbine. Often, regions with less-stringent air emission requirements profit from the same combustion technology as areas with stringent requirements because the same hardware is used.

The following emissions from a power station directly affect the environment:

- Combustion products (exhaust and ash)
- Waste heat
- Waste water
- Noise
- Radioactivity and nuclear waste (nuclear power stations only)

Exhaust gases can include the following components: H_2O , N_2 , O_2 , NO , NO_2 , CO_2 , CO , C_nH_m (unburned hydrocarbons, UHC), SO_2 , SO_3 , dust, fly ash, heavy metals, and chlorides.

H_2O , N_2 , and O_2 are harmless; the others, however, can negatively affect the environment. Concentration levels of these substances in the exhaust depend on the fuel composition and the type of installation.

However, the greater the efficiency of the installation the greater the dropoff in the proportion of emissions per unit of electrical energy produced.

Because most combined-cycle plants combust natural gas, they produce low exhaust emissions. Their high efficiency results in low emissions to the atmosphere per MWh of electrical power produced as well as a small amount of waste heat. The high excess air ratios typical of gas turbines enable practically complete combustion, which results in a very low level of unburned constituents such as CO or UHC in the exhaust. Because of the extremely low sulfur content of natural gas, SO_x (SO_2 , SO_3) emissions are negligible. Particulate emissions from combined-cycle power plants are low due to the restrictions of solid matter input imposed by the gas turbine. Both fuel and combustion air have to be thoroughly cleaned to protect the gas turbine from corrosion and blockage of cooling-air passageways in air-cooled blades and in the combustion chamber. Therefore, a combined-cycle plant can be considered environmentally friendly and well suited for use in heavily populated areas.

For plants that combust natural gas, the most relevant emissions in the exhaust are NO_x (NO and NO_2). NO_x emissions generate nitric acid (HNO_3) in the atmosphere which, together with sulfuric and sulfurous acids (H_2SO_4 , H_2SO_3), are factors that cause acid rain. CO_2 is created by burning fossil fuels and is considered responsible for global warming.

Reduction of NO_x Emissions

NO_x is produced in air in large quantities at very high temperature levels. Figure 10–1 shows NO_x concentrations in the air as a function of air temperature. The concentration shown is at equilibrium in air attained after an infinite time.

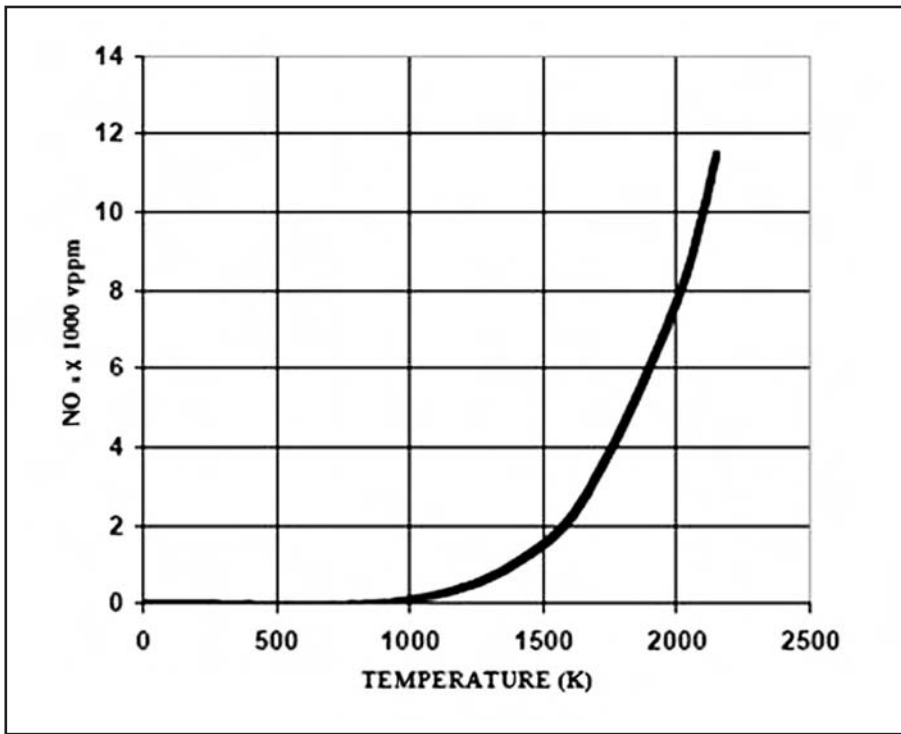


Figure 10–1 NO_x equilibrium as a function of air temperature

The situation in a gas turbine combustor is different because combustion of fuel in air takes place, and also because the high temperature residence time is fairly limited. The major factors affecting NO_x production in the combustor are:

- Fuel to air ratio of the combustion process
- Combustion pressure
- Air temperature in the combustion chamber
- Duration of the combustion process.

As seen in figure 10–1, NO_x is formed only when temperatures are high; this is the case in the flame in the combustor. The temperature of this flame depends on the fuel-to-air ratio of the flame and the air temperature in the combustion chamber, as shown in figure 10–2.

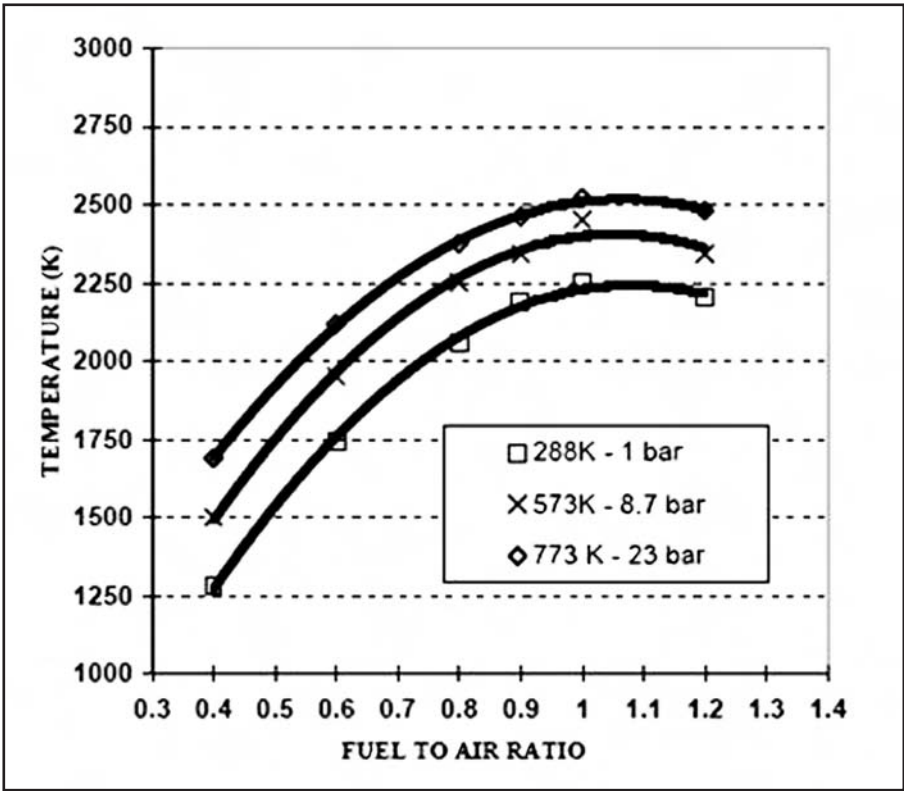


Figure 10-2 Flame temperature as a function of the fuel-to-air ratio and combustion air conditions

This temperature is highest in the case of stoichiometric combustion, (fuel-to-air ratio = 1). Figure 10-3 shows how concentrations of NO_x depend on the fuel-to-air ratio and the combustion air conditions.

It is evident that a peak is reached with a ratio of approximately 0.8. Above that level, the flame temperature is higher but there is less oxygen available to form NO_x because most of it is used by the combustion process. Below that level, NO_x decreases because of the abundance of excess air within the flame, thus lowering the flame temperature.

Very high fuel-to-air ratios are beneficial from the point of view of NO_x but are detrimental to combustion efficiency and cause the production of large amounts of CO and unburned hydrocarbons (UHC).

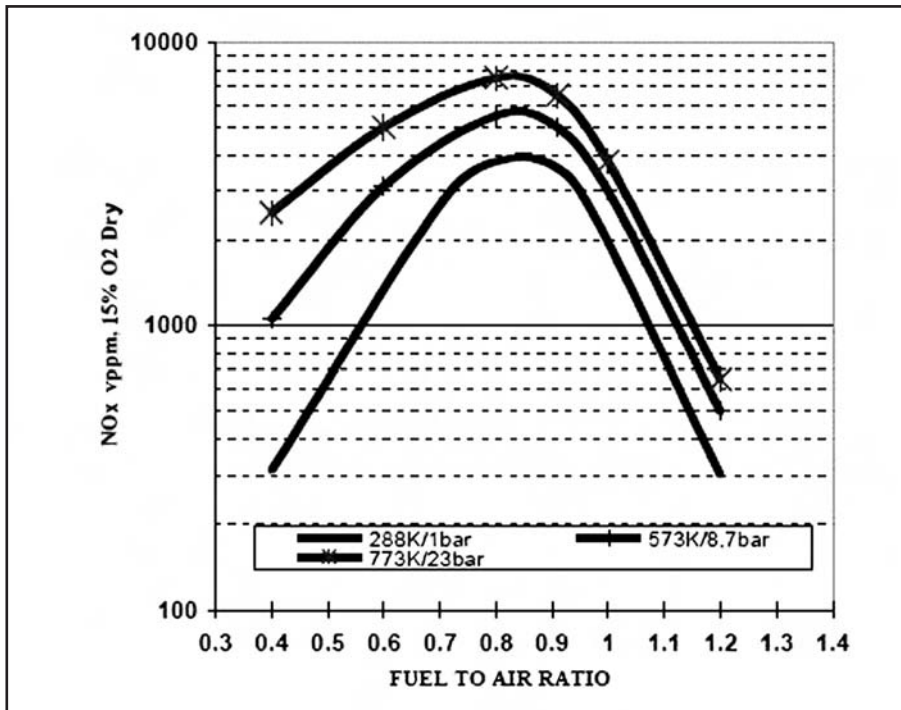


Figure 10-3 NO_x concentration as a function of fuel-to-air ratio and combustion air conditions

Conventional gas turbine combustors with diffusion burners were designed to operate with a fuel to air ratio of approximately 1 at full load, ensuring thorough, stable combustion over the entire load range. Obviously, NO_x emissions are high unless special precautions are taken. Nowadays burners operate at a lower fuel-to-air ratio that results in lower NO_x emissions.

The simplest way to reduce NO_x concentrations in these diffusion burners is to cool the flame by injecting water or steam into it. Figure 10-4 shows the reduction factors for NO_x emissions that can be attained as a function of the amount of water or steam injected. The amount of water or steam injected is indicated by the coefficient Ω (the ratio between the flows of water or steam and fuel). At a ratio of $\Omega = 1$, the typical reduction factor is approximately 6 with water and approximately 3 with steam. Water is more efficient than steam

because evaporation takes place in the flame at low temperature, which enhances cooling efficiency.

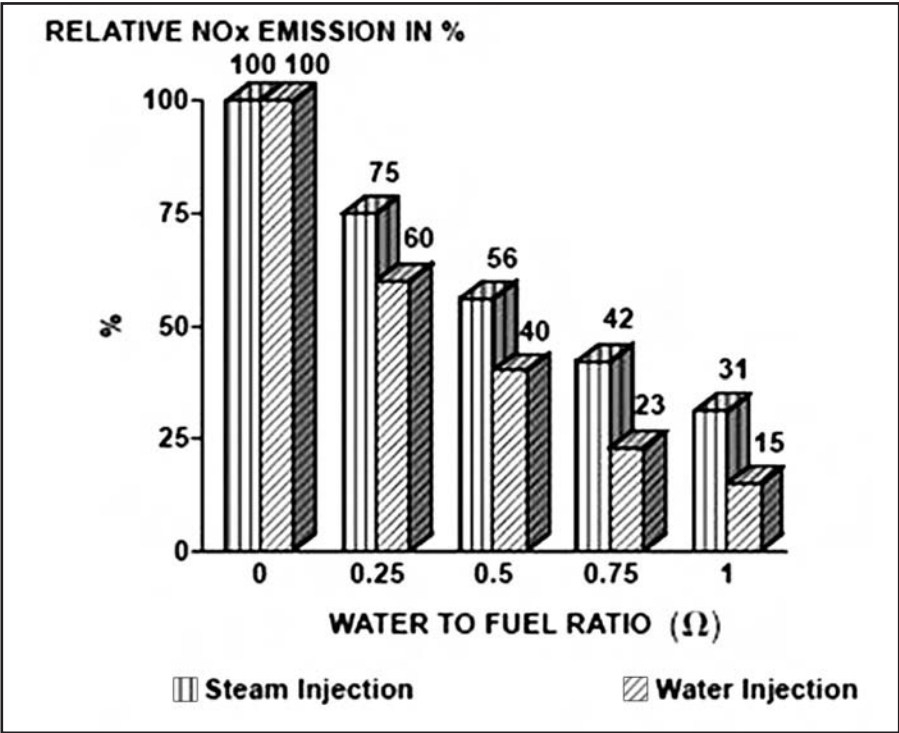


Figure 10–4 NO_x reduction factor as a function of the water or steam to fuel ratio in gas turbines with diffusion combustion

With this wet method, it is possible to attain NO_x levels as low as 40 ppm (15% O₂ dry) in the exhaust from gas-fired gas turbines or a combined cycle. Steam or water injection is a simple means to reduce NO_x emissions, but it does entail the following disadvantages:

- Large amounts of demineralized water are required.
- The efficiency of the combined-cycle plant is lower, particularly if water injection is implemented.

The fact that these methods can increase the plant output (especially with water injection) can be of interest in plants with a low number of

operating hours per year. In general, however, this does not compensate for the loss in efficiency and the high water consumption (for further details see chapter 5, figure 5–9).

The disadvantages inherent to steam or water injection have motivated all gas turbine manufacturers to develop combustors which achieve low NO_x levels with dry combustion (i.e., without injecting steam or water).

The principle behind keeping NO_x levels low is to always dilute the fuel with as much combustion air as possible to maintain a low flame temperature, and to keep the residence time in the hot combustion zone short. With this dry low- NO_x method and firing of natural gas, NO_x levels below 25 ppm (15% O_2 dry) are achieved by modern gas turbines with high firing temperatures and efficiencies.

Theoretically, there are two ways that reduced NO_x emissions can be achieved:

- Oxygen-lean combustion
- Oxygen-rich combustion

Figure 10–3 shows the dependence of NO_x concentrations on the fuel-to-air ratio. Despite a high-flame temperature, only a small amount of NO_x can form in oxygen-lean combustion because there is hardly any oxygen available to produce NO_x . However, to attain complete combustion there must be a second, follow-up combustion stage here in which almost no NO_x is 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 with dry low- NO_x burners, a different—and more effective—procedure is applied because of the high overall excess air ratio (2.5 to 3.5). It is called *combustion with excess air* and is the same principle as the injection of water or steam, where a large amount of excess air effectively cools the flame. The procedure is subject to limits due to consideration of flame stability. With excess air ratios exceeding approximately two, combustion cannot take place and the flame is completely extinguished. This risk does not exist while the gas turbine is at full load because there is not enough air available for the burner

to significantly exceed an excess air ratio of two. The remaining air is needed to cool the hot parts and the turbine blading. It is, however, a problem at part loads. For combustion to actually take place at the desired excess air ratio, the air and fuel must be mixed homogeneously with each other.

Figure 10–5 shows a typical low- NO_x burner. The air and natural gas are premixed in the burner and combusted downstream of the burner.

A vortex breakdown structure holds the flame in the free space. Characteristics of this burner include:

- No flashback due to the vortex breakdown structure
- Low fuel-to-air ratio (and, therefore, low NO_x)
- For liquid fuel, water injection is used to limit NO_x emissions

With this combustion technology there are modern gas turbines with high firing temperatures in operation with NO_x emissions clearly below 25 ppm NO_x at 15% O_2 .

Further development of the low- NO_x burners is proceeding toward even lower emission levels. Due to investments in combustion research, lower NO_x levels are attainable and potential for further improvement is assumed.

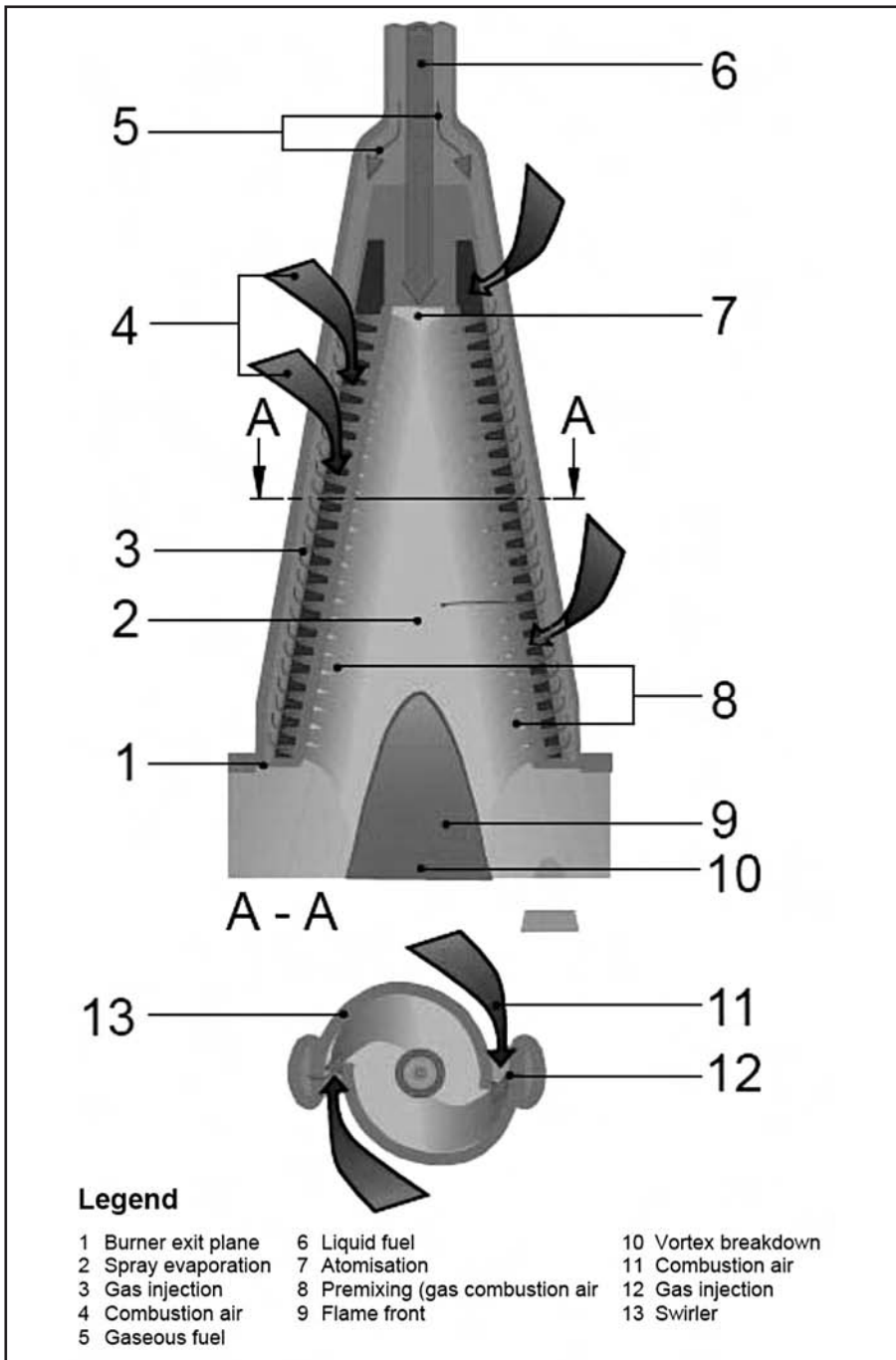


Figure 10-5 Cross section of a low NO_x burner

Another technical concept available on the market, which follows the same principles, is shown in figure 10–6.

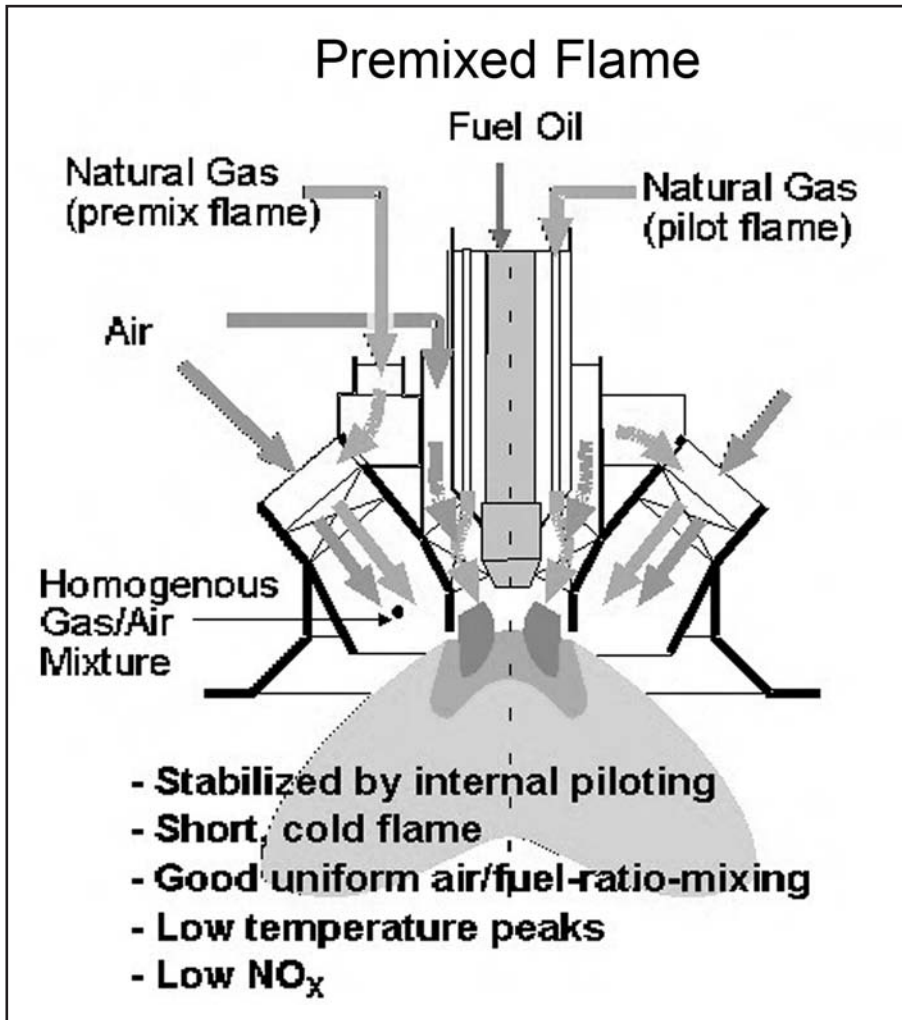


Figure 10–6 Cross section of the Siemens dry low NO_x burner

Local regulations in some parts of the United States and in Japan require NO_x emissions well below 25 ppm (15% O_2 dry). In these cases, it is generally necessary to install a reduction system in the heat recovery

steam generator (HRSG). Known as selective catalytic reduction (SCR), these systems inject ammonia (NH_3) into the exhaust upstream of a catalyst and can thereby remove up to approximately 85% of the NO_x in the exhaust leaving the gas turbine. The chemical reactions involved are as follows:



Technically these are well-proven systems, but they entail the following disadvantages:

- Investment costs are high; the HRSG is 10 to 30% more expensive.
- The use of ammonia is necessary, with a fraction of the ammonia passing through the SCR (ammonia slip).
- Power output and efficiency of the power plant are reduced by approximately 0.3% due to the increased back-pressure of the gas turbine.
- When firing oil in the gas turbine, sulfur in the fuel reacts to form ammonium sulfate, which precipitates in the cold end of the HRSG and further increases back pressure to the gas turbine. Power output and efficiency of the plant is further reduced.
- The HRSG requires periodical cleaning, and the waste must be disposed of.
- The catalyst must be installed in the evaporator section of the HRSG because the reaction takes place only in a temperature window of 300 to 400°C (572 to 752°F).
- Replacement costs are high.

Figure 10–7 shows a typical SCR system installed in a HRSG. NO_x levels of less than 5 ppm (15% O_2 dry) can be achieved by implementing a SCR system in conjunction with a gas turbine equipped with dry low- NO_x burners.

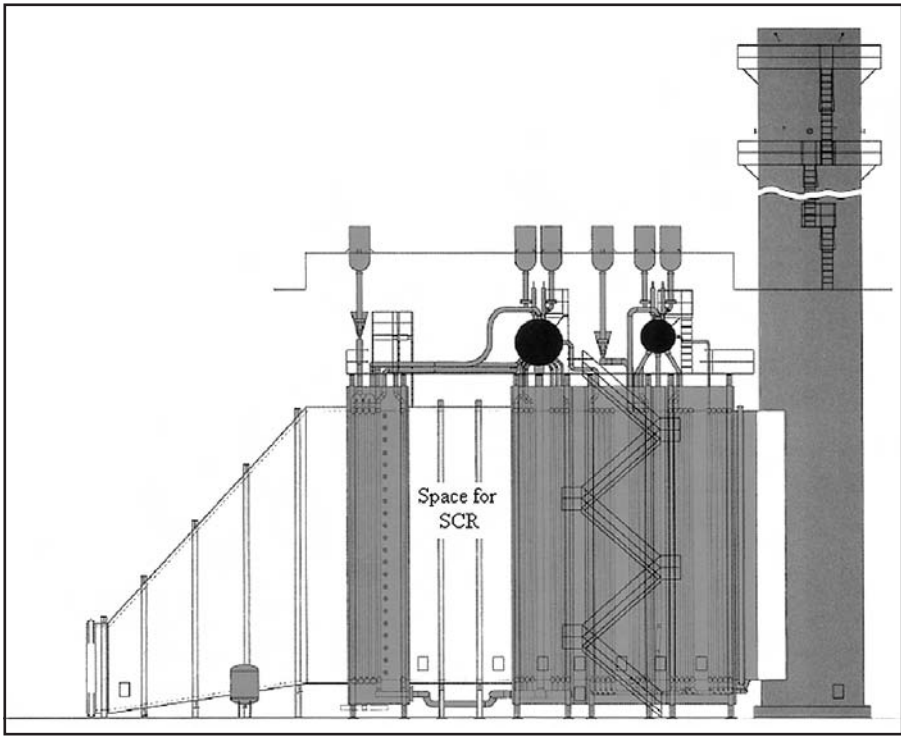


Figure 10–7 Heat recovery steam generator with selective catalytic reduction

SO_x Emissions

Concentrations of SO_2 and SO_3 produced depend only on the quality of the fuel. Approximately 70% of the gas turbines in operation use clean natural gas fuel; therefore, the SO_x emissions are negligible.

Liquid fuels contain between 0.05% sulfur in the case of sulfur-reduced No.2 oil (used only by few older gas turbines) and 2% sulfur for heavy or crude oil (used by approximately 4% of the gas turbine fleet). If the resulting SO_x emissions are not acceptable, the most economic way to reduce these emissions is to directly reduce the sulfur content in the fuel, either by chemically removing sulfur or by blending the fuel with a fuel with lower sulfur content.

CO₂ Emissions

Each fossil-fired power plant produces CO₂, which is considered responsible for global warming. However, a modern combined-cycle plant combusting natural gas produces approximately 40% of the CO₂ per MWh of electricity of a conventional coal-fired power plant. There are two reasons for this:

- The higher efficiency of the combined-cycle plant
- Use of natural gas fuel, which is mainly methane (CH₄), as opposed to coal (C)

During the 1990s, with deregulation of the power generation market in the United Kingdom, modern, high-efficiency, gas-fired combined-cycle plants replaced many old coal-fired power stations. This replacement meant that CO₂ production per MWh of electricity generated by these new power plants dropped to a third of the value for the coal-fired plants. This example shows that deregulation can have a strong positive ecological effect.

New concepts to reduce and even to avoid CO₂ emissions in power generation from coal will use combined-cycle plants as the major component of plants that first generate gas from coal by gasification followed by gas cleaning and the power generation process itself. These innovative Integrated Gasification Combined-Cycle plants (IGCC) are described in chapter 12.

Waste Heat Rejection

Another environmental concern is the waste heat that every thermal power station releases 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 useful output is produced. This reduces the amount of waste heat released to the environment.

In addition to the quantity of waste heat, however, the form in which the heat is transferred to the environment is also important. The effect is less severe if the power plant heats air instead of releasing its waste heat to a river or the sea. Conventional steam power plants often dissipate the waste heat to water for efficiency reasons. The most economic solution for combined-cycle plants is frequently to dissipate waste heat to the air through a wet cooling tower. Direct air cooling is also possible, but results in a reduction in output and efficiency together with increased costs due to the air-cooled condenser (see chapter 7).

A combined-cycle plant needs only half the cooling water of a conventional steam plant of the same output and a third of what is required for a nuclear power station.

A gas turbine usually requires practically no external cooling except for the lube oil and generator. This fact has contributed greatly to its widespread acceptance in countries where water is scarce. Because the steam portion of the combined-cycle plant accounts only for a third of the output, an air-cooled condenser is an economic solution for dry areas, underscoring the wide range of possible applications.

Table 10–1 shows the amount of waste heat that must be dissipated for various types of plants with 1000 MW electrical output. All steam cycles are cooled by river water or seawater.

Table 10–1 Comparison of the heat to be dissipated for various types of 1,000 MW station

Heat sink	Gas turbine [MW]	Combined cycle [MW]	Steam plant [MW]	Nuclear plant [MW]
Air / stack	1,500–2,000	130–180	70–100	0
Water	0	550–700	1,100–1,400	1,800–2,000

Noise Emissions

An environmental issue that is considered during the design and construction of a combined-cycle power plant is noise. This issue can be resolved using acoustic insulation and silencing equipment available today.

A distinction is made between *near field noise* and *far field noise*. Near field noise refers to the noise levels at the machinery. Far field noise often based on the plant site boundary and indicates the noise emitted to the environs. Main sources for this noise are the gas turbine inlet, the gas turbine exhaust (or HRSG), the stack and cooling tower, or an air-cooled condenser. Steam bypass operation during startup and shutdown is an additional source of noise.

Conclusion

Due to their low emissions levels, low cooling requirements, and noise levels capable of meeting stringent requirements, combined-cycle plants are considered environmentally friendly and are well suited for decentralized power generation in urban areas.

11 Developmental Trends

New technology developments and operating modes of combined-cycle power plants are being driven primarily by economic needs and—with growing importance—by ecological constraints. At the end of the day these ecological constraints will also result in a commercial impact on the competitiveness of the respective technical concepts.

This means in general that new competitive combined-cycle power plants have to generate electrical energy at lower cost than competing power stations in the different market segments of base and intermediate load with their fluctuating electricity prices today and in the foreseeable future.

This can be achieved by plants that offer:

- High energy conversion efficiency
- Low investment and operating costs resulting in low life-cycle cost
- High reliability, availability, and maintainability
- Higher plant flexibility
- Improved maintenance concepts
- A broad range of applicable fuels and fuel flexibility
- Environmentally compliant operation and preparations for increasing environmental requirements

As a matter of course all these features have to be optimized from a commercial point of view to achieve the ability to compete.

Seven major developmental trends can be identified to meet the aforementioned objectives.

- Increased gas turbine firing temperatures to enhance the efficiency of the gas turbine, and simultaneously of the water/steam cycle thanks to higher main steam parameters
- New technical gas turbine concepts that will achieve a higher efficiency
- Increased gas turbine power output to benefit from the economies of scale
- Enhanced load cycle capabilities to enable intermediate load and frequency control operation
- Reduced operating costs by applying remote control and service packages
- Less emissions, especially NO_x , to reduce environmental impact
- The development of hydrogen-fired gas turbines as a prerequisite
 - for new plant concepts using other fuels than natural gas and oil
 - and/or to improve environmental compatibility (CO_2 capture), (see chapters 12 and 13).

Increased Gas Turbine Firing Temperatures

The positive effect of high gas turbine inlet temperatures on the efficiency of the combined cycle and the resultant reduction in the fuel cost component in the cost of electricity is pushing gas turbine inlet temperatures higher.

It seems reasonable to look for further improvements from even higher gas turbine inlet temperatures that have become possible through the development of new materials and improved cooling systems.

Research projects in this area focus primarily on improved air cooling of the hot-gas path of the gas turbine. Alternative cooling technologies employing steam cooling, for example, have been announced by gas turbine manufacturers and implemented in both prototype and commercial plants, but, due to the increased complexity, without the expected breakthrough.

One main constraint on raising the temperature is the allowable strength of the blades and vanes. One major effort concentrates on the strength of the base alloy and its crystal structure, going from conventionally cast low-nickel to high-nickel alloys with a single crystal. In the future even more important will be the development of the coating technologies, protecting the base alloy not only against corrosion but also against the temperature of the flue gas (thermal barrier coating or TBC). Some companies are considering ceramic structures (CMC) at least for the vanes. The respective development programs are ongoing but the results do not meet expectations. In particular the required mechanical strength and parts life remains a challenge for further research. Figure 11–1 shows the increase of the gas turbine inlet temperatures over the course of time, and as a function of the specific development steps.

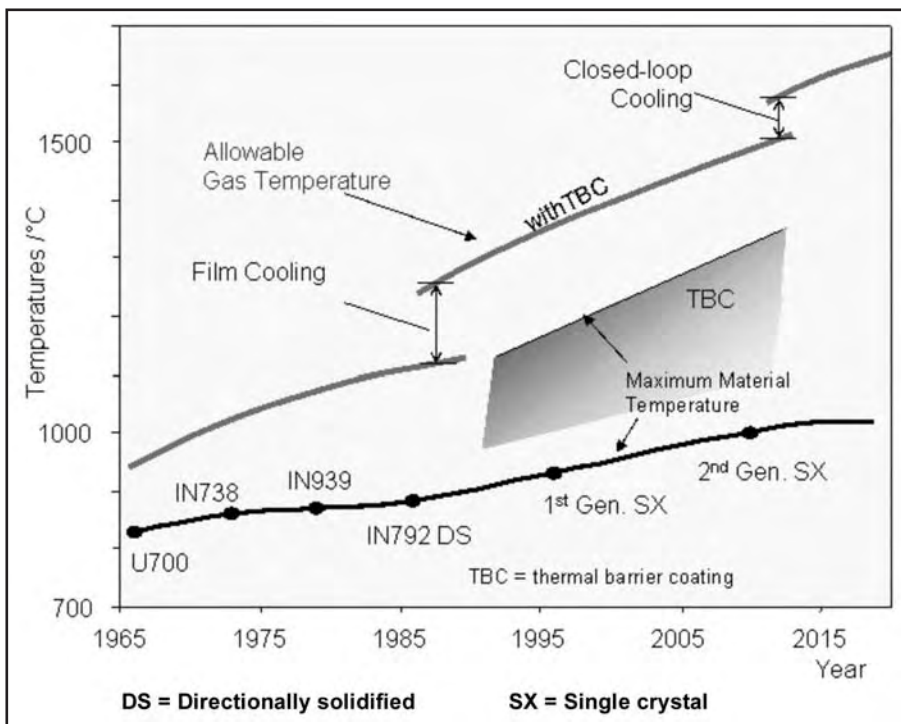


Figure 11–1 Chronology of the gas turbine inlet temperatures based on improved material and cooling technologies

Closed-steam cooling of stationary parts

The majority of gas turbines presently in operation have air-cooled blades and combustion chambers.

Cooling of the blading allows turbine inlet temperatures to be increased beyond the allowable material temperatures. If air cooling is employed, the cooling air first cools the blade internally and then exits onto the blade surface, providing a cooling film that separates the hot gas from the blade surface. However, mixing the two flows leads to a disturbance of the flow pattern and the cooling of the hot gas, which reduces efficiency. An alternative is to use steam as a cooling medium. Closed-steam cooling takes advantage of the fact that steam has a higher heat capacity than air, and doesn't deteriorate the expansion efficiency. Additionally, more air is available for the combustion process, which results in a lower flame temperature and lower NO_x levels.

Additional output is achieved by the increased mass flow through the turbine. Steam for cooling is available from the steam part of the water/steam cycle. It can be taken from the cold reheat line exiting from the steam turbine.

An increase in plant efficiency can be achieved with closed-steam cooling of the stationary parts, but this performance improvement has the following disadvantages:

- The gas turbine and steam turbine cycle get more interconnected, resulting in restriction or even prevention of operation of the plant as a simple-cycle gas turbine.
- Prolonged startup times of both gas turbine and whole plant leads to reduced operational flexibility.
- Higher complexity leads to slightly reduced reliability and availability figures.

Steam cooling of stationary and rotating parts

The improvement potential is even larger for steam cooling if the rotating parts are steam cooled as well.

A combined-cycle plant with a fully steam-cooled gas turbine is expected to offer higher efficiency and produces a larger specific power output per unit of inlet airflow. This could eventually result in lower specific costs, although maintenance costs are expected to be higher due to the higher blading costs. The following points are challenges for this type of gas turbine.

- Cooling of the thin front and rear ends of the blades
- Steam purity requirements for the cooling steam
- Leak tightness of the steam system (steam must be supplied to the gas turbine rotor and returned from there)

This type of plant is attractive from an economic point of view only if these challenges are resolved satisfactorily and the plant achieves the corresponding reliability. Also the disadvantages of steam cooling in general, mentioned in the previous paragraph should be taken in consideration. Whether the benefits of highly sophisticated and complex technology will be high enough to make it competitive is still an open question. The manufacturer is speaking about a *“...combined-cycle system capable of breaking the 60% efficiency barrier...”*

Larger compressors

Parallel to these developments, improvements are also being made to the compressor. The advantages offered by the higher gas temperatures can only be fully exploited if the pressure ratio of the machine is increased to an appropriate level. Increasing the airflow through the compressor also produces high unit ratings. With modern blading, compressors are able to handle volumetric flows that seemed utopian just a few years ago. With latest compressor models developed using modern design tools and incorporating long-term experiences with aero engines, it is possible to increase pressure ratios and reduce the number of compressor stages.

