Solar Thermal Water Heating

Introduction
Solar Thermal Systems for Water Heating
Solar Collectors
Pipes
Thermal Storage
Heat Demand and Solar Fraction

Table 3.1 *Thermodynamic Quantities for Thermal Calculations*

Name	*Symbol*	*Unit*
Heat, energy	Q	Ws (= J) or kWh
Heat flow	$\dot{Q}$	W
Temperature	ϑ	°C
Thermodynamic temperature	T	K (Kelvin, 0 K = –273.15°C)
Specific heat capacity	c	J/(kg K)
Thermal conductivity	λ	W/(m K)
Heat transition coefficient	k'	W/(m K)
Coefficient of heat transfer	k	W/(m^2 K)
Surface coefficient of heat transfer	α	W/(m^2 K)

Energy in form of *heat Q* is linked with the *heat flow* $\dot{Q}$.

$$Q = \int \dot{Q} \, \mathrm{d}t \tag{3.1}$$

Every temperature change $\Delta\vartheta$ also causes a *heat change* ΔQ. The change in heat can be calculated with the specific *heat capacity c* and the mass *m* of the affected material:

$$\Delta Q = c \cdot m \cdot \Delta\vartheta \tag{3.2}$$

$$T = \vartheta \cdot \tfrac{\mathrm{K}}{^\circ\mathrm{C}} + 273.15\ \mathrm{K} \tag{3.3}$$

The heat flow $\dot{Q}$ which causes the heat change for a constant heat capacity *c*, is:

$$\dot{Q} = \frac{\mathrm{d}Q}{\mathrm{d}t} = c \cdot \frac{\mathrm{d}m}{\mathrm{d}t} \cdot \Delta\vartheta + c \cdot m \cdot \frac{\mathrm{d}\Delta\vartheta}{\mathrm{d}t} \tag{3.4}$$

side there is a temperature ϑ_1, on the other ϑ_2. This temperature gradient generates a heat flow through the layers given by:

$$\dot{Q} = k \cdot A \cdot (\vartheta_2 - \vartheta_1) \tag{3.5}$$

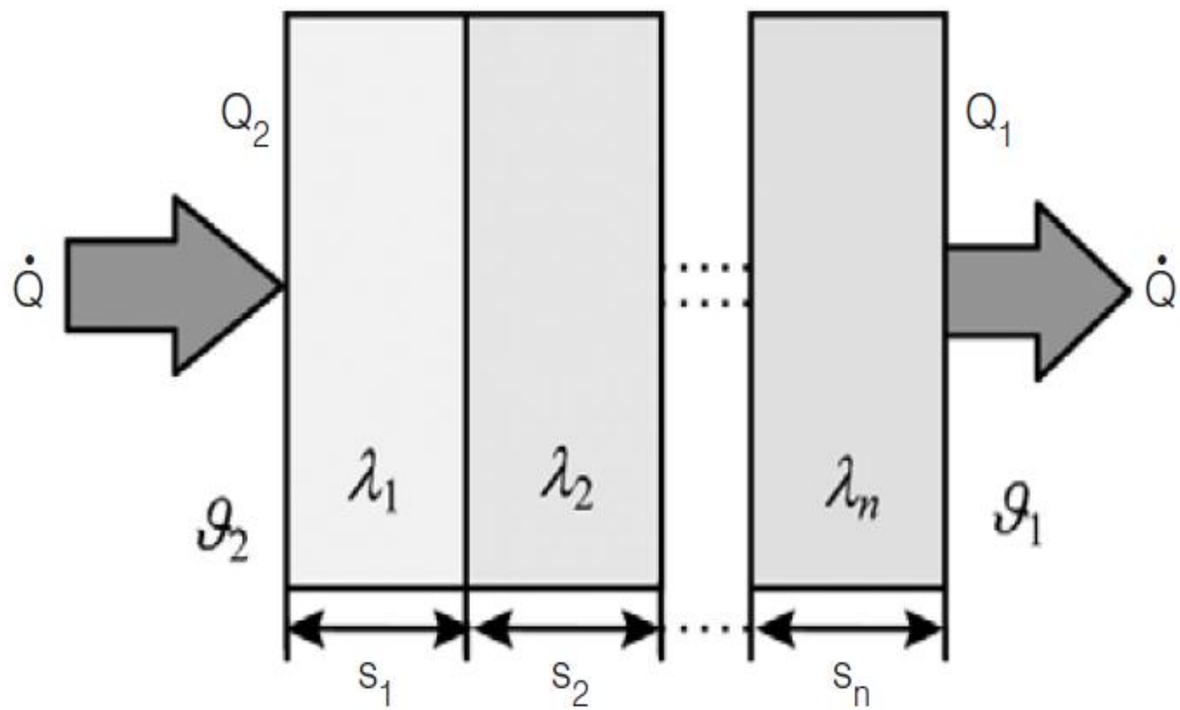


Figure 3.1 *Heat Transfer through* n *Layers with the Same Surface Area* A

The *coefficient of heat transfer*

$$k = \left(\frac{1}{\alpha_1} + \frac{1}{\alpha_2} + \sum_{i=1}^{n} \frac{s_i}{\lambda_i} \right)^{-1} \quad (3.6)$$

