

Energy Efficient Buildings with Solar and Geothermal Resources

Ursula Eicker



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Preface

One of the world's major challenges is the transformation of its energy system, which for a short period in human history has been based on fossil fuels. These resources are approaching their end and create serious environmental damages by emissions and long-term waste issues.

Renewable energy sources have always been available on Earth and can easily cover the planet's energy demand. New technologies in solar cell and wind turbine manufacturing, innovative materials and efficiency strategies support the transition to environmentally friendly energy systems.

Especially in urban areas, buildings are major energy consumers. All together they account for about 40% of final energy consumption worldwide and are responsible for about one third of overall CO₂ emissions. In urban structures, building energy consumption is typically twice as high as the need for transport energy, and the energy-saving potential is large. Up to 20% can be saved in the short term and within the next decades buildings should become climate neutral. In urban areas, solar technologies are the most suitable energy sources, as solar modules and collectors can be easily integrated into buildings. In denser urban structures, often the individual roof and facade surface areas are not sufficient to make each building zero energy. Here new concepts are required for the design of local supply systems in city quarters with adequate distribution networks and storage capacities.

Planners, engineers and researchers need fundamental knowledge to deal with fluctuating renewable energy sources, to design adequate storage systems and to integrate the energy systems in highly efficient buildings. To achieve efficiency goals, buildings need to use passive and low-energy resources such as solar gains, daylight, natural ventilation or geothermal heat exchange as intelligently as possible.

This new textbook on energy efficient buildings with solar and geothermal resources provides detailed insight into the design and physics of energy efficient buildings. It discusses the theoretical background of solar thermal cooling and heating, of photovoltaics and geothermal energy, and provides information on applications and costs. Many examples help to apply the theory to real praxis applications.

The reader as an engineer, physicist, energy planner, researcher, student or informed layman will profit from the textbook by acquiring in-depth knowledge of today's new energy systems and building concepts.

This book is based on the knowledge developed within 20 years of research at the Stuttgart University of Applied Sciences on buildings and renewable energy systems. The research centre Sustainable Energy Technologies has been successfully involved in many national and European research and demonstration projects on solar cooling and heating, geothermal energy use, simulation and energy management, zero energy buildings, photovoltaic system technology and many other topics.

Without the support of this research group with about 30 scientists, the broad subject range of the book would not have been possible. I would like to especially thank PhD students and now doctors of philosophy Dilay Kesten and Aysegül Tereci, who produced many results of the first two chapters of building energy efficiency in the urban context; PhD student Tobias Schulze who worked on natural ventilation, and Antoine Dalibard and Felix Thumm who developed the compression chiller and photovoltaic thermal collector models; Dr. Dirk Pietruschka, who did many simulations on solar cooling systems; Ruben Pesch for his contribution to geothermal energy analysis and Mariela Cotrado for her comparison of thermal and electric cooling; Eric Duminil for his very nice irradiance maps; and all the other members of our research team, who discussed the physics and applications of solar and geothermal energy use in buildings.

The layout and design of this book has been completely done by Silvio Barta, an excellent graphic designer, who is even interested in energy technologies and has provided many helpful comments not just on design, but on the contents of the book. Many thanks to his continuous and often tedious work on many details and design issues that are usually lost when concentrating on the contents.

Most heartfelt thanks are due to Juergen Schumacher, who continuously supports me in my work and life. It is with his simulation environment INSEL that most of the simulation results were obtained.

Ursula Eicker
Stuttgart, August 2013



1 Energy consumption of buildings

Figure 1.1 (previous page): Low-energy residential urban development in Scharnhauser Park, near Stuttgart, Germany (Photo: Ursula Pietzsch).



Figure 1.2 The ebök passive standard office building in Tübingen, Germany after (top), before (bottom left) and during (bottom right) renovation work (Photo: eboek GmbH, Tuebingen).

Buildings account today for about 40% of final energy consumption worldwide, and they are responsible for about one third of overall CO₂ emissions (36% in Europe, 39% in the USA, about 20% in China (IEA Study, 2008)). Especially in urban structures, the building energy consumption is typically twice as high as transport, e.g. approximately by a factor 2.2 in London. The energy-saving potential is large: in the short term (up to 2020), savings of 20% are expected within the European Union, and in the long term (up to 2050), buildings are supposed to be climate neutral. The improvement of building energy efficiency can be economically worthwhile today, as shown by a study of the Intergovernmental Panel on Climate Change: between 12% and 25% CO₂ emissions caused by heating and cooling and between 13% and 52% CO₂ emissions caused by electric lighting and equipment can be reduced economically until 2020.

The European Directive for the Energy Performance of Buildings (EPBD) adopted in 2002 is an attempt to unify the diverse national regulations, to define minimum common standards on buildings' energy performance and to provide certification and inspection rules for heating, cooling and ventilation plants. National energy efficiency action plans have been required since 2006, and European Member States must show how they intend to reach the 9% indicative energy savings target by 2016.

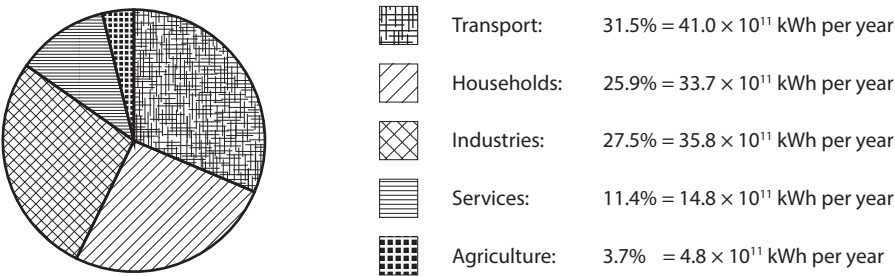


Figure 1.3 Distribution of end energy consumption within the European Union with a total value of 1.3×10^{13} kWh per year (European Environment Agency, 2009).

Average heat transfer coefficients for new buildings are today about 0.3 and 0.4 W m⁻² K⁻¹. In 2009 the European Union tightened the EPDB directive and now demands nearly zero energy standards for new buildings until 2020. Here the building energy demand is balanced, with the local renewable energy production resulting in a net zero energy demand. For public existing buildings, the zero energy standard will come into effect in 2018.

In moderate European climatic zones like Germany, 80% of residential building energy is consumed by heating, 12% for water heating and the remainder for other electricity consumption, communication and electric lighting. The high percentage of heat consumption is caused by low thermal insulation standards in existing buildings, in which today 90% and even in 2050 60% of residential space will be located. Today, about 1.4% of buildings in Europe are renovated each year energetically. With this rate, there will be a saving of about 40% until 2050 compared to 2005. If the renovation rate were raised to 2% per annum, the energy saving would go up to 74% until 2050.

With high heat insulation standards and the heat recovery ventilation concept of passive houses, a low limit of heat consumption has meanwhile been achieved, which is around 20 times lower than today's values. A crucial factor for low consumption of passive buildings was the

development of new glazing and window technologies, which enable the window to be a passive solar element and at the same time cause only low transmission heat losses. In new buildings with low heating requirements, other energy consumption in the form of electricity for lighting, power and air conditioning, as well as warm water in residential buildings, is becoming more and more dominant. Electricity consumption within the European Union is estimated to rise by 50% by 2020. In this area renewable sources of energy can make an important contribution to the supply of electricity and heat.

The majority of the world's new buildings are constructed in Asia. The Asian building sector accounts today for about 25% of the final energy consumption and is expected to rise to 32% in 2030 (World Energy Outlook, 2006). A World Bank study showed that China and India could cut their current energy consumption by 25% with cost-effective retrofitting of lighting, air conditioning, boilers and heat recovery. The Chinese Ministry of Construction states that 95% of all buildings are highly energy consuming and that energy consumption is currently two to three times that of developed countries in achieving the same comfort level (Building Energy Efficiency, an Asia Business Council Book, 2007).

1.1 Residential buildings

To limit transmission heat losses, the average heat loss coefficients of the building envelope are regulated in most countries. In China with severely cold regions (between 5500 and 8000 heating degree days), exterior walls for three-storey buildings are supposed to have U values below $0.33 \text{ W m}^{-2} \text{ K}^{-1}$, high-rise buildings below $0.48 \text{ W m}^{-2} \text{ K}^{-1}$, whereas in temperate regions, $0.5 \text{ W m}^{-2} \text{ K}^{-1}$ are sufficient. In Japan, today's wall U values are between 0.39 and $1.76 \text{ W m}^{-2} \text{ K}^{-1}$ depending on climatic condition, in Korea between 0.47 and $0.76 \text{ W m}^{-2} \text{ K}^{-1}$.

In Europe with its wide geographical extent of nearly 35° geographical latitude difference (36° in Greece, 70° in northern Scandinavia), a wide range of climatic boundary conditions are covered. In Helsinki (60.3° northern latitude), average exterior air temperatures reach -6°C in January, when southern cities such as Athens at 40° latitude still have averages of $+10^\circ\text{C}$. Consequently the building standards vary widely: whereas average heat transfer coefficients (U values) for detached houses are $1 \text{ W m}^{-2} \text{ K}^{-1}$ in Italy, they are only $0.4 \text{ W m}^{-2} \text{ K}^{-1}$ in Finland. The heating energy demand determined is comparable in both cases at about $50 \text{ kWh m}^{-2} \text{ a}^{-1}$. The necessary U value to achieve the passive house standard with less than $15 \text{ kWh m}^{-2} \text{ a}^{-1}$ heating energy demand for several climate zones are shown in Table 1.1.

U values / $\text{W m}^{-2} \text{ K}^{-1}$	Rome Passive building	Helsinki Passive building	Stockholm Passive building
Wall	0.13	0.08	0.08
Window	1.4	0.7	0.7
Roof	0.13	0.08	0.08
Ground	0.23	0.08	0.1
Mean	0.33	0.16	0.17

Table 1.1 U values required to reach passive standard for different European climates.

In Germany, the heat requirement of residential buildings is between 10 and $250 \text{ kWh m}^{-2} \text{ a}^{-1}$, depending on the insulation standards. Existing buildings with an average consumption of about

220 kWh m⁻² a⁻¹ have the highest energy-saving potentials by reduction of the transmission heat loss of the building envelope.

With an extremely good insulation of all outer surfaces, avoidance of thermal bridges on critical details like basement walls, attic, etc., as well as an air-proof constructed building shell and a controlled ventilation with heat recovery, the heating energy demand can be reduced to 10 - 15 kWh m⁻² a⁻¹.

Studies in Switzerland show that the additional investments to achieve the passive house standard are about 14% (Minergie-P Label) and for the low-energy standards about 6 - 9%. In Germany, with a high quantity of passive houses, the additional investment costs are denoted with 3 - 5%.

In comparison current low-energy houses in Germany have around 70 kWh m⁻² a⁻¹ heating demand. Several hundred row houses, all built after 2000, were surveyed as part of the European project POLYCITY. The building heating energy consumption varies strongly even for identical building types as a result of user influence, with standard deviation of 35% on average of the mean consumption.

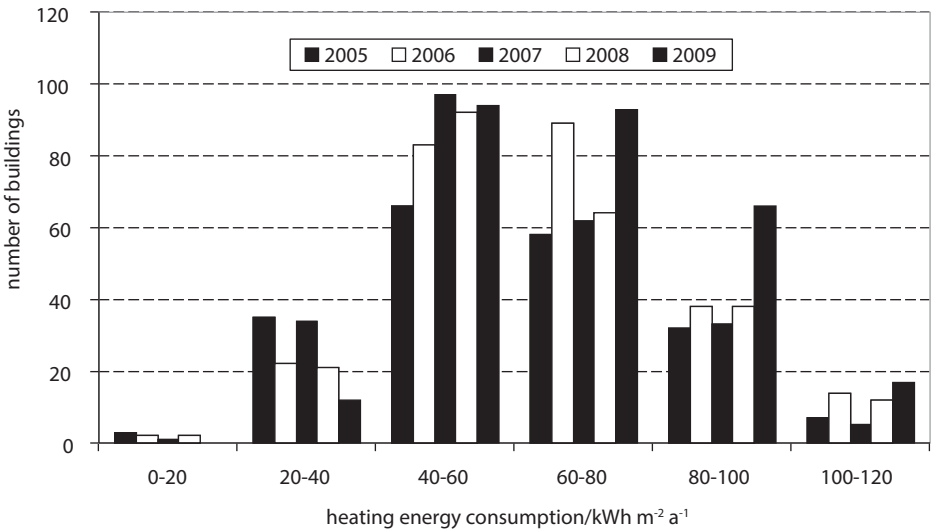


Figure 1.4 Distribution of heating energy consumption of all row houses in the study area supplied by district heating.

Independent of the standard of insulation, water heating is always necessary in residential buildings, and this lies between about 220 (low requirement) and 1750 kWh per person and year (high requirement), depending on the pattern of consumption. For the middle requirement range of 30 - 60 litres per person and day, with a warm-water temperature of 45°C, the result is an annual consumption of 440 - 880 kWh per person, i.e. 1760 - 3520 kWh for an average four-person household. Related to a square metre of heated residential space, in Germany a relatively low value of 12.5 kWh m⁻² a⁻¹ is used in the standards and in Switzerland 14 kWh m⁻² a⁻¹.

The average electricity consumption of private households, around 3600 kWh per household per year, is of a similar order of magnitude. Related to a square metre of heated residential space, an average value of $31 \text{ kWh m}^{-2} \text{ a}^{-1}$ is the result. An electricity-saving household needs only around 2000 kWh a^{-1} . In a passive building project in Darmstadt (Germany), consumptions of between 1400 and 2200 kWh per household per year were measured, which corresponds to an average value of $11.6 \text{ kWh m}^{-2} \text{ a}^{-1}$ (Passivhausinstitut, 1997).

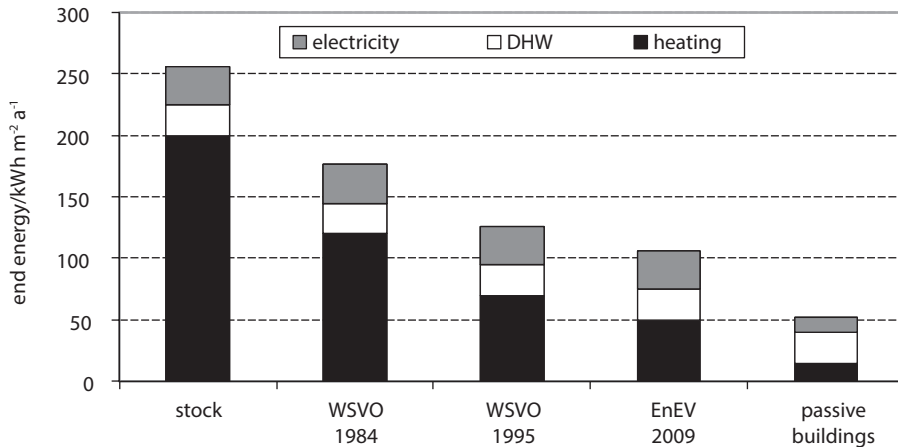


Figure 1.5 End energy consumption for the existing building stock and subsequent legal requirements in residential buildings per square metre of heated floor space in Germany.

According to the 1995 German legislation (WSVO 1995), new buildings were limited in their heating energy demand between 70 and $100 \text{ kWh m}^{-2} \text{ a}^{-1}$ for surface to volume ratios between 0.5 and 1.0 (the lower limit value is marked). Since energy-saving legislation changed in 2002 (EnEV), the building service side is also included in the energy balance and primary energy demand is regulated.

In 2007 energy certificates were introduced in the energy legislation. In 2009 the dependency of maximum demand from A/V ratio was discarded, and the demand was compared with reference buildings with the same geometry. The maximum allowed heating energy demand is currently between 50 and $80 \text{ kWh m}^{-2} \text{ a}^{-1}$. Further tightening of the energy legislation is expected because of the amendment of the European performance directive, so that in a few years, new buildings with passive house standard can be expected.

1.2 Office and administrative buildings

Existing office and administrative buildings have approximately the same consumption of heat as residential buildings and most have a higher electricity consumption.

Both heat and electricity consumption depend strongly on the building's use. In terms of the specific costs, electricity almost always dominates.

More than half of the running costs are accounted for by energy and technical service. A large part of the energy costs is due to ventilation and air conditioning. The VDI Directive

3807 defines the bases of calculation and the temperature normalisation for measured energy consumption values related to the gross floor area.

The AGES GmbH, Münster, provides an extensive and up-to-date database for buildings of different uses.

Heat consumption in administrative buildings can be reduced without difficulty, by improved thermal insulation, to low-energy standards, and even to a few kWh per square metres and year in a passive building. Related to average consumption in the stock, a reduction to 5 - 10% is possible. Electricity consumption dominates total energy consumption where the building shell is energy-optimised. Measured consumption was between 30 and 130 kWh m⁻² a⁻¹ (see also www.solarbau.de).

Detailed measurements over several years in the first passive office building in Germany (Weilheim/Teck), completed in 2000, illustrate ways of energy optimisation: passive house standard is realisable at low additional costs, the hot water consumption is insignificant in office buildings, and the electrical energy consumption for building services (ventilation, lighting, pumps) can be limited to low target values (< 15 kWh m⁻² a⁻¹).

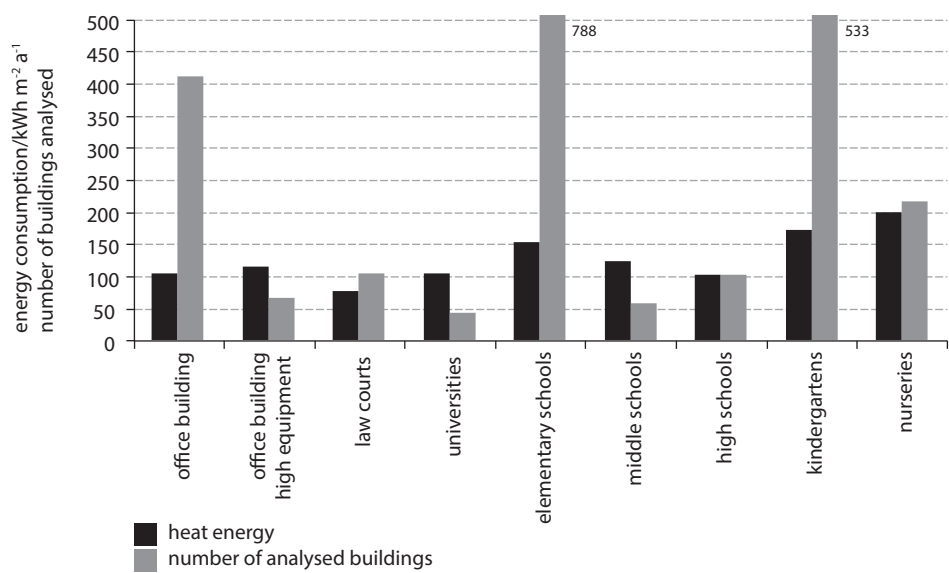


Figure 1.6 Statistical heating energy consumption values per m² of gross floor area from the AGES GmbH.

The main consumer of electric energy is office equipment, which is responsible for more than 40% of the total energy consumption, with a rising trend during the three measurement years even though energy-saving equipment was used.

While the passive house standard was confirmed by measurements, the measured values for total electricity consumption exceeded the planning value of 23.5 kWh m⁻² a⁻¹ by 45%.

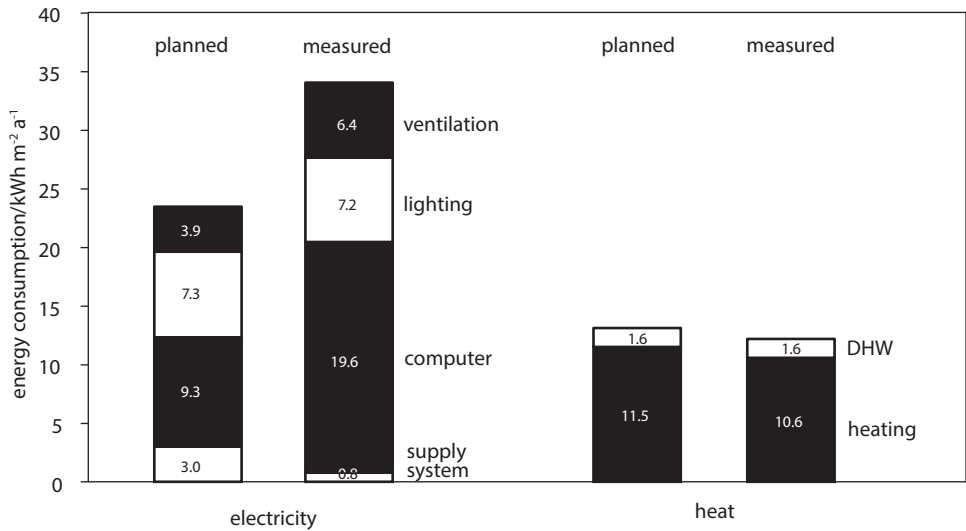


Figure 1.7 Measured consumption of electricity, heat and water heating in an office building with a passive house standard in Weilheim/Teck, Germany.

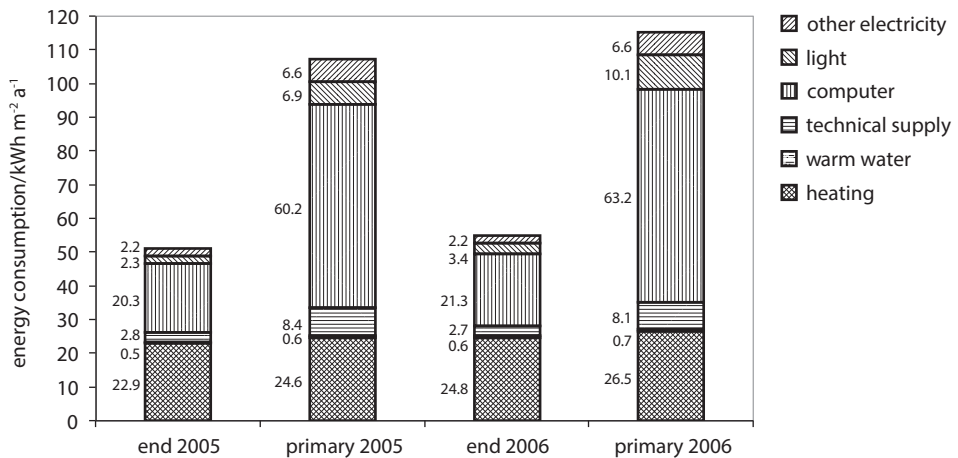


Figure 1.8 Measured end energy and primary energy consumption in the renovated offices of the engineering firm ebök in Tübingen, Germany.

It is also possible to achieve the passive house standard through building rehabilitation. Detailed measurements at an office building in Tübingen, Germany show that a very low thermal heat consumption of less than $25 \text{ kWh m}^{-2} \text{ a}^{-1}$ can be achieved, although not all building elements such as the ground floor can be well insulated due to the low ceiling heights.

Within the total consumption, particularly when considering primary energy, the electrical energy consumption dominates and is mainly caused by office equipment.

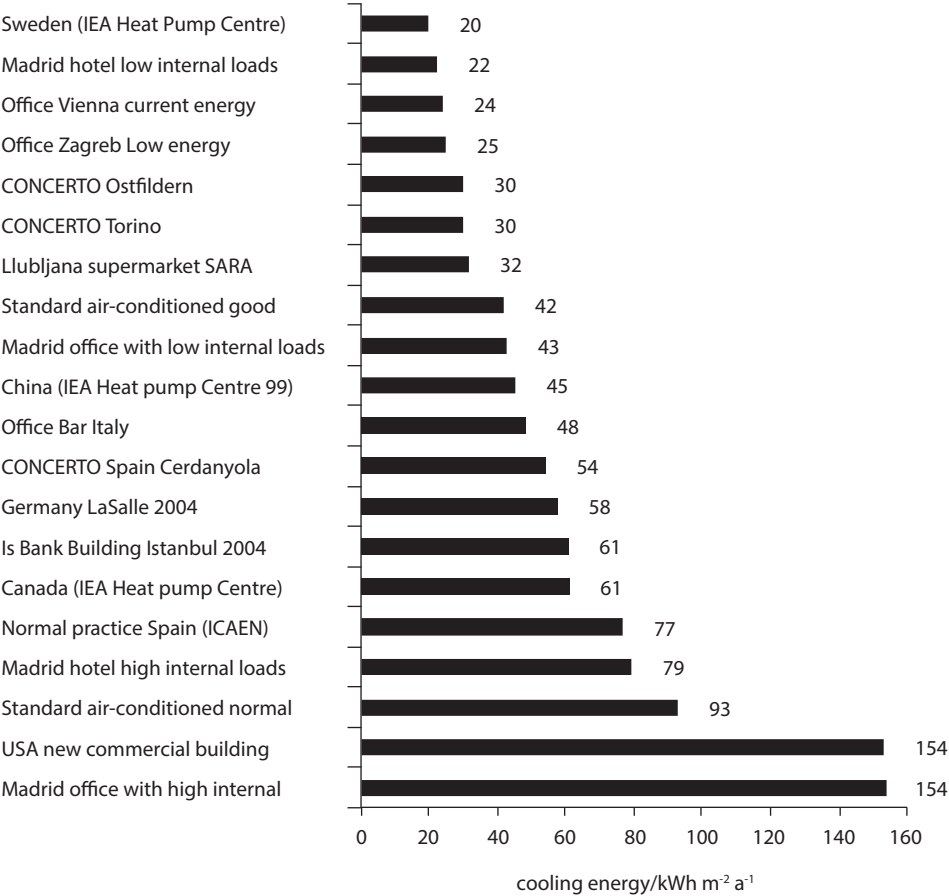


Figure 1.9 Survey of cooling energy requirements of buildings.

1.3 Air conditioning

Air conditioning in buildings and refrigeration is responsibly for about 15% of the total energy consumption worldwide, in hot and humid climates for 30% (Government Information Centre Hong Kong, 2004). Cooling energy is often required in commercial buildings, with the highest consumption worldwide in the USA with up to 150 kWh m⁻² a⁻¹. Breembroek and Lazáro (1999) quote values between 20 kWh m⁻² a⁻¹ for Sweden, 40 and 50 kWh m⁻² a⁻¹ for China and 61 kWh m⁻² a⁻¹ for Canada.

Our own overview of the cooling energy requirement of different building projects shows a typical cooling energy consumption for administrative buildings between 20 and 60 kWh m⁻² a⁻¹ in Europe.

In the food industry, the energy consumption for cooling is considerably higher with an energy consumption between 82 and 345 kWh per square metre of sales floor. The main

part of the required energy is not used for space conditioning, but rather for cooling food (O.Ö. Energiesparverband, 1996).

In Southern Europe, the installed cooling capacity is often dominated by the residential market. Although in Spain less than 10% of homes have air-conditioning systems, 71% of the installed cooling capacity is in the residential sector.

About 50% of internal loads are caused by office equipment such as computers, printers, photocopiers, etc., which leads to an area-related load of about $10 - 15 \text{ W m}^{-2}$. Modern office lighting has a typical connected load of $10 - 20 \text{ W m}^{-2}$ at a luminance level of $300 - 500 \text{ lx}$. The heat given off by people, around 5 W m^{-2} in an enclosed office or 7 W m^{-2} in open-plan offices, is also not negligible. Typical mid-range internal loads are around 30 W m^{-2} , resulting in a daily cooling energy of $200 \text{ Wh m}^{-2} \text{ day}^{-1}$, in the high range between 40 and 50 W m^{-2} and $300 \text{ Wh m}^{-2} \text{ day}^{-1}$.

Detailed own measurements in a passive standard office building in Weilheim, Germany, described above, show $30 - 35 \text{ W m}^{-2}$ internal loads, in a south office with two people and a computer workstation. In a north office with two computer workstations, the loads were about 50 W m^{-2} . The resulting daily intern loads in the south office were between 200 and 300 Wh m^{-2} compared with 400 and 500 Wh m^{-2} in the north office.

Further measurements in the context of the German funding programme SolarBau Monitor showed that for rather low total internal loads from about 92 to $188 \text{ Wh m}^{-2} \text{ day}^{-1}$, office equipment also clearly dominates.

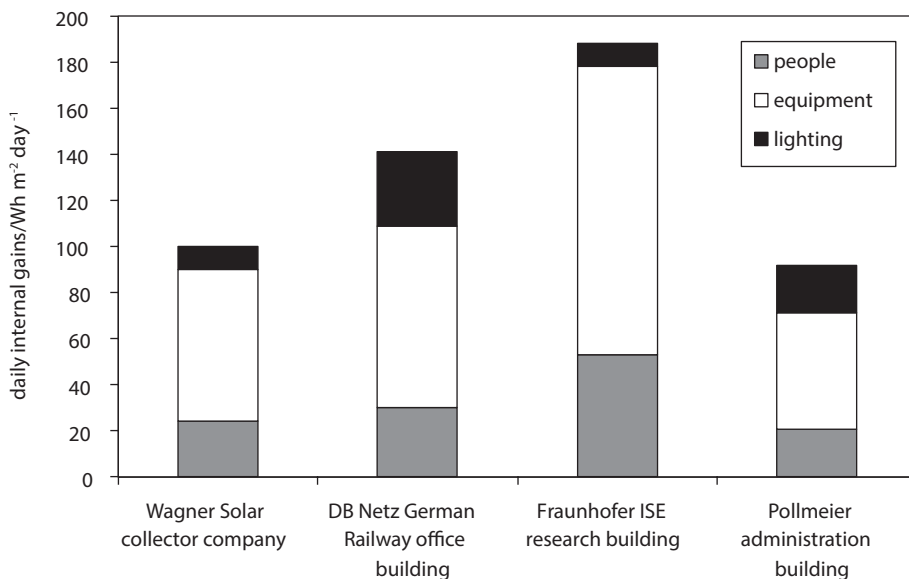


Figure 1.10 Measured distribution of internal gains in buildings of the SolarBau monitor project.

External loads depend greatly on the surface area of the glazing as well as the sun-protection concept. On a south-facing facade, a maximum irradiation of about 600 W m^{-2} occurs on a sunny summer day. The best external sun protection reduces this irradiation by 80%. Together with the

total energy transmission factor (g value) of low-e coated double glazing of typically 0.65, the transmitted external loads are about 78 W per square metre of glazing surface. In the case of a 3 m² glazing surface of an office room, the result is a load of 234 W, which creates an external load of 20 W m⁻² for a room surface area of 12 m². This situation is illustrated in Figure 1.11 for south-east- and west-facing facades in the summer.

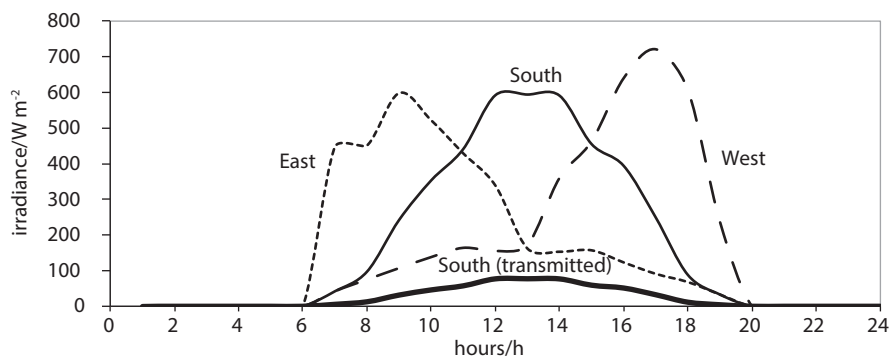


Figure 1.11 Diurnal variation of irradiance on different facade orientations and transmitted irradiance by a sun-protected south facade on a day in August (Stuttgart).

The shading coefficients of sun-protection devices depend particularly on the arrangement of the sun protection: external sun protection can reduce the energy transmission of solar radiation by 80%, whereas with internal sun protection a reduction of at most 60% is possible.

Sun shading system	Colour	Energy reduction coefficient/-
External sun shades	Bright	0.13 - 0.2
External sun shades	Dark	0.2 - 0.3
Internal sun shades	Bright	0.45 - 0.55
Reflection glazings	-	0.2 - 0.55

Table 1.2 Energy reduction coefficients of internal and external sun protection.

External loads depend on the relation of window surface to floor space as well as the chosen shading system. For area ratio between 0.1 and 0.7 the typical external loads are between 8 and 60 W m⁻² (Arsenal Research, 2007). Together with the internal loads there are 25 - 90 W m⁻² total cooling loads.

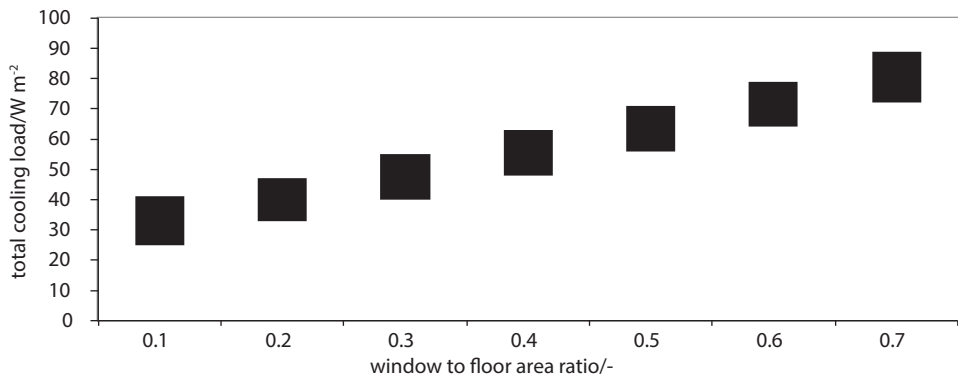


Figure 1.12 Cooling load as a function of the window to floor area ratio to net area.

In case of a very high energy-intensive use like computer centres or server rooms, the cooling loads could increase up to 1000 W m^{-2} .

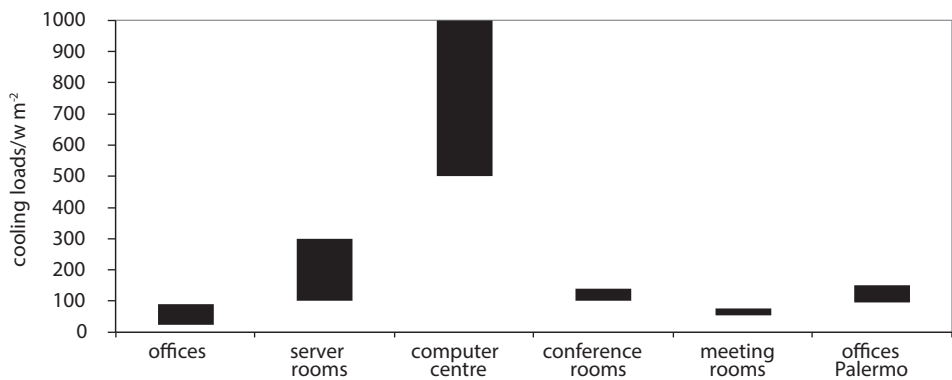


Figure 1.13 Typical cooling loads for buildings in Germany and offices in Palermo, Italy.

The sum of external and internal loads leads to an average cooling load in administrative buildings of about 50 W m^{-2} . The cooling load is dominated in many cases by the external loads.

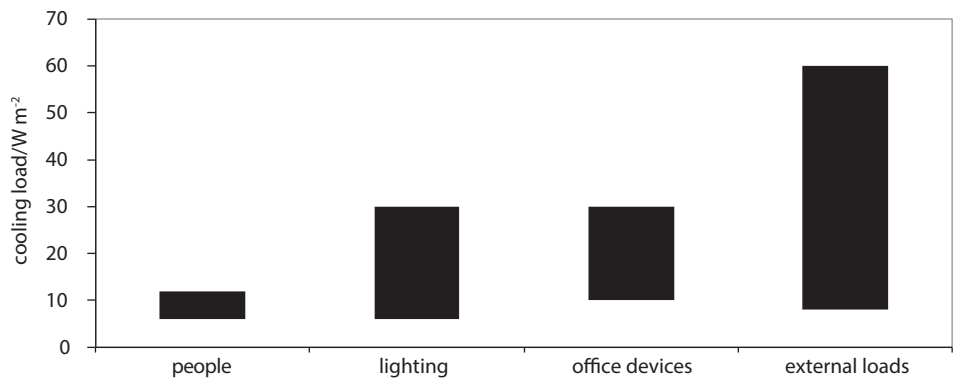


Figure 1.14 Cooling load distribution in office buildings.

1.4 Lighting electricity consumption

The average share of lighting in electricity consumption is 36% in administrative buildings, compared to only about 5% in the industrial sector. Due to the high luminous efficiency of daylight, the internal thermal loads due to lighting are in addition reduced, and thus also the problem of summer overheating in offices.

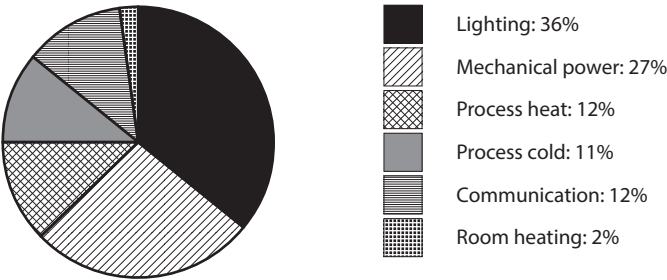


Figure 1.15 Average electricity consumption and lighting contribution for small consumers (trades, service sector and public sector) in Germany, 1998.

The lowest measured electricity consumption values for lighting are below $5 \text{ kWh m}^{-2} \text{ a}^{-1}$ in office buildings, e.g. the ZUB office building in Kassel, Germany, with facade high windows, a shallow room depth of 4.6 m and natural light-dependent dimming. During an intensive period of measurement evaluation, yearly values of $3.5 \text{ kWh m}^{-2} \text{ a}^{-1}$ were determined. For unregulated illumination in supermarkets or banks, the electrical consumption for lighting can reach values of $50 - 70 \text{ kWh m}^{-2} \text{ a}^{-1}$.

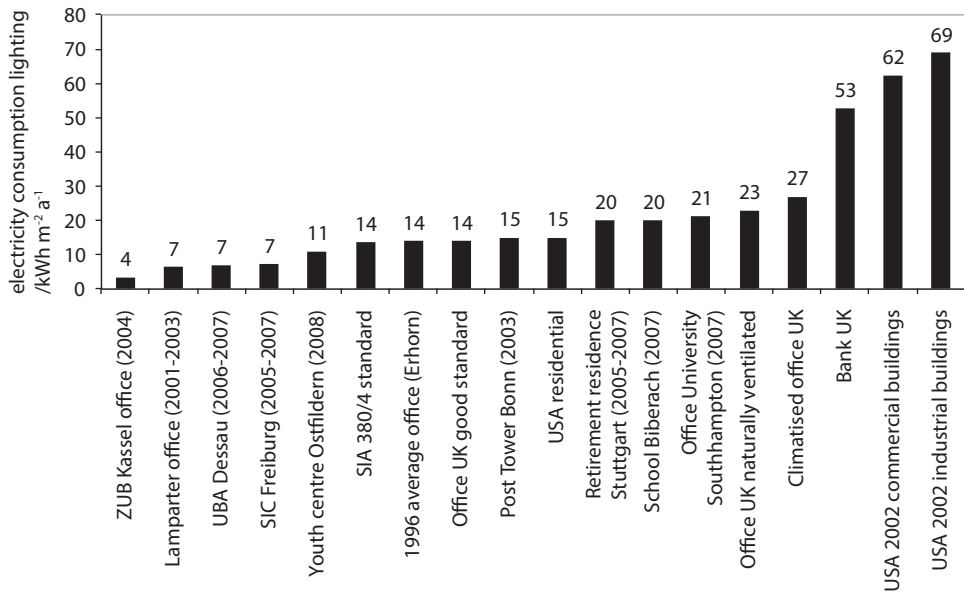


Figure 1.16 Electrical consumption for lighting in non-residential building types (measured values with years).

As part of one of the author's European research projects, an administration building of the University of Southampton completed in 2005 with 2600 m^2 was measured in detail. The electricity consumption for illumination was $21 \text{ kWh m}^{-2} \text{a}^{-1}$. An analysis of the daily and monthly consumption data demonstrated that there was no seasonal dependency in the lighting. The illumination consumption decreased only at the weekend by a factor of 10. The potential for electricity consumption savings through dimming dependent on natural light is clear.

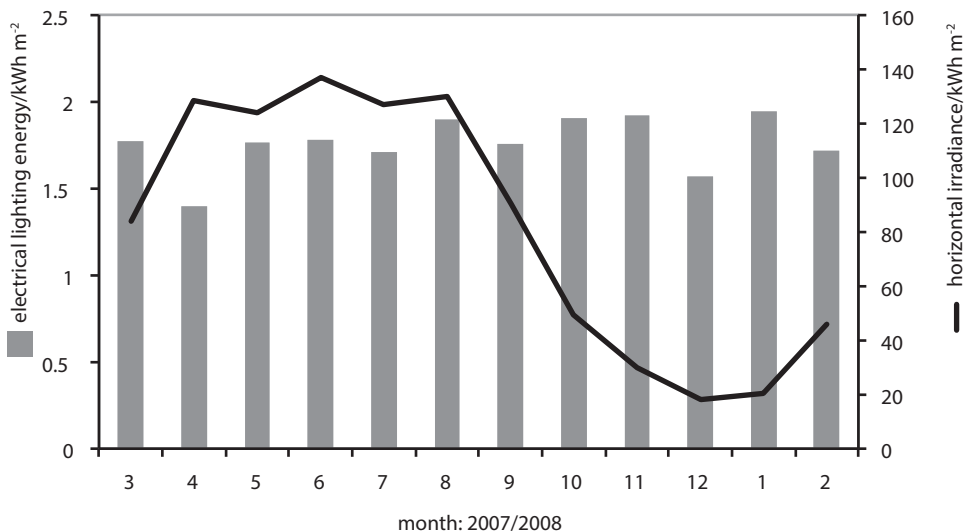


Figure 1.17 Monthly irradiance and measured illumination consumption for a new building in Great Britain.

In residential buildings, illumination contributes normally less than 10% to total electricity consumption. In Great Britain, 3% of electricity consumption was attributed to lighting.

1.5 Influence of the urban form on energy consumption of buildings

Today around 50% of the world's population are living in cities, which already account for 75% of all energy use. Urban areas with their industries and power stations are estimated to be responsible for about 60 - 70% of global CO₂ emissions (Satterthwaite, 2008). Residential buildings alone represent 63% of the total energy consumption in the European building sector and account for 10% of total greenhouse gas emissions (Balares et al., 2007).

The policies and urban master plan decisions on zoning influence the urban energy demand. The urban density refers to the number of people inhabiting a given urbanised area. Urban density can be influenced by increasing building depth, by increasing building height or reducing spacing or by increasing compactness.

The urban density can affect the total energy demand of a city in different ways, and these effects are complex and conflicting. Density supports district energy systems and infrastructure facilities are closer so it reduces also the energy requirement for distributing heat or cold. Steadman (1979) assumed that high-density linear development along transport routes would be more energy efficient than compact central development, as linear patterns allow better natural lighting, ventilation and passive solar gain. Building energy costs were estimated higher than travel energy demand. High density can increase energy demand due to limitation on natural ventilation, lighting and solar gains (Hui, 2001). In cooling-dominated climates, higher building densities reduce energy demand. On the other hand, in dense urban settlements the concentration of services reduces the need to travel large distances, and generally efficient transport is difficult to provide in the low-density cities. When comparing 10 major cities in the USA with 12 European cities, European cities were five times as dense, and the US cities consume 3.6 times as much transport energy per capita. A compact city is the most fuel efficient of all urban forms, with 43% less fuel consumption than 'business-as-usual' development (Newton, 1997).

The effects caused by urban design on heating, cooling and lighting energy demand of the spaces are different, and their influences are varying according to climatic conditions (see Table 1.3).

The main factors that affect daylight use and solar gains in buildings are the distance between buildings, the height of the facing building, the orientation of and the reflectance from the facing buildings, the size of openings and the size of the shading device. The daylight performance is especially significant for office buildings characterised by high lighting energy consumption and where the productivity of the employees is highly affected by lighting conditions. In residential buildings, electricity consumption for lighting is much less influenced by daylight performance due to a higher evening use profile.

Simulation models are often used to assess the annual energy consumption for electric lighting or the impact of daylight on the thermal behaviour of the building. The building energy performance in an urban context is evaluated by changing the height of the buildings while keeping the urban design density constant. To quantify the urban canopy layer assessments, the height to width ratio (aspect ratio) of the street canyon can be used.

	Heating	Cooling	Lighting
Height and width of buildings	Multi-storey buildings and compact forms reduce the heat losses from the building envelope.	Multi-storey buildings and compact forms reduce the heat gains from the building envelope.	Shading impact increases with increasing height to width ratio, causing higher artificial lighting requirement.
Street configuration	Density of the area increases the mutual shading and reduces the solar gains with increased heating and reduced cooling demand. District heating and cooling systems can be used efficiently in dense areas. Efficiency of solar heating and cooling systems decreases because of the shading effect of other buildings on the solar collecting area.		High density causes more daylight-controlled artificial lighting energy consumption at lower floor levels.
Thermal and optical properties of buildings	Low U value reduces the heat losses from building envelope.	Low U value reduces the heat gains from building envelope.	High reflection provides high illuminance levels and results in less electric lighting consumption.
	Low albedo and high absorption of the building envelope increase the urban heat island effect.		High visual transmission of glass provides high illuminance levels and results in less electric consumption.

Table 1.3 Urban effects on heating, cooling and lighting demand.

To analyse urban geometry configuration changes and building energy performance, the analysis should start from room scale that is a basic space of a building in an urban quarter. This is mainly important for electrical lighting requirements, which strongly depend on the location of a room within a building. In residential buildings, where energy efficient daylight is less important, it is not necessary to simulate individual rooms within a building.

When the electric lighting demand has been calculated, heating and cooling demand calculations can be done considering the thermal effects generated by artificial lighting. In this way, the total (heating, cooling and electric lighting) energy demand of the room is calculated. The building energy demand is obtained by adding up all rooms' energy demand in the building. Using the same approach, a complete urban site energy performance can be determined. The results presented have been obtained by two dissertations within the European PhD school CITYNET (Kesten, 2012 and Tereci, 2012).

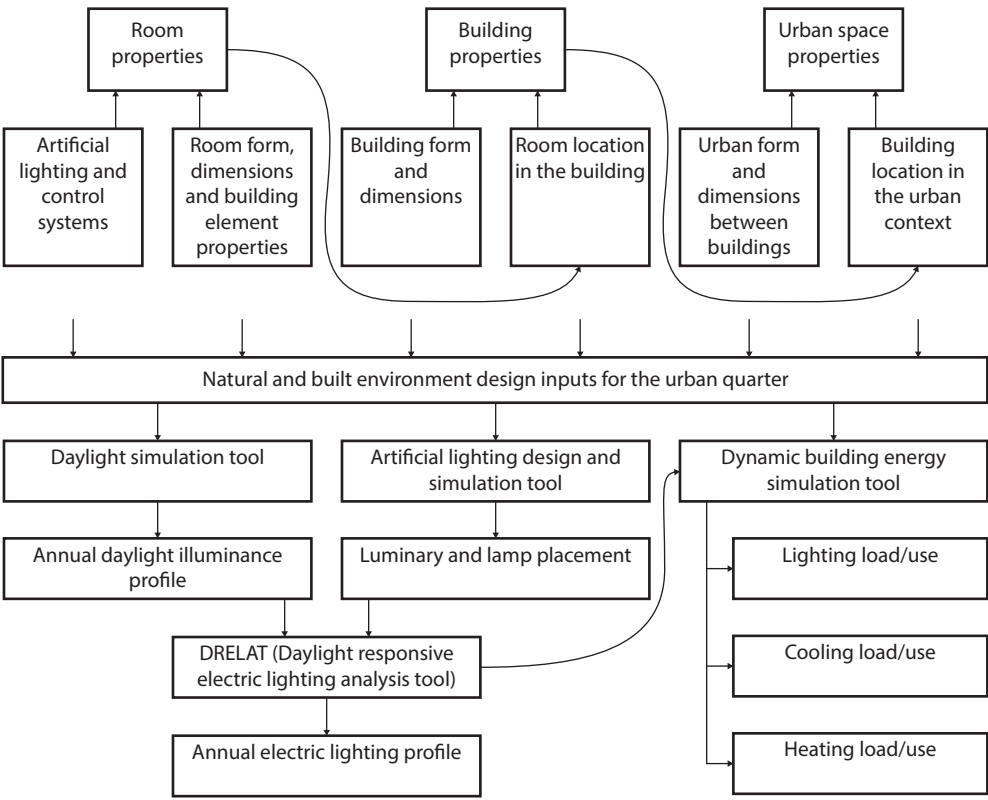


Figure 1.18 Analysis model for energy performance of buildings in an urban context (Kesten, 2012).

1.6 Office buildings in an urban context

The real urban texture is highly complex to compute. In order to limit these complexities, some archetypes were defined, and these simplified types are used especially for energy use studies. Two generic urban types are considered here: separated and continuous units. Separated unit, defined by geometrical ratios, are shown in Figure 1.19. Sixty different building configurations were analysed, corresponding to five levels of spacing distance (L_1/L_2), four levels of building depth (D/L_2), four levels of aspect ratios (H/W) and the cases without surrounding blocks. All simulations were conducted for all four cardinal directions: north, east, south and west.

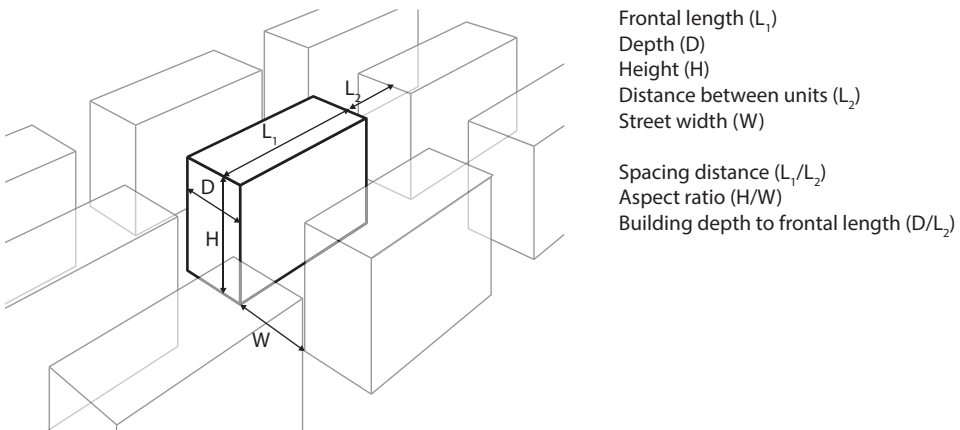


Figure 1.19 The form structure labels H , D and L_2 refer to the height, depth and frontal length of each unit, L_1 refers to the spacing between the units and W to the width of the street.

To calculate the daylight illuminance, the raytracing programme Radiance was used. The annual electricity consumption with a daylight-responsive control system was simulated in Radiance based on the lighting programme Daysim. The heating and cooling analysis including the annual electricity consumption with daylight-responsive control was carried out using the EnergyPlus simulation programme. EnergyPlus uses the hourly electric lighting schedules to integrate daylight-responsive electric lighting into thermal calculations.

The office building type is a cellular plan office, and the working schedule is weekdays from 08:30 am till 6:30 pm. The envelope was designed according to the German Energy Saving Ordinance 2009 (EnEV 2009). A pavilion type of urban generic form is used, which means a detached block of building. The three-storey reference building has a 10 m depth and a 20 m length. The reference office is located in the first floor – middle axis of the building and facing south. The dimensions of the office room are 2.5 m width, 4.5 m depth, 2.5 m height, and it has a 50% window to wall ratio.

Figure 1.20 shows total annual energy demand of the south-orientated sample office space under four different aspect ratio (H/W) scenarios in Southern German climatic conditions. The energy demand is strongly influenced by the H/W ratio when it changes from 0.5 to 1 and increases by 20%. The total energy demand slightly decreases with aspect ratios above 1. The increase of the heating and electric lighting energy demand is more than compensated by the decrease of the cooling energy demand. Therefore, the total energy demand decreases by 6%. On the other hand, the daylight-responsive electric lighting demand rises from 7 to 10 kWh m⁻² a⁻¹.

When decreasing the ratio of the lateral distance between buildings L_1 to the frontal length of the building L_2 from $L_1/L_2=1.5$ to $L_1/L_2=0.25$, which corresponds to an increase in urban site density from 17% to 29%, the heating and lighting demand slightly increases by 5% and the cooling demand decreases by 7%. For a given building height to width ratio (here $H/W=0.5$), the change of the L_1/L_2 ratio is not very relevant for blocking or increasing solar gains entering the space.

The annual energy demand is also not much affected by the building depth to the frontal length (D/L_2). The annual heat demand increases by less than 2% with increasing D/L_2 ratios, as the solar gains reduce. A similar tendency can be observed in the daylight-responsive electricity demand.

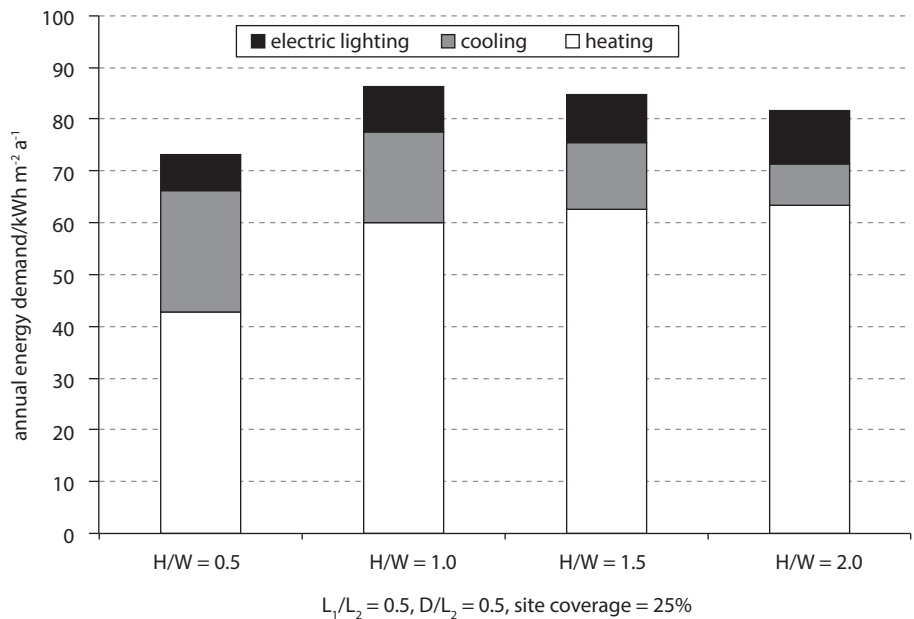


Figure 1.20 Annual energy demand of an office building room as a function of aspect ratios (H/W).

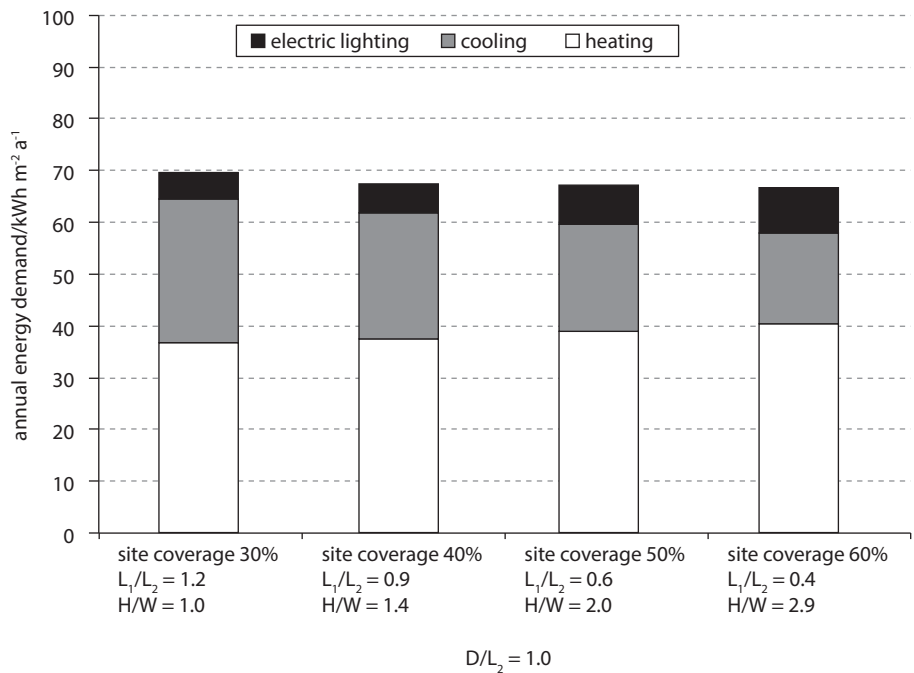


Figure 1.21 Heating, cooling and lighting demand of high-rise office blocks for different densities in Stuttgart.

High-rise office blocks with different site densities were also analysed for Southern German climatic conditions. The 10-storey reference building has 24.4 m depth, 24.4 m length and 30 m

height. The energy consumption of the office room's daylight-controlled artificial lighting was evaluated for different site coverage. The required illumination level of the office room is 500 lux (lx), and the artificial lighting system was designed to supply this level.

When the shading effect due to the surrounding buildings is taken into account, the electric lighting demand increases. The cooling energy demand is also affected by daylight-responsive controlled artificial lighting, as there is less heat gain generated by artificial lighting. In Figure 1.21, the effects caused by external obstructions can be observed. The shading effect results in less cooling requirement due to a decrease in solar gains. The simulated annual heating demand increases from 36 to 40 kWh m⁻² a⁻¹. At 60% site coverage, the annual cooling loads decrease to 17 kWh m⁻² a⁻¹, which is about 36% less than at 30% site coverage.

For the sample office room, the annual electric lighting demand was also evaluated for each site density as a function of floor height. On the first and fifth floor, the influence of site coverage is very important, whereas the top floor is no longer shaded by neighbouring buildings. This height sensitivity of lighting energy demand makes urban simulations complex.

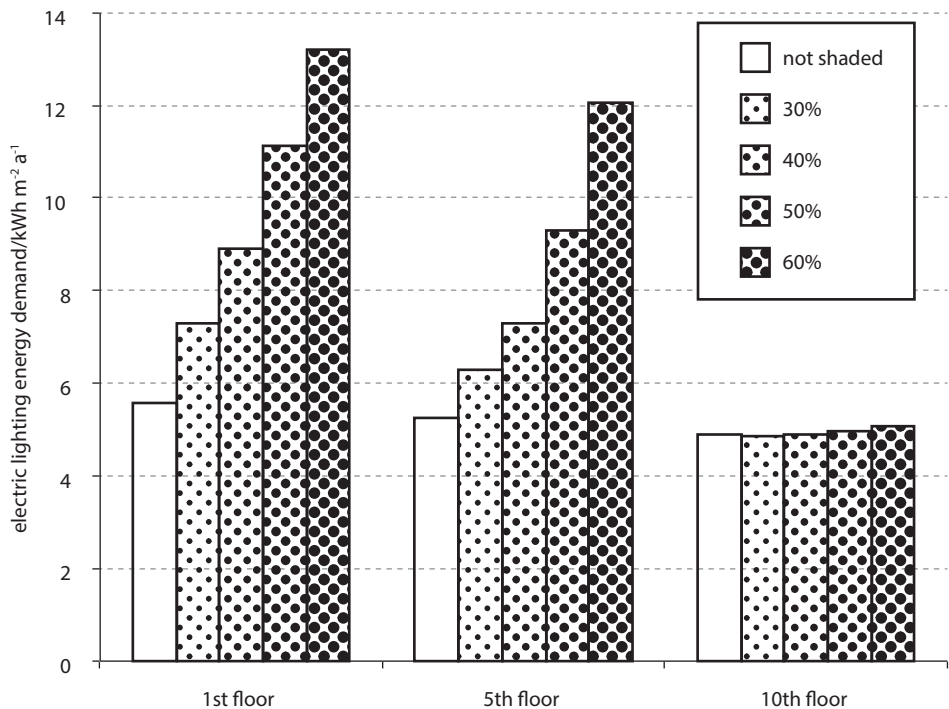


Figure 1.22 Daylight-responsive artificial electric lighting demand of office blocks for different site densities and different floor levels.

1.7 Residential buildings in an urban context

Settlement types for residential buildings were chosen using urban generic forms based on the work of Martin and March (1972). The generic urban forms have simple and repeatable characteristics, thus eliminating the complexities found in real urban sites and allowing for a more systematic comparative analysis of geometry and built form. These forms are represented in Figure 1.23.

For the analysis of residential buildings, the electricity consumption from lighting and appliances are included using given load profiles. Here no separate daylighting simulations are carried out, but the effects of shading on heating and cooling demand is considered. The residential settlement types are chosen as pavilion settlement types for one-family houses, apartment and high-rise blocks, as terrace settlement type for row houses and as pavilion courtyard settlement types for old houses.

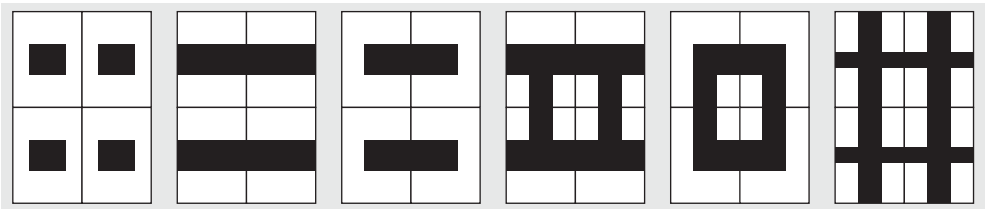


Figure 1.23 Generic urban zones based on the classification system of Martin and March (1972) with pavilions, slabs, terraces, terrace-courts, pavilion-courts and courts.

The urban shading effect is simulated with all obstructing buildings around the simulated one. The model includes the reflectance from the obstruction surfaces, and an albedo factor is defined for obstruction buildings and the ground. Site densities are defined as the ratio of built-up area to total area and were varied between 30% and 60%, as below 30% obstructions are nearly negligible and above 60% the buildings become unrealistically close (Tereci, 2012).

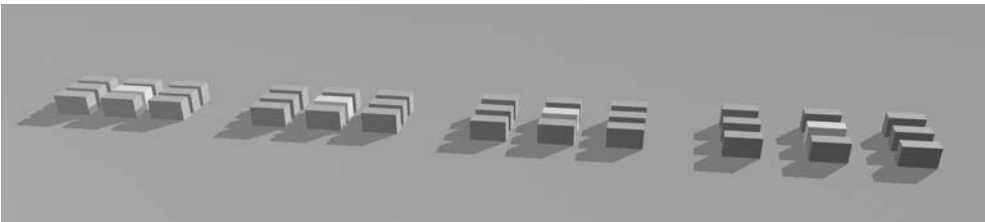


Figure 1.24 EnergyPlus models of multifamily buildings for shading simulations with site densities varying between 60% and 30%.

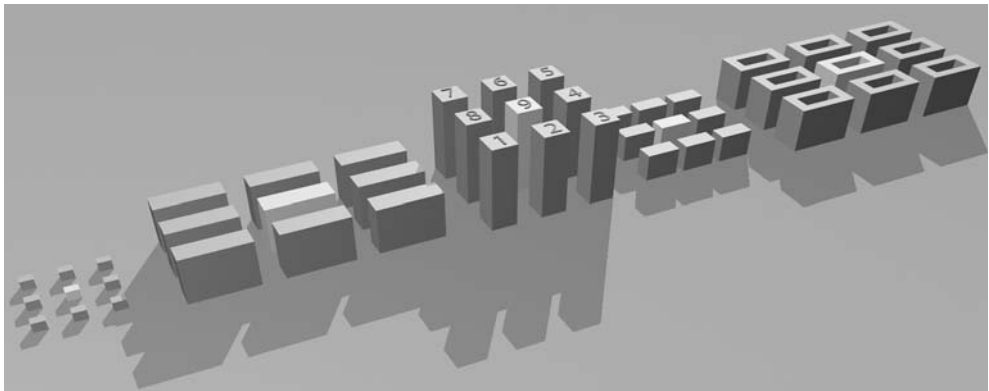


Figure 1.25 EnergyPlus models of building types and the urban quarter constituted of 9 generic building blocks from each building type. The distance between the buildings varies according to site densities. The building position numbers (1 - 9) are analogous for each cluster.

Geometry	Single-family house	Multifamily house	Old high-rise block	Row houses	Old apartment (courtyard)
Length	10.5 m	20 m	24.4 m	7 m	10 m
Width	10.5 m	14 m	24.4 m	10 m	10 m
Height	3.5 m	10.8 m	30 m	7 m	13 m
<i>U values/W m⁻² K⁻¹</i>					
Wall	0.5	0.3	0.88	0.5	1.45
Roof	0.3	0.3	0.97	0.3	1.3
Floor	0.22	0.22	0.85	0.22	2.6
Window	1.6	1.6	2.57	1.6	2.9

Table 1.4 Geometric properties and *U* values of building types.

The classification of the urban settlements traditionally relies on their population and activity type. German statistic data and measurements from a case study project near Stuttgart were applied for estimating the occupancy scenario (Strzalka et al., 2011). All residential buildings were simulated using the same occupation scenario.

The scenario is based on a family with four people with no daytime use of the flat except weekends. Every house has television, computer, washing machine, dishwasher, oven, fridge and microwave. It was assumed that appliances were used early in the morning and evening time for weekdays and weekend. EN ISO 13791 was taken as an input for internal gains from occupants. The lighting was defined with 13 W m⁻² and 40% of those are convective gains. The heating set point is 19°C and cooling set point is 26°C.

The heating, cooling, electricity and hot water demand for buildings without obstructions are simulated and serve as a baseline for comparison.

Reference buildings	Heating demand	Cooling demand	Electricity demand	Hot water demand
	per kWh m ⁻²			
Single-family house	72	10	23	28
Multifamily house	44	11	33	28
Old high-rise block	83	16	21	28
Old apartment with courtyard	168	4	25	28
Row houses	74	4	19	28

Table 1.5 Dynamic simulation results of reference buildings without shading effect using typical U values from the German building stock. The electricity demand shown includes appliances and a lighting energy demand of 5 kWh m⁻² a⁻¹.

The highest heating demand value of the old courtyard apartment is 168 kWh m⁻² a⁻¹ with a cooling demand of 4 kWh m⁻² a⁻¹. The minimum heating demand value of the multifamily house is 43.7 kWh m⁻² a⁻¹ with 11.4 kWh m⁻² a⁻¹ cooling demand. The thermal performance of the multifamily house is better than the other forms due to the most compact building form with the smallest outside wall area.

1.8 Site density effect

First the heating and cooling demand of a building type in the centre of an urban structure was analysed as a function of site density (building position 9 described above). In a second step the influence of the position of each building within the urban setting is shown.

For single-family houses with a building standard corresponding to the construction years between 1995 and 2001, the heating demand increases by 17% (to 84 kWh m⁻² a⁻¹) for a 60% site density compared with the unshaded situation (with 72 kWh m⁻² a⁻¹). Using an increased albedo of 0.7 for the surroundings instead of 0.2, the heating demand slightly reduces to 79 kWh m⁻² a⁻¹ for the 60% site coverage.

Similar results are obtained for the multifamily houses. Without any shading effect the heating consumption of multifamily house is 43.7 kWh m⁻² a⁻¹ and increases by 20% for 60% site coverage. The rather low cooling demand of 11.4 kWh m⁻² a⁻¹ reduces with site density to 6.4 kWh m⁻² a⁻¹, i.e. by 44%.

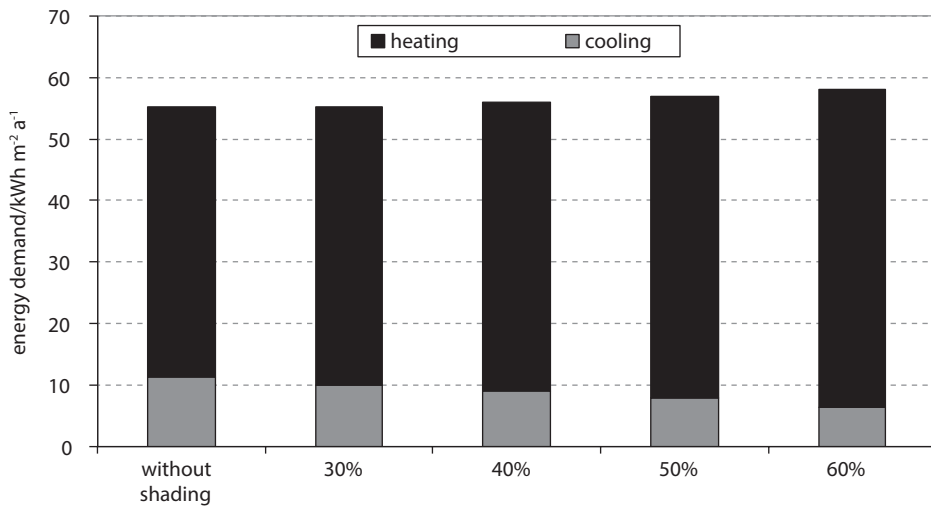


Figure 1.26 Average heating and cooling demand of multifamily houses with increasing urban site density under Stuttgart weather conditions. Similar tendencies are observed for the single-family or high-rise blocks.

The heating consumption of the high-rise blocks with today's high insulation standards is $53 \text{ kWh m}^{-2} \text{ a}^{-1}$ and cooling demand is $10 \text{ kWh m}^{-2} \text{ a}^{-1}$. The heating demand is 26% higher for 60% site coverage. For high-rise blocks constructed between 1981 and 1985, the heating demand of $83 \text{ kWh m}^{-2} \text{ a}^{-1}$ rises by 17% for 60% site coverage.

The heating consumption of row houses is so influenced by the surface area proportion exposed to ambient conditions. The heating consumption of corner houses is 17% higher than the other buildings in the middle of the row house block as shown in Figure 1.22. The heating demand increases with site density by 11% for the corner houses and 14% for the middle houses. The reflection effect with density almost does not influence the heating demand of the row houses.

Figure 1.28 shows the heating and cooling demand of existing, old apartment blocks within an urban courtyard form. The energy demand for heating is even stronger, affected by the position of the building in the block as for the row houses, as not only the surface area to ambient, but also mutual shading between the buildings, play a role. In order to quantify this effect, nine buildings with the same envelope properties were simulated and then the site density was varied.

The south-facing central buildings in the second row of the courtyard (buildings 7 and 8) have up to 80% higher heating demand than the first row buildings, as they are already shaded by the buildings within the courtyard. The influence of site density on the second row buildings (number 5 to 10) is only 1 - 4%.

The buildings in the first row with no mutual shading on their southern facade increase in heating demand with site density by 9 - 10% (buildings 1 - 4). When considering the position of one building type with a given building compactness (such as multifamily, single-family, courtyard, etc.) within the urban structure, the influence of shading becomes evident.

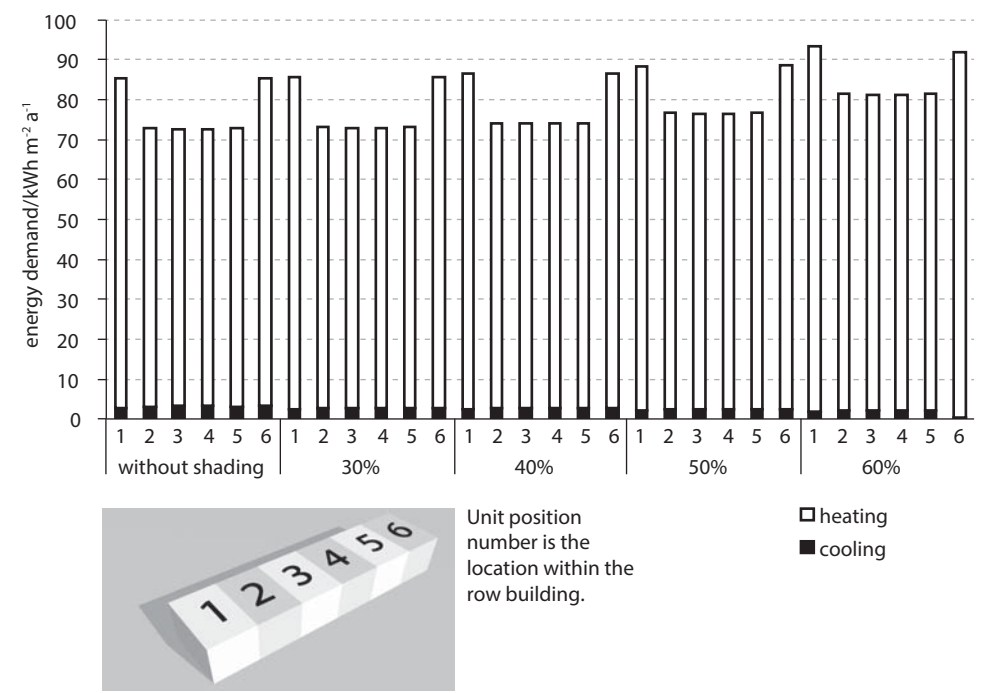


Figure 1.27 Dynamic simulation results of row houses with different densities.

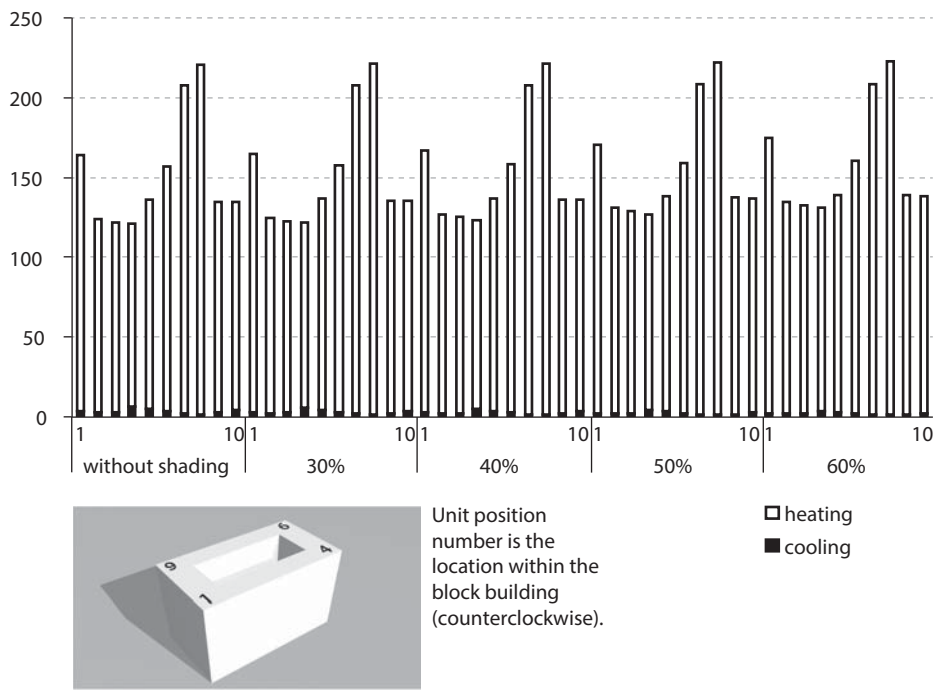


Figure 1.28 Dynamic simulation results of old apartment blocks within the courtyard urban form.

The most affected from mutual shading is the building located in the middle of the urban structure that was considered above (building 9) in Figure 1.25. Compared to the unshaded situation, its heating energy demand increases by 3% for 30% site density up to 20% for 60% site density.

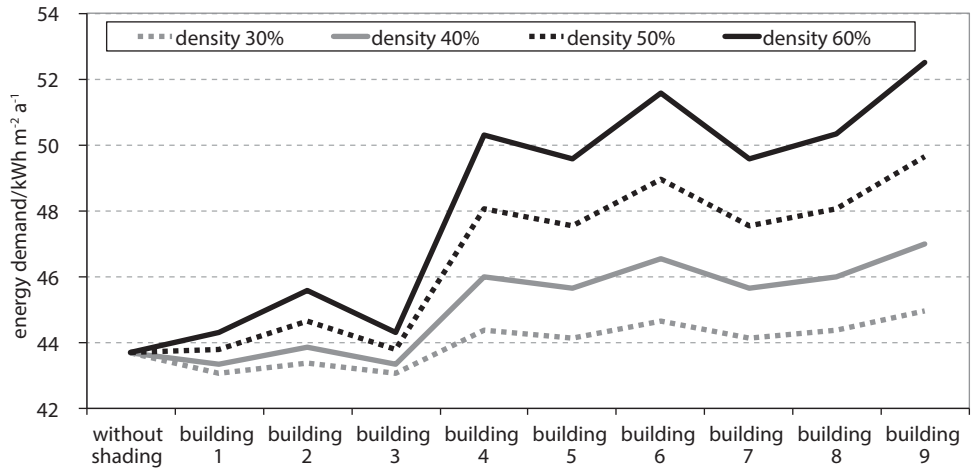


Figure 1.29 Heating demand of the multifamily house with different site coverage.

For the lower height single-family houses, the shading effect is less. Building 9 in the centre of the site has a 13% higher demand for 60% site coverage than the unshaded buildings.

1.9 Climate effect

To understand the weather impact together with site density effects, apartment blocks are simulated for Stuttgart/Germany, Ankara/Turkey and Hong Kong/China weather data.

In all climates the cooling demand decreases with increasing site density and the heating demand increases. In heating-dominated climates such as Germany, the overall energy demand for heating and cooling thus slightly increases by 6% for the highest site density. In Ankara/Turkey with comparable heating demand, but double the cooling demand than in Stuttgart/Germany, there is an optimum for the total energy demand at 30% site density. In Hong Kong with its cooling-dominated climate, the energy demand drops by 20% compared with the unshaded situation.

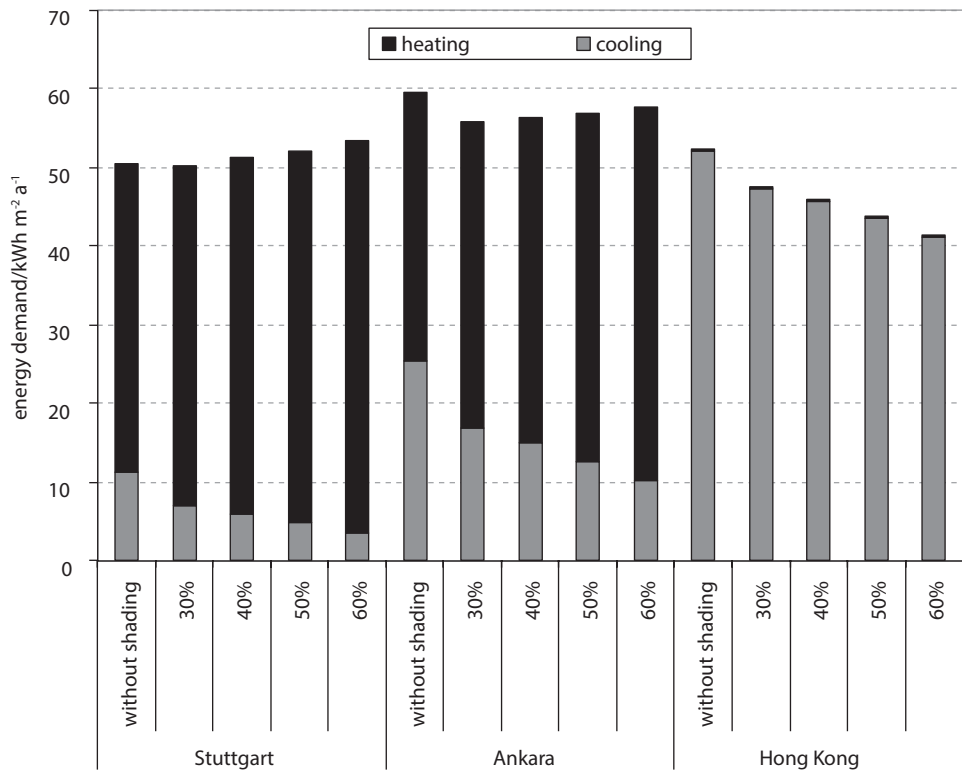


Figure 1.30 Comparison of heating and cooling demand of apartment blocks with U values according to the new German legislation (EnEV 2009) for different settlement densities in Stuttgart, Ankara and Hong Kong climate.

1.10 Albedo effects

The surface albedo can assume any value between 0.0 and 1.0 for a black body to perfect reflection of a white surface. Two albedo factors were evaluated: 0.3 for dark colour rough surfaces and 0.7 for light colour metallic or granite surfaces. When the albedo factor is high, the heating consumption slightly decreases by about 1% and cooling consumption increases. When only considering reflection from surfaces and not secondary effects such as heat island temperature increases, the Albedo factor does not have much influence on the heating performance of the buildings, the less so, the higher the buildings are.

In cooling-dominated climates, the influence of surface albedo is higher and reaches up to 10%, even when only simulating the additional short-wave irradiance increase by higher surface albedo.

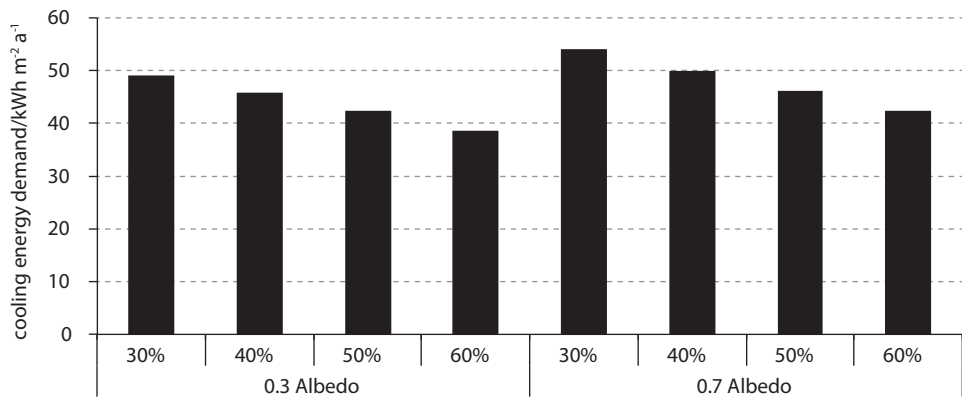


Figure 1.31 Cooling demand for high-rise blocks with different site coverage and 0.3 and 0.7 albedo factors in Hong Kong climate.

1.11 Thermal properties of the building envelope

To conclude the study, the relative importance of the urban density was compared to the properties of the building envelope and the compactness of the building.

First the transmission losses of the building components were reduced using U values corresponding to the typical standard values in Germany according to the construction year. Heat recovery for ventilation was not considered, i.e. the ventilation is always natural ventilation.

For all building forms and urban settings (single-family, multifamily, high-rise, courtyard), the influence of the building standard on heating demand is higher than the site density. The largest influence of envelope properties is obtained for the single-family houses with the highest surface to volume ratio. Here the heating demand decreases by 75% when the highest insulation standards are applied. For multifamily and high-rise blocks with lower surface to volume ratios, about 60 - 70% can be saved by increasing the insulation standards.

Building component	1948	1968	1978	1983	1994	2000	2007	2009	Low energy		Passive standard
									1	2	
Outside walls	1.7	1.4	1.03	0.8	0.6	0.5	0.3	0.24	0.23	0.15	0.13
Floor	0.8	0.8	1.0	0.8	0.6	0.5	0.3	0.24	0.3	0.26	0.19
Roof	0.8	0.8	0.6	0.5	0.3	0.3	0.3	0.24	0.25	0.19	0.15
Windows	5.89	2.72	2.72	2.72	1.93	1.82	1.73	1.3	1.2	0.9	0.8

Table 1.6 Envelope property variations according to construction year and building energy standards in Germany.

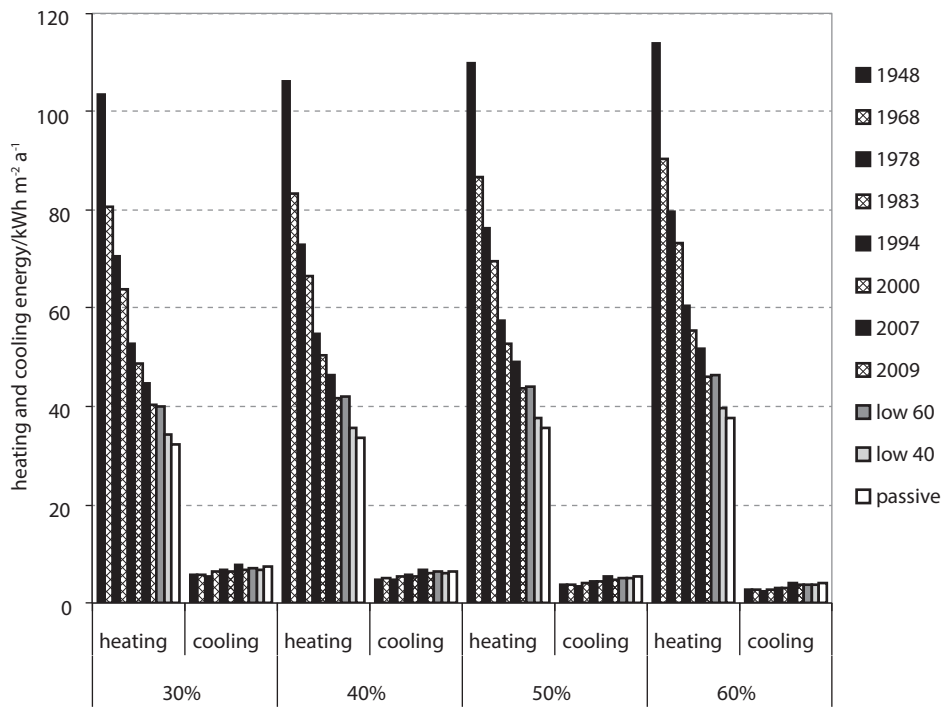


Figure 1.32 Heating and cooling demand of the multifamily house with different site coverage and envelope properties. The building ages indicate the thermal properties of typical German building stock, 'low 60' and 'low 40' means low-energy buildings with 60 or 40 kWh m⁻² a⁻¹ heating demand, the passive standard buildings are simulated without ventilation air heat recovery.

The influence of the envelope parameters was also evaluated for row houses and buildings within the urban courtyard structure. The corner houses of the row houses have higher energy demand in all conditions, as more envelope area is exposed to ambient air. Increasing insulation standards from low energy to passive house standards results in 25% less heating demand of the buildings. Similar results are obtained for the courtyard buildings with 20% reduction from the 2007 low energy to passive standard. Here the shading factor from the mutual obstruction by buildings has the strongest influence.

1.12 Solar gains and glazing

Direct solar gains are significantly influenced by the glazing ratio. The simplest and the most effective way for more solar gains is to have bigger glazing areas, which on the other hand also cause heat loss depending on the glazing type and U value. For all building types the heating and cooling demand was calculated with various glazing ratios for different orientations. At first the glazing ratio was only changed on one facade with fixed other glazing ratios. As a result, the heating demand increases strongest for the northern facade, but hardly so for a southern facade.

A single-family house with no windows on the north facade has 26% less heating demand than the single-family house with a totally glazed north facade ($65 \text{ kWh m}^{-2} \text{ a}^{-1}$ with no north facade glazing compared to $93 \text{ kWh m}^{-2} \text{ a}^{-1}$ for the fully glazed northern facade), whereas on the southern facade the demand reduces by 7% with reducing window area. With today's high-performance double or triple glazing, the heat demand on Southern facades no longer increase with higher glazing fractions.

In multifamily houses, the heating demand decreases from 67 to $38 \text{ kWh m}^{-2} \text{ a}^{-1}$ with decreasing northern glazing fraction, i.e. by 42%, and by 24% for high-rise buildings (from 62 to $47 \text{ kWh m}^{-2} \text{ a}^{-1}$).

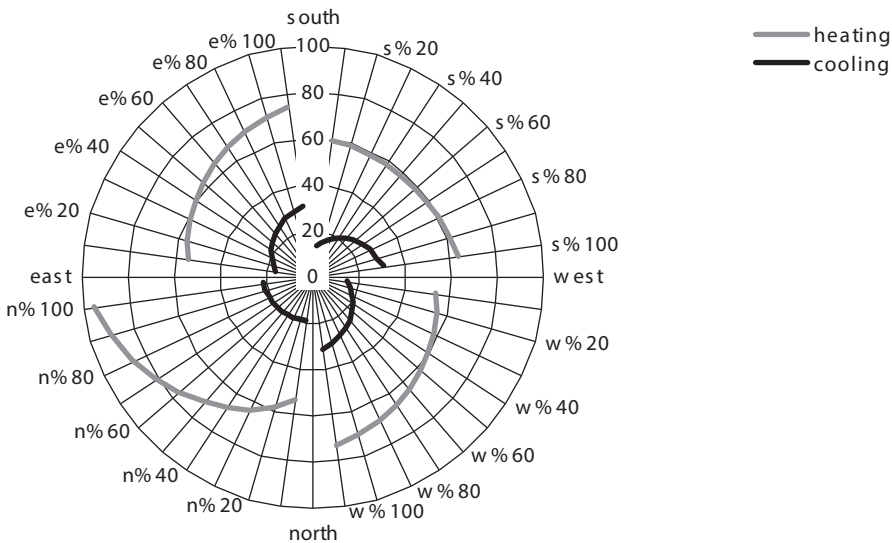


Figure 1.33 Heating and cooling demand (in $\text{kWh m}^{-2} \text{ a}^{-1}$) as a function of glazing ratio of a given facade orientation for a multifamily apartment block under Stuttgart climatic conditions.

In order to understand the importance of glazing ratios in relation to urban density, three types of glazing ratios were evaluated in four different site densities and the results are represented in Figure 1.35. As a reference scenario, typical glazing ratios from the German building typology were used for the different building types and their construction year (30% for the single-family building, 30% for the multifamily building, 32% for the high-rise apartment blocks, 16% for row house and 17% for the old apartment house). When the glazing ratio is increased on all facades, the heating demand increases with increasing site density. The cooling demand nearly doubles with high glazing fractions and again decreases with increasing site density. Higher site densities increase the heating demand by 12 - 18%, but reduce cooling by up to 50%.

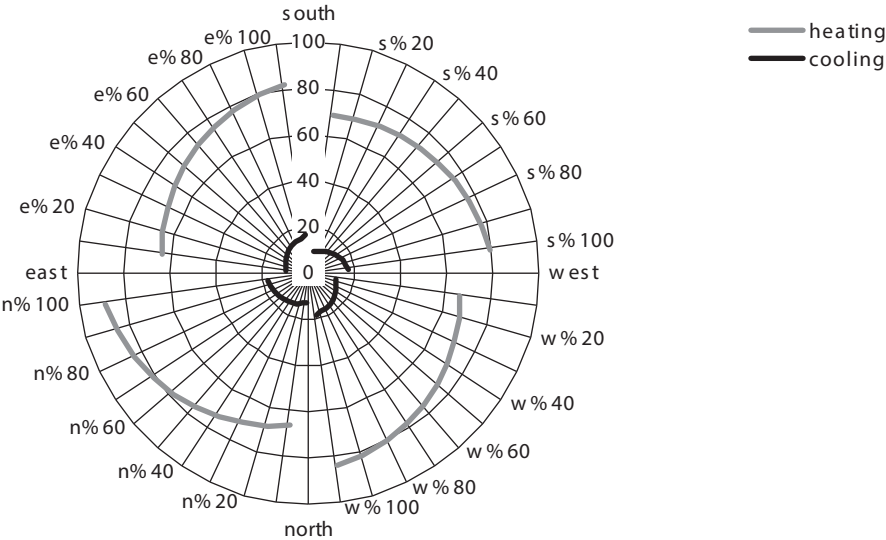


Figure 1.34 Heating and cooling demand as a function of glazing ratio effect for high-rise blocks under Stuttgart climatic conditions (in $\text{kWh m}^{-2} \text{a}^{-1}$).

1.13 Building typology and urban form

To evaluate the influence of building typology on the energy performance of an urban area, the same size urban district with different building types and equal density is evaluated. The envelope properties of the buildings are the same, and they are based on the German low-energy standard (EnEV 2007). The heating demand of multifamily houses is the lowest with $45 \text{ kWh m}^{-2} \text{a}^{-1}$, and apartment buildings within a courtyard form have the highest heating demand per square metre with $75 \text{ kWh m}^{-2} \text{a}^{-1}$, i.e. the building form is responsible for up to 35% heating energy reduction. The cooling demand also changes significantly with building typology, but is generally low in heating-dominated climates for residential buildings.

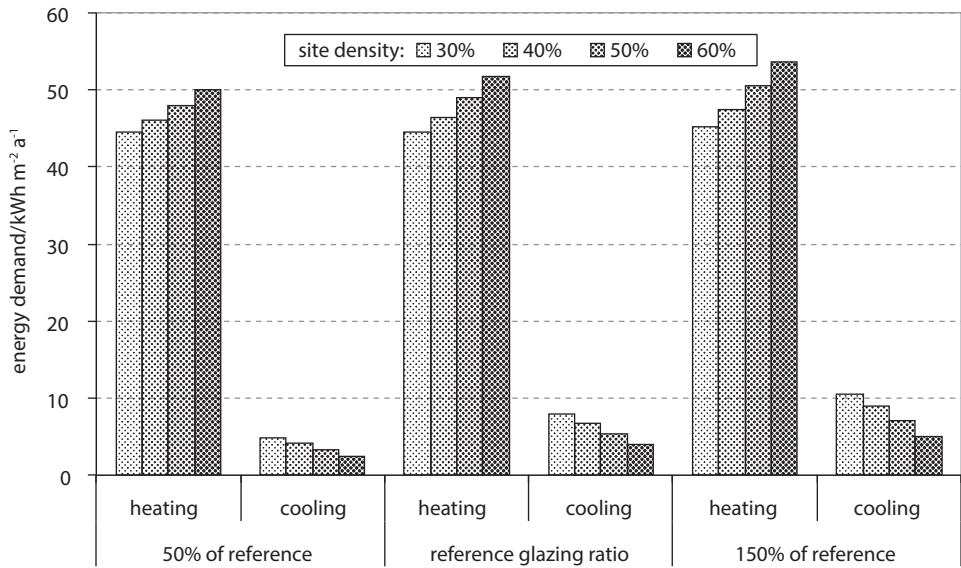


Figure 1.35 Influence of glazing ratio on heating and cooling demand for multifamily houses with different site densities.

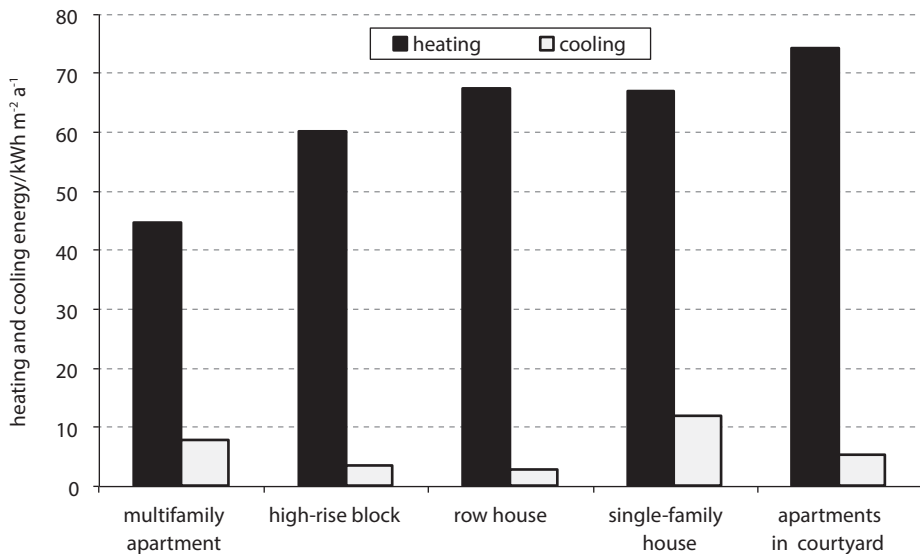


Figure 1.36 Heating and cooling energy demand for different building typologies with the same insulation standard.

Finally, the total primary energy performance for different building typologies and urban forms were evaluated for a given site area. Depending on the building typology chosen, different number

of buildings, flats and people can be placed on the site. For each building type, the heating and cooling demand was simulated. The site density chosen was 40%.

	Number of buildings	Flats	People	Conditioned floor area /m ²	Total heating energy demand /MWh	Total cooling energy demand /MWh
Apartment within courtyard block	9	360	1 440	36 000	2 689	191
Single-family houses	88	88	352	8 800	592	105
Multifamily house	31	372	1 488	34 720	1 555	271
High-rise blocks	14	560	2 240	83 342	5 034	283
Row houses	21	147	588	20 580	1 393	576

Table 1.7 Building type energy performance for a site coverage of 40%.

Delivery distribution losses of 10% were added to the demand of heating and cooling. Heating was provided to 80% by an electric heat pump with a COP of 4.0 and to 20% by a gas condensing boiler with 96% efficiency, cooling by an electric chiller with a COP of 3.0. Also auxiliary electrical energy for pumping was added. For lighting, 5 kWh m⁻² a⁻¹ electricity consumption was assumed. The total building primary energy demand thus includes heating and cooling with distribution losses and auxiliary electric energy and lighting. The CO₂ emission factor for natural gas was 0.202 t CO₂ MWh⁻¹ and for electricity 0.539 t CO₂ MWh⁻¹ (www.iea.org).

	Average heating demand /kWh m ⁻²	Average cooling demand /kWh m ⁻²	Average primary energy demand /kWh m ⁻²	CO ₂ Emission /t CO ₂	CO ₂ Emission per capita /t CO ₂ cap ⁻¹
Single-family houses	67	12	95	174	0.49
Multifamily houses	45	8	71	486	0.33
High-rise blocks	60	3	79	1291	0.58
Apartments with courtyard form	75	5	91	1668	1.10
Row houses	68	3	84	338	0.57

Table 1.8 Energy and CO₂ evaluation of the five building typologies in the same size urban area.

It can be seen that urban areas with multifamily houses have the lowest specific primary energy demand and per capita CO₂ emissions, single-family houses the highest. Due to the low density and

number of people in single-family urban areas, the absolute energy consumption and emissions are lowest for this urban structure.

1.14 Conclusions

The impact of urban configurations, building typology and building standards on energy consumption was evaluated. Site density and building envelope properties have significant effect on the site energy performance and should be carefully analysed in the urban design process.

Detailed dynamic thermal simulations including mutual shading of buildings show that heating energy increases by 17 (single-family building) to 25% (high-rise building) when increasing the site density from a moderate 30% to 60%, while the rather low-cooling demand decreases by up to 45%.

Exposure of walls to ambient air, i.e. the building compactness, is in the same order of magnitude; for example, a row house in the middle of the block has 17% less heating demand than the corner house. Courtyard forms lead to very high mutual shading between the buildings, so that the most disadvantaged apartments with hardly any solar gains have up to 80% higher heating demand. When varying the site density for courtyard from typologies between 30% and 60%, the increase in heating demand is smaller at 20% compared to an open space situation.

In cooling-dominated climates, the cooling demand decreases by about 20% with increasing density. While albedo effects are rather negligible for heating-dominated climates, it might increase the cooling demand by up to 10%. If the glazing ratios are increased on all facades, the heating demand typically increases by 10 - 20%, while changes only on the Southern facade do not affect the heating demand significantly, as gains and losses balance.

While in heating-dominated climates the combined heating and cooling energy demand slightly increases with site density (6%), increasing site densities decrease the total demand in cooling-dominated climates. In climates with comparable heating and cooling energy demand, there is an optimum site density between completely unshaded free spaces and high site densities.

The building compactness and insulation standard have the strongest influence on the heating and cooling demand. Compact multifamily houses have the lowest primary energy demand and CO₂ emissions per capita. High-rise urban settings can increase the person density, but at higher specific emissions.

For a given insulation standard, the building typology alone varies the heating and cooling demand by up to 60%. For a given building type, changing insulation standards also influence the heating and cooling demand in this order of magnitude.

For a given fixed size urban area, multifamily houses have the lowest specific primary energy demand and per capita CO₂ emissions, single-family houses the highest. Due to the low density and number of people in single-family urban areas, the absolute energy consumption and emissions are lowest for this urban structure.

For non-residential buildings with a higher demand in lighting electricity, the simulation results showed that between 4.5% and 35% of lighting electricity demand can be saved by the site design. For high-rise office buildings, the daylight-responsive electric lighting demand rises from 5 kWh m⁻² for 30% site coverage to 8 kWh m⁻² in 60% site coverage. If no daylight-responsive strategy is used, the electric lighting demand would be 36 kWh m⁻², which is five times as much in the worst case.

For cooling-dominated climate conditions, the site design should focus on reducing the cooling load, but the daylighting illuminance level and the electric lighting loads should also be considered.

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A photograph of a modern architectural courtyard. On the left, a white building with large windows is partially visible. A large, dark, angular overhang extends from the top left towards the center. In the background, a dark building with several vertical poles stands under a bright blue sky with scattered clouds. A bright sun is visible in the upper right, creating a strong lens flare and illuminating the scene. A water wall, consisting of numerous thin vertical pipes, runs across the middle ground, with water spraying upwards and outwards, creating a misty effect. The foreground is a paved area with dark, rectangular tiles. The overall atmosphere is bright and modern.

2 Part A: Passive solar

Figure 2.1 (previous page): Stadthaus municipal administration building in Scharnhäuser Park near Stuttgart, Germany (Photo: M. Seubert).



Figure 2.2 Elektror building in Scharnhäuser Park near Stuttgart, Germany with a concrete core activation for an efficient heat and cold distribution (Photo: M. Seubert).

Passive solar energy use contributes significantly to the energy demand of every building. The main energy supply is via short-wave solar irradiance transmitted by glazing, which provides daylight and is converted into heat by absorption on wall surfaces. This form of energy transfer is described here as **passive** and takes place solely by thermal conduction, solar irradiance, long-wave radiation and free convection, i.e. transfer is not line-bound and requires no auxiliary mechanical energy for moving a heat carrier.

Solar irradiance is absorbed without transport losses directly by the building shell or internal storage masses. Besides windows and the associated internal storage mass, the possibilities of passive use also include transparent thermal insulation on a heat-conducting external wall. Despite having a lower solar efficiency compared to windows, transparent thermal insulation in connection with a massive building component enables a temporal phase shift between irradiance and utilisation of heat. This characteristic reduces overheating problems that can occur with large glazing surfaces.

Unheated winter gardens rank among the classical forms of passive solar use. As elements placed in front of the building shell, they contribute to the building insulation, but at the same time they reduce the amount of solar irradiance available to the main building. Thus, the amount of both the daylight and the direct heat entry by solar irradiance into the adjoining heated rooms are clearly reduced. Furthermore, the indirect heating of glazed conservatories by adjoining rooms often leads to an increase in the heating requirement of buildings. Only with a very energy-conscious use of an unheated conservatory can energy gains for the building actually be achieved.

2.1 Passive solar use by glazing

Glazings are characterised by high transmittances for short-wave solar irradiance up to wave lengths of 2.5 μm and an impermeability to long-wave radiant heat emitted from building components with a maximum intensity at wavelengths of around 10 μm . This transmission characteristic results in a greenhouse effect and the conversion of the solar irradiance by absorption in structural elements into heat, whose long-wave radiation proportion is not transmitted through the glazing.

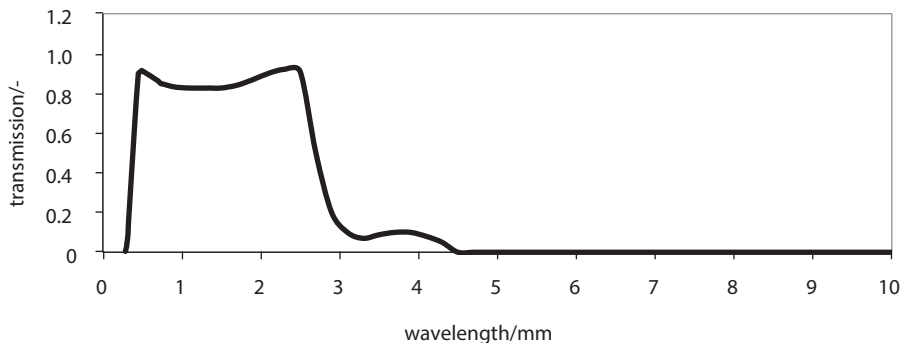


Figure 2.3 Wavelength-dependent transmittance of a single glazing.

Total energy transmittance of glazing

Aside from the direct transmission of the short-wave solar radiation with a transmittance τ , part of the irradiance is absorbed in the panes. Heating the panes causes a heat flow towards the room, which contributes to the total energy transmission factor (g value). The absorption coefficient of single glazing can be up to 50% for special heat absorbing single glazing. According to DIN EN 410, the secondary heat emission q_i is defined as the relation of the heat flow on the room side $\dot{Q}_i$ per square metre of window area A_w to the impacting solar radiation G . The value of q_i is calculated by solving heat balance equations. Transmission, absorption and reflection coefficients must be known for each pane of a multi-pane system. The total energy transmission factor then results from the relation of the total heat flow $\dot{Q}_{total} / A_w$ into the room to the irradiance:

$$g = \frac{\dot{Q}_{total} / A_w}{G} = \frac{\tau G + \dot{Q}_i / A_w}{G} = \tau + q_i \tag{2.1}$$

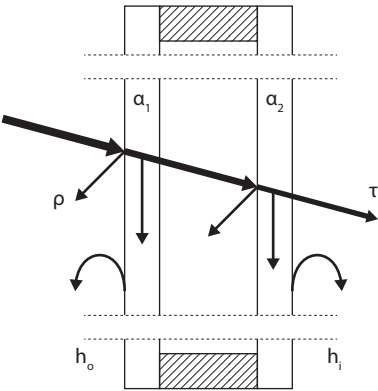


Figure 2.4 Transmission τ , reflection ρ , absorption α and heat transfer coefficients at the inside (h_i) and outside (h_o) of a double glazing.

Table 2.1 lists typical g values for various transparent building components.

Building component	g Value
Single glazing	0.87
Double glazing	0.76
Low-e double glazing (with a selective coating)	0.50 - 0.70
Triple glazing, normal	0.60 - 0.70
Triple glazing (with a double selective coating)	0.35 - 0.50
Sun protective glazing	0.20 - 0.50
Transparent insulation (100 - 120 mm; $0.8 \leq U_e \leq 0.9 \text{ W m}^{-2} \text{ K}^{-1}$)	0.35 - 0.60

Table 2.1 Typical g values for transparent building components.

The transmittance τ is calculated based on DIN EN 410 by integration of the wavelength-dependent transmission over the solar spectrum. With perpendicular incidence, transmittance is approximately 90% for an uncoated simple float glass and about 80% for a two-pane system. Transmittance falls and the total energy transmission factor of the two-pane system rarely exceeds 65% due to the metallic coating in thermally insulating, low-e glazing commonly used today on the room-side panes. The absorption factor α of the short-wave solar radiation is similarly calculated for each pane by integration over the spectrum, taking into account inter-reflections in multi-pane systems.

The solutions of the heat balance equations for single, double and triple glazing are indicated in EN 410. Using the example of single glazing, a heat balance can be described as follows: the intensity αG absorbed by the pane of surface A_w is divided into a heat flow inward $\dot{Q}_i / A_w$ and outward $\dot{Q}_o / A_w$.

$$\alpha G = \frac{\dot{Q}_i}{A_w} + \frac{\dot{Q}_o}{A_w} \quad (2.2)$$

These heat flows can be calculated with the help of the internal heat transfer coefficient h_i (standard value $7.7 \text{ W m}^{-2} \text{ K}^{-1}$) and external coefficient h_o (standard value $25 \text{ W m}^{-2} \text{ K}^{-1}$), and the temperature difference between pane surface T_s and room air T_i or outside air T_o .

$$\frac{\dot{Q}_i}{A_w} = h_i (T_s - T_i) \quad \frac{\dot{Q}_o}{A_w} = h_o (T_s - T_o) \quad (2.3)$$

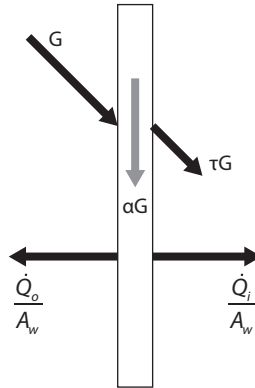


Figure 2.5 Irradiance G , absorption αG and secondary heat flows of single glazing outward and inward.