Higher main-steam parameters

Early combined-cycle plants had water/steam cycles with low steam parameters compared to conventional steam power plants. Main steam

pressures in a range of 50 to 80 bar (710 to 1,150 psig) and main steam temperatures of 450 to 500°C (842 to 932°F) were standard.

Modern, large-capacity combined-cycle power plants for power generation operate at a main steam pressure of up to 170 bar (2,310 psig) and a main steam temperature up to 565°C (1,049°F), with the latest figure as high as 600°C.

For steam turbines beyond approximately 100 MW capacity, reheat machines are now standard compared to the previous non-reheat units. This was made possible by the sequential firing technology or higher firing temperature of the gas turbines, which, as a result, provided higher exhaust temperatures and, therefore, higher inlet temperatures to the heat recovery steam generator (HRSG). Higher gas turbine exhaust temperatures yield a higher optimal main steam pressure based on the net present value (NPV) of the plant. In the near future it can be expected that these trends will continue. Large plants for power generation will have higher main steam pressures, increased main steam temperatures, and reheat for the steam cycle.

The following points must be considered when the main steam parameters are increased.

- Higher main steam temperatures require more expensive alloys in the HRSG, steam piping, and steam turbine. The gain in output must, therefore, justify this additional investment.
- Higher main steam pressures cause wall thickness to increase, which, in general, reduces thermal flexibility and increases cost. Once-through HRSGs will be installed more frequently to avoid the negative impact on thermal flexibility of the higher main steam pressures.
- High main steam pressure in combination with a reheat steam turbine reduces the main steam volume, which may result in a reduction in the efficiency of the high-pressure steam turbine due to shorter blading.

The last mentioned effect would result in a main steam pressure that is too far away from the thermodynamic optimum. To avoid this relative disadvantage either two gas turbines and HRSGs are combined with one steam turbine to the typical 2 + 1 configuration, or the high-pressure

steam turbine runs at higher speed, which reduces rotor diameter and allows an increase of blade length. A side benefit is the smaller casing dimensions, which improve thermal flexibility. Figure 11–2 shows a geared high-pressure turbine.

As the exhaust gas temperatures of the gas turbine rise further, the optimal cycle selection is affected.

The triple-pressure reheat cycle is often the optimal steam cycle for a gas turbine with an exhaust temperature of approximately 600°C ($1,112^{\circ}\text{F}$). The optimal cycle will move towards a single-pressure reheat cycle for gas turbines with an exhaust temperature of 750 to 800°C ($1,382$ to $1,472^{\circ}\text{F}$).

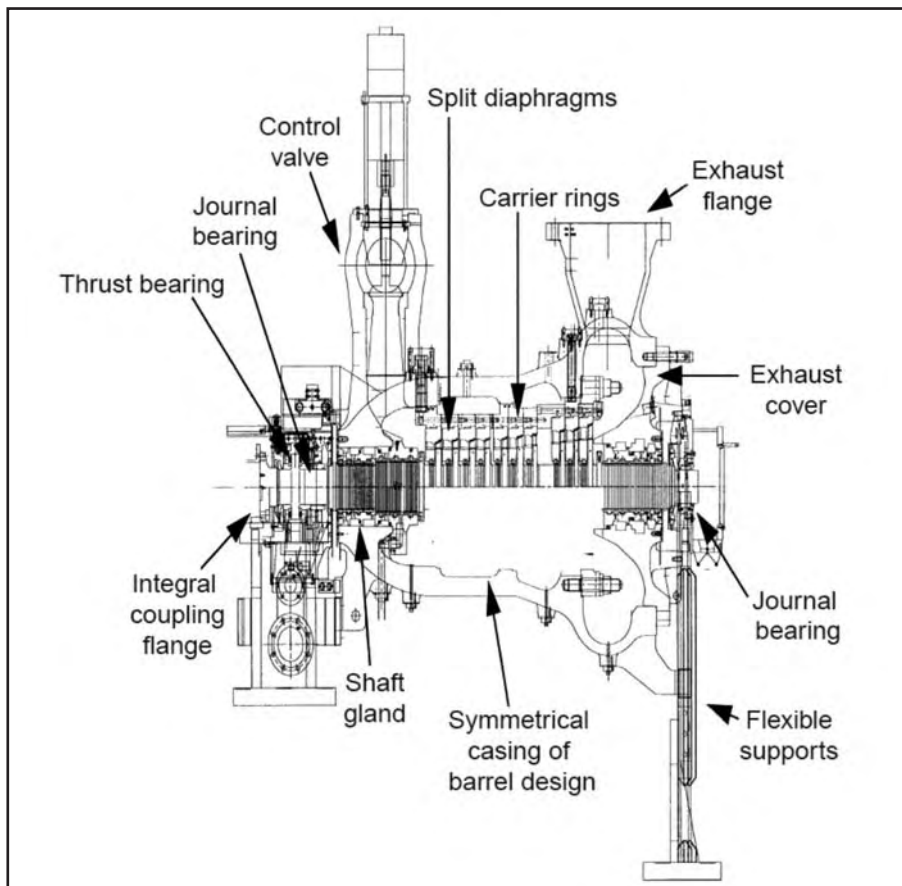


Figure 11–2 Geared high-pressure turbine

With these improvements combined-cycle plants with typical guaranteed net efficiency of around 58.5% are already in commercial operation, and the first modern plants achieving 60% are already in test and validation. New gas turbines will now be available with unit power capacities up to 340 MW for 50 Hz applications (figure 11–3) and 270 MW for 60 Hz applications leading to lower costs, so that combined-cycle plants will become more competitive for large power plants.



Figure 11–3 Siemens SGT5-8000H gas turbine

Plant operations and service—Plant flexibility

High operational flexibility—the ability of a power plant to achieve fast startup and to adjust load output quickly and predictably to changing market requirements, as well as high reliability and availability—are essential prerequisites to ensure economy of operation in a liberalized market.

Major factors limiting the load variation of an existing combined-cycle power plant are the allowed pressure and temperature transients of the steam turbine and the heat recovery steam generator, waiting times to establish required steam chemistry and warm-up times for the balance of plant and the main piping system. Those limitations also influence the fast startup capability of the gas turbine through required waiting times compared to a simple-cycle startup. Enabling combined-cycle power plants to improve operational flexibility requires a range of so-called fast cycling provisions. The most important ones are shown in figure 11–4, with the main emphasis on warm-up of the steam turbine with the first steam produced after start of the gas turbine and run-up during continuous pressure and temperature increase. As an example, these measures allow a startup time of less than 40 minutes.

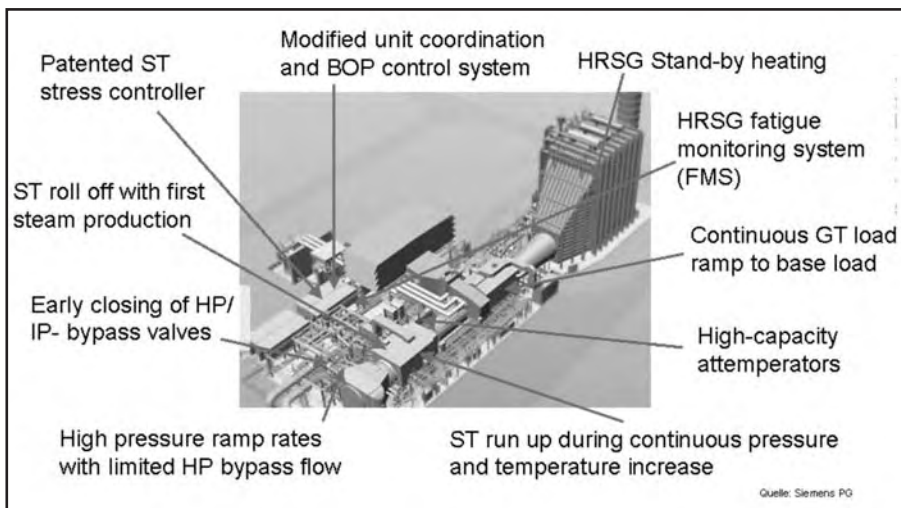


Figure 11–4 Plant impact for fast cycling capability

Maintenance and performance degradation optimization

The reliability and availability of a power plant are strongly dependent on an optimized operations and maintenance concept covering the whole lifetime of the plant. The original equipment manufacturers

(OEMs) offer a lot of service packages to support the power plant operator. Beyond the simple case-by-case or routine maintenance, refurbishment of the hot gas path parts, supply and installation of spare parts for components, these packages might be long-term and full-scope maintenance programs for components or the whole plant, and finally the complete operations and maintenance done by the OEM. To minimize the gas turbine performance degradation described in chapter 6, either the measures of the routine maintenance are applied or customized upgrade packages for the main components of the power plants, especially for the gas turbines, could be implemented. A classic example is the download of modern gas turbine technology into the older engines that have been operating for many years. That could be compressor upgrade to increase the air mass flow or upgrade of the hot gas path parts to make an increase of the firing temperature possible.

Incremental as well as disruptive technical improvements linked with aligned operational and service concepts are mandatory to ensure the competitive position of combined-cycle power plants in a market driven by changing economic and ecological challenges.

12 Integrated Gasification Combined Cycle

Environmental constraints as well as increasing prices of high-grade fossil fuels such as natural gas are major drivers that determine further development of fossil-fuel-fired power stations. One of the most attractive options to achieve extremely low environmental pollution is the Integrated Gasification Combined Cycle (IGCC). The IGCC concept, shown in figure 12–1, opens the well-proven combined-cycle concept to so-called *dirty fuels* such as coal, refinery residues, biomass, and wastes by adding gasification, air separation, and gas cleaning processes to the upstream gas turbine combustor.

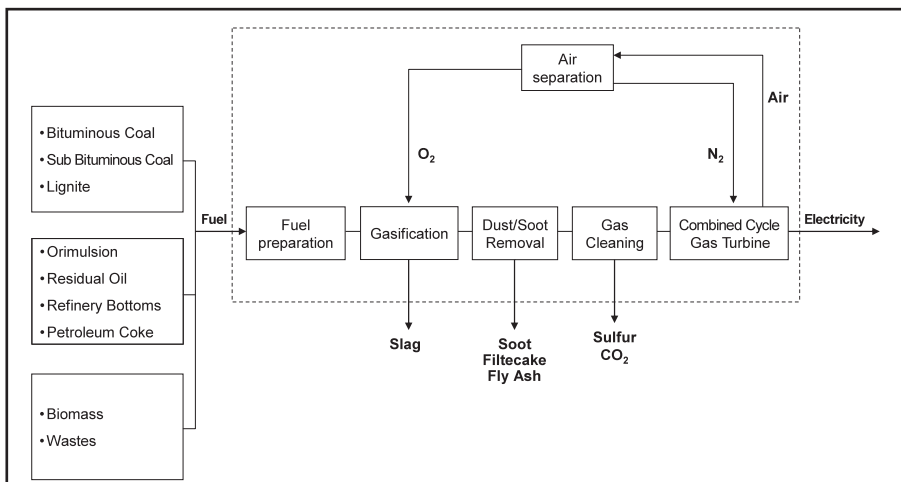


Figure 12–1 Principle IGCC concept

Gasification is basically a partial oxidation process that converts any carbonaceous feedstock such as coal, petroleum coke, heavy oil and oil tars, biomass, and waste streams into a gaseous product called synthesis gas (syngas). Syngas mainly consists of CO and H₂ (with some CO₂, H₂O, and contaminants), with the composition depending on fuel and type of gasifier (figure 12–2).

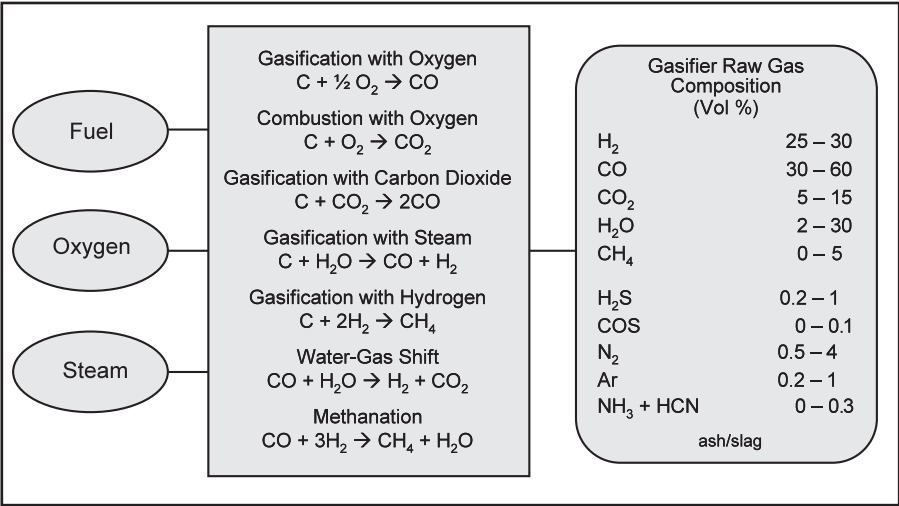


Figure 12–2 Gasification reactions

After gasification, the syngas is cleaned of its contaminants such as fly ash, alkali, chlorine, and sulfur compounds. If CO₂ capture is required, additional conversion and treatment steps can be foreseen. After leaving the gas cleaning section and before feeding the gas turbine combustor, the syngas is diluted with nitrogen and/or water vapor to moderate combustion condition and minimize thermal NO_x formation. Finally, combustion takes place in a combined-cycle power plant with a modified gas turbine. Such a system is inherently cleaner and more efficient than conventional pulverized coal (PC) fired plants. Current IGCC plants in operation have proven that syngas cleaning is a more efficient way than flue gas cleaning of conventional PC plants (table 12–1).

Table 12–1 Emission of existing coal based IGCC

IGCC	Start-up	PM [kg/MWhel]	NO _x [kg/MWhel]	SO _x [kg/MWhel]
Nuon Buggenum <i>The Netherlands</i>	01/94	0,005	0,318	0,200
Elcogas Puertollano <i>Spain</i>	12/97	0,020	0,399	0,068
Wabash River <i>USA</i>	10/95	0,045	0,494	0,490
Tampa / Polk Power <i>USA</i>	09/96	0,064	0,390	0,612
Advanced SPP	2010	0,030	0,530	0,400

Due to high investment cost, IGCC technology is primarily suitable for large centralized power plants with access to low-cost feedstock such as coal and petcoke. Moreover, petcoke with its high sulfur (>5% in weight) and vanadium content is suitable only for IGCC applications. Refinery residues such as heavy oils and tars are potential sources for hydrogen production used in refinery upgrading to produce diesel and petrol fuels. Therefore, refineries primarily focus on poly-generation concepts with hydrogen as the main product, whereas only surplus of syngas is used for electricity generation and export. Biomass and waste are best exploited using smaller plants close to their source or being co-gasified with coal in large IGCC plants to achieve better competitiveness.

The drawback of IGCC plants, however, is that they are significantly more expensive than conventional pulverised coal PC plants. Current IGCC plants in operation have suffered from relatively poor reliability and limited operational flexibility (e.g., long startup time). Nevertheless the demonstration plants built in the United States and Europe have proven that IGCC is a viable power generation technology (table 12–2). Improvements have been identified, and it is expected that further evolution of technology and commercialization will significantly reduce cost.

Table 12–2 Commercial solid fuel IGCC plants 2007

IGCC power plant	Buggenum	Wabash-River	Tampa	Puertollano	Vreshova
Location	Netherland	USA	USA	Spain	Czech Republic
Commissioning	1994	1995	1996	1998	1996 (2005)*
Net Power	253 MW	262 MW	250 MW	300 MW	351 (430) MW
Fuel/Coal	Hard coal/Biomass	Hard coal/Petcoke	Hard coal	Hard coal/Petcoke	Lignite
Gasifier	Prenflo	E-Gas (CoP)	GE	Shell	Sasol-Lurgi/SFG
Gasturbine	Siemens V94.2	GE 7FA	GE 7FA	Siemens V94.3	GE 9FE
Net Efficiency (LHV)	43 %	39 %	41 %	42 %	44 %

*Plant extension 2005

Gasification Technologies

Gasification became a commercial process in 1812 and was initially used to produce town gas for lighting and heating purposes. Since the early 1900s syngas has also been used as a chemical feedstock. Development of today’s large-scale gasification processes took place in the 1920s to the 1940s, mainly in Germany.

In a gasification reactor, first drying of the feedstock particles takes place. As the temperature rises, devolatilization occurs and tars, oils, phenols, and hydrocarbons are formed. With further increased temperature these devolatilization products are decomposed to lighter products such as CO, H₂, and CH₄. The fixed carbon, which remains after devolatilization, is gasified via reactions with O₂, H₂O, and CO₂. Beyond hydrocarbons other fuel constituent elements such as sulfur, nitrogen, and ash are of major consideration because they also react and their reaction products (e.g., H₂S, COS, HCN, NH₃, alkaline, chlorine, slag, and ash) have to be removed either out of the gasifier or downstream in a gas cleaning process.

Since the 1920s, three basic gasification technologies have been developed that are applicable for IGCC in principle: fixed-bed processes, fluidized-bed (FB) processes, and entrained-flow processes. All types of gasifier can be designed to operate with air or pure oxygen (>85%)

as gasification agent. Oxygen-blown gasification offers a number of advantages over air-blown systems because they promise a higher cold gas efficiency and carbon conversion rate and smaller plant component dimensions due to absence of nitrogen surplus. Gasifier technologies differ in temperature operating range, carbon conversion rate, particle size, and hydrocarbon content in raw gas starting from fixed-bed to entrained-flow gasification. A summary of technologies is given in figure 12–3.

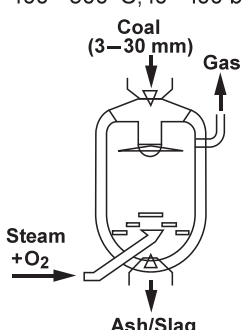
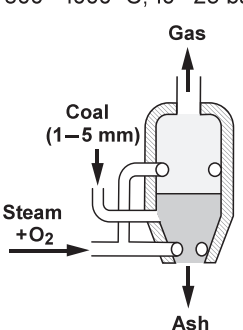
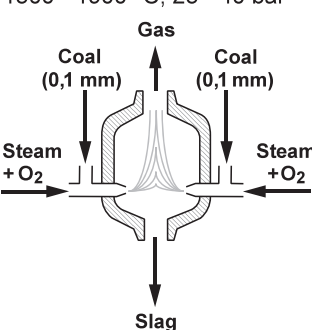
Fixed Bed	Fluidized Bed	Entrained Flow			
400–500 °C, 10–100 bar	800–1000 °C, 10–25 bar	1500–1900 °C, 25–40 bar			
					
Process Examples	Process Examples		Process Examples		
Lurgi Dry Ash; BGL	HTW	KRW	SHELL	SFG	GE
			(PRENFLO)	MHI	E-GAS
Autothermal gasification with oxygen		Autothermal gasification with air			

Figure 12–3 Basic technologies of gasification and gasifier technology vendors

Fixed-bed gasification is characterized by significant steam requirements for temperature control, long carbon conversion time, and a high content of higher hydrocarbons in the product such as tars and oils. The process is, therefore, not preferable for IGCC applications. Modern large-scale IGCC applications are based on FB or entrained-flow gasification concepts. FB gasification is characterized by gasification temperature below the ash melting point and subsequently lower oxygen consumption, which leads to a higher IGCC efficiency. Moreover as a consequence of lower gasification temperature, carbon conversion rate reduces resulting in extensive char recycle and significant

carbon loss. FB gasifiers are, therefore, suited only for high volatile coals such as lignite or Powder River Basin (PRB) sub bituminous coal.

Entrained-flow gasifiers operate with gasification temperature above ash-slagging conditions, achieve carbon conversion rates higher than 98%, and offer very short residence time leading to highest unit capacity. Although oxygen consumption is higher and IGCC efficiency lower compared to FB gasification, entrained-flow gasification systems are compatible with almost all kind of liquid and solid hydrocarbons.

There are numbers of solid-based entrained-flow gasifier types available, which differ in various features. They are summarized in figure 12–4 and explained in more detail following the following sections.

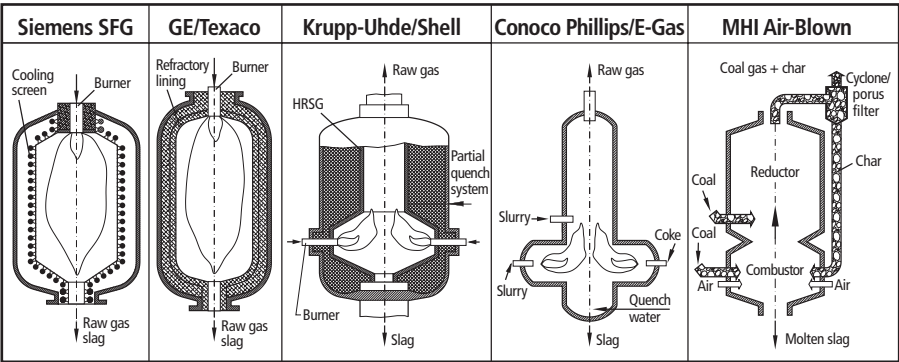


Figure 12–4 Entrained-flow gasifiers

Shell coal gasification process (SCGP)

- Single-stage gasification with dry feed
- O₂ blown
- Reactants flow upward and molten slag downwards; reactor employs a vertical tube membrane wall for cooling that generates IP steam
- Syngas cooling done by recycle of cooled syngas and further cooling by HP/IP steam generation in water tube cooler design

Advantage: Highest IGCC efficiency due to dry feed and heat recovery with HP/IP steam generation

Disadvantage: High investment caused by high alloy heat exchanger materials to prevent corrosion

GE coal gasification process (former Texaco)

- Single-stage gasification with slurry feed
- O₂ blown
- Reactants and molten slag flow downwards; reactor is equipped with refractory (ceramic brick liner)
- Syngas outlet temperature reduction is done by either full water quench or radiant and convective fire tube cooler

Advantage: Extensive commercial operating experience

Disadvantages: Limited lifetime of refractory and burner
 Long startup time due to need of refractory preheating
 Limited ability to handle low rank coals
 Relatively high oxygen consumption due to slurry feed

Siemens fuel gasifier (former GSP/Future Energy)

- Single-stage gasification with dry feed
- Reactants and molten slag flow downwards; reactor employs a spiral-wound coil-cooling screen
- Syngas cooling done by full and potentially also partial water quench; in case of partial water quench convective heat recovery for coal under development

Advantages: Low specific investment and oxygen consumption
 Proven for low-rank coals and most suitable for IGCC with CO₂ capture due to water quench
 Fast startup

Disadvantage: Limited operational experience up to 200 MW thermal throughput
 Lower efficiency due to quench cooling

ConocoPhillips E-Gas gasification process

- Two-stage gasification with slurry feed—first stage accomplishes exothermic oxidation and second stage provides syngas temperature reduction to 300–350°C by endothermic chemical and water quench (slurry); further cooling by fire tube syngas cooler
- Char/solid recycle to first stage
- Reactants flow upwards and molten slag downwards; reactor employs a refractory wall

Advantage: Relative high IGCC efficiency due to chemical quench and heat recovery

Disadvantages: Limited lifetime of refractory and burner
 Long startup time due to need of refractory preheating
 Limited ability to handle low-rank coals
 Limited suitability for CO₂ capture due to high methane content in syngas

MHI dry-feed gasifier

- Air-blown gasifier
- Two-stage gasification with dry feed—first stage is exothermic oxidation and second stage is chemical quench by endothermic chemical reaction
- Reactants flow upward and molten slag downwards; reactor employs a cooled membrane wall
- Syngas cooler

Advantage: High IGCC efficiency

Disadvantage: Technology in demonstration phase, no commercial reference yet

Syngas Cleaning Technologies

The syngas cleaning process is required to remove compounds, which harm the environment and/or lead to corrosion and thus a limited lifetime of components. Alkali compounds are especially responsible for high-temperature gas turbine corrosion and, to a very high extent, have to be removed. Table 12–3 provides gas turbine limitation for chemical impurities and dust.

Table 12–3 Allowed chemical impurities of gas turbine fuels (Siemens source)

Limits for chemical impurities (Fuel Weighting factor f=1)			downstream filter
Pollutant	Test/Check	Unit	Syngas
Dust (with natural gas), sediments (with EL distillate Fuel)	Total	DIN EN 12622	< 20
	$d < 2 \mu\text{m}$	VDI 2066 (1994)	< 18.5
	$2 < d < 10 \mu\text{m}$	ASTM D 2709	< 1.5
	$d > 10 \mu\text{m}$ $d > 25 \mu\text{m}$	& D 6304	< 0.002
Vanadium (V) (in case of fuel oil)	DIN 51790 ASTM D 3605	ppm (wt)	< 0.5
Lead (Pb) (in case of fuel oil)	DIN 51790 ASTM D 3605	ppm (wt)	< 0.1
Total of Sodium (Na) + potassium (K)	EPC method		< 0.5
	DIN 51790 ASTM D 3605	ppm (wt)	< 0.3
Calcium (Ca)	ASTM D 3605	ppm (wt)	< 10
Hydrogen sulfide (H_2S)	ASTM D 6228	ppm (vol)	< 100

Limits for fuel impurities (dust and ash, V, Pb, Na, K, Ca) are based on a lower heating value (LHV) of 42,000 kJ/kg.

Formula $X = \text{LHV}/42$ [X = pollutant content; LHV in MJ/kg] shall be used to correct for deviations in lower heating value

To treat and purify the syngas leaving the gasifier a number of cleaning steps are needed, starting with removal of soot, char, and ash particles. This is followed by halogen, ammonia, and alkaline removal, and finally with sulfur, mercury, metal carbonyl, and, if required, CO_2 capture.

Dedusting

The particulates removal as first step can either be done by water scrubbing or using dry filtration methods such as cyclones or ceramic candle filters. In case of high particle flow and significant unconverted carbon (char) containing syngas as it occurs in two-stage entrained-flow gasifiers (E-Gas, MHI) or FB (HTW), a dry removal and char recirculation into the gasifier reaction zone is applied. Bulk removal of char and fly ash is done by cyclones or/and ceramic candle filter systems, which are sensible against plugging but require regular cleaning by flushing. The water Venturi scrubber operates at syngas dew point and also washes some soluble vapors such as ammonia, HCN (hydrogen cyanide) halogen, and alkali contaminants. Therefore, current gasification systems employ water scrubbers for final mechanical cleaning and removal of soluble compounds. Gasifiers with full-water quench such as GE and Siemens apply water scrubbing only (figure 12–5), whereas gasifiers with heat recovery such as Shell (figure 12–6), E-Gas, FB, and so on normally use dry fly ash/char removal steps and water scrubbing for final dust and ammonia removal. Full quench systems, therefore, are simpler with regard to particle removal.

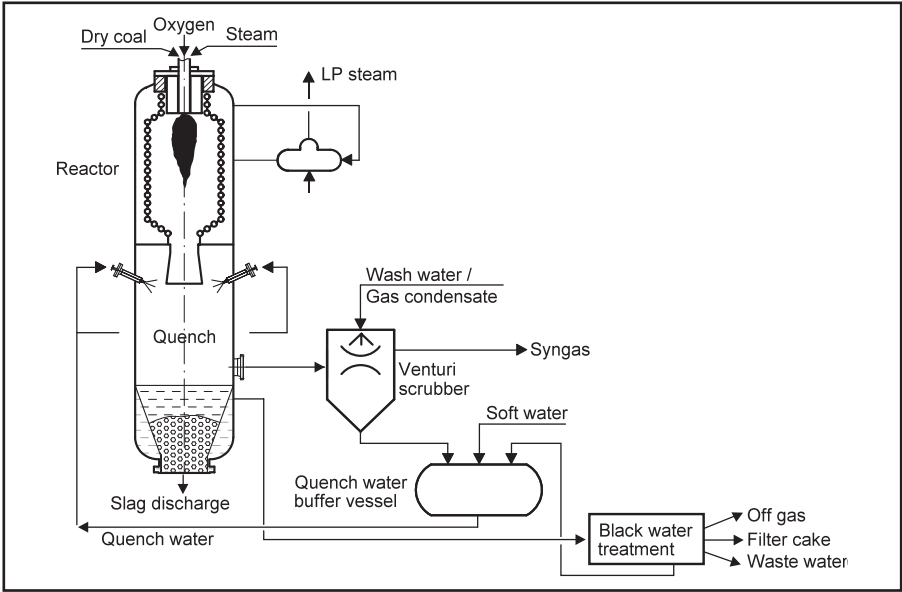


Figure 12–5 Schematic of Siemens full water quench gasifier with water scrubbing

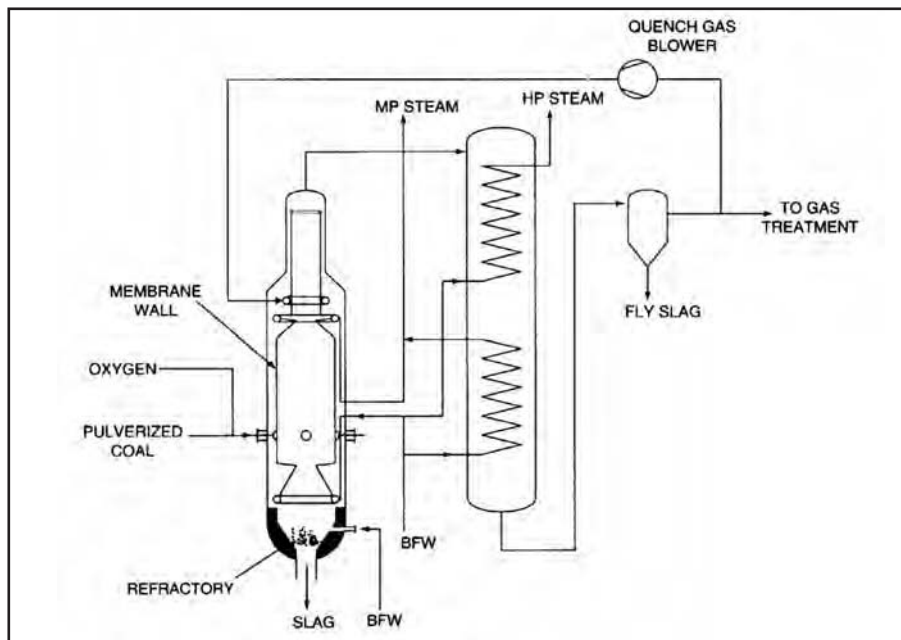


Figure 12–6 Schematic of shell coal gasification process with heat recovery and dry fly ash removal

Acid Gas Removal (AGR)

The next step in the syngas cleaning chain is acid gas removal to take out sulfur compounds such as H_2S (hydrogen sulfide) and COS (carbonyl sulfide), and if required also CO_2 . In case of CO_2 removal an additional catalytic shift reactor is needed to convert CO into CO_2 .

AGR systems currently employed in IGCC application are regenerable solvent-type processes (figure 12–7). The following three types of processes are known.

- Chemical solvent processes (e.g., Methyl-diethanolamine (MDEA), or ADIP)
- Physical solvent processes (e.g., Rectisol, Selexol)
- Mixed chemical/physical solvent processes (e.g., FLEXSORB)

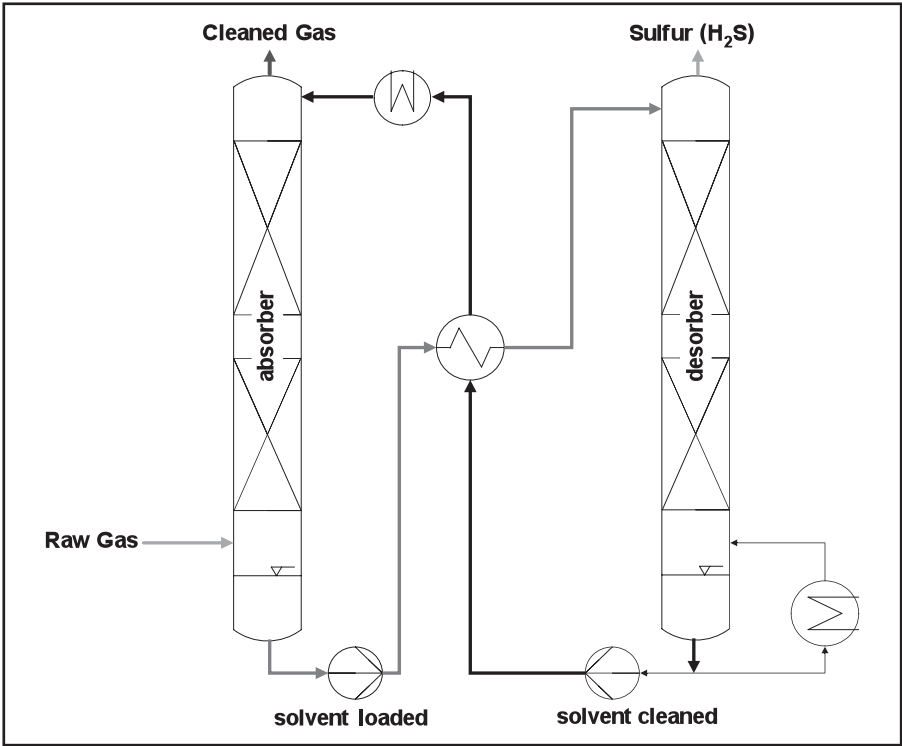


Figure 12–7 Regenerable solvent-type AGR process

AGR processes for IGCC application without CO₂ capture target to leave as much CO₂ as possible in the syngas to maximize the gas turbine syngas flow and power output. Thus, high selectivity for H₂S and COS is needed. Numerous studies concluded that IGCC application favors processes using MDEA solvent as a low capital cost option. Because MDEA is a selective H₂S removal process only, a COS hydrolysis (catalytical conversion to H₂S) needs to be applied before.



An alternative promising AGR process applied in several gasification projects is the SELEXOL process. SELEXOL as a physical solvent process is more effective with increasing syngas pressure because its solubility directly depends on partial pressure and follows Henry’s law.

The sour gas (H_2S enriched) leaving the desorption column of the AGR process is further processed in sulfur recovery and tail gas treating units. Sulfur recovery involves H_2S conversion into elemental sulfur or alternatively to sulfuric acid (H_2SO_4). However, the most common technology for sulfur recovery is the Claus process that produces liquid sulfur by means of multiple catalytic conversion. Today air- and oxygen-blown Claus processes are used. If the H_2S feed content is low, modified Claus processes such as CLINSULF can also be applied. The overall chemical Claus reaction is:



A simplified Claus process is shown in figure 12–8.

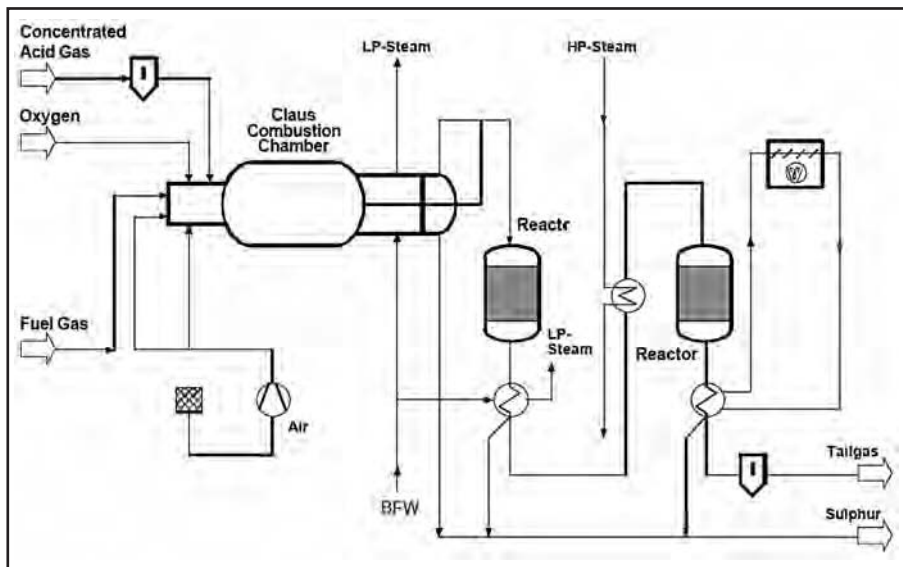


Figure 12–8 Claus process

Because Claus plants typically have sulfur recovery rates of less than 99%, the tail gas leaving the process still contains some sulfur and needs further processing. Depending on the sulfur emission level allowed, it can be fed to an incinerator, recycled upstream of the COS hydrolysis unit after hydrogenation of the Claus tail gas, or converted via additional hydrogenation/hydrolysis units.

Alternative and Advanced Gas Cleaning Technologies

Alternative processes to AGR and subsequent tail gas treating are direct oxidizing desulfurization methods such as SulFerox or LO-CAT. Both technologies use iron as oxygen carrier to directly convert H_2S into elemental sulfur, which is then filtered to produce sulfur filter cake.

Further technology development for syngas cleaning focuses on warm gas desulfurization processes, which target higher IGCC efficiency and cost reduction. These processes operate at temperatures of 400 to 500°C, which means that the syngas-cooling step can be limited. It allows feeding the gas turbine with syngas temperatures well above 400°C. The hot and warm gas reactor of such processes makes use of H_2S adsorption with metal oxide sorbents such as Zn or Ni. For continuous operation the reactants have to be regenerated by contact with air in a separate vessel. Although these developments have been tested successfully for sulfur removal, other contaminants (e.g., alkalis) are difficult to capture, and harm the gas turbine lifetime.

