Chapter 3
Thermofluid Engineering Basics

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## Nomenclature

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<th>Symbol</th>
<th>Meaning</th>
<th>Unit</th>
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</thead>
<tbody>
<tr>
<td>$A$</td>
<td>area</td>
<td>m²</td>
</tr>
<tr>
<td>$A_s$</td>
<td>surface area</td>
<td>m²</td>
</tr>
<tr>
<td>$c_p$</td>
<td>specific heat (capacity) at constant pressure</td>
<td>J/(kg K)</td>
</tr>
<tr>
<td>$C$</td>
<td>a constant</td>
<td></td>
</tr>
<tr>
<td>$E$</td>
<td>energy</td>
<td>J</td>
</tr>
<tr>
<td>$E_{b b}$</td>
<td>is the respective emissive power for a black body</td>
<td>W/m²</td>
</tr>
<tr>
<td>$E_\lambda$</td>
<td>emissive power of the investigated body with respect to $\lambda$</td>
<td>W/m²</td>
</tr>
<tr>
<td>$\dot{E}_g$</td>
<td>rate of energy production</td>
<td>W</td>
</tr>
<tr>
<td>$F$</td>
<td>force</td>
<td>N</td>
</tr>
<tr>
<td>$F_t$</td>
<td>tangential force</td>
<td>N</td>
</tr>
<tr>
<td>$g$</td>
<td>acceleration due to gravity</td>
<td>m/s²</td>
</tr>
<tr>
<td>$G$</td>
<td>irradiation</td>
<td>W/m²</td>
</tr>
<tr>
<td>$Gr$</td>
<td>Grashof number</td>
<td></td>
</tr>
<tr>
<td>$h$</td>
<td>altitude</td>
<td>m</td>
</tr>
<tr>
<td>$h_{c, h}$</td>
<td>convection enthalpy</td>
<td>J/kg</td>
</tr>
<tr>
<td>$I$</td>
<td>current</td>
<td>A (ampere)</td>
</tr>
<tr>
<td>$I_{b b}$</td>
<td>intensity of a black body</td>
<td>W/m² sr µm</td>
</tr>
<tr>
<td>$I_{\lambda, e}$</td>
<td>radiation intensity of the investigated body (with respect to $\lambda$)</td>
<td>W/m² sr µm</td>
</tr>
<tr>
<td>$k$</td>
<td>thermal conductivity</td>
<td>W/(m K)</td>
</tr>
<tr>
<td>$KE$</td>
<td>kinetic energy</td>
<td>J</td>
</tr>
<tr>
<td>$L$</td>
<td>length</td>
<td>m</td>
</tr>
<tr>
<td>$L_c$</td>
<td>characteristic length</td>
<td>m</td>
</tr>
<tr>
<td>$m$</td>
<td>mass of an object</td>
<td>kg</td>
</tr>
<tr>
<td>$N$</td>
<td>number of coulombs of electrical charge</td>
<td></td>
</tr>
<tr>
<td>$Nu$</td>
<td>Nusselt number</td>
<td></td>
</tr>
<tr>
<td>$p$</td>
<td>pressure</td>
<td>bar</td>
</tr>
<tr>
<td>$PE$</td>
<td>potential energy</td>
<td>J</td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandtl number</td>
<td></td>
</tr>
<tr>
<td>$\dot{q}$</td>
<td>rate of energy generation per unit volume of a volume element</td>
<td>W/m³</td>
</tr>
<tr>
<td>$\dot{q}_{12\text{conv}}$</td>
<td>convection heat transfer rate between the HTF and inside wall of the absorber pipe per unit receiver length</td>
<td>W/m</td>
</tr>
<tr>
<td>$\dot{q}_{56\text{conv}}$</td>
<td>convection heat transfer rate between the outer surface of the glass envelope to the atmosphere per unit receiver length</td>
<td>W/m</td>
</tr>
<tr>
<td>$Q$</td>
<td>heat (or total heat transfer)</td>
<td>J</td>
</tr>
<tr>
<td>$\dot{Q}$</td>
<td>heat flow (or heat transfer rate)</td>
<td>W</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{cond}}$</td>
<td>heat flow by conduction</td>
<td>W</td>
</tr>
<tr>
<td>$\dot{Q}_e$</td>
<td>rate at which energy is emitted</td>
<td>W</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{rad.net}}$</td>
<td>net rate of radiation exchange</td>
<td>W</td>
</tr>
<tr>
<td>$r$</td>
<td>radius</td>
<td>m</td>
</tr>
<tr>
<td>$R$</td>
<td>resistance</td>
<td>Ω (Ohm)</td>
</tr>
<tr>
<td>$Ra$</td>
<td>Rayleigh number</td>
<td></td>
</tr>
<tr>
<td>Symbol</td>
<td>Definition</td>
<td>Units</td>
</tr>
<tr>
<td>--------</td>
<td>------------</td>
<td>-------</td>
</tr>
<tr>
<td>Reynolds number</td>
<td>$Re$</td>
<td>-</td>
</tr>
<tr>
<td>Specific entropy</td>
<td>$s$</td>
<td>J/(kg K)</td>
</tr>
<tr>
<td>Entropy flow</td>
<td>$\dot{S}_Q$</td>
<td>J/(s K) or W/K</td>
</tr>
<tr>
<td>Entropy</td>
<td>$S$</td>
<td>J/K</td>
</tr>
<tr>
<td>Time</td>
<td>$t$</td>
<td>s</td>
</tr>
<tr>
<td>Temperature</td>
<td>$T$</td>
<td>°C or K</td>
</tr>
<tr>
<td>Surface temperature</td>
<td>$T_s$</td>
<td>°C or K</td>
</tr>
<tr>
<td>Temperature of the surroundings (enclosure)</td>
<td>$T_{sur}$</td>
<td>°C or K</td>
</tr>
<tr>
<td>Saturation temperature</td>
<td>$T_{sat}$</td>
<td>°C or K</td>
</tr>
<tr>
<td>Wall temperature</td>
<td>$T_{wall}$</td>
<td>°C or K</td>
</tr>
<tr>
<td>Specific internal energy</td>
<td>$u$</td>
<td>J/kg</td>
</tr>
<tr>
<td>Free stream velocity</td>
<td>$u_\infty$</td>
<td>m/s</td>
</tr>
<tr>
<td>Internal energy</td>
<td>$U$</td>
<td>J</td>
</tr>
<tr>
<td>Velocity</td>
<td>$v$</td>
<td>m/s</td>
</tr>
<tr>
<td>Volume</td>
<td>$V$</td>
<td>m³</td>
</tr>
<tr>
<td>Work</td>
<td>$W$</td>
<td>J</td>
</tr>
<tr>
<td>Electrical work done</td>
<td>$W_e$</td>
<td>J</td>
</tr>
<tr>
<td>Power, or rate of work</td>
<td>$\dot{W}$</td>
<td>W</td>
</tr>
<tr>
<td>Electrical power</td>
<td>$\dot{W}_e$</td>
<td>W</td>
</tr>
<tr>
<td>Characteristic length</td>
<td>$x$</td>
<td>m</td>
</tr>
<tr>
<td>Coordinate direction</td>
<td>$x_{fd,h}$</td>
<td>m</td>
</tr>
</tbody>
</table>