From the heat flow balance, first the pane surface temperature T_s is determined:

$$\begin{aligned}\alpha G &= (h_i + h_o)T_s - h_i T_i - h_o T_o \\ \Rightarrow T_s &= \frac{\alpha G + h_i T_i + h_o T_o}{h_i + h_o}\end{aligned}\quad (2.4)$$

With this pane surface temperature T_s , the heat flow inward can be calculated:

$$\frac{\dot{Q}_i}{A_w} = h_i (T_s - T_i) = \frac{h_i}{h_i + h_o} (\alpha G - h_o (T_i - T_o)) = \underbrace{\frac{h_i}{h_i + h_o} \alpha G}_{\text{secondary heatflow}} - \underbrace{\frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} (T_i - T_o)}_{\text{Transmission losses}} \quad (2.5)$$

Transmission heat losses of the glazing are calculated separately via the U value. Thus, for the definition of the secondary heat emission degree q_i , the ambient temperature can be set equal to the outside temperature. The result for q_i is:

$$q_i = \frac{\dot{Q}_i / A_w}{G} = \alpha \frac{h_i}{h_i + h_o} \quad \text{for } T_i = T_o \quad (2.6)$$

For a double glazing, the characteristic values are calculated similarly, although an additional heat balance for the outside pane must now be created. The absorption coefficient for the outside pane is defined as α_1 and for the internal pane as α_2 . The secondary heat emission degree also depends on the layer thicknesses s_1, s_2 and heat conductivities λ_1, λ_2 of the two panes and on the thermal resistance R_{air} of the standing air layer between the panes:

$$q_i = \frac{\left(\frac{\alpha_1 + \alpha_2}{h_o} + \frac{\alpha_2}{\frac{s_1 + R_{air} + s_2}{\lambda_1} + \frac{s_2}{\lambda_2}} \right)}{\frac{1}{h_i} + \frac{1}{h_o} + \frac{s_1 + R_{air} + s_2}{\lambda_1} + \frac{s_2}{\lambda_2}} \quad (2.7)$$

The solar radiation let through (transmitted by) the glazing into the room $\dot{Q}_{trans}$ results directly from the product of the g value and the solar irradiance:

$$\dot{Q}_{trans} = gG \quad (2.8)$$

Heat transfer coefficients of windows

The calorific loss through the window is deducted from the transmitted power, which is characterised by the heat transfer coefficient of the glazing U_g or of the entire window including the frame U_w . Double glazing with low emissivity-coated panes filled with heavy noble gases achieve a minimum U_g value of $1.0 \text{ W m}^{-2} \text{ K}^{-1}$, and triple glazing at best reaches a U_g value of $0.4 \text{ W m}^{-2} \text{ K}^{-1}$. Even at a glazing with a U_g value of $1.3 \text{ W m}^{-2} \text{ K}^{-1}$, a wooden or plastic frame increases the window's U_w value slightly. For passive house concepts, specially insulated expanded polystyrene frameworks must be used to avoid worsening the low glazing U values of triple glazing by the frame proportion.

The passive solar gain $\dot{Q}_u$ usable in a room results from the balance of losses and gains. The losses are calculated from the U_w value of the window of surface A_w and from the temperature difference between the room air T_i and the outside air T_o :

$$\frac{\dot{Q}_u}{A_w} = U_w (T_i - T_o) - gG \quad (2.9)$$

From the available energy balance, an effective U value U_{eff} can be defined, which is often used for monthly or annual balance calculations with mean temperature differences and irradiances.

$$U_{eff} = \frac{\dot{Q}_u}{A_w (T_i - T_o)} = U_w - g \frac{G}{T_i - T_o} \quad (2.10)$$

To calculate the overall thermal effect of windows, solar gains and losses or gains by heat transmission need to be balanced. Over a heating season, about $400 \text{ kWh m}^{-2} \text{ a}^{-1}$ of solar irradiance is available on a south-facing facade in Germany. The so-called heating degree days are obtained by multiplying the mean temperature difference between the inside and outside by the number of days in the heating season. There are about 3500 kelvin days per year on average in Germany. Thus, the maximum usable energy per square metre of glazing surface for a two-pane low-e coated glazing with a U_w value of $1 \text{ W m}^{-2} \text{ K}^{-1}$ and $g = 0.65$ is:

$$\begin{aligned} \frac{Q_u}{A_w} &= 0.65 \times 400 \times 10^3 \frac{\text{Wh}}{\text{m}^2 \text{a}} - 1.0 \frac{\text{W}}{\text{m}^2 \text{K}} \times 3500 \frac{\text{K days}}{\text{a}} \times 24 \frac{\text{h}}{\text{day}} \\ &= 260 \times 10^3 \frac{\text{Wh}}{\text{m}^2} - 84 \times 10^3 \frac{\text{Wh}}{\text{m}^2} = 176 \frac{\text{kWh}}{\text{m}^2} \end{aligned} \quad (2.11)$$

The amount of heat effectively usable in the room depends greatly on the storage capability of the structural elements inside. High passive solar heat gains can easily lead to overheating of the interior and thus do not contribute to covering the heating requirement.

According to the EN 832 monthly energy balance procedure for calculating the heating demand, the useful solar irradiance transmitted by windows is a function of the relation of the monthly gains to the transmission and ventilation heat losses. For low-energy buildings with heat-storing heavy building construction and an annual heating requirement between about 30 and $70 \text{ kWh m}^{-2} \text{ a}^{-1}$, the result is a flat minimum of the heating requirement for a window area proportion to the south-facing facade of approximately 25%. In administrative buildings with mostly higher internal loads, the window area proportion should be even lower to avoid overheating in the summer. The minimum heating requirement is obtained for 0 - 20% of the window area proportion for a lightweight building with a small storage capacity.

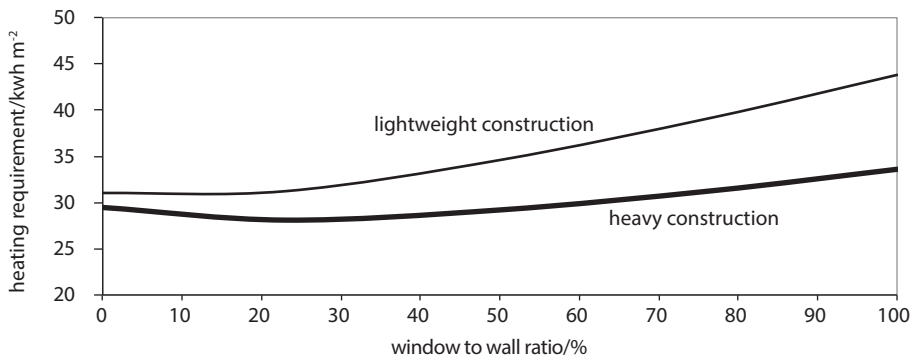


Figure 2.6 Influence of the window area proportion on the heating requirement for a low-energy building in Germany.

Shading factors

The product of total energy transmittance g and shading factor F determines the solar gains to a building.

$$g = g_w F \quad (2.12)$$

where g_w is the total energy transmission factor of the window and pane, and F is the area proportion of windows.

In the German standard DIN 4108, the following shading factors are used:

Shading equipment		Shading factor F_c
1	None	1.0
2	Interior and between glazing panes	
2.1	White or reflective surfaces with low transparency	0.7
2.2	Light colours and low transparency	0.8
2.3	Dark colours and high transparency	0.9
3	Exterior	
3.1	Blinds, rotating louvres, back-ventilated	0.25
3.2	Blinds, shutters	0.3
3.3	Awning, porch	0.3
3.4	Marquee, above and side-ventilated	0.4
3.5	Marquee, general	0.5

As too high solar gains lead to overheating of buildings, the product of energy transmittance and shading factor needs to be limited. The limiting values should be lower, if the building has a lightweight construction and cannot be naturally ventilated in the summer.

Recommendations according to DIN 4108-2 for maximum values of ($g_w F$) depending on natural ventilation and interior construction.

Ventilation	Light construction ($g_w F$)	Heavy construction ($g_w F$)
Without	0.12	0.14
With	0.17	0.25

New glazing systems

For flexible control of total energy transmission, glazing systems are being developed that modify their transparency as a function of temperature (thermo-tropic) or electrically controlled (electro-chromic).

Thermotropic layers are polymer mixtures or hydrogels which are inserted in a homogeneous mixture between two window panes, and reduce the light permeability by up to 75% with rising temperature.

Electrochrome thin films, for example, made from tungsten oxide, can be evaporated on window panes with conductive oxide coatings. The two thin-film electrodes are connected by a polymer ion conductor. When an external electrical field is applied and causes cations (e.g. Li^+) from the counter-electrode to accumulate onto the tungsten oxide, the transmittance falls dependent on wavelength to 10 - 20%. A tight seal at the edges is very important for long-term stability. The first commercially available glasses with a maximum surface of $0.9 \text{ m} \times 2.0 \text{ m}$ achieve a reduction of the total energy transmission factor from 44% in the bright status to 15% in the dark status (U value = $1.6 \text{ W m}^{-2} \text{ K}^{-1}$). In systems with a lower U value of $1.1 \text{ W m}^{-2} \text{ K}^{-1}$, the g value falls from 36% to 12%.

2.2 Transparent thermal insulation (TTI)

Since the early 1980s, transparent thermally insulated facade systems are available on the market. Compared to conventional thermal insulation in buildings, transparently insulated external walls can well utilise incoming solar radiation. The energy-saving potential for the application of such solar systems is high. For example, if a fifth of all existing building facades in Germany were equipped with transparent insulating systems, approximately 15% of the heat needed for room heating could be supplied. The technology is particularly interesting for the renovation of old buildings with heavy, very heat-conducting walls. Transparent insulation can be attached directly onto an external wall and a transparent plaster is used to protect the material from weather. The potentially high costs of glass composite structures can be significantly abated by foregoing complex frame constructions. Transparent thermal insulation can also be applied to provide daylighting of building interiors. The elements of the thermal insulation can scatter and direct light and provide for the even illumination of a room. Combined with very good heat-insulating properties, transparent thermal insulation has the potential for large-scale application on external building walls.

Operational principle

If short-wave solar radiation hits an external wall, the radiation is absorbed and converted into heat. Although the external surface warms up, most of the heat produced is transferred to the outside air. Only a small portion of the heat produced reaches the building interior. If a transparent

insulating layer is attached in front of the wall (in the simplest case, as a window pane), heat emission to the outside is made more difficult.

In addition to the transmission coefficient for solar radiation and the heat resistance of the transparent thermal insulation, the main parameters influencing the extent of the useful heat gain are the absorption coefficient, heat conductivity and storage capability of the adjoining wall. The wall itself produces a time delay in the heat flow such that the maximum values of the solar-induced heat flow reach the inside when the direct solar gains through the windows have already decreased and outside temperatures are falling. In addition, the thermal characteristics of the entire building play a role; however, above all, it is the heat-storing capability of the interior structural elements that aids in the avoidance of overheating.

For example, an external wall even with 10 cm external insulation under German climatic conditions will still exhibit annual heat losses of over 30 kWh per square metre. However, with the application of transparent thermal insulation, this same wall acts as a solar collector during the heating season and produces around 50 - 100 kWh m⁻² of useful heat for the building.

The heat transfer coefficient of an external wall insulated with transparent insulation results from the total of the thermal resistances of the existing external wall and the transparent insulation, with layer thickness s_{TTI} and heat conductivity λ_{TTI} of the transparent insulating material, layer thickness s_{wall} and heat conductivity λ_{wall} of the external wall and also the heat transfer coefficients inside h_i and outside h_o .

$$U_{wall+TTI} = \left(\frac{1}{h_o} + \frac{s_{TTI}}{\lambda_{TTI}} + \frac{s_{wall}}{\lambda_{wall}} + \frac{1}{h_i} \right)^{-1} \quad (2.13)$$

The wall absorber is characterised by the absorption coefficient α and the transparent insulation by the diffuse total energy transmission factor g_d . The U values of 10 cm TTI are typically about 0.8 W m⁻² K⁻¹, which are lower than the best double-glazed heat-protection glass, but still twice as high as the heat transfer coefficients of 10 cm of conventional insulating material with U values of 0.4 W m⁻² K⁻¹.

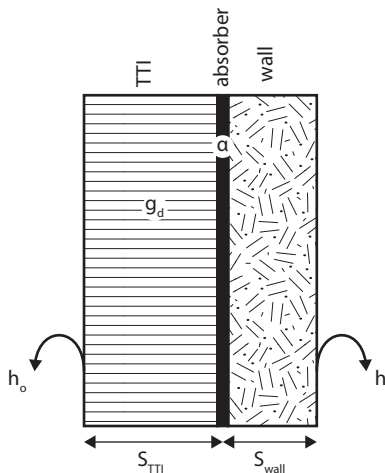


Figure 2.7 System structure of a transparently insulated wall.

Just as for the energy balance of windows, an effective U value can also be defined for transparent thermal insulation as the difference between losses and solar gains with efficiency η_0 .

$$U_{eff} = U_{wall+TTI} - \eta_0 \frac{G}{T_i - T_o} \quad (2.14)$$

Solar efficiencies η_0 up to 50% with a simultaneous low heat transfer coefficient lead to low or even negative effective U values. Given favourable wall orientation, negative U values can be achieved, which means the wall produces heat gains for the building. Weekly averaged measurements of a 10 cm transparently insulated building in the City of Freiburg, Germany exhibited effective U values between 0 and $-3.5 \text{ W m}^{-2} \text{ K}^{-1}$.

The solar efficiency η_0 corresponds to the total energy transmission of a glazing and consists of the g_d value of the transparent insulating material, the absorption factor of the absorber α and the proportion of the heat flow inward to the total heat flow. The heat flow from the absorber inward is calculated from the temperature node of the absorber at temperature T_a to the room air temperature T_i via the heat transfer coefficients of the wall.

$$U_{wall} = \left(\frac{s_{wall}}{\lambda_{wall}} + \frac{1}{h_i} \right)^{-1}$$

$$\frac{\dot{Q}_i}{A} = U_{wall} (T_a - T_i) \quad (2.15)$$

The heat flow from the absorber outward is calculated via the heat transfer coefficient U_{TTI} of the TTI material.

$$\frac{\dot{Q}_o}{A} = U_{TTI} (T_a - T_o) \quad (2.16)$$

The solar efficiency is then proportional to the ratio of the interior heat flux to the total heat flux as the sum of interior and exterior flux.

$$\eta_0 = \alpha g_d \frac{U_{wall} (T_a - T_i)}{U_{wall} (T_a - T_i) + U_{TTI} (T_a - T_o)} \quad (2.17)$$

Assuming identical temperatures inside and outside, a constant solar efficiency can be defined, which is very suitable for material comparisons and estimates of the energy yield.

$$\eta_0 = \alpha g_d \frac{U_{wall}}{U_{wall} + U_{TTI}} \quad (2.18)$$

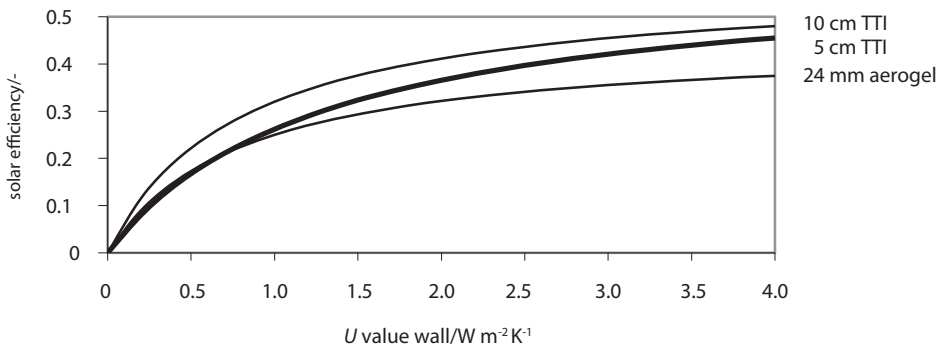


Figure 2.8 Solar efficiency of transparent insulation as a function of the heat transfer coefficient of the opaque wall.

With 5 cm transparent capillary tubes, the U_{TTI} value is $1.3 W m^{-2} K^{-1}$ at a g value of 0.67; with 10 cm it is $0.8 W m^{-2} K^{-1}$ and $g = 0.64$. The aerogel material shows a U_{TTI} value of $0.8 W m^{-2} K^{-1}$ with a very small layer thickness of 2.4 cm at a g value of 0.5.

For optimal use of solar heat, one must ensure that transparently insulated rooms do not overheat. Conventional window areas usually bring sufficiently high solar gains into the room during the day; additional gains from the TTI wall must be stored and then used in the evening hours. The temporal shift between heat production at the absorber and the maximum heat flow into the room rises with external wall thickness and also depends on the density, thermal capacity and heat conductivity of the wall. Phase shifts of 6 - 8 hours are achieved with a 24 cm brick or lime-sandstone wall while a little more than 5 hours are possible with concrete walls. With sufficiently thick wall constructions, the result is typically 100% efficiencies of the heat produced by the TTI wall within the core months of the heating season; in the transition months the efficiency is around 30%.

The high absorber temperatures of the external wall, which can reach peak values between 70 and 80°C, are effectively dampened at wall thicknesses over 20 cm and are rarely higher than 30°C at the interior surface, even in the summer. The absorber temperatures are lower with heavy, heat-conducting components than with light. The heat that develops can penetrate quickly into the heavy external wall and be led into the interior. Any thermal tensions are thus correspondingly low.

The thermal deformations of an external wall on an experimental house in Stuttgart were measured on a long-term basis. Compression stresses and slight swelling of the wall due to the high-temperature difference did not pose a problem. Fine cracks of about 1 - 2 mm in the plaster resulted from accelerated drying of the new building's brickwork dampness around the absorber; however, these cracks did not influence the load-carrying capacity of the wall. It is worth planning for defined joints at the edges of the TTI surfaces for new buildings. The use of shading systems such as blinds or shutters prevents heating of the external wall during the transition period and in the summer months, although the use of these systems is complex in terms of construction and maintenance. In order to avoid exposure to the sun during the transition or summer period, constructional shadings such as balconies or roof projections must be planned very carefully. Foregoing shading mechanisms is possible if the transmittance of the transparent insulation is

strongly angle-dependent such that with a high summer sun position having angles of incidence over 60° at the south-facing facade, less than 20% of the irradiance reaches the absorber wall. The transmittance of a transparent insulation system falls, for example, from approximately 50% with perpendicular incidence to 15% with a sun elevation angle of 60° .

Simulations for highly insulated low-energy buildings have shown that even when the south-facing facade is largely covered with a TTI heat insulation system, the excess heat in the summer can be expelled by night ventilation (Meyer, 1995). The number of hours with room temperatures over 26°C was below 300 hours per year under German conditions and only 50 hours higher than a conventionally insulated building.

On the other hand, at angles of incidence of 60° , transparent insulation with a glass covering can still have a very high total energy transmission factor of 35%. In this case, sun protection is unavoidable when the facade is largely covered. If nocturnal ventilation is not possible, e.g. in office buildings, shading of the TTI surfaces in the summer should likewise be provided.

The orientation of the transparently insulated facade is crucial both for energy gain in the winter and for protection against overheating in the summer. Facade orientations between south-east and south-west are suitable. Twice as much energy, some 400 kWh m^{-2} , falls on a south-facing facade in the heating season than on an east- or west-facing one. In addition, the low winter sun position facilitates good light transmission by the TTI material. In the summer, on the other hand, only the south-facing facade offers a certain natural sun protection with low transmittance.

Materials used and construction

TTI capillary or honeycomb structures are assembled from thin-walled plastic tubes and welded by a hot wire section or manufactured into strips of any width by extruder nozzles with an almost square cell cross section. The typical cell diameter is 3 mm.

Two polymer types are in use today: polymethyl methacrylates (PMMA) and polycarbonates (PCs). PMMA is characterised by high transmittance and by good UV stability. Due to the brittleness of the material and its poor fire retardance (class B3), PMMA is bound between window panes, which requires a complex mullion-transom construction costing between 400 and 750 € m^{-2} . The cost of the 10 - 12 cm transparent insulation is typically only around 50 € m^{-2} . It is the glazing and attachment, at approximately 250 € m^{-2} , plus shading items such as blinds at around 150 € m^{-2} , which drive up costs.

PCs are mechanically more stable and can be processed without the use of a glass covering; however, they are not very UV-resistant. PC fire retardance is better (class B1) than PMMA, and the material is temperature-resistant to about 125°C . PC materials can be used in heat-insulating compound systems. The covering plaster is an acryl adhesive mixed with 2.5 - 3.0 mm diameter glass balls, which is applied in the factory directly onto the capillary material. Additional UV absorbers can be likewise brought into the cover plaster. Such heat-insulating compound systems can be manufactured with substantially reduced costs of around 150 € m^{-2} , since there are no complex glazing and shading systems. The weight of capillary materials is around 30 kg m^{-3} .

Capillaries made of glass are manufactured like the polymer structures, but are complicated to produce due to high processing temperatures and the associated engineering problems. Glass capillaries are much more temperature- and UV-resistant, but they are also mechanically not very stable. The recycling ability of glass, which is also possible with PMMA, is advantageous. On the other hand, PCs are recyclable only with high energy expenditure and with quality losses.

For glazing systems with smaller thicknesses of 2 - 3 cm, aerogels are suitable. Aerogels are highly porous, open-pored solids made of silica gel that consist of more than 90% air and 10% silicate and exhibit a very low heat conductivity ($\lambda = 0.02 \text{ W m}^{-1} \text{ K}^{-1}$). Aerogels can easily be poured into the cavity of double glazing, are not inflammable, easy to dispose of and to recycle. Significant disadvantages include a light transmission only about half that of capillary materials and a sensitivity to water. Aerogel material absorbs water that penetrates into the edge network of a double glazing, and the sensitive structure is broken down by capillary forces.

Construction principles of TTI systems

Transparent heat-insulating systems are mainly used in two types of construction:

1. Mullion-transom or element construction with framed TTI panel elements. To avoid dirtying of the TTI materials, the external covers usually consists of highly transparent, iron-poor single glass panes. Element constructions are characterised by a higher degree of prefabrication, which has a potential for cost reduction. From the outside, it is often impossible to distinguish between element and a mullion-transom construction installation.
Shading mechanisms such as blinds or shutters are preferably inserted between the outside window pane and the TTI material. Lamella-type systems can also be used in front of the facade and are highly reliable when movements between open and closed positions are minimal or when maintained in a lowered position.
2. Heat-insulating compound system construction with frameless direct installation. The transparently plastered capillary structures are supplied with a fabric for attaching the plaster to the conventional insulation and fastened to the external wall with a black adhesive that serves as an absorber.

2.3 Heat storage by interior building elements

The heat storage capacity of the interior components is decisive for the degree of useful energy produced by both passive solar use via windows and transparent heat-insulating systems. Heating demand can only be reduced if solar gains do not lead to overheating of the interior. The heat storage capacity of interior components can be roughly estimated from the storage mass, the thermal capacity and the possible rise in temperature of the storage mass. Thus, for example, a solid concrete wall with a thickness d of 30 cm, a heat capacity c of $1 \text{ kJ kg}^{-1} \text{ K}^{-1}$ and a gross density ρ of 2100 kg m^{-3} can store, with a rise in temperature of 5°C , 0.875 kWh of heat per square metre of surface.

$$\frac{Q}{A} = \rho d c \Delta T = 2100 \frac{\text{kg}}{\text{m}^3} \times 0.3 \text{ m} \times 1.0 \frac{\text{kJ}}{\text{kgK}} \times 5 \text{ K} = 3150 \frac{\text{kJ}}{\text{m}^2} = 0.875 \frac{\text{kWh}}{\text{m}^2} \quad (2.19)$$

This approach assumes that the component is completely warmed or cooled to the temperature levels forming the basis of the calculation. It also assumes very high heat transfer coefficients and high heat conductivities, which in practice is not the case. To what extent the storage capacity can be used depends, apart from the materials' values, primarily on the duration of a rise in temperature.

If the amount of heat Q per surface A that flows in a given period into the wall is determined by dynamic calculation methods or by measurement, then an effective thickness d_{eff} wherein full storage capability is utilised can be calculated for the wall.

$$d_{eff} = \frac{Q}{A c \rho \Delta T} \quad (2.20)$$

During a 3-hour rise in temperature, a concrete wall (with $A = 1 \text{ m}^2$ surface) can, largely irrespective of its thickness, take up approximately 33 Wh for each kelvin of temperature rise (with heat take-up on both sides). This corresponds to an effective thickness of approximately 5 cm. With a 6-hour rise in temperature this value is approximately 9 cm.

If the heating up of a room is to be calculated, then rough estimates of the amount of heat Q flowing into the component can be calculated by using the heat transfer coefficient h_i and the temperature difference between the component surface $T_{s,1}$ (at the beginning of a time step Δt) and the room air T_r .

$$Q = h_i A \Delta t (T_r - T_{s,1}) \quad (2.21)$$

After the time step, the new temperature of the component $T_{s,2}$ results from the stored amount of heat $Q_s = Q$:

$$T_{s,2} = T_{s,1} + \frac{Q_s}{A c \rho d_{eff}} \quad (2.22)$$

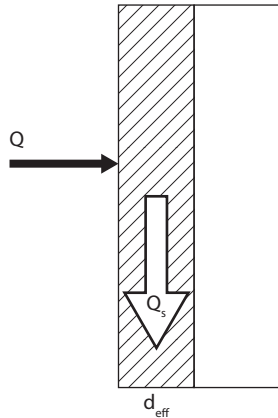


Figure 2.9 Amount of heat flowing into a component and the effective storage mass thickness d_{eff} .

This process is repeated with each time step. For a more exact calculation of the temporally variable temperature distribution, an energy balance for a volume element of the storage mass must be created, which leads to the classical thermal heat conduction equation. For simplicity, only one-dimensional temperature distributions will be derived, i.e. from the air via the surface into the component depth.

For passive solar energy use, the following boundary conditions play a role:

- the storage capacity of components during brief variations in temperature in the room caused by solar irradiance or air temperature modification,
- the potential for night cooling by utilisation of the periodic modification of the air temperature between day and night,
- the temperature amplitude and phase shift on the inner side of a transparently insulated wall.

In all cases, heat is only absorbed or dissipated via the surface of the component. There are no heat sources in the component interior, so very simple energy balances for each volume element results. From an entering heat flow $\dot{Q}_{in}$, part of the energy leads to the rise in temperature in the volume element (heat storage $\dot{Q}_{st}$) and the remainder is passed on by thermal conduction into the next element $\dot{Q}_{out}$.

$$\dot{Q}_{in} = \dot{Q}_{st} + \dot{Q}_{out} \quad (2.23)$$

Based on Fourier's law of thermal conduction, the heat flow $\dot{Q}_{in}$ entering through the surface A is proportional to the temperature gradient at the point x_0 .

$$\dot{Q}_{in} = -\lambda A \left. \frac{dT}{dx} \right|_{x_0} \quad (2.24)$$

The exiting heat flow $\dot{Q}_{out}$ at the point $x_0 + dx$ is, at constant heat conductivity λ , only different from $\dot{Q}_{in}$ if the temperature gradient has changed in the volume element, e.g. has become flatter due to heat storage in the element.

$$\dot{Q}_{out} = -\lambda A \left. \frac{dT}{dx} \right|_{x_0+dx} \quad (2.25)$$

A Taylor series expansion of the temperature gradient at the point $x_0 + dx$ leads to the following simplification, if all higher-order members are ignored:

$$\begin{aligned} \dot{Q}_{out} &= -\lambda A \left(\left. \frac{dT}{dx} \right|_{x_0} + \frac{d}{dx} \left(\left. \frac{dT}{dx} \right|_{x_0} \right) dx + \dots \right) \\ &\approx -\lambda A \left(\left. \frac{dT}{dx} \right|_{x_0} + \left. \frac{d^2T}{dx^2} \right|_{x_0} dx \right) \end{aligned} \quad (2.26)$$

The amount of heat stored in the volume element $dV = A dx$ is given by:

$$\dot{Q}_{st} = \rho dV c \frac{dT}{dt} \quad (2.27)$$

Thus, the energy balance equation leads to:

$$\begin{aligned}
 -\lambda A \frac{dT}{dx} \Big|_{x_0} &= \rho d V c \frac{dT}{dt} - \lambda A \frac{dT}{dx} \Big|_{x_0} - \lambda A \frac{d^2 T}{dx^2} dx \\
 \frac{\lambda}{\rho c} \frac{d^2 T}{dx^2} &= \frac{dT}{dt}
 \end{aligned} \tag{2.28}$$

where

$$a = \frac{\lambda}{\rho c} \left[\frac{m^2}{s} \right]$$

is termed the thermal diffusivity, which lies between $10^{-7} \text{ m}^2 \text{ s}^{-1}$ for wood and $10^{-4} \text{ m}^2 \text{ s}^{-1}$ for metals.

Component temperatures for sudden temperature increases

The differential equation can be solved by a product approach for periodic boundary conditions or with temperature-equalising processes, wherein one function is dependent only on time and the other only on place.

If a temperature jump on one side is given as a boundary condition, an approach with the Gauss error integral (or error function $\text{erf}(z)$) leads to a more a general solution than the product approach, as the initial temperature distribution $F(x)$ at the point in time $t = 0$ can assume any values. However, the temperature and time functions will no longer be separate.

$$T(x, t) = \frac{1}{\sqrt{\pi}} \frac{1}{\sqrt{4at}} \int_{-\infty}^{+\infty} F(\xi) \exp\left(-\frac{(\xi-x)^2}{4at}\right) d\xi \tag{2.29}$$

From this general approach, some solutions for simple boundary conditions can be represented analytically. For a concrete floor slab, the change in temperature at a certain depth (as an indication of the utilisation of the heat storage), the heat flow occurring through the surface, and the stored heat as a function of the duration of the temperature jump at the surface can all be examined. Far more relevant in practice is the case of a temperature jump in the room air, which is transferred via convection to the component surface. The mathematical derivation of this situation is more complex and is therefore discussed later along with solutions.

The temperature distribution for a component with a constant initial temperature T_c and a temperature jump at the component surface to zero for $t > 0$ is directly calculable from the Gauss error integral. With the substitution,

$$\eta = \frac{\xi - x}{\sqrt{4at}} \quad d\xi = d\eta \sqrt{4at} \quad F(\xi) = T_c$$

the result from Equation (2.29) is:

$$\frac{T(x, t)}{T_c} = \frac{2}{\sqrt{\pi}} \int_0^{\frac{x}{\sqrt{4at}}} \exp(-\eta^2) d\eta = \text{erf}\left(\frac{x}{\sqrt{4at}}\right) = \text{erf}(z) \tag{2.30}$$

The factor 2 results from splitting the integral from Equation (2.29) into two parts and the symmetry of the function $\exp(-\eta^2)$. Strictly, the solution not only applies to a semi-infinitely expanded body, but it can also be used for shorter time intervals (a few hours) for a finitely expanded wall.

The error function $\text{erf}(z)$ can be approximated with an error $< 2.5 \times 10^{-5}$ with an exponentially dampened polynomial function of third order (Wong, 1997).

With the auxiliary variable

$$p = \frac{1}{1 + 0.47047 \times z}$$

the approximation equation reads,

$$\text{erf}(z) = 1 - (0.3480242 \times p - 0.0958798 \times p^2 + 0.7478556 \times p^3) \times \exp(-z^2) \quad (2.31)$$

From the values of the error function, it can be directly determined which temperature $T(x, t)$ prevails at any point x and time t . If, on the other hand, the penetration of a given temperature ratio $T(x, t)$ to the initial temperature T_c at time t or depth x is to be determined, the equation must be solved through iterations for z , and x and t must be calculated from $z = x / \sqrt{4at}$. The graphical representation of the error function for direct reading of the z value from the function value $\text{erf}(z) = T(x, t)/T_c$ is represented below:

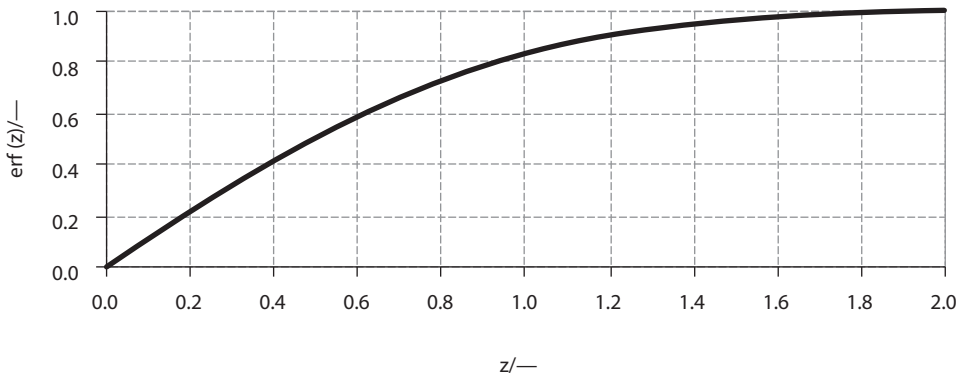


Figure 2.10 Error function $\text{erf}(z) = T(x,t)/T_c$ as a function of $z = x / \sqrt{4at}$.

The temperature jump must always take place from the initial temperature T_{c0} to the jump temperature $T_j = 0^\circ\text{C}$. If this is not the case ($T_j \neq 0$), a standardised initial temperature T_c is calculated from the temperature difference of the initial temperature T_{c0} and the jump temperature T_j : $T_c = T_{c0} - T_j$. The following calculations are always based on a standardised initial temperature T_c and a surface temperature of zero.

Example 2.1

On a 20°C concrete (or wooden) floor, a surface temperature jump to $T_j = +30^\circ\text{C}$ occurs. Over what period of time has the temperature at a depth of 20 cm risen by 5°C ?

Characteristic values of the components:

	Heat conductivity λ /W m ⁻¹ K ⁻¹	Density ρ /kg m ⁻³	Heat capacity c /kJ kg ⁻¹ K ⁻¹	Thermal diffusivity a /m ² s ⁻¹
Concrete	1.28	2200	0.879	0.66×10^{-6}
Wood	0.2	700	2.4	0.12×10^{-6}

In order to adapt the boundary condition to a temperature jump of the surface temperature to 0°C, the initial floor temperature $T_{c0} = 20^\circ\text{C}$ is replaced by the standardised initial temperature $T_c = T_{c0} - T_j = 20^\circ\text{C} - 30^\circ\text{C} = -10^\circ\text{C}$. A temperature rise $T(x, t)$ of 5°C at a component depth of 20 cm thus corresponds to a temperature condition

$$\frac{T(x=0.2\text{m}, t)}{T_c} = \frac{-5^\circ\text{C}}{-10^\circ\text{C}} = \text{erf}(z)$$

Calculated through iterations, the z value (which can also be read off from the error function figure shown above) is

$$z = 0.477 = \frac{x}{\sqrt{4at}}$$

From

$$t = \frac{x^2}{4az^2}$$

the result for the concrete ceiling is a temperature-rise time of 18.5 hours and 111 hours for the wooden ceiling.

If the temperature ratio $T(x,t)/T_c = \text{erf}(z)$ is represented as a function of time, it can be seen how quickly a surface temperature jump propagates into the component. For $x > 0$, the temperature at $t = 0$ is at first the constant initial temperature T_c and the temperature ratio is unity. With increasing time, the component temperature approaches the surface temperature of zero, i.e. the temperature ratio tends towards zero.

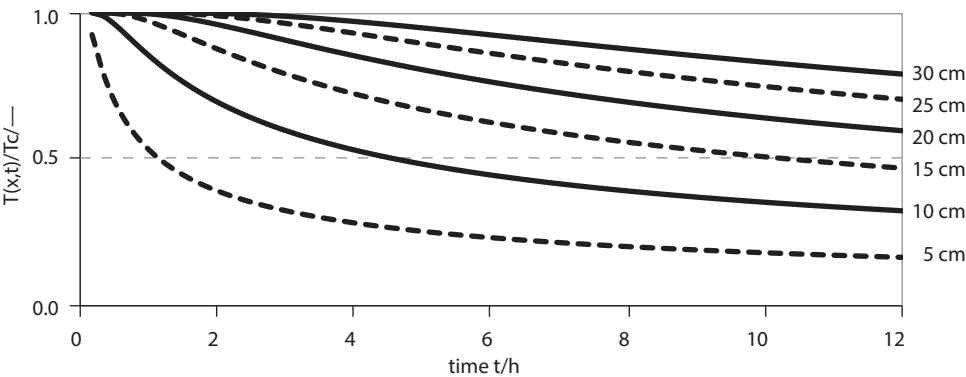


Figure 2.11 Temperature decrease (error function $\text{erf}(z) = T(x,t)/T_c$) as a function of time for concrete floors with thermal diffusivity $a = 0.66 \times 10^{-6} \text{ m}^2 \text{ s}^{-1}$.

After 12 hours, half of the surface temperature increase ($T(x, t)/T_c = 0.5$) is at a component depth of 16 cm for the concrete floor, which illustrates the limitation of the effectively usable storage capacity. Only with times $\gg 12$ hours, does the component temperature approach the surface value of zero ($T(x, t)/T_c \rightarrow 0$).

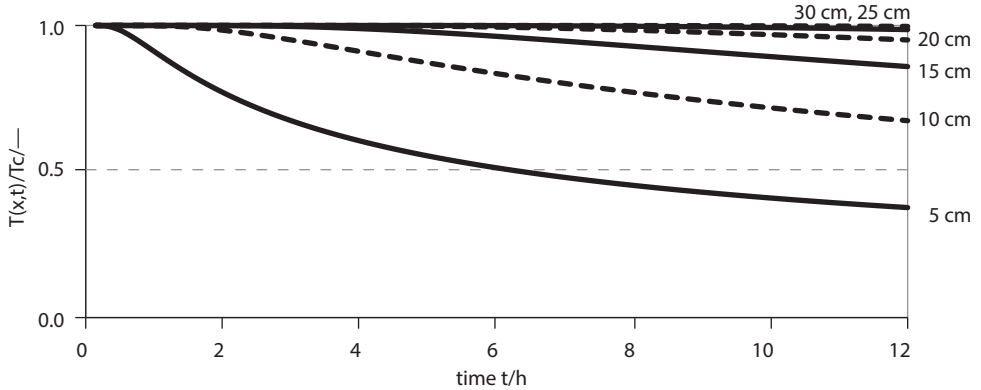


Figure 2.12 Temperature field in the component wood as a function of time.

For wood with a lower thermal diffusivity a , a surface temperature jump propagates even more slowly. After 12 hours, half of the surface temperature increase has only reached a depth of 7 cm. For component thicknesses over 20 cm, the temperature has not changed, even after 12 hours.

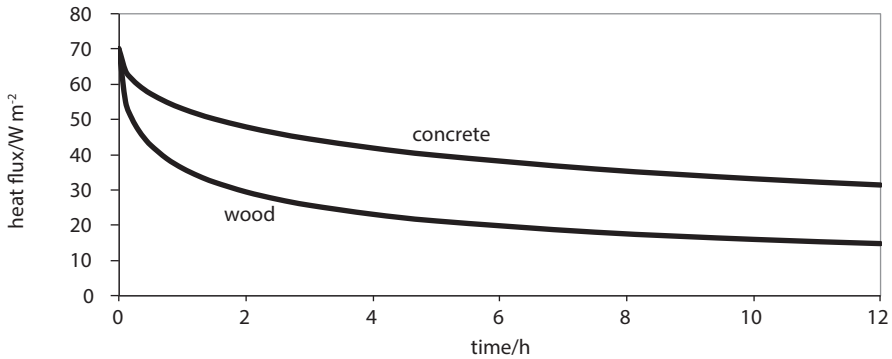


Figure 2.13 Heat flux as a function of time for a concrete and a wooden floor with a 10 K temperature jump.

The heat flow dQ/dt entering or leaving the surface A is proportional, based on Fourier's law, to the temperature gradient at the surface:

$$\begin{aligned} \frac{dQ}{dt} &= -\lambda A \left. \frac{dT}{dx} \right|_{x=0} = -\lambda A T_c \left. \frac{d}{dx} \operatorname{erf} \left(\frac{x}{\sqrt{4at}} \right) \right|_{x=0} \\ &= -\lambda A T_c \frac{1}{\sqrt{4at}} \frac{2}{\sqrt{\pi}} \exp \left(-\frac{x^2}{4at} \right) \Big|_{x=0} = -A \underbrace{\lambda \rho c}_{=1 \text{ für } x=0} \sqrt{\frac{1}{\pi}} T_c \frac{1}{\sqrt{t}} \end{aligned} \quad (2.32)$$

The heat flow is proportional to the so-called heat penetration coefficient $b = \sqrt{\lambda \rho c}$ and falls with $\sqrt{t}^{-1}$.

The integration of the heat flow over time results in the total amount of heat penetrating into the component when there is a surface temperature jump.

$$\frac{Q}{A} = \int \frac{dQ}{A} = \int_0^{t_0} \left(-\sqrt{\frac{\lambda \rho c}{\pi}} T_c \frac{1}{\sqrt{t}} \right) dt = -\frac{2}{\sqrt{\pi}} \sqrt{\lambda \rho c} \sqrt{t_0} T_c \quad (2.33)$$

Since the surface temperature for $t > 0$ has been set by definition to 0°C , for T_c the standardised initial temperature must be used again.

Example 2.2

Calculation of the amount of heat penetrating a component with a temperature jump at the surface of 10K for $t_0 = 12\text{h}$ for the two floors from Example 2.1.

The heat penetration coefficient $b = \sqrt{\lambda \rho c}$ for concrete is $1.573 \text{ kJ m}^{-2} \text{ K}^{-1} \text{ s}^{-1}$ and $0.579 \text{ kJ m}^{-2} \text{ K}^{-1} \text{ s}^{-1}$ for wood. Thus, within 12 hours an amount of heat of 3689 kJ m^{-2} ($= 1.02 \text{ kWh m}^{-2}$) is brought into the concrete ceiling and 0.38 kWh m^{-2} into the wooden ceiling.

The amount of energy stored within a component is, like the heat flow, directly proportional to the heat penetration coefficient b and to the temperature jump ΔT at the surface, but it also rises with the root of time.

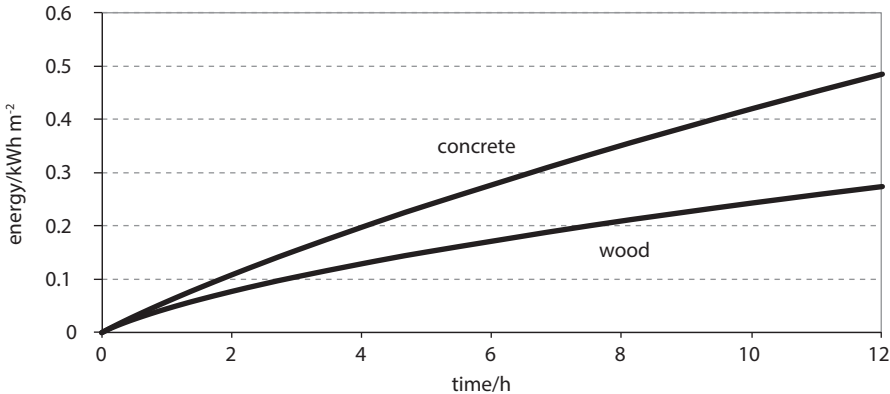


Figure 2.14 Amount of energy led into the component with a rise in temperature at the surface of 10 K.

Usually, however, it is not the surface temperature but the air temperature, which is known. Between the air and the surface temperature change there is a phase shift and a dampening of the amplitude. At a given air temperature T_o and a given heat transfer coefficient h between the air and the surface, the temperature field $T(x, t)$ can be calculated as follows (Gröber et al., 1988):

$$\frac{T(t, x)}{T_c} = \operatorname{erf}\left(\frac{x}{\sqrt{4at}}\right) + \exp\left(at\left(\frac{h}{\lambda}\right)^2 + \frac{h}{\lambda}x\right) \left(1 - \operatorname{erf}\left(\frac{x}{\sqrt{4at}} + \frac{h}{\lambda}\sqrt{at}\right)\right) \quad (2.34)$$

At the surface $x = 0$, therefore, the following temperature appears:

$$T(t, x=0) = T_c \exp\left(at\left(\frac{h}{\lambda}\right)^2\right) \left(1 - \operatorname{erf}\left(\frac{h}{\lambda}\sqrt{at}\right)\right) \quad (2.35)$$

Example 2.3

Calculation of the temperature of a concrete ceiling at the surface and at a depth of 5 cm for $t = 1 \text{ h}$ and $t = 10 \text{ h}$ with parameters from Example 2.1, if the air temperature, rather than the surface temperature, jumps to 30°C . The heat transfer coefficient h is $8 \text{ W m}^{-2} \text{ K}^{-1}$.

First, the surface temperature is calculated for $x = 0$ using Equation (2.34). For $t = 1 \text{ h}$, the temperature at the surface is: $T(t=1\text{h}, x=0)/T_c = 0.73$ with

$$\exp\left(at\left(\frac{h}{\lambda}\right)^2\right) = \exp\left(\underbrace{0.66 \times 10^{-6} \frac{\text{m}^2}{\text{s}} \times 1\text{h} \times 3600 \frac{\text{s}}{\text{h}} \times \left(\frac{8 \frac{\text{W}}{\text{m}^2 \text{K}}}{1.28 \frac{\text{W}}{\text{mK}}}\right)^2}_{0.0928}\right) = 1.097$$

and

$$\operatorname{erf}\left(\frac{h}{\lambda}\sqrt{at}\right) = \operatorname{erf}(0.304) = 0.33$$

With an air temperature jump of 10 K, the ratio of the surface temperature to the initial body temperature T_c after 1 hour is still at 73%, i.e. with the selected boundary condition for heating, the surface temperature has increased by $(1-0.73) \times 10 \text{ K} = 2.7 \text{ K}$. After $t = 10 \text{ h}$ the temperature ratio is 43.8%. The surface temperature has then increased by 5.6 K.

At a depth of 5 cm, the temperature ratio after $t = 1 \text{ h}$ is 0.903, i.e. the temperature has only increased by 1 K. After 10 hours, the temperature at a depth of 5 cm has increased by 4.33 K.

With air temperature modifications, the potential for heat storage with a limited duration of the temperature jump is clearly smaller than with direct impact of the temperature jump on the surface. After 12 hours the surface has only taken up 60% of the air temperature jump, i.e. the effective storage capacity sinks by 40%.

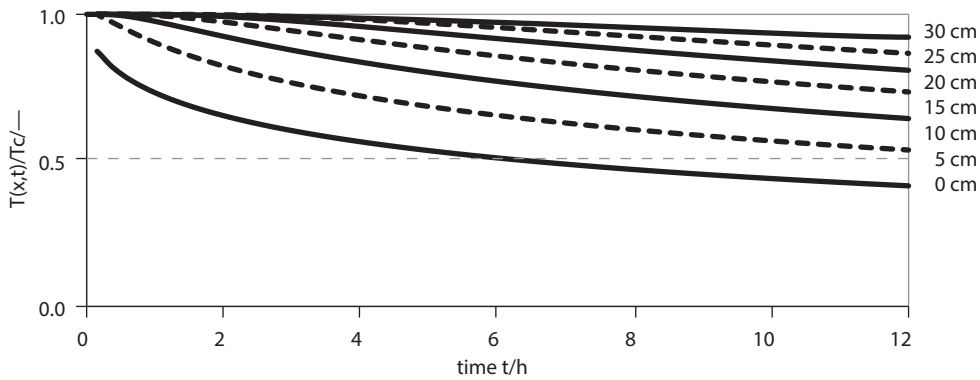


Figure 2.15 Temperature ratio as a function of time at a given air temperature for the material concrete.

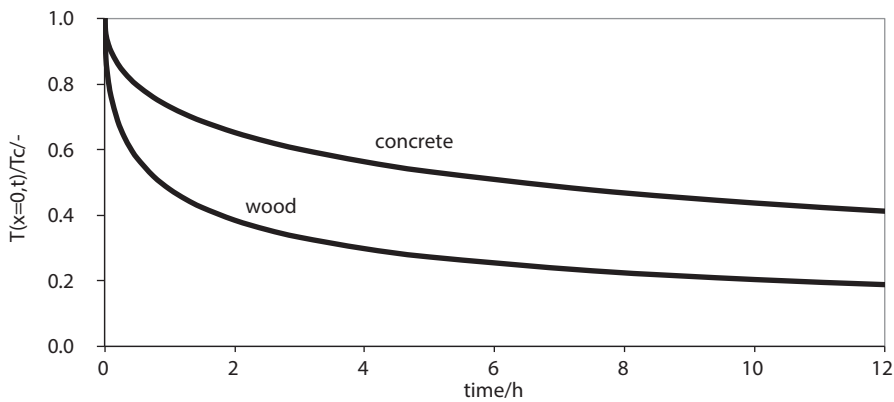


Figure 2.16 Relation of the surface temperature to the initial body temperature during an air temperature jump of 10 K.

The surface temperature changes during an air temperature jump essentially depend on the relation of the heat transfer coefficient h_i to the heat conductivity λ of the component. Component surfaces of materials with low heat conductivity clearly assume the air temperature faster (i.e. $T(x=0,t)/T_c$ becomes zero) than components that conduct heat well. Deep in the component, on the other hand, a temperature jump of air continues only very slowly with poorly conducting materials.

The heat flow density into the component can be obtained either from:

$$\frac{\dot{Q}}{A} = h_i (T_o - T_{x=0})$$

or

$$\frac{\dot{Q}}{A} = -\lambda \left. \frac{\partial T}{\partial x} \right|_{x=0}$$

With the common solution for an air temperature jump T_o to zero:

$$\frac{\dot{Q}}{A} = -h_i T_c \exp\left(\left(\frac{h}{\lambda}\right)^2 at\right) \left(1 - \operatorname{erf}\left(\left(\frac{h}{\lambda}\right) \sqrt{at}\right)\right) \quad (2.36)$$

Since the temperature difference of the surface $T(x=0)$ and air T_o (here zero) is larger at the concrete surface than at the wood surface, the larger heat flows and stored amounts of energy occur there.

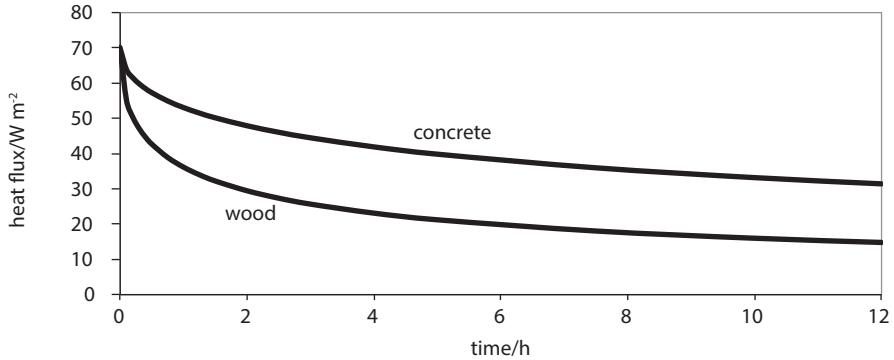


Figure 2.17 Heat flux into wood and concrete at an air temperature jump of 10 K.

To calculate the amount of heat Q for each unit area A that has flowed into the component until the point in time t_1 , the heat flux density must be integrated over the time, which is complicated to do analytically. It is simpler to calculate the heat flow density numerically in small time intervals and total afterwards.

$$\frac{Q}{A} = \frac{1}{A} \int_0^{t_1} \dot{Q} dt \approx \frac{1}{A} \sum_1^n \dot{Q}_i \Delta t \quad \text{with } \frac{t_1}{\Delta t} \quad (2.37)$$

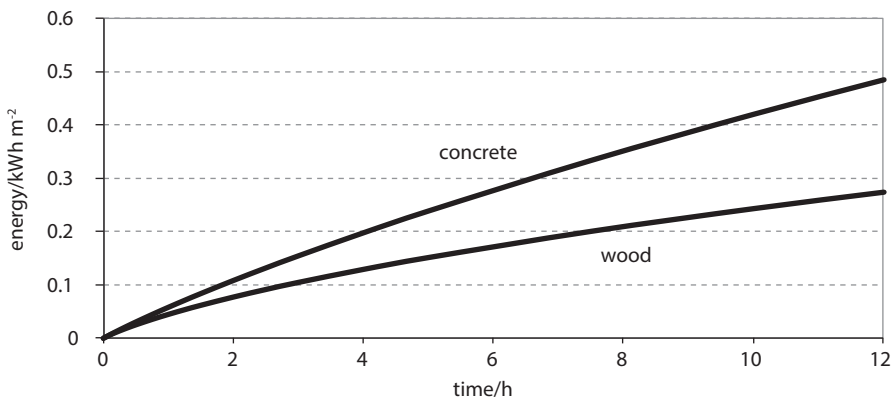


Figure 2.18 Amount of heat stored in the materials for an air temperature jump of 10 K.

Periodically variable temperatures

Apart from temperature jumps, periodic changes in temperature caused by the external climate are of particular importance for practical applications. The outside temperature T_o and the irradiance converted to a fictitious sol-air temperature can be approximated as periodic functions of time t with a period t_0 of 24 hours and an amplitude T_{om} .

The analytic solution of the heat conduction equation is, as with the temperature jump, simpler for a periodic boundary condition at the surface. First, the material temperatures of a semi-infinitely expanded component with periodically varying surface temperatures $T_s(t)$ will be calculated; only afterwards will they be generalised to the boundary condition of the air temperature. With a period duration t_0 and amplitude T_{sm} , the surface temperature $T_s(t)$ is set as follows:

$$T_s(t) = T_{sm} \cos\left(\frac{2\pi}{t_0} t\right) \quad (2.38)$$

Proceeding from a product approach for the temperature field $T(t, x) = \varphi(t)\psi(x)$, for the time function $\varphi(t)$ the complex exponential function $\exp(ipt) = \cos(pt) + i\sin(pt)$ is selected as the periodic function. With this approach for the time function and $d\varphi(t)/dt = ip\exp(ipt)$, the thermal conduction equation

$$a \frac{d^2 T}{dx^2} = \frac{dT}{dt}$$

becomes:

$$\begin{aligned} a \frac{d^2 \Psi(x)}{dx^2} \exp(ipt) - ip\psi(x)\exp(ipt) &= 0 \\ \Leftrightarrow \frac{d^2 \Psi(x)}{dx^2} - i \frac{p}{a} \psi(x) &= 0 \end{aligned} \quad (2.39)$$

with the solution

$$\psi(x) = C \exp\left(x \sqrt{-i \frac{p}{a}}\right) \quad (2.40)$$

and

$$T(t, x) = C \exp(ipt) \exp\left(x \sqrt{-i \frac{p}{a}}\right) \quad (2.41)$$

This can be split into a real and an imaginary part. After some rearrangements and use of the surface boundary condition at $x = 0$, the constant of the imaginary solution becomes zero and the real temperature field is:

$$T(x, t) = T_{sm} \exp\left(-x \sqrt{\frac{\pi}{at_0}}\right) \cos\left(\frac{2\pi}{t_0} t - x \sqrt{\frac{\pi}{at_0}}\right) \quad (2.42)$$

The wavelength x_l of the cosine function results in $x_l = 2\sqrt{\pi a t_0}$ from the relationship

$$x_l \sqrt{\pi / (a t_0)} = 2\pi$$

and the propagation rate of the wave is $v = x_0 / t_0 = 2\sqrt{\pi a / t_0}$. The subsiding exponential function dampens the amplitude of the wave with rising x . The temporal phase shift t_x of the temperature wave at component depth x compared to the surface $x = 0$ is:

$$\frac{2\pi}{t_0} t_x = x \sqrt{\frac{\pi}{a t_0}} \Rightarrow t_x = \frac{x}{2} \sqrt{\frac{t_0}{a \pi}} \quad (2.43)$$

A periodic change in temperature at a component surface occurs, for example, in transparently insulated components with absorption of solar radiation on the wall surface.

With equation Equation (2.42) the periodic change in temperature within the component can be calculated with exponentially dampened amplitude and period duration t_0 . The temperature field $T(x, t)$ is related for illustration purposes to the amplitude of the surface temperature fluctuation T_{sm} . Even at just 5 cm component depth, the surface amplitude of a concrete wall is reduced by 30%. The phase shift between the maximum surface temperature and the temperature at a depth of 5 cm is 1.5 h.

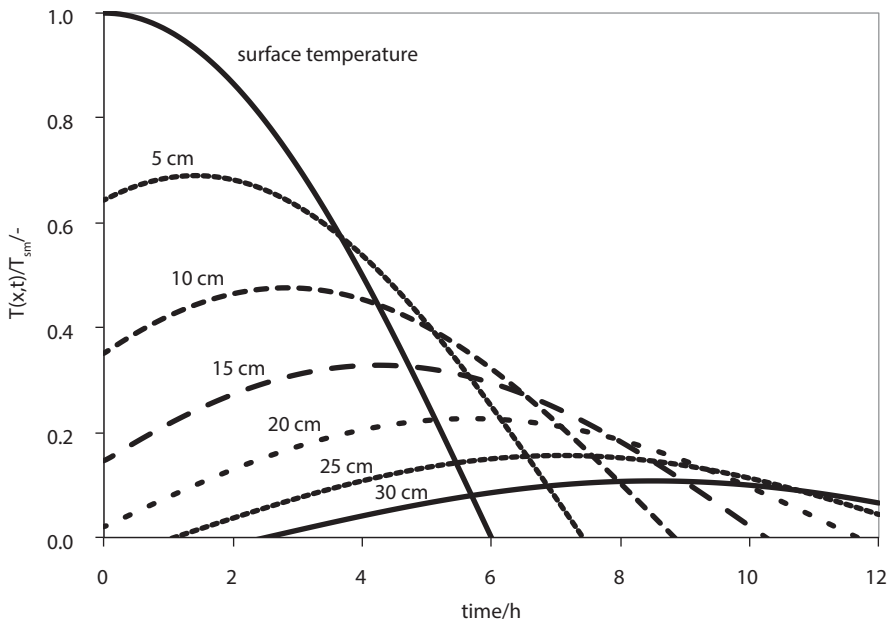


Figure 2.19 Variation in temperature $T(x,t)$ normalised to the amplitude T_{sm} around the average value as a function of time and component depth x in 5 cm steps for a concrete building component.

The phase shift between the maximum surface temperature and maximum temperature at 24 cm wall thickness is

$$t_x = \frac{0.24m}{2} \sqrt{\frac{24h \times 3600 \frac{s}{h}}{0.66 \times 10^{-6} \frac{m^2}{s} \times \pi}} = 6.8h$$

and 10.8 hours at 0.38 m wall thickness. The amplitude ratio is

$$\frac{T(x, t)}{T_{sm}} = \exp \left(-0.24m \sqrt{\frac{\pi}{0.66 \times 10^{-6} \frac{m^2}{s} \times 24h \times 3600 \frac{s}{h}}} \right) = 0.17$$

for the 24 cm wall and 0.06 for the 38 cm wall. The temperature amplitude, which amounts to 20 K around the average value of 30°C, is dampened to 3.4°C and 1.2°C, respectively.

The heat flow into a component with a periodic temperature boundary condition is again calculated using Fourier's law, from the temperature gradient at the surface:

$$\begin{aligned} \frac{dQ}{dt} &= -\lambda A \frac{dT}{dx} \Big|_{x=0} = -\lambda A \frac{d}{dx} \left(T_{sm} \exp \left(-x \sqrt{\frac{\pi}{at_0}} \right) \cos \left(\frac{2\pi}{t_0} t - x \sqrt{\frac{\pi}{at_0}} \right) \right) \Big|_{x=0} \\ &= -\lambda A T_{sm} \left[\underbrace{\exp \left(-x \sqrt{\frac{\pi}{at_0}} \right)}_1 \times \underbrace{-\sqrt{\frac{\pi}{at_0}}}_{\times} \times \cos \left(\frac{2\pi}{t_0} t - x \sqrt{\frac{\pi}{at_0}} \right) \right. \\ &\quad \left. + \underbrace{\exp \left(-x \sqrt{\frac{\pi}{at_0}} \right)}_1 \times \underbrace{-\sin \left(\frac{2\pi}{t_0} t - x \sqrt{\frac{\pi}{at_0}} \right)}_{\times} \times \underbrace{-\sqrt{\frac{\pi}{at_0}}}_{\times} \right] \Big|_{x=0} \\ &= \lambda A T_{sm} \sqrt{\frac{\pi}{at_0}} \left(\cos \left(\frac{2\pi}{t_0} t \right) - \sin \left(\frac{2\pi}{t_0} t \right) \right) \end{aligned} \quad (2.44)$$

The heat flow $\dot{Q}/A$ is proportional to the surface temperature amplitude T_{sm} .

Example 2.4

A building with concrete ceilings is to be passively cooled by night ventilation. The surface temperatures can be approximated using a cosine function with an average value of 22°C, a maximum deviation of $\pm 5^\circ\text{C}$ and a period duration $t_0 = 24\text{ h}$.

The heat flow removed, based on Equation (2.44), is a maximum of -67 W m^{-2} with a phase shift to the temperature of $t_0/8$, i.e. 3 hours.

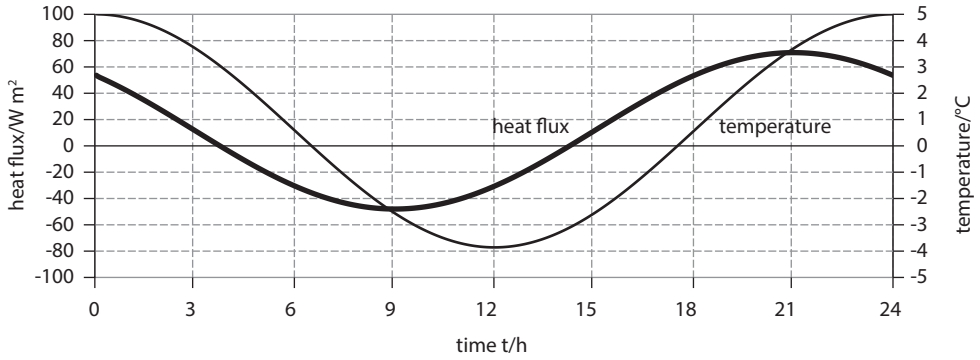


Figure 2.20 Heat flux into a concrete ceiling with a periodic surface temperature at amplitude $T_{sm} = 5^\circ\text{C}$.

The integration of the heat flux results in the surface-related energy Q/A . If the stored or removed amount of heat is to be calculated, the integration limits t_1 and t_2 must be selected in such a way that only positive or negative heat flows are integrated. From Example 2.4 it can be seen that a phase shift of $\pi/4$ exists between the maximum heat flow and maximum surface temperature, i.e. one-eighth of a period.

$$\begin{aligned} \frac{Q}{A} &= \lambda T_{sm} \sqrt{\frac{\pi}{at_0}} \int_{t_1}^{t_2} \left(\cos\left(\frac{2\pi}{t_0}t\right) - \sin\left(\frac{2\pi}{t_0}t\right) \right) dt \\ &= \lambda T_{sm} \sqrt{\frac{\pi}{at_0}} \frac{t_0}{2\pi} \left(\sin\left(\frac{2\pi t_2}{t_0}\right) - \sin\left(\frac{2\pi t_1}{t_0}\right) + \cos\left(\frac{2\pi t_2}{t_0}\right) - \cos\left(\frac{2\pi t_1}{t_0}\right) \right) \end{aligned} \quad (2.45)$$

The stored amount of heat of a half period results from the integral of the positive heat flows with a lower integration limit of $t_1 = 5/8 t_0$ and an upper limit $t_2 = 5/8 t_0 + t_0/2$, and the released amount of heat from the integral being between $t_1 = t_0/8$ and $5/8 t_0$.

Example 2.5

Calculation of the energy removed in a half period (12 hours) by night cooling with a surface temperature amplitude of 5 K.

From Equation (2.45), the result is an amount of energy removed by night cooling of -0.51 kWh m^{-2} at a lower integration limit of $t_1 = t_0/8$.

Normally, it is not the surface temperature T_s that is known, but rather the air temperature T_o with amplitude T_{om} and the heat transfer coefficient h between the air and the surface. The analytic solution corresponds to the solution with the surface temperature as a boundary condition from Equation (2.42), with the amplitude dampened by a factor η_0 and a phase shift ε_0 occurring due to thermal resistance between the air and the surface.

$$T(x, t) = T_{om} \eta_0 \exp\left(-x \sqrt{\frac{\pi}{at_0}}\right) \cos\left(\frac{2\pi}{t_0}t - \left(\varepsilon_0 + x \sqrt{\frac{\pi}{at_0}}\right)\right) \quad (2.46)$$

with

$$\eta_0 = \sqrt{\left(1 + 2\sqrt{\frac{\pi}{(h/\lambda)^2 at_0}} + 2\frac{\pi}{(h/\lambda)^2 at_0}\right)^{-1}}$$

and

$$\varepsilon_0 = \arctan\left(\left(1 + \sqrt{\frac{(h/\lambda)^2 at_0}{\pi}}\right)^{-1}\right)$$

At $x=0$ the surface temperature T_s is obtained with amplitude dampening η_0 and phase shift ε_0 .

$$T(x=0, t) = T_s = T_{om} \eta_0 \cos\left(\frac{2\pi}{t_0} t - \varepsilon_0\right) \quad (2.47)$$

Example 2.6

Calculation of the surface temperature amplitude and the phase shift ε_0 for the concrete ceiling in Example 2.4 with a heat transfer coefficient $h = 8 \text{ W m}^{-2} \text{ K}^{-1}$. The air temperature is to have an amplitude of $\pm 5 \text{ K}$ around an average value of 22°C .

With

$$\frac{\pi}{(h/\lambda)^2 at_0} = \frac{\pi}{\left(8 \frac{\text{W}}{\text{m}^2 \text{K}} / 1.28 \frac{\text{W}}{\text{mK}}\right)^2 0.66 \times 10^{-6} \frac{\text{m}^2}{\text{s}} \times 24 h \times 3600 \frac{\text{s}}{h}} = 1.41$$

the result is

$$\eta_0 = \sqrt{\frac{1}{1 + 2\sqrt{1.41} + 2 \times 1.41}} = 0.4$$

The maximum temperature amplitude at the surface is now only $T_s = 5\text{K} \times 0.4 = 2\text{K}$, and thus, the removable energy falls by 60%! The phase shift is

$$\varepsilon_0 = \arctan\left(\left(1 + \sqrt{\frac{1}{1.41}}\right)^{-1}\right) = 28.5^\circ$$

i.e. in a 24 hour period, scarcely 2 hours.

Influence of solar irradiance

The use of a simple energy balance model is recommended if the irradiance on a component surface is also to be considered in addition to fluctuations of air temperature. The energy balance model provides for the conversion of short-wave solar irradiance into a so-called sol-air temperature, which allows for the usage of the aforementioned analytic solutions to the thermal conduction equations.

The heat flow supplied to a component surface consists of the absorbed irradiance αG plus the heat flow transferred by the air (temperature T_o) to the surface (T_s) with the heat transfer coefficient h . This supplied heat flow is combined in a simple model, ignoring temperature-dependent modifications of the heat transfer coefficient h , into a purely temperature-dependent heat flow described by the sol-air temperature T_{so} .

$$\begin{aligned}\alpha G + h(T_o - T_s) &= h(T_{so} - T_s) \\ T_{so} &= T_o + \frac{\alpha G}{h}\end{aligned}\quad (2.48)$$

With the sol-air temperature T_{so} , the heat storage in components can then be calculated with the solutions of the thermal conduction equation already considered.

Example 2.7

Calculation of the sol-air temperature and the energy stored in a concrete floor, with periodically varying irradiance at an amplitude of 500 W m^{-2} and period $t_0 = 24 \text{ h}$ transmitted through the windows, an absorption coefficient of the floor of $\alpha = 0.6$, a room air and ceiling average temperature $T_o = 20^\circ\text{C}$ and a heat transfer coefficient $h = 8 \text{ W m}^{-2} \text{ K}^{-1}$. The maximum sol-air temperature is:

$$T_{so, \max} = 20^\circ\text{C} + \frac{0.6 \times 500 \frac{\text{W}}{\text{m}^2}}{8 \frac{\text{W}}{\text{m}^2 \text{K}}} = 57.5^\circ\text{C}$$

Via the fictitious sol-air temperature, first the surface temperature at the concrete surface is calculated, and from this, using Equation (2.45), the stored energy per square metre of surface is also obtained.

Using Example 2.6, $\eta_0 = 0.4$ and thus, the maximum air temperature fluctuation of $57.5^\circ\text{C} - 20^\circ\text{C} = 37.5^\circ\text{C}$ is reduced to a maximum surface temperature fluctuation of $T_{a, \max} = \eta_0 (T_{so, \max} - T_o) = 15^\circ\text{C}$. The energy stored during a half period is 1.54 kWh m^{-2} .

Altogether, however, an irradiance of

$$\int_0^{t_0/2} G dt = G_{\max} \int_0^{t_0/2} \sin\left(2\pi \frac{t}{t_0}\right) dt = \frac{G_{\max} t_0}{\pi} = 3.8 \frac{\text{kWh}}{\text{m}^2}$$

is transmitted through the glazing during the half period and 60% of it, i.e. 2.3 kWh m^{-2} , is absorbed by the concrete floor. The difference between the absorbed irradiance and stored heat has been transferred over the heat transfer coefficient h directly to the room air.

2 Part B: Natural ventilation

with Tobias Schulze

Figure 2.21 (previous page): Cerdanyola residential building near Barcelona, Spain with ventilated Trombe walls and cross ventilation for night cooling (Photo: POLYCITY).

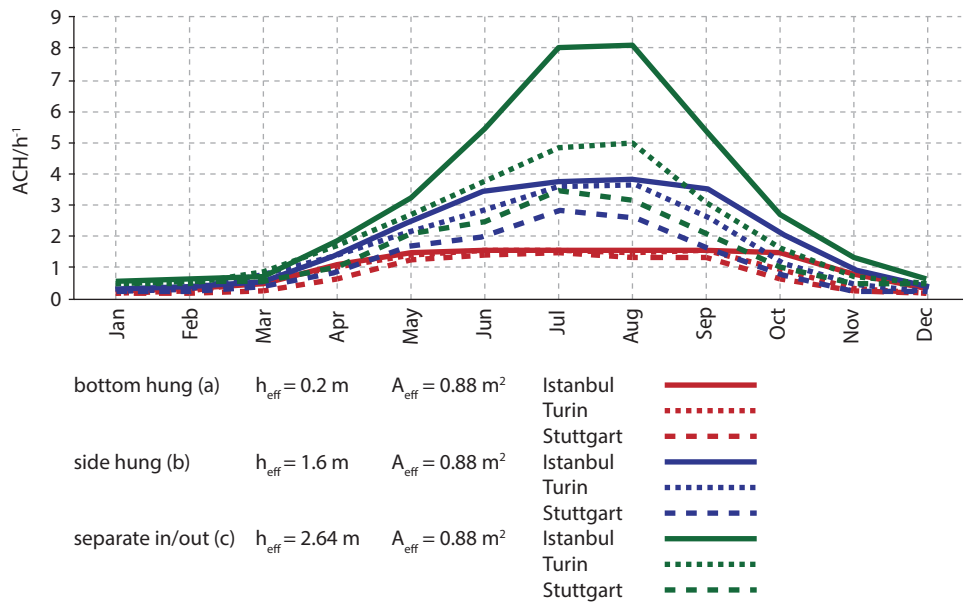


Figure 2.22 Simulated air exchange rates (monthly averages) for adaptive temperature-controlled single-sided ventilation over the course of the year (see 'Controlled single-sided ventilation with dynamic boundary conditions' on page 87).

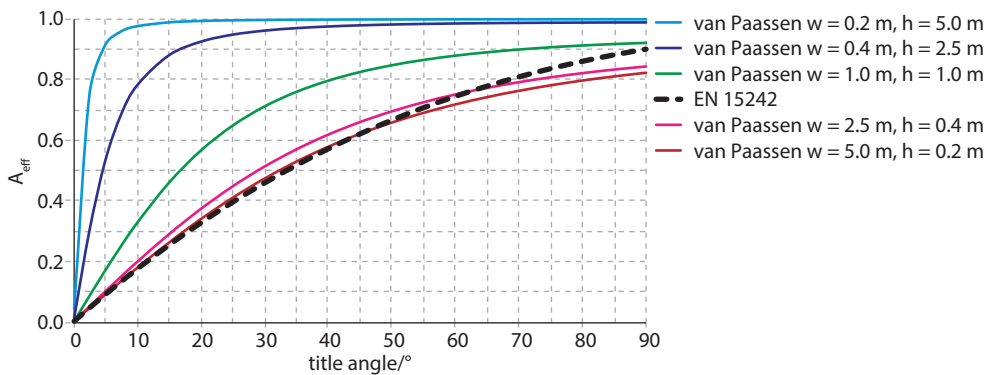


Figure 2.23 Effective opening area as a function of opening angle for bottom-hung tilting windows with an area of 1 m^2 (see 'Effective opening geometry' on page 77).

The two main functions of natural ventilation concepts are to provide good indoor air quality without any electricity demand for moving the air and to improve summer thermal comfort by increased daytime air speed and high night ventilation rates. The main drawback is the difficulty of winter heat recovery from the warm room air; the main advantage is the possibility to achieve high ventilation rates in summer for cooling without any energy need.

In the design of ventilation strategies, natural ventilation is often not considered, as there are no standards available, which support the planners in the design of natural ventilation openings, and few results are available on the performance of naturally ventilated buildings, especially when energy savings are concerned. This lack of experience and know-how of the main stakeholders such as architects, engineering consultant and building developers seems to be a major barrier to the use of natural ventilation. Many research projects tried to develop simplified methods for passive cooling applications to handle the complex task of simulating air flow rates under changing external pressure and temperature conditions.

Two design strategies for naturally ventilated buildings can be identified: when indoor air quality is the main priority, air flow rates tend to be low. When summer comfort is the priority, much higher air flow rates are required and significant thermal mass needs to be present for night ventilation. Simulations show that natural night ventilation is most suitable in buildings with sufficient and accessible thermal mass of about 75 - 100 kg per square metre of floor space. The internal gains have to be limited to about 30 W per m² of floor area.

Night ventilation can reduce the mean room temperature by about 1 - 3 K, depending on climate and ventilation strategy (1.2 K during working hours measured in building projects in Freiburg/Germany, between 1.5 and 2 K in La Rochelle/France, 1.8 - 3 K in Greece). Monitoring projects show that night ventilation works well in climates with large diurnal temperature differences, but cannot be recommended for humid climates with humidity ratios above 15 g per kg air.

Natural ventilation rates can reach much higher values than mechanical ventilation systems are typically designed for, especially when cross ventilation is possible. A range of studies using measurements and simulations in schools and offices showed air change rates between 5 and 22 per hour for cross ventilation and 1 to 4 for single-sided ventilation. The World Health Organisation recommends the use of natural ventilation in hospital isolation rooms with very high air exchange rates of 24 per hour, while general hospital areas should be ventilated with 8 air exchanges per hour.

Depending on the ventilation type, the openings should be roughly between 1% and 3% of the floor space: for single-sided ventilation, the opening cross section needs to be highest with about 3%, for cross ventilation 2% and for stack ventilation 1% stack area.

The control of natural ventilation openings either focuses on indoor air quality, on indoor temperature levels for free cooling modes or a mix of both. During occupation the control should also consider the draught risk (especially if the temperature difference between inside and outside is high) and inhabitants would not accept frequent adjustments. Different control strategies either based on room air temperature setpoints, floor slab temperature levels, degree hours or daily cooling uptake were compared for office buildings and showed very comparable results. The main issue is to provide high-enough opening areas for night ventilation and to limit the internal gains. Results from 12 office and educational building case studies showed that during non-occupancy, automatic control is necessary to cool down the building structure by night ventilation (Aggersholm, 2002). During occupancy, a distinction is made between cellular

and landscaped offices. While user control functions well in cellular offices, in landscaped offices, automatic control is also necessary by day, although the correct control strategy is not easy to establish, and users should still be able to override the automatic control.

Energy savings by natural ventilation can mostly only be evaluated, when simulation tools are used, as two identical buildings with different ventilation or climatisation strategies are rarely available for monitoring.

A 30% reduction of the cooling energy consumption and 40% reduction of the installed cooling capacity was predicted for a UK low-energy office building with a night stack ventilation air exchange of 10 per hour (Kolokotroni, 1999). Forty per cent reduction of the daily cooling demand was simulated for a high thermal mass office building in Belgium (Gratia, 2004). Night ventilation air changes of 8 per hour can reduce cooling requirements by 12 - 54%, depending on the temperature set point (Blondeau, 1997). Studies at the 23-storey Liberty Tower of Meiji University in Tokyo showed that about 17% of energy consumption for cooling is saved by the natural ventilation system.

Performance indicators for naturally ventilated buildings

The temperature excess method cumulates the hours with room air temperatures above a given setpoint and compares them with limiting values, for example, 10% of all office hours. The more sophisticated weighted temperature excess method includes additional parameters such as air velocity, internal vapour pressure, metabolism and clothing resistance. As these values are difficult to obtain, only the temperature excess method is considered in the following. Acceptable temperature setpoints should be calculated following the adaptive comfort limits, which are defined in the European standard EN 15251. In the standard, operative zone temperatures are used, which are the arithmetic mean of the zone mean air and the zone mean radiant temperature.

The operative room temperature is allowed to increase in naturally ventilated, non-air-conditioned buildings, with rising ambient air temperatures. Depending on the exponentially weighted running mean of the daily mean ambient air temperature series of the last week, recommended operative temperatures are calculated for different comfort categories. The criteria were obtained through investigations in office buildings with user-operated windows.

The running mean daily ambient temperature Θ_{rm} is obtained from the mean external temperatures of the day before Θ_{ed-1} and weighted temperatures of the previous days of the week.

$$\Theta_{rm} = \frac{(\Theta_{ed-1} + 0.8 \Theta_{ed-2} + 0.6 \Theta_{ed-3} + 0.5 \Theta_{ed-4} + 0.4 \Theta_{ed-5} + 0.3 \Theta_{ed-6} + 0.2 \Theta_{ed-7})}{3.8} \quad (2.49)$$

Using this mean temperature makes calculations of the admissible lower and upper boundaries of the operative temperature for different categories possible.

$$\begin{aligned} \Theta_{imax} &= 0.33\Theta_{rm} + 18.8 + 2 \quad \text{category 1 upper limit} \\ \Theta_{imax} &= 0.33\Theta_{rm} + 18.8 - 2 \quad \text{category 1 lower limit} \\ \Theta_{imax} &= 0.33\Theta_{rm} + 18.8 + 3 \quad \text{category 2 upper limit} \\ \Theta_{imax} &= 0.33\Theta_{rm} + 18.8 - 3 \quad \text{category 2 lower limit} \\ \Theta_{imax} &= 0.33\Theta_{rm} + 18.8 + 4 \quad \text{category 3 upper limit} \\ \Theta_{imax} &= 0.33\Theta_{rm} + 18.8 - 4 \quad \text{category 3 lower limit} \end{aligned} \quad (2.50)$$

For ambient temperatures $\theta_{rm} < 10^{\circ}\text{C}$ or $\theta_{rm} > 30^{\circ}\text{C}$, the upper comfort temperature level remains constant at the value of the boundary temperature; for the lower limit, fixed constant values are used for $\theta_{rm} < 15^{\circ}\text{C}$ or $\theta_{rm} > 30^{\circ}\text{C}$.

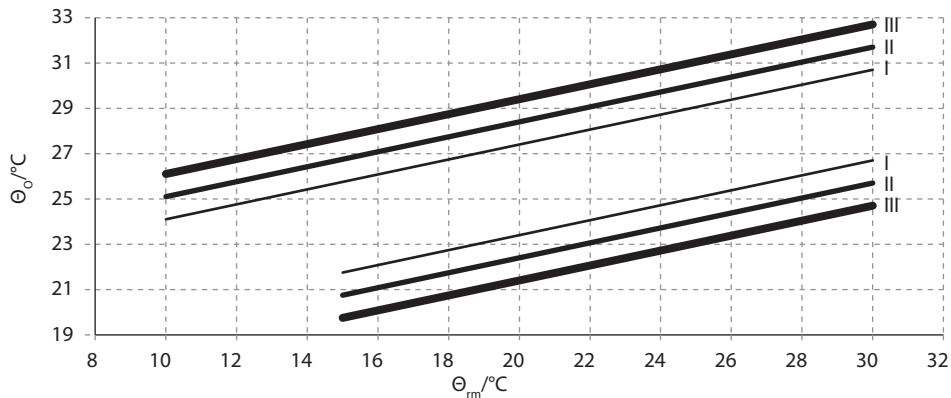


Figure 2.24 Design values of the operative room temperature for buildings without mechanical cooling as a function of the weighted ambient temperature according to EN 15251.

Category	Explanation
I	High level of expectation and is recommended for spaces occupied by very sensitive and fragile people with special requirements like handicapped, sick, very young children and elderly people
II	Normal level of expectation and should be used for new buildings and renovations
III	An acceptable, moderate level of expectation and may be used for existing buildings
IV	Values outside the criteria for the above categories. This category should only be accepted for a limited part of the year

Table 2.2 Description of the categories used in the adaptive thermal comfort model.

As an example, the annual adaptive thermal comfort limits were calculated with the ASHRAE IWEC typical climate data for three reference locations Istanbul/Turkey, Turin/Northern Italy and Stuttgart/Southern Germany. Operative temperature levels over 30°C are allowed for hot ambient air summer conditions (see, for example, Istanbul in Figure 2.25).

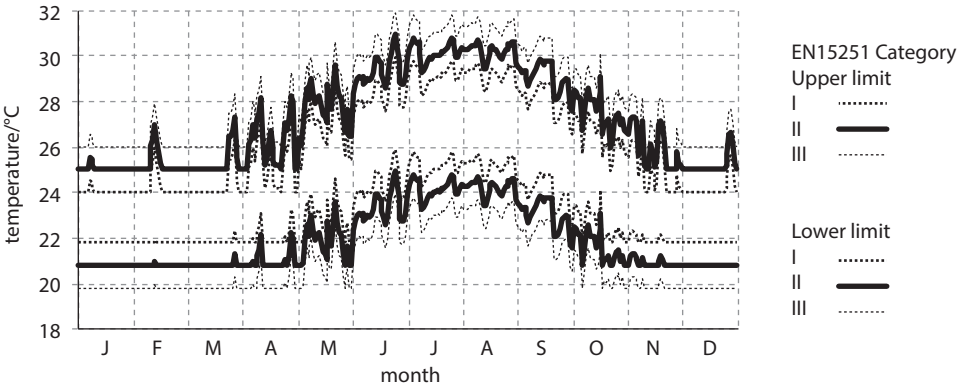


Figure 2.25 Limits of thermal comfort for naturally ventilated buildings in Istanbul according to EN 15251 calculated from sliding daily mean values for Istanbul climate data.

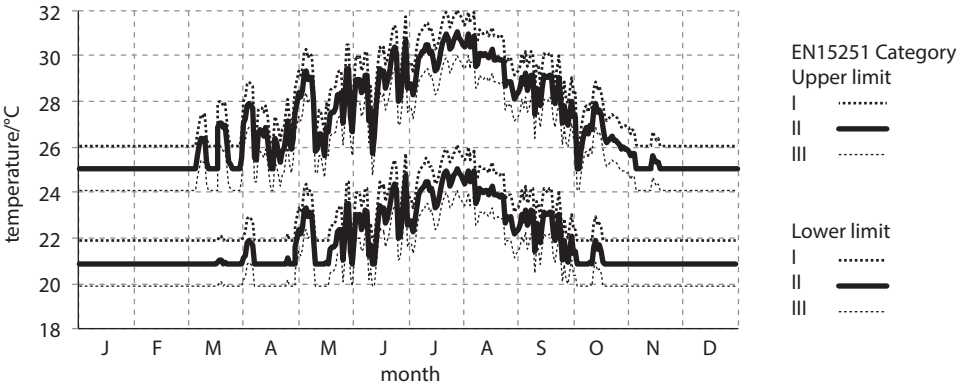


Figure 2.26 Limits of thermal comfort for naturally ventilated buildings according to EN 15251/calculated from sliding daily mean values for Turin climate data.

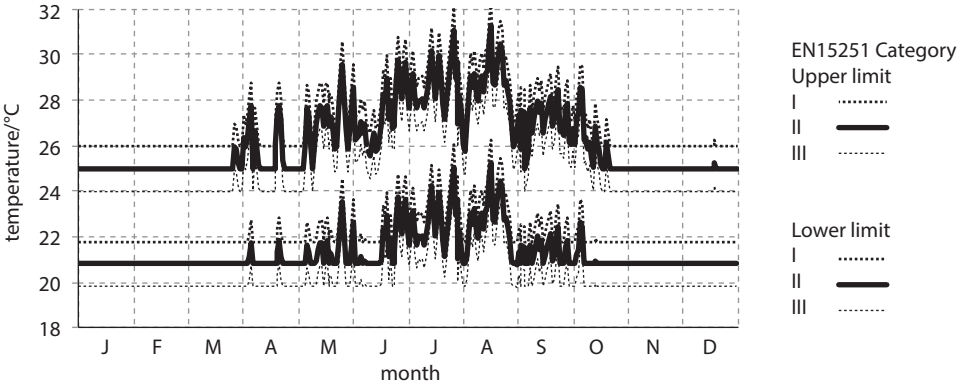


Figure 2.27 Limits of thermal comfort for naturally ventilated buildings according to EN 15251/calculated from sliding daily mean values for Stuttgart climate data.

Indoor air quality is usually evaluated by the CO_2 concentration indicator. There is, however, no agreement on the limit values for good air quality. According to EN 13779, high indoor air quality (IAQ) is achieved with less than 400 ppm above level of outdoor air, medium quality in a range between 400 to 600 ppm, moderate IAQ from 600 to 1000 ppm and low IAQ above 1000 ppm. The German Central Environment Ministry considers 1000 ppm in schools as hygienically good, from 1000 to 2000 ppm as hygienically noticeable and above 2000 ppm as not acceptable. EN 15251 uses 1000 ppm as the upper limit for the design of ventilation systems.

2.4 Analytical methods for volume-flow calculations

The driving forces for natural ventilation of rooms and buildings are pressure differences caused by buoyancy and wind. Ventilation rates are dependent on the magnitude and direction of these forces and the flow resistance of the flow path. The analytical calculation methods can be utilised to validate airflow network or computational fluid dynamics (CFDs) simulation results.

Buoyant flow

The volume flow caused by buoyancy forces through a single large opening (i.e. open window rather than cracks) or multiple openings in the building envelope is proportional to the square root of the pressure difference depending on the room air and the external air temperature.

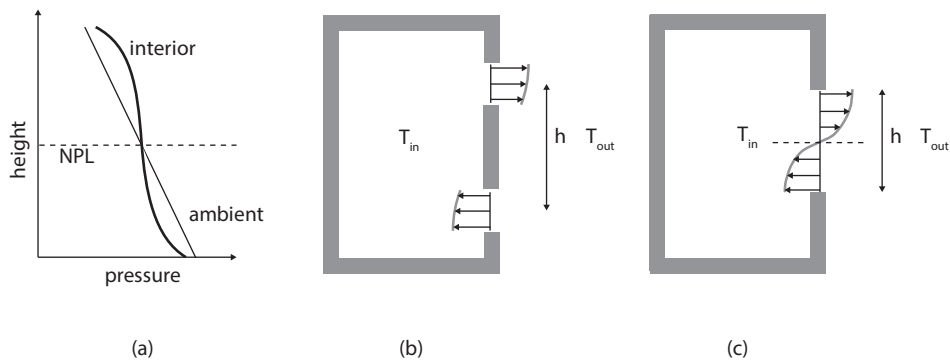


Figure 2.28 Internal and external pressure distribution of buoyant flow (a) causes air flow through a lower and upper opening (b) or through a single large opening (c).

Buoyant flow through two openings

For the ventilation through a lower and an upper equally sized opening and negligible internal resistance like in Figure 2.28 (b), the air volume flow rate can be estimated according to the following equation (Etheridge, 1996):

$$q_v = C_d A_{eff} \sqrt{g h_s \frac{\Delta T}{\frac{T_{in} + T_{out}}{2}}} \quad (2.51)$$

where

q_v volume flow/ $\text{m}^3 \text{s}^{-1}$

C_d the discharge coefficient of the openings, which are dependent on the opening geometry

A_{eff} effective cross-sectional inlet opening area if the outlet is equally sized/ m^2

g gravity acceleration $9.81/\text{m s}^{-2}$

h_s height difference between the midpoints of the openings, which is the stack height/ m

ΔT absolute temperature difference between the room and ambient air/ K

T_{in} room air temperature in kelvin/ K

T_{out} ambient air temperature in kelvin/ K

Values of the discharge coefficient are between 0.60 and 0.65 for sharp-edged external openings. For internal openings in partitioned buildings, which are mostly larger than the external openings, the discharge coefficient may be up to 1.

Buoyant flow through a single opening

The buoyant flow through a single opening like a window is bidirectional. For cold ambient conditions, the warm room air exits at the upper part of the opening and colder ambient air enters at the lower part of the opening. If the ambient air is warmer than the room air in summer conditions, the flow reverses. The flow direction changes at the height of the neutral pressure plane.

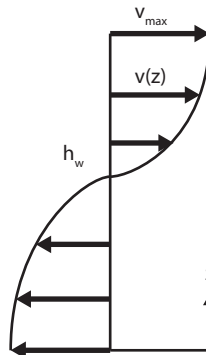


Figure 2.29 Flow profile for single-sided ventilation of a large opening.

For single-sided ventilation with a single large opening, the flow rate caused by buoyancy can be expressed by the following equation:

$$q_v = \frac{C_d A_{eff}}{3} \sqrt{g h_{eff} \frac{\Delta T}{T_{in} + T_{out}}} \quad (2.52)$$

where

A_{eff} effective cross-sectional opening area of the single opening/m²

h_{eff} effective opening height of the free area of the opening/m

The integration of the flow profile over the height results in the constant of 1/3 in this equation, as only part of the opening is available for air inlet, and the neutral pressure plane is assumed at half of the effective opening height level.

Wind-induced flow

The air flow due to wind pressure differences is dependent on the pressure coefficients C_p , which are influenced by the wind direction, and on the local wind speed at the height of the opening. The flow rate may be calculated by the following equation (Etheridge, 1996):

$$q_v = C_d A_{eff} u_w \sqrt{\frac{\Delta C_p}{2}} \quad (2.53)$$

where

u_w local wind speed if similar at both openings/m s⁻¹

ΔC_p difference of wind pressure coefficients

Values of the pressure coefficient depend on the building shape, the wind direction, the environment (nearby buildings and vegetation) and the specific location on the building surface (height and distance from the middle) (ASHRAE, 2009).

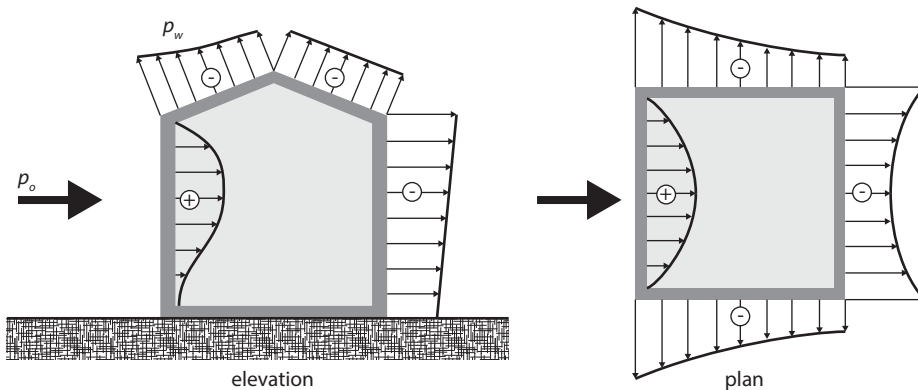


Figure 2.30 Wind pressure distribution on a pitched roof building.

C_p data can be obtained from primary or secondary sources. Primary sources are expensive and time-intensive full scale measurements, wind tunnel measurements or CFD simulations. Less reliable secondary sources are databases or analytical approaches which are based on

interpolation and extrapolation of generic knowledge and previously measured data and are therefore best applicable on standard building geometries.

As it is not practically possible to consider the complete complexity of the variation of the wind pressure coefficient during air flow simulations, a simplified method is used.

The static pressure coefficient is defined as:

$$C_p = \frac{p_w - p_o}{0,5 \rho u_w^2} \quad (2.54)$$

where

p_w static pressure on a point on the building/Pa

p_o reference static pressure in the open flow area/Pa

ρ density in the open flow area/kg m⁻³

u_w wind velocity in the open flow area at the building height or another reference height/m s⁻¹

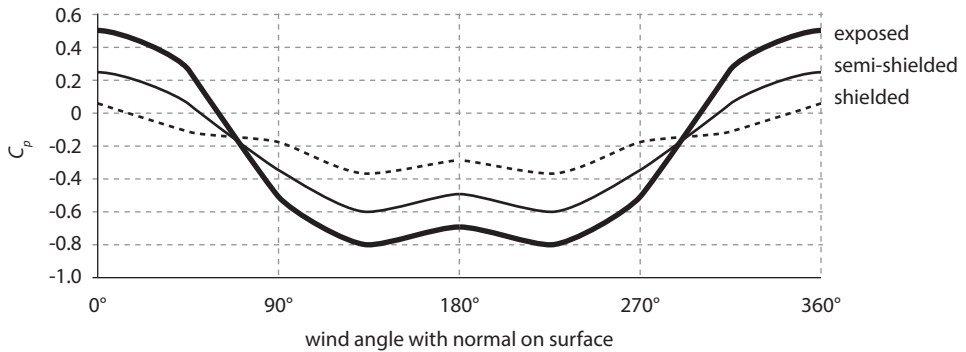


Figure 2.31 Comparison of wind pressure coefficient C_p on the facade for a low-rise building with different shielding conditions (Orme and Leksmono, 2002).

Combined flow

For ventilation with separate equally sized inlet and outlet openings and negligible internal resistance, the flow rate may be estimated for combined wind and buoyant flow by the following equation:

$$q_v = C_d A_{eff} \sqrt{\frac{C_{p,1} u_{w,1}^2 - C_{p,2} u_{w,2}^2}{2} \pm g h_s \frac{\Delta T}{\frac{T_{in} + T_{out}}{2}}} \quad (2.55)$$

where

- q_v volume flow/m³ s⁻¹
- C_d discharge coefficient of the openings
- A_{eff} effective cross-sectional inlet opening area if the outlet is equally sized/m²
- C_p wind pressure coefficients for the inlet and outlet
- u_w reference wind speed at local height of one opening (centroid) according to ASHRAE power-law calculation or EN 15242/m s⁻¹
- T temperature of the room (T_{in}) and ambient air (T_{out})/K
- g gravity acceleration g/9.81 m s⁻²
- h_s height difference between the midpoints of the openings, which is the stack height/m

The equation may be also applied on flows which are only wind or thermally driven by setting the wind speed or the stack height to zero. The solution of a combined wind and buoyant flow ($\pm$) is dependent on whether the opening level on the windward side is located higher than on the leeward side (-) or not (+). The internal flow resistance is not reflected in the formula.

Effective opening geometry

The definition of the effective opening area is not standardised in literature – some authors include the discharge coefficient (C_d), some do not, depending on the focus of their work.

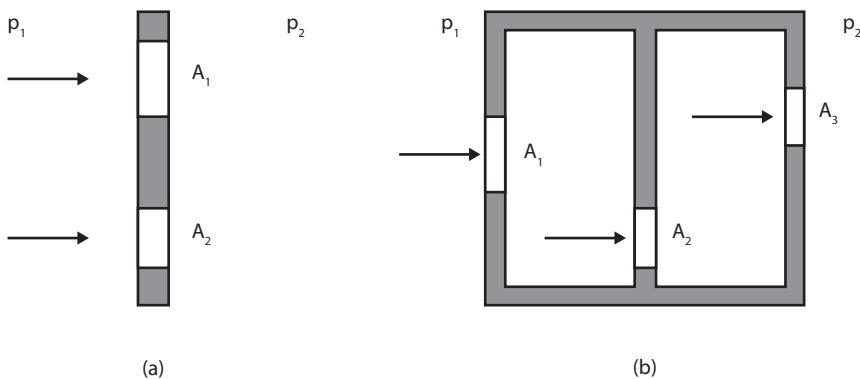


Figure 2.32 Exemplary geometrical representation of openings in (a) parallel and (b) serial arrangement.

For openings in a parallel arrangement (a), the effective opening surface ($A_{p,eff}$) is obtained from the sum of discharge coefficients and area products:

$$A_{p,eff} = C_{d,1}A_{eff,1} + C_{d,2}A_{eff,2} + \dots \quad (2.56)$$

$A_{p,eff}$ effective opening area in a parallel arrangement/m²

For openings in serial arrangement (b), the effective surface ($A_{s,eff}$) is calculated as follows:

$$\frac{1}{A_{s,eff}^2} = \frac{1}{A_{p,eff,1}^2} + \frac{1}{A_{p,eff,2}^2} + \dots \quad (2.57)$$

Simplified methods for the effective area calculation may be found in the literature, but more research is needed on the calculation of effective opening areas, for example, for pivot-hung tilted windows.

The standard EN 15242 makes the very simplified assumption that the volume flow through a tilted window only depends on the tilt angle and is independent of the ratio of height to width.

$$A_{eff} = C_k(\alpha) \times A_w \quad (2.58)$$

A_{eff} effective window opening area/m²

A_w surface area of completely opened window/m²

α opening angle (0 - 180°)

The coefficient $C_k(\alpha)$ can be calculated using a polynomial equation:

$$C_k(\alpha) = 2.60 \times 10^{-7} \alpha^3 - 1.19 \times 10^{-4} \alpha^2 + 1.86 \times 10^{-2} \alpha \quad (2.59)$$

A more precise formula also considering the ratio of height to width was developed by van Paassen (1998), which fits reasonably well to the EN 15242 formula for low height windows.

$$A_{eff} = \sqrt{\frac{1}{\frac{1}{(hw)^2} + \frac{1}{\left(2hw \sin\left(\frac{\varphi}{2}\right) + h^2 \sin(\varphi)\right)^2}}} \quad (2.60)$$

w window width/m

h window height/m

φ opening angle (0 - 90°)

The effective opening height for a bottom-hung window is then determined as follows (see Figure 2.23):

$$h_{eff} = \frac{A_{eff}}{w} \quad (2.61)$$

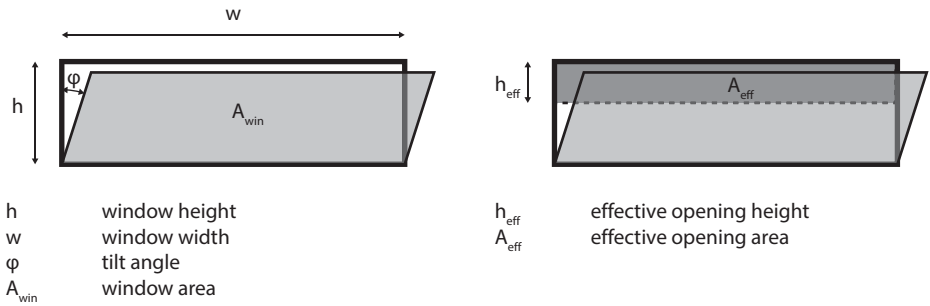


Figure 2.33 Geometrical representation of a tilted window and the resulting effective opening area and height.

2.5 Air flow network simulations

Air flow network (AFN) models consist of a set of nodes linked by air flow components such as cracks, ducts, fans, flow controllers, vertical large openings (windows and/or doors), horizontal openings (e.g. stairs) and passive stacks to obtain pressures and air flows. They can be used for simulating multizone air flows due to wind pressure and stack effects. The multizone airflow calculations when combined with dynamic building models are performed at the HVAC system time step which allows for modelling hybrid ventilation systems. With a well-mixed assumption, a building is subdivided into zones with homogeneous air properties. Large external openings like windows allow flow movement simultaneously in two different directions depending on stack effects and wind conditions, which is of special importance when simulating a single-sided ventilation strategy.

A simple AFN model can be used to determine ventilation potentials with unchanging boundary conditions to validate resulting flow rates against analytical solutions. Here only one zone is simulated with two linkages to outdoor. As the internal and external temperatures in each simulation run are stable throughout the whole run period (here one design day), there is no need to investigate the zone's heat loads as only the flow rates are of interest. If there is just one large opening for bidirectional flow, a second tiny crack, which is too small to influence the simulations, has to be integrated to fulfil the minimum requirements for an AFN with two linkages for a zone.

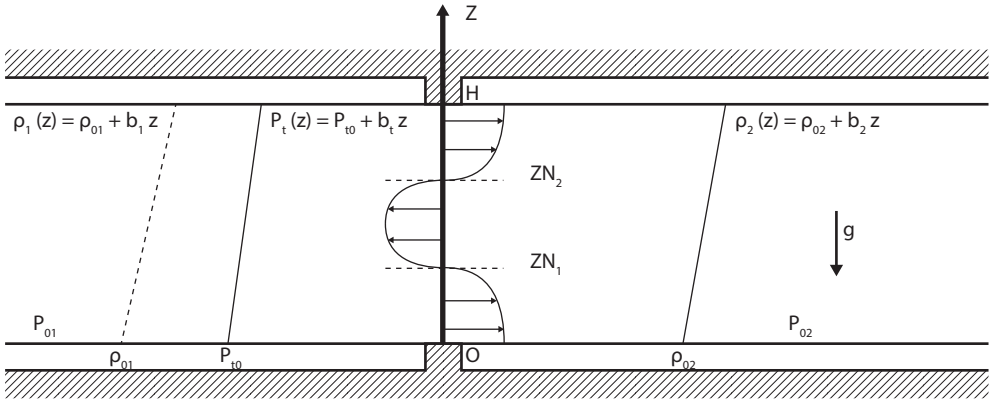


Figure 2.34 Bidirectional flow in network modelling.

For the investigation of thermal comfort and energy consumption, a far more complex AFN was developed for a reference office building, which consists of 18 office and 3 corridor zone nodes linked by 18 controlled doors and 2 stairway linkages with uncontrolled cracks. Wind pressure coefficients are input values. The local wind speed calculations are set to be calculated at opening height (centroid) depending on the roughness characteristics of the surrounding terrain based on the ASHRAE power-law calculation.

Internal cracks, doors and stairways allow zone air mixing. Doors and stairways (with $C_d=1$ as mentioned before) are opened during occupancy by 5%. The door opening time, furthermore, is controlled in a way not to allow cross ventilation for single, sided strategies. A horizontal opening object is utilised to simulate airflows across three floors with a sloping plane angle of 40° to allow staircase simulations

$$A_{eff} = A \cos \alpha (1 + \cos \alpha) \quad (2.62)$$

where α is the angle between the stair plane and horizontal opening plane.

The infiltration (Q_v in kg s^{-1}) as a function of the building's tightness is realised by the definition of a flow coefficient (C_Q in kg s^{-1} per m at 1 Pa pressure difference) and the flow exponent ($n = 0.65$ as default). The flow coefficient is a function of crack length (l_{crack} , here the sum of the uncontrolled window perimeters in the AFN), the building's net volume (V_b) and mean air density (ρ):

$$C_Q = \frac{1.5 \cdot V_b \cdot \rho}{50 \text{ Pa} \cdot 0.65 \cdot l_{crack}} \quad (2.63)$$

Reference office building

An office building used as a reference case is taken from the report 'Study on energy efficiency of innovative building, lighting and air-space concepts' (Sinnesbichler, H. and A. Koller, 2009). The thermal properties and the HVAC system correspond to the German energy-saving legal standard 2007. The chosen small administration building has a net floor area of 1146 m^2 on three floors and a surface-to-volume ratio (A/V) of 0.48 m^{-1} . The building has a flat roof, no basement and the offices on three floors are east or west orientated. The same building has been placed in three

different climates to evaluate the functionality of controlled natural ventilated building under various external conditions.

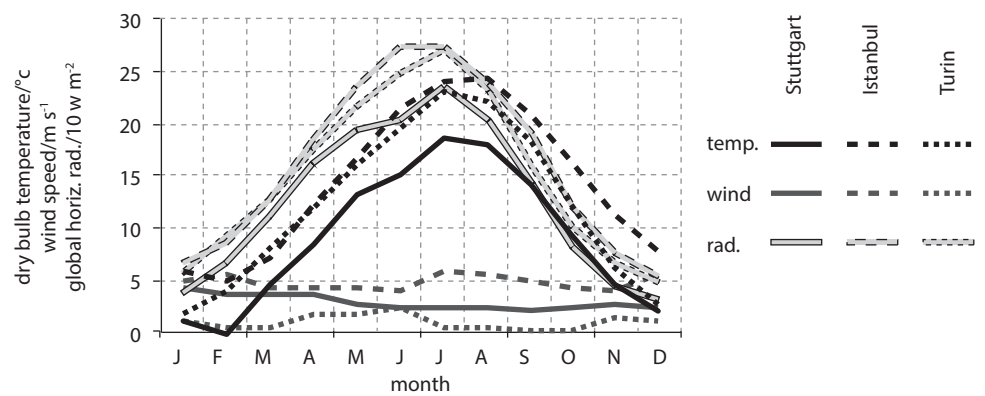


Figure 2.35 Monthly mean data for Stuttgart, Turin and Istanbul from ASHRAE IWECC typical climate files.

Wind pressure coefficients are extracted as secondary data from the AIVC database for buildings with up to three storeys, a width to length ratio of 2:1 surrounded by buildings with equal height.

To reduce the simulation effort, only part of the building with 18 identical offices connected by corridors are modelled as shown in Figure 2.36.

One vertical cut separates six offices and the corridors from the rest of the building and is defined with adiabatic properties, i.e. the same conditions to each side of the construction (inside and outside of the room), so that there is no temperature difference ΔT across the surface. All heat transfer into the surface is a result of the dynamic response of the construction and therefore an adiabatic wall or floor can still store energy if the construction materials are defined to include thermal mass.

Subsequently in the simulations, the base case is modified to show the influence of individual parameters (orientation, shading, lighting control, climate data, etc.) on the functionality of the system.

