

Power cycles

Principles of combustion cycles and efficient concepts

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Reference Document on
Best Available Techniques for
Large Combustion Plants
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revised and updated with new developments and proven industrial applications.

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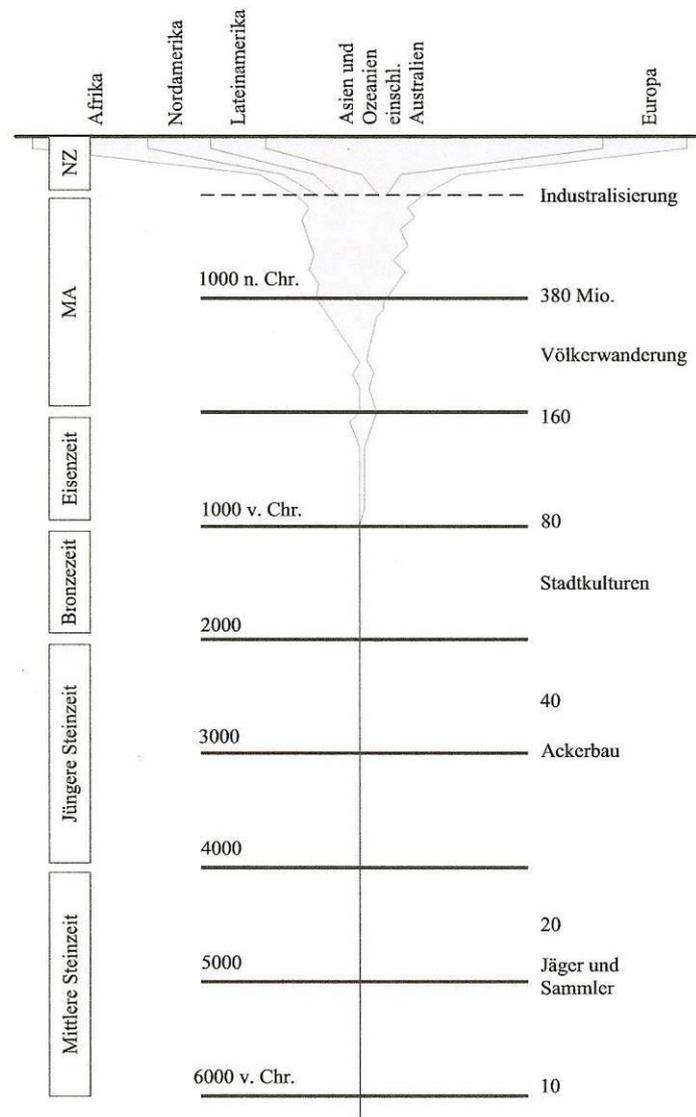
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1 Introduction

To assist better understanding of the rules that govern the transformation and use of energy, this document gives a short explanation of the principles and concepts of combustion cycles, as this is the theory on which power plant technology and large combustion installation design and techniques are based. For more detailed information on the theory of thermodynamic cycles of fossil fuel combustion and energy generation, reference can be made to a number of documents as mentioned in Annex, which also have been used as background material for this document.

However, before trying to understand how things work, and to think about how we could do things better, it might be appropriate to put the issue in the right context, so that we can get a feeling for the level on which all this is happening.



Entwicklung der Weltbevölkerung [IZE 1997]

Figure 1: Development of the world population

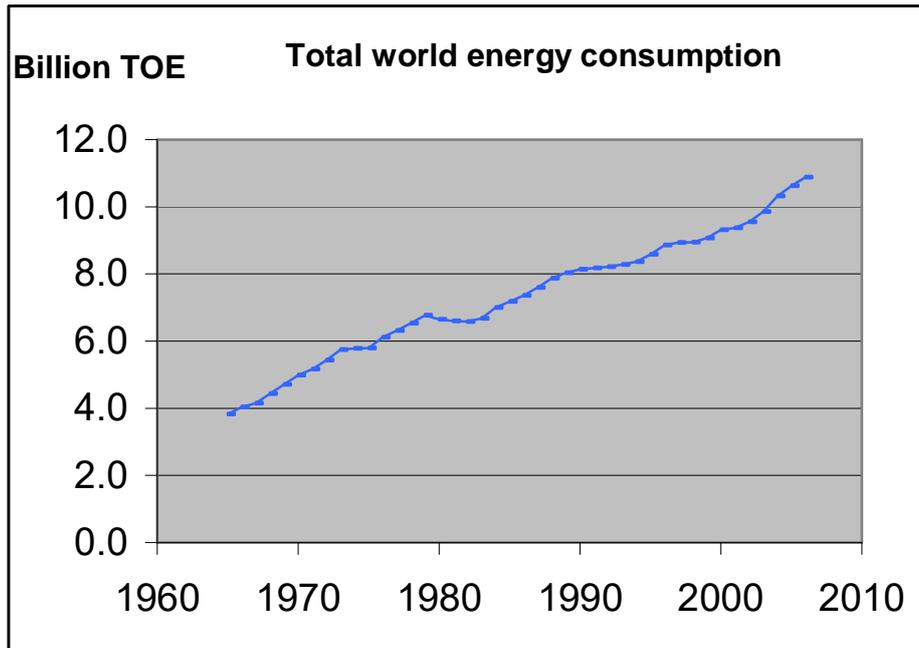


Figure 2 World energy consumption 1965 - 2005

Figure 1 [11] gives an overview of the development of this world's population.

Figure 2 shows the development of the world energy consumption in the last 40 years. In 1914, this consumption was only 0.6 Billion TOE.

It is obvious from these pictures already that things will never be the same as they were before. It should also be understood that, in nature, no development is known as being capable of growing at a linear rate forever, and certainly not at an exponential rate as we have seen in the past centuries. Every system in nature develops until it reaches a maximum growth rate, then this rate slows down, comes to a standstill, and turns into a negative figure until the system dies and becomes history.

All civilisations have been fascinated by the power of nature and have always been wondering about the rules governing them. And since the middle ages, scientists have monitored physical processes and have tried, more or less successfully, to set up models to describe the phenomena and to catch them in mathematic formulas, known as "the laws of thermodynamics". These will be explored further on, however, we should be aware of some very simple rules and principles to start with, as summarised by C.P. Snow:

1st Law: Energy can neither be created nor destroyed.

This means that we can never get more out of a system than what we put into it.

Hence : **"You can't win!"**

2nd Law: Heat flows from a hot object to a colder object and a given amount of heat cannot be changed complete into energy to do work.

Even though you put a certain amount of energy into a system you can't actually get that energy back as work.

Hence: **"You can't break even!"**

3rd Law: The energy of a system tends towards a constant as absolute zero is approached. (Entropy approaches a constant). Or stated another way: you can't reach absolute zero or, you can't remove all the energy from a system.

Hence: **"You can't get out of the game!"**

A more scientific description of the laws of thermodynamics will follow. They serve as a tool to guide ideas, concepts and machinery that are related to the transformation of one form of energy into another, the most attractive being transforming heat into work.

The first know idea in this field is two thousand years old and is known as the aeolipile, built in the 1st century by Heron of Alexandria, a Greek, living in Alexandria, Egypt in the years 10 to 70 AD. Then it took until 1680 when Denis Papin built a steam driven device.



Figure 3 The aeolipile of Heron of Alexandria

Heron's invention was not very successful. One can only guess why, but there might be a number of reasons that could explain the lack of interest. First, qualified craftsmanship was required to build the engine, but a sword was more useful for defending oneself at that time ó a matter of priorities. Second, a lot of wood was needed to keep the engine running whilst very little came out ó a matter of efficiency. Third, what came out was hot steam whilst people in those regions might have preferred a cold breeze ó a matter of basic needs.

These three elements: priorities, efficiency and basic needs are the key factors that still are dominating our present life and the way we are capable of dealing with them will be determining for our future.

And, unfortunately, today we are not doing very well in this respect. Figure 4 illustrates how we have been dealing with one of the scarcest resources of this earth: resources for energy. A particular feature of Figure 4 is that, further to being a well known Sankey diagram for the transformation of primary energy into practical forms for end-use, it also shows the losses involved for some applications. More complete diagram can be found in [15].

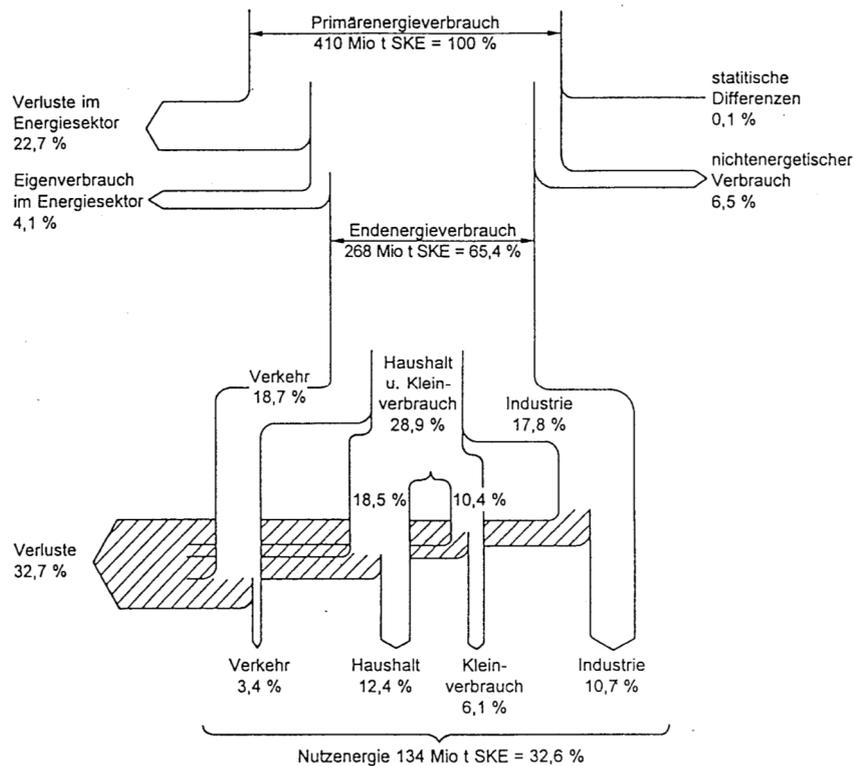


Figure 4: System efficiency and losses ó an example from Germany

First, the efficiency in generating electricity is (only) about 50%. The other 50% of the raw material is already lost before it reaches the end customer, enough reason to think very hard about how this process can be further improved and how renewable sources could be used. But we should not forget the second law of thermodynamics that implies that transformations of energy are not reversible which means, in simple words, that it might be easy to transform a noble form of energy into a less noble form, but not vice-versa. Or, all work can be transformed into heat, but not all heat can be transformed into work. We should think about how primary energy can be transformed into work (or electricity), without passing through the intermediate, less noble, form of heat. But we should also think very hard about how the efficiency of the appliances themselves could be improved so that they need less work or electricity, or about alternatives that would not require this very noble form of energy.

Second, the efficiency in transportation is less than 18%. This means that 82% of the transportation fuel (or 85% of the original primary energy) is not efficiently transformed into mechanical energy for moving and is lost. The efficiency for generating the transportation fuel is very high, but the way we use it is alarmingly low. There also, we will have to think very hard about how to increase the efficiency of the system and define a priority scale of what really is required to satisfy our needs.

A strategy for the future cannot disregard the three fundamental questions asked before:

- 1) What are the priorities in view of a sustainable future;
- 2) What are our basic needs, differentiated from what we all would like but not necessarily can afford;
- 3) How can we satisfy these needs in the most efficient way.

This document intends to give an overview of mature techniques developed in the past decades to meet the requirements of increased efficiency to transform primary energy into the most flexible forms such as mechanical energy (or work) and electricity, the rules governing these techniques and their industrial applications.

2 Thermodynamic principles

2.1 First law of thermodynamics

Thermodynamics is a branch of physics in which the concept of energy related concepts and their properties are studied. As is usually the case in physics, thermodynamics is based on induction: the laws, which describe the behaviour of energy, have been formulated on the basis of observations of reality. These are called the laws of thermodynamics.

According to the first law, energy is always conserved; it can be neither created nor destroyed. The energy of a system undergoing change (process) can be increased or decreased by exchange with the surroundings and can convert from one to another within that system. This is, therefore, simply a bookkeeping law that declares the exchange and convertibility of energy and sees to it that all energy is accounted for when a change occurs. The first law does not indicate whether conversions of energy from one form to another are or are not performed perfectly or whether some forms may be completely converted to others. Such limitations are left to the second law.