CO_2 capture becomes more and more important and leads to zero-emission IGCC concepts (ZEIGCC). CO_2 capture is done in two steps, shifting of CO to CO_2 and subsequently capturing the CO_2 by using AGR solvent processes. Future ZEIGCC concepts could also use more advanced membrane systems.

Depending on capture rate and selected gasification technology, two different shift reactor systems are commonly applied: high temperature sour gas shift (HT sour shift) or low temperature sweet gas shift (LT sweet shift). HT sour shift reactors achieve a CO conversion rate down to about 1 to 5 vol% residual CO, and LT sweet shift reactors down to about 0.05 to 0.5 vol% residual CO. Although LT sweet shift reactors are more efficient, their catalyst price is higher and operation much more complex. Thus, LT sweet shift reactors are applied only if a high carbon capture rate is required, or if the syngas has high methane content that cannot be converted to CO_2 .

The water gas shift reaction is described as:



An H_2O to CO ratio of about 1.1 to 1.3 is required to convert CO to CO_2 . The water gas shift reaction is an exothermic reaction producing sensible heat, which can be used for HP or IP steam generation. Hence chemical energy is disposed causing lower gas turbine throughput and significant efficiency losses compared to standard IGCC processes. The conversion steam can either be introduced by full water quench (GE, Siemens gasifier) or additional IP steam injection or syngas saturation. For IGCC with CO_2 capture entrained-flow gasifiers with water quench are the preferred solution because of simpler process design and lower investment cost.

After-shift reaction, the syngas consists mainly of H_2 and CO_2 , and the CO_2 can then be removed by a standard AGR process. It has been shown that CO_2 capture can most preferably be done by simultaneous sulfur and CO_2 removal using a physical absorption process (figure 12–9) such as Rectisol or Selexol. These processes consist of one column for CO_2 absorption and a second column for sulfur removal. The desorption process separates the $\text{H}_2\text{S}/\text{COS}$ from CO_2 and feeds the sulfur gas to the Claus plant. The CO_2 is sent to further compression or liquefaction units.

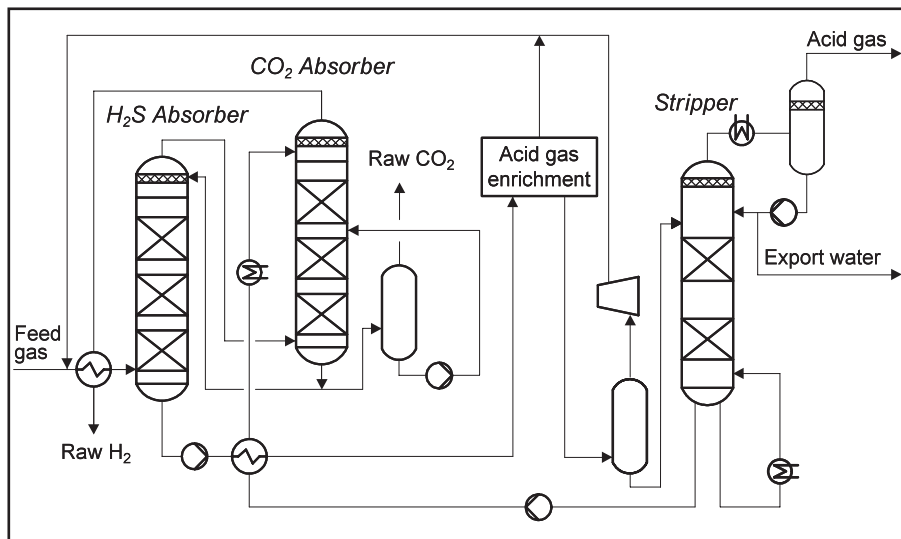


Figure 12–9 Physical absorption process for simultaneous sulfur and CO₂ removal

For IGCC application with CO₂ capture the Rectisol process is mainly considered today. The Selexol process shows a number of similarities to Rectisol, but leads to lower capture rate and an additional COS hydrolysis step caused by relatively poor selectivity between COS and CO₂. Another process available for CO₂ capture is an activated MDEA.

Air Separation Technology

Nowadays, almost all IGCC concepts are based on oxygen-blown gasifiers because they offer a number of advantages. The oxygen is supplied by an air separation unit (ASU). The oxygen is mainly used for gasification, and the pure nitrogen is used for feeding, pressurization, and flushing after shutdown. For IGCC cryogenic air separation units are used. Typical O₂ purities are approximately 95 vol% and N₂ purities are >99 vol%. Remaining nitrogen is used for syngas dilution and control of gas turbine NO_x emission. The cryogenic ASU (figure 12–10) consists of a cold box where oxygen and nitrogen are separated, a molecular sieve to remove CO₂ and water vapor and for downstream compression

Combined-Cycle Technology and Integration of Gas Turbine and Air Separation Unit

The combined cycle part of an IGCC is normally based on a three-pressure reheat cycle with additional water/steam side interfaces to a gasifier, gas cleaning, and air separation unit. Differences to standard combined-cycle configurations can be found in gas turbine modifications to accomplish compressor air bleed, higher turbine mass flow, and additional systems for syngas dilution and preheating. These adaptations are introduced to minimize NO_x emission and combustion instabilities but also to maximize IGCC efficiency. A syngas conditioning system is installed prior to the gas turbine inlet (figure 12–11). This system has the objective to first dilute the syngas with nitrogen and then to humidify the diluted gas with water vapor. Heat is needed for syngas saturation to evaporate the water, which can be recovered from gas turbine air bleed or supplied from the gas island or the combined cycle water/steam system. If air bleed is selected the sensible air heat is recovered in two steps, first for steam generation and second as heat supply for saturation. After syngas saturation, the diluted syngas is preheated to enhance overall IGCC efficiency.

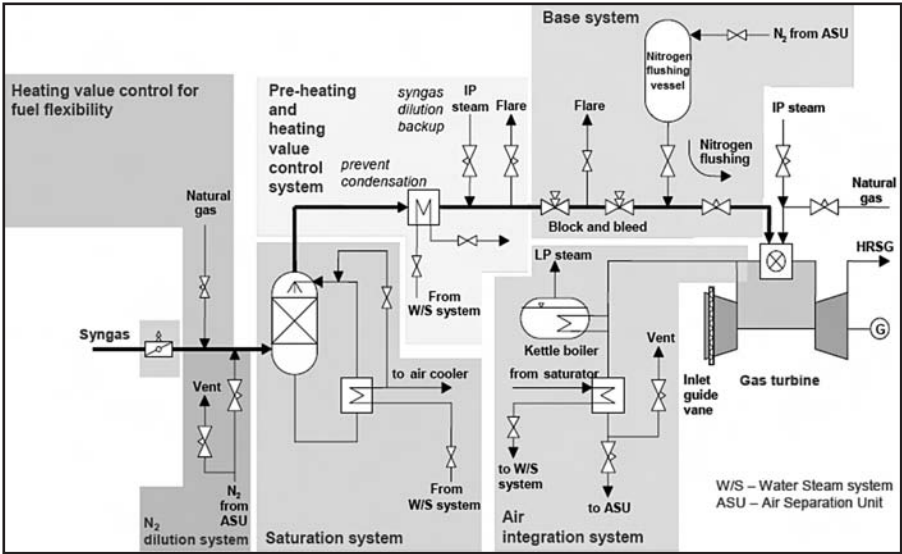


Figure 12–11 Gas turbine syngas conditioning system (Siemens concept)

Syngas dilution is primarily required for NO_x control. For this purpose, water vapor reduces flame temperature peaks and NO_x emission more efficiently than nitrogen because of its higher specific heat capacity (figure 12–12). The level of syngas saturation has a direct impact on IGCC efficiency and power output, which can be maximized by low temperature heat utilization through adding water of temperature between 150°C to 200°C up to a water content of approximately 10% (mass in syngas). In case a high syngas water content is required to reduce NO_x emission, the water temperature supplied to the saturator needs to be increased, which in turns lowers steam turbine power output and plant efficiency.

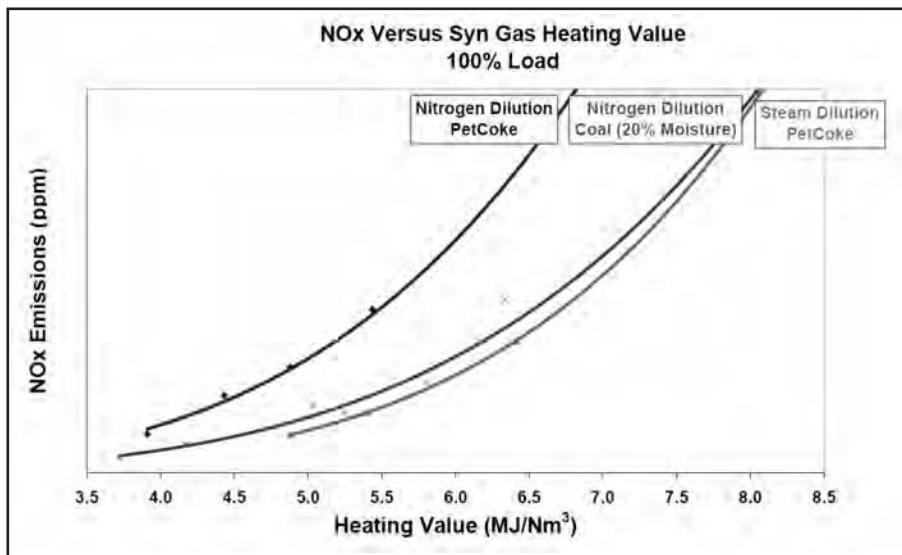


Figure 12–12 Effect of dilution and heating value on NO_x emission

After syngas dilution, saturation, and preheating, the syngas is fed to the gas turbine combustor. Compared to natural gas combustion, syngas combustion differs in various parameters, which require gas turbine modification with special focus on combustor design (figure 12–13). Advanced natural-gas-fired gas turbines are designed with pre-mix combustion systems to minimize thermal NO_x emission, but syngas contains significant amounts of hydrogen that cause higher flammability and risk of flashback and burner overheating.

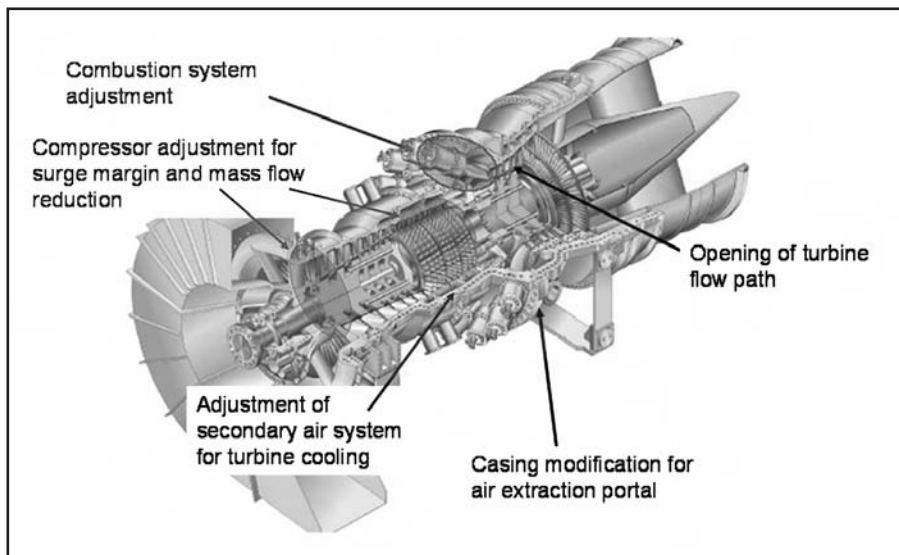


Figure 12–13 GT modification for syngas application

For these reasons, diffusion-based combustion systems are currently used for syngas utilization. Syngas pre-mix combustors are under development and might be available for the next generation of gas turbine applications. Although diffusion combustors offer higher flame stability than pre-mix combustors, the fuel/air mixture is inhomogeneous and leads to higher combustion temperatures and NO_x formation, respectively. Syngas dilution helps to bring NO_x emissions down to <25 ppm. By dilution, syngas heating values are in the range of 4 to 10 MJ/kg, depending on diluent type and turbine inlet temperature. Thus, large fuel pipes and flow control valves as well as an increase of turbine mass flow and corresponding imbalance of compressor and turbine section compared to the standard natural gas design have to be considered (figure 12–14). The higher turbine mass flow leads to a higher pressure drop over the turbine section and a higher compressor exit pressure.

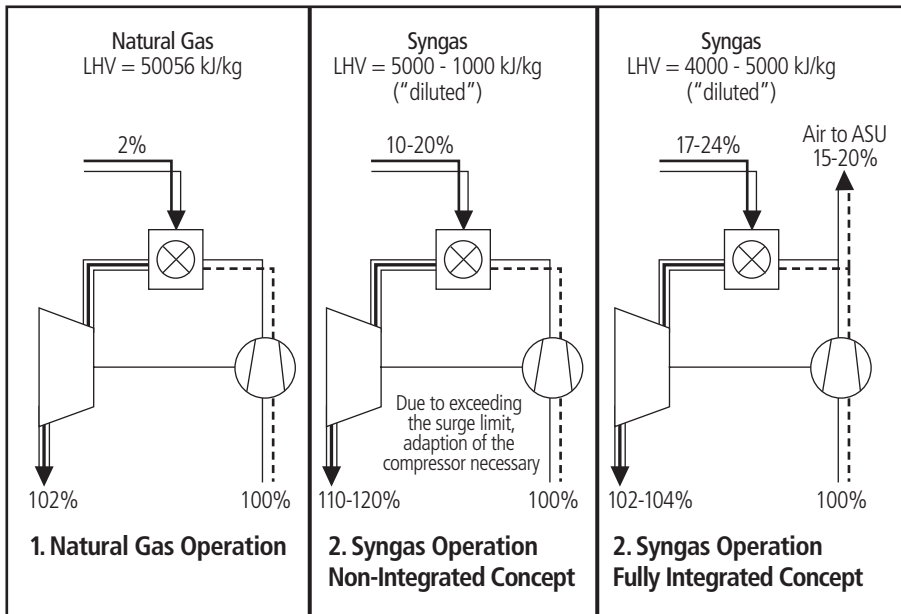


Figure 12-14 Gas turbine mass flow imbalance

With increasing compressor exit pressure, the surge limit of gas turbine compressors is approached or can even be exceeded. Under these conditions the risk of abrupt reversal airflow of compressor blades increases, which would harm the gas turbine compressor section. Beside surge limit, the increasing compressor exit pressure also affects the pressure profile of the gas turbine compressor and has an impact on the gas turbine cooling system. Therefore, compressor or turbine modification is required; for example, adding stages and staggering critical stages to enable high pressure ratio at lower mass flow, or opening the first turbine vane stage to reduce compressor exit pressure at given turbine mass flow. Alternative solutions to overcome these modifications are part load operation with throttled inlet guide vanes or air bleed to ASU at compressor exit. This has been done for both European IGCC plants at Buggenum and Puertollano where the gas turbine compressor supplies all the air needed in the air separation unit.

Conceptual Design of Solid Fuel IGCC Plants

Although IGCC application can be designed for various solid or liquid fuels and can be used for electricity generation only or co-production of chemicals, the main market is expected to push IGCC applications based on coal and used only for power generation.

Differences not only among gasifier technology but also various options of gas cleaning with and without CO₂ capture offer a number of IGCC concepts today. Four exemplary IGCC concepts with and without CO₂ capture are discussed in the following section, which covers coal diversity and differences in gasification technologies. HTW (high temperature winkler) and Shell gasification have been selected for demonstrating IGCC concepts without CO₂ capture, and Shell and Siemens gasifier technology for concepts with CO₂ capture.

IGCC Concept Based on Hard Coal and Shell Gasification and Siemens Combined-Cycle Technology

The Shell-based IGCC plant concept has been derived from the existing demonstration plant at Buggenum, but improved by using advanced gas turbine F-class technology as well as MDEA gas cleaning and partial air-side integration between ASU and gas turbine to enhance operational flexibility (figure 12–15). Heat and mass balance calculation are based on ISO ambient conditions, and a typical hard coal such as Douglas Premium is assumed.

The hard coal must be milled and dried before entering a Shell gasification process where it is converted to syngas. The hot syngas leaving the Shell gasification reactor is quenched with recycled raw gas down to about 800°C to 900°C and subsequently further cooled to about 250°C while generating high-pressure and intermediate-pressure steam. Particle removal takes place by use of ceramic filter units, and the raw gas is further processed by Venturi scrubbing to remove alkali and halogen contaminants. About 80% of the initial chemical heat

The oxygen needed for Shell gasification process is supplied by a conventional ASU process, which is integrated with the gas turbine on air and waste nitrogen side. All the waste nitrogen separated from compressed air returns for syngas dilution, and about 50% of the air input is supplied by the gas turbine compressor leading to a partially integrated concept. Energy losses by ASU have been estimated to about 5.3% and belong to cooling water consumption.

The diluted syngas enters the gas turbine combustor where the remaining chemical heat is converted to electricity and sensible heat of leaving flue gas. Based on original chemical heat input of coal the gas turbine efficiency can be estimated to about 27.3%, assuming a typical F-class engine with turbine inlet temperature of 1230°C (ISO). The flue gas leaving the gas turbine has a temperature of more than 600°C and is used for steam generation within a triple-pressure reheat HRSG configuration. The HP and IP saturated steam imported from the gasification island is fed to the HP and IP drum, and jointly superheated with internal generated steam fraction. To accommodate the saturated steam import from gasifier island, the HP/IP superheater section is oversized for secondary fuel operation and, therefore, higher water consumption for steam attemperation is needed if the gas turbine is operated with natural gas or fuel oil. In coal gas operation the steam turbine power output is about 45% higher due to steam generation in gasifier heat recovery and results to approximately 19.1% relative to entire chemical heat input (figure 12–16). To achieve the highest IGCC efficiency level while limiting NO_x emission, the gas turbine is integrated with the air separation unit on the air and nitrogen side. Nitrogen dilution is used to reduce NO_x emission, but increases turbine mass flow. To compensate the resulting turbine imbalance, part of compressed air is extracted and supplied to the air separation plant.

IGCC Concept Based on Lignite and HTW Gasification and Siemens Combined Cycle

The most mature pressurized FB technology is the high temperature winkler (HTW) process, which was tested on demonstration scale at Berrenrath, Germany. Because of its high efficiency potential overall performance estimations based on ISO condition, a blended German

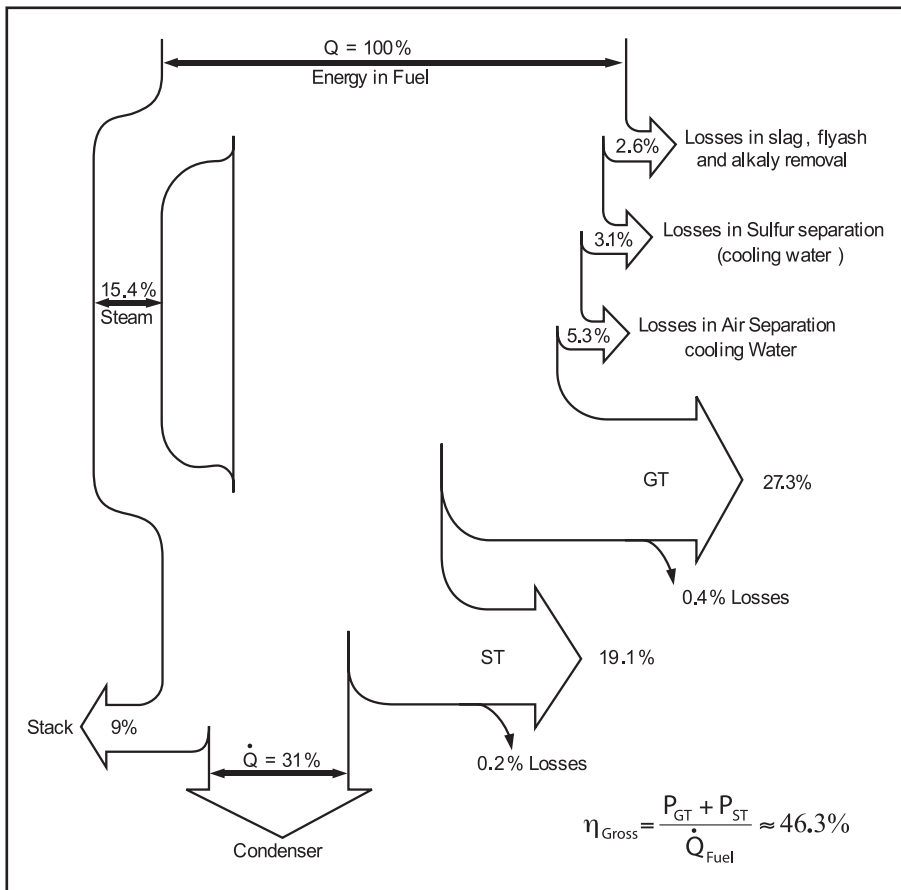


Figure 12–16 Sankey diagram of Shell-based IGCC concept

lignite coal and the Siemens combined-cycle technology have been performed. Although more efficient than entrained flow IGCC application, the FB technology is characterized by a limited carbon conversion rate and a carbon-containing ash leaving the bottom of the gasifier (figure 12–17). The bottom product cannot be disposed of and is used for HP steam generation in an additional fluidized bed combustor. The superheated steam is exported to the HP steam section of the combined cycle HRSG.

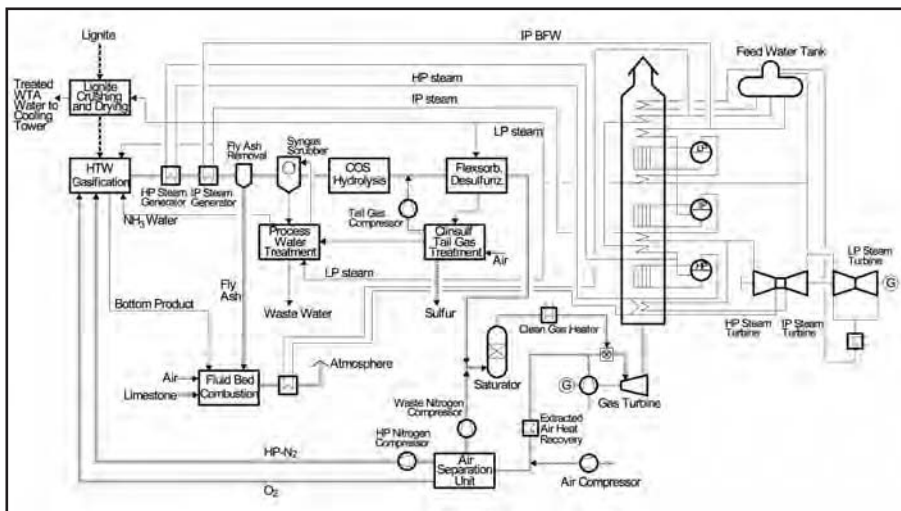


Figure 12–17 IGCC concept based on lignite and HTW gasification and Siemens combined cycle

Lignite applications suffer from the high moisture content of the fuel of typically more than 54%. Establishing an effective lignite gasification at the HTW gasifier a moisture level of <12% is required, which cannot be achieved by conventional coal milling and drying systems. Therefore, fluidized bed drying systems such as WTA from RWE have been developed to effectively treat the lignite before feeding the gasifier. The WTA process recycles the evaporated steam coming out of the drying process for heat input into the system after compression. For balancing energy losses, LP steam from combined cycle is used and integrated. The crushed and dried lignite leaving the WTA unit enters the fluidized bed of the gasifier and reacts with oxygen and IP steam at about 1000°C but below ash melting point. The produced hot syngas leaves the reaction zone and contains still significant amounts of carbon and fly ash, which are separated in a cyclone and re-injected into the fluidized bed. After coarse ash removal, the syngas is cooled down to about 250°C in a heat recovery generator where saturated HP/IP steam is produced. Downstream of the syngas cooler, an additional fine ash removal filter is situated to separate the remaining fine ash from the syngas stream. Total losses of coal conversion and mechanical cleaning process are about 1.4%, and 10.6% of chemical energy is converted to steam. Further treating

steps are needed after syngas cooling and de-dusting, starting with water scrubbing to remove trace solids, ammonia, and acids. Afterwards the syngas enters the COS hydrolysis unit and COS is converted into H_2S , which than is removed by a FLEXSORB SE+ absorption process. FLEXSORB SE+ represents a very selective chemical solvent well suited to remove low H_2S concentration. After desorption the H_2S is fed to a CLINSULF reactor (comparable to Claus process; see figure 12–8) and converted to elemental sulfur while the off gas leaving the CLINSULF unit is converted back to H_2S by hydrogenation and returns upstream to the FLEXSORB SE+ unit. Beyond sulfur removal the FLEXSORB unit also absorbs naphthalene, which is formed in the HTW gasifier due to its lower gasification temperature. The total energy losses of sulfur removal amounts to about 1% and of air separation to about 3.3%.

The combined cycle in case of HTW gasification represents a highly integrated system of importing and exporting steam and water on different pressure levels and temperature condition. Although the entire IGCC net efficiency level is higher than for Shell gasification (see table 12–4), the efficiency benefit is compensated by higher investment and system integration in case of lignite application. Gas turbine integration on the air and nitrogen side has been considered similar to the Shell case.

Table 12–4 Investment and efficiency of IGCC concepts

	Shell based IGCC (hard coal) (50 Hz)	HTW based IGCC (Lignite) (50 Hz)	Shell based IGCC with CO_2 capture (50 Hz)	SFG based IGCC with CO_2 capture (50 Hz)
Net power output	874 MW	826 MW	737 MW	681 MW
Gross power output	986 MW	931 MW	956 MW	864 MW
Gasification island	36%	36%	32%	23%
Gas cleaning	6%	9%	13%	15%
ASU	7%	5%	7%	8%
Combined Cycle	28%	27%	24%	28%
BoP and Owners cost	23%	23%	24%	26%
Spec. total costs	1500 €/KW	1600 €/KW	2050 €/KW	1850 €/KW
Net Efficiency (LHV)	46,3%	51,7%	35,9%	34,5%
Net Efficiency (HHV)	44,5%	43%	34,4%	33,1%
Coal	Douglas Premium 2	Lignite	Douglas Premium 2	Douglas Premium 2

IGCC Concept with CO₂ Capture Based on Hard Coal and SFG Gasification and Siemens Combined Cycle

The SFG technology is currently based on a dry feeding system and a full water quench for syngas outlet cooling to about 200°C and leads to a syngas water content of more than 50 vol% when the syngas leaves the gasifier quench section. SFG gasification with partial water quench system and heat recovery can also be applied for future application presently under development, which in general leads to higher IGCC efficiency levels. But if CO₂ capture is required more than 50% syngas water level is needed to perform the water gas shift reaction from CO into CO₂ and H₂. In this case a full water quench solution such as SFG gasification offers the most economical and simplest IGCC configuration. No or limited steam injection is needed to accommodate the water gas shift reaction and the Siemens IGCC concept with CO₂ capture comprises only few interfaces between the combined cycle and gas island, which are shown in figure 12–18.

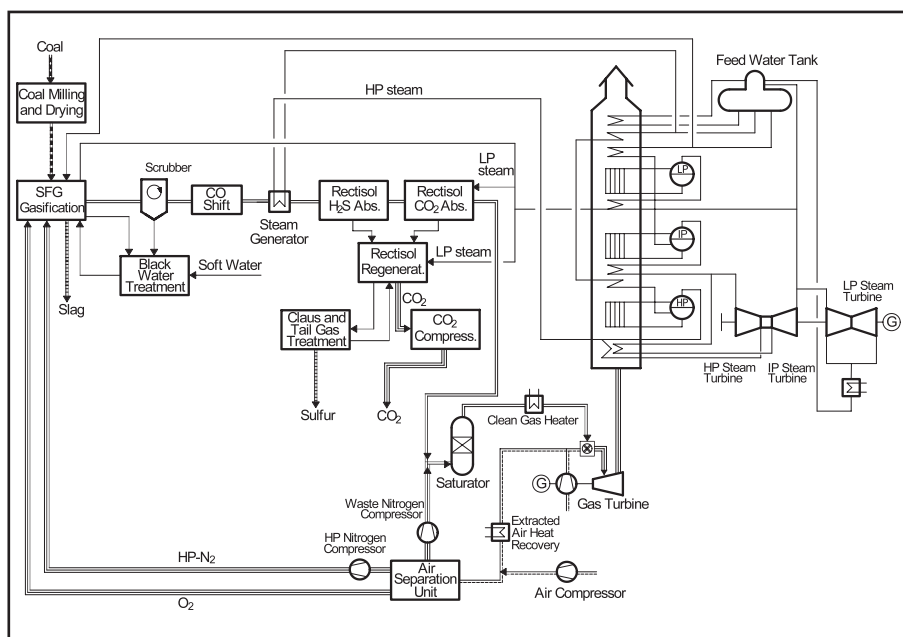


Figure 12-18 IGCC concept with CO₂ capture based on hard coal and Siemens fuel gasifier (SFG) and Siemens combined-cycle technology

ISO ambient condition, a typical hard coal such as Douglas Premium, and 90% CO₂ capture have been assumed for performance estimation of an IGCC concept based on SFG technology. The CO₂ is assumed to be compressed to about 110 bar.

Before gasification the hard coal is milled, dried, and fed to the gasifier where the coal reacts with oxygen and steam under lambda values of 0.4 to 0.5. After gasification and water quench, the syngas is cleaned by water scrubbing to remove particles and acids, and is then fed to a CO raw gas shift reactor to convert CO into CO₂ and H₂ by exothermic water gas reaction. After leaving the shift reactor, the syngas contains about 42 vol% H₂, 30 vol% CO₂ and still 24 vol% H₂O which depends on water/gas ratio and uses catalyst material (iron or cobalt molybdenum). The syngas temperature rises during shift reaction to more than 500°C and HP steam can be generated and exported to the combined cycle. After shift conversion, the CO₂, H₂S, and COS as well as other syngas traces are removed by a two stage Rectisol process. The sulfur compounds are absorbed in a first column and converted to elemental sulfur in an OxyClaus process followed by a tail gas treatment—hydrogenation—plant as sulfur recovery process. The CO₂ is removed in a second column, separated from solvent (methanol) by flashing on lower pressure levels, and compressed for transport and sequestration. The syngas leaving the Rectisol process contains of about 85 vol% hydrogen and needs to be diluted with nitrogen and water vapor (saturator) to control NO_x emission and reduce flame speed during gas turbine combustion. To enhance IGCC efficiency an additional syngas preheating step is assumed before the hydrogen-rich syngas enters the gas turbine combustor. Main energy losses are caused by CO conversion, CO₂ removal, and compression. The entire IGCC net efficiency level is estimated to be about 34.5% based on F-class gas turbine. Integration of gas turbine and air separation unit are considered to be comparable to IGCC cases without capture. The water steam cycle is based on a three-pressure reheat with only few interfaces to gasification and gas cleaning. Thus the combined-cycle operation does not change between syngas/hydrogen and secondary fuel operation.

IGCC Concept with CO₂ Capture Based on Hard Coal and Shell Gasification and Siemens Combined-Cycle Technology

The Shell gasification technology for IGCC with CO₂ capture corresponds to the standard Shell application, and performance estimations are based on ISO ambient condition, a typical hard coal and 90% CO₂ capture with CO₂ compression to about 110bar (figure 12–19). The syngas leaves the Shell gasification island after water scrubbing and contains than about 9 vol% water vapor, 57 vol% CO, and 23 vol% hydrogen, which needs to be converted to hydrogen and CO₂ for downstream CO₂ capture. To accommodate the water gas shift reaction the syngas needs to be saturated first to about 50 vol% water vapor. This can be done by adding a cooler/saturator cycle that transfers the heat for water evaporation from shift reactor outlet to inlet side and by additional IP steam injection to cover the remaining steam requirement. Thus the sensible heat generated during exothermic water gas reaction is used efficiently, but significant investment and higher plant integration need to be considered, too.

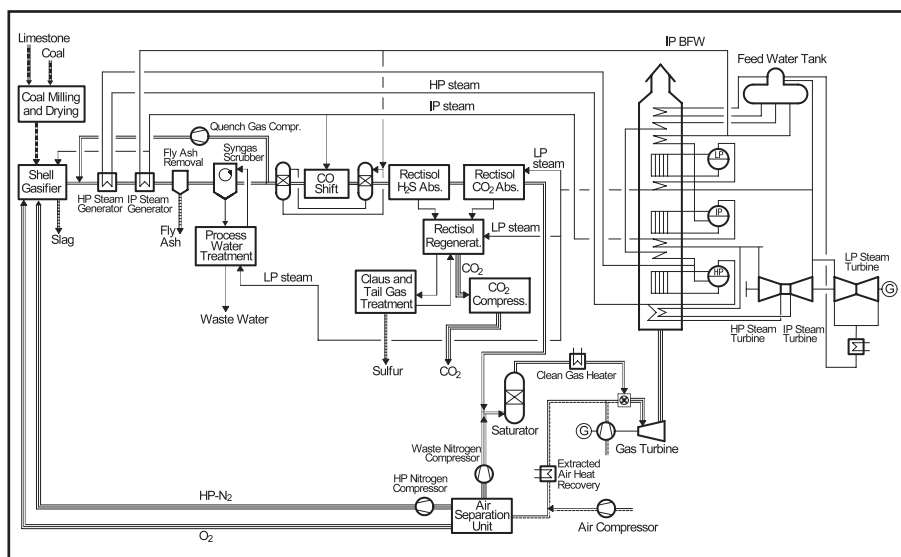


Figure 12-19 IGCC concept with CO₂ capture based on Shell gasification and Siemens combined-cycle technology

The shifted gas is then fed to a Rectisol process where CO_2 and sulfur compounds are removed in two separate columns. The process design of gas cleaning and conditioning corresponds to the SFG concept consisting of dilution, saturation, and preheating of hydrogen-rich syngas and the energy losses of CO shift, CO_2 , and sulfur removal are in the same range as the previously described SFG concept. Hence, the total efficiency is slightly higher and results in about 35.9% due to high temperature heat recovery from the raw gas (figure 12–20). But compared to the previous concept the Shell based IGCC design leads to higher investment and higher plant integration. The water steam cycle represents a three-pressure reheat design and is highly integrated between gasification and gas cleaning. Thus the heat exchanger surface of gas turbine heat recovery generator is basically designed for syngas operation leading to different operational concepts between syngas/hydrogen and secondary fuel operation.

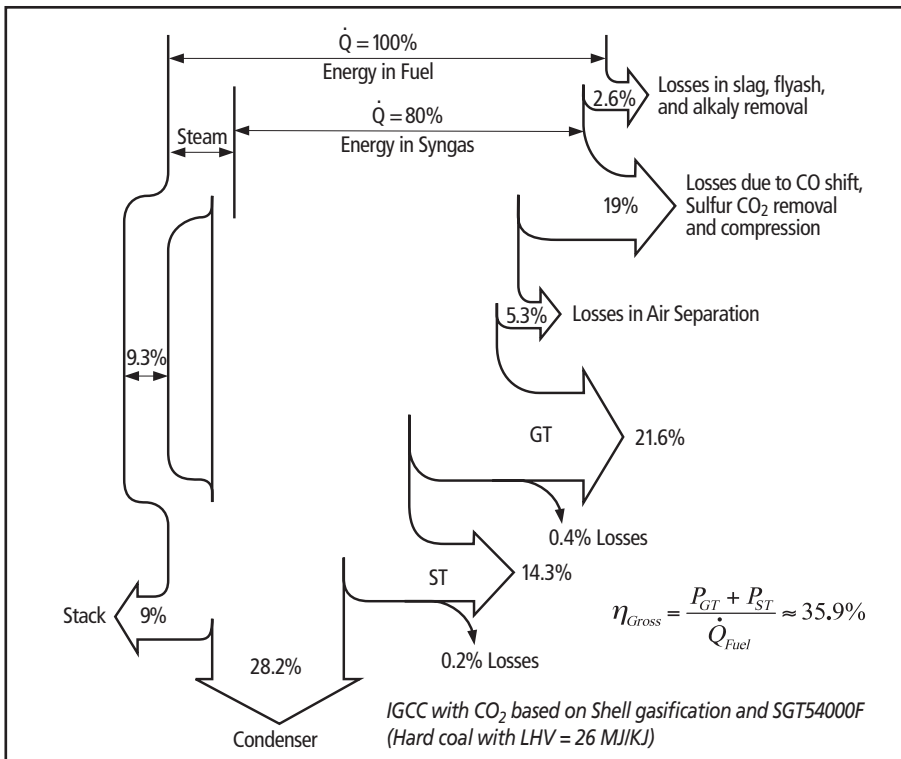


Figure 12–20 Sankey diagram of Shell-based IGCC concept with CO_2 capture

Other IGCC concepts with and without CO₂ capture

Apart from Shell, SFG, and HTW concepts with Siemens combined-cycle technology, other IGCC concepts based on GE energy or MHI gas turbine technology as well as CoP, GE and MHI gasification technology are well known. The IGCC concept of GE energy is primarily based on GE's gasification technology, which is currently in operation in Polk County, Florida, and consists of a coal slurry gasification process with radiant heat recovery for first syngas cooling. For gas cleaning a sulfinol washing unit and GE's advanced F-class gas turbine technology is considered. In case of configuration without CO₂ capture, the plant produces a net power output of about 640 MW; if the plant is designed for 90% CO₂ capture, about 555 MW can be achieved when a two-train concept is assumed.

Another well-known IGCC concept based on F class gas turbine technology is the CoP E-Gas IGCC plant. The two-train concept is equipped with either an MDEA gas cleaning unit for IGCC plant concepts without CO₂ capture that produces a net power output of 623 MW, or a sulfinol-based gas cleaning unit for configurations with CO₂ capture and 518 MW, respectively. Due to the higher syngas methane content of CoP E-Gas gasification, a CO₂ capture rate of 88% only is practicable. The third IGCC concept available today is based on MHI gas turbine and gasification technology. The first 250 MW demonstration plant is situated in Nakoso, Japan, and commercialization is planned with IGCC concepts of 500 to 650 MW sizes based on G-class gas turbine technology and an MHI two-stage air-blown gasifier.