	Controlled natural ventilation	Mechanical ventilation and cooling
Climatic data	Stuttgart, Turin, Istanbul (ASHRAE IWECC typical climate)	
Room geometry (net)	18 offices: floor net area = 29.40 m ² , volume = 80.50 m ³ , ceiling height = 2.72 m, facade area = 16.64 m ² 3 corridors: floor net area = 30.60 m ² , volume = 83.23 m ³ , ceiling height = 2.72 m, facade area = 6.00 m ²	
Insulation standard <i>U</i> values	opaque external façade = 0.31 W m ⁻² K ⁻¹ , windows = 1.20 W m ⁻² K ⁻¹ , roof = 0.17 W m ⁻² K ⁻¹ , slab = 0.32 W m ⁻² K ⁻¹	
External wall	external rendering (0.005 m)/mineral wool (0.10 m)/brick (0.24 m)/interior plaster (0.015 m)	
Internal ceiling and internal floor	reinforced concrete (0.20 m)/air space (R = 0.18 m ² K W ⁻¹)/acoustical ceiling (0.01 m)	pavement (0.05 m)/EPS (0.03 m)/reinforced concrete (0.20 m)/air space/acoustical ceiling (0.01 m)
Internal partition	gypsum plasterboard (0.012 m)/mineral wool (0.08)/gypsum plasterboard (0.012 m)	
External flat roof	loose bulk (0.05 m)/bitumen sheeting (0.005 m)/EPS (0.20 m)/reinforced concrete (0.20 m)/air space (R = 0.18 m ² K W ⁻¹)/acoustical ceiling (0.01 m)	
External slab	foam glass (0.08 m)/reinforced concrete (0.15 m)/bitumen sheeting (0.005 m)/EPS (0.03 m)/pavement (0.05 m)	
Ground temperature	calculated with the E+ auxiliary programme 'slab -3d- ground heat transfer tool'	
Building air tightness	1.5 ACH at 50 Pa pressure difference between inside and outside ($n_{50} = 1.5 \text{ h}^{-1}$)	
Wind pressure coefficient data	wind angle	0° 45° 90° 135° 180° 225° 270° 315°
	<i>C_p</i> east facade	-0.30 0.15 0.18 0.15 -0.30 -0.32 -0.20 -0.32
	<i>C_p</i> west facade	-0.30 -0.32 -0.20 -0.32 -0.30 0.15 0.18 0.15
Glazing (double)	height = 1.60 m, area = 8.00 m ² , heat transfer coefficient <i>U_g</i> value = 1.20 W m ⁻² K ⁻¹ , total energy transmittance <i>g</i> value (SHGC) = 0.65, visible light transmittance <i>LT</i> value = 0.78	
Occupancy	120 W per person (office work), 2 people, 6 full hours of use from 7 am to 6 pm, 5 days per week	
Office equipment	7 W m ⁻² , 7 am - 6 pm, 5 days per week	
Shading devices	external venetian blinds: slat angle = 45°, slat width = 5 cm, slat separation = 4 cm, reflectivity = 50%	
Setpoint shading	on/off setpoint (250 W m ⁻²) depending on solar intensity on facade	
Lighting system	16.3 W m ⁻² surface mount, 18% luminous efficiency (fluorescent tube), 7 am - 6 pm, 5 days per week,	
Setpoint lighting	on/off 500 lx at 0.8 m height in the centre of the room	
Heating system	net heating demand (such as district heating with $\eta = 100\%$)	
Setpoints heating	21°C (5 am - 6 pm)/17°C (6 pm - 5 am)	
Cooling system	-	net cooling demand calculated
Setpoint cooling	-	25.0°C (5 am - 6 pm), 30.0°C (6 pm - 5 am)

	Controlled natural ventilation	Mechanical ventilation and cooling
Ventilation system	single-sided ventilation/sliding-window	supply and exhaust air fan (975 Pa each pressure drop)
Opening geometry	per office: $A_{\text{eff}} = 0.88\text{m}^2$ (fully opened)/ $h_{\text{eff}} = 1.6\text{ m}$	-
Setpoints fresh air supply	500 ppm	12.5 l person ⁻¹

Table 2.3 Overview of the properties of the base case office-type building.

For the first estimation of required effective opening areas as a fraction of the net floor area, a simple graphical design tool developed in the context of the NatVent project is used (van Paassen et al., 1998). The focus is on cooling of buildings, which is a function of internal heat gains, the sun protection system, the thermal mass of the building, the control strategy and the ventilation type (single-sided, cross, stack). For the given office room, the internal loads consist of an average heat gain through people of 4.5 W m^{-2} , office equipment of 7.0 W m^{-2} , lighting of 6.3 W m^{-2} in June and 16.1 W m^{-2} in December, during occupancy time and Istanbul solar radiation climatic data. This is a rather low total load value in the cooling season. The office room has external shading and the storage mass is high.

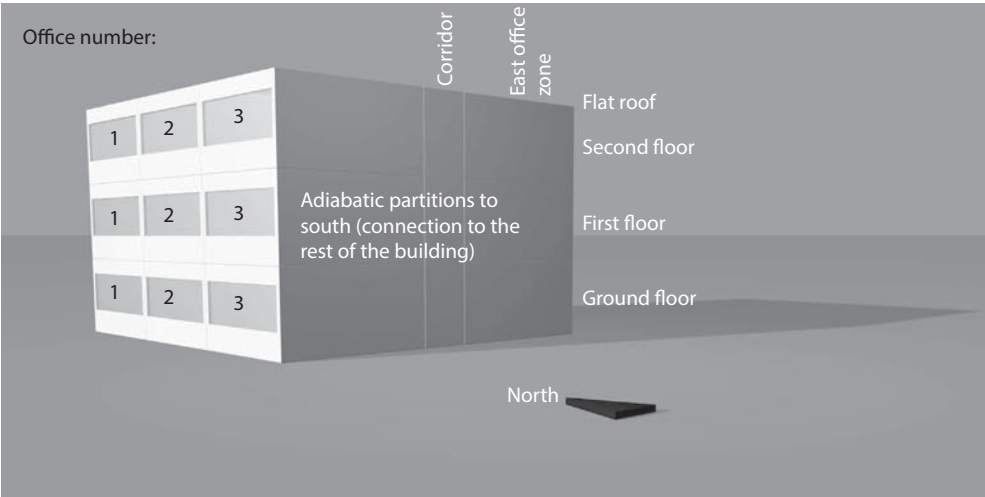


Figure 2.36 Geometrical representation of the simulation model.

2.6 Ventilation potentials

The potentials of different ventilation openings and room geometries are examined first for a constant inside temperature (20°C) and variations of, for example, outside air temperatures. The ventilation potential for these fixed boundary conditions is evaluated for single-sided ventilation in a two-person office room with a floor area of 29.4 m^2 . For cross and stack ventilation, a large open plan office with four people and a total floor area of 71.5 m^2 is taken. The effective opening areas are obtained from the simple graphical pre-design tool of the NatVent study, and corresponding

window tilt angles for pivot-hung windows from the equations of van Paassen are shown in Table 2.4.

h/m	$A_{\text{eff}} = 0.88 \text{ m}^2$ (single) (3% of 29.4 m²) w = 5.0 m → h_{eff} = 0.18 m			$A_{\text{eff}} = 2 \times 0.72 \text{ m}^2$ (cross) (2% of 71.5 m²) w = 5.0 m → h_{eff} = 0.14 m			$A_{\text{eff}} = 2 \times 1.07 \text{ m}^2$ (stack) (3% of 71.5 m²) w = 5.0 m → h_{eff} = 0.21 m		
	$\varphi/^\circ$	$A_{\text{eff}}/A_{\text{win}}$		$\varphi/^\circ$	$A_{\text{eff}}/A_{\text{win}}$		$\varphi/^\circ$	$A_{\text{eff}}/A_{\text{win}}$	
0.40	26.3	44%		20.6	36%		34.3	53%	
0.50	19.7	35%		15.7	29%		24.9	43%	
0.75	12.1	23%		9.8	19%		14.9	29%	
1.00	8.5	18%		7.0	14%		10.5	21%	
1.25	6.5	14%		5.3	12%		8.0	17%	
1.50	5.2	12%		4.3	10%		6.4	14%	
1.75	4.3	10%		3.5	8%		5.2	12%	
2.00	3.6	9%		3.0	7%		4.4	11%	
2.25	3.1	9%		2.5	6%		3.8	10%	
2.50	2.7	9%		2.2	6%		3.3	9%	

Table 2.4 Tilt angle as a function of the window height with constant effective opening area and window width for bottom-hung windows according to van Paassen et al.

Single-sided ventilation

The ventilation potential for single-sided ventilation is evaluated for the two-person office room with a floor area of 29.4 m². It depends on the effective height of the opening (or the height difference of two openings), the discharge coefficient and the effective opening area.

The influence of the wind pressure is very low, because for standard floor heights, the wind pressure increases only slightly over the openings. The rise of wind pressure with height is generally contrary to the thermally induced pressure differences. Local turbulences caused by the wind may increase the air flow to a certain degree. In the calculations for single-sided ventilation (analytical and network simulation) here, only thermally induced pressure differences are taken into account.

The base case effective opening area for the single-sided ventilation of the two-person office described above is 3% of the floor space area, which corresponds to an effective aperture area of 0.88 m² (see Figure 2.37, Figure 2.38 and Figure 2.39).

The three opening configurations' bottom-hung window (case (a)), sliding window (case (b)) and the arrangement with two openings are the basis of the calculations. The bottom-hung window (a) is chosen with a height of 0.40 m following the recommendations of the German Electrical and Electronic Manufacturers' Association (ZVEI) as a popular size in real application.

Opening geometries can be modelled by converting window geometries with tilt angles to effective values (Table 2.4, first column) which then can be used as input for the analytical and AFN simulation calculations.

First the network simulation is validated using available analytical solutions described above. The E+ sizing period design day object was used instead of a weather file to create a steady

state simulation with unchanging external temperatures in each run and the internal temperature was fixed by an ideal loads zone air object with a setpoint of 20°C for heating and cooling. Very good agreement was obtained for all cases (Figure 2.40). The calculated flow rates at 10 K temperature difference correspond to 3 ACH (air change rate) for the bottom-hung window (case (a)), 6 ACH for the sliding window (case (b)) and 12 ACH for the two openings (case (c)).

As high natural ventilation air exchanges are mainly needed when the temperature differences between room and ambient air are rather low, for example, in summer, Figure 2.41 shows the calculated air flow rates for small temperature differences up to 5 K. In the most critical case of a low height bottom-hung window (case (a)) and a deep room of nearly 10 m, the air exchange rates drop to below 2 and are no longer sufficient for passive cooling. The effect of opening geometry, of the effective opening area, the room depth and the temperature differences ($\Delta T = T_{in} - T_{out}$) on achievable flow rates were again analysed for the three different opening arrangements, here with 5 different effective opening areas and 10 different temperature differences. Subsequently the room depth (assuming an ideal air mixing behaviour) of the type office room is varied to determine the achievable air exchange rates. A change of effective opening area for a given opening configuration is obtained for a fixed effective opening height and variable width. As the volume-flow calculation method is independent of the room depth, the air exchange rate (ACH) is calculated as a direct function of the room air volume.

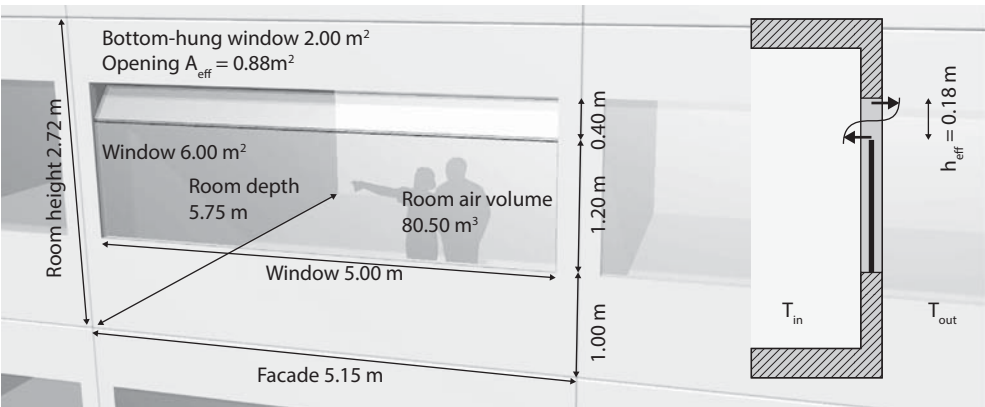


Figure 2.37 Base case (a) for single-sided ventilation with partially opened tilted bottom-hung window with an effective opening height of 0.18 m.

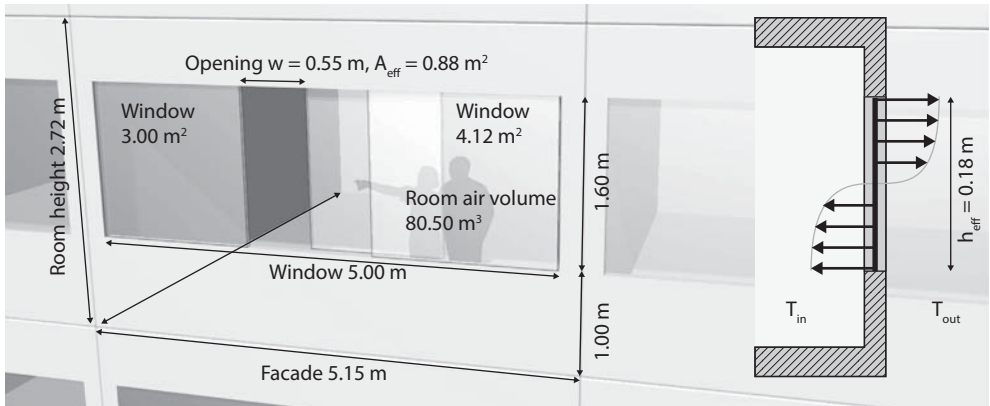


Figure 2.38 Base case (b) for single-sided ventilation with partially opened horizontally sliding window with an opening width of 0.55 m.

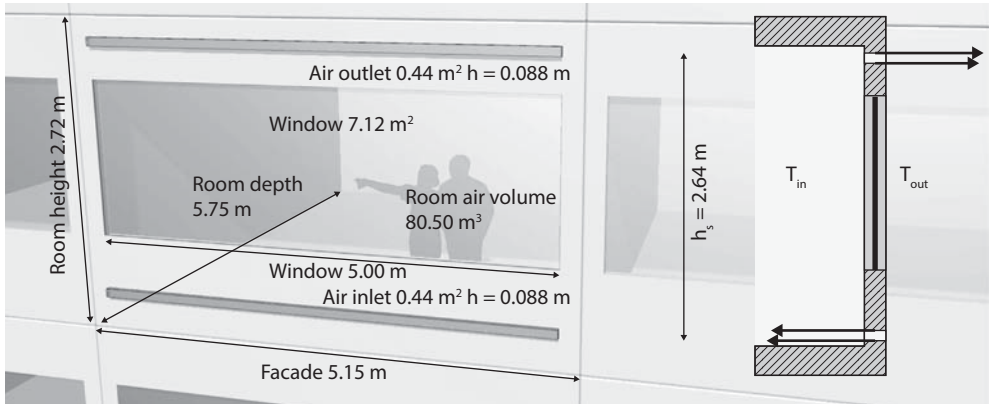


Figure 2.39 Base case (c) for single-sided ventilation with two openings.

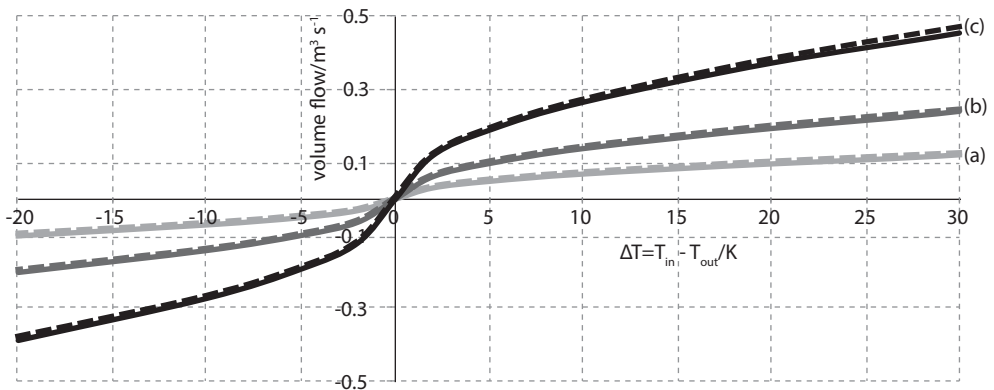


Figure 2.40 Buoyant volume flow as a function of temperature difference for the three opening configurations; solid lines from EnergyPlus simulations, dotted lines are analytical solutions.

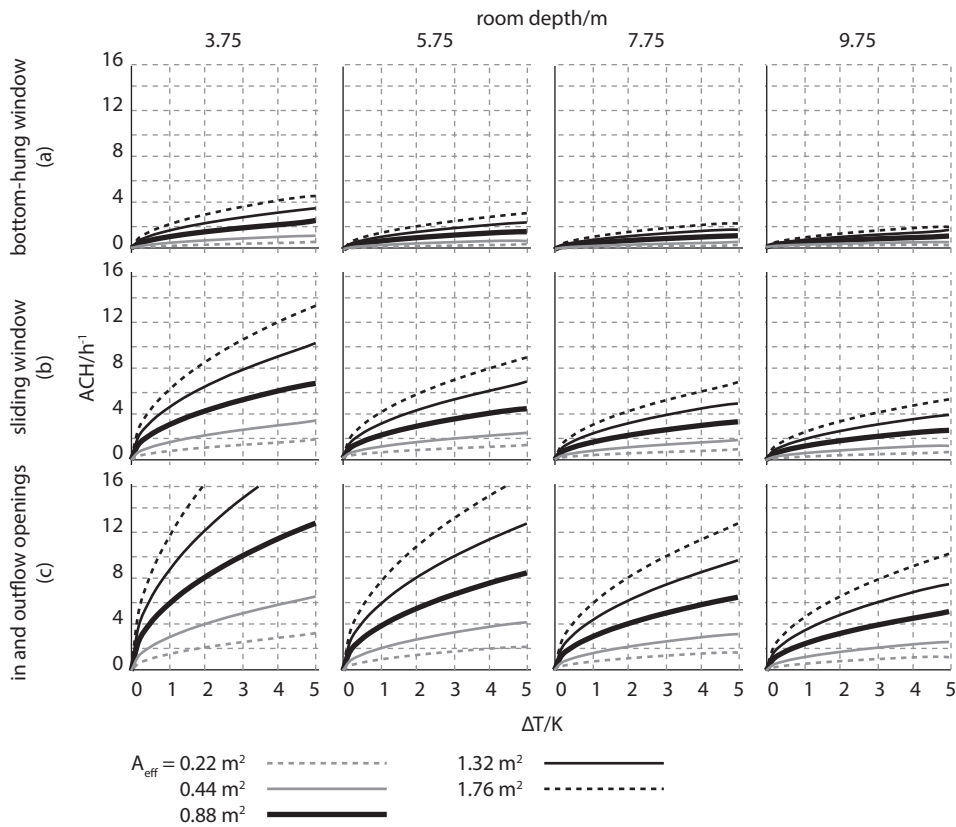


Figure 2.41 Analytical results of the variant matrix single-sided ventilation with small temperature differences between room and ambient air. The base case is a room depth of 5.75 m with an effective opening area of 0.88 m².

Controlled single-sided ventilation with dynamic boundary conditions

For a dynamic analysis of the potential for single-sided buoyancy-driven ventilation, the entire system with controls has to be simulated because the ventilation is not driven just by external conditions, but by an interaction of internal and external conditions, i.e. ambient and indoor temperature. The ventilation control strategy will be described later for the combined air flow and thermal simulations. Figure 2.41 shows how the achievable air change rates for buoyant flow depend on the opening geometry (effective opening height and width) of the chosen design ((a), (b) or (c)) with a constant maximum effective opening area for all three designs. In winter, ventilation is used solely to limit the CO₂ levels.

Air exchange rates increase with higher passive cooling demands in Istanbul and Turin compared to Stuttgart. The bottom-hung windows only give a limited air change rate, which is not sufficient for passive cooling in most climates.

Cross ventilation

The cross ventilation potentials are calculated for the entire width of a floor as an open office with no internal partition walls (71.5 m² in total) by combining two small offices and the corridor in-between to one larger zone. By the use of opposite arrangement of the openings (east and west facades), these cases may be suitable for deeper spaces. There are four different strategies modelled. The **purely wind-driven (d) strategy** has two bottom-hung windows at the same height of opposite facade orientation. The base case has 2% effective opening area relative to the floor area of the two offices plus the corridor (opening geometry can be designed according Table 2.4, second column). The second approach models a **purely buoyancy-driven (e) strategy**. Here again two bottom-hung windows with opposite facade orientation are modelled but on different height, one close to the ceiling level and one close to the floor level with no wind pressure difference. The effective opening areas are dimensioned as for the single-sided buoyant ventilation (3% of the floor area, Table 2.4). The third case for cross ventilation gives an insight into **combined buoyancy and wind pressure (f) effects**. The inlet opening (low level) is mostly modelled on the windward side, thus adding thermal forces and wind forces. The opening area corresponds to case (d). Finally cross ventilation scenario with **external chimney (g)** is modelled, which again is purely buoyancy driven but with a higher level difference than in case (d). Here a bottom-hung window is the air inlet and opposite, 5 metres higher (height difference corresponds to mid of supply-opening level to exhaust level), an external chimney provides the outlet. The cross section of the chimney is 1% of the entire floor area.

Table 2.5 summarises the calculated air changes for fixed boundary conditions. Cross ventilation has generally quite high air exchange rates, which vary between 5.0 and 8.6 h⁻¹ for all base case scenarios. Again a good agreement was obtained between simulation and analytical calculation. The interpretation of the table results works as follows for the wind-driven cases (d): the air exchange in the base case at 3 m s⁻¹ wind speed is 8 h⁻¹. The maximum air exchange with the east-/west-orientated building is achieved for a south-westerly wind direction. The wind speeds are meteorological wind speeds corresponding to the situation in open country at 10 m height. In the city centre, the air exchange rates reduce from 8.0 to 4.7 h⁻¹ on the first floor level due to the terrain. Due to the increasing wind speed with altitude, the air change rate increases with increasing height of the openings on the various floors of the building. Without local shielding by adjacent buildings, the air change rates increases from 8.0 to 10.4 h⁻¹.

Cross ventilation with dynamic boundary condition

Simulations were also performed to obtain the frequency distribution of air changes in cross-ventilated buildings with openings dimensioned according to case (d). The simulated case is the purely wind-driven case with no heating nor cooling system applied and has two windows on the east and west facades with an effective opening area of 0.72 m² each. The opening is set to constantly 100% open. Hourly values are calculated for the base case in the 1st floor in urban terrain wind profile, using wind pressure coefficients from literature (see Table 2.1) with a shielding effect with adjacent buildings of equal height.

In general the air change rates are high throughout most hours of the year for the climates in Istanbul and Stuttgart, less so in Turin, increasing on the open country compared to urban sites and decreasing for the lower storeys in a building.

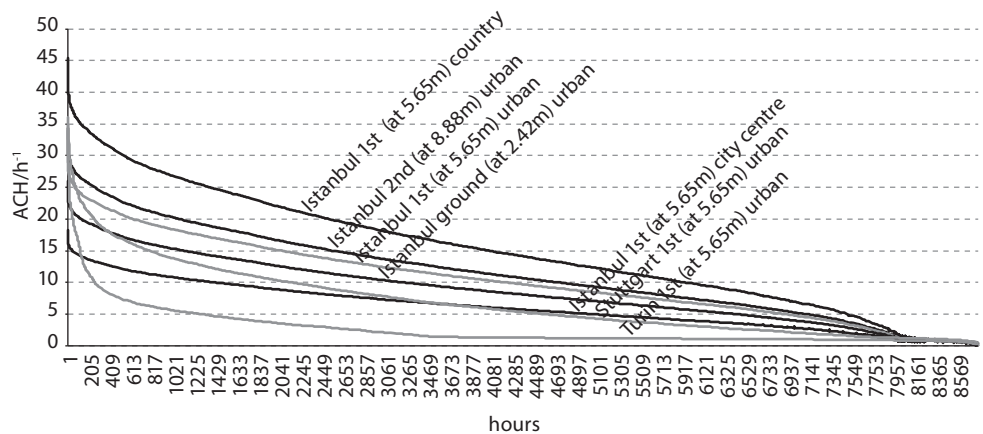


Figure 2.42 Hourly ACH frequency distribution for wind-driven cross ventilation depending on the floor level, terrain and local annual wind data.

2.7 Thermal comfort and energy savings in office rooms with controlled natural ventilation

Apart from the previously discussed air exchange rates, which directly influence air quality, thermal comfort is a crucial indicator to evaluate natural ventilation concepts. If thermal comfort can be guaranteed, then significant cooling and ventilation energy savings can be obtained.

In the following calculations with the base case for controlled, natural ventilation is defined for an opening type (b), i.e. single-sided ventilation with a sliding window with an effective opening area of 0.88 m^2 . The single-sided ventilation is buoyancy driven and wind pressure is not relevant here. To cover heating loads, an ideal system is assumed. The setpoint temperature of the heating system corresponds to the lower operative comfort temperatures for naturally ventilated buildings Category II. In winter operation that is 20.8°C .

A mechanical reference system with active cooling and ventilation (fans) is used to determine the annual energy savings for the controlled natural ventilation by comparing the energy consumptions. The mechanical ventilation system has no heat recovery. The energy required for distribution of the cooling and heating energy in buildings is not considered (pumps, etc.). The setpoint temperature of the heating system is 21°C mean air temperature. The setpoint mean air temperature for cooling is 25°C during the period of operation, which is the maximum temperature of EN 15251 Category II minus 1 K for air-conditioned offices and 30°C outside the operating hours.

Table 2.5 Air change rates for cross ventilation for a range of temperature differences and wind speed. Further columns show the influence of wind direction, terrain, floor height and surrounding buildings and have to be compared with the base case scenario.

Cross ventilation strategy	1)	$\Delta T/K$	ACH	$u_{met}/m\ s^{-1}$	ACH	Wind direction/ $^{\circ}$	ACH	A_{eff}/m^2	ACH	Terrain ²⁾	ACH	Floor	ACH	Adjacent building height	ACH	ACH	
																	*
Case (d) Wind only $A_{eff} = 2 \cdot 0.72\ m^2$ $h_S = 0.00\ m$		0.0				90 E	7.2	7.2									
						135 SE	8.0	8.0									
						180 S	0.0	0.0							none	10.4	10.4
						185 SSW	2.7	2.7							half	9.8	9.7
						195 SSW	4.6	4.6							equal	8.0	8.0
			3.0	8.0	8.0	225 SW	8.0	8.0	0.72	8.0	8.0						
Wind only $A_{eff} = 2 \cdot 0.72\ m^2$ $h_S = 0.00\ m$						270 W	7.2	7.2	0.36	4.0	4.0						
										0.18	2.0	2.0					
										0.09	1.0	1.0					
										0.05	0.5	0.5					
Case (e) Buoyancy only $A_{eff} = 2 \cdot 1.07\ m^2$ $h_S = 2.64\ m$																	
			0.0	0.0	0.0				1.07	5.4	5.4						
			1.0	3.8	3.8				0.54	2.7	2.7						
			2.0	5.4	5.4				0.27	1.4	1.4						
			3.0	6.7	6.6				0.13	0.7	0.7						
			4.0	7.7	7.7				0.07	0.3	0.3						
			5.0	8.7	8.6												

Cross ventila- tion strategy	1)	ΔT/K		ACH		u _{met} /m s ⁻¹		ACH		A _{eff} /m ²		ACH		Wind direction/°		ACH		Terrain 2)		ACH		Floor		ACH		Adjacent building height		ACH	
		*	**	*	**	*	**	*	**	*	**	*	**	*	**	*	**	*	**	*	**	*	**	*	**	*	**	*	**
Case (f) Buoyancy and wind A _{eff} = 2 · 0.72 m ² h _S = 2.64 m																													
Case (g) Chimney A _{stack} = 0.72 m ²																													

Control strategies for naturally ventilated offices

Control strategies for natural ventilation serve to ensure acceptable air quality, compliance with the limits of comfort and a best possible energy efficient building operation. Energy efficiency is achieved by means of passive cooling and by optimal control during winter operation. A simple EMS (Energy Management System) control algorithm could consist of the following three components:

1. Rules on indoor air quality with CO₂ sensors:
If the CO₂ level exceeds the content of outdoor air by 500 ppm, intense ventilation occurs during operating hours. Intense ventilation is defined here as brief opening of 50% of effective opening areas (to avoid too high-temperature jumps) for a few minutes. In the simulation that corresponds to a simulation time step of 5 minutes.
2. Rules on thermal comfort:
If the mean room air temperature is higher than the outside air temperature and the operative room temperature exceeds the lower limit of comfort, Category II ventilation is on. The ventilation setpoint temperature is given by the operative temperature level exceeding the lower limit of comfort. For different seasons the following algorithms are implemented:
 - In the heating season with a running mean outside temperature of less than 10°C, the ventilation setpoint temperature at day and night is increased by 4 K to minimise the risk of overcooling with high inside to outside temperature differences.
 - In the intermediate season with a running mean outside temperature of more than 10°C and less than 15°C, the ventilation setpoint temperature at day and night is increased by 2 K and with a running mean outside temperature of more than 15°C and less than 20°C by 0.5 K to minimise the risk of overcooling, resulting in unnecessary heating.
3. Rules to reduce the risk of drafts:
The opening areas are modulated as a function of the temperature difference between indoor and ambient air. The lower limit of the temperature difference for the maximum ventilation opening factor is 5 K and the upper limit for the minimum aperture is 20 K. This rule serves to reduce the discomfort due to ambient outside air and thus the risk of drafts. Besides, the modulation also reduces the risk of overcooling in the heating season.

2.8 Weekly simulations with dynamic boundary conditions

Design week simulations give insight into the dynamic functioning of natural ventilation systems and also serve to analyse the opening control. Seven simulations within the cooling period between 22 and 29 June were performed with the climate data from Istanbul. Considered are the operative room temperatures in relation to the limits of comfort and the ambient temperatures, the CO₂ level and the air exchange rates.

Mechanical ventilation

The first weekly profile was simulated for mechanical ventilation for indoor air quality with minimal air change rates of 12.5 l s⁻¹ person⁻¹ according to EN 13779 in a two-person office. There is no cooling system available, which results in high operative room temperatures.

The results show a room air temperature for a room without shading above 50°C, with shading around 40°C, which is far above all admissible comfort temperatures for both mechanically and naturally ventilated offices. The carbon dioxide content of the indoor air with the demand-

driven system with $12.5 \text{ l s}^{-1} \text{ person}^{-1}$ has a maximum of about 310 ppm, which is mostly below the standard value category IDA 1 ($\leq 300 \text{ ppm}$ according to EN 13779) and thus may be classified as low.

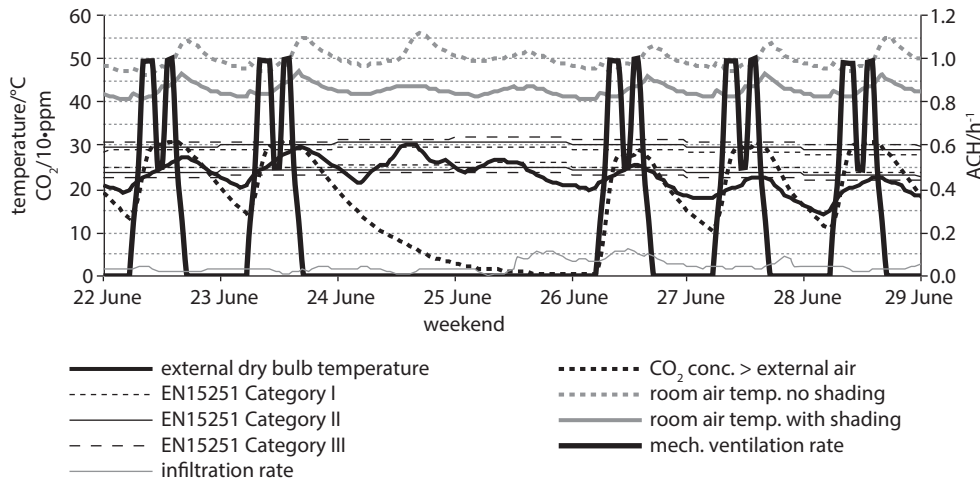


Figure 2.43 Simulation results of the mechanical ventilation case and no active cooling 1st floor office in Istanbul.

2.9 Natural single-sided ventilation with sliding windows

For single-sided natural ventilation, different scenarios are possible for ventilation control. The first scenario analysed is uncontrolled natural ventilation, which here means that the opening area is open at all times to 100%, which results in high air exchange rates and often too low indoor temperatures even in summer. The air exchange rates are in a range up to about 6 h^{-1} . The CO_2 concentrations are very low due to the high air exchange with a maximum value of about 100 ppm above ambient level.

A second scenario is the uncontrolled night-time ventilation, which is defined with openings open at 100% at night and closed during occupancy. Here the temperatures are still unacceptably high, because daily heat loads are not removed by ventilation and the thermal mass is not sufficient. Due to the closed window during occupancy times, unacceptable, too high CO_2 concentrations are obtained (up to 1800 ppm above external levels). Only the infiltration through the building envelope of approximately 0.1 h^{-1} is not enough.

Daytime ventilation is defined as the period in which offices are occupied (weekdays only). The openings here are open during the day at 100% and closed at night. The pure daytime ventilation is suitable for building ventilation to supply fresh air to the rooms. The CO_2 concentrations are classified as low due to a continuous air exchange during the occupancy and a value below 100 ppm. The operative room temperatures are often outside the comfort zone in accordance with EN 15251 because the operative temperature is always above the outdoor temperature, which demonstrates the limits of natural daytime ventilation especially at temperature peaks.

The next scenario examined is a temperature-controlled scenario with fixed setpoint control. Here the ventilation openings are opened when the room temperature is greater than the

ambient temperature and the room temperature exceeds 20°C. The temperature limits (Category 2) are far less exceeded than in the two cases before. CO₂ levels are constantly low. This control seems to be a good choice, but depends on climate, the season, the construction and the ventilation potential of the design.

To overcome these limits, in the final case ventilation, openings are opened when the room temperature is greater than the ambient temperature and the adaptive, operative room temperature setpoint is exceeded according to the definitions in Control strategies for naturally ventilated offices on page 92.

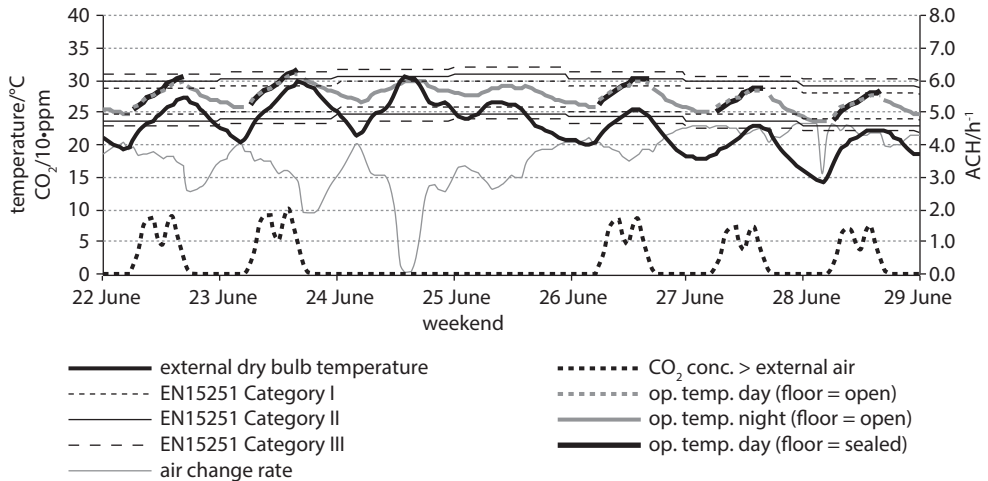


Figure 2.44 Simulation results of the adaptive setpoint control, single-sided natural variation with sliding window 1st floor offices Istanbul.

The adaptive setpoint temperature control for night and day ventilation ensures that the upper and the lower comfort limits are rarely exceeded. This control is more robust in terms of adapting the flow rates depending on the climate, season, building construction and on the ventilation potentials of the opening design. In the example, the exposition of the thermal mass of the concrete floor reduces peak temperatures by about 0.5 - 1.0 K. To have a higher contribution of thermal mass, the designer must take a closer look on the diurnal capacity of the mass as a heat sink and also on the achievable convection coefficient for heat exchange of the mass depending e.g. on the orientation of the mass, the local flow speed, surface roughness and temperature differences.

Case description	Mech. ref.						Controlled natural ventilated						
	Energy /kWh m ⁻² a ⁻¹						Ventilation ¹⁾ ACH/h ⁻¹			Thermal comfort T _{in} > EN15251 Cat. II			
	Climate ³⁾	Heating ⁴⁾	Cooling ⁴⁾	Ventilation ⁵⁾	Lighting ⁵⁾	Heating ⁴⁾	Jul - Aug ²⁾			/h /degree h			
							Mean ⁶⁾	Max ⁷⁾	Min ⁷⁾	Mean ⁶⁾	Mean ⁶⁾	Max ⁷⁾	Min ⁷⁾
Base case building (see)	S	10	22	4	22	8	2.8	2.9	2.6	16	2	12	0
	T	6	36	4	21	5	3.6	3.7	3.5	80	60	101	18
	I	2	44	4	20	2	3.6	3.8	3.4	188	115	219	25
No external shading devices on windows	S	9	39	4	16	7	3.7	3.8	3.6	141	123	257	37
	T	5	59	4	13	4	4.2	4.3	4.2	299	462	596	292
	I	2	69	4	11	1	4.2	4.4	4.0	560	893	1130	595
Dimmed lighting to 500 lx at centre desk level	S	19	13	4	14	11	2.6	2.7	2.4	5	0	0	0
	T	8	32	4	13	6	3.4	3.5	3.3	40	24	42	6
	I	3	39	4	12	2	3.5	3.7	3.3	102	40	83	5
50% Glazed area (only 4m ² instead of 8m ² per room)	S	7	19	4	32	5	2.5	2.6	2.3	10	1	2	0
	T	4	32	4	32	3	3.3	3.4	3.2	33	19	37	4
	I	1	38	4	30	2	3.4	3.5	3.2	76	27	60	4
Building is N-S orientated (instead of E-W)	S	9	21	4	22	8	2.7	2.8	2.5	15	2	9	0
	T	5	35	4	22	4	3.5	3.6	3.4	67	47	99	9
	I	2	42	4	19	2	3.5	3.7	3.3	147	70	116	18
Fresh air supply only controlled by CO ₂ sensor (ON 7 am - 6 pm)	S	7	26	2	22	8	0.2	0.3	0.1	1599	10355	12454	6992
	T	4	38	2	21	4	0.2	0.3	0.1	1841	14931	17360	10930
	I	1	45	2	20	1	0.2	0.3	0.1	1999	18076	20757	13680
No ventilation system (only infiltration)	S	4	29	0	22	4	0.1	0.2	0.0	1782	13908	16727	9263
	T	2	40	0	21	2	0.1	0.2	0.0	2052	19233	22629	13758
	I	1	47	0	20	1	0.1	0.2	0.0	2296	22751	26485	16643
½ Opening area (reduced 50% width - h _{eff} not effected)	S	as base case			22	8	2.0	2.0	1.9	56	20	38	0
	T	as base case			21	4	2.1	2.2	2.1	312	452	642	242
	I	as base case			20	2	2.2	2.3	1.9	597	843	1276	413
Bottom-hung window with h _{eff} = 0.178 m (see Figure 2.37)	S	as base case			22	8	1.4	1.5	1.2	282	248	617	26
	T	as base case			21	4	1.4	1.6	1.3	603	1388	1974	733
	I	as base case			20	2	1.5	1.7	1.2	898	2166	3186	1106
No night-time ventilation (OFF 6 pm - 7 am)	S	as base case			22	8	1.4	1.5	1.3	227	110	158	39
	T	as base case			21	4	1.5	1.5	1.4	565	1056	1373	651
	I	as base case			20	2	1.5	1.6	1.3	879	2089	2768	1300

Case description	Mech. ref.					Controlled natural ventilated							
	Energy /kWh m ⁻² a ⁻¹					Ventilation 1) ACH/h ⁻¹ Jul - Aug 2)			Thermal comfort T _{in} > EN15251 Cat. II				
	Climate 3)	Heating 4)	Cooling 4)	Ventilation 5)	Lighting 5)	Heating 4)	Mean 6)	Max 7)	Min 7)	Mean 6)	Mean 6)	Max 7)	Min 7)
Sealed floor like in mech. vent. reference case (see)	S	as base case		22	8	2.8	2.9	2.6	36	14	26	0	
	T	as base case		21	5	3.6	3.7	3.5	120	108	167	36	
	I	as base case		20	2	3.6	3.8	3.5	256	214	348	67	
Two openings vent. with height difference of h _s = 2,64 m (see Figure 2.40)	S	as base case		22	10	3.7	4.0	3.4	8	1	3	0	
	T	as base case		21	7	5.3	5.6	5.1	21	6	19	0	
	I	as base case		20	2	6.8	8.8	5.5	26	8	19	0	
Fresh air supply with constant 1.5 ACH (ON 7 am - 6 pm)	S	25	16	5	22								
	T	16	34	5	21								
	I	10	44	5	20								
Cooling setpoint at 26/30°C (instead of 25° day/30° night)	S	10	19	4	22								
	T	6	33	4	21								
	I	2	41	4	20								
Setpoints = limits Cat. II for nat. vent. buildings	S	10	16	4	22								
	T	6	27	4	21								
	I	2	33	4	20								
1) Controlled natural ventilation including infiltration according to building tightness						4) Heating demand							
2) 24-h mean during two hottest months						5) Electricity							
3) ASHRAE IWEK weather data: Stuttgart/Turin/Istanbul						6) 18 office zones mean							
						7) 1 office zone max/min							

Table 2.6 Dynamic simulation outputs for mechanical and natural ventilated/cooled design cases.

2.10 Annual simulations

Dynamic annual simulation gives insight into the overall functionality of controlled natural ventilation throughout the year. Indicators used to evaluate the functionality are the energy consumption compared to a mechanical ventilation and cooling system and especially the compliance with the thermal comfort limits for the users.

The results refer to the office-type building from Table 2.3 and represent the base cases like described before, with sliding windows for natural ventilation. The parameter changes investigated are the building orientation, the building location (Stuttgart/Germany, Turin/Italy and Istanbul/Turkey), the use of dimmers for daylight control of electric lighting, the disabling of external solar shading, the percentage of facade glazing surface, the thermal mass to be activated (exposed or sealed double floors), the setpoint temperatures, the ventilation design (e.g. bottom-

hung or two openings instead of the sliding window) or the ventilation strategy. Minimum and maximum values for naturally ventilated buildings refer to the 18 office zones simulated and their different thermal behaviour in respect to their window orientation, exposure (roofs, north facade, ground slab) and the different infiltration rates depending on their floor level, while the mean value represents all 18 office rooms.

Energy consumption of air-conditioned office rooms

The evaluation of the reference office building with mechanical cooling and ventilation only considers the energy consumption, which can be compared with the consumption of controlled natural ventilation in order to evaluate energy savings. An evaluation of thermal comfort and indoor air quality is not necessary, because the setpoints are always reached with the mechanical system.

The low heating loads are due to the high-quality building design following the German standard EnEV 2007 with low U values, the absence of thermal bridges, the high solar transmittance, the relatively high internal heat gains and the good building airtightness.

The simulation results in Table 2.6 show that:

- External, controlled sun shades are very effective in reducing the solar heat gains, and hence the cooling loads from 39, 59 and 69 kWh m⁻² a⁻¹ without shading for Stuttgart, Turin and Istanbul down to 22, 36 and 44 kWh m⁻² a⁻¹ with shading.
- Daylight controlled dimming of electrical lights with a setpoint of 500 lx on the desk level strongly reduces the electricity consumption for lighting from 20 to 22 kWh m⁻² a⁻¹ for the base cases to 12 - 14 kWh m⁻² a⁻¹, the internal heat gains and thus the cooling loads but slightly increase the heating loads.
- An adaptive supply of fresh air with CO₂ sensors reduces the power consumption of the fans for mechanical ventilation from 4 to 2 kWh m⁻² a⁻¹.

Thermal comfort of controlled natural ventilation office rooms

Apart from the energy consumption, thermal comfort criteria play a major role to evaluate the function of controlled naturally ventilated office rooms. The thermal comfort is evaluated with the excess frequency (in hours) and amplitude (in operative temperature degrees hours) of the comfort criteria during office occupation according to EN 15251 Class II. If the amplitude is smaller than the number of hours, then the mean excess is less than 1 K. As expected, the upper comfort limit is usually exceeded in the hottest months July and August. In the base case for Istanbul the comfort range is exceeded in the hottest month of August at about 100 hours on average less than 1 K. A drop below the comfort criteria will not take place throughout the year, since the heating system is correspondingly controlled.

The thermal comfort results for all variations simulated are shown in Table 2.6. Depending on the case, the comfort limit is exceeded more or less often. All cases with less temperature excess than the base case therefore benefit the users in the cooling season. The table indicates which parameters have positive or negative influence regarding the maximum comfort operative temperatures for naturally ventilated buildings.

The thermal comfort results can be summarised as follows:

- With proper design, the use of controlled natural ventilation shows a good functionality. The comfort limits of the EN 15251 Category II are exceeded only very rarely (at best only 0 degree hours per year in Stuttgart and 8 degree hours Istanbul).
- Night ventilation, especially in combination with a heavy building construction, supports the compliance with the adaptive temperature limits significantly.
- The reduction of internal (lighting) and external heat sources (solar radiation) is of crucial importance. This can be achieved through the use of an external sun protection and adaptive light dimming.
- Care must be taken to establish adequate ventilation rates. They are highly dependent on the natural ventilation design. In the base case they are at about 3 - 4 ACH in summer with reasonable comfort results. If the opening width is reduced to 50%, this will reduce the ACH to about 2 h^{-1} – a value, that is not enough to cool passively in hot summer climate (Turin and Istanbul). Also a bottom-hung window with its small effective opening height is not a good design for single-sided ventilation to cool with air exchange rates of only about 1.5 h^{-1} . If the potentials are increased, like with two openings at different heights for buoyant flow, it also reduces the risk of overheating.
- Different climatic conditions for the cases considered have a great influence but may be compensated by design adaptations.
- The orientation of the room affects its functionality. Proper external shading devices must be applied.

Energy consumption for naturally ventilated office rooms

Due to passive cooling and ventilation with controlled natural ventilation, there is no energy consumption for cooling and ventilation. No hybrid strategies were considered.

The potential energy savings depend on the conditions of the mechanically ventilated and cooled offices chosen for comparison. Compared to the air-conditioned office, the energy consumption results for naturally ventilated offices can be summarised as follows:

- Due to the elimination of active cooling and ventilation electrical energy, the overall energy consumption of naturally ventilated spaces is very low.
- A reduction in artificial lighting electricity by daylight-controlled dimming has a positive effect on thermal comfort and a great influence on the total energy consumption of the rooms.
- With the use of CO_2 sensors to control the fresh air supply, the heating energy consumption is very low.
- The non-existence of heat recovery is not important for these high-quality office buildings with a very low winter heat demand.

Energy savings of purely passively cooled and ventilated office spaces are significant. In the considered cases with external shading devices, the reduction of cooling energy demand is between 13 and 22 kWh m^{-2} per year in Stuttgart, 32 and 36 kWh m^{-2} per year in Turin and 38 and 44 kWh m^{-2} per year in Istanbul. In addition, electricity savings of 4 kWh m^{-2} per year occur in the base case, as no fans are required.

Additionally to the 18-zone model, the same building, but with a simple, less reliable, single room zone model, was simulated for three different German locations. In the simulation of the base case, the location is Frankfurt and only the window-facade is treated as a surface to the

exterior. The ceiling, the floor and the internal partitions are modelled with adiabatic properties. A summary of simulation cases for different locations, orientations and lighting control strategies is shown below in Figure 2.45. The variations in the model are indicated below the x-axis. In the model variant 'exposed to outside' also a side wall is treated as an opaque facade area and the ceiling like a flat roof with heat transfer to the outside. The thermal storage properties of the thermal mass are preserved in all variants.

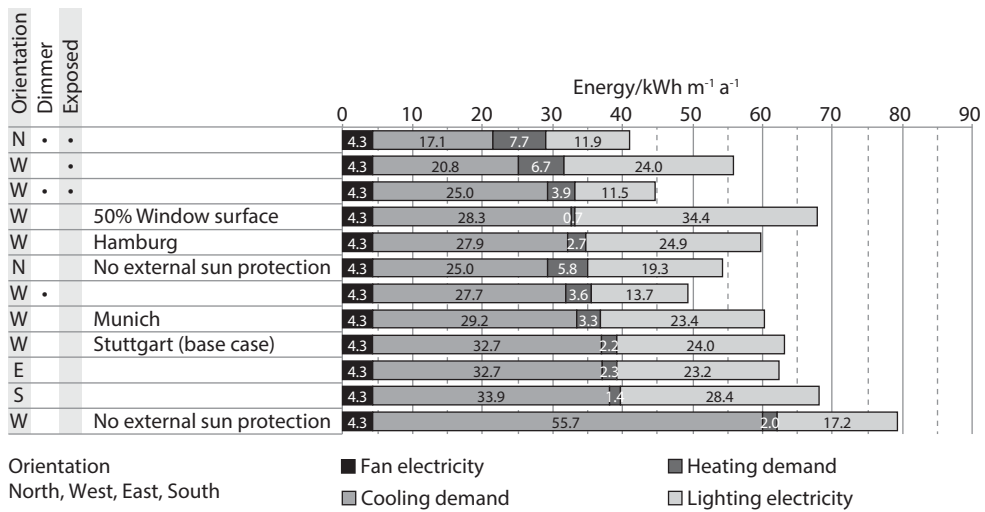


Figure 2.45 Energy demand for heating and cooling as well as electricity for lighting and ventilation for different building conditions in Germany.

Conclusions for natural ventilation

The potential energy savings, indoor air quality and thermal comfort performance of different natural ventilation designs and control strategies was evaluated. A low-energy office building was used for the performance evaluation under different climatic conditions.

It could be shown that cross ventilation and chimney-supported buoyancy-driven natural ventilation result in high air exchange rates, which improve summer comfort in moderate climates, but need control of the openings to prevent against overcooling of rooms. Single-sided ventilation works best, if the openings are high or split at different height levels. For bottom-hung windows with effective opening heights as low as 0.18 m, the air exchange rates are mostly high enough to provide good indoor air quality, but cannot guarantee good thermal comfort conditions in the summer.

Whether thermal comfort can be achieved with natural ventilation strongly depends on the thermal load situation of the room, which again is a function of building heat losses and gains, sun shading performance, internal loads, etc. The higher the loads, the greater the needed effective opening area and height.

2 Part C: Daylighting of buildings

Figure 2.46 (previous page): Interior of the Stadthaus city hall in Scharnhauser Park near Stuttgart, Germany (Photo: Barta).



Figure 2.47 The Cerdanyola residential building near Barcelona, Spain under construction (Photo: POLYCITY).



Figure 2.48 External horizontal shading elements (Photo: Eicker).

Lighting technology deals primarily with the supply of sufficient, glare-free lighting for workplaces and dwellings. Light, however, also assumes the important function of orientation in the interior and enables reference to the exterior. These qualities are supplied predominantly by daylight and contribute crucially to visual comfort. The human eye is optimally adapted to the visible spectral range of solar radiation, so that with short-wave solar irradiance there is a higher luminous efficiency per watt of power than with most types of artificial light. Efficient daylight use thus contributes directly to reducing energy consumption, in particular in administrative buildings.

Energy-saving lighting concepts allow the reduction of installed power for artificial lighting to about 6 - 8 Watt per square metre, while often artificial lighting power levels in office buildings lie between 10 and 20 W m⁻². The required illumination levels strongly depend on the visual tasks.

Type of room	Required illumination /lx	Specific electrical power /W m ⁻²
Parking garage	75	2
Residential dwellings	50	
Adjoining rooms, corridors	100	3 - 5
Stairwell, coat room, restaurant	200	4 - 7
Showrooms, multi-purpose halls, workshops, offices with work stations exclusively near windows	300	6 - 8
Office spaces, school room, kitchen, workshop for fine works	500	7 - 20
Operating room	1000	15 - 30

Table 2.7 Required illumination levels according to EN 12 646 and typical specific power ratings.

Daylight is predominantly used via conventional window openings. Special components are available for glare and sun protection, as well as for light distribution to the depth of the room. Particularly for computer work spaces, glare shielding assumes a crucial role in daylight technology. Glare—shielding systems should be based on the criteria of brightness reduction, light permeability, visual contact outward and light guidance far into the room. Textile anti-glare blinds reduce the brightness by their partially translucent layers, but cannot redirect the light.

The decrease in glare luminance is proportional to the light permeability with absorption and reflection glass. With glare-reducing blinds, the user has more control. Transparency and visual contact outward can be improved by perforation.

Apart from open lamella systems, specially reflecting lamella profiles are also available for the space between panes, consisting of two mirror profiles opposite one another. The light reflected upwards by the lower mirror profile is, with a steep angle of incidence, reflected out from the pane by the mirror in front (sun protection). With a flat angle of incidence, light is channelled inwards by the second mirror located at the back. While the profiles mainly have a sun-protection function with a high sun position, a light-guiding effect can be obtained when the sun is low. Lamellas between non-ventilated glass panes, however, often cause high glass surface temperatures and thus add to the cooling load of the building.

Prism-profile panels use the total reflection of the light either to block it (sun-protection function) or to channel it. They are usually inserted between the panes. Prism systems are

translucent and therefore only used in the skylight area or the overhead area. Sun-protection prisms reduce the sky brightness and thus the glare problem by a factor of 100 even with an overcast sky.

Artificial light sources

Fifty per cent of all artificial light sources are fluorescent lamps, consisting of a glass tube with two wolfram electrodes with an emitting surface at the ends. The gas filling is a mixture of rare gases such as argon or krypton (at a pressure of about 70 Pa) and mercury (at 1 Pa pressure). The mercury atoms emit due to the electric discharge in the ultraviolet region (mainly at 185 and 254 nm) and excite fluorescent material on the tube walls (halogen phosphate or others) to give a continuous visible spectrum. Fluorescent tubes are not well suited for external applications as the light flux decreases at -10°C to 25% of its normal value at 20°C . Compact fluorescent lights are folded fluorescent lights with lifetimes over 10 000 h. Fluorescent lights have photometric radiation equivalents of $50 - 88 \text{ lm W}^{-1}$, clearly lower than daylight. Electrical light bulbs with hot wolfram wires provide a very low light flux of $6 - 16$ lumen per watt power. Halogen lights are slightly better, as the pressure and operating temperatures are higher within the glass tube, and evaporated wolfram from the hot wire no longer deposits on the light bulb walls. Light-emitting diodes (LED) today provide very good luminous efficiencies at a wide spectral range.

Light source	Description	Power <i>P/W</i>	Photometric radiation equivalent <i>k/lm W⁻¹</i>
(Electric) light bulb	Wolfram glowing wire 2800 K	15 - 200	6 - 16
Halogen light	Wolfram wire at 3000 K with higher gas pressure and halogen addition, quartz glass	15 - 200	8 - 20
Fluorescent lamp	Argon/krypton + Hg filled	18 - 58	50 - 88
Compact fluorescent lamps	Argon/krypton + Hg filled	5 - 55	50 - 88
Sodium vapour lamp	Widening of double emission line at 589/90 nm at high pressure	180	150
Light-emitting diodes		40 - 80 mW	20 - 100
LED white	AlInGaN/Phosphor	20 mA, 4V	20
LED green	506, 530, 571 nm, AlInGaAsN	30, 54, 14	
LED red	611, 658 nm, AlInGa	20 mA, 2V	102, 38

Table 2.8 Photometric radiation equivalent of artificial light sources.

Daylighting of interior spaces

The illuminance level during daylight use in the interior is typically between 2% and 5% of the exterior illuminance, corresponding to a mean illuminance of $200 - 500 \text{ lx}$ in the interior.

The relation of the interior illuminance E_i to the horizontal exterior illuminance E_o is termed the daylight coefficient D .

$$D = \frac{E_i}{E_o} \quad (2.64)$$

A room lit by daylight, for which no perceptible difference in brightness between outside and inside is to exist, must have a daylight coefficient of at least 10%, i.e. illuminances of 1000 - 3000 lx or more. The visual sensitivity of the eye is almost constant in relation to further brightness increases. The illuminance E is defined by the ratio of the light flux Φ (lumen) and the illuminated surface area A (m^2).

$$E = \frac{\Phi}{A} \left[\frac{lm}{m^2} = lx \right] \quad (2.65)$$

In residential buildings, the minimum daylight requirement is essentially characterised by the avoidance of the impression of a dark room, which corresponds to daylight coefficients of around 0.9% halfway into the room. For offices in which all workstations are near windows, an illuminance of 300 lx is sufficient. In offices with computer workstations, an illuminance 500 lx is necessary and for open-plan offices or drawing workstations, 750 lx is required.

Example 2.8

Calculation of the mean illuminance for a 3 m high office of surface area $4 \text{ m} \times 5 \text{ m}$, with a window area of 9 m^2 with 40% total transmission (including the framework proportion). The outside available illuminance on the vertical window area on an overcast day is to be 4500 lx.

- Light flux onto the entire window area:

$$4500 \text{ lx} \times 9 \text{ m}^2 = 40\,500 \text{ lm}$$

- Transmitted light flux into the office:

$$40\,500 \text{ lm} \times 0.4 = 16\,200 \text{ lm}$$

- Light flux related to a square metre of office surface:

$$\frac{16\,200 \text{ lm}}{20 \text{ m}^2} = 810 \text{ lx}$$

It is not the quantity of light that is the problem, but rather the unfavourable distribution of the daylight. Purely side-illuminated rooms are characterised by an almost exponential fall in the daylight coefficient, so that glare problems occur near the windows and illuminances are too low with increasing room depth.

For the use of daylight, glazing in the upper window section that enables good deep illumination of the room are particularly favourable. A centrally arranged 1 m high window in a 3 m high room leads to an exponential fall in the daylight coefficient with increasing room depth. An arrangement of the same window height just below the ceiling, on the other hand, increases the daylight coefficient in the room depth. The highest daylight coefficients are obtained with a window front over the entire room height.

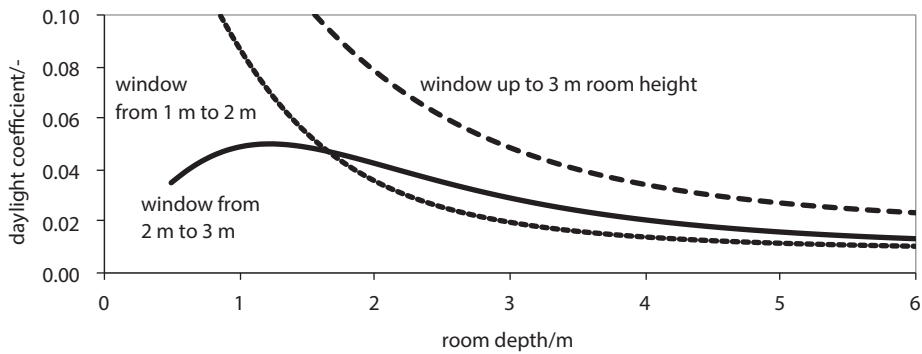


Figure 2.49 Daylight coefficient in a 3 m high side-illuminated room for a floor-to-ceiling window, a centrally positioned window at 1 m height and a window located within the area below the ceiling.

Solar irradiance and light flux

Optical radiation is part of the electromagnetic radiation with wavelengths above 1 nm (upper boundary of Röntgen radiation) up to 1 mm (lower boundary of radio waves). The radiated power P transported by the electromagnetic waves can be calculated using the Poynting vector as a cross product of electrical and magnetic field vectors integrated over a closed surface A surrounding the radiating source.

$$\begin{aligned}\vec{S} &= \vec{E} \times \vec{H} \\ P &= \oint_A \vec{S} \cdot d\vec{A}\end{aligned}\quad (2.66)$$

Due to the high temperature of the sun's surface, solar irradiance covers a wide spectral range from about 0.3 to 4.0 μm . The measured radiated power per square metre is the irradiance G (W m^{-2}) and is a so-called radiometric quantity. As the human eye is only sensitive to a small spectral range of 0.38 - 0.78 μm , the radiometric units have to be converted to eye sensitivity weighted photometric quantities. The light flux Φ (lumen) as a photometric quantity is the sensitivity weighted power. If the light flux is determined per square metre of surface, the illuminance E (lumen m^{-2}) is obtained.

Physiological-optical basics

The human eye is an almost spherical object of 24 mm diameter and about 26 mm length. The pigmented iris functions as a lens with varying opening diameters between 2 and 8 mm, depending on an object's distance and average luminance. The maximum sensitivity of the retina cells during the day is $555 \times 10^{-9} \text{ m}$ (555 nm), i.e. in the green colour area, falling to zero in the short-wave area below 380 nm and in the long-wave area above 780 nm. This spectral area is called visible light. In some international norms, the spectral boundaries for the visible part are rounded to 400 - 800 nm.

About 6.5 million cone cells, responsible for colour vision, are concentrated in the centre of the retina and contain three different dye stuffs, with maximum spectral sensitivities of 419 nm, 531 nm and 558 nm. During illumination, the dyes chemically change (isomerisation) and require around 6 minutes for regeneration. The rod cells, which are responsible for night vision, have a

maximum spectral sensitivity at 496 nm. Rod cells are extremely light sensitive due to their large numbers (120 million) and their characteristic coupling of different cells of up to 32 cells on one nerve end. The pigment regeneration time is around 30 minutes.

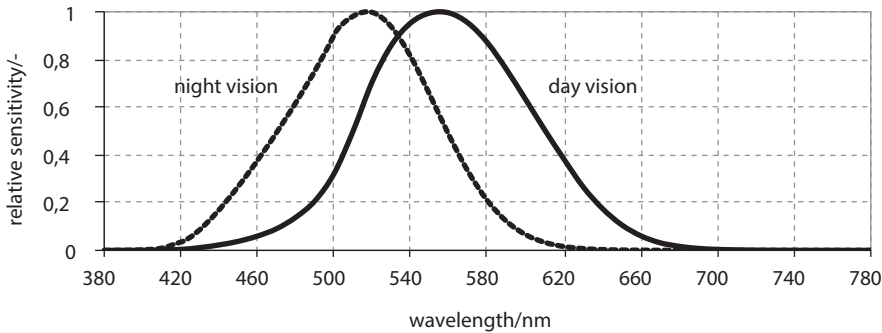


Figure 2.50 Relative spectral sensitivity $V(\lambda)$ of the eye for day vision with retina cells termed cones and for night vision with rods.

Daylight has many medicinal properties. It suppresses the production of melatonin (sleep hormone), affects the circadian system (sleep/wake cycle), helps to reduce accidents and improves performance and the ability to learn. Lack of daylight causes disturbances in metabolism, hormone regulation and various autonomic processes. The artificial lighting of workplaces is an important cause of the so-called sick building syndrome. Working under adverse lighting conditions can result in faster fatigue, headaches and poor concentration.

The hormone melatonin influences the fatigue during the day, and lack of the hormone can result in so-called winter depression. The control of melatonin production is mediated by special receptors called the circadian sensors, which are also in the human retina. The relative spectral sensitivity of the receptors is 380 - 580 nm with a maximum at 460 nm. The sensors react only above an illuminance level of about 2000 lx, which is above the normal artificial lighting levels, but around the range daylight provides near a window.

Photometric radiation equivalent

The relative eye sensitivity $V(\lambda)$ at a given wavelength λ must be known in order to convert an energetic radiant flux Φ_e (index e for energetic) in units of watt into a light flux Φ_v (index v for visual) in lumen, as well as an absolute conversion factor between the units, called the photometric radiation equivalent k_{max} . For each wavelength λ , the light flux $\Phi_{v,\lambda}$ is converted from the spectral radiating power $\Phi_{e,\lambda}$:

$$\begin{aligned}\phi_{v,l} &= k_{max} V(\lambda) \phi_{e,l} \\ k_{max} &= \frac{\text{maximum light flux / lm}}{\text{radiative power / W}}\end{aligned}\quad (2.67)$$

The conversion factor k_{max} results from the definition of the photometric SI unit candela (cd), which describes the light flux Φ per spatial angle Ω (in steradian sr), i.e. the luminous intensity I :

$$I = \frac{\Phi}{\Omega} \quad [cd] = \left[\frac{lm}{sr} \right] \quad (2.68)$$

For the definition of the luminous intensity I in candela, a black cavity emitter (platinum with a melting temperature of 2044.9 K) has historically been used, as its spectral radiant emittance can be calculated using Planck's law of radiation. The electromagnetic radiation is a function of wavelength and has to be integrated over the whole spectrum to obtain the total emitted power. Planck's law is given by:

$$E(\lambda, T) = \frac{2\pi hc^2}{\lambda^5} \left(\exp\left(\frac{hc}{\lambda kT}\right) - 1 \right)^{-1} \quad (2.69)$$

where

λ denotes the wavelength,

h is the Planck constant $6.6260755 \times 10^{-34}$ Js,

c is the speed of light in vacuum $299\,792\,458$ m s⁻¹,

k is the Boltzmann constant 1.380658×10^{-23} J K⁻¹, and

T the temperature of the blackbody in kelvin.

A candela was defined as the light flux per spatial angle, which 1/60 cm² of the surface of the black cavity platinum emitter emits. The emitted power of this surface, calculated using Planck's law of radiation, is 1/673 W.

Today, a candela is defined as the light flux of a monochromatic radiation source with the somewhat smaller power of 1/683 W, which radiates with a frequency of 540×10^{12} Hz, i.e. 555 nm, into a spatial angle of one steradian. Thus, an emitted power of 1 W per steradian results in a luminous intensity of 683 cd or 683 lumen per steradian.

Since the wavelength 555 nm corresponds to the maximum relative eye sensitivity, the maximum photometric radiation equivalent is today $k_{max} = 683$ lm W⁻¹.

Example 2.9

Calculation of the light flux $\Phi_{v,\lambda}$ for a light source of 1 W power with the wavelength $\lambda = 633$ nm (red helium-neon laser) and with $\lambda = 588$ nm (yellow resonance line of the sodium vapour low pressure lamp). The relative spectral sensitivity is $V(633\text{ nm}) = 0.25$ or $V(588\text{ nm}) = 0.77$. Based on Equation (2.67), the light flux is

$$\Phi_{v,633\text{ nm}} = 683 \frac{\text{lm}}{\text{W}} \times 0.25 \times 1\text{ W} = 170.75 \text{ lm}$$

or

$$\Phi_{v,588\text{ nm}} = 683 \frac{\text{lm}}{\text{W}} \times 0.77 \times 1\text{ W} = 525.8 \text{ lm}$$

To determine the light flux and illuminance on a surface of any radiation source, the entire spectrum must be weighted with the spectral sensitivity of the eye.

A single value of the photometric radiation equivalent for a given spectrum (of the sun or a lamp) is obtained by converting the energetic radiant flux for each wavelength into a light flux, integrating it over the visible area of the spectrum and standardising the value on the integrated total radiation flow of the source of light. A source of light is thus characterised by the following radiation equivalent:

$$k = \frac{\Phi_v}{\Phi_e} = \frac{k_{max} \int_{380\text{ nm}}^{780\text{ nm}} \Phi_{e\lambda} V(\lambda) d\lambda}{\int_0^\infty \Phi_{e\lambda} d\lambda} \tag{2.70}$$

For an evenly overcast sky, a radiation equivalent of $k = 115 \text{ lm W}^{-1}$ results based on DIN 5034. The radiation equivalent varies, however, with cloud thickness, the vapour content, the height of the sun, etc., and can be between 90 and 120 lm W^{-1} . The diffuse radiation of the clear sky can assume values over 140 lm W^{-1} , and values between 50 and 120 lm W^{-1} have been measured for the direct component. If one relates the light flux to a square metre of recipient surface, the illuminance E is obtained with the unit lumen per square metre (abbreviated to lux [lx]).

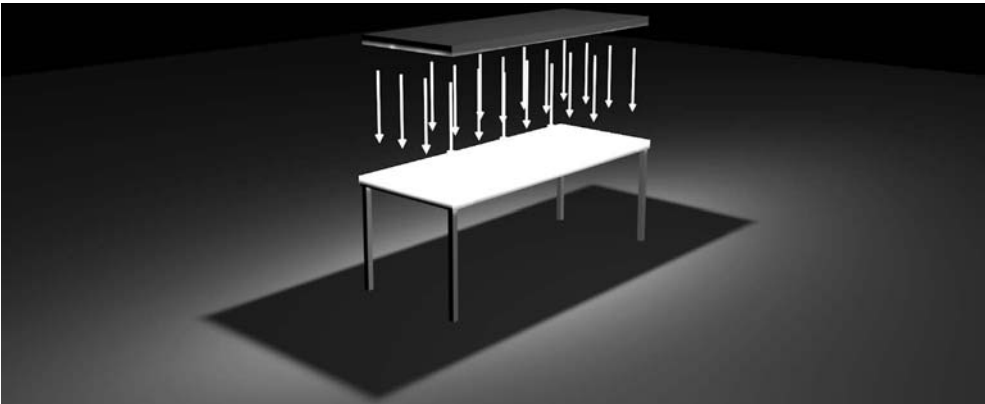


Figure 2.51 Light flux and illuminance on a horizontal surface.

Light source	Photometric radiation equivalent $k/\text{lm W}^{-1}$
Evenly overcast sky	115
Diffuse radiation of the clear sky	140
Direct component of the clear sky	50 - 120

Table 2.9 Photometric radiation equivalent.

An exterior lighting level of 120 000 lx is achieved on a horizontal surface during full exposure to the sun (1000 W m^{-2}) with a photometric radiation equivalent of 120 lm W^{-1} .

$$\Phi_v = k\Phi_e = 120 \frac{\text{lm}}{\text{W}} \times 1000 \frac{\text{W}}{\text{m}^2} = 120000 \frac{\text{lm}}{\text{m}^2} = 120000 \text{ lx} \tag{2.71}$$

The annual average value of the exterior illuminance is around 10 000 lx during the day.

2.11 Luminance and illuminance

The light flux onto a surface and thus the illuminance E_v , defined as light flux per square metre, can be calculated in the simplest case at a given solar irradiance directly from the radiating power and the photometric radiation equivalent. Since luminance distributions rising to the zenith are mostly used in lighting engineering, the light flux onto a randomly inclined surface must be calculated from the contributions of the luminance of different spatial angles of the sky's diffuse radiation. In the interior as well, the light flux onto a recipient surface A_r consists of the total of the light fluxes of radiation-emitting or radiation-reflecting sender surfaces A_s at different spatial angles.

The task now is to integrate the light fluxes received by a surface over all spatial angle areas and thus, to calculate the illuminance E for different luminance distributions. For this, a spatial angle $d\Omega$ is constructed by a two-dimensional surface element dA , which is at a distance r from the sending or receiving surface.

$$d\Omega = \frac{dA}{r^2} \quad (2.72)$$

The unit of the three-dimensional spatial angle is steradian (sr) and corresponds to the definition of a two-dimensional angle in radian (rad), which is constructed by a circular arc with radius r . A simple spatial angle is, for example, given by a surface A_{sp} on a sphere, which is calculated as a function of the sphere radius r and of the height h of the cut-out sphere segment.

$$\Omega = \frac{A_{sp}}{r^2} = \frac{2\pi r h}{r^2} = \frac{2\pi r^2 (1 - \cos\theta)}{r^2} = 2\pi(1 - \cos\theta) \quad (2.73)$$

Here the angle θ describes half the opening angle of the spherical cone. The full spatial angle of a hemisphere with half an opening angle $\theta = 90^\circ$ is therefore 2π , and of a full sphere 4π . The differential spatial angle of the spherical cone is obtained from the above:

$$d\Omega = \frac{dA_{sp}}{r^2} = d(2\pi(1 - \cos\theta)) = 2\pi \sin\theta d\theta \quad (2.74)$$

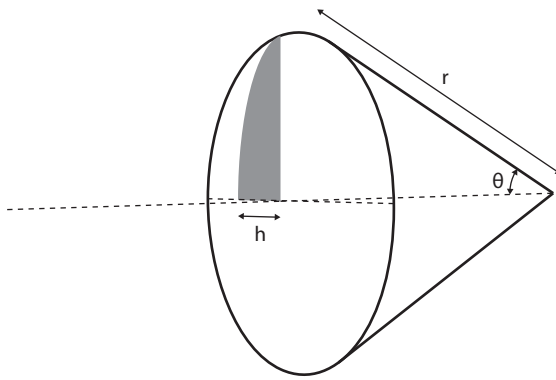


Figure 2.52 Spherical cone spatial angle.

Alternatively, in spherical polar coordinates, the two-dimensional surface of the sphere with radius r is given by the product of the surface sides $r d\theta$ and $r \sin\theta d\gamma$. The angle θ corresponds to the half of the opening angle of the spherical cone. In case of a surface on the sky dome, the angle θ corresponds to the zenith angle θ_z .

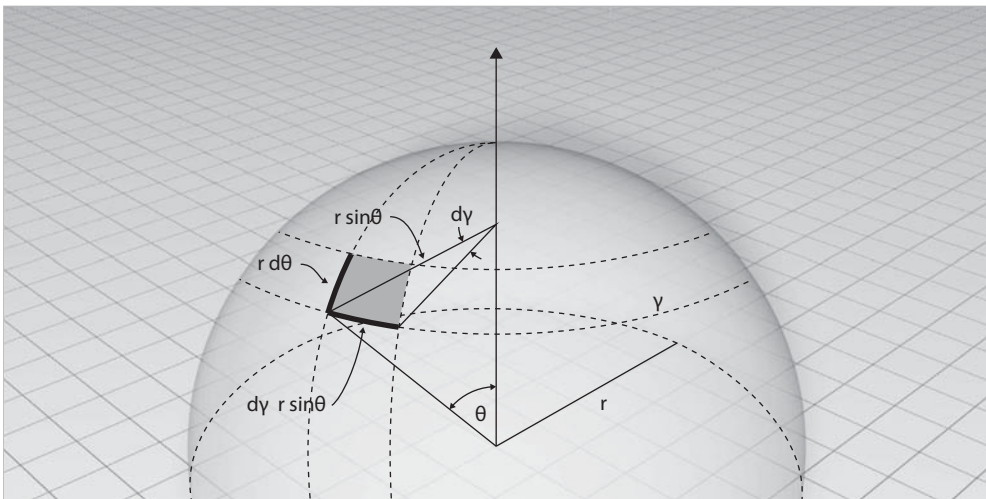


Figure 2.53 Solid angle element of a ring zone on a sphere of the sky.

The solid angle $d\Omega$ is thus:

$$d\Omega = \frac{r \sin\theta d\theta r d\gamma}{r^2} = \sin\theta d\theta d\gamma \quad (2.75)$$

The solid angle of a ring zone on a sphere is obtained from

$$\Omega = \int_{\gamma=0}^{2\pi} \int_{\theta_1}^{\theta_2} \sin\theta d\theta d\gamma = -2\pi(\cos\theta_2 - \cos\theta_1) \quad (2.76)$$

The solid angle of a hemisphere is obtained by integrating from $\theta_1 = 0$ up to $\theta_2 = \pi/2$:

$$\Omega = \int_{\gamma=0}^{2\pi} \int_0^{\pi/2} \sin\theta d\theta d\gamma = -2\pi \left(\cos\frac{\pi}{2} - \cos 0 \right) = 2\pi \quad (2.77)$$

A full sphere requires integration from $\theta_1 = 0$ until $\theta_2 = \pi$, so that the resulting solid angle is 4π .

If the light flux of a luminating or radiation-reflecting surface is to be calculated into a certain spatial angle, then the orientation of the surface relative to the direction of light emission must first be determined. If the radiant emission is not perpendicular to the sending surface element dA_s , but instead at an angle θ_s , then effectively only the smaller surface $dA_s \cos \theta_s$ is visible.

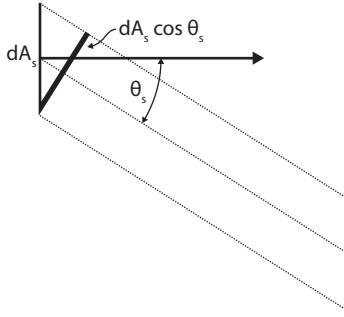


Figure 2.54 Effective size of the radiant two-dimensional element towards the direction of emission θ_s .

The light flux $d\Phi$, which a two-dimensional surface area dA_s sends at the angle θ_s into a receiver spatial angle $d\Omega_r$ is termed luminance L and describes the brightness of the surface.

$$L = \frac{d^2\Phi}{d\Omega_r dA_s \cos\theta_s} = \frac{dI}{dA_s \cos\theta_s} \left[\frac{\text{lm}}{\text{sr m}^2} = \frac{\text{cd}}{\text{m}^2} \right] \quad (2.78)$$

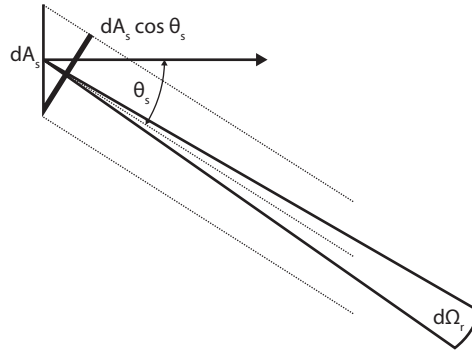


Figure 2.55 Luminance of a two-dimensional area dA_s radiating at an angle θ_s into a spatial angle $d\Omega_r$

The light flux $d\Phi$ (lm) into a spatial angle $d\Omega$ (sr) is called luminous intensity I .

The unit of the luminous intensity I is candela (cd) as the ratio of light flux in lumen divided by the spatial angle in steradian (sr). The unit candela is the basic unit of lighting and is one of eight SI units.

$$I = \frac{d\Phi}{d\Omega} \left[\frac{\text{lm}}{\text{sr}} = \text{cd} \right] \quad (2.79)$$

The luminous intensity is the product from luminance and the surface area dA_s , which radiates with an angle θ_s to the surface normal.

$$I = L dA_s \cos \theta_s \quad (2.80)$$

In many cases, the luminance L is independent of the angle θ_s , i.e. the surface appears equally bright independent of the viewing angle. Such surfaces are termed Lambert radiators. From Equation (2.80), it follows that the luminous intensity I of Lambert radiators must decrease with the cosine of the angle θ_s :

$$I(\theta_s) = I(0) \cos \theta \quad (2.81)$$

Rough, diffusely reflecting surfaces such as gypsum walls and paper behave similarly to Lambert radiators.

In order to obtain the light flux of a sender surface A_s into a receiver spatial angle Ω_r , integration must take place over the differential spatial angles $d\Omega_r$, with consideration given to the effective surface size through the cosine of the angle θ_s between the sender surface and the respective receiver spatial angle element.

$$\Phi = \int_{\Omega_r} \int_{A_s} L dA_s \cos \theta_s d\Omega_r \quad (2.82)$$

The spatial angle $d\Omega_r$ is constructed by the receiving two-dimensional surface dA_r ; for example, the work surface or even the pupil opening surface of the eye. Of course, the receiving surface dA_r does not need to be perpendicular to the spatial angle in each case and therefore, effectively only $dA_r \cos \theta_r$ is available for radiation reception.

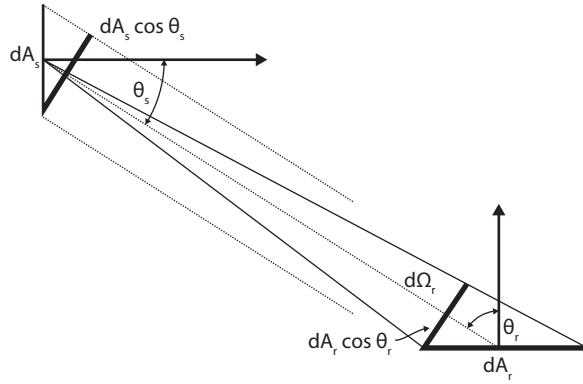


Figure 2.56 Light flux of a radiating two-dimensional surface dA_s onto a recipient surface dA_r .

Thus, the light flux falling from the radiating sender surface A_s onto the receiving surface A_r results in:

$$\Phi_{s \rightarrow r} = \int_{\Omega_r} \int_{A_s} L_s dA_s \cos \theta_s d\Omega_r = \int_{A_r} \int_{A_s} L_s dA_s \cos \theta_s \frac{dA_r \cos \theta_r}{r^2} \quad (2.83)$$

If, conversely, the view is from the receiving surface A_r towards the light flux from the sending surface A_s with its spatial angle Ω_s , the received light flux must correspond to the emitted light flux for reasons of energy conservation, and thus, the Basic Law of Photometry is obtained:

$$\Phi_{r \leftarrow s} = \int_{\Omega_s} \int_{A_r} L_s dA_r \cos \theta_r d\Omega_s = \int_{A_s} \int_{A_r} L_s dA_r \cos \theta_r \frac{dA_s \cos \theta_s}{r^2} = \Phi_{s \rightarrow r} \quad (2.84)$$

The cosine of the angle θ_r refers to the normal of the receiving surface dA_r , and the spatial angle $d\Omega_s$ is constructed from the two-dimensional surface area of the sender dA_s .

The illuminance E_r on the receiving surface dA_r for any sender surfaces dA_s with luminance L_s results from Equation (2.84):

$$E_r = \frac{d\Phi_r}{dA_r} = \int_{\Omega_s} L_s \cos \theta_r d\Omega_s = \int_{A_s} L_s \cos \theta_r \frac{dA_s \cos \theta_s}{r^2} \quad (2.85)$$

Example 2.10

The luminance of the sun is about $1.6 \times 10^9 \text{ cd m}^{-2}$ and its angular extension is $\delta = 0.5^\circ = 8.73 \times 10^{-3} \text{ rad}$. Calculate the illuminance onto a surface normal to the sun's rays and onto a horizontal surface when the sun's elevation is 50° .

Solution: The spatial angle is calculated from half the opening angle given above using Equation (2.74). From this, the illuminance is obtained for normal incidence, i.e.:

$$E_r = \int_{\Omega_s} L_s \cos \theta_r d\Omega_s = \int_0^{0.25^\circ} L_s 2\pi \sin \theta_s d\theta_s = L_s 2\pi - \cos \theta_s \Big|_0^{0.25^\circ} = 1.6 \times 10^9 \frac{\text{cd}}{\text{m}^2} \times 2\pi - \cos \theta_s \Big|_0^{0.25^\circ} = 95698 \frac{\text{lm}}{\text{m}^2}$$

Alternatively, the integrated spatial angle from Equation (2.73) can be directly used.

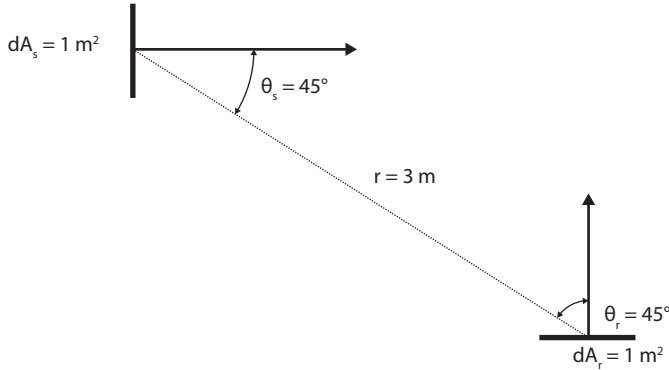
$$E_r = 1.6 \times 10^9 \frac{\text{cd}}{\text{m}^2} 5.98 \times 10^{-5} \text{sr} = 95698 \frac{\text{lm}}{\text{m}^2}$$

For the horizontal surface, the incidence angle is equal to the zenith angle, so that both the differential spatial angle and the incidence angle can be calculated using the zenith angle.

$$E_r = \int_{\Omega_s} L_s \cos \theta_r d\Omega_s = \int_0^{0.25^\circ} L_s \cos \theta_s 2\pi \sin \theta_s d\theta_s = L_s 2\pi \frac{1}{2} \sin^2(\theta_s) \Big|_0^{0.25^\circ} = 61562 \frac{\text{lm}}{\text{m}^2}$$

Example 2.11

Calculation of the illuminance of a 1 m² horizontal work surface illuminated by a 1 m² vertical window with a luminance of 4500 cd m⁻² at an average angle of 45° and 3 m distance.



If the differential surface areas are considered as finite differences, then Equation (2.85) is simplified to:

$$E_r \approx \frac{\Delta \Phi_r}{\Delta A_r} = L_s \cos \theta_r \frac{\Delta A_s \cos \theta_s}{r^2}$$

Both the radiating and receiving surface normal vectors are at 45° to the spatial angle.

$$E_r = \frac{\Delta \Phi_r}{\Delta A_r} = 4500 \frac{\text{cd}}{\text{m}^2} \cos 45^\circ \frac{1 \text{m}^2 \cos 45^\circ}{(3 \text{m})^2} = 250 \text{lx}$$

Luminance and adaptation of the eye

The luminances of sources of light or radiation-reflecting surfaces vary over a very wide range of values.

Light source	Luminance/cd m ⁻²
Sun	1.6×10^9
Moon	2 500
Clear sky	2 000 - 12 000
Overcast sky	1 000 - 6 000
Light bulb	20 000 - 50 000
Compact fluorescent lamp	9 000 - 25 000
Candle flame	7 000
Paper in a well lit office	250
Computer screen	20 - 200
Lower limit of light sensitivity	10^{-5}

Table 2.10 Luminances of sources of light.

The human eye adapts to the mean outside brightness L_o in the visual field. This is termed adaptation and takes place via modification of the pupil surface A_p , which expands with falling brightness from 2 mm diameter to a maximum of about 8 mm (Hentschel, 2002). The pupil surface, which varies as a logarithmic function of the luminance, enables adjustment of the penetrating light flux by a factor of 16.

The luminance reaching the eye and striking the retina is let through with transmittance τ_p . Brightness sensitivity is determined by the number of photons that strike a two-dimensional element of the retina, i.e. by the illuminance on the retina. This is obtained from the mean luminance of the visual field, which is obtained by the spatial angle $d\Omega_s$ of the pupil surface A_p at a distance r between the lens and retina $d\Omega_s = dA_p / r^2$.

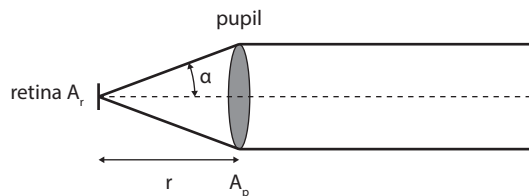


Figure 2.57 Cross section of the eye with distance r between the lens of the pupil and retina.

$$E_{retina} = \frac{d\Phi}{dA_r} = \int_{\Omega_s} L_o \tau_p \cos \theta_r d\Omega_s = \int_0^{2\pi} \int_0^\alpha L_o \tau_p \cos \theta_r \sin \theta_r d\theta_r d\gamma = L_o \tau_p \pi \sin^2 \alpha = L_o \tau_p \pi \frac{d^2}{4r^2} = \text{const } L_o A_p \quad (2.86)$$

Besides the opening area of the pupil (A_p), which depends on the adaptation status of the eye, the brightness sensitivity is determined solely by the luminance striking the eye. The luminance is thus the most important quantity in lighting engineering.

Although equal light fluxes produce equal brightness, to establish a brightness scale is a complex psychophysical process. If the difference between the luminance level L and the luminance level L_a to which the eye is adapted, is approximately between $0.5 \times L_a$ and $2 \times L_a$, the brightness B is proportional to the logarithm of the luminance flux (Fechner's law). Brightness can then be defined by the relation:

$$B = c \int_{L_1}^{L_2} \frac{dL}{L} = c(\ln L_2 - \ln L_1) \quad (2.87)$$

Luminance is now mostly used rather than brightness, although the eye can detect equal brightness quite reliably. However, to establish a quantitative scale of brightness that determines, for example, when one surface is twice as bright as another, is much more difficult.

Luminance of light reflecting surfaces

If a surface reflects the incident light flux on its surface A_r with a reflection coefficient ρ , the luminance of this surface can be easily calculated for diffuse reflection. The surface sends out a reflected luminance into the half space as a sending surface.

$$E_e \rho = \rho \frac{d\Phi_e}{dA_e} \quad (2.88)$$

For a Lambert radiator with constant luminance radiating into the half space, one obtains:

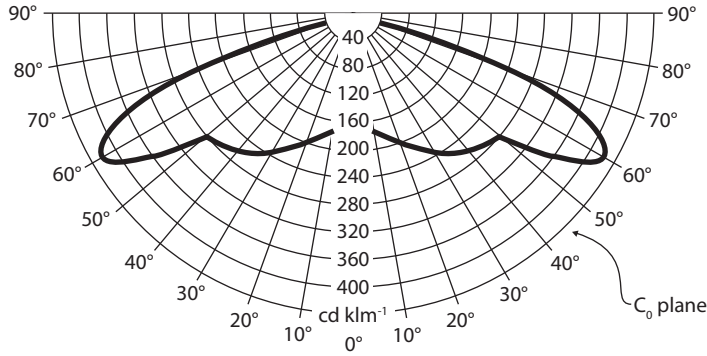
$$L_{\text{reflected}} = \frac{E_e \rho}{\int_0^{2\pi} \int_0^{\pi/2} \cos \theta_s \sin \theta_s d\theta_s d\gamma} \quad (2.89)$$

with

$$\int_0^{2\pi} \int_0^{\pi/2} \cos \theta_s \sin \theta_s d\theta_s d\gamma = 2\pi \left. \frac{\sin^2 \theta}{2} \right|_0^{\pi/2} = \pi$$

Distribution of the luminous intensity of artificial light sources

The luminance of lamps or lights is rarely constant in all directions in space. Luminous intensity distributions are indicated by the manufacturers in polar diagrams in different sections. Usually the absolute luminous intensity in candelas refers to a fixed lamp light flux.



Light distribution curve in the C_0 plane, which is perpendicular to the radiating surface (in candelas per Kilolumen). If in Equation (2.85) the luminance of the sender surface is replaced by the luminous intensity per unit area with

$$L_s = \frac{d^2\Phi_s}{d\Omega_r dA_s \cos\theta_s} = \frac{dl(\theta)}{dA_s \cos\theta_s} \quad (2.90)$$

the result for the illuminance is:

$$E_r = \frac{d\Phi}{dA_r} = \int_{\Omega_s} L_s \cos\theta_r d\Omega_s = \int_{A_s} \frac{dl(\theta_s)}{dA_s \cos\theta_s} \cos\theta_r \frac{dA_s \cos\theta_s}{r^2} = \int_{A_s} \frac{dl(\theta_s)}{r^2} \cos\theta_r \quad (2.91)$$

with θ_r as the angle between the receiving surface and solid angle $d\Omega_s$, under which the sender is seen.

With the usual constancy of the luminous intensity over the radiating surface A_s , the photometric distance law is obtained:

$$E_r = \frac{d\Phi}{dA_r} = \frac{l(\theta_s)}{r^2} \cos\theta_r \quad (2.92)$$

However, this functional characteristic of the illuminance, which decreases in inverse proportion to the square of the distance, only applies starting from the so-called photometric minimum distance, which is about 10 times as large as the largest linear dimension of the lighting surface.

This is due to the changing incidence angles, under which the receiver surface sees a large sender surface at close distance. If one calculates, for example, the luminous intensity of a circular sender with radius h with a given luminance L_s , the intensity in the direction of the surface normal is given per definition by the product of luminance and sender surface, i.e.

$$I_s = L_s \pi h^2 \quad (2.93)$$

The illuminance on the receiving surface for half an opening angle α , under which the sender is seen, is calculated as follows:

$$E_r = \frac{d\Phi}{dA_r} = \int_{\Omega_s} L_s \cos \theta_r d\Omega_s = \int_0^{2\pi} \int_0^\alpha L_s \cos \theta_r \sin \theta_r d\theta_r d\gamma = L_s \pi \sin^2 \alpha = L_s \pi \frac{h^2}{h^2 + r^2}$$

Using the photometric distance law, the intensity at normal incidence would be calculated from the illuminance as a function of distance r between sender and receiver.

$$I_s^* = E_r r^2 = L_s \pi \frac{h^2}{h^2 + r^2} r^2 \quad (2.94)$$

As the real intensity is given by $I_s = L_s \pi h^2$, the relative error is:

$$\delta = \frac{I_s - I_s^*}{I_s} = \frac{h^2}{h^2 + r^2} \quad (2.95)$$

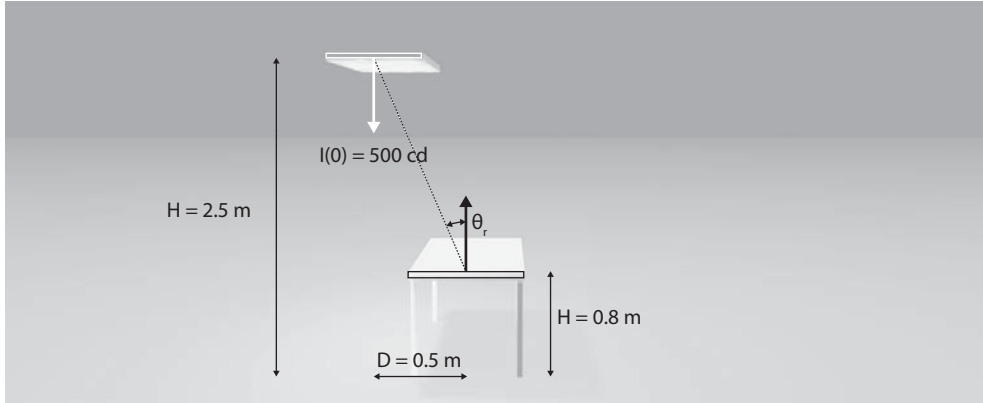
The ratio of distance r to sender radius h , for a maximum admissible error is then given by:

$$\frac{r}{h} = \sqrt{\frac{1-\delta}{\delta}} \quad (2.96)$$

For an error of 1%, a ratio of distance to radius of 9.95 is the result.

Example 2.12

Calculation of the illuminance on an 80 cm high work surface lit by a Lambert radiating light source at a height of 2.5 m and a lateral distance of 0.5 m with a luminous intensity $I(0) = 500 \text{ cd}$.



The angle θ_r between the recipient-surface normal and the light is obtained from the lamp height and lateral distance.

$$\theta_r = \arctan\left(\frac{0.5\text{m}}{1.7\text{m}}\right) = 16.4^\circ$$

The distance r is 1.77 m. The luminous intensity of the light source in direction of the recipient surface is:

$$I(0)\cos\theta_s = 500\text{ cd} \times \cos 16.4 = 480\text{ cd}$$

The result is an illuminance of

$$E = \frac{I(\theta_s)\cos\theta_r}{r^2} = \frac{480\text{ cd} \times \cos(16.4^\circ)}{(1.77\text{ m})^2} = 147\text{ lx}$$

Example 2.13

In an office work space a luminance of 80 cd m^{-2} by reflection from a paper with 70% reflectance should be generated. The light source is 1 metre above the work surface. What is the required intensity of this light source?

From the required luminance, first the illumination is calculated on the plane of the paper:

$$E_e = \frac{L_s \pi}{\rho} = 359\text{ lx}$$

The intensity of light in the vertical direction can then be calculated with the photometric law of distance:

$$I(0) = Er^2 = 359\text{ cd}$$

Units and definitions

An overview over the definitions and units in lighting technology is given in the following table.

Photometric unit	Symbol	Definition	Unit
Luminous energy	Q	$Q = \int \Phi \, dt$	lm s
Luminous flux	Φ	-	lm
Luminous emittance	M	$M = \frac{d\Phi}{dA_s}$	lm m ⁻²
Luminous intensity	I	$I = \frac{d\Phi}{d\Omega}$	cd
Luminance	L	$L = \frac{d\Phi}{d\Omega dA_s \cos \theta_1}$	cd m ⁻²
Illuminance	E	$E = \int L \cos \theta_r \, d\Omega_s$	lx
Luminous exposure	H	$H = \frac{dQ}{dA_r}$	lx s

Table 2.11 Units in lighting technology.

Artificial and daylight sources can be characterised by the following efficiencies:

Efficiency	Symbol	Definition	Unit
Radiative efficiency	η_e	Φ_e / P	-(W/W)
Luminous efficiency	η_v	Φ_v / P	lm W ⁻¹
Photometric radiation equivalent	k	$\frac{\Phi_v}{\Phi_e}$	lm W ⁻¹
Optical efficiency	σ	$\int \frac{d\Phi_v}{d\lambda} d\lambda / \Phi_e$	-
Visual efficiency	ν	$\frac{\Phi_v}{k_m \Phi_e}$	-

Table 2.12 Efficiency definitions in lighting technology.

2.12 Visual performance and quality of lighting

Visual performance is understood to be the recognition of shapes and contrasts. The basic function of the visual system is to provide visual acuity, depth of field, contrast sensitivity, shape and speed of perception, colour sensitivity and the speed of adjustment of the eye to changing distances. Here, the performance of the eye depends to a considerable extent on the luminance level.

The highest visual performance lies at a fixed viewing direction within a very narrow solid angle of 1 - 2° (mapping to the fovea/fovea centralis) and is called foveal vision. Visual objects including contours, brightness and colour differences are fixed in foveal vision.

In regard to the performance within a space, foveal vision indeed provides focus, but only within a very limited view of the visual object. Peripheral vision outside the visual axis, on the other hand, provides information about the area and is very important for the assessment of visual comfort. The highest brightness sensitivity lies in an annular zone at a distance of about 10 - 20° from the axis of the eye. The light sensitivity decreases towards the periphery and extends laterally about 90°, up to about 60° and down to about 70°. Here lie the physiological limits of the field of vision.

The visual acuity decreases sharply at the borders of the visual field. The inner peripheral vision extends to a 45° deviation from the visual axis and determines the level of adaptation. Light sources within this region of the visual field having a higher luminance than the visual object result in a decrease in contrast or glare.

The outer peripheral vision encompasses the remaining solid angle up to the 90° limit of the visual field. Although visual acuity further decreases in this area, it is essential for spatial perception and orientation.

Although peripheral vision does not provide sharp images, it is of high importance for spatial orientation and the early detection of hazards (i.e. steps, doorways). A higher visual comfort supports peripheral vision by accentuating features such as brightness or colour distribution. Daylight also supports the peripheral vision, as the rods responsible for perception of surroundings are more sensitive in the blue wavelength range than cones. Daylighting allows for better orientation not only due to its dynamics, but also because of its shorter wavelength spectrum.

Lighting quality consists of several features such as adequate illumination, a balanced luminance distribution, a limitation of glare, well-coordinated light direction and shading quality resulting in true colour representation, as well as lamps free of flicker.

Luminance contrast and glare

Apart from the supply of sufficient illuminance, the main issues in administrative buildings are problems with computer workplaces, which require glare-free illumination and low contrast. A very uneven distribution of luminance produces disturbances, which in the worst case impairs the visual function (physiological or disability glare). The contrast between the object and the background luminance is reduced by scattered luminance that enters the eye sideways, decreasing contrast sensitivity of the eye and its ability to distinguish between nearby objects.

So that an object can be detected, a luminance difference ΔL_o of the object to the environment has to be given. This required luminance difference increases with increasing luminance levels of the environment L , to which the eye adapts.

The so-called veiling luminance L_s leads to an increase in adaptation level $L_s + L$, so that the visual object with a given luminance difference ΔL_o is no longer perceived. In order to perceive

it again, the luminance difference must be increased to the value ΔL_b . The percent threshold increment T of ΔL_o to ΔL_b is a measure of the physiological glare. High T values mean strong glare. In highly glare-free lighting systems, the luminance difference increases by 7 - 10%.

Absolute glare occurs at luminance levels above 10^4 cd m^{-2} and objects can no longer be distinguished within the visual field.

Psychological or discomfort glare is a sensation of disturbance without a real reduction in the eye's capability. The contrast relation between object luminance and surrounding luminance, which is experienced as pleasant, depends on the absolute level of the surrounding luminance and thus on eye adaptation. The higher the surrounding luminance is, the lower the subjectively perceived brightness of an object luminance. Raising the average luminance can reduce this relative glare caused by too large of a luminance contrast in the visual field, which occurs most frequently in lighting technology.

The average luminance of a computer monitor is between 150 and 500 cd m^{-2} , while the luminance of a day-lit window is about 4500 cd m^{-2} , even with an overcast sky. The luminance contrast of up to 1:30 is so far over the acceptable contrast ration of 1:15 that glare reduction measures must be taken. Ideal luminance contrasts between the visual task and darker immediate surroundings are only 3:1, with more distant surroundings 10:1 and should not exceed values over 20 to 40:1.

To well perceive contrast, i.e. luminance differences on a well-lit work area, there must be a minimum work surface luminance. The required luminance increases with the complexity of the visual task, lying between 10 and 80 cd m^{-2} for very coarse to very fine visual tasks, respectively.

Glare evaluation is performed according to procedures known as *UGR* (Unified Glare Rating). The *UGR* value as a measure of the glare is the ratio of the direct glare from lights in the interior space to the overall brightness or luminance of the background.

The glare is then calculated for the entire lighting system for an observer's position and is thus dependent on the location of the observer. The lower the value of the *UGR*, the lower the level of glare. In addition, an elevation angle is specified as the critical angle above which the light source has a luminance of 1000 cd m^{-2} or more.

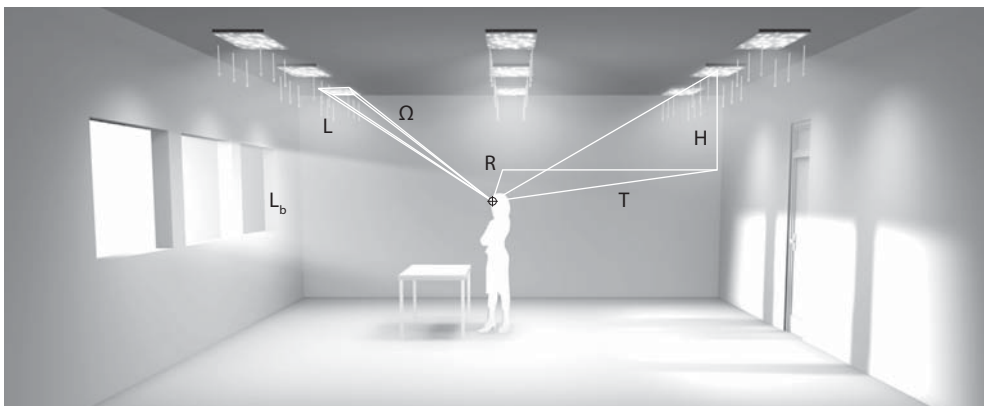


Figure 2.58 Calculating the glare index.

The glare index is calculated from the background luminance L_b and the respective luminance L of the light sources in the solid angle seen from an observer (see Figure 2.58).

$$UGR = 8 \log \left(\frac{0.25}{L_b} \sum \frac{L^2 \Omega}{p^2} \right) \quad (2.97)$$

The position index p indicates the position of the light and is greater the more peripheral the light source is in the visual field. When looking directly into the light source $p = 1$, the largest value is 17. The glare index UGR increases with the square of the light source luminance L , increases linearly with the solid angle of the illuminating surface and is inversely proportional to the environment luminance L_b . The glare index in rooms with computer monitors and office workstations, regardless of the light intensity, should not be greater than 19.

A directly seen light source of $10\,000 \text{ cd m}^{-2}$ with 1° diameter, which is within an environment with a background luminance of 100 cd m^{-2} , has a UGR value of 30.2 ($\Omega = 2.39 \times 10^{-4} \text{ sr}$, $p = 1$) and represents a very high level of glare. According to EN 12464, the following are the permitted maximum UGR values:

Room types and visual tasks	Maximum UGR value
Traffic areas and corridors, hallways and station platforms	28
Assembly works depending on the visual task	16 - 25
Offices	19
CAD rooms	16

Table 2.13 Maximum glare indices for various uses and visual tasks.

Contrast sensitivity and visual acuity

The contrast sensitivity $L/\Delta L$ indicates the ability to perceive luminance differences in the visual field. $L/\Delta L$ initially increases with environment luminance levels L up to about 100 cd m^{-2} and then remains relatively constant at a value of about 60. In this upper range of adaptation luminance, $L/\Delta L$ can be assumed as constant. In this range the so-called Weber–Fechner law defines a brightness dH , which is proportional to dL/L , and the integration yields a logarithmic function of the brightness with luminous intensity.

$$\int_{H_1}^{H_2} dH = H_2 - H_1 = c \int_{L_1}^{L_2} \frac{dL}{L} = c (\ln L_2 - \ln L_1) \quad (2.98)$$

or

$$H = c_1 \ln L + c_2 \quad (2.99)$$

Although the contrast sensitivity improves with higher adaptation luminance levels, the required luminance difference between visual object and environment increases with increasing background luminance levels. If the environment luminance is reduced, the visual object appears brighter.

Visual acuity means the ability to recognise shapes and colour details. Visual acuity increases with increasing luminance levels.

Light direction and object reproduction

The objection reproduction or shadiness describes the property of the lighting system affecting the reproduction of spatial objects. The object reproduction is defined by the ratio of illuminance levels on the horizontal to the vertical plane.

Too much directional light in interior rooms should be avoided, since it not only promotes the formation of shadows, but also the emergence of reflections on non-matte surfaces. Also, light distribution that is too diffuse should be avoided because of the reduction in orientation capacity and a differentiated impression of the space.

The shadiness characteristic is good in rooms with skylights or in rooms with windows on two sides of the room. In contrast, the effects of shadiness in rooms with windows on only one side strongly depend on the viewing direction.

The vertical illuminance that is generated only by artificial light is usually lower than in outside spaces. The vertical illuminance should be at least one-third of the planned horizontal illuminance, i.e. in an office with 500 lx horizontal illuminance, the vertical illuminance should be about 170 lx. For work spaces near a window, the vertical illuminance is over a large part of the day almost 10 times higher than provided by artificial light.

Luminous colour and colour rendering

The colour provided by daylight is variable because the spectral distribution depends on the clearness of the sky and the position of the sun. This variability is usually perceived as pleasant. The colour rendering of daylight is very good and only a few artificial light sources can approximate its quality. The colour rendering of light sources are characterised by the colour rendering index R_a . The highest value with the most natural colour reproduction is $R_a = 100$.

The colour rendering index of a light source is determined by determining the colour shift with respect to a reference light source with eight different colour samples of frequently occurring test colours. The value of each colour sample is added together with 1/8 weight for colour rendering index:

Light source	Colour rendering index
Daylight	100
Fluorescent lamp, white deluxe	85 - 100
White fluorescent lamp	70 - 84
White LED	70 - 95
Fluorescent lamp	50 - 90
Metal halide lamp	60 - 95
Sodium vapour high pressure lamps, warm white	80 - 85
Mercury vapour high pressure lamp	45
High pressure sodium vapour lamp, standard	18 - 30

Table 2.14 Colour rendering index of light sources.

Depending on the visual task, the most suitable light source is selected on the basis of colour reproduction. In areas with the highest standards of colour fidelity (graphics and design), only lamps with a colour rendering index above 90 can be used. The colour rendering index in offices and workstations should not fall below 80.

2.13 Light measurements

The human eye is capable of comparing illuminances of adjacent surfaces with an accuracy of about 2%. This was quantitatively first used in the so-called grease photometer of Bunsen, where luminous intensities were determined by shining two light sources from opposite sides on a paper screen partly covered with grease. If one light source has a known light intensity I_1 , the other light intensity I_2 can be determined by varying the distance d from the paper surface, until no difference in illuminance E is detected by the eye.

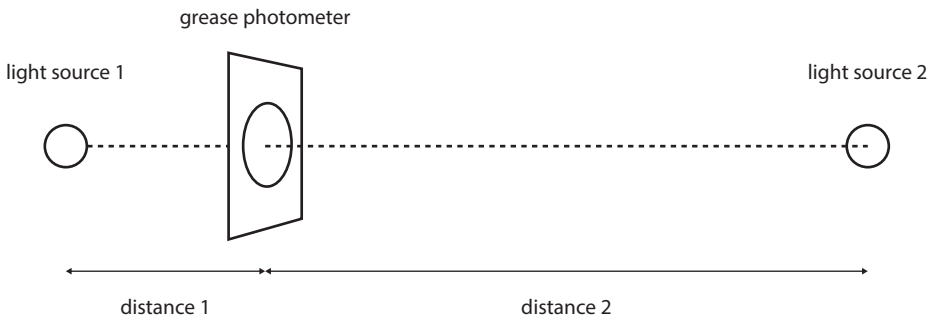


Figure 2.59 Determination of light intensities by illuminance comparison on a greasy screen.

$$E = \frac{I_1}{d_1^2} = \frac{I_2}{d_2^2} \quad (2.100)$$

Photometric receivers today measure light or radiation either by using thermocouples or thermopiles, or by photo-electric techniques. The most important sensors are silicon photodiodes, which at short-circuit current operation linearly measure irradiance over about six orders of magnitude. Very low light fluxes can be measured with a photomultiplier tube.

Charged coupled devices within CCD cameras have up to 2048×2048 individual sensors organised in a matrix, where electrons are generated within the sensor pixels proportionally to the illuminance level and time. At a frequency of 20 MHz, the whole matrix can be analysed within fractions of a second. The dynamic range of CCD receivers is very large. CCD cameras can be used to measure luminance or spectral distributions of large areas.