2.2 Second law of thermodynamics

As pointed out in Section 2.1, energy is a conserved property, and no process is known to have taken place in violation of the first law of thermodynamics. Therefore, it is reasonable to conclude that a process must satisfy the first law to occur. However, as explained below, satisfying the first law alone does not ensure that a thermodynamic process will actually take place.

Processes proceed in a certain direction and not in the reverse direction. The first law places no restriction on the direction of a process, but satisfying the first law does not ensure that the process will actually occur. This inadequacy of the first law to identify whether a process can take place is remedied by introducing another general principle, the second law of Thermodynamics.

The use of the second law of thermodynamics is not limited to identifying the direction of processes, however. The second law also ensures that energy has not only quantity but also quality. The first law is concerned with the quantity of energy and the transformations of energy from one form to another with no regard to its quality. Preserving the quality of energy is a major concern to engineers, and the second law provides the necessary means to determine the quality as well as the degree of degradation of energy during a process. The above statements serve as a starting point for the following formulation of the second law:

- energy consists of two components, one part is totally convertible into other forms of energy (called exergy), and one part is not at all convertible (called anergy)
- in any real thermodynamic process, part of the exergy is irreversibly converted into anergy.

2.3 Enthalpy and entropy

Enthalpy represents the amount of energy possessed by a thermodynamic system for transfer between itself and its environment. For example, in a chemical reaction, the change of enthalpy of the system is the heat of the reaction. In a phase change, such as from a liquid to a gas as boiling water in a steam generator, the enthalpy of the system is the heat of vaporisation. In a simple temperature change, the change of enthalpy with each degree is the capacity of the system at constant pressure. Mathematically, enthalpy H is identified as $U + PV$, where U is internal energy, P is pressure and V is volume.

Rudolph Clausius first introduced the term 'entropy' in 1865. He had noticed that a certain ratio of energy was constant in reversible, or ideal, heat cycles, where a cycle in general is a series of processes that begins and ends at the same state and thus can repeat indefinitely, or as long as needed. The ratio was heat exchanged to absolute temperature. Clausius decided that the conserved ratio must correspond to a real, physical quantity, and he named it 'entropy'

For a closed system, entropy can be defined as a quantitative measure of the amount of energy not available to do work. So it is a negative kind of quantity, the opposite of available energy, or in other words a measure of disorder or randomness.

2.4 The concept of reversibility

A reversible process is defined as a process that can be reversed without leaving any trace on the surroundings. That is, both the system and the surroundings are returned to their initial states at the end of the reverse process. This is possible only if the net heat exchange and net work exchange between the system and the surroundings are zero for the combined (original and reverse) processes. Processes that are not reversible are called irreversible processes.

It should be pointed out that a system can be restored to its initial state following a thermodynamic process, regardless of whether the process is reversible or irreversible. But for reversible processes, this restoration is made without leaving any net change on the surroundings, whereas for irreversible processes, the surroundings usually do some work on the system and, therefore, will not return to their original state.

Reversible processes do not actually occur in nature. They are merely idealisations of actual processes. Reversible processes can be viewed as theoretical limits for the corresponding irreversible ones. Some thermodynamic processes are more irreversible than others. It is not possible to have a reversible process, but it is possible to approach to it. The more closely a reversible process is approximated, the more work delivered by a work-producing device or the less work required by a work-consuming device.

The concept of reversible processes leads to the definition of second-law of thermodynamic, the efficiency of a thermodynamic related to the degree of approximation to the corresponding reversible processes. This enables us to comparison of the performance of different devices that are designed to do the same task on the basis of their efficiencies. The better the design, the lower the irreversibility and the higher the efficiency.

The factors that cause a process to be irreversible are called irreversibilities. They include friction, unrestrained expansion, mixing of two gases, heat transfer across a finite temperature difference, electric resistance, inelastic deformation of solids, and chemical reactions. The presence of any of these effects renders a process irreversible. A reversible process involves none of these.

2.5 The ideal cycle (Carnot cycle)

One ideal cycle in which the heat is taken in at a constant upper temperature (T_a) and rejected at a constant lower temperature (T_b) is that suggested by Sadi Carnot. It consists of two reversible adiabatic (no heat is exchanged with the surrounding environment) (isentropic- no change in entropy) processes. When the working fluid is considered as water, or water vapour, the two isothermal (no change in temperature) processes are easily obtained by heating and cooling at constant pressure and temperature while the fluid is a wet vapour. The cycle is represented on the temperature-entropy ($T-s$) diagram for steam in Figure 5 conceptual sketch of the process.

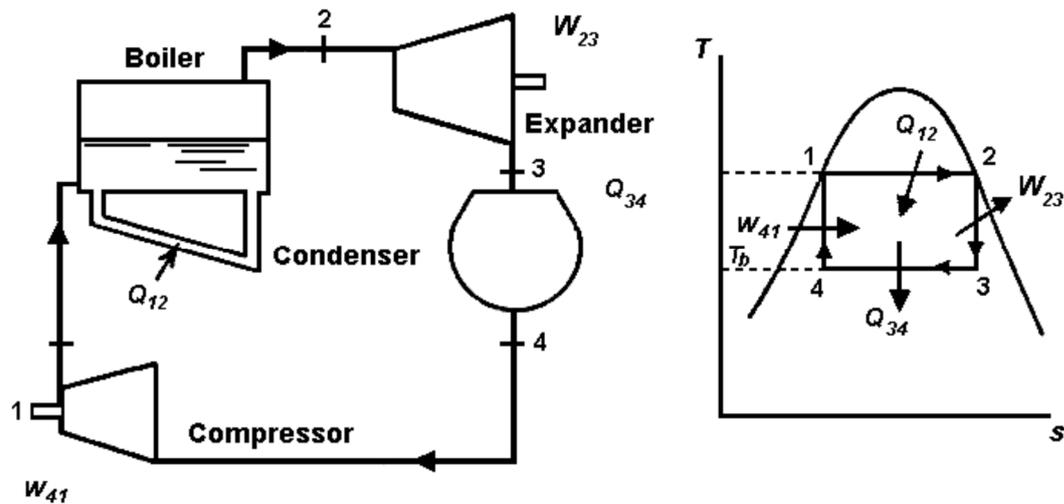


Figure 5: Ideal Carnot cycle

In the Carnot cycle, saturated water in state 1 (1) is evaporated in a boiler at constant pressure to form saturated steam in state 2 (2) with the addition of heat (Q_{12}). The steam is then expanded isentropically to state 3 (3) while doing work in a turbine or reciprocating engine (W_{23}). After expansion, the steam is then partially condensed at constant pressure while heat is rejected. Condensation is stopped at state 4 (4) where $s_4 = s_1$. Finally the steam is compressed isentropically in the rotary or reciprocating compressor to state 1 (1). The Carnot cycle is the most efficient cycle that can be executed between a thermal energy source at temperature T_a and temperature at sink T_b . Its thermal efficiency is expressed as:

$$\eta_{th \text{ Carnot}} = 1 - T_b/T_a$$

Reversible isothermal heat transfer is very difficult to achieve in reality because it would require very large heat exchangers and it would take a very long time. Therefore, it is not practical to build an engine that would operate on a cycle that closely approximates the Carnot cycle.

The real value of the Carnot cycle comes from its being a standard against which the actual or other ideal cycles can be compared. The thermal efficiency of the Carnot cycle is a function of the temperatures sink and source, and the thermal efficiency relation for the Carnot cycle conveys an important message that is applicable to all cycles.

Thermal efficiency increases with an increase in the average temperature at which heat is added to the system or with a decrease in the average temperature at which heat is rejected from the system.

The overall thermal efficiency of a power plant is suitably measured by the proportion of latent energy in the fuel that is converted into useful mechanical work. The overall efficiency can be expressed as the product of two efficiencies:

- a) the combustion efficiency, which expresses the proportion of latent energy in the fuel, transferred as heat to the working fluid
- b) the cycle efficiency, which expresses the proportion of this, heat which is converted into mechanical work.

2.6 Properties of vapour (water vapour)

An important step in design of steam generating equipment and its operational cycle application, either for power or heat transfer or both, is to establish reliable information on steam properties.

Steam tables currently in use are the 1967 ASME Steam Tables based on agreements reached at the Sixth International Conference on the Properties of Steam

The steam tables include the corresponding pressure-temperature equilibrium values between the liquid and vapour phases referred to as "saturation". Steam heated beyond saturation to higher temperatures is "superheated steam". Water heated to temperatures below saturation (subcooled water) is referred to as "compressed water" in the steam tables. Properties for the superheated steam and compressed water ranges are given in the steam tables, which also includes the intensive properties of specific volume, specific enthalpy, and specific entropy for specified temperatures, pressures and states (liquid or gas). Intensive properties are those which are independent of mass; they are also independent of the type of process or any past history.

These are the state and thermodynamic properties required for numerical solutions to design and performance problems involving steam for heat transfer and power. The special significance of the important properties of enthalpy is a consequence of the laws of thermodynamics.

Customarily, the boiler industry uses 300 K and 10133 Pa (or 1.013 bar) as the zero enthalpy of air and combustion products, although this practice is not widespread in other related engineering fields.

A more general reference is one atmosphere pressure (101325 Pa) and 25 °C (298.15 K). This is referred to as the standard reference point for listing the heats of formation of compounds from their elements in their standard states, latent heats of phase changes, free energy, and other important thermodynamic quantities.

Less commonly, a reference temperature of 0°C (273.15 K) is used.

From the above, it is obvious that it is important to mention the reference state used when referring to thermodynamic properties.

3 The Rankine cycle, the standard cycle for steam power plants

There are two major reasons why the Carnot cycle is not used in practice. Firstly, because it has a low work ratio. Secondly, because of practical difficulties associated with compression. It would be difficult to control the condensation process so that it stopped at stage 4 (4) (Figure 1), and then carry out the compression of a very wet vapour efficiently. The liquid tends to separate out from the vapour and the compressor would have to deal with a non-homogeneous mixture. Moreover, the volume of the fluid is high and the compressor would be comparable in size and cost with the turbine. It is comparatively easy, on the other hand, to condense the vapour completely, and compress the liquid to boiler pressure in a small feed pump.

Another impracticality associated with the Carnot cycle can be eliminated by superheating the steam in the boiler, as shown schematically on a T-s diagram in Figure 6. The cycle that results is the Rankine cycle, which is the ideal cycle for vapour power plants.

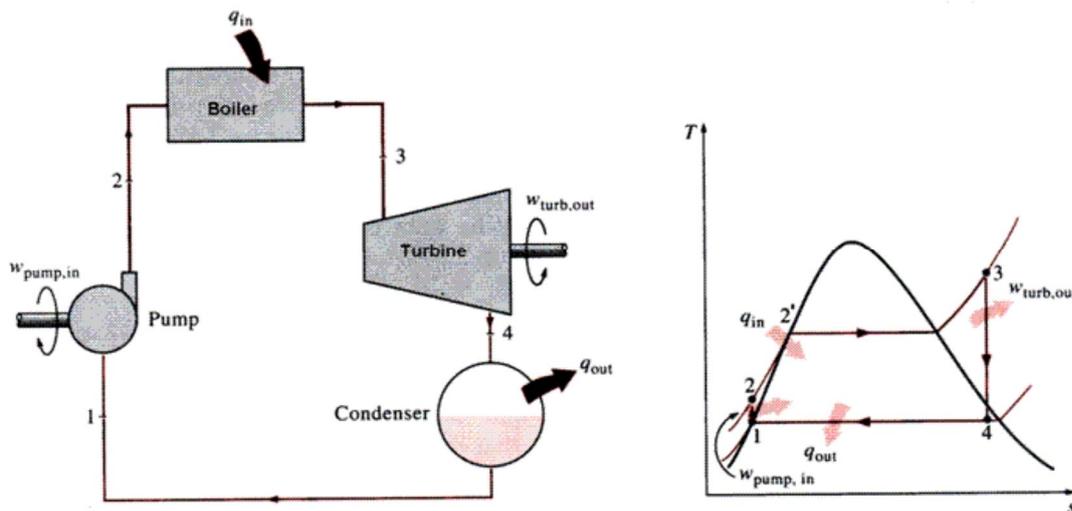


Figure 6: The simple ideal Rankine cycle

The ideal Rankine cycle does not involve any internal irreversibilities and consists of the following four processes:

- stage 1 δ stage 2 (1 δ 2) Isentropic compression in a pump
- stage 2 δ stage 3 (2 δ 3) Heat addition in a boiler at constant pressure
- stage 3 δ stage 4 (3 δ 4) Isentropic expansion in a turbine
- stage 4 δ stage 1 (4 δ 1) Heat rejection in a condenser at constant pressure

It is evident without calculation that the efficiency of this cycle will be less than that of the Carnot cycle operating between the same temperatures, because all the heat supplied is not transferred at the higher temperature. Some heat is added while the temperature of the working fluid varies from state 2 (2) to state 2 δ (2 δ). From the comparison between the areas of the two cycles, that the net work output per kg of steam is greater in the Rankine cycle.