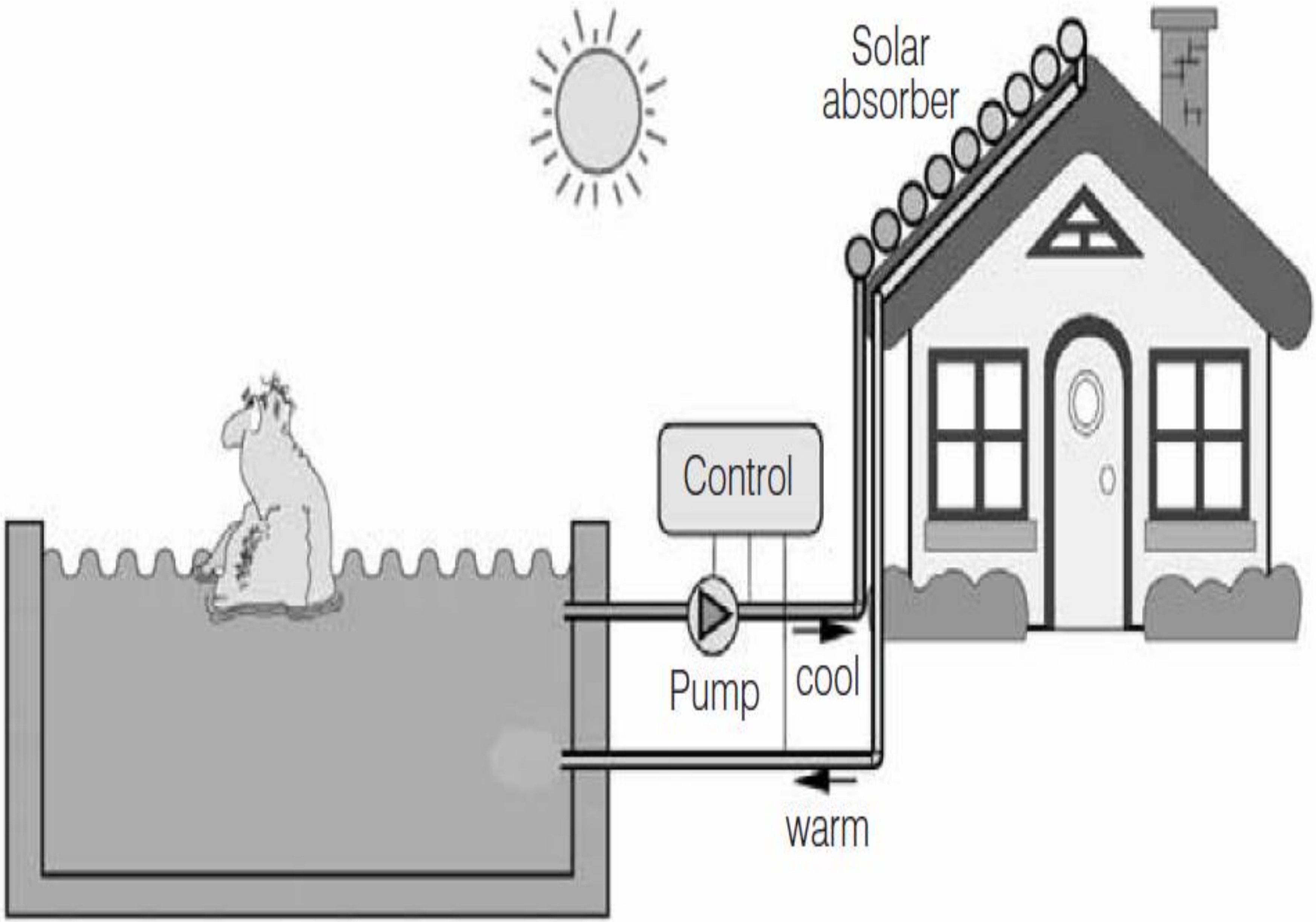
can be calculated with the surface coefficient of heat transfer α_1 and α_2 of both sides, the thermal conductivity λ_i and the layer thickness s_i of all n layers. Table 3.3 shows the heat conductivity λ of various materials.

Solar thermal swimming pool heating

Swimming pools in moderate climatic zones usually need heating systems; otherwise they are usable for only a few weeks per year. For instance, about 500,000 swimming pools have been built in Germany. Since average ambient temperatures are below 20°C even in summer, there is a huge potential for solar pool heating. In many cases simple solar swimming pool heating systems have already become competitive with conventional heating systems.

The *absorber material* is usually plastic. However, the material must be resistant to degradation caused by ultraviolet sunlight and chlorinated pool water. Some suitable materials are polyethylene (PE), polypropylene (PP) and ethylene propylene diene monomer (EPDM). EPDM has a longer lifetime but also costs more. PVC should not be used for ecological reasons – it can emit highly toxic dioxins if it burns.

The *pump* of the system should only operate if the absorber can achieve a temperature rise of the pool water. If the pump operates under very cloudy conditions or during the night, the pool water cools down in the absorber, which now acts as a radiator. A simple two-step controller with hysteresis can avoid this; if sensors detect that the temperature difference between pool and absorber is above a certain threshold level, the pump is switched on.


Solar
absorber
Control
Pump
cool
warm

The typical *heat demand of outdoor swimming pools* in moderate climates is between 150 kWh and 450 kWh per square metre of pool surface. A well-designed solar heating system can maintain a base temperature of 23°C, and thus a fossil heating system is not necessary. For a pool with a surface area of 2000 m^2, a solar heating system can avoid the burning of 75,000 litres of fuel oil and the production of 150,000 kg CO_2 (boiler efficiency 80 per cent) every season. Covering the pool during the night can minimize the heat losses and save additional energy.

