Advanced CSP Teaching Materials

Chapter 4
Thermal Power Plants

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# Nomenclature

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<th>Meaning</th>
<th>Unit</th>
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<tr>
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<tr>
<td>$\vec{B}$</td>
<td>magnetic field</td>
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<tr>
<td>$c_p$</td>
<td>specific heat at constant pressure</td>
<td>J/(kg K)</td>
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<td>$c_v$</td>
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<td>$q$</td>
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<td>$Q_C$</td>
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<td>$T$</td>
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<td>$\Delta$</td>
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<tr>
<td>$\eta$</td>
<td>efficiency</td>
<td>-</td>
</tr>
<tr>
<td>$\vartheta$</td>
<td>polar wheel angle</td>
<td></td>
</tr>
<tr>
<td>$\mu$</td>
<td>cooling tower efficiency</td>
<td>-</td>
</tr>
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<td>$\nu_s$</td>
<td>synchronous rotational frequency of rotator</td>
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<td>$\phi$</td>
<td>constant relative humidity curves</td>
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<tr>
<td></td>
<td>angular velocity</td>
<td>rad/s</td>
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Subscripts

da  air
atm  atmosphere
c  centripetal
comp  compressor
e  electrical
f  liquid
in  input
int  internally
L  Lorentz
max  maximum
out  output
rev  reversible
th  thermal
turb  turbine
v  vapour
wb  wet-bulb

Overbar
→  vectorial

Acronyms
AC  alternating current
C  compressor
CC  combustion chamber
DC  direct-cooled
CSP  Concentrating Solar Power
HTF  heat transfer fluid
G  generator
HRSG  heat recovery steam generator
ISCCS  Integrated Solar Combined Cycle Systems
kW  kilowatt
kWh  kilowatt hour
M  motor
MVA  megavolt ampere
MW  megawatt
ppb  parts per billion
ppm  parts per million
r  reaction
T  turbine
Summary

This chapter provides the fundamentals of power plant theory and technology for CSP plant applications. Three well known power cycles which are available for electricity and/or heat generation are introduced. Following this, many conventional power plant components are described as well as special designs for the application in CSP plants.

4 Thermal Power Plants

Key questions

- What power cycles are available for CSP plants?
- What components are found in steam and gas-turbine power plants and how do they work?
- What cooling systems are available for CSP plants?
- Is water treatment important in power plants?
- How does a steam turbine work? What types are there?
- How does a steam generator (boiler) work? What types are there?
- How does a gas turbine work? What types are there? What are the common combustor types used in Europe and USA?
- How does an electric generator work?
4.1 Power Cycles

There are several different power cycles available for electricity and/or heat generation. The most known cycles are the steam Rankine cycle, Brayton cycle and the Stirling cycle. All these cycles are used in CSP applications. The Rankine cycle is required whenever a CSP plant is used for generating steam (e.g. solar tower, linear Fresnel and parabolic trough technology), a Brayton cycle is required if a gas is expanded in a gas turbine (e.g. application in a hybridised parabolic trough power plant) and a Stirling cycle describes the thermodynamic processes of the Dish technology. To begin with, the well-known Carnot cycle is introduced which is a reversible cycle. Reversible cycles in general provide upper limits on the performance of real cycles.

4.1.1 Carnot Cycle

The Carnot cycle is probably the most-widely known reversible cycle. The Carnot cycle consists of four reversible processes, of which two are isothermal and two are adiabatic. The Carnot cycle can be executed either in a closed or a steady-flow system. Such a closed system can, for example, consist of a gas contained in an adiabatic piston-cylinder device [1, p. 247]. A $p$-$v$ diagram for a Carnot gas power cycle is shown in Figure 1 and the piston-cylinder assembly with details of the processes taking place inside is shown in Figure 2. As illustrated in Figure 2, the wiggly arrows show the direction of the heat transfer. Furthermore, there is an insulating stand as well as a hot and a cold reservoir. The temperature of the hot reservoir is denoted with $T_H$ and the temperature of the cold reservoir is denoted with $T_C$. The piston and cylinder walls are nonconducting. The four processes are as follows, with the initial condition being that the piston-cylinder assembly is on the insulating stand and the cycle begins from state 1 [2, p. 239]:

- **Process 1 → 2:** An adiabatic compression of the gas takes place until state 2 is reached where the temperature is $T_H$.
- **Process 2 → 3:** The piston-cylinder assembly is in close contact with the hot reservoir at temperature $T_H$. Isothermal expansion of the gas takes place while energy $Q_H$ is passed from the hot reservoir to the gas by heat transfer.
- **Process 3 → 4:** The piston-cylinder assembly is again placed on the insulating stand and the gas is being expanded adiabatically to state 4 which is at temperature $T_C$.
- **Process 4 → 1:** The piston-cylinder assembly is in close contact with the cold reservoir at temperature $T_C$. Isothermal compression of the gas takes place until state 1 is reached. During the compression, the gas discharges energy $Q_C$ to the cold reservoir by heat transfer.
For process \(2 \rightarrow 3\) to be reversible during the heat transfer, the temperature difference between the gas temperature and the hot reservoir’s temperature must be vanishingly small. Both the temperature of the hot reservoir and the cold reservoir remains constant, meaning that the temperature of the gas also remains constant during the processes \(2 \rightarrow 3\) and \(4 \rightarrow 1\).

The \(p-v\) diagram shown in Figure 1 also gives information about the work done by each of the internally reversible processes, which is represented by an area:

- The area under the adiabatic process line \(1-2\) represents the work done per unit of mass to compress the gas in this process.
- The areas under process lines \(2-3\) and \(3-4\) represent the work done per unit of mass by the gas as it expands in these processes.
• The area under process line 4-1 is the work done per unit of mass to compress the gas in this process

The coloured area within the four process lines is the net work developed by the cycle per unit of mass [2, p. 239].

The Carnot cycle not only applied to processes of closed systems as was presented with the piston-cylinder assembly. A schematic and the accompanying $p$-$v$ diagram of a Carnot cycle executed by water steadily circulating through a series of four interconnected components that has features in common with a simple steam power plant is shown in Figure 3. The water undergoes a phase change from liquid to vapour at constant temperature $T_H$ as it passes through the boiler as a result of heat transfer from the hot reservoir. Due to the fact that the temperature remains constant, the pressure also remains constant during the phase change. The steam which is generated by the boiler is then expanded adiabatically by the steam turbine and work is being developed. As the steam is expanded, the pressure of the steam drops and so does the steam temperature. When the steam exits the steam turbine, the steam has temperature of the cold reservoir $T_C$, at which it enters the condenser. In the condenser, heat is transferred from the steam to the cold reservoir and some of the steam condenses at constant temperature $T_C$. In this process, the temperature of the cold reservoir $T_C$ remains constant and so does the pressure. In the last process to complete the cycle, a pump, or compressor, adiabatically pumps the two-phase liquid-steam mixture obtained from the condenser to the boiler entrance (state 4). In this process (i.e. process 3-4), a work input is required to increase the pressure. As the pressure rises, the temperature also increases, namely from $T_C$ to $T_H$ [2, p. 240]

![Figure 3: Carnot steam cycle](image)

In the examples given so far, only $p$-$v$ diagrams illustrating the Carnot cycle have been shown. Figure 4 shows the Carnot cycle in a $T$-$s$ diagram.
The Carnot efficiency is the thermal efficiency of a system undergoing a reversible power cycle while operating between thermal reservoirs at temperatures $T_H$ and $T_C$, and can be determined according to equation (1):

$$\eta_{\text{max}} = 1 - \frac{T_C}{T_H}$$  \hspace{1cm} (1)

When interpreting equation (1) it becomes evident that the Carnot efficiency increases when $T_H$ increases and/or $T_C$ decreases [2, p. 234].

A plot of the Carnot efficiency versus the temperature of the hot reservoir ($T_H$) for a fixed temperature of the cold reservoir ($T_C = 298$ K) is shown in Figure 5. The temperature of the cold reservoir ($T_C = 298$ K) for constructing the plot was chosen in recognition that in actual power cycles, energy is discharged by heat transfer at about the ambient temperature or the temperature of the cooling water from a nearby river or lake [2, p. 234].
For steam power plants, for example, the Carnot cycle poses several impracticalities. An explanation for this will be given in a comparison of the Rankine cycle with the Carnot cycle in the following section about the Rankine cycle.

4.1.2 Rankine Cycle

This section discusses the Rankine steam cycle and compares it to the Carnot cycle. Furthermore, possibilities of improving the thermal efficiency of the basic Rankine cycle are presented, irreversibilities are discussed and two important diagrams used in thermodynamics are briefly introduced.

(1) Rankine Cycle

Steam power cycles are based on the so-called ideal Rankine cycle as shown in Figure 6 in its simplest form (1-2-3-4-1). Considering the neglecting of irreversibilities, the four internally reversible processes are [2, p. 395]:

Process 1 → 2: Isentropic expansion of the working fluid in the steam turbine from saturated vapour pressure to the condenser pressure

Process 2 → 3: Heat rejection by the working fluid in a condenser at constant pressure. At state 3 saturated liquid is obtained

Process 3 → 4: Isentropic compression of the working fluid by the pump

Process 4 → 1: Heat addition for the evaporation of the working fluid in a boiler at constant working fluid pressure
Figure 6: T-s Diagram of an ideal Rankine cycle [2, p. 395]

Figure 7 shows the components in a power plant that are associated with the Rankine cycle (the numbering system is adapted to the above stated processes).

Figure 7: Components associated to the Rankine cycle processes, edited from [1, p. 378]

Comparison of the ideal Rankine cycle with the Carnot cycle
Figure 8 shows the ideal Rankine cycle (1-2-3-4-4′-1) and the Carnot cycle (1-2-3′-4′-1) together in one diagram for the purpose of illustration. Both cycles have the same maximum temperature $T_H$ and minimum temperature $T_C$. The Rankine cycle, however, has a lower thermal efficiency than the
Carnot cycle. This is easily understood because in the Rankine cycle the average temperature between 4 and 4’ is less than $T_H$. Although the Carnot cycle has a better thermal efficiency it has several impracticalities as a model for the simple steam power cycle. In conventional steam power plants, hot products from the combustion of fuel provide the heat which is transferred to the working fluid in a boiler. This takes place at approximately constant pressure. It is generally desired to exploit fully the energy which is released by combustion. To do so, the hot products must transfer as much of their heat to the working fluid. In other words, the hot products should be cooled as much as possible. In the Rankine cycle, the first portion of the heating process is realised by cooling the combustion products below the maximum temperature $T_H$, as illustrated by process (4-4’). This is totally different to the Carnot cycle where the combustion products would be cooled at the most to $T_H$ and not lower. This means that the Carnot cycle only a smaller portion of the energy from the combustion products. Another problem associated with the Carnot cycle poses the two-phase liquid-vapour mixture at state 3’ which needs to be pumped. Significant practical problems exist in developing pumps that can handle two-phase liquid-vapour mixtures. The Rankine cycle does not encounter this problem as here the steam from the steam turbine’s exhaust is condensed completely and pumps have no problem with handling a liquid. The processes of pumping from 3-4 and to heat at constant-pressure without work from 4-4’ can be closely achieved in practice [2, p. 401].

**Figure 8:** Illustration for comparing the ideal Rankine cycle with the Carnot cycle [2, p. 401]

*Thermal efficiency improvements*

The thermal efficiency of the described basic Rankine cycle can be improved by

- increasing the boiler pressure
- reheating
- superheating the steam to high temperatures
• regenerative heating of feedwater
• lowering the condensers pressure

The basic idea behind these measures are to 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. For example, lowering the condensers pressure leads to a lower temperature $T_{C,average}$ [1, p. 384].

For the first three cases (increasing the boiler pressure, reheating and superheating the steam to high temperatures), the result is a higher average steam temperature. Remembering definition of the Carnot efficiency, the thermal efficiency is then higher. Some of the above mentioned improvements are described in more detail including also a reheat-regenerative cycle, which combines both the reheating of steam and the regenerative heating of feedwater.

**Superheating**

Figure 6 illustrates the possibility of superheating the steam, as shown in cycle 1’–2’–3–4–1’ [2, p. 395]. In power plants it is generally required to superheat and possibly also to reheat the steam. Superheating is necessary due to a mechanical reason, but there are also several thermodynamical advantages associated with it. If not superheated adequately, condensate can form in the piping and turbine. Liquid droplets contained in steam are a great problem for steam turbine blades as they cause erosion and are a major contributor to wear. Hence the live steam must have a steam quality greater than 90% in the operation of power plants (with exception of nuclear power plants where special steam turbines can handle lower quality steam) [1, p. 377]. It is important that the steam quality remains high throughout the steam passages through the turbine stages. Figure 6 shows the expansion of saturated steam from state 1 – 2 and of superheated steam from state 1’ – 2’. It is clear that the steam quality is higher at exhaust state 2’ than at state 2 and that superheating steam reduces the risk of obtaining low steam quality at the turbine exhaust [2, p. 406]. Moreover, superheating the steam multiplies the steam volume (depending on the temperature) at constant pressure and raises the efficiency of the steam power cycle [3, p. 19].

**Reheating**

The material properties of the steam cycle components set a limit to the maximal permissible steam temperature and therefore reheating is an adequate solution to further reduce the risk of obtaining wet steam at the turbine exit. Moreover, reheating of steam, as shown in Figure 9, is also a common method of increasing the efficiency of a steam power cycle as the average temperature during the heating process is increased. In a reheat cycle the steam is not expanded to the condenser pressure in a single stage, rather it is expanded to a pressure level between the boiler and condenser pressures in a first-stage high-pressure turbine (process 1–2). The steam is then directed back to the steam generator and reheated to temperature $T_3$ (process 2–3). Then, the steam is expanded to condenser pressure (process 3–4) in a second-stage low-pressure turbine. The main advantages of reheating are that the efficiency of the steam cycle and also the quality of the steam at the turbine exhaust are increased. The improvement of the steam quality by reheating is shown in the subsequent T-s diagram (Figure 9). State 4 shows the steam quality at the exhaust of the turbine of the reheat cycle. State 4’ shows that when no reheat cycle is used, the steam quality is lower than in state 4 [2, p. 406].
The average temperature during the heating process can be increased by increasing the number of expansion and reheat stages, as shown in Figure 10. The more reheat stages are used, the more the expansion and reheat processes come close to an isothermal process at the maximum temperature. However, using more than two reheat stages is not practical [1, p. 389].

**Figure 9:** Ideal Rankine cycle with reheating [2, 406]

**Figure 10:** Multiple reheat stages increase the average temperature at which heat is transferred [1, p. 389]

*Regeneration*

Regeneration is the regenerative heating of feedwater, which can be realised with open or closed feedwater heaters. The open feedwater heater is a direct contact-type heat exchanger. The following explanation of a regenerative Rankine cycle with one open feedwater heater is given with reference to Figure 11.
The steam, at state 1, enters the first-stage turbine and is expanded to state 2. At state 2, a fraction of the total steam flow is extracted, or bled, into an open feedwater heater. The feedwater heater operates at the extraction pressure $p_2$. The remaining steam is expanded through the second-stage turbine to state 3. Heat removal occurs in the condenser in process 3-4 and the steam is condensed to saturated liquid, state 4. In process 4-5, the saturated liquid is then pumped to the extraction pressure so that it can be fed into the feedwater heater. Then, a single mixed stream exits the feedwater heater at state 6. In the example shown in Figure 11, the mass flow rates of the streams entering the feedwater heater are chosen so that the stream exiting the feedwater heater is a saturated liquid at the extraction pressure. The pressure of the saturated liquid is then raised as it is pumped to the steam generator pressure (process 6-7). At state 7, the saturated liquid enters the steam generator. In the steam generator the working fluid is heated from state 7 to state 1.

Figure 11 shows also a T-s diagram on which the following discussion is based. Because of the used regeneration, heat addition is required only from state 7 to state 1. If no regeneration were used heat addition would take place from state a to state 1. Making use of regeneration lowers the energy required for the evaporation and superheating of the steam, meaning that the fossil fuel or another energy source requirement is lowered, which is the aim of the regenerative Rankine cycle. As mentioned, only a portion of the steam flows is expanded in the second-stage turbine (process 2-3), which results in less work being developed. “In practice, operating conditions are chosen so that the reduction in heat added more than offsets the decrease in the net work developed, resulting in an increased thermal efficiency in regenerative power plants” [2, pp.412-413].