3.1 The externally irreversible Rankine cycle

The actual vapour power cycle differs from the ideal Rankine cycle, as illustrated in Figure 7, as a result of irreversibilities in various components. Fluid friction and undesired heat loss to the surroundings are the two most common sources of irreversibilities.

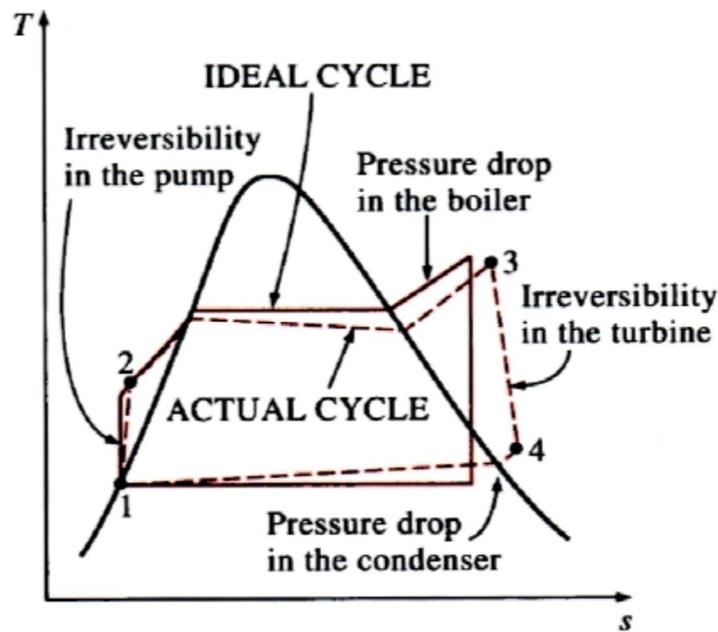


Figure 7: Deviation of an actual vapour power cycle from the ideal Rankine cycle

Fluid friction causes pressure drops in the boiler, the condenser, and the piping between various components. As a result, steam leaves the boiler at a somewhat lower pressure. Also, the pressure at the turbine inlet is somewhat lower than that at the boiler exit due to the pressure drop in the connecting pipes. The pressure drop in the condenser is usually very small. To compensate for these pressure drops, the water must be pumped to a sufficiently higher pressure than the ideal cycle needs. This requires a larger pump and larger work input to the pump.

The other major source of irreversibility is the heat loss from the steam to the surroundings as the steam flows through various components. To maintain the same level of net work output, more heat needs to be transferred to the steam in the boiler to compensate for these undesired heat losses. As a result, the efficiency of the cycle decreases.

Of particular importance are the irreversibilities occurring within the pump and the turbine. A pump requires a more work input, and a turbine produces less work output as a result of irreversibilities.

Other factors also need to be considered in the analysis of actual vapour power cycles. In actual condensers, for example, the liquid is usually sub-cooled to prevent the onset of cavitation, the rapid vaporisation and condensation of the fluid at the low pressure side of the pump impeller, which may eventually destroy it. Additional losses occur at the bearings between the moving parts as a result of friction. Steam that leaks out during the cycle and air that leaks into the condenser represent two other sources of loss. All these power consumed by the auxiliary have to be considered in evaluating the performance of actual power plants.

3.2 Efficiency improvement of the Rankine cycle

The basic idea behind all the modifications to increase the thermal efficiency of a power cycle is the same: increase the average temperature at which heat is transferred to the working fluid in the boiler, or decrease the average temperature at which heat is rejected from the working fluid in the condenser. That is, the average fluid temperature should be as high as possible during heat addition and as low as possible during heat rejection. Next, three ways of accomplishing this for the simple ideal Rankine cycle are presented below.

A) Lowering the condenser pressure

Steam exists as a saturated mixture in the condenser at the saturation temperature corresponding to the pressure inside the condenser. Therefore, lowering the operating pressure of the condenser automatically lowers the temperature at which heat is rejected.

The effect of lowering the condenser pressure on the Rankine cycle efficiency is illustrated on a T - s diagram in Figure 8. For comparison purposes, the turbine inlet state is maintained. The shaded area on this diagram represents the increase in net work output as a result of lowering the condenser pressure from state 4 to state 4'. The heat input requirements also increase (represented by the area under curve state 2' to state 2, but this increase is very small).

Thus, the overall effect of lowering the condenser pressure is an increase in the thermal efficiency of the cycle.

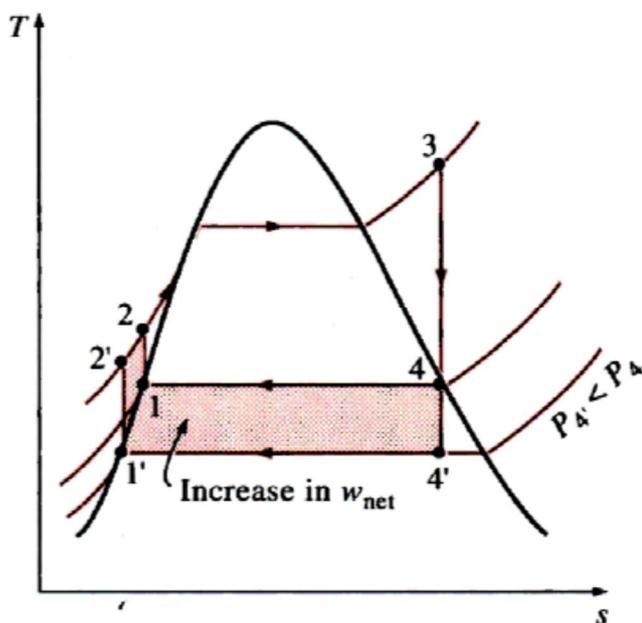


Figure 8: The effect of lowering the condenser pressure of the ideal Rankine cycle

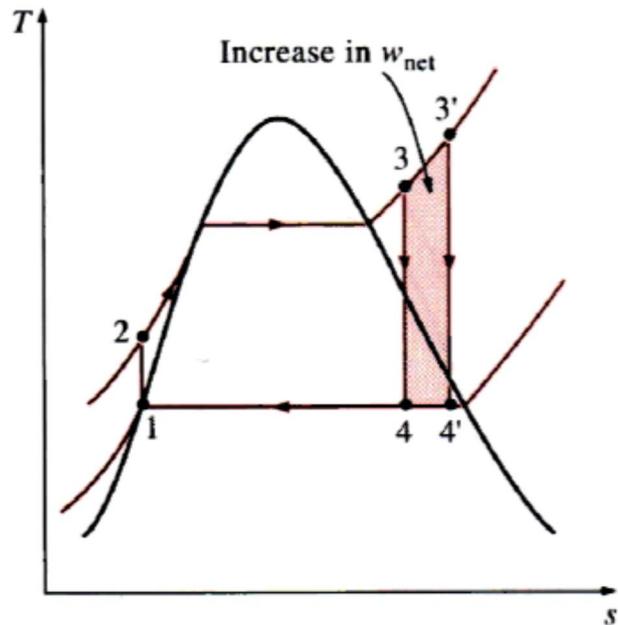


Figure 9: The effect of superheating the steam to higher temperatures in the ideal Rankine cycle

B) Superheating the steam to high temperatures

The average temperature at which heat is added to the steam can be increased without increasing the boiler pressure by superheating the steam to high temperatures. The effect of superheating on the performance of vapour power cycles is illustrated on a T - s diagram in Figure 9. The shaded area on this diagram represents the increase in the net work. The total area under the process curve state 3 to state 3' represents the increase in the heat input.

Thus, both the net work and heat input increase as a result of superheating the steam to a higher temperature. The overall effect is an increase in thermal efficiency, however, since the average temperature at which heat is added increases.

Superheating the steam to higher temperatures has another very desirable effect: it decreases the moisture content of the steam at the turbine exit, as can be seen from the T - s diagram (the quality at state 4' is higher than that at state 4).

C) Increasing the boiler pressure. Supercritical cycles.

Another way of increasing the average temperature during the heat addition process is to increase the operating pressure of the boiler (also called supercritical cycles), which automatically raises the temperature at which boiling takes place. This, in turn, raises the average temperature at which heat is added to the steam and thus raises the thermal efficiency of the cycle.

The effect of increasing the boiler pressure on the performance of vapour power cycles is illustrated on a T - s diagram in Figure 10. Notice that for a fixed turbine inlet temperature, the cycle shifts to the left and the moisture content of steam at the turbine exit increases. This undesirable side effect can be corrected, however, by reheating the steam, as discussed in Section 3.3.

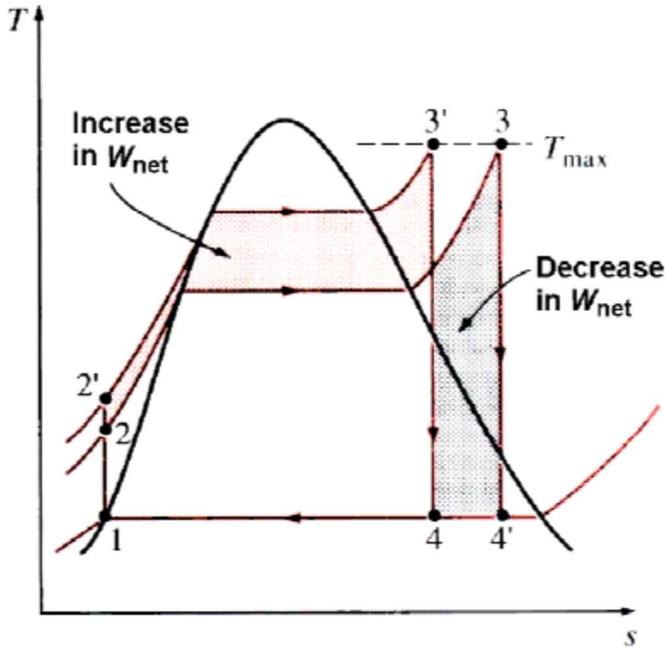


Figure 10: The effect of increasing the boiler pressure in the ideal Rankine cycle

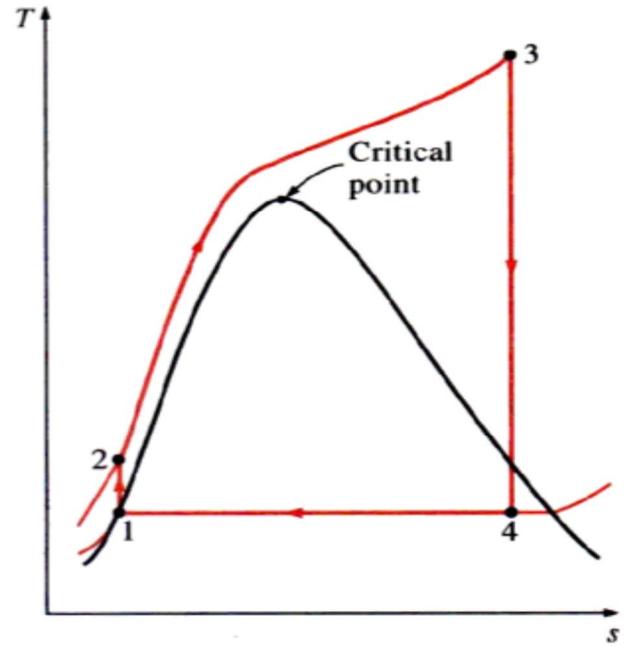


Figure 11: A supercritical Rankine cycle

Operating pressures of boilers have gradually increased over the years from about 125 bar in the seventies to over 300 bar today, generating enough steam to produce a net power output of 1000 MW or more. Today, many modern steam power plants operate at supercritical pressures ($P > 221$ bar) (see Figure 11).

3.3 Reheat

The ideal reheat Rankine cycle (Figure 12) differs from the simple ideal Rankine cycle in that the expansion process takes place in two stages. In the first stage (the high pressure turbine), steam is expanded isentropically to an intermediate pressure and sent back to the boiler where it is reheated at constant pressure, usually to the inlet temperature of the first turbine stage. Steam then expands isentropically in the second stage (low pressure turbine) to the condenser pressure.