Summary and Investment Cost of IGCC Concepts

Current IGCC concepts consist of different gasifier, gas cleaning, and combined cycle technologies, and their competitiveness is very much depending on feedstock and CO₂ capture rate. Under predefined condition the four selected IGCC concepts represent optimal solutions. For lignite application with CO₂ capture SFG and Shell based IGCC solutions are suitable, too. Alternative IGCC solutions such as GE and CoP E-Gas™-based IGCC application are preferred for high-rank coals and not optimal for low-rank coals because of their slurry feeding limitations.

Today's investment cost calculation for IGCC application are based on front-end engineering design (FEED) studies, which leads to uncertainties of about $\pm 20\%$. IGCC still presents a new power generation technology with only few demonstration plants in operation. Thus cost basis for components, units, and contingencies are tentative and estimated on database available 2006. Moreover, the price increase of about 30% to 40% due to economy growth between 2005 and 2007 resulted in higher cost uncertainties, and delayed a number of investment decisions.

It can be concluded that a number of IGCC concepts available today vary with regard to performance and investment costs, depending on selected technology, coal quality, and emission requirements; 12% to 30% (with CO₂ capture) of gross power output is internally consumed for syngas generation and gas compression. Therefore, efficiency numbers of IGCC application are generally not applicable, depending on selected technology, component design, and integration features. Although natural gas compression cannot be neglected as well for natural-gas-fired combined-cycle plants, it is typically not considered for efficiency calculation.

IGCC in total suffers on high investment but also offers a high potential for cost reduction and further efficiency improvement when the latest features of gas turbine, air separation, and gas cleaning developments are introduced. IGCC applications are promising for future fossil power generation, focusing on low emission and capability for CO₂ capture.

13 Carbon Dioxide Capture and Storage

After decades of expert and public-level discussion, the existence of global warming and its reasons are undisputed to a far extent today. Increasing ambient temperatures, more frequent heat waves, and disastrous storms and floods are indications. Among all natural or technical gases, which are assessed to be responsible for the global warming effect, CO_2 is the most important one. This results, to a lesser extent, from its physical characteristics, but more from the high amount emitted from all carbon conversion processes. The world population has grown more than four times since the beginning of the 20th century, resulting in steeply rising energy consumption (figure 13–1). In addition, the per-capita energy demand increased as a consequence of the growing prosperity in the industrialized countries. As most of the energy was and is based on fossil (carbon-containing) fuels, the atmospheric carbon dioxide (CO_2) concentration accumulated from 280 up to some 370 ppmv today. In parallel, the global average temperature rose by about 0.8K. According to the Intergovernmental Panel of Climate Change (IPCC) fourth assessment report, a doubling of the CO_2 concentration results in an increase of the global mean temperature by 2 to 4.5K, with a best estimate of 3K.

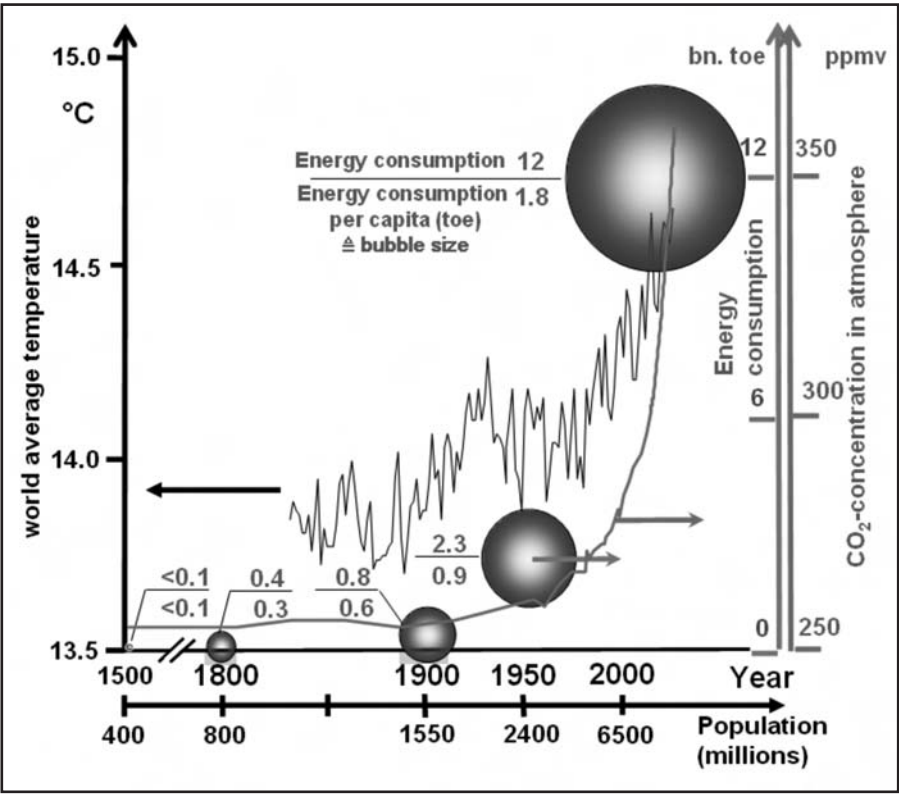


Figure 13-1 Historical trend of energy consumption and global warming

To meet the challenging targets of significant CO₂-emissions reduction a diversified package of measures is needed.

Methods for CO₂-Reduction

In addition to measures, which are aiming at directly reducing the CO₂ concentration in the atmosphere such as biological sinks (photosynthesis, reforestation), there are two fundamental pathways for mitigating the CO₂ problem.

- Efficient production and use of energy
- Substitution of high-carbon fossil fuels by low-carbon/ carbon-free fuels

The combination of the two thermodynamical processes—Joule (Bryton) and Clausius-Rankine cycle (combined cycle)—results in high efficiencies and consequently low CO₂, nitrogen, and sulphur oxides emissions. There is a direct correlation among power plant efficiency, saving of primary energy, and emission of CO₂ in case of fossil-fuel-fired technologies (figures 13–2 and 13–3). Most of the combined-cycle plants in operation today are based on natural gas and consist mainly of methane, the fossil fuel with the highest hydrogen to carbon ratio.

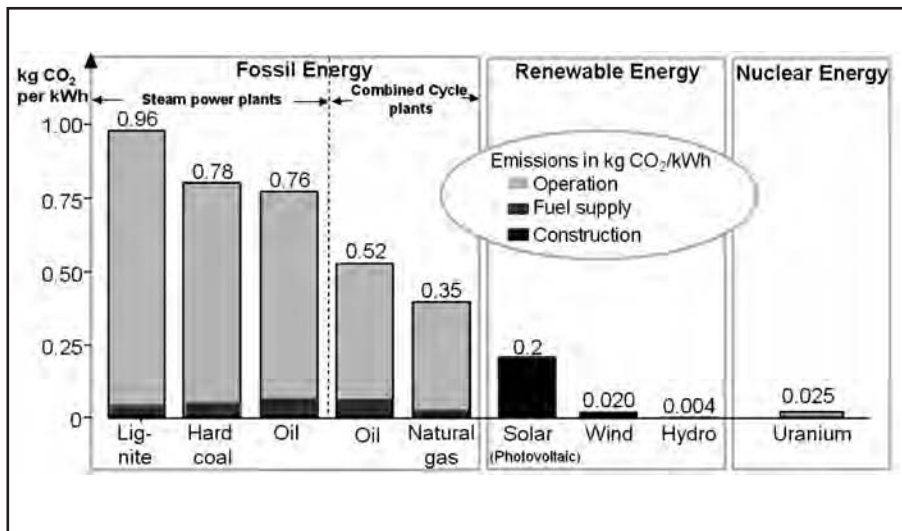


Figure 13–2 CO₂-emissions mitigating by fuel switch

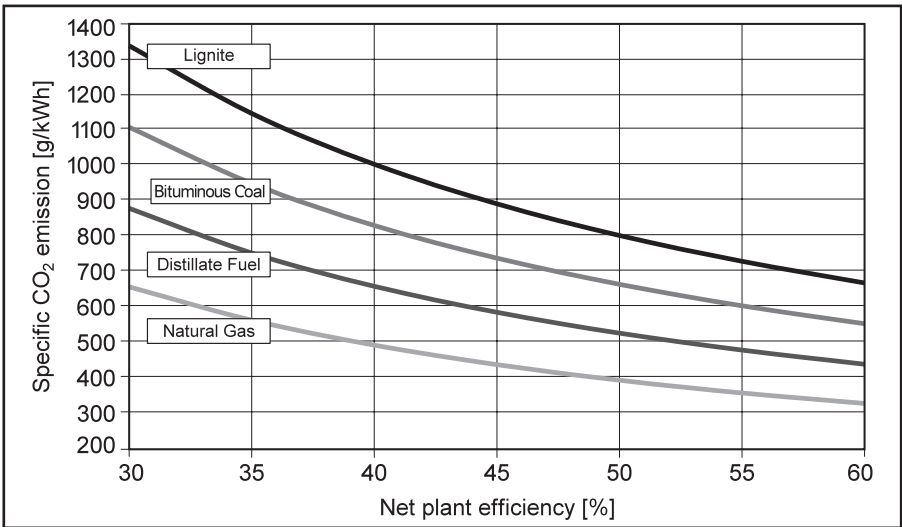


Figure 13–3 Typical specific CO₂-emissions from fossil fuels

Both effects, high efficiency and use of low-carbon fuel, are of advantage with a view to global warming mitigation. On the other side, at least the conventional resources of natural gas are clearly limited and a significant and lopsided switch towards natural gas might be contrary to a safe, independent, and sustained energy supply for many countries worldwide. For this reason, switching to less carbon-intensive fossil fuels doesn't have the potential to significantly reduce the emission of greenhouse gases. In fact, power generation from coal, which is widespread worldwide and available for long-term, has to be taken in an environmental benign way. Beside the development of high-efficient steam-power plants also the combined-cycle technology is to be applied for coal (e.g., Integrated Gasification Combined Cycle, or IGCC).

Reducing the emissions of CO₂ to the atmosphere—carbon capture and storage (CCS)

For that, the CO₂ molecule is in-situ captured and stored. Storage in this connection means the isolation of the CO₂ from the atmosphere for a long time. Carbon capture and storage (CCS) could allow the continued use of fossil fuels, also coal, whereas other CO₂-free energy sources are developed and applied. However, CCS reduces the efficiency

of power generation, and, consequently, such measures need more primary energy. In addition, investment and operational costs rise.

CCS is considered first for large-point CO_2 sources, central power generation plants, and large CO_2 emitting industries such as refineries or chemical plants. In principle, the separated CO_2 can be reused as feedstock for synthesizing chemicals or fuels, possibly using CO_2 -free produced hydrogen. But these applications are not expected to contribute to a significant abatement of CO_2 -emissions. Main options for CO_2 -storage are geological formations, selected areas of the worldwide oceans, and depleted oil and gas fields. An application, which is already a commercial option, is to use the CO_2 as driving medium in the enhanced oil recovery (EOR). By injecting CO_2 the pressure of the oil field is enhanced, and density and viscosity of the oil is reduced. Thus, EOR can potentially increase the total recovered production from an oil reservoir by 10 to 30%. In a similar way CO_2 could be used for enhanced gas and coal-bed methane production.

The concepts for CO_2 separation are divided in three fundamental categories.

- Capture of the carbon fraction before being combusted (precombustion capture)
- Capture of the combustion product CO_2 (post-combustion capture)
- Combustion of the fuel with nearly pure O_2 to get CO_2 enriched flue gas (Oxyfuel fired concepts)

Separation of CO_2 from Gas Mixtures

There are various applications for CO_2 removal in the chemical industry or the long-term purification of natural gas. However, the requirements for CO_2 capture processes when being implemented into power generation concepts are partly different and more challenging: The behaviour of the power plant caused by the grid demand is very dynamic, and the volume flows to be handled are higher. In addition,

oxygen and trace elements such as sulphur or alkali compounds may cause degradation of the solvent.

Sorption

The CO₂ is loaded to a liquid (absorbent) or solid (adsorbent), and released under changed temperature and/or pressure conditions (desorption). With view to the reversible bonding mechanism it is differentiated between physical and chemical sorption with a smooth transition. Physical absorption depends strongly on temperature, and can be used in a very effective way at increased total gas pressures and high content of the compounds to be removed. Due to Henry's law, which is exactly valid for diluted solutions only, the amount of removed substance is nearly proportional to its partial pressure (figure 13–4). As a rule chemical absorbents are more efficient at low gas pressures.

Hereby, an aqueous alkaline solvent, usually an amine, reversibly reacts with the acid CO₂ gas such as for monoethanolamine (MEA):

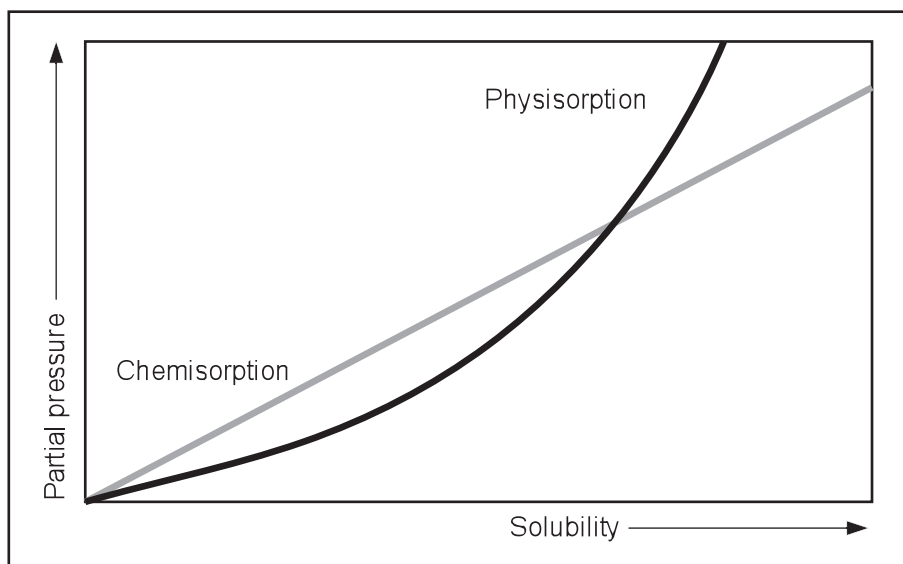


Figure 13–4 Typical trend of solubility in physical and chemical washing agents

Besides cycling the solvent, the main energy consumption is caused by the desorption unit. Depending on capture rate, desorption temperature (reboiler temperature, see figure 13–11) and solvent flow rate, the steam consumption such as for the MEA process was found equal to around 4 MJ/kg CO₂. As a consequence of degradation and loss in activity, some makeup is always needed, which contributes to the operating costs. Particularly for CO₂ from flue gases (post-combustion), new or modified solvents are under investigation for high CO₂ loading, high lifetime, and low energy demand.

Membranes

Membranes for gas separation are thin layers made from porous materials, which are selectively permeable for gases. Plant concepts based on membranes are considered to be less complex and more cost-effective. CO₂-selective membranes are already applied in industry, such as for the treatment of natural gas, but have to be further developed for large-scale plants, high stability, and availability, as well as the application at high temperatures.

Generation of Oxygen

Precombustion and Oxyfuel fired CCS concepts use highly oxygen-enriched air, which results in lower dilution and, thus, improved reaction conditions as well as reduced dimensions of apparatus and pipings (lower investment). On the other side an additional process unit for air separation is needed, which makes the overall plant more complex and increases the internal power demand.

Cryogenic temperature air separation unit

Today, large-scale air separation necessary to provide precombustion or Oxyfuel plants in the order of hundreds of megawatts are based on proven air distillation at cryogenic temperatures. For reduced power demand the product oxygen fraction may be pumped instead of being

compressed. Single-train plants are available for more than 4,000 t/d with typical 95% to beyond 99% purity of the oxygen fraction.

Ion transport membranes

For these innovative concepts the oxygen is transported as an ion through a membrane of adapted lattice structure (figure 13–5). These ion transport membranes (ITM, or specifically OTM in case of oxygen) are highly selective. The active thin layer is manufactured from ceramic material and applied on a porous carrier for mechanical stability. For sufficient separation capacity the ionic mechanism needs high operating temperatures of about 850°C, which can be realized, for example, by directly heating up the air through combustion.

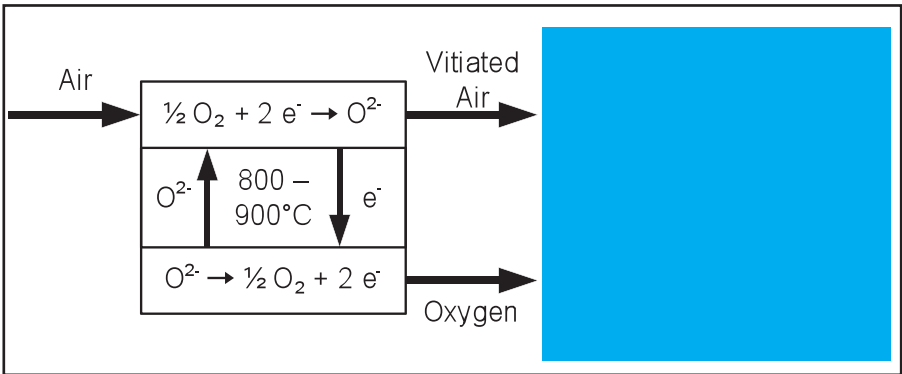


Figure 13–5 Principle of ion transport membrane

Chemical looping

In the chemical looping concept there is no direct contact of combustion air and fuel. In fact, the oxygen is fixed on a solid carrier medium and then moved to another, separate area where it is used for oxidizing. The reduced carrier is cycled back to the air to be loaded again (figure 13–6). As a consequence, no NO_x formations can occur. Typical materials are metals such as iron, nickel, copper and manganese. The process is currently in the state of a pilot plant.

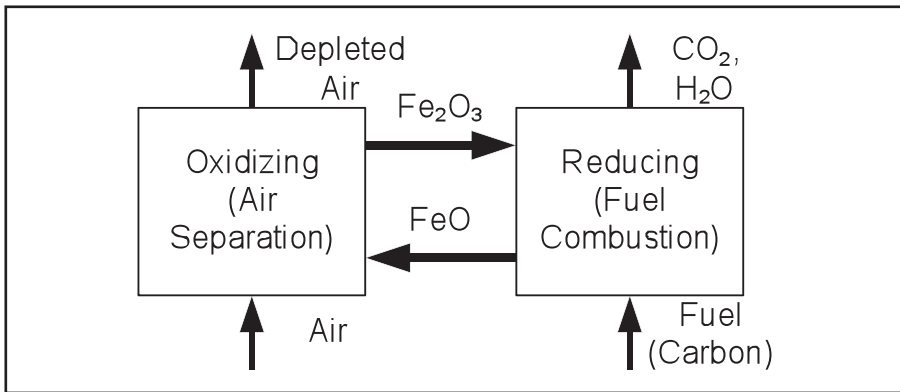


Figure 13-6 Principle of chemical looping combustion

Precombustion Capture

The feedstock is processed in a way to get the carbon content ready for separation. This is realized by reaction of the carbon to CO, which is then converted to CO₂ and hydrogen (figure 13-7).

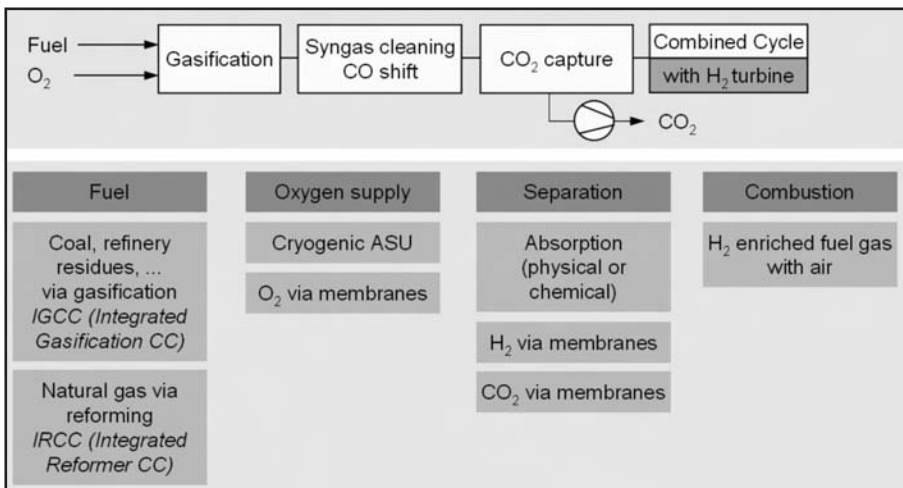


Figure 13-7 Schema of precombustion CO₂ capture

Conversion of carbon to CO

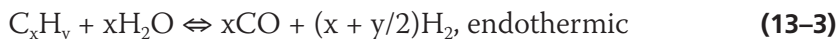
There are two reaction paths resulting in the synthesis gas (syngas).

Partial oxidation (POX), gasification. Reaction with oxygen; typically 1250 to 1500°C (entrained-flow gasification).



The syngas is cooled by quenching (water, recycled gas) and/or through a heat recovery system. By using nearly pure oxygen instead of air, the volumes to be handled at downstream CO₂ separation are reduced.

Steam reforming. Reaction with H₂O; typically 800 to 900°C.



The process is catalysed and indirectly heated. A waste heat system is used for cooling down the products to the temperature needed for the following shift reaction. Steam reforming followed by pressure-swing absorption is today's way to produce highly pure hydrogen from natural gas.

Autothermal reforming (ATR). Combination of partial oxidation and steam reforming (figures 13–2 and 13–3); typically 950 to 1050°C.

The reaction takes place in a refractory-lined vessel containing a nickel catalyst bed. The heat needed for the reforming process is directly produced from partially oxidizing the feedstock. There is no indirect heating by combustion, which would need a second separate CO₂ capture plant when aiming for high capture rates in total. In addition, compared to steam reforming, the process is more compact, favorable for large-scale plants, and the investment costs for the reactor are lower. To prevent carbon soot deposition on the catalyst and to limit methane slip, steam has to be added in excess of the stoichiometric requirements. ATR needs an oxygen generation plant, which also provides nitrogen, if necessary, for moderating the combustion conditions.

Conversion of CO to CO₂ (CO shift). The temperature level for this process depends on the applied catalyst (180 to 500 C). Its sensibility to sulphur compounds needs the shift to be performed up- (raw gas shift) or downstream (clean gas shift) of the desulphurisation unit. To cope with the excess heat of the reaction, a two-stage adiabatic reactor system with intermediate cooling is applied.



Separation of the CO₂

Gasification and downstream combustion of the syngas in a gas turbine take place under elevated pressure levels. Compared to post-combustion capture, the CO₂ partial pressure is higher in the syngas. Thus, the CO₂ separation unit is smaller in size and the specific energy consumption is lower. Physical solvents such as methanol (Rectisol process) are most suitable to such conditions. Depending on the applied shift systems and the requirements for transport and storage, both sulphur compounds and CO₂ are washed out together or selectively.

Use of the remaining H₂ enriched gas for power generation

Independent of the feedstock, also for coal as a solid or liquid hydrocarbon, the precombustion decarbonization process ends with a hydrogen-enriched fuel in gaseous state usable in a gas turbine. This is why precombustion concepts are predestined to be integrated into a combined-cycle process:

In the case of natural gas, the concepts are known as Integrated Reforming Combined Cycle (IRCC) (figure 13–8). For coal or refinery residues, they are called Zero Emission Integrated Gasification Combined Cycle (ZEIGCC). The basic IGCC process is described in Chapter 12. In principle, the modifications needed for carbon capture are limited to the gas treatment and the gas turbine combustion system. But as a consequence mainly of the high steam demand for the CO shift, the interactions between the water/steam cycle and the gas island

are also affected. By reducing this steam export, gasification systems applying a water quench system for cooling the raw gas downstream of the gasifier may be advantageous for ZEIGCC concepts.

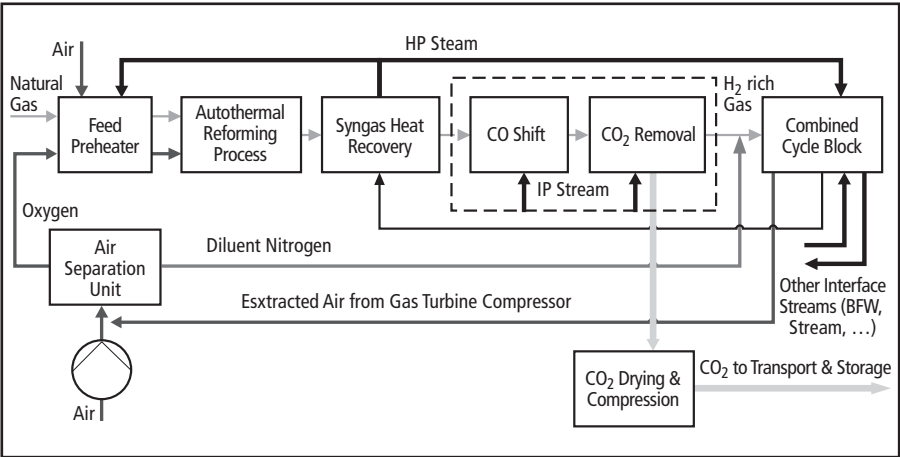


Figure 13–8 Combined autothermal reforming and CO₂-capture

Table 13–1 shows typical compositions of the decarbonized, undiluted synthesis gas from the CO₂ removal for different processes and feedstocks. For the use in the combustion systems of modern high-temperature gas turbines, such hydrogen-enriched gases have to be conditioned.

Table 13–1 Typical hydrogen enriched syngases downstream gas cleaning

		IRCC Natural Gas / ATR	ZEIGCC Hard Coal / SCGP	ZEIGCC Lignite / HTW
H ₂	vol%	95.2	85.3	81.7
CO	vol%	0.7	4.8	0.8
CO ₂	vol%	0.5	0.5	0.1
CH ₄	vol%	1.6		6.6
N ₂ + Ar	vol%	1.4	9.4	10.9
H ₂ O	vol%	0.6		
LHV	MJ/kg	77.2	36.4	41.4

The different combustion behavior of hydrogen compared to the standard fuel natural gas results in special requirements for burner design and gas turbine operation (Table 13–2).

Table 13–2 Characteristics of typical gas turbine fuels

		CH ₄	H ₂	CO
LHV	MJ/kg	50.0	119.9	10.1
	MJ/m ³ STP	35.8	10.2	12.6
Flame speed in air	cm/s	43	350	20
Stoichiometric combustion temperature	K	2227	2370	2374
Density	kg/m ³ STP	0.74	0.09	1.25
Specific heat	kJ/kg K	2.18	14.24	1.05
Flammability limits	vol%	5–15	4–75	12.5–74

Important for the geometric burner design is the substantially smaller density leading to higher volumetric flow rates. The use of today's natural gas burners for hydrogen-rich gases would lead to high fuel injection velocities and high fuel-side pressure losses. Thus, the burner nozzles need to be redesigned. An additional crucial characteristic is the very high reactivity and laminar flame speed of hydrogen. Special burner modifications are necessary to avoid the risk of flame flashbacks and pre-ignition. These modifications ensure stable combustion at low NO_x formation. For this, a homogeneous fuel/air mixture and the avoidance of recirculation areas within the burner are of particular importance. These are the specifications for the development and optimization of premix burner concepts, applied for innovative high-temperature gas turbines. The design shown in figure 13–9 is derived from the Siemens standard natural gas premix burner. A concentric passage is added for enabling the high volumetric throughput of the hydrogen-rich gas, feeding directly downstream to the diagonal swirler into the fuel/air reaction zone.

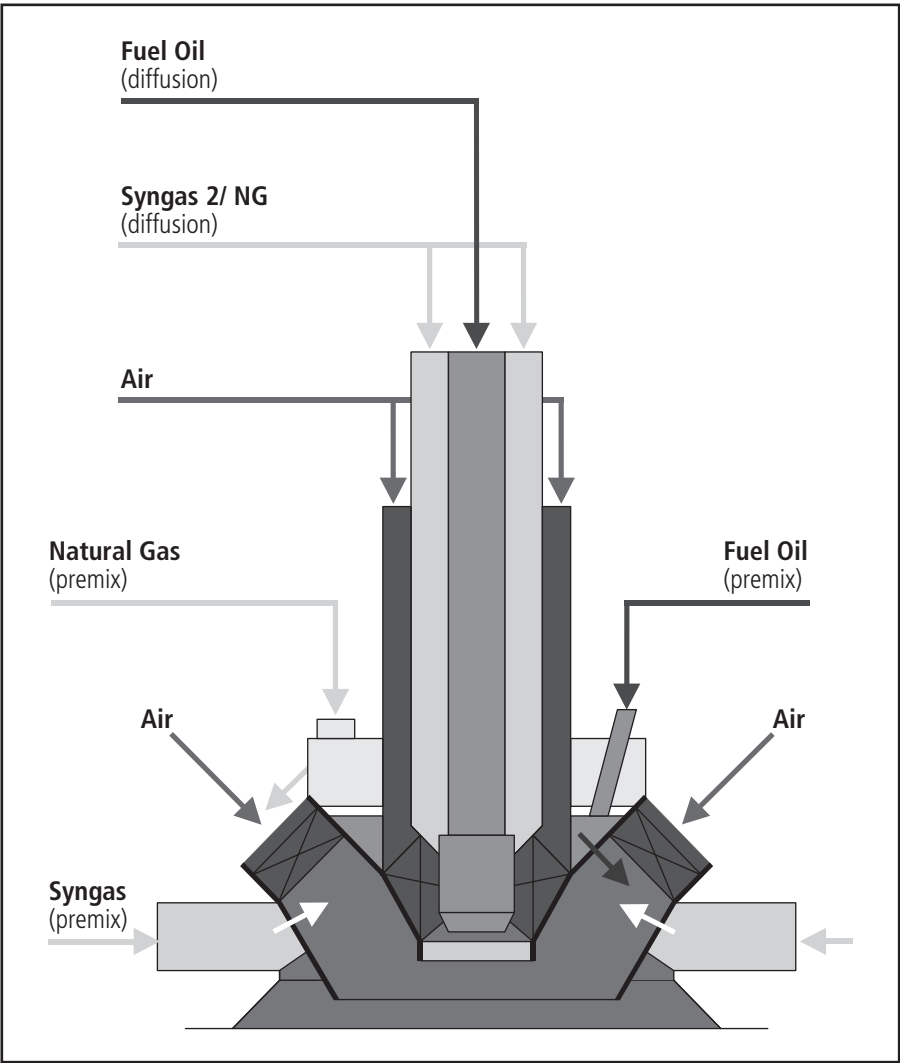


Figure 13–9 Concept of an innovative pre-mix burner for hydrogen-rich syngases

The development of gas turbines for decarbonized fuels is pushed by all manufacturers and supported by different national and international funding programs. One of the first large research and development programs initiated by the European Commission was the Enhanced CO₂ Capture (ENCAP) project, which started in 2004 and put together a great number of partners from research institutes, universities, manufacturers, and utilities.

Capture of the CO₂ after Combustion (Post-combustion)

The carbon content reacts to CO₂ through air-blown, hyperstoichiometric combustion (with possible traces of CO), which is separated for storage. The remaining flue gas, which mainly consists of nitrogen and H₂O, is discharged to the atmosphere (figure 13–10).

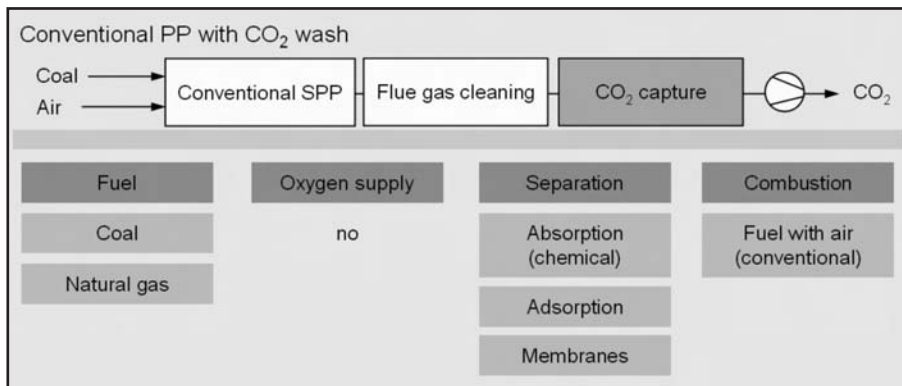


Figure 13–10 Schema of post-combustion CO₂-capture

Post-combustion capture methods can be applied to all power generation concepts because they are located in the flue gas stream at the end of the process chain. Thus, this capture method is best qualified for being incorporated into conventional coal-fired steam power plant concepts, for which no other CO₂ capture paths are available. In principle, also retrofitting of existing plants is an option for post-combustion capture. In case of natural-gas-fired combined-cycle plants, post-combustion capture has to compete with the two other capture methods—IRCC and Oxyfuel firing.

The exhaust gases to be treated are at atmospheric pressure, of high volume flow, and relative low in CO₂ content, particularly in the case of a natural-gas-fired combined-cycle plant. For CO₂ separation under such conditions, absorption processes applying mainly chemical sorbents are preferred today. Figure 13–11 shows the flow sheet of such

a sorption cycle. The CO_2 is transferred to the counter-flowing solvent in the absorption column, which leaves the column at its bottom and is pumped to the top of the desorber unit. For regenerating the chemical solvent, such as breaking its bonds with CO_2 , heat at 100 to 140°C is needed, which is extracted from the water steam/cycle to the desorption unit. The medium for heat transfer is steam at the corresponding temperature level for desorption. In the current concepts, about half to two-thirds of the LP steam are extracted from the IP to LP crossover pipe. The condensate is fed back to the water/steam cycle downstream of the condenser. The resulting loss in expansion energy available for power production is mainly responsible for the process' penalty. All thermodynamic, operational, and constructive modifications following from this steam extraction (for example, the LP steam turbine is then operated in part load only) have to be taken into account when considering retrofitting an existing power plant.

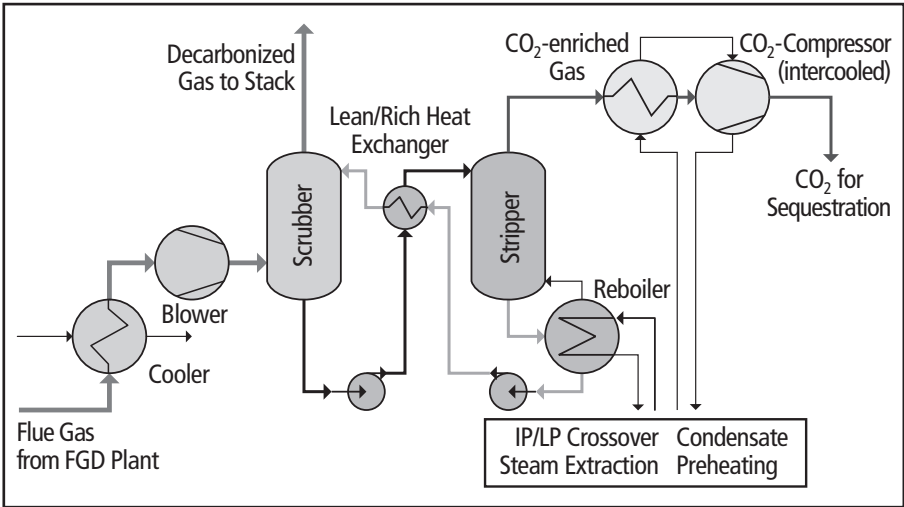


Figure 13–11 Post-combustion absorption process

Beside the conditions previously mentioned and, of course, the specific load rate and energy needed for desorption, a criterion for solvent selection is the potential degradation that is caused, for example, by oxygen and possible corrosiveness. Although various chemisorption agents and processes are already used in the chemical and other industries, predominantly aqueous solvents of amines (table 13–3), the different requirements need further development and demonstration.

Table 13–3 Commercially available CO₂ absorption systems for post-combustion applications

Kerr-McGee/ABB Lummus Crest Process (US)	MEA (aqueous 15–20 wt %) + inhibitor
Fluor Daniel ECONAMINE™ Process	MEA (aqueous 30 wt %), with inhibitor
Kansai Electric Power Co., Mitsubishi Heavy Industries, Ltd.	Sterically-hindered amine (KS solvents)

For example, in the large European research and development project CO₂ capture and storage (CASTOR), several solvents are tested for CO₂ removal in a flue gas side-stream of the Elsam coal-fired power station in Esbjerg/Denmark. Against the background of obligations for CO₂ reduction and the increasing application of EOR, several projects are pushed for developing CO₂ capture from natural-gas-fired combined cycles mainly in Norway, such as Karstoe and Tjeldbergodden (Statoil).

The SARGAS cycle (a concept belonging to the Norwegian-based company SARGAS AS) describes a post-combustion concept where the CO₂ is advantageously separated under elevated pressure (figure 13–12). It is a combined cycle applying an open gas turbine, which is externally fired in a pressurized boiler. The flue gas is cooled, decarbonized, and cleaned from NO_x. After being heated up again, it is expanded through the turbine unit. About 15 to 20% of the total power is generated by the gas turbine. The rest is generated in the steam cycle, which is heated by the combustion chamber.