To provide the correct visible spectrum to the detector, either integral filtering or partial filtering can be used. Integral filtering is the combining of a range of different filters completely covering the sensor surface (f_1 -error is 1.5%), while partial filtering involves the combining different filters only partially covering the sensor (1% f_1 -error). The f_1 -error is an energy-weighted error using the spectral sensitivity of the eye and a standardised light source (at 2856 K) divided by the energy-weighted measured curve of the filters. If the incident light flux is not perpendicular to the detector surface, additional reflection losses occur, which can be compensated by round domes or additional transmitting surfaces on the detector sides. Further measurement errors are due

to linearity, temperature coefficients, etc. The best photometric receivers have a total error of 3% (highest accuracy class) up to 20% (lowest accuracy class).

2.14 Sky luminous intensity models

To calculate the daylight distribution in a room, models of the luminous intensity distribution of the sky are required. The light flux onto any recipient surfaces, for example, windows, can be calculated from the luminous intensity distribution, the effective surface and the solid angle.

From meteorological databases, hourly direct and diffuse irradiance can be obtained for a large number of locations. The direct beam irradiance can be directly converted to luminous intensity using an empirical correlation for the photometric radiation equivalent as a function of sun height.

For the diffuse part of the irradiance, different models with increasing levels of complexity are available to describe the radiance for clear, intermediate and overcast skies. In the simplest case, the radiance is considered to be constant over the sky dome (isotropic sky) or to vary only with elevation angle, but not with the azimuth (Moon and Spencer sky for overcast conditions). For the clear sky diffuse radiance, both the elevation and azimuth angles of the sky dome point and the sun need to be taken into account.

Overcast skies

In the simplest model, the luminance of the sky is set as constant, corresponding to an isotropic sky model. The illuminance on a receiver surface is calculated using Equation (2.85):

$$E_r = \frac{d\Phi_r}{dA_r} = \int_{\Omega_s} L_s \cos \theta_r d\Omega_s$$

The solid angle $d\Omega_s$ can be calculated using Equation (2.75). On a horizontal receiver, the angle of incidence θ_r is equal to the zenith angle of the sky patch and the following illuminance results:

$$E_r = \iint L_s \cos \theta_r \sin \theta_r d\theta_r d\gamma = 2\pi L_s \int_0^{\pi/2} \cos \theta_r \sin \theta_r d\theta_r = 2\pi L_s \left. \frac{\sin^2 \theta_r}{2} \right|_0^{\pi/2} = \pi L_s \quad (2.101)$$

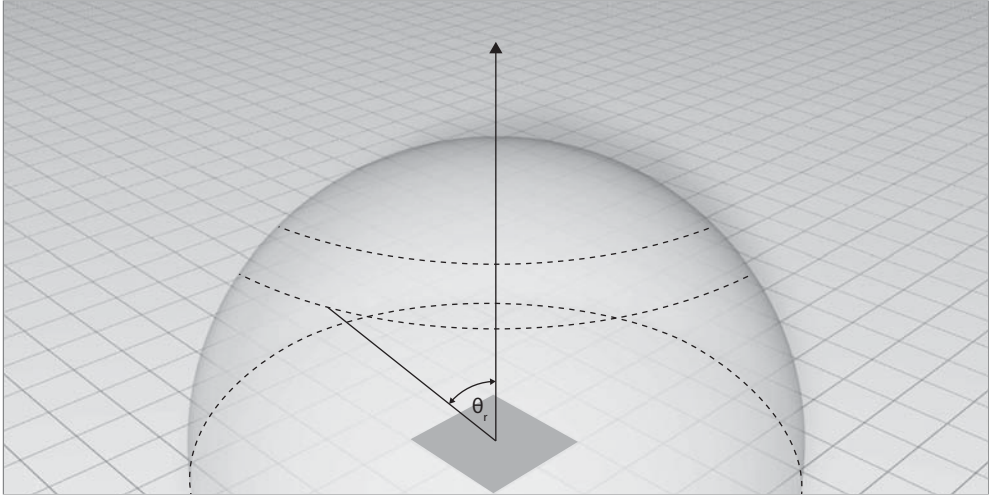


Figure 2.60 Projection of the sky dome with constant luminance onto a horizontal recipient surface.

The sky luminance can be calculated from the horizontal irradiance (in W m^{-2}) by converting the irradiance via the photometric radiation equivalent into illuminance and then dividing the illuminance by π .

Example 2.14

Calculation of the isotropic luminance of the sky for a horizontal irradiance of the overcast sky of 100 W m^{-2} and a photometric radiation equivalent of 115 lm W^{-1} .

The illuminance on the horizontal surface is $11\,500 \text{ lm m}^{-2}$. The illuminance then corresponds to a sky luminance of

$$L_s = \frac{E_r}{\pi} = 3660 \frac{\text{cd}}{\text{m}^2}$$

Apart from the isotropic luminance distribution, standardised distributions of the international lighting engineering commission (CIE) for a clear and an overcast sky are used. The overcast, so-called Moon and Spencer sky is characterised by a rise in the luminance L with elevation angle α , with the zenith luminance L_z assuming three times the value of the horizon luminance.

$$L(\alpha) = \frac{1}{3} L_z (1 + 2 \sin \alpha) \quad (2.102)$$

The zenith luminance for an overcast sky is a function of the sun height angle α_s :

$$L_z = \frac{9}{7} \pi (300 + 21000 \sin \alpha_s) \quad (2.103)$$

For a horizontal surface, the illuminance is calculated using the angle of incidence θ_r on the horizontal (which corresponds to the zenith angle θ_z) instead of the elevation angle α .

$$\begin{aligned}
 E_{r,h} &= \int_0^{2\pi} \int_0^{\pi/2} \left(\frac{1}{3} L_z (1 + 2 \cos \theta_z) \right) \cos \theta_r \sin \theta_z d\theta_z d\gamma \\
 &= \frac{2\pi}{3} L_z \int_0^{\pi/2} (1 + 2 \cos \theta_z) \cos \theta_z \sin \theta_z d\theta_z \\
 &= \frac{2\pi}{3} L_z \left(\int_0^{\pi/2} \cos \theta_z \sin \theta_z d\theta_z + \int_0^{\pi/2} 2 \cos^2 \theta_z \sin \theta_z d\theta_z \right) \\
 &= \frac{2\pi}{3} L_z \left(\left[\frac{1}{2} \sin^2 \theta_z \right]_0^{\pi/2} + 2 \left(-\frac{1}{3} \cos^3 \theta_z \right) \Big|_0^{\pi/2} \right) = \frac{L_z \pi}{3} \left(1 + \frac{4}{3} \right) = L_z \frac{7}{9} \pi
 \end{aligned} \tag{2.104}$$

For vertical surfaces, the solid angle must be selected in such a way that the angles of incidence on the vertical, which change with the azimuth for a given zenith angle, can be taken into account. The best choice is to calculate sky solid angles always with polar coordinates using the zenith angle and azimuth.

The angle of incidence θ_r between the solid angle element of the sky with the coordinates zenith angle θ_z and azimuth γ and the vertical surface normal (angle of inclination $\beta = 90^\circ$) is calculated with the known sun-position equations. Since the luminance does not depend on the azimuth in the Moon and Spencer sky model, the vertical surface can be arbitrarily orientated in azimuth direction. For simplification, the surface azimuth $\gamma_s = 0$ is selected.

$$\cos \theta_r = \cos \theta_z \cos \underbrace{\beta}_{90^\circ} + \sin \theta_z \underbrace{\sin \beta}_1 \cos \left(\gamma - \underbrace{\gamma_s}_0 \right) = \sin \theta_z \cos \gamma \tag{2.105}$$

The integration over the solid angle is carried out with the integration limits of the zenith angle from 0 to $\pi/2$ and of the azimuth of $-\pi/2$ to $+\pi/2$, since only half the sky is seen from the vertical surface.

$$\begin{aligned}
 E_{r,v} &= \int_{d\Omega} L(\theta_z) \cos \theta_r d\Omega = \int_{-\pi/2}^{\pi/2} \int_0^{\pi/2} \left(\frac{1}{3} L_z (1 + 2 \cos \theta_z) \right) \underbrace{\sin \theta_z \cos \gamma}_{\cos \theta_r} \underbrace{\sin \theta_z d\theta_z d\gamma}_{d\Omega} \\
 &= \frac{1}{3} L_z \left(\int_{-\pi/2}^{\pi/2} \int_0^{\pi/2} \sin^2 \theta_z d\theta_z \cos \gamma d\gamma + \int_{-\pi/2}^{\pi/2} \int_0^{\pi/2} 2 \cos \theta_z \sin^2 \theta_z d\theta_z \cos \gamma d\gamma \right) \\
 &= \frac{1}{3} L_z \left(\left(\underbrace{\frac{1}{2} \theta_z}_{\pi/4} - \underbrace{\frac{1}{4} \sin(2\theta_z)}_0 \right) \Big|_0^{\pi/2} \underbrace{\sin \gamma}_{\frac{\pi}{2}} \Big|_{-\pi/2}^{\pi/2} + 2 \left(\underbrace{\frac{1}{3} \sin^3(\theta_z)}_{1/3} \right) \Big|_0^{\pi/2} \underbrace{\sin \gamma}_{\frac{\pi}{2}} \Big|_{-\pi/2}^{\pi/2} \right) \\
 &= \frac{1}{3} L_z \left(\frac{\pi 2}{4} + 2 \frac{1}{3} 2 \right) = L_z \pi \underbrace{\left(\frac{1}{6} + \frac{4}{9\pi} \right)}_{0.308}
 \end{aligned} \tag{2.106}$$

In contrast to the isotropic sky model, in which from a vertical surface exactly half of the horizontal illuminance is seen, only 40% of the horizontal value is obtained in the Moon and Spencer sky model.

$$\frac{E_{r,v}}{E_{r,h}} = \frac{L_z \pi \left(\frac{1}{6} + \frac{4}{9\pi} \right)}{L_z \pi 7/9} = 0.4 \quad (2.107)$$

Clear sky diffuse component

The luminance distribution of a perfectly clear and cloudless atmosphere has been standardised by the CIE as a function of the sun's zenith angle θ_s and azimuth γ_s , the point's P zenith angle θ_p and azimuth γ_p and the angle η between the point on the sky dome and the sun.

$$\frac{L_p}{L_z} = \frac{f_1(\eta)f_2(\theta_p)}{f_1(\theta_s)f_2(0)} \quad (2.108)$$

With

$$f_1(x) = 0.91 + 10 \exp(-3x) + 0.45 \cos^2(x)$$

$$f_2(x) = 1 - \exp\left(\frac{-0.32}{\cos(x)}\right)$$

The angle between the point on the sky dome and the sun is calculated using

$$\eta = \arccos(\cos\theta_s \cos\theta_p + \sin\theta_s \sin\theta_p \cos(\gamma_s - \gamma_p))$$

For polluted atmospheric conditions, the function f_1 becomes

$$f_1(x) = 0.856 + 16 \exp(-3x) + 0.3 \cos^2(x)$$

The zenith luminance L_z for clear sky conditions has been proposed by Nakamura et al. (1985) as a function of sun height α_s .

$$L_z = 100 + 600(\tan\alpha_s)^{1.1}$$

2.15 Daylight distribution in interior spaces

With the asymmetrical luminance distribution of the Moon and Spencer sky, the daylight distribution in the interior can now be calculated. First, as an overview, an illustration will be made showing from which zenith angles the main part of the light flux falls onto a vertical or horizontal glazing surface, in order to make simple estimations of the depth of daylight illumination of rooms.

Using Equation (2.106) the illuminance on a vertical glazing for a zenith angle range of $\theta_{z,1}$ to $\theta_{z,2}$ can be calculated as follows (angles in arc measure):

$$E_v|_{\theta_{z,1}}^{\theta_{z,2}} = \frac{1}{3}L_z \left((\theta_{z,2} - \theta_{z,1}) - \frac{1}{2}(\sin(2\theta_{z,2}) - \sin(2\theta_{z,1})) + \frac{4}{3}(\sin^3(\theta_{z,2}) - \sin^3(\theta_{z,1})) \right) \tag{2.109}$$

The luminance falls with rising zenith angle, but the average angles of incidence onto the vertical surface become smaller, and the solid angles of the spherical ring zones with constant zenith angle steps (for example, 15°) likewise become larger with rising zenith angle. All three effects taken together lead to the fact that despite the highest luminance in the zenith, the light flux from the high zenith angle areas is most relevant.

Zenith angle /°	Luminance ratio L/Lz/-	Normalised vertical illuminance Ev/Lz/sr
0 - 15	0.99	0.01
15 - 30	0.95	0.07
30 - 45	0.86	0.17
45 - 60	0.74	0.24
60 - 75	0.59	0.26
75 - 90	0.42	0.21

Table 2.15 Decrease of illuminance and increase of solid angle and vertical illuminance with zenith angle.

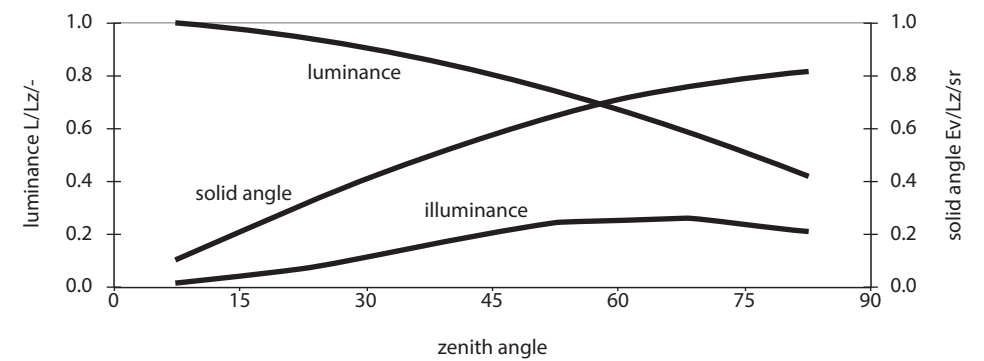


Figure 2.61 Reduction in luminance, and increase in the solid angle as well as in the vertical illuminance with the zenith angle.

The maximum of the vertical illuminance, here standardised on the zenith luminance L_z , comes from the zenith angle interval of $60^\circ - 90^\circ$. The total of the vertical illuminance over all zenith angle areas results in:

$$\sum \frac{E_v|_{\theta_{z,1}}^{\theta_{z,2}}}{L_z} = 0.986 = \pi \left(\frac{1}{6} + \frac{4}{9\pi} \right)$$

and again the result from Equation (2.109) is obtained. The angle of incidence is, based on Equation (2.105), given by $\cos \theta_r = \sin \theta_z \cos \gamma$, and is thus not constant for a given zenith angle interval, as the azimuth γ varies from $-\pi/2$ to $+\pi/2$, respectively. The average angle of incidence onto the vertical surface falls from 83.7° in the zenith angle interval of $0 - 15^\circ$ to 51.3° for the interval of $75 - 90^\circ$.

When light passes through the window, the curve shifts to even higher zenith angles, since the reflection losses for steeply incident light are large. As a rule of thumb for the design of window openings, it follows that for sufficient illumination in depth, at least the lower 30° of the sky should be seen. After all, $(0.21 + 0.26)/0.986 = 0.48$, or 48% of the entire illuminance of the sky hemisphere originates from these zenith angle intervals ($60 - 90^\circ$).

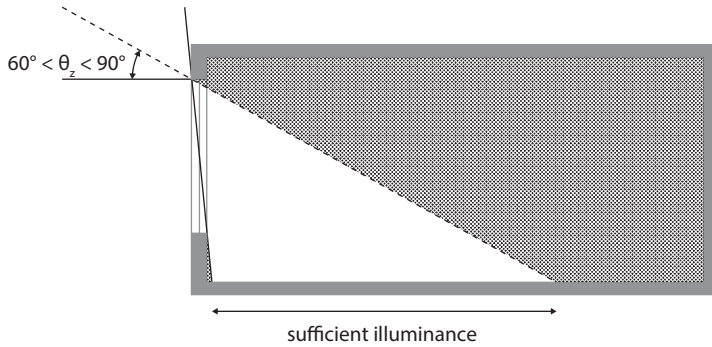


Figure 2.62 Sufficient illuminance of a room.

High-positioned windows without obstructions therefore result in good depth illumination. Narrow, high windows up to the ceiling are clearly better than broad strip windows. With horizontal skylights, the illuminance is calculated from different zenith angle areas using Equation (2.104):

$$E_{r,h}|_{\theta_{z,1}}^{\theta_{z,2}} = \frac{2\pi}{3} L_z \left(\frac{1}{2} (\sin^2 \theta_{z,2} - \sin^2 \theta_{z,1}) - \frac{2}{3} (\cos^3 \theta_{z,2} - \cos^3 \theta_{z,1}) \right)$$

The maximum of the horizontal illuminance normalised with the zenith luminance is now in the zenith angle area of 30 to 45° . A total of 31% of the entire illuminance comes from the zenith angle area $0 - 30^\circ$.



Figure 2.63 Normalised vertical illuminance on a horizontal glazing with an overcast sky, depending on the zenith angle.

Illuminating a room with skylights is sufficient if no more than a zenith angle area of 0 - 30° is cut off.

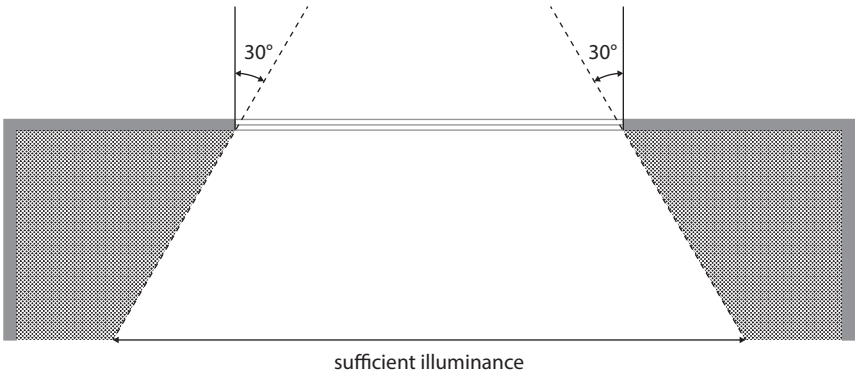


Figure 2.64 Sufficient lighting of a room using skylights.

Calculation of daylight coefficients

The daylight coefficient D as the relation of interior illuminance to exterior illuminance is defined as standard for two measuring points at a height of 0.85 m, and 1 m away from the side walls. The daylight coefficient consists of a skylight proportion D_{sky} , a proportion of diffuse reflection of shading obstructions D_{sh} and an interior reflection proportion D_r .

$$D = D_{sky} + D_{sh} + D_r \tag{2.110}$$

Since about 1920, graphic methods have been used to determine the proportion of the sky D_{sky} seen from the work surface and of the shading external obstructions D_{sh} , which only reflect light. The so-called Waldram diagram contains the projection of the sky dome onto a horizontal surface and takes into account the luminance increase to the zenith. Shading buildings obstructing the horizon

are included with their solid angle and reflection coefficient. The more complex inter-reflections in the interior (interior reflection proportion D_i) were only later included in the calculation of the daylight coefficient and are today usually calculated in a simplified way by the so-called split flux method, in which only skylight reflections from the interior floor and the lower wall sections and the reflection of ground reflected light from the ceiling and the upper wall sections are regarded separately.

The daylight coefficient is determined first in dependence on the room geometry from the raw dimensions of the window openings (index r) and then multiplied by light-reducing factors (window transmittance τ , a framework proportion factor k_1 , a dirt factor k_2 and a correction factor accounting for reflections for non-vertical incidence k_3).

$$D = (D_{sky,r} + D_{sh,r} + D_{r,r}) \tau k_1 k_2 k_3 \quad (2.111)$$

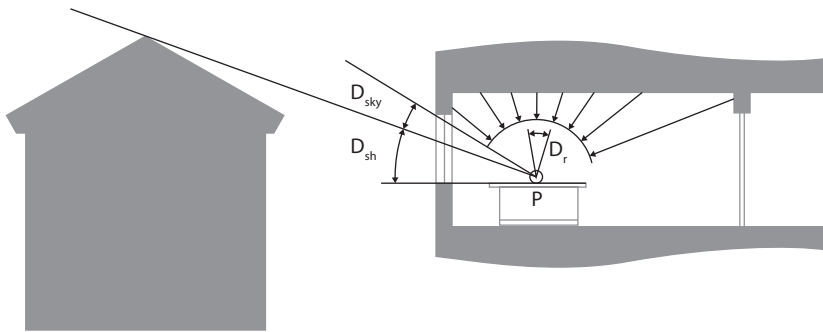


Figure 2.65 Shares of daylight factor (from DIN 5034 Part 3).

For the calculation of the skylight proportion seen from the point of reference P , the effective elevation angle α_w of the window's upper edge and the lateral delimitations of the window by a left and right azimuth angle γ_{wl} and γ_{wr} must first be determined.

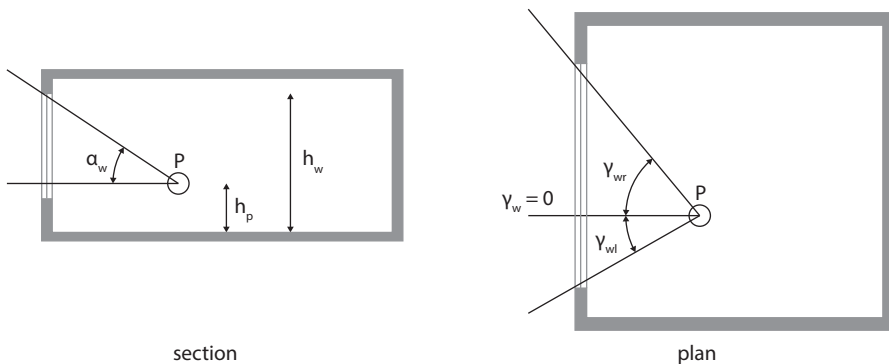


Figure 2.66 Geometrical characteristics of the window in cross section and plane view.

To determine the effective elevation angle α_w of the window, first the maximum elevation angle $\alpha_{w,max}$ is determined using the window height h_w (upper edge) and the shortest distance between the window and point of observation. Then the lateral reduction of the elevation angle with the azimuth γ is calculated.

$$\alpha_w = \arctan(\tan \alpha_{w,max} \cos \gamma) \quad (2.112)$$

To take obstructions into account, the obstruction elevation angles of external shading objects must be known as a function of the azimuth. The illuminance on a horizontal surface in the unshaded exterior is $E_{h,o} = L_z \pi / 9$, based on Equation (2.104). The illuminance on a horizontal surface in the interior can likewise be calculated, taking into account the reduced height and azimuth angles of the window:

$$E_{h,i} = \frac{L_z}{3} \left(\int_{\gamma_{wf}}^{\gamma_{wr}} d\gamma \left(\frac{1}{2} \sin^2(\theta_z) \right) \right)_{\theta_{z,1}}^{\theta_{z,2}} + \int_{\gamma_{wf}}^{\gamma_{wr}} d\gamma \left(-\frac{2}{3} \cos^3(\theta_z) \right)_{\theta_{z,1}}^{\theta_{z,2}} \quad (2.113)$$

The zenith angle is replaced by the window height angle $\alpha_w = \pi/2 - \theta_z$, with

$$\begin{aligned} \cos(\theta_z) &= \sin(\alpha_w), \quad \sin(\theta_z) = \cos(\alpha_w) \\ \sin^2(\theta_z) &= \cos^2(\alpha_w) = 1 - \sin^2(\alpha_w) \end{aligned}$$

The delimitation angles for the luminance integral are the left and right azimuth angle of the window opening γ_{wl} and γ_{wr} . The lower elevation angle is the obstruction elevation angle α_{sh} (corresponding to the larger zenith angle $\theta_{z,2}$) and the upper elevation angle is the window height angle α_w (corresponding to $\theta_{z,1}$).

$$\begin{aligned} E_{h,i} &= \frac{L_z}{3} \left(\int_{\gamma_{wl}}^{\gamma_{wr}} \left(\left(\frac{1}{2} (\sin^2(\theta_{z,2}) - \sin^2(\theta_{z,1})) \right) - \frac{2}{3} (\cos^3(\theta_{z,2}) - \cos^3(\theta_{z,1})) \right) d\gamma \right) \\ &= \frac{L_z}{3} \left(\int_{\gamma_{wl}}^{\gamma_{wr}} \left(\left(\frac{1}{2} (-\sin^2(\alpha_{sh}) + \sin^2(\alpha_w)) \right) - \frac{2}{3} (\sin^3(\alpha_{sh}) - \sin^3(\alpha_w)) \right) d\gamma \right) \end{aligned} \quad (2.114)$$

Thus, the daylight coefficient results in:

$$\begin{aligned} D_{sky,r} &= \frac{E_{h,i}}{E_{h,o}} \\ &= \frac{\frac{L_z}{3} \left(\int_{\gamma_{wl}}^{\gamma_{wr}} \left(\left(\frac{1}{2} (-\sin^2(\alpha_{sh}) + \sin^2(\alpha_w)) \right) - \frac{2}{3} (\sin^3(\alpha_{sh}) - \sin^3(\alpha_w)) \right) d\gamma \right)}{\frac{L_z}{9} \pi} \\ &= \frac{3}{7\pi} \int_{\gamma_{wl}}^{\gamma_{wr}} \left(\frac{2}{3} (\sin^3 \alpha_w(\gamma) - \sin^3 \alpha_{sh}(\gamma)) + \frac{1}{2} (\sin^2 \alpha_w(\gamma) - \sin^2 \alpha_{sh}(\gamma)) \right) d\gamma \end{aligned} \quad (2.115)$$

The externally reflected proportion $D_{sh,r}$ results as a function of the obstruction angles and the reflection coefficient ρ_{sh} of the obstruction (typically assumed with 20%), by integration from the

elevation angle $\alpha = 0$ up to the obstruction elevation angle α_{sh} as well as over the azimuth angles of the obstruction $\gamma_{sh,l}$ and $\gamma_{sh,r}$. In the German standard DIN 5034, the external reflection proportion is reduced by a factor of 0.75.

$$D_{sh,r} = 0.75 \rho_{sh} \frac{3}{7\pi} \int_{\gamma_{sh,l}}^{\gamma_{sh,r}} \left(\frac{2}{3} \sin^3 \alpha_{sh} + \frac{1}{2} \sin^2 \alpha_{sh} \right) d\gamma \quad (2.116)$$

The interior reflection proportion calculated using the split flux method is calculated depending on the surface-weighted reflection coefficient of the floor, the lower part of the wall ρ_{fw} (without window walls, lower part of wall to height of window centre), as well as on the corresponding surface-weighted reflection coefficient of the ceiling and upper wall sections ρ_{cw} (likewise without window walls). In contrast, the average reflection coefficient of the room $\bar{\rho}$ includes all walls.

First the two light fluxes from the sky and from ground reflection are determined. The luminous flux from the sky onto a vertical surface $\Phi_{V,sky}$ is given by the product of vertical illuminance $E_{V,sky}$ and window area A_W , where the vertical illuminance is calculated using Equation (2.106). If there are obstructions from surrounding buildings, the integration angles for zenith and azimuth are adapted.

$$\Phi_{V,sky} = A_W E_{V,sky} \quad (2.117)$$

This luminous flux meets the floor and the lower wall surfaces. The first reflection of the luminous flux results in the light flux Φ_1 .

$$\Phi_1 = \rho_{\text{floor-wall}} A_W E_{V,sky} \quad (2.118)$$

Similarly, the light flux from ground reflection $\Phi_{V,ground}$ hits the ceiling and upper walls and is then reflected:

$$\Phi_2 = \rho_{\text{ceiling-wall}} \Phi_{V,ground} \quad (2.119)$$

The ground is considered a fully diffuse reflector with a reflectance ρ_{ground} and a specific emittance M . This is calculated from the incident sky horizontal illuminance using Equation (2.104), again considering the correct integration angles, if surrounding objects are present.

$$M = \rho_{ground} E_{H,sky} \quad (2.120)$$

The corresponding luminance of the ground reflection is thus given by

$$L_{ground} = \frac{\rho_{ground} E_{H,sky}}{\pi} \quad (2.121)$$

To calculate, how much of the ground emission is seen by the vertical facade, Equation (2.106) is used. If the horizon is obstructed by surrounding objects, the integration angles have to be limited. In case of an unobstructed horizon, the following result is obtained.

$$\begin{aligned}
 E_{v,ground} &= \int_{d\Omega} L \cos \theta_r d\Omega = L \int_{-\gamma_l}^{\gamma_r} \int_0^{\theta_{ground}} \underbrace{\sin \theta \cos \gamma}_{\cos \theta_r} \underbrace{\sin \theta d\theta d\gamma}_{d\Omega} \\
 &= L \left(\int_{-\pi/2}^{\pi/2} \int_0^{\pi/2} \sin^2 \theta_z d\theta_z \cos \gamma d\gamma \right) \\
 &= L \left(\left(\underbrace{\frac{1}{2} \theta_z}_{\pi/4} - \underbrace{\frac{1}{4} \sin(2\theta_z)}_0 \right) \bigg|_0^{\pi/2} \underbrace{\sin \gamma}_{\pi/2} \bigg|_{-\pi/2}^{\pi/2} \right) \\
 &= \frac{\rho_{ground} E_{H,sky}}{\pi} \left(\frac{\pi}{2} \right) = \frac{\rho_{ground} E_{H,sky}}{2}
 \end{aligned} \tag{2.122}$$

These two light fluxes are now indefinitely re-reflected using the area weighted reflectance of the room (all surrounding surface areas). The sum of the reflection components results in the limit value of the geometric series:

$$\bar{\rho} + \bar{\rho}^2 + \bar{\rho}^3 + \bar{\rho}^4 + \dots = \frac{1}{1 - \bar{\rho}} \tag{2.123}$$

And thus, the whole interior light reflected light flux is given by:

$$\Phi_r = \frac{1}{1 - \bar{\rho}} \left(\Phi_{v,sky} \rho_{floor-wall} + \Phi_{v,ground} \rho_{ceiling-wall} \right) \tag{2.124}$$

The external light fluxes onto the vertical facade from the sky and ground are obtained as described before by integrating over the solid angle of the sky up to the obstructions and of the solid angle of the ground.

The illuminance by internal reflection is obtained by relating this light flux to the floor surface area A_R :

$$E_r = \frac{1}{1 - \bar{\rho}} \frac{A_w \left(E_{v,sky} \rho_{floor-wall} + E_{v,ground} \rho_{ceiling-wall} \right)}{A_R} \tag{2.125}$$

The daylight factor is calculated by dividing the internal illuminance by the external horizontal illuminance.

$$D_{r,r} = \frac{E_r}{E_{H,sky}} = \frac{1}{1 - \bar{\rho}} \frac{A_w \left(\frac{E_{v,sky}}{E_{H,sky}} \rho_{floor-wall} + \frac{\rho_{ground}}{2} \rho_{ceiling-wall} \right)}{A_R} \tag{2.126}$$

The German standard DIN 5034 uses so-called window factors used to calculate internal reflection, which use empirical equations for the ratio of illuminances and a constant floor surface reflectance of 20%.

$$D_{r,r} = \frac{\sum b_w h_w}{A_{room}} \frac{\bar{\rho}}{1 - \bar{\rho}^2} (f_{up} \rho_{fw} + f_{low} \rho_{cw}) \quad (2.127)$$

where

A_R room surface (m²)

w_w, h_w window width and height (m)

The upper window factor f_{up} describes the integrated luminance of the Moon and Spencer sky model on the vertical surface, dependent upon an average obstruction angle α (obstruction elevation angle in arc measure, measured from the window centre). The lower window factor f_{low} takes into account the diffuse radiation reflected by the floor.

$$\begin{aligned} f_{up}(\alpha) &= 0.3188 - 0.1822 \sin \alpha + 0.0773 \cos(2\alpha) \\ f_{low}(\alpha) &= 0.03286 \cos \alpha' - 0.03638 \alpha' + 0.01819 \sin(2\alpha') + 0.06714 \end{aligned} \quad (2.128)$$

with $\alpha' = \arctan(2 \tan \alpha)$

Example 2.15

Calculation of the daylight coefficient of a side-illuminated room without obstructions, with window transmittance $\tau = 0.65$, a glazing proportion of 80%, a dirt factor $k_2 = 0.9$ (low contamination) and a factor $k_3 = 0.85$ to take account of the non-vertical incidence angle of the irradiance.

Room geometry

Width, W	4 m
Depth, T	6 m
Height, H	3 m
Height of window upper edge, h_w	2.5 m
Height of window bottom edge, h_{wb}	0.85 m
Width of window, w_w	4 m

Reflection coefficients of the surfaces:

ρ_{floor}	0.3
ρ_{ceiling}	0.7
ρ_{wall}	0.5
ρ_{windows}	0.15

From this results a surface-weighted reflection coefficient of

$$\bar{\rho} = \frac{\rho_{\text{floor}} A_{\text{floor}} + \rho_{\text{ceiling}} A_{\text{ceiling}} + \rho_{\text{wall}} A_{\text{wall}} + \rho_{\text{window}} A_{\text{window}} + \rho_{\text{wall}} A_{\text{wall, window}}}{\sum A}$$

$$= \frac{0.3 \times 24m^2 + 0.7 \times 24m^2 + 0.5 \times 48m^2 + 0.15 \times 6.6m^2 + 0.5 \times 5.4m^2}{108m^2} = 0.48 \quad (2.129)$$

Without obstructions the window factors are $f_{\text{up}} = 0.3961$ and $f_{\text{low}} = 0.1$. Thus, the interior reflection proportion becomes $D_{r,r} = 0.008$, less than 1%!

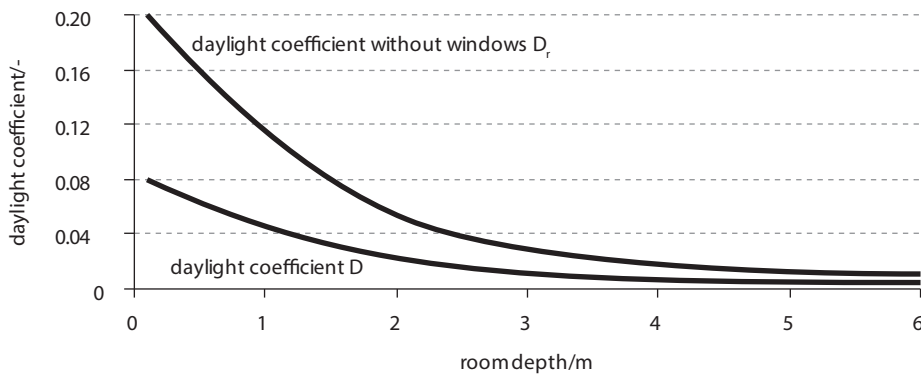


Figure 2.67 Daylight coefficient of the side-illuminated room.

If the reflection coefficient of the walls is increased to 0.7, the interior reflection proportion rises to 1.28%. Raising the window's upper edge to room height (3 m) increases the interior reflection proportion further to 1.7%.

If the reduction of the daylight coefficient by transmittance, framework proportion, etc. is taken into account (a factor of 0.4!), a daylight coefficient of about 1% is obtained at a depth of 4 m.

2.16 Calculation of daylight availability in buildings

To calculate the daylight availability in buildings and the energy consumption for additional electrical lighting, simulations have to be performed for every hour of the year. As the most simple approach, one could consider the daylight coefficients calculated once for a given room situation. If the external illuminance is known for every hour of the year by using photometric radiation equivalents, then the internal illuminance could be directly calculated from the product of daylight coefficient and external illuminance.

A first problem here is that the direct beam component of the external illuminance is a function of sun height and sun azimuth, i.e. a clear geometric position, whereas the daylight coefficient is obtained from the integration of a sky luminance distribution, limited by the window side and height angles. This clearly indicates that the direct beam component needs to be treated separately from the diffuse component; for each point in the room, the geometry determines whether direct solar radiance falls on the point or not.

The second difficulty arises when sky luminance distributions change with time, because the daylight factors as described above are obtained from the integration of one fixed distribution (mostly the Moon and Spencer sky for overcast conditions). It is well known, however, that for clear and intermediate skies, the luminance distributions are different from the overcast conditions.

A solution to the problem is to treat the sky as a mixture of clear and overcast conditions, depending on the diffuse fraction for that hour. The overcast component of the daylight coefficient is constant and needs to be calculated only once. The clear sky component has to be calculated anew for every sun position. The contribution of the direct beam is also calculated separately. Once the different components of the daylight factor have been determined, the diffuse and direct component of the irradiance are converted to illuminance values using photometric radiation equivalents and multiplied by the different daylight factors to obtain the internal illuminance. The useful daylight illumination (*UDI*) is defined by illuminance levels between 100 and 2000 lx. These limits were determined from investigations in day-lit offices, whereby the users could activate shading devices. Visual discomfort usually occurs above 2000 lx and shading devices are activated.

Direct beam illuminance

The horizontal beam irradiance usually given in meteorological databases is first converted to direct normal irradiance by dividing through the solar zenith angle and multiplying by the cosine of the incidence angle onto the receiving (mostly horizontal) surface. The limited window opening is taken into account by a factor δ_{ijk} , which determines, whether the receiving room node j of working surface i 'sees' the sun through the window k .

$$E_{b, direct} = k_s G_{b,h} \frac{\cos\theta}{\cos\theta_z} \delta_{ijk} \quad (2.130)$$

This means that the daylight factor for direct illuminance D_d is either 0 or 1, depending on the value of δ_{ijk} .

The photometric radiation equivalent for the direct beam is given by Aydinli (1981) as a function of the solar altitude angle α_s :

$$\begin{aligned} k_s = & 17.71 + 4.4585\alpha_s - 8.7563 \times 10^{-2} \alpha_s^2 + 7.3948 \times 10^{-4} \alpha_s^3 \\ & - 2.167 \times 10^{-6} \alpha_s^4 - 8.14132 \times 10^{-10} \alpha_s^5 \end{aligned} \quad (2.131)$$

Diffuse illuminance

The relative amounts of overcast and clear sky diffuse contributions are parameterised by a so-called nebulousity index C_i , which is essentially a function of diffuse to global irradiance and the solar altitude angle α_s :

$$C_i = \frac{1 - G_{diff} / G_{global}}{1 - 0.12(\sin \alpha_s)^{-0.82}} \quad (2.132)$$

Using this index, the external unobstructed diffuse illuminance on a horizontal surface is obtained from:

$$E_{diff, external} = C_i G_{d,h} k_{cl} + (1 - C_i) G_{d,h} k_{oc} \quad (2.133)$$

The photometric radiation equivalent for the clear sky diffuse component has been measured by a range of researchers and varies over a great range between 84 and 146. For European sky conditions, a typical value is $k_{cl} = 130 \text{ lm W}^{-1}$.

For overcast sky conditions, the luminous efficacy varies with solar altitude α_s , although often a constant value of $k_{oc} = 115 \text{ lm W}^{-1}$ is used.

The photometric radiation equivalent can be calculated using:

$$k_{oc} = (91.2 + 0.702\alpha_s - 0.0063\alpha_s^2) \times (1.22 - 7.96 \exp(-4)\Delta + 7.96 \exp(-7)\Delta^2) \quad (2.134)$$

and

$$\Delta = \frac{G_{global,h}}{\sin \alpha_s}$$

To calculate the internal illuminance, the two luminance distributions have to be integrated over the window opening angles. For the overcast conditions, the known formulas for the daylight coefficients can be used:

$$D_{sky, oc, r} = \frac{3}{7\pi} \int_{\gamma_{wl}}^{\gamma_{wr}} \left(\frac{2}{3} (\sin^3 \alpha_w(\gamma) - \sin^3 \alpha_{sh}(\gamma)) + \frac{1}{2} (\sin^2 \alpha_w(\gamma) - \sin^2 \alpha_{sh}(\gamma)) \right) d\gamma$$

For the clear sky part, both the illuminance and the daylight coefficient need to be calculated:

$$\begin{aligned} E_{sky, cl, r} &= \int_{\gamma_{wl}}^{\gamma_{wr}} \int_{\pi/2-\alpha_w}^{\pi/2-\alpha_{sh}} L_p(\theta_s, \gamma_s, \theta_p, \gamma_p) \cos \theta_p \underbrace{\sin \theta_p d\theta_p d\gamma_p}_{d\Omega} \\ &= \int_{\gamma_{wl}}^{\gamma_{wr}} \int_{\pi/2-\alpha_w}^{\pi/2-\alpha_{sh}} L_z \frac{f_1(\theta_s, \gamma_s, \theta_p, \gamma_p) f_2(\theta_p)}{f_1(\theta_s) f_2(0)} \cos \theta_p \underbrace{\sin \theta_p d\theta_p d\gamma_p}_{d\Omega} \end{aligned}$$

and

$$D_{sky, cl, r} = \frac{\int_{\gamma_{wl}}^{\gamma_{wr}} \int_{\pi/2-\alpha_w}^{\pi/2-\alpha_{sh}} \frac{f_1(\theta_s, \gamma_s, \theta_p, \gamma_p) f_2(\theta_p)}{f_1(\theta_s) f_2(0)} \cos \theta_p \sin \theta_p d\theta_p d\gamma_p}{\int_0^{2\pi} \int_0^{\pi/2} \frac{f_1(\theta_s, \gamma_s, \theta_p, \gamma_p) f_2(\theta_p)}{f_1(\theta_s) f_2(0)} \cos \theta_p \sin \theta_p d\theta_p d\gamma_p}$$

The internal illuminance from the sky contribution is then obtained from:

$$E_{sky, r} = C_i G_{d, h} D_{sky, cl, r} k_{cl} + (1 - C_i) G_{d, h} D_{sky, oc, r} k_{oc} \quad (2.135)$$

To simplify matters, the contribution of the obstructed sky daylight coefficient through shading is calculated as before only for a completely overcast sky.

$$D_{sh, r} = \rho_{sh} \frac{3}{7\pi} \int_{\gamma_{sh, l}}^{\gamma_{sh, r}} \left(\frac{2}{3} \sin^3 \alpha_{sh} + \frac{1}{2} \sin^2 \alpha_{sh} \right) d\gamma$$

and

$$E_{sh, r} = G_{d, h} D_{sh, r} k_{oc}$$

The internally reflected component is calculated using the split flux method described before. The ground reflection coefficient is ρ_{ground} and the surface orientation of the window opening β is usually 90° .

$$\begin{aligned} E_{int, r} &= \frac{\sum b_w h_w}{A_{room}} \frac{\bar{\rho}}{1 - \bar{\rho}^2} (f_{top} \rho_{fw} + f_{bottom} \rho_{cw}) \\ f_{top} &= C_i G_{d, h} D_{sky, cl, r} k_{cl} + (1 - C_i) G_{d, h} D_{sky, oc, r} k_{oc} + E_{b, direct} \\ f_{bottom} &= (E_{diff, external} + E_{b, direct}) \rho_{ground} \frac{1 - \cos \beta}{2} \end{aligned}$$

As a result, the total internal illuminance is now given by

$$E_{int, r} = E_{b, r} + E_{sky, r} + E_{sh, r} + E_{int, r} \quad (2.136)$$

2.17 Standardisation and calculation methods

The European standard EN 12464 'Light and Illumination' defines lighting level requirements for the task area and immediate surroundings, thus allowing to specify different zones for lighting.

Furthermore, workplace regulations are often applicable, for example, the European Workplace Directive (89/654/EEC). It requires workplaces to receive sufficient natural and artificial light for health and safety protection. In addition, windows, skylights and glass partition walls, depending on the type of work and the work space, should be provided with protection against excessive sun exposure. Also the need for an optical connection to the outside world has influenced the design of workplaces and window openings to a considerable extent.

The energy demand for lighting can be calculated by a simple method in accordance with Part 4 of the German standard DIN V 18 599 'Useful and Final Energy Demand for Lighting.'

Basically, with the use of daylight as well as lighting control systems, the cost of electricity for lighting purposes should be minimised.

The three-step process classifies the supply of daylight by calculating the daylight factor for the building shell openings on the basis of geometric data regardless of the climate and the location of the building. In the second stage, the influence of the facade elements (transmittance, frame shares, etc.) is taken into account. In the third step, the calculated daylight supply is correlated with other variables such as geographic data and information about the artificial lighting system to obtain the electricity demand for artificial lighting. The method allows in the early planning stage the comparison and optimisation of different daylighting systems and calculates the energy savings achievable in combination with the other parts of the building energy balance (heating, cooling).

The standard DIN V 18599, Part 4, distinguishes between daylight-supplied ($A_{DL,i}$) and not daylight-supplied ($A_{NDL,i}$) areas. The depth of a daylight supplied area $a_{DL,max}$ is calculated from the window height and working plane height.

$$a_{DL,max} = 2.5(h_{window} - h_{workplane}) \quad (2.137)$$

The width of the supplied areas results from the window width plus a marginal strip of a quarter of the depth of the supplied areas.

For a detailed calculation, first the areas with significantly different daylight supplies must be defined, which do not necessarily coincide with the building zones. Alternatively, an average daylight factor can be calculated in a simplified way as a function of the Room Depth Index, an index of transparency and an obstruction index for vertical facades.

For example, for skylights, there is a simplified calculation formula:

$$\bar{D} = D_a \tau_{65} k_1 k_2 k_3 \frac{\sum A_o}{A_f} \eta_R \quad (2.138)$$

with

A_o	Surface of the openings (building raw dimensions)/m ²
A_f	Floor area/m ²
D_a	External daylight factor/%
τ_{D65}	Light transmission factor for standard daylight spectrum D65/-
k_1	Reduction factor for frames/-
k_2	Reduction factor for dirt/-
k_3	Reduction factor for non-normal incidence of light (0.85)/-
η_R	Room efficiency factor/-

The external daylight factor D_a is defined as follows:

$$D_a = \frac{E_F}{E_H} \quad (2.139)$$

with

E_F	illuminance on the outside of the light opening with a completely overcast sky/lx
E_H	Horizontal illuminance under overcast sky/lx.

The determination of the external daylight factor depends on the slope of the light entry opening and is equal to 1 for horizontal surfaces. The room efficiency η_R depends on the flux distribution, room geometry and the degrees of reflection in the room and is also used in artificial lighting design.

Light can directly or indirectly reach the workplace via the reflection at the room surfaces and furnishings. The efficiency of a lighting system strongly depends on the light reflectance from the ceiling, floor, walls and furniture. Up to 50% more light flux must be emitted by the sources to achieve the same work area illuminance in a darkly painted room. The greater the proportion of luminous flux that reaches directly onto the illuminated work surface, the greater the room efficiency.

The closer the light source is to the work area, the more light can fall onto the working plane. As a measure of this ratio, the room index k is used:

$$k = \frac{ab}{h(a+b)} \quad (2.140)$$

where a and b are the room length and width, and h is the difference in room height and height of the working plane. The room efficiency is then determined using the room index and the reflectance of the ceiling, floor and walls.

After the calculation of the average daylight coefficient, the daylight supply of the building is classified by one of the four classification groups.

Criterion daylight coefficient D	Classification of daylight supply
$6 \leq D^{1)}$	Good
$4 \leq D < 6\%$	Middle
$2 \leq D < 4\%$	Low
$0 \leq D < 2\%$	None

1) where $D > 10\%$ should be avoided to avoid thermal overheating in summer

Table 2.16 Classification of the daylight factor according to DIN V 18599-4.

The electricity demand for artificial lighting is then relatively simple to determine. Based on the required illuminance in the area under consideration and the type of light sources used, a specific electric power rating in $W\ m^{-2}$ is determined and multiplied by the area served and the operating time of the lights. However, the calculation of the operating time is complex, as it depends on both the amount of daylight as well as usage.

Daylighting factor $C_{DL, supply}$

On the basis of the determined daylighting classification, the maintenance value of illuminance E_m and the orientation and inclination of the light-transmissive areas, the daylight contribution factor $C_{DL, supply}$ is taken from Table 18, DIN V 18599-4. The daylight supply factor takes into account the light transmission of the glazing (if sun protection is not enabled) and an assessment of the sunscreen solution (if sun protection is enabled).

Partial operation daylight factor F_{DL}

The partial operation daylight factor is determined by a variety of parameters, for example, room size, room geometry, window size, orientation, height of the working plane, shading effects of obstructions, glare protection system, type of glazing, usage hours for day and night (diurnal arrangement of working hours), periods of absence, automatic control systems and so on. The values are typically between 0.4 and 0.8, i.e. 40 - 80% of the hours of use, the artificial light is turned on.

Taking into account the correction factor for daylight-dependent lighting control systems, $C_{DL,con}$, with values from Table 19 of the DIN V 18599-4, the partial operation daylight factor F_{DL} is calculated as follows:

$$F_{DL} = 1 - C_{DL, supply} C_{DL, con} \quad (2.141)$$

Effective operating time $t_{eff, Day, DL}$

The effective operating time in the daylight area during daytime is given by:

$$t_{eff, day, DL} = t_{day} F_{DL} F_{pres} \quad (2.142)$$

Where, t_{day} are the annual hours of use in the daytime (DIN V 18599-10), and F_{DL} is the partial operation factor described above.

The presence operational factor F_{pres} takes into account the presence of users and a potential detection of the presence by measurements.

$$F_{pres} = 1 - C_A C_{pres, con} \quad (2.143)$$

The value for user presence C_A is given in DIN V 18599-10. The value for the presence control $C_{pres, con}$ according to Table 22, DIN V 18599-4 is 50% without occupancy sensors and 95% with occupancy sensors or in rooms that are constantly occupied.

Electrical performance evaluation p

The electrical-rated power p for a chosen lighting system is determined as:

$$p = p_{lx} E_m k_A k_L k_R \quad (2.144)$$

with

p_{lx}	specific installed capacity related to 1lx (Table 1 DIN V 18599-4)
E_m	maintenance value of illuminance according to usage profile (DIN V 18599-10)
k_A	correction factor for the visual task area (DIN V 18599-10)
k_L	correction factor for lamp and ballast (Table 2 DIN V 18599-4)
k_R	correction factor for the spatial geometry (Table 3 DIN V 18599-4)

The electrical rating capacity can be quickly determined using a table, which essentially lists parameters such as the type of lighting (direct, direct-indirect, indirect), the lamps and ballast, and the influence of the room geometry. It can also be identified by an efficiency factor method or by professional lighting system planning. The electrical power rating should be determined in accordance with the requirements specified in DIN EN 12464-1 on the Lighting of Work Places.

Final energy demand for lighting Q_l

The annual energy demand for lighting Q_l taking into account a supply of daylight is then calculated for the N zones of a building, with the J subdivided daylighting areas:

$$Q_l = \sum_{n=1}^N \sum_{j=1}^J p_j \left(A_{DL,j} (t_{eff,day,DL,j} + t_{eff,night,j}) + A_{NDL,j} (t_{eff,day,NTL,j} + t_{eff,night,j}) \right) \quad (2.145)$$

with

p_j	electrical power rating of the lighting system/W m ⁻²
A_{DL}	daylight supplied area/m ²
A_{NDL}	not daylight supplied area/m ²
$t_{eff,day,(D)TL}$	annual effective operating time in hours during daytime
$t_{eff,night}$	annual utilisation in hours during night-time

2.18 Determination of needed artificial light sources

The lighting efficiency η_l is given by the product of luminous efficiency of the light source η_{LB} and the room efficiency η_R . It is defined as the ratio of useful luminous flux onto the work surface Φ_{use} to the total luminous flux emitted by the lamps installed within a luminaire $\Sigma\Phi$.

$$\eta_l = \frac{\Phi_{use}}{\Sigma\Phi} = \frac{\bar{E}A_{use}}{\Sigma\Phi} \quad (2.146)$$

The luminous efficiency is the ratio of luminous flux of the lamps within a luminaire $\Sigma\Phi$ to the light flux leaving the luminaire Φ_{lum} and is mainly influenced by the geometry and reflection coefficient of the luminous fixture.

$$\eta_{lum} = \frac{\Phi_{lum}}{\Sigma\Phi} \quad (2.147)$$

The room efficiency is defined as the ratio of the useful luminous flux onto the work surface Φ_{use} to the light flux leaving the luminaires Φ_{lum} .

$$\eta_r = \frac{\Phi_{use}}{\Phi_{lum}} \quad (2.148)$$

The lighting efficiency is thus the product of luminous efficiency and room efficiency.

$$\eta_l = \frac{\Phi_{use}}{\Phi_{lum}} \cdot \frac{\Phi_{lum}}{\Sigma\Phi} = \eta_{lum} \eta_r \quad (2.149)$$

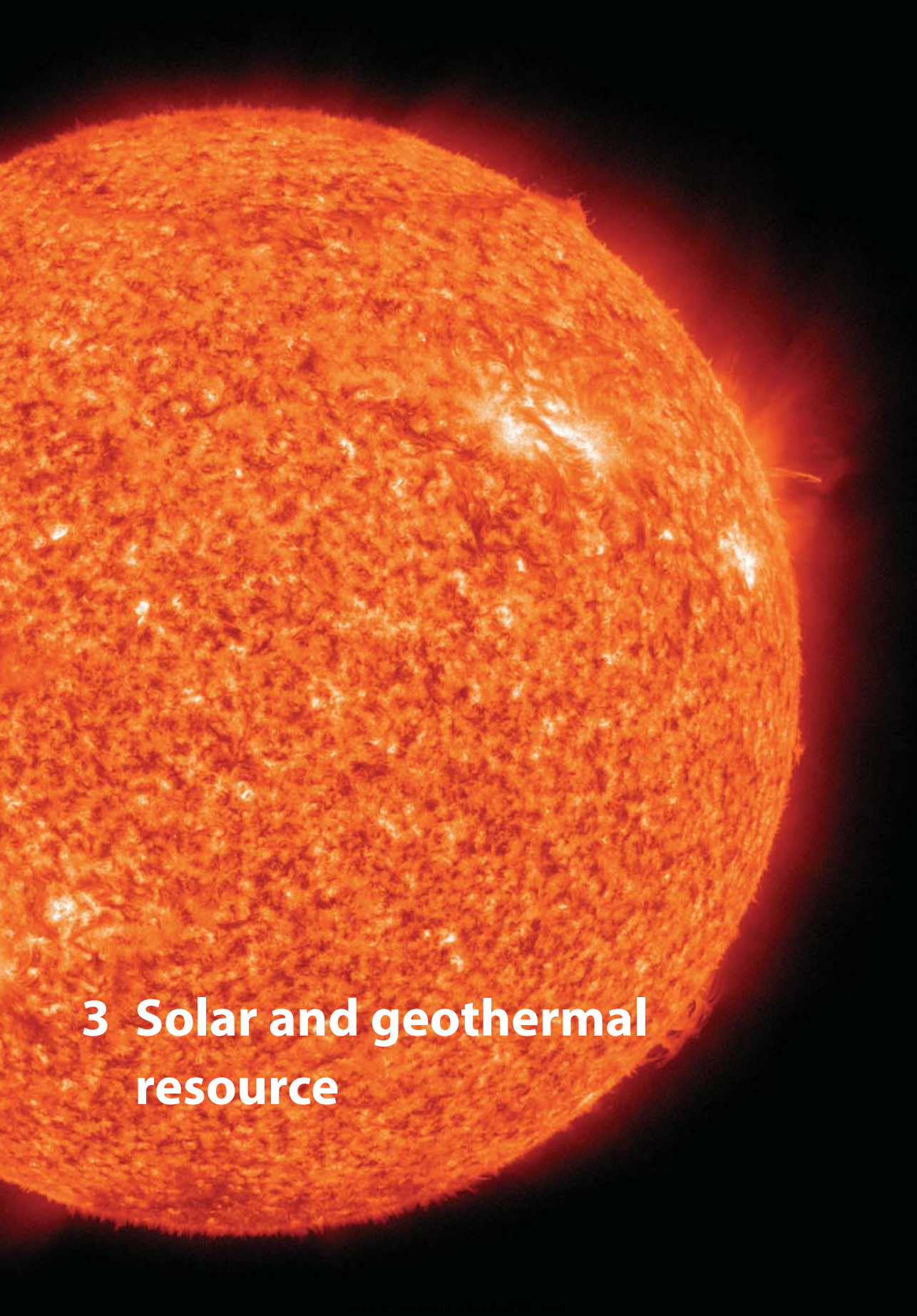
From this, the number of luminaires n required to achieve a mean $\bar{E}$ illuminance level is calculated.

$$n = \frac{\bar{E}A}{\eta_r \eta_{LB} z \Phi} \quad (2.150)$$

where z is the number of lamps per luminaire and Φ the luminous flux of a single lamp.

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3 Solar and geothermal resource

Figure 3.1 (previous page): The Sun photographed by the Atmospheric Imaging Assembly (AIA 304) of NASA's Solar Dynamics Observatory (SDO). This is a false-colour image of the Sun observed in the extreme ultraviolet region of the spectrum.

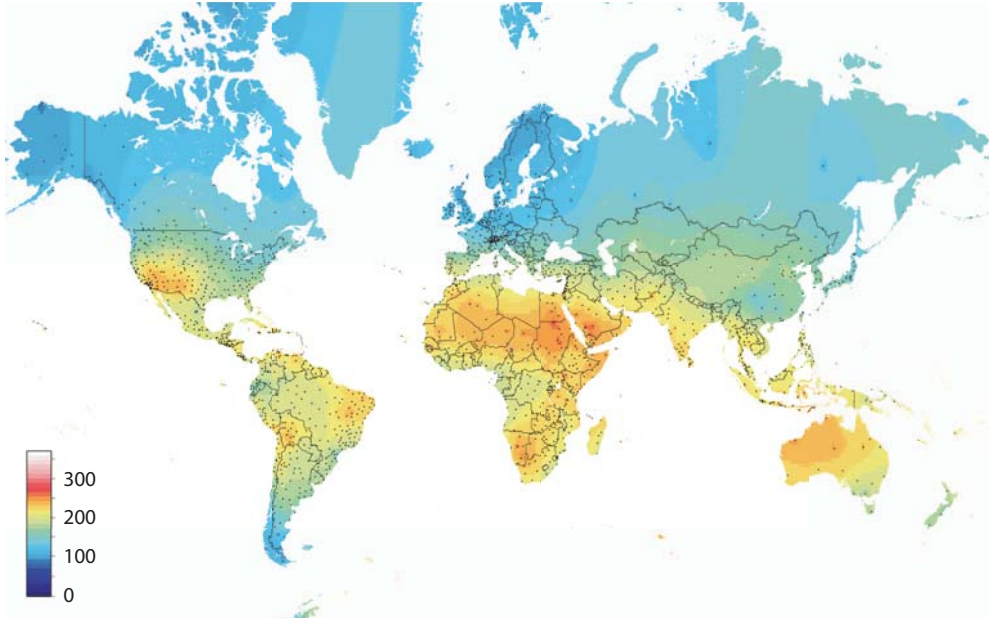


Figure 3.2 A map with annual average irradiated solar power (in watt per square metre) has been generated using the INSEL meteorological database. If the average power is multiplied with the number of hours per year (8760 hours), the annual horizontal irradiance per square metre is obtained.

The annual solar irradiance on a horizontal surface is highest near the Earth's equator, where the incidence angles are close to vertical all year, the air mass to be crossed is the smallest and the cloud cover is low. In these regions the short-wave irradiance reaches up to $2100 \text{ kWh m}^{-2} \text{ a}^{-1}$.

Under moderate European climatic conditions, for example, Germany, a square metre of horizontal Earth surface receives between 925 kWh m^{-2} of solar irradiance annually in the north (54° latitude), and 1170 kWh m^{-2} in the south (47°), with a daily average of around 3 kWh m^{-2} . Direct solar irradiance accounts for just about 50%, while the remainder consists of diffuse irradiance from the atmosphere. In southern Europe, the annual irradiance on a horizontal surface can reach up to 1770 kWh m^{-2} (Almeria/Spain at 37° northern latitude). In northern Europe, between 60 and 70° geographical latitudes, the irradiance drops from 990 kWh m^{-2} in Helsinki to a low of 700 kWh m^{-2} in northern Norway, where there is no sun for 4 months in winter. Asian locations such as Shanghai with 31° Northern latitude has $1375 \text{ kWh m}^{-2} \text{ a}^{-1}$, comparable to central European situations; Mumbai at 19° Northern latitude already reaches $1880 \text{ kWh m}^{-2} \text{ a}^{-1}$.

For the dimensioning and yield prognosis of active and passive solar technology in buildings, it is often not sufficiently precise to determine only the monthly or annual solar irradiance on a roof or a facade surface and then multiply this by the system efficiency. The dynamic storage behaviour of building components and heat stores is decisive for the solar contribution to energy requirements, particularly concerning the thermal use of solar energy (active and passive).

For system simulations, a temporal resolution of the solar irradiance of 1 hour has proved a good compromise between computational accuracy and computational time. In what follows, time series of hourly average irradiance will be presented. On the basis of the purely geometry-dependent hourly irradiance on a surface outside the atmosphere, the so-called extra-terrestrial irradiance, absorption and dispersion in the atmosphere are taken into account using statistical methods. Allocation of the irradiance into a direct proportion and irradiance diffused by atmospheric dispersion enables the conversion of horizontal irradiance on surfaces orientated at any angle. The effect of shading, which plays a significant role particularly in the urban space, can be determined from the geometrical relationship between recipient surfaces and points in the sky.

3.1 Extra-terrestrial solar irradiance

Power and spectral distribution of solar irradiance

The radiating power of the sun results from a process in which four hydrogen nuclei fuse into a helium nucleus. The resulting loss of mass, totalling 4.3 million tonnes per second is transferred into a freed power of $3.845 \times 10^{26} \text{ W}$. This energy, released at extremely high temperatures ($> 10^7 \text{ K}$), is transferred by radiation and convection to the outer photosphere.

Extra-terrestrial irradiance develops predominantly in the photosphere, which is composed of inhomogeneous gases of low density. The photosphere consists of strongly ionised gases, which constantly recombine with free electrons and whose kinetic energy is transferred into a continuous irradiance spectrum. Over it is a so-called reversal layer of about 100 km in thickness, which contains almost all the elements in the Earth's crust. The chromosphere, which consists of hydrogen and helium, and is about 2500 km thick, forms the sun's atmosphere together with the reversal layer. The corona, which expands far into the solar system, is a gas layer much hotter than the chromosphere (Iqbal, 1983). If the sun is regarded as a black body, an equivalent irradiance temperature, T_e , can be calculated from the specific radiant emittance M , on the basis of the Stefan-Boltzmann law (with the Boltzmann constant $\sigma = 5.67051 \times 10^{-8} \text{ W m}^{-2} \text{ K}^{-4}$). The specific

radiant emittance is defined as the relationship of total radiating power Φ and sun surface A_s ($6.0874 \times 10^{12} \text{ km}^2$).

$$M(T_s) = \frac{\Phi}{A_s} = \sigma T_s^4 = \frac{3.845 \times 10^{26} \text{ W}}{6.0874 \times 10^{18} \text{ m}^2} = 63.11 \frac{\text{MW}}{\text{m}^2}$$

$$\Leftrightarrow T_s = \sqrt[4]{\frac{M}{\sigma}} = 5777 \text{ K} \quad (3.1)$$

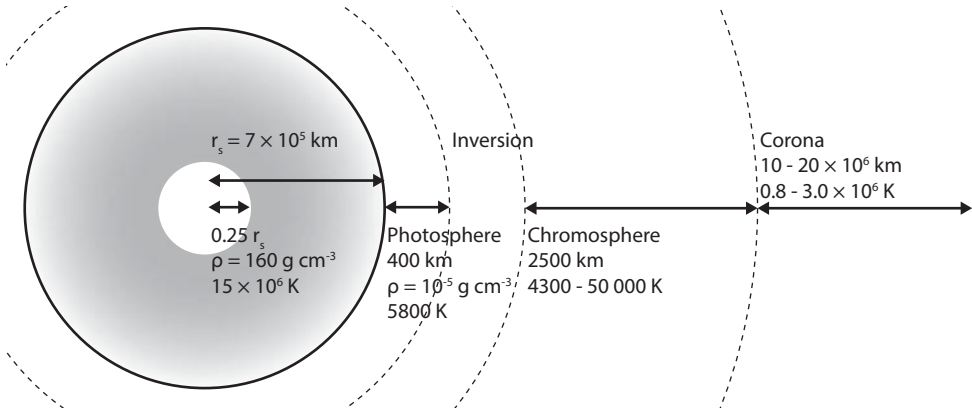


Figure 3.3 Structure of the sun (not to scale).

The irradiance G_{sc} outside the terrestrial atmosphere, known as the solar constant, can be calculated from the entire radiant emittance of the sun, which is the product of specific radiant emittance and sun surface MA_s . By relating this radiant emittance to a square metre of the spherical surface A_{se} , which is formed by the radius of the distance Earth-sun, the solar constant can be obtained. The average distance between the Earth and the sun is $r_0 = 1.496 \times 10^{11} \text{ m}$, which is referred to as an astronomical unit (AU).

$$G_{sc} = M \frac{A_s}{A_{se}} = M \left(\frac{r_s}{r_0} \right)^2 = 63.11 \left(\frac{6.9598 \times 10^8}{1.4959789 \times 10^{11}} \right)^2 \frac{\text{MW}}{\text{m}^2} = 1367 \frac{\text{W}}{\text{m}^2} \quad (3.2)$$

The sun's radiating power of $3.845 \times 10^{26} \text{ W}$ is diluted by the squared relationship of its radius r_s to the sun-earth distance r_0 (factor 2.16×10^{-5}).

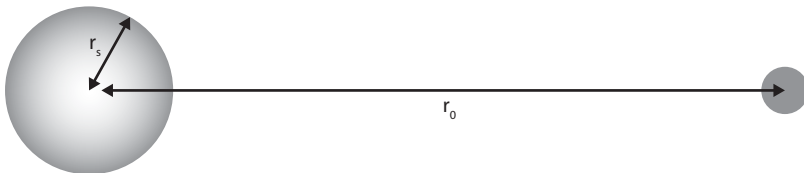


Figure 3.4 Sun radius and Sun-Earth distance.

As a first approximation, the solar spectrum corresponds to a black emitter radiating at 5777 K. The deviations between the spectral distribution of a black emitter and measured extra-terrestrial irradiance are caused by absorption and dispersion in the outer, cooler layers of the photosphere, which, apart from hydrogen and helium, contains about 2% by mass of heavy elements. Altogether 20 000 accurately measured absorption lines in the solar spectrum can be observed.

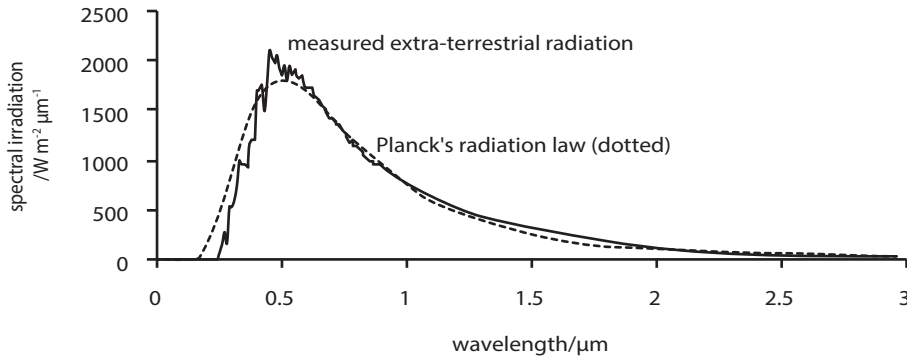


Figure 3.5 Measured and calculated spectral distribution of the sun.

The spectral radiant emittance of a thermal emitter G_λ (in $\text{W m}^{-2} \mu\text{m}^{-1}$) is calculated using Planck's irradiance law as a function of the temperature T (K) and wavelength λ (μm):

$$G_\lambda = \frac{C_1}{\lambda^5 \left(\exp(C_2 / (\lambda T)) - 1 \right)} \quad (3.3)$$

with constants $C_1 = 3.7427 \times 10^8 \text{ W } \mu\text{m}^4 \text{ m}^{-2}$ and $C_2 = 1.4388 \times 10^4 \mu\text{m K}$. At a sun surface temperature of 5777 K, the calculated spectrum is illustrated in Figure 3.5. If the extra-terrestrial irradiance is integrated over the wavelength, the cumulative irradiance power can be calculated. Only 48% of the extra-terrestrial irradiance intensity is in the visible range of 380 - 780 nm ($1 \text{ nm} = 10^{-9} \text{ m}$). Apart from ultraviolet irradiance ($< 380 \text{ nm}$), which accounts for 6.4% of the total intensity, 45.6% is given off in the upper infrared. The irradiance is negligible above 3000 nm.

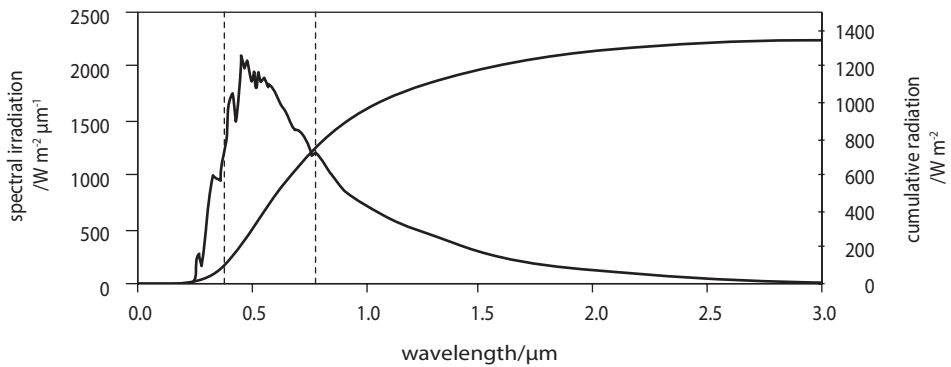


Figure 3.6 Spectral intensity and cumulative power of extra-terrestrial irradiance.

Total ultraviolet irradiance below $0.38 \mu\text{m}$ is about 92.6 W m^{-2} . The visible area within the broken lines has a total power of 660 W m^{-2} , and the remainder of the total irradiance of 1367 W m^{-2} is infrared.

3.2 Sun–Earth geometry

The orbit of the Earth around the sun in the so-called ecliptic plane is slightly elliptical, with a minimum distance of 0.983 AU on 3 January and a maximum distance of 1.017 AU on 4 July.

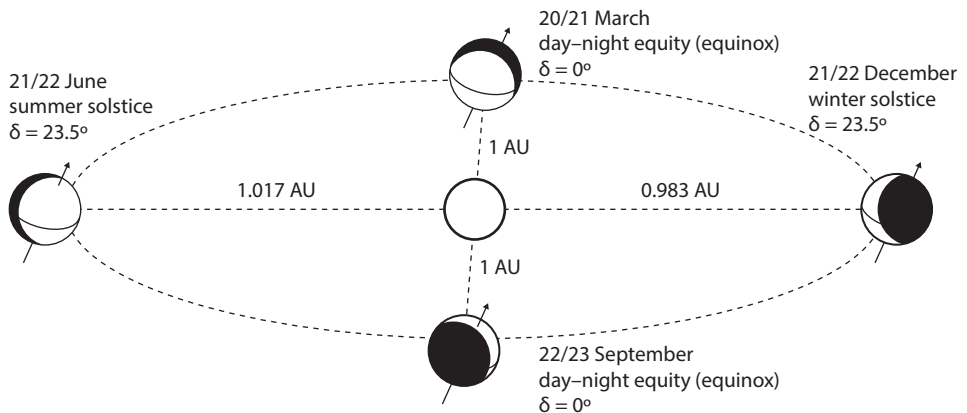


Figure 3.7 Ecliptic plane and position of the Earth at the winter and summer solstices, and also at the spring and autumn equinoxes.

The changing distance leads to a fluctuation of the extra-terrestrial irradiance on a normal surface G_{en} of approximately $\pm 3\%$. For a given day-number n , the irradiance can be calculated with an error <

0.3% by a simple approximation formula (Duffie and Beckmann, 1980) or more exactly, by a Fourier series expansion (Spencer, 1971).

$$G_{en} = G_{sc} \left(\frac{r_0}{r} \right)^2 = G_{sc} \left(1 + 0.033 \cos \left(\frac{360n}{365} \right) \right) \quad (3.4)$$

$$G_{en} = G_{sc} \left(1.000110 + 0.034221 \cos B + 0.001280 \sin B \right. \\ \left. + 0.000719 \cos 2B + 0.000077 \sin 2B \right) \quad (3.5)$$

with

$$B = 360^\circ \frac{n-1}{365}$$

3.3 Equator coordinates

The Earth rotates around its polar axis, which is inclined at 23.45° to the normal of the ecliptic plane. The daily irradiance fluctuations are caused by the rotation on the polar axis; the seasonal variations are caused by the inclination of the polar axis relative to the sun. Let us first analyse the seasonal modifications of the irradiance, which result from the movement of the Earth around the sun with the polar axis in a constant position in space.

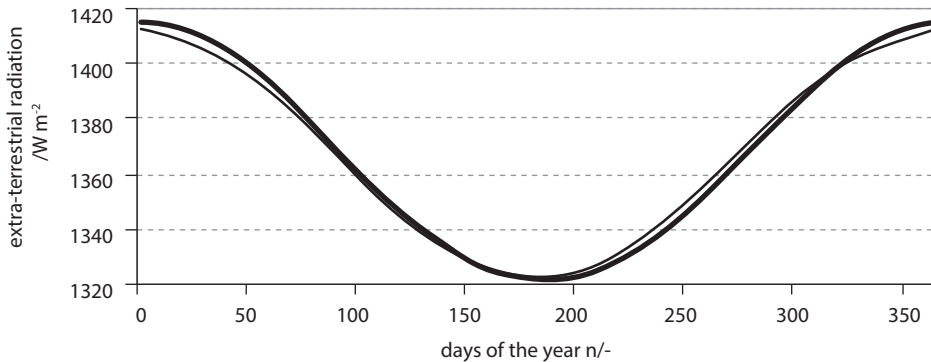


Figure 3.8 Variation of the extra-terrestrial irradiance on a normal surface calculated by Fourier expansion (thick curve) and by the approximation equation (thin curve).

Declination

If the sun is theoretically 'observed' from the centre of the Earth during the annual circulation of the earth around the sun, the angle between sun direction and equatorial plane constantly changes. This angle is called the declination and is positively defined in the northern hemisphere. Positive declination angles (sun above the equator) up to a maximum of 23.45° characterise the summer in the northern hemisphere, while negative declination angles up to -23.45° characterise the winter and vice versa for the Southern hemisphere. At the autumn and spring equinoxes, the declination is 0° .

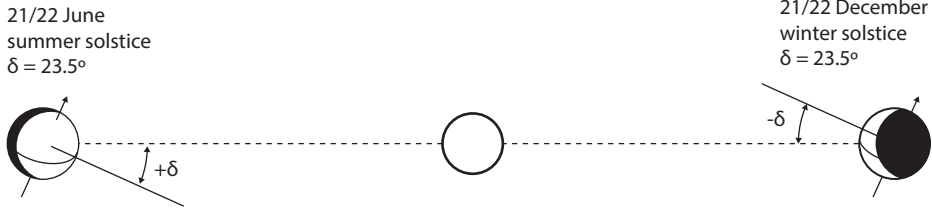


Figure 3.9 Positive and negative declination angles in summer and winter in the northern hemisphere.

The change in declination in 1 day is at most 0.5° and is often ignored. The declination can be calculated by a simple approximation equation or by a Fourier series:

$$\delta = 23.45 \sin \left(\frac{360}{365} (284 + n) \right) \quad (3.6)$$

$$\delta = \left(\begin{array}{l} 0.006918 - 0.399912 \cos B + 0.070257 \sin B - 0.006758 \cos 2B \\ + 0.000907 \sin 2B - 0.002697 \cos 3B + 0.00148 \sin 3B \end{array} \right) \frac{180^\circ}{\pi} \quad (3.7)$$

Hour angle and equation of time

While the declination clearly determines the position of the Earth in the ecliptic plane relative to the sun, the hour angle characterises the daily irradiance fluctuations due to the Earth's rotation. The hour angle is defined as the angle between local longitude and the longitude at which the sun is at its zenith. By hour angle $\omega = 0$ is meant the highest position of the sun during the day (the so-called upper culmination), i.e. in the northern hemisphere, the time when the sun is exactly in the south. The true solar time or the true local time TLT is 12.00 h, at the highest position of the sun.

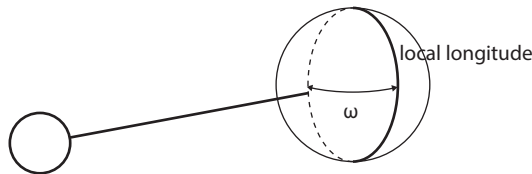


Figure 3.10 Hour angle ω of the sun as an angle deviation between local longitude and longitude with the sun at its zenith.

A sun's day of $\omega = 0^\circ$ to $\omega = 360^\circ$ is the period between two consecutive days in which the sun crosses the local meridian (degree of longitude of the observer) in each case. Since the earth does not move at a constant rate in its elliptical orbit, one sun-day is not exactly 24 hours long. At the nearest point to the sun (perihelion on 3 January), the velocity is highest, according to Kepler's second law, i.e. the sun-day is at its shortest; at the furthest point (aphelion on 4 July) it is smallest. This velocity-determined whole-year period is overlain by a second deviation with biannual periodicity,

which results from the projection of sections of orbit of equal length on the ecliptic plane onto sections of orbit of different lengths on the equatorial plane.

In order to calculate the hour angle and thus, the true solar time TLT from the local time-of-day with a day length fixed to 24 h, the local time must be corrected by this temporal deviation (equation of time E_t in minutes).

$$E_t = 229.2 \left(0.000075 + 0.001868 \cos B - 0.032077 \sin B - 0.014615 \cos 2B - 0.040849 \sin 2B \right) \quad (3.8)$$

A further correction of local time is necessary, as only zones of around 15° longitude have their own standard time; there is not one for each degree of longitude. In most Western European countries, Central European Time (CET) applies, which corresponds to 15° longitude east. For each degree of deviation between local (L_{local}) and Standard-Time Meridian (L_{zone}), a longitudinal correction L_c of 4 minutes must be taken into account ($360^\circ / (24 \text{ h} \times 60 \text{ min h}^{-1})$).

$$L_c = 4(L_{zone} - L_{local}) \text{ [min]} \quad (3.9)$$

The true local time TLT (or solar time) therefore results from the standard time corrected by the constant longitude correction, as well as by the equation of time. During Central European Summer Time, a further hour must be deducted.

$$\begin{aligned} TLT &= CET - L_c + E_t \\ &= CET_s - 1\text{h} - L_c + E_t \end{aligned} \quad (3.10)$$

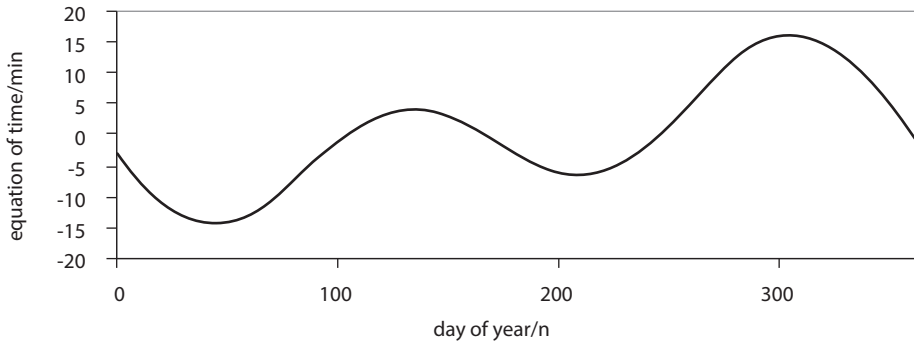


Figure 3.11 Equation of time as a deviation of the true solar time from universal time.

The hour angle ω results directly from the solar time. An hour angle of 15° corresponds to a 1 hour deviation from solar noon (in the morning negatively, in the afternoon positively).

$$\omega = (TLT - 12.00\text{h}) \times \frac{15^\circ}{\text{h}} \quad (3.11)$$

Example 3.1

Calculation of the hour angles of the sun at 12.00 h local time in Stuttgart (longitude 9.2° east) on 1 February, 1 July and 1 October (taking summer time into consideration).

For Stuttgart the result is a constant longitude correction of $L_c = 4 \text{ min/}^\circ \times (15^\circ - 9.2^\circ) = 23.2 \text{ min} = 0.387 \text{ h}$.

Date	Day number n	$B/^\circ$	E_t/min	E_t/h	TLT	$\omega/^\circ$
1 February	32	30.57	-13.1	-0.218	11.39h	-9.08
1 July (CEST)	182	178.52	-3.5	-0.058	10.55h	-21.68
1 October (CEST)	274	269.26	+10.5	0.175	10.79h	-18.18

For the calculation of the TLT both longitude correction and equation of time were converted decimally into hours.

3.4 Horizon coordinates

From the declination and hour angle, the sun's height α_s and azimuth γ_s can be determined for each location with latitude Φ . For these angles, described as horizon coordinates, the azimuth γ_s for the north direction is defined as 0° , east $+90^\circ$, south $+180^\circ$ and west 270° , and the elevation angle α_s is determined from the horizontal plane. The complementary angle to the elevation angle is the zenith angle θ_z , which at the same time represents the angle of incidence of direct solar irradiance on a horizontal surface.

$$\cos \theta_z = \sin \delta \sin \Phi + \cos \delta \cos \Phi \cos \omega = \sin \alpha_s \quad (3.12)$$

$$\gamma_s = \begin{cases} 180^\circ - \arccos\left(\frac{\sin \alpha_s \sin \Phi - \sin \delta}{\cos \alpha_s \cos \Phi}\right) & \text{for } TLT \leq 12.00\text{h} \\ 180^\circ + \arccos\left(\frac{\sin \alpha_s \sin \Phi - \sin \delta}{\cos \alpha_s \cos \Phi}\right) & \text{for } TLT > 12.00\text{h} \end{cases} \quad (3.13)$$

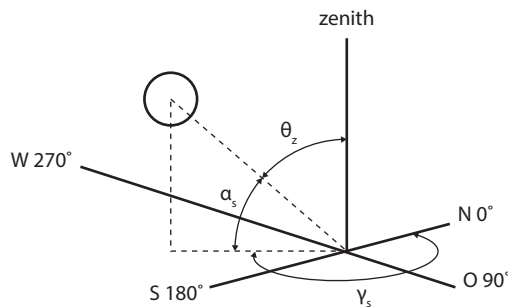


Figure 3.12 Zenith angle θ_z , elevation angle α_s and azimuth angle γ_s of the sun.

Example 3.2

a) Calculate the angles of the solar azimuth between the autumn and spring equinoxes in the Northern Hemisphere: between 23 September and 21 March, the solar azimuth always

lies between east and west, i.e. 90° and 270° . This results directly from Equation (3.13), as the declination is $\delta = 0^\circ$ and the elevation angle $\alpha_s = 0^\circ$ at sunrise and sunset, and thus, $\arccos(0) = 90^\circ$.

b) Determine the solar azimuth at the equator at 12.00 h *TLT* at the winter and summer solstice in the northern hemisphere:

- 21 December: Declination angle -23.45° , i.e. the sun is below the equator and is thus exactly in the south: $\gamma_s = 180^\circ$.
- 21 June: Declination angle $+23.45^\circ$, i.e. the sun is above the equator and is thus exactly in the north: $\gamma_s = 0^\circ$.

Sunrise time and length of day

At sunrise and sunset the zenith angle θ_z is exactly 90° . From it the hour angle ω_s results, based on Equation (3.12).

$$\cos \omega_s = -\frac{\sin \phi \sin \delta}{\cos \phi \cos \delta} = -\tan \phi \tan \delta \quad (3.14)$$

The number of daylight hours N can be calculated from the hour angle, since the hour angle changes by 15° every hour. The factor 2 results from taking into consideration morning and afternoon hours.

$$N = \frac{2}{15^\circ} \arccos(-\tan \phi \tan \delta) \quad (3.15)$$

Example 3.3

Calculation of the hour angles of sunrise and of the number of daylight hours for 1 February, 1 July and 1 October in Stuttgart (48.8° northern latitude).

	1 February	1 July	1 October
Declination $\delta/^\circ$	-17.5	23.1	-4.2
$\omega_s/^\circ$	-68.4	-119.1	-85.2
N/h	9 h 11 min	15 h 57 min	11 h 22 min

Sun-position diagrams

For the illustration of the sun's height and azimuth angles over the course of the year, at a given location, sun-position diagrams with either Cartesian or polar coordinates can be used. Cartesian coordinates with the elevation angle as a function of the azimuth are particularly suitable for the representation of shading horizons. Shaded objects can easily be entered into the sun-position diagram with the respective azimuth and elevation angles, and the times of shading can be directly seen.

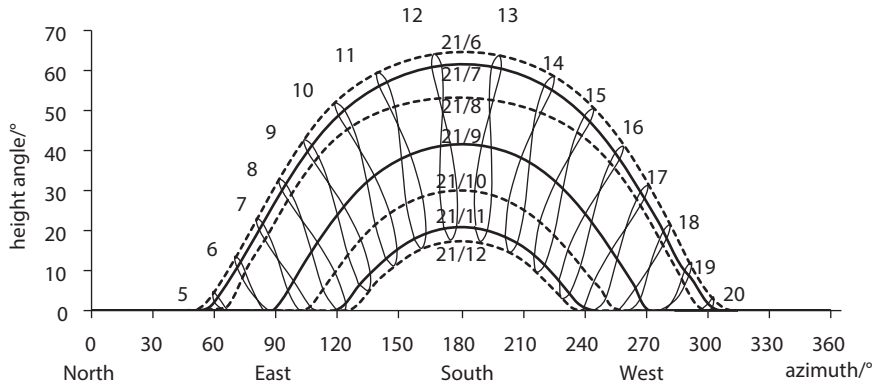


Figure 3.13 Sun-position diagrams in Cartesian coordinates for Stuttgart.

The elevation angles calculated for the second half of the year can be used symmetrically for the first half (21 July corresponds to 21 May, etc.). The lines of same local time arise from the equation of time, here CET (the so-called analemma). While Cartesian coordinates illustrate the elevation angles of the sun and can include the height angles shading objects, polar diagrams make clear the azimuth angles of the sun and the azimuth location of shading objects.

The outer circle corresponds to an elevation angle of zero, i.e. the horizon, and the centre of the polar diagram corresponds to the zenith.

Angle of incidence on randomly inclined surfaces

The angle of incidence of direct rays on an inclined receptor surface depends on the angle of inclination β of the surface to the horizontal, and also on the surface azimuth γ . From a given position of the sun (zenith angle θ_z and solar azimuth γ_s), the angle of incidence can be calculated from the horizon coordinates θ_z and γ_s :

$$\cos \theta = \cos \theta_z \cos \beta + \sin \theta_z \sin \beta \cos(\gamma_s - \gamma) \quad (3.16)$$

or, when using equator coordinates δ and ω ,

$$\begin{aligned} \cos \theta = & (\cos \beta \sin \Phi + \cos \Phi \cos \gamma \sin \beta) \sin \delta \\ & + (\cos \beta \cos \Phi - \sin \Phi \cos \gamma \sin \beta) \cos \delta \cos \omega \\ & - \sin \gamma \sin \beta \cos \delta \sin \omega \end{aligned} \quad (3.17)$$

Simplifications of the formula result e.g. from horizontal surfaces with $\beta = 0^\circ$ (Equation (3.12) of the zenith angle) or vertical and south-orientated facade surfaces ($\beta = 90^\circ$, $\gamma = 180^\circ$):

$$\cos \theta = -\cos \Phi \sin \delta + \sin \Phi \cos \delta \cos \omega \quad (3.18)$$

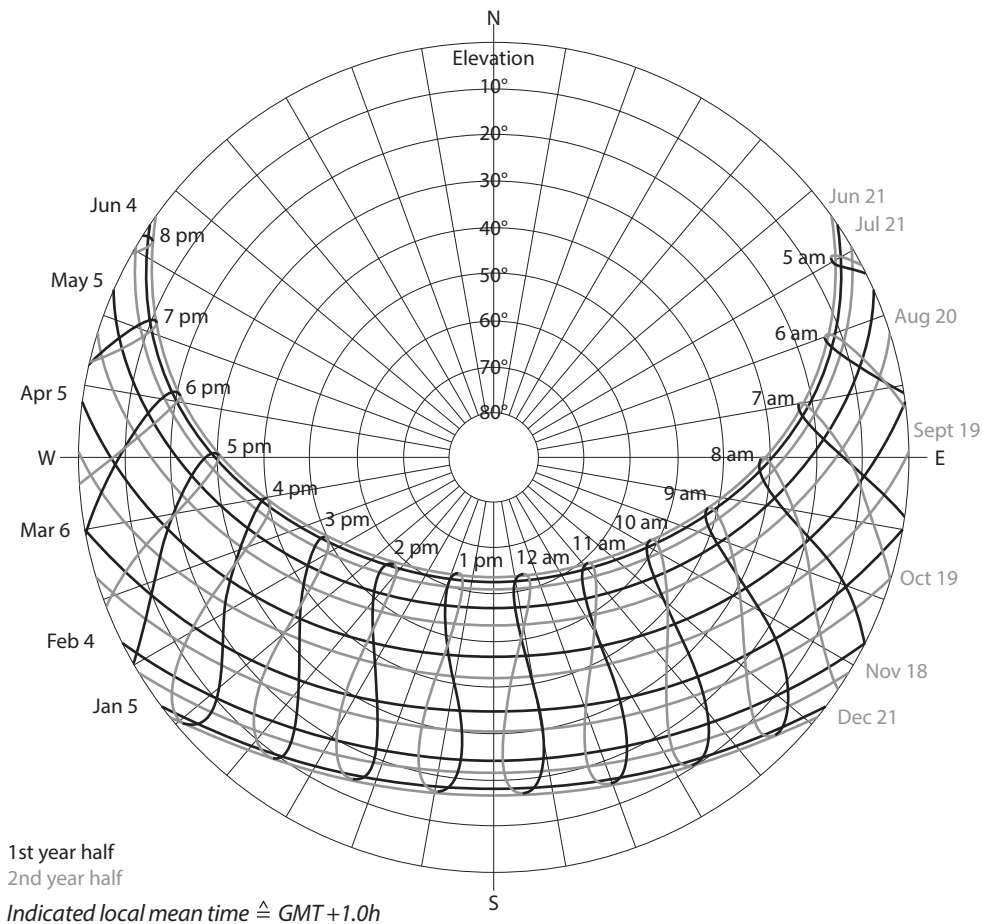


Figure 3.14 Sun-position diagram in polar coordinates for Stuttgart (University of Bochum, 1999).

At solar noon with $\omega = 0^\circ$, the following formula applies for a south-orientated surface:

$$\theta_{noon} = |\Phi - \delta - \beta| \tag{3.19}$$

For a horizontal surface with $\beta = 0^\circ$ the maximum and minimum positions of the sun in a year can thus be quickly determined. The highest position in Stuttgart on 21 July with a declination of 23.45° results for a zenith angle θ_z of $48.8^\circ - 23.45^\circ = 25.35^\circ$, which corresponds to a sun height of 64.65° . The lowest position of the sun at solar noon is with a declination of 23.45° on 21 December and a zenith angle of $48.8^\circ - (-23.45^\circ) = 72.25^\circ$, i.e. a sun height of 17.75° .

Example 3.4

Calculation of the angle of incidence on a 10° inclined surface with a surface azimuth of 160° on 1 October at 11.00 h *TLT* in Stuttgart.

The zenith angle is $\theta_z = 54.6^\circ$, the sun azimuth $\gamma_s = 161.5^\circ$. Thus, an angle of incidence of 44.6° results. The same result is obtained by use of the equator coordinates $\delta = -4.2^\circ$ and $\omega = -15^\circ$.

3.5 Atmospheric transmission and spectral irradiance

While extra-terrestrial irradiance values on randomly orientated surfaces are solely geometry-dependent and thus simple to calculate, the passage of rays through the atmosphere is so complex that simple procedures like the use of monthly turbidity factors result in extremely inaccurate irradiance values. Particularly for dynamic system simulations of active or passive solar components, hourly resolved irradiance values of the direct and diffuse irradiance are necessary that must indicate the statistical characteristics typical of the location. Since long-term measured or partly synthesised irradiance time sequences are available for only a few locations worldwide (e.g. test reference years for 12 German climate zones), more and more statistical procedures are becoming generally accepted that produce hourly irradiance values from monthly average ones. After a short illustration of the main irradiance absorption and dispersion mechanisms of the atmosphere, the following deals mainly with the statistical procedures of irradiance calculation.

Extra-terrestrial irradiance is weakened in the atmosphere by absorption and reflection and is partially converted by dispersion into diffuse irradiance. The relative air mass m (also termed AM), which solar irradiance has to pass through, gives the relation of atmospheric thickness for a given zenith angle, i.e. $d_{atm} / \cos \theta_z$, to the simple thickness of the atmosphere d_{atm} in the local zenith, and can be calculated for a homogeneous atmosphere by a simple approximation formula:

$$m = \frac{d_{atm} / \cos \theta_z}{d_{atm}} = \frac{1}{\cos \theta_z} \quad (3.20)$$

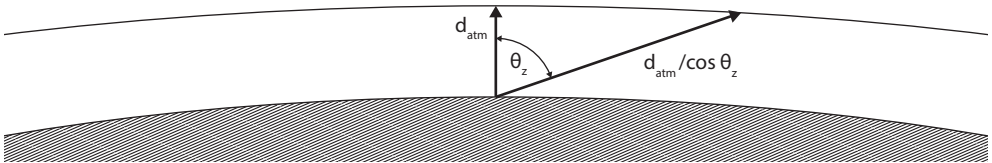


Figure 3.15 Definition of air mass m from atmospheric thickness and zenith angle.

Short-wave irradiance is scattered by air molecules, whose diameter are small in relation to the wavelength of light (around 10^{-10} m), proportionally to $1/\lambda^4$. Above around $0.6 \mu\text{m}$, the so-called Rayleigh scattering is negligible. Mie dispersion by larger dust particles (aerosols) with diameters of around 10^{-9} m reduces the transmittance with a weaker wavelength dependence. Ozone absorbs solar irradiance almost completely below $\lambda = 0.29 \mu\text{m}$ and more weakly to around $0.7 \mu\text{m}$. Water vapour absorbs in the infrared, with pronounced absorption bands at 1.0 , 1.4 and $1.8 \mu\text{m}$. Above $2.5 \mu\text{m}$, almost the entire irradiance is absorbed by CO_2 and H_2O .

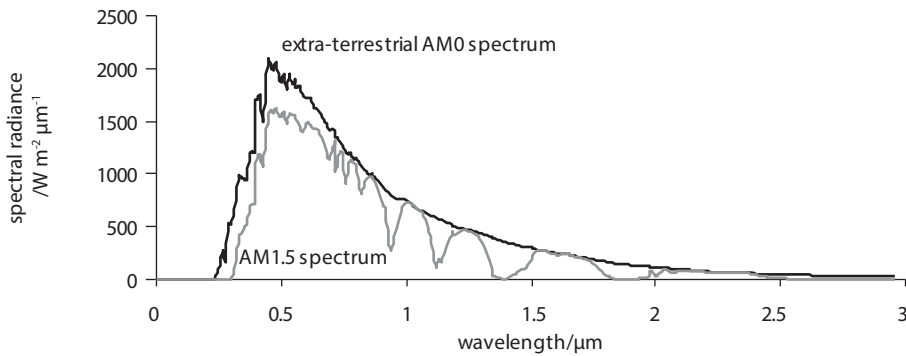


Figure 3.16 An extra-terrestrial spectrum with zero air mass (AM0) and a terrestrial spectrum with air mass 1.5 (AM1.5).

Spectral modelling

A semi-empirical model was developed by (Nann, 1992) to simulate short-wave irradiance spectra from only three readily available meteorological data: global and diffuse irradiance (alternatively direct irradiance), and dew point temperature (alternatively relative humidity and ambient temperature). Over 3 years, 100 000 solar spectra were recorded at the Centre for Solar Energy and Hydrogen Research in Stuttgart/Germany in the wavelength range from 300 to 1100 nm. These measured spectra were utilised to develop the semi-empirical model SEDES2 that calculates hemispherical solar spectral irradiance on a south-tilted surface. The use of atmospheric surface pressure instead of site elevation slightly improves calculations. The model is implemented in the simulation environment INSEL (www.insel.eu) and simulates hemispherical irradiance from 300 to 1400 nm with 10 nm resolution using hourly input data of irradiance and dew point. SEDES2 consists of the clear sky approximation code SPCTRAL2, a normalisation procedure and cloud cover modifiers derived from statistical analysis of the measured spectra. There are four cloud cover modifiers for each wavelength corresponding to four zenith angle ranges. The cloud cover modifiers are slightly site dependent mainly because of particular site turbidity and surface albedo.

The simple cloudless-sky irradiance model called SPCTRAL2 was developed using comparisons with more complex models and limited outdoor measurements. SPCTRAL2 calculates irradiance at 122 wavelengths that were selected to minimise the number of wavelength calculations while retaining the major structure in the spectrum. SPCTRAL2 is a one-layer model, which considers the atmosphere as a uniform layer with single scattering only.

Input to the model includes latitude, longitude, day of the year, solar zenith angle, aerosol optical depth at 0.5 μm (turbidity), perceptible water vapour, surface pressure, ground reflectance and direct and diffuse irradiance.

The model uses data of extra-terrestrial spectral irradiance at 122 wavelengths, together with 122 absorption coefficients for water vapour, ozone and uniform gases. It then calculates the transmission losses through the atmosphere for the direct beam spectral irradiance and the diffuse fraction from atmospheric scattering.

The transmission losses by Rayleigh scattering depend on wavelength and air mass and are obtained from

$$\tau_R = \exp\left(-m / \lambda^4 \times 115.6406 - 1.3366 / \lambda^2\right) \quad (3.21)$$

The transmission loss from water vapour absorption is given by

$$\tau_W = \exp\left(-2.385 \alpha_W(\lambda) W m\right) \frac{1}{\left(1.0 + 20.07 \alpha_W(\lambda) W m\right)^{0.45}} \quad (3.22)$$

The water vapour column W of the atmosphere is calculated according to Wright and Perez using the dew point temperature T_{dew}

$$\ln\left(\frac{W}{cm}\right) = -0.0756 + 0.0693 T_{dew} \quad (3.23)$$

The transmission of uniform gases is obtained as follows:

$$\tau_U = \exp\left(-1.41 \alpha_U(\lambda) W m\right) \frac{1}{\left(1.0 + 118.3 \alpha_U(\lambda) W m\right)^{0.45}} \quad (3.24)$$

The wavelength-dependent extinction by aerosols is generally calculated using the Angstrom formula.

$$\delta_A(\lambda) = \beta_A \left(\frac{\lambda}{1\mu m}\right)^{-\alpha_A} \quad (3.25)$$

where

δ_A is the wavelength-dependent optical thickness of the aerosols, which is calculated using the Angstrom turbidity coefficient β_A and a typical value of 1.3 for α_A , which can vary between 0.5 and 2.5, depending on aerosol size.

SPCTRAL2 reaches a precision of a few percent ages for the direct beam part and less than 7% for diffuse irradiance for weak absorption. Spectral measurements carried out in Stuttgart on clear days showed that best agreement between model and measurement was obtained for on optical aerosol thickness for $\delta_A(500\text{ nm}) = 0.27$ and $a = 1.14$. The optical thickness is thus given by:

$$\delta_A(\lambda) = 0.27 \left(\frac{\lambda}{0.5\mu m}\right)^{-1.14} \quad (3.26)$$

and the transmission coefficient from aerosols by the negative exponent of the product of optical thickness and air mass:

$$\tau_A = \exp\left(-\delta_A(\lambda) m\right) \quad (3.27)$$

The aerosol single-scattering albedo is the ratio of scattering to extinction and an important parameter in climate modelling. In the SPCTRAL2 model, the aerosol single-scattering albedo at

400 nm is 0.945 with a variation factor of 0.095 for all other wavelengths. The single-scattering albedo is obtained from:

$$SSA = 0.095 \exp \left(-0.095 \log \left(\frac{\lambda}{0.4} \right)^2 \right) \quad (3.28)$$

Total aerosol scattering is then calculated from:

$$s_A = \exp(-SSA \delta_A(\lambda) m) \quad (3.29)$$

The aerosol asymmetry factor a_{sym} is 0.65 for rural aerosol models, 0 would be isotropic. The forward-scattered ratio for any air mass is given by:

$$F_s = 1.0 - 0.5 \exp(A_{fs} + B_{fs} \cos \theta_z) \cos \theta_z \quad (3.30)$$

with

$$\begin{aligned} A_{fs} &= A_{LG} (1.459 + A_{LG} (0.1595 + A_{LG} 0.4129)) \\ B_{fs} &= A_{LG} (0.073 + A_{LG} (-0.3824 - A_{LG} 0.5874)) \\ A_{LG} &= \log(1 - a_{sym}) \end{aligned}$$

Latitude, longitude and day of the year are used to calculate the amount of ozone, which varies with location and time of year.

$$\tau_o = \exp(-\alpha_o(\lambda) O m_o) \quad (3.31)$$

The ozone mass m_o is obtained from the height of the ozone layer and the zenith angle θ on a horizontal plane and is approximated empirically:

$$m_o = \frac{(1 + 22/6370)}{\left((\cos \theta)^2 + 2 \times 22/6370 \right)^{0.5}} \quad (3.32)$$

For the Northern hemisphere and Eastern (positive) latitudes, the factor O is given as a function of day number, latitude and longitude.

$$O = \left(235 + \left(150 + 40 \sin \left(\frac{0.9864 \times 3.14}{180 \times n_{day} - 30} \right) \right) + 20 \sin \left(\frac{3 \times 3.14}{180 \times (long + 20)} \right) \right) \times \left(\sin \left(\frac{1.28 \times 3.14}{180 \times lat} \right) \right)^2 \times 0.001 \quad (3.33)$$

For Western (negative) latitudes on the Northern hemisphere, the addition of the number 20 in the longitude expression is left out.

The transmitted direct normal irradiance is then calculated from the product of the extra-terrestrial spectral irradiance $I_{\lambda,ex}$ and the transmission coefficients described above.

$$I_{\tau,nor}(\lambda) = I_{\lambda,ex} \tau_R \tau_W \tau_U \tau_O \tau_A \quad (3.34)$$

The transmitted direct irradiance on a horizontal plane is given by:

$$I_{\tau,hor}(\lambda) = I_{\tau,nor}(\lambda) \cos(\theta_z) \quad (3.35)$$

To calculate the diffuse spectral irradiance, the scattering processes and ground diffuse reflection need to be considered.

The intensity of Rayleigh scattering is highest in forward and backward direction of the rays and minimal at 90° angle. The diffuse contribution of Rayleigh scattering is obtained from:

$$I_{D,R} = I_{\lambda,ex} \cos(\theta_z) \tau_O \tau_W \tau_U \tau_A (1 - \tau_R^{0.95})^{0.5} \quad (3.36)$$

The aerosol contribution to the diffuse component is given by:

$$I_{D,A} = I_{\lambda,ex} \cos(\theta_z) \tau_O \tau_W \tau_U \tau_A \tau_R^{1.5} (1 - s_A) F_s \quad (3.37)$$

With a typical ground albedo $\rho = 0.2$, both the transmitted direct beam and the transmitted Rayleigh and aerosol-scattered components are reflected back. The atmosphere then reflects with an atmospheric reflection coefficient ρ_A , which again contains reflection from the aerosols, the water vapour and the uniform gases.

The atmospheric reflection coefficient is obtained from:

$$\rho_A = \tau_{UP} \tau_{WP} s_{AP} (0.5(1 - \tau_{RP})) - (1 - F_s) \tau_{RP} (1 - s_A) \quad (3.38)$$

using:

$$\tau_{RP} = \exp\left(\frac{-1.8}{\lambda^4}\right) \left(115.6406 - \frac{1.3366}{\lambda^2}\right) \quad (3.39)$$

$$\tau_{WP} = \frac{\exp(-0.2385 \alpha_W(\lambda) W 1.8)}{(1.0 + 20.07 \alpha_W(\lambda) W 1.8)^{0.45}} \quad (3.40)$$

$$\tau_{UP} = \frac{\exp(-1.41 \alpha_U(\lambda) 1.8)}{(1.0 + 118.3 \alpha_U(\lambda) 1.8)^{0.45}} \quad (3.41)$$

The single scattering albedo is calculated for a fixed air mass of 1.8.

$$s_A = \exp(-SSA \delta_A(\lambda) 1.8) \quad (3.42)$$

$$s_{AP} = \exp(-(1 - SSA) \delta_A(\lambda) 1.8) \quad (3.43)$$

This gives the diffuse multi-reflection component between ground and sky:

$$I_{Dp} = \left(I_{\tau,nor} \cos \theta_z + (I_{D,R} + I_{D,A}) \right) \frac{\rho \rho_A}{1 - \rho \rho_A} \quad (3.44)$$

The total horizontal wavelength-dependent irradiance is then given by:

$$I_{hor}(\lambda) = I_{\tau,DIR}(\lambda) + I_{DIFF}(\lambda) = I_{\tau,nor}(\lambda) \cos \theta_z + (I_{D,R}(\lambda) + I_{D,A}(\lambda) + I_{D,p}(\lambda)) \quad (3.45)$$

Cloud cover is calculated using empirical spectrally dependent cloud correction factors and multiplied with the clear sky spectrum. The parameter to choose the best cloud cover coefficients is a modified clearness index k_t' , which is defined as the ratio of measured global horizontal irradiance and the integral of the simulated clear sky spectrum.

$$k_t' = \frac{I_h}{\int_{300nm}^{4000nm} I(\lambda) d\lambda} \quad (3.46)$$

This normalisation already removes the dependence of spectral irradiance on water content and air mass. The cloud cover correction functions have been determined from 3 years of continuous spectral measurements in Stuttgart and are a function of zenith angle and clearness index.

$$CCM(\lambda, \theta_z, k_t') = A_0(\lambda) + A_1(\lambda) \frac{1}{\cos \theta_z} + B_0(\lambda) k_t' + B_1(\lambda) \frac{k_t'}{\cos \theta_z} + C_0(\lambda) k_t'^2 + C_1(\lambda) \frac{k_t'^2}{\cos \theta_z} \quad (3.47)$$

A total of 74 parameters for the wavelength range of 320 - 1050 nm have been implemented in the simulation environment INSEL (www.insel.eu). Below and above these wavelengths the cloud cover modifiers are kept constant at the boundary values.

The main influences of cloud cover can be summarised as follows:

- The relative transmission of clouds increases below 650 nm with decreasing wavelength. This effect increases with decreasing clearness index, i.e. higher optical thickness of water vapour.
- The absorption around the water vapour absorption band at 938 nm increases with cloud cover, i.e. the cloud cover modifier function drops.

The cloud cover modification function increases above 1.0 for wavelengths between about 700 and 900 nm. Here multiple reflections from the ground with high reflections above the chlorophyll absorption at 650 nm lead to higher spectral contributions.

The cloud cover modifiers were calculated exemplary for day 180, 12:00 am, for clearness index variations between 0.11 and 0.99. The above-described effects can be clearly seen, especially for low clearness indices.

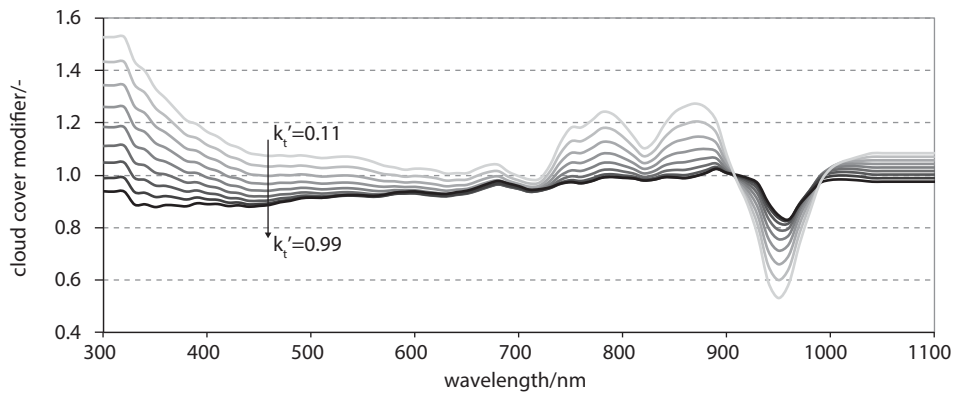


Figure 3.17 Cloud cover modifiers for clearness index variations between 0.11 (top line) and 0.99 (bottom line) for day 180, 12 am.