The incorporation of the single reheat in a modern power plant improves the cycle efficiency by 4 to 5 per cent by increasing the average temperature at which heat is added to the steam. The average temperature during the reheat process can be increased by increasing the number of expansion and reheat stages. As the number of stages is increased, the expansion and reheat processes approach an isothermal process at the maximum temperature. The use of more than two reheat stages, however, is not practical.

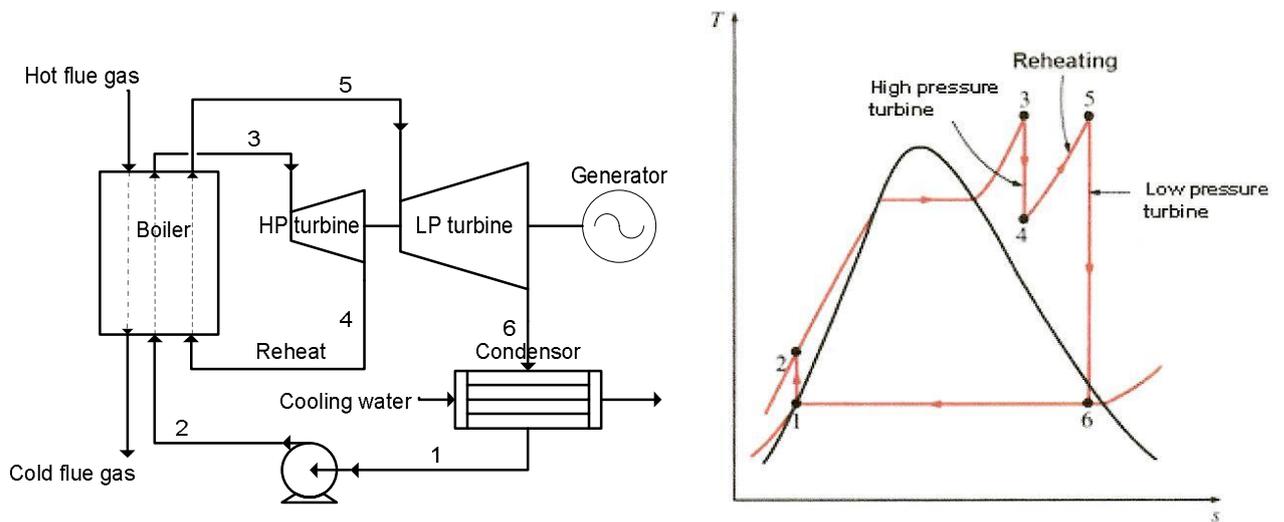


Figure 12: The ideal reheat Rankine cycle

3.4 Regeneration

A practical regeneration process in steam power plants is accomplished by extracting, or “bleeding” steam from the turbine at various points. This steam, which could have produced more work by expanding further in the turbine, is used to heat the feed-water instead. The device where the feed-water is heated by regeneration is called a “regenerator” or a “feed-water heater”

Regeneration not only improves cycle efficiency, but also provides a convenient means of deaerating the feed-water (removing the air that leaks in at the condenser) to prevent corrosion in the boiler. It also helps control the large volume flowrate of the steam at the final stages of the turbine (due to the large specific volumes at low pressures). Therefore, regeneration is used in all modern steam power plants since its introduction in the early 1920s.

A feed-water heater is basically a heat-exchanger where heat is transferred from the steam to the feed-water either by mixing the two fluid streams (open feed-water heaters) or without mixing them (closed feed-water heaters).

The schematic of a steam power plant with one closed feed-water heater and the $T-s$ diagram of the cycle are shown in Figure 13. In an ideal closed feed-water heater, the feed-water is heated to the exit temperature of the extracted steam, which ideally leaves the heater as a saturated liquid at the extraction pressure. In actual power plants, the feed-water leaves the heater below the exit temperature of the extracted steam because a temperature difference of at least a few degrees is required for any effective heat transfer to take place.

The condensed steam is then either pumped to the feed-water line or routed to another heater or to the condenser through a device called a “trap”. A trap allows the liquid to be throttled to a lower pressure region but traps the vapour. The enthalpy of steam remains constant during this throttling process.

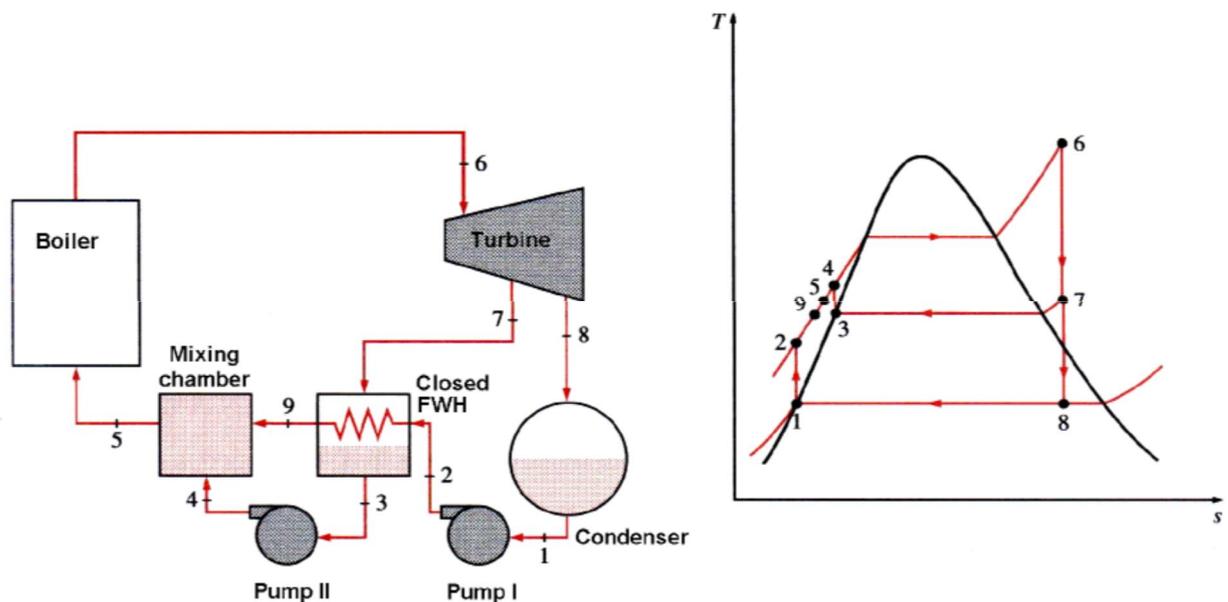


Figure 13: The ideal regenerative Rankine cycle with a closed feed-water heater

4 The Joule or Brayton cycle as the standard cycle for gas turbines

4.1 The ideal Brayton cycle

George Brayton first proposed the Brayton cycle for use in the reciprocating oil burning engine that he developed around 1870. Today, it is used for gas turbines only where both the compression and expansion processes take place in rotating machinery. Gas turbines usually operate on an open cycle. Fresh air at ambient conditions is drawn into the compressor where its temperature and pressure are raised. The high pressure air proceeds into the combustion chamber where the fuel is burned at constant pressure. The resulting high temperature gases then enter the turbine where they expand to the atmospheric pressure, thus producing power.

The exhaust gases leaving the turbine are thrown out (not recirculated), causing the cycle to be classified as an open cycle.

The open gas turbine cycle described above can be modelled as a closed cycle, as shown in Figure 14, by utilising the air-standard assumptions where deductions about the performance uses air as the working fluid. Here, the compression and expansion processes remain the same, but the combustion process is replaced by a constant pressure heat addition process from an external source, and the exhaust process is replaced by a constant pressure heat rejection process to ambient air.

The ideal cycle that the working fluid undergoes in this closed loop is the Joule or Brayton cycle, which is made up of four internally reversible processes:

- stage 1 ó stage 2 (12) Isentropic compression in a pump
- stage 2 ó stage 3 (23) Heat addition in a boiler at constant pressure
- stage 3 ó stage 4 (34) Isentropic expansion in a turbine
- stage 4 ó stage 1 (41) Heat rejection in a condenser at constant pressure

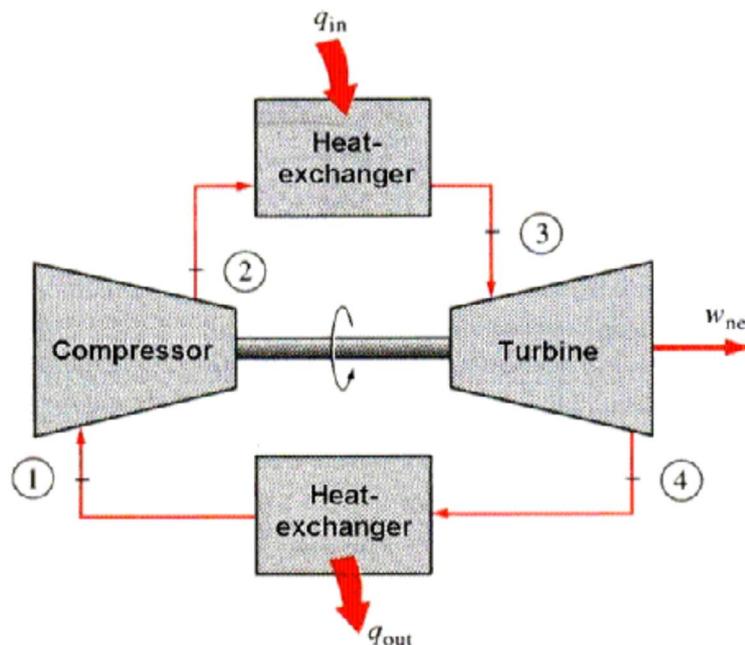


Figure 14: A closed cycle gas turbine engine

The T - s and P - v diagrams of an ideal Brayton cycle are shown in Figure 15. Notice that all four processes of the Brayton cycle are executed in steady flow devices; thus they should be analysed as steady flow processes.

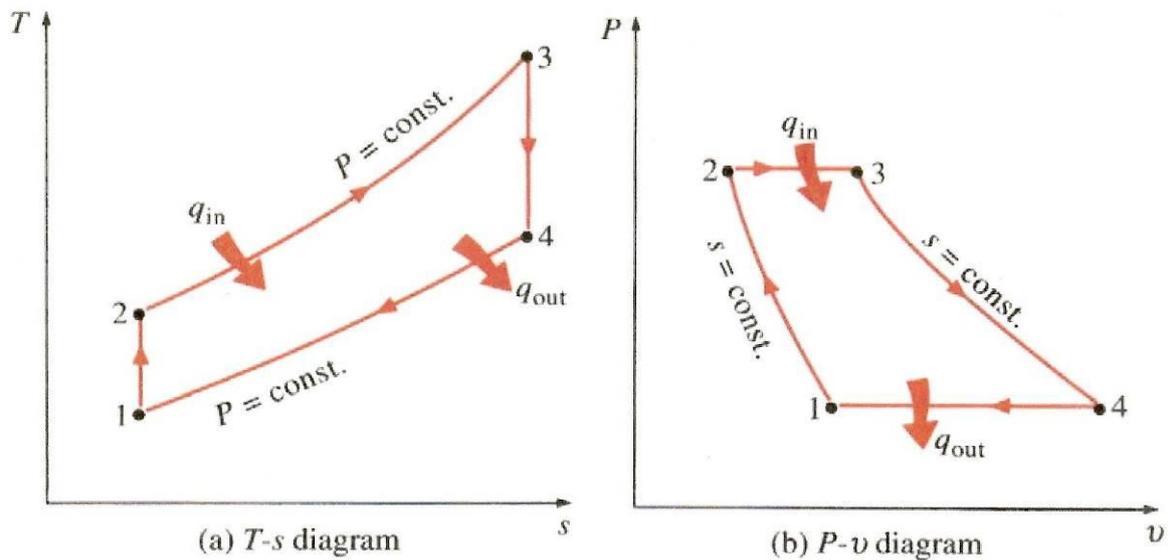


Figure 15: T - s and P - v diagrams for the ideal Brayton cycle

The thermal efficiency of a gas turbine engine depends on the allowable maximum gas temperature at the turbine inlet. Raising the turbine inlet temperature from 900 to 1200 °C increases the power output by 71 % and the thermal efficiency by 26 %. Significant advances, such as coating the turbine blades with ceramic layers and cooling the blades with the discharge air from the compressor have been made during the last two decades. As a result, today's gas turbines can withstand temperatures as high as 1425 °C at the turbine inlet, and gas turbine power plants have efficiencies well over 30 %.