**Greek letters**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal diffusivity</td>
<td>$\alpha$</td>
<td>m²/s</td>
</tr>
<tr>
<td>Convection heat transfer coefficient</td>
<td>$\beta$</td>
<td>W/(m² K)</td>
</tr>
<tr>
<td>Absorptivity</td>
<td>$\delta$</td>
<td>-</td>
</tr>
<tr>
<td>Volumetric thermal expansion coefficient</td>
<td>$\delta$</td>
<td>1/K</td>
</tr>
<tr>
<td>Difference operator</td>
<td>$\delta$</td>
<td>-</td>
</tr>
<tr>
<td>Thermal boundary layer</td>
<td>$\delta_s$</td>
<td>m</td>
</tr>
<tr>
<td>Difference operator</td>
<td>$\Delta$</td>
<td>-</td>
</tr>
<tr>
<td>Emissivity</td>
<td>$\varepsilon$</td>
<td>-</td>
</tr>
<tr>
<td>Zenith angle</td>
<td>$\theta$</td>
<td>rad</td>
</tr>
<tr>
<td>Temperature</td>
<td>$\theta_s$</td>
<td>°C or K</td>
</tr>
<tr>
<td>Surface temperature</td>
<td>$\theta_\infty$</td>
<td>°C or K</td>
</tr>
<tr>
<td>Temperature of a fluid at a far distance from the surface</td>
<td>$\lambda$</td>
<td>-</td>
</tr>
<tr>
<td>Wavelength</td>
<td>$\mu$</td>
<td>μm</td>
</tr>
<tr>
<td>Dynamic viscosity</td>
<td>$\nu$</td>
<td>kg/(m s)</td>
</tr>
<tr>
<td>Kinematic viscosity</td>
<td>$\rho$</td>
<td>m²/s</td>
</tr>
<tr>
<td>Mass density</td>
<td>$\rho C_p$</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Heat capacity</td>
<td>$\rho C_p$</td>
<td>J/m³ K</td>
</tr>
<tr>
<td>Stefan-Boltzmann constant</td>
<td>$\sigma$</td>
<td>W/(m² K⁴)</td>
</tr>
<tr>
<td>Torque</td>
<td>$\tau$</td>
<td>Nm</td>
</tr>
<tr>
<td>Transmissivity</td>
<td>$\phi$</td>
<td>-</td>
</tr>
<tr>
<td>Azimuth angle</td>
<td>$\omega$</td>
<td>rad</td>
</tr>
<tr>
<td>Angular velocity</td>
<td>$\omega$</td>
<td>rad/s</td>
</tr>
</tbody>
</table>
Subscripts
abs
absorbed
e
electrical
irr
irreversible
k
convective
p
pipe
rad
radiation
ref
reflected
rev
reversible
s
surface
sur
surrounding
t
torque
tr
transmitted
∞
free stream conditions
far from a surface
x
coordinate direction
y
coordinate direction
z
coordinate direction

Overbar
–
average condition
time mean
→
vectorial

Coordinate Systems
r,φ,z
cylindrical coordinates
x,y,z
Cartesian coordinates

Other
∂
partial derivative

Summary
This chapter provides the thermofluid engineering basics. The chapter is divided into the sections
Thermodynamics, Energy Conversion Processes in CSP Plants and Selected Heat Transfer Topics.
CSP-related topics are included in the section Selected Heat Transfer Topics.
3 Thermofluid Engineering Basics

Key questions

- What do the first three laws of thermodynamics state?
- What forms of heat transfer are there?
- Does fluid in parabolic trough collector absorber tubes flow laminar or turbulent?
- What boiling mechanisms exist?
- How and why can heat be transported in absolute vacuum?
- What is a blackbody?

3.1 Thermodynamics

This section deals with the basics of thermodynamics. The name Thermodynamics is composed of the two Greek words therme (heat) and dynamis (power), which, back in the ancient time, describe the efforts made to convert heat into power. Thermodynamics is therefore a science of energy [1]. The essential laws, definitions and equations will be given.

There are two process energy forms, namely work $W$ and heat $Q$. The most important stored energy forms are potential ($PE$), kinetic ($KE$), internal, chemical, magnetic and nuclear energy. The relation between these forms of energy, for example the observation that energy can be transformed from one form into another, is given by the First Law of Thermodynamics.

3.1.1 First Law of Thermodynamics

The First Law of Thermodynamics states that “energy can neither be created nor destroyed; it can only change forms”. Hence the First Law can be summarised by the statement: “Energy is conserved” [1, p. 158].

There are two types of systems: closed systems and open systems or control volumes.

The term closed system describes a system which is defined by boundaries. No mass is transferred in or out through the system boundaries. A special type of closed system is the isolated system which allows neither mass nor energy to pass through its boundaries.

An open system, often referred to as control volume, is a system with arbitrary boundaries where mass as well as energy may pass. The boundaries can be real or imaginary, fixed or moving. This is illustrated in the below figure. The control volume encloses a thermodynamic device which can be e.g. a turbine, a compressor, or a pump. In most cases the control volume is best placed inside such a device [2, p. 2].
A comparison of a closed system and a control volume can be seen below. Examples of closed systems are a pressure cooker or a bomb calorimeter.

By considering the First Law and incorporating the relationships of energy and work, an energy balance of a closed system can be expressed [2, pp. 52, 54, 68, 69]:

\[
E_2 - E_1 = Q - W
\]  

(1)

where \( E_2 - E_1 \) (\( \Delta E \)) is the “change in the amount of energy contained within a system during some time interval”.

Alternatively the energy balance can be written as:

\[
\Delta KE + \Delta PE + \Delta U = Q - W \quad \text{[J]},
\]

(2)

where,

\( \Delta KE \) is the change in kinetic energy,

\[
\Delta KE = KE_2 - KE_1 = \frac{1}{2} m(v_2^2 - v_1^2) \quad \text{[J]}
\]

(3)

where \( v_2 \) and \( v_1 \) are the final and initial velocity, respectively,

\( \Delta PE \) is the change in potential energy.
\[ \Delta PE = PE_2 - PE_1 = mg(h_2 - h_1) \] \[ \text{[J]} \] (4)

where \( m \) is the mass of the object, \( g \) is the acceleration due to gravity, and \( h_2 \) and \( h_1 \) are the final and initial altitude of the object, respectively.

\( \Delta U \) is the change in internal energy \([1]\),

\[ \Delta U = U_2 - U_1 = m(u_2 - u_1) \] \[ \text{[J]} \] (5)

where \( U_2 \) and \( U_1 \) are the final and initial internal energy, respectively, \( m \) is the mass of the object, and \( u_2 \) and \( u_1 \) are the final and initial specific internal energy.

The time rate form of the energy balance is:

\[ \frac{dE}{dt} = \dot{Q} - \dot{W} \] \[ \text{[W]} \] (6)

which can also be expressed as

\[ \frac{dKE}{dt} + \frac{dPE}{dt} + \frac{dU}{dt} = \dot{Q} - \dot{W} \] \[ \text{[W]} \] (7)

**Transfer of Energy by Work**

The rate of energy transfer by work is called *power*. The symbol of power is \( \dot{W} \) where the dot on top of the symbol indicates time rate. The rate of energy transfer by work is equal to the product of the force \( F \) and the velocity \( v \) at the point of application of the force (if the force is constant):

\[ \dot{W} = F \cdot v \] \[ \text{[W]} \] (8)

This equation can be integrated from time \( t_1 \) to time \( t_2 \) to get the total work \( W \) during the time interval:

\[ W = \int_{t_1}^{t_2} \dot{W} \, dt = \int_{t_1}^{t_2} F \cdot v \, dt \] \[ \text{[J]} \] (9)

Other forms of power include *Power Transmission by a Shaft* and *Electrical Power*.

**Power Transmission by a Shaft:** In a power plant, there are several components where the power transmission is performed by a shaft such as motors driving pumps, blowers and butterfly valves. The equation for the power is:

\[ \dot{W} = F_i \cdot v = \frac{\tau}{r} \cdot r \cdot \omega = \tau \cdot \omega \] \[ \text{[W]} \] (10)

where \( \tau \) is the torque and \( \omega \) the angular velocity. The torque can also be expressed as a product of the tangential force \( F_i \) and the shaft radius \( r \), \( \tau = F_i \cdot r \). Rearranging the equation for the tangential force, \( F_i = \tau / r \). The velocity \( v \) can be expressed with the product of the shaft radius and the angular velocity \([2, \text{pp. 38, 45}]\).
Electrical Power [1, p. 127]: Electrical work on a system is performed when electrons cross the system boundary. Electrons passing through a wire that is placed into an electric field move under the effect of electromotive forces and hence perform work. In the case that \( N \) coulombs of electrical charge move through a potential difference \( V \), then the equation of the electrical work done is given by

\[
W_e = V \cdot N \quad \text{[J]} \quad (11)
\]

Expressed as electrical power the formula is

\[
\dot{W}_e = V \cdot I = I^2 \cdot R = V^2 / R \quad \text{[W]} \quad (12)
\]

where the electrical power \( \dot{W}_e \) is given in the unit Watt \([W]\), the current \( I \) in Ampere \([A]\), the potential difference \( V \) in volt \([V]\) and the resistance \( R \) in Ohm \([\Omega]\). The current \( I \) is the number of electrical charges flowing per unit time.