The mean irradiance intensity weighted spectra for the Southern German location of Munich have been simulated. Whereas the mean monthly summer spectrum corresponds well to the AM1.5 spectrum, the December spectrum has a much lower short-wave contribution, which is due to higher scattering of short-wave irradiance at high air masses.

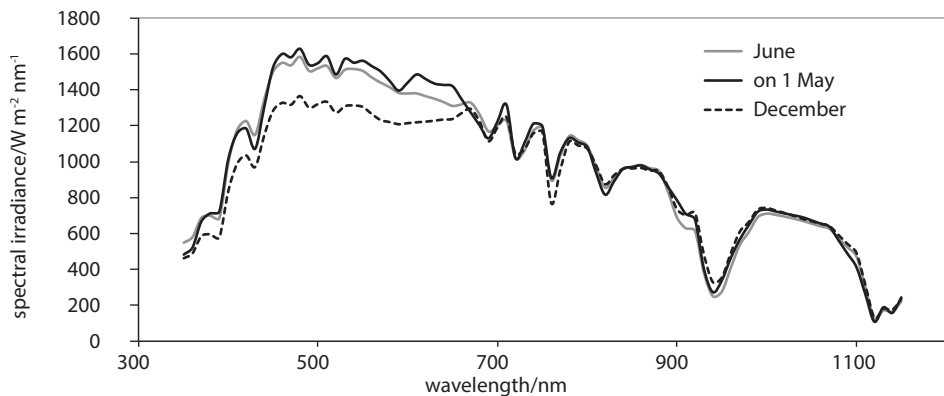


Figure 3.18 Simulated mean monthly spectral irradiance as a function of wavelength for the German location Munich.

The simplest calculation models of irradiance at the earth's surface combine all the above wavelength-dependent effects in a single number, the so-called turbidity factor T , which does not take into account the statistical fluctuations of terrestrial irradiance. T is indicated in some standards as a monthly average value, and varies from a minimum of 3.8 in January to 6.3 in September for German climatic conditions. With a clear sky, the direct irradiance on a normal

surface is air mass-corrected from the extra-terrestrial irradiance by the sun-height angle α_s , and pressure-corrected by the height of the location above sea level H (m):

$$G_n = G_{en} \exp \left(- \frac{T}{\underbrace{(0.9 + 9.4 \sin \alpha_s)}_{\text{air mass correction}} \underbrace{\exp(H/8000)}_{\text{pressure correction}}} \right) \quad (3.48)$$

The proportion of diffuse irradiance in a clear sky can likewise be calculated as a function of the sun's height and the turbidity factor. The conversion for inclined surfaces takes place by means of tabulated correction factors.

For an overcast sky the model becomes even simpler. Daylighting calculations are often based on symmetrical irradiance or luminous intensity distributions with a maximum at the local zenith and decrease towards the horizon. The irradiance intensity in the zenith L_{ez} is determined by:

$$L_{ez} = 1.068 + 74.7 \sin \alpha_s \left[\frac{W}{m^2 \text{ sr}} \right] \quad (3.49)$$

If the irradiance intensity is integrated over the entire sky hemisphere, the following formula is obtained for the horizontal irradiance.

$$G_h = 2.609 + 182.609 \sin \alpha_s \left[\frac{W}{m^2} \right] \quad (3.50)$$

While these simple procedures might be sufficient for the design of daylight systems under overcast sky conditions, more exact meteorological data records are necessary for the sizing of solar energy systems.

3.6 Statistical production of hourly irradiance data records

The statistical procedures described below enable the production of a series of hourly irradiance values, a so-called time series, proceeding from a monthly average irradiance value. In order to eliminate the deterministic proportion of the irradiance on the Earth's surface, which is determined by extra-terrestrial irradiance and the respective position of the sun, the clearness index is used as a statistical variable. The clearness index k_t is defined as the relation of terrestrial to extra-terrestrial irradiance on a horizontal surface, calculated for 1 hour or totalled over the hours of a day or month.

$$k_t = \frac{\sum G_h}{\sum G_{eh}} \quad (3.51)$$

First, daily values are produced by an autoregressive procedure from the given monthly clearness index, and afterwards, hourly values are calculated.

Daily average values from monthly average values

Two observations of long-term irradiance data form the basis for the statistical production of daily averages from monthly average values:

- Each daily irradiance average value correlates only to the preceding daily value.
- The probability distribution of daily clearness indices around the monthly average value is only determined by the average clearness index. For example, clear months have only small dispersions of the daily values around the average value, and vice versa.

The time series of daily clearness indices can be calculated either by Markov transition-matrices or by autoregressive procedures. Autoregressive procedures are more generally used, since not only correlations with the preceding daily value, but also with values of more than a one-day interval, can be taken into account. For a normally distributed random variable Z_d with an average value of zero and a standard deviation of $\sigma = 1$, the new Z_d values from the correlated preceding values as well as a noise term r_d result. The order n of the regression procedure gives the correlations which must be considered with values of more than a one-day interval.

$$Z_d = \rho_1 Z_{d-1} + \rho_2 Z_{d-2} + \dots + \rho_n Z_{d-n} + r_d \quad (3.52)$$

For the clearness index calculation, a first-order autoregressive procedure is sufficiently exact, since the daily averages correlate particularly with the preceding day, and days which are longer ago have hardly any influence on the clearness index. ρ_1 defines the autocorrelation coefficient for an interval of one day ($n = 1$). For a given time series of N random variables Z_d , the autocorrelation coefficients ρ_n with time interval n can be calculated:

$$\rho_n = \frac{\sum_{i=1}^{N-n} (Z_i - \bar{Z})(Z_{i+n} - \bar{Z})}{\sum_{i=1}^N (Z_i - \bar{Z})^2} \quad (3.53)$$

However, since time series are to be synthesised here and, therefore, their correlation characteristics are unknown, the autocorrelation coefficient of first order ρ_1 has to be given as a parameter. From meteorological data investigations by Gordon and Reddy (1988), it is evident that ρ_1 can vary between 0 and 0.6, depending upon location, but in most cases, a value of 0.3 is a good approximation.

The statistical noise r_d is, with an average value of zero, normally distributed, and defined with a standard deviation

$$\sigma' = \sqrt{1 - \rho_1^2}$$

The noise is calculated from a random number sequence z from the value-area [0,1]:

$$r_d = \sigma' (z^{0.135} - (1-z)^{0.135}) / 0.1975 \quad (3.54)$$

Therefore, for the production of the time series of the normally distributed variables Z_d , the autocorrelation coefficient ρ_1 is given, the random variable Z_d is initialised with $Z_0 = 0$ and for each time step with the noise term r_d , the next value is calculated.

Gordon and Reddy have shown, however, that the daily clearness indices are not normally distributed around the mean monthly value, but that the probability function above the mean clearness index sinks faster than the Gaussian distribution. Therefore the time series of the

Gaussian-distributed random variables Z_d , produced in the first step, must be converted into the non-Gaussian-distributed variable X_d .

For the random variable X_d , the relation of the daily mean clearness index k_{td} to the monthly average value $\overline{k_{tm}}$ is chosen.

$$X_d = \frac{k_{td}}{\overline{k_{tm}}} \quad (3.55)$$

The empirical probability function $P(X_d)$ determined by Gordon and Reddy describes with good accuracy the distribution of the daily clearness indices. The only parameter is the standard deviation of the daily values σ_{X_d} for the respective location, which depends only on the monthly average value $\overline{k_{tm}}$.

$$P(X_d) = AX_d^n \left(1 - \frac{X_d}{X_{max}} \right) \quad (3.56)$$

where

$$n = -2.5 + 0.5 \sqrt{9 + \frac{8}{\sigma_{X_d}^2}} \quad (3.57)$$

$$X_{max} = \frac{n+3}{n+1} \quad (3.58)$$

$$A = \frac{(n+1)(n+2)}{(X_{max})^{n+1}} \quad (3.59)$$

The dispersion σ_{X_d} of the daily values around the monthly average value decreases with rising monthly clearness index at the location, and can be described by a linear function:

$$\sigma_{X_d}^2 = \begin{cases} 0.1926 & \text{for } \overline{k_{tm}} \leq 0.2 \\ \max\{0.01, (0.269 - 0.382\overline{k_{tm}})\} & \text{for } \overline{k_{tm}} > 0.2 \end{cases} \quad (3.60)$$

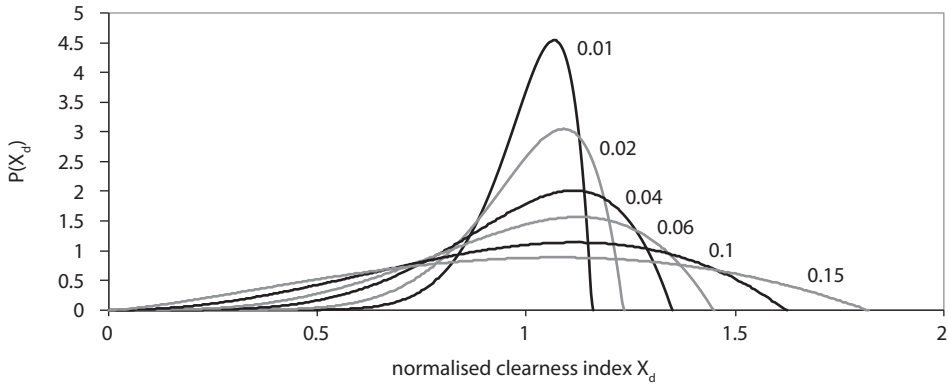


Figure 3.19 Gordon–Reddy distribution function of the standardised daily clearness indices around the monthly average value as a function of the variance σ^2 .

In clear months where $\sigma^2 < 0.1$ the daily clearness indices above the monthly average value ($X_d = 1$) drop very sharply.

The initially assumed Gaussian-distributed variable Z_d is converted by a transformation known as Gaussian mapping into the actual probability distribution $P(X_d)$ and so finally the actual time sequence for X_d is obtained. In Gaussian mapping, first the cumulated distribution $F(Z_d)$ of the Gaussian-distributed random variables Z_d is calculated:

$$F(Z_d) = \frac{1}{2} \left(1 \pm \sqrt{1 - \exp\left(\frac{-2Z_d^2}{\pi}\right)} \right) \quad (3.61)$$

with a positive sign for $Z_d > 0$ and a negative sign for $Z_d < 0$.

Following this, the value $F(Z_d)$ is equated with the cumulated value of the non-Gaussian distributed variables $F(X_d)$, and from this, the associated X_d value is unambiguously determined. For X_d an implicit equation results, which must be solved iteratively:

$$AX_d^{n+1} = \frac{F(Z_d)}{1 - \frac{X_d}{(n+2)X_{max}}} \quad (3.62)$$

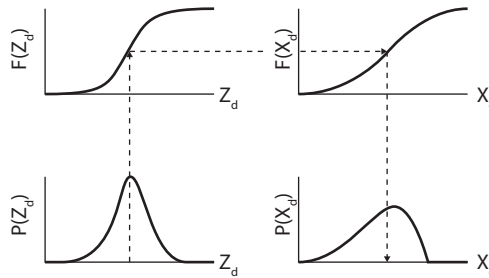


Figure 3.20 Conversion of the normally distributed random variables Z_d into Gordon–Reddy distributed random variables X_d on the basis of Gaussian Mapping.

From the thus calculated X_d values, a series of daily values of the clearness index is obtained, with the probability distribution $P(X_d)$ and autocorrelation coefficient ρ_1 .

Example 3.5

Calculation of the first six daily k_{td} values for a mean monthly $\overline{k_{tm}}$ value of 0.5 for the month of July ($m = 7$) with an autocorrelation coefficient ρ_1 of 0.3.

Day number	d = 1	2	3	4	5	6
Random number z [0,1]	0.3	0.1	0.9	0.65	0.2	0.5
Noise term r_d	-0.495	-1.22	1.22	0.365	-0.799	0
Z_d	-0.495	-1.37	0.811	0.608	-0.617	-0.185
$F(Z_d)$	0.31	0.082	0.792	0.729	0.268	0.4266
X_d	0.87	0.57	1.25	1.2	0.83	0.97
k_{td}	0.434	0.285	0.625	0.6	0.415	0.485

with $\sigma^2_{X_d} = 0.078$, $X_{max} = 1.53$, $n = 2.78$ and $A = 3.63$.

A 1 year time sequence of daily irradiated energy on a horizontal receptor surface in Stuttgart (48.8° northern latitude) shows the fluctuations of daily irradiance produced by the auto regression method. The yearly total irradiance is about 1190 kWh m⁻².

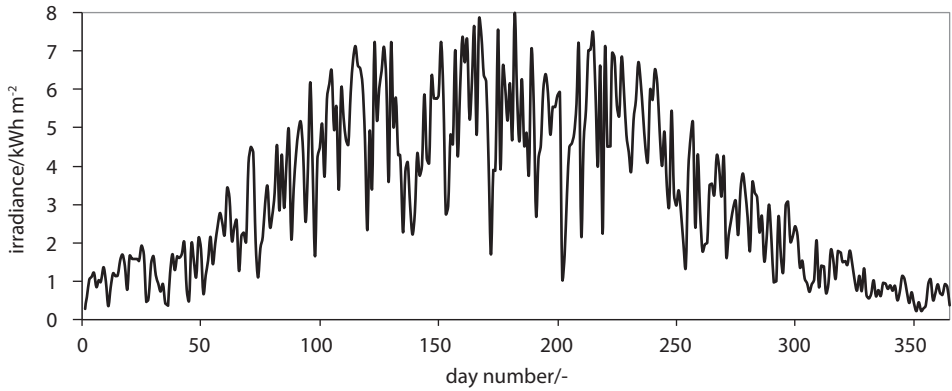


Figure 3.21 Daily irradiance on a horizontal surface in Stuttgart.

Hourly average values from daily average values

Hourly values can likewise be calculated from daily averages by an autoregressive model. Analysis of long-term hourly irradiance data shows that significant correlations exist only between directly consecutive hours. Since the hourly average values of the clearness index are normally distributed, the first-order autoregressive procedure can be used directly.

$$y_h = \rho_1 y_{h-1} + r_h \quad (3.63)$$

In order to eliminate the deterministic characteristics of the hourly clearness index, i.e. dependency on sun height, first the expected clearness index $\langle k_{th} \rangle$ of the respective hour is calculated, which depends on the average daily clearness index k_{td} and the zenith angle θ_z . The variable y_h then describes the difference, normalised by the standard deviation, between the hourly, statistically produced clearness index k_{th} and the expected clearness index $\langle k_{th} \rangle$ of the respective hour.

$$y_h = \frac{k_{th} - \langle k_{th}(k_{td}) \rangle}{\sigma_h} \quad (3.64)$$

The noise r_h is calculated according to Equation (3.54) with standard deviation

$$\sigma' = \sqrt{(1 - \rho_1^2)}$$

The autocorrelation coefficient ρ_1 depends only slightly on the average daily clearness index and is about 0.38,

$$\rho_1 = 0.38 + 0.06 \cos(7.4k_{td} - 2.5) \quad (3.65)$$

where the cosine term has to be calculated in arc measure.

Crucial for the quality of the model is the parameterising of the expected hourly clearness index, which is then varied by the random number procedure. Aguiar and Collares-Pereira (1992) have shown that very good results can be obtained with an angle-dependent exponential function.

$$\langle k_{th} \rangle = \lambda + \varepsilon \exp(-\kappa / \sin \alpha_s) \quad (3.66)$$

The parameters of the exponential function are functions of the average daily clearness index. They are empirically derived from a database of one African and 13 European weather stations.

$$\lambda = -0.19 + 1.12k_{td} + 0.24 \exp(-8k_{td}) \quad (3.67)$$

$$\varepsilon = 0.32 - 1.6(k_{td} - 0.5)^2 \quad (3.68)$$

$$\kappa = 0.19 + 2.27k_{td}^2 - 2.51k_{td}^3 \quad (3.69)$$

The standard deviation σ_h of the average hourly clearness index k_{th} is likewise empirically derived from the database values and is represented by a sun-height-dependent exponential function.

$$\begin{aligned} \sigma_h &= A \exp(B(1 - \sin \alpha_s)) \\ A &= 0.14 \exp(-20(k_{td} - 0.35)^2) \\ B &= 3(k_{td} - 0.45)^2 + 16k_{td}^5 \end{aligned} \quad (3.70)$$

The starting value y_0 for the first hour after sunrise is initialised with zero, and the next value is calculated with the autocorrelation coefficient ρ_1 and the noise r_h according to Equation (3.63). The hourly average value of the clearness index results from the definition of the variable y_h .

$$k_{th} = \langle k_{th} \rangle + \sigma_h y_h \quad (3.71)$$

Meaningful k_{th} values must be zero or greater and be below a maximum clearness index for extremely clear sky conditions. A simple approximation for the maximum k_{th} value is

$$k_{th,max} = 0.88 \cos(\pi(h - 12.5)/30) \quad (3.72)$$

For k_{th} values outside the required range of values, new random numbers must be generated.

Example 3.6

Calculation of hourly k_{th} values (solar time-of-day) for the fourth day of the preceding example (4 July = day 185) with an average daily clearness index of 0.6 in Stuttgart.

The hour angle for sunrise and sunset is $\omega = 118.8^\circ$, i.e. the first hour is between 4 am and 5 am. First, those values which need only be calculated once are determined. These include:

1. The autocorrelation coefficient ρ_1 :

$$\rho_1 = 0.38 + 0.06 \cos(7.4k_{td} - 2.5) = 0.44$$

2. The standard deviation σ' of the noise term r_h :

$$\sigma' = \sqrt{(1 - \rho_1^2)} = 0.898$$

3. The parameters for the expected hourly value:

$$\lambda = -0.19 + 1.12k_{td} + 0.24 \exp(-8k_{td}) = 0.485$$

$$\varepsilon = 0.32 - 1.6(k_{td} - 0.5)^2 = 0.304$$

$$\kappa = 0.19 + 2.27k_{td}^2 - 2.51k_{td}^3 = 0.465$$

4. The parameters A, B for the standard deviation of the hourly k_{th} values:

$$A = 0.14 \exp(-20(k_{td} - 0.35)^2) = 0.0397$$

$$B = 3(k_{td} - 0.45)^2 + 16k_{td}^5 = 1.3229$$

With the autocorrelation coefficient and the noise term r_h , which can be calculated via random numbers z , the time series of the variables y_h can be obtained. Then for each hour the expected value $\langle k_{th} \rangle$ and the standard deviation σ_h can be calculated via the sun's height α_s at mid-hour, and thus, from y_h the time series of the hourly clearness index k_{th} can also be calculated.

The random variable $y_h = 1$ is initialised with zero, since no correlation exists with the hourly value before sunrise $y_h = 0$.

h	TLT	z	r_h	y_h	α_s	$\langle k_{th} \rangle$	σ_h	k_{th}
1	4.30h	-	-	0	3.5	0.485	0.138	0.485
2	5.30h	0.3	-0.468	-0.468	12.3	0.519	0.112	0.467
3	6.30h	0.1	-1.15	-1.356	21.9	0.572	0.091	0.529
4	7.30h	0.2	-0.753	-1.349	31.7	0.61	0.074	0.509
5	8.30h	0.9	1.15	0.557	41.5	0.635	0.062	0.552
6	9.30h	0.8	0.753	0.998	50.8	0.652	0.053	0.681

While the expected values of the hourly clearness indices constantly rise as expected with the sun's height, the statistical clearness index k_{th} displays clear fluctuations.

With the thus calculated hourly clearness indices, the global irradiance on a horizontal surface can also be calculated. Extra-terrestrial irradiance on a normal surface for day 185 is 1321.9 W m^{-2} , i.e. on a horizontal surface for the second hour after sunrise with $\alpha_s = 12.32^\circ$, making 282.1 W m^{-2} altogether. With the calculated clearness index of 0.467, a total of 131.7 W m^{-2} is obtained for global horizontal irradiance.

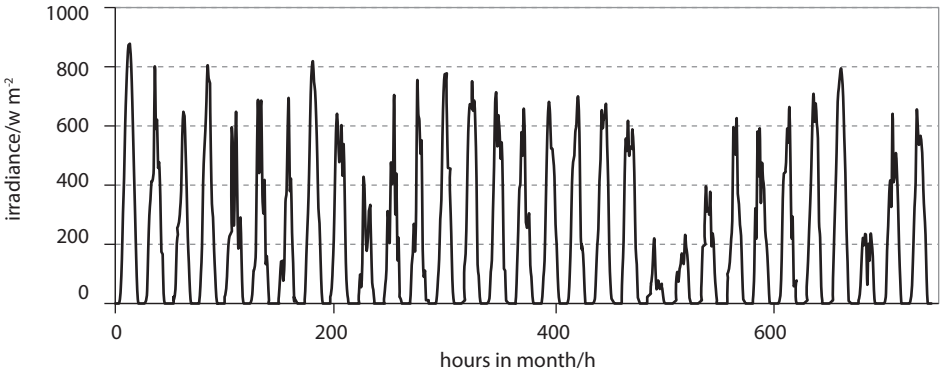


Figure 3.22 Average hourly irradiance on a horizontal surface in Stuttgart in July. The monthly irradiated energy is about 153 kWh m⁻².

3.7 Global irradiance and irradiance on inclined surfaces

For a first design of a solar energy system, it is usually sufficient to consider the annual solar irradiance. As an example, the following map shows the annual horizontal irradiance values in Europe, Western Asia and Northern Africa.

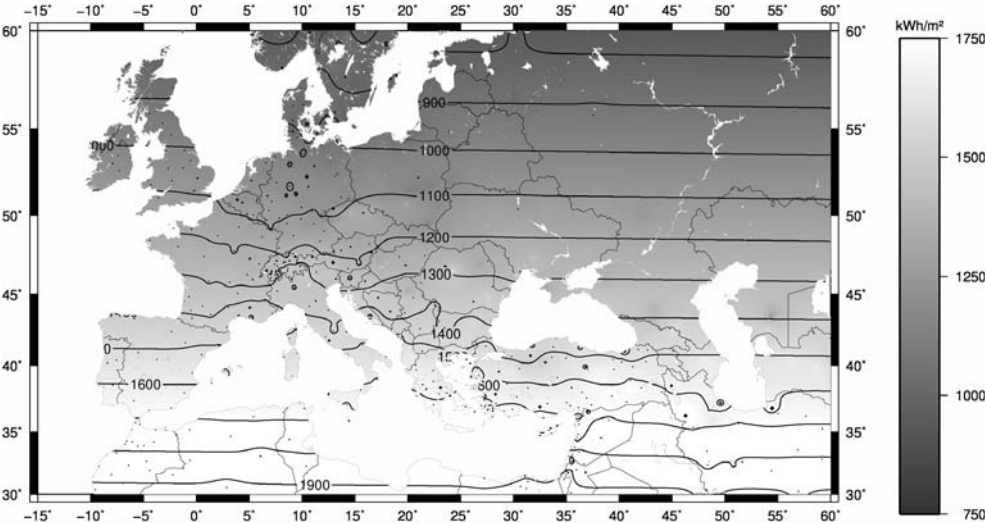


Figure 3.23 Annual irradiation totals (kWh m⁻²) on a horizontal surface in Europe from the database of the software environment INSEL (www.insel.eu).

Direct and diffuse irradiance

Due to the scattering of extra-terrestrial irradiance in the atmosphere, a diffuse irradiance proportion G_d always occurs in addition to direct irradiance G_b (index b : beam). The irradiance on the horizontal in W m^{-2} is called global irradiance (G_h):

$$G_h = G_{bh} + G_{dh} \quad (3.73)$$

Direct and diffuse irradiance on the horizontal are supplied in many meteorological data records as measured hourly values. If irradiance is determined by statistical procedures, at first only the global irradiance is obtained.

The diffuse irradiance proportion correlates directly, however, with the hourly clearness index. An empirical correlation for 15 locations in North America, Europe and Australia has been determined by Erbs et al. (1982).

$$G_{dh} = \begin{cases} G_h \times (1.0 - 0.09k_{th}) & \text{for } k_{th} \leq 0.22 \\ G_h \times \left(\begin{aligned} &0.9511 - 0.1604k_{th} + 4.388k_{th}^2 \\ &- 16.638k_{th}^3 + 12.336k_{th}^4 \end{aligned} \right) & \text{for } 0.22 < k_{th} < 0.8 \\ G_h \times 0.165 & \text{for } k_{th} \geq 0.8 \end{cases} \quad (3.74)$$

Conversion of global irradiance to inclined surfaces

The conversion of horizontal global irradiance to inclined surfaces must proceed separately for direct and diffuse irradiance. While the intensity of the direct irradiance depends only on the angle of incidence on the recipient surface, different conversion procedures exist for diffuse irradiance.

The simplest model, by Liu and Jordan (1960), proceeds from even, isotropic, diffuse irradiance in the sky, which with rising surface inclination results in ever smaller visible proportions. The total irradiance reflected by the ground is likewise assumed to be isotropically distributed. More exact models divide the diffuse irradiance in such a way that apart from the isotropically distributed celestial irradiance, the brightening around the sun (circumsolar irradiance) and also the horizon brightening caused by dispersion are taken into account (Perez et al., 1987). The ray-tracing software programmes used in light technology or the standard sky models for daylight calculations calculate the irradiance intensity for each point of the sky hemisphere (Brunger and Hooper, 1993).

An isotropic diffuse irradiance model

The total irradiance on the inclined surface G_t (index t : tilted) results from the sum of direct irradiance, the isotropic sky diffuse irradiance and also the ground reflection proportion with the reflection coefficient ρ .

$$F_{\text{surface-sky}} = \frac{1 + \cos \beta}{2} \quad (3.75)$$

The direct irradiance proportion on a horizontal plane is converted via the cosine of the zenith angle to a surface normal to the sun, and multiplied afterwards by the cosine of the angle of incidence to the inclined surface.

The other diffuse irradiance components are converted with the help of form factors F between the receptor surface and the sky or ground. The form factors depend on the angle of inclination β of the surface to the horizontal.

$$F_{\text{surface-ground}} = \frac{1 - \cos \beta}{2} \quad (3.76)$$

$$G_t = G_{bh} \frac{\cos \theta}{\cos \theta_z} + \underbrace{G_{dh} \frac{1 + \cos \beta}{2}}_{G_{d,sky}} + \underbrace{G_h \rho \frac{1 - \cos \beta}{2}}_{G_{d,ground}} \quad (3.77)$$

With these geometrical relations, the total irradiance can be calculated on a randomly inclined surface.

$$G_t = G_{bh} \frac{\cos \theta}{\cos \theta_z} + \underbrace{G_{dh} \frac{1 + \cos \beta}{2}}_{G_{d,sky}} + \underbrace{G_h \rho \frac{1 - \cos \beta}{2}}_{G_{d,ground}} \quad (3.78)$$

Example 3.7

Calculation of the terrestrial irradiance with the isotropically diffuse model for a surface normally orientated to the sun for the second hour after sunrise on day 185 (see Example 3.6), i.e. $\theta = 0^\circ$ and $\beta = \theta_z = 77.7^\circ$.

Global horizontal irradiance:

$$G_h = 131.7 \text{ W m}^{-2}$$

With the clearness index $k_{th} = 0.467$, the result is a value of 95.5 W m^{-2} for horizontal diffuse irradiance, based on the empirical correlation of Erbs et al. (1982), and thus, a horizontal direct irradiance proportion of 36.2 W m^{-2} .

Direct irradiance on the normally orientated surface:

$$G_{bt} = 169.6 \text{ W m}^{-2}$$

Diffuse proportion:

$$G_{d,sky} = 58.3 \text{ W m}^{-2}$$

Diffuse ground reflection at $\rho = 0.2$:

$$G_{d,ground} = 10.3 \text{ W m}^{-2}$$

The irradiance on the inclined surface is thus:

$$G_t = 238.5 \text{ W m}^{-2}$$

Diffuse irradiance model based on Perez

In the Perez model, the isotropically assumed sky irradiance G_{iso} is overlain by a circumsolar proportion G_{cir} as well as by a term for the horizon brightening G_{hor} .

$$G_{dt} = \underbrace{G_{iso} + G_{cir} + G_{hor}}_{G_{d,sky}} + G_{d,ground} \quad (3.79)$$

On inclined surfaces, the total irradiance reflected at the ground increases the diffuse proportion by $G_{d,ground}$.

The circumsolar irradiance is caused by strong forward scattering of the aerosols. The horizon brightening results from scattering at the large air mass of the atmosphere near the horizon, as seen by the observer, and is usually present when the sky is clear.

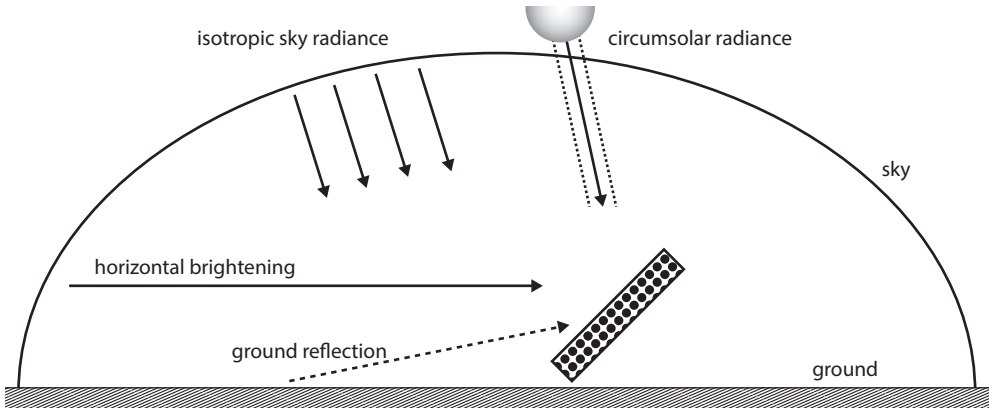


Figure 3.24 Allocation of the irradiance into different components.

$$G_{dt} = \underbrace{G_{dh}(1-F_1)\frac{1+\cos\beta}{2}}_{G_{iso}} + \underbrace{G_{dh}F_1\frac{\cos\theta}{\cos\theta_z}}_{G_{cir}} + \underbrace{G_{dh}F_2\sin\beta}_{G_{hor}} + \underbrace{G_h\rho\frac{1-\cos\beta}{2}}_{G_{d,ground}} \quad (3.80)$$

The isotropic irradiance with the form factor $(1 + \cos \beta)/2$ is therefore reduced by the proportion of F_1 of circumsolar irradiance (first term). The circumsolar irradiance with the proportion F_1 is, like the direct irradiance, converted to the inclined surface. F_2 describes the horizon irradiance, which can be both positive (horizon brightening) and negative (darkening).

The coefficients F_1 and F_2 are parameterised by the zenith angle, a clearness index ε and the brightness coefficient Δ . The clearness index ε essentially shows the ratio of direct to diffuse irradiance (with G_{bn} as direct irradiance on a normal surface), i.e. small clearness indices mean overcast skies and vice versa. Through brightness as a ratio of the air mass-corrected diffuse irradiance G_{dh} to the extra-terrestrial irradiance on a normal surface G_{en} , the fact that even very clear skies can be dark (deep blue, small Δ) and overcast skies very bright (large Δ) is taken into account.

$$\varepsilon = \frac{1 + \frac{G_{bn}}{G_{dh}} + 5.535 \times 10^{-6} \theta_z^3}{1 + 5.535 \times 10^{-6} \theta_z^3} \quad (3.81)$$

$$\Delta = \frac{m G_{dh}}{G_{en}} \quad (3.82)$$

The air mass m is, as before, $m = 1/\cos \theta_z$.

The brightness coefficients F_1 and F_2 are empirically derived coefficients from measurements of different sky conditions. They are calculated using the clearness index ε .

$$F_1 = f_{11} + f_{12}\Delta + \frac{\pi\theta_z}{180} f_{13} \quad (3.83)$$

For F_1 , only positive values are used (otherwise $F_1 = 0$).

$$F_2 = f_{21} + f_{22}\Delta + \frac{\pi\theta_z}{180}f_{23} \quad (3.84)$$

Clearness index $\varepsilon/-$	f_{11}	f_{12}	f_{13}	f_{21}	f_{22}	f_{23}
≤ 1.056	0.041	0.621	-0.105	-0.040	0.074	-0.031
$1.056 < \varepsilon \leq 1.253$	0.054	0.966	-0.166	-0.016	0.114	-0.045
$1.253 < \varepsilon \leq 1.586$	0.227	0.866	-0.250	0.069	-0.002	-0.062
$1.586 < \varepsilon \leq 2.134$	0.486	0.670	-0.373	0.148	-0.137	-0.056
$2.134 < \varepsilon \leq 3.230$	0.819	0.106	-0.465	0.268	-0.497	-0.029
$3.230 < \varepsilon \leq 5.980$	1.020	-0.260	-0.514	0.306	-0.804	0.046
$5.980 < \varepsilon \leq 10.080$	1.009	-0.708	-0.433	0.287	-1.286	0.166
> 10.080	0.936	-1.121	-0.352	0.226	-2.449	0.383

Table 3.1 Perez coefficients for the calculation of the anisotropic diffuse irradiance.

Thus, for the entire irradiance on the inclined surface, the following is obtained:

$$G_t = \frac{G_{bh}}{\cos\theta_z} \cos\theta + G_{dh}(1-F_1) \frac{1+\cos\beta}{2} + G_{dh}F_1 \frac{\cos\theta}{\cos\theta_z} + G_{dh}F_2 \sin\beta + G_h \rho \frac{1-\cos\beta}{2} \quad (3.85)$$

Example 3.8

Calculation of the irradiance on the 77.7° inclined surface from the last example with the Perez model.

$$\varepsilon = 1.5$$

$$\Delta = 0.34$$

$$\theta_z = 77.7^\circ$$

$$F_1 = f_{11} + f_{12}\Delta + \frac{\pi\theta_z}{180}f_{13} = 0.227 + 0.866 \times 0.34 + \pi \times 77.7 / 180 \times -0.25 = 0.232$$

$$F_2 = f_{21} + f_{22}\Delta + \frac{\pi\theta_z}{180}f_{23} = 0.069 + (-0.002) \times 0.34 + \pi \times 77.7 / 180 \times (-0.062) = -0.0157$$

Isotropic sky component:

$$G_{dh}(1-F_1) \frac{1+\cos\beta}{2} = 95.5 \frac{W}{m^2} (1-0.232) \frac{1+\cos 77.7}{2} = 44.5 \frac{W}{m^2}$$

Circumsolar proportion:

$$G_{dh}F_1 \frac{\cos\theta}{\cos\theta_z} = 95.5 \frac{W}{m^2} \times 0.232 \times \frac{1}{\cos 77.7} = 104 \frac{W}{m^2}$$

Horizon proportion:

$$G_{dh} F_2 \sin \beta = 95.5 \frac{W}{m^2} \times -0.0157 \times \sin 77.7 = -1.46 \frac{W}{m^2}$$

$G_t = 169.6 W m^{-2} + 147 W m^{-2} = 316.6 W m^{-2}$, thus 33% more than calculated via the isotropic model.

Using the procedures outlined above, the annual irradiated energy for randomly orientated surfaces can be calculated. This energy, represented as a function of the height and azimuth angle, enables quick determination of the energy supply on randomly orientated surfaces. The optimal annual energy yield for the Northern Hemisphere is obtained on a south-orientated surface with an angle of inclination β of approximately the geographical latitude ϕ minus 10° . With energy losses of only 5%, deviations in the azimuth angle of $\pm 35^\circ - 40^\circ$ from south, and in the angle of inclination of $\pm 15^\circ - 20^\circ$ of the optimal angle can be tolerated.

The maximum annual irradiance is achieved in the northern hemisphere on south-facing surfaces inclined at an angle of the geographical latitude minus about 10° . In Stuttgart, the maximum irradiation on a 38° inclined south-facing surface is $1200 \text{ kWh m}^{-2} \text{ a}^{-1}$. A deviation from south orientation of $+ \text{ or } - 50^\circ$ leads to an annual irradiation reduction of 10%. A south-facing facade receives about 72% of the maximum possible irradiation G_{max} (defined in Figure 3.25 as 100%).

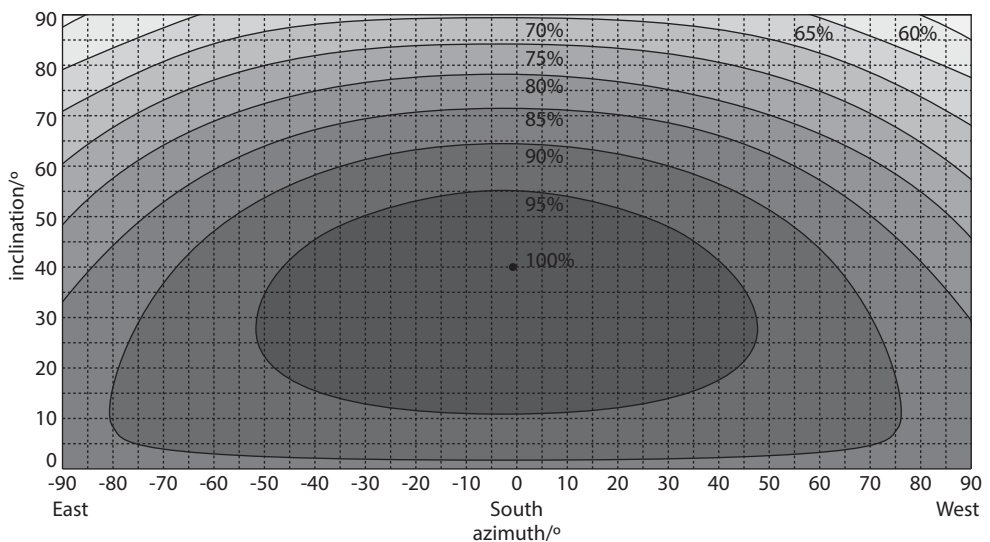


Figure 3.25 Annual irradiance as a function of surface azimuth and angle of inclination in Stuttgart, Germany.

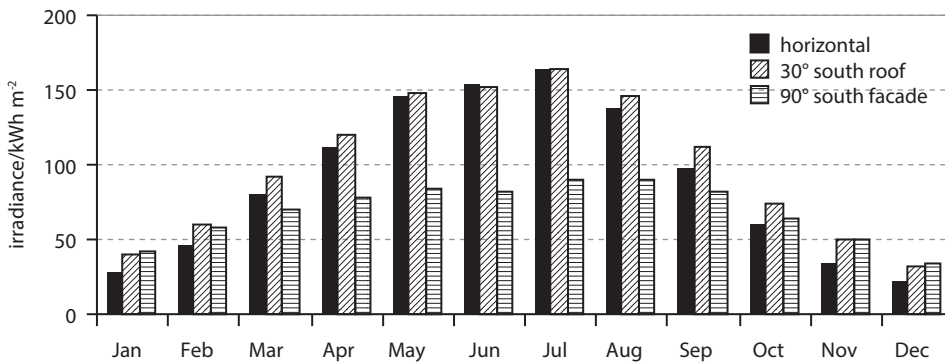


Figure 3.26 Monthly irradiance of differently inclined surfaces in Stuttgart.

Measurement techniques for solar irradiance

Global irradiance and irradiance on inclined surfaces is measured by pyranometers, with errors under 5%. Pyranometers measure irradiance by means of thermocouples, from the temperature difference of an irradiance-absorbing blackened surface and the housing. Due to the hemispherical glass cover, the angle dependence of the signal is small, and moreover, sensitivity is almost independent of wavelength. The voltage level of the thermocouples is low, typically around 5×10^{-6} V per W m^{-2} of irradiance. With more exact calibration, measuring accuracies of up to $\pm 1\%$ can be achieved.

Photovoltaic solar cells are cheaper detectors. However, spectral sensitivity and the temperature dependence of the measuring signal, plus reduced efficiency at small irradiances, lead to measuring errors of over 10%. For the measurement of the mean monthly irradiance, PV cells are nonetheless quite suitable.

The diffuse irradiance can be measured with a pyranometer, with shadowing rings to block off direct irradiance. The shadowing ring is adjusted for declination and degree of latitude, and must also be adjusted for seasonal changes in declination (every 2 - 3 days, depending on the width of the ring). The reduction of diffuse irradiance by the ring is corrected by a variable correction factor between 1.05 and 1.2.

3.8 Shading

When using solar technology in urban areas, it cannot be assumed that solar irradiance strikes receptor surfaces unhindered. Apart from temporary shading of the direct rays by nearby objects, buildings or vegetation, it is seldom the case that a free horizon for diffuse irradiance is available, in particular when facades are used.

For a representation of the shadowing effects of obstructing buildings, all objects are first represented by surface polygons and their corner points. The shadowing caused by a building is then calculated for each corner point by means of the sun vector and the points of shading are again connected to a polygon.

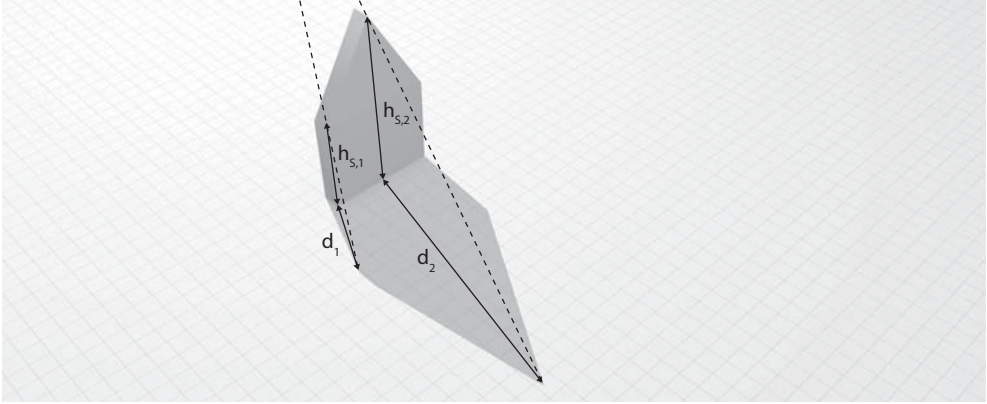


Figure 3.27 Construction of the shadowing from the surface polygon of the blocking building with heights of $h_{s,i}$.

It is simpler, however, to analyse the shading using horizon coordinates (elevation angles and azimuth) from a point receptor. With large receptor surfaces a subdivision of the surfaces is possible.

From the difference between building height h_s and observer height h_o , and also from the building's distance from the point of observation d , the elevation angle α_s for a given azimuth angle γ_s of the shading building can be calculated.

$$\alpha_s = \arctan\left(\frac{h_s - h_o}{d}\right) \quad (3.86)$$

The thus obtained value pairs γ_s and α_s form a polygonal sequence which can be entered in a sun-position diagram, and which enables direct reading of the times when shading of direct irradiance occurs.

For the reduction of diffuse irradiance, the surface between the horizon line of elevation angle $\alpha = 0^\circ$ and the shading contour must be calculated. This shading surface reduces the isotropically assumed diffuse irradiance proportion of the sky by the given angle area.

For this, the shading contour for an azimuth angle area γ_1 to γ_2 is divided into even stages, which can be described by a straight line equation (Quaschnig, 1998):

$$\alpha(\gamma) = m\gamma + c \quad (3.87)$$

where m indicates the gradient of the straight lines with

$$m = \frac{\alpha_2 - \alpha_1}{\gamma_2 - \gamma_1}$$

and the constant c results from equating the gradient m and the gradient for $\gamma = 0$:

$$\frac{\alpha_2 - \alpha_1}{\gamma_2 - \gamma_1} = \frac{\alpha_1 - c}{\gamma_1 - 0} \Rightarrow c = \frac{\alpha_1\gamma_2 - \alpha_2\gamma_1}{\gamma_2 - \gamma_1} \quad (3.88)$$

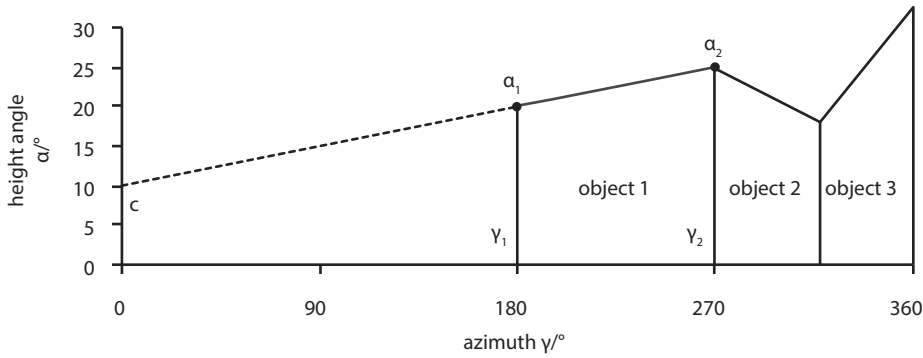


Figure 3.28 Contours of three blocking buildings.

Here, the constant c is determined from the elevation angles of the shading for a linear section between the azimuth angles γ_1 and γ_2 of the first building.

The luminance of each shaded celestial point $L_e(\alpha, \gamma)$ is projected onto the recipient surface with the cosine of the angle of incidence θ , and integrated over the total area. The luminance is defined as the radiant flux emitted by a two-dimensional element dA into a spatial angle $d\Omega$. With an isotropic sky, the luminance $L_{e,iso}$ is constant at each celestial point, and can be easily calculated from the horizontal diffuse irradiance.

$$L_{e,iso} = \frac{G_{dh}}{\pi} \left[\frac{W}{m^2 sr} \right] \quad (3.89)$$

A small section of the celestial sphere in horizon coordinates can be expressed as $dA = \cos \alpha \, d\alpha \, d\gamma$. The shaded proportion of diffuse celestial radiance, seen from an inclined surface $G_{dt,S}$, results from the luminance $L_{e,iso}$ multiplied by the cosine of the angle of incidence and of the sky section. The limits of integration are for an azimuth area γ_1 to γ_2 between the horizon ($\alpha = 0^\circ$) and the straight line shading contour ($m\gamma + c$):

$$\begin{aligned} G_{dt,S} &= L_{e,iso} \int_{\gamma_1}^{\gamma_2} \int_0^{m\gamma+c} \cos \theta \cos \alpha \, d\alpha \, d\gamma \\ &= L_{e,iso} \int_{\gamma_1}^{\gamma_2} \int_0^{m\gamma+c} (\sin \alpha \cos \beta + \cos \alpha \sin \beta \cos(\gamma - \gamma_f)) \cos \alpha \, d\alpha \, d\gamma \end{aligned} \quad (3.90)$$

In the case of a horizontal receptor surface (angle of inclination $\beta = 0^\circ$ and surface azimuth $\gamma = 0^\circ$), the projection factor is $\cos \theta = \cos \theta_z = \sin \alpha$, and the integral is simplified to:

$$G_{dh,S} = L_{e,iso} \int_{\gamma_1}^{\gamma_2} \int_0^{m\gamma+c} \sin \alpha \cos \alpha \, d\alpha \, d\gamma = L_{e,iso} \frac{1}{2} \int_{\gamma_1}^{\gamma_2} \sin^2(m\gamma + c) \, d\gamma \quad (3.91)$$

For straight horizontal shading contours $m = 0$, i.e. the elevation angle $\alpha_1 = \alpha_2$. On the basis of Equation (3.88) therefore, $c = \alpha_1$ and for the integral, the result is the simple solution:

$$G_{dh,S} = \frac{1}{2} L_{e,iso} (\gamma_2 - \gamma_1) \sin^2 \alpha_1 \quad \text{for } m = 0 \quad (3.92)$$

Also, the integral can be solved for horizontal receivers:

$$G_{dh,S} = \frac{1}{2} L_{e,iso} (\gamma_2 - \gamma_1) \left(\frac{1}{2} + \frac{1}{4} \frac{\sin 2\alpha_1 - \sin 2\alpha_2}{\alpha_2 - \alpha_1} \right) \quad \text{for } m \neq 0 \quad (3.93)$$

The angles are to be entered here in arc measures.

The solution of the integral for the surface with an angle of inclination β and surface azimuth γ from Equation (3.90) is only possible for gradients of the shading contour $m \neq 0$ on a case-by-case basis, using complex formulae (Quaschnig, 1996). Therefore, only the solution for $m = 0$ is shown, with which the real contours can be approximated in small sections.

$$G_{dt,S} = \frac{1}{2} L_{e,iso} \left(\cos \beta (\gamma_2 - \gamma_1) \sin^2 \alpha + \sin \beta (\alpha + \sin^2 \alpha) (\sin(\gamma_2 - \gamma) - \sin(\gamma_1 - \gamma)) \right) \quad (3.94)$$

For a given polygonal sequence of the shading contour, the diffuse irradiance proportions of the shaded sections are totalled and an overall shading factor is formed, by which the diffuse irradiance is reduced:

$$S_d = \frac{\sum G_{d,S}}{G_d} \quad (3.95)$$

Example 3.9

On a south-orientated roof area, a solar plant is to be erected with the low edge H_0 having a height of 8 m. The shading contour of two facing multifamily houses (Building 1 is 10 m high and Building 2 is 18 m high) with $d = 14$ m distance is to be entered in the sun-position diagram, and the reduction of diffuse and direct irradiance is calculated.

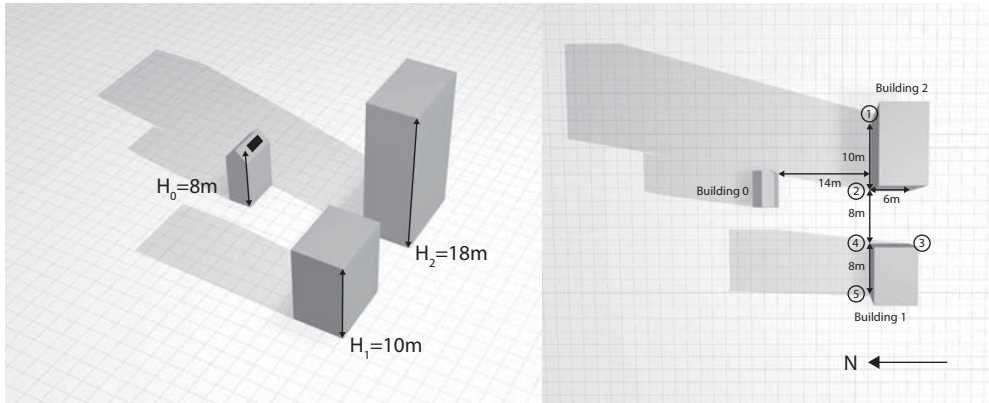


Figure 3.29 Geometry of the shading situation.

For increasing shading azimuth angles, the following value pairs of γ_s , α_s are determined from Figure 3.28:

Point	Distance d/m	Shading object height h_s/m	Azimuth $\gamma_s/^\circ$	Shading height angle $\alpha_s/^\circ$
1	17.2	18	144.5	30.2
2	14	18	180	35.5
3	21.5	10	201	5.3
4	16.1	10	209.7	7.1
5	21.3	10	228.8	5.4

From the values shown in the above table it can be seen that in Stuttgart, for example, shading of the direct irradiance by Building 2 occurs during the winter months (October until March) from about 10 to 12 am. Building 1, on the other hand, with its small elevation angles, does not contribute to the shading. When entering the shading contour into the sun-position diagram, a linear extrapolation between the corner points of the shading is carried out, in order to simplify the diffuse irradiance calculation which follows.

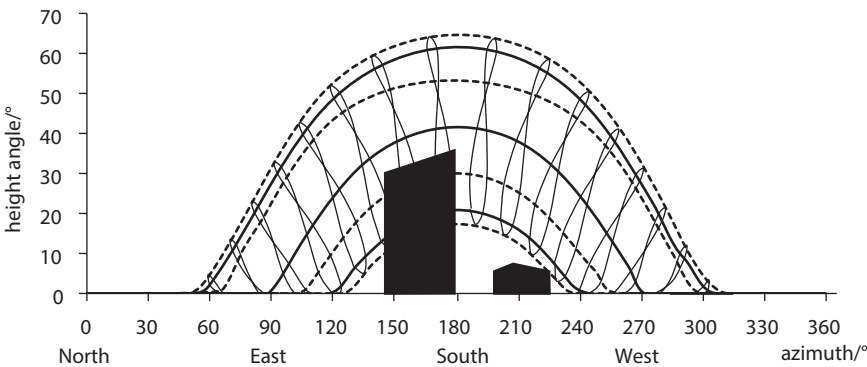


Figure 3.30 Sun-course diagram for Stuttgart with shading buildings drawn in.

For Building 2, the shading factor S_d of the diffuse irradiance for a horizontal receptor surface and for a surface with a roof pitch angle of 45° must be calculated as follows. Although the absolute value of the diffuse irradiance is not necessary for the shading factor, since the radiances shorten in Equation (3.95), a value of 300 W m^{-2} is given, in order to be able to calculate the absolute irradiance loss due to shading based on Equation (3.92).

For the horizontal receptor surface with $m \neq 0$, a shaded diffuse proportion of 9 W m^{-2} is obtained, and thus, a shading factor of 3%. For the 45° inclined surface, the shaded diffuse proportion for $m = 0$ and a mean elevation angle of the shading of

$$(\alpha_{s1} + \alpha_{s2})/2 = 32.85^\circ$$

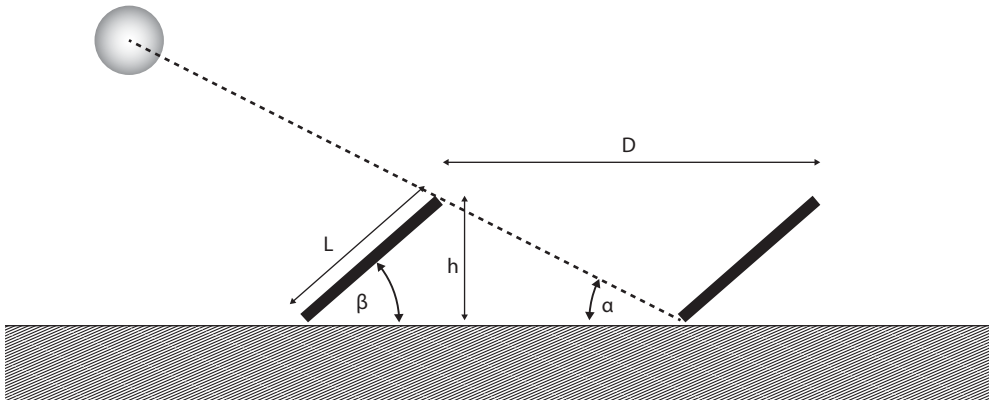
totals 19.8 W m^{-2} and the shading factor is 7%. For the calculation of the shading factor of the inclined surface, the shaded sky proportion was related to the diffuse irradiance to the inclined surface, i.e. to

$$G_{dh} \left(\frac{1 + \cos \beta}{2} \right)$$

here 256 W m^{-2} .

Example 3.10

On a flat roof with a limited surface, the largest possible south-orientated solar plant is to be installed in Stuttgart at a latitude of 48.8° . As a criterion for an acceptable shading situation, it is assumed that at solar noon on the winter solstice (21 December) no shading takes place. The relation of the distance of the collectors D to the collector length L with a given surface angle of inclination β is to be determined.



The sun-height angle α at solar noon on 21 December is 17.8° .

The distance D between the collector bases for this position of the sun is calculated as the total of the distances $L \cos \beta$ and $h \cot \alpha$.

$$D = L \cos \beta + h \cot \alpha = L \cos \beta + L \sin \beta \frac{\cos \alpha}{\sin \alpha}$$

For an angle of inclination β , for example, 40° , the relation of distance to length should be at least as shown below:

$$\frac{D}{L} = \cos \beta + \sin \beta \cot \alpha = 2.77$$

3.9 Temperature time series modelling

Ambient temperature time series

The generation of ambient temperature data has been the object of several studies. A weather data generator that provides both hourly solar radiation and ambient temperature time series was firstly developed by Degelman (1976). Knight (1991) improved this programme and established an easy method to generate hourly ambient temperature data from monthly average values using widely location-independent correlations. A general shortcoming of these models is that cross-correlations between solar radiation and ambient temperature cannot be reproduced.

On the other hand, studies taking into account cross-correlations were mostly based on empirical data sets for only one or two locations. An example of this approach is given by Hokoi et al., 1990. They used an ARMAX model that directly reproduces cross-correlations between the stochastic components of radiation and temperature in Tokyo. It is shown that the model parameters can be estimated by Kalman filters. For summer and winter seasons, different models and parameters are obtained. As this method leads to a large number of site-specific coefficients for Fourier series and ARMAX-models, it seems to be reasonable only if measured ambient temperature data for the given location are available.

This section presents a method for the generation of time series showing this characteristic, which is based on solar radiation data, and calculates ambient temperature time series from the given synthetic solar radiation values (Schumacher, 2011). The algorithm requires only the monthly averages of the global horizontal radiation and ambient temperature as input parameters, which are available for many locations.

The temperature time series may be separated into a component that is strongly affected by the daily solar radiation, an advective component caused by air movement without a significant correlation with solar radiation, and a small residual component which mainly is governed by stochastic fluctuations.

It has been shown that if the monthly average daily temperature pattern is known, the daily pattern of the solar component may be determined by three characteristic temperature values: the midnight temperature at the beginning of the day, the sunrise temperature and the temperature for the hour when the monthly average temperature pattern has its maximum.

The advective component, given by the variation of the midnight temperature, can be synthetically derived from a random generation model for the difference between two subsequent midnight temperatures. Furthermore, once the midnight temperatures are known, the other two characteristic temperature values may be estimated from empirical correlations with the daily clearness index. Based on these relationships, a synthesis is suggested by (Schumacher, 2011) which only requires the geographical latitude, the monthly and the daily averages of the clearness index and the monthly average temperature values as input parameters for the generation of hourly ambient temperature data.

As clouds and other atmospheric constituents (water vapour, aerosols, etc.) reduce solar irradiance at the surface and terrestrial net emissions, a cloudy sky leads to a lower maximum temperature at daytime and to a higher minimum at night. So the daily temperature amplitude mainly is a function of the transparency of the atmosphere, which is commonly given by the clearness index k_t , defined as the ratio of the total solar radiation on a horizontal surface to the extra-terrestrial radiation on a horizontal surface. Generally, the variation of ambient temperature follows the annual and diurnal variation of solar irradiance with a certain time lag and the mean

daily minimum of ambient temperature occurring at sunrise hour, while the mean daily maximum is reached in the first hours following high noon or in the early afternoon. For the annual variations, the time lag between solar radiation and temperature is typically one month. The daily temperature pattern resulting from these effects is modified by additional meteorological influences. For coastal sites with distinctive thermally driven wind systems, the daily maximum of ambient temperature is reached before noon due to the strong cooling effect of the sea wind. A similar effect is caused by daily precipitation in tropical regions. The annual variation of the monthly average temperature increases with the geographical latitude and the continentality of the climate. While this annual variation is small for the equatorial region, the temperature values at locations in high latitudes are characterised by high differences between summer and winter seasons. Because of the much slower time response of physical exchange processes in water, the annual variation of the monthly average temperature is substantially smaller in regions with maritime climate than in continental areas with the same latitude.

The surface temperature affects both the magnitude and the distribution of ambient temperature over the day. Over large areas of water, the temperature reaches its daily maximum typically half an hour after solar noon. Over continental areas with high insolation, this delay is up to 3 hours. These differences are due to the high specific heat capacity c_p of water, which is about five times higher than for sandy soils.

The amplitude of the monthly average daily temperature pattern ranges from 1K (over oceans) to 14 - 16 K in continental deserts in low latitudes (Heyer, 1981). It depends both on the solar elevation and on the cloudiness. The high elevation at noon in summer time leads to a bigger amplitude than in winter. These relationships between radiation and ambient temperature are strictly valid only for long-term averages. Considering particular hours or days, the influence of air mass transports will also affect the variation of ambient temperature. At continental sites, the supply of warm or cold air masses can cause the temperature to rise or fall more than 10 K within a day.

Calculation of monthly mean values of hourly ambient temperature

Generally, ambient temperature follows a cyclic variation over the course of a day, leading to significant differences between monthly average daytime and night-time temperatures. In order to identify deterministic and stochastic components of hourly ambient temperature time series, the monthly average values for a particular hour of the day must be known. As these values are not available for most locations, approximation methods must be used. In the following, an approach which was developed by Erbs (1984) is used. However, the location-independent Erbs formulas often result in systematic errors.

The Erbs relations

Erbs showed that the monthly average daily temperature courses have the same shape for different locations and months. He defined a 'standardised diurnal variation of monthly average ambient temperature':

$$\Theta(h) = \frac{\bar{T}_m(h) - \bar{T}_m}{A_m} \quad (3.96)$$

by subtracting the monthly average ambient temperature from the monthly average hourly values and dividing the residual differences by the peak-to-peak amplitude A_m of the monthly average daily temperature course:

$$A_m = \bar{T}_m^{max}(h) - \bar{T}_m^{min}(h) \quad (3.97)$$

The values were found to be quite similar for nine locations of the North American continent. A Fourier series was fitted to the average curve calculated from all data sets for these locations:

$$\begin{aligned} \Theta(h) = & 0.4632 \cos(t^* - 3.805) + 0.0984 \cos(2t^* - 0.36) \\ & + 0.0168 \cos(3t^* - 0.822) + 0.0138 \cos(4t^* - 3.513) \end{aligned} \quad (3.98)$$

where

$$t^* = \frac{2\pi(h-1)}{24}$$

It was mentioned earlier that the amplitude is mainly a function of the clearness index. Using the same database as stated above, Erbs extracted an empirical linear relationship between A_m and $k_{t,m}$

$$A_m = (25.8 k_{t,m} - 5.23)^\circ\text{C} \quad (3.99)$$

which he claimed to be a reasonable approximation for all locations where only the monthly average ambient temperature is available.

Equation (3.99) gives a good approximation for A_m only for continental regions in middle latitudes. For locations in a maritime climate with typically small temperature amplitudes, even in cases of high monthly clearness indices, the approximation provides significantly higher values than calculated from measured data.

Construction of hourly ambient temperature sequences

A universal and improved accuracy method for the characterisation of daily temperature patterns was implemented in the simulation environment INSEL using the following method:

A linear advective component was separated from the hourly ambient temperature values. The residual values could then be identified as the part of the daily temperature variations, which is mainly forced by solar radiation. For this component a daily temperature amplitude was defined, which is correlated with the daily clearness index and the sunrise temperature. Furthermore, a linear relationship between the sunrise temperature and the previous midnight temperature was set up. With the midnight temperature, the sunrise temperature, the daily amplitude and the daily clearness index, an average temperature pattern for the given day could be constructed by scaling the monthly average hourly temperature values. Synthetic time series were finally obtained by summing up this average temperature pattern, the advective component and a residual stochastic part represented by an autoregressive model.

The synthetisation procedure necessarily requires the following commonly well-known input parameters:

- the geographical latitude of the respective site
- 12 monthly clearness index values $k_{t,m}$ for every year (if the year-to-year variability of $k_{t,m}$ is neglected, the long-term average values may be used)
- 365 (or 366) daily clearness index values for every year and 12 long-term monthly average values of ambient temperature $\bar{T}_m$

Optional parameters include:

- 12 long-term monthly average values of the daily maximum temperature $\bar{T}_m^{max}$ and
- 12 long-term monthly average values of the daily minimum temperature $\bar{T}_m^{min}$

which are not strictly necessary, but very useful to improve the accuracy of the correlation equations used by the procedure.

From the monthly average values, firstly the monthly average daily temperature pattern is estimated by a Fourier series fit and an approximation for the monthly amplitude is calculated. Secondly, a sequence of daily temperature patterns for each month is generated as follows:

1. For every day a value for ΔT_{24} , defined as the difference between two subsequent midnight temperatures, is calculated. It is composed of a small deterministic part and a stochastic component.
2. The advective component of the daily temperature pattern is obtained from a linear interpolation between the midnight temperatures at the beginning and the end of the day, which are fixed once the ΔT_{24} value for the given day is known.
3. With the midnight temperature, the daily and the monthly clearness index values and the monthly average temperature values at midnight and sunrise hour, the sunrise temperature of the solar component is determined.
4. With the monthly averages of the midnight and the sunrise temperature, the monthly amplitude, the daily and the monthly clearness index values, the daily amplitude of the solar component is calculated. Then the temperature value for the monthly maximum hour is calculated.
5. Then all the other 20 values for the given day are estimated by scaling and shifting the normalised monthly average daily temperature pattern.
6. A small additional stochastic noise component for every hour of the day is obtained from a first-order autoregressive model.
7. Finally, the hourly ambient temperature values for the given day are calculated by summing up the solar component, the advective component and the stochastic noise component for each hour. Based on this algorithm, a computer programme was implemented in the simulation environment INSEL (www.insel.eu), which automatically generates a series of hourly ambient temperature data from the input parameters stated above.

Sky Temperature modelling

The long-wave radiation of the sky corresponds to a black body radiation from the atmosphere except for a strong dip between 8 and 14 μm . In this so-called 'atmospheric window', the atmosphere is nearly transparent, particularly on very dry air conditions, and long-wave radiative exchange takes place with the space at a temperature level of 2.7279 K.

The sky temperature is equivalent to the temperature of a black body, which emits the same power as the real sky. The effective sky temperature lies typically 10 - 20 K under the

ambient temperature. About 90% of the long-wave radiation is caused by the water vapour in the first kilometre over the Earth's surface (Bliss, 1961). Other atmospheric gases, like CO₂ or ozone, contribute to only a small amount of the long-wave sky radiation.

The net radiant flux between sky and a horizontal radiative body on the Earth's surface may be calculated either by the temperature differences between Earth T and sky T_{sky} or via an effective emissivity ϵ_h of the sky.

The radiant flux Q_{net} between a body with the surface A and an emissivity ϵ with the sky at a temperature T_{sky} is:

$$\dot{Q}_{net} = \epsilon \sigma A (T^4 - T_{sky}^4) \quad (3.100)$$

with σ as the Stefan–Boltzmann constant.

In this case, the sky is regarded as a black body ($\epsilon_{sky}=1$). The sky temperature is proportional to the radiant flux of the sky Q_{sky} (W m²):

$$T_{sky} = \left(\frac{\dot{Q}_{sky}}{\sigma} \right)^{\frac{1}{4}} \quad (3.101)$$

Alternatively, the sky temperature is set equal to the ambient temperature T_0 and the radiant flux is described by the sky emissivity ($\epsilon_{sky} < 1$) of a grey body:

$$T_{sky} = \epsilon_{sky}^{\frac{1}{4}} T_0 \quad (3.102)$$

The combination of the last two equation results in:

$$\epsilon_{sky} = \frac{\dot{Q}_{sky}}{\sigma T_0^4} \quad (3.103)$$

Many models, based on empirical or semi-empirical correlations are available in the literature, most of them for clear sky conditions.

Models for a clear sky

The simplest models calculate the sky temperature using only the ambient temperature (Swinbank, 1963, Fuentes, 1987). Advanced models consider the important influence of the water content using the dew point temperature T_{DP} or the water vapour partial pressure p_v .

Berdahl and Martin (1984) have also introduced additional correction factors for the difference between black night sky and bright day sky, as well as the height of the observation station.

Authors	Correlation
Swinbank	$T_{\text{sky}} = 0.0552 (T_0)^{1.5}$
Bliss	$\epsilon_{\text{sky}} = 0.8004 + 0.00396 T_{\text{DP}}$
Elsasser	$\epsilon_{\text{sky}} = 0.21 + 0.22 \ln(p_v)$
Berdahl (night sky)	$\epsilon_{\text{sky}} = 0.741 + 0.0062 T_{\text{DP}}$
Berdahl (daylight sky)	$\epsilon_{\text{sky}} = 0.727 + 0.0060 T_{\text{DP}}$
Berdahl (medium)	$\epsilon_{\text{sky}} = 0.734 + 0.0061 T_{\text{DP}}$
Clark and Allen	$\epsilon_{\text{sky}} = 0.787 + 0.0028 T_{\text{DP}}$
Berger et al.	$\epsilon_{\text{sky}} = 0.770 + 0.0038 T_{\text{DP}}$
Brunt	$T_{\text{sky}} = (0.564 + 0.059 (p_v)^{0.5})^{0.25}$ $T_{\text{sky}} = (0.527 + 0.065 (p_v)^{0.5})^{0.25}$
Berdahl and Martin	$\epsilon_{\text{sky}} = 0.711 + 0.56 (T_{\text{DP}}/100) + 0.73 (T_{\text{DP}}/100)^2$
Chen	$\epsilon_{\text{sky}} = 0.736 + 0.00571 T_{\text{DP}} + 0.3318 \times 10^{-5} \times T_{\text{DP}}^2$
Berdahl and Martin (mod)	$\epsilon_{\text{sky}} = 0.711 + 0.56(T_{\text{DP}}/100) + 0.73 (T_{\text{DP}}/100)^2 + 0.013 \cos 15t + 0.00012 (p_{\text{atm}} - p_0)$
p_v is the water vapour partial pressure in Pa. T_{DP} is the temperature of dew point in the air in K. p_{atm} is the atmospheric pressure in Pa. p_0 is the pressure on sea level in Pa (typical 105 Pa)	

Table 3.2 Models for a clear sky.

Models for cloudy skies

Clouds have a very high influence on the sky temperature. Dense, low stratus clouds have a relatively high radiation temperature, while thin, high cirrus clouds contribute little to the long-wave sky radiation. For a complete overcast sky, the ISO 6946 proposes to set the sky temperature and the ambient temperature equal.

In case of a not completely overcast sky, the sky temperature would be between the ambient temperature and the temperature of a clear sky, calculated using the Swinbank formula, for example. An analysis in the USA of 68 cities gave a mean clearness index of 0.61, which was used as a parameter for the cloudiness. As on a clear day, the clearness index is about 0.9, the solar irradiation is reduced on average by 32% for cloudy conditions. The same approach is selected for the sky temperature. The result is a weighted sky temperature with 32% cloudy conditions using the ambient temperature and with 68% from clear sky conditions using the temperature formulae according to Swinbank and Fuentes, 1987.

$$T_h = 0.68 \left(0.0552 T_0^{1.5} \right) + 0.32 T_0 \quad (3.104)$$

Kasten et al. (1980) has introduced a cloudiness index C_{cloud} between 0 and 1 to characterise the part of the sky covered by clouds. C_{cloud} is calculated from the ratio of global radiation G to the diffuse radiation G_d .

$$C_{cloud} = \left(1.4286 \frac{G_d}{G} - 0.3 \right)^{0.5} \quad (3.105)$$

The sky emissivity for cloudy sky is then calculated with the emissivity of the clear sky.

$$\varepsilon_{sky, cloud} = \varepsilon_{sky, clear} + 0.8(1 - \varepsilon_{sky, clear})C_{cloud} \quad (3.106)$$

Useful correlations for the modification of the radiant flux R_{net} by cloudiness were developed by Martin (1989). Depending on the cloudiness index C_{cloud} , the radiant flux is modified as follows:

$$\dot{Q}_{sky, cloud} = (1 + 0.0224C_{cloud} - 0.0035C_{cloud}^2 + 0.00028C_{cloud}^3) \dot{Q}_{sky, clear} \quad (3.107)$$

If the cloud formations are known more accurately, the radiant flux can be determined more precisely (Oke, 1987).

$$\dot{Q}_{sky, cloud} = (1 + aC_{cloud}^2) \dot{Q}_{sky, clear} \quad (3.108)$$

Cloud type	Typical height/km	Coefficient a
Cirrus	12.2	0.04
Cirrostratus	8.39	0.08
Altostratus	3.66	0.17
Altostratus	2.14	0.20
Cumulus		0.20
Stratocumulus	1.22	0.22
Stratus	0.46	0.24
Fog	0	0.25

Table 3.3 Cloud type and coefficient for calculating the radiant flux.

Aubinet (1994) has introduced a clearness index K_0 as the ratio of global radiation G_h to the extra-terrestrial radiation G_0 :

$$K_0 = \frac{G_h}{G_0} \quad (3.109)$$

The sky temperature is then calculated from:

$$T_{sky} = 94 + 12.6 \ln p_v - 13K_0 + 0.341T_0 \quad (3.110)$$

Perraudreau (1986) calculates the radiation of cloudy skies from a fog-index:

$$I_p = \frac{1 - \frac{G_{d,h}}{G_h}}{1 - \frac{G_{d,clear}}{G_{clear}}} \quad (3.111)$$

where,

$G_{d,h}$ is the diffuse radiation on a horizontal surface/ W m^{-2}

G_h is the global radiation on a horizontal surface/ W m^{-2}

$G_{d,clear}$ is the diffuse radiation for a clear sky using the Bird model/ W m^{-2}

G_{clear} is the direct radiation on a horizontal surface for a clear sky by the Bird model/ W m^{-2}

The Bird model is a diffuse radiation model for a clear sky (1981). The fog-index is used for the calculation of a cloudiness index ($N=0$ for clear and $N=8$ for completely covered sky):

$$\begin{aligned} N &= 8 & \text{if } I_p \leq 0.07 \\ N &= INT \left(\sqrt[3]{\frac{1-I_p}{0.825}} + 0.5 \right) & \text{if } 0.07 < I_p < 1 \\ N &= 0 & \text{if } I_p \geq 1 \end{aligned}$$

3.10 Geothermal resource

Low depth geothermal energy is well suited for heating and cooling applications in buildings, as the underground temperature has a constant level around the annual average air temperature at a depth of about 15 m and then increases by about 3 K per 100 metres. In most climates, the annual average temperature is above ambient air temperature in winter so that the geothermal energy can be used as a heat source for heating applications. In summer, the geothermal temperature levels are usually lower than ambient air temperatures, and the heat exchangers can be used as a heat sink for cooling applications.

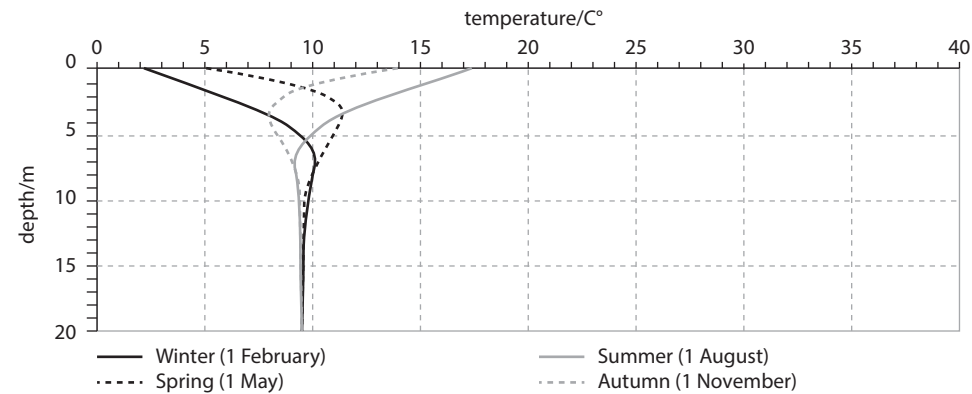


Figure 3.31 Development of soil temperatures for a moderate German climate with 9.7°C annual average ambient air temperature.

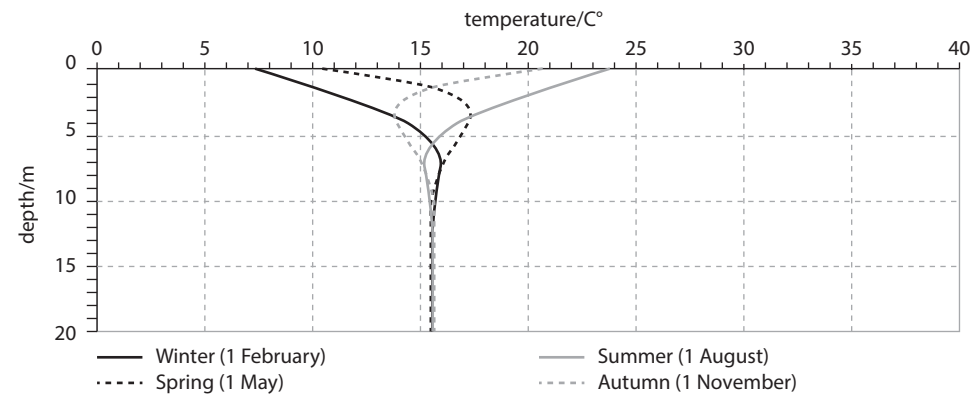


Figure 3.32 Development of soil temperatures for the Italian location Gela with 15.5°C annual average ambient air temperature.

In climates with high average air temperature levels and little temperature fluctuations as in many Asian countries, the use of geothermal energy for heating and cooling is very limited (see, for example, Figure 3.34 for Bangkok, Thailand). Between the ground level and about 15 m depth, the temperature levels strongly fluctuate with time, especially in climates with a high seasonable temperature variation (see, for example, Figure 3.31 for a moderate European climate of Stuttgart).

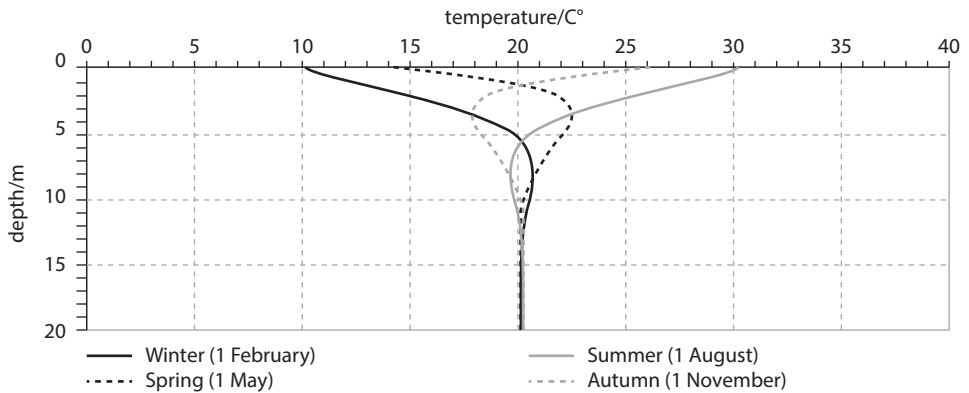


Figure 3.33 Development of soil temperatures for the American location Austin/Texas with 20.1°C annual average ambient air temperature.

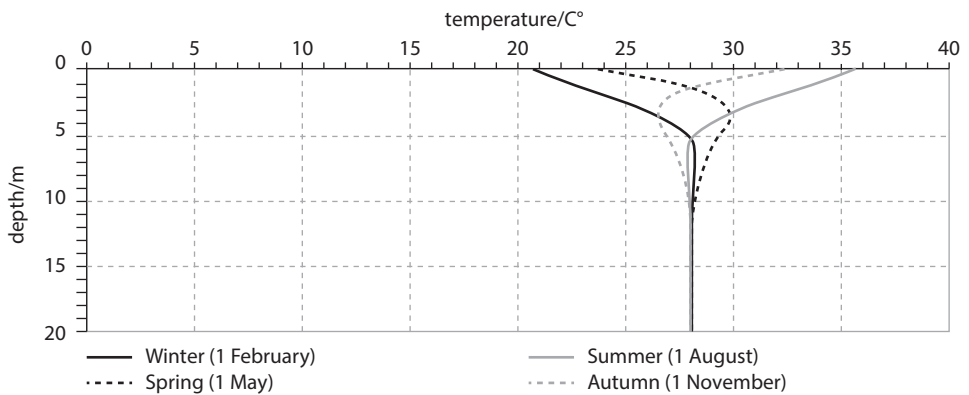


Figure 3.34 Development of soil temperatures for the hot Bangkok/Thailand location with 28.1°C annual average ambient air temperature.

The geothermal heat flux from the 3000 to 6000 K centre of the Earth is very small at 0.06 W m^{-2} ; only about 0.3% of the solar energy absorbed. The heat generated in the Earth's core is mainly attributed to natural radioactive decay. Temperature levels at the Earth's inner to outer core boundary at around 3500 kilometres depth are estimated to be about 5000°C. In around 80 - 100 km depth temperature levels in highly viscous or partially molten rocks are between 650 and 1 200°C. In some regions with geothermal anomalies, the near surface temperature gradient can be as high as 10 K per 100 m.

The undisturbed temperature of the soil T_{soil} depends on the thermal properties of the soil. Only the upper part of the ground is influenced by the ambient temperature. The annual course

of the temperature of a homogeneous soil can be calculated as a function of the thermal depth d_{th} , which is defined as:

$$d_{th} = d \sqrt{\frac{\pi \rho c_p}{\lambda L_a}} \quad (3.112)$$

where ρ , c_p and λ are the density, heat capacity, and heat conductivity of the soil, respectively. L_a denotes the length of a year. Assuming a sinusoidal change of the ambient temperature with an annual mean value $T_0(a)$ at a maximum monthly mean temperature $T_{0,m}$ the soil temperature is approximately:

$$T_{soil}(t) = T_{0,m} + (T_{0,m} - T_0(a)) \exp(-d) \cos(d_{th} - 2\pi t / L_a) \quad (3.113)$$

Three-year measurements of the undisturbed soil temperature for a project with a 2.8 m deep air to earth heat exchanger used for office air preheating and cooling in Germany showed reasonable agreement between measured soil temperatures and the simple sinusoidal approach.

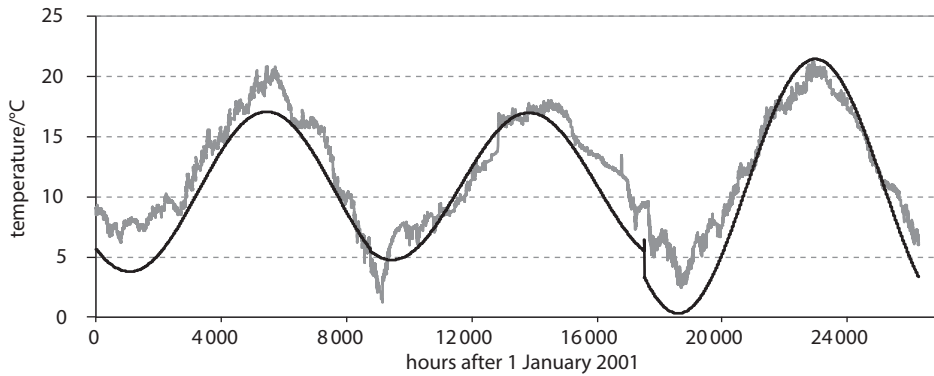


Figure 3.35 Measured (grey) and simulated (black) soil temperature at 2.8 m depth in Weilheim/Germany.

Typical values for the thermal properties of some soil materials are given in Table 3.4.

	$\rho/10^3 \text{ kg m}^{-3}$	$\lambda/W \text{ m}^{-1} \text{ K}^{-1}$	$c_p/\text{kJ kg}^{-1} \text{ K}^{-1}$
Sedimentary rocks			
Limestone	2.6 - 2.7	2.5 - 4.0	0.81 - 0.89
Marl	2.5 - 2.6	1.5 - 3.5	0.88
Quartzite	2.7	3.6 - 6.6	0.78 - 0.81
Salt	2.1 - 2.2	5.3 - 6.4	0.56
Sandstone	2.2 - 2.7	1.3 - 5.1	0.73 - 1.04
Claystone, siltstone	2.5 - 2.6	1.1 - 3.5	0.84 - 0.92
Unconsolidated rocks			
Gravel, dry	2.7 - 2.8	0.4 - 0.5	0.52 - 0.57
Gravel, wet	2.7	1.8	0.89
Sand, dry	2.6 - 2.7	0.3 - 0.8	0.50 - 0.59
Sand, wet	2.6 - 2.7	1.7 - 5.0	0.85 - 1.07

Table 3.4 Thermal properties of typical soil materials at approximately 20°C.

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4 Solar thermal heating

A low-angle, perspective shot of a solar thermal power plant under construction in a desert. The foreground and middle ground are filled with rows of heliostats (mirrors) mounted on a complex metal frame. The mirrors are highly reflective, showing bright highlights from the sun. The metal frame consists of numerous diagonal and horizontal beams. In the background, a small figure of a worker is visible on the left, providing a sense of scale. The desert landscape and a clear blue sky are visible in the distance.

Figure 4.1 (previous page): Installation in Doha, Qatar (Photo: Industrial Solar).



Figure 4.2 Combi-storage with solar and auxiliary heat exchanger as well as finned stainless steel heat exchanger for drinking water heating (Photo: Eric Duminil).

Heating and domestic hot water (DHW) requirements often dominate final energy consumption in many moderate climates, for example, with 42% in Europe, which corresponds to 3300 TWh primary energy. The solar thermal contribution is still low in most countries and often limited to DHW. Most solar thermal systems today are installed in single-family houses for hot water supply and in some cases for heating support, but they are usually not connected to a heating network. As seasonal storage is still expensive; the fraction of solar thermal energy of the total heat demand thus stays low at levels of 10 - 20%. To reach higher renewable fractions of the demand for heating and hot water supply, the solar thermal energy contribution with good storage potential has to increase significantly. Other renewable heating sources such as biomass have a limited potential in most countries.

Building codes are used more and more to increase the renewable fraction of the heating demand, for example, since 2009 in the German renewable-energy-law for heat (EEWärmeG), which requires that in new buildings 15% of the thermal energy demand for heating and cooling is covered by solar energy, unless biomass is used with 50% fraction or other efficiency measures are taken. In some federal state like Baden-Württemberg there is additionally the obligation to use renewable heat, if the heating system is replaced. With 0.04 m² collector surface per square metre living space, the requirements are already fulfilled.

Solar thermal collectors are used for domestic water heating, support of space heating of buildings, swimming pool heating, process heat generation and for solar cooling.

The dominating application area is still in detached or two-family houses. However, in most countries, more than half of the flats and hence the heat demand are in apartment buildings (53% in Germany). Until 2050, solar thermal energy could cover about 25% of the heat demand in an industrialised country such as Germany. This corresponds to a collector surface of 250 million square metres, given that the heat demand has reduced to 50% by then.

To reach these goals a structural change from today's individual heating systems (88% today in Germany) to network-based supply is necessary. Solar thermal energy systems can cover higher fractions of the energy demand if they are integrated in large solar-supported heating networks with or without seasonal storage.

Some very large solar thermal systems have been installed in district heating networks, mainly in Denmark, Sweden, Austria and Germany. Especially if cogeneration loses importance in networks with high renewable electricity input, for example, in Denmark with very high wind energy capacity, large solar thermal systems are very attractive. Seven new plants between 5 000 and 10 000 m² collector surfaces have been constructed in Denmark since 2006, for example, the Braedstrup plant with 8 000 m² and 8% solar fraction. The costs of solar heat are around 4 Euro cents per kWh and is thus very competitive with fossil fuel heat production.

Also, solar process heat has a high potential, for example, in Germany alone with about 100 GW substitutable thermal power. Double-glazed collectors with reduced heat losses are available on the market for applications of a temperature range over 100°C.

In the field of collectors, technological developments are still taking place, aimed at improving selective coatings of absorbers using sputter technology, optimising the heat transfer from the absorber to the tubes through improved welding or soldering technologies (ultrasonic or plasma welding), reducing glazing reflection losses through antireflective layers, optimising vacuum collector geometry and introducing low-level concentrations of the irradiance (CPC collectors).

In systems engineering, work is taking place on simplifications of the ducting system (common ducts including sensor cables within the same insulation), on improved heat exchangers, on new strategies for avoiding overheating, on special pumps for solar operation generating high pressures at low volume flows and low electricity consumption and on new frost protection fluids, which can handle the high-temperature levels generated in vacuum tube collectors.

In storage technology, combined heating/freshwater storage is increasingly being developed, enabling solar support for conventional heating in addition to solar heating of freshwater. The market share of combined heating/DHW collector systems has reached about 20% of the total collector surface area sold in Europe; in countries such as Austria it is up to 50%. New stratified loading systems and various storage tank geometries are under development.

4.1 Markets and economics

The main markets today are in China (87.5 GW_{th}), Europe (28.5 GW_{th}), and the USA and Canada (15.1 GW_{th}), which together account for 86% of the total installations. The rest of the market is shared between Japan (4.4 GW_{th}), Australia and New Zealand (4.4 GW_{th}), some Asian countries (4 GW_{th}), Central and South American countries (3.8 GW_{th}), the Middle East represented by Israel and Jordan (3.3 GW_{th}) and some African countries (0.9 GW_{th}) (data from Euroheat & Power and Renewable Heating and Cooling Platform Conference 2012).

The dominant collector type is the vacuum tube collector, which represents more than 50% of the global market, whereas flat-plate collector cover about one-third of the market. Unglazed plastic collectors account for 12% of the market, and air collectors represent the smallest share with 0.8%.

There are about 150 large-scale plants above 500 m² or 350 kW_{th} in operation in Europe with a total installed capacity of 160 MW_{th}. The biggest plants for solar-assisted district heating are located in Denmark with 13 MW_{th} (18 300 m²) and Sweden with 7 MW_{th} (10 000 m²). The largest reported solar thermal system for industrial process heat was installed in 2007 in China. The 9 MW_{th} (13 000 m²) plant produces heat for a textile company.

Since the mid-1980s, the annual installed collector surface increased from 25 000 to more than 2.2 million m² or 1.5 GW installed capacity in Germany as the largest market in Europe. Europe now has about 5 million m² of newly installed collector area every year. The European Solar Thermal Technology Platform (ESTTP) assumes that by 2030, about 50% of the overall heat consumption of the EU can be covered by solar thermal energy.

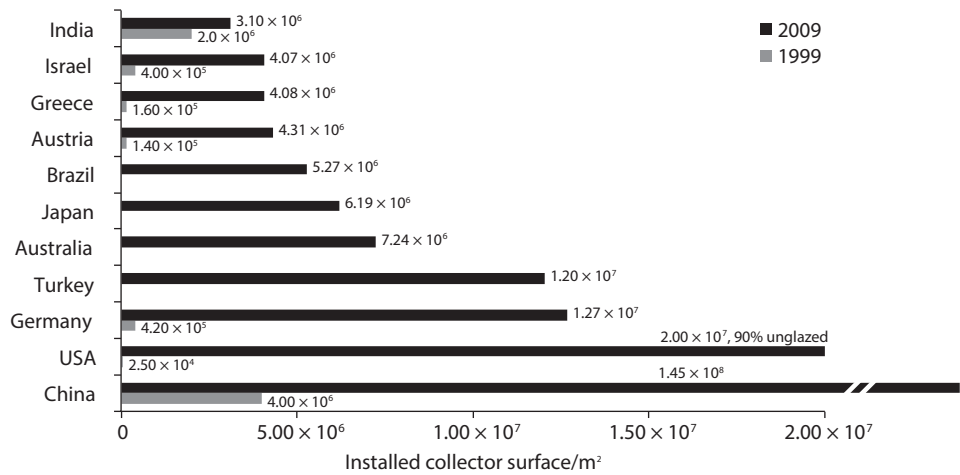


Figure 4.3 Installed solar thermal collector surface area in 1999 and 2009.

To judge the performance of solar thermal systems, some benchmarking figures are required for comparison.

The energy requirement factor defines the necessary energy need for a desired benefit (for example, heating). If the necessary energy input is converted to primary energy (with a factor e_{el} for electricity and e_{fuel} for fossile fuels), this factor can be used to convert the calculated heating demand (benefit) in primary energy.

$$e_{PE} = \frac{E_{el} e_{el} + E_{fuel} e_{fuel}}{Q_{heat} + Q_{DHW}} \quad (4.1)$$

The primary energy requirement factor has been calculated by the Fraunhofer ISE Institute for two building standards, an old building with high heating supply temperatures and a rehabilitated building with low supply temperatures. While the primary energy requirement for a gas condensing boiler is 1.24 for both buildings, the combination of a gas boiler and a solar thermal system with 12 m² reduces to 0.93 in the old building or even 0.53 in a rehabilitated building. This is mainly due to the better performance of the solar thermal system in a building with low supply temperatures and shows that the primary energy requirement factor is not independent of the building standard.

Alternatively, relatively primary energy savings can be calculated compared to a reference system.

To compare the economical performance of different systems, several benchmarking figures can be used, such as investment cost, static or dynamic amortisation time, internal rate of return, etc.

For an investor, a good method to compare systems is to calculate the life cycle costs of different systems using the capital value method, where all future costs are discounted to the present value. The total life cycle costs of an investment can then be compared to the benefit of placing the available capital on the capital market.

For a building owner, the annuity method is more useful, as the owner mainly wants to know what the resulting regular payments are. The costs per saved primary energy can then be calculated.

Solar thermal system costs

A standard flat-plate collector in Germany costs from 120 to 450 Euro m⁻² collector surface and provides annually around 300 - 550 kWh m⁻² solar heat. The lowest costs quoted are about 70 Euro m⁻². Vacuum tube collectors still cost between 500 and 900 Euro m⁻² for small plants. For plants with more than 100 m² collector surface, prices around 300 - 500 Euro m⁻² are possible. On the international market, however, prices of vacuum tube collectors of only \$50 m⁻² are also quoted. Parabolic trough concentrating collectors produced in India are today 120 Euro m⁻² (Thermax company information).

The system costs for small domestic systems in Germany are between 600 and 1500 Euro m⁻² collector surface depending on manufacturer and type. They have not significantly decreased during the last decade, so that small systems in Germany still cost 900 Euro per square metre, of which more than 50% are installation costs. Here a cost reduction potential of 10 - 35% seems possible, if systems can be mounted more easily and function without maintenance. System prices vary largely between countries: in European countries they are typically between 5000 and 7000 Euro total costs, whereas countries such as Turkey have very low average system prices of about 600 Euro (with 37% for installation costs) or Brazil with a system price of about 600 Euro and only 10% installation costs.

Apart from decreasing installation costs, cost reduction potentials seem possible for material costs, if low stagnation temperatures enable the use of cheaper materials, if thermal losses are reduced and if good electronic controllers check the system performance.

At investment costs of 4000 - 6000 Euro for a solar plant for DHW production and a service life of 20 years, the heat production costs for a small plant without subsidies in a moderate European climate are between 0.14 and 0.30 Euro kWh⁻¹. With regard to one square metre useful building floor area, additional investment costs of about 35 - 70 Euro m⁻² are necessary for the solar thermal system, i.e. only a very low percentage of total building costs. For large-scale solar installations over 100 m² gross collector surface, system costs of about 450 - 600 Euro m⁻² gross collector surface are realistic depending on storage size. This results in heat energy prices between 0.05 and 0.15 Euro kWh⁻¹, which can be already competitive today. The energy payback period of a solar plant for DHW supply lies between 12 and 24 months, the economic amortisation still takes about 13 years for small systems.

The best cost-benefit ratio is obtained for large-scale solar installations with short-term storage. For collector surfaces of around 1000 m², the system costs are around 375 - 465 Euro per m² of collector surface, including planning and value-added tax (for example, in the German projects Stuttgart-Burgholzof, Goettingen, Neckarsulm). In the context of the German funding programme, "Solarthermie 2000", the determined real solar heat costs are between 0.1 and 0.13 Euro kWh⁻¹. Here, too, the collector costs plus assembly and pipework each account for one-third of the total costs; the storage and system control costs clearly fall with size. Operation and maintenance costs in large district heating networks are only about 0.5 Euro MWh⁻¹.

Solar district heating systems with long-term heat stores installed in Germany had higher system costs of 620 Euro m⁻² including planning and value-added tax. Here, however, the solar fraction of the total heat requirement lies between 40% and 70%, while solar heated drinking-water supply in small- or large-scale installations covers at most 20% of the total heat requirement.

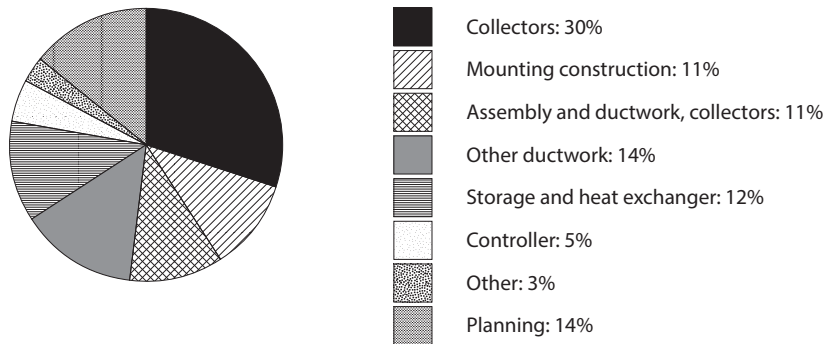


Figure 4.4 Cost distribution of 35 large-scale solar thermal installations with short-term stores from the German demonstration programme "Solarthermie 2000".

Solar pool heating

Solar pool heating can take place by using cost-effective absorbers, because the water temperature is normally under 30°C. The pool is used as storage for solar energy. An additional storage tank is not needed. Solar plants for heating open-air pools can run economically with heat prices between 0.04 and 0.07 Euro kWh⁻¹ under German conditions.

The pool water flows directly through the absorber and is heated by 2 - 4 K. Normally 0.5 - 0.8 m² absorber surface per m² water surface is enough. At small temperature increases, high flow rates of 80 - 110 l m⁻² h⁻² are usual. The annual heat gains are about 250 - 300 kWh m⁻² a⁻¹ under German climatic conditions.

Plastic absorbers can be installed cost-effectively directly on flat roofs or roofs with small gradients without framework. Without installation, one square metre collector surface costs around 35 Euro. For a large plant with 1500 m² collector surface and total costs of about 100 Euro per m² including planning, installation and pipework, a heat price under 0.05 Euro per kWh can be reached and thus, the payback period is under 5 years. Out of 3500 public open-air pools in Germany, 800 are equipped with solar plants.

4.2 System overview

Solar thermal systems can be constructed both for decentralised water heating with small collector surfaces of 4 - 8 m² and 300 - 500 litres storage capacity, and for central drinking-water heating in housing estates, hospitals, sports halls, etc., with collector surfaces of over 100 m² and storage capacities of 10 m³ or more. Decentralised heating support requires system sizes of around 10 - 20 m² and buffer storage volumes of 1 - 2 m³ volume.

When high solar fractions of 50% or more are wanted, seasonal storage is necessary. A central supply of solar-produced heating energy for housing estates (solar district heating) with seasonal storage requires at least 10 times more storage capacity, namely 1 - 2 m³ of seasonal storage per square metre of collector surface instead of 0.05 - 0.1 m³ for decentralised applications.

Solar-thermal systems are modularly structured with collector units of around 2.5 - 10 m² surface, which can replace conventional roofing material and take over the insulation function of the roofing. Completely prefabricated collector roofs including rafters and thermal insulation offer an economical solution, in particular for large-scale installations.

In contrast to photovoltaics, where modular system expansions are possible, all the components of a solar thermal system must be matched exactly with each other in the planning phase, since the pipework sizes and the pumping power are usually reduced to a low limit for economic reasons, and leave little scope for modular extensions. Storage and heat exchangers are, again with cost in mind, likewise designed for the collector rated output; they can, however, be complemented by additional units.

For buildings, solar thermal heating of drinking water and heating support are especially relevant. Concentrating collectors are interesting for solar cooling or process heat production. Some concentrating systems are now available which can be easily roof-mounted. Other high-temperature systems such as solar tower power plants require large open spaces and are not relevant for integration in buildings or urban areas.

Thermal collector types

Thermal solar collectors can be distinguished by their construction type and by the heat transfer medium. Most of the collectors use water as transfer medium, enriched with antifreeze, and some systems run with air.

The collector types differ primarily by the heat losses between the absorbers and their surroundings.

Glazed covers reduce the long-wave radiation and convection; an additional evacuation between absorber and glazing avoids almost completely the convection losses.

While in Germany, so far only 12% of the collector market are vacuum tubes, in China, the biggest customer country for solar thermal energy systems, almost all collectors are vacuum tubes.

Swimming pool absorbers

The simplest type of collector is a non-covered absorber for swimming pool heating, generally made of UV-resistant plastic (polyethylene [PE], polypropylene [PP] or ethyl propylene dien monomers [EPDM]). PP absorbers are not frost-proof when filled. During winter months, the water-filled absorber have to be emptied. The durability of PP absorbers can be more than 20 years. The lifetime of EPDM plastics is over 30 years. This absorber is also frost-proof when filled.

The heat transfer coefficient or U value of the front (U_f) in the direction of the incident irradiance is calculated from the reciprocal value of the thermal resistance R_{a-o} between the black absorber with temperature T_a and ambient air T_o . The thermal resistance R_{a-o} consists of the parallel heat transfer resistances for radiation ($1/h_r$) and wind-speed dependent convection ($1/h_c$); it is in the region of $0.04 \text{ m}^2 \text{ K W}^{-1}$. From it results a heat transfer coefficient U_f of $25 \text{ W m}^{-2} \text{ K}^{-1}$.

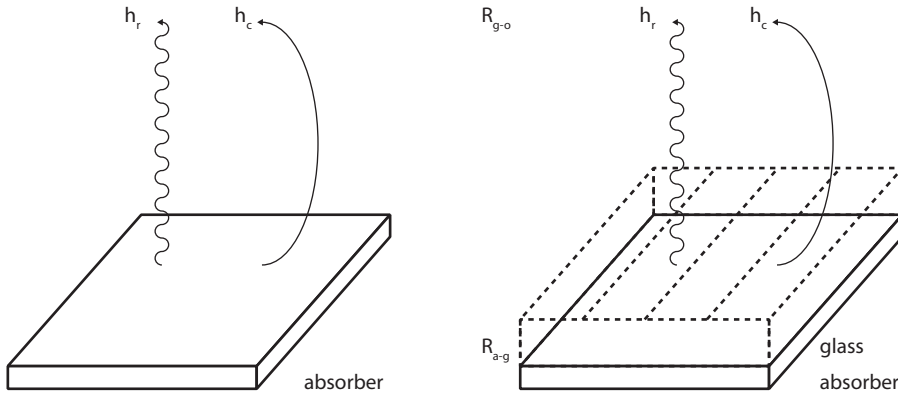


Figure 4.5 Heat losses of a non-covered black absorber (left) and heat loss through the transparent cover of a collector (right).

$$U_f = \frac{1}{R_{a-o}} = \frac{1}{1/(h_r + h_c)} = h_r + h_c \approx 25 \frac{W}{m^2 K} \quad (4.2)$$

Flat-plate collectors

With most thermal collectors, the calorific loss is reduced by a transparent glass covering of the absorber. The thermal resistance of the standing air layer between the absorber and the glass covering R_{a-g} , of around $0.1 - 0.2 \text{ m}^2 \text{ K W}^{-1}$, adds to the outside thermal resistance R_{g-o} between the cover glass and the ambient air. R_{g-o} consists again of a radiation and a convective proportion, and corresponds roughly to the total resistance R_{a-o} of the non-covered collector. Altogether the result is a heat transfer coefficient of around $5 - 6 \text{ W m}^{-2} \text{ K}^{-1}$.

$$U_f = \frac{1}{R_{a-g} + R_{g-o}} \approx \frac{1}{0.15 + 0.04} = 5.3 \frac{W}{m^2 K} \quad (4.3)$$

If the absorber is selectively coated, the radiation exchange between the absorber and the transparent cover is significantly reduced, and the U_f value falls to around $3 - 3.5 \text{ W m}^{-2} \text{ K}^{-1}$.

The flat-plate collector converts absorbed short-wave direct and diffuse solar radiation into available heat, under deduction of heat loss by the transparent cover as well as the insulated backside (Figure 4.6).

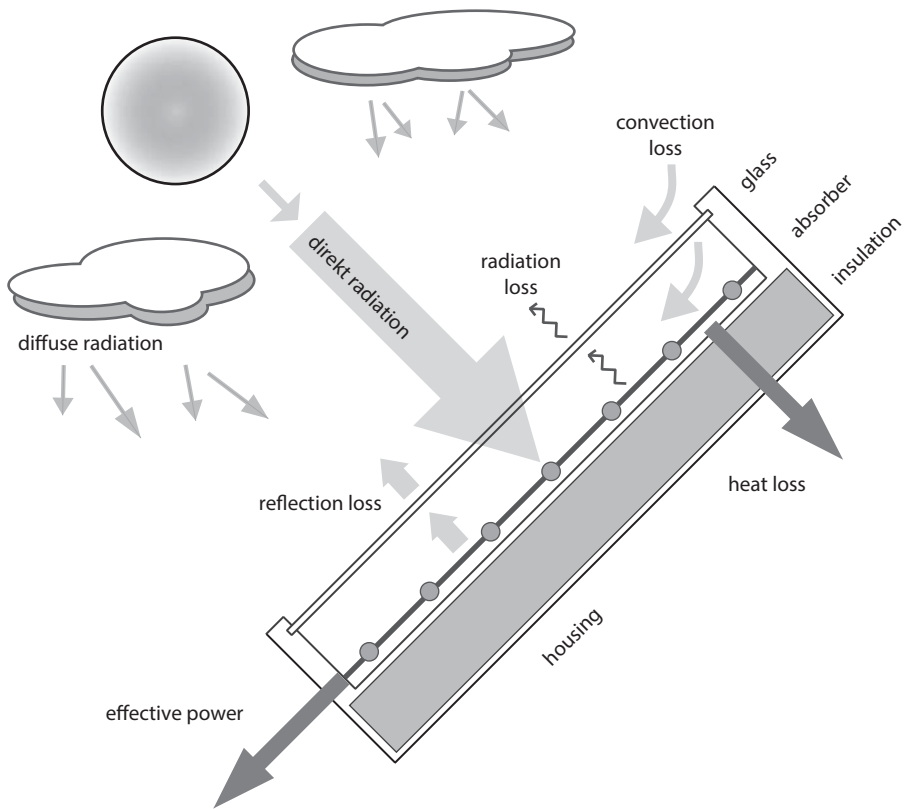


Figure 4.6 Section through a flat-plate collector with effective power balance due to irradiation gain and heat loss.

Vacuum tube collectors

In vacuum collectors the absorbers are also separated thermally from their surroundings by a transparent cover. In addition to the reduction in radiation exchange, convective heat transport is prevented by a very good vacuum of approximately 10^{-3} - 10^{-2} Pa, and the thermal resistance between absorber and cover rises to just about $1 \text{ m}^2 \text{ K W}^{-1}$. The result is U_f values of vacuum collectors of around $1 \text{ W m}^{-2} \text{ K}^{-1}$.

A range of different vacuum tube geometries is available on the market. The most compact vacuum tube collector including compound parabolic concentrator (CPC) mirrors is only 45 mm thick. The absorber is either a standard finned tube design within an evacuated glass tube or is coated directly on the inner surface of a double glass tube, with the heat transferred to liquid circulating in separate tubes on the inside of the double glass.

The heat transport either takes place by direct fluid flow through the absorber tubes or indirectly by a heat pipe. The heat pipe principle consists of a refrigerant, which transports heat from the absorber to the heat exchanger at the upper collector end. The refrigerant, e.g. water or alcohol, evaporates by the heat of irradiance absorption and moves up in the heat pipe. The heat

transfer fluid, which flows through the other side of a heat exchanger, takes up the heat and the refrigerant condenses. Because of the collector inclination, the liquid refrigerant flows down to the bottom again (Figure 4.7).

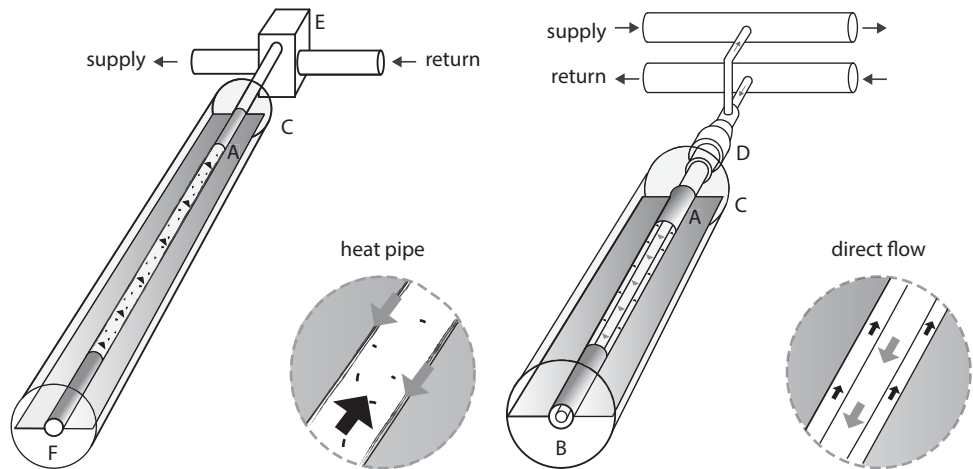


Figure 4.7 Vacuum tube collectors with heat pipe and direct flow through. (A, absorber; B, double tube; C, vacuum glass tube; D, tube connections; E, heat exchanger with condenser; and F, heat pipe with water or alcohol filling).