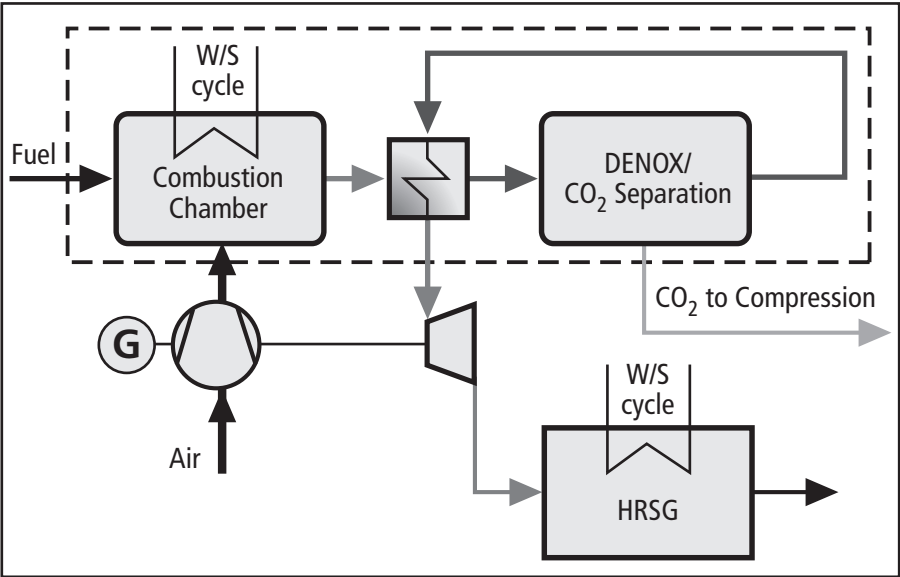


Figure 13–12 Post combustion CO₂ capture under pressure

Oxyfuel Processes

From the basic idea, Oxyfuel concepts are post-combustion processes, as well. However, by using nearly pure oxygen for the combustion, the flue gas is enriched in CO₂ and consists to a great extent of CO₂ and H₂O, which is condensed to get the CO₂ for depositing. Thus, strictly speaking, it is not the CO₂ that is separated. As there are modifications and additional steps needed up- and downstream the combustion, these Oxyfuel concepts are also called integrated CO₂ capture (figure 13–13).

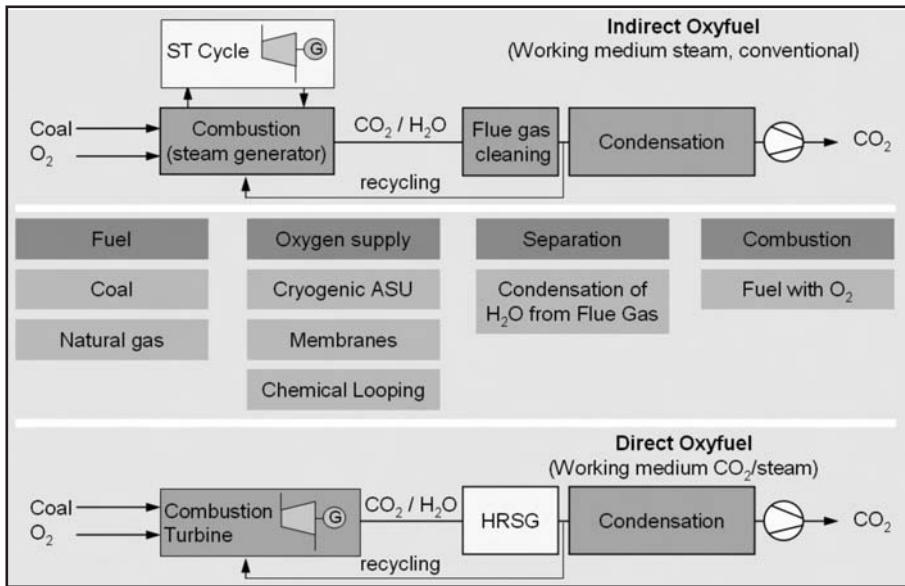


Figure 13–13 Schema of Oxyfuel firing CO₂ capture

For reducing the combustion temperature, which for pure oxygen is about 3,500°C, down to the level of conventional boiler or gas turbine conditions, either parts of CO₂ have to be recycled or water has to be injected.

With view to the medium used in the working machine, two systems are distinguished: indirect Oxyfuel systems where heat is generated in an oxygen-blown combustion and indirectly transferred to the working medium, and direct Oxyfuel systems, where the CO₂/H₂O flue gas is directly used in a turbine.

There are several Oxyfuel-fired concepts proposed or under development. Some of these are discussed in the following sections.

In the Graz cycle (Graz University of Technology) the fuel is burnt with the stoichiometric mass flow of oxygen, which is produced externally (over the fence), such as in a cryogenic air separation unit (figure 13–14). Steam and recycled CO₂ are used for cooling purposes. The flue gas is expanded in two steps with an intermediate heat recovery

system, and is divided into water and CO_2 in the condenser. A part is then recompressed to the combustor. The water is used as medium for the Rankine cycle. The gas turbine working medium is a mixture of CO_2 and steam. It has to be essentially redeveloped because its characteristics are clearly different from the working fluid of a standard gas turbine.

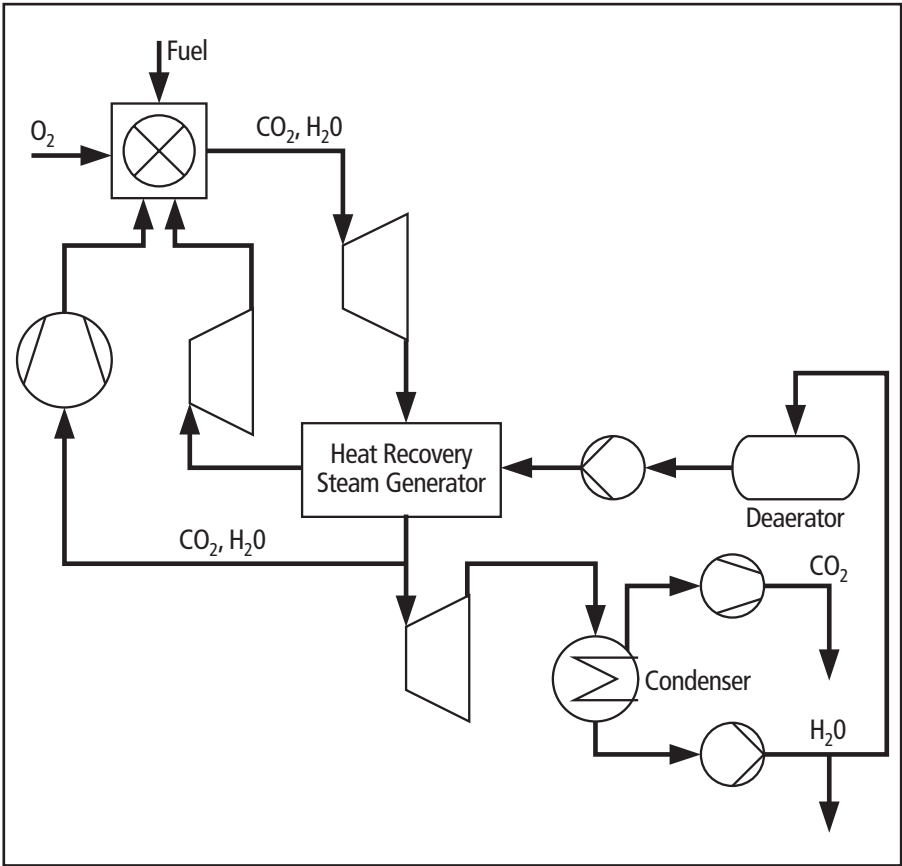


Figure 13–14 Concept of the Graz cycle

The CES cycle (Clean Energy Systems Inc.) uses a modified rocket burner, which is fired by a gaseous fuel and oxygen (figure 13–15). Water that has been separated from the flue gas is recycled for cooling this gas generator. The hot exhaust gas is directly expanded through a multistage turbine to generate power (direct Oxyfuel), or alternatively

can generate steam in a heat recovery steam generator to be used in a water/steam cycle (indirect Oxyfuel). H_2O is condensed after the cooling step.

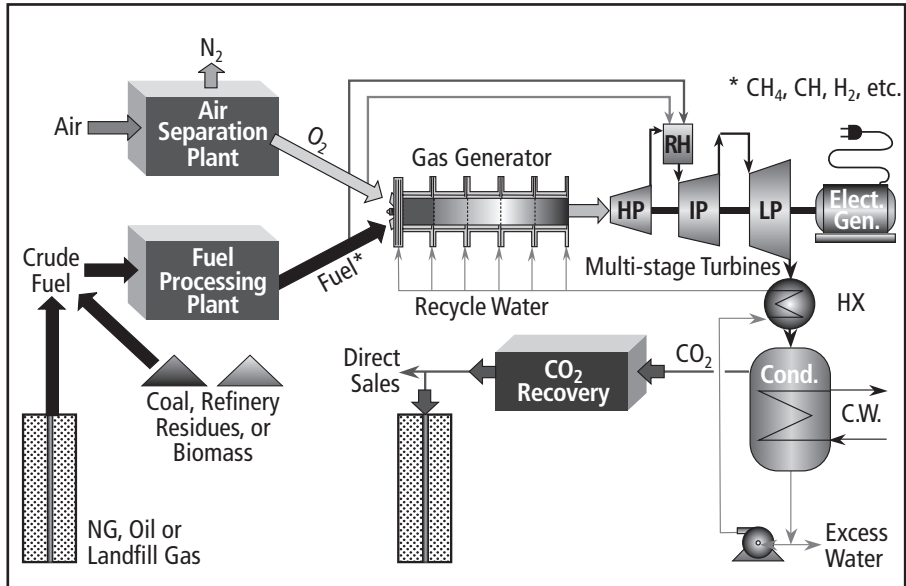


Figure 13–15 Direct Oxyfuel fired CO_2 capture, CES cycle

The Advanced Zero Emission Power plant (AZEP) (Norsk Hydro) combines two nonstandard gas turbines (figure 13–16). The first one is a hot air turbine in principle, where the air is oxygen-depleted by an ion transport membrane and indirectly heated by Oxyfuel combustion (indirect Oxyfuel). This turbine is combined with a water/steam cycle. The combustion products, CO_2 and steam, are used to drive the second turbine, which is operated in simply cycle mode (direct Oxyfuel). CO_2 is separated by condensing the water content of the flue gas. The separation process is supported by continuously reducing the oxygen concentration on the product side by the reaction and by continuously diluting with $\text{CO}_2/\text{H}_2\text{O}$ (sweep gas).

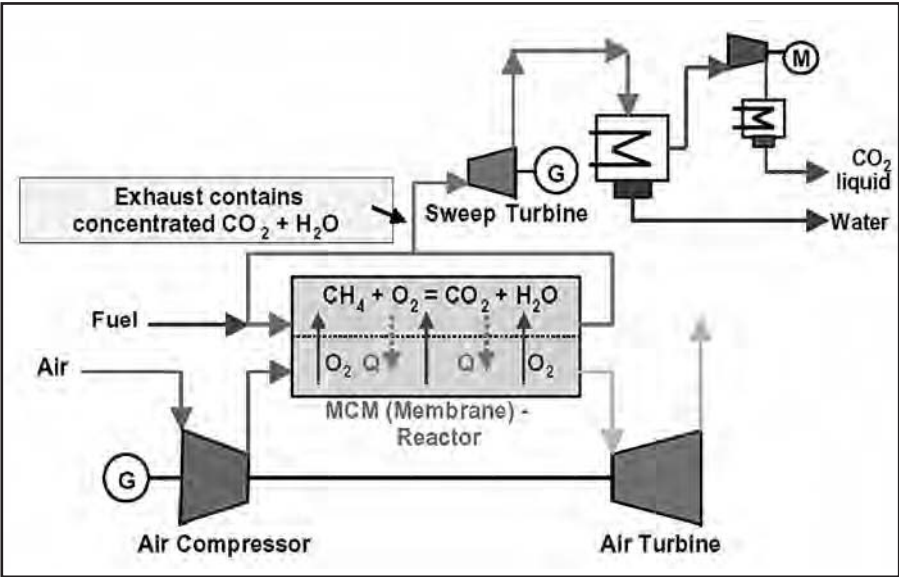


Figure 13–16 Principle of the AZEP process

The German OXYCOAL-AC project (led by the technical university RWTH Aachen) aims at using such a membrane-based Oxyfuel concept for coal-dust-fired power plants.

A promising, long-term option is the zero emission solid oxide fuel cell (ZESOFCC) concept. It combines an air-blown gas turbine with the high-efficient fuel cell process (figure 13–17). The solid oxide fuel cell (SOFC) has to be modified for combustion of the anode exhaust gas with oxygen. That way, the completely burnt flue gas consists of CO_2 and H_2O only, which is expanded in a specially designed turbine.

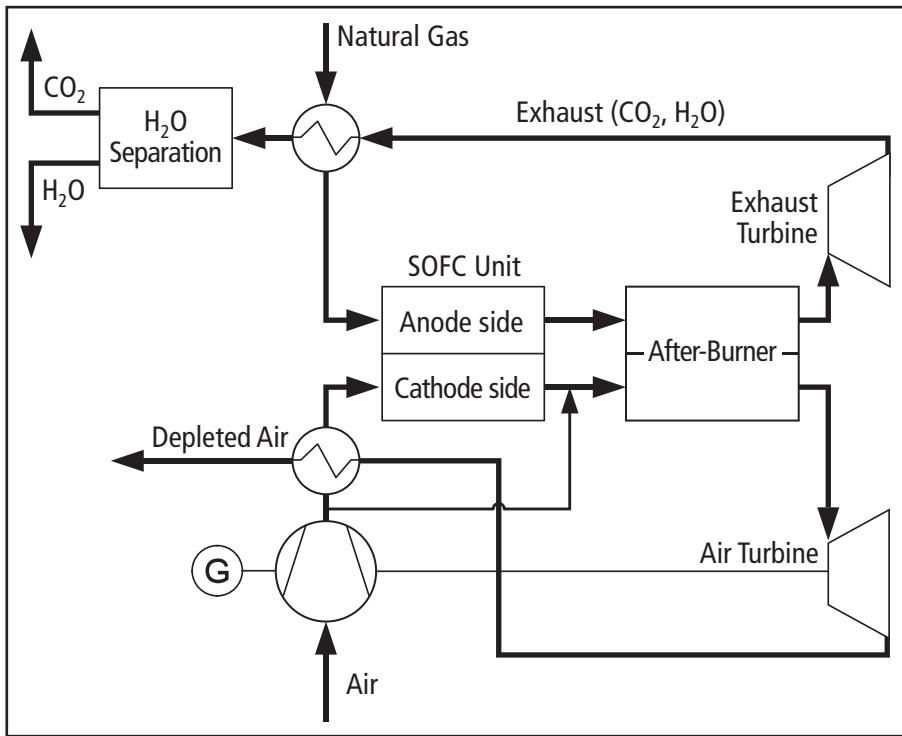


Figure 13–17 Principle of the ZESOF process

Impact on Efficiency and Economy

Because of the additional process units and power demand, CO₂ capture leads to reduced power generation efficiency and increased costs. Various studies are made to quantify these impacts for the different CCS concepts resulting in a large bandwidth because of the different boundary conditions and steps of development, or uncertainties when designing novel processes in general. In fact, certain information about plant performance and costs will be available from the ongoing first demonstration projects only.

Based on public research and development as well as on utility and manufacturer in-house studies, a comprehensive technology benchmark has been done by the ZEP Working Group 1 (European technology platform for zero emission fossil fuel power plants) encompassing technologies, which are expected to be in use in large applications by

2020. By an efficiency penalty of about 10 percentage points (reference 58%) combined-cycle concepts on natural gas are in the same range as hard-coal-fired power plants. No difference regarding efficiency results for the three principle paths pre- and postcombustion and Oxyfuel. However, the latter are assumed to lead to a 100% CO₂ capture rate (natural gas), whereas for post-combustion 85% and for precombustion 93% are the bases. With view to the CO₂ avoiding costs, CCS makes sense first of all for base-load operated coal plants (figure 13–18 top) for two reasons. For coal fired plants the specific CO₂ concentrations are higher and, as a consequence of the lower fuel costs, the reduction in efficiency for the CCS case results in a lower impact on the additional costs. On the other side, the amount of CO₂ produced per MWh electricity is lower for gas-fired plants. As a consequence the tendency levels off when focusing on the power generation costs (figure 13–18 bottom).

A comparison focusing on various natural-gas-fired gas turbine cycles has been published by the Norwegian independent research organization SINTEF. From that, based on 57% net plant efficiency for the reference case the near term pre- (air-blown ATR) and post-combustion (MEA) concepts lead to a drop down to 47% and 48%, respectively. In the same range are Oxyfuel processes applying a cryogenic air separation unit with the potential up to 49% for an advanced version of the Graz cycle (S-Graz). Higher efficiencies between 50 to 53% are evaluated for Oxyfuel processes using future membranes for the oxygen production or chemical looping combustion. The study also included the highly advanced combined SOFC/ gas turbine cycle with an efficiency potential of 67%, which, however, needs clearly long-term development for large-scale applications.

In addition to thermodynamics and the cost situation, further aspects are to be considered when evaluating CCS technologies, such as the quality of the CO₂ stream with view to its application/storage, the plant size, the complexity of the concept that might significantly effect the availability, or the suitability for retrofitting to existing plants. And, of course, it is not only the capture technology that influences the economy of CCS. For example, the absolute fuel price has a significant influence on the CO₂ avoidance costs. Therefore, according to European Technology Platform for Zero Emission Fossil Fuel Power Plants (ETP-ZEP) the decision on future capture plants will be driven primarily by the fuel choice and only second by the capture technology.

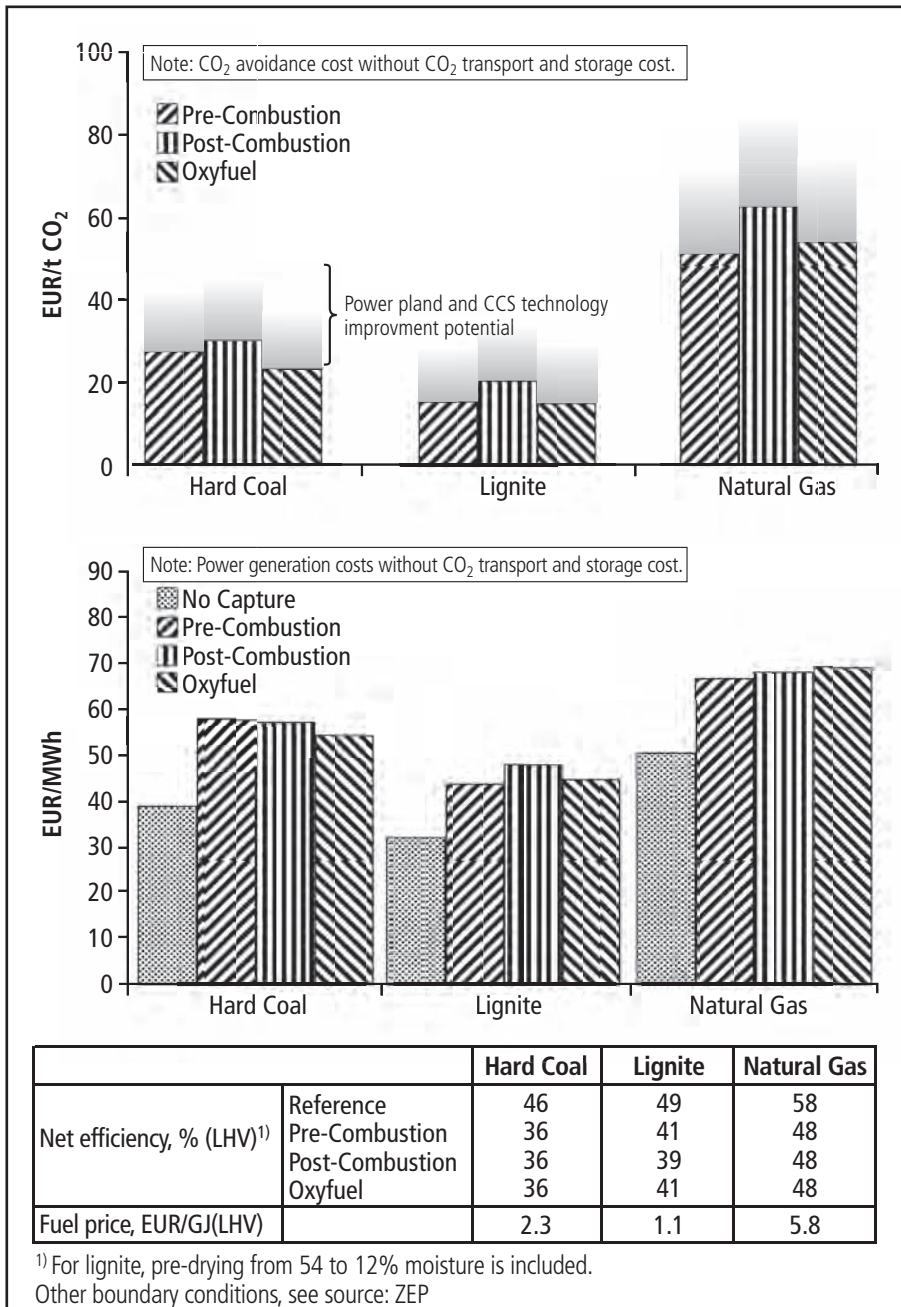


Figure 13-18 Expected CO₂ avoidance and power generation costs for industrial-scale power plants in operation by 2020 (Source: ZEP)

Outlook

For gas-fired combined-cycle plants a diversity of CO₂ capture concepts are under progress particularly for Oxyfuel combustion. In case of direct Oxyfuel the working medium, a mixture of mainly CO₂ and H₂O, significantly differs from the conventional flue gas from air-blown combustion. This means that the specific mass flow and compression/expansion work as well as the turbine outlet temperature and the design of blades and dimensions of gas passages. Thus, for the breakthrough of direct Oxyfuel in this sector the core component, the gas turbo-set, has to be fundamentally redesigned and optimally adapted. For the turbine manufacturer such a costly and time-consuming development requires a positive economical perspective as well as the support by national and international funding organizations. Altogether, this concept is expected to be a more long-term CO₂ capture option.

To apply CCS measures in power generation beside a technology push, incentives from market or policy are needed as well as a broad public acceptance of CO₂ storage. Increasing endeavours on the development of CCS techniques are made worldwide. Several large-scale demonstration plants for the different concepts will be built in the next decade (table 13–4). For future power plant projects the obligation is under discussion to include CO₂ capture or at least to be ready for CCS; this means, for example, to provide sufficient space for later installing the additional systems or to consider future steam extraction for the desorption units. Moreover, the reduction in power must be taken into account, or the systems have to be oversized for the period without capture. In CO₂-free operation mode, potential failures of the sequestration units, such as the CO₂ compressor, and their impact on the total plant operation must be handled.

Table 13–4 Selection of European CCS demonstration projects (Source: The World Energy Book, issue 3)

Country	Start-up	Owner	Capture	Storage
Denmark	2005	Castor project, FP6	Post-combustion; experimental; coal-fired plant	A few '000 t/y; no storage
Germany	2008	Vattenfall	Oxyfuel boiler; coal-fired plant; 30 MW	Pilot scale; no storage
France	2008	Total	Oxyfuel boiler; 30 MW, gases and liquids	Pilot scale; Lacq plant
Norway	2009	Stratkraft	Gas-fired power plant Karstoe, 400 MW	EOR (Field not specified)
UK	2009	Progressive Energy	Pre-combustion 800 MW; IGCC	—
Netherlands	2008	SEQ International, ONS Energy	Oxyfuel; gas-fired plant; 350 MW	EOR/EGR Drahten
UK	2010	BP/Scottish and Southern Energy	Pre-combustion; gas-fired plant; 350 MW	EOR Miller field
UK	2010	Powerfuel	Pre-combustion; IGCC; 900 MW	—
Norway	2011	Shell/Statoil	Gas-fired power plant; 860 MW	EOR Draugen/Heidrun oilfields
UK	2011	E.On	Pre-combustion; IGCC	Offshore
Germany	2014	RWE	Pre-combustion; IGCC; 450 MW	Geological storage
UK	2016	RWE	Coal-fired power plant retrofit	To be determined

14 Typical Combined-Cycle Plants

Table 14–1 shows how the technologies described in previous chapters are applied from the different suppliers in the global electric power industry.

Table 14–1 Combined-Cycle Plant examples—overview

Plant Name	Country	Supplier	Net-Output MW	Configuration	Remark
Taranaki	NZ	ABB*	360	single-shaft	
Monterrey	MEX	ABB*	2×242	2 single-shaft	
Phu My 3	VN	Siemens	716	multishaft	
Palos de la Frontera	E	Siemens	3×380	3 single-shaft	
Arcos III	E	GE	823	multishaft	
Diemen	NL	ABB*	250	multishaft	Cogeneration
Shuweihat IWPP	UAE	Siemens	1495	multishaft	Desalination
Vado Ligure	IT	Ansaldo	780	multishaft	Repowering
Puertollano	E	Siemens	300	multishaft	IGCC
Monthel	CH	KAM/Siemens	55	multishaft	Cogeneration

*ABB (today ALSTOM)

The Taranaki Combined-Cycle Plant, New Zealand

The 360-MW Taranaki combined-cycle power plant (figure 14–1) is owned by Stratford Power, Ltd., New Zealand's first independent power producer, and supplies electricity as a baseload facility connected to the North Island transmission grid.



Figure 14–1 View of Taranaki combined-cycle plant

The plant is located about 50 kilometers from New Plymouth on a site 270 meters above sea level. Initially the plant will fire only natural gas taken from the Maui gas field, but the design enables conversion to fire oil in the future, should this prove necessary.

The power plant consists of a single-shaft block with one ABB GT26 gas turbine, one ABB natural circulation HRSG, and one ABB two-casing steam turbine, as well as the necessary auxiliary equipment. Due to the lack of cooling water on the site a wet cooling tower has been installed.

A high-efficiency triple-pressure reheat cycle was chosen for the plant, with HP live-steam at 103 bar (1479 psig)/568°C (1,054°F), reheat live steam at 24 bar (333 psig)/568°C (1054°F), and LP steam at 4.0 bar (44 psig)/saturated. The gas is preheated to further increase the efficiency (figure 14–2).

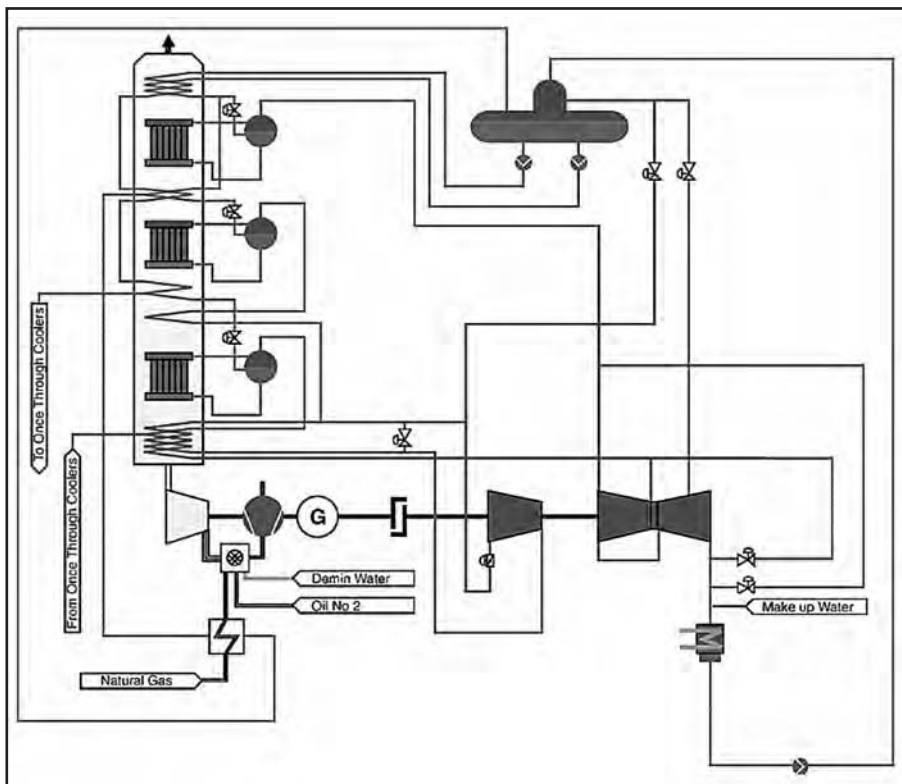


Figure 14–2 Process diagram of Taranaki combined-cycle power plant

Table 14–2 summarizes the performance data of the Taranaki combined-cycle power plant.

Table 14–2 Main technical data of Taranaki combined cycle power plant

Plant Configuration:		KA 26-1
Number of blocks		1
Number of gas turbines/block		1
Main fuel		Natural gas
Ambient Conditions:		
Ambient temperature	°C / °F	11.6 / 53
Ambient pressure	mbar	981
Relative humidity	%	84
Cooling type		Cooling tower
Performance data per block:		
Total fuel input to gas turbines	MW	615
Gross output	MW	359.9
Gross efficiency (LHV)	%	58.5
Auxiliary consumption and losses	MW	6.1
Net power output	MW	353.8
Net efficiency (LHV)	%	57.5
Heat rate (LHV)	kJ/kWh	6261
Heat rate (LHV)	BTU/kWh	5934
Process heat	MJ/s	0

The Taranaki plant follows ABB’s standard single-shaft arrangement with the generator located in between the gas turbine and steam turbine (figure 14–3). The steam turbine can be decoupled from the generator by means of a self-shifting clutch for startup and shutdown. The condenser is axial to the steam turbine—both are at ground level—eliminating the need for a steam turbine table. The hydrogen-cooled generator is mounted on a skid, together with its lube oil system and hydrogen coolers. The skid is mounted on transverse rails so it can be moved to the side for rotor inspections. The arrangement is very compact, with a footprint of the combined-cycle block measuring about 38 by 96 meters.

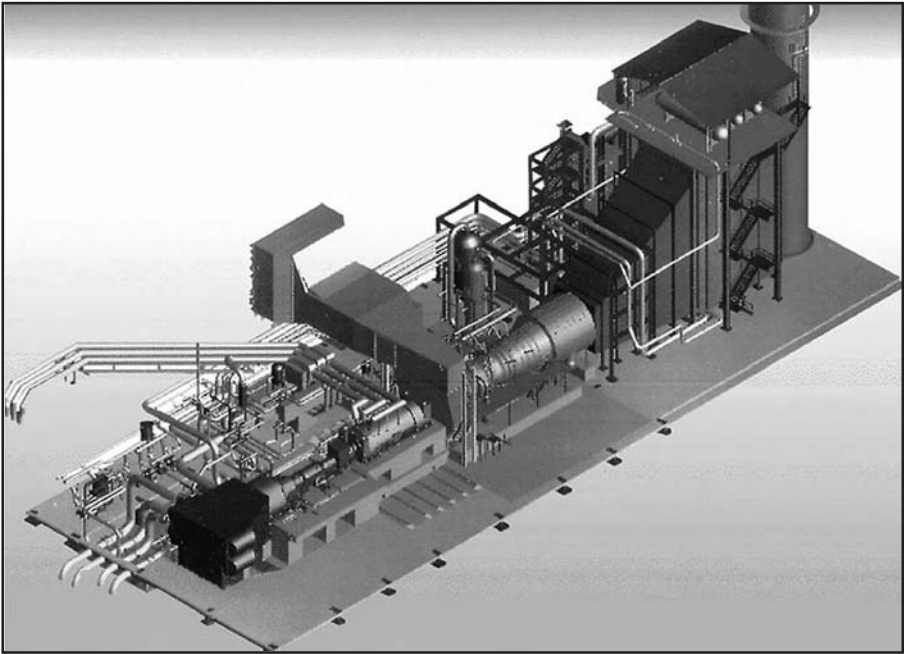


Figure 14–3 Arrangement of Taranaki combined-cycle power plant

This plant is the first in a series of ABB-standard reference single-shaft units for the 50 Hz market. Some of the follow-up orders are in the United Kingdom, Chile, and Japan.

ABB had also the responsibility for the operation and maintenance of the plant over the first six years. An online monitoring system is connecting the plant to the ABB headquarters in Switzerland, where direct assistance will always be available to the plant operators.

The 480 MW Combined-Cycle Power Plant, Monterrey, Mexico

The Monterrey combined-cycle power plant has been ordered by Comisión Federal de Electricidad (CFE), the national utility of Mexico, and achieved commercial operation in the year 2000. The plant, located

in Monterrey (Nuevo Leon), is designed for pure electrical power generation with a net output of 484 MW and 56% net efficiency, making it one of the most efficient thermal power plants in Mexico.

The plant is based on ABB's KA24-1 standard reference plant and is among the first units of a series of installations in the 60 Hz market, applying the same design.

The gas turbines are equipped with the latest low NO_x technology to minimize emissions. Effluents are basically nil, qualifying it as a zero-discharge site. An air-cooled condenser uses ambient air as heat sink to the cycle, keeping water consumption to a minimum.

The Monterrey power plant is made up of two identical 242 MW blocks, each comprising one 160 MW ABB GT24 gas turbine, one ABB CE once-through HRSG, and one ABB 90 MW double-casing reheat steam turbine unit. State-of-the-art combined-cycle plants are mostly designed with a triple-pressure reheat cycle to achieve high efficiency, but not so this plant. As explained in chapter 5, a simpler double-pressure reheat cycle was chosen, resulting in almost the same high net efficiency level.

The hot exhaust gases of the GT24 are used to generate steam at 160 bar (2310 psig) for best efficiency. Producing steam at this pressure helps to avoid an IP system, therefore, simplifying the cycle. The once-through HRSG avoids a thick-walled HP drum and results in a high thermal flexibility. The LP part is directly fed from the condenser. Steam of 160 bar (2310 psig) results in small steam volumes that can be efficiently expanded in the geared HP steam turbine. The barrel-type HP steam turbine design provides high thermal flexibility.

This simpler, dual-pressure reheat plant exceeds 56% net efficiency with an air-cooled condenser at 30°C (86°F) ambient temperature. The gas turbine and the steam turbine drive a common air-cooled generator installed between them. Although the gas turbine is rigidly coupled to the generator, the steam turbine is equipped with a self-shifting and synchronizing clutch, enabling startup of the gas turbine independent from the steam turbine. The steam turbine clutch engages automatically during startup as soon as the speed of the steam turbine reaches that of the generator.

The chosen cycle for the Monterrey plant produces HP steam of 160 bar (2310 psig)/565°C (1049°F) and LP steam of 7 bar (87 psig)/320°C (608°F), which are both fed into the steam turbine (figure 14–4). After partial expansion of the HP steam in the HP-steam turbine, it is reheated in the HRSG at 37 bar (522 psig) to 565°C (1049°F).

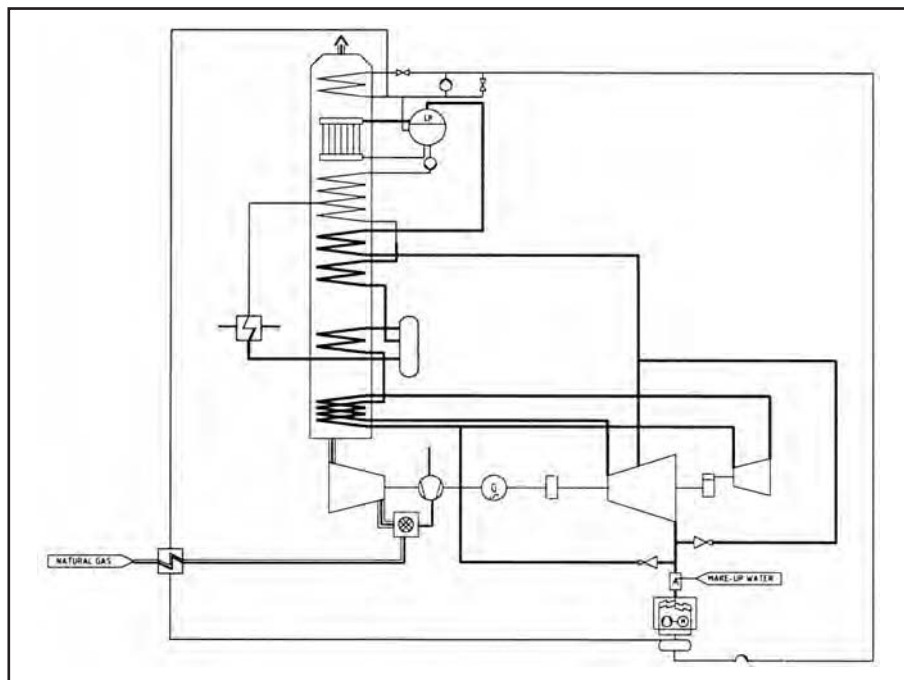


Figure 14–4 Process diagram for Monterrey combined-cycle power plant

The hot exhaust gases are thereby cooled from approximately 650°C (1202°F) to below 100°C (212°F) before being exhausted through the stack. Due to lack of water at this site an air-cooled condenser is used to condense the steam at the outlet of the steam turbine. The auxiliaries are cooled by air blast coolers.

Table 14–3 summarizes the performance data of the Monterrey combined-cycle power plant. As shown in the table, more than 56% of the fuel energy is converted to electrical power, despite the air-cooled condenser operating at 30°C (86°F) ambient temperature that reduces the steam turbine output and increases auxiliary consumption. By this

means, plant water consumption is kept to a minimum and no water body is heated up by the dissipated heat. NO_x levels in the exhaust are below 25 ppm.

Table 14–3 Main technical data of Monterrey combined cycle power plant

Plant Configuration:		2 × KA 24-1
Number of blocks		2
Number of gas turbines/block		1
Main fuel		Natural gas
Ambient Conditions:		
Ambient temperature	°C / °F	30 / 86
Ambient pressure	mbar	969
Relative humidity	%	60
Cooling type		Air-cooled condenser
Performance data per block:		
Total fuel input to gas turbines	MW	431.3
Gross output	MW	249.9
Gross efficiency (LHV)	%	57.9
Auxiliary consumption and losses	MW	7.8
Net power output	MW	242.1
Net efficiency (LHV)	%	56.1
Heat rate (LHV)	kJ/kWh	6413
Heat rate (LHV)	BTU/kWh	6078
Process heat	MJ/s	0

Each gas turbine, and the corresponding steam turbine, is arranged indoors in a common machine house, with sufficient laydown area for inspections. The two blocks are fitted in separate buildings. The HRSGs are arranged outdoors. The balance of the plant is arranged to provide short pipe and cable routing to the different balance of plant equipment (figure 14–5).