Since the mid-seventies, gas-turbine power plants have been used in the power generation industry to cover emergencies and peak periods because of their relatively low cost and quick response time. Gas turbines are also used in conjunction with gas fired power plants, the exhaust gases of the gas turbine serving as combustion air for the conventional part (see topping cycle) and since the nineties in combined cycle gas turbine systems (CCGT) and in co-generation schemes.

4.2 Non-ideal Brayton cycle

The real gas turbine cycle differs from the ideal Brayton cycle for several reasons. For one thing, some pressure drop during the heat addition and rejection processes is inevitable. More importantly, the actual work input to the compressor will be more, and the actual work output of the turbine will be less because of irreversibilities such as friction and non-quasi-equilibrium operation conditions of these devices.

As soon as compressor and turbine inefficiencies are introduced, as indicated in Figure 16, where states $2a$ ($2a$) and stage $4a$ ($4a$) are the real exit states of the compressor and the turbine, respectively, and $2s$ and $4s$ are the corresponding states for the isentropic case, the importance of a high work ratio becomes obvious. With a low value of T_3 , the difference between the turbine work and compressor work becomes very small, and a slight decrease in the turbine work and increase in the compressor work is sufficient to reduce the work output, and cycle efficiency, to zero.

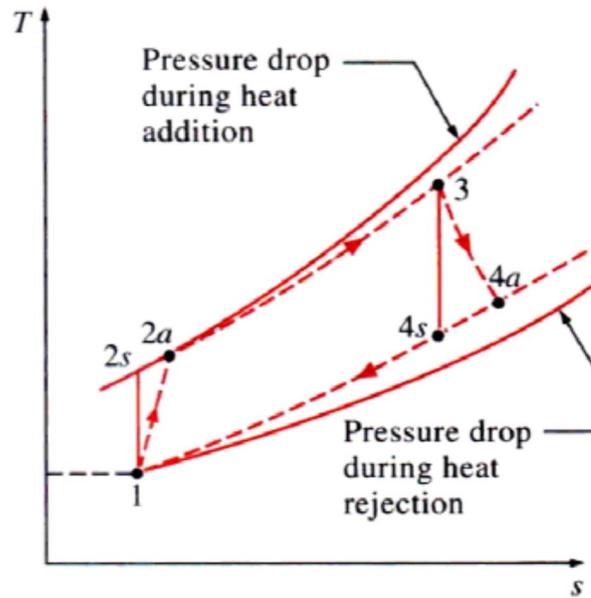


Figure 16: The deviation of an actual gas turbine cycle from the ideal Brayton cycle as a result of irreversibilities

The efficiency of the irreversible cycle can be shown to be not only a function of pressure ratio (r_p), but also of T_3 (Figure 17). There is an optimum pressure ratio for a maximum cycle efficiency as well as an optimum pressure ratio for maximum specific work output (although these optimum pressure ratios are not the same). The ideal efficiency curve is also shown in Figure 17, to emphasise the marked reduction in efficiency caused by irreversibilities in the compressor and turbine. The following sections deal with the more important modifications to the simple cycle which may be adapted to improve both the ideal efficiency and work ratio.

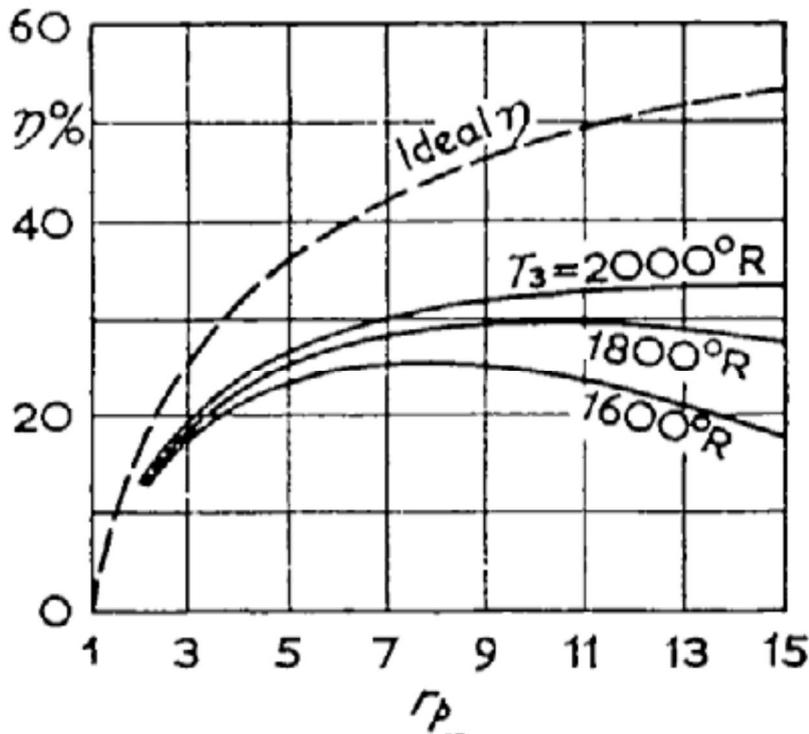


Figure 17: Thermal efficiency of the Brayton cycle as a function of pressure ratio (r_p) and temperature (T_3)

4.3 Regeneration

In the recuperative (or regenerative) cycle, a counter-flow heat-exchanger transfers exhaust heat to the compressed air before it enters the combustor. This is shown schematically in Figure 18. The amount of fuel needed to heat the air to combustion temperature is reduced by up to 25 %.

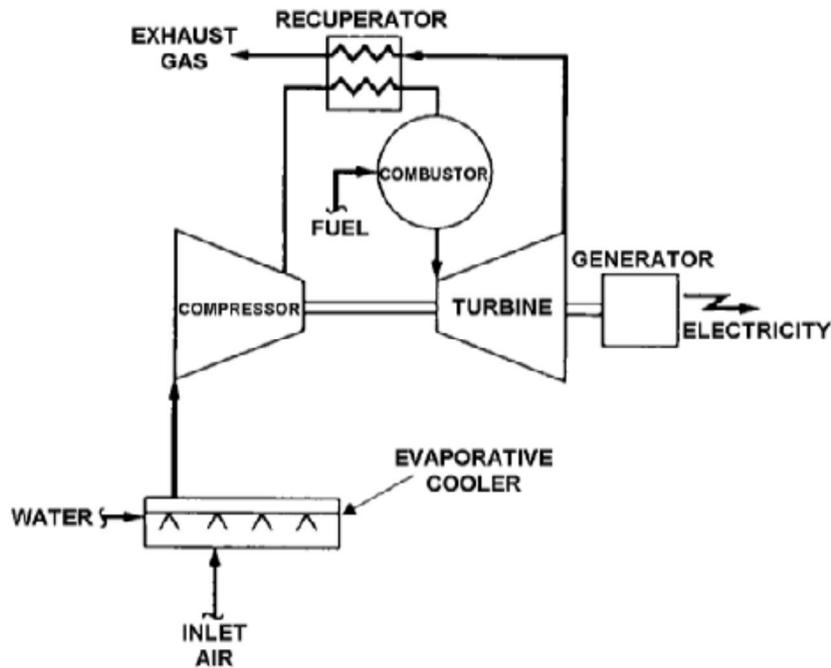


Figure 18: Evaporative cooling and recuperative cycles

4.4 Compressor intercooling

In the intercooled cycle, a heat-exchanger is placed in the air path between low and high pressure sections of the compressor, as shown in Figure 19. Compression of cool air requires less work than compression of warm air. This heat-exchanger is designed to cool the air and reduce the amount of work required in the high pressure section of the compressor, increasing both power output and efficiency. The intercooler can be a direct contact (evaporative) or an extended surface type heat-exchanger.

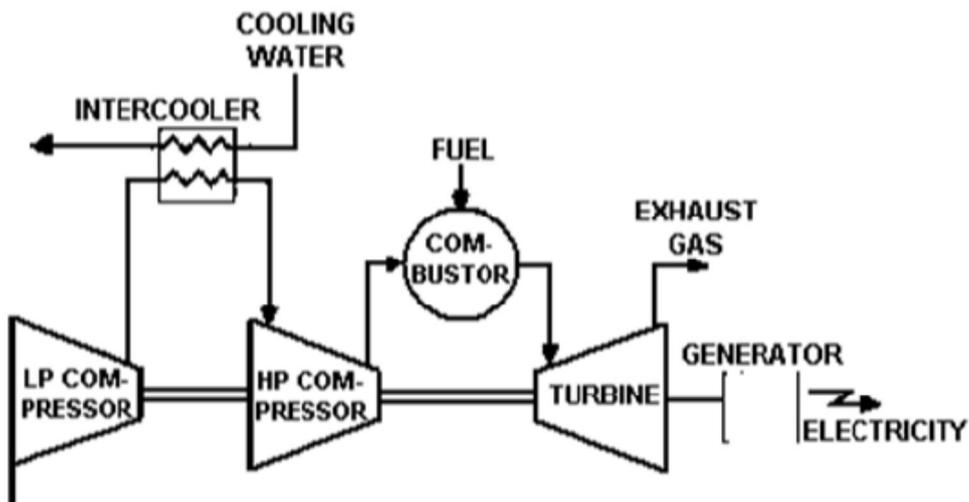


Figure 19: Intercooled cycle

4.5 Turbine reheat

The gas turbine reheat cycle is analogous to the steam turbine reheat cycle. The hot gases are partially expanded through the turbine, reheated by a second stage combustor, and returned to the turbine. This cycle is more complex than the standard gas turbine cycle.

Figure 20 illustrates a schematic representation of this cycle and the characteristics are as follows:

- the reheat cycle is more efficient
- to protect the reheat combustor, the high pressure turbine is required to lower the temperature of the hot gases entering the reheat combustor
- in general, it is necessary to increase the fuel/air ratio to increase specific power. Metallurgy and cooling technology limit the turbine inlet temperature. The two-stage combustor allows more fuel to be injected without exceeding this temperature limit. This increases the exhaust temperature and makes the reheat combustion turbine more suitable for the combined cycle
- the reheat combustion turbines higher specific power results in less residual oxygen in the exhaust sections which are smaller than those for a simple cycle gas turbine of the same output.

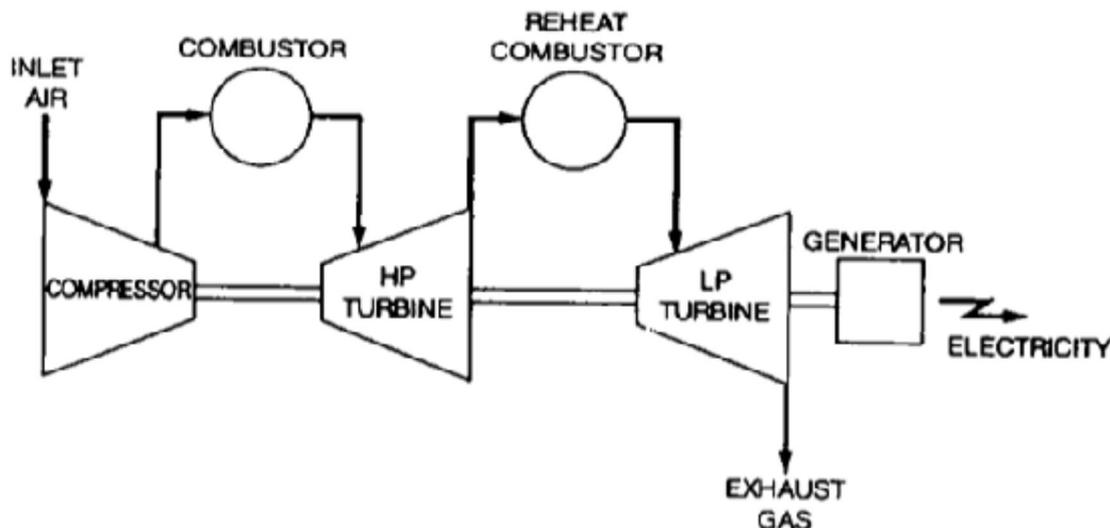


Figure 20: Reheat cycle

5 The Stirling motor

The Stirling engine is one of the few applications that are working in a closed cycle. It was invented in 1816 by Rev. Robert Stirling of Scotland. The Stirling is a very simple engine, and was often billed as a safe alternative to a steam engine (since there is no boiler to explode, which, in those times was not so unusual). It enjoyed some success in industrial applications, and in small appliances like fans and water pumps, but was eclipsed by the advent of inexpensive electric motors. Since it can run on any source of heat, it now holds promise for alternative fuel engines, solar power, geothermal power, etc.