The electrical work done during a time interval \( t_1 \) to \( t_2 \) is given by the formula

\[
W_e = \int_{t_1}^{t_2} VI \ dt \quad \text{[J]} \quad (13)
\]

Mechanical work

Mechanical work is done when a force moves an object through a distance. As an example, a person exerting a force to move a shopping trolley from one place to another is performing work [1, pp. 127-129].

When a constant force \( F \) acts on a body displacing it a distance \( s \), then the work done can be expressed by the formula [1 pp. 127-129]:

\[
W = Fs \quad \text{[J]} \quad (14)
\]

In the case the force varies while moving the body, then the work done is calculated by integrating the differential amounts of work [1 pp. 127-129].

\[
W = \int_{s_1}^{s_2} F \ ds \quad \text{[J]} \quad (15)
\]

Mechanical work exists in various forms. A very common form is the mechanical work of an expansion - compression process in a piston-cylinder that finds application for example in car engines. This is known as moving-boundary work.

In the example of an expansion-compression system, as shown in Figure 3, the differential of work \( dW \) can be described as [1, p. 129]:

\[
dW = F \ ds = pA \ ds = p \ dV \quad \text{[J]} \quad (16)
\]

where \( ds \) is the differential amount of displacement and \( dV \) is the differential amount of change in volume of the piston, \( p \) is the pressure and \( A \) is the area of the piston face. Moreover, it is shown in
the p-V diagram that the area under the process curve is “equal, in magnitude, to the work done
during a quasi-equilibrium expansion or compression process of a closed system”. The total work
done \( W \) can be obtained by integrating for a change of volume from \( V_1 \) to \( V_2 \):

\[
W = \int_{V_1}^{V_2} p \, dV \quad [J]
\]

(17)

**Figure 3**: Expansion or compression of a gas or liquid in a piston-cylinder [1, p.129]

► Experiments

– Write the statement that defines the First Law of Thermodynamics.

### 3.1.2 Second Law of Thermodynamics

Before defining the Second Law of Thermodynamics, some of the key findings about
thermodynamic processes are given below [3, p. 84]:

– all real processes are irreversible
– processes that include friction are irreversible

---

1 Reversible and irreversible processes: “A process is called irreversible if the system and all parts
of its surroundings cannot be exactly restored to their respective initial states after the process has
occurred” e.g. the “heat transfer through a finite temperature difference”, an “unrestrained expan-
sion of a gas or liquid to a lower pressure” or a “spontaneous chemical reaction”. “A process or sys-
tem is reversible if the system and all parts of its surroundings can be exactly restored to their re-
spective initial states after the process has taken place”. In other words, a reversible process is de-
\[\text{defined as a process which can be reversed without leaving any history on surroundings. Reversible}
\]-processes are purely hypothetical [2, pp. 219, 220, 222].
- processes in which substances are mixed are irreversible
- technical combustion processes are irreversible
- the adiabatic\textsuperscript{2} expansion of a gas without work performance is irreversible

There are three common statements which express the Second Law.

Kelvin-Planck statement: "It is impossible for any device that operates on a cycle to receive heat from a single [heat] reservoir and produce a net amount of work." This means that a heat engine\textsuperscript{3} must exchange heat both with a low-temperature sink and a high-temperature source to ensure continuous operation [1, p. 231]. It also implies that heat cannot be completely converted into work.

Clausius statement: "It is impossible for any system to operate in such a way that the sole result would be an energy transfer by heat from a cooler to a hotter body." The statement, however, does not rule out the possibility to transfer heat from a colder to a hotter body, but simply states that it would require a certain work input. As an example, refrigerators or heat pumps use a work input to achieve the transfer of heat from a colder to a hotter body [2, p. 216].

Entropy statement: "It is impossible for any system to operate in a way that entropy is destroyed" Different to energy, which, as described by the First Law, is always conserved, the property entropy is generated within any systems in which irreversibilities such as friction are present [2, p. 219].

The Second Law of Thermodynamics can also be described by a corollary known as the Clausius inequality, which is applicable to any cycle. The Clausius inequality does not consider the body or bodies, from which the cycle receives energy by heat transfer or to which the cycle rejects energy by heat transfer [2, p. 241].

The Clausius inequality states that for any thermodynamic cycle

\[
\oint \left( \frac{\delta Q}{T} \right)_b \leq 0 \tag{18}
\]

where $\delta Q$ represents the heat transfer at a part of the system boundary during a portion of the cycle, and $T$ is the absolute temperature at that part of the boundary. The subscript $b$ serves as a reminder that the integrand is evaluated at the boundary of the system executing the cycle. The symbol $\oint$ indicates that the integral is to be performed over all parts of the boundary and over the entire cycle [2, pp. 241-242].

An equivalent expression is

\[2\textsuperscript{Adiabatic: A system process within its system boundary, which does not undergo a heat transfer with its surroundings is called adiabatic [2, p. 49]. The word adiabatic comes from the Greek language and translates to "not to be passed" [1, p. 123], which, in thermodynamics, means "without heat transfer" [2, p. 49].

\textsuperscript{3} The term “heat engine” describes amongst other things a steam power cycle.
where $\sigma_{\text{cycle}}$ can be interpreted as representing the “strength” of the inequality and is a measure of the effect of the irreversibilities present within the system executing the cycle [2, p. 242]. The cyclic integral of $\delta Q / T$ is always less than or equal to zero [1, p. 274]. The value of $\sigma_{\text{cycle}}$ gives information about the nature of the cycle which is executed by a system [2, p. 242]:

- $\sigma_{\text{cycle}} = 0$ no irreversibilities present within system
- $\sigma_{\text{cycle}} > 0$ irreversibilities present within system
- $\sigma_{\text{cycle}} < 0$ impossible

$\sigma_{\text{cycle}}$ therefore is the entropy generated by internal irreversibilities during the cycle.

The Clausius inequality provides the basis for further development of the entropy, entropy generation and entropy balance concepts. The formulation of the Second Law with entropy as a property is given in the following section [4, p. 97-101].

For quantitatively formulating the Second Law the entropy was introduced by Clausius. The First Law is formulated with an energy balance and the Second Law with the entropy balance. The following postulates formulate the Second Law, which are described with equations in more detail.

- “Every system has an extensive property ‘entropy $S’”
- “The entropy of a system changes with
  - heat transfer over the system boundary
  - the transfer of matter over the system boundary (in open systems)
  - entropy generation due to irreversible processes within the system”
- The entropy flow $\dot{S}_Q$ passing the system boundary is

$$\dot{S}_Q = \frac{\dot{Q}}{T}$$

where $\dot{Q}$ is the heat flow and $T$ is the positive thermodynamic temperature at the system boundary.
- “The entropy generated within the system due to irreversibilities is positive” and is only zero for reversible processes of the system.

The entropy flow $\dot{S}_Q$ has the unit [J/(s K) or W/K] and the entropy $S$ the unit [J/K]. The specific entropy $s = \frac{S}{m}$ has the unit [J/(kg K)].

For closed systems the change in entropy and entropy flow is given with the equations

$$\Delta S = \Delta S_Q + \Delta S_{\text{irr}} \quad \text{and} \quad \dot{S} = \dot{S}_Q + \dot{S}_{\text{irr}}$$

where $\text{irr}$ stands for irreversible.
If $\dot{S}_{irr} > 0$, then the process is irreversible and if $\dot{S}_{irr} = 0$, then the process is reversible. It is impossible that $\dot{S}_{irr} < 0$, as entropy cannot be destroyed.

Example: Entropy generation in an electric oven
An electric oven works with the basic principle of electric resistance. A metal heating wire inside the oven is connected to the mains. On switching on the appliance current flows through the wire upon which it heats up due to its resistance. The temperature is regulated by varying the current.

Consider the steady temperature of 180°C (= 453.15K) inside the oven. The oven’s heat flow is $\dot{Q} = -\dot{W}_e = -600$ W. Calculate the entropy flow.

As the temperature of the oven is considered to be steady it follows that

$$\dot{S} = \frac{dS}{dt} = \dot{S}_Q + \dot{S}_{irr} = 0$$
(22)

$$\dot{S}_{irr} = -\dot{S}_Q = -\frac{\dot{Q}}{T} = -\frac{\dot{W}_e}{T} = -\frac{600}{453.15} = 1.32 \text{ W/K}$$
(23)

As $\dot{S}_{irr} > 0$, the process is irreversible. However, the dissipated heat (which was produced by the electric power) should not be considered as a loss, but rather as a measure for non-convertibility of the heat back into useful power [4, pp. 100].