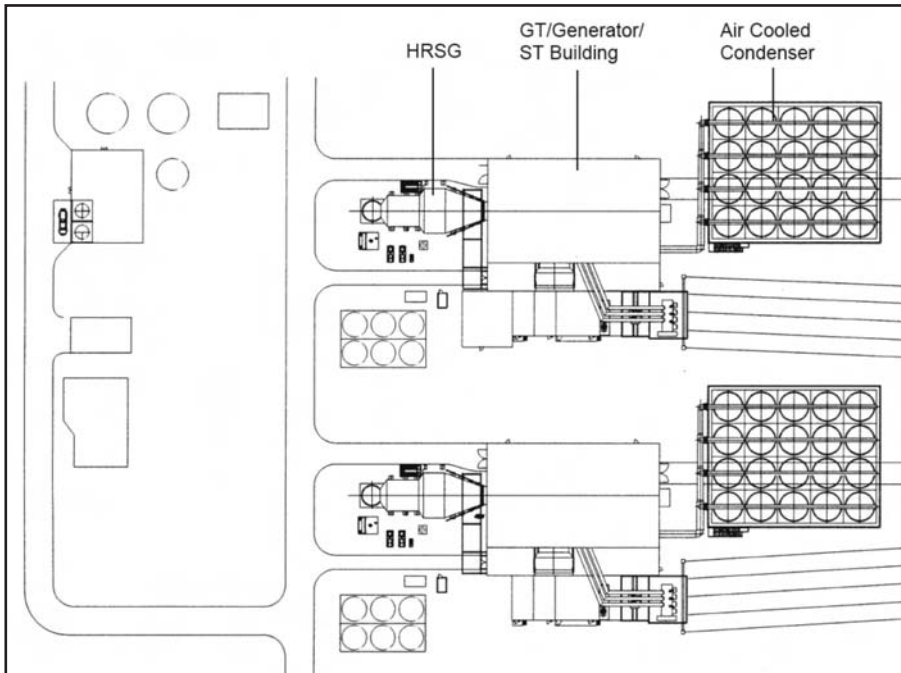


Figure 14–5 Layout of the Monterrey combined-cycle power plant

The 716-MW Combined-Cycle Power Plant Phu My 3, Vietnam

The Phu My 3 combined-cycle plant was built as a build-operate-transfer (BOT) project owned by the PhuMy 3 BOT Power Company. This project-specific company is a consortium of oil and gas major BP, SembCorp Utilities (majority owned by the Singapore government), and the Japanese companies Kyushu Electric and Sojitz (acting as one partner in the consortium). Each of the partners has a one-third interest. A power purchase agreement with Electricity of Vietnam (EVN) is scheduled to run for 20 years, after which ownership will be transferred to the state-owned utility.

Phu My 3 is located 70 km southeast of Ho Chi Minh City near the village of Phu My on an industrial site that already accommodates three plants (figure 14–6).



Figure 14–6 View of the Phu My 3 combined-cycle power plant

In May 2001, the consortium signed an EPC contract with Siemens Power Generation, including a 12-year maintenance agreement. Construction work began in late 2001, and commercial operation started in March 2004.

For achieving best performance at the specified ambient conditions in Southeast Asia, a triple-pressure, single-reheat cycle was selected. The plant rated capacity at ISO conditions is 760 MW with a design efficiency of 58% at the generator terminals.

The multishaft combined-cycle configuration is made up of standardized modules, the main modules being the two gas turbine generator sets, two outdoor heat recovery steam generators, one steam generator set, water-steam cycle system, instrumentation and control equipment, electrical power system as well as pre-designed, pre-engineered buildings or civil structures (figure 14–7).

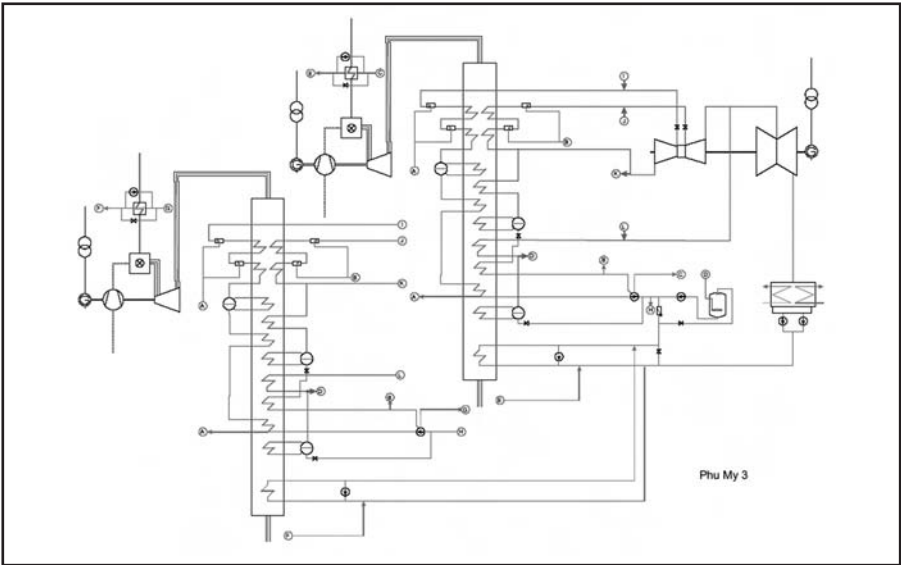


Figure 14–7 Process diagram of the Phu My 3 combined-cycle power plant

The arrangement of the plant is based on the Siemens combined-cycle reference power plant SCC5-4000F 2x1 (figure 14–8).

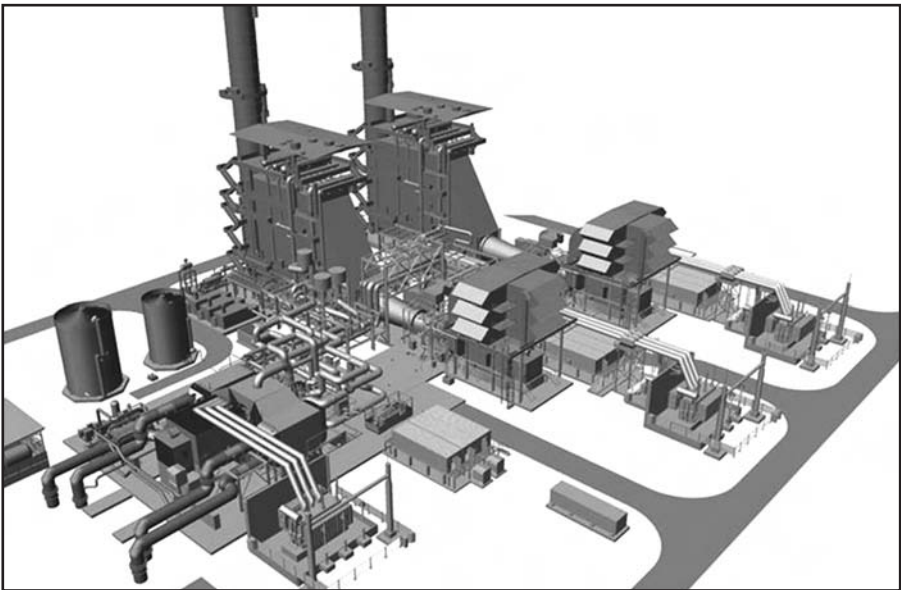


Figure 14–8 Arrangement of the combined-cycle reference power plant SCC5-4000F 2x1

The selected gas turbine is a Siemens SGT5-4000F machine, a heavy-duty gas turbine, structured as a completely pre-engineered unit. It is a single-shaft machine with horizontally split casings, disk-type rotor with center tie bolt and cold end drive. The combustion system consists of an annular combustion chamber with 24 hybrid burners. The burners are designed to operate on both gas and liquid fuel in dry low NO_x mode without water injection. The system also has provisions for water/fuel oil emulsion injection for NO_x control in diffusion mode.

Pollutant emissions are negligible when burning natural gas, and CO emissions in the load range above 60% are below 10 ppm, which is typical for gas turbines featuring combustion chambers with ceramic tiles.

The steam turbine, which drives an air-cooled generator, is a three-stage, reheat, two-casing machine. It is built as a two casing turbine with a combined opposed-flow HP/IP turbine and a double-flow LP turbine.

The main technical data are shown in table 14–4.

Table 14–4 Main technical data of Phu My 3 combined cycle power plant

Plant Configuration:		SCC5-4000F 2×1
Number of blocks		1
Number of gas turbines/block		2
Main fuel		Gas
Ambient Conditions:		
Ambient temperature	°C / °F	30 / 86
Ambient pressure	mbar	1013
Relative humidity	%	82
Cooling type—cooling water temperature	°C / °F	once-through 29 / 84
Performance data per block:		
Total fuel input to gas turbines	MW	1261.2
Gross output	MW	729.4
Gross efficiency (LHV)	%	57.84
Auxiliary consumption and losses	MW	12.60
Net power output	MW	716.8
Net efficiency (LHV)	%	56.84
Heat rate (LHV)	kJ/kWh	6334
Heat rate (LHV)	BTU/kWh	6003
Process heat	MJ/s	0

The 1200-MW Combined-Cycle Power Plant Palos de la Frontera, Spain.

The 3x 400-MW combined-cycle power plant Palos de la Frontera is owned by Union Fenosa Generacion (UFG), a fully owned subsidiary of the Spanish utility Union Fenosa. The plant is located in Andalusia, 100 kilometers southwest of Seville. Handover of the third unit took place in June, 2005.

The Palos de la Frontera CCPP consists of three separately housed Siemens SCC5-4000F combined-cycle blocks, each made up of a gas turbine, generator, and steam turbine in a single-shaft configuration (figure 14–9).



Figure 14–9 View of the Palos de la Frontera combined-cycle power plant

The modularized Reference Power Plant concept developed by Siemens is implemented for the three units. This concept is characterized by optimum standardization of plant systems and components in line with the specific needs of the power plant operators. Special customer requests can be met by using a modularized building-block system.

Costs can thus be saved and the plant can be tailored to the respective customer. A special feature of the single-shaft technology used in the three Spanish plants is the arrangement of the key components. Only one generator is required, which is arranged on a common shaft between the gas and the steam turbine. The single-shaft design allows low investment costs, short construction periods, and highest efficiency levels as well as high operational flexibility (figure 14–10).

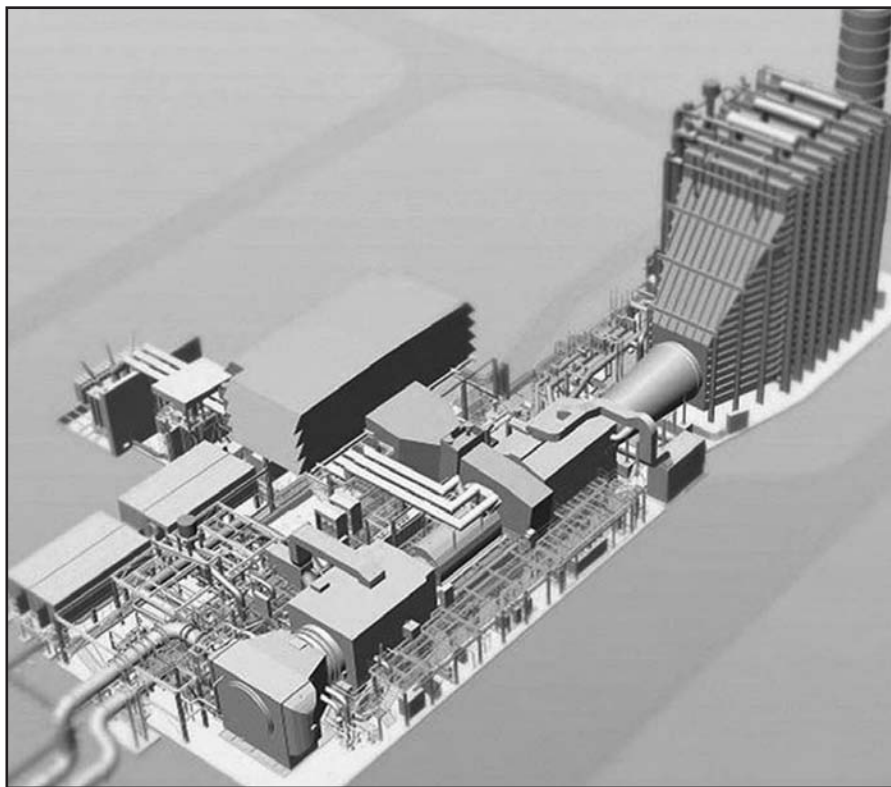


Figure 14–10 Layout of the Palos de la Frontera combined-cycle power plant

The gas turbines are Siemens SGT5-4000F (formerly V94.3A) machines. At site conditions, each gas turbine has a power output of 266 MW and an electrical efficiency of 38.6%. The rotor has a hollow shaft with disks that are interlocked via Hirth serrations and axially fixed via a central tie-bolt. The compressor has 15 stages with variable

inlet guide vanes for optimized flow control. The turbines feature an annular combustion chamber with 24 low NO_x hybrid burners.

The heat-recovery steam generators (HRSGs) are located outdoors next to the turbine buildings. The HRSGs supplied by Doosan are horizontal, natural circulation units with three pressure levels and reheat. The pipe bundles, which receive the exhaust gas from the gas turbine, are arranged vertically (figure 14–11).

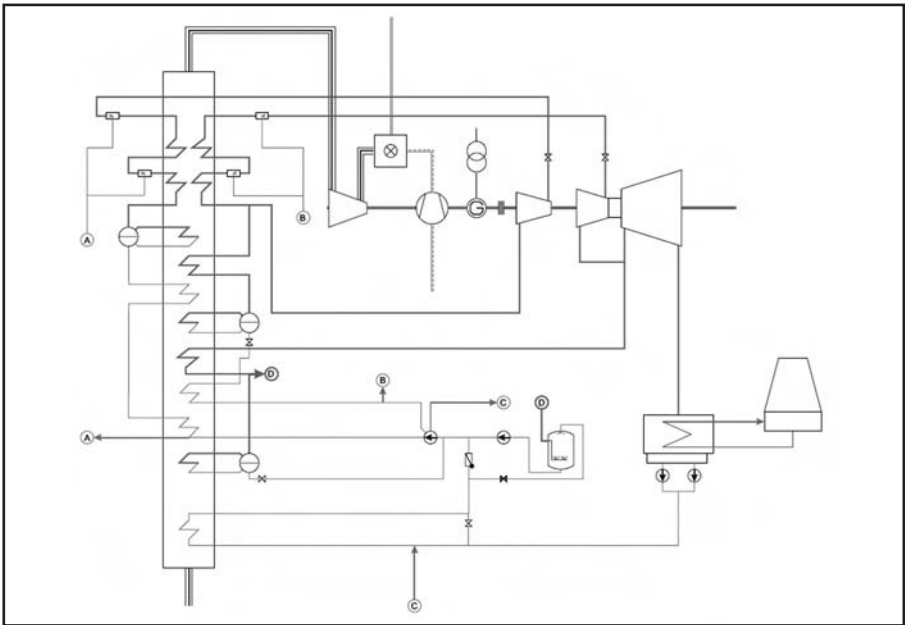


Figure 14–11 Process diagram of the Palos de la Frontera combined-cycle power plant (one unit)

The plant is operating with an average efficiency of 56%. For 2006, the plant achieved an availability of more than 98% (even when accounting for scheduled outages) and produced 7,023 GWh of power. Typically, the plant operates at nearly full load in the summer and winter, but it operates on a stop-start basis in the spring and fall according to the national grid demand and electricity market price.

In today's market, the two main areas of plant operator interest are in achieving minimum load while meeting emission limits and in realizing

maximum output with minimum startup time. Accordingly, Siemens has been adapting its technology to improve the cycling capability of the entire plant and not only individual components. Siemens has applied this design philosophy to develop what they refer to as advanced fast cycling (FACY) capability that was first fully tested for hot-start operation at Palos in 2005 and has since been sold into a number of power stations.

Fast hot-start capability (after an eight hour shutdown) means that full load can be achieved in less than 40 minutes. During the startup procedure, the steam turbine is run up to load on cold steam in parallel with the gas turbine. Siemens designed a new type of Benson boiler to cope with the much higher thermal and mechanical stresses imposed on the boiler. Additional stresses on the steam turbine from a fast hot start are within the range that is included in the lifetime calculation.

Table 14–5 summarizes the performance data of the Palos de la Frontera combined cycle power plant

Table 14–5 Main technical data of Palos de la Frontera combined cycle power plant

Plant Configuration:		SCC5-4000F 1S
Number of blocks		3
Number of gas turbines/block		1
Main fuel		Natural gas
Ambient Conditions:		
Ambient temperature	°C / °F	18.3 / 64.9
Ambient pressure	mbar	1015
Relative humidity	%	65
Cooling type		Cooling tower
Performance data per block:		
Total fuel input to gas turbines	MW	673
Gross output	MW	386.6
Gross efficiency (LHV)	%	57.5
Auxiliary consumption and losses	MW	4.7
Net power output	MW	381.9
Net efficiency (LHV)	%	56.8
Heat rate (LHV)	kJ/kWh	6343
Heat rate (LHV)	BTU/kWh	6012
Process heat	MJ/s	0

The Arcos III Combined-Cycle Plant, Spain

Arcos de la Frontera Grupo III in Cadiz, Spain, is the third in an assembly of plants that make up a significant investment and largest power plant in the 100-year history of Spanish energy producer, Iberdrola Generacion, S.A.—also owner and operator of the plant.

In commercial operation since March 2006, Arcos III generates more than 820 MWnet of power for the Spanish electricity grid. The plant is located just between the city of Sevilla and the Port of Algeciras, a heavy industrial area with a high demand for energy.

Arcos I and II began commercial service in January 2005, and each of the groups' 109FA (single-shaft units), together with Arcos III, has a common infrastructure with an administration building/control room, 400kV substation, demineralized and makeup water systems, effluents plant, compressed air and auxiliary steam, natural gas regulation station, and other components. The gas turbine, steam turbine, and generator of each combined-cycle unit are located inside a building where the condenser is also housed (figure 14–12).



Figure 14–12 View of the Arcos combined-cycle power plant

The GE 209FB (2-to-1 multishaft) combined-cycle configuration at Arcos III consists of two GE frame 9FB gas turbine generators, one GE 209D steam turbine, three GE 330H generators, and two heat recovery steam generators. The steam turbine has a single high-pressure/intermediate-pressure (HP/IP) section and one dual-flow low-pressure (LP) section with control and shutoff valves for each sections. The HP steam supply is at 1800 psia (124 bar) and 1050°F (565°C) and the exhaust pressure is 1.73 inches Hg (0.059 bar).

Using natural gas, Arcos III is operating at a combined-cycle efficiency of around 58%, placing it among the world's most efficient combined-cycle power stations. The plant reached another milestone in October 2007, when it surpassed 8000 hours of operation.

Arcos III was the commercial launch site for GE's 50 hertz, 9FB gas turbine technology, a project commissioned by Iberdrola Ingeniería y Construcción. The frame 9FB is among the world's most advanced, air-cooled, 50-hertz gas turbines. Addressing the need for cleaner power, the frame 9FB gas turbines are equipped with GE's advanced, Dry Low NO_x 2+ combustion systems, which limit NO_x emissions to 25 parts per million or less. Table 14–6 summarizes the performance of Arcos III combined-cycle power plant.

Table 14–6 Main technical data of Arcos III combined-cycle power plant

Plant Configuration:		STAG 209FB
Number of blocks		1
Number of gas turbines		2
Number of steam turbines		1
Main fuel		Natural gas
Ambient Design Conditions:		
Ambient temperature	°C / °F	17.9 / 64.2
Ambient pressure	mbar	1005
Relative humidity	%	69
Cooling type		Cooling tower
Performance data per block:		
Total fuel input to gas turbines	MW	1423
Gross output	MW	838
Gross efficiency (LHV)	%	58.9
Auxiliary consumption and losses	MW	15
Net power output	MW	823
Net efficiency (LHV)	%	57.8
Gross heat rate (LHV)	kJ/kWh	6112.5
Gross heat rate (LHV)	BTU/kWh	5794
NO _x emissions	ppm	12
CO emissions*	ppm	0.6

*corrected at 15% O₂

The Arcos III project is designed to produce one-third of the CO₂ emitted by a conventional coal plant and 10 times lower levels of NO_x, while not emitting any SO₂.

The units also feature GE's SPEEDTRONIC™ Mark VI turbine control systems, which offers complete integrated control, protection, and monitoring for generator and mechanical drive applications of gas and steam turbines.

There are currently 43 GE Frame 9FB gas turbines operating or committed for projects worldwide, including 16 in the Iberian region.

The GE 209D steam turbine at Arcos III features 48-inch, last-stage buckets, the industry's largest steel, full-speed (3000 rpm) last-stage

15 Conclusion

The thermodynamic advantages of the combined-cycle concept enable efficiencies to be reached far above those of other types of thermal power plants. This technology above all others can fully exploit the high-temperature potential of modern gas turbines and the low-temperature cold end of the steam cycle. Coupled with low investment and operating costs, short delivery times, and high operational flexibility, these factors ensure an overall low cost of electricity.

The thermodynamic advantages are also beneficial in cogeneration applications, especially where high electrical output is required because of the stable electrical output contribution of the gas turbine, which is not influenced by the steam process. The possibility of supplementary firing in the HRSG provides even greater operational flexibility where variations in the steam extraction demand are required.

Combined-cycle plants are suitable for daily cycling operation due to short startup times as well as for continuous baseload operation. Part-load efficiencies are also high due to the control of the gas turbine mass flow using variable inlet guide vanes in the compressor.

Combined cycles can be cooled by a cooling tower, a direct water-cooling system, or air-cooled condensers ensuring a wide range of applications. Where water is scarce, they are advantageous because the cooling requirement per unit of electricity produced is low because the main cooling requirement applies only to the steam process (one-third of total output).

Combined-cycle technology is one of the most environmentally acceptable large-scale power plant technologies in use today. Fuelled mainly by natural gas, they are ideally suited for use in heavily populated regions due to low emission levels for pollutants such as nitrous oxides (NO_x), carbon monoxide (CO), and particles. High efficiency means less fuel must be burned for each unit of electricity produced, which also contributes to these low emission levels.

The carbon dioxide emissions (CO_2) are the lowest of all fossil-fired power plants. In a modern gas-fired combined-cycle plant they are only 40% of those of a coal-fired plant with the same output. This is, of course, a great advantage with today's awareness of climatic change as well as an economical advantage when CO_2 -emission certificates have to be acquired.

Today's combined-cycle plants have reached a very high technical standard: net efficiencies of up to 59% are achievable. Block sizes of over 400 MW are state of the art. Nevertheless the development in combined-cycle plants continues through the advancement of the constituent components, which are now being developed specifically for use in combined-cycle applications. This has influenced the gas turbine in particular, with a trend towards pressure ratios and turbine inlet temperatures optimized for combined-cycle applications and a move towards more innovative machine concepts such as sequential combustion or even steam cooling.

Although the fuel flexibility of combined-cycle plants is limited to gases and some oils, this is becoming less significant as global gas distribution increases bringing natural gas to many countries that do not have their own natural reserves. Gas is either transported through pipe or shipped as liquid (LNG).

A wide range of combined-cycle power plant concepts is available and selection of a cycle concept is made according to the criteria of a specific project. Now that the combined-cycle is established as one of the main global power generation technologies, ideal cycle solutions are emerging matched to certain sets of criteria. Manufacturers have recognized in this a potential for the development of a range of standard plants. This leads to shorter delivery times, faster permitting, and lower risk to the investor due to proven components and systems.

These and other factors point to a continuing dependence on combined-cycle technology for generating a main part of the world's electrical energy well into the future. Especially if the concept of carbon capture and storage (CCS) gets imposed, combined-cycle plants with integrated coal gasification (IGCC) could be the standard coal fired stations of the future.

About the Authors



Rolf Kehlhofer is an internationally recognized expert in the energy industry and, more specifically, in the field of combined-cycle power plants. The founder and managing partner of The Energy Consulting Group Ltd. in Zurich, Kehlhofer previously served as president of ABB Power Generation Switzerland. A Swiss citizen, Kehlhofer is a graduate of the Swiss Federal Institute of Technology. During his career, his contributions have resulted in nearly 60,000 megawatts of generated power worldwide.



Bert Rukes joined Siemens in 1978 and has held various leadership positions during his tenure there. He currently oversees plant design and development, and nuclear and steam operations. During his distinguished career, Rukes received the Siemens Top Innovator award for outstanding innovations in fossil power plant solutions. He is a member of VDI (the Association of German Engineers) and is an Honorary Fellow of the International Association of the Properties of Water and Steam. A mechanical engineering graduate at the University of Applied Sciences in Düsseldorf and the Technical University in Berlin, Rukes also holds a doctorate in mechanical engineering.



Frank Hannemann has a degree in Thermal Engineering from the Technical University of Zittau. After ten years with Siemens Power Generation, he became Director of IGCC and innovative power plant cycles with special focus on CO₂ free technologies. He is currently head of engineering at Siemens Fuel Gasification Technology GmbH and is responsible for technology development and engineering activities for gasification projects. Hannemann has written articles for numerous publications about integrated gasification combined-cycle plants (IGCC). He is a holder of several patents and his inventions have been named by Siemens AG as one of the most attractive solutions for future power generation.



Franz J. Stirnimann has held the position of senior consultant at The Energy Consulting Group Ltd. in Switzerland since 2005. He started his career as a process engineer and spent more than 25 years in the field of engineering for thermal power plants, mainly for combined-cycle plants within ABB Power Generation and later ALSTOM Power. In 1994 he became vice president for Combined Cycle Plant Engineering with worldwide responsibilities. During his career, he has been involved in the design and execution of all types of combined-cycle plants as well as development activities for plant standardization. A Swiss citizen, Stirnimann has a degree in mechanical engineering from the Swiss University of Applied Science and Arts in Lucerne.

List of Figures

Figure 1–1	Simplified flow diagram of a combined cycle	2
Figure 2–1	Market development since 1975 of new power plants sold per year	9
Figure 3–1	Breakdown of the capital requirement for combined-cycle power plant	19
Figure 3–2	The cost of fuels over the years	23
Figure 3–3	Comparison of cost of electricity for base-load operation	29
Figure 3–4	Comparison of cost of electricity for intermediate load	30
Figure 3–5	Influence of fuel cost on the cost of electricity	31
Figure 3–6	Influence of the equivalent utilization time on the cost of electricity	32
Figure 3–7	Influence of interest rate on the cost of electricity	32
Figure 4–1	Temperature/entropy diagrams for various cycles	37
Figure 4–2	Gas turbine inlet temperature (TIT) definitions	38
Figure 4–3	The efficiency of a simple-cycle GT and a combined cycle plant as function of the gas turbine inlet temperature and pressure ratio	42
Figure 4–4a and 4–4b	The efficiency of a simple-cycle GT with single-stage combustion as a function of turbine inlet temperature (TIT) and turbine exhaust temperature	43
Figure 4–5a and 4–5b	The efficiency of a simple-cycle GT with sequential combustion as a function of the turbine inlet tempera- ture (TIT) and the turbine exhaust temperature	44
Figure 5–1	Selection of a combined-cycle power plant concept	46
Figure 5–2	Evolutionary change in combined-cycle design philosophy	47
Figure 5–3	Standardization approach	48
Figure 5–4	Entropy/temperature diagram for a gas turbine process at two different ambient air temperatures	52
Figure 5–5	Relative efficiency of gas turbine, steam process, and combined cycle as function of the air temperature	53
Figure 5–6	Relative power output of a gas turbine, steam turbine, and combined cycle as function of the air temperature	54

Figure 5-7	Relative power output of gas turbine, steam turbine, combined cycle, and relative air pressure versus elevation above sea level	55
Figure 5-8	Relative power output and efficiency of gas turbine and combined cycle as function of relative humidity . . .	56
Figure 5-9	Effect of water and steam injection on relative combined-cycle power	59
Figure 5-10	Effect of condenser pressure on steam turbine output . . .	62
Figure 5-11	Temperature of cooling medium versus condenser pressure for different types of cooling systems	63
Figure 5-12	Flow diagram to show fuel preheating	65
Figure 5-13	Steam turbine output and HRSG efficiency versus gas turbine exhaust temperature for a single-pressure cycle . . .	68
Figure 5-14	Ratio of steam turbine output of a dual pressure compared to a single-pressure cycle as a function of the gas turbine exhaust temperature	69
Figure 5-15	Energy/temperature diagram for an idealized heat exchanger	71
Figure 5-16	Flow diagram of a single-pressure cycle	73
Figure 5-17	Energy/temperature diagram of a single-pressure HRSG	75
Figure 5-18	Heat balance for a single-pressure cycle	76
Figure 5-19	Energy flow diagram for the single-pressure combined-cycle power	77
Figure 5-20	Effect of live steam pressure on steam turbine output for a single-pressure cycle (including steam turbine exhaust moisture content and HRSG efficiency)	78
Figure 5-21	Energy/temperature diagram of a single-pressure HRSG with live-steam pressure of 40 and 105 bar (566 and 1508 psig)	79
Figure 5-22	Effect of live-steam pressure on condenser waste heat at constant condenser pressure	80
Figure 5-23	Effect of live-steam temperature on steam turbine output for a single-pressure cycle with 105 bar (1508 psig) live-steam pressure (including HRSG efficiency and steam turbine exhaust moisture content) . . .	82
Figure 5-24	Effect of pinch point on relative steam turbine power output and relative HRSG heating surface	83

Figure 5–25	Influence of HRSG backpressure on combined-cycle output and efficiency, GT output and efficiency, and HRSG surface	85
Figure 5–26	Effect of feedwater temperature on steam turbine output and HRSG efficiency for cycles with one stage of preheating	87
Figure 5–27	Energy/temperature diagram for a single-pressure HRSG	88
Figure 5–28	Energy/temperature diagram for a conventional boiler	89
Figure 5–29	Flow diagram of a single-pressure cycle with LP preheating loop for high sulfur fuels	92
Figure 5–30	Flow diagram of a dual-pressure cycle for high sulfur fuel	93
Figure 5–31	Effect of feedwater temperature and number of preheating stages on steam turbine output of a dual-pressure cycle	94
Figure 5–32	Flow diagram of a dual-pressure cycle with low sulfur fuel	95
Figure 5–33	Heat balance for a dual-pressure cycle with low sulfur fuel	96
Figure 5–34	Energy flow diagram for a dual-pressure combined-cycle plant	97
Figure 5–35	Energy/temperature diagram for a dual-pressure HRSG	98
Figure 5–36	Effect of the HP and LP pressure on steam turbine output and exhaust moisture content for a dual-pressure cycle	101
Figure 5–37	Effect of LP pressure on HRSG efficiency for a dual-pressure cycle	102
Figure 5–38	Effect of HP and LP steam temperature on steam turbine output for a dual-pressure cycle	103
Figure 5–39	Effect of HP and LP pinch point on steam turbine output and relative HRSG surface for a dual-pressure cycle	104
Figure 5–40	Flow diagram of a triple-pressure cycle	106
Figure 5–41	Heat balance of a triple-pressure cycle	107

Figure 5-42	Energy/temperature diagram of a triple-pressure HRSG	108
Figure 5-43	Energy flow diagram of a triple-pressure combined-cycle plant	109
Figure 5-44	Steam turbine output and exhaust moisture content versus HP and IP pressure for triple-pressure cycles at constant LP pressure (5 bar)	110
Figure 5-45	Effect of LP pressure on steam turbine output and relative HRSG surface for triple-pressure cycles at constant HP (105 bar) and IP (25 bar)	111
Figure 5-46	Live-steam temperature optimization for a triple-pressure cycle	112
Figure 5-47	Temperature/entropy diagram showing the effect of “mild reheat” on the steam turbine expansion line	113
Figure 5-48	Effect of HP and IP evaporator pinch point on steam turbine output and relative HRSG surface for a triple-pressure cycle with constant LP pinch point	114
Figure 5-49	Flow diagram of a triple-pressure reheat cycle	116
Figure 5-50	Heat balance for a triple-pressure reheat cycle	117
Figure 5-51	Temperature/entropy diagram showing the effect of full reheat on the steam turbine expansion line	118
Figure 5-52	Energy flow diagram for a triple-pressure reheat combined-cycle plant	118
Figure 5-53	Energy/temperature diagram for a triple-pressure reheat HRSG	119
Figure 5-54	Steam turbine output and HRSG surface versus HP and reheat pressure for a triple-pressure reheat cycle at constant HP and reheat temperature (568/568°C)	120
Figure 5-55	Steam turbine output versus HP and reheat steam temperature for a triple-pressure reheat cycle at constant HP (120 bar), IP (30 bar) and LP (5 bar) pressure	122
Figure 5-56	Flow diagram of a high-pressure reheat cycle with a HP once-through HRSG and a drum-type LP section	123
Figure 5-57	Energy/temperature diagram for 647°C (A), 750°C (B) and 1000°C (C) exhaust gas temperature entering the HRSG	126

Figure 5–58	Effect of temperature after supplementary firing on power output and efficiency relative to that of a single-pressure cycle	127
Figure 5–59	Heat balance for a single-pressure cycle with supplementary firing	128
Figure 5–60	Performance of different combined cycles over the exhaust gas temperature	130
Figure 5–61	Influence of various parameters/measure on combined-cycle output and efficiency	132
Figure 5–62	Impact of HP and reheat steam parameters (pressure and temperature) on combined-cycle net efficiency ..	133
Figure 6–1	Simplified flow diagram of a cogeneration cycle with a back pressure turbine	137
Figure 6–2	Flow diagram of a cogeneration cycle with an extraction/condensing steam turbine	138
Figure 6–3	Flow diagram of a cogeneration cycle with no steam turbine	139
Figure 6–4	Heat balance for a single-pressure cogeneration cycle with supplementary firing	140
Figure 6–5	Effect of process steam pressure on relative combined-cycle power output and power coefficient for a single-pressure cycle with 750°C supplementary firing	142
Figure 6–6	Effect of power coefficient on electrical efficiency and fuel utilization for a single-pressure cycle with 750°C supplementary firing	143
Figure 6–7	Flow diagram of a cogeneration cycle with a dual-pressure HRSG	144
Figure 6–8	Comparison of 1-stage and 3-stage heating of district heating water	146
Figure 6–9	Heat balance for a cycle with two stages of district heating	147
Figure 6–10	Flow diagram of a district heating/condensing cycle ..	148
Figure 6–11	Flow diagram of a cycle coupled with a seawater desalination plant	150
Figure 6–12	Flow diagram of a conventional non-reheat steam power plant	152
Figure 6–13	Flow diagram of a combined-cycle plant using an existing steam turbine	152

Figure 6–14	Flow diagram of a gas turbine combined with a conventional steam cycle (fully fired combined-cycle plant)	155
Figure 6–15	Flow diagram of a parallel-fired combined-cycle plant	156
Figure 6–16	Flow diagram of a PFBC process	160
Figure 6–17	Flow diagram of a STIG cycle	161
Figure 6–18	Flow diagram of a turbo STIG cycle	162
Figure 6–19	Flow diagram of a HAT cycle	163
Figure 7–1	Turbine Inlet Temperature	167
Figure 7–2	Industrial Trent derived from the aero Trent 800	169
Figure 7–3	Heavy-duty industrial gas turbine	170
Figure 7–4	Gas turbine with sequential combustion	171
Figure 7–5	Types of losses contributing to overall performance degradation	175
Figure 7–6	Forced circulation heat recovery steam generator	184
Figure 7–7	Natural circulation heat recovery steam generator	185
Figure 7–8	Principle of drum-type and once-through evaporation	187
Figure 7–9	Three-pressure reheat once-through HRSG with drum-type LP and IP sections	188
Figure 7–10	Supplementary fired heat recovery steam generator	193
Figure 7–11	Different single-shaft combined-cycle configurations	197
Figure 7–12	Cross section of a 142 MW reheat steam turbine with a separate HP turbine and a combined IP/LP turbine with axial exhaust	199
Figure 7–13	Cross section of a two-casing steam turbine with geared HP turbine	200
Figure 7–14	Cutaway drawing of an air-cooled generator for use in combined-cycle power plants	202
Figure 7–15	Single-line diagram	203
Figure 7–16	Typical arrangement of an air-cooled condenser	204
Figure 7–17	Typical arrangement of a wet cell cooling tower	205
Figure 7–18	Principle of hybrid cooling tower	206
Figure 8–1	Hierarchic levels of automation	213
Figure 8–2	Standard layout for a modern combined-cycle power plant control room	214
Figure 8–3	Principle diagram for a combined-cycle load control system	216

Figure 8–4	Typical combined-cycle droop characteristic of a GT load controller	218
Figure 8–5	Closed control loops in a combined-cycle plant	219
Figure 9–1	Sliding pressure diagram	227
Figure 9–2	Effect of condenser vacuum on combined-cycle efficiency	231
Figure 9–3	Effect of frequency on relative combined-cycle output and efficiency for full-load operation	232
Figure 9–4	Effect of fuel composition and lower heating value on combined-cycle output and efficiency (base-load, gas-operation with wet cooling tower)	234
Figure 9–5	Part-load efficiency of gas turbine and combined cycle	235
Figure 9–6	Ratio of steam turbine and gas turbine output and live-steam data of a combined-cycle plant at part load	236
Figure 9–7	Part-load efficiency of combined-cycle plant with four single-shaft blocks	238
Figure 9–8	Performance guarantee comparison	241
Figure 9–9	Expected non-recoverable combined-cycle power plant degradation of power output and efficiency with GT operating on clean fuels	242
Figure 9–10	Gas turbine compressor efficiency as function of days operation (or operation hours)	243
Figure 9–11	Relative power increase of a combined-cycle power plant as function of ambient temperature and humidity	247
Figure 9–12	Typical arrangement of a fogging system in the GT air intake	248
Figure 9–13	Air inlet cooling process with chiller in Mollier diagram	250
Figure 9–14	Typical diagram of a chiller system	251
Figure 9–15	Startup curve for a 250–400 MW class combined cycle after eight hours standstill	256
Figure 9–16	Startup curve for a 250–400 MW class combined cycle after 48 hours standstill	257
Figure 9–17	Startup curve for a 250–400 MW class combined cycle after 120 hours standstill	257
Figure 9–18	Combined-cycle shutdown curve	258
Figure 9–19	Switch over diagram from gas to oil operation	259
Figure 10–1	NO _x equilibrium as a function of air temperature	263