To reduce the heat losses and mechanical stress at the metal/glass connection, the so-called “Sidney” vacuum tube with an evacuated double glass was developed, where the absorber consists of a coating of the inner glass. To use the complete round absorber surface, often additional reflectors, so-called CPC collectors are used (Figure 4.8). The external mirror is subject to weather conditions. Alternatively the reflectors can be installed on the inside of the external glass tube.

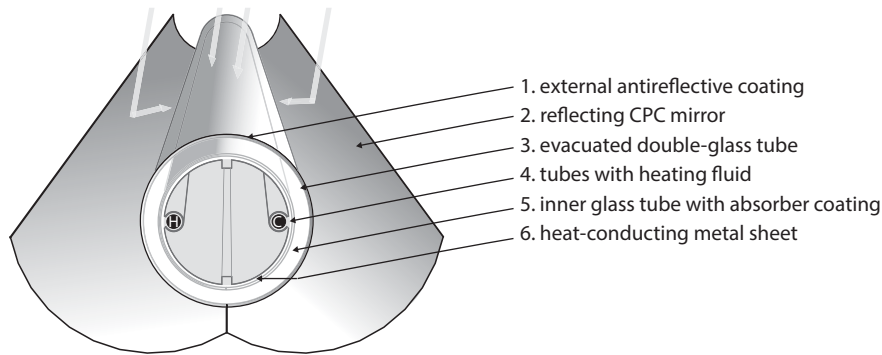


Figure 4.8 Double-glass vacuum tube collector with CPC reflector.

Collector type	Frontal heat resistance between absorber and ambient	Front heat transfer coefficient $U_f/\text{W m}^{-2} \text{K}^{-1}$
Uncovered swimming pool absorber	Direct radiative and convective heat exchange with environment	> 20
Flat-plate collector with uncoated black absorber	Standing air layer between absorber and transparent cover	5 - 6
Selectively coated flat-plate collector	Standing air layer with reduced radiative exchange	3 - 3.5
Selectively coated vacuum tube collector	Reduced convective and radiative heat exchange between absorber and transparent cover	1 - 1.5

Table 4.1 Classification of thermal collectors by the front U_f value.

Parabolic concentrating collectors

Parabolic concentrators are especially useful for high-temperature applications from about 100°C to several hundred degree Celsius, where their efficiency is higher than that of vacuum tube collectors. For double or triple effect solar cooling systems, temperature levels of 150 - 250°C are required, where parabolic concentrators can be efficiently used. Some first designs exist to place parabolic concentrators on building roofs. As they depend on direct irradiance, climates with a high direct irradiance proportion are advantageous. At low investment costs of about 300 Euro/m², the solar heat costs are as low as 0.045 Euro kWh⁻¹ on a Turkish site with 1900 kWh m⁻² direct normal irradiance, and 0.11 Euro kWh⁻¹ on a southern German site with 890 kWh m⁻² (Milow, 2002). In India, the company Thermax produces parabolic concentrating collectors for 120 Euro m⁻².

Fresnel collectors

Concentration can also be achieved using so-called Fresnel lenses, where the parabolic shape of the concentrator is broken down into small reflecting strips, which are tilted on a horizontal frame with various orientations. The advantage of Fresnel collectors are the high area coverage and very low wind loads, which makes them especially adapted for building integration. Modular concentrating collectors can be constructed with a height of about 4 m for the secondary concentrator and receiver tube at an area weight of 40 kg m⁻². The peak power is about 500 W m⁻².

Simple solar thermal collector model

The available power and efficiency of thermal collectors is determined by the optical characteristics of the transparent cover and the absorber, and by the calorific losses between absorber and environment. If the mean absorber temperature T_a is known, the surface-related available power is obtained simply by looking at an energy balance. The irradiance transmitted through the cover with a transmission coefficient τ and absorbed with an absorption coefficient α minus the calorific losses results in the available useful power $\dot{Q}_u$ per square metre of collector surface A .

$$\frac{\dot{Q}_u}{A} = G\tau\alpha - U_t(T_a - T_o) \quad (4.4)$$

The efficiency of the solar thermal collector is given by the ratio of the area-related available heating power divided by the irradiance G in the plane of the collector.

$$\eta = \frac{\dot{Q}_u}{AG} = \underbrace{\tau\alpha}_{\eta_0} - U_t \frac{(T_a - T_o)}{G} \quad (4.5)$$

η_0 represents the optical efficiency of the collector. The heat transfer coefficient of the collector U_t consists of the losses at the collector front, back and sides. The ratio of the temperature difference between absorber and environment $T_a - T_o$ and irradiance G is called the reduced parameter.

The temperature on the absorber sheet T_a is a complicated function of the distance from the heat-removing fluid tubes and the tube length, so an average value can only be determined very laboriously from a measured temperature distribution. What is measurable, however, is the fluid inlet temperature into the collector or the mean fluid temperature, which at not too low flow rates is given by the arithmetical average value between entry and exit temperatures. The representation of the available heating power as a function of the fluid inlet temperature is the most useful for system simulations, since the fluid inlet temperature is given by the storage return temperature.

In collector tests the efficiency characteristic is given as a function of the mean fluid temperature. The efficiency factor F' is obtained from the analytical solution of the differential equation of the collector's heat balance on the absorber. F' is defined as the ratio of real thermal power to the theoretical maximum power if the absorber surface would be on the lower temperature level of the mean fluid temperature T_m . In reality, the absorber temperature is higher than the fluid temperature due to heat resistances between the absorber and the fluid.

$$\eta = \underbrace{F' \tau\alpha}_{\eta_{0,eff}} - F' U_t \frac{(T_m - T_o)}{G}$$

Furthermore, the heat losses $F' U_t$ are not constant, but increase with temperature, as the radiant heat exchange rises with T^4 . This non-linear temperature dependence is often simply approached with:

$$F' U_t = a_1 + a_2 (T_m - T_o)$$

As a result, the following simple collector efficiency equation is obtained:

$$\eta = \eta_{0,eff} - \frac{a_1 \Delta T}{G} - \frac{a_2 \Delta T^2}{G}$$

The parameters of this equation are determined experimentally following the standard EN 12975-1. Typical values for a flat-plate collector related to the absorber surface area are $\eta_{0,eff}$ between 75 and 85%, a linear heat loss coefficient a_1 between 2.6 and 4.0 W m⁻² K⁻¹ and a quadratic loss coefficient a_2 between 0.006 and 0.022 W m⁻² K⁻¹. For vacuum tubes, typical optical efficiencies are between 62 and 77 %, a_1 is between 0.8 and 1.7 W m⁻² K⁻¹ and a_2 between 0.001 and 0.02 W m⁻² K⁻¹.

Some characteristics of commercial unglazed swimming pool absorbers, flat-plate and vacuum tube collectors are shown below. The best operation ranges can be seen from the diagram. With increasing temperature difference between fluid and ambient air, the efficiency of single-glazed flat-plate collectors drops more strongly than for vacuum tube collectors. The intersection

of the efficiency with the ordinate gives the effective optical efficiency $\eta_{0,eff}$ at zero temperature difference between the mean fluid temperature and ambient air ($\Delta T = 0 \text{ K}$). For a given temperature difference to ambient air, the efficiency rises with increasing irradiance (the so-called reduced parameter $\Delta T/G$ is smaller).

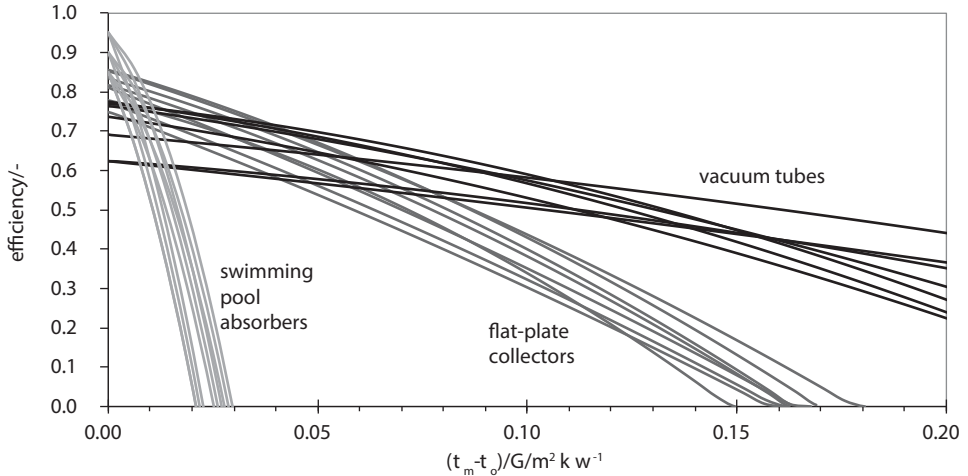


Figure 4.9 Efficiency of commercially available unglazed, flat-plate and vacuum tube collectors.

If the irradiance is not normal to the collector plane, reflection losses increase. This can be considered by introducing incidence angle modifiers (IAMs), which indicate the additional reflection losses relative to normal incidence of the direct and diffuse irradiance.

Installation

Flat-plate collectors can be installed as in-roof solution with a good aesthetics and no additional wind load. In cases of installations on existing roofs, often, on-roof solutions are chosen with collector attachments with rafter roof hooks with a distance to the roof covering of about 6 - 8 cm. On flat roofs, often, frame-mounted systems are built. In case of large solar plants, roof-tilted systems are 20 - 30% more expensive than the roof integration with large surface collectors. The gross roof surface is only used by 50 - 60% for tilted systems on flat roofs, because mutual shading must be avoided. On pitched roofs, the utilisation can go up to 90%. Because of the wind loads, the statics requirements are significantly higher. For collector systems with more than 50 m^2 , the collector modules should be bigger than 5 m^2 . Complete collector roofs are reliable and cost-effective because of the very fast installation within 1 - 2 days also for very large system with several hundred square metres. To guarantee for the waterproof quality of the roof, the pitch of the roof should be not less than 20° .

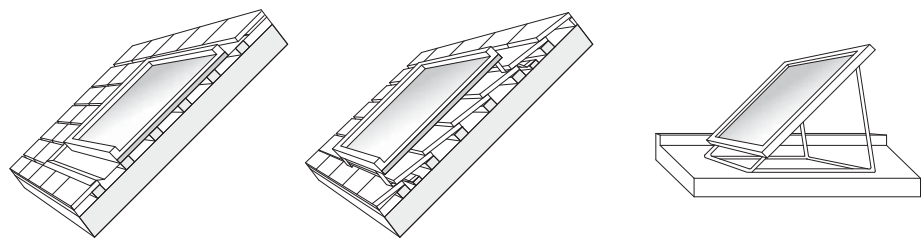


Figure 4.10 Installation types for flat-plate collectors on the roof: in-roof, on-roof and tilted frame-mounted system for flat roofs.

4.3 Systems engineering

Domestic hot water production

The systems engineering of a standard solar plant for heating drinking water comprises the main components collector and storage as well as the solar circuit hydraulics and safety system. For small drinking-water systems under 10 m² collector surface area, simple sizing rules have proved satisfactory, proceeding from an average warm-water consumption of around 30 - 60 litres per person and day, and typical collector yields of 350 - 450 kWh m⁻² a⁻¹ in moderate climates with global horizontal irradiance levels of about 1000 kWh m⁻². Such systems contain the following components:

Component	Dimensions
Flat-plate collector	1.25 - 1.5 m ² per person
Vacuum tube collector	1.00 - 1.2 m ² per person
Storage tank (pressurised)	40 - 70 l per m ² collector surface area
Heat exchanger in collector circuit (integrated in storage tank)	30 - 40 W K ⁻¹ power per m ² collector surface (corresponds to 0.15 - 0.20 m ² surface area)
Pipework	15 mm external diameter up to 8 m ² collector surface and 50 m pipe length
Solar station with pump, pressure gauge, security valve, expansion vessel	25 - 80 W pumping power (frequency controlled special solar pump is best, otherwise smallest available heating system pump)
Controller (temperature-difference measurement between collector and storage tank, maximum temperature limitation storage tank)	Relay for pump control

Table 4.2 Components of a standard solar plant for domestic hot water production.

With a system of these dimensions, an annual solar fraction of 40 - 60% of the DHW consumption can be obtained, about 70 - 100% in the summer and 10 - 20% in the winter.

Besides the selection of the collector and storage, the hydraulic system of the collector circuit with ducting, valves and pump has to be dimensioned, and also the safety engineering with the pressure level, expansion vessel and excess-pressure protection has to be worked out. With a standard solar plant for DHW heating, all hydraulic and safety-related functions are brought together in a pre-mounted solar station.

Heating support with solar thermal systems

Between the monthly solar irradiance or collector yield maxima and the heating requirement of buildings, typically a half-year phase shift exists. Sensible heat stores have, however, only a small heat storage capacity of a few days. This means that decentralised heating support systems should be designed to cover no more than 15 - 30% of the heating requirement, as otherwise summer overheating, longer standstill periods and falling surface-specific collector yields are inevitable. Collector surfaces between 10 and 20 m² per housing unit can be recommended for DHW and heating support with no significant specific yield reduction.

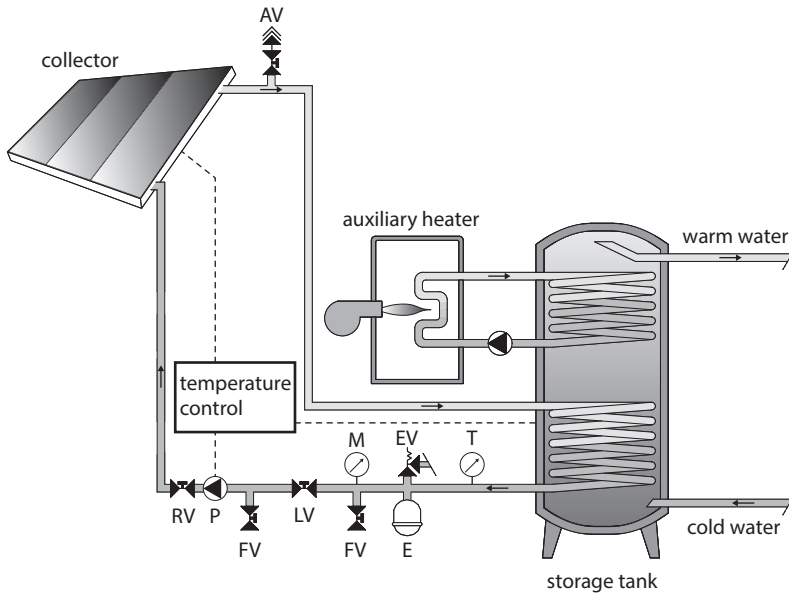


Figure 4.11 Standard solar plant for domestic hot water supply – the system consists of a collector, storage, temperature-difference regulation and the following components: AV, air vent; T, temperature sensor; E, expansion vessel; EV, excess pressure valve; FV, filling/emptying valves; M, manometer; LV, lockable valve; P pump; and RV, a lockable non-return valve.

At collector areas below 30 m², only short-term storage is necessary, and the system yield barely improves with storage size, as long as a minimum storage volume of about 0.8 to 1.5 m³ is available. More important for system yield is the correct hydraulic connection of the conventional heater to the storage volume: if, for example, the return flow to the boiler is connected at the bottom of the buffer store, the temperature stratification will be destroyed due to the high boiler volume flows and yield will drop by about 5 - 6 %.

The measured energy consumption and demand calculations for different insulation standards of an apartment building constructed in 2005 in Ostfildern, Germany with 1600 m² floor space shows the phase shift between solar production and energy demand. A solar plant dimensioned for DHW with 60 m² (corresponds to 1.5 m² per person) covers the entire hot water demand during five summer months. An increase of the solar thermal plant to 150 m² for heating support (with storage volumes between 3.6 and 9 m³), strongly overproduces during these five summer months.

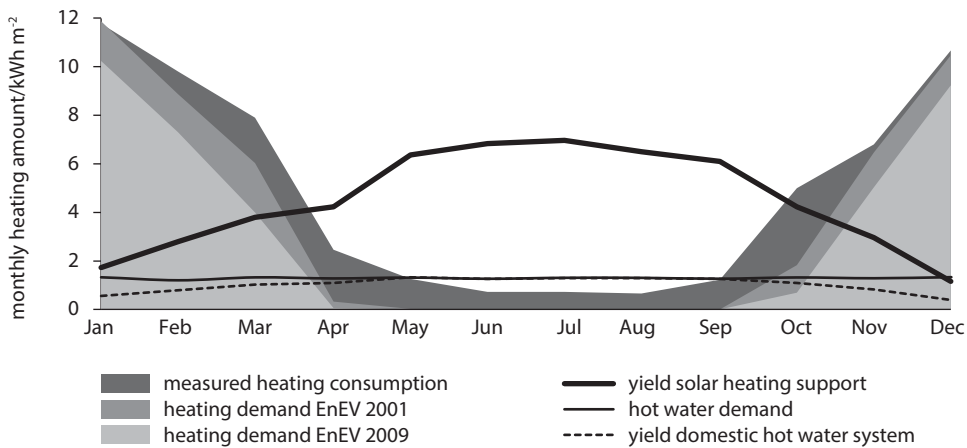


Figure 4.12 Heating and hot water demand of an apartment building with useful solar energy for hot water and the possible yield of a heating support system. Besides the measured values of the heat consumption, the calculated data of the German building standard EnEV 2001 and today's low-energy building standard EnEv 2009, are shown.

Solar circuit hydraulics

Pumps

For the collector circuit, conventional circulating heating pumps with a low power annular gap motor (< 100 W electrically) are still often used. These are optimally dimensioned for one-family or two-family houses, depending on the amount of heat needed, with flow rates between 1 and $4 \text{ m}^3 \text{ h}^{-1}$. Thermal solar plants for DHW are, however, usually driven with small flow rates, between $0.1 - 0.5 \text{ m}^3 \text{ h}^{-1}$, i.e. with only 10% of the delivery of standard heating pumps. With such small deliveries, the efficiency of a heating pump is between 2% and 7%, while with optimal sizing, an efficiency of up to 20% can be achieved. The small efficiencies lead to an insignificant electricity consumption of approximately 100 kWh a^{-1} in a small system with 2000 kWh yield per year, i.e. in terms of primary energy, 15% (300 kWh) is used for pump energy. Regulating the pump speed reduces annual power input by 50%.

The pumping height of a standard heating pump is relatively small, a maximum of $4 - 5 \text{ m}$. When using such pumps, de-aeration of the collector circuit at its highest point is important for the operation of the system. High supply pressures with small volume flow deliveries are obtained on the other hand with gear pumps with wet or dry rotating motors, which provide a pumping height of 35 m at a flow rate of $0.1 \text{ m}^3 \text{ h}^{-1}$, so the de-aeration valve, which is usually difficult to access in roof-mounted systems, is unnecessary, and the initial air cushion after filling the system can be fully squeezed out of the collector circuit.

Heat transfer liquid

The heat transfer liquid in the solar circuit must be provided with antifreeze in cold winter climates to avoid freezing of the external pipework system. Mixtures of water and antifreeze

agents (glycols, e.g. polypropylene glycol) are used; they are provided with different inhibitor salts for corrosion protection. Mixing proportions of approximately 40% antifreeze to 60% water are usual, causing the freezing temperature to sink to -20°C . With an increasing proportion of antifreeze the viscosity of the heat transfer liquid rises by a factor of 3 - 5, which must be taken into account in pressure-loss calculations, and which lowers the heat capacity by 10 - 20%.

With solar plants with high standstill temperatures (vacuum tube collectors), great attention must be paid to the temperature reliability of the antifreeze.

With very careful hydraulic design and a specific control strategy, it is possible not to use antifreeze fluid and to circulate storage water through the collectors in case of frost. This causes annual heat loss of about 2 - 4%, but saves electricity for the pumps due to the lower viscosity of pure water and only one pump for the solar circuit. It is also easier to integrate the solar thermal plant into district heating networks or heating distribution systems of large buildings. Vacuum tube collectors with low heat losses are a prerequisite for the good functionality of the concept.

Overheating protection and expansion vessels

The standstill temperatures of flat-plate collectors is around $160 - 200^{\circ}\text{C}$, for vacuum tube collectors between 200 and 300°C . Temperature-resistant thermal insulation has to be used and soft soldering of the collector loop pipework must be avoided. An advantage of vacuum tube collectors with heat pipes are the limitation of maximum standstill temperatures to about $130 - 150^{\circ}\text{C}$. At these temperature levels, the complete refrigerant content of the absorber is evaporated and located in the condensor.

The expansion vessel is an essential component of overheating protection, since, when the system is at a standstill, with the usual system pressure at 3 to 6×10^5 Pa, the solar circuit liquid boils in the collector. The expansion vessel in small systems has to take up not only the volume increase of the solar circuit liquid of approximately 10%, but also the complete evaporating collector fluid content V_c of 0.5 - 2 litres per square metre of collector surface.

Operational experience with large solar plants has shown that during standstill, evaporated heat transfer medium can be found in the whole collector field, in the collector field pipework and in about one-third of the collector circuit pipework. Sometimes steam is even found in the pipes up to the storage tank location. It makes sense to design the expansion vessel for that volume and if necessary to dimension the safety valve for a higher release pressure to reduce fluid evaporation events (up to 8 bar). The membrane expansion vessel must not be connected from below, otherwise air can collect in the vessel, which can no longer be removed.

The maximum volume increase ΔV of the solar liquid at system standstill is calculated from the maximum occurring temperature difference between the standstill temperature T_s and the temperature during initial filling T_o , from the volume expansion coefficient of the fluid β' , from the volume of the collector circuit contents V_{cc} and from the evaporated collector contents V_c .

$$\Delta V = \beta'(T_s - T_o)V_{cc} + V_c \quad (4.6)$$

The volume expansion coefficient β' rises with temperature and for water in the temperature range, $60 - 80^{\circ}\text{C}$ is about $5.87 \times 10^{-4} \text{ K}^{-1}$, for antifreeze mixtures, around $10 \times 10^{-4} \text{ K}^{-1}$. The volume of the collector circuit V_{cc} is calculated for the pipes and heat exchanger without the fluid contents of the collectors.

The gas cushion of the diaphragm expansion vessel with content V_g is compressed from the selected pre-pressure p_{min} when filling the system, to the maximum operating pressure p_{max} at

system standstill. To avoid air entry, the pre-pressure is selected to be about $0.2 - 0.5 \times 10^5$ Pa higher than the static pressure. This results from the difference in height between the collector field and expansion vessel Δh [m] with a static increase in pressure of 10^4 Pa per metre.

$$p_{min} = (\Delta h + 5) \times 10^4 \text{ [Pa]} \quad (4.7)$$

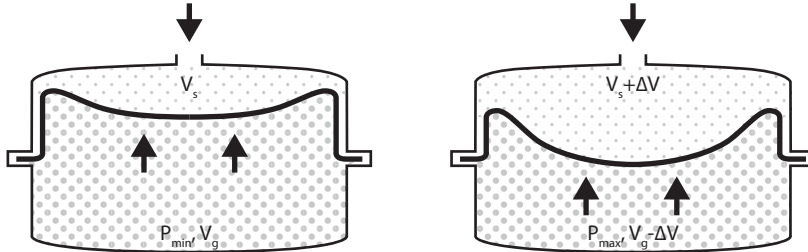


Figure 4.13 Expansion vessel at minimum and maximum operating pressures.

The maximum operating pressure p_{max} is given by the design pressure of the safety overflow valves. The volume of the expansion tank V_{total} must be designed in such a way that the volume compression of the gas volume V_g is sufficient, due to the difference between minimum and maximum operating pressure, to absorb the volume increase ΔV on the liquid side of the diaphragm. Based on the ideal gas equation, the result is, assuming a constant temperature:

$$p_{min} V_g = p_{max} (V_g - \Delta V)$$

$$V_g = \frac{p_{max}}{p_{max} - p_{min}} \Delta V \quad (4.8)$$

In addition to the necessary gas volume to absorb the volume increase, the liquid side of the expansion vessel always includes some safety water volume V_s , so that at very low temperatures the system pressure does not drop. This safety volume should be about 1 - 2% of the entire system volume, i.e. the collector circuit and collector itself ($V_{cc} + V_c$):

$$V_{total} = V_g + V_s \quad (4.9)$$

In large systems ($> 100 \text{ m}^2$), closed pressure expansion vessels are designed only for the volume increase of the liquid. Steam produced in the collector is blown off through safety or overflow valves, and the heat distribution medium is collected in a pressure-free receptacle. After the collectors have cooled, a filling pump takes over the automatic feedback of the liquid to the solar circuit.

The non-return valve *RV* is necessary for the avoidance of nocturnal cooling of the storage tank through a thermosyphonic rise of the solar circuit fluid.

According to commissioning procedures, for example, described in the German standard DIN 18380, the plant has to be entirely emptied before filling. Water filled into the plant for cleaning and pressure test has to be completely removed from the hydraulic system. Because of the configuration of the collector field, often a complete emptying of the plant is not possible. To

remove the water in the collectors also, during filling with antifreeze, the drain valve remains open until coloured heat transfer medium comes out.

Example 4.1

Calculation of the volume of an expansion tank for a 15 m² solar plant with a height difference between the expansion tank and collector field of 20 m and a safety valve with a maximum operating pressure of 4.5×10^5 Pa. For the system the following collector circuit volume is assumed:

Heat exchanger liquid content with 4.6 m ² of heat transfer surface:	3.6 litres
Pipe volume at a total of 40 m pipe length and 22 mm external pipe diameter (DN22):	12.6 litres

The volume expansion of the collector circuit contents V_{cc} for $\beta' = 10 \times 10^{-4} \text{ K}^{-1}$ and a standstill temperature T_s of flat-plate collectors of about 180°C ($T_o = 15^\circ\text{C}$) is about $10 \times 10^{-4} \text{ K}^{-1} \times (180 - 15) \text{ K} \times 16.2 \text{ l} = 2.57 \text{ l}$.

The evaporating collector content V_c is about 15 litres, so that $\Delta V = 17.67 \text{ litres}$. The safety water volume $V_s = 0.01 \times (3.6 + 12.6 + 15) \text{ l} = 0.31 \text{ l}$.

The minimal operating pressure given by the height difference is 2.5×10^5 Pa, so the expansion tank volume is

$$V_{total} = \frac{4.5 \times 10^5}{4.5 \times 10^5 - 2.5 \times 10^5} 17.67 \text{ l} + 0.31 = 40.1 \text{ l}$$

Heat exchangers

For heat transfer between the collector primary circuit and the storage tank, internal heat exchangers are usually chosen for small systems for reasons of cost and space. Gilled or plain pipe heat exchangers have a specific heat transfer rate of 200 - 500 W m⁻² K⁻¹.

With maximum surface areas of 1 - 2 m² within 300 - 500 litre storage tanks and a mean temperature difference of 5 K, internal heat exchangers are limited in their performance to about 5 kW. Thus, in larger systems, external heat exchangers requiring a second pumping circuit have to be used. Through the forced convection on both sides of the heat-transferring surface, the transfer rates of counter-current plate heat exchangers rise from 1000 to 4000 W m⁻², so at plate distances of a few millimetres per cube metre construction volume, very high power density can be transferred.

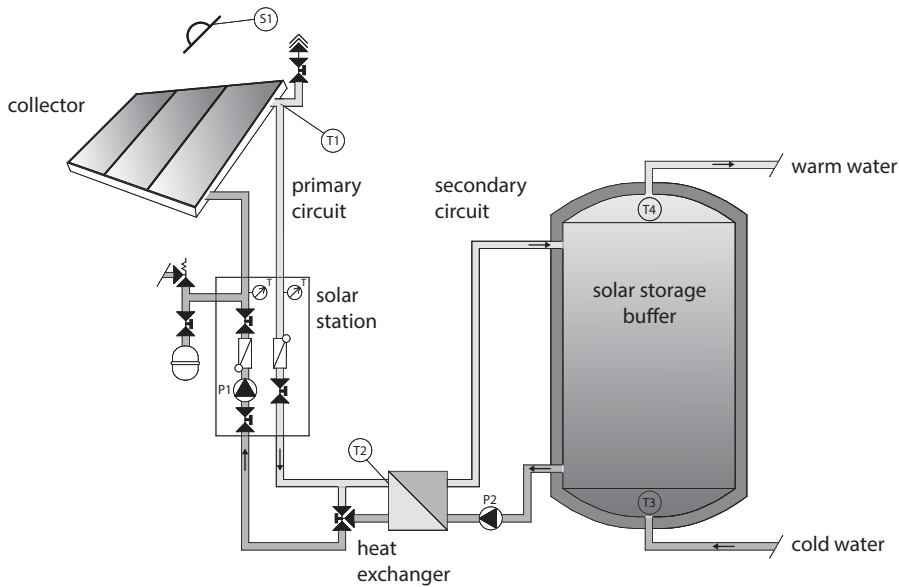


Figure 4.14 Circuit with external heat exchanger.

For the secondary circuit, a further pump is necessary, as well as an expansion tank and safety valves.

Collector interconnecting and pressure losses

For an even flow through the collectors at high flow rates, and thus good heat transfer between absorbers and fluid, serial interconnecting of collectors is favourable. A limiting factor is the pressure loss

$$\Delta p = \sum \xi \frac{\rho}{2} v^2$$

which rises as a square of the flow velocity $v = \dot{V} / A_s$. This determines the electrical power P_{el} of the solar circuit pump. The flow velocity is determined by the mass flow $\dot{m} = \rho \dot{V}$ in the collector circuit, which is about 10 - 15 kg per m² collector surface and hour in so-called low flow systems. In high flow systems it is between 30 and 60 kg m⁻² h⁻¹ and flows through the tubing cross section A_s (m²).

The pressure drops are determined with the help of pressure loss factors ξ , which are calculated from friction factors in pipes and from tabulated factors for various fittings. For bends and T-fittings, an overall addition to the pipe pressure losses of 50% is often assumed. The pressure losses of pumps, heat counters, etc. are determined from data sheets.

$$P_{el} = \frac{\dot{V} \Delta p}{\eta} = \frac{\dot{V} \sum \xi \frac{\rho}{2} \left(\frac{\dot{V}}{A_s} \right)^2}{\eta} \quad (4.10)$$

The efficiency η of small solar pumps at 100 W electrical output is around 2 - 7% and rises up to 70 - 80% for high-power dry rotor pumps. With small systems of up to 10 m² collector surface and 50 m pipe length, the pressure losses in the collector are less than 0.2×10^5 Pa and in the rest of the solar circuit about 0.3×10^5 Pa, so without a pressure loss calculation, small three-stage heating pumps ($P_{el} < 100$ W) can be used.

With medium system sizes of approximately 15 - 40 m², typical total pressure losses are between 0.3 and 0.8×10^5 Pa, a third of which is caused by the collector field, the pipes and the heat exchanger, respectively. In the collector field, the pressure loss should be limited to a maximum of 0.3×10^5 Pa by combined parallel/series connection. In systems between 40 and 100 m², a collector field pressure loss of 0.4×10^5 Pa is acceptable; the rest of the solar circuit should remain limited to 0.6×10^5 Pa. In large-scale installations over 100 m², typical pressure losses of 0.7×10^5 Pa in the collector field and 0.9×10^5 Pa in the solar circuit can be expected.

The electrical annual coefficient of performance as a ratio of available thermal energy yield to annual electrical auxiliary energy is typically around 20 for small solar systems; for bigger systems with highly efficient pumps, it is up to 50 (Wesselak and Schabbach, 2009).

Example 4.2

Calculation of the total flow rate of a small system of 10 m² with a heating pump of 80 W maximum power at 7% efficiency and a typical pressure drop of 0.5×10^5 Pa. Determine the pump power of a larger installation with 100 m² of collector surface and a specific flow of 30 kg m⁻² h⁻¹ with a pressure drop of 1.6×10^5 Pa.

With a pump efficiency of 7% and an electrical power of 80 W, the resulting flow rate is calculated as follows:

$$\dot{V} = \frac{\eta P_{el}}{\Delta p} = \frac{0.07 \times 80 \text{ W}}{0.5 \times 10^5 \text{ Pa}} = 1.1 \times 10^{-4} \frac{\text{m}^3}{\text{s}} = 0.4 \frac{\text{m}^3}{\text{h}}$$

With a 10 m² system that is about 40 l m⁻² h⁻¹, which corresponds to a typical flow of a standard system. For a large-scale installation with an assumed pump efficiency of 40%, the electrical power is

$$P_{el} = \frac{\dot{V} \Delta p}{\eta} = \frac{\frac{3 \text{ m}^3}{3600 \text{ s}} \times 1.6 \times 10^5 \text{ Pa}}{0.4} = 333.3 \text{ W}$$

Controllers

The usual temperature-difference controllers for warm-water systems control the solar circuit pump via a relay switch, as a function of the temperature difference between the collector exit and the store at the height of the heat exchanger. Temperature-difference regulation is usually provided by hysteresis, to avoid frequent switching of the pump during flow-begin and subsequent temperature reduction (switching-on temperature difference around 5 K, switching-off temperature difference 2 - 3 K).

Heat storage

For DHW production in single-family houses a storage volume of about 200 - 500 litres is sufficient. The costs for such small heat stores are about 3000 Euro per m^3 . Large solar plants with a rather small solar fraction are often installed with short-term, i.e. daily, storage. Cost-effective standard steel storage tanks with a maximum volume of 5 m^3 can be connected in series up to three units. The costs for such large heat stores are about 700 - 1500 Euro per m^3 . In solar thermal plants with several-day storage, steel buffer storage with up to 200 m^3 volume can be installed.

The energy densities of water storage are about 60 kWh m^{-3} , depending on the usable temperature differences between the hottest supply and coldest still useful temperature.

Latent heat storages based on salt hydrates or paraffin have about twice the energy density than water. Thermo-chemical storages are based on adsorption processes with silica gel or zeolite with a two to three times higher energy storage density. Chemical reactions can have up to 10 times more energy density than water storage. By dehydration of salt hydrates in the temperature range of $100 - 150^\circ\text{C}$, storage energy densities of about 630 kWh m^{-3} are achieved. First prototypes are tested under laboratory conditions.

Short-term stores in solar thermal systems, for heating drinking water and heating support, are predominantly steel tanks with pressure levels between 2 and $6 \times 10^5 \text{ Pa}$. For direct drinking-water storage, these are either enamelled on the inside or made of high-grade steel. The stored drinking or heating water has a thermal capacity of $4190 \text{ J kg}^{-1}\text{K}^{-1} = 1.16 \text{ Wh kg}^{-1} \text{ K}^{-1}$, so at a usable temperature difference of about 40°C , an energy quantity of

$$Q_{st} = mc_p \Delta T = 1 \text{ kg} \times 1.16 \text{ Wh} / \text{kgK} \times 40 \text{ K} = 46.4 \text{ Wh} \quad (4.11)$$

per litre of storage volume can be stored. If a heater buffer store is loaded to 80°C and a supply temperature of 40°C can still be used in low-temperature heating systems, 46 kWh of usable energy can be stored with a 1000 litre store. In a building with a heating power requirement of 5 kW , about 10 h of the heating energy requirement can be covered, i.e. not even a full day.

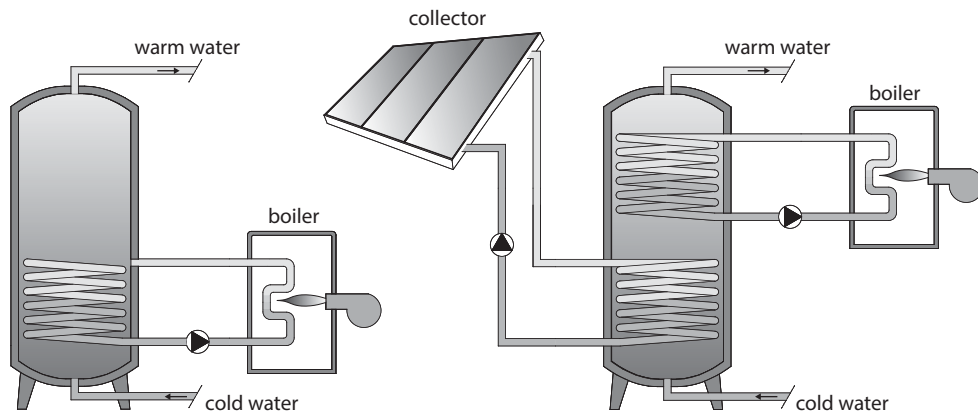


Figure 4.15 Conventional drinking-water storage with one heat exchanger (left) and solar-storage with two heat exchangers (right).

Drinking-water stores for solar plants are equipped, in contrast to conventional stores, with two heat exchangers, one of which must be situated in the lower, cool storage area, in order to utilise even small rises in temperature at the collector. The second internal heat exchanger is necessary for auxiliary heating in the upper third of the store (the standby section). From the solar heat transfer in the lower storage area alone, and the subsequent free convection of the heated liquid within the drinking-water store, a high system inertia is the result, so usable temperatures of $> 40^{\circ}\text{C}$ are only achieved after approximately 4 hours of irradiance.

Because of the fixed arrangement of the auxiliary heat exchanger, a certain fixed storage volume is always on a high-temperature level. This leads to unnecessary heat losses particularly at small tap rates. Studies in Denmark with variable auxiliary heating volumes show that depending on the tap profile, 5 - 35% more solar yield can be reached with intelligent storage loading (Furbo et al., 2005).

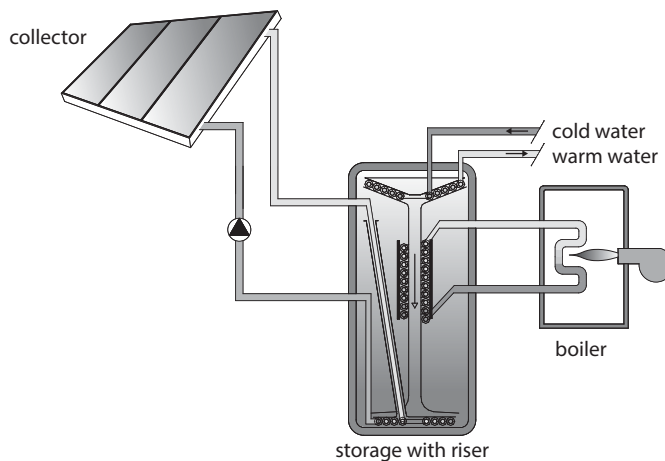


Figure 4.16 Drinking-water storage with a riser to bring solar heat quickly into the upper storage area.

In more flexible storage concepts, the solar-produced heat is led directly into the upper storage area to reduce system inertia. This is possible, for example, via a riser situated in the store. The warm collector water is led through a tube in a riser from above to below; the storage water warms up first in the riser around the collector tubing and flows up into the upper storage area with the drinking-water heat exchanger. The cooled storage water at the heat exchanger sinks in a second conduit pipe back into the lower storage area. The drinking-water is warmed up via an internal heat exchanger when flowing through the upper storage area.

Alternatively, the warm collector water can be directed at high temperatures directly to the upper storage area, via external switching with three-way valves. At low collector temperatures, the solar heat is brought into the colder lower storage area.

Apart from heating drinking water, such heating-water-filled buffer stores enable a simple solar heating support, since from the middle storage area heat can be drawn off directly for space heating. The hydraulic connection of the buffer store to the conventional heating system must, however, be carefully planned for the preservation of the temperature stratification and for

negative consequences on the efficiency of condensation boilers, if return temperatures are too high.

For DHW production in combination with buffer storage, also tank in tank systems are available, where a drinking-water suitable stainless steel small tank or tube system is inserted in the heating buffer storage.

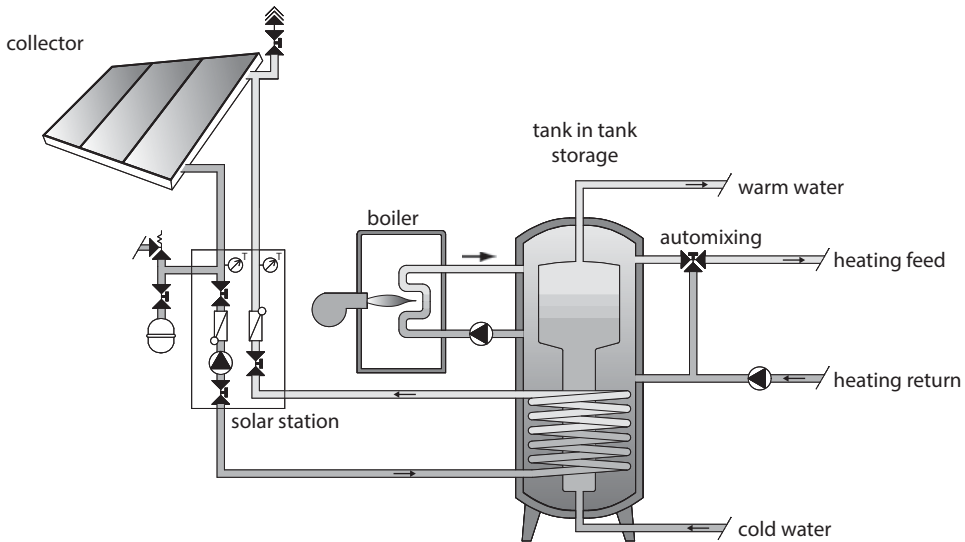


Figure 4.17 Combined drinking-water and hot water storage as tank-in-tank system.

A stratified transfer of the solar heat is also possible through internal risers with diaphragm flaps, which enable a release of warmed fluid of low density only at a height at which the surrounding store fluid also has a low density and thus a high temperature.

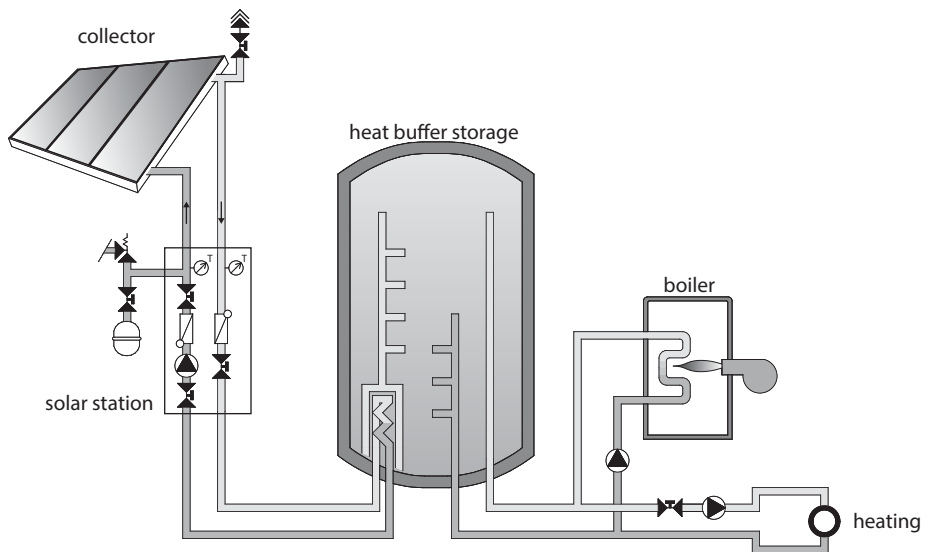


Figure 4.18 Internal stratified loading of a heat buffer store.

The riser with diaphragm flaps is heated through an internal heat exchanger by the solar collector warm-water supply flow. The heating circuit supply is taken from the upper storage area and stepped in again depending on the heating return temperature. Alternatively, the feed can be controlled externally via temperature-dependent three-way valves. Decisive for the temperature stratification is, apart from the feeding of the solar heat, above all the heat extraction or supply of the heating system, which is always connected with high mass flows. The heating supply is connected to the upper storage area; the colder heating return should, if possible, be led into the store by a step-loading installation, so that at low-temperature differences of the space heating and relatively high return temperatures, there is no direct feed into the lower storage area. Through the high heating water mass flows, the lower storage area without a step-loader is heated very quickly to the heating return temperature, i.e. at least 30°C, often to 40 - 70°C. A heating return temperature rise through the store is not recommended if a condensing boiler is used whose condensation potential depends largely on low return temperatures which fall below the condensation point.

The heating of drinking water by internal heat exchangers is no problem as long as the heat exchangers contact the whole store from bottom to top. Integrated small drinking-water stores, if placed only in the upper standby section, can destroy the temperature stratification if provision is not made for flow of the cooled storage water into the lower area, by special diverting pipes in the store.

As an alternative to internal high-grade steel heat exchangers, use is also made of external plate-type heat exchangers with such a high transfer power that cold water passing through can be warmed up to the required temperature. In order to achieve as even an outlet temperature as possible, the external heating water pump is revolution-adjusted as a function of the warm-water flow. In addition, a mixing valve provides an even outlet temperature and an upper temperature limit.

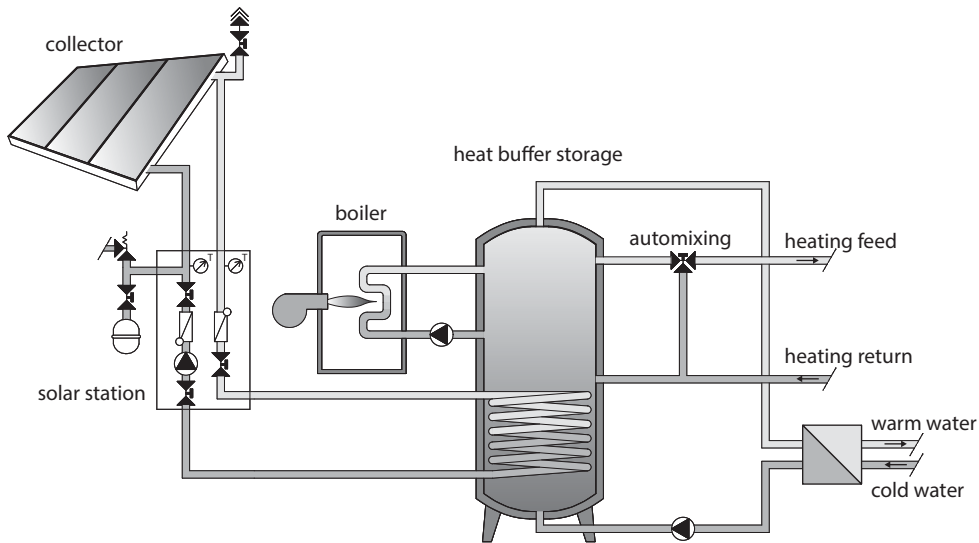


Figure 4.19 Solar plant for heating support and freshwater station.

The primary side of such a freshwater station is supplied by the buffer storage, the secondary side describes the consumer side. For standard hygienic requirements, a temperature level of 45°C on the secondary side is sufficient; for higher requirements 60°C can be used. The maximum achievable secondary volume flow at a given consumer temperature level, for example, 45°C, depends on the temperature level available on the buffer store side and increases with increasing buffer store temperature.

For high secondary flow rates of 7 - 14 litres per minute (typical flow rate for showers), the temperature fluctuations of current freshwater modules are below 1 K and thus without comfort problems. Especially at high primary side temperature levels and very low secondary mass flows, the temperature fluctuations can be higher. Minimum secondary volume flows of about 1 - 2 litres per minute can, in some cases, produce higher temperature fluctuations of several kelvin. The heat losses of freshwater modules are between 2% and 3% for insulated stations; between 1% and 10% additional heat losses can be caused if return temperatures from the freshwater station are high, i.e. the heat exchanger are underdimensioned.

Many theoretical investigations were done by the Institute of Thermodynamics of Stuttgart University. An analysis of the solar yield of five storage concepts for the heating support of a low-energy building with 15 m² of collector surface and 1050 litres of storage volume has shown that only the concept of an integrated drinking-water store in the upper storage area leads to a clear yield reduction of 13% compared with the best system. All other concepts – a two-store system, a system with external throughflow heating of drinking-water, a system with an integrated drinking-water store reaching from bottom to top, and a step-loading store with external heating of drinking-water – produce yields of approximately 300 kWh m⁻² a⁻¹ under German climatic conditions, deviating by less than 3% from each other. If no storage is used, the system yield drops by about 20%.

Piping and circulation losses

Piping calorific losses arise during temporary operation of the collector and warm-water withdrawal with heating up and cooling down periods, as well as from constant heat losses in circulation pipes. The energy required for heating up pipes and water content from a given temperature T_o to the desired temperature level T can easily be calculated via the mass and heat capacity of the pipe (m_p, c_p) and of the heat distribution medium (m_f, c_f):

$$Q_h = (m_p c_p + m_f c_f)(T - T_o) \quad (4.12)$$

The heat losses of a circulation pipe are calculated via the length-related U_l value ($W m^{-1} K^{-1}$) of the insulated pipe:

$$Q_c = U_l l (T - T_o) t_c \quad (4.13)$$

where t_c are the operating hours of the circulation pump and l is the length of the circulation pipe.

Example 4.3

Calculation of the heating up losses of a DN15 (15 mm exterior diameter) collector pipe filled with antifreeze, from 10 to 50°C.

Heat capacity of copper c_p :	0.39 kJ kg ⁻¹ K ⁻¹
Density of copper ρ_p :	8867 kg m ⁻³
Heat capacity of fluid c_f :	3.5 kJ kg ⁻¹ K ⁻¹
Density of fluid ρ_f :	1060 kg m ⁻³
Pipe length l :	30 m
Outer diameter d_{po} :	0.015 m
Inner diameter d_{pi} :	0.013 m

$$m_p = \rho_p V_p = \rho \frac{\pi(d_{po}^2 - d_{pi}^2)}{4} l = 11.7 \text{ kg} \quad m_f = \rho_f V_f = \rho \frac{\pi d_{pi}^2}{4} l = 4.22 \text{ kg}$$

$$Q_h = (m_p c_p + m_f c_f)(T - T_o) = \left(4.56 \frac{\text{kJ}}{\text{K}} + 14.77 \frac{\text{kJ}}{\text{K}} \right) (40 \text{ K}) = 773.2 \text{ kJ} = 215 \text{ Wh}$$

Example 4.4

Calculation of the circulation losses of a 30 m long, DN15 pipe with 50°C warm water, to room air at 20°C, with a daily working time of 10 hours. The pipe has 30 mm of insulation ($\lambda = 0.04 W m^{-1} K^{-1}$).

$$U_l = \frac{\pi}{\frac{1}{2\lambda} \ln \frac{d_o}{d_i} + \frac{1}{h_i d_i}} = \frac{\pi}{\frac{1}{2 \times 0.04} \ln \frac{0.075}{0.015} + \frac{1}{8 \times 0.075}} = 0.144 \frac{W}{mK}$$

$$Q_c = U_l l (T - T_o) t_c = 0.144 \frac{W}{mK} \times 30 \text{ m} \times 30 \text{ K} \times 10 \text{ h} = 1298 \text{ Wh}$$

The insulation of the connection tubes has to be temperature-resistant under standstill conditions of a solar thermal system. This excludes thermoplastic insulation materials made of polyethylene with melting temperatures around 100°C. With synthetic rubber, the temperature resistance depends on the exact material composition. Nitrile rubber hardens at temperatures above +105°C. Insulating materials based on EPDM or acrylate type of rubber can be used at much higher temperatures. Such materials are suitable for temperatures up to +150°C under constant load, and allow temperatures up to +175°C, for a short time.

Insulating materials for solar pipes also have to be weather proof and protected against birds. For open-cell material like mineral wool in outdoor areas, there is a risk of moisture penetration. Ten per cent moisture content results in a 2.5-fold increase of the thermal conductivity of a mineral fibre insulation. Synthetic rubber insulations on the other hand have a closed-cell structure and are not hygroscopic, so that the insulating effect is permanently secured.

So-called Twin Tubes combine copper pipe or corrugated pipes made of stainless steel with insulation and electric cable for the solar circuit sensor in one system.

Thermal expansion

Due to the high-temperature differences in a solar thermal system, the thermal expansion of the materials for piping has to be considered. The thermal expansion coefficient α shows the relative length variation of the material with the temperature.

$$\alpha = \frac{1}{L} \frac{dL}{dT} \quad (4.14)$$

If α is assumed as temperature independent, the length variation results:

$$L = L_0 \exp(\alpha \Delta T) \quad (4.15)$$

With a thermal expansion coefficient of copper of 16.5×10^{-6} per kelvin, a 10 m long pipe warming up by 50°C extends by 8.2 mm. If this extension has to be taken up by the pipe, tensions and cracks can occur. Expansion bends in the form of piping loop can absorb length variation, but often they are not installed because of the space they need. Expansion compensators can be constructed as corrugated pipe bellows or with two interleaved pipe sockets. The positioning of fixed and sliding points near to the compensator's end is important.

Auxiliary heating systems

For the auxiliary heating in solar thermal plants, either fossil- or biomass-fired boilers or electric heat pumps are used. For combination with heat pumps, several system configurations are possible: either the solar plant and the heat pump heat the same storage (drinking water or buffer storage), or the thermal energy of the solar plant is used as a source for the evaporator of the heat pump. As a result, a considerable increase of the seasonal performance factor of a heat pump is possible. The solar thermal plant can also partially substitute the operation of the heat pump for DHW production, which runs at lower efficiency due to the high-temperature levels required for hot water production. Furthermore, the solar heat increases the source temperature of the heat pump which results in higher coefficients of performance. In the German Renewable Energy Heating Law (EEWärmeG), the coefficient of performance of air/water heat pumps in new buildings has to be at least 3.5, in existing buildings 3.3, for brine/water heat pumps 4.0. Here the solar thermal plant can help to reach the challenging target value. An analysis by Lang (2009)

showed that with a solar thermal plant with 15 m² collector surface, the seasonal performance factor of a heat pump for DHW and thermal support for the heat pump evaporator rises by 5%. If the thermal yield of the solar plant is added completely to the heat pump system, a seasonal performance rise of about 35% is the result.

4.4 Large solar plants for heating drinking water with short-term stores

While the main market for small solar thermal systems still is the detached and semidetached residential housing sector, large solar plants offer clear cost-savings potential. Large solar plants for water heating with short time buffer storages are typically designed for a solar fraction of about 10 - 20% and avoid standstill at low demand periods. For simultaneous central space heating, four-pipe-networks can be used as they offer better control of the different temperature ranges for water and space heating. To reduce costs and heat losses, two-pipe-networks are very efficient especially in multi-storey apartment buildings. The supply temperatures are in this case between 55 and 65°C, return temperatures can be kept under 35°C with a good hydraulic balance, and as a result, a high overall efficiency is obtained. Detailed measurements of 10 solar-supported heat networks show that a solar fraction between 12% and 20% and a payback period of 10 - 25 years can be reached for a plant service life of at least 25 years (Fink et al., 2006).

Large central solar plants for heating drinking water, with collector surfaces of over 100 m², are particularly suitable for buildings with a continuously high warm-water requirement of over 5 m³ per day, such as hospitals, nursing homes, old people's homes and large housing estates. All large-scale installations have central buffer storage with storage contents > 5 m³; they differ mainly in the concept of the heat transfer to the drinking water.

Drinking water can be preheated on the throughflow principle via an external heat exchanger with conventional auxiliary heating in a drinking-water store.

Since cold water temperatures around 10°C are available at the consumption-side entry to the external heat exchanger, the return temperature into the buffer store is low in this concept, and the solar plant yield is up to 10% higher than with classical load storage concepts. Apart from control issues caused by strongly varying tapping rates, the circulation losses cannot be covered via the solar buffer store with this concept.

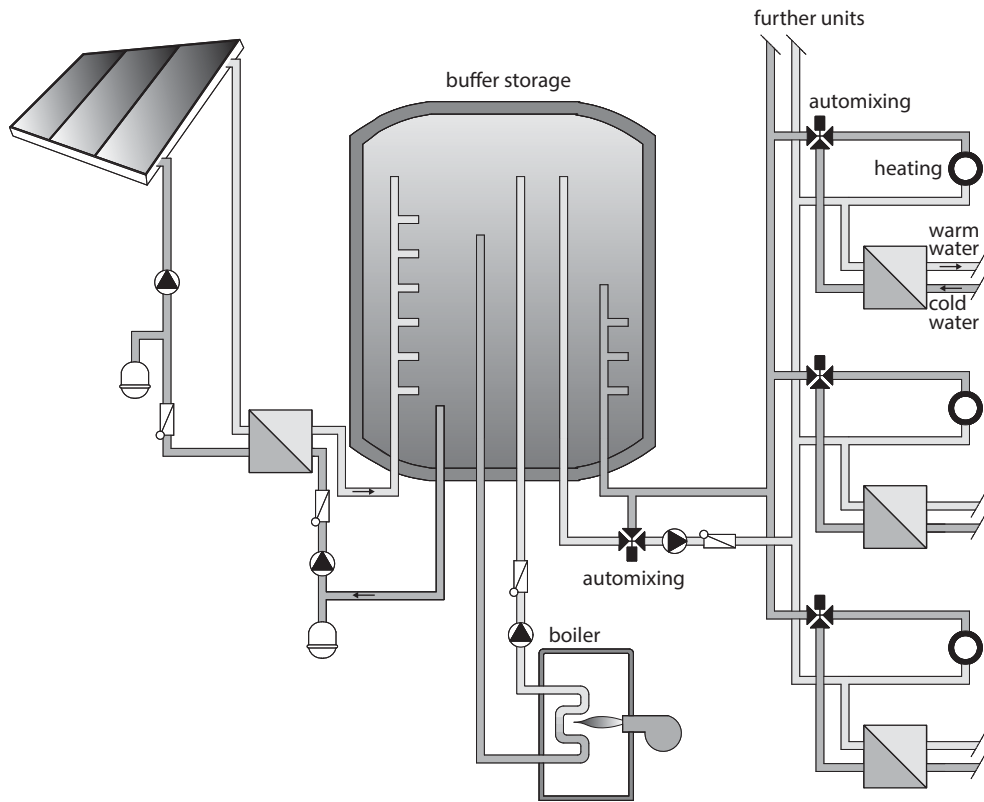


Figure 4.20 Schematic system overview for large solar plants at apartment buildings for heating support with freshwater stations.

The classical load storage concept is based on the transport of heat from the buffer store into a drinking-water store, as soon as the temperature level in the buffer is above the drinking-water temperature in the store. In retrofit systems, existing stores can be used as auxiliary heating stores and the additional drinking-water store for the solar plant is purely a preheating store. Here too, the lack of cover of circulation losses is unfavourable.

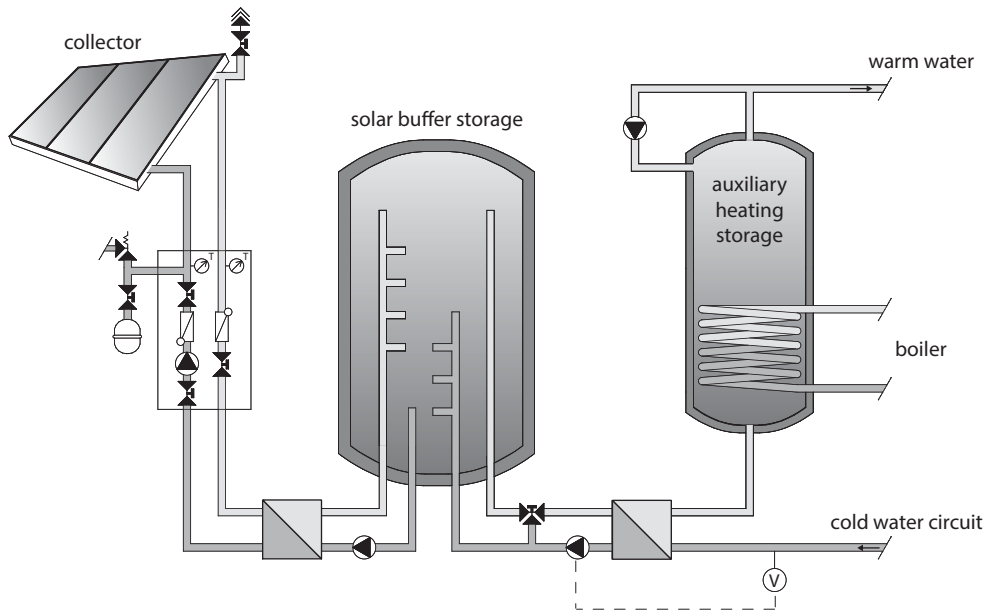


Figure 4.21 Drinking water preheating via external heat exchangers regulated by the measured drinking-water flow.

If a large drinking-water store is used for preheating and auxiliary heating, the circulation losses can also be covered by solar energy. The preheating area is either designed as a separate store or as a large vertical store combined with the auxiliary heating store (see Figure 4.21). In order to be able to through-heat the preheating area to avoid the formation of legionellae, heat can be taken out of the upper storage area.

Store loading control

The load regulation of the buffer store is done either by means of a temperature-difference control between the collector and lower storage area, as in small systems, or frequently by combined irradiance/temperature-difference control. Since in large solar plants the heat transfer to the buffer takes place via external heat exchangers and often large pipe lengths are available, not only the collector outlet temperature T_1 but also the collector circuit temperature T_2 at the heat exchanger entrance must be measured since it is relevant for the heat transfer to the buffer store.

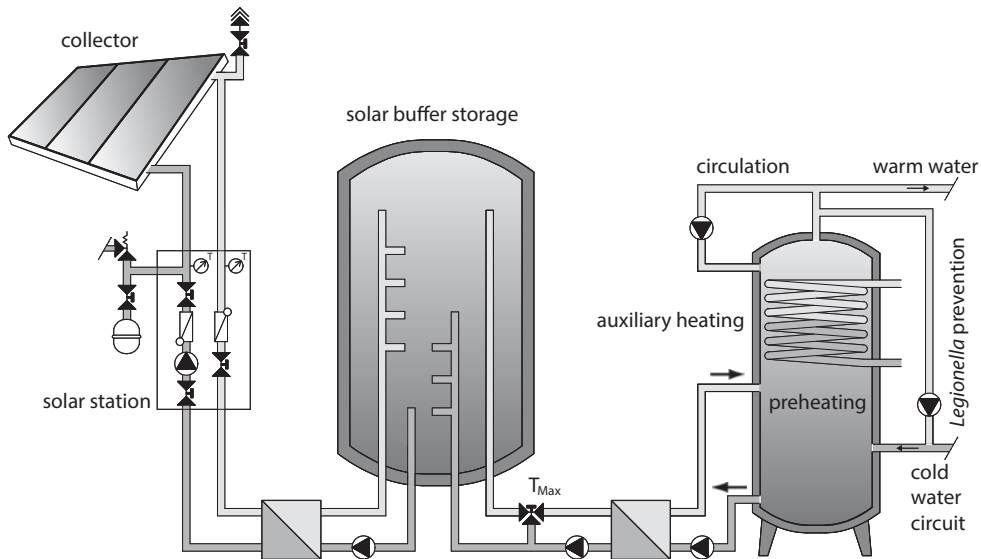


Figure 4.22 Transfer from a central buffer store to a drinking-water store.

To switch on the collector circuit pump, either a temperature signal of the collector or a radiation sensor is used. The control strategy reads:

- P_1 on, if $T_1 - T_3 > 8 \text{ K}$;
- P_2 on, if $T_2 - T_3 > 7 \text{ K}$ and if P_1 is on.

In pure temperature-difference control, the collector circuit pump P_1 is thus turned on if the collector temperature T_1 is about 8 K above the lower store temperature T_3 . With long external pipes, a minimum pump operating time of some minutes is necessary, so that the warmed collector liquid reaches the heat exchanger entrance T_2 . To be sure of avoiding freezing of the heat exchanger on the heating-circuit side, a bypass must be added before the heat exchanger, if external pipes are long. The storage pump P_2 does not operate till T_2 is also about 7 K above the lower store temperature T_3 and the collector pump P_1 is operating at the same time. Thus at high room temperatures, the storage pump will not be activated by a high temperature T_2 , at night, for example. As in small systems, the power-off temperature difference should clearly be lower than the switching-on temperature difference, at about 2 - 3 K.

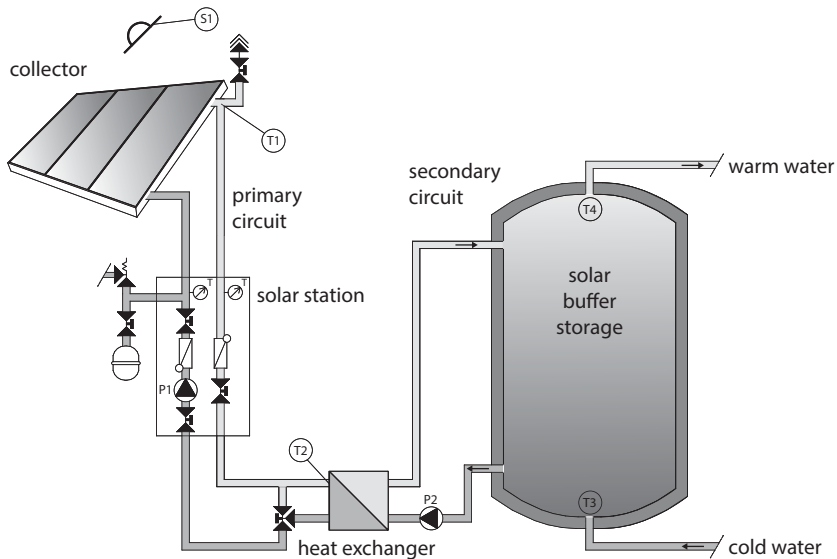


Figure 4.23 Sensors for storage loading control.

Alternatively to the collector temperature T_1 , a radiation sensor S_1 with a typical switching-on irradiance of 200 W m^{-2} can be used for switching the collector circuit pump. Again, the storage pump P_2 only operates if the heat exchanger temperature is around 7 K above the lower store temperature. If the storage pump does not switch on during the start-up minimum operating time of the collector pump, switch-off occurs when a lower irradiance value of about 150 W m^{-2} is reached. With a running storage pump, however, only the temperature difference between the heat exchanger entrance and the store temperature should be used as a power-off criterion, and not the irradiance signal.

When the maximum temperature T_4 of the store is attained, not only the storage pump but also the collector pump must be switched off. Thus, the collector goes into standstill, and the collector liquid evaporates and is pressed into the expansion tank, or blown off in a controlled way. Since the components near the collector (aeration valve, sensors, thermal insulation) are designed for high standstill temperatures, no problems arise. If, however, the collector pump is not switched off, high-temperature collector fluid circulates in the entire collector circuit, leading to a very strong thermal stress of all components.

Discharge control

The highest control demand exists for heating drinking water on the throughflow principle. Since the best heat transfer efficiencies result when there is the same flow on the buffer store side and the tapping side, the buffer store discharge pump P_3 must be directly revolution-adjusted as a function of the tapping flow rate. The regulation signal is received either by a direct volume-flow measurement, or indirectly by a dynamic temperature-difference measurement.

For example, a sensor immersed in the cold water intake (T_5) reacts far faster to changes in fluid temperature than an external pipe temperature sensor: when withdrawing water, this

different reaction rate leads to an initially high-temperature-difference signal, which can be used for volume flow measurement.

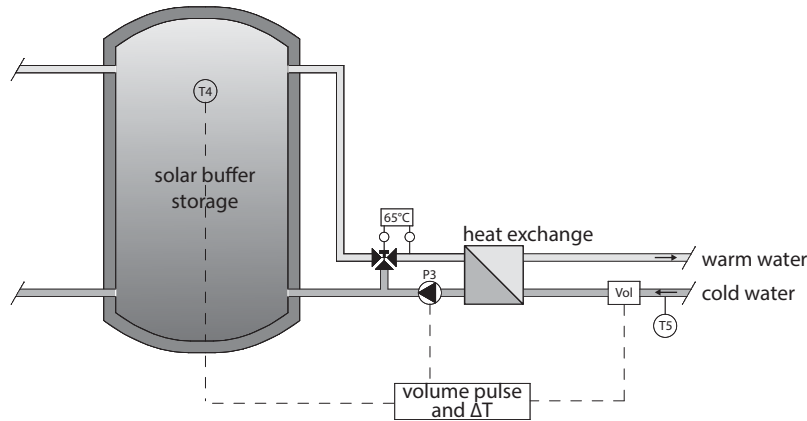


Figure 4.24 Discharge control of the buffer store for heating drinking water in a throughflow mode.

Transfer from the solar buffer store into a drinking-water store is simpler and more robust via an external heat exchanger regulated by a temperature-difference measurement between the upper buffer store temperature T_4 and the lower drinking water store temperature T_5 .

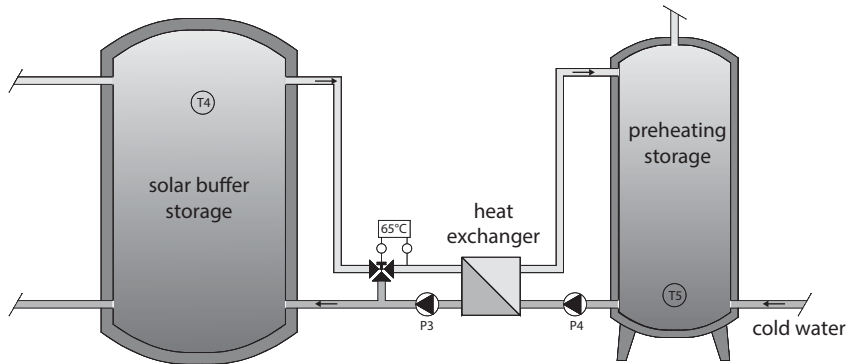


Figure 4.25 Discharge control for heat transfer to a drinking-water store.

The buffer store discharge pump P_3 and the drinking-water loading pump P_4 operate if the buffer store temperature T_4 is about 7 K above the lower temperature in the preheating store T_5 . At a temperature difference of less than 3 K, both pumps are switched off.

Design of large solar plants

In large solar plants, strict sizing of the collector surface to the minimum summer requirement has worked satisfactorily, to avoid standstill times of the collector field if possible, and to offer economical solutions at high solar yields. With such low collector surfaces, typical yearly solar fractions of around 20% are achieved, while in small systems with a higher specific system cost reduction per added square metre of collector surface, an oversizing in the summer with solar fractions of around 60% is better.

Consumption measurements in a number of building types show that during the summer, low demand periods have to be expected due to vacations; these periods cause a consumption reduction of up to 50% of the average value. Though consumption in a hospital is around 50 - 60 litres daily, per full occupancy person, in a vacation month only 35 litres are measured, related to the same occupancy number. Taking sizing for these low demand periods into account, design can proceed on the basis of the following values.

	Residential building	Hospital	Old people's home	Student dormitories
Specific warm-water consumption in litres per person and day	20 - 25	30 - 35	30 - 35	20 - 25

Table 4.3 Specific warm-water consumption in litres per full-occupancy person per day at 60°C, determined from summer vacation low demand periods (Croy, 2000).

On the basis of daily total drinking-water throughput thus calculated, the collector surface can be estimated. Annual yield calculations resulted in minimum solar heating costs of 0.13 Euro kWh⁻¹ at an utilisation of 70 litres of warm-water requirement per square metre of collector surface. Larger collector surfaces mean lower utilisation of the solar plant with higher standstill times, a smaller yield and higher system costs. In contrast, higher utilisation, i.e. undersizing of the collector surface, is no problem. The specific solar buffer volume per square metre of collector surface depends both on the uniformity of the tapping profile and on the degree of utilisation of the system. With an even tapping profile in multifamily houses, hospitals, old people's homes, etc. and a utilisation of 70 litres of warm water per square metre of collector surface, buffer sizing of 40 - 50 litres per m² of collector is sufficient. With larger collector surfaces, i.e. a lower utilisation of 40 - 50 l m⁻², the buffer volume should also increase to 60 - 70 l m⁻².

In buildings with greatly reduced consumption, e.g. at weekends (industrial premises), buffer sizing of 70 - 100 l m⁻² is recommended, depending on the degree of utilisation.

Degree of utilisation /litres per m ² of collector surface	Evenness of the tapping profile	System efficiency /%	Specific buffer volume /litres per m ² of collector surface
70	Even tapping profile of a multifamily house	47	40 - 50
40	Even tapping profile of a multifamily house	37	60 - 70
70	Tapping profile of an industrial premise with no weekend consumption	36 - 38	60 - 80

40	Tapping profile of an industrial premise with no weekend consumption	26 - 28	70 - 100

Table 4.4 System efficiency and buffer store volume depending on the degree of utilisation and the evenness of the tapping profile.

4.5 Solar district heating

Solar district heating means a central heating supply to large residential or commercial premises. A solar district heating network consists of a central heating installation with a buffer store and a boiler for auxiliary heating, a heat distribution network, building heat transfer stations for heating and warm water, plus the collector field, often distributed decentrally on the buildings. In addition to the short-term buffer store in the central heating installation, the summer excess heat can also be stored in a seasonal storage, and the solar fraction can be increased.

Energy can be supplied both for heating drinking water and for heating support via the solar district heating network. For an approximate design, both the heating and the DHW requirement must be known. The following reference values can be used for approximate sizing.

	Solar district heating with short-term store	Solar district heating with long-term store
Minimum plant size	From 30 to 40 housing units or 60 persons, respectively	From 100 to 150 housing units (70 m ² each)
Flat-plate collector surface	0.8 - 1.2 m ² per person	1.4 - 2.4 m ² per MWh yearly heating demand 0.14 - 0.2 m ² per m ² heated surface
Store volume (water equivalent)	0.05 - 0.1 m ³ per m ²	1.5 - 4 m ³ per MWh 1.4 - 2.1 m ³ per m ² flat-plate collector
Usable solar energy	350 - 500 kWh m ⁻² a ⁻¹	230 - 350 kWh m ⁻² a ⁻¹
Solar fraction	Drinking water: 50% Total: 10 - 20%	Total: 40 - 70%

Table 4.5 Approximate sizing of solar district heating systems under German climatic conditions (Hahne et al., 1998).

District heating systems differ in their types of heat distribution networks. If the DHW is produced centrally, the supply and return pipe for heating as well as a warm-water pipe and a circulation pipe for drinking-water supply are led to each building from the central heating installation (4 conductor network). Supply and return of the collector field need two further pipes for the central heating installation (4 + 2 conductor network). Due to high circulation losses of the drinking-water pipes, such a concept is recommendable only for small systems with 20 - 30 housing units.

With larger district heating systems, the use of a second building heat transfer station for drinking-water heating is preferable to reduce circulation losses. The heat distribution network is reduced to two conductors, which must maintain all year the necessary temperature of 60 - 70°C for heating drinking water. Furthermore, as above, two pipes are necessary for the collector field (2 + 2 conductor network). The heat exchanger of the warm-water transfer station can either be

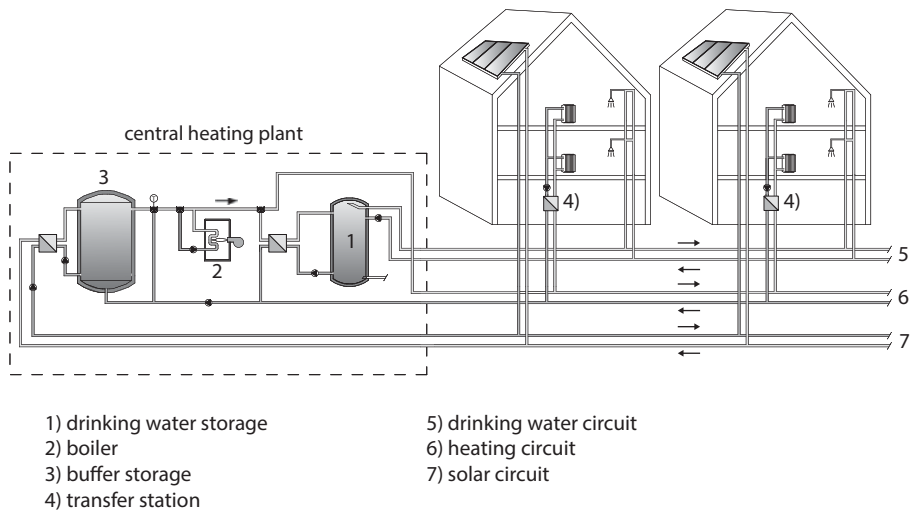


Figure 4.26 4 + 2 conductor network: supply and return heating with building heat transfer station, warm-water circuit for direct heating of drinking water, plus two pipes for the collector field.

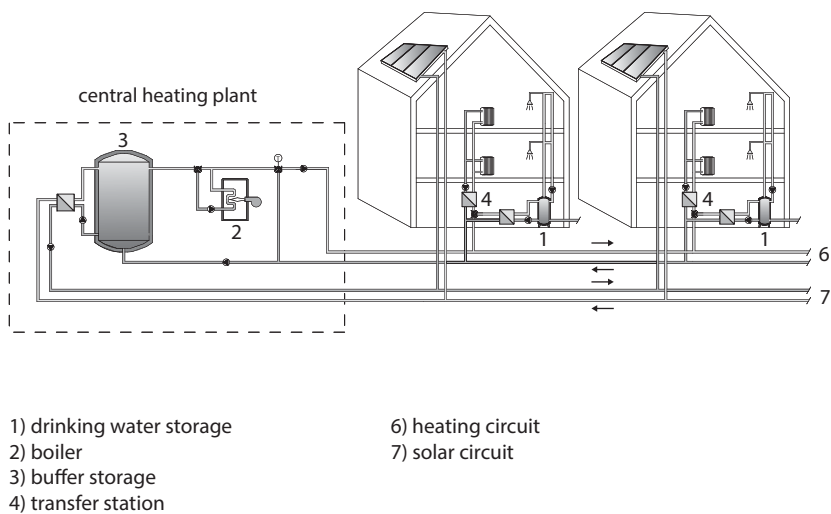


Figure 4.27 2 + 2 conductor network: supply and return pipes for the building heat transfer stations for heating and domestic hot water.

designed for throughflow heating of warm water (with small consumption in one- or two-family houses) or else for transfer to drinking-water stores.

A further pipe can be saved if only the hot collector supply (on the secondary circuit side) is led into the buffer store of the central heating installation, while the solar return pipe coincides with the heat return pipe. Via a building heat transfer station on the solar side, the heating or warm-water return can be warmed up directly in the solar transfer station (return temperature rise) and flows only then via the solar supply pipe to the central heating installation, if heating is required in the building. In this case the heat return pipe is not used. When heat is required without solar energy, hot buffer water is drawn via the heat supply pipe from the upper storage area, and fed without return rise by the solar plant via the heat return pipe back into the buffer. If no heating requirement exists and if the solar plant is supplying energy, the direction of flow is reversed in the heat return pipe, and stored water is drawn from the lower buffer store area to be heated in the solar transfer station, and returned via the solar supply pipe to the buffer.

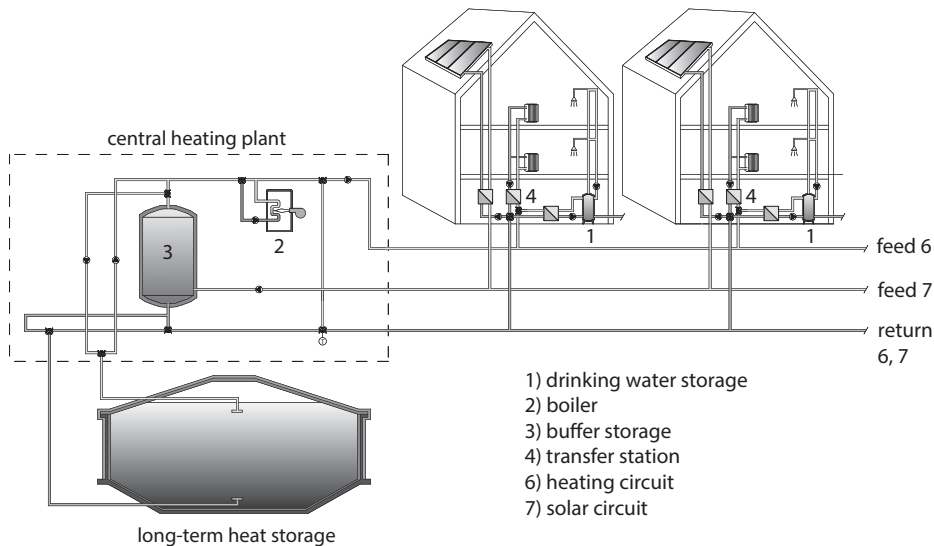


Figure 4.28 Three-conductor network with heating supply, secondary solar supply and a shared solar and heating return pipe. With a long-term heat store, solar heat can be stored seasonally.

Auxiliary heating systems can either be installed in series (see above) or parallel to the buffer storage. For a parallel integration a part of the buffer volume is constantly reheated, so that altogether higher heat losses result and the solar yield is reduced because of the higher storage

temperature. But a parallel integration is useful, if there are only few consumers in the network with high power fluctuations and the power requirement thus strongly fluctuates. Also a series connected boiler switches on and off more often, if the buffer storage temperature is only a few degree below the requested supply temperature.

Solar plants with several days storage can reach solar fractions up to 35%. In combination with a heat pump, the solar fraction can rise up to 40 - 50%.

Higher solar fractions can only be reached with seasonal storage. For seasonal storage with hot water, about 1.5 - 2.5 m³ per square metre collector surface are required, for storages with low heat capacity like gravel-water-storage, about 2.5 - 4 m³ per m². Geothermal storage needs around 8 - 10 m³ of soil per m² and aquifer storage 4 - 6 m³ per m² collector surface. Hot water storages with reinforced concrete must be covered with stainless steel metal sheets. Special concrete mixtures with very low water vapour permeability were first used at a large project with 2750 m³ storage volume in Hannover, Germany (2000) and are meanwhile state-of-the-art. Alternatively, glass fibre-reinforced plastic mixtures can be used in pressure-less storages with maximum temperatures of 95°C. Gravel-water storages with approximately 50% higher storage volume are sealed with plastic foils which limits the maximum temperature to 90°C.

Since the mid-1990s over 10 large solar plants with seasonal storage have been built in Germany with solar fractions between 40% and 50%. Project experience from Friedrichshafen with 4 050 m² collector surface and 12 000 m³ hot water storage as well as from Neckarsulm with 5670 m² plus a large earth heat exchanger storage with 63 360 m³ shows that the solar fraction for total heat requirement (drinking water and heating) can be rather high. In 2007, for 390 housing units, 33% coverage in Friedrichshafen and 45% for 300 housing units in Neckarsulm was measured (Bauer et al., 2009). Because of network distribution and storage losses, the specific collector heat production is mostly lower than in systems for DHW heating. The thermal energy generated by the solar system was between 218 and 238 kWh m⁻²a.

District heating networks with solar thermal plants have to be designed for low supply temperatures of about 55 - 65°C and as low as possible return temperatures about 30°C. A high-temperature difference reduces the required pump power consumption for distribution. A gliding supply temperature reduces the distribution loss in the network.

A short characterisation of the most important available storage concepts as well as approximate ratios of storage volume to collector surface at 50% solar cover are taken from Kübler and Fisch (1998) and Hahne et al. (1998).

Operational experiences and relevant standards

The service life of a solar thermal collector plant is at least 30 years, for electronic components around 20 years. The warranties are often limited at 10 - 12 years.

Prescriptions for safety, testing and efficiency are described in the European standards EN 12975, EN 12976 and ENV 12977. The collector test method according to EN 12975 includes a test of thermal performance (stationary and quasi dynamically) as well as evaluating the reliability and durability (internal pressure testing of the absorber, high-temperature resistance,

Store type	Hot water store	Gravel/water store	Earth heat exchanger store
Storage concept	Container store or earth-basin store	Earth-basin with gravel/water filling without a load-bearing cover construction	Heat transfer tubes in the ground, to a maximum depth of 150 m
Construction	Reinforced concrete, steel- or glass fibre-reinforced plastic, or a pit with a cover and lid, stainless steel- or sheeting cover	Cover with plastic sheeting	U-shaped coaxial plastic pipes with 1.5 – 3 m separation
Maximum/minimum volumes	Max. 100 000 m ³ , the largest store designed so far being 28 000 m ³	-	>100 000 m ³ due to high lateral heat losses
Heat insulation	15 - 30 cm at the lid and the store walls, and also under the store if the pressure can be withstood	As with water stores	Only in the covering layer 5 - 10 m from the surface
Store volume/flat-plate collector surface	1.5 - 2.5 m ³ per m ²	2.5 - 4 m ³ per m ²	8 - 10 m ³ per m ²
Approximate costs (Euro per m ³) at 20 000 m ³ storage volume	70 - 80	65 - 85	25
Other characteristics	Container store costly	With a gravel proportion of 60 - 70 vol. %, around 50% larger building volume as a water store	Easily constructed

Table 4.6 Characteristics of long-term heat stores.

fast external and internal temperature change, rainwater, mechanical load, standstill temperature determination). Storages are evaluated according to EN 12977-3. System tests for prefabricated systems are described in EN 12976, system tests for customer-specific systems in ENV 12977. The energetic evaluation of solar thermal systems is calculated on the basis of DIN 18599-8.

For large-scale installations, the Steam Boiler Code is furthermore relevant, which controls the security technology and acceptance tests of the plant.

Like conventional heating systems, solar plants require regular maintenance. The most frequently occurring defects in practice are problems during installation and initial operation (leakages in pipes, insufficient aeration) as well as unsatisfactory control strategies and hydraulic problems. The main defects are not caused by the components (14% of all defects), but are rather due to planning (44%) or workmanship mistakes (42%) (Keilholz, 2008).

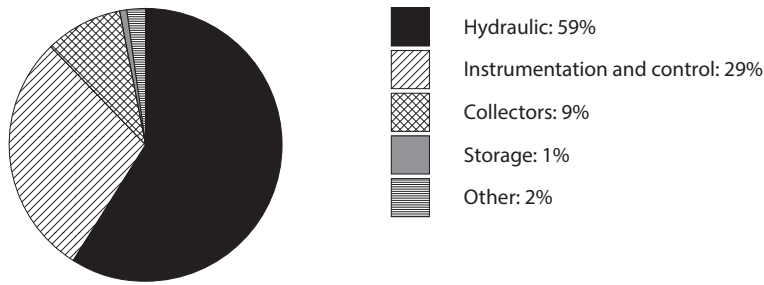


Figure 4.29 Relative frequency (in %) of defects in thermal solar plants.

The occurrence of leaky fittings, especially on the roof, can be reduced by the use of larger collector units. Problems with air in the system are reduced by having as high flow rates as possible, i.e. series connection of the collectors. Aeration valves must be fitted at the highest places, and air cushions avoided; alternatively, pumps with a high increase in pressure can be installed to force air through the entire cycle to the pump aeration valve.

In the event of damage, the construction company of the thermal solar plant is responsible. For that, the state-of-the-art engineering practice is decisive, which has the approval of the majority of independent experts. These can, but do not have to, be congruent with the available standards, which often take years to develop and be finalised.

For example, according to EN 12975, the stress test is performed with 1 kPa m⁻² vertical to the collector housing, while supporting structure standards such as the German DIN 1055 assumes considerably higher loads. Furthermore, it is known from experience that snow can exercise diagonal loads so that in case of a collector damage, the reference to EN 12975 is not enough to justify the damage (Keilholz, 2008).

4.6 Modelling of thermal collectors

Steady-state collector model

In the simplest static case without consideration of the thermal mass of the collector, the collector equation is given by

$$\dot{q}_{col} = \eta_0 (K_{\Theta b}(\Theta)G_b + K_{\Theta d}(\Theta)G_d) - a_1(T_{col,m} - T_0) - a_2(T_{col,m} - T_0)^2 \quad (4.16)$$

Often the incident angle modifiers for the beam and for the diffuse irradiation, which account for angular dependent reflection and absorption losses, are either neglected or used to first calculate the irradiance reduction by non-vertical incidence. Directly using the total solar irradiation on the tilted collector field of G_t , the equation simplifies to

$$\dot{q}_{col} = \eta_0 G_t - a_1(T_{col,m} - T_0) - a_2(T_{col,m} - T_0)^2 \quad (4.17)$$

and the overall collector efficiency is:

$$\eta_{col} = \eta_0 - a_1 \frac{(T_{col,m} - T_0)}{G_t} - a_2 \frac{(T_{col,m} - T_0)^2}{G_t} \quad (4.18)$$

If the collector mean temperature $T_{col,m}$ is approximated by the mean temperature of the collector inlet $T_{col,in}$ and collector outlet $T_{col,out}$

$$T_{col,m} = \frac{T_{col,in} + T_{col,out}}{2} \quad (4.19)$$

the useful heating power of the collector is given by

$$\dot{Q}_{col} = \eta_{col} G_t A = \dot{m}_{fluid} c_{fluid} (T_{col,out} - T_{col,in}) \quad (4.20)$$

The solution of Equation (4.18) and Equation (4.20) for the collector outlet temperature leads to a quadratic equation with the solutions

$$T_{col,out\ 1/2} = -\frac{p}{2} \pm \sqrt{\left(\frac{p}{2}\right)^2 - q} \quad (4.21)$$

and the coefficients:

$$p = \frac{2a_1}{a_2} + 2T_{col,in} - 4T_0 + \frac{2\dot{m}_{fluid} c_{fluid}}{A \cdot a_2}$$

$$q = \left(\frac{2a_1}{a_2} - 4T_0 + T_{col,in} - \frac{2\dot{m}_{fluid} c_{fluid}}{A \cdot a_2} \right) T_{col,in} - 4 \cdot \left(\frac{G_t \eta_0}{a_2} + \frac{a_1 T_0}{a_2} + T_0^2 \right)$$

This simple model requires only the efficiency parameters of the collector optical efficiency η_0 , linear heat loss coefficient a_1 , and quadratic heat loss coefficient a_2 together with the reference gross aperture or absorber area. These values are provided on the product data sheet of the collectors.

Simple one node dynamic collector model

With the new collector test procedure of EN 12975-2:2001 – Part 2, the specific thermal capacity of the collectors is determined in the form of an area-related effective thermal capacity C_{eff} given in $\text{kJ m}^{-2} \text{K}^{-1}$. This thermal mass is considered in the collector equation in the general form:

$$\dot{q}_{col} = \eta_0 (K_{\Theta b}(\Theta) \cdot G_b + K_{\Theta d}(\Theta) G_d) - a_1 (T_{col,m} - T_0) - a_2 (T_{col,m} - T_0)^2 - C_{eff} \frac{dT_{col,m}}{dt} \quad (4.22)$$

If we again assume that the incident angle modifiers $K_{\Theta b}(\Theta)$ and $K_{\Theta d}(\Theta)$ are considered externally we get

$$\dot{q}_{col} = \eta_0 G_t - a_1 (T_{col,m} - T_0) - a_2 (T_{col,m} - T_0)^2 - C_{eff} \frac{dT_{col,m}}{dt} \quad (4.23)$$

If we further assume the collector mean temperature $T_{col,m}$ to be approximately equal to the mean temperature of the collector inlet and outlet temperature

$$T_{col,m}(t) = \frac{T_{col,in} + T_{col,out}}{2} \quad (4.24)$$

the collector equation can be written as

$$\begin{aligned} \frac{\dot{m}_{fluid} c_{fluid}}{A} (T_{col,out}(t) - T_{col,in}) &= \eta_0 G_t - a_1 (T_{col,m}(t-1) - T_0) \\ &\quad - a_2 (T_{col,m}(t-1) - T_0)^2 - C_{eff} \frac{dT_{col,m}(t)}{dt} \\ \frac{2\dot{m}_{fluid} c_{fluid}}{A} (T_{col,m}(t) - T_{col,in}) &= \eta_0 G_t - a_1 (T_{col,m}(t-1) - T_0) \\ &\quad - a_2 (T_{col,m}(t-1) - T_0)^2 - C_{eff} \frac{dT_{col,m}(t)}{dt} \\ C_{eff} \frac{dT_{col,m}(t)}{dt} &= \eta_0 G_t - a_1 (T_{col,m}(t-1) - T_0) - a_2 (T_{col,m}(t-1) - T_0)^2 \\ &\quad + \frac{2\dot{m}_{fluid} c_{fluid} T_{col,in}}{A} - \frac{2\dot{m}_{fluid} c_{fluid}}{A} T_{col,m}(t) \end{aligned} \quad (4.25)$$

With the simplifications

$$\begin{aligned} a &= \eta_0 G_t - a_1 (T_{col,m}(t-1) - T_0) - a_2 (T_{col,m}(t-1) - T_0)^2 + \frac{2\dot{m}_{fluid} c_{fluid} T_{col,in}}{A} \\ b &= \frac{2\dot{m}_{fluid} c_{fluid}}{A} \end{aligned} \quad (4.26)$$

this differential equation has the form:

$$C_{eff} \frac{dT_{col,m}(t)}{dt} = a - b T_{col,m}(t) \quad (4.27)$$

Considering the initial condition: $T_{col,m}(0) = T_{col,m,0}$ the solution of this equation is:

$$T_{col,m}(t) = \frac{e^{-\frac{bt}{C_{eff}}} \left(b T_{col,m,0} + a e^{\frac{bt}{C_{eff}}} - a \right)}{b} \quad (4.28)$$

The collector outlet temperature is then calculated from Equation (4.24).

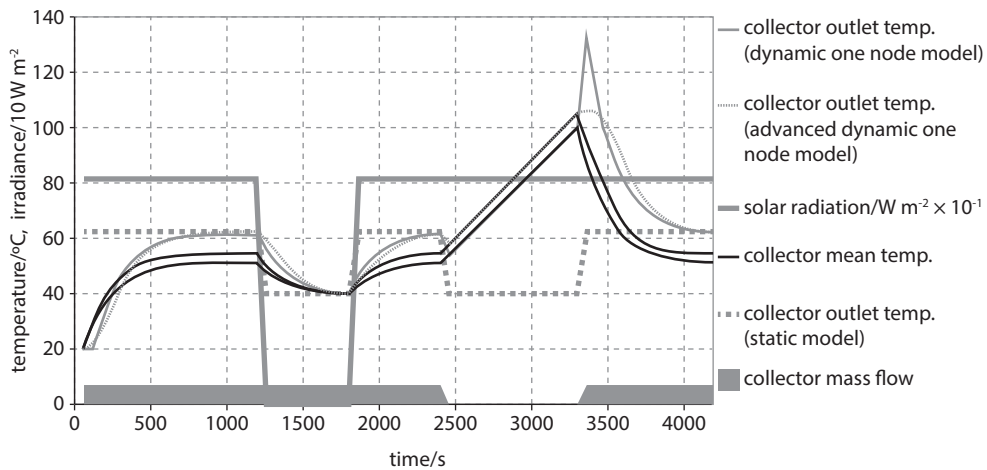


Figure 4.30 Comparison of the performance of a steady state and a one node dynamic collector model.

The difference between the steady-state model and the dynamic one node model is clearly visible. The static model results in an immediate temperature increase to the steady-state outlet temperature as soon as solar irradiation is on the collector plane and drops immediately down to the collector inlet temperature as soon the solar radiation is off. For the dynamic one node model the temperature increases and decreases much slower, but finally reaches the same outlet temperature as the static model. However, if after a period without mass flow in the collector, the collector pump is switched on, a large peak in the outlet temperature of the simple dynamic one node collector model is visible. This temperature peak increases significantly above the collector mean temperature before the pump is switched on. The reason for this unrealistic behaviour is that during the standstill phase the collector temperature increases quite fast to a high value. When the collector pump is switched on, the collector outlet temperature is then simply calculated from the cold collector inlet temperature and the collector mean temperature. Due to a large temperature difference between collector inlet temperature and collector mean temperature, a very high outlet temperature is calculated from Equation (4.24). To overcome this problem, an improved model was developed which simply divides the collector model internally into a number of simple finite segments with the same absorber area (whole absorber area divided by number of segments). The calculations are then done for each of the segments where the segment inlet temperature is set equal to the segment outlet temperature of the segment before. With this advanced dynamic one node model, the peak after a phase without collector mass flow is avoided and a realistic behaviour is simulated.

To be able to determine analytically the available energy at a given fluid inlet temperature or mean fluid temperature, the temperature distribution on the absorber sheet metal must first be calculated as the solution of a thermal conduction problem. Subsequently the local fluid temperature is calculated by the heat transfer to the fluid. At a given mass flow, the entire rise in temperature and available energy can then be obtained by integration over the flow length, and represented as a function of the fluid inlet temperature.

Temperature distribution of the absorber

To determine the conversion of the absorbed irradiance to available power, the heat flow caused by the temperature gradient on the sheet metal towards the fluid tube must be calculated.

The temperature distribution on the absorber sheet metal transverse to the direction of flow is calculated by the thermal conduction characteristics of the sheet metal and by the convective heat transfer coefficient between sheet metal and fluid. The calculation of the temperature distribution takes place via the solution of the energy balance equation for a point on the absorber sheet metal. The absorbed irradiance $G\tau\alpha$ is transferred, on the one hand, into calorific losses to the environment, and on the other hand, into a heat flux from the absorber sheet metal towards the fluid tube. Following Fourier's law of thermal conduction, this heat flow $\dot{Q}|_x$ at point x on the sheet metal is proportional to the temperature gradient and to the cross-section area, which is calculated from the plate thickness δ and the unit length $l = 1$ in the direction of flow. Due to the small absorber plate thickness, the thermal conduction problem is regarded as only one-dimensional, i.e. along the sheet metal.

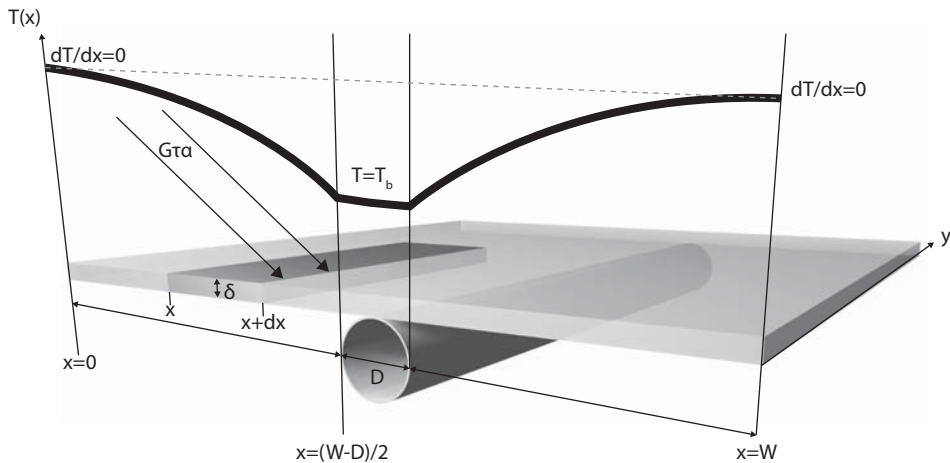


Figure 4.31 Geometry of the absorber sheet metal with fluid pipe and surface element for energy balance.

The energy surplus between the absorbed radiation and heat losses leads to a change in the temperature gradient within a surface element dx and is led off laterally to the fluid pipes over the cross-sectional area $l\delta$. Using a Taylor series development, the difference of the heat flows results in:

$$\begin{aligned}\dot{Q}|_x - \dot{Q}|_{x+dx} &= \left(-\lambda(l\delta) \frac{dT}{dx} \right) \Big|_{x_0} - \left(-\lambda(l\delta) \frac{dT}{dx} \right) \Big|_{x_0+\Delta x} \\ &= -\lambda(l\delta) \left(\frac{dT}{dx} \Big|_{x_0} - \left(\frac{dT}{dx} \Big|_{x_0} + \frac{d^2T}{dx^2} \Big|_{x_0} dx + \dots \right) \right) = \lambda(l\delta) \frac{d^2T}{dx^2} \Big|_{x_0} dx\end{aligned}\quad (4.29)$$

The power balance for the surface element $l dx$ (with unitary length $l = 1$ in the direction of flow y) of the absorber sheet metal is thus:

$$\begin{aligned} G\tau\alpha(l dx) - U_t(T - T_o)(l dx) + \lambda(l\delta)\frac{d^2T}{dx^2}dx &= 0 \\ \Leftrightarrow G\tau\alpha - U_t(T - T_o) + \lambda\delta\frac{d^2T}{dx^2} &= 0 \quad \left[\frac{W}{m^2} \right] \end{aligned} \quad (4.30)$$

From this a second-order differential equation for the temperature field is obtained.

$$\frac{d^2T}{dx^2} = \frac{U_t}{\lambda\delta} \left(T - T_o - \frac{G\tau\alpha}{U_t} \right) \quad (4.31)$$

As the zero point of the x -axis, the midpoint between two fluid pipes is chosen, at which the temperature is maximum and the temperature gradient is zero (first boundary condition):

$$\left. \frac{dT}{dx} \right|_{x=0} = 0 \quad (4.32)$$

As the second boundary condition, a temperature of the sheet metal over the fluid pipe is assumed. The geometry of the fluid pipe with lateral sheet metal corresponds to a classical cooling fin in which the sheet metal, in contrast to the collector, serves not to absorb heat but to lead it off. The temperature over the fluid pipe is, as with a cooling fin problem, characterised with base temperature T_b . This temperature can be eliminated in a second step, after determination of the useful energy led to the pipe and the convective heat transfer to the fluid.

$$T|_{x=(W-D)/2} = T_b \quad (4.33)$$

The differential equation can be solved by substitution of the variables

$$\Psi = T - T_o - G\tau\alpha / U_t$$

and

$$m = \sqrt{U_t / (\lambda\delta)} \quad [m^{-1}]$$

by using the constants

$$\frac{d^2\Psi}{dx^2} - m^2\Psi = 0 \quad (4.34)$$

with the boundary conditions

$$\left. \frac{d\Psi}{dx} \right|_{x=0} = 0 \quad (4.35)$$

$$\Psi|_{x=(W-D)/2} = T_b - T_o - \frac{G\tau\alpha}{U_t} \quad (4.36)$$

and the general solution

$$\Psi = C_1 \sinh(mx) + C_2 \cosh(mx) \quad (4.37)$$

From the boundary conditions the constants C_1 and C_2 are obtained.

$$\left. \frac{d\Psi}{dx} \right|_{x=0} = C_1 m \underbrace{\cosh(0)}_1 + C_2 m \underbrace{\sinh(0)}_0 = 0 \Rightarrow C_1 = 0 \quad (4.38)$$

$$\begin{aligned} \Psi|_{x=(W-D)/2} &= \underbrace{C_1}_0 \sinh(m(W-D)/2) + C_2 \cosh(m(W-D)/2) = T_b - T_o - \frac{G\tau\alpha}{U_t} \\ \Rightarrow C_2 &= \frac{T_b - T_o - \frac{G\tau\alpha}{U_t}}{\cosh(m(W-D)/2)} \end{aligned} \quad (4.39)$$

After back substitution, the result for the temperature on the absorber sheet metal is

$$T(x) = T_o + \frac{G\tau\alpha}{U_t} + \left(T_b - T_o - \frac{G\tau\alpha}{U_t} \right) \frac{\cosh(mx)}{\cosh(m(W-D)/2)} \quad (4.40)$$

From the temperature distribution, and using Fourier's law, the amount of heat at the point $x = (W-D)/2$ led from both sheet metal sides to the fluid pipe can now be determined. As the temperature distribution right and left of the fluid pipe is symmetric, the amount of heat calculated on one side can simply be doubled. The heat flow is characterised by .

$$\begin{aligned} \dot{Q}_{fin} &= 2 \times \left(-\lambda(l\delta) \left. \frac{dT}{dx} \right|_{x=(W-D)/2} \right) \\ &= -2\lambda(l\delta) \left(T_b - T_o - \frac{G\tau\alpha}{U_t} \right) \frac{\sinh(m(W-D)/2)}{\cosh(m(W-D)/2)} \times m \\ &= 2 \frac{\lambda(l\delta)m}{U_t} (G\tau\alpha - U_t(T_b - T_o)) \tanh(m(W-D)/2) \\ &= (W-D)l (G\tau\alpha - U_t(T_b - T_o)) \frac{\tanh(m(W-D)/2)}{m(W-D)/2} \end{aligned} \quad (4.41)$$

with

$$\frac{\lambda\delta m}{U_t} = \frac{1}{m}$$

used in accordance with the definition.

The fin efficiency F shows the ratio of the actual heat flow $\dot{Q}_{fin}$, based on Equation (4.41), to the ideal heat flow which would result if all the sheet metal were at the lower base temperature with correspondingly low heat losses.

$$F = \frac{\tanh(m(W-D)/2)}{m(W-D)/2} \quad (4.42)$$

$$\dot{Q}_{fin} = (W-D)F(G\tau\alpha - U_t(T_b - T_o)) \quad [W] \quad (4.43)$$

In addition to the heat flow led to the pipe from both sides, the irradiance directly absorbed via the external diameter D of the pipe must also be taken into account.

$$\dot{Q}_{tube} = DI(G\tau\alpha - U_t(T_b - T_o)) \quad [W] \quad (4.44)$$

The sum of these two heat fluxes is transferred convectively to the fluid and finally produces the useful power per pipe in the direction of flow.

$$\dot{Q}_{fin} + \dot{Q}_{tube} = ((W-D)F + D)(G\tau\alpha - U_t(T_b - T_o)) = \dot{Q}_{n(N=1)} \quad [W] \quad (4.45)$$

The thermal resistance between the sheet metal over the fluid tube with a base temperature T_b and the fluid consists of a convective part between the tube sheet and fluid, with thermal resistance $1/h_f$ for a tubing inside diameter of D_i and of the contact resistance between the sheet metal and tube. The convective heat transfer coefficient h_f is about $100 \text{ W m}^{-2} \text{ K}^{-1}$ with a laminar flow and reaches values of $300 - 1000 \text{ W m}^{-2} \text{ K}^{-1}$ with a turbulent flow. The effective contact conductivity is determined from the conductivity of the contact material, the width of the contact b_{con} and the thickness d_{con} , and is usually negligible in today's absorber constructions:

$$\lambda_{con, eff} = \frac{\lambda_{con} b_{con}}{d_{con}} \left[\frac{W}{mK} \right]$$

$$\dot{Q}_{u(N=1)} = \frac{1}{\left(\frac{1}{h_f \pi D_i l} + \frac{1}{\lambda_{con, eff} l} \right)} (T_b - T_f) \quad [W] \quad (4.46)$$

The base temperature can now be eliminated by equating the available power from Equation (4.45). The available power $[W]$ of an absorber pipe (number $N = 1$) for unit flow length l can be represented as a function of the local fluid temperature.

$$\dot{Q}_{u(N=1)} = \frac{1/U_t}{\frac{1}{h_f \pi D_i l} + \frac{1}{\lambda_{con, eff} l} + \frac{1}{((W-D)F + D)U_t l}} (G\tau\alpha - U_t(T_f - T_o)) \quad (4.47)$$

Collector efficiency factor F'

The collector efficiency factor F' introduced in the reference textbook on solar engineering from Duffie and Beckmann is the ratio of thermal resistances from Equation (4.47), which arises as a result of normalisation to the surface area of an absorber strip of width W and unit length $l = 1$:

$$F' = \frac{1/U_t}{W \left(\frac{1}{h_f \pi D_i} + \frac{1}{\lambda_{con,eff}} + \frac{1}{((W-D)F + D)U_t} \right)} \quad (4.48)$$

The efficiency factor indicates the ratio of the actual available power to the higher available power, which would result for an absorber sheet metal at the low fluid temperature (with correspondingly smaller calorific losses).

$$\dot{Q}_{u(N=1)} = W F' (G \tau \alpha - U_t (T_f - T_o)) \quad [W] \quad (4.49)$$

Heat dissipation factor F_R

The available energy calculated so far describes the heat supply from the absorber sheet metal to the fluid at a point y on the collector. This heat supply leads to a local rise in temperature of the fluid, which depends on the mass flow through the fluid pipe.