As a rule of thumb, the *size of the solar absorber surface* should be 50–80 per cent of the pool surface; however, this depends significantly on the climate. Experience from previous installations or computer simulations can provide more exact values for the system designer. The absorber costs are of the order €100/m^2. Usually the costs of solar heating systems are lower than the costs of fossil fuel heating systems. Only if the outdoor swimming pool is to be operated all year round or if the pool temperature has to be rather high would an auxiliary fossil heating system reduce costs compared to solar energy solutions alone.

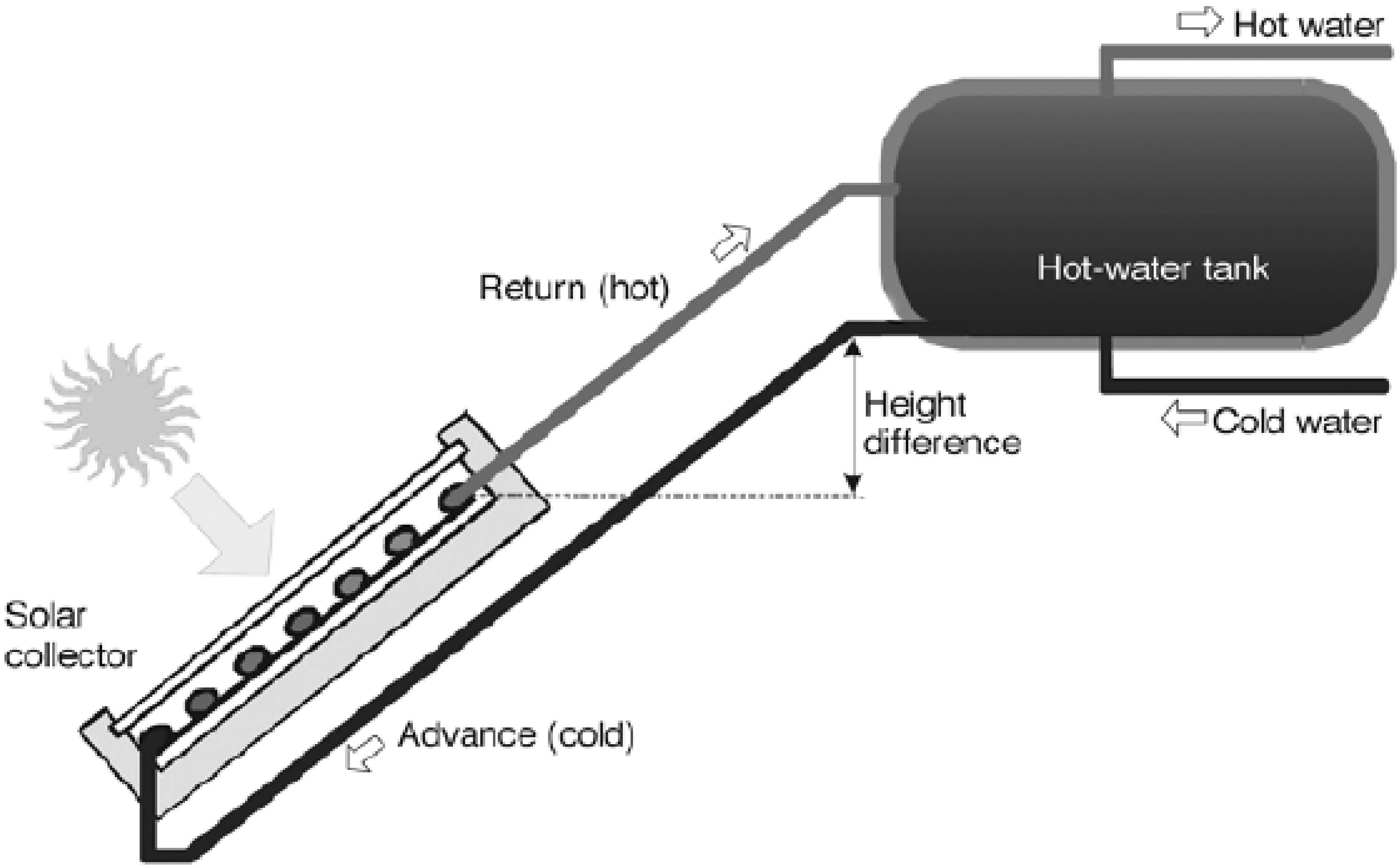


Figure 3.3 *Schematic of a Thermosyphon System*

Thermosyphon systems

single-circuit system.

a *double-circuit system*

Systems with forced circulation

Two *temperature sensors* monitor

Two *temperature sensors* monitor the temperatures in the solar collector and the storage tank. If the collector temperature is above the tank temperature by a certain threshold, the control starts the pump. The pump moves the heat transfer fluid in the solar cycle. The switch-on temperature difference is normally between 5 and 10°C. If the temperature difference decreases below a second threshold, the control switches the pump off again. The choice of both thresholds must ensure that the pump does not continually switch on and off during low irradiance conditions.

Conventional circulation *pumps*

are usually designed for flow rates of 30–50 litre/h per square metre of solar collector area

The pump usually runs at the alternating voltage of the public grid. It is also possible to use DC motors to drive the pump. A small photovoltaic system can provide the electrical energy needed. In that case, all of the energy for the system comes from the sun.