As mentioned earlier, regenerative feedwater heating can also be realised with closed feedwater heaters. A regenerative Rankine cycle with one closed feedwater heater and a corresponding T-s diagram is shown in Figure 12. The closed feedwater heaters are shell-and-tube recuperators. The feedwater flows through the tubes. The steam extracted from the steam turbine after the first-stage condenses on the outside of these tubes. The feedwater temperature is thereby increased. The following explains the T-s diagram:
Initially at state 1, the total flow is expanded in the first-stage turbine to state 2. Then, a portion of the steam flow is bled into the closed feedwater heater, which condenses the steam. The saturated liquid then exits the feedwater heater at the same pressure as the extracted steam at state 7. The saturated liquid, now called condensate, is then trapped into the condenser. As mentioned, only a portion of the steam was extracted from the steam turbine after the first-stage. The rest is expanded in the second-stage turbine and is then condensed in the condenser. In the condenser, the initially separated flows are reunited again. The expansion from state 7 to state 8 through the trap is irreversible. This is shown by a dashed line on the T-s diagram. All the condensate at state 4 is then pumped to the steam generator pressure and enters the feedwater heater at state 5. The condensate increases the temperature of the feedwater, and the feedwater exits at state 6. The feedwater is then pumped through the steam generator. The working fluid is heated in the steam generator at constant pressure from state 6 to state 1. The cycle is now complete. Although the closed heater shown in Figure 12 operates with no pressure drop in either stream, there is a source of irreversibility due to the stream-to-stream temperature difference [2, pp. 416-417].

![Figure 12: Regenerative Rankine cycle with one closed feedwater heater][2, p. 417]

**Reheat-regenerative cycle**

A reheat-regenerative cycle with two feedwater heaters (one open and one closed heater) is shown in Figure 13 together with the corresponding T-s diagram and example values for pressures and temperatures at the different states.
(2) Irreversibilities
The compression and expansion processes as shown, for example, in Figure 9 are reversible and therefore idealised. In reversible processes the change in entropy is zero ($\Delta s = 0$); in actual cycles, as shown in Figure 14 (left illustration), irreversibilities are involved (shown by the dashed lines in $T$-$s$ diagram) and lead to an increase of entropy greater than zero ($\Delta s > 0$). The compression and expansion process is therefore no longer at constant entropy.

Two common sources of irreversibilities are fluid friction and heat loss to the surroundings. Fluid friction causes pressure drops in the boiler, the condenser, and the piping between various components. For example, the pressure at the turbine inlet is lower than at the boiler exit as a result of the pressure drop in the connecting piping. The pressure drop in the condenser is usually very small. These pressure drops are compensated by pumping the water to a sufficiently higher pressure than would be required by the ideal cycle. Thus, a larger pump and larger work input to the pump is needed. As mentioned before, another common source of irreversibility is heat loss to the surroundings. This, in particular, refers to the heat loss of the steam which flows through various components. Due to these undesired heat losses more heat must be transferred to the steam in the boiler so that the level of net work output is kept the same. This leads to a decrease in cycle efficiency [1, p. 382].

Irreversibilities are also occurring in the pump and the steam turbine and these are of particular importance. Due to the irreversibilities, a pump will need a greater work input, and a steam turbine produces a smaller work output. If no irreversibilities exist, the flow through a pump and a steam
turbine is isentropic. This is shown in Figure 14 (right illustration), where states 2a and 4a are the actual exit states of the pump and the steam turbine, respectively [1, pp. 381-382]

Figure 14: Deviation of actual steam power cycle from the ideal Rankine cycle (left); The effect of pump and turbine irreversibilities on the ideal Rankine cycle (right) [1, p. 382]

(3) Diagrams
In thermodynamics, the two most commonly used diagrams for the Second Law analysis are the temperature-entropy ($T$-$s$) and enthalpy-entropy ($h$-$s$) diagrams. The following illustration shows both diagrams.

Figure 15: T-s diagram (left) and h-s diagram (right) [2, p.260]
The \( T-s \) diagram is a very common diagram which is used for the analysis of energy transfer system cycles. In a \( T-s \) diagram the heat \( Q \) transferred during an internally reversible process is given by the area under the respective process curve, i.e.

\[
Q_{\text{int,rev}} = \int_{1}^{2} T \, dS
\]  

(2)
as shown in Figure 16 below [1, p. 286].

![Figure 16: Heat transferred in internally reversible processes, depicted in the T-s diagram [1, p. 286]](image)

The \( h-s \) diagram for water, also called Mollier chart (named after the German scientist Richard Mollier) [1, p. 287], was created for evaluating properties at superheated vapour states and for two-phase liquid-vapour mixtures. The chart shows lines of constant steam quality\(^1\) in the two-phase liquid-vapour region and the superheated vapour states as well as constant temperature and constant pressure lines [2, p. 260].

**Exercises**

- What are the 4 processes of the Carnot cycle?
- What are the 4 processes of the ideal Rankine cycle?
- Describe the methods of improving the thermal efficiency of a Rankine cycle.
- Write the formula for calculating the Carnot efficiency.
- Why are water droplets damaging steam turbine blades?

\(^1\) The steam quality, denoted with \( x \), is defined as the mass of steam present per unit mass of steam-water mixture [4]. The value of \( x \) lies within \( 0 - 1 \), where 0 represents 0% steam (i.e. 100% liquid) and 1 represents 100% saturated steam.
4.1.3 Brayton Cycle

The Brayton cycle is defined as an ideal gas turbine cycle and serves as a reference cycle. Dissipation and friction losses, and pressure drops in heat exchangers or combustion chamber are neglected.

(1) Processes

Gas turbines commonly operate in open cycles: ambient air is continuously sucked in by the compressor, which compresses it before it enters a combustion chamber. Inside the combustion chamber the air is mixed with fuel and the mix is then combusted. The combustion products reach elevated temperatures and are expanded in a gas turbine. In a last step the combustion products are discharged to the surroundings. Not all of the turbine work can be used for electricity generation (or other purposes) – a portion must be used for driving the compressor. The ratio of the compressor work to the turbine work is called back work ratio [2, p. 461].

The open cycle as just described can be modelled as a closed cycle by utilising the air-standard assumptions.

![Figure 17: Simple gas turbine as open cycle (left) and closed cycle (right) design, edited from [2, pg. 461]](image)

The closed Brayton cycle consists of the following four processes [1, p. 368]:

- Isentropic compression in the compressor (1-2)
- Isobaric addition of heat from combustion process (2-3)
- Isentropic expansion in the turbine (3-4)
- Isobaric removal of heat (4-1)

The term isobaric defines a condition of constant pressure in a system.

The thermodynamics of each process shall now be discussed in more detail using the T-s diagram of Figure 18, as a reference. The figure shows example values of temperatures and pressures. The
pressure ratio $r_p$ is defined as the ratio of the pressure of the working fluid at the exit of the gas turbine compressor $p_2$ to the pressure of the working fluid at the entry of the gas turbine compressor $p_1$.

$$r_p = \frac{p_2}{p_1} \quad (3)$$

In the diagram a pressure ratio $r_p = 8$ is shown. For an open Brayton cycle, the pressure $p_1$ is the atmospheric pressure.

![T-s Diagram for an ideal Brayton cycle](https://via.placeholder.com/150)

**Figure 18:** T-s Diagram for an ideal Brayton cycle [1, p. 371]

**a) Isentropic compression by the compressor (1-2)**

The task of the compressor is to compress the air (raise the air pressure) before it enters the combustion chamber. The specific compression work input, $w_{\text{comp,in}}$, is determined by the specific enthalpies of states 1 and 2:

$$w_{\text{comp,in}} = h_2 - h_1 = c_p(T_2 - T_1) \quad (4)$$

The T-s diagram shows that the temperature increases together with the pressure during the compression process. There is no change in entropy as neither heat is supplied nor removed, nor dissipation occurs, hence the term *isentropic compression* [1, p.372].

**b) Isobaric addition of heat from combustion process (2-3)**

In the ideal cycle, the combustion of the gas in the combustion chamber takes place at constant pressure (isobaric). The T-s diagram illustrates how the entropy and the temperature of the working fluid have been increased during the addition of heat in the combustion chamber [1, p.372].

The specific heat input, $q_{\text{in}}$, is determined by the specific enthalpies:
\[ q_{in} = h_3 - h_2 = c_p(T_3 - T_2) \]

where \( c_p \) is the specific heat at constant pressure, J/(kg K).

c) Isentropic expansion (3-4)
The expansion process takes place at constant entropy (isentropic). The pressure and temperature decreases during the expansion from state 3 to 4. The specific work output, \( w_{\text{turb,out}} \), of the turbine is determined by the specific enthalpies [1, p.372]:
\[ w_{\text{turb,out}} = h_4 - h_3 = c_p(T_4 - T_3) \]

\( d) \) Isobaric heat removal (4-1)
The heat removal, \( q_{out} \), takes place at constant pressure and leads to a reduction in the specific volume. The temperature of the working fluid is decreased and the entropy is reduced to that of state 1, as shown in the T-s diagram [1, p.372].
\[ q_{out} = h_1 - h_4 = c_p(T_1 - T_4) \]

(2) Thermal efficiency of the Brayton cycle

The formulae for calculating the thermal efficiency, \( \eta_{th,Brayton} \), of the cycle are given below [1, pp. 368, 372]:
\[ \eta_{th,Brayton} = \frac{w_{\text{net}}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{c_p(T_4 - T_1)}{c_p(T_3 - T_2)} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)} \]

where \( w_{\text{net}} = w_{\text{turb,out}} - w_{\text{comp,in}} \)

Processes 1-2 and 3-4 are isentropic, and \( p_2 = p_3 \) and \( p_4 = p_1 \). This leads to
\[ \frac{T_2}{T_1} = \left( \frac{p_2}{p_1} \right)^{(k-1)/k} = \left( \frac{p_3}{p_4} \right)^{(k-1)/k} = \frac{T_3}{T_4} \]

Substituting these equations into the thermal efficiency relation and simplifying gives
\[ \eta_{th,Brayton} = 1 - \frac{1}{r_p^{(k-1)/k}} \]

where \( k \) is the specific heat ratio [-] of the working fluid, i.e. the ratio of the specific heat at constant pressure, \( c_p \), divided by the specific heat at constant volume, \( c_v \):
\[ k = \frac{c_p}{c_v} \]
(3) **Real cycle**

Figure 19 shows a diagram illustrating the deviation of an actual gas-turbine cycle from the ideal Brayton cycle as a result of irreversibilities. Some pressure drop occurs during the processes of heat addition and heat rejection, which is inevitable. More importantly, due to the irreversibilities, the actual work input to the compressor will be more, and the actual work output from the turbine will less [1, pp. 372-373]. The deviation of an actual gas-turbine cycle from the ideal Brayton cycle due to the irreversibilities is shown by the dashed lines in Figure 19. In the real cycle, irreversibilities cause an increase in entropy in the compression (1-2a) and expansion (3-4a) processes. Processes (1-2s) and (3-4s) represent the isentropic processes of an ideal Brayton cycle.

![Figure 19: Deviation of an actual gas-turbine cycle from the ideal Brayton cycle as a result of irreversibilities [1, p. 373]](image)

**Exercises**

- The closed Brayton cycle consists of 4 processes. Name all 4 processes.
4.1.4 Stirling Cycle

In this section we shall explain the principle of operation of the Stirling engine and present a functional description of different constructive solutions. Stirling engines are engines that approximate the Stirling cycle. They convert thermal energy in mechanical energy. As they work with external heat supply, they can be operated with many different kinds and qualities of heat sources. In particular, they are appropriate to be driven by (concentrated) solar radiation.

History
The Stirling engine was invented in 1816 by Robert Stirling. It was the result of the search of a heat engine that would be a safer alternative to the steam engine, which at that time represented a security problem because their boilers frequently exploded causing many injuries and fatalities. However, only few Stirling engines were built and they still were prone to failures, so that they did not always satisfy the expectations of their constructors. In particular, they finally could not compete with steam engines. The first solar application of record was by the British-American inventor John Ericsson in 1872. Small non-industrial applications were found in the second half of the 19th century and during the 20th century. In the latter, Philips produced Stirling engines to operate their radios in areas without central electricity supply. The Stirling engines were used as electricity producers that were easily to handle and transportable. Recently, the Stirling engine has gained more attention especially because of realised and planned solar applications and because of their application in smaller cogeneration units.

![Figure 20: Philips Stirling engine [5]](image)

Stirling Cycle
The Stirling cycle is a thermodynamic cycle in which thermal energy is transformed into mechanical energy. As in other such cycle processes, a working gas is compressed at lower temperatures and expanded at higher temperatures. As the expansion at higher temperatures releases more mechanical energy than is needed for the compression at lower temperatures, the total cycle releases a surplus of mechanical energy, such that a net conversion of heat into mechanical work takes place. Besides the isothermal compression and expansion at different temperature levels, the Stirling cycle includes isochoric heating and cooling that are realised in
order to change from one temperature level to the other. So, the cycle consists of the following four processes:

- Process 1-2: Isothermal compression of the working gas under heat release at low temperature.
- Process 2-3: Isochoric heating of the working gas.
- Process 3-4: Isothermal expansion of the working gas under heat absorption at high temperature.
- Process 4-1: Isochoric cooling of the working gas.

The ideal cycle (all processes are reversible, i.e. there is no dissipation) is represented in the following pressure-volume and entropy-temperature diagrams.

![Figure 21: Ideal Stirling process in the p-V and T-s diagram](image)

The energy balance of the working gas over the cycle is the following: the working gas receives thermal energy at the processes 3-4 and 2-3, mechanical work is done on it at the process 1-2, it releases thermal energy at the processes 1-2 and 4-1 and it does mechanical work on its surroundings at the process 3-4. According to the First Law of Thermodynamics, the balance is zero:²

$$Q_{34} + Q_{23} + W_{12} - |Q_{12} + Q_{41} + W_{34}| = 0$$  \hspace{1cm} (13)

If we change the system boundaries and do not consider the working gas as the system, but the Stirling engine (in this case an ideal one, which – what is impossible, as we will see – realises the ideal Stirling cycle as represented in the diagrams), then (13) can be reduced to:

² In the following equation, the values are considered as positive or negative depending on whether it is energy supplied to the system (positive) or energy released by the system (negative). The system in this case is the working gas. In the following this convention is always respected. Only when we speak of the “released work” we refer to the amount, which is always positive (the negative sign is already expressed, so to speak, by the attribute “released”).
\[ Q_{34} + W_{12} - |Q_{12} + W_{34}| = 0 \] (14)

The reduction consists in omitting the heat absorbed in the isochoric heating (process 2-3) and released in the isochoric cooling (process 4-1). The heat amount needed in these two processes is exactly the same. It may remain within the Stirling engine, changing its bearer within the engine. Hence, this heat amount has to be omitted in the energy balance if we take the whole engine as the system under consideration and if we choose an engine that retains the heat that is exchanged in the two isochoric processes within its boundaries. As we will see below, it is the regenerator, which has the specific task to maintain this amount of energy within the engine.

The amount of released usable mechanical energy is the difference of the amount of mechanical energy that is released in the process 3-4 and the mechanical energy supplied in the process 1-2. According to (14), it is equal to the difference between the supplied thermal energy in the process 3-4 and the amount of the released thermal energy in the process 1-2:

\[ W = |Q_{34} - W_{12}| = |Q_{34} - |Q_{12}| = 0 \] (15)

\( W \) is represented in the two diagrams as the area between the four points.

The efficiency of the cycle is determined as the ratio of the released mechanical energy \( W \) to the supplied thermal energy (at process 3-4):

\[ \eta = \frac{W}{Q_{34}} \] (16)

which in the case of the ideal cycle amounts to:

\[ \eta = 1 - \frac{T_L}{T_H} \] (17)

This is, according to Carnot’s theorem, the highest possible efficiency for any heat engine.