The following equations require the explanation of the below described properties:

The enthalpy $H$ and specific enthalpy $h$ are a measure for the total energy of a mass. Enthalpy $H$ is the sum of the internal energy $U$ and the product of pressure $p$ and volume $V$, which is the work done [1, p. 81]. The internal energy $U$ of a system is the sum of all the microscopic forms of energy. The microscopic forms of energy are those a system possesses as a whole with respect to some outside reference frame, such as kinetic and potential energies [1, p. 32].

The enthalpy per unit mass is the sum of the specific enthalpy $h$, is the sum of the internal energy per unit mass $u$ and the product of pressure $p$ and specific volume $v$ [1, p. 81].

The definition of the enthalpy $H$:

$$H = U + pV \quad \text{[J]} \quad (24)$$

and the specific enthalpy $h$:

$$h = u + pv \quad \text{[J/kg]} \quad (25)$$

To express the entropy with state variables, i.e. measurable variables such as temperature and volume, only the reversible process is considered, i.e. $\dot{S}_{irr} = 0$. The equation (20) is then written as

$$\dot{S}_Q = \frac{\dot{Q}_\text{rev}}{T}$$
(26)

where rev stands for reversible.

The heat flow $\dot{Q}_\text{rev}$ is described according to the First Law [4, pp. 125-126]:

---

4 The term enthalpy comes from the Greek language and means to heat.
\[ \dot{Q}_{rev} = \frac{dU}{dt} + p \frac{dV}{dt} \]  
\hfill (27)

The differential \( dS \) of the entropy then becomes
\[ dS = \frac{dU + pdV}{T} = \frac{1}{T} dU + \frac{p}{T} dV \]  
\hfill (28)

and for the specific entropy:
\[ ds = \frac{1}{T} du + \frac{p}{T} dv \]  
\hfill (29)

Reformulating equation (29) for \( T ds \), an equation known as the Gibbs equation is formed:
\[ T ds = du + p \ dv \]  
\hfill (30)

The Gibbs equation can also be written as
\[ T ds = dh - v dp \]  
\hfill (31)

because \( dh - v dp = du + p \ dv \).

Prove by differentiation and rearrangement of the differential of the specific enthalpy:

The specific enthalpy \( h \) is given by
\[ h = u + pv \]  
\hfill (32)

The differential of the specific enthalpy is
\[ dh = du + p \ dv + v \ dp \]  
\hfill (33)

Rearranging this
\[ dh - v \ dp = du + p \ dv \]  
\hfill (34)

which proves that either the left or right side of the equation can be substituted in equation (33) and is equal to \( T ds \).

### 3.1.3 Third Law of Thermodynamics

The Third Law of Thermodynamics states that “the entropy of a pure crystalline substance is zero at the absolute zero of temperature, 0K” [2, p. 739].

► Exercises

- State the First Law of Thermodynamics.
– State 1 of the 3 statements that defines the Second Law of Thermodynamics.
– State the Third Law of Thermodynamics.
– What does the Clausius inequality state?

### 3.2 Energy Conversion Processes in CSP Plants

Figure 4 shows a simplified CSP plant. For any CSP plant the same principle applies: The heat source is the solar irradiation from the sun. A collector field reflects and focuses (concentrates) the irradiation on an absorbing surface which then heats up. Depending on the system, the heat is either utilised for heating a heat transfer fluid, which is then passed to a steam generator (in figure: heater) for generating steam or it is used for direct steam generation. Whichever method is applied, the heat is used for generating steam. The steam drives a steam turbine, which in turn drives a generator for generating electricity.

![Figure 4: Principle of utilising solar energy in a CSP plant for the production of electricity, edited from [5]](image)

### 3.3 Selected Heat Transfer Topics

This section deals with the heat transfer modes as well as selected topics that are of particular interest for solar thermal power plants. More heat transfer equations are given in chapters about CSP technologies such as Parabolic Trough Technology and Solar Tower Technology.
3.3.1 Transfer of Energy by Heat

The net rate of heat transfer is denoted by $\dot{Q}$. The total amount of energy transferred $Q$ is calculated by integration over the time interval $t_1$ to $t_2$ [2, pp. 48-49]:

$$Q = \int_{t_1}^{t_2} \dot{Q} \, dt \quad [J] \quad (35)$$

3.3.2 Heat Transfer Modes

The energy balance for any solar thermal system is defined by the sum of the heat transfer events in the system. Energy transfer by heat may occur as conduction, convection (natural or forced), or thermal radiation. These three heat transfer modes are discussed in this section.

a) Conduction

The transfer of energy by conduction can take place in solids, liquids and gases. Within these, conduction can be understood as the transfer of energy from more energetic particles to subsequent less energetic particles [6, p. 3]. When doing a conduction analysis the aim is to find the temperature field or temperature distribution (temperature variation) within the analysed medium (e.g. a solid). In order to perform such an analysis it is necessary to define a differential control volume and to identify the heat transfer processes. Figure 5 shows the differential control volume for conduction analysis using Cartesian coordinates. The heat transfer rate at any position within the medium, and also on the surface, can be calculated with Fourier’s law. An energy flow entering into plane $dydz$ is described with the equation

$$d\dot{Q}_x = -k \frac{\partial \vartheta}{\partial x} dydz \quad (36)$$

where the proportionality constant $k$ is the thermal conductivity [W/(m K)] and is a material property, and $\vartheta$ is the temperature in K at the considered coordinate [7, pp. 9-10].

An energy flow leaving out of the plane $dydz$ [7, p. 10] is described by the equation

$$\dot{Q}_{x+dx} = d\dot{Q}_x + \frac{\partial}{\partial x}(d\dot{Q}_x) \, dx \quad (37)$$

Substituting for $d\dot{Q}_x$ the equation becomes:

$$\dot{Q}_{x+dx} = d\dot{Q}_x + \frac{\partial}{\partial x} \left(-k \frac{\partial \vartheta}{\partial x}\right) dx dydz \quad (38)$$

The same expressions hold true also for the other two coordinate directions $y$ and $z$.

Within the volume element there may additionally be an energy source producing heat and the rate of energy generation is described by

$$\dot{E}_x = \dot{q} \, dx dydz \quad [W] \quad (39)$$

where $\dot{q}$ is the energy generation rate per unit volume of the volume element and has the unit [W/m$^3$] [6, p. 47].
The rate of increase of energy stored within a control volume is given by

\[ \dot{E}_m = \rho c_p \frac{\partial \theta}{\partial t} \, dx\,dy\,dz \quad [W] \quad (40) \]

where the term \( \rho c_p \frac{\partial \theta}{\partial t} \) is the time rate of change of the internal energy of the medium per unit volume [6, p. 48]. According to the First Law of Thermodynamics (conservation of energy), the energy balance is always preserved and is expressed with

\[ \dot{E}_{in} + \dot{E}_g - \dot{E}_{out} = \dot{E}_{st} \quad [W] \quad (41) \]

where \( \dot{E}_{in} \) and \( \dot{E}_{out} \) are the rate of energy transfer into and out of a control volume, respectively [6, p. 48].

The energy balance of the control volume \( dx\,dy\,dz \) is given by

\[ \left[ \frac{\partial}{\partial x} \left( k \frac{\partial \theta}{\partial x} \right) + \frac{\partial}{\partial y} \left( k \frac{\partial \theta}{\partial y} \right) + \frac{\partial}{\partial z} \left( k \frac{\partial \theta}{\partial z} \right) \right] + \dot{q} = \rho c_p \frac{\partial \theta}{\partial t} \quad (42) \]

For cylindrical bodies the energy balance is expressed as

\[ \left[ \frac{1}{r} \frac{\partial}{\partial r} \left( kr \frac{\partial \theta}{\partial r} \right) + \frac{1}{r^2} \frac{\partial}{\partial \phi} \left( k \frac{\partial \theta}{\partial \phi} \right) + \frac{\partial}{\partial z} \left( k \frac{\partial \theta}{\partial z} \right) \right] + \dot{q} = \rho c_p \frac{\partial \theta}{\partial t} \quad (43) \]
with reference to Figure 6 [6, pp. 49-50]. The equations (42) and (43) assume that in a homogeneous control volume, $k$ is the same in all directions.