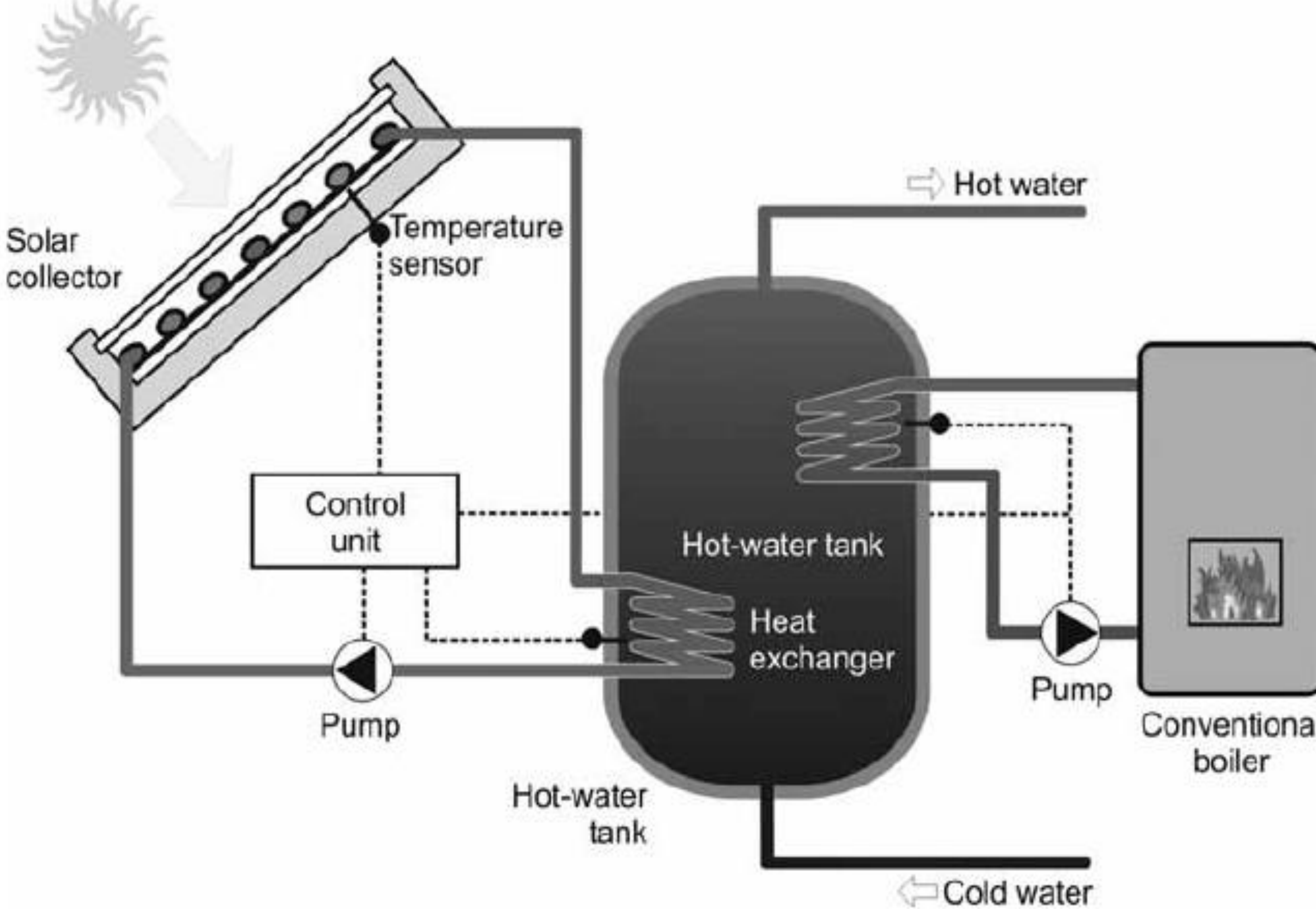


Figure 3.4 *Schematic of a Double-Cycle System with Forced Circulation*

- integral storage collector systems
- flat-plate collectors
- evacuated flat-plate collectors
- evacuated tube collectors.

Integral collector storage systems

New so-called *transparent insulation materials* (TIM) brought a solution to these problems (Lien et al, 1997; Manz et al, 1997). These materials have a slightly lower transmittance compared to low-iron solar safety glass. However, the heat transition coefficient is significantly lower so that the heat losses are reduced to levels acceptable for ICS systems. Table 3.4 compares various conventional and TIM covers.