Figure 10–2	Flame temperature as a function of the fuel-to-air ratio and combustion air conditions	264
Figure 10–3	NO _x concentration as a function of fuel-to-air ratio and combustion air conditions	265
Figure 10–4	NO _x reduction factor as a function of the water or steam-to-fuel ratio in gas turbines with diffusion combustion	266
Figure 10–5	Cross section of a low NO _x burner	269
Figure 10–6	Cross section of the Siemens dry low NO _x burner	270
Figure 10–7	Heat recovery steam generator with selective catalytic reduction	272
Figure 11–1	Chronology of the gas turbine inlet temperatures based on improved material and cooling technologies	279
Figure 11–2	Geared high-pressure turbine	283
Figure 11–3	Siemens SGT5-8000H gas turbine	284
Figure 11–4	Plant impact for fast cycling capability	285
Figure 12–1	Principle IGCC concept	287
Figure 12–2	Gasification reactions	288
Figure 12–3	Basic technologies of gasification and gasifier technology vendors	291
Figure 12–4	Entrained flow gasifiers	292
Figure 12–5	Schematic of Siemens full-water quench gasifier with water scrubbing	296
Figure 12–6	Schematic of shell coal gasification process with heat recovery and dry fly ash removal	297
Figure 12–7	Regenerable solvent-type AGR process	298
Figure 12–8	Claus process	299
Figure 12–9	Physical absorption process for simultaneous sulfur and CO ₂ removal	302
Figure 12–10	ASU process	303
Figure 12–11	Gas turbine syngas conditioning system (Siemens concept)	304
Figure 12–12	Effect of dilution and heating value on NO _x emission	305
Figure 12–13	Effect of dilution and heating value on NO _x emission	306
Figure 12–14	Gas turbine mass flow imbalance	307
Figure 12–15	IGCC concept based on hard coal and Shell gasification and Siemens combined-cycle technology	309
Figure 12–16	Sankey diagram of Shell-based IGCC concept	311

Figure 12–17	IGCC concept based on lignite and HTW gasification and Siemens combined cycle	312
Figure 12–18	IGCC concept with CO ₂ capture based on hard coal and Siemens fuel gasifier (SFG) and Siemens combined-cycle technology	314
Figure 12–19	IGCC concept with CO ₂ capture based on Shell gasification and Siemens combined-cycle technology	316
Figure 12–20	Sankey diagram of shell-based IGCC concept with CO ₂ capture	317
Figure 13–1	Historical trend of energy consumption and global warming	322
Figure 13–2	CO ₂ -emissions mitigating by fuel switch	323
Figure 13–3	Typical specific CO ₂ -emissions from fossil fuels	324
Figure 13–4	Typical trend of solubility in physical and chemical washing agents	326
Figure 13–5	Principle of ion transport membrane	328
Figure 13–6	Principle of chemical looping combustion	329
Figure 13–7	Schema of pre-combustion CO ₂ -capture	329
Figure 13–8	Combined autothermal reforming and CO ₂ -capture	332
Figure 13–9	Concept of an innovative premix burner for hydrogen-rich syngases	334
Figure 13–10	Schema of post-combustion CO ₂ -capture	335
Figure 13–11	Post-combustion absorption process	336
Figure 13–12	Post-combustion CO ₂ capture under pressure	338
Figure 13–13	Schema of Oxyfuel firing CO ₂ capture	339
Figure 13–14	Concept of the Graz cycle	340
Figure 13–15	Direct Oxyfuel fired CO ₂ capture, CES cycle	341
Figure 13–16	Principle of the AZEP process	342
Figure 13–17	Principle of the ZESOFc process	343
Figure 13–18	Expected CO ₂ avoidance and power generation costs for industrial-scale power plants in operation by 2020 (Source: ZEP)	345
Figure 14–1	View of Taranaki combined-cycle plant	350
Figure 14–2	Process diagram of Taranaki combined-cycle power plant	351
Figure 14–3	Arrangement of Taranaki combined-cycle power plant	353

Figure 14–4	Process diagram for Monterrey combined-cycle power plant	355
Figure 14–5	Layout of the Monterrey combined-cycle power plant	357
Figure 14–6	View of the Phu My 3 combined-cycle power plant	358
Figure 14–7	Process diagram of the Phu My 3 combined-cycle power plant	359
Figure 14–8	Arrangement of the combined-cycle reference power plant SCC5-4000F 2×1	359
Figure 14–9	View of the Palos de la Frontera combined-cycle power plant	361
Figure 14–10	Layout of the Palos de la Frontera combined-cycle power plant	362
Figure 14–11	Process diagram of the Palos de la Frontera combined-cycle power plant (one unit)	363
Figure 14–12	View of the Arcos combined-cycle power plant	365
Figure 14–13	View of Diemen combined-cycle cogeneration plant	368
Figure 14–14	Process diagram of Diemen combined-cycle cogeneration plant	369
Figure 14–15	General arrangement of Diemen combined-cycle cogeneration plant	371
Figure 14–16	View of the Shuwei hat S1 plant	372
Figure 14–17	Process diagram of the Shuwei hat S1 plant	373
Figure 14–18	Vado Ligure, old layout	375
Figure 14–19	Vado Ligure, new layout	377
Figure 14–20	Thermal process of repowered plant	380
Figure 14–21	CAD Image of the repowered plant	381
Figure 14–22	Flow scheme of Puertollano IGCC plant	382
Figure 14–23	Main technical data of the IGCC Puertollano	384
Figure 14–24	Availability between 1996 and 2003	384
Figure 14–25	Process diagram of the Monthel cogeneration power plant	386
Figure 14–26	Plant arrangement	386
Figure A–1	Calculation of operating and part-load behavior: Method for solving the system of equations Conversion of the main units used in this book Conversion formulæ	402 403 403

List of Tables

Table 2–1	Installed capacity 2007	7
Table 2–2	World net electricity generation by type 2004 (TWh)	8
Table 2–3	World fuel consumption for fossil-fuelled power plants	8
Table 2–4	Fossil fuels: proved reserves vs. yearly consumption worldwide in 2007	10
Table 3–1	Specific price of various power plants in US\$/kW	19
Table 3–2	Net efficiency of various power plants	21
Table 3–3	Fuel flexibility of various power plants	22
Table 3–4	Variable operating and maintenance costs for various power plants of different sizes	24
Table 3–5	Fixed operation and maintenance costs for various power plants of different sizes	25
Table 3–6	Availability and reliability of various power plants	26
Table 3–7	Construction times for various power plants	28
Table 3–8	Inputs for economical comparison	29
Table 4–1	Thermodynamic comparison of gas turbine, steam turbine, and combined-cycle processes	36
Table 4–2	Allowable reduction in steam process efficiency as a function of gas turbine efficiency ($\eta_{ST} = 0.30$)	41
Table 5–1	Comparison of combined-cycle performance data for different cooling systems and ambient temperatures	61
Table 5–2	Possible fuels for combined-cycle applications	66
Table 5–3	Comparison multishaft versus single shaft	67
Table 5–4	Performance comparison of different cycle concepts (natural gas with low sulfur content and GT exhaust gas temperature of 647°C (1197°F))	129
Table 6–1	Conversion of a 500 MW steam turbine power plant into a parallel-fired combined-cycle plant	157
Table 7–1	Main characteristic data of modern gas turbines for power generation	171
Table 7–2	Critical fuel properties	176
Table 7–3	Typical composition of fuel gases for gas turbines	180

Table 7–4	Change of boundary conditions for steam turbines in combined-cycle plants	196
Table 8–1	Operation of steam turbine control loops in a single-shaft combined-cycle plant	222
Table 9–1	Overview of GT air inlet cooling systems	246
Table 9–2	Max ambient temperature and corresponding cooling potentials at different locations	246
Table 9–3	Typical example of combined cycle power improvement with chiller	252
Table 9–4	Specific investment costs for additional output of air inlet cooling system	253
Table 9–5	Expected startup times for a 400 MW combined-cycle plant	255
Table 10–1	Comparison of the heat to be dissipated for various types of 1,000 MW stations	274
Table 12–1	Emission of existing coal-based IGCC	289
Table 12–2	Commercial solid fuel IGCC plants 2007	290
Table 12–3	Allowed chemical impurities of gas turbine fuels (Siemens source)	295
Table 12–4	Investment and efficiency of IGCC concepts	313
Table 13–1	Typical hydrogen-enriched syngases downstream gas cleaning	332
Table 13–2	Characteristics of typical gas turbine fuels	333
Table 13–3	Commercially available CO ₂ absorption systems for post-combustion applications	337
Table 13–4	Selection of European CCS demonstration projects (source: The World Energy Book, issue 3)	347
Table 14–1	Combined-Cycle Plant examples—overview	349
Table 14–2	Main technical data of Taranaki combined-cycle power plant	352
Table 14–3	Main technical data of Monterrey combined-cycle power plant	356
Table 14–4	Main technical data of Phu My 3 combined-cycle power plant	360
Table 14–5	Main technical data of Palos de la Frontera combined-cycle power plant	364
Table 14–6	Main technical data of Arcos III combined-cycle power plant	367

Table 14–7	Main technical data of Diemen combined-cycle cogeneration plant	370
Table 14–8	Main technical data of the Shuweihat S1 Independent Water and Power Plant (IWPP)	374
Table 14–9	Main technical data of Vado Ligure combined-cycle power plant	378
Table 14–10	Main technical data of Monthel cogeneration plant	387

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Appendix A

Calculation of the Operating Performance of Combined-Cycle Installations

When determining the performance data of a combined cycle, the gas turbine data are usually given and only the HRSG and steam turbine can be individually calculated. This appendix shows the main steps involved in such a calculation process, starting with the HRSG.

1. Equations for the heat exchangers of the HRSG

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:

$$\sum \dot{m} = 0 \quad (\text{A-1})$$

The impulse equation can be simplified into:

$$\Delta p = f(\text{geometry}) \quad (\text{A-2})$$

However, because the pressure loss both in the economizer and in the evaporator has a negligible influence on the energy equations, the assumption

$$\Delta p = 0 \quad (\text{A-3})$$

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:

$$d\dot{Q} = k \cdot \Delta t \cdot \pi \cdot d \cdot dx \quad (\text{A-4})$$

If it is assumed that the heat transfer coefficient k remains constant over the entire length of the heat exchanger (economizer or evaporator), equation A-4 becomes:

$$\dot{Q} = k \cdot S \int_0^L \Delta t(x) \quad (\text{A-5})$$

In general, the expression cannot be integrated. The heat exchanger must, therefore, be dealt with in small elements.

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:

$$\int_0^L \Delta t(x) = \frac{\Delta t_{\text{Inlet}} - \Delta t_{\text{Outlet}}}{\ln \left(\frac{\Delta t_{\text{Inlet}}}{\Delta t_{\text{Outlet}}} \right)} = \Delta t_m \quad (\text{A-6})$$

This average value can also be used for a superheater, an evaporator, or a recuperator. The heat exchangers do not, in fact, operate in accordance with an ideal counterflow principle, but the errors remain negligible.

Substituting equation A-6 into equation A-5 yields:

$$\dot{Q} = k \cdot S \cdot \Delta t_m \quad (\text{A-7})$$

From equation A-1, the amount of heat exchanged can be expressed as follows:

$$\dot{Q} = \dot{m}_S \cdot \Delta h_S = \dot{m}_G \cdot \Delta h_G \quad (\text{A-8})$$

At the design point, equations A-7 and A-8 become:

$$\dot{Q}_0 = k_0 \cdot S \cdot \Delta t_{m0} \quad (\text{A-9})$$

$$\dot{Q}_0 = \dot{m}_{S0} \cdot \Delta h_{S0} = \dot{m}_{G0} \cdot \Delta h_{G0} \quad (\text{A-10})$$

Dividing equation A-7 by equation A-9 and equation A-8 by A-10 yields the formulae:

$$\frac{\dot{Q}}{\dot{Q}_0} = \frac{k \cdot \Delta t_m}{k_0 \cdot \Delta t_{m0}} \quad (\text{A-11})$$

$$\frac{\dot{Q}}{\dot{Q}_0} = \frac{\dot{m}_G \cdot \Delta h_G}{\dot{m}_{G0} \cdot \Delta h_{G0}} \quad (\text{A-12})$$

Subtracting equation A-12 from A-11 produces:

$$\frac{\Delta t_m}{\Delta t_{m0}} = \frac{k_0 \cdot \dot{m}_G \cdot \Delta h_G}{k \cdot \dot{m}_{G0} \cdot \Delta h_{G0}} \quad (\text{A-13})$$

This is the nondimensional, global equation of heat transfer for the heat exchanger. If, in addition, equation A-8 is taken into consideration and the heat transfer coefficient k is known, a system of equations is obtained that defines the heat exchanger.

2. Finding the heat transfer coefficient of HRSG sections

The heat transfer coefficient can be calculated using the following equation:

$$k = \frac{1}{\frac{1}{\alpha_G} + \frac{d_1}{2 \cdot \lambda} \cdot \ln \left(\frac{d_1}{d_2} \right) + \frac{d_1}{d_2 \cdot \alpha_S}} \quad (\text{A-14})$$

However, the relative values k/k_0 appear in the heat transfer equation. From this:

$$K = \frac{k}{k_0} = \frac{\frac{1}{\alpha_{G0}} + \frac{d_1}{d_2 \cdot \alpha_{S0}}}{\frac{1}{\alpha_G} + \frac{d_1}{d_2 \cdot \alpha_S}} \quad (\text{A-15})$$

The heat transfer coefficients on the gas side of the economizer and the evaporator (α_G) are from 0.1 to 0.01 times as large as those on the steam side (α_S). Moreover, both values always shift in the same direction (++, --).

For these reasons, the following approximation can be used:

$$K = \frac{k}{k_0} = \frac{\alpha_G}{\alpha_{G0}} \quad (\text{A-16})$$

The α -value on the gas end can be calculated as follows using the Nusselt number:

$$\text{Nu}_G = c \cdot \text{Re}^m \cdot \text{Pr}^n = \frac{\alpha_G \cdot d_1}{\lambda_G} \quad (\text{A-17})$$

Here, c , m , and n are constants that depend mainly upon the geometry involved. From this, the following expression is obtained:

$$\alpha_G = c' \cdot \lambda_G \cdot \text{Re}^m \cdot \text{Pr}^n \quad (\text{A-18})$$

If this is substituted into equation (A-16), the geometric constant c' disappears:

$$K = \frac{\lambda \cdot \text{Re}^m \cdot \text{Pr}^n}{\lambda_{G0} \cdot \text{Re}_0^m \cdot \text{Pr}_0^n} \quad (\text{A-19})$$

For gases, the Prandtl number is almost exactly a constant; therefore:

$$K = \frac{\lambda}{\lambda_{G0}} \cdot \left(\frac{\text{Re}}{\text{Re}_0} \right)^m \quad (\text{A-20})$$

For the Reynolds number, the following expression applies:

$$\text{Re} = \frac{c_G \cdot \rho_G \cdot d_1}{\mu_G} \quad (\text{A-21})$$

By substituting $\dot{m}_G/S$ for $c_G \cdot \rho_G$, one obtains:

$$\text{Re} = \frac{\dot{m}_G \cdot d_1}{\mu_G \cdot S} \quad (\text{A-22})$$

Then, substituting this expression into equation (A-20), the geometric parameters disappear:

$$K = \frac{\lambda_G}{\lambda_{G0}} \cdot \left(\frac{\dot{m}_G \cdot \mu_{G0}}{\dot{m}_{G0} \cdot \mu_G} \right)^m \quad (\text{A-23})$$

If the mass flow is constant, all that remains is:

$$K = \frac{\lambda_G}{\lambda_{G0}} \cdot \left(\frac{\mu_{G0}}{\mu_G} \right)^m \quad (\text{A-24})$$

For m , one can use 0.57 for HRSG tubes that are staggered and 0.62 for inline pipes.

The value of the expression $K = \frac{\lambda_G}{\lambda_{G0}} \cdot \left(\frac{\mu_{G0}}{\mu_G} \right)^m$ does not vary notably

and depends mainly on the properties of the gas. It can be replaced with the following approximation:

$$\frac{\lambda_G}{\lambda_{G0}} \cdot \left(\frac{\mu_{G0}}{\mu_G} \right)^m = 1 - (\bar{t}_0 - \bar{t}) \cdot 5 \cdot 10^{-4} \text{ (in SI-Units)} \quad (\text{A-25})$$

$\bar{t}_0$ and $\bar{t}$ are the average gas temperatures along the heat exchanger in the design and operating point. This produces the relative value of K :

$$K = \left(\frac{\dot{m}_G}{\dot{m}_{G0}} \right)^m \cdot 1 - (\bar{t}_0 - \bar{t}) \cdot 5 \cdot 10^{-4} \quad (\text{A-26})$$

In this equation, only the exponent m depends to a slight extent on the boiler geometry.

It is more complicated to calculate an exact K-value for the superheater because the heat transfer on the steam side is lower than that in the evaporator.

As soon as all these equations are available for all parts of the boiler, the HRSG is defined mathematically. Similar equations can also be formulated for calculating the condenser. With this methodology it is possible to calculate off-design points of operation without having to know the actual geometry of the boiler; only the design point has to be known.

3. The steam turbine

Most steam turbines in combined-cycle plants operate in sliding pressure mode and 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 swallowing capacity and one for its efficiency. The swallowing capacity can be defined using the Law of Cones (ellipse law).

$$\left(\frac{\dot{m}_S}{\dot{m}_{S0}} \right) = \frac{\bar{v} \cdot p_\alpha}{v_0 \cdot p_{\alpha 0}} \sqrt{\frac{p_{\alpha 0} \cdot v_{\alpha 0}}{p_\alpha \cdot v_\alpha}} \cdot \sqrt{\frac{1 - \left(\frac{p_\omega}{p_\alpha} \right)^{\frac{n+1}{n}}}{1 - \left(\frac{p_{\omega 0}}{p_{\alpha 0}} \right)^{\frac{n+1}{n}}}} \quad (\text{A-27})$$

In condensing steam turbines, the pressure ratio is always very small due to the low pressure at the steam turbine exhaust p_ω . If simplified, this makes it possible to replace the quadratic expression with 1. The ratio of the swallowing capacities is as well close to 1.

What remains is:

$$\left(\frac{\dot{m}_S}{\dot{m}_{S0}} \right) = \sqrt{\frac{p_\alpha \cdot v_{\alpha 0}}{p_{\alpha 0} \cdot v_\alpha}} \quad (\text{A-28})$$

At a constant rotational speed, the efficiency of a stage depends only upon the enthalpy drop involved. In part-load operation, however, no important change occurs in that drop 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 the design point.

The following formulae are used to calculate efficiency. For parts of the turbine operating in the superheated zone:

$$\eta_{\text{pol,dry}} = \text{constant}$$

For parts in the wet steam region:

$$\eta_{\text{pol}} = \eta_{\text{pol-dry}} - \frac{(1 - x_{\alpha}) + (1 - x_{\omega})}{2} \quad (\text{A-29})$$

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 isentropic efficiency:

$$\eta_{\text{is}} = \frac{1 - \left(\frac{P_{\omega}}{P_{\alpha}} \right)^{\frac{x-1}{x} \eta_{\text{pol}}}}{1 - \left(\frac{P_{\omega}}{P_{\alpha}} \right)^{\frac{x-1}{x}}} \quad (\text{A-30})$$

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 considering dummy piston, steam turbine exhaust, mechanical, and generator losses. The actual losses are based on the steam turbine type, size, and live-steam pressure. As a guidance the following losses can be considered:

- Piston losses

For reaction type steam turbines, the piston losses account for 400 to 1000 kW mechanical equivalent losses. On the other hand impulse type steam turbines by their nature do not need a piston to compensate the axial thrust and would only have gland steam losses accounting for approx. 0.2% of the mechanical output.

- Steam turbine exhaust losses
Steam velocity at the steam turbine exhaust cause enthalpy losses, generally in the range of 20 to 35 kJ/kg
- Mechanical losses
Bearing losses and other mechanical losses normally account for 0.3% mechanical losses
- Generator losses
The generator efficiency varies from 97 to 99% dependent on the steam turbine size

4. Solving the system of equations

All the equations in the HRSG, the steam turbine, and so on create a system that can be solved only by iteration.

The following values are known:

- Thermodynamic data at 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 cycle

Figure A–1 shows the method to find the solution. Starting with the superheater a first estimate for live-steam temperature and pressure is made. Using the Law of Cones and the energy equation, the live-steam flow and the gas temperature after the superheater are found. 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 solved.

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 to find a new approximation for feedwater temperature if the pressure in the feedwater tank varies. The boiler is then recalculated with this new value. Finally, the condenser pressure and extraction flow are determined in another iteration. Then, from these new values, the power output of the steam turbine can be determined.

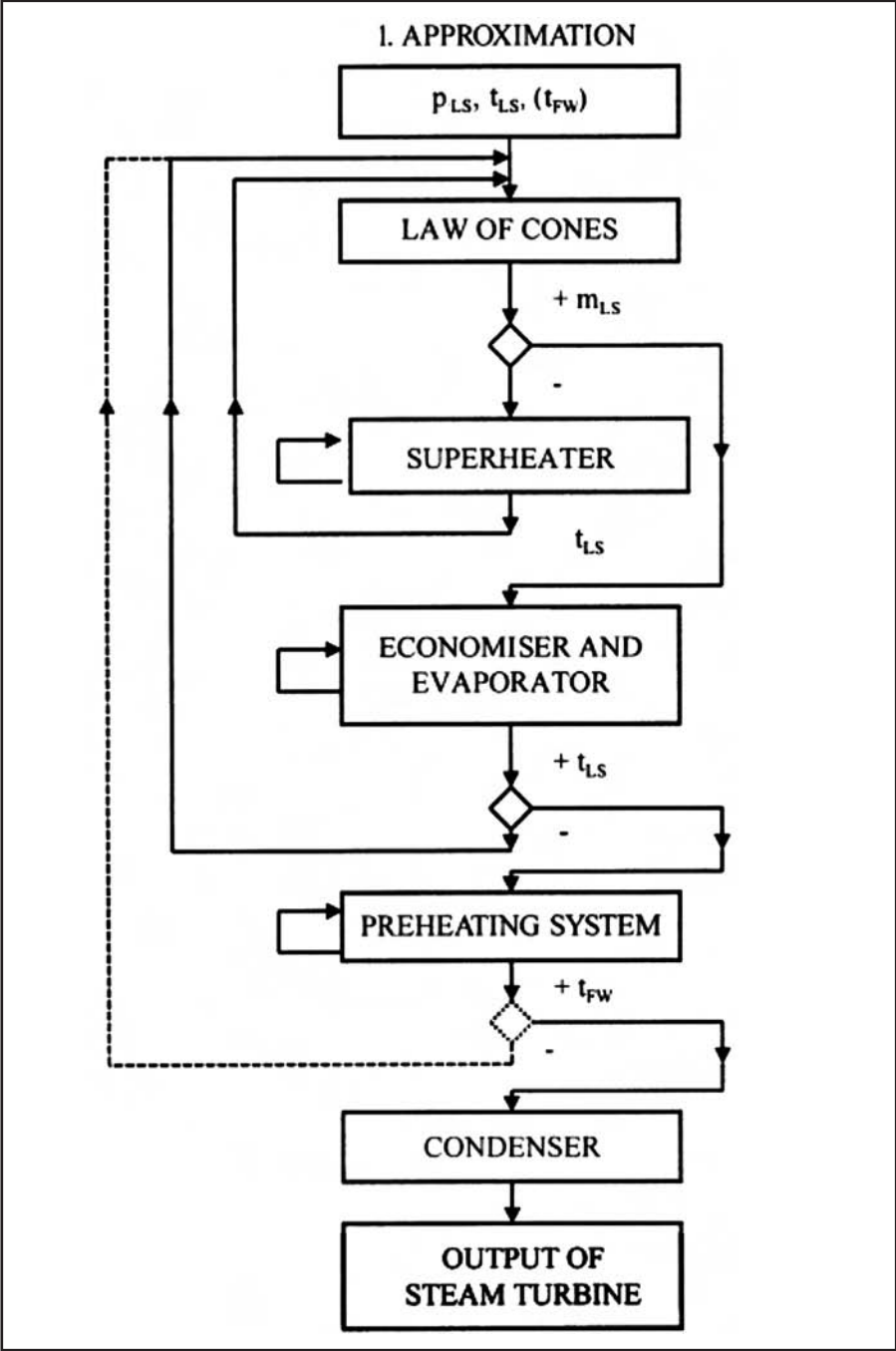


Figure A-1 Calculation of operating and part-load behavior: method for solving the system of equations

INDEX

Index Terms

Links

A

Abu Dhabi Water and Electricity Authority (ADWEA)	371
acid gas removal (AGR)	
alternatives to	300
for IGCC	298
processes for	297
advanced fast cycling (FACY) capability	364
Advanced Zero Emission Power plant (AZEP)	
process	341
aeroderivative gas turbine	169
efficiency of	168
aging	172
air	
combustion, supply of	154
gas turbine historical trends in	
compressor and flow of	167
air filter replacement	174
air pressure. <i>See</i> ambient air pressure	
air separation unit (ASU). <i>See also</i>	
cryogenic ASU	
future of	303
in Puertollano IGCC plant	383
Shell gasification and	310
use of	302

Index Terms

Links

air temperature. <i>See</i> ambient air temperature			
air-cooled blades	280		
air-cooling system. <i>See</i> direct air-cooling system			
alkali compounds	295		
alpha-value. <i>See</i> power coefficient			
alternative fuels	159		
alternative working media	164		
ambient air pressure			
operating behavior and	229		
power output impacted by site elevation and	54		
ambient air temperature			
combined-cycle plant efficiency and	52		
evaporative cooling systems and	246		
gas turbine efficiency and	51		
NO _x emissions and	262		
operating behavior and	229		
power output impacted by	53		
steam process efficiency impacted by	52		
ambient conditions			
gas turbines and	50		
steam turbines and	50		
ammonia	3	271	296
bottoming cycle, water mixing with	164		
amortization	11		
production costs impacted by	12		
Ansaldo Energia	379		
approach temperature	75		
HRSG and	84		

This page has been reformatted by Knovel to provide easier navigation.

Index Terms

Links

Arcos III combined-cycle plant	
efficiency of	367
history of	365
multishaft combined-cycle plant	
configuration of	366
technical data of	367
ash-forming fuels	
contaminants in	180
ignition problems for	181
automation. <i>See also</i> hierarchic levels of	
automation	
in base load operation	213
complexity of	211
autothermal reforming (ATR)	330
precombustion CO ₂ capture and	332
availability	
determining	26
of power plants	25
average electrical efficiency	14

B

back-pressure turbine	
cogeneration with	137
fuel utilization and	143
balance of plant (BOP)	208
base load operation	13
automation in	213
electricity costs, power plant comparisons	
for	29

biomass power plant. *See* power plants

This page has been reformatted by Knovel to provide easier navigation.

Index Terms

Links

blast furnace gas	179
boiler control	208
bottoming cycle	
ammonia mixing with water in	164
defining	2
water/steam cycle and	2
BTU natural gas	175
characteristics of	179
build-operate-transfer (BOT) project	357
butane	179
bypass stacks	
economics and	208
gas turbines and	207
steam turbines and	207

C

capital cost	29
elements impacting	14
capture and storage (CASTOR)	337
carbon capture and storage (CCS).	
<i>See also</i> Oxyfuel	
demonstration projects for	346
economics, future estimations for	345
efficiency and	324
EOR and	325
studies on	343
carbon dioxide (CO ₂) capture. <i>See also</i>	
Oxyfuel; postcombustion CO ₂ capture;	
precombustion CO ₂ capture	
efficiency/economics of	343

This page has been reformatted by Knovel to provide easier navigation.

Index Terms

Links

carbon dioxide (CO ₂) capture (<i>Cont.</i>)		
IGCC concept based on hard coal, SFG, Siemens combined-cycle and	314	
IGCC concept based on hard coal, Shell gasification, Siemens combined-cycle and	316	
carbon dioxide (CO ₂) emissions		
combined-cycle plants producing	273	390
from fossil fuels	323	
global warming and	321	
power plant economics impacted by	33	
reducing	323	
carbon dioxide (CO ₂) separation		
categories of	325	
from gas mixtures	325	
membranes and	327	
precombustion CO ₂ capture and	331	
sorption and	326	
carbon (CO) shift	330	
Carnot efficiency	53	
calculating	35	
of combined-cycle plants	36	
gas turbines and	36	
of steam turbines	36	
casing. <i>See</i> cold-casing design; hot-casing design		
CES cycle	340	
chemical looping	328	

Index Terms

Links

chiller	245	
advantages of using	249	
combined-cycle plant power output		
improvement with	251	
control system for	252	
economics and	250	
GT air inlet cooling with	249	
TES with	251	
typical diagram for	251	
circulation		
HRSG and forced	184	
HRSG and natural	185	
Claus process	299	309
CLINSULF	299	313
closed control loops. <i>See also</i> drum level control		
in dual-pressure cycle	219	
feedwater temperature and	220	
live-steam pressure and	221	
live-steam temperature and	220	
load control/frequency control in	216	
steam turbines, operation of	222	
types of	214	
closed-circuit air cooling generators	201	
closed-steam cooling	280	
clutch	198	
coal. <i>See also</i> hard coal; shell coal gasification		
process; steam power plant		
availability of	9	
proved reserves v. yearly consumption of	10	

Index Terms

Links

coal plants	8
<i>See also</i> pulverized coal (PC) plants	
cogeneration. <i>See also</i> Diemen combined-cycle cogeneration plant; Monthel cogeneration plant	
with back-pressure turbine	137
categories of	135
design for	144
dual-pressure cycle HRSG and	144
evaluating	139
with extraction/condensing steam turbine	138
heat balance for single-pressure cycle with supplementary firing and	140
without steam turbine	139
supplementary firing HRSG in	136
thermal energy in	135
cold-casing design	188
combined-cycle plants. <i>See also</i> Arcos III combined-cycle plant; Diemen combined-cycle cogeneration plant; dual-pressure cycle; exhaust gas; fully fired combined-cycle plant; Monterrey combined-cycle plant; Monthel cogeneration plant; multishaft combined-cycle plants; Palos de la Frontera combined-cycle plant; parallel-fired combined-cycle plant; Phu My 3 combined-cycle plant; power plants; Puertollano IGCC plant; Shuweit S1 Independent Water and Power Plant; Siemens combined-cycle;	

<u>Index Terms</u>	<u>Links</u>			
combined-cycle plants (<i>Cont.</i>)				
single-pressure cycle; single-shaft combined-cycle plants; Taranaki combined-cycle plant;				
Vado Liguire power station				
additional equipment for	209			
advantages of	3			
alternative fuel applications for	159			
alternative working media for	164			
ambient air temperature impacting				
efficiency of	52			
capital requirement breakdown for	19			
Carnot efficiency of	36			
chiller improving power output in	251			
CO ₂ emissions produced in	273	390		
competitive standing of	17			
concept selection for	45			
control room layout in	213			
conventional steam turbines v.	195			
cooling media configurations for	204	207	389	
defining	1			
degradation typical to	174	242		
design philosophy evolution in	47			
development trends in	277			
efficiency improvements for	33			
efficiency, with added heat in HRSG of	38			
efficiency, without added heat in HRSG of	40			
emissions from	262			
environment, impact of	6	275	390	
evaporative cooling system for	204			

Index Terms

Links

combined-cycle plants (*Cont.*)

evaporative cooling system, power output		
increase in	246	
feedwater preheating in	86	
flow diagram of	2	
frequency impacting power output in	232	
fuel flexibility of	390	
fuel types for	66	
future trends in	133	390
gas turbine efficiency and	234	
gas turbine's application in	172	
gas turbines converted to	158	
gas turbines optimized for	42	
global examples of	349	
installed capacity of	7	
investors, popularity of	24	
legislation and selection of	57	
LHV of	3	
load control droop characteristic in	217	
off-design behavior corrections for	228	
off-design behavior of	225	
PFBC applications for	159	
price comparisons for	18	
process steam pressure impacting power		
coefficient/power output in	141	
reheat impacting efficiency of	133	
seawater desalination plants coupled with	149	
shutdown for	258	
startup time importance to	196	
startup time ranges of	253	

Index Terms

Links

combined-cycle plants (<i>Cont.</i>)		
steam injection, water injection		
impacting	58	
steam process efficiency equation for	39	
technical standard of	390	
temperature/entropy diagrams for	37	
testing procedures for	238	
thermodynamics of	35	389
TIT impacting efficiency in	41	
waste heat reduced in	273	
water/steam cycle design challenges for	70	
combustion. <i>See also</i> oxygen-lean combustion;		
oxygen-rich combustion; postcombustion		
CO ₂ capture; precombustion CO ₂ capture;		
sequential combustion; staged combustion;		
stoichiometric combustion		
air supply for	154	
gas turbines and	166	
combustion with excess air	267	
Comisión Federal de Electricidad (CFE)	353	
compressor		
development of	165	
exit pressure and	306	
fouling in	172	
gas turbine historical trends in air flow		
of	167	
improving	281	
on-line washing/off-line washing for	173	
power consumed by	51	
pressure increased by	65	

Index Terms

Links

condenser. <i>See also</i> hotwell	
deaeration in	99
live-steam pressure impacting	80
reusing	153
condenser pressure	
cooling medium temperature v.	62
economics impacted by	62
condenser vacuum	56
efficiency impacted by	230
ConocoPhillips E-Gas gasification process	294
construction	
economics, interest and time for	27
of HRSG without supplementary firing	183
power plant comparisons on time for	27
contaminants	180
control room	213
control system. <i>See also</i> closed control loops;	
frequency control; load control	
for chiller	252
for evaporative cooling system	252
features of	212
for fogging system	252
hierarchic levels of automation in	212
for hotwell	222
necessity of	208
for power augmentation	223
for process energy	223
role of	211
for supplementary firing	223
for Taranaki combined-cycle plant	353

Index Terms

Links

control system (<i>Cont.</i>)			
VIGV/TIT and	215		
conventional boiler			
energy/temperature diagram for			
conventional	89		
feedwater preheating in	89		
gas turbine repowering, adapting	154		
conversions table	403		
converter gas	179		
cooling media. <i>See also</i> closed-steam cooling;			
GT air inlet cooling; steam cooling			
combined-cycle plant configurations for	204	207	389
condenser pressure v. temperature of	62		
dry cooling system with ACC for	61		
options for	60		
reusing	153		
temperature impacting	61		
TIT chronology aided by	279		
water for	60		
cooling water temperature	59	204	
operating behavior and	230		
CoP E-Gas IGCC plant	318		
COREX gas	179		
correction factors	239		
corrosion. <i>See also</i> low-temperature corrosion			
alkali compounds causing	295		
gas turbines impacted by	175		
costs. <i>See</i> capital cost; electricity costs; fuel			
costs; investment costs; operation and			
maintenance costs; production costs			

This page has been reformatted by Knovel to provide easier navigation.

Index Terms

Links

credit	11
power plants and	49
cryogenic ASU	
IGCC, process of	302
oxygen generation and	327
customer requirements	45
determining	48
customization	48
cycle concept	60
cycle performance summary	130

D

damper	207	
deaeration		
in condenser	99	
importance of	98	
leakages and	98	
types of	99	
deaerator	91	
eliminating	99	
debt-to-equity ratio	14	
IPPs and	12	
dedusting	296	
degradation. <i>See also</i> aging; corrosion; fouling		
combined-cycle plant, typical	174	242
gas turbines, causes of	172	242
maintenance for limiting	285	
performance value impacted by	241	
preventing	173	

Index Terms

Links

deregulation	
electric power market and	5
reliability and	26
desalination plants	149
<i>See also</i> multistage flash	
desalination units; seawater desalination plants	
Shuweit S1 IWPP as	373
design. <i>See also</i> cold-casing design; drum-type	
design; hot-casing design; off-design	
behavior; once-through design	
for cogeneration	144
combined-cycle plants, evolution of	47
combined-cycle plants, water/steam cycle	
challenges for	70
of Diemen combined-cycle cogeneration	
plant	368
district-heating power plants, criteria	
for	146
for economizer	191
HRSG, horizontal/vertical comparisons	
for	185
HRSG, optimizing	190
PFBC, complexities of	160
of Puertollano IGCC plant	382
single-pressure cycle, parameters of	77
of Taranaki combined-cycle plant	350
of Vado Ligure power station	378
design point	227
development time	28

Index Terms

Links

Diemen combined-cycle cogeneration plant	
design of	368
efficiency of	369
general arrangement of	371
process diagram of	369
technical data of	370
dilution	305
direct air-cooling system (ACC)	
arrangement of	204
cooling media using dry cooling system	
with	61
dirty fuels	287
district-heating power plants	135
design criteria for	146
extraction/condensing steam turbines	
in	148
heat balance for dual-pressure cycle and	147
process of	145
steam extraction, pressure stages in	145
drive level	212
droop characteristic	217
drum level control	219
drum-type design	188
evaporation in.	187
dry cooling system	204
cooling media using ACC and	61
dual-pressure cycle	
closed control loops in	219
cogeneration with HRSG in	144
district-heating power plant's heat balance for	147

This page has been reformatted by Knovel to provide easier navigation.