Assuming that $\dot{q} = 0$, the one-dimensional conduction through a plain wall is calculated according to the equation

$$\frac{\partial}{\partial x} \left( k \frac{\partial \vartheta}{\partial x} \right) = k \frac{\partial^2 \vartheta}{\partial x^2} = \rho c_p \frac{\partial \vartheta}{\partial t} \quad (44)$$

For steady state operation $\vartheta \neq \vartheta(t)$ and $k \frac{\partial^2 \vartheta}{\partial x^2} = 0$. This implies that the temperature distribution is linear, as described by the equation of a straight line

$$\vartheta = C_1 x + C_2 \quad (45)$$

where $C_1$ and $C_2$ are constants and $x$ is the distance. The equation for the heat flow by conduction for steady state operation is expressed as

$$\dot{Q}_{\text{cond}} = k \cdot A \cdot \frac{\vartheta_2 - \vartheta_1}{\Delta x} \quad [\text{W}] \quad (46)$$

where $(\vartheta_2 - \vartheta_1)/\Delta x$ is the temperature gradient in the $x$-axis direction and $A$ is the heat exchanging surface area [m$^2$] normal to the direction of heat flow, with reference to Figure 7 [1, p. 743].
When the wall is composed of \( n \) layers, the following equation holds true:

\[
\dot{Q}_{\text{cond}} = \frac{A}{\sum_{i=1}^{n} \frac{\delta_i}{k_i}} (\vartheta_1 - \vartheta_{n+1}) \quad (47)
\]

where \( \delta \) is the layer thickness.

Example for a wall with three layers:

\[
\dot{Q}_{\text{cond}} = \frac{A}{\frac{\delta_1}{k_1} + \frac{\delta_2}{k_2} + \frac{\delta_3}{k_3}} (\vartheta_1 - \vartheta_4)
\]

For cylindrical walls, the below equation is used:

\[
\dot{Q}_{\text{cond}} = \frac{1}{2\pi L} \frac{A}{\sum_{i=1}^{n} k_i} \frac{1}{\sum_{i=1}^{n-1} \ln \frac{r_{i+1}}{r_i}} (\vartheta_1 - \vartheta_{n+1}) \quad (48)
\]
b) Convection

The thermal losses due to convection are defined as:

$$\dot{Q}_{\text{conv}} = h \cdot A_s \cdot (\vartheta_s - \vartheta_\infty) \quad [W] \quad (49)$$

where $h$ is the convection heat transfer coefficient [W/(m²K)], $A_s$ is the surface area normal to the direction of heat transfer by convection, $\vartheta_s$ is the surface temperature and $\vartheta_\infty$ is the temperature of the fluid at a far distance from the surface [1, p. 722]. The convection heat transfer coefficient can be defined as the rate of heat transfer between a solid surface and a fluid per unit surface area per unit temperature difference [1, p. 855].

Figure 10 shows the convective heat transfer from a hot surface to the surrounding air. Convection can either occur naturally or forced (e.g. by a fan) as shown in Figure 11.

---

5 The bar above the $h$ indicates the average value over the total surface of the considered body. This depends on the flow type, geometry as well as fluid and thermal properties.
The convection heat transfer coefficient $h$ is determined experimentally and is contained in the equations for calculating the Nusselt number. The Nusselt number is a dimensionless temperature gradient at the surface and it gives a measure of the convection heat transfer occurring at the surface [8, p. 371]. The Nusselt number can be determined for various geometries such as flat plates, cylinders and other shapes. When the convection is forced, $Nu$ is calculated with the Reynolds and Prandtl numbers, $Nu = f(Re, Pr)$, while for natural convection, $Nu$ is calculated with the Prandtl and Grashof numbers, $Nu = f(Gr, Pr)$. The expression of the Nusselt number is:

$$Nu = \frac{h L_c}{k}$$  \hspace{1cm} (50)

where $L_c$ is a characteristic length and $k$ is the thermal conductivity of the fluid.

A thermal boundary layer develops when a fluid at a specified temperature flows over a surface that is at a different temperature [1, p. 857]. With reference to Figure 12 which shows a thermal boundary layer on a flat plate, the thermal boundary layer, here denoted as $\delta_t$, gets thicker along the length of the flat plate. The increase of the thickness of the boundary layer is explained by the following consideration. A fluid flows at a uniform temperature of $T_\infty$ over an isothermal plate at temperature $T_s$. In this example, the uniform fluid temperature $T_\infty$ is higher than the plate’s temperature $T_s$. The fluid particles in the layer adjacent to the surface will reach thermal equilibrium with the plate and therefore assume the temperature of the plate’s surface $T_s$. An energy exchange of these fluid particles with the particles in the adjoining-fluid layer will take place, which continues on and on also in the adjoining-fluid layers. As a result of this, a temperature profile will develop in the flow field that ranges from $T_s$ at the surface to $T_\infty$ sufficiently far from the surface. The thermal boundary layer is therefore the flow region over the surface in which the temperature variation in the direction normal to the surface is significant.
For example, the Nusselt equation for a flat plate for laminar flow is

$$Nu_x = \frac{\tilde{h}_x L_c}{k} = 0.332 \frac{Re_x^{0.5}}{Pr^{1/3}}$$

($$0.6 < Pr < 10; \text{ Re}_x < \text{ Re}_{x,\text{critical}} \approx 2 \cdot 10^5$$).

The Nusselt numbers for natural convection are not dependent on the geometry of the body, but rather on the orientation. The equations for the natural and forced Nusselt numbers are determined based on experimental results.

The convection heat transfer coefficient $$\tilde{h}$$ is obtained from a dimensionless relationship with the Nusselt number $$Nu$$:

$$\tilde{h} = \frac{Nu \cdot k}{L_c}$$  \quad [W/m²K]  \quad (51)

The equation for calculating the Nusselt number

The Reynolds number is the dimensionless ratio of inertia to viscous forces [1, p. 858]:

$$\text{Re}_x = \frac{u_{\infty} x}{\nu} = \frac{\rho u_{\infty} x}{\mu}$$  \quad (52)

where $$\nu = \frac{\mu}{\rho}$$ is the kinematic viscosity of the fluid in the unit [m²/s], $$u_{\infty}$$ is the free stream velocity [m/s], $$\mu$$ is the dynamic viscosity [kg/(m s)], $$x$$ is the characteristic length of the geometry [m] and $$\rho$$ the mass density of the fluid [kg/m³].

The thermal diffusivity $$\alpha$$ is a material property useful in transient heat conduction analyses. As the name suggests, the thermal diffusivity expresses how quickly heat diffuses through a material. Definition of thermal diffusivity:

$$\alpha = \frac{k}{\rho C_p}$$  \quad [m²/s]  \quad (53)

where the heat capacity $$\rho C_p$$ in the unit of [J/(m³K)] expresses the amount of energy which a material can store per unit volume and per unit temperature difference[1, p. 719].

---

*Figure 12: Thermal boundary layer on a flat plate [1, p. 857]*
The Prandtl number $Pr$ is the dimensionless ratio of kinematic viscosity $\nu$ to thermal diffusivity $\alpha$ [1, p. 857]:

$$Pr = \frac{\nu}{\alpha} = \frac{\mu}{k} \times \frac{c_p}{\alpha} \quad [-] \quad (54)$$

The Grashof number $Gr$ is the dimensionless ratio of buoyant to viscous forces [1, pp. 908, 911]:

$$Gr = \frac{g \cdot \beta \cdot (\vartheta_s - \vartheta_\infty) \cdot L^3}{\nu^2} \quad [-] \quad (55)$$

where $g$ is the gravitational acceleration [m/s$^2$], $\beta$ is the volumetric thermal expansion coefficient which approximates to $\beta = \frac{1}{T}$ [1/K] for ideal gases and $L$ is the characteristic length of the body [m].

The Rayleigh number $Ra$ is the product of the Grashof and Prandtl numbers and is dimensionless:

$$Ra = Gr \cdot Pr \quad [-] \quad (56)$$

Convection heat transfer in CSP applications:

Convection (as is also radiation) plays an important role for CSP technologies. The following section is based on the source [9] and provides only an example of heat transfer due to convection in receiver’s in parabolic trough systems. The heat transfer due to convection from the glass envelope to the atmosphere, the flow of a fluid inside a tube as well as the heat transfer due to convection between a heat transfer fluid and an absorber of a parabolic trough power plant is discussed.