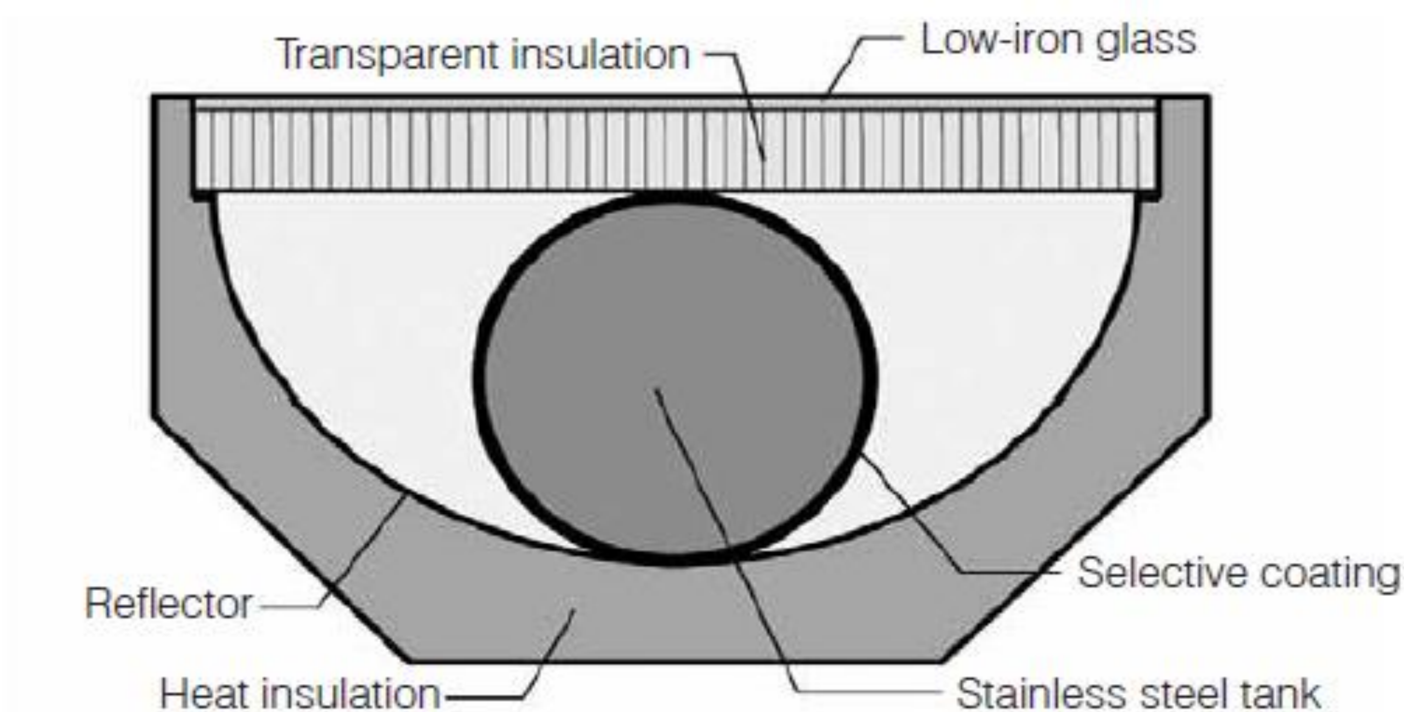


Figure 3.5 *Cross-section through an Integral Collector Storage System*

Flat-plate collectors

- transparent cover
- collector housing
- absorber.

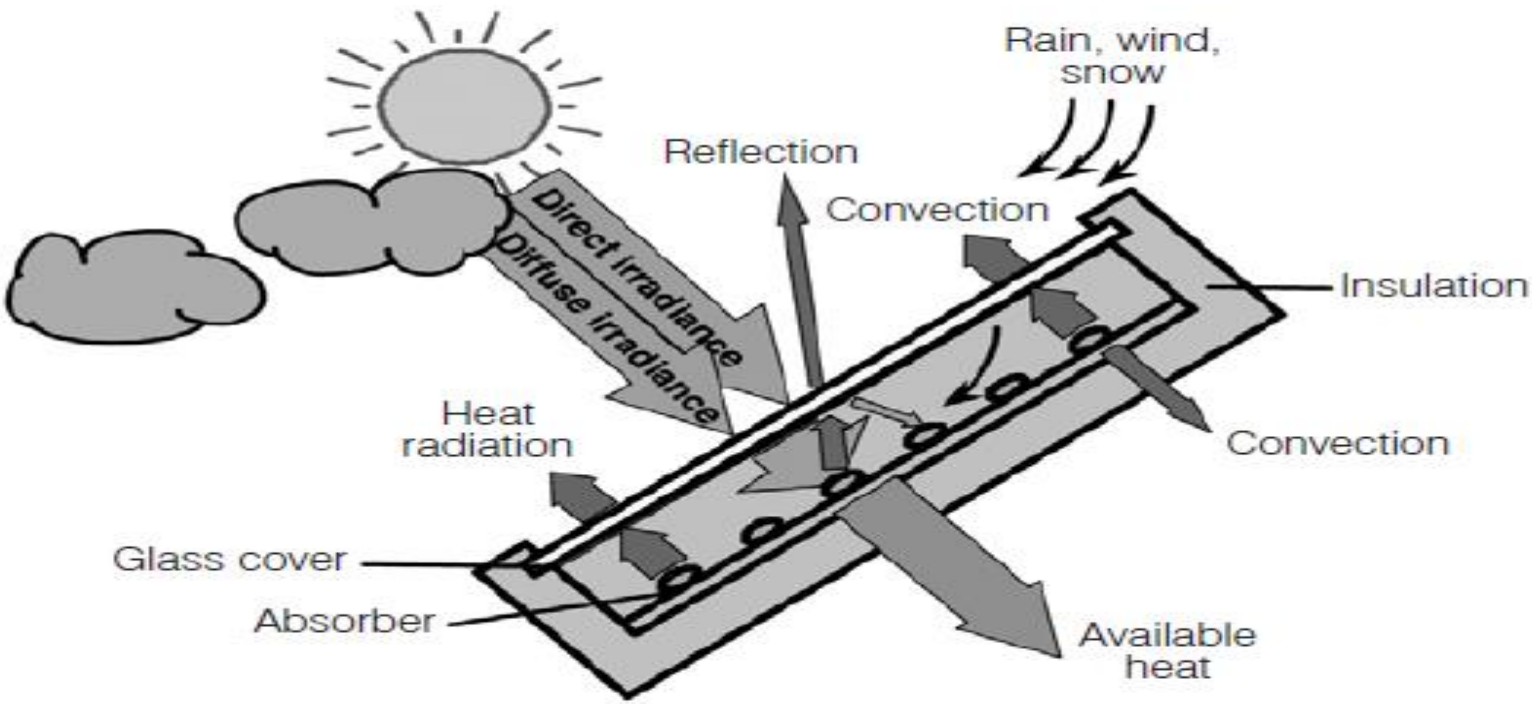


Figure 3.6 *Processes in a Flat-plate Collector*

The *front glass cover* reflects and absorbs a small part of the solar radiant power Φ_e as shown in Figure 3.8; however, the majority of the solar radiation passes through the glass. The reflectance ρ, absorptance α and transmittance τ can describe these processes. The sum of these three values must always be equal to one:

$$\rho+\alpha+\tau=1 \tag{3.7}$$

The corresponding radiant powers are:

$$\Phi_e=\Phi_\rho+\Phi_\alpha+\Phi_\tau=\rho\cdot\Phi_e+\alpha\cdot\Phi_e+\tau\cdot\Phi_e \tag{3.8}$$