The calculation of the rise in temperature over the complete collector length then enables the calculation of the available useful power from the collector. For an absorber pipe, the following energy balance results for a total mass flow through the collector $\dot{m}$, i.e. a mass flow per pipe of $\dot{m}/N$:

$$\left(\frac{\dot{m}}{N} \right) c_p \frac{dT_f}{dy} - W F' (G \tau \alpha - U_t (T_f - T_o)) = 0 \quad (4.50)$$

with the boundary condition

$$T_f|_{y=0} = T_{f,in} \quad (4.51)$$

The differential equation is solved by substitution

$$\Psi = G \tau \alpha - U_t (T_f - T_o) \quad (4.52)$$

and separation of the variables:

$$\frac{d\Psi}{\Psi} = -U_t \frac{N W F'}{\dot{m} c_p} dy \quad (4.53)$$

$$\Psi = C_1 \exp \left(-\frac{U_t N W F'}{\dot{m} c_p} y \right) \quad (4.54)$$

The boundary condition results in

$$\Psi|_{y=0} = C_1 = G\tau\alpha - U_t(T_{f,in} - T_o) \quad (4.55)$$

and thus for the fluid temperature at any point y in the direction of flow:

$$T_f(y) = T_o + \frac{G\pi\alpha}{U_t} + \left((T_{f,in} - T_o) - \frac{G\pi\alpha}{U_t} \right) \exp\left(-\frac{U_t NWF'}{\dot{m}c_p} y \right) \quad (4.56)$$

To determine the fluid output temperature, the collector length L is used for y . The product of the number of tubes N , absorber strip width W and collector length L thereby corresponds to the collector surface A .

$$T_{f,out} = T_o + \frac{G\pi\alpha}{U_t} + \left((T_{f,in} - T_o) - \frac{G\pi\alpha}{U_t} \right) \exp\left(-\frac{U_t F'A}{\dot{m}c_p} \right) \quad (4.57)$$

The available power of the collector can thus be represented analytically as a function of the fluid inlet temperature, the ambient temperature and the irradiance.

$$\begin{aligned} \dot{Q} &= \dot{m}c_p(T_{f,out} - T_{f,in}) \\ &= \dot{m}c_p \left(T_o + \frac{G\pi\alpha}{U_t} + \left((T_{f,in} - T_o) - \frac{G\pi\alpha}{U_t} \right) \exp\left(-\frac{U_t F'A}{\dot{m}c_p} \right) - T_{f,in} \right) \\ &= \left(\frac{\dot{m}c_p}{U_t} + \dot{m}c_p \frac{\left((T_{f,in} - T_o) - \frac{G\pi\alpha}{U_t} \right) \exp\left(-\frac{U_t F'A}{\dot{m}c_p} \right)}{G\pi\alpha - U_t(T_{f,in} - T_o)} \right) G\pi\alpha - U_t(T_{f,in} - T_o) \\ &= \left(\frac{\dot{m}c_p}{U_t} \left(1 - \exp\left(-\frac{U_t F'A}{\dot{m}c_p} \right) \right) \right) G\pi\alpha - U_t(T_{f,in} - T_o) \end{aligned} \quad (4.58)$$

The first term of the equation, normalised to the collector surface A , is defined after Duffie and Beckmann as the heat dissipation factor F_R . It indicates the ratio of the actual available power to the attainable available power, if the complete absorber were at the cold fluid inlet temperature.

$$F_R = \frac{\dot{m}c_p}{AU_t} \left(1 - \exp\left(-\frac{U_t F'A}{\dot{m}c_p} \right) \right) \quad (4.59)$$

The heat dissipation factor, which depends on the mass flow and geometrically on the collector efficiency factor, leads to a simple available-power equation:

$$\dot{Q}_u = AF_R(G\tau\alpha - U_t(T_{f,in} - T_o)) \quad (4.60)$$

The thermal efficiency is determined by the ratio of available power per square metre collector surface and irradiance.

$$\eta = \frac{\dot{Q}_u / A}{G} \quad (4.61)$$

The mean fluid temperature of the collector is obtained by integrating the fluid temperature using Equation (4.56) over the collector length.

$$\bar{T}_f = \frac{1}{L} \int_0^L T_f(y) dy = T_{f,in} + \frac{\dot{Q}_u \left(1 - \frac{F_R}{F'}\right)}{AF_R U_t} \quad (4.62)$$

The mean absorber temperature is obtained by equating the available power equation as a function of the input temperature and as a function of the mean absorber temperature.

$$\bar{T}_a = T_{f,in} + \frac{\dot{Q}_u / A}{U_t F_R} (1 - F_R) \quad (4.63)$$

Example 4.5

Calculation of the available power and temperatures of a flat-plate collector at $G = 800 \text{ W m}^{-2}$ irradiance, ambient temperature $T_o = 10^\circ\text{C}$ and an inlet temperature into the collector $T_{f,in}$ from the lower part of the storage tank at 30°C . To represent the influence of the mass flow on the available power and the temperature conditions, calculations are done for a low-flow system with $\dot{m} = 10 \text{ kg m}^{-2} \text{ h}^{-1}$ and also for a standard system with $\dot{m} = 50 \text{ kg m}^{-2} \text{ h}^{-1}$.

The optical efficiency is $\eta_o = \tau\alpha = 0.9 \times 0.9 = 0.81$ and the entire calorific loss U_t is $4 \text{ W m}^{-2} \text{ K}^{-1}$. The other data of the collector are as follows:

Width of the absorber strip W	15 cm
Length of the absorber strip L	2.5 m
External pipe diameter D (DN8, 1 mm wall strength)	8 mm
Heat conductivity of the absorber sheet metal λ_{copper}	$385 \text{ W m}^{-1} \text{ K}^{-1}$
Sheet thickness δ	0.5 mm
Contact resistance $1/\lambda_{\text{con,eff}}$	0
Heat transfer coefficient h_f	$1000 \text{ W m}^{-2} \text{ K}^{-1}$

Calculations:

Fin efficiency

$F = 0.966$ with

$$m = \sqrt{\frac{U_t}{\lambda \delta}} = 4.56 \text{ [m}^{-1}\text{]}$$

Efficiency factor

$$F' = \frac{1/4}{0.15(0.053 + 1.722)} = 0.94$$

Heat dissipation factor F_R

$$F_R \Big|_{\dot{m}=10 \frac{\text{kg}}{\text{m}^2 \text{h}}} = 2.9(1 - \exp(-0.94 / 2.9)) = 0.8$$

$$F_R \Big|_{\dot{m}=50 \frac{\text{kg}}{\text{m}^2 \text{h}}} = 14.5(1 - \exp(-0.94 / 14.5)) = 0.91$$

The heat dissipation factor F_R is mass-flow dependent.

Available power:

$$\frac{\dot{Q}_u}{A} \Big|_{\dot{m}=10 \frac{\text{kg}}{\text{m}^2 \text{h}}} = 0.8(800 \times 0.81 - 4(30 - 10)) = 454.4 \frac{\text{W}}{\text{m}^2}$$

$$\frac{\dot{Q}_u}{A} \Big|_{\dot{m}=50 \frac{\text{kg}}{\text{m}^2 \text{h}}} = 516.9 \frac{\text{W}}{\text{m}^2}$$

The available power and efficiency improve with rising mass flow. At typical flow conditions of collectors between 10 and 50 $\text{kg m}^{-2} \text{h}^{-1}$, the efficiency varies by 12%.

Efficiency:

$$\eta \Big|_{\dot{m}=10 \frac{\text{kg}}{\text{m}^2 \text{h}}} = 0.57$$

$$\eta \Big|_{\dot{m}=50 \frac{\text{kg}}{\text{m}^2 \text{h}}} = 0.64$$

Outlet temperature: $T_{f,\text{in}} = 69^\circ\text{C}$ (39°C)

The advantage of the smaller mass flows shows up particularly in the outlet temperature. During a single flow-through with the low-flow system, a rise in temperature of 39 K is achieved, with the higher throughflows only 9 K.

Mean fluid temperature: $\bar{T}_f = 51.1^\circ\text{C}$ (31.5°C)

Mean absorber temperature: $\bar{T}_a = 58.4^\circ\text{C}$ (42.8°C)

Between the mean absorber temperature and mean fluid temperature there is a difference of 7.3 or 8.3 K, respectively.

Heat losses of thermal collectors

The thermal collectors available on the market are today produced almost exclusively with a single transparent cover, or even no cover. The heat transfer coefficient between the absorber and ambient air via the collector front (U_f) cannot be set as constant, since the temperatures of the absorber cover a large range. For each given absorber temperature, therefore, the convective and radiant heat transfer coefficients should be calculated iteratively and the U_f value of the front determined. The heat transfer coefficients through the insulated collector rear U_b and side panels U_s can, however, be regarded as constant and calculated from the layer thickness s [m], heat

conductivity λ [$\text{W m}^{-1} \text{K}^{-1}$] of the insulating material, and from the outside thermal resistance $1/h_a$ [$\text{m}^2 \text{K W}^{-1}$] between the insulating material and the environment.

$$U_b = U_s = \left(\frac{s}{\lambda} + \frac{1}{h_a} \right)^{-1} \quad (4.64)$$

The outside thermal resistance consists of a radiation-dependent and a convective, wind velocity-dependent proportion ($1/h_a = 1/(h_c + h_r)$). It is quite sufficient for the insulated parts to calculate using a standard value of outside thermal resistance of $1/h_a = 0.04 \text{ m}^2 \text{K W}^{-1}$. If all heat transfer coefficients are calculated from the absorber against the ambient temperature, then the parallel heat transfer coefficients can be added to the total loss coefficient U_t . The losses through the collector sides with the small side panel surface A_s are related here to the aperture surface of the collector A .

$$U_t = U_f + U_b + U_s \frac{A_s}{A} \quad (4.65)$$

Heat transfer coefficient of the transparent cover U_f

The heat transfer coefficient of the transparent cover consists of the total of the thermal resistances between the absorber and the environment. The thermal resistances between absorber and glazing R_{a-g} and between glazing and environment R_{g-o} are calculated from the parallel heat transfers for convection h_c and radiation h_r .

$$R_{a-g} = \frac{1}{h_{c,a-g} + h_{r,a-g}} \quad R_{g-o} = \frac{1}{h_{c,g-o} + h_{r,g-o}} \quad (4.66)$$

$$U_f = \frac{1}{R_{a-g} + R_{g-o}} \quad (4.67)$$

The temperature-dependent heat transfer coefficients are first calculated assuming a cover plate temperature T_g . As the heat flow $\dot{Q}_f$ over the collector front between absorber and environment,

$$\frac{\dot{Q}_f}{A} = U_f (T_a - T_o) \quad (4.68)$$

equals the heat flow between absorber and cover,

$$\frac{\dot{Q}_{a \rightarrow g}}{A} = (h_{c,a-g} + h_{r,a-g}) (T_a - T_g) \quad (4.69)$$

a new cover glass temperature can be calculated by equating the heat flows.

$$\begin{aligned} \frac{\dot{Q}_f}{A} &= U_f (T_a - T_o) = (h_{c,a-g} + h_{r,a-g}) (T_a - T_g) = \frac{\dot{Q}_{a \rightarrow g}}{A} \\ \Rightarrow T_g &= T_a - \frac{U_f (T_a - T_o)}{(h_{c,a-g} + h_{r,a-g})} \end{aligned} \quad (4.70)$$

In the following, the equations for the necessary heat transfer coefficients are discussed.

Heat transfer coefficient for radiation h_r

Using the Stefan–Boltzmann law, from a diffusely radiating black surface A_1 [m^2] with an emission coefficient $\varepsilon = 1$, the radiating power

$$\dot{Q}_1 = A_1 \sigma T_1^4 \quad (4.71)$$

is emitted, the Stefan–Boltzmann constant being $\sigma = 5.67 \times 10^{-8} \text{ W m}^{-2} \text{ K}^{-4}$. A part Φ_{12} of this radiation power can be absorbed by a second black surface A_2 placed at random in the hemisphere. A_2 for its part also emits radiation as a function of the temperature T_2 , with $\dot{Q}_2 = A_2 \sigma T_2^4$ of which the first surface absorbs a part Φ_{21} . The net radiation exchange $\dot{Q}$ between surfaces A_1 and A_2 results from the difference between the radiation to the respective other surface and the returning radiation.

$$\dot{Q} = \dot{Q}_{1 \rightarrow 2} - \dot{Q}_{2 \rightarrow 1} = \underbrace{\Phi_{12} A_1 \sigma T_1^4}_{\text{radiative flux } A_1 \text{ to } A_2} - \underbrace{\Phi_{21} A_2 \sigma T_2^4}_{\text{radiative flux } A_2 \text{ to } A_1} \quad (4.72)$$

As both surfaces have been defined as black emitters with absorption and emission coefficients of 1, no reflection of the radiation at the surfaces takes place.

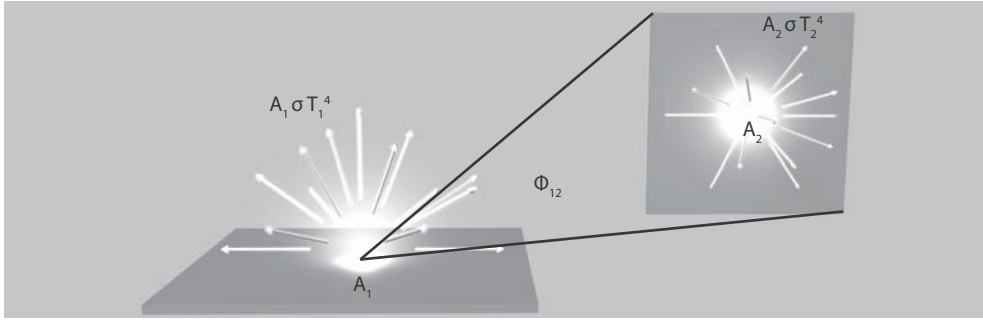


Figure 4.32 Radiant flux of a small (differential) two-dimensional element A_1 at temperature T_1 to the surface element A_2 , which absorbs the proportion Φ_{12} and in turn emits radiation.

If the two surfaces are at the same temperature, the net radiant heat exchange equals zero and the reciprocity condition for the form factors Φ_{12} and Φ_{21} is obtained:

$$\Phi_{12} A_1 = \Phi_{21} A_2 \quad (4.73)$$

The heat flow by radiation exchange between two black emitters is thus given by:

$$\dot{Q} = \Phi_{12} A_1 \sigma (T_1^4 - T_2^4) = \Phi_{21} A_2 \sigma (T_1^4 - T_2^4) \quad (4.74)$$

The form factors Φ_{ij} as a geometrical ratio of the radiation-receiving surface A_j to the entire hemisphere, into which power is irradiated, are generally complicated functions of the surface geometry.

For the calculation of the heat transfer coefficients in solar thermal collectors, two cases are of special importance; the radiation exchange between two flat-parallel plates (absorber and transparent cover) and the radiation exchange between the flat cover and the sky hemisphere. If with the parallel plates it is approximately assumed that these are infinitely expanded, the entire radiant flux of surface A_1 is absorbed by surface A_2 and vice versa, i.e. $\Phi_{12} = \Phi_{21} = 1$.

In the radiation exchange of a flat surface with the sky hemisphere, the radiation emitted by surface A_1 is completely received by the sky hemisphere, i.e. $\Phi_{12} = 1$. Conversely the proportion which the small surface A_1 sees of the total radiation of the sky hemisphere is very small. Based on the reciprocity condition from Equation (4.73), here $\Phi_{21} = A_1 / A_2 \approx 0$ for $A_1 \ll A_2$.

Normally the emission or absorption coefficient of surfaces is less than one (grey emitters), so the radiation received from the second surface with $(1 - \varepsilon_2) = (1 - \varepsilon_2)$ is partly back-reflected, and in turn can be absorbed by surface A_1 with ε_1 and reflected back with $(1 - \varepsilon_1)$ etc..

To create a radiation energy balance for surface A_1 , the radiation $\dot{Q}_{1 \rightarrow 2 \rightarrow 1}$ reflected by surface A_2 and absorbed again by A_1 must be deducted from the radiated energy quantity $\dot{Q}_1$. Furthermore, the radiation $\dot{Q}_{2 \rightarrow 1}$ emitted by surface A_2 and absorbed by A_1 must be treated as an energy gain. This balance will now be discussed for the simple case of flat-parallel plates with a form factor of 1. For radiation $\dot{Q}_1$ with intensity emitted by surface A_1 , the following amount of heat returns:

$$\begin{aligned} \frac{\dot{Q}_{1 \rightarrow 2 \rightarrow 1}}{\dot{Q}_1} &= \underbrace{(1 - \varepsilon_2)}_{\text{first reflexion from } A_2} \times \underbrace{\varepsilon_1}_{\text{absorption at } A_1} + \underbrace{(1 - \varepsilon_2)(1 - \varepsilon_1)}_{\text{first back reflexion from } A_1} \times \underbrace{(1 - \varepsilon_2)\varepsilon_1}_{\text{second reflexion from } A_2 \text{ absorbed by } A_1} \\ &+ \underbrace{(1 - \varepsilon_2)(1 - \varepsilon_1)(1 - \varepsilon_2)}_{\text{second reflexion from } A_1} \underbrace{(1 - \varepsilon_1)}_{\text{third reflexion from } A_2 \text{ absorbed by } A_1} \times \underbrace{(1 - \varepsilon_2)\varepsilon_1}_{\text{third reflexion from } A_2 \text{ absorbed by } A_1} + \dots \\ &= (1 - \varepsilon_2)\varepsilon_1 \left(1 + (1 - \varepsilon_2)(1 - \varepsilon_1) + ((1 - \varepsilon_2)(1 - \varepsilon_1))^2 + \dots \right) = \frac{(1 - \varepsilon_2)\varepsilon_1}{1 - (1 - \varepsilon_2)(1 - \varepsilon_1)} \end{aligned} \quad (4.75)$$

In addition, surface A_2 emits the intensity $\dot{Q}_2$, of which surface A_1 absorbs a proportion ε_1 and reflects $(1 - \varepsilon_1)$. After multiple reflections the infinite sequence of the radiation absorbed by A_1 results in:

$$\frac{\dot{Q}_{2 \rightarrow 1}}{\dot{Q}_2} = \varepsilon_1 + (1 - \varepsilon_1)(1 - \varepsilon_2)\varepsilon_1 + \dots = \frac{\varepsilon_1}{1 - (1 - \varepsilon_1)(1 - \varepsilon_2)} \quad (4.76)$$

The energy balance of surface A_1 thus results for the radiation exchange between flat-parallel plates with $A_1 = A_2$ and $\Phi_{12} = \Phi_{21} = 1$.

$$\begin{aligned} \dot{Q}_{1, \text{net}} &= \dot{Q}_1 - \dot{Q}_{1 \rightarrow 2 \rightarrow 1} - \dot{Q}_{2 \rightarrow 1} \\ &= \dot{Q}_1 - \dot{Q}_1 \frac{(1 - \varepsilon_2)\varepsilon_1}{1 - (1 - \varepsilon_2)(1 - \varepsilon_1)} - \dot{Q}_2 \frac{\varepsilon_1}{1 - (1 - \varepsilon_2)(1 - \varepsilon_1)} \\ &= \frac{\dot{Q}_1 \varepsilon_2 - \dot{Q}_2 \varepsilon_1}{\varepsilon_1 + \varepsilon_2 - \varepsilon_1 \varepsilon_2} \\ &= \frac{\varepsilon_1 \sigma A_1 T_1^4 - \varepsilon_2 \sigma A_2 T_2^4}{\varepsilon_1 + \varepsilon_2 - \varepsilon_1 \varepsilon_2} = \frac{1}{\frac{1}{\varepsilon_2} + \frac{1}{\varepsilon_1} - 1} \sigma A (T_1^4 - T_2^4) \end{aligned} \quad (4.77)$$

This heat flow equation can be linearised by extracting the temperature difference $T_1 - T_2$ and thus the heat transfer coefficient for radiation h_r is determined.

$$\dot{Q} = h_r A (T_1 - T_2) \quad (4.78)$$

$$h_r = \frac{\sigma}{1/\varepsilon_1 + 1/\varepsilon_2 - 1} (T_1^2 + T_2^2)(T_1 + T_2) \quad (4.79)$$

For the calculation of the radiation exchange between the absorber at temperature T_a and glazing at temperature T_g , $T_1 = T_a$ and $T_2 = T_g$ are set.

Selective coating

The long-wave emission coefficients of black absorber colour are typically 95%, those of glass 88%. The emissivity can be reduced if a solar-radiation absorbing coating is applied on a substrate with a small emission coefficient, for example, a metal. Galvanisation of metals with black chrome is increasingly replaced by environmentally friendly cathodic sputtering methods, which also consume only one-tenth of the energy required for galvanisation. Emissivities of 5% can be reached, while short-wave irradiance absorption stays at 95%.

If one or more form factors Φ_{ij} are not equal to one, the energy balances must be modified accordingly. This can be analysed by the example of the radiation exchange of a surface against the sky hemisphere.

The radiation $\dot{Q}_1 = \varepsilon_1 \sigma A_1 T_1^4$ emitted by surface A_1 meets the sky hemisphere with form factor $\Phi_{12} = 1$ and is reflected with $(1 - \varepsilon_2)$. Now, however, only a fraction of the irradiance reflected from the sky reaches surface A_1 , i.e. $\Phi_{21} = A_1/A_2$. The infinite sequence of the radiation reabsorbed by A_1 is thus modified as follows:

$$\begin{aligned} \frac{\dot{Q}_{1 \rightarrow 2 \rightarrow 1}}{\dot{Q}_1} &= \underbrace{(1 - \varepsilon_2)}_{1. \text{ refl. from } A_2} \underbrace{\frac{A_1}{A_2} \varepsilon_1}_{1. \text{ abs. from } A_1} + \underbrace{(1 - \varepsilon_2)}_{1. \text{ refl. from } A_2} \underbrace{\left(1 - \frac{A_1}{A_2}\right) \varepsilon_1}_{1. \text{ refl. from } A_1} \underbrace{(1 - \varepsilon_2)}_{2. \text{ refl. from } A_2} \underbrace{\frac{A_1}{A_2} \varepsilon_1}_{2. \text{ abs. from } A_1} + \dots \\ &= \frac{(1 - \varepsilon_2) \varepsilon_1 \frac{A_1}{A_2}}{1 - (1 - \varepsilon_2) \left(1 - \frac{A_1}{A_2} \varepsilon_1\right)} \end{aligned} \quad (4.80)$$

In a similar way, the radiation emitted from A_2 towards A_1 is modified by the factor A_1/A_2 .

$$\frac{\dot{Q}_{2 \rightarrow 1}}{\dot{Q}_2} = \frac{\varepsilon_1 \frac{A_1}{A_2}}{1 - \left(1 - \frac{A_1}{A_2} \varepsilon_1\right) (1 - \varepsilon_2)} \quad (4.81)$$

The net radiant flux of surface A_1 is thus given by:

$$\dot{Q}_{1, \text{net}} = \frac{1}{\frac{A_1}{A_2} \frac{1}{\varepsilon_2} + \frac{1}{\varepsilon_1} - \frac{A_1}{A_2}} \sigma A_1 (T_1^4 - T_2^4) \quad (4.82)$$

from which Equation (4.77) results as a special case for equal surfaces $A_1 = A_2$. If surface A_1 is very much smaller than surface A_2 , then Equation (4.82) is simplified to:

$$\dot{Q}_{i,net} \approx \varepsilon_1 \sigma A_1 (T_1^4 - T_2^4) \text{ for } A_1 \ll A_2 \quad (4.83)$$

If the sky temperature is defined with T_{sky} ($T_2 = T_{sky}$), the result for the heat transfer coefficient between cover ($T_1 = T_g$) and sky is:

$$h_r = \sigma \varepsilon_1 (T_g^2 + T_{sky}^2) (T_g + T_{sky}) \quad (4.84)$$

The sky temperature takes into account the transparency of the atmosphere, i.e. missing back radiation in the wavelength range between approximately 8 - 14 μm . The sky temperature can be calculated from the dew point temperature T_{dp} [K]:

$$T_{sky} = T_o \left(0.8 + \frac{T_{dp} - 273}{250} \right)^{0.25} \quad (4.85)$$

As a simpler equation without calculation of the water vapour partial pressure and the dew point temperature, the modified Swinbank equation [K] can be used (Fuentes, 1987):

$$T_{sky} = 0.037536 T_o^{1.5} + 0.32 T_o \quad (4.86)$$

To be able to calculate the heat flows as usual by the temperature difference between glass cover T_g and ambient temperature T_o , the heat transfer coefficient for radiation is normalised to this temperature difference.

$$h_r = \sigma \varepsilon_1 (T_g^2 + T_{sky}^2) (T_g + T_{sky}) \left(\frac{T_g - T_{sky}}{T_g - T_o} \right) \quad (4.87)$$

Convective heat transfer coefficients h_c

Convective heat transfer occurs in the form of natural convection between the absorber and transparent cover, and as forced convection by wind forces between the cover and the environment. The characteristic nominal value for the convective heat transfer is always the Nußelt number Nu , from which the convective heat transfer coefficient can be calculated as a function of a characteristic length L and of the heat conductivity λ of the air.

$$Nu = \frac{h_c L}{\lambda} \quad (4.88)$$

Free convection in a standing air layer

In a standing air layer between flat-parallel plates, the plate distance d is used as a characteristic length L for the calculation of h_c . For the calculation of the Nußelt number, there exists a set of empirical correlations for flat-plate collector geometries with a temperature gradient ΔT between absorber and cover and different collector angles of inclination β . Based on Duffie and Beckmann

(1980), the following equation can be used up to collector angles of inclination of 75°; for angles of inclination over 75° the function value of 75° is retained.

$$Nu = 1 + 1.44 \left[1 - \frac{1708(\sin 1.8\beta)^{1.6}}{Ra \cos \beta} \right]^+ \left[1 - \frac{1708}{Ra \cos \beta} \right]^+ + \left[\left(\frac{Ra \cos \beta}{5830} \right)^{1/3} - 1 \right]^+ \quad (4.89)$$

The plus sign of the bracketed term means that only positive results are to be used; with negative bracketed terms, the term is set at zero. The Rayleigh number describes the buoyancy by the thermally caused density variations and is given by the product of the Grashof (Gr) and Prandtl (Pr) numbers:

$$Ra = Gr Pr = \frac{g\beta'\Delta T L^3}{\nu^2} \times \frac{\nu \rho c_p}{\lambda} = \frac{g\beta'\Delta T L^3}{\nu \alpha} \quad (4.90)$$

where

- g Gravitation constant [m/s^2]
- $\beta' = 1/T$ Volume expansion coefficient of ideal gases [K^{-1}]
- ΔT Temperature difference between the panels [K]
- L Characteristic length, here the panel separation d [m]
- ν Kinematic viscosity [m^2/s]
- $\alpha = \lambda / \rho c_p$ Temperature conductivity [m^2/s]

Forced convection by wind

The convective heat flow at the glass cover of a collector is mainly caused by wind, i.e. forced convection. A smaller proportion results from free convection between the pane and the ambient temperature. A good approximation is given by the following overlay of the two heat transfer coefficients:

$$h_c = \sqrt[3]{h_{c,w}^3 + h_{c,free}^3} \quad (4.91)$$

The forced convection transition coefficient $h_{c,w}$ is calculated from a Nußelt correlation for a plate with flow parallel to the plate with a turbulent boundary layer.

$$Nu_{turb} = \frac{0.037 Re^{0.8} Pr}{1 + 2.443 Re^{-0.1} \left(Pr^{\frac{2}{3}} - 1 \right)} \Rightarrow h_{c,w} = \frac{Nu_{turb} \lambda}{L} \quad (4.92)$$

The Reynolds number $Re = v_w * L / \nu$ results from the wind velocity v_w and the plate length L under parallel flow conditions from. For the material values Nu , ν , ρ , λ and c_p , the temperature of the ambient air is used. The heat transfer coefficient $h_{c,w}$ is likewise calculated with the panel length L as a characteristic length. For the calculation of the free convection proportion, the glass temperature T_g and the ambient temperature T_o must be known.

$$h_{c,free} = 1.78 (T_g - T_o)^{1/3} \quad [W m^{-2} K^{-1}] \quad (4.93)$$

A simplified approach takes only the wind velocity into account, so

$$h_{c,w} = 4.214 + 3.575 v_w \quad (4.94)$$

Example 4.6

Calculation of the front U value U_f of collectors with low emissivity ($\varepsilon = 0.1$) and with high emissivity without selective coating ($\varepsilon = 0.9$) of the absorber under the following boundary conditions:

Absorber temperature	$T_a = 70^\circ\text{C}$
Ambient temperature	$T_o = 10^\circ\text{C}$
Wind velocity	$v_w = 3 \text{ m/s}$
Emission coefficient of transparent cover	$\varepsilon_g = 0.88$
Plate distance absorber glass	$d = 2.5 \text{ cm}$
Collector angle of inclination	$\beta = 45^\circ$
Assumed glass pane temperature for first iteration	$T_g = 40^\circ\text{C}$

1st iteration:

The heat transfer coefficient for radiation $h_{r,a-g}$ between absorber and glass is $0.79 \text{ W m}^{-2} \text{ K}^{-1}$ for the selectively coated absorber, and $6.44 \text{ W m}^{-2} \text{ K}^{-1}$ for the black absorber. The convective heat transfer coefficient of the standing air layer is determined by the material values of the mean air temperature of $(70 + 40)/2 = 55^\circ\text{C}$ (where $\lambda = 0.0286 \text{ W m}^{-1} \text{ K}^{-1}$, $\nu = 1.85 \times 10^{-5} \text{ m}^2 \text{ s}^{-1}$, $\rho = 1.045 \text{ kg m}^{-3}$, $cp = 1.009 \text{ kJ kg}^{-1} \text{ K}^{-1}$). The Prandtl number is 0.71 and the Grashof number 43787 at a mean temperature difference of 30 K between absorber and assumed pane temperature, so the Rayleigh number is 31089. From this results a Nußelt number of 2.78 and a convective heat transfer coefficient $h_{c,a-g} = 2.9 \text{ W m}^{-2} \text{ K}^{-1}$.

Between the glass cover and the sky, according to Swinbank, a radiant heat transfer coefficient of $h_{r,g-o} = 7.23 \text{ W m}^{-2} \text{ K}^{-1}$ is obtained at a sky temperature of 269 K, and for the wind-velocity dependent convective heat transfer based on simplified Equation (4.94) $h_{c,w} = 14.9 \text{ W m}^{-2} \text{ K}^{-1}$.

From this results the heat transfer coefficient of the transparent cover after the first iteration of

$$U_f = \frac{1}{\frac{1}{2.9 + 0.79} + \frac{1}{14.9 + 7.23}} = 3.17 \text{ W m}^{-2} \text{ K}^{-1}$$

for the selectively coated absorber, and $6.57 \text{ W m}^{-2} \text{ K}^{-1}$ for the black absorber.

With these values the new pane temperature is calculated:

$$T_{g,new} = 70^\circ\text{C} - \frac{3.17 \text{ W m}^{-2} \text{ K}^{-1} \times 60^\circ\text{C}}{2.9 \text{ W m}^{-2} \text{ K}^{-1} + 0.79 \text{ W m}^{-2} \text{ K}^{-1}} = 18.6^\circ\text{C}$$

or 27.8°C for the non-selective absorber. The temperature is clearly lower than the originally set temperature of 40°C . With this temperature the heat transfer coefficients are again calculated. After the end of iteration the following values are obtained:

Parameter	Selective coating	Non-selective
$h_{r,a-g}$	$0.72 \text{ W m}^{-2}\text{K}^{-1}$	$6.08 \text{ W m}^{-2}\text{K}^{-1}$
$h_{c,a-g}$	$3.4 \text{ W m}^{-2}\text{K}^{-1}$	$3.2 \text{ W m}^{-2}\text{K}^{-1}$
$h_{r,g-o}$	$11.88 \text{ W m}^{-2}\text{K}^{-1}$	$8.3 \text{ W m}^{-2}\text{K}^{-1}$
$h_{c,w}$	$14.9 \text{ W m}^{-2}\text{K}^{-1}$	$14.9 \text{ W m}^{-2}\text{K}^{-1}$
T_g	18.0°C	27.1°C
U_f	$3.56 \text{ W m}^{-2}\text{K}^{-1}$	$6.64 \text{ W m}^{-2}\text{K}^{-1}$

Optical characteristics of transparent covers and absorber materials

The transmittance of a transparent cover results from the reflection losses at the boundary surfaces and the absorption losses in the cover material itself. The reflection losses depend on the refractive index of the cover material and on the mechanical structuring of the cover. The lowest reflection losses are achieved if an antireflective coating with a refractive index is used, which has a value between the materials 'air' and the cover material and if the glass is very plane. The mechanical structuring, which is still often used to avoid glare from direct irradiance reflections and to optically hide the interior of the collector, increases reflection losses especially at higher incidence angles. Compared to a plane, non-structured glass, the annual additional reflection losses of a structured glass are up to 6% higher.

Absorption of the cover material can be reduced by a lower iron content of the glass, which increases transmission from typically 84% for standard float glass to 89 – 91% for 'solar glass'. A low-iron glass with antireflection coating can achieve 96% transmission at zero incidence angle.

At the boundary surface of two media with different refractive indices (here, for example, air and glass), the reflection characteristics of the surface can be calculated from constancy conditions for the electrical and magnetic field. For this, the almost unpolarised natural light is divided into two components, which strike the boundary surface parallel or perpendicularly to the plane of incidence. The square of the field strengths results in the radiating power with the associated coefficient of reflection r .

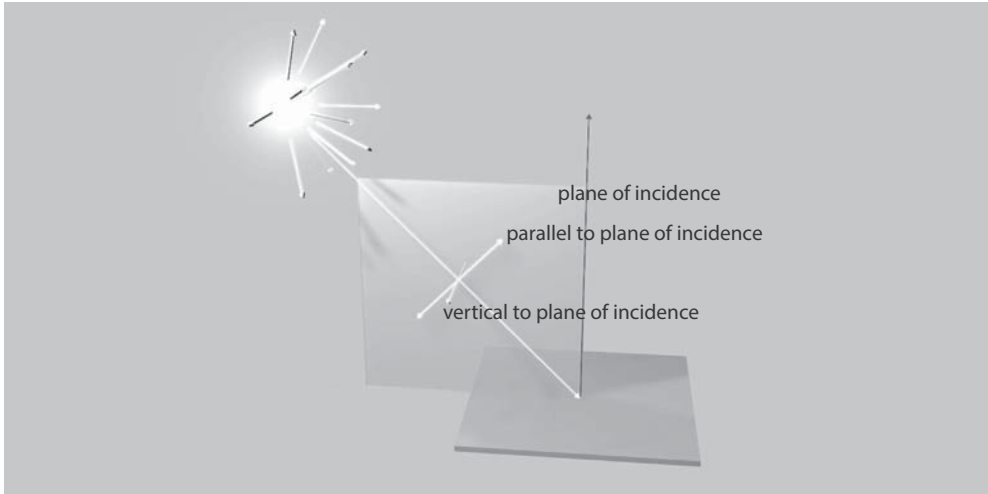


Figure 4.33 Polarisation directions parallel and perpendicular to the plane of incidence.

The plane of incidence is defined by the surface normal and the irradiance vector. The reflected radiating power G_r in relation to the incident power G_i results from the average value between parallel and perpendicularly polarised components, which must first be calculated separately.

$$r = \frac{G_r}{G_i} = \frac{1}{2}(r_{\perp} + r_{\parallel}) \quad (4.95)$$

The reflection factors are calculated according to the Fresnel formulae from the angle of incidence θ_1 and the angle of refraction in the material θ_2 .

$$r_{\perp} = \frac{\sin^2(\theta_2 - \theta_1)}{\sin^2(\theta_2 + \theta_1)} \quad (4.96)$$

$$r_{\parallel} = \frac{\tan^2(\theta_2 - \theta_1)}{\tan^2(\theta_2 + \theta_1)} \quad (4.97)$$

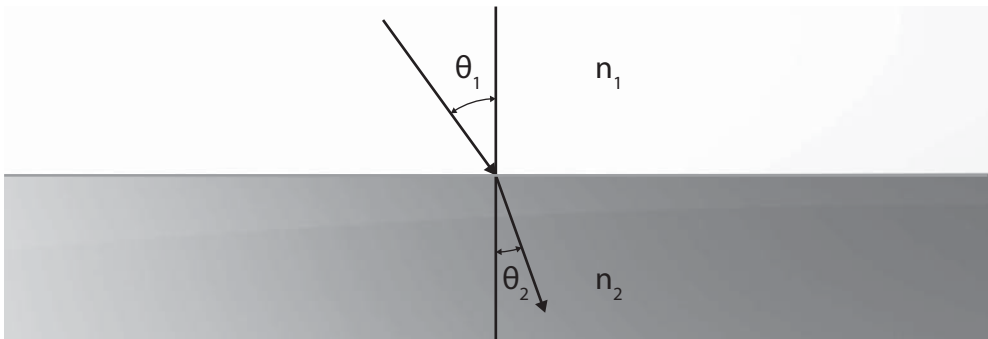


Figure 4.34 Illustration of the angles and refractive indices.

The angles are, according to Snell's law, a function of the refractive indices n_1 and n_2 :

$$\frac{n_1}{n_2} = \frac{\sin \theta_2}{\sin \theta_1} \tag{4.98}$$

With perpendicular irradiance, both angles are zero, and the reflection coefficient of the boundary surface becomes

$$r(0) = \left(\frac{n_1 - n_2}{n_1 + n_2} \right)^2 \tag{4.99}$$

The reflection factors at the boundary surface are equal for perpendicular irradiance, then decrease to zero as a function of the angle of incidence θ_1 with parallel polarised light, and finally rise to one at parallel incidence. With perpendicular polarisation the reflection factor constantly rises.

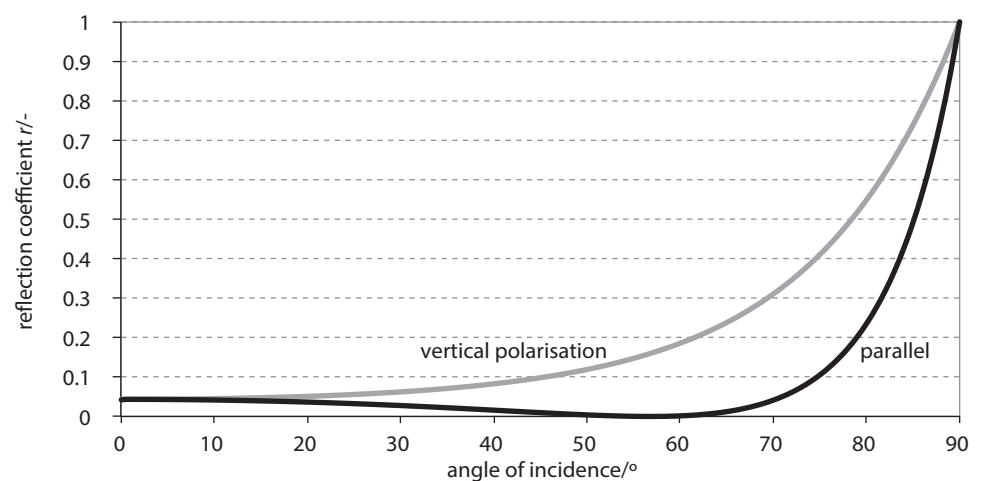


Figure 4.35 Reflection coefficient r for parallel and perpendicularly polarised incoming radiation after Fresnel.

Material	Refractive index
Air	1.0
Glass	1.526
Polycarbonate	1.6
Polymethyl methacrylate (Plexiglas)	1.49

Table 4.7 Refractive indices of transparent materials.

Absorption in the material

The reduction in radiation intensity in the cover material itself (dG) is proportional to the absolute intensity G of the radiation and to the extinction coefficient K of the material:

$$\begin{aligned}dG &= -GK dx \\ G &= C \exp(-K(x - x_0))\end{aligned}\tag{4.100}$$

The boundary condition is given by the incident intensity G_0 at the point $x = x_0$. The distance travelled in the material results from the thickness of the material L and the cosine of the angle in the material θ_2 to $L/\cos \theta_2$, so the transmittance τ_a of the material (without surface reflections) results from the intensity of the irradiance G_t after a single ray passage to the entering intensity G_0 at x_0 .

$$\tau_a = \frac{G_t}{G_0} = \exp\left(-K \frac{L}{\cos \theta_2}\right)\tag{4.101}$$

Material	Extinction coefficient K/m^{-1}
Solar glass	4
Typical window glass	30
Absorbing sun protection glass	130 - 270

Table 4.8 Extinction coefficients of transparent materials.

Transmission and reflection coefficients of the transparent cover

If the passage of an incoming ray is followed, taking into account the reflection losses at the entrance boundary surface in the material, then the intensity of the first outgoing ray is obtained, including further reflection losses at the exit boundary surface between material and air. The rays reflected at the exit boundary surface are followed further and lead finally, after further reflections, to further exiting rays of lower intensity. The overall transmittance results as the ratio of an infinite series of the exiting radiation intensity to the incident radiation.

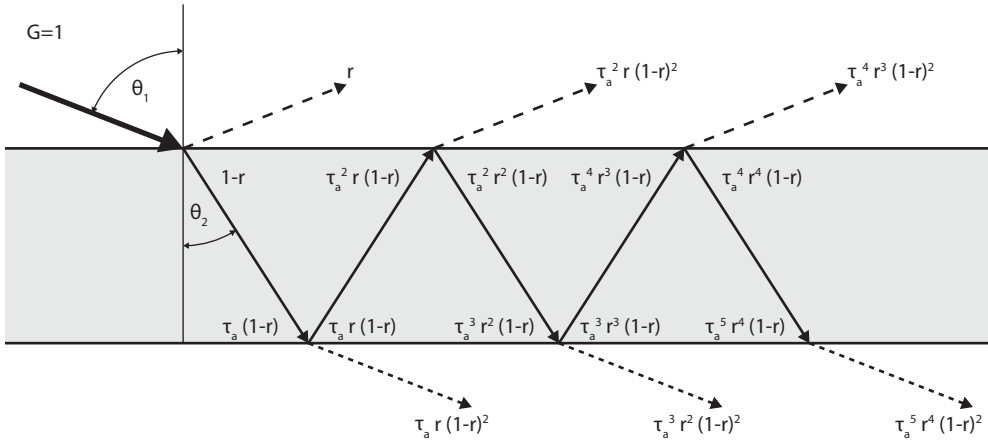


Figure 4.36 Transmission and reflection of a single cover material.

Similarly the entire reflection coefficient is calculated from the infinite series of the reflected rays. Here the transmission or reflection coefficients must be calculated separately for both polarisation directions, and afterwards can be averaged arithmetically for unpolarised natural light.

$$\begin{aligned}\tau_{\perp} &= (1-r_{\perp})^2 \tau_a \sum_{n=0}^{\infty} r_{\perp}^{2n} \tau_a^{2n} = \frac{\tau_a (1-r_{\perp})^2}{1-(r_{\perp} \tau_a)^2} \\ \tau_{\parallel} &= \frac{\tau_a (1-r_{\parallel})^2}{1-(r_{\parallel} \tau_a)^2} \\ \tau &= \frac{\tau_{\perp} + \tau_{\parallel}}{2}\end{aligned}\quad (4.102)$$

$$\begin{aligned}\rho_{\perp} &= r_{\perp} + r_{\perp} (1-r_{\perp})^2 \tau_a^2 \sum_{n=0}^{\infty} r_{\perp}^{2n} \tau_a^{2n} = r_{\perp} + \frac{r_{\perp} (1-r_{\perp})^2 \tau_a^2}{1-(r_{\perp} \tau_a)^2} \\ \rho_{\parallel} &= r_{\parallel} + \frac{r_{\parallel} (1-r_{\parallel})^2 \tau_a^2}{1-(r_{\parallel} \tau_a)^2} \\ \rho &= \frac{\rho_{\perp} + \rho_{\parallel}}{2}\end{aligned}\quad (4.103)$$

The absorption coefficient of the material can likewise be calculated via an infinite series or directly from the transmission and reflection degrees.

$$\alpha_{\perp} = 1 - \tau_{\perp} - \rho_{\perp} = (1-r_{\perp}) \tau_a \sum_{n=0}^{\infty} (1+r_{\perp}^n \tau_a^n) \quad (4.104)$$

Example 4.7

Calculation of the transmission and reflection degrees of a single-glazed, 4 mm thick solar glass cover (with $K = 4 \text{ m}^{-1}$ and $n = 1.526$) for angles of incidence of 0° and 60° .

With perpendicular incidence, the reflection coefficient for perpendicular and parallel polarised light is $r(0) = 0.043$. With $\tau_a = 0.984$:

$$\tau(0) = \frac{0.98(1-0.043)}{1-(0.98 \times 0.043)^2} = 0.9$$

$$\rho(0) = 0.043 + \frac{0.043(1-0.043)^2 0.98^2}{1-(0.043 \times 0.98)^2} = 0.081$$

For non-perpendicular incidence, both polarisation directions must be regarded separately. For $\theta_1 = 60^\circ$ is $\theta_2 = 35^\circ$. Thus, and $\tau_a = 0.98$. The transmission coefficients are and the reflection factors $\rho_\perp = 0.31$, $\rho_\parallel = 0.0028$ and $\rho = 0.16$, i.e. twice as high as with perpendicular incidence.

Absorption and transmission - absorption product

The absorption factor of absorber materials can be regarded with good accuracy as angle-independent.

Material	Absorption coefficient α	Emissivity ϵ
Black paint	0.95	0.95
Black chrome on nickel (galvanically applied selective coating)	0.95	0.1
TiNOx (vacuum sputter coating)	0.94	0.038

Table 4.9 Absorption coefficients and emissivities of typical absorber materials.

Taking into account multiple reflections between the transparent cover and absorber, the result is, from the infinite series, an effective transmission absorption product:

$$(\tau\alpha) = \tau\alpha \sum_{n=0}^{\infty} ((1-\alpha)\rho)^n = \frac{\tau\alpha}{1-(1-\alpha)\rho} \quad (4.105)$$

where ρ is the reflection factor for light reflected diffusely at the underside of the glass (this can be calculated simply for an average angle of incidence of 60°).

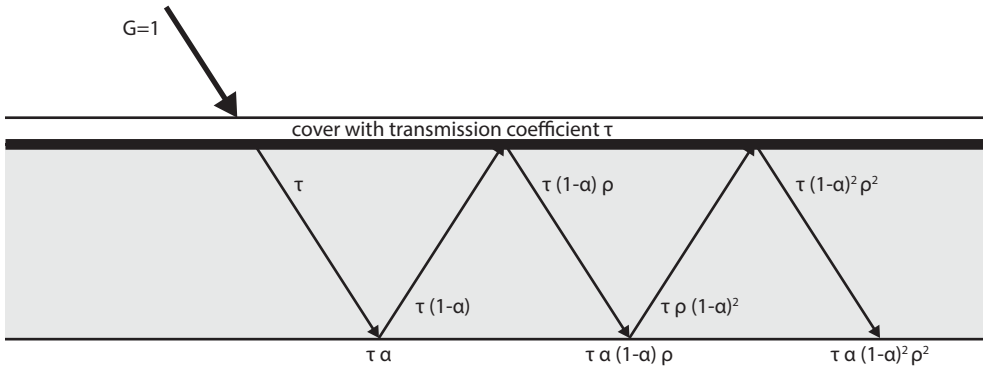


Figure 4.37 Effective transmission-absorption product of a single-glazed collector.

4.7 Storage modelling

The simplest model for a heat store is a fully mixed storage tank with a homogeneous temperature level.

If no energy is supplied or withdrawn from the storage tank, the heat losses can then be easily calculated. They consist of the heat transfer losses of the thermal insulation and also the convective losses by fluid circulation through connections and armatures.

The determination of the heat loss as a product of the effective heat transfer coefficient U_{eff} and external surface A of the store ($U_{eff}A$) is possible by a simple energy balance: if no warm water is withdrawn from the store after homogeneous heating to a temperature level T_{st} , the temperature of the store drops due to the calorific losses of the shell of the store to the ambient temperature T_o and by free convection in the tank connections.

$$mc \frac{dT_{st}}{dt} = -(U_{eff}A)(T_{st}(t) - T_o) \quad (4.106)$$

As a boundary condition at the point in time $t=0$, the storage temperature is given after heating up the store: from the exponential drop in storage temperature, the effective calorific loss of the store can be determined in W/K .

$$T_{st}(t) = (T_{st,0} - T_o) \exp\left(-\frac{(U_{eff}A)}{mc}t\right) + T_o \quad (4.107)$$

Example 4.8

Determination of the effective calorific losses of a 750 litre heater buffer store from measured storage temperature values on a night without direct heat withdrawal. The room temperature, where the store is placed, is constant at 13°C.

The measured values are plotted as a ratio of temperature differences against time, here in minutes: the difference between the time-dependent storage temperature minus ambient

temperature is divided by the difference between initial storage minus ambient temperature. From the exponential function, the heat loss coefficient can be derived.

$$\frac{T_{st}(t) - T_o}{T_{st,0} - T_o} = \exp\left(-\frac{(U_{eff}A)t}{mc}\right) = \exp(-0.0001 \times t)$$

For the store with a water content $m = 750 \text{ kg}$ and a heat capacity of water of $1.16 \text{ Wh kg}^{-1} \text{ K}^{-1} \times 60 \text{ min/h} = 69.9 \text{ W min kg}^{-1} \text{ K}^{-1}$, an effective calorific loss of $U_{eff}A = 10^{-4} \times mc = 5.22 \text{ W K}^{-1}$ results from the exponential coefficient of 10^{-4} min^{-1} .

Time/min	0	30	60	90	120	150	180	210	240	270
Storage temperature/°C	49.2	49.15	49	48.85	48.75	48.6	48.5	48.4	48.25	48.15
Time/min	300	330	360	390	420	450	480	510	540	570
Storage temperature/°C	48.05	47.95	47.8	47.7	47.55	47.45	47.3	47.2	47.1	46.95

Table 4.10 Development of storage temperature levels in a 750 litre storage tank with no charging or discharging.

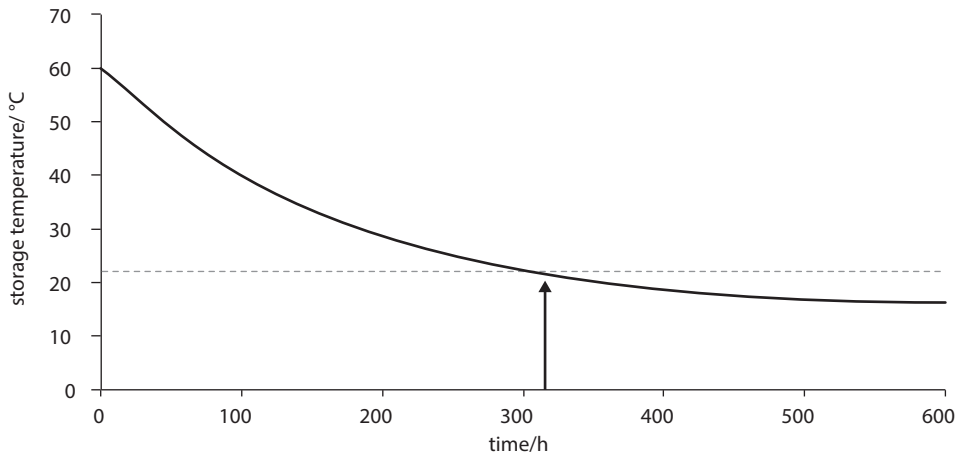


Figure 4.38 Exponential drop in storage temperature. After 308 h, i.e. some 13 days, the temperature has fallen to $1/e$ of the initial value.

Typical heat loss values for a 400 litre drinking water store are between 1.7 and 3 W K^{-1} , depending upon the standard of insulation and type of pipe connections, for a 1000 litre store between 3.7 and 5.5 W K^{-1} . Substantially higher loss coefficients indicate missing non-return valves and strong convection of the storage water through the connection pipes. The effective calorific loss of the heater buffer store in the example is thus in the upper range of the loss values.

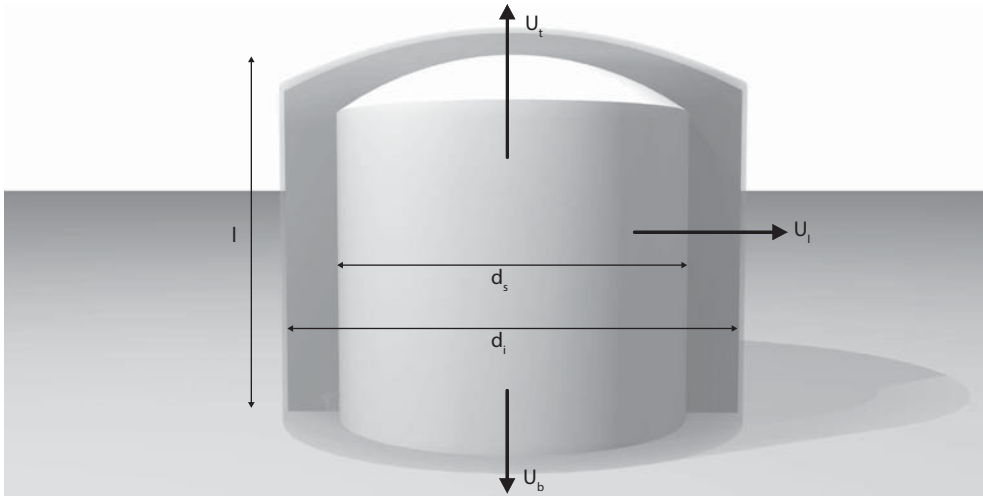


Figure 4.39 Dimensions and heat loss coefficients of solar storage.

Solar stores are often cylinder geometries with flat ball caps as covers and bases. For simplification, the ball caps can be approximated in small short-term stores by flat surfaces. The U values of the store top cover U_t and bottom U_b are calculated from the layer thickness s and heat conductivity λ of the thermal insulation (if present) as well as from the thermal resistance between the thermal insulation and room air h_i^{-1} (standard value $0.13 \text{ m}^2 \text{ K W}^{-1}$).

$$U_t = \left(\frac{s}{\lambda} + \frac{1}{h_i} \right)^{-1} \quad U_b = \left(\frac{1}{h_i} \right)^{-1} \quad (4.108)$$

The length-related U_l value of the standing cylinder results from the solution of the stationary thermal heat conduction equation in cylindrical coordinates and depends on the total external diameter of the store d_s and d_i as the diameter of the store including the thermal insulation. The thermal resistance between the fluid and the store wall can be neglected when calculating the U_l value.

$$U_l = \frac{\pi}{\frac{1}{2\lambda} \ln \frac{d_i}{d_s} + \frac{1}{h_i d_i}} \quad (4.109)$$

The entire calorific loss of the store per kelvin temperature difference to the ambient air thus results in:

$$UA = U_l l + U_t A_t + U_b A_b \quad (4.110)$$

Example 4.9

Calculation of the calorific losses of the external surface of the above 750 litre store with the following geometry:

Store height l	2.03 m
Store diameter d_s	0.69 m
120 mm polyurethane foam insulation with heat conductivity λ	0.04 W m ⁻¹ K ⁻¹
Heat transfer coefficient to the room h_i	8 W m ⁻² K ⁻¹

The bottom of the store is not insulated.

From the above values, the total diameter with insulation $d_i = 0.93$ m results, thus a length-related U_l value of 0.81 W m⁻¹ K⁻¹ and for the given store height a calorific loss U_l of 1.65 W K⁻¹. In addition there are calorific losses at the top and the bottom, both with a surface of 0.37 m². The heat transfer coefficient of the insulated storage top is

$$U_t = \frac{1}{s/\lambda + 1/h_i} = 0.32 \frac{W}{m^2 K}$$

and of the uninsulated bottom

$$U_b = \frac{1}{1/h_i} = 8 \frac{W}{m^2 K}$$

The entire calorific loss of the external surface is thus:

$$UA = U_l l + U_t A_t + U_b A_b = 1.65 W / K + 0.12 W / K + 2.96 W / K = 4.73 W / K$$

Although the bottom constitutes only 7% of the total external surface of 5.14 m², its calorific loss coefficient dominates. With good temperature stratification of the store, however, the temperature at the store's lower surface, typically 20°C, is clearly lower than the top cover area temperature of about 60°C, so the total calorific loss $\dot{Q}_l$ at a mean storage temperature of 40°C is distributed roughly thus:

$$\dot{Q}_l = \underbrace{1.65 W / K \times (40 - 15) K}_{41.25 W} + \underbrace{0.12 W / K \times (60 - 15) K}_{5.4 W} + \underbrace{2.96 W / K \times (20 - 15) K}_{14.8 W} = 61.45 W$$

For the calculation of the calorific losses of stores, a simple energy balance (Equation (4.106)) was used for the temperature decrease of the stored water as a function of time. A prerequisite for the analytic solution of the differential equation was the assumption of a homogeneously mixed store, with no heat supply or withdrawal from the store during cooling. In normal operation, however, apart from the transmission heat losses from the store $\dot{Q}_t$, the solar collector and the auxiliary heat source supply the amount of heat $\dot{Q}_c$ and $\dot{Q}_h$, and a heat load $\dot{Q}_l$ is withdrawn by consumers for warm water or heating.

The temporal development of the store temperature T_s results from the energy balance:

$$mc \frac{dT_s}{dt} = \dot{Q}_c + \dot{Q}_h - \dot{Q}_l - \dot{Q}_t \quad (4.111)$$

If the parameter δ_c or δ_l are used for the operation of the collector circuit or the load circuit of consumers, with the values 1 in operation and 0 at pump standstill or zero load, the energy balance can be represented as a function of the temperatures. The return temperature into the collector and the withdrawal temperature for consumers (consumer supply temperature) equals the store

temperature T_s for a homogeneously mixed storage tank. The return temperature of the consumers $T_{l,return}$ is (in warm-water systems) the cold water temperature, in heating support systems, the return temperature of the heat distribution. The outlet temperature $T_{f,out}$ from the collector (with mass flow $\dot{m}_c$) is above the store temperature during operation of the collector circuit. For homogeneously mixed stores with temperature T_s , the following energy balance applies:

$$(\dot{m}c)_s \frac{dT_s}{dt} = \delta_c (\dot{m}c)_c (T_{f,out} - T_s) + \dot{Q}_h - \delta_l (\dot{m}c)_l (T_s - T_{l,return}) - U_{eff} A (T_s - T_o) \quad (4.112)$$

Since neither the load profile with the operating conditions δ_l nor the auxiliary heating $\dot{Q}_h$ are analytical functions of time, the above differential equation cannot be solved analytically. A simple forward difference method enables the calculation of the store temperature at the time step $n+1$ from the values of the preceding time step n .

$$T_{s,n+1} = T_{s,n} + \frac{\Delta t}{(\dot{m}c)_s} \left(\delta_c (\dot{m}c)_c (T_{f,out,n} - T_{s,n}) + \dot{Q}_{h,n} - \delta_l (\dot{m}c)_l (T_{s,n} - T_{l,return,n}) - U_{eff} A (T_{s,n} - T_{o,n}) \right) \quad (4.113)$$

Far more favourable for solar operation is, however, a thermally non-mixed store. A realistic storage model must take into account a temperature stratification. For this, the store is divided across its height L into several layers. For each layer an energy balance is created, which as above contains the supply of solar heat and auxiliary energy, plus calorific losses and possible heat dissipation by consumers. In addition to these terms, the amount of heat $\dot{Q}_{free}$ is exchanged in layer i with the surrounding layers $i-1$ and $i+1$ by natural convection and thermal conduction. Exact mathematical modelling of the convection current is complex. In the simplest approximation, thermal conduction and convection are summarised in an effective vertical heat conductivity λ_{eff} and the heat flow between layer $i-1$ and i , or from i to $i+1$, is calculated using the Fourier equation. The net heat flow for layer i of height z and cross section A_q results from the difference of the two heat flows.

$$\begin{aligned} \dot{Q}_{free} &= \dot{Q}_{free, i-1 \rightarrow i} - \dot{Q}_{free, i \rightarrow i+1} = -A_q \frac{\lambda_{eff}}{z} (T_{s,i} - T_{s,i-1}) - \left(-A_q \frac{\lambda_{eff}}{z} (T_{s,i+1} - T_{s,i}) \right) \\ &= A_q \frac{\lambda_{eff}}{z} (T_{s,i+1} - 2T_{s,i} + T_{s,i-1}) \end{aligned} \quad (4.114)$$

The vertical temperature stratification in the store is diminished by conduction and convection described by the effective heat conductivity. The heat losses of the upper standby section leads to an increased heat requirement for auxiliary heating. Also the heat transfer to the lower storage sections leads to reduced collector operation, as low collector temperatures cannot be used for warmed up lower storage volumes. The effective heat conductivity with good stores without internal installations according to measurements taken by the University of Stuttgart corresponds to the heat conductivity of water ($\lambda = 0.644 \text{ W m}^{-1} \text{ K}^{-1}$ at 50°C). With good stores with internal heat exchangers, λ_{eff} is higher at $1 - 1.5 \text{ W m}^{-1} \text{ K}^{-1}$. Furthermore, amounts of heat $\dot{Q}_{for}$ are exchanged between the layers by forced convection, depending on the mass flow balance of the store. Since the external storage tank fittings are connected with only a few layers, a separate balance has to be

created for each layer. In the simplest case without external links in a layer, the thermal capacity flows are $(\dot{m}c)_{i-1} = (\dot{m}c)_i$ and the heat flow by forced convection is given by:

$$\begin{aligned}\dot{Q}_{for} &= (\dot{m}c)_{i-1}(T_{i-1} - T_i) + (\dot{m}c)_i(T_i - T_{i+1}) \\ &= (\dot{m}c)_i(T_{i-1} - T_{i+1})\end{aligned}\quad (4.115)$$

For layers with external connections, the external heat flows must also be taken into account, and the total of the mass flows for each layer must be zero. The total energy balance for a layer is then:

$$mc \frac{dT_{s,i}}{dt} = \dot{Q}_{c,i} + \dot{Q}_{h,i} - \dot{Q}_{l,i} - \dot{Q}_{t,i} + \dot{Q}_{free,i-1,i+1} + \dot{Q}_{for,i-1,i+1} \quad (4.116)$$

With the terms of forced and free convection, a coupling of the equations for layer i with the two layers $i-1$ and $i+1$ occurs. The solution to the set of equations is substantially simplified if in each time step only the energy entry from the preceding layer $i-1$ is considered, which leads to a change of temperature in the node i , so the store temperatures $T_{s,i}$ can be calculated downward successively.

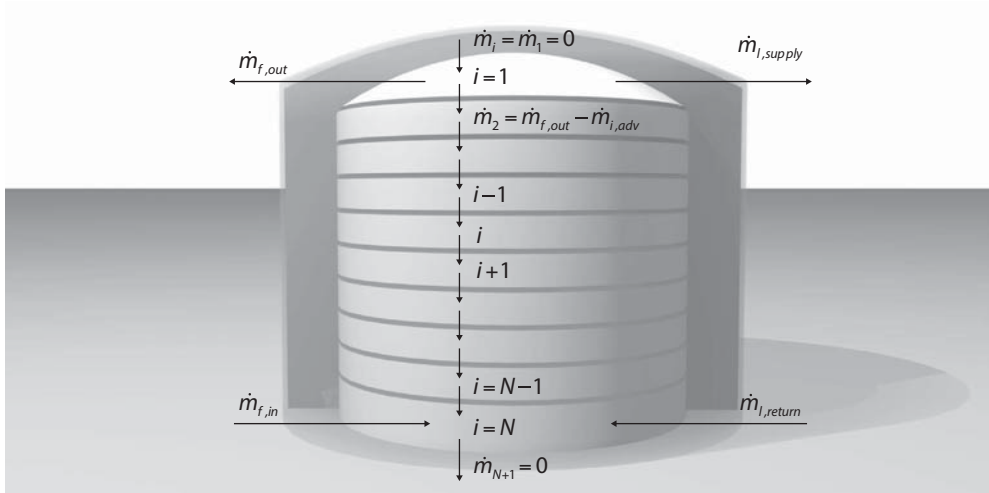


Figure 4.40 Stratified store with N layers.

For the calculation of the energy exchange between the layers from top ($i=1$) to bottom ($i=N$), first the effective mass flows are determined. If, for example, two connections for the collector circuit with mass flow $\dot{m}_c$ and two links for the load circuit with mass flow $\dot{m}_l$ are selected, with flow in opposite directions, the effective mass flow between the layers $\dot{m}_i$ is formed from the difference of the two mass flows. For the first and last layer the effective mass flow is zero.

$$\dot{m}_i = \dot{m}_c - \dot{m}_l \quad \text{for } i=2, N \quad (4.117)$$

$$\dot{m}_i = 0 \quad \text{for } i=1 \text{ and } i=N+1 \quad (4.118)$$

A positive effective mass flow with energy entry from layer $i-1$ in layer i is taken into account by parameter $\delta_i^+ = 1$ (otherwise $\delta_i^+ = 0$). A negative effective mass flow from layer $i+1$, i.e. dominance

of the load mass flow and thus cooling of layer i , is taken into account by parameter δ_i^- . Thus the energy balance for the temperature node i reads:

$$\begin{aligned} (m_i c) \frac{dT_{s,i}}{dt} = & \delta_i^c (\dot{m} c) (T_{f,out} - T_{s,i}) + \dot{Q}_{h,i} - \delta_i^l (\dot{m} c) (T_{s,i} - T_{l,return}) \\ & - U_{eff} A_i (T_{s,i} - T_o) + \delta_i^+ \dot{m}_i c (T_{s,i-1} - T_{s,i}) + \delta_i^- \dot{m}_{i+1} c (T_{s,i} - T_{s,i+1}) \\ & - A_q \frac{\lambda_{eff}}{Z} (T_{s,i} - T_{s,i+1}) \end{aligned} \quad (4.119)$$

with A_i as the exterior surface and A_q as the cross-section area of the respective node, and:

$$\delta_i^c = \begin{cases} 1 & \text{for } i=1 \\ 0 & \text{for } i \neq 1 \end{cases} \quad \text{i.e. collector supply into the top layer,}$$

$$\delta_i^l = \begin{cases} 1 & \text{for } i=N \\ 0 & \text{for } i \neq N \end{cases} \quad \text{i.e. load return into the bottom layer,}$$

$$\delta_i^+ = \begin{cases} 1 & \text{for } \dot{m}_i > 0 \\ 0 & \text{for } \dot{m}_i \leq 0 \end{cases} \quad \text{i.e. energy input from layer } i-1 \text{ to layer } i$$

$$\delta_i^- = \begin{cases} 1 & \text{for } \dot{m}_{i+1} < 0 \\ 0 & \text{for } \dot{m}_{i+1} \geq 0 \end{cases} \quad \text{i.e. energy input from layer } i+1 \text{ to layer } i$$

Example 4.10

Calculation of the temperature distribution for a 750-litre buffer store, loaded with a mass flow from the collector field of 150 kg h^{-1} and a constant collector fluid outlet temperature $T_{f,out}$ of 60°C . The heat transfer of the store walls is to be $0.5 \text{ W m}^{-2} \text{ K}^{-1}$, and the effective heat conductivity $1 \text{ W m}^{-1} \text{ K}^{-1}$. The store diameter d_s is 0.69 m , the height 2.0 m . The ambient temperature T_o and also the initial temperature of the store is 12°C . Auxiliary heating is switched off, and no heat is withdrawn from the store, i.e. all load terms are zero.

For the highest temperature node, the term of the effective heat conductivity is omitted. The collector mass flow at temperature $T_{f,out}$ is brought into the highest node, so $\delta_f^c = 1$. In the first time step the calorific losses to the environment, at the same initial store temperature as ambient

temperature, are still zero. With a time step of 60 seconds, there is a rise in temperature in the highest node of:

$$\Delta T_{s,1} = \frac{\Delta t}{60s} \left(\underbrace{m_j}_{150} \underbrace{c}_{4190 \text{ J/kgK}} \right)_s \left(\underbrace{\delta_1^c}_{1} \left(\underbrace{\dot{m}}_{0.04167 \text{ kg/s}} \right)_k \left(\underbrace{T_{f,out}}_{60^\circ\text{C}} - \underbrace{T_{s,1}}_{12^\circ\text{C}} \right) + \underbrace{\dot{Q}_{h,1}}_0 - \underbrace{\delta_1^l(\dot{m}c)}_0 (T_{s,1} - T_{l,return}) - \underbrace{U}_{0.5 \text{ W/m}^2\text{K}} \underbrace{A_1}_{1.24 \text{ m}^2} \left(\underbrace{T_{s,1}}_{12^\circ\text{C}} - \underbrace{T_o}_{12^\circ\text{C}} \right) + \underbrace{\delta_1^+ \dot{m}c}_0 (T_{s,i-1} - T_{s,i}) + \underbrace{\delta_i^- \dot{m}c}_{0} (T_{s,i} - T_{s,i+1}) \right) = 0.8^\circ\text{C}$$

The second node is warmed by effective thermal conduction and forced convection from the first layer, with the forced convection clearly more significant.

$$\Delta T_{s,2} = \frac{\Delta t}{60s} \left(\underbrace{m_j}_{150} \underbrace{c}_{4190 \text{ J/kgK}} \right)_s \left(\underbrace{\delta_2^c}_{0} \left(\underbrace{\dot{m}}_{0.04167 \text{ kg/s}} \right)_k \left(\underbrace{T_{f,out}}_{60^\circ\text{C}} - \underbrace{T_{s,2}}_{12^\circ\text{C}} \right) + \underbrace{\dot{Q}_{h,2}}_0 - \underbrace{\delta_2^l(\dot{m}c)}_0 (T_{s,2} - T_{l,return}) - \underbrace{U}_{0.5 \text{ W/m}^2\text{K}} \underbrace{A_1}_{0.8696 \text{ m}^2} \left(\underbrace{T_{s,2}}_{12^\circ\text{C}} - \underbrace{T_o}_{12^\circ\text{C}} \right) + \underbrace{\delta_2^+ \dot{m}c}_1 \left(\underbrace{T_{s,i-1}}_{12.8^\circ\text{C}} - \underbrace{T_{s,i}}_{12^\circ\text{C}} \right) + \underbrace{\delta_i^- \dot{m}c}_0 (T_{s,i} - T_{s,i+1}) - \underbrace{A_q}_{0.3739 \text{ m}^2} \frac{\lambda_{eff}}{0.4 \text{ m}} \left(\underbrace{T_{s,2}}_{12.0^\circ\text{C}} - \underbrace{T_{s,1}}_{12.8^\circ\text{C}} \right) \right) = 0.0134^\circ\text{C}$$

The temperatures represented as a function of time clearly show the temperature stratification of the store on loading from above by the collector:

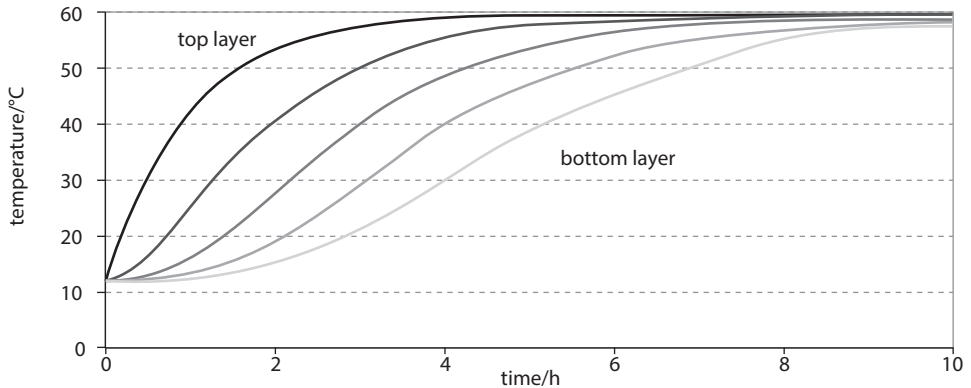


Figure 4.41 Temperature gradient in the store at different heights.

If at the same time a load mass flow (e.g. the half mass flow of the collector of 75 kg h^{-1} at a load return temperature of 12°C) is withdrawn from the store, the temperature gradients flatten accordingly.

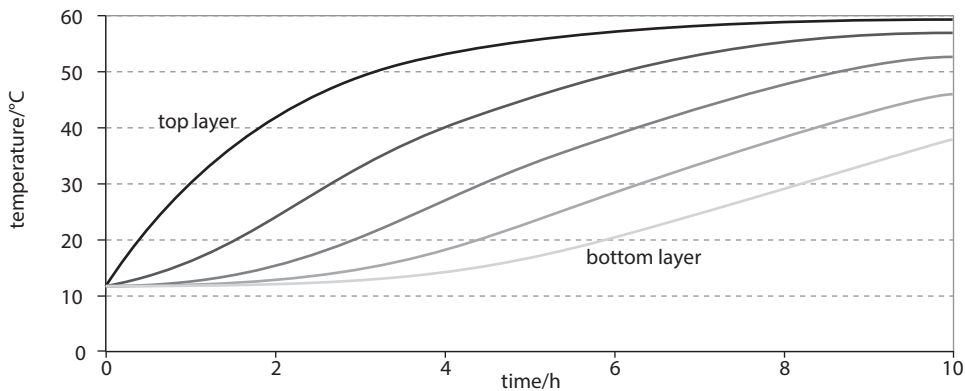


Figure 4.42 Temperature gradient of a layer store with a load mass flow of 75 kg h^{-1} and a collector mass flow of 150 kg h^{-1} .

4.8 Solar air collectors

Solar air collectors are thermal collectors which use air as a heat distribution medium and enable much simpler safety and system engineering by avoiding freeze-protection and overheating problems. In low-energy and passive house development with low heating requirements and air distribution systems for controlled ventilation, a certain renaissance of air heating has taken place, since with small air volume flows covering the heating requirement, thermal comfort can now be ensured.

Solar air collectors are particularly suitable for integration into building shells, since the air collector, being a well-insulated element at the rear, fulfils the thermal demands on external construction components, and at the front is in keeping with usual building materials by a glass cover or by trapezoidal sheet metal constructions. Due to the use of air, possible collector leakages are not problematic even for buildings with integrated warm facade constructions.

However, the large dimensions of the air ducts (due to the low thermal heat capacity of air and the lack of direct storage possibilities for the heat produced) are a negative factor. For reasons of cost and efficiency, special stone stores have not gained acceptance in buildings. More suitable is the activation of storage masses in the building itself (hypocaust systems) or heat storage in conventional hot water tanks by means of an air to water heat exchanger. At present, only a few commercial air collector systems are available, but they are all examples of interesting building-integration solutions.

Collector types

Solar air collectors differ mainly in the type of absorber cover and the air circulation along the absorber. Simple and economical air collector systems for preheating outside air do without a transparent cover and suck air in through fine perforations in the absorber sheet metal. At the same time, the trapezoidal sheet metal construction of the absorber sheet metal serves as a weather shell for the building.

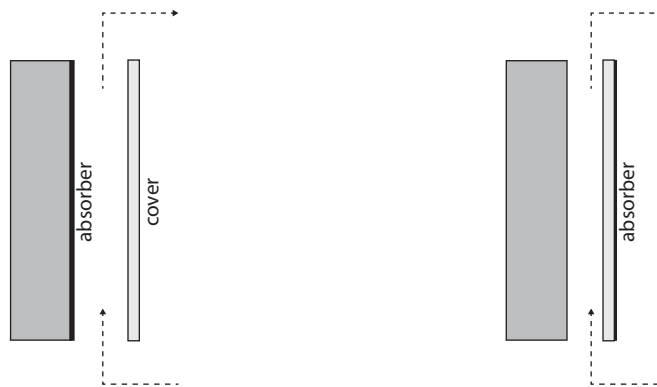


Figure 4.43 A transparently covered air collector as a Trombe wall type (left), and as an underflowed absorber (right).

In transparently covered absorber systems, the air is either led between the cover and absorber (so called Trombe-wall) or, with better thermal characteristics, led under the absorber sheet metal (underflowed absorber). Absorbers flowed-around on both sides, or porous ones, have not gained market acceptance.

The calorific losses of the absorber to the environment depend, as with water collectors, on the cover system. Unglazed absorbers are comparable to swimming pool collectors and are used for applications with small rises in temperature. Calorific losses of underflowed absorbers with a standing air layer between the absorber and cover are calculated like the thermal water collectors. With overflowed absorbers, the flow rate-dependent convective heat transfer coefficients in the air gap enter the calculation. Calculation of the useful thermal energy is completely different, since a thermal conduction problem between the absorber sheet metal and the fluid tube no longer has to be solved; the entire absorber sheet metal transfers heat convectively to the air.

Due to the substantially poorer heat transfer characteristics of air compared to water, absorber ribs are used in most collector systems for surface enlargement. Here the design with continuous, even ribs is the technically simplest and most economical.

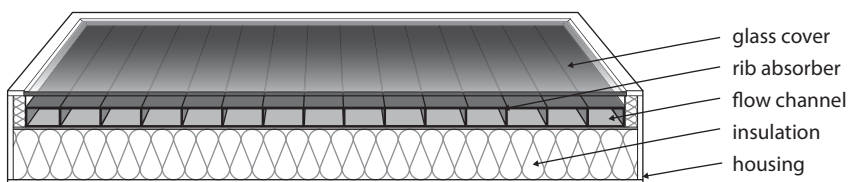


Figure 4.44 Transparently covered air collector with an underflowed absorber.

Systems engineering

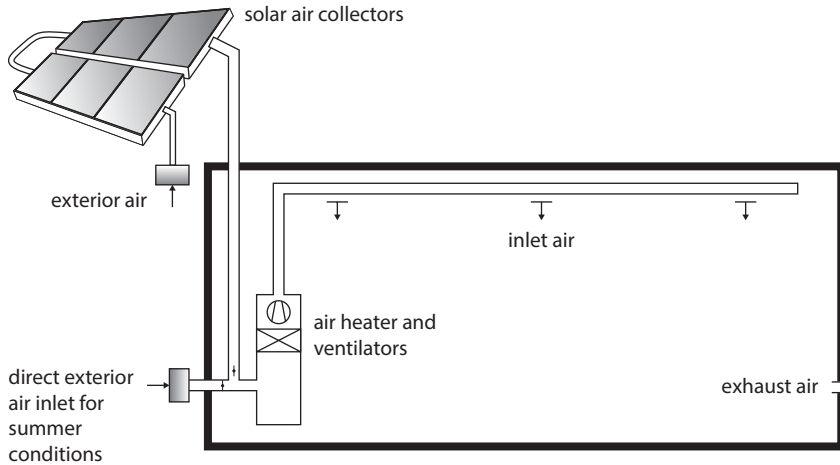


Figure 4.45 System sketch of the preheating of fresh air.

Preheating of fresh air

The most favourable use of air collectors is, as with water-based systems, the preheating of outside air with altogether low absorber and air temperatures and thus small calorific losses. Systems for air preheating are used for heating fresh air in buildings without heat recovery systems. With decentralised air collector fields within the parapet area of a facade with an air intake into the space behind, the entry and exit air distribution system always necessary for heat recovery can be omitted. Even in an energy-optimised low-energy building, a commercial air collector can produce heating energy savings of between 150 kWh m^{-2} in lightweight building and 210 kWh m^{-2} in a building with massive construction and thus high storage capacity under German climatic conditions. Flow rates recommended by the manufacturers are typically $60 \text{ m}^3 \text{ m}^{-2} \text{ h}^{-1}$.

Direct air heating (fresh air/re-circulating air)

Apart from purely fresh air preheating, the warm air produced in air collectors can also be used for direct heating, if the outlet temperatures are at least 5 K above the room temperature. Systems for direct heating are usually operated with lower specific flow rates, to guarantee high rises in temperature even with low irradiances in winter ($20 - 40 \text{ m}^3 \text{ m}^{-2} \text{ h}^{-1}$).

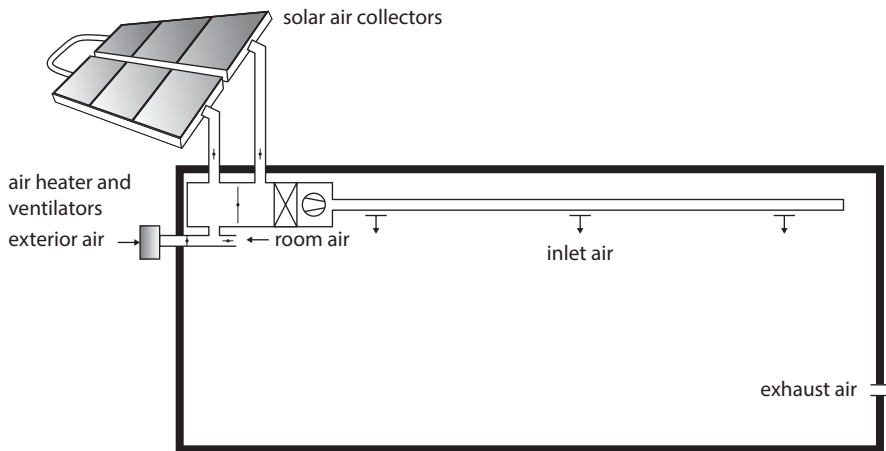


Figure 4.46 System sketch of air heating with air collectors and mixed-air operation.

Indirect air heating with hypocausts

When there is a high heating requirement in rooms, the amounts of air necessary to provide the heating can lead to uncomfortably high injection rates. To improve thermal comfort, the solar-heated warm air can be led into ceiling or wall cavities in a closed cycle (so-called hypocausts), and the room can be warmed by radiant heat.

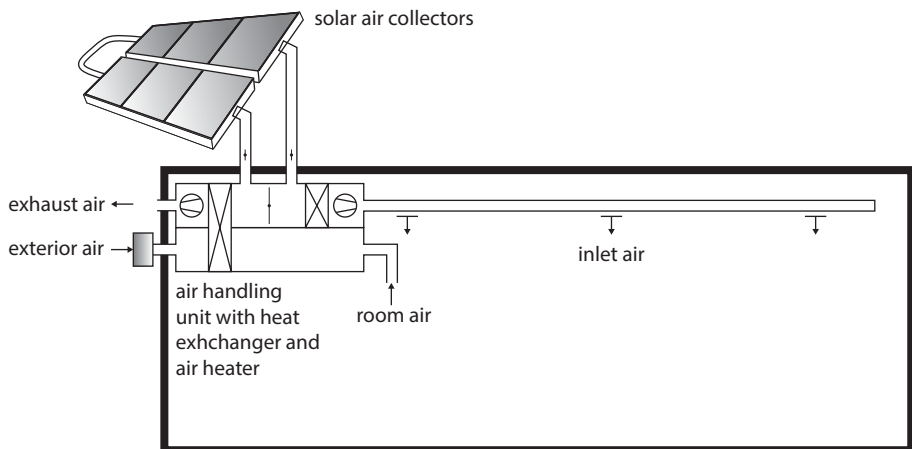


Figure 4.47 Combination of air collectors with heat recovery systems.

Combination with heat recovery systems

The combination of an air collector system with heat recovery from the space exhaust air reduces the heating energy production by the air collector. During fresh air preheating, the air collector

and heat recovery system compete. With an air collector between the outside air inlet and the heat recovery unit, savings of 25 - 60 kWh per square metre of collector are possible. If the collector is placed behind the heat recovery unit to provide additional temperature rises of the room inlet air, the air collector system is energetically more favourable. In a low-energy building, between 60 and 110 kWh of heating energy per square metre of air collector surface can be saved in this way.

4.9 Calculation of the available thermal power of solar air collectors

Temperature-dependent material properties of air

In solar air heaters, rises in air temperature of 10 - 60 K usually occur in the direction of flow, and around 30 K (unribbed) to 10 K (ribbed) perpendicularly to the direction of flow between the absorber and rear wall, depending on the heat exchange surface.

The functions for calculating the kinematic viscosity, heat conductivity, specific thermal capacity and density of the fluid are polynomial fits to the numerical values of dry air at a constant pressure ($p = 10^5 \text{ Pa}$) in the temperature range of 273 - 373 K. The characteristic fluid properties are calculated as a function of the mean fluid temperature $T_{f,m}$ [K].

Kinematic viscosity:

$$\nu = (0.09485 (T_{f,m} - 273.15) + 13.278) \times 10^{-6} \text{ [m}^2/\text{s}] \quad (4.120)$$

Heat conductivity:

$$\lambda = (0.02795 (T_{f,m} - 273.15) + 24.558) \times 10^{-3} \text{ [W/mK]} \quad (4.121)$$

Specific thermal capacity:

$$c_p = 1006 + 0.05 (T_{f,m} - 273.15) \text{ [J/kgK]} \quad (4.122)$$

The density ρ [kg m^{-3}] and the heat expansion coefficient β' [K^{-1}] are determined from the ideal gas equations.

$$\rho = \frac{p}{R T_{f,m}} = \frac{10^5 \text{ N/m}^2}{287.1 \text{ J/kgK} \times T_{f,m}} = \frac{348.3}{T_{f,m}} \quad (4.123)$$

$$\beta' = \frac{1}{T_{f,m}} \quad (4.124)$$

Energy balance and collector efficiency factor

Calculation of the thermal and optical characteristics of the transparent cover of an air collector with an underflowed absorber follows the method already described for water collectors, as the geometry of the glazing, the standing air layer and the absorber is identical. The heat transfer between the absorber and the fluid now takes place only perpendicularly to the absorber level. The geometry-dependent collector efficiency factor can thus be deduced from a stationary energy balance of three temperature nodes.

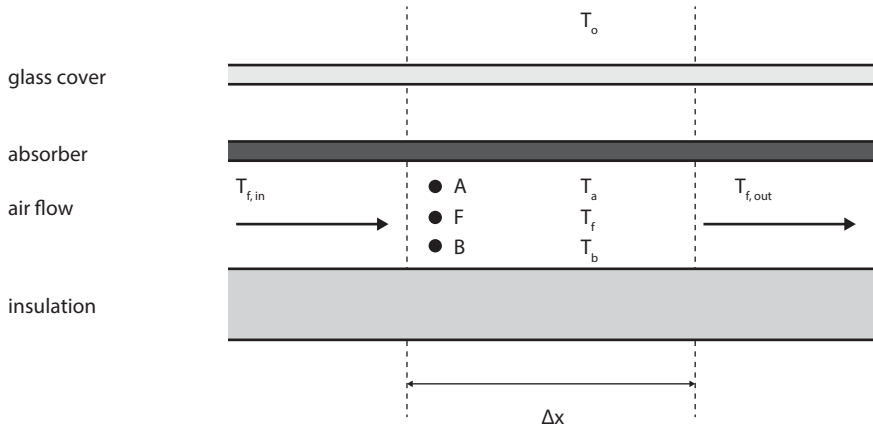


Figure 4.48 Designation of the nodes and temperatures in a solar air heater with an underflowed absorber (absorber width B).

At node A of the absorber, the irradiance G let through with the transmission coefficient τ is absorbed by the absorber sheet metal with the absorption coefficient α . The calorific losses through the transparent cover against the exterior temperature T_o are calculated by the heat transfer coefficient U_f . Convective heat with the heat transfer coefficient $h_{c,a-f}$ is transferred to the heat distribution medium fluid with temperature T_f and heat is exchanged by the radiation heat transfer coefficient $h_{r,a-b}$ with the rear wall of the flow channel with temperature T_b .

$$G(\tau\alpha) - U_f(T_a - T_o) - h_{c,a-f}(T_a - T_f) - h_{r,a-b}(T_a - T_b) = 0 \quad (4.125)$$

At node f of the heat distribution medium fluid, the available power $\dot{Q}_u$ is produced over the collector width B and the distance Δx in the direction of flow. This power consists of the amount of heat transferred convectively from the absorber (with $h_{c,a-f}$) and from the rear (with $h_{c,b-f}$).

$$\dot{Q}_u - h_{c,a-f}B\Delta x(T_a - T_f) - h_{c,b-f}B\Delta x(T_b - T_f) = 0 \quad (4.126)$$

At node B of the rear wall of the flow channel, heat is radiated from the absorber sheet metal with a radiative heat transfer coefficient $h_{r,b-a}$ and at the same time heat is transferred convectively to the fluid. Calorific losses to the ambient air develop via the collector rear with the heat loss coefficient U_b .

$$h_{r,b-a}(T_a - T_b) - h_{c,b-f}(T_b - T_f) - U_b(T_b - T_o) = 0 \quad (4.127)$$

Equation (4.125) and Equation (4.127) are solved for the absorber and rear wall temperatures T_a and T_b as a function of the fluid and ambient temperatures T_f and T_o , and these are inserted into the useful power Equation (4.126). Thus, a conditional equation for the available power $\dot{Q}_u$ is obtained which depends only on the fluid and ambient temperatures,

$$\dot{Q}_u = AF'(G(\tau\alpha) - U_f(T_f - T_o)) \quad (4.128)$$

the collector efficiency factor F' being given by:

$$F' = \left(1 + \frac{U_t}{h_{c,a-f} + \left(\frac{1}{h_{c,b-f}} + \frac{1}{h_{r,a-b}} \right)^{-1}} \right)^{-1} \quad (4.129)$$

and U_t represents the total of the heat transfer coefficients over the front, back and sides. The functional dependency of all characteristics on temperatures and flow rates is discussed in the following section.

Convective heat transfer in air collectors

The collector efficiency factor is mainly dependent on the convective heat transfer coefficients. These vary over a wide range, approximately $5 - 50 \text{ W m}^{-2} \text{ K}^{-1}$, depending on the flow rate (laminar or turbulent current) and the ribbing of the absorber. The order of magnitude of the convective heat transfer coefficient determines the possible thermal efficiency and is a crucial criterion in selecting a type of air collector, whether with flowed-through absorbers, back-ventilated absorbers (for example, PV modules) with flat-parallel gap geometry or ribbed air collectors with very small gap dimensions of a few centimetres.

In active solar energy systems, the heat distribution medium air is moved by fans, so consideration of the forced convection has priority. In back-ventilated facade systems with large gap depths of between 0.1 and 1 m, the flow rates are often so small, however, that the buoyancy term is not negligible, and a proportion of free convection must be calculated.

In such systems with a large distance between the channel-limiting surfaces (cavity facade), heat transfer calculations for a separated individual plate geometry result in better results than for gap geometry. Besides, free convection between flat-parallel panels also dominates the thermal resistance of transparent collector covers, with which commercial air collectors are thermally insulated against outside air. The calculation of the convective heat transfer coefficients in the standing air layer between the absorber and glass cover has been discussed already in the analysis of flat-plate water collectors above.

Asymmetrical heating of the air duct is common to active solar systems; the solar radiation-absorbing absorber side is clearly warmer than the thermally insulated rear, which closes the air duct either against outside air or, with warm facade constructions, from room air. The convective heat transfer coefficient h_c is directly proportional to the dimensionless Nußelt number Nu , which depends both on the material properties of the fluid and on forces of inertia and friction. For a range of geometries and temperature and flow conditions, experimentally determined correlations for Nußelt numbers Nu are available in the literature, which enable calculation of the convective heat transfer coefficient using the heat conductivity of the fluid $\lambda [\text{W m}^{-1} \text{ K}^{-1}]$ and a geometry-dependent characteristic length $L [\text{m}]$.

$$h_c = \frac{Nu \lambda}{L} \quad (4.130)$$

For the usual case of air collector operation with a fan-driven forced flow within the laminar or turbulent flow regime, the characteristic length L is given by the hydraulic diameter of the flow

channel d_h , defined as the relation of the quadruple duct cross section A to the circumference U . For a rectangular channel of height (= panel distance) H and width W the result is:

$$d_h = \frac{4A}{U} = \frac{4WH}{2(W+H)} = \frac{2WH}{(W+H)} \approx 2H \quad \text{for } \frac{H}{W} \ll 1 \quad (4.131)$$

The types of convective heat transfer in air collector systems, within each case one application example, are shown below.

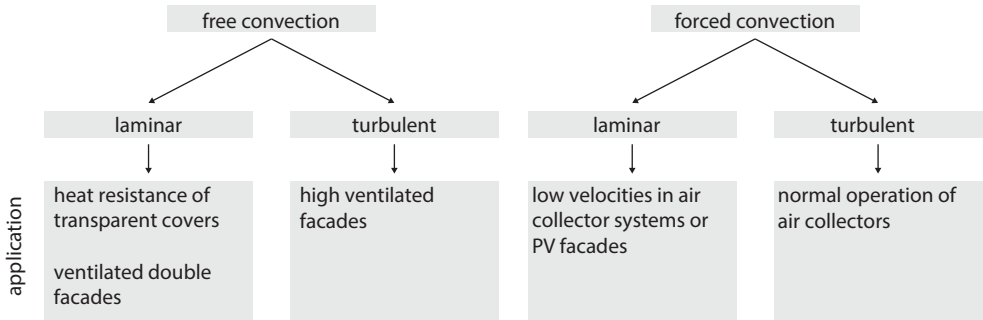


Figure 4.49 Convection mechanisms in air collectors.

In the following section, only classical air collector configurations with flat-parallel panels or ribbed absorbers will be analysed, whose heat transfer is dominated by forced convection (laminar or turbulent).

Laminar flow

For laminar forced flow in air ducts with one-sided heating, a set of Nußelt correlations can be found in the literature. The Nußelt numbers are generally constant in a fully developed laminar flow; with the boundary condition of constant heat flow, a value of $Nu = 5.4$ can be used. In the intake area of the flow channel, the Nußelt numbers are, however clearly higher and are approximated as a function of the flow path x with functions of the type:

$$Nu = c_0 + c_1 \left(\frac{x}{d_h Re Pr} \right)^{-c_2} \quad (4.132)$$

The thermal intake length x_{th} is defined as the length within which the Nußelt number has fallen to a value 1.05 times higher than for the fully developed current. The intake lengths are generally very large for laminar flow.

$$x_{th} = x_{th}^* Re Pr d_h \quad (4.133)$$

According to Merker and Eiglmeier (1999) the factor x_{th}^* is between 0.0335 and 0.053, depending on the boundary condition. The Reynolds number $Re = v d_h / \nu$ is proportional to the flow velocity v , and the Prandtl number $Pr = \nu c_p \rho / \lambda$ to the temperature-dependent material properties of the fluid.

Example 4.11

Calculation of the thermal intake length for an air collector channel with 0.0275 m hydraulic diameter and a flow velocity v of 1 m s⁻¹ at a mean fluid temperature of 40°C. For the boundary condition of an intake developing both thermally and hydrodynamically, $x_{th}^* = 0.053$.

Dynamic viscosity	ν [m ² s ⁻¹]	1.707 × 10 ⁻⁵
Heat conductivity	λ [W m ⁻¹ K ⁻¹]	0.02568
Heat capacity	c_p [J kg ⁻¹ K ⁻¹]	1008
Density	ρ [kg m ⁻³]	1.1123
	Re [-]	1610
	Pr [-]	0.7455

This results in a thermal intake length x_{th} of 1.68 m.

The constants of the Nußelt correlation from Equation (4.132), c_0 , c_1 and c_2 depend on the dimensionless flow path $x^* = x/(Re Pr dh)$ (Shah and London, 1978). According to investigations into flat-ribbed air heaters, the relation of the flow channel width W to the channel height H is used as a parameter (Altfeld, 1985). As a boundary condition, a constant heat flow in the direction of flow and a constant temperature in the circumferential direction, i.e. along the rib, are assumed, and the critical Reynolds number for the transition from a laminar to a turbulent current is set at 3100.

Parameter	c_0	c_1	c_2	Reference
$x^* \leq 0.00325$	0	0.5895	0.5	Shah and London
$0.00325 < x^* \leq 0.045$	0	2.614	0.24	Shah and London
$x^* > 0.045$	5.39	0	-	Shah and London
$W/H = 0.5$	4.11	0.0777	0.8212	Altfeld
$W/H = 1.0$	3.6	0.1028	0.7782	Altfeld

Table 4.11 Coefficients for Nußelt correlations.

An extended equation by Merker and Eiglmeier is likewise taken up in the comparison of the Nußelt correlations represented in Figure 4.47. The average values of the Nußelt numbers, integrated from 0.1 to 2.5 m, are indicated in the legend and hardly differ.

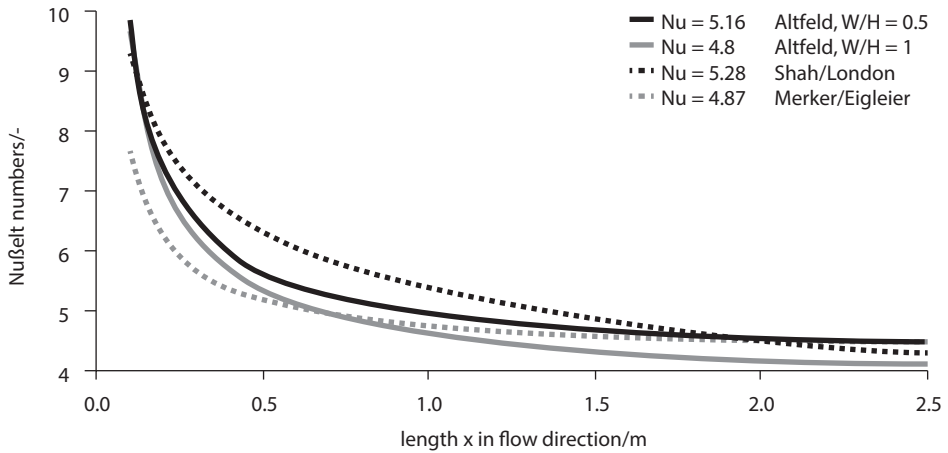


Figure 4.50 Nußelt numbers as a function of the distance x from the intake into an air collector for laminar flow.

Since to calculate the convective heat transfer for the efficiency factor F' only the mean Nußelt number over the collector length is needed and the average values of the different correlations hardly differ, it is recommended to calculate the integral for a correlation and determine directly the mean Nußelt number as a function of the collector length. If, for example, the Altfield correlation is used with:

$W/H = 0.5$

$$Nu = 4.11 + 0.0777 \left(\frac{x}{d_h Re Pr} \right)^{-0.8212} \quad (4.134)$$

the average value can be calculated directly from the integral:

$$Nu_m = \frac{\int_{L_0}^{L_k} Nu(x) dx}{L_k - L_0} = 4.11 + \frac{0.4345}{(d_h Re Pr)^{-0.8212}} (L_k^{0.1788} - L_0^{0.1788}) \quad (4.135)$$

Since the Nußelt numbers for $x \rightarrow 0$ become infinitely large, for the calculation of the mean Nußelt number, integration should only be carried out from, for example, $L_0 = 0.1 \text{ m}$.

Example 4.12

Calculation of the mean Nußelt number for a collector of flow channel length 2.5 m for the collector in Example 4.11 at a flow rate of 1 m s^{-1} .

At a Reynolds number of 1610, laminar flow conditions prevail. At integration limits of 0.1 m and 2.5 m, the mean Nußelt number results in $Nu_m = 5.758$. From this a mean heat transfer coefficient h_c of $5.38 \text{ W m}^{-2} \text{ K}^{-1}$ is calculated.

Forced turbulent flow

With turbulent flow, the influence of the intake area is smaller than with laminar flow. The influence of the boundary conditions and the channel geometry is also small with turbulent flow. As a general equation for calculating the Nußelt numbers with turbulent flow, the modified Petukhov equation for $Re > 3100$ can be used:

$$Nu = \frac{(Re - 1000)Pr \frac{f}{8}}{1 + 12.7 \sqrt{\frac{f}{8}} (Pr^{2/3} - 1)} \left(1 + \left(\frac{d_h}{L_c} \right)^{2/3} \right) \quad (4.136)$$

with $f = (0.791nRe - 1.64)^{-2}$ as the pressure loss coefficient and the term $1 + (d_h/L_c)^{2/3}$ is a correction term for short channels with a channel length L_c .

A simplified Nußelt correlation is indicated by Tan and Charters (1970) for asymmetrically heated flat-parallel panels and a fully developed turbulent current, which likewise contains a correction term for short channels.

$$Nu = 0.0158Re^{0.8} + (0.00181Re + 2.92) \exp(-0.03795L_c/d_h) \quad (4.137)$$

Both correlations produce sufficiently exact values for the convective heat transfer coefficients.

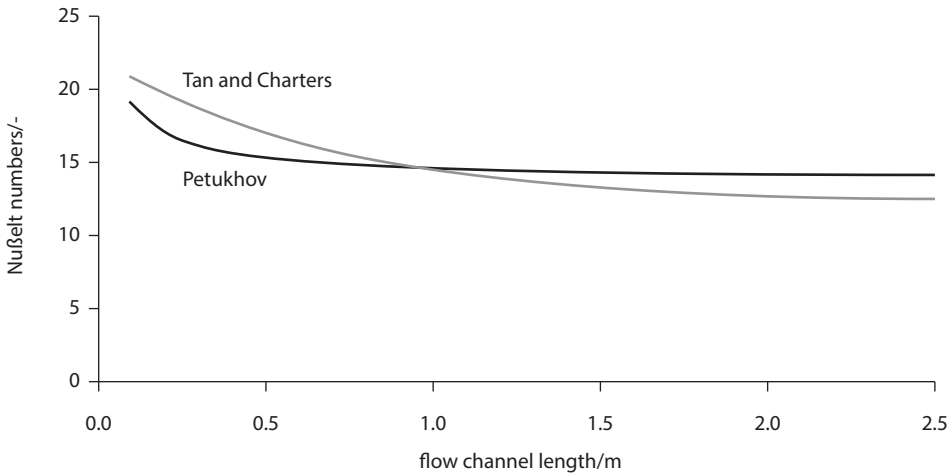


Figure 4.51 Nußelt correlations for forced convection.

With the above Nußelt correlations, the heat transfer coefficient of ribbed absorbers can be calculated.

Example 4.13

Calculation of the convective heat transfer coefficient h_c for the 2.5 m long collector in Example 4.11 at a flow rate of 2.5 m s^{-1} .

The Reynolds number for the flow channel of 0.0275 m hydraulic diameter is 4026. This results in a pressure loss coefficient $f = 0.041$ and a Nußelt number 14.53, based on the Petukhov

equation. The convective heat transfer coefficient for a turbulent current is $13.57 \text{ W m}^{-2} \text{ K}^{-1}$, i.e. almost three times as high as with the laminar flow.

Rib efficiency

The heat transfer coefficient h_c calculated so far refers to the complete heat transfer surface of the air duct confinement surfaces A_{total} , which for each flow channel is composed of the absorber underside and channel rear wall of width W and the two lateral ribbed surfaces of height H . For a unit length in the direction of flow, the total area of the flow channel is:

$$A_{total} = 2(W + H) \times 1 \quad (4.138)$$

Since the surface reference for the available power calculation is the absorber surface, a surface normalisation for the convective heat transfer coefficient must be carried out. Furthermore, it is taken into account, by means of the so-called rib efficiency, that the temperatures of the ribs drop from the absorber temperature level.

The rib efficiency is deduced similarly to the temperature distribution calculation of the absorber sheet metal with water-throughflowed collectors, assuming ideal thermal contact of the rib to the absorber underside, a constant convective heat transfer coefficient and neglect of heat dissipation at the tip of the rib.

$$\eta_{Ri} = \frac{\tanh(mH)}{mH} \quad (4.139)$$

with

$$m = \sqrt{\frac{h_c}{\lambda_{Ri}} \frac{U_{Ri}}{A_{cross}}} \quad (4.140)$$

and h_c as the convective heat transfer coefficient related to the total area and λ_{Ri} as the heat conductivity of the sheet metal. The relation of the circumference of the rib U_{Ri} of rib thickness t to the cross section of the rib A_{cross} results from:

$$\frac{U_{Ri}}{A_{cross}} = \frac{2(H+t)}{Ht} \quad (4.141)$$

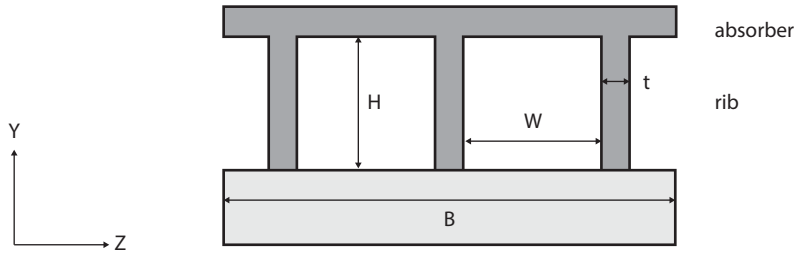


Figure 4.52 Geometry of a ribbed absorber.

The surface relation of the heat-transferring ribbed surface $A_{Ri} = 2H$ to the entire channel confinement surface A_{total} is defined with β , following Altfeld (1985).

$$\beta = \frac{A_{Ri}}{A_{total}} = \frac{2H}{2(W+H)} = \frac{H}{(W+H)} \quad (4.142)$$

and the relation of the surface between the ribs A_W to the absorber surface A is defined with γ .

$$\gamma = \frac{A_W}{A} = \frac{W}{W+t} \quad (4.143)$$

The heat flux from the absorber underside and rib to the fluid $\dot{Q}_a$, related to the absorber surface A , is described by the heat transfer coefficient $h_{c,a-f}$ which results from the rib efficiency and the surface factors from the already calculated h_c .

$$\frac{\dot{Q}_a}{A} = h_{c,a-f} (T_a - T_f) \quad (4.144)$$

with

$$h_{c,a-f} = \frac{\gamma}{1-\beta} (1 + \beta(2\eta_{Ri} - 1)) h_c \quad (4.145)$$

For the rear of the flow channel, the convective heat transfer coefficient is reduced by the factor γ , i.e. by the effective heat-transferring surface portion of the rear.

$$h_{c,b-f} = \gamma h_c \quad (4.146)$$

The two heat transfer coefficients $h_{c,a-f}$ and $h_{c,b-f}$ can be inserted into the equations of the collector efficiency factor.

Example 4.14

Calculation of the convective heat transfer coefficients $h_{c,a-f}$ and $h_{c,b-f}$ of the underflowed ribbed air collector already considered, with flow rates of 1 m s^{-1} or 2.5 m s^{-1} . To calculate the surface factors β and γ , the exact geometry of the flow channel is needed:

Rib height H of the channel:	0.028 m
Rib thickness t :	0.0014 m
Rib distance (= channel width) W :	0.027 m
Heat conductivity of the rib sheet metal λ_{Ri} :	$238 \text{ W m}^{-1} \text{ K}^{-1}$

$$\beta = 0.5091, \gamma = 0.9507, U_{Ri}/(\lambda_{Ri} A_{cross}) = 6.3$$

Flow velocity $/\text{m s}^{-1}$	h_c $/\text{W m}^{-2} \text{ K}^{-1}$	m	Rib efficiency η_{Ri}	$h_{c,a-f}$ $/\text{W m}^{-2} \text{ K}^{-1}$	$h_{c,b-f}$ $/\text{W m}^{-2} \text{ K}^{-1}$
1	5.38	5.82	0.991	15.6	5.1
2.5	13.57	9.25	0.978	39.1	12.9

Thermal efficiency of air collectors

With the convective heat transfer coefficients determined above, the heat transfer coefficient for radiation h_r and the total heat transfer coefficient U_t between the absorber and environment, the efficiency factor F' can be calculated using Equation (4.129). The heat transfer coefficient for radiation is calculated on the simplified assumption that the absorber and gap rear wall can be described as infinitely expanded flat-parallel surfaces, and the ribs are not considered.