Now, let’s have a look at the released work:

\[ W = Q_{34} - |Q_{12}| = Q_{34} + Q_{12} \] (18)

The heat absorbed or released in an isothermal process is equal to the work that the gas does or that the surroundings do on the gas:

\[ Q_{34} = -W_{34} \text{ and } Q_{12} = -W_{12} \] (19)

Taking into consideration the ideal gas equation \( pV = mRT \) and considering that \( W = \int pdV \), the work in the isothermal processes is determined as follows:

\[ W_{12} = \int_{V_1}^{V_2} pdV = \int_{V_1}^{V_2} \frac{mRT}{V} dV = mRT_L \ln\left(\frac{V_2}{V_1}\right) \] (20)

\[ W_{34} = \int_{V_3}^{V_4} pdV = \int_{V_3}^{V_4} \frac{mRT_H}{V} dV = mRT_H \ln\left(\frac{V_4}{V_3}\right) \] (21)

Taking into account that \( V_1 = V_4 = V_{1,4} \) and \( V_2 = V_3 = V_{2,3} \) the total work is:

\[ W = Q_{34} + Q_{12} = mRT_H \ln\left(\frac{V_4}{V_3}\right) + mRT_L \ln\left(\frac{V_2}{V_1}\right) \]

\[ = mR(T_H - T_L) \ln\left(\frac{V_{1,4}}{V_{2,3}}\right) = mR(T_H - T_L) \ln\left(\frac{V_1}{V_2}\right) \]
We see that the generated mechanical work depends on the temperature difference \((T_H - T_L)\), on the compression ratio \(\frac{V_{1,4}}{V_{2,3}}\) and on the gas mass. As gas mass, volume and temperature are related to pressure, in these dependencies is included also the dependency on pressure.

\[
W = mR(T_H - T_L) \ln\left(\frac{V_{1,4}}{V_{2,3}}\right)
\]  

(22)

**Stirling Engine:**

The Stirling cycle is technically approximated in the Stirling engine. Stirling engines work with external heat supply. They do not need internal combustion and they can be driven by many kinds of heat sources. They can use low quality fuels and they can use heat sources that do not depend on any combustion. Contrary to internal combustion heat engines, the working gas is isolated from the heat source. That’s why the working gas can be maintained always within the engine and need not be changed between the cycles.

The fact that Stirling engines are driven by external heat supply makes them appropriate for solar thermal applications. Furthermore, as they can combine different heat sources in one application, they are also appropriate for hybrid operation.

External heat supply has further advantages compared to internal combustion engines: First, Stirling engines tend to be more reliable with lower maintenance requirements and, second, their operation is quieter. On the other hand, external heat supply implies also that Stirling engines need efficient heat exchangers in order to transport thermal energy over the engine boundaries.

A particular component of many Stirling engines is the regenerator, which stores and releases thermal energy periodically. An ideal regenerator stores the heat that is released by the working gas in the process 4-1 (isochoric cooling) and it gives the same amount of heat back to the working gas in the process 2-3 (isochoric heating). It is an optional component that raises considerably the efficiency of the engine maintaining heat within the system that otherwise would be exchanged with the environment and thereby raise the heat flow from the high temperature reservoir to the low temperature reservoir without any additional gain of mechanical work.

**Approximation of the Stirling cycle in Stirling engines**

The following figure represents the four processes of the cycle illustrating also the role of the regenerator, which has a high temperature at stages 1 and 2, in which it stores thermal energy, and a low temperature at stages 3 and 4, in which the working gas contains the thermal energy that is stored in the regenerator in the other stages.
This figure is still an idealised representation of the working of a Stirling engine. Real engines approximate this schema, but they do not reach it completely.

There are different constructive solutions of Stirling engines, which have the objective to realise more or less exactly the Stirling cycle. Generally they have two pistons in order to be able to carry out compression and expansion as well as (approximate) isochoric cooling and heating through gas displacement.

We can distinguish between kinematic and free-piston Stirling engines. The pistons in a kinematic Stirling engine are mechanically connected to a rotating output shaft, contrary to the free-piston Stirling, where they bounce alternately between the space containing the working gas and a spring (mechanical or gas spring). Kinematic Stirling engines, again, are divided into alpha Stirling engines, beta Stirling engines and gamma Stirling engines.

In the following we shall explain how the functioning of kinematic Stirling engines (alpha and beta type) approximate the Stirling cycle.

Alpha Stirling engines contain two cylinders, one hot and one cold. The hot cylinder is situated inside a high temperature heat exchanger and the cold cylinder is situated inside a low temperature heat exchanger. The hot cylinder represents the expansion space for the working gas and the cold cylinder the compression space. The following four illustrations show how the alpha Stirling engine approximates the Stirling cycle:
The major part of the gas is in the cold cylinder. A compression of the working gas at a low temperature level takes place. The regenerator has stored thermal energy. This process is roughly the process 1-2 in the idealised representation (isothermal compression).

The gas has reached its minimal volume and it moves towards the warm cylinder. In its way it flows through the regenerator and absorbs heat. In the warm cylinder it is heated additionally. This process approximates the process 2-3 in the idealised representation (isochoric heating).
The system is in the working stage. The gas is heated in the warm cylinder. It expands and drives the flywheel. The regenerator has released thermal energy to the working gas. This process is roughly the process 3-4 in the idealised representation (isothermal expansion).

The gas has reached its maximal volume and it moves towards the cold cylinder. In its way it flows through the regenerator and releases heat to it. In the cold cylinder it is cooled additionally. This process approximates the process 4-1 in the idealised representation (isochoric cooling).

Beta Stirling engines have only one cylinder that contains a displacer piston besides the power piston. The displacer piston is a loose fit and does not extract any power from the expanding gas but only serves to shuttle the working gas from the hot heat exchanger to the cold heat exchanger. Quite in an analogue way to the alpha Stirling engine, the working gas expands and pushes the power piston when it flows to the hot end of the cylinder. When it flows to the cold end of the cylinder it contracts and the momentum of the machine pushes the power piston the other way to compress the gas. The gas flow is caused by the movement of the displacer piston. The displacer piston normally has at the same time the function of the regenerator. The following four illustrations show how the beta Stirling engine approximates the Stirling cycle:
The major part of the gas is in the cold part of the cylinder. The power piston moves downward and compresses the working gas at a low temperature level. The regenerator (incorporated in the displacer piston) has stored thermal energy. This process is roughly the process 1-2 in the idealised representation (isothermal compression).

The gas has reached its minimal volume. The displacer piston moves upward and the gas moves through it towards the warm end of the cylinder. It absorbs heat from the regenerator and from the warm end of the cylinder. This process approximates the process 2-3 in the idealised representation (isochoric heating).
The system is in the working stage. The gas is heated in the warm part of the cylinder. It expands and drives the flywheel. The regenerator has released thermal energy to the working gas. This process is roughly the process 3-4 in the idealised representation (isothermal expansion).

The gas has reached its maximal volume. The displacer piston moves downward and the gas moves towards the cold end of the cylinder and releases its way heat to the regenerator. In the cold end of the cylinder the gas is cooled additionally. This process approximates the process 4-1 in the idealised representation (isochoric cooling).

The gamma Stirling engine resembles the beta Stirling in the sense that it has a displacer piston additionally to the power piston. But, contrary to the beta Stirling, the two pistons are in two different cylinders.

The characteristic constructive difference of free-piston Stirling engines in comparison to kinematic Stirling engines is that the pistons are not connected to an output shaft. The power piston oscillates between the space that contains the working gas and a spring (mechanical or gas spring). To extract power, a magnet may be attached to the power piston, which moves along stationary
coils and induces electric voltage in them. Free-piston engines are quite a new development from the 1960s.

The thermodynamic processes that are realised in Stirling engines resemble the ideal Stirling cycle, but they do not represent it exactly. One important consequence is that the efficiency of the conversion of thermal energy to mechanical work in the real cycle is lower than in the ideal process with the same upper and lower temperature limits and the same gas volumes and masses. The following $p$-$V$ diagram, which compares the real to the ideal cycle, illustrates this. The $p$-$V$ curve of the real cycle reaches only in few points the closed $p$-$V$ curve of the ideal cycle. Especially the angles of the $p$-$V$ curve of the ideal cycle are not reached. Consequently, the area enclosed by the curve of the real process, which indicates the released work, is smaller than the enclosed area of the ideal cycle. That means, the released work in the real cycle is less than the released work in the ideal cycle (for the same temperature range and the same gas volumes and masses).

![Diagram](image-url)

**Figure 31:** Ideal Stirling process compared to the real Stirling process in $p$-$V$ diagram

The causes of this difference are the following:

- **Dissipation losses:** As in any mechanical process, friction cannot be reduced to zero, so that dissipation losses never can be excluded. This does not only affect the mechanical components like the pistons but also the working gas that may suffer pressure losses. Additionally, there may be gas losses during the operation of the engine.

- **Continuous piston movement:** In the ideal cycle, the movement of the pistons is considered as discontinuous. However, in realised engines (like in the mentioned alpha and beta Stirling engines), the piston movement is continuous. This means that the process steps are not strictly separated from each other, but they overlap each other. For instance, there is no strict isochoric process because the pistons in the illustrated engine geometries maintain always their movement with sinusoidal velocity characteristics, which modifies continuously the gas volume. Discontinuous piston movements would imply complicated crank mechanisms and very high accelerations and forces.

- **Permanent temperature changes:** There are no strict isothermal processes.

- **Regenerator efficiency:** As in any device, the efficiency of the regenerator cannot be 100%. That means there is always an energy loss in comparison to the ideal process because of
imperfect recuperation of the thermal energy that is released in the process that approximates the isochoric cooling.

- Heat loss through the engine material: There is heat transfer through the cylinder(s) that does not contribute to the energy conversion process.

- Adiabatic losses: If a Stirling engine works with high frequency, the high velocity of compression and expansion processes may provoke that these processes are far from being isothermal and that they approximate adiabatic processes with corresponding pressure losses in the expansion and pressure rises in the compression processes.

- Clearance volume: There will be always some “lost” volume in the system, in the cylinders, connecting pieces, regenerator etc., which impedes that the whole working gas is subject to the heating and cooling processes and which reduces the compression ratio.\(^3\)

**Some specific constructive requirements**

- The regenerator has to have high thermal capacity, low thermal conductivity parallel to the fluid flow, small volume and it should introduce a very low friction to the working fluid. A good equilibrium of these characteristics, which are partially contrary to each other, is reached by a stack of fine metal wire meshes, with low porosity to reduce dead space and with the wire axes perpendicular to the gas flow to reduce conduction in that direction and to maximise convection heat transfer.

- Favourable characteristics of the working gas are the following: Very important is a low heat capacity so that a low amount of transferred heat leads to a large increase in pressure. Low viscosity is favourable for a quick functioning without much friction losses. High thermal conductivity helps to accelerate heat transfer processes. Hydrogen combines most perfectly the mentioned characteristics and is used therefore in a number of engines. However, it also has some disadvantages: Hydrogen leaks quite easily through some materials so that the materials have to be chosen carefully and hermetic pressure vessel seals are necessary to maintain the pressure without replacement of lost gas. Additionally, hydrogen may embrittle metals and, of course, it is a flammable gas, which is a safety concern if gas is released from the engine. Helium also combines quite well the mentioned characteristics. It functions close to the efficiency and power density of hydrogen and it lacks the disadvantages of hydrogen. One the other hand, it is more expensive than hydrogen. Helium is used in most technically advanced Stirling engines. Air or nitrogen is also used as working gas. They provide a lower power density, but they minimise gas containment problems and supply costs.

**Thermodynamic operation range**

As indicated above, the efficiency of a Stirling engine, as any heat engine, depends importantly on the ratio of upper to lower operating temperature. A small ratio of the two temperatures implicates a low Carnot efficiency. Additionally, the released work depends on the temperature difference between the hot and the cold part of the engine (see (22)).

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\(^3\) The compression ratio in internal combustion engines is typically between 9 and 22. Stirling engines reach only a compression ratio from 2 to 3 (see Schleder 2004, [6, p. 32]).
Now, as the lower temperature normally is fixed in accordance with the environmental conditions, the high temperature may be the only manageable parameter. A high upper temperature is, thus, favourable in a double sense: First, it rises the Carnot efficiency of the Stirling cycle and, second, a high temperature difference rises the generated mechanical work (all other things being equal). And there is even a third aspect in favour of a high upper temperature: Larger temperature differences allow higher energy exchange rates by heat transfer through the engine boundaries. The Stirling engine, as a heat engine with external heat supply, depends on quick heat transfer through the engine boundaries, so that high temperature differences are also very important in order to allow a high energy flow. In solar dish/Stirling systems, the typical operating temperatures at the high temperature level are in the range from 650°C to 800°C.

Another important parameter that influences the work output of a Stirling engine is the compression ratio. A bigger concentration ratio leads to a higher work output (logarithmic dependency, see (22)). Stirling engines normally operate at a compression ratio of 2 to 3.\(^4\)

Finally, the gas mass is also a parameter that has an influence on the work output. As expressed by (22), the generated work per cycle is proportional to the involved air mass. As the volume of the engine is fixed, this parameter can be modified by different working pressures. Higher pressures lead to higher work output per cycle. Typically the working pressure level in Stirling engines is in the range of 5 to 20MPa (which leads frequently to sealing problems, a common difficulty in Stirling engines). Typical Stirling engines that operate at the specified thermodynamic conditions have efficiencies around 30% to 40%.\(^5\)

**Economical aspects**

Stirling engines have normally higher costs per unit power (€/kW) than internal combustion engines. Lower power density and higher material costs provoke this difference. For instance, Stirling engines need especially highly efficient heat exchangers, which in some cases have to resist quite high temperatures and which elevate considerably the system cost. Also the dissipation of the waste heat requires additional radiators. The costs per unit energy generated (€/kWh), however, may be more similar. Especially in smaller applications (up to 100 kW), Stirling engines may be economically competitive. Typical applications are micro combined heat and power generation, water pumping, astronautics and, most important in our context, generation of electrical energy from renewable energy sources.

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\(^4\) See Schleder 2004, [6, p. 32]
\(^5\) See Stine/Diver 1994, [7, p. 9]
4.2 Conventional Power Plant Technologies

This section describes the main components that are found in any thermoelectric power plant. In large-scale power plants (e.g. a 600 MW coal-fired power plant) many components are very large in size. In small to medium scale power plants the components will be considerably smaller, but nevertheless work with the same principle of operation. Concentrating solar power (CSP) plants are usually small to medium size and are practically identical to conventional power plants with the exception that they have “solar firing”.

4.2.1 Steam Power Cycle Components

This chapter gives a basic overview of the components of the water-steam cycle as can be found in a conventional power plant. Some of the components found in this subsection are discussed in more detail in the succeeding chapters.

In the following section an example of a simple water-steam cycle of a conventional power plant as shown below in Figure 32 is described.

![Figure 32: Example of a water-steam cycle (created with the software Cycle Tempo)](image-url)
A water-steam cycle consists of the following main components (according to the component numbering in Figure 32):

1. Steam generator (boiler)
2. Steam turbine
3. Feedwater pumps
4. Condenser
5. Coolant pumps
6. Heat sink (e.g. cooling tower, heat exchanger)
7. Feedwater tank and deaerator assembly
8. Condensate pumps
9. Steam pressure control system (pressure-reducing valve)
10. Generator

The functions of some of the main components in the list are described in this section. Stand-alone sections for the steam generator and steam turbine follow this section:

(1) Condenser
The condenser’s primary function is to condense the exhaust steam of the steam turbine or the bypassed\(^6\) steam. The condensate is collected in the *hotwell*, the bottom part of the condenser. Within the water-steam cycle the condensate is pumped through the condensate preheater\(^7\) (not shown in Figure 32) and finally into the feedwater tank by means of the condensate pumps. The condensed steam is therefore continuously recovered as condensate and fed back into the system.

The pressure in the condenser depends on the temperature to which the condensate can be cooled down to. This in turn depends on the temperature of the cooling fluid (e.g. river water), which ideally has a low temperature. In power plants the condenser pressure lies in the region of 0.03 – 0.1 bar absolute pressure, depending on the ambient temperature.

The condenser pressure is also referred to as *turbine backpressure*. The more the condensate temperature is reduced, the lower the turbine backpressure will be [8]. With the condenser pressure at below atmospheric pressure the live steam can be expanded in the turbine with a greater enthalpy difference than if the condenser were at ambient or higher than ambient pressure; thus the efficiency of the Rankine cycle is increased. Moreover, it is important that the product in the condenser is fully liquid, as the condensate pumps are designed to handle liquid only.

The condenser also has other important functions, which are worth mentioning [8]:

\(^6\) The steam is bypassed around the steam turbine into the condenser when starting up or shutting down the power plant, or when there is a steam turbine trip. (Note: The bypass is not included in the schematic)

\(^7\) The condensate preheater is a component of the steam generator
• The condenser collects the condensate in the hotwell providing short-term storage for condensate.
• Generally, deaeration of condensate, feedwater and make-up water\(^8\) plays a very important role in a power plant. Deaeration of the condensate is also provided by the condenser.