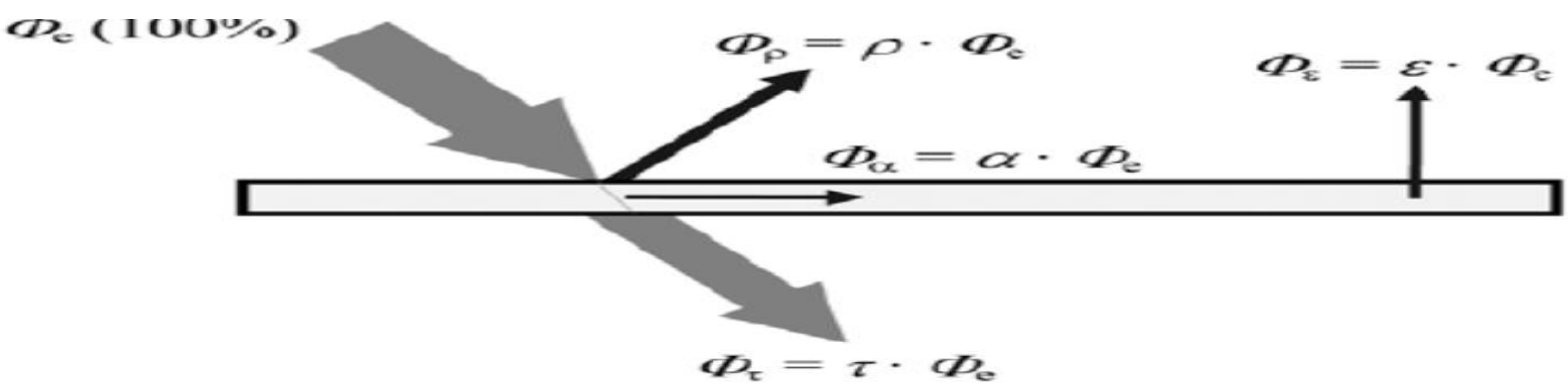
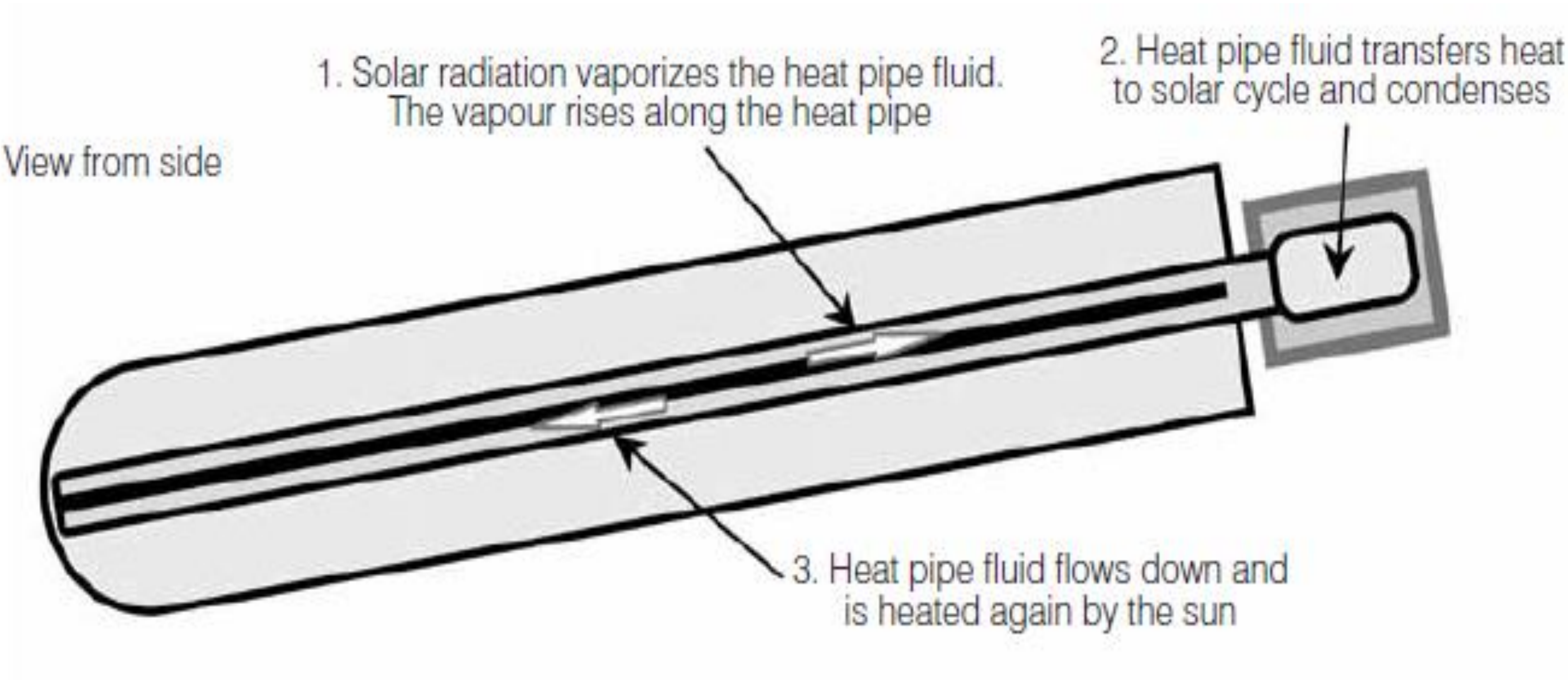