The symbols and subscripts given in the following equations correspond to Figure 13, which illustrates a one-dimensional energy balance of a receiver with and without glass envelope. The radiation and conduction heat transfer rates are not discussed in this section.
The greatest source of heat loss of a parabolic trough collector is the convection heat transfer from the glass envelope to the atmosphere. According to Newton’s law of cooling

\[ \dot{q}_{\text{conv}} = h_{56} \pi D_5 (T_5 - T_6) \]  

[W/m]  \hspace{1cm} (57)

with

\[ h_{56} = \frac{k_{56}}{D_5} Nu_{D5} \]  \hspace{1cm} (58)

where \( \dot{q}_{\text{conv}} \) is the convection heat transfer rate between the outer surface of the glass envelope to the atmosphere per unit receiver length [W/m], \( h_{56} \) is the convection heat transfer coefficient for air at \((T_5 + T_6)/2\) [W/(mK)], \( D_5 \) is the glass envelope outer diameter [m], \( T_5 \) is the glass envelope outer surface temperature [°C], \( T_6 \) is the ambient temperature [°C], \( k_{56} \) is the thermal conductance of air at \((T_5 + T_6)/2\) [W/(mK)] and \( Nu_{D5} \) is the average Nusselt number based on the glass envelope outer diameter.

The Nusselt number is calculated depending whether windy or wind free conditions apply. If there is wind, forced convection takes place and if there is no wind then natural convection takes place. If there is no wind, then the following equation can be used for estimating the Nusselt number (developed by Churchill and Chu) [9].

\[ Nu_{D5} = \left\{ 0.60 + \frac{0.387 Ra_{D5}^{1/6}}{1 + (0.559/Pr_{56})^{9/16}} \right\}^{2} \]  \hspace{1cm} (59)

with
\[ Ra_{D5} = \frac{g \beta (T_5 - T_6) D_5^3}{\alpha_{56} \nu_{56}} \] (60)

\[ \beta = \frac{1}{T_{56}} \] (61)

\[ Pr_{56} = \frac{\nu_{56}}{\alpha_{56}} \] (62)

where \( Ra_{D5} \) is the Rayleigh number [-] for air based on the glass envelope outer diameter, \( D_5 \), \( g \) is the gravitational acceleration \([m/s^2]\), \( \beta \) is the volumetric thermal expansion coefficient for ideal gases \([1/K]\), \( \alpha_{56} \) is the thermal diffusivity for air at \( T_{56} \) \([m^2/s]\), \( Pr_{56} \) is the Prandtl number [-] for air at \( T_{56} \), \( \nu_{56} \) is the kinematic viscosity for air \([m^2/s]\) at \( T_{56} \) and \( T_{56} \) is the film temperature \((T_5 + T_6)/2 \) [K].

The equation is valid for a range of the Rayleigh number between \( 10^5 < Ra_{D5} < 10^{12} \) and also assumes a long isothermal horizontal cylinder. Moreover, all the fluid properties are determined at the film temperature, \((T_5 + T_6)/2\).

For windy conditions, the Nusselt number is estimated using the following equation which was developed by Zhukauskas [9]:

\[ \bar{Nu}_{D5} = C Re_{D5}^m Pr_6^n \left( \frac{Pr_6}{Pr_5} \right)^{\frac{1}{4}} \] (63)

with

<table>
<thead>
<tr>
<th>( Re_0 )</th>
<th>( C )</th>
<th>( m )</th>
</tr>
</thead>
<tbody>
<tr>
<td>40</td>
<td>0.75</td>
<td>0.4</td>
</tr>
<tr>
<td>( 40 ) – ( 1 \cdot 10^3 )</td>
<td>0.51</td>
<td>0.5</td>
</tr>
<tr>
<td>( 1 \cdot 10^3 ) – ( 2 \cdot 10^5 )</td>
<td>0.26</td>
<td>0.6</td>
</tr>
<tr>
<td>( 2 \cdot 10^5 ) – ( 1 \cdot 10^6 )</td>
<td>0.076</td>
<td>0.7</td>
</tr>
</tbody>
</table>

and

\[ n = 0.37, \text{ for } Pr \leq 10 \]

\[ n = 0.36, \text{ for } Pr > 10 \]

This correlation is valid for \( 0.7 < Pr_6 < 500 \), and \( 1 < Re_{D5} < 10^6 \).

All fluid properties with the exception of the Prandtl number \( Pr_5 \) are evaluated at atmospheric temperature, \( T_6 \). The Prandtl number \( Pr_5 \) is evaluated at glass envelope outer surface temperature, \( T_5 \) [9].
The following part discusses the flow of a fluid inside a tube and the convection heat transfer between a heat transfer fluid (HTF) and an absorber of a parabolic trough power plant: Figure 14 shows a circular tube of radius $r_0$ through which a fluid flows in a laminar flow. The fluid enters the tube with a uniform velocity $u$. The fluid gets in contact with the tube’s inner surface, upon which viscous effects arise and a boundary layer also forms which gets larger in the direction of $x$. As the boundary layer gets larger, the inviscid flow region shrinks and concludes with boundary layer merger at the centreline. With the boundary layer merger at the centreline, the viscous effects extend over the entire cross section and the velocity profile no longer changes with increasing $x$. One now speaks of a fully developed region. The distance from the entrance of the tube until the condition of a fully developed region exists is called hydrodynamic entry length, $x_{fd,h}$. [8, pp. 486-487].

![Figure 14: Laminar, hydrodynamic boundary layer development in a circular tube](8, p. 486)

When dealing with internal flows the Reynolds number gives information about whether the flow is laminar or turbulent. In a fully developed flow, the flow is said to be turbulent if it takes values beginning with and beyond the critical Reynolds number, $Re_{D,c} \approx 2300$. It should be noted that only with much larger Reynolds numbers ($Re_D \approx 10,000$) fully turbulent conditions are reached [8, p. 487].

The following figure (Figure 15) considers thermal effects when the fluid enters the tube at a uniform temperature $T(r,0)$ which has a lower temperature than the surface temperature of the tube wall. Due to this temperature difference, convection heat transfer occurs and this leads to the development of a thermal boundary layer. The tube surface condition can be fixed by imposing either a uniform temperature (i.e. $T_s$ is constant) or a uniform heat flux (i.e. $q_s$ is constant). This leads to a thermally fully developed condition being eventually reached. It should be noted that the fully developed temperature profile $T(r,x)$ can have different shapes according to whether a uniform surface temperature or heat flux is maintained [8, p. 487].
A turbulent flow characteristic is commonly utilized in practice for the reason that the heat transfer coefficients are higher [1, p. 883].

The following part deals with convection heat transfer from the receiver to the HTF of a parabolic trough collector and is based on source [9].

According to Newton’s law of cooling, the convection heat transfer rate between the HTF and inside wall of the absorber pipe per unit receiver length is calculated with the following equation:

\[ q_{\text{conv}} = h_1 D_2 \pi (T_2 - T_1) \]  \hspace{1cm} \text{[W/m]} \hspace{1cm} (64)

where \( h_1 \) is the convection heat transfer coefficient of the HTF at \( T_1 \) \([\text{W/(m}^2\text{K)}]\), \( D_2 \) is the inside diameter of the absorber pipe [m], \( T_2 \) is the inside surface temperature of absorber pipe [°C] and \( T_1 \) is the mean (bulk) temperature of the HTF (°C) [9].

The convection heat transfer coefficient of the HTF \( h_1 \) is

\[ h_1 = \frac{N_{\text{D2}} k_1}{D_2} \]  \hspace{1cm} (65)

where \( h_1 \) has the unit [W/(m²K)], \( k_1 \) is the thermal conductance of the HTF at \( T_1 \) [W/(mK)] and \( N_{\text{D2}} \) is the Nusselt number based on \( D_2 \).

Many different equations to calculate the Nusselt number exist and they differ greatly depending on the form of the object around or inside which a fluid flows. In parabolic trough collector’s HCEs, the fluid (e.g. thermal oil or water) flows inside the tubular HCE. When typical parabolic trough power plant operating conditions apply, the fluid flow is well within the region of turbulent flow. At operating conditions like off-solar hours or at times when HCE heat losses are evaluated on a test platform, transitional or laminar flow conditions may also apply because of the viscosity of the HTF at lower temperatures [9].