Condensers are characterised as **surface condensers** and **jet condensers**. A surface condenser, which is a water-cooled shell and tube heat exchanger, is shown in Figure 33.

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**Exercises**

- What is the main function of a condenser?

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\(^8\) Make-up water is water provided by the water treatment facility. Make-up water is required to compensate the water losses in the water-steam cycle [9]. Feedwater is consists of condensate and make-up water as both are feed into the feedwater tank.
(2) Feedwater Tank and Deaerator Assembly

The feedwater tank and deaerator assembly are located before the steam generator. Within the feedwater tank, feedwater is treated as well as heated before it is pumped to the steam generator. In steam power plants, ordinary potable water is not suitable for a steam generator and must be treated, which is explained below. The heating of the feedwater is important in order to prevent thermal shock to the steam generator.

Figure 34 shows such a feedwater tank and deaerator assembly. It has an

- *inlet* for
  - condensate, soft water (permeate) from the water treatment facility, heating steam, water treatment chemicals

- *outlet* for
  - feedwater, exhaust vapours.

The indication *LIS* stands for level indication and *TC* for temperature control.

![Figure 34: Example of a feedwater tank and deaerator assembly, edited from [11, p. 25]](image)

*a) Task of the Feedwater Deaerator*

The feedwater deaerator assembly is a mechanical device needed for driving unwanted gases such as oxygen (O\(_2\)) and carbon dioxide (CO\(_2\)) out of the feedwater and for heating the feedwater before it enters the steam generator. Because of their corrosion capacity O\(_2\) and CO\(_2\) gases dissolved in water must be removed from the condensate and make-up water before it is pumped to the economiser and evaporator section of the steam generator.

In the case of oxygen and carbon dioxide dissolved in feedwater, very small quantities of these gases result in a great damage to the boiler as well as to the water and steam piping systems in the
form of metal pitting [12, p. 2.183]. Corrosion leads to a reduction in the metal thickness of the tubes or shells and eventually in a necessary lowering of the operating pressure for safety reasons [13]. Oxygen leads to so-called oxygen attacks and carbon dioxide to acid attacks. Oxygen, however, is the bigger contributor to corrosion. Oxygen dissolved in water becomes very aggressive when it is heated. Oxygen reacts with the internal surface of the boiler to form corrosive components on the metal surface. Oxygen attacks can cause damage to steam drums, mud drums, boiler headers and condensate piping.

Corrosion by acid attacks occurs at feedwater pH-values below 8.5. As a result of the high pressure and temperature in the boiler, carbonate alkalinity contained in the boiler water is converted to carbon dioxide (CO₂). The steam takes the CO₂ into the live steam piping system. When the steam is condensed, the CO₂ dissolves in the water forming carbonic acid (H₂CO₃). Carbonic acid reduces the pH-value of the condensate (it becomes more acidic), which can lead to acid attacks in the entire water-steam piping [9]. Water treatment is extremely important in power plants and the necessity for it cannot be overstated. A power plant cannot utilise ordinary potable water because it contains impurities like minerals, dissolved gases, solids and salts which would do damage to pre-boiler, boiler, steam and condensate systems [14]. The following table lists some of the major harmful impurities, the problems they cause and the method of chemically treating them:

<table>
<thead>
<tr>
<th>Impurity (Chemical Formula)</th>
<th>Problems</th>
<th>Common Chemical Treatment</th>
</tr>
</thead>
<tbody>
<tr>
<td>Alkalinity (HCO₃⁻, CO₃²⁻ and CaCO₃)</td>
<td>Carryover of feedwater into steam, produces CO₂ in steam leading to formation of carbonic acid (acid attack)</td>
<td>Neutralizing amines, filming amines, combination of both, and lime-soda.</td>
</tr>
<tr>
<td>Hardness (calcium and magnesium salts, CaCO₃)</td>
<td>Primary source of scale in heat exchange equipment</td>
<td>Lime softening, phosphate, chelates and polymers</td>
</tr>
<tr>
<td>Iron (Fe³⁺ and Fe²⁺)</td>
<td>Causes boiler and water line deposits</td>
<td>Phosphate, chelates and polymers</td>
</tr>
<tr>
<td>Oxygen (O₂)</td>
<td>Corrosion of water lines, boiler, return lines, heat exchanger equipments, etc. (oxygen attack)</td>
<td>Oxygen scavengers, filming amines and deaeration</td>
</tr>
<tr>
<td>pH</td>
<td>Corrosion occurs when pH drops below 8.5</td>
<td>pH can be lowered by addition of acids and increased by addition of alkalies</td>
</tr>
<tr>
<td>Hydrogen Sulfide (H₂S)</td>
<td>Corrosion</td>
<td>Chlorination</td>
</tr>
<tr>
<td>Silica (SiO₂)</td>
<td>Scale in boilers and cooling water systems</td>
<td>Lime softening</td>
</tr>
</tbody>
</table>

Every power plant has a water treatment facility to replace the water that is “lost” during operation. Water is lost in several places in the water-steam cycle, for example from the boiler. The problem with the boiler is that the salts dissolved in the water cannot entirely be removed by water treatment and as salt is not steam-volatile, the salt concentration in the boiler increases during operation. As a consequence, the salty water must be continuously removed from the boiler to reduce the salt concentration. The salty water cannot be reused and therefore is replaced with make-up water from the water treatment facility during operation. Additionally, suspended solids are carried by the feedwater and enter the boiler. If left inside the boiler, the concentration of the suspended solids will increase and can form sludge, which degrades both the boiler efficiency and the heat transfer.
Moreover, suspended solids can be carried with the steam into the piping system and cause damage to piping, steam traps and possibly also process equipment. Therefore the suspended solids are removed by carrying out a boiler blowdown (the suspended solids are removed by opening the blowdown valve).

As seen in the previous table, there are several water treatment methods available. In a water treatment facility, the city water is passed through several such stages until the water quality is met.

A water treatment facility therefore has the following positive effects on the power plant [9]:

- Keeps boiler efficiency high (due to reduced formation of scale) and reduces fuel requirements and costs
- Minimises maintenance and downtime, which leads to a reduction of operating and maintenance costs
- Water treatment protects equipment from corrosion and extends equipment lifetime

In addition it is important to mention that for power plant systems it is crucial that steam is available at the point of use in the correct quantity (throughput) and at the correct pressure and temperature. Moreover the steam is required to be clean and dry; air and other incondensable gases must be driven out of the steam leaving only small, allowable residual amounts (the amounts depending on the individual power plant) [15, p. 2.4.2]. For more information regarding the dryness of steam see “Introduction to Steam Generators (also called boilers)” of this chapter.

b) Mechanical Deaerators Types
There are two types of mechanical deaerators, the vacuum and pressure deaerator. In vacuum deaerators the vacuum is created using steam-jet ejectors or mechanical pumps. This method of expelling the gases from the deaerator reduces the content of $O_2$ to approximately 0.2 ppm and $CO_2$ to 2 to 10 ppm. Such deaerators are frequently used for protecting anionic demineraliser resins from taking damage. The pressure-type deaerator finds preference in applications for heated feedwater streams. This deaerator type is also known as deaerating heater and serves as contact heater. The pressure in the tank is kept at about 1.2 bar a (0.2 bar g)$^9$, equivalent to a saturation temperature of 105°C. Saturated steam from the boiler or the steam turbine is passed through a pressure-reducing valve and fed into the water tank of the deaerator. There are two ways of injecting the steam. One possibility is the steam injection through a perforated pipe, which is submerged under the water near the bottom of the tank. The second possibility is the injection through a nozzle at the deaerator head, where the condensate and make-up water flows in. This way, the steam removes the gases from the water at an early stage. The water - now freed from the gases - passes into the feedwater tank through a perforated pipe together with the steam. For both systems the condensate and make-up water enter the feedwater tank at the top of the deaerator head and is broken up into small drops e.g. by cascading the water over a series of perforated trays, by spraying or a combination of the two. The steam gets into contact with the hottest water first and thoroughly scrubs out the last remaining fraction of gas. The gas is carried along with the steam to the top of the deaerator and is vented into the atmosphere by a valve. The direction of the steam

$^9$ The pressure “bar a” is the absolute pressure and “bar g” is the gauge pressure (The relation is: bar a = bar g + 1)
may be cross-flow, down-flow or countercurrent. The deaerated (de-gassed) water is collected in the feedwater tank below the deaerator head. Due to the heating by the steam the deaerated water has a temperature of about 105°C, which is the saturation temperature at the pressure of 1.2 bar a. The content of O₂ can be reduced below 7 ppb, a level at which oxygen is nearly undetectable. Moreover, the feedwater deaerator is equipped with a water level control system, a steam pressure control system (pressure-reducing valve), a feedwater level gauge and a feedwater outlet at the bottom of the vessel [12, 2.183], [15, p. 3.21.4].

The deaerator removes most of the gases. The traces of gases that are left in the water, however, are still very harmful, so further treatment is necessary. This is achieved chemically with oxygen binding chemicals, referred to as oxygen scavengers.

The feedwater must always be kept in the alkaline range at a pH value depending on the boiler design. Chemicals such as sodium hydroxide NaOH and trisodium phosphate Na₃PO₄ are fed into the feedwater to control the pH value [12, p. 2.185].

Other chemicals which bind oxygen that can be used are hydrazine (N₂H₄) and sodium sulphite (Na₂SO₃).

Reaction with sodium sulphite [11, p. 27]:
\[
H_2O + O_2 + 2Na_2SO_3 \rightarrow H_2O + 2Na_2SO_4
\]  
(23)

Reaction with hydrazine [11, p. 27]:
\[
H_2O + O_2 + N_2H_4 \rightarrow 3H_2O + N_2
\]  
(24)

► Exercises
- What is the main function of a feedwater deaerator assembly?
- Why are the gases O₂ and CO₂ so harmful to components such as the steam generator?
- Name at least 2 positive effects of a water treatment facility on the power plant?

(3) Power Plant Cooling
According to the Second Law of Thermodynamics only a portion of the heat released by the combustion of fuel can be converted to useful shaft work for electricity generation. This means that there is residual heat that must be rejected in an energy sink, which is at a lower temperature [12, p. 2.137]. The methods of integrating cooling in a power plant and types of cooling systems are explained below.

Residual heat is given off by the exhaust steam leaving the steam turbine or bypass. The exhaust steam entering the condenser immediately condenses the moment it comes into contact with piping inside the condenser, through which coolant is circulated. Due to a higher temperature, the exhaust steam transfers the heat to the coolant. The liquid product of the condensed exhaust steam is called condensate and is collected in the hotwell at the bottom of the condenser.
Fossil fuel fired and nuclear power plants mostly use evaporative water cooling for several reasons: The cooling efficiency depends on the coolant temperature. Evaporative water cooling is more efficient than dry cooling because the water temperatures vary far less than the air temperatures. Hence, the evaporative water cooling method is steadier throughout the year than the dry cooling method. Another reason for using evaporative water cooling is the lower capital cost. Whenever possible, thermoelectric power plants are built close or next to rivers, lakes or the sea. Solar thermal power plants, however, are often located in desert regions with limited access to large amounts of water that would be required for evaporative water cooling. Drawing water from a distant source or the purifying of low-quality water for the use in solar thermal power plants may increase costs, making dry cooling more attractive [16, p. 4]. When addressing water limitations and environmental regulations, air cooling can be used for new thermoelectric power plants. Using air cooling eliminates over 90% of the water usage, but at a penalty of a less electric energy being provided per year and higher electricity costs [16, p. 5].

There are several different cooling systems available that can be used for removing the heat from the coolant, such that it can be recirculated in the condenser. Some of the cooling systems are introduced and some information on the water consumption in existing parabolic trough power plants is given. Following the section on cooling systems, a short introduction to psychometric charts is given. Psychometric charts are used, for example, for the calculation of wet and hybrid cooling systems. What makes the charts very useful is that they include several important properties of moist air in a single graph.

Cooling systems:

a) Once-Through Cooling Systems

Once-through cooling systems can be used when a power plant is located next to rivers, lakes or the sea. If, for example, water is taken from a river, the river’s water is pumped as cooling water through the pipes inside the condenser, removing the heat from the waste steam. The heated cooling water is then simply returned to the river, as shown in Figure 35 [12, p. 2.138]. This cooling method is typically not available for solar thermal power plants, as these are located mostly in deserts where water is scarce.
b) Cooling Towers
For cooling towers two technologies are available: dry and wet cooling towers. This section deals with wet-cooling towers only.
A cooling tower is a specialised heat exchanger which uses the principle of evaporation cooling for removing the waste heat from the coolant. Cooling is realised with the two fluids air and water, which are brought into direct contact with each other [12, p. 2.139]. The water supply is secured from nearby rivers, lakes or the sea.
The evaporation process is a natural phenomenon which takes place until the medium air has reached a state when it is fully saturated with moisture (humidity) [17].

There are two basic types of cooling towers:

- Forced draught (realised with fans, see example Figure 36)
- Natural draught (see example Figure 37)
There are certain differences in the operating efficiency of the two types. The cooling effect of natural-draught towers depends on the temperature gradients between the two media water and air as well as the wind forces, whereas for forced draught towers the cooling effect can be manipulated by the tower’s fans [18].

Cooling towers are rated in terms of approach and range, where
- the approach is the difference in temperature between the cooled-water temperature and the wet-bulb temperature $T_{wb}$ of the air entering the tower
- the range is the temperature difference between the water inlet and exit states

A cooling tower is based on evaporative cooling. Hence, the maximum cooling tower efficiency is limited by the wet-bulb temperature $T_{wb}$ of the cooling air.

The cooling tower efficiency $\mu = \frac{(T_{in} - T_{out}) \cdot 100}{T_{in} - T_{wb}}$, \hspace{1cm} (25)

where $\mu$ is the cooling tower efficiency (common range between 70 – 75%), $T_{in}$ is the inlet temperature of water [°C] to the tower, $T_{out}$ is the outlet temperature of water [°C] from the tower, and $T_{wb}$ is the wet bulb temperature of air [°C].

The temperature difference between inlet and outlet water ($T_{in} - T_{out}$) is normally in the range of 10 – 15°C [18].

General principle of operation: The cooling water is sprayed into the large space inside the cooling tower (to increase the heat transfer surface area of the water) and is cooled by the ambient air, which passes through the tower either by a forced-draught or natural-draught. Approximately 3 – 5% of the cooling water evaporates [19] and is taken up by the passing air until the air has reached a saturated state. The non-evaporated, cooled cooling water is collected at the bottom of the tower.
As a lot of cooling water was evaporated, new cooling water must be added (e.g. from a river). Ultimately, the cooling efficiency depends on the wet-bulb temperature\textsuperscript{10} (i.e. to what temperature the cooling water can be cooled down to) and the temperature of the replacement river water. The colder the air and the water, the better is the cooling effect. For power plants with wet-cooling towers, the efficiency of the entire plant depends on the difference in the steam and condensate temperatures. The lower the cooling water temperature returned to the condenser is, the higher the overall power plant efficiency will be. Note: The temperature dependency shows that cold winter months provide the most suitable conditions for plant operation. However, freezing weather conditions can be problematic during plant operation requiring anti-freeze measures to be taken by the operator.

A cooling tower type that combines the principle of operation of a forced-draught and natural-draught tower is shown in Figure 38.

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{fan-assisted_natural-draught_tower.png}
\caption{Cutaway diagram of a fan-assisted natural-draught tower [12, p. 2.145]}
\end{figure}

The wet-cooling method has been applied to existing solar thermal power plants. In regions where water is scarce, a combination of wet-cooling and dry-cooling in a dry-wet hybrid cooling system may be favoured. For details see the section \textit{Wet-Dry Hybrid Cooling}.

c) \textit{Air-cooled condensers}
Air-cooled condensers exist in two basic types: dry-cooled and wet-surface cooled.

In dry-cooled direct air-cooled condensers, as shown in Figure 39, the turbine exhaust steam flows through tubes that are equipped with external fins (the fins increase the heat transfer area of the tube). The heat is transferred to the air that is blown around the tubes [12, p. 2.153].