Figure 3.8 *Processes at the Collector Front Glass Cover*

Evacuated tube collectors

The high vacuum inside the closed *glass tube* of the evacuated tube collector is easier to maintain over a long period of time than that in an evacuated flat-plate collector. Glass tubes can resist the ambient air pressure due to their shape so that no supports are necessary between the back and front sides.



Collector performance and collector efficiency

The collector converts solar irradiance E, which is transmitted through the front glass cover with transmittance τ onto the collector surface A_C, directly to heat. The power output of the solar collector $\dot{Q}_{out}$ is reduced by losses due to reflection $\dot{Q}_{ref}$, convection $\dot{Q}_{conv}$ and heat radiation $\dot{Q}_{rad}$

$$\dot{Q}_{out} = \tau \cdot E \cdot A_C - \dot{Q}_{ref} - \dot{Q}_{conv} - \dot{Q}_{rad} \tag{3.13}$$

The convection losses $\dot{Q}_{conv}$ and heat radiation losses $\dot{Q}_{rad}$ can be combined as $\dot{Q}_{RC}$. The heat radiation losses $\dot{Q}_{rad}$ of selective absorbers are much lower than the radiation losses of non-selective absorbers as described above. A vacuum between the front cover and the absorber can reduce the convection losses $\dot{Q}_{conv}$ as described for the evacuated flat-plate and tube collector. The reflection losses $\dot{Q}_{ref}$ can be estimated using the reflectance ρ from the irradiance passing the glass front cover.

With

$$\dot{Q}_{RC} = \dot{Q}_{conv} + \dot{Q}_{rad}$$

and

$$\dot{Q}_{ref} = \tau \cdot \rho \cdot E \cdot A_C$$

and the collector power output becomes:

$$\dot{Q}_{out} = \tau \cdot E \cdot A_C \cdot (1-\rho) - \dot{Q}_{RC} \quad (3.14)$$

Using the absorptance $\alpha = 1 - \rho$ of the absorber, the equation reduces to:

$$\dot{Q}_{out} = \tau \cdot \alpha \cdot E \cdot A_C - \dot{Q}_{RC} = \eta_0 \cdot E \cdot A_C - \dot{Q}_{RC} \quad (3.15)$$

with $\eta_0 = \alpha \cdot \tau$ (3.16)

η_0 is called the *optical efficiency*. It describes the collector efficiency without any losses due to convection or heat radiation. This is only the case if the absorber temperature is equal to the ambient temperature.

The thermal losses Q_{RC} depend on the collector temperature ϑ_C and the ambient temperature ϑ_A as well as on the coefficients *a*, respectively, a_1 and a_2:

$$\dot{Q}_{RC} = a_1 \cdot A_C \cdot (\vartheta_C - \vartheta_A) + a_2 \cdot A_C \cdot (\vartheta_C - \vartheta_A)^2 \approx a \cdot A_C \cdot (\vartheta_C - \vartheta_A) \quad (3.17)$$

The *collector efficiency* η_C can be calculated using the power output of the solar collector $\dot{Q}_{out}$ as well as the solar irradiance E, which reaches the collector surfaces A_C.

With $\eta_C = \frac{\dot{Q}_{out}}{E \cdot A_C} = \eta_0 - \frac{\dot{Q}_{RC}}{E \cdot A_C}$, the *collector efficiency* becomes:

$$\eta_C = \eta_0 - \frac{a_1 \cdot (\vartheta_C - \vartheta_A) + a_2 \cdot (\vartheta_C - \vartheta_A)^2}{E} \approx \eta_0 - \frac{a \cdot (\vartheta_C - \vartheta_A)}{E} \quad (3.18)$$

PIPES

collector flow rate $\dot{m}$

$$\dot{m} = \frac{\dot{Q}_{out}}{c \cdot \Delta\vartheta_{HTF}}$$

$\Delta\vartheta_{HTF}$ of the heat transfer fluid between collector inlet and outlet:

$$\dot{m} = \frac{\eta_0 \cdot E \cdot A_C - a_1 \cdot A_C \cdot (\vartheta_C - \vartheta_A) - a_2 \cdot A_C \cdot (\vartheta_C - \vartheta_A)^2}{c \cdot \Delta\vartheta_{HTF}}$$

the collector flow rate with respect to the collector or absorber surface is:

$$\dot{m}' = \frac{\dot{m}}{A_C} = \frac{\eta_0 \cdot E - a_1 \cdot (\vartheta_C - \vartheta_A) - a_2 \cdot (\vartheta_C - \vartheta_A)^2}{c \cdot \Delta\vartheta_{HTF}} \approx \frac{\eta_0 \cdot E - a \cdot (\vartheta_C - \vartheta_A)}{c \cdot \Delta\vartheta_{HTF}} \quad (3.21)$$

volume flow $\dot{V}$

$$\dot{V} = \frac{1}{\rho} \cdot \dot{m}$$

The cross-sectional area A_p of the pipes in the collector cycle and the flow velocity v_p of the heat transfer medium defines the necessary *pipe diameter* d_p using $\dot{V} = A_p \cdot v_p = \pi \cdot \frac{1}{4} \cdot d_p^2 \cdot v_p$ as:

$$d_p = \sqrt{\frac{4 \cdot \dot{m}}{\rho \cdot v_p \cdot \pi}} = \sqrt{\frac{4 \cdot \dot{m}' \cdot A_C}{\rho \cdot v_p \cdot \pi}}$$