Stirling engines feature a completely closed system in which the working gas (usually air but sometimes helium or hydrogen) is alternately heated and cooled by shifting the gas to different temperature locations within the system.

In the two-cylinder (or *alpha configured*) Stirling, one cylinder is kept hot while the other is kept cool. In the illustration the lower-left cylinder is heated by burning fuel. The other cylinder is kept cool by an air cooled heat sink (a.k.a. cooling fins).

The Stirling cycle may be thought of as four different phases: expansion, transfer, contraction, and transfer as shown in Figure 21. [14]

Expansion. At this point, most of the gas in the system has just been driven into the hot cylinder. The gas heats and expands driving both pistons inward.

Transfer. At this point, the gas has expanded (about 3 times in this example). Most of the gas (about 2/3rds) is still located in the hot cylinder. Flywheel momentum carries the crankshaft the next 90 degrees, transferring the bulk of the gas to the cool cylinder.

Contraction. Now the majority of the expanded gas has been shifted to the cool cylinder. It cools and contracts, drawing both pistons outward.

Transfer. The now contracted gas is still located in the cool cylinder. Flywheel momentum carries the crank another 90 degrees, transferring the gas back to the hot cylinder to complete the cycle.

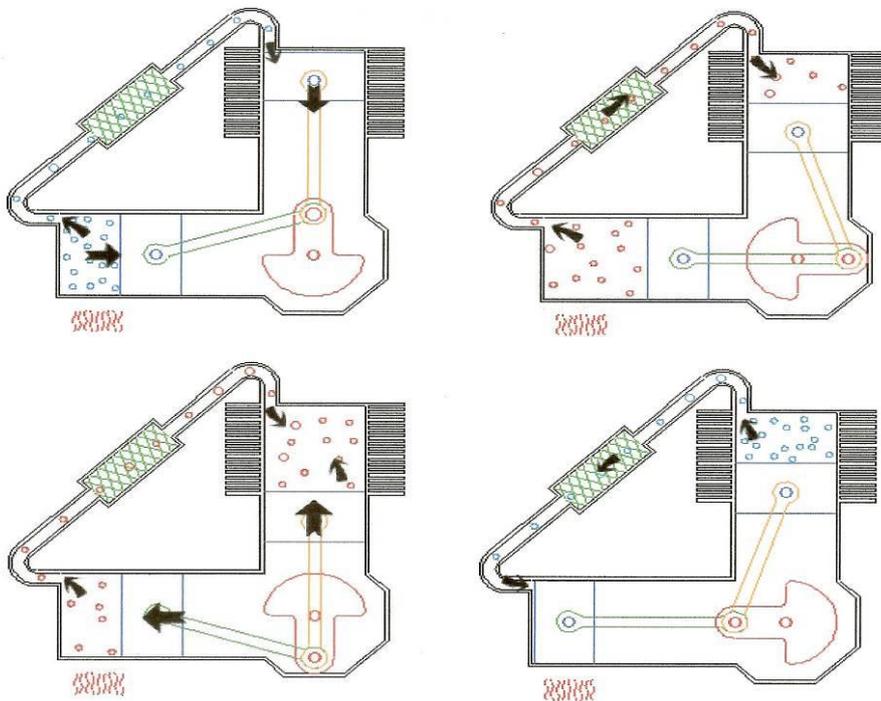


Figure 21: Stirling engine cycles

This engine also features a *regenerator*, illustrated by the chamber containing the green hatch lines. The regenerator is constructed of material that readily conducts heat and has a high surface area (a mesh of closely spaced thin metal plates for example). When hot gas is transferred to the cool cylinder, it is just driven through the regenerator, where a portion of the heat is deposited. When the cool gas is transferred back, this heat is reclaimed; thus the regenerator "pre heats" and "pre cools" the working gas, significantly improving efficiency.

Today, the Stirling engine seems to enjoy a revival, also for domestic application in co-generation schemes.

6 Combined cycles

As described in Section 2.5, the Carnot cycle is the most efficient cycle that can be executed.

Naturally, the efficiencies of real processes are lower since there are losses involved. The process efficiency can be improved by raising the maximum temperature in the cycle, releasing the waste heat at a lower temperature, or by improving the process to minimise the internal exoegetic losses.

The interest in combined cycles arises particularly from these considerations. By its nature, no single cycle can make all of these 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 high. In the steam cycle, the maximum process temperature is much lower than the gas turbine process, but the exhaust heat is returned to the environment at a low temperature. A combination of the two cycles leads to the 'combined cycle' shown in Figure 22. As illustrated in Table 1 combining a gas turbine and a steam turbine thus offers the best possible basis for a high efficiency thermal process.

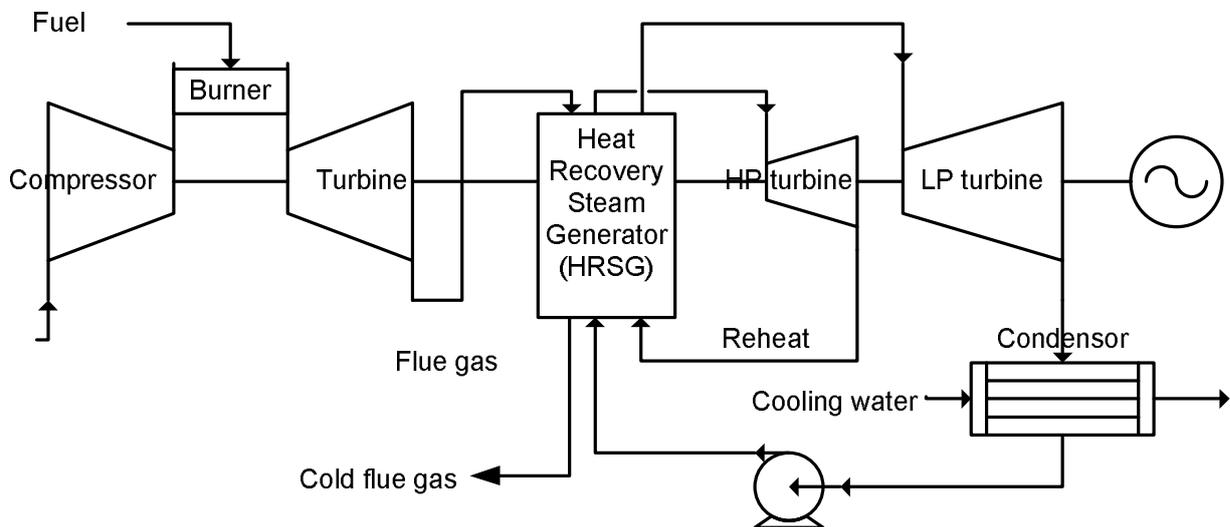


Figure 22: The CCGT ó Combined Cycle Gas Turbine

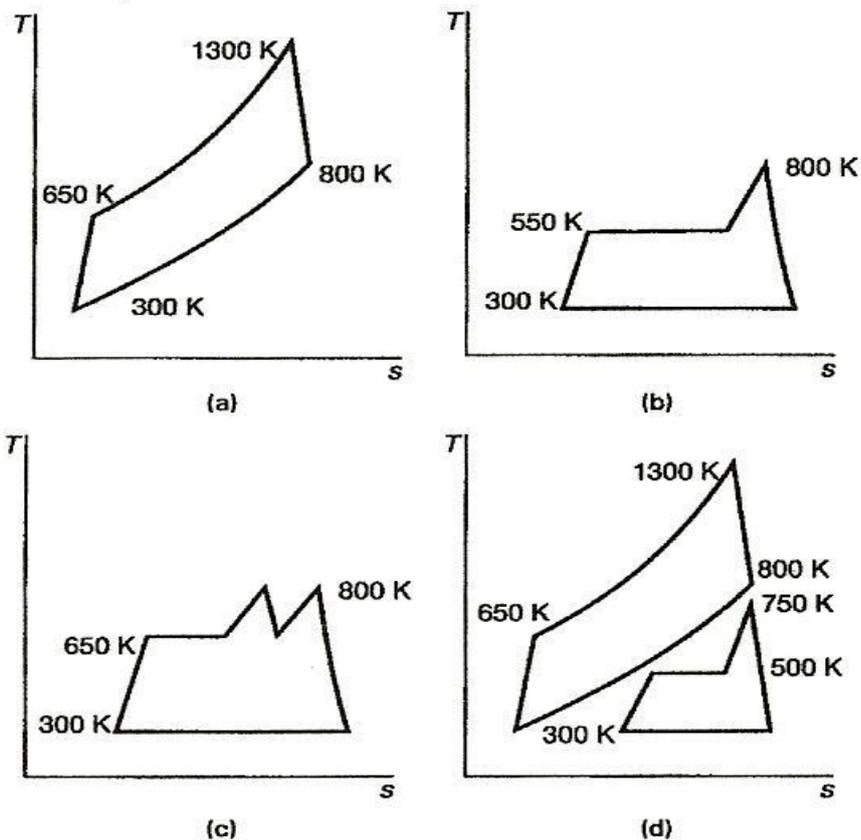
Parameters	Gas turbine power plant (cycle a)	Steam turbine power plant (cycle b)	Steam turbine power plant with reheating (cycle c)	Combined cycle power plant (cycle d)
Average temperature of heat supplied (K)	1000	600	680	1000
Average temperature of dissipated heat (K)	520	300	300	300
Carnot efficiency (%)	48	50	50	70

Table 1a: Thermodynamic comparison of gas turbine, steam turbine and combined cycle processes [50, Korobitsyn, 1998]

The last line in Table 1 shows the Carnot efficiencies of the various processes. Although that is not the case in reality, this table 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 of around 20 points lower than that of a good combined cycle plant. For combined cycle power plants, actual efficiencies are around 75 % of the Carnot efficiency. The differences between the actual efficiencies attained by a combined cycle power plant and the other processes are, therefore, not quite as large as illustrated in Table 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.

Recent developments in gas turbine technology have made the combined gas-steam cycle economically very attractive, due to better mastering of the turbine technique, new materials and more efficient manufacturing processes and the economic attractiveness of the higher thermodynamic efficiency. Consequently, all new power plants operate on combined cycles, and many existing steam plants are retrofitted into combined cycle power plants by adding a gas turbine in a topping cycle as described hereafter. Thermal efficiencies well over 40 % are reported as a result of conversion.

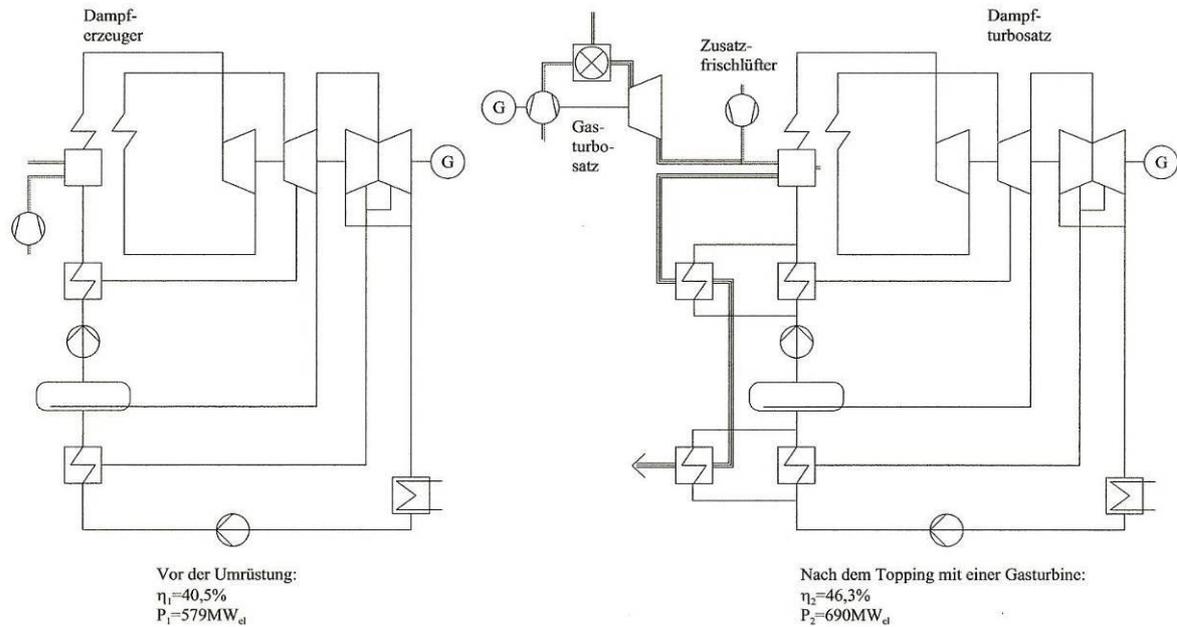


Plant	(a)	(b)	(c)	(d)
Average temperature of heat supplied, K	1000	600	680	1000
Average temperature of heat rejected, K	520	300	300	300
Carnot efficiency, %	48	50	50	70

Table 1b: Thermodynamic comparison of gas turbine, steam turbine and combined cycle processes ó combined diagrams [50, Korobitsyn, 1998]

7 Gas turbine as topping cycle and as hot utility provider in industrial processes

A particular application is the use of gas turbines in combined cycle mode by using the flue gas out of the expansion turbine as preheated combustion air for high temperature furnaces and also as revamp in existing conventional gas fired power plants (Figure 23). Indeed, the flue gas out of the gas turbine has still an oxygen content of around 15%, which suffices to be used as combustion air.