\textsuperscript{10} The wet-bulb temperature is read from a wet-bulb thermometer, which is an ordinary liquid-in-glass thermometer whose bulb is enclosed by a wick moistened with water or is wrapped by wet muslin. As the water evaporates from the thermometer, the thermometer cools down as the evaporation takes up heat. The term dry-bulb temperature refers to the temperature of air measured by a thermometer which is freely exposed to the air but is shielded from radiation and moisture. Due to the evaporative cooling, the temperature indicated by the wet-bulb-thermometer is always lower than the temperature indicated by the thermometer measuring the dry-bulb temperature, unless the relative humidity of the surroundings is 100%. Often a wet-bulb thermometer is mounted together with a dry-bulb thermometer to form an instrument called a psychrometer [20], [21], [22], [2. p. 675].
The wet-surface cooled condenser is described according to Figure 40. The exhaust steam from the steam turbine flows through tube bundles. The steam is condensed by a mixture of air and water which is cascading over the tube bundles. The water takes up the heat from the steam. The heat from the cascading water is transferred to the air stream by vaporisation. In a next step, water contained in the air stream is removed as much as possible by forcing the air stream to turn 180°. Finally, the air is discharged vertically by means of fan assistance. This is accomplished at a high air velocity in order to minimise recirculation [23].

\[\text{Figure 40: Wet-surface cooled condenser [23]}\]

\textit{d) Air-cooled heat exchangers}, as shown in Figure 41, work as follows: The hot coolant flows from the condenser to the air-cooled heat exchanger(s) in a single pipe. The coolant flow is then divided up and passed through a large number of small finned pipes in order to increase the heat transfer surface area. Fans located at the top of the heat exchanger induce a forced-draught air flow. The air passes around the coolant pipes removing the heat by forced convection.
e) Wet-Dry Hybrid Cooling:
Wet-dry hybrid cooling is the combination of wet and dry cooling and can be used at locations where local water resources are available but limited [25], hence this is an option for the application in solar thermal power plants. Several recently constructed plants use a combination of wet and dry cooling. The dry-cooling is realised with an air cooler – which is usually dimensioned smaller than for a plant using air-cooling only. The dry-cooling system is the primary heat rejection system. The performance of a dry-cooled system falls at high air temperatures, unlike the performance of the wet-cooling system, which remains unaffected by the ambient temperature variations. Hence, the wet cooling system must be run in parallel mode during the hot summer months to compensate the low efficiency of the dry-cooler when air temperatures are very high [16, pp. 5, 15]. A hybrid wet/dry cooling system working in parallel mode is shown in Figure 42.

There are two broad categories of hybrid wet/dry cooling systems. One is aiming to abate plume and the other to reduce the water consumption. Plume abatement is the reduction of water vapour plume from a wet-cooling tower such that it no longer appears or to avoid icing on nearby roads during freezing conditions in winter periods. Plume abatement is generally not considered as a problem for CSP technologies as they are usually located in desert regions and therefore does not pose any risk to motorists. The second category is far more important for CSP systems: reduction of the water consumption compared to wet-cooled systems and a cooling performance enhancement at high ambient temperatures compared to dry-cooled plants [16, p. 14].
f) **Heller System:**
The exhaust steam from the steam turbine is condensed in a direct-cooled (DC) jet condenser, as shown in Figure 43. The jet condenser injects cooled (condensate) water (coming from the cooling tower) via many nozzles directly into the exhaust steam and condenses it. The (hot) condensate is pumped to the cooling tower, as illustrated by Figure 44, where it is air-cooled in a natural or mechanical draught tower. A portion of the (hot) condensate is branched off before reaching the cooling tower for the use in the water-steam cycle [26].
Examples of water consumption in existing parabolic trough power plants:

Water consumption of the Andasol I cooling system: The cooling water consumption of Andasol I (50 MW_e), which is equipped with wet cooling towers, amounts to about 870,000 m³/year or 4.8 m³/MWh.\(^\text{11}\)

Water consumption of SEGS: The total water consumption of a 30 MW_e SEGS power plant is about 3.8 m³/MWh. 91.8% of the consumption is used for wet cooling (i.e. 3.5 m³/MWh), 6.1% for steam cycle water demand and 2% for reflector cleaning.

Psychometric charts:

As mentioned, psychometric charts are used, for example, for the calculation of wet and hybrid cooling systems. In SI units, the following properties are included in the graph:

- dry-bulb temperature [°C]
- humidity ratio, \(\omega\) [kg, or g, of water vapour per kg of dry air]
- vapour pressure, \(p_v\) [bar]
- constant relative humidity curves, \(\phi\) [%]
- mixture enthalpy per unit mass of dry air in the mixture: \( h_a + \omega h_v\) [kJ/kg], where \( h_a\) is the enthalpy of the dry air and \( h_v\) is the enthalpy of the water vapour
- wet-bulb temperature, \(T_{wb}\) [°C]
- volume per unit mass of dry air, \(v\) or \(V/m_a\) [m³/kg]

---

\(^{11}\) See Solar Millennium 2008.
Note that the lines of constant wet-bulb temperature are approximately lines of constant enthalpy per unit mass of dry air.

A psychometric chart for illustrating where in the graph the above mentioned properties can be found is shown in Figure 45. A psychometric chart for 1 atm (SI units) is shown in Figure 46.

![Psychometric chart](image)

**Figure 45:** Psychometric chart (without values) [2, p. 677]
Calculation problem: Cooling of a power plant by a cooling tower [1, pp. 448-449]

Problem:
Cooling water leaves the condenser of a power plant and enters a wet cooling tower at 35°C at a rate of 100 kg/s. The water is cooled to 22°C in the cooling tower by air that enters the tower at 1 atm, 20°C, and 60% relative humidity and leaves saturated at 30°C. Neglecting the power input to the fan, determine (a) the volume flow rate of air into the cooling tower and (b) the mass flow rate of the required make-up water.

Solution:
The entire cooling tower is taken to be the system, which is shown schematically in Figure 47. The mass flow rate of liquid water decreases by an amount equal to the amount of water that vaporises in the tower during the cooling process. The water lost through evaporation must be made up later in the cycle to maintain steady operation.
Assumptions: 1) Steady operation conditions exist and thus the mass flow rate of dry air remains constant during the entire process. 2) Dry air and the water vapour are ideal gases. 3) The kinetic and potential energy changes are negligible. 4) The cooling tower is adiabatic.
Analysis: Applying the mass and energy balances on the cooling tower gives
Dry air mass balance: 

\[ \dot{m}_{a_1} = \dot{m}_{a_2} = \dot{m}_a \]  

Water mass balance: 

\[ \dot{m}_1 + \dot{m}_a \omega_1 = \dot{m}_4 + \dot{m}_a \omega_2 \]  

or 

\[ \dot{m}_3 - \dot{m}_4 = \dot{m}_a (\omega_2 - \omega_1) = \dot{m}_{\text{makeup}} \]  

Energy balance: 

\[ \sum \dot{m}_a h_{in} = \sum \dot{m}_{out} h_{out} \rightarrow \dot{m}_a h_1 + \dot{m}_3 h_3 = \dot{m}_a h_2 + \dot{m}_4 h_4 \]  

or 

\[ \dot{m}_3 h_3 = \dot{m}_a (h_2 - h_1) + (\dot{m}_3 - \dot{m}_{\text{makeup}}) h_4 \]  

Solving for \( \dot{m}_a \) gives 

\[ \dot{m}_a = \frac{\dot{m}_3 (h_3 - h_4)}{(h_2 - h_1) - (\omega_2 - \omega_1) h_4} \]  

Figure 47: Schematic of a wet cooling tower [1, p. 448]

The method for locating the required properties in the psychrometric chart for doing the calculating is given with reference to Figure 48. The psychrometric chart given in Figure 48 has coloured lines to support the explanation. At first, the properties for the air entering the tower are to be taken from the graph. As a first step, locate the dry bulb temperature of 20°C on the x-axis. Then draw a vertical line to the interception with the line of 60% relative humidity (red line in graph). From the interception point draw a horizontal line to the right-hand y-axis to read off the humidity ratio (red line in graph). Next, draw a line from the interception point parallel to the lines of the wet bulb temperature until the axis of the specific enthalpy of moist air is touched (magenta line in graph). Finally, draw a line from the interception point parallel to the lines of volume per unit mass of dry air (brown line in graph). The obtained values are: 

\[ h_1 = 42.2 \text{ kJ/kg dry air} \]
$\omega_1 = 0.0087 \text{ kg H}_2\text{O/kg dry air}$
$v_1 = 0.842 \text{ m}^3/\text{kg dry air}$

Repeat the process for the air leaving the tower at 30°C in a saturated state to obtain $h_2$ and $\omega_2$ (orange lines).

$h_2 = 100.0 \text{ kJ/kg dry air}$
$\omega_2 = 0.0273 \text{ kg H}_2\text{O/kg dry air}$

![Figure 48: Psychrometric chart for calculation problem, edited from [2, p. 677]](image)

According to equation (31), the enthalpy $h_4$ of the cooled water leaving cooling tower at 22°C and the enthalpy $h_3$ of the cooling water entering the wet cooling tower at 35°C are required. For obtaining these values Table 2 is used and the column of the enthalpy for a saturated liquid $h_f$ is of interest here.

$h_4 = h_{f@35°C} = 146.68 \text{ kJ/kg H}_2\text{O}$
$h_3 = h_{f@22°C} = 92.33 \text{ kJ/kg H}_2\text{O}$
Table 2: Saturated water — Temperature table [1, p. 1088]

| Temp., °C | Sat. press., P
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>0.01</td>
<td>0.6113</td>
</tr>
<tr>
<td>5</td>
<td>0.8721</td>
</tr>
<tr>
<td>10</td>
<td>1.2276</td>
</tr>
<tr>
<td>15</td>
<td>1.7051</td>
</tr>
<tr>
<td>20</td>
<td>2.3390</td>
</tr>
<tr>
<td>25</td>
<td>3.169</td>
</tr>
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<td>30</td>
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<td>50</td>
<td>12.349</td>
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<td>55</td>
<td>15.759</td>
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<td>60</td>
<td>19.940</td>
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<td>65</td>
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<td>31.19</td>
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<td>85</td>
<td>57.83</td>
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<tr>
<td>90</td>
<td>70.14</td>
</tr>
<tr>
<td>95</td>
<td>84.55</td>
</tr>
</tbody>
</table>

Substituting into equation (31),

\[
\dot{m}_u = \frac{(100 \text{ kg/s})\left[(146.68 - 92.33) \text{kJ/kg}\right]}{[10000.0 - 42.2] \text{kJ/kg} - [0.0273 - 0.0087] \text{(92.33)kJ/kg}] = 96.9 \text{ kg/s}
\]

Giving the answer to part (a), the volume flow rate of air into the cooling tower becomes

\[
\dot{V}_1 = \dot{m}_u v_1 = (96.9 \text{ kg/s})(0.842 \text{ m}^3/\text{kg}) = 81.6 \text{ m}^3/\text{s}
\]

(b) The mass flow rate of the required make-up water is determined from

\[
\dot{m}_{\text{makeup}} = \dot{m}_u (\omega_2 - \omega_1) = (96.9 \text{ kg/s})(0.0273 - 0.0087) = 1.80 \text{ kg/s}
\]
Exercises

- Name at least 3 cooling technologies and make a sketch for each.
- Explain how the hybrid wet/dry parallel cooling system works.
- Which cooling technology can be used for a solar thermal power plant if local water resources are available but limited?
- What properties are included in a psychometric chart?

4.2.2 Steam Generators and Heat Exchangers

This section is divided into conventional steam generators, direct steam generation in CSP plants, heat exchangers and steam generators in CSP plants.

(1) Conventional Steam Generators

Steam is generated for many different applications. In power plants, the generated steam drives a steam turbine, which is coupled to a generator. Other applications, such as those found in the chemical industry, may require the steam for heating processes only. Many different boiler designs exist and the design depends on the application. Boilers found in power plants can be designed very large in size and very high (currently up to 170 m high, e.g. Lippendorf power plant, Germany). Large boilers occupy an entire building, which is called boiler house.

A steam generator (boiler) can be defined as a closed vessel in which water under pressure is transformed into steam by the application of heat [27]. Note that there is no difference in the terms steam generator and boiler. However, it is common nowadays that the term boiler refers to all the systems that receive water and supply steam.

A conventional steam generator usually consists of a condensate preheater, an economiser (feedwater preheater), evaporator and superheater stages. In a combustion process, a fossil fuel is burned (e.g. gas or coal) and (hot) flue gas is produced. The flue gas is then used for generating steam. Depending on the boiler design, thermal radiation can also be utilised. The flue gas is first passed in counterflow through the superheater stages, then through the evaporator, the economiser and finally the condensate preheater. The flue gas enters the superheater stages with the highest temperature and leaves the steam generator after the condensate preheater with the lowest temperature. The main components of a steam generator (superheaters, evaporator, economiser, and condensate preheater) are briefly explained and some steam generator designs are introduced.

Superheaters: Steam is superheated when it is heated above its saturation temperature. After it is superheated, the steam is called live steam or main steam. The pressure in the superheater corresponds to that in the evaporator. In a power plant the maximum temperature of the steam depends on [3, p. 19]:

- the material of the superheaters, as the material sets the limit to the temperature of the hot gas passing through the superheater stages
- the highest admissible operating temperature of the steam turbine.

Evaporator: In the evaporator water is vapourised to saturated steam.
Economiser: The economiser is sometimes also referred to as eco or feedwater preheater. The economiser is a counterflow heat exchanger for energy recovery downstream of the superheater and reheater. By design, steam is usually not generated inside the economiser’s tubes. The economiser increases the feedwater temperature before it enters the steam drum of a watertube boiler [12, p. 2.40] or the evaporator of a firetube boiler.

Condensate preheater: The condensate preheater is a counterflow heat exchanger for energy recovery in which the condensate is heated to a temperature of about 85°C. This is a typical temperature at which the condensate is supplied to the deaerator of the feedwater tank. Keeping the condensate temperature constant at 85°C is very important. If the condensate temperatures were much higher than 85°C then the tank pressure would be raised. As mentioned earlier in this chapter, a feedwater tank is operated at a pressure of about 1.2 bar a (0.2 bar g), equivalent to a saturation temperature of 105°C. Generally the pressure in the tank can be maintained at a constant level by increasing or decreasing the steam flow of the steam required for deaeration. Hence, in the case of a higher-than-normal condensate temperature, the only way to control the tank pressure is to throttle down the valve controlling the steam flow, as less steam needs to be made available for achieving the setpoint pressure. This, in turn, leads to the negative effect that the steam flow rate reduces to such a low value that proper dispersal at the steam nozzle can no longer be ensured [15, p. 3.21.7] i.e. the quality of deaeration would be affected and more gases would remain dissolved in the water.

Steam generator designs:

Steam generators can be divided into two basic types:

- Firetube boiler
- Watertube boiler

a) Firetube boiler

Firetube boilers are built up with a large water tank and exist in various designs with different combinations of the tube design.

There are two classifications of firetube boilers: Boilers with

- external furnace
- internal furnace (e.g. a burner)

Boilers with external furnace: A boiler with external furnace uses the heat from a gas, e.g. the exhaust gas from a gas turbine. An example of a design configuration of a firetube boiler with external furnace is shown in Figure 49, featuring a two-pass dry back configuration. One or more large first pass furnace tubes and a large number of small second pass tubes pass through the water [15]. This boiler design is therefore a type of heat recovery steam generator (HRSG), which requires an external furnace because steam is generated solely by passing hot gases through its tubes. Heat recovery steam generators are used in (conventional) combined-cycle plants, but they

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12 The firetube boiler is also known as ‘smoke tube boiler’ or ‘shell boiler’ [9, p. 3.2.2].
can also be used in solar tower power plants with air receiver technology\textsuperscript{13} and Integrated Solar Combined Cycle Systems (ISCCS)\textsuperscript{14}. More details on HRSG are given in the section following the descriptions on watertube boilers.

Boilers with internal furnace: Figure 50 shows an economic boiler with three passes, wet back and internal furnace (it is equipped with a burner) [15].

Firetube boilers cannot be used in large power plants as their design limits them to lower capacities, pressures and temperatures. The operating pressure is limited to approximately 35 bar

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\textsuperscript{13} Solar tower power plant with air receiver technology can be hybridised, for example, with a gas turbine. See chapter “Solar Tower Technology“ for more detailed information.

\textsuperscript{14} An ISCCS plant is a combined-cycle plant into which a parabolic trough power plant is integrated. See chapter “Parabolic Trough Technology“ for more detailed information.
(today’s watertube boilers can handle pressures up to approximately 270 bar). Figure 51 shows a modern packaged boiler. Here, the numerous tubes of the pass(es) can be seen very clearly. Nowadays it is typical to design firetube boilers in the packaged design [12, p. 2.59].