The convection heat transfer from the receiver to the HTF for turbulent and transitional flow conditions, i.e. at Reynolds numbers > 2300, the following equation developed by Gnielinski [1976] is used:
The equation is used for both the turbulent and transitional flow conditions. For transitional flow conditions the Reynolds numbers are in the range of 2300 and 4000. Furthermore, the correlation adjusts for fluid property variations between the absorber wall temperature and bulk fluid temperature. The correlation is valid for $0.5 < Pr_1 < 2000$ and $2300 < Re_{D2} < 5 \cdot 10^6$. With exception for $Pr_2$, all fluid properties are evaluated at the mean HTF temperature $T_1$. The correlation assumes uniform heat flux and temperature, and assumes the absorber has a smooth inner surface [9].

All the one-dimensional versions of the HCE heat transfer codes are able to model laminar HTF flow. For a laminar flows calculation the Reynolds number is $< 2300$ and the Nusselt number will be constant. The Nusselt number for pipe flow is $Nu = 4.36$ [Incropera and DeWitt 1990]. For the case of annulus flow, the value depends on the ratio between the pipe insert diameter and absorber inside diameter, $D_p/D_2$. Pipe and annulus flow values were derived assuming constant heat flux [9].

**Boiling mechanisms**

The following section gives information on different types of boiling mechanisms and also discusses the tube dryout problem in direct steam generating parabolic trough power plants.

Mechanism of heat transfer at pool boiling process:

The transmission of heat, principally driven by natural convection, from the heat surface to a boiling fluid, is called pool boiling. This type of boiling occurs in shell and tube heat exchangers, in which the HTF is on the tube side and water/steam is on the shell side. The hot HTF is used to evaporate water. The following figure shows a characteristic boiling curve.
The boiling process can be subdivided into single phase natural convection (before first bubbles arise), subcooled boiling, nucleate boiling and film boiling.

Boiling starts when the boiling temperature is reached. Even though the fluid temperature above the heat transfer surface is subcooled, first bubbles arise due to local overheating. This is called subcooled boiling. The bubbles do not leave the liquid, but they cool down, implode and condense. If the temperature of the heat surface raises more, the fluid reaches nucleate boiling. The whole water body reaches boiling temperature and the steam bubbles traverse it completely. The water is
mixed continuously and a high heat flux is reached. That’s why nucleate boiling is the preferred evaporation type in power plants.

![Figure 18: Nucleate boiling](image)

When the overheat temperature of the heat surface is raised further, the fluid reaches the critical heat flux (burnout point) and suddenly gets into film boiling. The region between critical heat flux and film boiling is called transition boiling. At transition boiling bubbles combine to clusters. Hereby the heating surface is only partly wetted. So the transmittable heat flux decreases until it reaches a minimum (Leidenfrost point). At this point the steam clusters are combined to a close steam cushion across the complete heat surface: film boiling. The steam cushion is like an insulation layer between the heat surface and the medium which has to be evaporated. In consequence the heat flux decreases significantly. Film boiling has to be avoided in order to achieve an efficient heat transfer and also in order to prevent the fusing of the heat transfer surface.

![Figure 19: Film boiling](image)
Flow boiling in horizontal tubes:
The water, which is evaporated in the receiver tubes, passes through several flow types during the evaporation process. It flows as a combination of water and steam. The flow types are the following:

In case of two phase flow, the vapour phase is much faster than the liquid phase. Therefore, and because of gravitational effects, a separation of the liquid and the vapour phase can happen in horizontal evaporation tubes. Drying out of the heat surface (tube wall) leads to small heat fluxes. This happens in case of stratified, wave, slug and dispersed flow. In case of bubbly, plug and annular flow the whole inner surface of the absorber tubes is covered with water, so that high heat fluxes are reached.

In Figure 21 a schematic of the flow types and the corresponding heat transfer coefficients (average values over the whole tube circumference) in a horizontal evaporator tube is shown. Note that the convection heat transfer coefficient is denoted by $\alpha$ in the figure, but is normally denoted by $h$ in English language literature.

The evaporation process passes through all flow types. At the beginning of a collector loop no water is evaporated, so $\alpha_{LO}$ holds, i.e. the heat transfer coefficient of liquid flow. When the boiling temperature is reached, bubbling and convective boiling starts. The convective heat transfer coefficient $\alpha(z)_k$ rises quickly. Subsequently, plug flow and stratified flow occur. The convective heat transfer coefficient drops because the tube surface is not completely covered with liquid water. Later, more and more waves arise in the fluid and $\alpha(z)_k$ rises again slightly. The next stage is slug flow. The waves become so big that in parts of the tube the whole surface is wetted so that the convective heat transfer coefficient increases again sharply. Slug flow is problematic due to its high oscillation potential. After slug flow, the steam gets so fast that the liquid phase is pushed to the wall and the fluid reaches annular flow. Hereby the surface is completely covered by water until there is not enough water and the flow is converted into dispersed flow. This is the last stage until the water is completely evaporated.

In the figure, the heat transfer coefficient at nucleate boiling $\alpha(z)_B$ is plotted, too. The overall heat transfer coefficient is a function of both $\alpha(z)_k$ and $\alpha(z)_B$. When subcooled boiling starts $\alpha(z)_B$ increases sharply. At the end it falls down to $\alpha_{GO}$, which is the heat transfer coefficient of pure steam flow [16].
Dryout (burnout) in parabolic trough receivers (heat collecting elements):

Many parabolic trough power plants using thermal oil as HTF have been built. When using thermal oil, a heat exchanger for generating steam in a water-steam cycle is automatically required; this leads to efficiency losses and extra costs. The concept of direct steam generation in the receiver tubes is regarded as a promising option to push on the competitiveness of this technology. Using water directly results in lower investment and operating costs. Moreover, water reduces the environmental risk and fire hazards in the case of leaks, as it is non-toxic and non-flammable. A further advantage is that only a single cycle – the water-steam cycle – is required, so the performance can be enhanced as there are no longer the thermodynamic losses associated with the oil–water/steam heat exchanger [17].

Amongst other problems, some disadvantages of direct steam generation in line-focus collectors include [18, p. 1]:

- the incomplete understanding of the boiling phenomena (including stability) in horizontal tubes,
- the possibility of tube dryout\textsuperscript{6} and damage to the selective surface,
- the lack of freeze protection compared to heat-transfer fluid systems,

\textsuperscript{6} The dryout phenomenon is also called burnout. Dryout causes the damage of the pipe wall in sections where there are excessive wall temperatures.
• the difficulty of integrating a thermal storage system into the plant

Dryout is also a problem for (vertically arranged) direct steam tube receivers in solar tower power plants.

c) Radiation

Physically radiation is an electromagnetic wave, having a certain wavelength and frequency. Both can range from zero to infinity. For CSP applications the range of the thermal radiation is important.

Thermal radiation is emitted by all matter whose temperature is above 0 K as a result of energy transitions of molecules, atoms and electrons. The wavelength $\lambda$ is in the range between 100 $\mu$m and 0.1 $\mu$m, namely the ranges of infrared radiation, visible light and some of the ultraviolet radiation range. Practically interesting for thermal radiation is the temperature range between 20°C and approximately 6000°C.

Unlike convection, radiation requires no intervening medium to propagate and can occur in a vacuum. The rate at which energy is emitted, $\dot{Q}_r$, from a surface area $A$ is quantified by a modified form of the Stefan-Boltzmann Law [6, p. 9]:

$$\dot{Q}_r = \varepsilon \sigma A T_s^4 \quad [W] \quad (68)$$

where $\sigma$ is the Stefan-Boltzmann constant $\sigma = 5.67 \times 10^{-8} \text{ W/(m}^2\text{K}^4)$, $\varepsilon$ is the emissivity $(0 \leq \varepsilon \leq 1)$, and $T_s$ is the absolute temperature of the surface of the body.

The equation of the net rate of radiation exchange between a surface at temperature $T_s$ and the surroundings (enclosure) at temperature $T_{sur}$, as illustrated in Figure 23, is given by [6, p. 10]:

$$\dot{Q}_{rad,net} = \varepsilon \sigma A \left(T_s^4 - T_{sur}^4 \right) \quad [W] \quad (69)$$
In CSP applications there is a special interest in the radiation properties of surfaces. Considering a receiver tube, a surface is needed which absorbs as much radiation as possible, while it does not reflect or emit significant amounts of energy (at least not in the same range of wavelengths as for the incident irradiation).