Flußdiagramm der Kraftwerksanlage Eemscentrale 2 in Groningen

Figure 23: Gas turbine for topping cycle in a gas fired power station

8 Co-generation (CHP)

In all the cycles discussed so far, the sole purpose was to convert a portion of the heat transferred to the working fluid to work, which is the most valuable form of energy. The remaining portion of the heat is rejected to rivers, lakes, oceans, or to the air as waste heat, because its quality (or grade) is too low to be of any practical use.

Many systems or devices, however, require energy input in the form of heat, called process heat, which might be needed in a wide range of temperature levels between 50 °C and 1500 °C. Heat up to 120 °C can be supplied by hot water circuits, heat up to 250 °C usually is supplied by steam at corresponding saturation pressures between 1 bar (100 °C) and 45 bar (250 °C). Heat up to 500 °C can be supplied by flue gas from gas turbines, heat up to 1500 °C must be supplied by burning fuel without and using the flue gas at high temperature. Fuel can be coal, oil, natural gas, or another fuel in a furnace, depending upon the particular needs.

The temperature in classic furnaces is typically very high (around 1500 °C), and thus the energy in the furnace is of very high quality. This high quality energy is transferred to evaporate water at a certain temperature level to produce steam and to superheat the steam for appropriate transportation. This steam generation process occurs at temperatures that are relatively low compared to the flame temperatures and represents a highly irreversible process. Associated with this irreversibility is, of course, a loss in work potential and thus a loss of exergy that is the scarce resource and must, therefore, be used sparingly. So it is simply not wise to use high quality energy to accomplish a task that could be accomplished with low quality energy.

A plant which producing electricity whilst meeting the process heat requirements of certain industrial processes, is called a co-generation plant. In general, co-generation means the simultaneous production of more than one useful form of energy (such as process heat and electric power). Either a steam turbine (Rankine) cycle or a gas turbine (Brayton) cycle or even a combined cycle can be used as the power cycle in a co-generation plant.

8.1 The back-pressure cogeneration plant

The schematic of an ideal steam turbine co-generation plant is shown in Figure 24. It is the classic back-pressure steam generation plant. In the absence of a condenser, no heat is rejected from this plant as waste heat. In other words, all the energy transferred to the steam in the boiler is utilised as either process heat or electric power.

To take account of the latter benefit, the total efficiency, perhaps better called the energy utilisation factor (EUF), is a more adequate measure of performance than the simple work.

The ideal steam turbine co-generation plant described above is not necessarily the most practical because it cannot always adjust to very wide variations in power and process heat loads. The schematic of a more practical (but more complex) co-generation plant is shown in Figure 25. It is the back-pressure co-generation plant with additional condensing section. Under normal operation, some steam is extracted from the turbine at some predetermined intermediate pressure state P_6 . The rest of the steam expands to the condenser pressure state P_7 and is then cooled at constant pressure. The heat rejected from the condenser represents the waste heat for the cycle.

At times of high demand for process heat, all the steam is routed to the process heating units and none to the condenser. The waste heat is zero in this mode. If this is not sufficient, some steam leaving the boiler is throttled by an expansion or pressure reducing valve (PRV) to the extraction pressure P_6 and is directed to the process heating unit. Maximum process heating is realised when all the steam leaving the boiler passes through the PRV. No power is produced in this mode. When there is no demand for

process heat, all the steam passes through the turbine and the condenser, and the co-generation plant operates as an ordinary steam power plant.

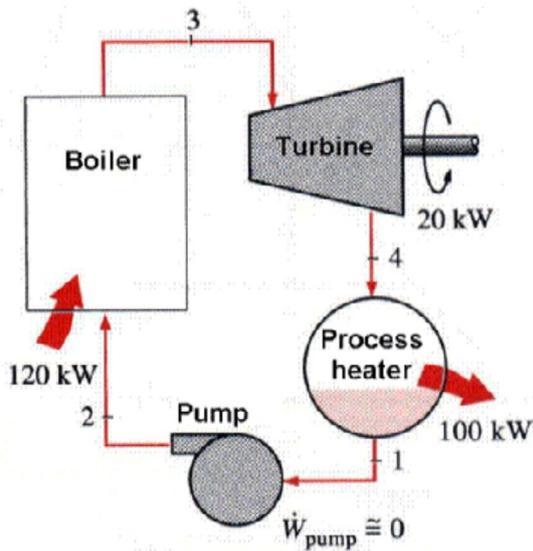


Figure 24: An ideal co-generation plant

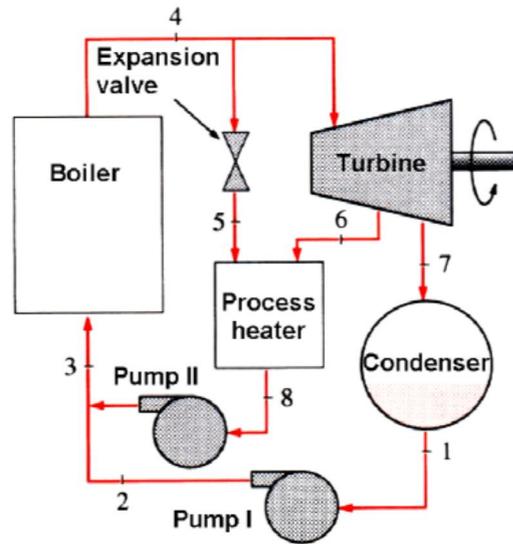


Figure 25: Co-generation plant with adjustable loads

Under optimum conditions, a co-generation plant simulates the ideal co-generation plant discussed earlier. That is, all the steam expands in the turbine to the extraction pressure and continues to the process heating units. This condition may not always be achieved in practice because of the constant variations in the process heat and power loads. But the plant should be designed so that the optimum operating conditions are approximated most of the time for the heat demand in the first place. The heat is the primer product of a co-generation plant; the associated power is a övery valuable - öby-productö.

It should be emphasized that öco-generationö means simultaneous production of heat and power. In a co-generation plant with condensing section at full load conditions, it is no longer a matter of producing heat AND power, but rather heat OR power: steam that is condensed is no longer available as useful process heat and process heat extracted does not produce condensing power. In such plant, only a part of the power produces is really öco-generatedö power.

It should also be mentioned that prior art systems installed at large industrial premises delivered steam in steam distribution systems at various pressure levels, adapted for particular temperature ranges. Examples are: the system with boiler steam conditions of 120 bar, 525 °C and steam distribution levels of 45 bar (250 °C), 16 bar (200 °C), 5 bar (150 °C) and 1.2 bar (100 °C), or the system with boiler steam conditions of 90 bar, 400 °C and steam distribution levels of 30 bar (225 °C), 10 bar (175 °C) and 3 bar (125 °C). Between each temperature level and the next lower one, additional power could be generated in back-pressure, making lower pressure steam less expensive and thus creating an incentive on the user side for using the most appropriate steam level.

8.2 The combined cycle cogeneration plant

Development of the aero-derivative gas turbines, derived from the engines used in the aeronautics lead to applications in the power industry, using the engines for back-up and for peak power generation and also in process industries using the engines to replace conventional power systems for cogeneration.

The primary advantage of such replacement is the much higher achievable power-to-steam ratio, with a significant increase of the exergy efficiency. In the conventional back-pressure steam plant, power to steam ratios of 0.05 to 0.20 kWh/ton of steam are achievable. With a gas turbine upfront, this ratio can be increased up to 0.40 to 0.50 kWh/ton of steam. As an example, the frequently used LM6000 gas turbine based co-generation system generates 40 MW and 100 t/h of steam. An economic disadvantage, however, is the reduced flexibility with regard to the choice of input fuel and the higher price for the more "noble" natural gas.

Figure 26 shows a combined cycle co-generation plant with additional condensing section. Smaller plants usually will not have such condensing section since a condensing turbine is relatively expensive for small sizes.

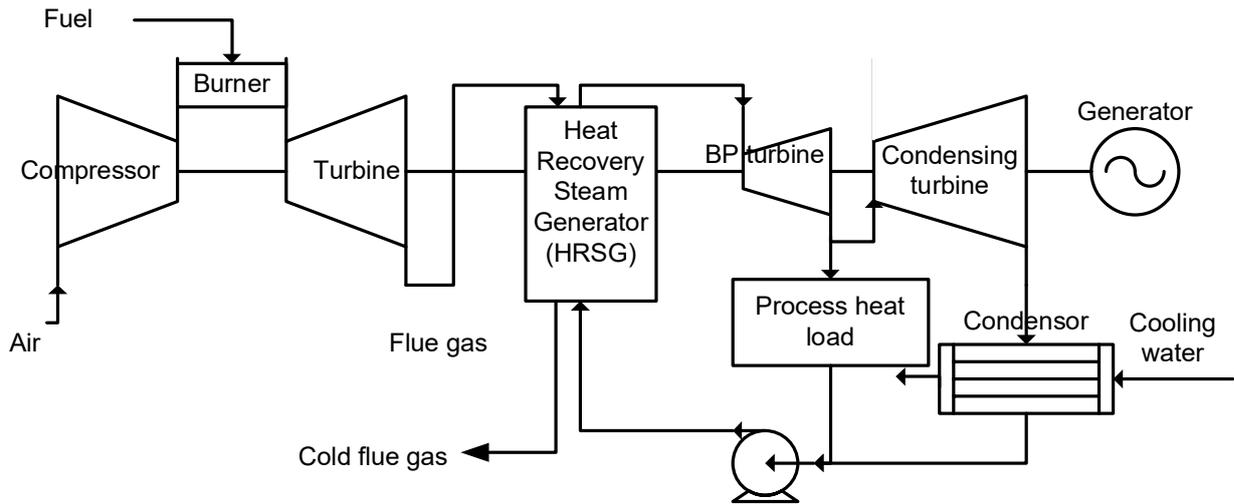


Figure 26: A combined cycle co-generation plant

Gas turbines are also used in co-generation mode by using the flue gas out of the expansion turbine directly as hot utility to supply process heat requirements at high temperature levels from 250 °C to 500 °C. Typical applications are found in the petrochemical, chemical and fertiliser industry.

8.3 The combined cycle cogeneration plant with supplementary firing

A practical way to increase the flexibility of a combined cycle co-generation plant is to add a burner system to the HRSG and to burn additional fuel in the HRSG in order to raise the temperature and to increase the energy content of the flue gas. A process scheme is shown in Figure 27. Supplementary firing typically enables the amount of generated steam to be doubled. Such scheme offers the additional advantage that also other fuels, including waste fuels, can be burned at higher thermodynamic efficiency than what would be achievable otherwise.