Figure 51: Modern packaged boiler [15, p. 3.1.9]

b) Watertube boiler

In watertube boilers, the water and evaporated steam flows through a great number of small-diameter pipes. The smaller dimensions allow for much higher inner pressure loads. When combusting conventional fuel hot gas is produced, that is continuously passed around these water-carrying tubes for evaporation.

Water tube boilers are predominantly used in large-scale power plants and operate with pressures and temperatures beyond 180 bar and 500°C, respectively.

A watertube boiler consists of many components. The major components of a coal-fired subcritical-pressure watertube boiler are shown in the illustration of Figure 52 and include a furnace (for combusting the fossil fuel), steam superheater (primary and secondary), steam reheater, economiser, steam drum and an attemperator or steam temperature control system.

A steam drum is required by subcritical boilers. In subcritical boilers, the water can only be partially evaporated to steam on any one pass through the furnace wall circuits. It is therefore necessary to separate steam and water in the steam drum. Depending on the required live steam quality the withdrawn saturated steam either is used directly or will be superheated in a superheater. The water remaining in the drum is mixed with the feedwater that replaces the withdrawn steam and is returned to the boiling circuits where it is heated further. Once-through, or drumless, boiler designs do not require a steam drum because the water is completely evaporated and superheated within one pass through the furnace and superheater stages.

The attemperator or steam temperature control system controls the steam temperature. Usually this is achieved by injection of cold liquid water into the live steam. A less common method is to use a cooling loop [12, pp. 2.34-2.40].
There are three classifications of watertube boilers:

- Horizontal straight tube boilers
- Bent tube boilers
- Cyclone fired boilers

For all watertube boilers applies that the watertubes are fitted to the walls of the boiler. The flow through the pipes is realised by the steam-water circulation loop. In watertube boilers – those that are not once-through design – the water is circulated from a steam drum located at the top of the boiler, down through pipes called downcomers to the bottom of the boiler. From there the pipes enter the furnace and the water is heated up in the waterwall tubes. The circuit is closed by the steam drum. There are many boiler designs, where the driving force of circulation is convection, or natural circulation, i.e. no pump is required as sufficient flow to adequately cool the waterwall tubes is provided solely by density differences of the water/steam flow. Other boiler designs use pumps for the circulation either because it is needed or more economical for the power plant. This is referred to forced circulation [12, p. 2.35]. The two illustrations Figure 53 and Figure 54 show the steam-water circulation loop for natural and forced circulation, respectively.
Watertube boilers are also designed as HRSG, for example for utilising the energy from the exhaust gas of a gas turbine. Such HRSG can be used in solar tower power plants with air receiver technology and Integrated Solar Combined Cycle Systems (ISCCS). Details on such HRSG are given in the following section.

(2) Heat Recovery Steam Generator (HRSG) designs for Combined-Cycle Plants, Solar Tower Power Plants with air-receiver technology and Integrated Solar Combined Cycle Systems (ISCCS)

In this section heat recovery steam generators for application in combined-cycle plants are introduced. Such HRSG can also be integrated in solar tower power plants with air-receiver technology and (parabolic trough) Integrated Solar Combined Cycle Systems (ISCCS).

A HRSG is basically a counterflow heat exchanger. It is composed of a series of superheater, boiler and economiser sections, which are arranged in such a way that heat recovery is maximised. Originally, the HRSG was designed especially for the use with gas turbines. HRSG handle large gas volumes with minimum pressure drop, minimising the impact on the gas turbine efficiency. A HRSG can be designed such that the gas flow is vertical and in cross-flow over the tube bundles or that the gas flow is horizontal and in cross-flow over the tube bundles. The most common, however, is the horizontal gas flow. Natural and forced-circulation systems are used for the boiler sections. Single-pressure operation of HRSG is the most common. However, multiple (up to three) pressure flow loop units are becoming more prevalent, especially on larger units where energy efficiency is particularly important [12, p. 2.57]. It is possible to also make use of auxiliary or supplemental fuel firing in a duct burner so that the steam production is increased, for the control of the steam superheat temperature or to meet process steam requirements. High-pressure boiler circuits used for power generation are capable of producing up to 76 kg/s of steam at 541°C [12, p. 2.57].
Figure 55 shows a forced circulation and a natural circulation HRSG.

A HRSG with forced circulation is a vertical HRSG, which has the tube bundles arranged horizontally (in cross-flow to the gas flow). A forced-circulation of the boiler water is realised with circulating pumps so that a positive circulation through the evaporator sections is maintained. The exhaust gas flows at first horizontally and then changes the direction to vertical. In vertical direction, the exhaust gas flows around the horizontally arranged tube bundles.

A HRSG with natural circulation is a horizontal HRSG, which has the tube bundles arranged vertically (in cross-flow to the gas flow). The circulation of the boiler water occurs entirely naturally by gravity, based on the density difference of water and boiler water mixtures. The exhaust gas flows horizontally around the vertically arranged tube bundles [28, pp. 184-185].

A HRSG can be used in a solar tower power plant which works with the open or closed volumetric receiver technology. Such receiver types use air as HTF.

![Figure 55: Forced circulation HRSG (left); Natural circulation HRSG (right) [28, pp. 184-185]](image)

A sectional view of a multipressure HRSG for gas turbine exhaust applications is shown in Figure 56 [12, p. 2.58].
In the once-through design of the HRSG (Figure 57) the water is preheated in the economiser, evaporated in the evaporator and superheated in the superheater in a single passage. This is different to a HRSG with a drum, where water is only partially evaporated to steam [28, pp. 186-187].

Figure 56: Sectional view of a multipressure HRSG for gas turbine exhaust applications [12, p. 2.58]

Figure 57: Principle of drum type and once-through evaporation [28, p.187]

An example of a temperature profile for a single-pressure HRSG is shown in Figure 58.
(3) Direct Steam Generation in CSP plants

Direct steam generation has been realised in the parabolic trough, linear Fresnel and solar tower technology. However, these are no classical boiler systems and hence direct steam generation is discussed in their respective chapters.

In direct steam generation systems the steam is generated directly in the absorber tubes in the solar field. Instead of thermo oil, water (alternatively organic media) is pumped through the receiver tubes where it is evaporated. There is only one fluid cycle compared to two cycles such as in indirect steam generation systems. A schematic of a parabolic trough power plant with direct steam generation is shown in Figure 59.
(4) Heat Exchangers and Steam Generators in CSP Plants

Current parabolic trough power plant designs use thermal oil as heat transfer fluid or directly generate steam in the solar field. In recent developments, molten salt as heat transfer fluid is also explored.

Parabolic trough power plants using thermal oil as heat transfer fluid, such as the Andasol type plants in Spain, deploy an oil/water heat exchanger to transfer the heat to the Rankine cycle. Similarly, a molten-salt/water heat exchanger is required by both solar tower and, in future, parabolic trough power plants which use molten salt as heat transfer medium.

*Heat exchangers in parabolic trough power plants*

Shell and tube heat exchangers are usually built as U-type or header-type heat exchanger. The header-type heater, which is also known as the snake heater, was traditionally used in coal-fired power plants. The U-type heater is applied in systems with low tube-side pressures and flow rates, while the header-type heaters are usually implemented in such with high pressures and flow rates. A header type heat exchanger with header pipes instead of tube sheets allows considerably better thermo-elastic operating behaviour at start-up and turbine trip [30]. This is very advantageous for power plants with daily cycling like parabolic trough power plants. An example of a U-Type and header-type heat exchanger is given in Figure 60. An example for a tube bundle of a shell and tube heat exchanger is shown in Figure 61.
To ensure an efficient heat transfer and long lifetime, the tube material has to fulfil some requirements such as high thermal conductivity, resistance against thermal stress, thermo-chemical stability, pressure stability and corrosion resistance. In particular, the tube material should be compatible with the HTF and the working fluid [32]. Typical materials are copper alloy, stainless
Steel, low carbon steel, titanium, nickel and nickel alloys (Inconel, Hastelloy). Inappropriate materials could result in leakages between the shell and tube side.

Steam generator train for parabolic trough power plants
Steam generators are usually divided into different parts that are operated at different temperature levels. The steam generator usually consists of an economiser, a drum type evaporator and a superheater. These components are found in the following schematic for a parabolic trough power plant which, for example, uses oil as heat transfer fluid. Furthermore, a re heater is also employed in this example.

![Process flow sheet for indirect steam generation](image)

Figure 62: Process flow sheet for indirect steam generation [33]

To give an example, Figure 63 shows the system configuration of the Californian SEGS VIII (80 MW_e) plant. The power block is based on a reheat-turbine concept with solar steam generation and solar superheating. An auxiliary natural gas heater is integrated. The gross steam cycle efficiency at the design point is 38% ¹⁶. The HTF is heated up in the collector field and is pumped into the steam generator with a temperature of 391°C. The steam generator is divided into three heat exchangers in series - a feed water heater (preheater), an evaporator and a steam superheater. The generated live steam parameters are 371°C and 100 bar at the entry of the high pressure steam turbine. These steam parameters are common in large parabolic trough power plants. After passing the high pressure turbine the steam is reheated from 206°C to 371°C and has a pressure of 18 bar. The reheated steam is expanded through the low pressure turbine and is condensed in the condenser, which is cooled by a wet cooling tower [35].

¹⁵ These are trademarks of the firms Special Metals Corporation and Haynes International.
¹⁶ This is valid for the solar mode as well as for the gas mode, because the turbine inlet steam has identical characteristics for both modes. See Johansson/Burnham 1993 [34].
In many parabolic trough plants there are two steam generator trains in parallel. Figure 64 shows a parallel 50 MW_{e} steam generator train for parabolic trough power plants. The components (economiser, evaporator and superheater) that are presented in the following are part of the same steam generator train.
An economiser is a power plant component used to preheat the feedwater. Economisers are usually designed as shell and tube heat exchangers in which the HTF is on the shell side, and the feedwater inside the tube, as shown in Figure 65 and Figure 66. Feedwater preheating has thermodynamic advantages and improves the system efficiency.

In the evaporator the water is evaporated. Saturated or wet steam is generated that afterwards is superheated in the superheater. The inner heat exchanger of the evaporator is usually of header type with HTF on the tube side and water on the shell side. The evaporator consists of a bottom drum and a top drum. The bottom drum is the evaporator drum with the coil inside. The top drum is the drum where the steam is accumulated. The drums have several connections, which secure a high circulation ratio. Such an evaporator is shown in Figure 67 and Figure 68.
The superheater is the last component of the high pressure heater train. It is used to convert wet or saturated steam into dry steam (superheated steam). The superheater is usually a header type heat exchanger with HTF on the shell side and steam on the tube side. The coil design has to be very robust and flexible securing a very high start-up ramp and long lifetime. Such a superheater is shown in Figure 69 and Figure 70.

Steam generator train – design parameters:
The design and the dimensioning of the steam generators depend on the following parameters:
required steam flow
heat exchange surface
temperature level and mass flow of the heat source (HTF)
required steam parameters (temperature, pressure)
properties of heat transfer fluid and working fluid
material properties of tubes and drum
geometry of the heat transfer surface
positioning and design of the tube bundle
flow conditions (laminar or turbulent)

Figure 71: Design and dimensioning parameters for steam generators

At the end of the different components of the steam generator train, the temperature difference of the water/steam and the HTF should be small in order to reach high steam temperatures and in order to not make an efficient use of the heat that is stored in the HTF. But, small temperature differences require large heat exchange surfaces, which increases the cost. An appropriate trade-off between plant efficiency and material costs has to be found.

Figure 72: Heat exchanger size and plant efficiency
Solar tower receiver designs

Many different receiver designs using different HTF exist. One type is a receiver for generating both saturated and superheated steam, as presented in Figure 73.

![Image of receiver for generating both saturated and superheated steam](image)

**Figure 73:** Receiver for generating both saturated and superheated steam [33]

A well known example of a water/steam receiver used in a solar tower power plant is the receiver which was used in the plant Solar One (Barstow, USA). The Solar One receiver and a system diagram of the plant are described in the chapter Solar Tower Technology.

**Exercises**

- What are the two basic types of steam generators?
- What are the main design differences between a watertube and fire tube boiler?
- What is the definition of a steam generator (boiler)?
- Explain the function of an economiser.
- What are the design and the dimensioning of the steam generators?
4.2.3 Steam Turbines

The function of the steam turbine is to utilise the energy in the steam for shaft rotation by means of expanding the high-pressure live steam to the condenser pressure. A generator is connected with the turbine shaft to convert the mechanical energy (from shaft rotation) into electrical energy (electricity generation).

(1) Steam Turbine Types

There are two types of steam turbines, impulse and reaction turbines. Historically, steam turbines are categorised as impulse and reaction turbines from the thermodynamics point of view and into axial and radial flow turbines according to the flow direction of the steam [36, p. 205]. The terms impulse and reaction refer to the type of forces that are acting on the turbine inducing shaft rotation [37].

a) Impulse Turbine

The impulse turbine works according to Newton’s Second Law of Motion\textsuperscript{17}. In the turbine, impulsive forces are exerted on the vanes\textsuperscript{18}, which divert or change the flow of a fluid that is passing over it. The windmill is an example for a very basic impulse turbine. Windmills utilise the kinetic energy of the wind and convert it into mechanical power [37]. The impulse turbine may have several stages of fixed and moving vanes. A jet of steam is directed from the fixed vanes, which act like nozzles, onto the moving vanes. The fixed vanes convert the pressure of the fluid into velocity. In this process there is little or no pressure drop across its moving vanes as the passage between them has a constant cross-section area as shown in Figure 74. There is, however, a pressure drop over each of the fixed rows. Hence, due to the constant cross-section area there is no acceleration of the fluid at any point in the moving vanes [36, pp. 3-4].

The velocities shown in Figure 74 are the following: $C_1$ is the absolute velocity at the inlet and $C_2$ the absolute velocity at the outlet of the moving vane, $U$ is the vane rotation speed and $W_1$ the relative velocity at the inlet and $W_2$ the relative velocity at the outlet of the moving vane [38]. The numbering 0, 1 and 2 above the vanes correspond to the pressures given in the h-s diagrams in Figure 82.

\textsuperscript{17} Impulse = Change of Momentum and Impulsive Force = Rate of Change in Momentum; This is expressed with the equation: $F = \dot{m} \Delta v$, where $F$ is the impulsive force, $\dot{m}$ the mass flow rate (kg/s) and $\Delta v$ the change in velocity of the fluid (m/s). Because $\Delta v$ is the change in velocity in the direction of motion, we obtain the force for shaft rotation [37].

\textsuperscript{18} A vane is also called bucket or blade.
Figure 74: Arrangement of the vanes of a fully loaded multi-stage impulse turbine, edited from [38]
The theory of utilising the impulse in a steam turbine is shown below [36, p. 4].

1. If the turbine rotor is locked, the steam jet exerts its maximum force on the blades. As the blades are locked and do not move, no work is done.

2. If the blade is moving at ¼ of the steam jet velocity, the force on the blades is reduced, but some work is done by moving the blades.

3. Maximum work is done when the blades are moving at ½ jet speed. Relative velocity of steam leaving blades is zero.

4. The impulse principle is shown in the above figure.

Figure 79 shows the operating principle of an impulse turbine stage and the associated velocity diagram.
b) Reaction Turbine

In reaction turbines, reaction forces are exerted because the fluid velocity is changed. Newton's Third Law, which states that for every action there is an equal and opposite reaction, therefore applies for this type of turbine. The turbine is designed such that the fluid is accelerated by changing in the cross-sectional area between the passages of the vanes. Due to the principle of conservation of energy, the fluid pressure decreases when the velocity increases [37].

The expansion of the steam is realised in both the fixed and moving vanes. The moving vanes have two purposes, to utilise the energy exerted by the steam jet coming from the fixed vanes and to act as nozzles themselves [36].

Although called reaction turbine, the turbine makes use of both reaction and impulsive forces. The fixed vanes accelerate the steam resulting in a pressure drop in that particular stage. The steam passes onto the moving vanes, which also accelerate the steam and further cause a drop in pressure. Therefore, reaction forces (due to the change in fluid velocity), but also impulsive forces (due to the rate of change in momentum as the vanes change the direction of the fluid) are exerted on the moving vanes, both contribute to the rotational work being done. Reaction forces of about 50% are usual for reaction turbines [37]. This means that the pressure drop in the fixed and moving vanes is approximately identical.