Index Terms

Links

energy flow diagram for	97
energy/temperature diagram for HRSG in	98
exhaust temperature, single-pressure cycle v.	69
heat balance for	96
for high sulfur fuels	93
live-steam pressure in	100
live-steam temperature in	102
for low sulfur fuels	95
moisture content, triple-pressure cycle	
compared to	113
pinch point of HP/LP evaporator in	104
steam turbines, single-pressure cycle v.	96
dual-pressure reheat cycle	
energy flow diagram for	123
performance benefit in	123
dynamic behavior	226

E

economics

bypass stacks and	208
CCS, future estimations for	345
chiller and	250
CO ₂ capture and	343
CO ₂ emissions impacting power plants	
and	33
condenser pressure impacting	62
construction time and	27
environmental restraints impacting	277
higher live-steam pressure and	80
main-steam parameters increased and	282

Index Terms

Links

economics (*Cont.*)

operational flexibility and	284
sliding pressure and	226
STIG cycles and	161
TIT increases and	278
of turbo STIG cycle	162

economizer

design for	191
role of	74
temperature changes in	88

efficiency. *See also* average electrical efficiency;

Carnot efficiency; gas turbine efficiency; net

efficiency; steam process efficiency

of aeroderivative gas turbine	168
ambient air temperature impacting	
combined-cycle plant and	52
of Arcos III combined-cycle plant	367
CCS and	324
closed-steam cooling aiding	280
CO ₂ capture and	343
combined-cycle plants, improving	33
combined-cycle plants, TIT impacting	41
of combined-cycle plants with added heat	
in HRSG	38
of combined-cycle plants without added	
heat in HRSG	40
condenser vacuum impacting	230
cycle performance summary for	130
of Diemen combined-cycle cogeneration	
plant	369

Index Terms

Links

efficiency (*Cont.*)

fuel costs, importance of	19	
gas turbines, pressure ratio impacting	41	65
of HAT cycle	163	
of HTW process	310	
of IGCC concepts	313	
LP impacting HRSG's	101	
of Monterrey combined-cycle plant	354	
of Monthel cogeneration plant	385	
of Oxyfuel	344	
of Palos de la Frontera combined-cycle plant	361	
part-load behavior improvements for reheat impacting combined-cycle plants and	237	
repowering increasing	133	
of Shell gasification	151	
steam cooling aiding	317	
steam injection impacting	280	
steam turbine equations for	57	
temperature, after supplementary firing, impacting	399	
water injection, problems with	128	130

ELCOGAS	266	
---------	-----	--

electric power market

deregulation and	382	
priorities of	5	
production costs, risks in	5	

electrical corrections	6	
------------------------	---	--

Index Terms

Links

electricity		
power plant's type impacted by value of	13	
production cost of	14	
storage challenges with	5	
world net generation, by type, of	8	
electricity costs		
calculating	28	33
equivalent utilization time impacting	31	
fuel costs impacting	30	
interest rate impacting	31	
power plant comparisons, base load		
operation and	29	
power plant comparisons, intermediate		
load operation and	30	
Electricity of Vietnam (EVN)	357	
electricity power demand	48	
emissions	6	
<i>See also</i> carbon dioxide (CO ₂)		
emissions; nitrous oxide (NO _x) emissions;		
SO _x emissions		
combined-cycle plants and	262	
of IGCC	289	
legislation limiting	57	
Switzerland legislation on	387	
types of	261	
Vado Liguire power station and	381	
energetic loses		
exergetic loses compared to	35	
steam turbines, exergetic losses v.	78	

Index Terms

Links

energy consumption	327	331	
global warming, rise in	321		
energy/temperature diagram			
for conventional boiler	89		
for dual-pressure cycle HRSG	98		
for heat exchanger	71		
for HRSG and supplementary firing	126		
for single-pressure cycle HRSG	75	79	88
for triple-pressure cycle HRSG	108		
for triple-pressure reheat cycle HRSG	119		
engineering procurement and construction			
contract (EPC)	13	47	
Enhanced CO ₂ Capture (ENCAP) project	334		
enhanced oil recovery (EOR)	337		
CCS and	325		
entrained-flow gasifiers	290		
environment. <i>See also</i> global warming			
combined-cycle plants' impact on	6	275	390
economics impacted by restraints of	277		
exhaust gas impacting	261		
IGCC and	287		
leakages impacting	164		
waste heat form impacting	274		
equivalent utilization time			
determining	16		
electricity costs impacted by	31		
European Technology Platform for Zero Emission			
Fossil Fuel Power Plants (ETP-ZEP)	344		

<u>Index Terms</u>	<u>Links</u>
evaporation	149
once-through design/drum-type design	
and	187
evaporative cooling systems	
ambient air temperature and	246
for combined-cycle plants	204
combined-cycle plants, power output	
increase in	246
control system for	252
fogging system compared to	247
goal of	246
plume abatement concept for	206
evaporator	
dual-pressure cycle, pinch point in	
HP/LP and	104
role of	74
triple-pressure cycle, pinch point in	
HP/LP and	114
evaporator loop	93
feedwater preheating with	91
exergetic losses	
energetic losses compared to	35
steam turbines, energetic losses v.	78
exergy transfer	
improving	123
in triple-pressure reheat cycle	117
exhaust gas	74
environment impacted by	261
evaluating	131
HRSG, pressure loss in	84

Index Terms

Links

exhaust gas (<i>Cont.</i>)	
water/steam cycle's heat transfer v.	91
exhaust losses	400
exhaust temperature	
dual-pressure cycle v. single-pressure cycle	
and	69
gas turbines impacted by	68
exit pressure	306
extraction/condensing steam turbines	
cogeneration with	138
in district-heating power plants	148
F	
far field noise	275
fast cycling provisions	285
<i>See also</i> advanced fast	
cycling (FACY) capability	
feedwater preheating	
combined-cycle plant and	86
conventional boiler and	89
evaporator loop for	91
HRSG impacted by	86
recirculation and	90
in steam power plant	86
steam turbines impacted by	94
feedwater temperature	
closed control loops and	220
HRSG and control for	221

Index Terms

Links

financing structure	
creating	14
terms of	13
finned tubing	189
fixed costs	17
power plants, O&M costs and	25
fixed-bed processes	290
flame temperature	
NO _x emissions, gas turbines and	263
in stoichiometric combustion	264
flashing	315
in seawater desalination plants	149
flexibility. <i>See</i> operational flexibility	
FLEXSORB unit	313
fluidized-bed (FB) processes	290
fly ash	296
fogging system	245
<i>See also</i> high-fogging system	
control system for	252
evaporative cooling system compared to	247
principles of	247
typical arrangement	248
fossil fuels	
CO ₂ emissions from	323
consumption of	8
price volatility of	23
proved reserves v. yearly consumption for	10
fouling	
in compressor	172
of turbines	174

Index Terms

Links

frequency		
combined-cycle power output impacted by	232	
operating behavior and	231	
frequency control	216	
closed control loops and	216	
load control compared to	217	
front-end engineering design (FEED) studies	319	
fuel. <i>See also</i> alternative fuels; dirty fuels;		
gaseous fuels; high sulfur fuels; liquid fuels;		
low sulfur fuels		
availability of	9	
combined-cycle plant, flexibility of	390	
combined-cycle plant, types of	66	
critical properties of	176	
for gas turbines	175	
LHV's importance in	64	233
off-design behavior and	233	
power plants flexibility of	22	
power plants, selecting	21	
preheating	64	
proved reserves v. yearly consumption of	10	
fuel changeover		
diagram for	260	
duration of	259	
purpose of	258	
steps for	259	
fuel costs		
average electrical efficiency and	14	
efficiency's importance to	19	
electricity costs impacted by	30	

Index Terms

Links

fuel costs (<i>Cont.</i>)	
volatility of	23
fuel utilization	
back-pressure turbine and	143
determining	139
single-pressure cycle, power coefficient	
impacting	142
fuel-to-air ratio	263
fully fired combined-cycle plant	154
functional group level	212

G

gas mixtures	325
gas turbine efficiency	
ambient air temperature and	51
combined-cycle plants and	234
increasing	40
part-load behavior and	235
steam process efficiency impacting	41
gas turbines. <i>See also</i> aeroderivative gas turbine;	
GT air inlet cooling; heavy-duty industrial gas	
turbines; Siemens SGT5-8000H gas turbine;	
steam-injected gas turbine (STIG) cycles;	
turbine inlet temperature	
ambient conditions and	50
bypass stacks and	207
capacity improvements of	284
Carnot efficiency of	36
categories of	168
combined-cycle plants, application of	172

Index Terms

Links

gas turbines (*Cont.*)

combined-cycle plants converted from	158	
combined-cycle plants with optimized	42	
combustion and	166	
conventional boiler adaptation for		
repowering	154	
corrosion impacting	175	
degradation causes in	172	242
exhaust temperature impacting	68	
fuel for	175	
high-fogging system impacting	248	
HRSG's function for	183	
ignition and	254	
inspections for	172	
load jumps and	218	
multi-shaft v. single shaft	67	
NO _x emissions, flame temperature in	263	
plant concept solution, selecting	66	
power output data for	171	
pressure ratio impacting efficiency in	41	65
price comparisons for	18	
process of	165	
reheat impacting	114	
sequential combustion and	170	
sequential combustion's advantages for	44	
in single-pressure cycle	171	
startup quickness of	253	
syngas utilization in modified	305	
TIT, compressor air flow historical trends in	167	
TIT increased in development of	165	

Index Terms

Links

gas turbines (<i>Cont.</i>)	
turbine mass flow imbalance in	306
gaseous fuels	
typical composition of	180
Wobbe index for comparing	177
gasification. <i>See also</i> high temperature winkler (HTW) process; Integrated gasification combined-cycle plants; partial oxidation (POX) gasification	
basic technologies for	290
ConocoPhillips E-Gas gasification process for	294
GE coal gasification process for	293
history of	290
MHI dry-feed gasifier for	294
SCGP for	292
SFG and	293
syngas process in	288
GE coal gasification process	293
GE energy, IGCC concept	318
geared high-pressure turbine	282
generator losses	400
generators. <i>See</i> heat recovery steam generator	
GE's SPEEDTRONIC	367
global warming	1
CO ₂ emissions and	321
energy consumption impacting	321
natural gas mitigating	324
NO _x emissions and	262
Graz cycle	339

Index Terms

Links

GT air inlet cooling

chiller impacting	249
investment costs of	253
overview of	246
types of	245

H

hard coal

IGCC concept based on CO ₂ capture, SFG, Siemens combined-cycle and	314
IGCC concept based on CO ₂ capture, Shell gasification, Siemens combined-cycle and	316

heat balance

for cogeneration, single-pressure cycle with supplementary firing	140
for district-heating power plant with dual-pressure cycle	147
for dual-pressure cycle	96
performance guarantees, computer model for	240
for single-pressure cycle	76
for single-pressure cycle with supplementary firing	129
for triple-pressure cycle	106
for triple-pressure reheat cycle	116

heat exchanger(s). *See also* economizer;

evaporator; superheater

energy exchange in idealized	70
energy/temperature diagram for	71

This page has been reformatted by Knovel to provide easier navigation.

Index Terms

Links

heat exchanger(s). (*Cont.*)

HRSG, equations for 393

HRSG, sections of 74

heat recovery steam generator (HRSG).

See also economizer; evaporator; superheater

approach temperature and 84

cogeneration with dual-pressure cycle
and 144

cogeneration with supplementary firing in 136

cold-casing design for 188

combined-cycleplants' efficiency, added heat in 38

combined-cycle plants' efficiency, without
added heat in 40

construction without supplementary
firing for 183

contradictory conditions for 189

design, horizontal/vertical comparisons
for 185

design optimization for 190

dual-pressure cycle, energy/temperature
diagram for 98

energy/temperature diagram for
supplementary firing and 126

exhaust gas pressure loss in 84

feedwater preheating impacting 86

feedwater temperature control and 221

finned tubing for 189

forced circulation in 184

gas turbines, function of 183

heat exchanger equations for 393

Index Terms

Links

heat recovery steam generator (HRSG) (*Cont.*)

heat exchanger sections of	74		
heat transfer coefficient for	395		
heat transfer factors for	71		
heat utilization improved in	64		
horizontal type of	185		
hot-casing design for	188		
limited supplementary firing in	192		
low-temperature corrosion in	90	190	
LP impacting efficiency of	101		
maximum supplementary firing in	194		
natural circulation in	185		
once-through design for	186		
in Palos de la Frontera combined-cycle plant	363		
pinch point determining heating surface of	75	83	
pinch points in modern	190		
with SCR	271		
in single-pressure cycle	74		
single-pressure cycle, energy/temperature diagram for	75	79	88
startup challenges with	190		
supplementary firing and	125		
temperature control and	220		
triple-pressure cycle, energy/temperature diagram for	108		
triple-pressure reheat cycle, energy/ temperature diagram for	119		
vertical type of	184		
heat transfer coefficient	395		

This page has been reformatted by Knovel to provide easier navigation.

Index Terms

Links

heavy-duty industrial gas turbines	170
developments in	169
power output of	168
Henry' s law	326
hierarchic levels of automation	212
high sulfur fuels	91
dual-pressure cycle for	93
high temperature winkler (HTW) process	
efficiency of	310
IGCC concept based on lignite, Siemens	
combined-cycle and	311
high volatile liquid fuels	182
higher heating value (HHV)	313
determining	20
high-fogging system	
gas turbines impacted by	248
potential of	252
power augmentation increased in	248
high-pressure (HP)	57
dual-pressure cycle, pinch point in	
evaporator with	104
pinch point, LP's relation to	105
steam turbine power output impacted by	
LP and	100
triple-pressure cycle, pinch point in	
evaporator with	114
hot-casing design	188
hotwell	
control system for	222
steam turbines and	99

Index Terms

Links

HT sour shift	300	
humid air turbine (HAT) cycle		
challenges of	164	
efficiency of	163	
humidity. <i>See</i> relative humidity		
hybrid cooling tower	206	
hydrogen	177	305
precombustion CO ₂ capture and behavior of	333	
hydrogen-cooled generators	201	
I		
ignition		
ash-forming fuels, problems with	181	
in gas turbines	254	
independent power producers (IPPs)		
debt-to-equity ratio and	12	
risks and	11	
SPC used by	13	
indices	406	
indirect air-cooling system	204	
industrial power stations	135	
steam extraction in	136	
inlet guide vane (IGV)	258	
inspections		
gas turbines and	172	
intervals between	168	
installed capacity		
of combined-cycle plants	7	
power plant evolution in	9	
of steam turbines	7	

Index Terms

Links

integrated CO ₂ capture. <i>See</i> Oxyfuel		
integrated gasification combined-cycle plants		
(IGCC)	1	
<i>See also</i> CoP E-Gas IGCC plant;		
Puertollano IGCC plant; Zero Emission		
Integrated Gasification Combined Cycle		
AGR processes for	298	
CO ₂ capture, hard coal, SFG, Siemens		
combined cycle concept for	314	
CO ₂ capture, hard coal, Shell gasification,		
Siemens combined cycle concept for	316	
concepts for	308	
cryogenic ASU process in	302	
demonstration plants for	289	
efficiency/investment cost of concepts for	313	
emissions of	289	
environment and	287	
GE energy concept for	318	
investment costs of	289	319
lignite, HTW process, Siemens combined		
cycle concept for	311	
Rectisol/Selexol and	302	
Shell gasification, Siemens combined-cycle		
concept for	308	
syngas conditioning system in	304	
Integrated Reforming Combined Cycle (IRCC)	331	
interest		
construction time and	27	
production costs impacted by	12	

<u>Index Terms</u>	<u>Links</u>
interest rate	13
electricity costs impacted by	31
intergovernmental Panel of Climate Change (IPCC)	321
intermediate load operation	13
electricity costs, power plant comparisons for	30
intermediate-pressure (IP)	57
internal rate of return (IRR)	11
investment costs	
elements impacting	14
of GT air inlet cooling	253
of IGCC	289
of IGCC concepts	313 319
PFBC, complexities with	160
of Shell gasification	317
investors	
combined-cycle plants' popularity with	24
targets of	11
ion transport membranes	328
Italy. <i>See</i> Vado Ligure power station	
J	
Joule and Clausius-Ranking cycle (combined cycle)	323
L	
Law of Cones	400
leakages	
deaeration and	98

Index Terms

Links

leakages (*Cont.*)

environment impacted by 164

legislation 261

combined-cycle plant selection and 57

emissions limited in 57

NO_x emissions regulated by 270

of Switzerland on emissions 387

light distillate fuel 175

characteristics of 180

lignite 311

liquefied natural gas (LNG) 19

characteristics of 178

liquefied petroleum gas (LPG) 177 179

liquid fuels 180

live-steam pressure

closed control loops and 221

condenser impacted by 80

dual-pressure cycle and 100

economic advantages of higher 80

moisture content and 79

part-load behavior and 236

steam turbines, effect of 77 199

triple-pressure cycle and 110

triple-pressure reheat cycle and 120

live-steam temperature

closed control loop and 220

dual-pressure cycle and 102

part-load behavior and 236

steam turbines, impact of increasing 81

triple-pressure cycle and 111

This page has been reformatted by Knovel to provide easier navigation.

Index Terms

Links

live-steam temperature (<i>Cont.</i>)		
triple-pressure reheat cycle and	122	
load control		
closed control loops and	216	
combined-cycle plant, droop		
characteristic of	217	
frequency control compared to	217	
supplementary firing and	217	
load factor		
measuring	16	
purpose of	14	
load jump	225	
gas turbines and	218	
LO-CAT	300	
long-term sales contracts (LTSCs)	11	
low boiling liquid fuels	182	
low sulfur fuels	95	
lower heating value (LHV)		
of combined-cycle plants	3	
determining	20	
fuel, importance of	64	233
low-NO _x burner. <i>See also</i> Siemens dry		
low-NO _x burner		
characteristics of	268	
cross section of	269	
low-pressure (LP)	57	
dual-pressure cycle, pinch point in		
evaporator with	104	
HRSG efficiency impacted by	101	
pinch point, HP's relation to	105	

Index Terms

Links

low-pressure (LP) (<i>Cont.</i>)		
steam turbine power output impacted by		
HP and	100	
triple-pressure cycle, pinch point in		
evaporator with	114	
low-temperature corrosion	90	190
LT sweet shift	300	

M

machine level	212	
main-steam parameters		
economics of increasing	282	
geared high-pressure turbine and	282	
historical improvement of	281	
maintenance	285	
<i>See also</i> operation and maintenance costs		
makeup water	100	
market price of electricity	11	
mechanical draft cooling tower	205	
mechanical losses	400	
membranes	327	
<i>See also</i> ion transport membranes		
methanol	182	
Mexico. <i>See</i> Monterrey combined-cycle plant		
MHI dry-feed gasifier	294	
moisture content	78	
live-steam pressure impacting	79	
reheat and	115	
triple-pressure cycle compared to dual-		
pressure cycle for	113	

This page has been reformatted by Knovel to provide easier navigation.

Index Terms

Links

moisture content (<i>Cont.</i>)	
triple-pressure reheat cycle decreasing	118
Monterrey combined-cycle plant	
CFE ordering of	353
composition of	354
efficiency of	354
layout of	356
performance data for	355
process diagram for	355
Monthel cogeneration plant	
arrangement of	386
efficiency of	385
process diagram for	386
technical data of	387
multishaft combined-cycle plants	
Arcos III combined-cycle plant	
configuration as	366
configuration for	197
Phu My 3 combined-cycle plant as	358
multistage flash desalination units (MSF)	373

N

natural draft cooling tower	205
natural gas	175
See also liquefied natural gas availability of	9
global warming mitigation with	324
pressure fluctuations with	177
proved reserves v. yearly consumption of	10
natural gas premix burner	333
near field noise	275

This page has been reformatted by Knovel to provide easier navigation.

Index Terms

Links

net efficiency	
defining	20
power plant comparisons on	20
net present value (NPV)	282
Netherlands. <i>See</i> Diemen combined-cycle cogeneration plant	
New Zealand. <i>See</i> Taranaki combined- cycle plant	
nitrous oxide (NO _x) emissions. <i>See also</i> low-NO _x burner	
ambient air temperature and	262
combustion with excess air reducing	267
fuel-to-air ratio and	263
gas turbines, flame temperature and	263
global warming and	262
legislation regulating	270
reducing	57
SCR disadvantages for reducing	271
staged combustion reducing	267
syngas dilution controlling	305
water injection/steam injection and	265
noise emissions	261
types of	275
nuclear power plant. <i>See</i> power plants	
nuclear waste	261

O

off-design behavior	
combined-cycle plants and	225
combined-cycle plants, corrections for	228

This page has been reformatted by Knovel to provide easier navigation.

Index Terms

Links

off-design behavior (<i>Cont.</i>)	
fuel and	233
off-line washing	173
oil	
availability of	9
proved reserves v. yearly consumption of	10
oil-soluble contaminants	181
once-through design	188
evaporation in	187
HRSG with	186
once-through water cooling	204
configuration/role of	206
on-line washing	173
open cycle gas turbine	172
advantages of	3
open-circuit air cooling generators	201
operating behavior	
ambient air pressure and	229
ambient air temperature and	229
calculating	226
cooling water temperature and	230
design point solution for calculating	227
electrical corrections for	231
frequency impacting	231
parameters for correcting	228
power factor and	232
process energy and	233
relative humidity and	229

Index Terms

Links

operation and maintenance (O&M) costs	15	29
power plants, fixed costs and	25	
power plants, variable costs and	24	
operational flexibility		
economics and	284	
fast cycling provisions for	285	
original equipment manufacturers (OEMs)	285	
overpressure deaeration	99	
Oxyfuel	327	
AZEP process for	341	
CES cycle concept for	340	
direct/indirect systems in	339	
efficiency of	344	
Graz cycle concept for	339	
processes of	338	
ZESOFc concept for	342	
oxygen generation		
chemical looping and	328	
cryogenic ASU and	327	
ion transport membranes and	328	
oxygen-lean combustion	267	
oxygen-rich combustion	267	

P

Palos de la Frontera combined-cycle plant	
efficiency of	361
HRSGs in	363
layout of	362
location of	361
technical data of	364

Index Terms

Links

parallel-fired combined-cycle plant	
operating modes for	157
reheat in	156
repowering and	155
steam turbine converted to	157
partial oxidation (POX) gasification	330
part-load behavior	
calculating	226
efficiency improvements for	237
gas turbine efficiency with	235
live-steam pressure/live-steam	
temperature with	236
reheat and	236
single-shaft combined-cycle plants and	237
peak load operation	13
performance guarantees	
comparison for	241
heat balance computer model for	240
measuring	238
power output correction factors for	239
performance monitoring	244
performance test	241
performance value	
degradation impacting	241
monitoring	244
petcoke	289
Phu My 3 combined-cycle plant	
as BOT project	357
location of	358
as multishaft combined-cycle plant	358

Index Terms

Links

Phu My 3 combined-cycle plant (<i>Cont.</i>)		
Siemens combined-cycle and		
arrangement of	359	
technical data of	360	
pinch point		
in HP/LP evaporator, dual-pressure cycle	104	
in HP/LP evaporator, triple-pressure		
cycle	114	
HP/LP interrelation in	105	
in HRSG	190	
HRSG heating surface determined by	75	83
steam turbine power output, impact of	83	
piston losses	399	
plant concept solution	45	132
gas turbine selection for	66	
plume abatement	206	
population growth	321	
postcombustion CO ₂ capture		
methods for	335	
SARGAS cycle for	337	
sorption and	335	
sorption systems for	337	
power augmentation		
applications for	245	
control system for	223	
goal of	244	
high-fogging system increasing	248	

Index Terms

Links

power coefficient	
determining	139
parameters impacting	141
process steam pressure impacting	
combined-cycle plants and	141
single-pressure cycle, fuel utilization	
impacted by	142
supplementary firing impacting	143
power factor	232
power island	67
benefits of	47
power output	
ambient air pressure, site elevation	
impacting	54
ambient air temperature impacting	53
chiller improving combined-cycle plant	
and	251
cycle performance summary for	130
evaporative cooling system in combined-	
cycle plant increasing	246
frequency impacting combined-cycle	
plants and	232
gas turbines, data for	171
of heavy-duty industrial gas turbines	168
HP, LP impacting steam turbines and	100
performance guarantees, correction	
factors for	239
pinch point impacting steam turbines and	83
process steam pressure impacting	
combined-cycle plants and	141

Index Terms

Links

power output (*Cont.*)

relative humidity and	56	
steam injection, water injection		
impacting	58	
of steam turbines	72	399
temperature, after supplementary firing,		
impacting	128	130

power plants. *See also* combined-cycle

plants; district-heating power plants; gas
turbines; industrial power stations; seawater
desalination plants; steam power plant;
steam turbines

availability of	25	
base load operation, electricity cost		
comparisons for	29	
CO ₂ emissions impacting economics of	33	
construction time comparisons for	27	
credit and	49	
electricity value impacting choice of	13	
fuel flexibility of	22	
fuel selection considerations for	21	
installed capacity evolution of	9	
intermediate load operation, electricity		
cost comparisons for	30	
net efficiency comparisons for	20	
O&M costs, fixed costs for	25	
O&M costs, variable costs for	24	
present value determined for	15	
reliability of	25	
risks for	12	

Index Terms

Links

power plants (<i>Cont.</i>)		
as trading tool	225	
waste heat dissipation in various	274	
precombustion CO ₂ capture	329	
ATR and	332	
CO ₂ separation and	331	
hydrogen behavior in	333	
syngases and	332	
preheating. <i>See also</i> feedwater preheating		
fuel and	64	
increasing stages of	94	
present value	15	
pressure. <i>See also</i> ambient air pressure;		
condenser pressure; dual-pressure cycle;		
high-pressure; intermediate-pressure;		
live-steam pressure; low-pressure; process		
steam pressure; single-pressure cycle;		
triple-pressure cycle		
compressor increasing	65	
HRSG exhaust gas, loss in	84	
natural gas, fluctuations in	177	
steam extraction, district-heating power		
plants, stages of	145	
superheater, water/steam cycle drop in	84	
pressure ratio	51	165
gas turbine efficiency impacted by	41	65
pressurized fluidized-bed combustion (PFBC)		
combined-cycle plant applications with	159	
design/investment cost complexities for	160	

Index Terms

Links

process energy	
control system for	223
operating behavior and	233
process steam pressure	141
production costs	
amortization, interest impacting	12
electric power market and	6
of electricity	14
propane	179
Puertollano IGCC plant	
ASU in	383
availability issues of	384
design of	382
technical data of	384
pulverized coal (PC) plants	288
purge time	254

R

radioactivity	261
recirculation	
feedwater preheating with	90
loop for	124
Rectisol	302
reheat. <i>See also</i> dual-pressure reheat cycle;	
triple-pressure reheat cycle	
combined-cycle efficiency impacted by	133
gas turbines impacted by	114
moisture content and	115
in parallel-fired combined cycle plant	156
part-load behavior and	236

This page has been reformatted by Knovel to provide easier navigation.

Index Terms

Links

reheat (<i>Cont.</i>)		
steam turbine expansion line impacted by	113	118
relative humidity		
operating behavior and	229	
power output and	56	
reliability		
deregulation and	26	
determining	26	
of power plants	25	
repowering		
efficiency increased in	151	
with fully fired combined-cycle plant	154	
gas turbines, conventional boiler		
adaptation for	154	
in parallel-fired combined-cycle plant	155	
steam turbine before/after	151	
steam turbine, problems with	153	
steam turbines, equipment reuse in	153	
Vado Liguire power station, thermal		
process after	379	
resources	60	
<i>See also</i> cooling media; fuel		
return on equity (ROE)	11	
risks		
electric power market and	6	
IPPs mitigating	11	
power plants and	12	

Index Terms

Links

S

SARGAS cycle	337
seawater desalination plants	135
combined-cycle plants coupled with	149
flashing in	149
selective catalytic reduction (SCR)	271
SELEXOL	298
IGCC and	302
separator	123
sequential combustion	
gas turbine's advantages with	44
gas turbines with	170
shell coal gasification process (SCGP)	292
Shell gasification	
ASU for	310
diagram of	311
efficiency/investment costs of	317
IGCC concept based on CO ₂ capture,	
hard coal, Siemens combined-cycle	
and	316
IGCC concept based on Siemens	
combined-cycle and	308
shutdown	258
Shuweit S1 Independent Water and Power	
Plant (IWPP)	
as desalination plant	373
functioning of	372
location of	371
technical data of	374

Index Terms

Links

Siemens combined-cycle			
IGCC concept based on CO ₂ capture,			
hard coal, SFG and	314		
IGCC concept based on CO ₂ capture,			
hard coal, Shell gasification and	316		
IGCC concept based on lignite, HTW			
process and	311		
IGCC concept based on Shell gasification			
and	308		
Phu My 3 combined-cycle plant			
arrangement and	359		
Siemens dry low-NO _x burner	270		
Siemens fuel gasifier (SFG)			
gasification with	293		
IGCC concept based on CO ₂ capture,			
hard coal, Siemens combined-cycle			
and	314		
Siemens SGT5-8000H gas turbine	284		
single-line diagram	202		
single-pressure cycle			
design parameters of	77		
energy flow diagram for	77		
energy/temperature diagram for HRSG in	75	79	88
example of	76		
exhaust temperature, dual-pressure cycle v.	69		
flow diagram for	73		
gas turbines in	171		
heat balance for	76		
heat balance for cogeneration with			
supplementary firing in	140		

Index Terms

Links

single-pressure cycle (<i>Cont.</i>)	
heat balance for supplementary firing in	129
HRSG in	74
power coefficient impacting fuel	
utilization in	142
steam turbines, dual-pressure cycle v.	96
single-shaft combined-cycle plants	
clutch and	198
configuration for	197
cross section of	199
part-load behavior and	237
single-line diagram for	202
steam turbines in	198
SINTEF	344
site conditions	45
site elevation	54
site-related factors	45
<i>See also</i> ambient air	
pressure; ambient air temperature; ambient	
conditions; relative humidity	
location and	49
sliding pressure	226
solid oxide fuel cell (SOFC)	342
sorption	
CO ₂ separation and	326
postcombustion CO ₂ capture and	335
postcombustion CO ₂ capture, systems for	337
SO _x emissions	272

Index Terms

Links

Spain. <i>See</i> Arcos III combined-cycle plant;		
Palos de la Frontera combined-cycle plant;		
Puertollano IGCC plant		
Special purpose company (SPC)	13	
stack loses	109	
staged combustion	267	
standardization	132	361
customization compared to	48	
standstill		
startup 8 hours	256	
startup 48 hours	257	
startup 120 hours	257	
startup and	254	
startup. <i>See also</i> ignition		
8 hours standstill, curve for	256	
48 hours standstill, curve for	257	
120 hours standstill, curve for	257	
combined-cycle plants and	196	
combined-cycle plants' range of times for	253	
gas turbines, quickness of	253	
HRSG, challenges with	190	
procedure for	254	
standstill impacting	254	
for steam turbines	255	
times/values for	255	
steam bypass	221	
steam cooling	280	

Index Terms

Links

steam extraction		
cycle with	138	
district-heating power plants, pressure		
stages for	145	
industrial power stations and	136	
steam injection		
combined-cycle plant, impact of	58	
efficiency and	57	
NO _x emissions and	265	
power output impacted by	58	
steam parameters. <i>See</i> main-steam parameters		
steam power plant. <i>See also</i> power plants		
feedwater preheating in	86	
price comparisons for	18	
steam process efficiency		
ambient air temperature impacting	52	
combined-cycle plant, equation for	39	
gas turbine efficiency impacting	41	
steam reforming	330	
steam turbine expansion line		
reheat impacting	113	118
in triple-pressure reheat cycle	122	
steam turbines. <i>See also</i> back-pressure turbine;		
extraction/condensing steam turbines;		
water/steam cycle		
ambient conditions and	50	
bypass stack and	207	
Carnot efficiency of	36	
characteristics of	196	
closed control loops operation for	222	

Index Terms

Links

steam turbines (*Cont.*)

cogeneration with no	139	
combined-cycle plants v. conventional	195	
efficiency equations for	399	
energetic losses v. exergetic losses in	78	
exhaust losses for	400	
feedwater preheating impacting	94	
generator losses in	400	
hotwell in	99	
HP, LP impacting power output of	100	
installed capacity of	7	
live-steam pressure impacting	77	199
live-steam temperature impacting	81	
mechanical losses for	400	
parallel-fired combined-cycle plant		
converted from	157	
pinch point impacting power output of	83	
piston losses and	399	
power output of	72	399
repowering, before/after for	151	
repowering, equipment reuse in	153	
repowering problems in	153	
single-pressure cycle v. dual-pressure cycle		
in	96	
in single-shaft combined-cycle plants	198	
startup for	255	
swallowing capacity of	398	

Index Terms

Links

steam-injected gas turbine (STIG) cycles.

See also turbo STIG cycle

economics of 161

energy flow diagram for 161

stoichiometric combustion 166

flame temperature in 264

stress controller. *See* turbine stress controller

Sulferox 300

Sulfur 64 159

See also high sulfur fuels; low

sulfur fuels; SO_x emissions

recovery 299

superheater

role of 74

simplicity v. 103

water/steam cycle pressure drop in 84

supplementary firing

advantages of 125

cogeneration, HRSG with 136

control system for 223

energy/temperature diagram for HRSG

and 126

heat balance for cogeneration, single-

pressure cycle with 140

heat balance for single-pressure cycle

with 129

HRSG and 125

HRSG construction without 183

HRSG with limited 192

HRSG with maximum 194

Index Terms

Links

supplementary firing (<i>Cont.</i>)		
load control and	217	
power coefficient impacted by	143	
temperature impacting power output,		
efficiency after	128	130
swallowing capacity	398	
sweep gas	341	
Switzerland	387	
<i>See also</i> Monthel cogeneration plant		
symbols	405	
syngas conditioning system	304	
syngases	176	
advanced cleaning technology for	300	
characteristics of	179	
cleaning techniques for	295	
dedusting and	296	
gas turbine modification for utilizing	305	
gasification process with	288	
NO _x emissions, dilution of	305	
precombustion CO ₂ capture and	332	
water gas shift reaction and	301	
system of equations	400	

T

Taranaki combined-cycle plant	
arrangement of	352
control system for	353
design of	350
process diagram for	351
technical data of	352

Index Terms

Links

temperature. <i>See also</i> ambient air temperature;		
approach temperature; cooling water		
temperature; energy/temperature diagram;		
exhaust temperature; feedwater temperature;		
flame temperature; high temperature winkler		
(HTW) process; live-steam temperature;		
low-temperature corrosion; pinch point		
combined-cycle plants, diagrams on		
entropy and	37	
condenser pressure v. cooling media and	62	
cooling media impacted by	61	
economizer, changes in	88	
HRSG and controlling	220	
power output/efficiency, after		
supplementary firing impacted by	128	130
terawatt hour (TWh)	8	
thermal barrier coating (TBC)	279	
thermal cycles	1	
consumption breakdown of	8	
popularity of	8	
thermal energy	135	
thermal energy storage (TES)	251	
thermal process	379	
thermodynamics	35	389
Tolling Agreement	12	
top gas	179	
topping cycle	2	
trading tool	225	

Index Terms

Links

triple-pressure cycle

adding	105	
energy flow diagram for	105	109
energy/temperature diagram for HRSG in	108	
heat balance for	106	
live-steam pressure in	110	
live-steam temperature in	111	
moisture content, dual-pressure cycle		
compared to	113	
pinch point of HP/LP evaporator in	114	

triple-pressure reheat cycle

energy flow diagram for	116	119
energy/temperature diagram for HRSG in	119	
exergy transfer in	117	
heat balance for	116	
live-steam pressure in	120	
live-steam temperature in	122	
moisture content decreased in	118	
performance benefit in	123	
steam turbine expansion line in	122	

tubing. *See* finned tubing

turbine inlet temperature (TIT) 67.

See also GT air inlet cooling

chronology of cooling media improving	279
combined-cycle plant efficiency impacted by	41
control system and	215
definitions	38
economics of increasing	278
gas turbine development increasing	165
gas turbine historical trends in	167

Index Terms

Links

turbine inlet temperature (TIT) (*Cont.*)

importance of 44

turbine mass flow 306

turbine stress controller (TSC) 254

turbo STIG cycle 162

U

unconverted carbon (char) 296

Union Fenosa Generacion (UFG) 361

unit level 212

United Arab Emirates. *See* Shuweit S1

Independent Water and Power Plant

V

vacuum deaeration 99

Vado Ligure power station

design of 378

emissions in new 381

main equipment supply for 379

new layout for 376

pre-existing situation of 375

repowering, thermal process of 379

technical data of 378

vaporized liquids 179

variable costs 17

power plants, O&M costs and 24

variable inlet guide vane (VIGV) 215

Vietnam. *See* Phu My 3 combined-cycle plant

Index Terms

Links

W

waste heat	261
combined-cycle plants reducing	273
environment impacted by form of	274
power plants' dissipation of	274
waste water	261
water. <i>See also</i> cooling water temperature;	
makeup water; waste water	
bottoming cycle, ammonia mixing with	164
for cooling media	60
water gas shift reaction	301
water injection	57
combined-cycle plant, impact of	58
efficiency problems with	266
NO _x emissions and	265
water-cooled generators	201
water-soluble contaminants	181
water/steam cycle	
bottoming cycle and	2
combined-cycle plant design challenges	
with	70
concepts for	72
evaluating	131
exhaust gas' heat transfer v.	91
superheater, pressure drop in	84
wet cell cooling tower	63
Wobbe index	179
gaseous fuel comparison with	177

Index Terms

Links

Z

Zero Emission Integrated Gasification	
Combined Cycle (ZEIGCC)	331
zero emission solid oxide fuel cell (ZESOFc) concept	342