An idealised body which is used as reference for such analyses is the so-called blackbody. A blackbody is both a perfect absorber and a perfect emitter, meaning that no surface at the same temperature and wavelength can emit more energy than a blackbody. Furthermore the emitted radiation is uniformly distributed around the surface of the blackbody; this means that a blackbody is a diffuse emitter. The term “black” must not be confounded with the colour of black paint or a black surface.

What human eyes see black means only that no visible light is emitted or reflected by the body which is looked at. In fact this may differ in other wavelength ranges. Snow for example seems to be white to the human eye, which means it reflects all incoming light. However it is black for infrared radiation.

Before continuing the term solid angle must be defined to understand the subsequent sections.

For the solid angle $\omega$ can be said that the area of a surface on a sphere of unit radius is equivalent in magnitude to the solid angle it subtends. This means for that sphere which has a surface area of $4\pi$, the solid angle is $4\pi$ sr (steradians). Consequently $\omega$ is $2\pi$ sr for a hemisphere with radius one. The differential solid angle is expressed as

$$d\omega = \frac{dS}{r^2} = \sin \theta d\theta d\phi.$$  (70)
Where \( r \) is the sphere radius, \( \theta \) is the zenith angle, \( \phi \) is the azimuth angle and \( S \) is the respective area on the surface of the sphere [1, pp. 960-961].

\[
\int_{\omega} = \int_{\text{Hemisphere}} = \int_{0}^{\frac{\pi}{2}} \int_{0}^{2\pi} \sin \theta \ d\theta \ d\phi = 2\pi
\]

**Figure 25:** The emission of radiation from a differential surface element into the surrounding hemispherical space through a differential solid angle [1, p. 961]

For engineers and technicians in the field of solar thermal power plants the terms emissivity \( \varepsilon \), absorptivity \( \alpha \) (\( 0 \leq \alpha \leq 1 \)), reflectivity \( \rho \) and transmissivity \( \tau \) (\( 0 \leq \tau \leq 1 \)) are important properties of CSP materials.

At a given temperature the emissivity is the ratio of the radiation emitted by a surface to the radiation emitted by a blackbody that has the same temperature as the surface [1, p. 66].

A blackbody has an emissivity of one, all other bodies are in the range \( 0 \leq \varepsilon \leq 1 \).

It has to be distinguished between the following terms:

- **Spectral directional emissivity**
  
  The emissivity of a real surface varies with temperature, wavelength and direction, thus it is not uniformly distributed.

  \[
  \varepsilon_{\lambda,\theta}(\lambda, \theta, \phi, T) = \frac{I_{\lambda,\theta}(\lambda, \theta, \phi, T)}{I_{b\lambda}(\lambda, T)} \tag{71}
  \]

  The spectral directional emissivity respects the wavelength \( \lambda \) [\( \mu \text{m} \)] and the direction \( \theta, \phi \), where \( \theta \) is the zenith angle and \( \phi \) is the azimuth angle. Since a blackbody is a uniform emitter it has no azimuth or zenith angle dependence. \( I_{\lambda,\varepsilon} \) is the radiation intensity of the investigated body [ \( \text{W/m}^2 \cdot \text{sr} \cdot \mu\text{m} \) ] (with respect to \( \lambda \)) and \( I_{b\lambda} \) is the respective intensity of a black body in [ \( \text{W/m}^2 \cdot \text{sr} \cdot \mu\text{m} \) ].

- **Total directional emissivity**
  
  When integrating the spectral directional emissivity over all wavelengths the total directional emissivity is obtained.
Spectral hemispherical emissivity
More practical than the total directional emissivity is the spectral hemispherical emissivity, which averages the emissivity over all directions.

\[ \epsilon_\lambda(T) = \frac{E_\lambda(T)}{E_{b\lambda}(T)} \]  

(73)

Here \( E_\lambda \) is the emissive power of the investigated body [W/m\(^2\)] with respect to \( \lambda \) and \( E_{b\lambda} \) is the respective emissive power for a black body [W/m\(^2\)].

Total hemispherical emissivity
Analogue to the total directional emissivity the total hemispherical emissivity is gained by integrating over all wavelengths in all directions.

\[ \epsilon(T) = \frac{E(T)}{E_b(T)} \]  

(74)

According to the Stefan-Boltzmann Law, the higher the temperature of a given surface is, the more energy is emitted by that surface.

However, considering the example of the receiver tube again, if the temperature shall be increased then the surface must absorb energy first. Absorptivity is a measure of how much energy of the incident irradiation \( G \) [W/m\(^2\)] is actually absorbed by a surface or body. In the same way the reflectivity as well as the transmissivity is defined.

\[
\alpha = \frac{G_{abs}}{G}, \quad \rho = \frac{G_{ref}}{G}, \quad \tau = \frac{G_{tr}}{G},
\]  

(75, 76, 77)

Here \( G_{abs} \), \( G_{ref} \) and \( G_{tr} \) are the respective amounts of energy that are absorbed, reflected or transmitted, respectively. \( \alpha \), \( \rho \) and \( \tau \) are always in the range between zero and one and their sum must be one.

\[ \alpha + \rho + \tau = 1 \]  

(78)

Considering opaque surfaces like receiver tubes no transmission can take place and \( \tau = 0 \) the above equation simplifies to
\[ \alpha + \rho = 1. \]  

(79)

As shown for the emissivity \( \alpha, \rho, \tau \) of a real surface depend on wavelength and direction.

- Spectral directional absorptivity and reflectivity

\[
\alpha_{\lambda, \theta, \phi}(\lambda, \theta, \phi) = \frac{I_{\lambda, \text{abs}}(\lambda, \theta, \phi)}{I_{\lambda, \text{abs}}(\lambda, \theta, \phi)} \quad \text{and} \quad \rho_{\lambda, \theta, \phi}(\lambda, \theta, \phi) = \frac{I_{\lambda, \text{ref}}(\lambda, \theta, \phi)}{I_{\lambda, \text{abs}}(\lambda, \theta, \phi)}
\]

(80, 81)

Here \( I_{\lambda, \lambda} \) is the incident radiation intensity in [W/m²]

- Spectral hemispherical absorptivity, reflectivity and transmissivity

\[
\alpha_{\lambda}(\lambda) = \frac{G_{\lambda, \text{abs}}(\lambda)}{G_{\lambda}(\lambda)} \quad \rho_{\lambda}(\lambda) = \frac{G_{\lambda, \text{ref}}(\lambda)}{G_{\lambda}(\lambda)} \quad \tau_{\lambda}(\lambda) = \frac{G_{\lambda, \text{tr}}(\lambda)}{G_{\lambda}(\lambda)}
\]

(82, 83, 84)

If integrated over the wavelength \( \alpha, \rho, \tau \) is obtained again.

Considering a blackbody Kirchhoff’s law states that

\[ \epsilon_{\lambda}(T) = \alpha_{\lambda}(T). \]  

(85)

This makes it possible to easily calculate the radiation properties \( \alpha, \varepsilon, \rho \) of an opaque body when only one of them is known, since

\[ \rho = 1 - \alpha. \]  

(86)

Depending on the application different properties of a surface are desired. The glass tube of a receiver for example should have a transmissivity near one for the wavelength range of light and a high reflectivity for infrared radiation. For the actual receiver surface the opposite properties are preferred, here the absorptivity should be near one and consequently transmissivity and reflectivity should be near zero [1, pp. 966-973]. This can be achieved with selective coatings as shown in the below table. Further details can be found in the chapter **Parabolic Trough Technology**.

### Table 2: Absorptivity, transmissivity and reflectivity for some absorber materials [19, p. 104]

<table>
<thead>
<tr>
<th>Material or coating</th>
<th>Visible range</th>
<th>Infrared range</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( \alpha = \varepsilon )</td>
<td>( \tau )</td>
</tr>
<tr>
<td>Non-selective absorber</td>
<td>0.97</td>
<td>0</td>
</tr>
<tr>
<td>Black chrome</td>
<td>0.95</td>
<td>0</td>
</tr>
<tr>
<td>Cermet</td>
<td>0.95</td>
<td>0</td>
</tr>
<tr>
<td>Tinox</td>
<td>0.95</td>
<td>0</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>► Exercises</th>
</tr>
</thead>
<tbody>
<tr>
<td>– What are the three heat transfer modes?</td>
</tr>
<tr>
<td>– What properties describe a blackbody?</td>
</tr>
<tr>
<td>– In which three ways radiation can interact with a body or surface?</td>
</tr>
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