Figure 80 shows the operating principle of a reaction turbine stage and the associated velocity diagram.
Shown in Figure 81 is the arrangement of the vanes of a multi-stage reaction turbine. For the description of the velocities, see the description used for Figure 74. The numbering 0, 1 and 2 above the vanes correspond to the pressures given in the h-s diagrams in Figure 82.

**Comparison of an Impulse Turbine stage with a Reaction Turbine stage**

In Figure 82, a stage of an impulse turbine and a stage of a reaction turbine is shown together with velocity triangles and the corresponding h-s diagrams. If the reaction \( r = 0 \), then the turbine is an impulse turbine and if \( r > 0 \), then the turbine is a reaction turbine. In the fixed vane of an impulse turbine stage, the entire pressure gradient is transformed into velocity. In the stage of a reaction
turbine, the pressure gradient is transformed into velocity to one half in the fixed vane and one half in the moving vane.

Figure 82: Comparison of an impulse turbine stage with a reaction turbine stage [39]

- **Exercises**
  - What steam turbine types exist?
  - How does a reaction turbine work?
  - How does an impulse turbine work?
(2) Dimensioning of Steam Turbines

The steam turbines in large power plants are very big and are composed of several stages. Hence, a steam turbine needs to be accommodated in a large building. The steam is usually not expanded in a single turbine, rather steam turbines are designed in configurations that typically include a high, medium (or intermediate) and low pressure turbine (see Figure 84).

![Figure 83: Steam turbine built by the Japanese company Hitachi](image)

Many CSP plants operate in a daily cycle involving a start-up and shut-down of the plant because their thermal energy storage does not allow for round-the-clock operation. In addition, in the case of (partial) shading of the collector field, the steam turbine has to be operated in part load. Hence such CSP plants require specially adapted steam turbines for dealing with the varying conditions and quick load changes. The blading, for instance, has to be designed in such a way that high efficiencies are reached over a wide range of operation modes, and the turbine has to withstand the high thermal stresses due to the frequent temperature variation because of frequent start-up and shut-down.
In the last 20 years turbines were developed for the special CSP requirements. Siemens, for instance, designed appropriate turbines for the power range from 1.5 to 250 MW.

<table>
<thead>
<tr>
<th>Type</th>
<th>Steam parameters</th>
<th>Output (MW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>SST-110</td>
<td>130 bar, 530°C</td>
<td></td>
</tr>
<tr>
<td>SST-120</td>
<td>120 bar, 530°C</td>
<td></td>
</tr>
<tr>
<td>SST-300</td>
<td>120 bar, 520°C</td>
<td></td>
</tr>
<tr>
<td>SST-400</td>
<td>140 bar, 540°C</td>
<td></td>
</tr>
<tr>
<td>SST-600</td>
<td>140 bar, 540°C</td>
<td></td>
</tr>
<tr>
<td>SST-700</td>
<td>165 bar, 585°C</td>
<td>Dual casing / reheat or non-reheat</td>
</tr>
<tr>
<td>SST-800 &amp; SST-500</td>
<td>140 bar, 540°C</td>
<td>Single casing / reheat or non-reheat</td>
</tr>
<tr>
<td>SST-900</td>
<td>165 bar, 585°C</td>
<td></td>
</tr>
</tbody>
</table>

**Figure 85:** Steam turbines suitable for parabolic trough power plants [42]

The following figure shows the rotor of the Siemens SST-700 turbine with dual casing machinery train and with intermediate superheating. This turbine was integrated, for instance, in the Andasol I and II plants in Spain.

**Figure 86:** Rotor assembly of Siemens SST-700 [42]

The turbine/generator set of Andasol III was manufactured by MAN Diesel & Turbo in Oberhausen, Germany. The turbine weighs 160 tonnes and consists also of a high-pressure and a low-pressure module with dual casing machinery train with intermediate superheating.
Figure 87: MAN turbo steam turbine: high pressure and low pressure turbines [43]
4.2.4 Gas Turbines

Gas turbines are mainly used in a combined-cycle configuration. Apart from the use in conventional combined-cycle plants, gas turbines are also of particular interest for the application in CSP technologies, i.e. a CSP plant can be hybridised with a gas turbine. This includes the hybridisation of a solar tower with air receiver technology and ISCCS plants.

Power generation gas turbines can be classified into two categories [28, p. 168]:
- Heavy-duty industrial gas turbines, which were originally derived from the steam turbine or jet industry
- Aeroderivative gas turbines, comprising a jet turbine modified for industrial duty and frequently incorporating a separate power turbine

Gas turbines are further distinguished in two types, those with:
- open cycle and internal heat supply
- closed cycle and external heat supply

An open cycle gas turbine power plant consists fundamentally of a
- compressor
- combustion chamber
- turbine

A closed cycle gas turbine power plant consists fundamentally of a
- compressor
- combustion chamber & heat exchanger (closed cycle)
- turbine

(1) Open and closed cycle gas turbines

a) Open cycle gas turbines
In open cycles, ambient air is sucked into the compressor where it is being compressed. The gas is added and burned in the combustion chamber where the temperature is raised. The hot gas is then expanded in the turbine and exhausted into the atmosphere. The working fluid air must be constantly replaced in the process. The majority of gas turbine power plants work on the open cycle principle [44, p. 268].

In order to improve the efficiency of the gas turbine, intercoolers, reheaters and regenerators (heat exchangers) can be included in the cycle.

Figure 88 below shows a schematic of an open loop gas turbine. The symbol \( M \) stands for motor (which is used for the start-up phase), \( C \) for compressor, \( CC \) for combustion chamber, \( T \) for turbine and \( G \) for generator [38, p. 60].
The following two figures (Figure 89 and Figure 90) illustrate a single and a twin-shaft gas turbine system.
The pressure ratio of modern industrial gas turbine plants typically lies within the range of 17:1 to 35:1. In comparison, modern aircraft engines have a pressure ratio exceeding 40:1 [45, p. 35].

Some advantages and disadvantages of the open cycle gas turbine are given below [44, pp. 268-269]:

Advantages:
1) *No warm-up time required* – every gas turbine has a motor for start-up, which brings the turbine to its rated operational speed. The gas turbine handles without problems the rapid temperature increase after fuel ignition.
2) *Highly effective in covering peak load demand* due to its short start-up time and take-up of the load frequency.
3) *Diversity of fuel* – gas turbines handle a large range of hydrocarbon fuels, including high-octane petrol and heavy diesel oils.
4) *Little space requirements*
5) *No cooling water required* (except for those gas turbines designed with an intercooler). Cooling is realised with the ambient air. Most of the air sucked in by the compressor is used for cooling i.e. only a small portion is provided to the combustion chamber.
6) *Application in a solar tower power plant and dish system possible*

Disadvantages:
1) *Low part load efficiency* – compressor power requirements are considerably high.
2) *Sensitive to weather conditions* i.e. changes in atmospheric air temperature, pressure and humidity.
3) *Dust filtering system requires close monitoring* – filtering of dust is highly important to prevent damage to the compressor blades (dust causes erosion and deposits on blades). Moreover it must be ensured that the filter is never clogged.
4) *Unit rating limited* as backpressure equal to atmospheric pressure
b) **Closed cycle gas turbines**

In closed cycles the working fluid air or another suitable gas is continuously recirculated. After the working fluid has been compressed, the temperature is raised in a heater at constant pressure. The heating is realised with an external heat source. In the next stage the gas which is at high-pressure and high-temperature is expanded in the turbine. The expanded gas is then cooled down to its initial temperature in a cooler that uses an external cooling source. The closed cycle is completed by passing the air back to the compressor, where the recirculation begins [44, p. 269]. Figure 91 shows a closed cycle gas turbine plant.

![Figure 91: Closed cycle gas turbine plant [44]](image)

As with the open cycle design, the efficiency of closed cycle gas turbines can be improved when intercoolers, reheaters and regenerators (heat exchangers) are included in the cycle.

Some advantages and disadvantages of the closed cycle gas turbine are given below [44, pp. 269-271]:

**Advantages:**

1) *No compressor blade erosion or fouling* because the working fluid is already clean and is reused in the closed cycle. The filtering of air is therefore not required. As the closed cycle completely eliminates the problem of corrosion and abrasion, the life of the gas turbine plant is extended.

2) *Low maintenance expenses* for the above reasons and *high reliability* of the plant.
3) Load requirements can be adapted by varying the absolute pressure and mass flow of the working fluid (the pressure ratio, temperature and air velocity are controlled to remain at near constant level).

4) Open cycle gas turbines are bound to backpressure equal to the atmospheric pressure, whilst the backpressure of a closed cycle gas turbine can be adjusted to higher pressures. This has the advantage that the unit rating can be increased roughly in proportion to the backpressure increase.

5) High density of working fluid when internal pressure range is increased: The higher the density, the better the heat transfer properties in the heat exchanger.

6) Low quality fuels can be used in the combustion chamber and heat exchanger.

Disadvantages:

1) Design more complicated due to higher internal pressures. This involves higher quality materials leading to an increase in investment costs.
2) External cooling with considerable amount of cooling water is required in pre-cooler
3) Large sized heat exchanger required for heating of working fluid
4) Poor response to load variations compared to open cycle gas turbine plant

Exercises

– Describe the working principle of an open cycle gas turbine.
– Name at least 4 advantages and 3 disadvantages of an open cycle gas turbine.
– Describe the working principle of a closed cycle gas turbine.
– Name at least 4 advantages and 3 disadvantages of a closed cycle gas turbine.
– What components does a gas turbine power plant fundamentally consist of?

(2) Dimension of industrial gas turbines

Usually heavy-duty industrial gas turbines are designed as single-shaft machines when used to drive generators for generating outputs greater than approximately 30 MWₑ. Modern heavy-duty industrial gas turbines produce electricity up to 340 MWₑ under ISO conditions¹⁹.

¹⁹ Gas turbine manufacturers quote the performance of the gas turbines at ISO ambient conditions of 15°C, 1.013 bar, and 60% relative humidity [28, p. 50]
Figure 92: Heavy-duty industrial gas turbine designed for an output of 340 MW\textsubscript{e} [28, p. 170]

The main characteristic data of modern gas turbines for power generation is shown in Table 3.

### Table 3: Main characteristic data of modern gas turbines for power generation [28, p. 171]

<table>
<thead>
<tr>
<th>Power Output (ISO condition)</th>
<th>Up to 340 MW</th>
</tr>
</thead>
<tbody>
<tr>
<td>Efficiency (ISO conditions)</td>
<td>34 – 40%</td>
</tr>
<tr>
<td>Gas turbine inlet temperature (ISO 2314)</td>
<td>1100 – 1350°C</td>
</tr>
<tr>
<td>Exhaust gas temperature</td>
<td>450 – 650°C</td>
</tr>
<tr>
<td>Exhaust gas flow</td>
<td>50 – 820 kg/s</td>
</tr>
</tbody>
</table>
(3) **Methods for increasing the efficiency**

There are three common ways to increase the efficiency of a gas turbine plant:

- using a regenerator
- using a reheat combustor
- compression with intercooling

**Regenerator:**

The temperature of the exhaust gas leaving the gas turbine is far above ambient temperature. Instead of discarding the exhaust gas directly to the surroundings, a heat exchanger called a regenerator can be used, which has the purpose of preheating the air exiting the compressor before entering the combustor. Using a regenerator reduces the amount of fuel required for the combustion. Figure 93 shows a regenerative air-standard gas turbine cycle and the corresponding T-s diagram. The regenerator shown is a counterflow heat exchanger. A counterflow heat exchanger works in such a way that the hot turbine exhaust gas and the cooler air leaving the compressor pass in opposite directions. The T-s diagram shows that the turbine exhaust gas is cooled from state 4 to state y, while the air exiting the compressor is heated from state 2 to state x. When utilising the heat from the exhaust gas, then the external heat source of the combustor only needs to increase the air temperature from state x to state 3, and not from state 2 to state 3 if no regenerator were used [2, p. 472].

![Figure 93: Regenerative air-standard gas turbine cycle [2, p. 472]](image)

**Reheat:**

The temperature of the gaseous combustion products which exit the combustor must be limited before entering the turbine due to metallurgical reasons. The temperature can be controlled by providing more air than is required for burning the fuel in the combustor, i.e. excess air. As a result, the gases exiting the combustor still contain sufficient air to support the combustion of additional fuel. Some gas turbine power plants take advantage of the excess air by using a multistage turbine with a reheat combustor between the first and second turbine stage. With this combination the net work per unit of mass flow can be increased.

Figure 94 shows an ideal gas turbine with two turbine stages and with reheat, and the corresponding T-s diagram. With reference to the T-s diagram, the first stage expands the gas from state 3 to state a. The gas is then reheated at constant pressure from state a to state b. A second
expansion occurs from state b to state 4. The T-s diagram shows also the ideal Brayton cycle without reheat, 1-2-3-4’-1, for comparison. As can be seen of the T-s diagram, the lines of constant pressure diverge slightly with increasing entropy. Because of this, the total work of the two-stage turbine is greater than that of a single expansion from state 3 to 4’. For this reason, the net work for the reheat cycle is greater than for a cycle without reheat. Although the net work increases with reheat, it does not necessarily mean that the cycle thermal efficiency would increase. This is due to a greater total heat addition that would be required. As can further be seen on the T-s diagram, the temperature of the exhaust gas is higher at state 4 than at state 4’. This enhances the potential for regeneration. Using reheat and regeneration together increases the thermal efficiency significantly [2, p. 477].

Figure 94: Ideal gas turbine with reheat [2, p. 477]

*Compression with intercooling:*

To increase the net work of a gas turbine, the compressor work input can be reduced by means of multistage compression with intercooling. Figure 95 shows a two-stage compression with intercooling and the corresponding p-v and T-s diagrams. The intercooling takes place between compressor stage 1 and compressor stage 2. An internally reversible process is considered here.

The following processes occur:

- Process 1-c is an isentropic compression taking place from state 1 to state c. At state c the pressure is $p_c$
- In process c-d, constant-pressure cooling occurs from temperature $T_c$ to $T_d$
- Process d-2 is an isentropic compression from state d to state 2

The p-v diagram shows the shaded area 1-c-d-2-a-b-1, which is the work input per unit of mass flow. The p-v diagram also depicts the processes without intercooling. In this case the gas would be compressed isentropically in a single stage from state 1 to state 2’ and the work is given by the enclosed area 1-2’-a-b-1. Intercooling leads to a reduction in work. This is represented by a crosshatched area on the p-v diagram [2, p. 479]
Regenerative gas turbine with intercooling and reheat:
When reheat between turbine stages and intercooling between compressor stages is used, two important advantages are provided: An increase in the net work output, and an enhancement of the potential for regeneration. Therefore, when reheat and intercooling are used together with regeneration, then a substantial improvement in performance can be realised. Such an arrangement with intercooling, reheating and regeneration is shown in Figure 96. The gas turbine shown has two compression and two turbine stages. The corresponding T-s diagram is drawn to indicate irreversibilities in the compressor and turbine stages. The diagram does not show the pressure drops which normally would occur when the working fluid passes through the intercooler, regenerator, and combustors.
Exercises

- Name and explain in detail the three methods of increasing the efficiency of the gas turbine plant.
(4) Pressure and temperature characteristics
Figure 97 shows the pressure and temperature characteristics of an example gas turbine. Here the compressed air pressures are 12 – 15 bar at a temperature in the combustion chamber of nearly 1150°C. The temperature of the exhaust gas is between 450 – 550°C.

![Figure 97: Pressure and temperature characteristics of a gas turbine, edited from [46], [38]](image)

(5) Combustors
Generally, all combustors have the same functions, but differ in their arrangement. Their function is to increase the temperature of high-pressure gas by burning fuel. The combustor of a gas turbine requires only little air inside the combustion chamber and uses most (90%) for cooling and mixing. The air is diffused prior entering the combustion chamber. Newer combustor designs use steam for cooling [45, p. 33].

Three typical combustor designs exist:
- Tubular (single can)
- Tubo-annular
- Annular

The tubular (single can) and tubo-annular combustor types are described in more detail in the following section.