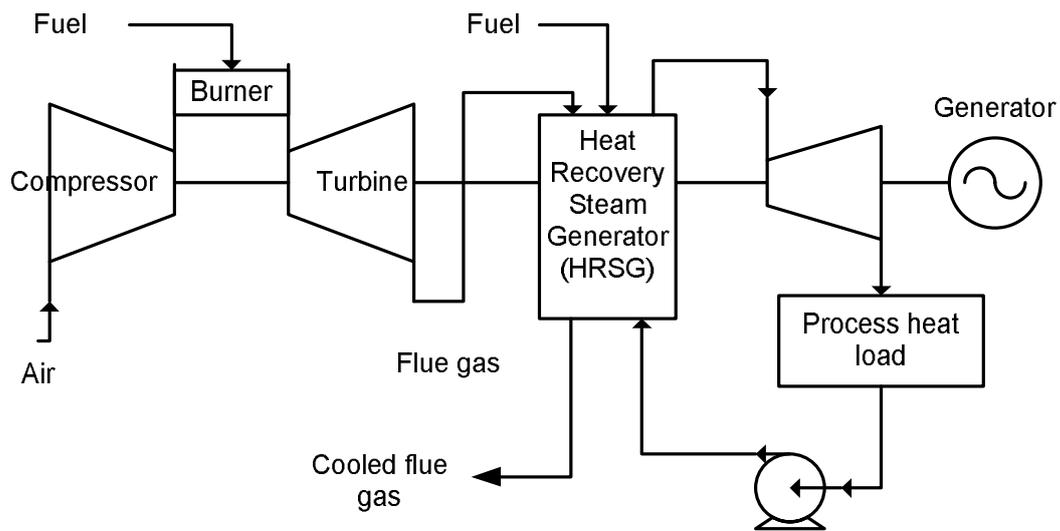


Figure 27: A combined cycle co-generation plant with additional firing

A typical application in the steel industry is the use of blast furnace off-gas that favourably can be used as additional fuel for supplementary firing. This technique competes successfully with the direct use of said off-gas in a gas turbine, which requires significant compression power to bring the off-gas up to the pressure required for the gas turbine burner system and which has the additional disadvantage that the gas turbine system is no longer available from the shelves and requires significant adjustment to suit the conditions of the off-gas fuel.

9 Technical options to remove CO₂ from flue-gases

Given current technology, increasing the thermal efficiency of processes and techniques - generating energy is the most important measure in reducing the amount of greenhouse gases emitted per unit of energy produced. Efficiency increases are limited by various factors so that, even with increased efficiency, significant amounts of CO₂ will be emitted. To reduce the emissions of CO₂ further, different technical options are currently under development or are at a research stage and might be available in the future. These technical options are well described by the IEA (International Energy Agency), [12], IEA, 1992 and will be briefly presented in this annex of the document.

9.1 Absorption techniques to remove CO₂ from flue-gases

Absorption of CO₂ is a technique used in the chemical industry for the commercial production of CO₂. The amounts of CO₂ currently produced are small in comparison with the total amount of CO₂ that could be removed from the flue-gas in the global sector-generating energy. Therefore such absorption techniques would have to be introduced on a much larger scale.

Typically three basic types of absorption systems are possible: chemical, physical and hybrid systems. In chemical absorption systems, the CO₂ reacts with chemical solvents to form a weakly bonded intermediate compound which is broken down by the application of heat, regenerating the original solvent and producing a CO₂ stream. Typical solvents are amine- or carbonate-based, such as MEA, diethanolamine (DEA), ammonia and hot potassium carbonate. These processes can be used at low CO₂ partial pressures, but the flue-gas must be free of SO₂, O₂ hydrocarbons and particulates because of operating problems in the absorber.

The CO₂ can also be physically absorbed in a solvent and then regenerated using heat and/or pressure reduction. Typical solvents are dimethylether of polyethylene glycol and cold methanol, which are applied at high pressure. At low pressures, the chemical absorption processes are economic.

Hybrid solvents combine the best characteristics of both the chemical and physical solvents and are usually composed of a number of complementary solvents.

All the absorption processes operate in essentially the same manner by scrubbing the flue-gas in absorption towers to collect the CO₂ and then regenerating the solvent and releasing the CO₂.

9.2 Adsorption techniques to remove CO₂ from flue-gases

Solid adsorption methods employ a physical attraction between the gas and -active sites on the solid, whereas solid absorption methods employ a chemical reaction to capture the target gas.

There are several adsorption methods used commercially in the process industries that may be applicable for removing CO₂ from power plant flue-gases. These employ adsorbent beds of alumina and zeolite molecular sieves (natural or manufactured aluminosilicate).

There are various methods of regeneration. Pressure swing regeneration or adsorption (PSA) involves lowering the pressure in the vessel containing the saturated bed until trapped gases are pulled off the bed. The regeneration cycles are relatively short and are typically measured in seconds. Thermal (or temperature) Swing Adsorption employs high temperature regeneration gas to drive off the trapped gas. The regeneration cycles are quite long (measured in hours) and result in larger quantities of adsorbent being required than with PSA systems.

9.3 Cryogenic techniques to remove CO₂ from flue-gases

Cryogenic separation involves cooling the gases to very low temperatures so that frozen CO₂ can be separated. The potential advantages of the process include the possibility of direct disposal of CO₂ ice (e.g. in the deep ocean) and the high purity of the separated gas which is close to 100 %. The

disadvantage includes the high energy input required for reaching the cryogenic temperature. The major energy requirements are incurred during the compression stage so that improvements in this area would greatly reduce the overall power plant efficiency loss [40, Soria, et al., 1998].

9.4 Membrane techniques to remove CO₂ from flue-gases

In membrane separation, an appropriate membrane is used to separate the flue-gases into a CO₂ rich and lean gas stream. Two membrane operations are possible, gas separation and gas absorption. The combination of MEA and membranes is said to be the best option.

Gas separation membranes rely on a difference in physical or chemical interaction between components present in a gas mixture with the membrane material causing one component to permeate faster through the membrane than others.

Gas absorption membranes are membranes which are used as contacting devices between a gas flow and a liquid flow. The separation is caused by the presence of an absorption liquid on one side of the membrane. In contrast with gas separation membranes, it is not essential that the membrane has any selectivity at all.

9.5 The Carnol technique to remove CO₂ from flue-gases

The Carnol system, which is under development in the US, combines CO₂ removal from coal-fired power plants with methanol generation. It thus eliminates the need for CO₂ disposal. Part of the CO₂ emitted is used for methanol production, and some of the carbon is separated in solid form during the process. As such, it can either be stored or sold. The proposed system also uses the waste heat from the methanol production to decrease the energy needs for CO₂ extraction from the stack gases [13, Soria, et al., 1998]. This process is just in the research stage.

9.6 Comparison of the different CO₂ removal options

A comparison of the different technical options that might in the future be available for CO₂ removal from large combustion plants has been carried out by the IEA Greenhouse Gas R&D Programme [12], IEA, 1992. The four following power generation schemes have been studied:

- 1) a modern pulverised coal-fired power plant equipped with flue-gas desulphurisation (FGD) and operating with a subcritical high temperature steam cycle, assuming an efficiency of 40 %
- 2) a natural gas fired combined cycle power plant assuming an overall electrical efficiency of 52 %
- 3) an integrated gasification combined cycle (IGCC) power plant in which a coal slurry is fed to an oxygen blown gasifier of the entrained flow type assuming an efficiency of 42 %
- 4) a power plant burning pulverised coal in oxygen using recycled CO₂ to moderate the combustion temperature, assuming an efficiency of 33 %. It should be noted that this technique must be regarded as a long-term one.

As a base case for each of the techniques mentioned above, a 500 MW_e power plant, with a 35 year working life has been assumed. Furthermore, the plants are assumed to be seawater cooled and located on the coast in Western Europe. The ambient conditions are 15 °C, 1.013 bar, relative humidity 60 % and the cooling water is available at 15 °C. The design fuel is an internationally traded Australian bituminous coal of 0.86 % sulphur content from the Drayton mine. Plant emissions are controlled according to EU Directives. Also 90 % removal of sulphur dioxide (SO₂) has been specified.

For the gas turbine combined cycle case study, two gas turbines are used to obtain a net power output of approximately 500 MW with fuel being a typical Brent field natural gas.

The results of this study are presented in the Table 2 and Table 3. The information on costs was originally given in US dollars, nominally set in the third quarter of 1992. For the purpose of this document, the cost has been converted to Euro using the exchange rate from October 2000. The

accuracy is believed to be around +/- 30 %, so results should be taken only as an indication. It should also be noted that these figures do not take into account the cost of carbon dioxide disposal, which will be a necessary measure for most of the techniques discussed.

Performance parameter	CO2 removal technique	Pulverised coal power plant with FGD	Gas turbine combined cycle	Integrated gasification combined cycle	Pulverised coal combustion in oxygen using recycled CO2
Changes in the net efficiency by applying different CO2 removal techniques according to a given reference case	Reference efficiency without CO2 removal	40%	52%	42%	33%
	Absorption	29%	42%	28% (Note 2)	30%
	Adsorption PSA	28%	33%	26%	29%
	Adsorption TSA	29%	39%	29%	-
	Cryogenic technique	NA	NA	36%	27%
	Separation membrane	31%	31%	26%	31%
CO2 Captured	Absorption membrane+ MEA	30%	47%	32%	30%
	Absorption	90%	85%	90%	99%
	Adsorption PSA, TSA	95%	95%	95%	95%
	Cryogenic technique	-	-	85%	85%
	Separation membrane	80%	80%	80%	80%
CO2 in product	Absorption membrane+ MEA	80%	80%	80%	80%
	Absorption	99.2%	99.4%	99.8%	96%
	Adsorption PSA, TSA	75%	50%	60%	97%
	Cryogenic technique			97%	99%
	Separation membrane	55%	16%	30%	97%
	Absorption membrane+ MEA	55%	16%	30%	97%
Note: Based on an IEA study to demonstrate costs of the possible removal options which might be considered in the future					
Note 2: Integrated gasification combined cycle using solvents may lead to 36% efficiency instead of 42% as a reference					
Note 3: The decrease in efficiency for combined cycles is much higher due to high compression requirements					

Table 2: Comparison of different combustion techniques with and without CO2 removal [41, IEA, 1992]

Performance parameter	CO2 removal technique	Pulverised coal power plant with FGD	Gas turbine combined cycle	Integrated gasification combined cycle	Pulverised coal combustion in oxygen using recycled CO2
Cost per tonne of CO2 avoided (EUR/tonne)	Absorption	40	63	99	18
	Adsorption PSA	96	623	235	24
	Adsorption TSA	302	459	413	-
	Cryogenic technique	NA	NA	26	29
	Separation membrane	53	384	143	10
	Absorption membrane+ MEA	51	35	48	18
Specific investment cost (EUR/kW)	Reference case without CO2 removal	1213	805	1790	2344
	Absorption	2112	1567	3731	3557
	Adsorption PSA	1569	1376	2465	2510
	Adsorption TSA	2363	1779	3475	-
	Cryogenic technique	-	-	2763	4125
	Separation membrane	2411	3573	5567	2537
	Absorption membrane+ MEA	1885	-	3137	-
Note: Based on an IEA study to demonstrate costs of the possible removal options which might be considered in the future					

Table 3: Comparison of different combustion techniques with and without CO2 removal [41, IEA, 1992]

The disposal of carbon dioxide is an integral part of a complete strategy to avoid major greenhouse gas emissions from plants-generating energy in the future, but it is not an immediate operational issue for a large combustion plant and is, therefore, not described in this document.

However, possible options for large-scale disposal of CO₂ are:

- disposal in the deep ocean
- disposal in deep aquifers
- use for enhanced oil recovery (EOR)
- disposal in exhausted gas and oil reservoirs.

10 Appendix 1 - References

- [1] EC Reference Document on Best Available Techniques for Large Combustion Plants July 2006
- [2] Electrabel (1996). "Combined Heat and Power. A way towards rational use of energy".
- [3] Korobitsyn, M. A. (1998). "New and Advanced Energy Conversion Technologies. Analysis of Cogeneration, Combined and Integrated Cycles".
- [4] El-Wakil, M. M. (1984). "Power plant Technology".
- [5] Cortés, V. J. and Plumed, A. (2000). "Principles and Concepts of Combustion Gasification and Thermodynamic Cycles".
- [6] Çengel, Y. A. and Boles, M. A. (1994). "Thermodynamics, An Engineering Approach".
- [7] Rogers, G. F. C. and Mayhew, Y. R. (1967). "Engineering Thermodynamics, Work and Heat Transfer".
- [8] Eurelectric (2001). "EURELECTRIC proposal for a Best Available Techniques Reference Document for Large Combustion Plants".
- [9] Siemens (2000). "Benson boilers for maximum cost-effectiveness in power plants", Power Generation Group (KWU).
- [10] COGEN Europe (1999). "A Guide to Cogeneration".
- [11] Umweltschutztechnik I - Kapitel VI - Rationeller Energieeinsatz - Prof. Dr. techn. R. Leithner Institut für Wärme und Brennstofftechnik TU Braunschweig, Wintersemester 2004/2005
- [12] IEA (1992). "Carbon dioxide capture from power stations".
- [13] Soria, A.; Schrattenholzer, L. and Akutsu, N. (1998). "Energy Technology Strategy 1995 - 2030: Opportunities arising from the threat of climate change".
- [14] Information at www.Keveney.com
- [15] Energieflussbild Deutschland zum Jahr 1995. Quelle: RWE