Piping heat-up losses

To heat pipes with mass m_P and heat capacity c_P as well as the heat transfer fluid with mass m_{HTF} and heat capacity c_{HFT} from temperature ϑ_1 to temperature ϑ_2, heat $Q_{Pheatup}$ is needed for n heat-up cycles:

$$Q_{Pheatup} = n \cdot (m_P \cdot c_P + m_{HTF} \cdot c_{HTF}) \cdot (\vartheta_2 - \vartheta_1) = n \cdot (m \cdot c)_{eff} \cdot (\vartheta_2 - \vartheta_1) \tag{3.27}$$

Piping circulation losses

$$Q_{circ} = k' \cdot l \cdot t_{circ} \cdot (\vartheta_{HTF} - \vartheta_A) \tag{3.28}$$

The *heat transition coefficient*

$$k' = \frac{\pi}{\frac{1}{2\lambda} \ln \frac{d_1}{d_P} + \frac{1}{\alpha \cdot d_1}} \tag{3.29}$$

If the regulator stops the circulation in the collector cycle, the pipes and the heat transfer medium cool down again. At a time t_1 with an ambient temperature of ϑ_A and a heat transfer fluid temperature of $\vartheta_{HTF}(t_1)$, the stored heat in the pipes is:

$$Q(t_1) = (c \cdot m)_{eff} \cdot (\vartheta_{HTF}(t_1) - \vartheta_A) \tag{3.30}$$

This heat is reduced by the heat flow:

$$\dot{Q} = k' \cdot l \cdot (\vartheta_{HTF}(t) - \vartheta_A) \tag{3.31}$$

The *stored heat* at time t_2 becomes:

$$Q(t_2) = Q(t_1) - k' l \cdot (\vartheta_{\mathrm{HTF}}(t_1) - \vartheta_{\mathrm{A}}) \cdot (t_2 - t_1) \tag{3.32}$$

with the resulting heat transfer fluid temperature:

$$\vartheta_{\mathrm{HTF}}(t_2) = \frac{Q(t_2)}{(c \cdot m)_{\mathrm{eff}}} + \vartheta_{\mathrm{A}} = \left(1 - \frac{k' l \cdot (t_2 - t_1)}{(c \cdot m)_{\mathrm{eff}}}\right) \cdot (\vartheta_{\mathrm{HTF}}(t_1) - \vartheta_{\mathrm{A}}) + \vartheta_{\mathrm{A}} \tag{3.33}$$

$$\Delta t = \frac{t_2 - t_1}{n} \tag{3.34}$$

the temperature of the heat transfer fluid becomes:

$$\vartheta_{\mathrm{HTF}}(t_2) = \left(1 - \frac{k' l \cdot \Delta t}{(c \cdot m)_{\mathrm{eff}}}\right)^n \cdot (\vartheta_{\mathrm{HTF}}(t_1) - \vartheta_{\mathrm{A}}) + \vartheta_{\mathrm{A}} \tag{3.35}$$

Finally, $\Delta t \to 0$, $t_1 = 0$ and $t_2 = t$ gives:

$$\vartheta_{\mathrm{HTF}}(t) = \exp\left(-\frac{k' l}{(c \cdot m)_{\mathrm{eff}}} \cdot t\right) \cdot (\vartheta_{\mathrm{HTF0}} - \vartheta_{\mathrm{A}}) + \vartheta_{\mathrm{A}} \tag{3.36}$$

Thermal Storage

- short-term storage systems (daily cycle)
- long-term storage (inter-seasonal storage) systems.

Large storage systems can be:

- artificial storage basins
- rock caverns (cavities in rocks)
- aquifer storage (groundwater storage)
- ground and rock storage.

storage systems can be divided into different temperature ranges:

- low-temperature storage systems for temperatures below 100°C
- medium-temperature storage systems for temperatures between 100°C and 500°C
- high-temperature storage systems for temperatures above 500°C.

there are different types of heat storage such as:

- storage of sensible (noticeable) heat
- storage of latent heat (storage due to changes in physical state)
- thermo-chemical energy storage.

Domestic hot water storage tanks

The heat storage capacity of a hot water tank is:

$$Q = c \cdot m \cdot (\vartheta_S - \vartheta_A)$$

The *storage losses* $\dot{Q}_S$ of a cylindrical and spherical hot water storage tank are the sum of the losses $\dot{Q}_{S,cyl}$ of the cylindrical part and the losses $\dot{Q}_{S,sphere}$ of both spherical caps:

$$\dot{Q}_S = \dot{Q}_{S,cyl} + 2 \cdot \dot{Q}_{S,sphere} \qquad (3.38)$$

The losses in the cylindrical part

$$\dot{Q}_{S,cyl} = k' \cdot l_{cyl} \cdot (\vartheta_S - \vartheta_A)$$

can be calculated similarly to the losses of the pipes in the previous sections with the heat transition coefficient k', the length l_{cyl} and the difference between the average storage temperature ϑ_S and the ambient temperature ϑ_A.