The tubular single can side combustor design is preferred by many European industrial gas turbine manufacturers. These large single combustors have the advantage that its design is quite simple and it has a long operation life because of its low heat-release rates. The produced combustor sizes range from small units (15.24 cm in diameter and 0.3 m in height) to far larger units (3 m in diameter and 3 – 12 m in height). In large combustors, special tiles as liners are used. These can easily be replaced, in the event that some are damaged [45, p. 36]. A typical single can side combustor is shown in Figure 98 and is illustrated with parts of a gas turbine in Figure 99. Tubular
Combustors can be of “straight-through” or “reverse-flow” design. The large single-can combustors are mostly of the reverse-flow design. In the reverse-flow design the air enters the turbine through the annulus which is located between the combustor can and the hot gas pipe and from there passes between the liner and the combustor can. It then enters the combustion region at various points of entry. Only about 10% of the air is required to enter the combustion zone, about 30 – 40% of the air is required for cooling and the remaining air is used in the dilution zone [45, p. 388].

Figure 98: Typical single can side combustor [45, p. 40]

Figure 99: Single can combustor (from Brown Boveri Turbomachinery, Inc.) [45, p. 387]
US American companies, on the other hand, design industrial gas turbines with a tubo-annular or can-annular type combustor, as shown in Figure 100. The tubo-annular type combustor is the most commonly used combustor type in gas turbines. An illustration of an industrial-type can-annular combustor and some parts of a gas turbine are shown Figure 101. The preference for using tubo-annular or can-annular type combustors are because of ease of maintenance and the better temperature distribution compared to side single-can combustor types and the design can be “straight-through” or “reverse-flow”. A typical reverse flow can-annular combustor is shown in Figure 102. Most of the tubo-annular or can-annular type combustors are of the reverse-flow design in industrial gas turbines [45, p. 389].

![Tubo-annular or can-annular combustor for a heavy-duty gas turbine](image1)

**Figure 100**: Tubo-annular or can-annular combustor for a heavy-duty gas turbine (from General Electric Company) [45, p. 388]

![Industrial-type can-annular combustor](image2)

**Figure 101**: Industrial-type can-annular combustor (from Solar Turbines Incorporated) [45, p. 391]
Figure 102: Typical reverse flow can-annular combustor [45, p. 37]

► Exercises

– Name the 3 typical combustor designs.
– Which combustor type is preferred by European companies and which by US American companies?
Electric Generators

Introduction

Electric generators in the most general sense convert some kind of energy to electrical energy. In a solar thermal power plant, where radiation is converted to heat, which is converted to mechanical energy, the generator converts mechanical energy to electrical energy. Indeed, “generator” is commonly understood in a narrower sense as a device that converts mechanical energy, in most cases rotational energy, to electrical energy. Such generators work on the basis of electromagnetic induction.

Electromagnetic induction is the production of voltage across a conductor if this conductor is moved through a magnetic field. The existence of induction can be understood on the basis of the Lorentz force that is exerted on electrical charges that move through a magnetic field. If a conductor is moved through a magnetic field, such that the lines of magnetic flux are subtended, the Lorentz force moves the charged particles (for instance electrons) in the conductor, which causes a voltage between the ends of the conductor.

Figure 103: Basic configuration for electric induction

The Lorentz force is determined as:

$$\vec{F}_L = q(\vec{v} \times \vec{B}),$$

where \( q \) is the electric charge of the particles in the conductor, \( \vec{v} \) the vectorial velocity of them and \( \vec{B} \) the magnetic field.

In order to use the Lorentz force for the generation of electrical power, thus, the following elements are needed: A magnetic field must be provided by permanent magnets or by electromagnets. It can be static or moving, i.e. (in the common generator geometry) rotating. The generator must contain conductor windings, the so-called armature, in which the electric current is induced. In case of a static magnetic field, the armature has to be moved, i.e. (in the normal generator geometry) it has to rotate. As the mechanical energy is supplied by rotation, generators have to contain a stationary and a rotating component, i.e. a stator and a rotator.

According to the type of current, which is to be generated, the armature has to meet certain conditions. If three-phase alternating current is to be produced, which is the case we will consider
here, then, in the simplest case, the armature contains three 120°-shifted coils, such that the induced alternating currents in them are also 120°-phase shifted.

There are two common types of electrical generators, that convert mechanical energy (in form of rotational energy) to electrical energy: synchronous and asynchronous generators. They do the inverse energy conversion process, which is done by electric motors. Indeed, many devices can be used both as generators and as motors. That’s why they are also called, more generally, synchronous and asynchronous machines.

**Synchronous and asynchronous generators**

A synchronous machine uses a stable magnetic field, which is provided by permanent magnets or, with direct current (the exciting current), by electromagnets. Concerning the relative movement between the magnetic field and the armature, it is physically the same whether the first is moved and the latter is fixed or vice versa. So, synchronous machines can use a stationary magnetic field, provided by the stator, and a rotating armature, or a static armature in the stator and a revolving magnetic field. The first type are revolving-field machines (or stationary-armature machines), the second type are stationary-field machines (or revolving-armature machines). In power plants, revolving-field machines dominate. That’s why the illustrations show only machines of this type.

If a synchronous machine is operated as a motor, a three-phase AC is supplied to the armature and a rotating magnetic field evolves. This rotating field interacts with the machine’s stable magnetic field, which generates a torque and drives the rotator. If a synchronous machine is operated as a generator, the rotator is moved and a three-phase AC is induced in the armature.

The difference between motor operation and generator operation is reflected in a phase difference between the rotating magnetic field of the rotator and the rotating stator field, the so-called polar wheel angle. The rotating magnetic field of the rotator and the rotating stator field rotate always at the same frequency, but there may be a phase difference between them. This phase difference is correlated to the question whether the machine is used as a motor or as a generator: In motor operation, the supplied stator field is ahead of the rotator field, and in generator operation, the rotator field is ahead of the stator field.

The polar wheel angle \( \vartheta \), i.e. the phase difference between the rotating stator field and the rotator field, has to be maintained within the stability range of \(-90^\circ < \vartheta < 90^\circ\). Outside this range, the machine gets instable (being in motor operation, it stops, being in generator operation, it overaccelerates). The polar wheel angle is a measure of the converted power or the generated or supplied torque. At 0° the torque is 0, at 90° the breakdown torque is reached. This point marks the maximum power the machine can accept. If the torque supplied by the turbine gets higher, the generator is over-reved and most probably destroyed.
The synchronous machine is called so because the rotator rotates in accordance with the frequency of the supplied (in motor operation) or generated (in generator operation) three-phase AC: In case of one-pole-pair machines (see below), the rotational frequency of the rotator is the same as the three-phase AC. In the general case, the ratio between rotational speed of the rotator and frequency of the AC is fixed (see below). This means that in a grid-connected operation of a synchronous machine the rotational speed of its rotator is determined by the grid frequency and by the constructive features of the machine and can not vary if the grid frequency does not vary.

Asynchronous machines are very simple and reliable generators. They consist of a stator where a three-phase alternating current produces a rotating magnetic field (contrary to the synchronous machine, where the magnetic field is provided by permant magnets or by direct current). The stator with its three-phase winding has principally the same structure like the stator of a revolving-field synchronous machine. The very simple rotator, which has the form of a cage rotator, consists (quite simply) of a number of copper or aluminium bars, which are connected electrically by aluminium end rings. It is quite different from the rotator of the synchronous machine, which (in the case of a revolving-field machine) has to provide a stable magnetic field.

If an asynchronous machines works as a motor, the rotating current is connected to the stator and produces a rotating magnetic field. This rotating field induces a strong current in the bars of the rotator, such that the rotator develops its own (variable) magnetic field. The interrelation between these magnetic fields provokes electromagnetic forces that drag the rotator along. In order to maintain the rotator in movement, it is necessary that the stator field rotates slightly faster than the rotator. That means that there is a slip (frequency difference) between the rotating field and the rotation of the rotator. The asynchronous machine is called so because of this difference between the rotational frequencies of the rotating field and the rotator. The rotator does not rotate in accordance with the rotating field of the stator and its rotational speed is not determined by the latter, in fact, the rotational speed of the rotator changes if the mechanical load of the motor changes.

If the asynchronous machine works as a generator, the slip is inverted, i.e. the rotator rotates faster than the magnetic field. In this case, the electromagnetic forces, which are caused by the rotating field, and the field generated by the induced current in the rotor provoke a moment that tries to slow down the rotor. In order to maintain the rotation of the rotor, a torque has to act, i.e. mechanical energy has to be supplied. This supplied energy is converted to electric energy in the form of a reinforcement of the current in the stator. In the stator, the provision of the magnetic field

---

Figure 104: Stable motor and generator
operation range of a synchronous machine
by rotating current takes place as well as the induction of the generated electrical energy. This is also a difference to the synchronous machine, where the magnetic field is provided in one place (e.g. in the rotator) and the electric energy is generated in another place (e.g. in the stator). The difference between the rotator speed and the rotating speed of the stator field is a measure for the converted power or the generated or supplied torque. It is in the range of one to ten percent. As in the synchronous machine, there is a breakdown torque which indicates the maximum power that is accepted by the machine. If the supplied torque (in generator operation) gets higher than the breakdown torque, then the machine over-revs.

As already mentioned, synchronous and asynchronous generators show the following difference when connected to a grid: The rotator of a synchronous generator always rotates at a frequency that is dictated by the grid. It rotates, thus, at a constant frequency. The rotational frequency of the rotator of an asynchronous generator, on the contrary, can vary. Variations in the torque increases or decreases the speed of the rotator. This is an advantage in case of frequent torque variations because the generator reacts smoothly and there will be less wear and tear in the gear box. That’s why asynchronous generators are very common in wind power plants and smaller hydro power plants, where there are more fluctuations in the supplied torque. In thermal power plants, as for instance in solar thermal power plants, where the supplied mechanical power can be regulated very well and where nonintentional fluctuations will not happen, synchronous generators are used. Also in large hydro power plants, synchronous generators are used.

Synchronous generators have a very valuable advantage over asynchronous generators: They can generate reactive power, while asynchronous generators cannot.\[^{20}\] This means

\[^{20}\text{Reactive power is the portion of power flow in an alternating current transmission system that is temporarily stored in the form of electric or magnetic fields in inductive and capacitive elements of the grid and which is returned to the source. It oscillates with the AC frequency between the inductive and capacitive elements and the power source.}\]
that synchronous generators can be used for reactive power control, which is important in order to guarantee efficient power flows.

Synchronous generators in solar thermal power plants
Synchronous generators are used in thermal power plants. Among synchronous generators we can distinguish between turbogenerators and salient-pole generators. Turbogenerators have a massive cylindrical rotator that has slots in which the windings for the exciting current are located. Salient-pole generators, on the other hand, have an axis with salient poles. This constructive difference is related to an operative difference: Turbogenerators are operated at high rotational frequencies, while salient-pole generators are operated at lower rotational frequencies. The rotators of turbogenerators have a massive form in order to resist the high centripedal forces that result at high rotational speeds. Remember that the centripetal force is proportional to the square of the rotational speed \( F_c = m \omega^2 r \). As the centripetal force is proportional to the radius, the rotator diameter of turbomachines is quite small compared to the rotator diameter of salient-pole generators. On the other hand, turbogenerators are longer than salient-pole generators of the same power class.

The difference of rotational frequency is related to a difference in the number of pole pairs \( p \). The number \( p \) refers to the number of pole pairs (or half of the number of poles) of the exciting current, which in the case of revolving-field machines are located in the rotator. Additionally, each phase has \( p \) separated coils in the armature. Now, as the frequency of the generated three-phase AC of a grid-connected generator has to be constant (in accordance with the grid frequency), the number of pole pairs has an effect on the rotational frequency of the rotator. If the pole pair number is 1, then at each revolution one complete period is generated in each armature coil. If the pole pair number is 2, than half a revolution generates a complete period in each coil, etc. A generator with the pole pair number 2, then, has to be driven with half of the frequency in relation to a generator with the pole pair number 1, if the grid frequency is the same. A generator with the pole pair number 3 has to be driven with one third of the frequency of a one-pole-pair machine, etc.

\[ p = 1 \quad \begin{array}{c} 1 \\ 2' \\ 3' \\ 2 \\ 3 \\ 1' \end{array} \quad p = 2 \quad \begin{array}{c} 1 \\ 2' \\ 3' \\ 1'' \end{array} \]

**Figure 106:** Synchronous machine with one pole pair (left) and with two pole pairs (right)

The general interrelation between pole pair number \( p \), grid frequency \( f \), and the synchronous rotational frequency of the rotator \( v_s \) in [Hz] is the following:

\[ v_s = \frac{f}{p} \quad (33) \]

or, the rotational speed \( n_s \) in [min\(^{-1}\)] is:

\[ n_s = 60 \frac{1}{\text{min}} \cdot v_s = 60 \frac{s}{\text{min}} \cdot \frac{f}{p} \quad (34) \]
If we consider a 50Hz grid, then a generator with one pole pair has a rotational speed of 50Hz or 3000min⁻¹. A generator with two pole pairs has a rotational speed of 1500min⁻¹. A generator with three pole pairs has a rotational speed of 1000min⁻¹, etc. Turbogenerators have normally one or two pole pairs. Salient-pole generators normally have more pole pairs (between 3 and 50).

Steam and gas turbines are most efficient if they are operated at high frequencies. In many cases these turbines are directly connected to the generator and require, thus, generators that allow high rotational frequencies. That’s why turbogenerators are used in solar thermal power plants as well as in fossil fuel power plants and in nuclear power plants. Salient-pole generators are used in hydro power plants, where the turbines are run at a lower rotational speed.

![Synchronous generator](image)

**Figure 107**: Turbogenerator and salient-pole generator

Turbogenerators have to be cooled. Air, hydrogen and water are used for cooling. Air cooling is used in smaller and medium sized units. There is a tendency to use air cooling more and more in larger units, at the moment until a capacity of 400MVA.²¹ Air cooling is the most cost-effective solution. However, air has some disadvantages like a lower specific heat capacity and a lower heat conductivity than hydrogen or water. That’s why larger turbogenerators are cooled with hydrogen, hydrogen (rotator) and water (stator) or completely with water (ordered by size).²² For the size that is needed for CSP plants (realized until 80MW, planned until 250MW), air cooling is sufficient.

---

²¹ The capacity of generators is generally indicated in VA, i.e. volt-ampere, which is the unit of the apparent power. The apparent power is the combination of the active power (the power which works at the load) and the reactive power (see preceding footnote).

²² See Ginet et al. [47]
Problems

1. What is the rotational speed of a steam turbine that is directly connected to a two-pole synchronous generator, when the power plant is connected to the national grid
   a) in Germany
   b) in the USA?

2. a) On which properties does the voltage in the assembly in Figure 103 depend? What could be done in order to increase the voltage?
   b) Image that the conductor loop is closed with a load. What could be done additionally to the measures mentioned in (a) to increase the generated power?
   c) What does this mean for a synchronous generator? Which of the mentioned measure would be applicable to enhance the power of a grid-connected synchronous generator and which cannot be applied?

Answers

1. a) \( f = 50 \text{Hz}, \quad n_s = 60 \frac{\text{rpm}}{\min} \cdot \frac{f}{p} = 60 \frac{\text{rpm}}{\min} \cdot \frac{50}{2} = 1500 \text{min}^{-1} \)
   b) \( f = 60 \text{Hz}, \quad n_s = 60 \frac{\text{rpm}}{\min} \cdot \frac{60}{2} = 1800 \text{min}^{-1} \)

2. a) i) A stronger magnetic field increases the generated voltage. The stronger field can be reached by stronger magnets or by a smaller distance between them.
   ii) Faster rotation of the conductor loop increases the generated voltage.
   iii) Changing the conductor loop by a coil, i.e. by a series connection of many loops, increases the voltage.
   iv) A larger conductor loop (longer and/or wider) increases the generated voltage.

   b) v) If the loop is closed, the loop could be parallely connected with more loops, in order to increase the generated electric power.

   c) i), iii) and v) are applicable. In the case of (i) stronger magnetic fields have to be generated through higher exciting currents in order to increase the generated power.
   ii) Modification of the rotating speed is not applicable to grid-connected synchronous machines because their rotational speed is fixed.
   iv) First, we have to take into account that the geometry of a synchronous generator is quite different from the assembly in Figure 103. If we consider a revolving-field generator, than the magnetic field is generated by a revolving coil in the rotator and the armature is in the stator. But, increasing sizes allow higher power generation also in this geometry because of higher speeds at higher diameters (at constant rotational frequencies) and because of longer loops in longer stators.
However, certain limits have to be respected in the case of high-speed turbogenerators: The diameter cannot be increased at will because of the centripetal forces, which are proportional to the diameter. Also the length is limited because of high gravitational loads and because of the difficulty to balance very long rotators. However, turbogenerators are quite long in relation to their diameter.
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