Due to the typically small emissivities of the duct confinement surfaces ($\epsilon_a, \epsilon_b = 0.04 - 0.1$) and the small temperature difference between the surfaces, typically 5 K, these assumptions produce sufficiently exact results (T_a and T_b in kelvin).

$$h_r = \frac{\sigma}{\frac{1}{\epsilon_a} + \frac{1}{\epsilon_b} - 1} (T_a^2 + T_b^2) (T_a + T_b) \quad (4.147)$$

The heat transfer coefficient U_t is calculated, as with the flat-plate collector, in simplified fashion as the total of the front, side and rear wall losses, i.e. of the temperature node of the absorber to the environment; the side and rear wall losses U_s and U_b are temperature-independent and can be set as constant.

$$U_t = U_f + U_b + U_s \quad (4.148)$$

The heat transfer coefficient through the transparent cover takes into account wind influences ($h_{c,w}$) and radiation losses to the sky (h_{g-sky}) and is, as with the flat-plate collectors, calculated iteratively as a function of the glass cover temperature T_g .

$$U_f = \frac{1}{\frac{1}{h_{c,a-g} + h_{r,a-g}} + \frac{1}{h_{c,w} + h_{r,g-sky}}} \quad (4.149)$$

Since all the heat transfer coefficients are temperature-dependent, first temperatures for all surfaces and the mean fluid temperature must be given. With the initial temperature field, all coefficients are then calculated, the collector efficiency factor F' is determined and the available power is calculated as a function of the fluid input temperature:

$$\dot{Q}_n = AF_R \left(G(\tau\alpha) - U_t (T_{f,in} - T_o) \right) \quad (4.150)$$

with

$$F_R = \frac{\dot{m}c_p}{AU_t} \left(1 - \exp \left(- \frac{U_t F' A}{\dot{m}c_p} \right) \right)$$

From the available power, the mean temperatures for the absorber T_a , flow channel rear wall T_b , fluid T_f and glass cover T_g are then calculated similarly to water-throughflow collectors.

$$\bar{T}_a = T_{f,in} + \frac{\dot{Q}_n}{AU_t F_R} (1 - F_R)$$

$$\bar{T}_f = T_{f,in} + \frac{\dot{Q}_n \left(1 - \frac{F_R}{F'} \right)}{AF_R U_t}$$

$$T_g = T_a - \frac{U_f (T_a - T_o)}{(h_{c,a-g} + h_{r,a-g})}$$

The temperature of the flow channel rear wall is calculated by resolving the energy balance Equation (4.127) with the mean fluid temperature used for T_f .

$$T_b = \frac{h_{c,b-f} T_f + U_b T_o - h_{r,b-a} T_a}{h_{c,b-f} + U_b - h_{r,b-a}} \quad (4.151)$$

With these temperatures, in the next iteration, new heat transfer coefficients are calculated. The iteration is continued until the change in the temperature field becomes negligibly small.

Example 4.15

Calculation of the available power, the outlet temperatures and the thermal efficiency for a facade-integrated air collector at 800 W m^{-2} irradiance and an ambient temperature of 10°C . The ambient temperature equals the inlet temperature $T_{f,in}$ in the collector. The air duct geometry corresponds to

the examples already calculated with a collector length of 2.5 m, and the heat transfer coefficient of the rear U_b is constant with $0.65 \text{ W m}^{-2} \text{ K}^{-1}$ (side losses ignored).

No	Boundary condition	U_t / $\text{W m}^{-2} \text{ K}^{-1}$	F' /-	F_R /-	$\dot{Q}_u/A$ / W m^{-2}	$\bar{T}_a$ / K	$\bar{T}_f$ / K	$T_{f,out}$ / K	η [-]
1	$v = 1 \text{ m s}^{-1}$ $v_{wind} = 3 \text{ m s}^{-1}$ $\epsilon_a = 0.9$	6.6	0.70	0.59	375	49.9	26.0	40.2	0.47
2	$v = 1 \text{ m s}^{-1}$ $v_{wind} = 3 \text{ m s}^{-1}$ $\epsilon_a = 0.1$	4.2	0.79	0.69	442	56.9	28.8	46.0	0.55
3	$v = 2.5 \text{ m s}^{-1}$ $v_{wind} = 3 \text{ m s}^{-1}$ $\epsilon_a = 0.9$	6.2	0.87	0.80	514	30.3	18.3	26.1	0.64
4	$v = 2.5 \text{ m s}^{-1}$ $v_{wind} = 1 \text{ m s}^{-1}$ $\epsilon_a = 0.9$	5.4	0.89	0.82	528	30.8	18.5	26.6	0.66
5	$v = 2.5 \text{ m s}^{-1}$ $v_{wind} = 1 \text{ m s}^{-1}$ $\epsilon_a = 0.1$	3.55	0.92	0.88	562	32.1	19.9	27.7	0.70

Table 4.12 Result table for air collector efficiencies.

From the results, the efficiency rise with the change of the flow from laminar (No. 1, 2) into turbulent conditions (No. 3-5) is clearly evident. The first two simulations with laminar flow differ by the emission coefficient of the absorber ϵ_a , which in the case of the selective coating is 0.1 and with the black absorber 0.9. Due to the selective coating, the heat transfer coefficient falls from $6.6 - 4.2 \text{ W m}^{-2} \text{ K}^{-1}$ and the efficiency rises by 17%. At higher flow rates of 2.5 m s^{-1} , the influence of the outside wind velocity was examined. A reduction of 3 to 1 m s^{-1} leads to a reduction of the U_t value of $0.8 \text{ W m}^{-2} \text{ K}^{-1}$, and an efficiency improvement of 3%. The selective coating brings a further 6% efficiency improvement.

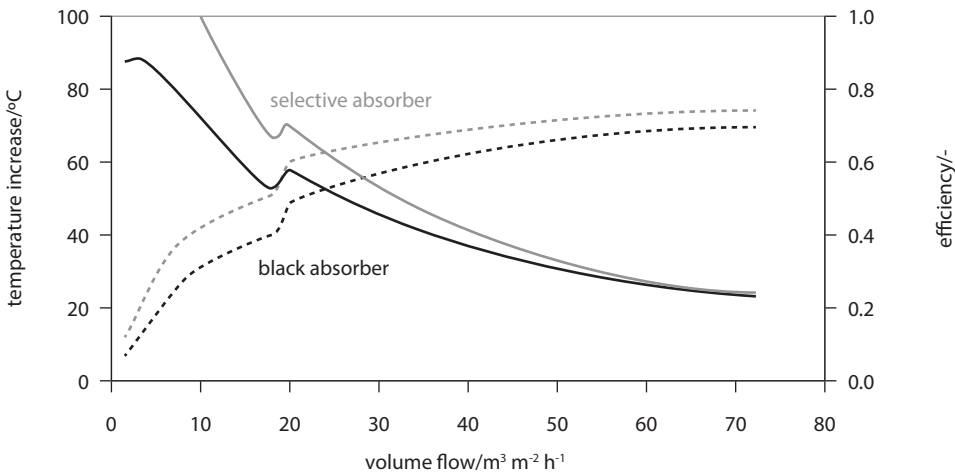


Figure 4.53 Rise in temperature (continuous lines) and efficiency (broken lines) of an air collector.

The boundary conditions for these temperature-rise or efficiency calculations as a function of the specific flow rate (in $\text{m}^3 \text{h}^{-1}$ per m^2 of collector surface) are irradiance of 800 W/m^2 , 10°C ambient temperature and 3 m s^{-1} wind velocity.

4.10 Design of the air circuit

The volumetric air flow of the collector depends on the desired application. While during pure fresh air preheating, high flow rates of $> 60 \text{ m}^3 \text{m}^{-2} \text{h}^{-1}$ with good thermal efficiency are favourable, direct heating or hypocaust applications require high rises in temperature and thus low specific flow rates ($20 - 40 \text{ m}^3 \text{m}^{-2} \text{h}^{-1}$).

With direct heating applications, the outlet temperature determined by the surface-specific flow rate must be limited for reasons of comfort, in buildings to 45°C , in industrial applications to a maximum of 60°C . The temperature limitation can take place either by flow rate regulation or by the addition of cold air.

Collector pressure losses

For interconnecting the collectors it is best to select as long a collector series as possible, to reduce the connection channels and thus system costs. At the same time, by series connection at a given total volume flow $\dot{V}$ the flow velocity v in the air ducts and thus the convective heat transfer is increased, since the throughflowed duct cross-section surface A_{cross} is small.

A limiting factor for the number of collectors switchable in series is the pressure loss by friction Δp_R , which at laminar flow rises linearly and with turbulent flow as a square of the flow velocity. The friction pressure loss is calculated from the coefficient of friction λ , the length l , the hydraulic diameter d_h and the dynamic pressure $\rho/2 v^2$.

$$\Delta p_R = \frac{\lambda l}{d_h} \frac{\rho}{2} v^2 \quad (4.152)$$

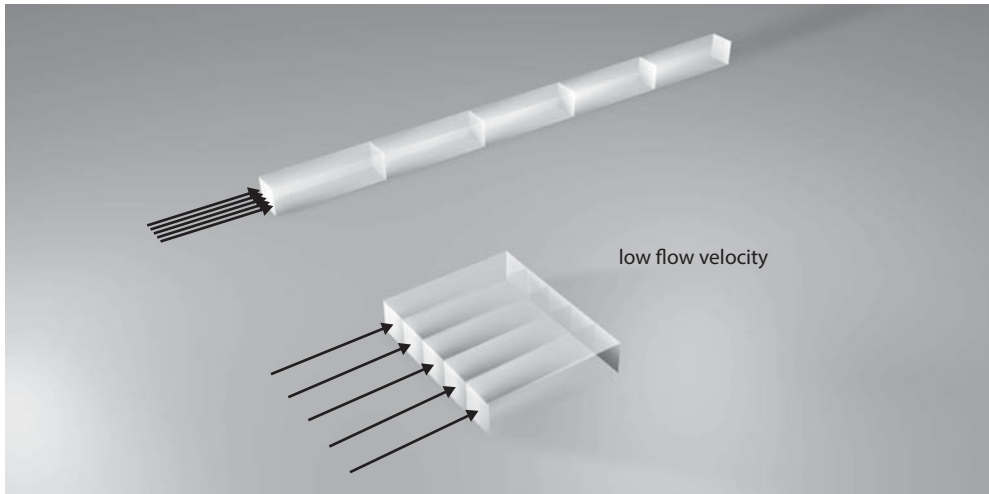


Figure 4.54 Series and parallel connection of collectors.

With laminar flow, the coefficient of friction λ is in inverse proportion to the Reynolds number Re , so the result is the linear connection between the pressure loss and the flow rate.

$$\lambda = \frac{64}{Re} = \frac{64\nu}{vd_h} \quad (4.153)$$

With turbulent flow, the coefficient of friction depends on the roughness of the wall. The ducts of a commercial air collector have, for example, an absolute roughness of $\varepsilon = 0.15 \text{ mm}$, comparable to that of galvanised steel tubes. The pipe friction number is determined iteratively as a function of the roughness and the Reynolds number both for hydraulically smooth and rough pipes:

$$\frac{1}{\sqrt{\lambda}} = -2 \log \left(\frac{2.51}{Re\sqrt{\lambda}} + \frac{\varepsilon}{3.72d_h} \right) \quad (4.154)$$

Example 4.16

Calculation of the pressure loss of the above air collector with $d_h = 0.027 \text{ m}$ at flow velocities of 1 m s^{-1} , 2.5 m s^{-1} and 5 m s^{-1} and a mean air temperature of 40°C .

Velocity v / m s^{-1}	Reynolds number Re /-	Friction coefficient λ /-	Pressure drop Δp_R / Pa m^{-1}
1	1583	0.04	0.82
2.5	3957	0.045	5.8
5	7914	0.038	19.6

4.11

In air duct systems, besides pipe friction losses, consideration must also be given to pressure losses through individual resistances Δp_z resulting from changes of direction and cross section and from branchings in the duct system. For turbulent flow conditions, these flow resistances are caused by the formation of eddies and are proportional to the square of the mean flow velocity. The dimensionless coefficient of drag ξ is tabulated in heating and climate-technical manuals for all usual components, and varies over a wide value range of approximately 10^{-2} (steady narrowings or widenings) to 10 (non-return valve, branches, etc.).

$$\Delta p_z = \xi \rho \frac{v^2}{2} \quad (4.155)$$

The dimensions of the installations and of the air ducts result from given values for maximum flow velocities, which should not be exceeded due to sound-related problems and to high pressure losses as shown in Table 4.12:

Duct type	Velocity
Small air systems in residential buildings up to 500 m ³ h ⁻¹ , tubes behind inlet/outlet valves	3 - 4 m s ⁻¹
Medium air systems, connecting ducts and air distribution ducts	4 - 8 m s ⁻¹
Large air systems and collecting channels	8 m s ⁻¹

Table 4.13 Typical values for air flow velocities.

At a given flow rate and a given maximum velocity, the duct cross-section surface and the pipe diameter are calculated by $A_{\text{cross}} = \pi d^2 / 4 = \dot{V} / v$. Typical distributions of pressure losses in large systems with short distribution pipes in the building are about 50% in the collector field itself, 15% in the collecting ducts and 35% in the building.

After determination of the total volume flow and the pressure loss calculation of the duct with the highest losses, the electrical power required by the fan can be determined. This is proportional to the flow rate and pressure loss, and depends on the fan and motor efficiency.

$$P_{\text{el}} = \frac{\dot{V} \Delta p}{\eta} \quad (4.156)$$

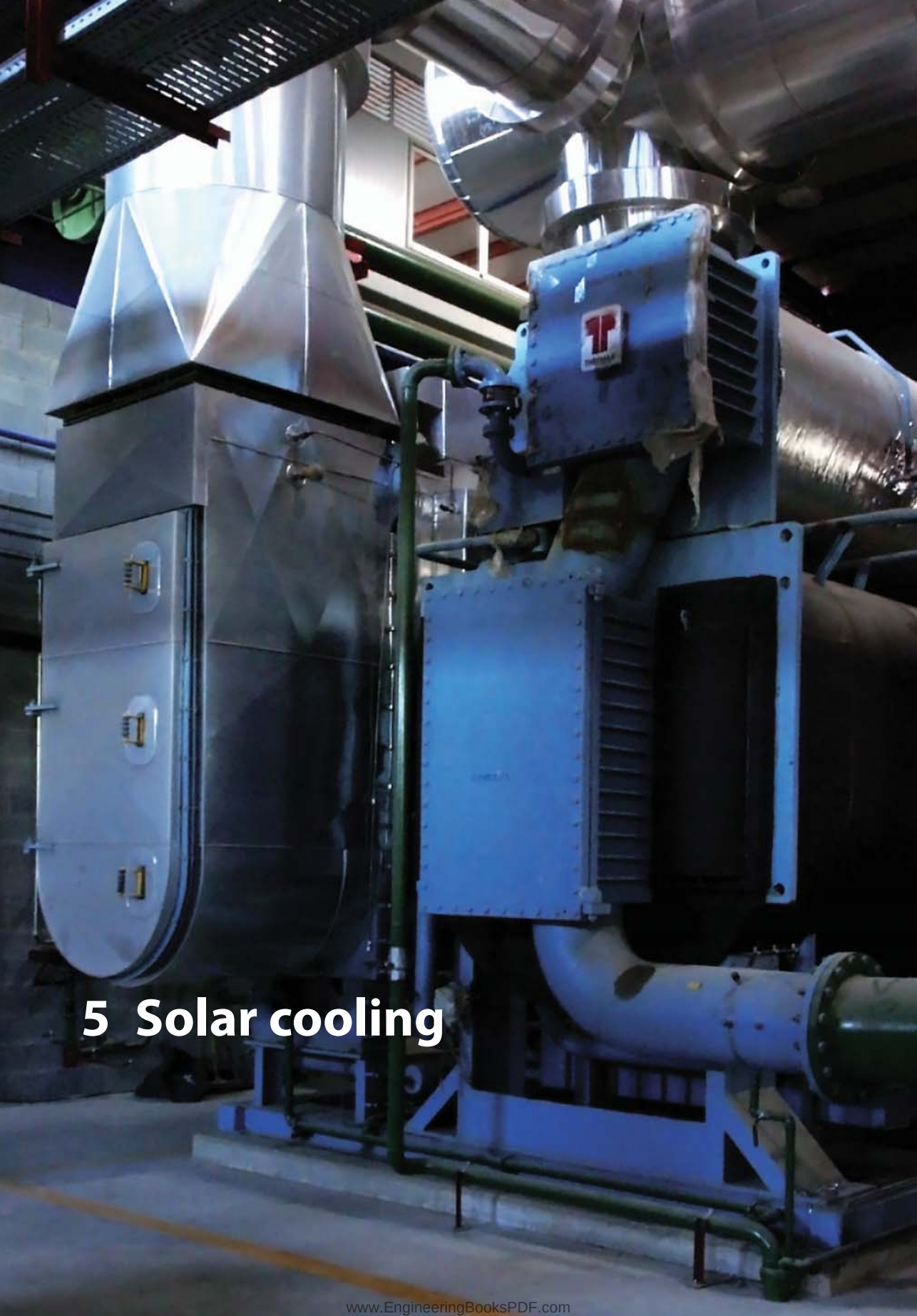
Volume flow rate /m ³ h ⁻¹	Ventilator efficiency η_V /-	Motor efficiency η_M /-	Total efficiency η /-
up to 300	0.4 - 0.5	0.8	0.32 - 0.4
400 - 1 000	0.6 - 0.7	0.8	0.48 - 0.56
2 000 - 5 000	0.7 - 0.8	0.8	0.56 - 0.64
6 000 - 10 000	up to 0.85	0.82	up to 0.7

Table 4.14 Typical values for ventilator efficiencies.

The total pressure difference produced by the fan is used up in the connected duct system by duct friction and individual resistances. With parallel air strands, duct cross sections and installations must be dimensioned in such a way that the pressure loss is as high as in the main line. If this basic requirement of the calculation is not adhered to, the flow rates adjust in such a way on operation of the system that the requirement of equal pressure losses is fulfilled; duct systems are therefore self-regulating. This automatic modification of the flow rates has the consequence that in inaccurately calculated duct systems the design flow rates do not flow in the individual parallel strands, and the modified total volume stream can lead to power modifications in central humidifiers, air heaters and the like.

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5 Solar cooling

Figure 5.1 (previous page): An absorption cooling machine in Cerdanyola near Barcelona, Spain (Photo: Eicker).



Figure 5.2 Vacuum tube collectors on the Presseamt in Berlin, Germany (Photo: Eicker).

To cover the air-conditioning and cooling requirement in buildings, both electrically and thermally driven cooling machines are available. Conventional refrigeration technology is dominated by compression chillers, where about 82 million room air-conditioning units are built annually worldwide, plus about 35 million mobile systems (Jarn, 2009). Medium and large capacity absorption cooling machines are manufactured in comparatively small numbers, with around 10 000 systems produced per year, of which 85% are produced in Asia. In Germany, a total of around 3000 absorption cooling machines are operating, with a cooling capacity of around 1200 MW. The proportion of absorption cooling machines worldwide is at around 8% (Lamers et al., 2008).

The market penetration in Europe is rather low (8.6 million units in 2008), but it has a high growth rate, increasing from 3 m² air conditioned space per person in the year 2000 to 6 m² per person in the year 2020. For every thousand people in Germany, there are 0.6 electrical air-conditioning units installed, in Spain 14 and in Italy 12 units per thousand inhabitants (Rolles, 2004).

Air-conditioning technology in moderate climates such as Germany is almost exclusively utilised in the commercial sector. This corresponds to around 40 000 GWh of electricity consumption for office air conditioning alone. According to the annual reports on office service charges of the consulting company Jones Lang LaSalle in Germany, about half of all offices now have air conditioning. Other authors indicate smaller percentages of 20 - 30% in offices, of 30% in both shops and hotel rooms, 5% in both supermarkets and hospital rooms and 1 - 2% in retirement homes (Rolles, 2004). According to the Austrian Energy Agency (2005), 90% of new office buildings in Europe are air-conditioned.

In the USA, around 76% of offices are air-conditioned (EIA, 2000); in Japan almost 100% of office buildings and 85% of residential buildings are air-conditioned. Due to climatic changes in Europe as well as the general trend towards more comfort, the IEA expects a growth of cooled office area of 65 million m² per year until the year 2020 in Europe or 12.7% per annum. The highest growth rates with up to 50% are expected in Italy and Spain.

Solar cooling applications are now entering the market. About 1000 solar cooling systems have been installed worldwide, many within recent years. Especially large systems with several thousand square metres of collector surface are economically attractive and can be financed using a contracting model.

In the whole of Australia there are about 10 installed solar cooling projects, most of which are with parabolic trough collectors. Also in China there are only a few installed solar cooling projects. In contrast, more installations can be found in countries with large solar thermal markets despite rather low cooling requirements, e.g. 20 installations mainly in office buildings in Austria. Thirty installations are now running in Italy, most of which are absorption cooling machines.

At the same time, there are several million square metres of solar collector area installed each year in China alone. The solar thermal industry is one of the drivers of solar thermal cooling, as in heating-dominated climates there is always an excess of heat in summer. Also waste heat from cogeneration is often available at low cost in summer.

Thermal cooling can replace electric compression cooling and reduce the summer loading of the electricity grid. Not only solar energy but also surplus heat from biomass cogeneration, Stirling motors, micro-gas turbines or other waste heat sources offer environmentally friendly alternatives to electricity-operated cooling. The greatest barriers for the spread of solar cooling are the high investment costs alongside low electricity prices.

Solar technologies in buildings can supply photovoltaic electricity for compression chillers or solar thermal heat for sorption chillers. The necessary surface area of the photovoltaic generator can be easily estimated via the average summer electricity requirement for air-conditioning of administrative buildings. For cooling energy requirements of typically 30 - 60 kWh m⁻² in moderate climates, the amount of electricity needed for compression chillers is about 10 to 20 kWh m⁻². With an annual PV generation efficiency of 15%, the generated PV electricity per square metre is between 105 kWh per square metre for a location with 700 kWh m⁻² irradiance during summer months and 210 kWh m⁻² for a high irradiance location with 1400 kWh m⁻² during the cooling season. This results in a conditioned office area of 5 - 21 m² for each square metre of PV installed depending on cooling demand and solar production. For 1000 m² of office area, about 50 - 200 m² of PV surface are necessary for air conditioning alone. In addition there are electricity requirements for lighting and equipment.

If thermal cooling systems are used with an energy efficiency ratio (EER) (cold to heat) of 0.7 and a solar thermal efficiency of 40%, the low irradiance location with a cooling demand of 30 - 60 kWh m⁻² gives a solar thermal cooling yield of 196 kWh m⁻², which covers between 3.3 and 6.5 m² of office surface, slightly less than the PV technology. In this simple estimation storage is not yet considered.

5.1 Introduction to the technologies

The sorption cooling market is dominated by absorption refrigeration technologies with the working pairs water-lithium bromide (LiBr) or ammonia-water, which produce cooling energy by a closed cycle process.

The principle of operation for closed cycle processes such as compression, absorption or adsorption is the same: in each case, a refrigerant (e.g. water, ammonia, carbon dioxide) is evaporated at low pressure and low temperature. The heat needed for this evaporation or boiling process is taken from the rooms to be cooled, and this is designated as usable cooling energy. In order for the refrigerant to condense once again at ambient or cooling tower temperatures, the pressure must be increased to bring the refrigerant vapour molecules close together. In compression cycles this pressure increase is done mechanically. For the sorption processes two steps are required: in absorption machines the gaseous refrigerant is dissolved in a solution and then the solution is heated to drive out the refrigerant at high pressure. In adsorption technology the refrigerant water is physically adsorbed to a solid such as silica gel or zeolite, also with subsequent heating and expulsion of the coolant at high pressure. With the increasing accumulation of the refrigerant water molecules, the heat from adsorption approaches zero, so that only condensation heat has to be removed. Chilled water of 6°C can be produced by the cyclic process with water as refrigerant in closed adsorption chillers.

Open sorption plants use the supply air directly as a cooling carrier. The physical adsorption of water by silica gel or absorption in salt solutions such as lithium chloride or calcium chloride serves to dry the air in this process. Thereafter, cooling takes place by direct evaporative humidification of the dried air, which has been pre-cooled with humidified room exhaust air via a heat exchanger. The thermal driving energy is necessary for the regeneration of the sorbent, i.e. for the desorption of the water. With open sorption, the process limits air temperatures to about 16°C minimum, so the area of application is for air conditioning. In this process, too, the drive temperatures can be very low (60 - 70°C).

Energy efficiency ratios

Single effect absorption chillers produce about 0.6 - 0.7 kW of cooling power per kW of heat (EER = 0.6 - 0.7). In water/lithium bromide absorption systems, double effect generators with a high-temperature section are available on the market for steam or natural gas heating and use of condensation heat for a low-temperature generator. In a double effect process, the EER rises from 1.1 to 1.3. Triple effect cycles can give EERs of 1.8 - 2.1.

With ammonia-water absorption chillers, two-stage machines are not available due to extremely high system pressures, so the performance figures remain limited to approximately 0.6. The EER of closed adsorption chillers depends on the available coolant temperature and can also achieve values between 0.6 and 0.7. With open sorption air-conditioning system, the state of the outside air influences the possible EER. With dry outside air, the air-conditioning system can be operated purely by evaporative cooling, so no thermal energy is necessary and the EER approaches infinity. With very damp outside air, significant amounts of heat are needed to dry the sorption material. Typical EERs are between 0.5 and 1.0.

For a comparison of system types, the utilised primary energy per kW cooling capacity can be looked at (primary energy resource factor according to the European Standard prEN 15316). Electrical compression chillers with a typical EER of 3.0 require 0.33 kW electrical power (Q_{el}) of each kW of cooling (Q_{cool}). With a primary energy factor (PEF_{el}) of 2.7 for electricity generation, this results in much less than 1 kW of primary energy for each kW of cooling power, i.e. the primary energy ratio is 1.1.

$$PER_{compression} = \frac{Q_{cool}}{Q_{el} \cdot PEF_{el}} = \frac{1kW}{0.33kW \cdot 2.7} = 1.1 \quad (5.1)$$

A single effect thermal cooling system with an EER of 0.7 requires 1.42 kW of heat input. If this heat is provided by conventional energy sources, for example, gas with a primary energy factor of about 1.1, the primary energy ratio is 0.66 even when no auxiliary electricity consumption is considered, i.e. worse than electrical cooling.

$$PER = \frac{Q_{cool}}{Q_{el} \cdot PEF_{el} + Q_{h,add} \cdot PEF_{gas}} = \frac{1kW}{0kW + 1.42kW} = 0.66 \quad (5.2)$$

To reach higher primary energy ratios than electrical chillers, less than 1 kW of primary energy for heating should be used, i.e. solar or waste heat sources are needed to improve the energy efficiency. Also in thermal cooling systems auxiliary cooling should be chosen instead of auxiliary heating due to the better EER of auxiliary compression chillers.

Along with improving primary energy ratios for thermal cooling, in recent years, there has been an increased focus on the electricity requirements of auxiliary devices such as pumps, cooling tower, etc. In relation to the produced cooling energy, the current electrical EER for auxiliary energy is often only between 3 and 6.

$$EER_{el} = \frac{Q_{cool}}{Q_{el,aux}} \quad (5.3)$$

It is, however, quite possible for EER values in thermal cooling systems to be over 10 if all pressure losses are consistently reduced and the regulation strategies optimised for the cooling tower operation. A study of a 15 kW cooling machine showed that when the cooling

tower's ventilator volume flow rate control was missing, the primary energy advantages disappear for the solar thermal cooling in comparison to good electric systems. To be able to achieve an energetic advantage over compression cooling, the solar coverage rate must account for at least 70%, better still 80%. In the case of a dry cooling tower, then, if energetic advantages are to be attained at all, the cooling tower ventilator volume flow rate must be regulated. With an efficient solar thermal cooling system with optimised control systems, almost 50% of the primary energy can be saved when compared with electrical cooling.

Technology	Absorption cooling with water-lithium bromide	Absorption cooling with ammonia-water	Closed cycle adsorption water-silica gel	Open cycle desiccant cooling
Refrigerant	H ₂ O	NH ₃	H ₂ O	-
Sorbent	LiBr	H ₂ O	Silica gel	Silica gel/LiCl
Chilling carrier	Water	Water-glycol	Water	Air
Chilling temperature	6 - 20°C	-60° to +20°C	6 - 20°C	16 - 20°C
Heating temperature	70 - 110°C	70 - 140°C	55 - 100°C	55 - 100°C
Cooling water temperature	25 - 40°C	25 - 50°C	25 - 35°C	Not required
Cooling power range	10 - 12000 kW	10 - 10000 kW	7 - 350 kW	6 - 300 kW
EER/-	0.6 - 0.8	0.5 - 0.7	0.6 - 0.7	0.5 - 1.0
Approximate chiller investment costs per kW cooling power	200 - 1200 € kW ⁻¹	400 - 1250 € kW ⁻¹	350 - 1500 € kW ⁻¹	1500 - 2000 € kW ⁻¹ (approx. 6000 € per 1000 m ³ h ⁻¹)

Table 5.1 Overview of solar thermally powered cooling and air-conditioning processes. The expense summary is derived mostly from an EU study from the Intelligent Energy for Europe programme (Lamers et al., 2008).

5.2 Technology trends

Absorption cooling technology

Absorption technology has been known of since the beginning the 19th century and has again attracted growing interest in recent years. This is due not only to the ability of using waste heat and thermal solar energy to save primary energy in cooling, but also heat pump applications.

While cooling production with absorption chillers has been common for decades, heat pump applications have only become meaningful in recent years. Gas-driven absorption heat pumps achieve coefficients of performance of approximately 1.5, i.e. 1 kWh of the fuel heating energy is converted into 1.5 kWh of useful heat by adding low exergy ambient heat. This conversion ratio is significantly higher than the efficiency of condensing boilers presently available on the market of around 100%.

Absorption chillers have been mass-produced since the 1960s. Arkla Industries alone (today Robur SpA) produced over 300 000 small chillers with cooling power of 10.5 - 17 kW. Current

chillers cover a large power range, from 5 to over 5 000 kW. Several new machines in the low power range below 20 kW have been developed in the last decade. There is still room for improvement of the chiller technologies, especially when low driving temperatures, low pressure drops in the components and, as a result, low auxiliary electricity demands are in the focus. A high efficiency 50 kW LiBr/H₂O chiller was developed by a collaboration of the German research centres ZAE Bayern and TU Berlin with Vattenfall. The machine has an EER of 0.8 over a wide range of load conditions (from 25% to 150% of nominal load) at an electrical EER of about 20.

Ammonia-water-based absorption technology makes it possible for heat to be taken up or cold to be produced at temperatures less than 0°C with heat rejection at over 45°C, i.e. with a relatively high-temperature hub. Such machines are suitable for summer cooling operation with dry heat rejection at condenser temperatures over 40°C. Yet high-temperature hubs require higher drive temperatures for the generator. Some projects with ammonia-water machines have been therefore carried out with concentrating collectors with drive temperatures of over 150°C to allow dry heat rejection at high ambient temperatures.

Fresnel or parabolic trough concentrating collectors are increasingly used and are also combined with double effect lithium bromide-water chillers to take advantage of high EERs. Lightweight parabolic trough collectors have been used in several Australian projects in order to air-condition shopping centres, cinemas or similar buildings. Fresnel collectors make optimal use of limited rooftop area and provide minimal wind loads with the narrow, rotating, partitioned mirrors.

Further increases in EER are possible with triple effect absorption chillers which are now available on the market. Triple effect machines achieve EERs between 1.6 and 2.1 at full-load, depending on the cooling water temperature.

Today, high power chillers are mainly produced with LiBr technology. One reason is the increasing application of double effect absorption machines, in which first the refrigerant is expelled at high temperatures and the condensation heat is used for further refrigerant desorption at lower temperatures and pressures. In solar energy operation, however, double effect chillers can only be operated with concentrating thermal collectors. Today the costs of solar-powered double effect cooling are not much lower than those of single effect systems (due to high collector and storage costs), despite the better performance figure.

A range of solar technology manufacturers or system developers have complete packages available for solar cooling. An integrated control system in the solar cooling kit simplifies the interaction of thermal solar installation with the chiller and thereby avoids faulty operation.

Adsorption cooling

Several companies are now producing small adsorption cooling machines with around 7 - 15 kW cooling capacity. The machines can be operated with dry heat rejection; however, for peak loads, spray evaporation of water supports the heat rejection system. *Legionella* build-up can be avoided by the complete drying of the heat rejection system. Adsorption chillers with small capacities below 10 kW are available with silica gel or zeolite as an adsorbent. Machines are designed either for very low heating temperatures of 55 - 65°C, or for high operating temperatures, yet with high heat rejection temperatures of up to 42°C. They therefore offer very interesting areas of application.

Larger adsorption machines mainly from Japanese manufacturers are operating reliably in different demonstration facilities. Average thermal EERs of 0.43 for a 70 kW adsorber were measured at a clinic at Freiburg University. The solar coverage rate of the 171 m² vacuum tube

installation only reached 28% yearly due to long operation hours during night-time. Since 2008 at the company Festo in Esslingen/Germany, three adsorption chiller machines each with 350 kW cooling capacity have been measured in detail. The operating energy is provided from a vacuum tube collector field with 1330 m² collector area as well as from surplus heat. The average monthly EERs for the system are between 0.37 and 0.53. The electrical EER for the cooling tower operation and pump electricity of the system is between 2.7 and 3.5.

Open sorption air conditioning

Open sorption cooling is often used in projects with ventilation air requirements. The air drying is done with either solid or liquid sorption materials; the cooling effect is achieved with evaporation of water.

Machines with double-stage sorption air conditioning, with high dehumidifying capacities and EER are being developed for humid climatic regions. With the use of two sorption wheels in series, EERs of 1.0 can be achieved at 80°C regeneration temperature and extremely humid ambient air conditions (35°C outside temperature and 23 g kg⁻¹ humidity). Through the combination of silica gel with a liquid sorption medium such as lithium chloride, the dehumidifying capacity at a given regeneration temperatures can be raised by 20 - 30% (Wang, 2009). In 2007, the Shanghai Jiao Tong University built a 10 kW machine which also attained thermal EERs of 1.0. The electrical EER for ventilators, pumps and other electricity consumers was 8.3. With higher outside air temperature and humidity of 35°C and 23.2 g kg⁻¹, the system with two sorption rotors enabled the condition of the supply air to be at 25°C and 17.1 g kg⁻¹. In China, a cost of only 1000 Euro per kW is estimated for a purely air-based 5 kW solar-powered sorption system.

Liquid sorbents offer the advantage of continuous air drying with the possibility of simultaneously releasing heat. Therefore, in contrast to open sorption with solid adsorbents, the drying process can take place almost isothermally.

Mean thermal EERs of almost 1.2 have been achieved in a commercial liquid sorption system. In 2006, a 350 kW capacity liquid sorption system for air drying was installed in Singapore which was based on the work of The Bavarian Centre for Applied Energy Research (ZAE Bayern). With a total air flow volume of 13 000 m³ h⁻¹ and with an outside temperature of 33°C and humidity level of 21 g kg⁻¹, a dehumidifying capacity of more than 10 g kg⁻¹ could be attained. Also in the USA, liquid sorption systems with especially low flowthrough amounts have been developed and are currently being tested in field tests.

Economic feasibility and system dimensions

Operational experience has been gathered in a variety of demonstration projects conducted in Germany and Europe, which have led to improvements in control strategies and system planning. With careful planning, implementation and operational optimisation, solar thermal cooling systems can already today be economically feasible with contracting models. The system cost today for small capacity solar thermal cooling is about 4000 Euro per kW of installed capacity. Within the coming years it is expected that the price will sink to about 3000 Euro per kW.

Inclusive of all expenses (including the system space requirements, commissioning, monitoring and planning), costs of 5346 Euro kW⁻¹ were documented for a system constructed in 2009 in France with a 35 kW cooling capacity and 90 m² of solar tube collectors, and 8267 Euro kW⁻¹ for a smaller adsorption cooling system with 7.5 kW cooling capacity (Mugnier, 2010).

In one of the first solar cooling projects for a low-energy building in Shanghai, China, a low specific price of 1750 Euros kW^{-1} was obtained, including the collectors (Wisions, 2007). For installations with very large capacities in the Megawatt capacity range such as at the company Festo, total investment costs of 1300 Euro kW^{-1} cooling power have been achieved. However, in this case available surplus heat is used alongside the solar thermally provided heat, which only has a low solar fraction, keeping investment costs low.

There exist high market barriers to the introduction of solar cooling systems, especially when there are low electricity costs (e.g. 8 - 10 Eurocents per kWh in Australia or in some of the Mediterranean countries) and minimal costs for conventional systems (approximately 300 Euro per kW).

The influence of buildings on the dimensioning and financial feasibility of solar cooling systems with absorption cooling technology was investigated in detail with the help of the simulation environment programme INSEL (www.insel.eu). For example, the result in the location Madrid was a total of 913 full load hours at low internal loads of 4 W m^{-2} . The same building in Stuttgart provided a result of just 313 full load hours. Only with considerably higher internal loads, such as 20 W m^{-2} , the full load hours increase to almost 2000 hours (in Madrid). If the building orientation is changed from south to east or west, there is a shift of the maximum cooling load peaks. The total energy demand for the cooling sinks slightly when orientated towards the east or west.

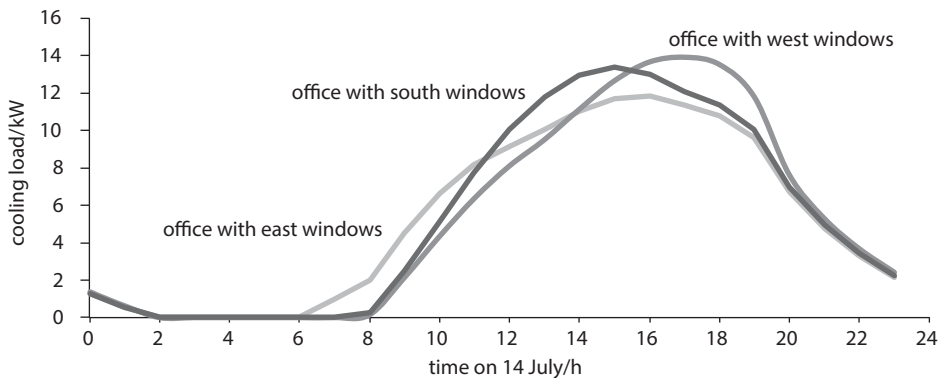


Figure 5.3 The cooling load curve on a summer day with different building orientations (Location: Madrid).

Although each building variation must be equipped with the same peak cooling power, the size of the necessary collector area depends on the time dependence of the cooling loads as well as on the chosen control strategy. For the same location, a larger collector area is required when a greater cooling energy demand is required.

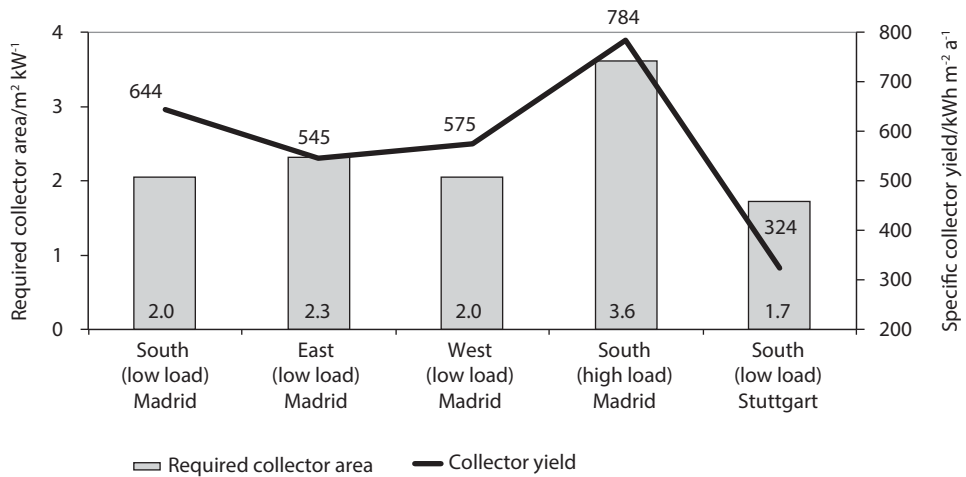


Figure 5.4 Required collector area for the locations Madrid/Spain and Stuttgart/Germany with low and high internal loads and different building orientations. The orientation given shows where the main window front of the offices are located.

The costs for cooling production are dominated by the investment and installation costs of the collector system and the absorption chiller machines (each accounting for about 40% of the total costs). For longer system operating times, the costs related to the amount of cooling energy drop considerably (to less than 200 Euro per MWh in Madrid). The higher actual cooling costs for the location Stuttgart are therefore mainly due to the low utilisation of the machine. Already today, large solar cooling systems for industrial usage have enabled very low heating prices of 30 Euro per MWh for solar thermal energy to be attained. The total costs for solar cooling energy can reach costs close to conventional cooling systems at 100 - 150 Euro per MWh when electricity consumption is reduced as much as possible and full load hours are high.

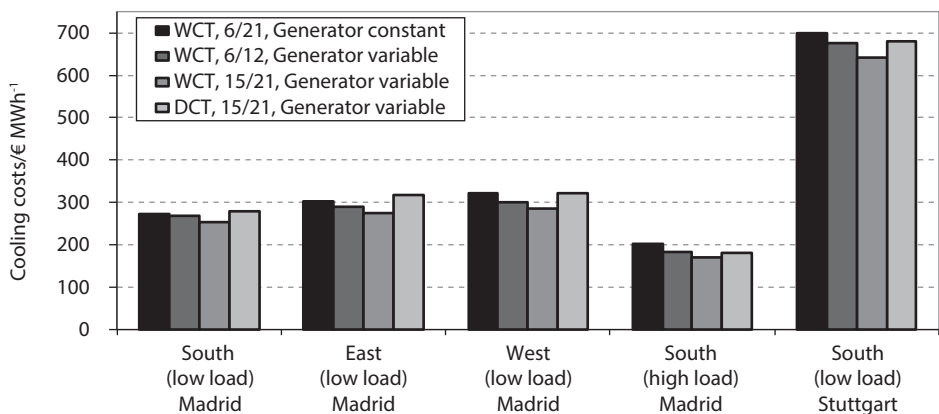


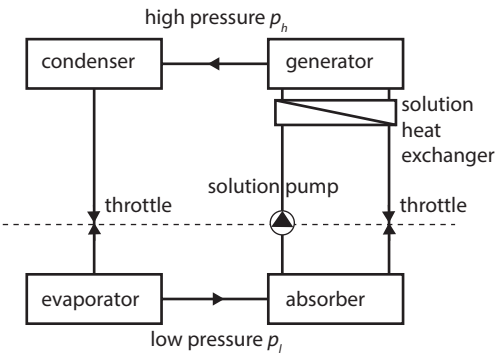
Figure 5.5 Cooling costs for offices in the locations Madrid and Stuttgart with low versus high internal loads, different orientations and with various control strategies and system configurations: wet cooling tower (WCT) or dry cooling tower (DCT), cooling distribution at 6/12°C and 15/21°C as well as constant or variable generator temperature.

5.3 The absorption cooling process and its components

Absorption chillers differ from electrically powered compression refrigerant plants in the substitution of the mechanical compressor by a thermal compressor. The compressor has the function of bringing the evaporated refrigerant to such a high pressure that it can be condensed at high temperatures, and in the cyclic process be returned again as liquid to the evaporator. In an absorption chiller the compressor process is replaced by absorption of the evaporated refrigerant in a solvent (for example, water or LiBr) and subsequent boiling in the generator at high pressure.

The refrigerant-poor solution from the generator is pumped back into the absorber, where it can take up refrigerant vapour again from the evaporator. A continuous cooling process can be maintained by circulating liquid sorbent, a substantial advantage over adsorption technology with intermittent refrigerant adsorption on the solid silica gel.

Absorption cooler



Compression cooler

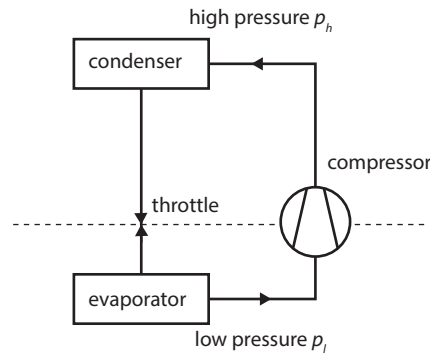


Figure 5.6 Components of the absorption chiller compared with an electrical compression chiller.

Through the representation of the components in the isostere diagram (isosteres being pressure lines with constant concentration of solution ξ as parameter), the individual process steps can be characterised. On the high pressure side with pressure p_h are the condenser and generator; on the low pressure side with pressure level p_l are the evaporator and absorber. In the evaporator and condenser the refrigerant concentration is 100%. The lowest refrigerant concentration in the solution is produced in the generator (right isostere).

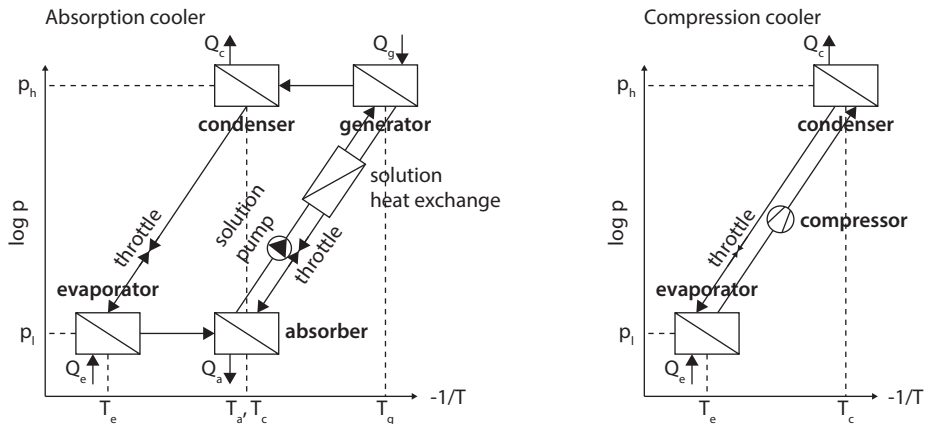


Figure 5.7 Representation of the absorption and compression cooling process in the $\log p - 1/T$ diagram.

The arrangement of the components in an absorption chiller is determined by the common pressure level of the generator and condenser on the one hand, and of the evaporator and absorber on the other. The generator and condenser are located in common upper chambers, while the evaporator and absorber are arranged in the lower area of the machine.

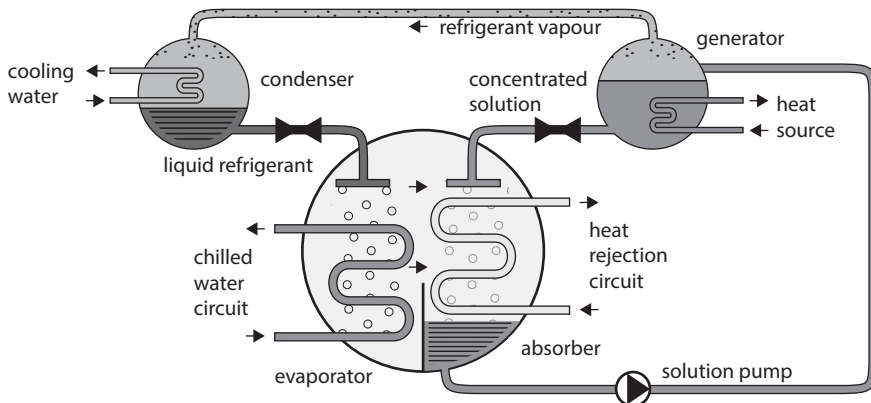


Figure 5.8 Arrangement of the components of an absorption chiller on a LiBr water basis.

The most commonly used pairs of working materials are ammonia-water and water-LiBr, with ammonia or water as refrigerants and water or LiBr as solvents. The thermodynamic properties of the refrigerant determine the possible temperature range of the machines. The drive temperatures for desorption are between 70 and 140°C, depending on the technology and temperature levels of cold and cooling water. During absorption, solution heat is released, which must be removed via a cooling (heat rejection) circuit.

The evaporator temperature can be lowered in ammonia chillers to -60°C, so that industrial cooling processes are possible. While ammonia boils at -33°C at a pressure of 105 Pa and can thus be used for cooling and air conditioning, the refrigerant water is limited to pure air conditioning with evaporator temperatures over 0°C.

In LiBr-water systems, the extremely low refrigerant pressure in the evaporator of 872 Pa at 5°C and the small pressure increase to 7375 Pa at +40°C in the condenser is favourable for a small solution pump power, but this means that the machine has to be evacuated. With the small water vapour pressure, very high flow rates must circulate to produce the cooling output, and large cross sections must be used to reduce pressure losses. In a 100 kW chiller, 145 kg of water must be evaporated every hour, which at the low system pressure of under 1000 Pa corresponds to a flow rate of $21\,300\text{ m}^3\text{ h}^{-1}$. In contrast, 286 kg of ammonia would have to be evaporated for the same cooling output, which at the high evaporator pressure of about $5 \times 10^5\text{ Pa}$ corresponds to a flow rate of only $80\text{ m}^3\text{ h}^{-1}$.

In LiBr systems the refrigerant concentration in the solution must not drop too sharply, otherwise crystallisation of the solvent occurs. Due to the rather poor solubility of water in LiBr, the absorber and condenser are usually water-cooled.

The large difference in boiling points between the refrigerant and solvent is an advantage of water-LiBr systems, so that when the refrigerant is expelled from the solution, pure refrigerant vapour develops. The difference in boiling points between ammonia and water on the other hand is only 133 K, so that when expulsion occurs, water vapour is always produced and must be separated again in a rectifying column.

If ammonia is used as a refrigerant, a high vapour pressure of $4.85 \times 10^5\text{ Pa}$ develops at evaporator temperatures of +5°C. To liquify the refrigerant at condenser temperatures of 40°C, the ammonia pressure in the generator must be brought to about $15 \times 10^5\text{ Pa}$, i.e. ammonia absorption chillers must be constructed for high system pressures and use powerful solution pumps.

Double effect absorption cooling process

In a double effect absorption process, two generators are operated at different temperature levels. The high-temperature generator 1 with process temperatures of around 150°C produces refrigerant vapour at a high pressure level. This refrigerant vapour condenses at temperatures sufficient to operate a second generator 2. The refrigerant-rich solution is thus pumped first to the medium-temperature generator, where some refrigerant is expelled and then to the high-temperature generator, where the solution is further depleted of refrigerant.

The pressure level and thus the temperature in the second generator must be high enough to achieve condensation in the second air- or water-cooled condenser. The condensate from both condensers is expanded into a common evaporator at the low pressure level, where the cooling power is produced.

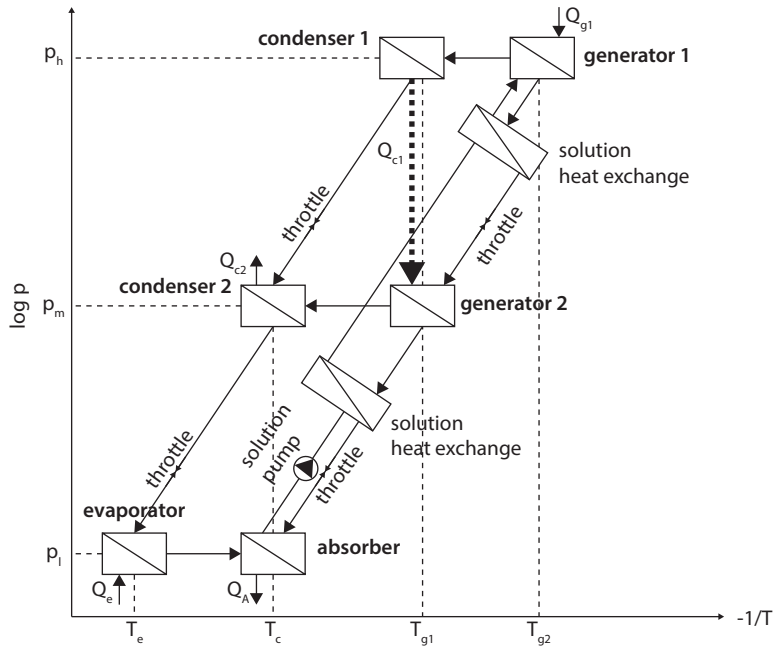


Figure 5.9 Process of double-effect absorption chiller.

The generator heat must be supplied at a high temperature (above 110°C). Mainly concentrating collectors are used for such applications. Even small parabolic concentrators are now available for installation on flat building roofs and are attractive alternatives to vacuum tube collectors. These concentrating collectors depend on direct beam irradiance, which varies from about $700 \text{ kWh m}^{-2} \text{ a}^{-1}$ in Germany (732 in Cologne, 894 in Würzburg) to about $1300 \text{ kWh m}^{-2} \text{ a}^{-1}$ in southern Europe.

By using condensation heat, the performance figure can clearly be improved from approximately 0.7 for single-effect processes to 1.3 for double-effect absorption chillers and 1.7 for triple effect.

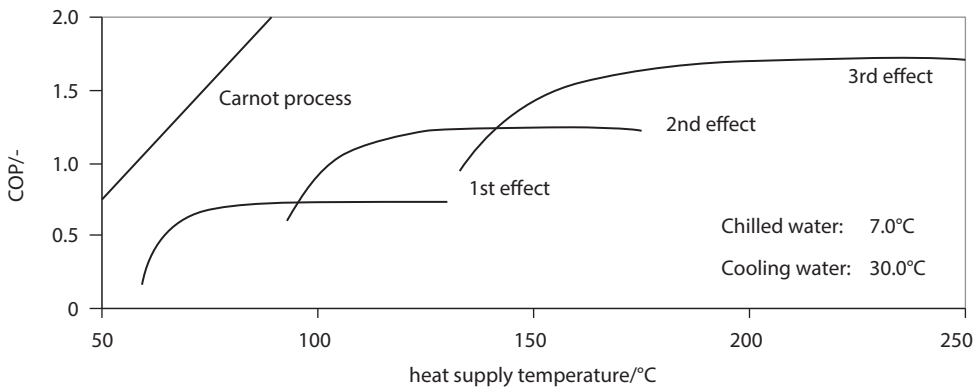


Figure 5.10 Comparison of coefficients of performance for multistage absorption chillers (Grossmann, 2002).

Operating experience in Turkey showed a maximum EER of 1.5 using a Chinese-produced double-effect absorption chiller, powered by parabolic trough collectors producing steam at 144°C.

5.4 Components of absorption chillers

Evaporator and condenser

Evaporator and condenser are conventional components whose heat uptake or release takes place either by air or a liquid circuit. If the refrigerant in the condenser is to condense at high ambient temperatures, the vapour pressure must be correspondingly high. The freed condensation heat is used in heat pump applications for heating purposes, and dissipated in cooling processes to the environment. Before the condensed refrigerant enters the evaporator, the pressure must be reduced to the low evaporator pressure. This is usually done by a throttle valve. Only in diffusion-absorption chillers does an auxiliary gas such as H_2 or He provide the pressure balance between the high and low pressure sides.

The refrigerant evaporates only because the pressure in the evaporator is reduced by the constant suction of the compressor or by the absorption in the solvent. Only if the saturation temperature of the respective vapour pressure is under the ambient temperature can heat be taken up from the environment, i.e. can cooling take place. The volume flow rate of the compressor or absorber is regulated in such a way that the evaporator pressure remains constant.

Absorber

The refrigerant-poor solution flows back into the absorber from the generator. The refrigerant vapour produced in the evaporator is absorbed there as a function of the absorber temperature and solvent concentration. The evaporator and absorber are at the same refrigerant pressure level. At low evaporator temperatures and correspondingly low vapour pressures, the absorber temperature and the concentration ξ of the refrigerant in the solution must not become too high, since otherwise no more absorption takes place.

The refrigerant-poor solution in the absorber must constantly take up the refrigerant produced in the evaporator, since otherwise the evaporator pressure would rise. Through refrigerant absorption the concentration of refrigerant vapour in the solution rises. The concentration

modification between the rich and the poor solution ξ_r and ξ_p is called the degassing width. It is usually between 10% and 25% in the $\text{NH}_3/\text{H}_2\text{O}$ system, and about 4 - 6% in the $\text{H}_2\text{O}/\text{LiBr}$ system. The necessary solvent mass flows are determined from a mass balance at the absorber. The mass flow of the rich solution $\dot{m}_r$ consists of the sum of supplied refrigerant vapour $\dot{m}_v$ and the poor solution $\dot{m}_p$ back-pumped from the generator:

$$\dot{m}_r = \dot{m}_v + \dot{m}_p \quad (5.4)$$

The associated refrigerant mass flows depend on the concentration of the refrigerant in the rich and poor solution ξ_r and ξ_p , and on the concentration of the refrigerant in the gaseous state, i.e. the purity of the vapour ξ_v , which is 1.0 in $\text{H}_2\text{O}/\text{LiBr}$ systems, but lower than 1.0 for ammonia-water working pairs.

$$\dot{m}_r \xi_r = \dot{m}_v \xi_v + \dot{m}_p \xi_p \quad (5.5)$$

The mass flow relation between the rich solution and the refrigerant vapour is defined as the specific recirculation factor f , and for ammonia systems is between 10 and 30.

$$f = \frac{\dot{m}_r}{\dot{m}_v} = \frac{\xi_v - \xi_p}{\xi_r - \xi_p} \quad (5.6)$$

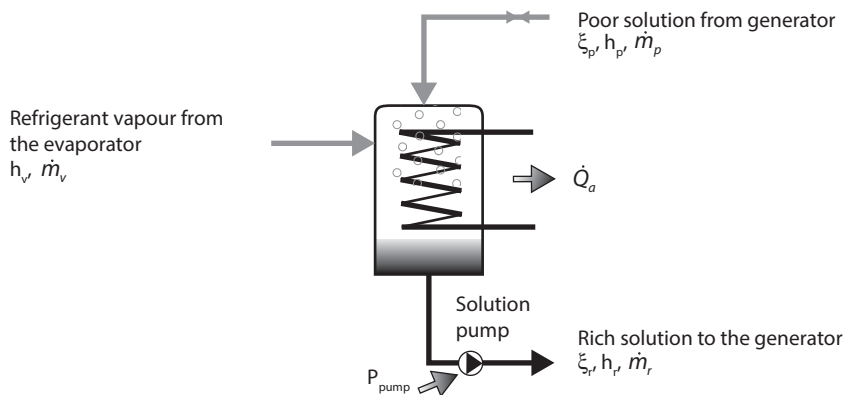


Figure 5.11 Mass balances at the absorber, in the absorber the solution heat $\dot{Q}_a$ is released.

Example 5.1

Calculation of the recirculation rate and the necessary solvent circulation for an evaporator power of 100 kW, if the poor solution comes from the generator at a concentration of $\xi_p = 0.3$ and the degassing width is limited to 10%.

The recirculation relation f is only a function of the degassing width. Assuming pure refrigerant vapour ($\xi_v = 1$) results in a mass flow relation

$$f = \frac{1 - 0.3}{0.4 - 0.3} = 7$$

i.e. seven times the quantity of solvent must be circulated in relation to the refrigerant vapour. For water as refrigerant the LiBr solvent mass flow is thus $7 \times 144.6 \text{ kg h}^{-1} = 1012 \text{ kg h}^{-1}$. For ammonia as refrigerant, $7 \times 286 \text{ kg h}^{-1} = 2003 \text{ kg h}^{-1}$ must be circulated.

Generator

The absorbed refrigerant is expelled again from the generator by heating the refrigerant-rich solution. Conventional heating by gas or other fossil sources of energy can be replaced by thermal solar energy. For operation with solar energy the necessary temperature level in the generator is crucial; it can be well over 100°C with unfavourable boundary conditions, in particular a low solvent concentration. Although today's vacuum collectors can certainly supply such temperature levels with acceptable efficiency, solar use is better at lower temperatures. The physical principles for determining the necessary temperatures and performance figures are discussed in the following sections.

5.5 Physical principles of the absorption process

Vapour pressure and solution concentration

The operational principle of the thermal compressor is based on the fact that the vapour pressure of the pure refrigerant drops if it is absorbed in a more highly boiling liquid. While pure ammonia at $+5^\circ\text{C}$ already produces a vapour pressure of about $5 \times 10^5 \text{ Pa}$, in a solution with 50% water the vapour pressure drops to $1.3 \times 10^5 \text{ Pa}$ at the same temperature; the absorbent fulfils the suction function of the compressor.

Conversely, high temperatures must be produced in the generator to expel the refrigerant from the solution. The smaller the refrigerant concentration in the rich solution, the higher the generator temperature required to produce the vapour pressure for liquefaction.

In thermal equilibrium, a saturation vapour pressure p_s rises over a pure liquid, depending solely on the temperature. As a function of the evaporation enthalpy of the pure working material, an exponential rise of the saturation vapour pressure with the negative reciprocal value of the temperature $-1/T$ results, based on Clausius–Clapeyron. If the vapour pressure over a solution is calculated, the coefficients of the Clausius–Clapeyron equation are modified accordingly.

Ammonia – Water

For pure ammonia, this results in an approximation solution of Bourseau (1986) with the logarithmic function

$$\log_{10} p_s = a - \frac{b}{T} \quad (5.7)$$

with the coefficients $a = 10.018$ and $b = 1204.3$ for pressures up to $25 \times 10^5 \text{ Pa}$ (T in kelvin).

The low pressure level p_l of the chiller is determined by the saturation vapour pressure at the desired evaporator temperature.

Example 5.2

Vapour pressure of NH_3 at evaporator temperatures T_e of -10°C and $+5^\circ\text{C}$, and at condenser temperatures T_c of $+30^\circ\text{C}$ and 50°C .

Component	Temperature/K	Vapour pressure NH ₃ /Pa
Evaporator	263	2.76×10^5
Evaporator	278	4.88×10^5
Condenser	303	11.1×10^5
Condenser	323	19.6×10^5

Due to the high vapour pressures of ammonia, the flow rates of the evaporated refrigerant are clearly lower than the flow rates of the water vapour in LiBr systems, despite higher vapour mass flows. From the pressure levels in the evaporator p_e and the vapour mass flow $\dot{m}_v$, the vapour volume flow $\dot{V}_v$ can be calculated at a given temperature [K] from the ideal gas equation:

$$p_v \dot{V}_v = \dot{m}_v R_s T$$

The specific gas constants R_s of water vapour and ammonia are calculated from the general gas constant $R = 8.314 \text{ J mol}^{-1} \text{ K}^{-1}$ using the mol mass M : $R_s = R/M$ and are, with mol masses of 18 g mol^{-1} for water and 17 g mol^{-1} for NH_3 , $R_s(\text{H}_2\text{O}) = 462 \text{ J kg}^{-1} \text{ K}^{-1}$ or $R_s(\text{NH}_3) = 489 \text{ J kg}^{-1} \text{ K}^{-1}$.

Example 5.3

Calculation of the flow rate of refrigerant vapour in the evaporator for water and ammonia for 100 kW evaporator cooling capacity.

The pressure level at 5°C evaporator temperature is, using Equation (5.7), $4.88 \times 10^5 \text{ Pa}$ for ammonia and from Equation (5.16) 872 Pa for water. Water: from the mass flow of 145 kg/h or 0.04 kg/s, a very high flow rate of

$$\dot{V}_v = \frac{\dot{m}_v R_s T}{p_e} = \frac{0.04 \frac{\text{kg}}{\text{s}} 462 \frac{\text{J}}{\text{kg K}} 278 \text{ K}}{872 \text{ Pa}} = 5.9 \frac{\text{m}^3}{\text{s}} = 21298 \frac{\text{m}^3}{\text{h}}$$

is due to the very low evaporator pressure. Ammonia: from the almost twice as high mass flow of 286 kg h^{-1} , a flow rate of only $80 \text{ m}^3 \text{ h}^{-1}$ results, due to the very high evaporator pressure of $4.85 \times 10^5 \text{ Pa}$.

The temperature level of the condenser determines the high pressure level p_h in the absorption chiller. If the expelled refrigerant is to be liquefied even at high ambient temperatures in an air-cooled condenser, the pressure level of the system rises sharply (typically $p_h > 15 \times 10^5 \text{ Pa}$ for ammonia chillers).

As a function of the refrigerant concentration ξ in the solution, coefficients a and b of the Clausius–Clapeyron vapour pressure equation are modified as follows for ammonia/water:

$$\begin{aligned} \log_{10} p &= a - b/T \\ a &= 10.44 - 1.767\xi + 0.9823\xi^2 + 0.3627\xi^3 \\ b &= 2013.8 - 2155.7\xi + 1540.9\xi^2 - 194.7\xi^3 \end{aligned} \quad (5.8)$$

From the vapour pressure curve of the pure refrigerant, the high and low pressure levels p_h and p_l can thus be determined at condenser and evaporator temperatures T_c and T_e . On the low pressure side, the maximum refrigerant concentration in the solution at which absorption can still occur

is determined by the absorber temperature T_a . If the absorber is at low temperatures, absorption can occur at higher refrigerant concentrations of solution. The generator temperature T_g on the high-pressure side determines the minimum refrigerant solution concentration and thus the degassing width as the concentration difference between the rich solution in the absorber and the poor solution in the generator.

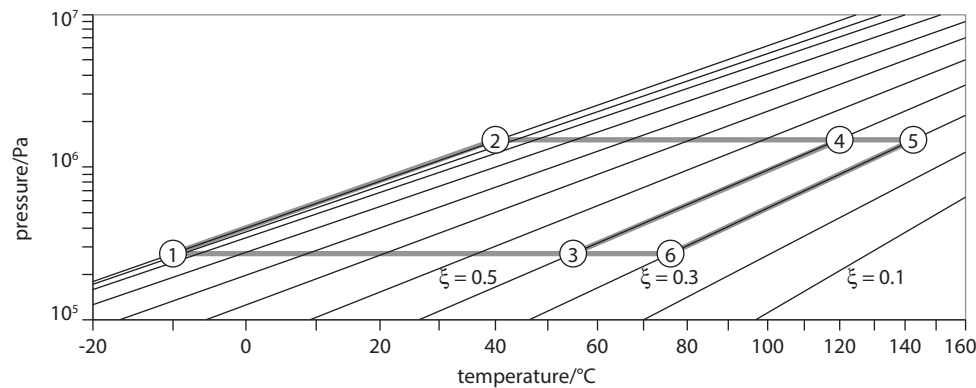


Figure 5.12 Vapour pressure curves over ammonia-water solutions in the log p - 1/T diagram.

The concentration of solution varies in 10% steps from 0.1 to 1 (pure refrigerant corresponds to $\xi = 1$, left curve). The process runs via the evaporator (status point 1), condenser (2), the absorption of the refrigerant in the solution (3), the entry of the rich solution into the generator (4), the poor solution at the end of the expulsion process (5) and the poor solution cooled by the solution heat exchanger before re-entry into the absorber (6). The generator temperature can be calculated for given concentrations of solution from the modified Clausius equation:

$$T = \left(\frac{1}{b(\xi)} \times (a(\xi) - \log_{10} p) \right)^{-1} \quad (5.9)$$

The maximum generator temperature results when refrigerant vapour has already been expelled from the solution, i.e. at the end of the expulsion process. The refrigerant vapour in the generator is then in equilibrium with the refrigerant-poor solution leaving the generator.

Example 5.4

Calculation of the generator temperatures required to produce an ammonia vapour pressure of 20×10^5 Pa, if the concentration of the poor solution leaving the generator ξ_p is 0.2, 0.35 and 0.5.

Solution concentration ξ_p /-	Generator temperature T_g /°C
0.2	156
0.35	120.6
0.5	92.5

High concentrations of solution are therefore favourable for lowering the generator temperature and adjusting it to solar operation. High concentrations of solution, on the other hand, also mean high vapour pressures in the absorber and thus a restriction in suction efficiency of the absorber. If the evaporator temperatures are to be very low, i.e. produce low vapour pressures, the concentration of solution must be correspondingly low to maintain the evaporator pressure at a given absorber temperature.

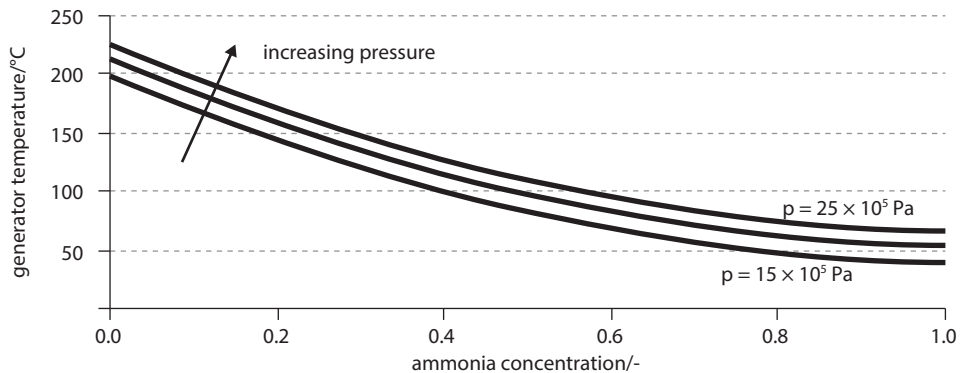


Figure 5.13 Necessary generator temperature as a function of the concentration of the refrigerant ammonia in the solution.

The pressure level is given by the condenser temperature and varies here in 5×10^5 Pa steps. With rising condenser pressure, the required generator temperatures also rise.

Example 5.5

In an absorber with a heat rejection temperature of 40°C , the ammonia refrigerant vapour must still be taken up at an evaporator temperature of -10°C . The maximum concentration of solution is determined by resolving the modified Clausius equation for ξ .

- Vapour pressure at -10°C : 2.75×10^5 Pa
- Maximum concentration of solution: $\xi = 0.38$

Refrigerant vapour concentration for ammonia – water systems

If the boiling points of the refrigerant and solution are close (as in ammonia-water systems), a significant proportion of the solvent is co-evaporated in the generator (not a problem in LiBr systems). The lower the refrigerant concentration of the rich solution, the higher are the required generator temperatures and the more solvent is evaporated. Two problems result from this co-evaporation: on the one hand, aqueous refrigerant can no longer be completely evaporated, and on the other hand, a substantial energy quantity is used for water evaporation, which reduces the performance. The purity of the ammonia evaporated in the generator is characterised by the vapour concentration ξ_v , with $\xi_v = 1$ representing pure refrigerant. The higher the refrigerant concentration in the liquid solution ξ_l , the higher the concentration of refrigerant in the vapour. The vapour concentration depends not only on the refrigerant concentration in the solution, but also on the temperature or the total pressure p necessary for liquefaction in the condenser. At high temperatures and pressures, more solvent is co-evaporated and the vapour purity falls.

To calculate the vapour purity (and afterwards the enthalpy), for better curve fitting a division is made into two pressure ranges: the low-pressure range to 5.52×10^5 Pa and the high pressure range for $p > 5.52 \times 10^5$ Pa.

$$\xi_v = 1.0 - (1.0 - \xi_l)^R \quad (5.10)$$

The parameter R for the high pressure range $p > 5.52 \times 10^5$ Pa is

$$R = 7.1588 - 0.6171 \times 10^{-6} p + (((10.7490 \xi_l - 17.8690) \xi_l + 4.0297) \xi_l - 1.3086) \xi_l + 0.3715 \times 10^{-6} p \xi_l \quad (5.11)$$

and for the low-pressure range $p < 5.52 \times 10^5$ Pa:

$$R = (((108.485 \xi_l - 229.009) \xi_l + 155.247) \xi_l - 41.0442) \xi_l + 11.2925 - 4.532 \times 10^{-6} p + 3.0934 \times 10^{-6} p \xi_l^2 \quad (5.12)$$

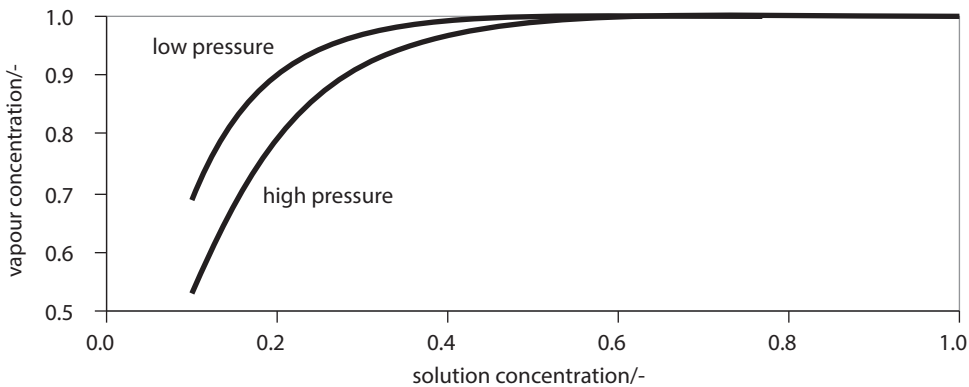


Figure 5.14 Vapour concentration as a function of the concentration of solution.

If the temperature is given instead of the pressure p , the pressure is determined first according to the modified Clausius equation (5.8).

The pressure curves in the low-pressure range (here represented in 0.5×10^5 Pa steps from 0.5 to 5.0×10^5 Pa) coincide, as do the pressure curves in the high-pressure range (here of $10 - 20 \times 10^5$ Pa in 10^5 Pa steps).

With Equation (5.10) the equilibrium concentration between the solution and vapour can be calculated. Very low concentrations of solution, which are produced at high generator temperatures and which are necessary for very low evaporator temperatures, are in equilibrium with impure vapour, from which high proportions of solvents must be still condensed out.

Example 5.6

Calculation of the equilibrium concentration between liquid and vapour at 20×10^5 Pa total pressure for concentrations of solution of 0.2 and 0.5.

The higher concentrations of solution lead directly to a higher vapour purity and thus to lower rectification heat losses.

Solution concentration/-	Vapour purity/-
0.2	0.7395
0.5	0.9807

Water – Lithium Bromide

The absorption cooling process of water-LiBr systems takes place at very low absolute pressure levels of a few hundred at the low pressure side up to a few thousand Pascal at the high pressure side. The minimum refrigeration concentration of solution is limited to 35%, since otherwise crystallisation of the solvent LiBr occurs.

The temperature-dependent saturation vapour pressure of water $p_s(T)$ is again calculated using by the Clausius–Clapeyron equation, which describes the equilibrium between liquid and vapour.

$$\frac{dp_s}{p_s} = \frac{h_e(T)}{RT^2} dT$$

The solution depends on the selected approximation for the temperature dependence of the evaporation enthalpy $h_e(T)$. This describes the enthalpy difference of vapour h_v and liquid h_l .

$$h_e(T) = h_v(T) - h_l(T) \quad (5.13)$$

The decrease in evaporation enthalpy $h_e(T)/\text{kJ kg}^{-1}$ with temperature $T/^\circ\text{C}$ results from the following approximation equations:

$$h_v(T) = 2501.482 + 1.789736T + 8.957546 \times 10^{-4}T^2 - 1.300254 \times 10^{-5}T^3 \quad (5.14)$$

$$h_l(T) = -2.25 \times 10^{-2} + 4.2063437T - 6.014696 \times 10^{-4}T^2 + 4.381537 \times 10^{-6}T^3 \quad (5.15)$$

For the temperature range $10^\circ < T < 200^\circ\text{C}$, the error in this approximation is less than 0.04%. The saturation water vapour pressure is then approximated for the range $0^\circ\text{C} < T < 100^\circ\text{C}$, with an error of under 0.02% (T in $^\circ\text{C}$, p in Pa):

$$p_s(T) = 611 \exp \left(\frac{-1.91275 \times 10^{-4} + 7.258 \times 10^{-2}T - 2.939 \times 10^{-4}T^2}{+9.841 \times 10^{-7}T^3 - 1.92 \times 10^{-9}T^4} \right) \quad (5.16)$$

The vapour pressure is again reduced if the water vapour is in contact with a salt solution.

For the calculation of the concentration of the LiBr solution in dependence of the temperature and surrounding vapour pressure a correlation provided by McNeely (1979) can be used, which describes the solution temperature in dependence of the concentration and surrounding water vapour pressure. Since this equation cannot be explicitly solved, the concentration is calculated in an iteration process (Regula Falsi), where the concentration is varied until the temperature of the solution is equal to the know temperature value. The equation used is of the type

$$t_{\text{LiBr}}(\xi, p) = A(\xi) + B(\xi)t_s(p) \quad (5.17)$$

with

$$A(\xi)=\sum_{i=0}^{10}a_i\xi^i$$
$$B(\xi)=\sum_{i=0}^{10}b_i\xi^i$$

i	a _i	b _i
0	0.0	1.0
1	1.6634856 × 10 ¹	-6.8242821 × 10 ⁻²
2	-5.5338169 × 10 ²	5.8736190 × 10 ⁰
3	1.1228336 × 10 ⁴	-1.0278186 × 10 ²
4	-1.1028390 × 10 ⁵	9.3032374 × 10 ²
5	6.2109464 × 10 ⁵	-4.8223940 × 10 ³
6	-2.1112567 × 10 ⁶	1.5189038 × 10 ⁴
7	4.3851901 × 10 ⁶	-2.9412863 × 10 ⁴
8	-5.4098115 × 10 ⁶	3.4100528 × 10 ⁴
9	3.6266742 × 10 ⁶	-2.1671480 × 10 ⁴
10	-1.0153059 × 10 ⁶	5.7995604 × 10 ³

Table 5.2 Coefficients for the calculation of the temperature and solution concentration of an aqueous water-lithium bromide solution.

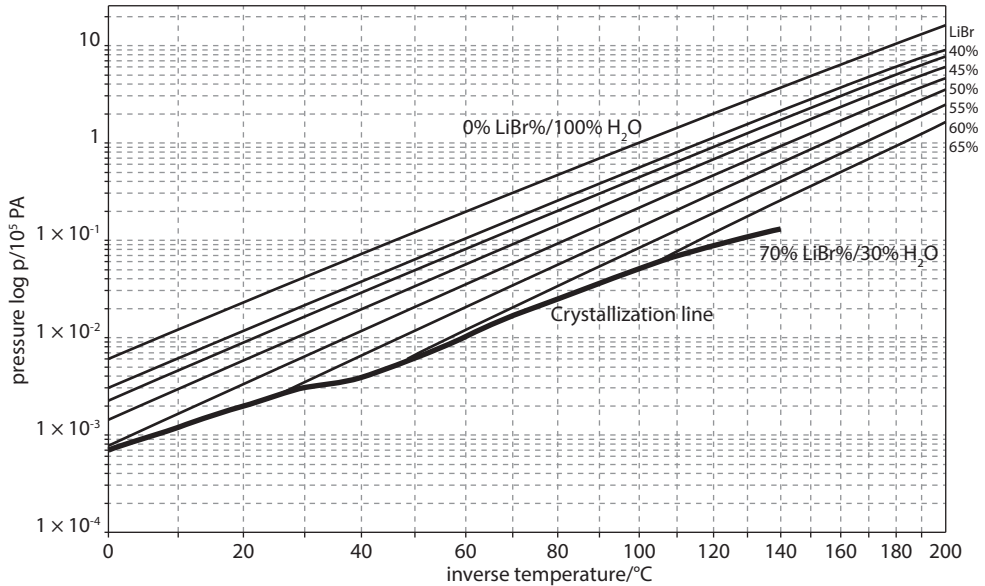


Figure 5.15 Logarithm of the vapour pressure of water over LiBr salt solutions plotted against the inverse of the temperature.

Evaporation enthalpy of absorption refrigerants

The evaporation enthalpy h_e (i.e. the amount of heat taken up at constant pressure) of the refrigerant water is almost twice as high as h_e of ammonia, for example, at an evaporator temperature of +5°C, 2489 kJ kg⁻¹ (water) as opposed to 1258 kJ kg⁻¹ (ammonia). The evaporation enthalpy h_e results from the enthalpy difference between vapour and liquid and is temperature-dependent $h_e = h_v - h_l$. For pure water vapour the approximation Equation (5.14) and Equation (5.15) are used.

Ammonia-Water

The specific evaporation enthalpy of pure ammonia is calculated with approximation formulae for the high and low pressure range. The equations for the high pressure range are only indicated in literature starting from 17.23×10^5 Pa, but they can however be applied with sufficient accuracy from the upper boundary of the low-pressure range at 5.52×10^5 Pa (temperature T in °C).

High pressure range:

$$5.52 \times 10^5 < p \leq 2.24 \times 10^6 \text{ Pa}$$

$$h_l = 6.7702 + 4.7182T \quad (5.18)$$

$$h_v = 1290.28542 + 19.4669 \times 10^{-9} (1.8T + 32)^4 \quad (5.19)$$

Low-pressure range:

$$3.45 \times 10^5 \text{ Pa} < p \leq 5.52 \times 10^5 \text{ Pa}$$

$$h_l = -1.4356 + 4.5705T \quad (5.20)$$

$$h_v = 1234.944 + (1.8T + 32) \times \left(0.9672 + (1.8T + 32) \left(-11.5081 \times 10^{-6} (1.8T + 32) + 3.4775 \times 10^{-3} \right) \right) \quad (5.21)$$

Example 5.7

Calculation of the refrigerant mass flow $\dot{m}_r$ of ammonia or water for a cooling output of the evaporator $\dot{Q}_e$ of 100 kW and an evaporator temperature of 5°C:

In the evaporator the refrigerant is evaporated in the low-pressure range. For water a mass flow of $0.04 \text{ kg s}^{-1} = 145 \text{ kg h}^{-1}$ results at an evaporation enthalpy

$$h_e = 2510 \frac{\text{kJ}}{\text{kg}} - 21 \frac{\text{kJ}}{\text{kg}} = 2489 \frac{\text{kJ}}{\text{kg}}$$

For ammonia with an evaporation enthalpy

$$h_e = 1279 \frac{\text{kJ}}{\text{kg}} - 21 \frac{\text{kJ}}{\text{kg}} = 1258 \frac{\text{kJ}}{\text{kg}}$$

a mass flow of $0.079 \text{ kg s}^{-1} = 286 \text{ kg h}^{-1}$ results.

The enthalpy of the solution depends on the refrigerant concentration in the solution ξ_l and on the temperature or pressure. In the following approximation equations, a differentiation is again made between the high- and low-pressure ranges.

For the high-pressure range with $p > 5.52 \times 10^5 \text{ Pa}$, the enthalpy h_l in kJ kg^{-1} at the temperature T in $^\circ\text{C}$ is calculated as follows:

$$h_l = -94.7974 + 4.7182T + (((1306.7740\xi_l - 4487.8637)\xi_l + 5450.0471)\xi_l - 1926.7160)\xi_l - 240.6738)\xi_l \quad (5.22)$$

For the low-pressure range $p \leq 5.52 \times 10^5 \text{ Pa}$ applies:

$$h_l = -51.7296 + 4.5705T + (((-1526.79\xi_l + 3160.60)\xi_l - 1158.988)\xi_l - 424.528)\xi_l \quad (5.23)$$

For the curve fitting of the vapour enthalpy, in the high pressure range an additional difference is made between low and high concentrations of solution. The vapour enthalpy depends not only on the temperature (or pressure) but also on the vapour purity ξ_v , which is calculated using Equation (5.10).

For the high-pressure range with $p > 5.52 \times 10^5 \text{ Pa}$ and a low concentration of solution between $0.1 \leq \xi_l \leq 0.36$, the vapour enthalpy is given by:

$$\begin{aligned} h_v = & 2506.6744 + 19.4669 \times 10^{-9} (1.8T + 32)^4 + (((-3122.7354\xi_v + 6871.3435)\xi_v \\ & - 5780.3107)\xi_v + 910.2483)\xi_v + (((-8.7804 \times 10^{-5} (1.8T + 32) \\ & + 0.0634)(1.8T + 32) - 13.8220)(1.8T + 32))(1 - \xi_v) \\ & + 1.2714(1.8T + 32)(1 - \xi_v)^2 \end{aligned} \quad (5.24)$$

For $\xi_l > 0.36$:

$$\begin{aligned} h_v = & 2113.2189 + 19.4669 \times 10^{-9} (1.8T + 32)^4 + (((0.1599XVT + 4.8363)XVT \\ & + 57.7705)XVT + 336.3805)XVT + (((-8.7804 \times 10^{-5} (1.8T + 32) \\ & + 0.0634)(1.8T + 32) - 13.8220)(1.8T + 32))(1 - \xi_v) \\ & + 1.2714(1.8T + 32)(1 - \xi_v)^2 \end{aligned} \quad (5.25)$$

with

$$\begin{aligned} XVT &= \ln(1 - \xi_v) \quad \text{for } \xi_v < 0.99996 \\ XVT &= \ln(0.00004) \quad \text{for } \xi_v \geq 0.99996 \end{aligned}$$

The enthalpy equations in the low-pressure range are valid for the entire concentration range of the liquid.

For $p < 5.52 \times 10^5 \text{ Pa}$ the vapour enthalpy is given by:

$$\begin{aligned} h_v = & 1234.944 + ((-11.5081 \times 10^{-6} (1.8T + 32) + 3.4775 \times 10^{-3})(1.8T + 32) \\ & + 0.9672)(1.8T + 32) + (((9.4324 \times 10^{-5} (1.8T + 32) \\ & - 0.0675)(1.8T + 32) + 15.7951)(1.8T + 32))(1 - \xi_v) \end{aligned} \quad (5.26)$$

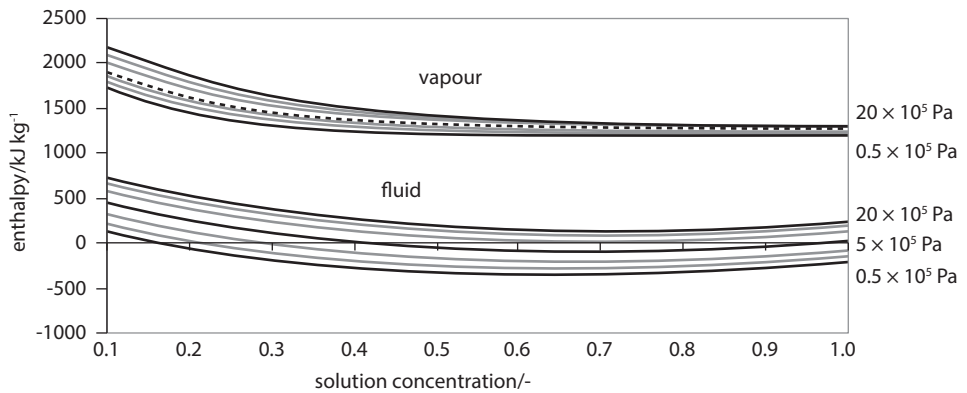


Figure 5.16 Enthalpy of ammonia-water mixtures in liquid and vaporous states as a function of the solution concentration.

In the curves illustrated, the parameter pressure rises from 0.5×10^5 Pa via 1×10^5 Pa, 2×10^5 Pa, 5×10^5 Pa, 10×10^5 Pa, 15×10^5 Pa and 20×10^5 Pa. From the enthalpy diagram as a function of the concentration of solution the enthalpy of the vapour is known (the vapour is in equilibrium with the solution), however, it is not evident what refrigerant concentration is contained in the vapour. To determine this, the vapour enthalpy is plotted not over the concentration of solution, but over the vapour concentration.

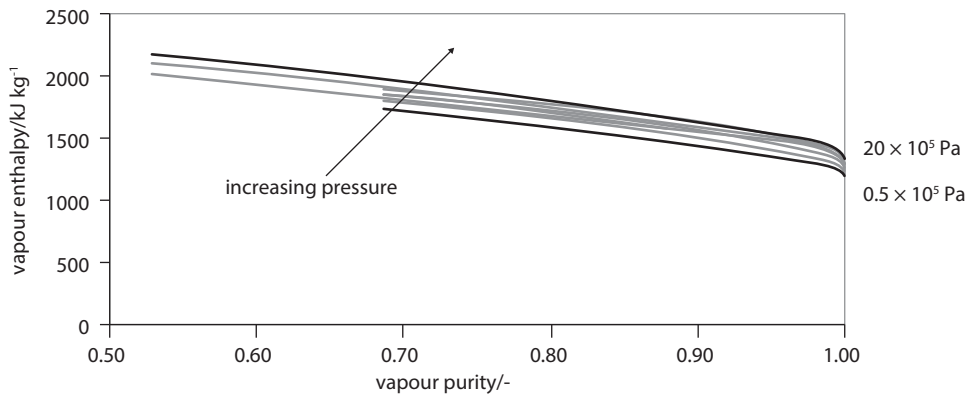


Figure 5.17 Enthalpy of the vapour as a function of the vapour concentration (= vapour purity) of the refrigerant ammonia. The pressure level rises from 0.5×10^5 Pa through 1, 2, 5, 10 to 20×10^5 Pa.

If the two enthalpy diagrams are combined, the vapour purity resulting for a given solution concentration can be determined at the same vapour enthalpy.

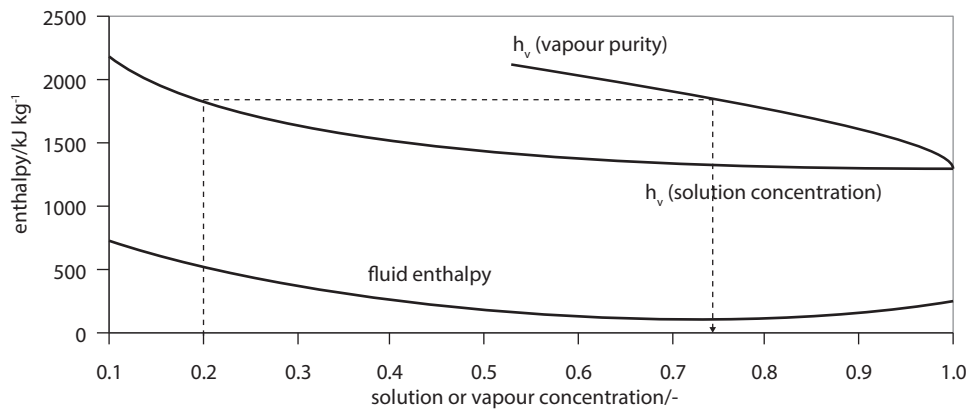


Figure 5.18 Combined enthalpy diagram for the fluid and vapour enthalpy as a function of the concentration of the solution or of the vapour concentration of the refrigerant.

For the concentration of solution of 20% shown, the vapour enthalpy is determined as a function of the concentration of solution at 1867 kJ kg^{-1} (for a total pressure of $20 \times 10^5 \text{ Pa}$). This vapour enthalpy also results at a vapour concentration of 74%, so the solution is in equilibrium at exactly this vapour concentration. From the enthalpy differences between vapour and fluid the energy quantity transferred can be calculated for each component of the absorption process.

Water – Lithium Bromide

The enthalpy of an aqueous lithium-bromide solution is calculated according to the following empirical equation of (McNeely 1979):

$$h_{\text{LiBr}+\text{H}_2\text{O}}(t,\xi)=\xi h_{\text{LiBr}}(t)+(1-\xi)\cdot h_{\text{H}_2\text{O}}(t)+\Delta h(t,\xi) \tag{5.27}$$

$$h_{\text{LiBr}}(t)=\sum_{i=0}^4 a_i \cdot t^i \tag{5.28}$$

$$\Delta h=\xi(1-\xi)\sum_{i=0}^4\sum_{j=0}^3 b_{ij}\cdot(2\xi-1)^j\cdot t^i \tag{5.29}$$

i	a_i	b_{i0}	b_{i1}	b_{i2}	b_{i3}
0	508.668	-1021.61	+36.8773	-0.186051	-7.51277×10^{-6}
1	-18.6241	- 533.308	+40.2847	-0.191198	0.0
2	$+9.85946 \times 10^{-2}$	+483.62	+39.9142	-0.199213	0.0
3	$- 2.50979 \times 10^{-5}$	+1155.13	+33.3572	-0.178258	0.0
4	$+4.15801 \times 10^{-8}$	+640.622	+13.1032	-0.077510	0.0

Table 5.3 Coefficients for the enthalpy calculations of McNeely.

5.6 Energy balances and performance figures of an absorption chiller

Ideal performance figures

According to the first law of thermodynamics – the conservation of energy principle – it stands that the uptaken thermal energy in the evaporator and in the generator must be equal to the rejected heat in the condenser and absorber.

$$\dot{Q}_E + \dot{Q}_G = \dot{Q}_C + \dot{Q}_A \quad (5.30)$$

The EER is defined as the ratio of the usable cooling energy $\dot{Q}_E$ (uptaken thermal energy in the evaporator from the room to be cooled) and the supplied generator power $\dot{Q}_G$.

$$EER_{th} = \frac{\dot{Q}_E}{\dot{Q}_G} \quad (5.31)$$

While the first law allows any kind of energy conversion, the second law of thermodynamics limits the possible energy conversion processes. Energy forms such as mechanical work, electrical energy, kinetic and potential energy can be completely converted into each other in reversible processes, as well as be transformed into limited convertible energies such as inner energy, enthalpy or heat. These completely convertible energies are referred to as exergy. In contrast, heat flows that are added or removed at a specific temperature T are associated principally with an entropy flow.

$$\dot{S}_Q = \frac{\dot{Q}}{T} \quad (5.32)$$

This entropy is not usually conserved but increases with all irreversible processes. The entropy balance equation has a source term or an entropy production rate $\dot{S}_{irr}$, which at a minimum is zero for reversible processes but is otherwise greater than zero. In a stationary case, the sum of all the added or removed entropy amounts to the system correspond to this entropy production rate.

$$\frac{dS}{dT} = \sum_i \frac{\dot{Q}_i}{T} + \dot{S}_{irr} = 0 \quad (5.33)$$

$$\dot{S}_{irr} = - \sum_i \frac{\dot{Q}_i}{T} \geq 0 \quad (5.34)$$

If heat is supplied to a system at a high temperature (positive sign and small entropy flow), then a minimum amount of heat at a lower temperature must accordingly be removed so that the entropy production is zero or larger than zero.

Heat is thereby not unlimitedly convertible in entropy free work (= exergy), a part is given off to the surroundings as so-called anergy. No more work can be produced from this anergy.

The first law of thermodynamics can be formulated as such, so that the sum of exergy and anergy remains constant for any process. The second law states that exergy is converted into anergy for any irreversible process, and the conversion from anergy into exergy is impossible.

Also, for the exergy balance, there is a source term, namely, the exergy loss term $\dot{E}_{loss}$, which at a minimum is zero for reversible processes or greater than zero for irreversible processes. In a stationary case it follows that:

$$\sum_i \dot{E}_{i, in} = \sum_i \dot{E}_{i, out} + \dot{E}_{loss} \quad \text{with } \dot{E}_{loss} \geq 0 \quad (5.35)$$

The calculation of exergy is reached through the combination of the first and second laws.

If, for example, heat is taken up at a high-temperature level T and is released at an ambient temperature T_0 , mechanical work W can be produced from this. The first law is:

$$\dot{Q} + \dot{Q}_0 + W = 0 \quad (5.36)$$

The entropy balance equation for this process is

$$\frac{\dot{Q}}{T} + \frac{\dot{Q}_0}{T_0} + \dot{S}_{irr} = 0 \quad (5.37)$$

The heat flow $\dot{Q}_0$ is eliminated because it is calculated out of the entropy balance equation and inserted into the energy balance.

$$\dot{Q}_0 = -\dot{Q} \frac{T_0}{T} - T_0 \dot{S}_{irr} \quad (5.38)$$

$$-W = \dot{Q} \left(1 - \frac{T_0}{T} \right) - T_0 \dot{S}_{irr} \quad (5.39)$$

Accordingly, in a reversible process ($\dot{S}_{irr} = 0$), the maximum recoverable work from the heat flow is limited due to the Carnot efficiency, which becomes higher with a greater temperature difference between the hot heat uptaking side and the low heat releasing side.

$$\eta_c = 1 - \frac{T_0}{T} \quad (5.40)$$

The maximum share of work of a heat flow is denoted as exergy.

$$\dot{E} = \dot{Q} \left(1 - \frac{T_0}{T} \right) \quad (5.41)$$

This share is calculated from the temperature differences to the ambient temperature T_0 where the heat is released as pure anergy to the surroundings. For example, in an absorption chiller, the released heat of the condenser and absorber is pure anergy.

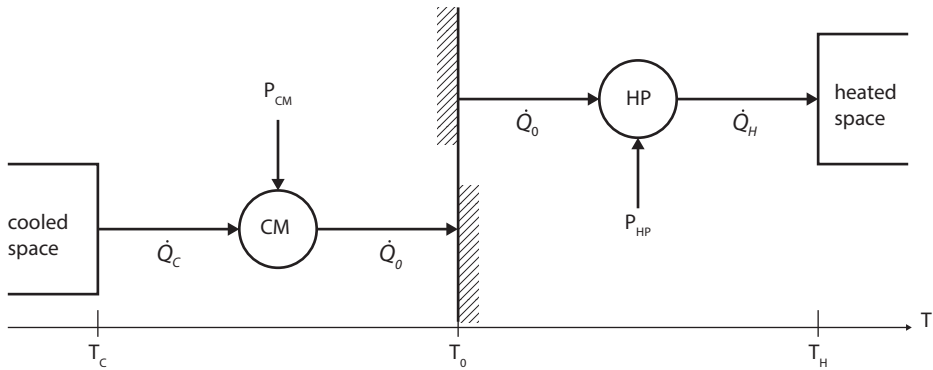


Figure 5.19 Temperature levels and power balances of a cooling machine and heat pump.

Firstly, the required exergy should be calculated that is necessary for heating and cooling activities.

In the heating mode, the heat flow goes from the higher heated room temperature T_H to the lower ambient temperature T_0 . The thereby lost exergy $\dot{E}_{loss}$ must be constantly replaced, i.e. supplied to the room. In the best case, this can take place with a reversible heat pump, which provides exactly this exergy and the rest of the heat flow for heating from anergy, i.e. it takes surrounding heat.

In the case of the heat pump, the supplied exergy is the mechanical (or thermal) operational power of the heat pump, which compensates for the heat loss. Because only anergy is taken from the surrounding, the released exergy is equal to zero. The supplied exergy is therefore the share of thermal heat calculated from the Carnot efficiency:

$$\dot{E}_{in} = \dot{E}_{loss} = \dot{Q}_H \left(1 - \frac{T_0}{T_H} \right) = \dot{Q}_H \eta_c \quad (5.42)$$

The remaining share of thermal heat is anergy $\dot{B}$ taken up from the surrounding.

$$\dot{B} = \dot{Q}_H - \dot{E}_{in} = \dot{Q}_H \left(1 - \left(1 - \frac{T_0}{T_H} \right) \right) = \dot{Q}_H \frac{T_0}{T_H} \quad (5.43)$$

For cooling activities, the temperature T_C of the room to be cooled is less than the ambient temperature. Heat flows from the ambient temperature through the not ideally insulated walls of the cooled room and thereby loses exergy, which must be provided from the cooling machine.

$$\dot{E} = \dot{Q}_{cool} \left(1 - \frac{T_0}{T_C} \right) \quad (5.44)$$

The Carnot efficiency will be negative, i.e. if the heat flows into the cooled room with a positive sign, the exergy flow will be negative – the cooled room loses exergy.

Or considered in reverse: if during cooling a heat flow, i.e. the cooling power $\dot{Q}_C$, is removed from the room (with a negative sign), exergy (positive) must be supplied. As before, the anergy share results from

$$\dot{B} = \frac{T_0}{T_C} \dot{Q}_{Cool} \quad (5.45)$$

The anergy is larger than the actual negative cooling power, that is exactly by the share of the exergy loss through the irreversible heat transfer in the wall. The cooling power from the room to be cooled is rejected to ambient temperature, i.e. consists completely of anergy, and the exergy balance is given by

$$\dot{E}_{in} = \dot{E}_{loss} = \dot{Q}_{Cool} \left(1 - \frac{T_0}{T_C} \right) = \dot{Q}_{Cool} \eta_C \quad (5.46)$$

For an electric compression cooling machine without losses the exergy input is the electrical operational power.

The EER as a relation between usable cooling capacity to input operational power is therefore given by

$$EER_{rev} = \left| \frac{\dot{Q}_{Cool}}{P_{el}} \right| = \left| \frac{\dot{Q}_{Cool}}{\dot{Q}_{Cool} \left(1 - \frac{T_0}{T_C} \right)} \right| = \frac{T_C}{T_0 - T_C} \quad (5.47)$$

For a thermal cooling machine, the electric compressor is replaced with a so-called thermal compressor, which works between a high generator temperature T_G and an ambient temperature T_0 .

For a reversible process, the supplied exergy in the generator would be exactly enough to cover the required exergy in the cool room.

$$\dot{Q}_G \left(1 - \frac{T_0}{T_G} \right) = \left| \dot{Q}_{Cool} \left(1 - \frac{T_0}{T_C} \right) \right| = |\dot{Q}_{Cool}| \frac{T_0 - T_C}{T_C} \quad (5.48)$$

Derived from this, the EER of the ideal thermally driven cooling process is given by

$$EER = \frac{|\dot{Q}_{Cool}|}{\dot{Q}_G} = \frac{\left(1 - \frac{T_0}{T_G} \right)}{\left(\frac{T_0 - T_C}{T_C} \right)} = \frac{T_C}{T_0 - T_C} \frac{T_G - T_0}{T_G} \quad (5.49)$$

Good EER values result especially when the condenser and absorber temperatures can be kept low because on the one hand, the temperature hub of the thermal engine increases between the absorber and generator and on the other hand, the temperature difference between the evaporator and condenser for an efficient cooling machine cyclical process remains low.

The real EER results from the product of the ideal EER and the exergetic efficiency.

$$EER_{real} = EER_{Carnot} \times \zeta \quad (5.50)$$

The exergetic efficiency is defined as the ratio of the minimum required exergy for the cooling process (this can be denoted as the useful exergy flow) to the total utilised exergy.

$$\zeta = \frac{\text{useful exergy flow}}{\text{total exergy flow}} = \frac{\dot{E}_{min}}{\dot{E}_{real}} \quad (5.51)$$

Real electric cooling machines or heat pumps have exergetic efficiencies of about 50%.

For a complete assessment of the EERs, the utilised energy for the process should be assessed on a primary energy basis. The primary energy EER results from the primary energy efficiency of the chosen energy carrier. Today, for electricity this efficiency is at about 37%, for heating at about 90%.

$$EER_{primary} = EER_{Carnot} \times \zeta \times \eta_{primary} \quad (5.52)$$

The same result is achieved if energy and entropy balances are considered. In an ideal absorption process, the cyclic process of the refrigerant is regarded as loss-free and thermodynamically reversible. Based on the principle of conservation of energy, the heat taken up in the evaporator and in the generator must equal the delivered heat in the condenser and absorber.

$$Q_E + Q_G = Q_C + Q_A \quad (5.53)$$

Since an ideal process occurs reversibly, the entropy remains constant according to the second law. The reduction of entropy in the condenser equates to the entropy increase in the evaporator, and the entropy reduction in the absorber equates to the entropy increase in the generator.

$$\frac{\dot{Q}_C}{T_C} = \frac{\dot{Q}_E}{T_E} \quad (5.54)$$

$$\frac{\dot{Q}_A}{T_A} = \frac{\dot{Q}_G}{T_G} \quad (5.55)$$

The EER for cooling or heating can be represented by reforming the above equations together with the energy balance equation once again as a temperature ratio:

$$EER_{Cooling} = \frac{\dot{Q}_E}{\dot{Q}_G} = \frac{T_G - T_A}{T_G} \times \frac{T_E}{T_C - T_E} \quad (5.56)$$

$$EER_{Heating} = \frac{\dot{Q}_A + \dot{Q}_C}{\dot{Q}_G} = 1 + \frac{T_G - T_A}{T_G} \times \frac{T_E}{T_C - T_E} = 1 + EER_{Cooling} \quad (5.57)$$

Example 5.8

Calculation of the ideal EERs for an absorption cooling machine with the following temperature ratios:

Temperature/°C	Example 1	Example 2	Example 3	Example 4	Example 5
Evaporator T_E	-10	-10	+5	+5	+5
Condenser T_C	30	50	50	30	50
Absorber T_A	30	50	50	30	50
Generator T_G	140	140	140	100	100
$EER_{Cooling}$	1.75	0.95	1.34	2.09	0.83
$EER_{Heating}$	2.75	1.95	2.34	3.09	1.83

Good EERs result if the condenser and absorber temperatures can be kept low, since on the one hand, the temperature lifting capacity of the thermal machine rises between the absorber and generator, and on the other hand, the temperature difference between the evaporator and condenser for an efficient chiller cyclic process remains small.

Example 5.9

An absorption cooling machine is supplied with 100 kW heat at 90°C in the generator. With this, it takes 70 kW cooling energy from the room to be cooled at 6°C and rejects heat to the surroundings at 30°C. From these values the exergy loss should be calculated, as well as the maximum cooling capacity for a reversible process.

The exergy input to the generator is calculated with the Carnot efficiency:

$$\dot{E}_G = \dot{Q}_G \left(1 - \frac{T_0}{T_G} \right) = 100 \text{ kW} \left(1 - \frac{303 \text{ K}}{363 \text{ K}} \right) = 16.5 \text{ kW}$$

In order to remove heat from the cool room, exergy must be supplied:

$$\dot{E}_{Cool} = \dot{Q}_{Cool} \left(1 - \frac{T_0}{T_E} \right) = 70 \text{ kW} \left(1 - \frac{303 \text{ K}}{279 \text{ K}} \right) = -6 \text{ kW}$$

The exergy loss is derived from the exergy balance equation. The exiting exergy is zero because the rejected heat is emitted at ambient temperature.

$$\sum_i \dot{E}_{i,in} = \dot{E}_G + \dot{E}_{Cool} = \dot{E}_{loss} = 10.5 \text{ kW}$$

Accordingly, of the 16.5 kW of supplied exergy, 10.5 kW is lost.

If the cooling machine would work reversibly, then the complete supplied exergy flow could be used for the cooling production.

$$\dot{Q}_G \left(1 - \frac{T_0}{T_G} \right) = \left| \dot{Q}_{Cool,rev} \left(1 - \frac{T_0}{T_E} \right) \right|$$

The reversible maximum cooling capacity would therefore be

$$\dot{Q}_{\text{Cool, rev}} = \dot{Q}_G \frac{T_G - T_0}{T_G} \times \frac{T_E}{T_0 - T_E} = 1.92 \dot{Q}_G = 192 \text{ kW}$$

i.e. 2.75 times higher than the actual cooling capacity.

The reversible EER is

$$EER_{\text{Cool, rev}} = \frac{T_E}{T_0 - T_E} \frac{T_G - T_0}{T_G} = \frac{279 \text{ K}}{303 \text{ K} - 279 \text{ K}} \frac{363 \text{ K} - 303 \text{ K}}{303 \text{ K}} = 1.92$$

The ratio of the minimum required exergy for cooling to the actual utilised exergy of the generator heat equates to the exergy efficiency. This is 36.3% so that the real EER amounts to 0.7.

$$\zeta = \frac{|\dot{E}_{\text{Cool}}|}{\dot{E}_G} = \frac{6 \text{ kW}}{16.5 \text{ kW}} = 0.36$$

Example 5.10

In comparison to absorption cooling, the exergetic efficiency of an electrical compression cooling machine should be determined, which operates at the same temperature conditions as the above example, namely, 6°C cold water and 30°C cooling (=heat rejection) water temperature. For the cooling capacity of 70 kW at an EER of 3.0, an electrical power of 23.3 kW is taken up.

For the required exergy of 6 kW in order to keep the cool room at 6°C, 23.3 kW of exergy is supplied via the electrical power. The exergetic efficiency is around 25.7% and therefore lower than the absorption cooling machine. The reversible EER is, however,

$$EER_{\text{Cool, rev}} = \frac{T_E}{T_C - T_E} = \frac{279 \text{ K}}{303 \text{ K} - 279 \text{ K}} = 11.62$$

and therefore clearly higher than for the absorption cooling machine. The real EER is higher than for the absorption cooling machine. The primary energy assessed EER is

$$EER_{\text{primary}} = EER_{\text{Carnot}} \times \zeta \times \eta_{\text{primary}} = 11.62 \times 0.257 \times 0.37 = 1.1$$

The primary energy EER of thermal cooling is lower at

$$EER_{\text{primary}} = 2.3 \times 0.375 \times 0.9 = 0.78$$

Real performance figures and enthalpy balances

In an ideal absorption cooling process, Example 5.8 shows that EERs over 1.0 can readily occur. In a real chiller, however, irreversible processes occur during absorption of the refrigerant in the solution. The real EERs depend furthermore on whether the freed amounts of heat in the absorber, condenser and rectifier can be recovered and supplied to the process again.

To calculate the real performance figures, the enthalpies of the liquid (solvent + absorbed refrigerant) and of the vapour (refrigerant plus co-evaporated solvent) for the different equilibrium concentrations in the absorber and generator must be known. For pure ammonia in the evaporator and condenser, the enthalpies are calculated using Equation (5.18), Equation (5.19), Equation (5.20) and Equation (5.21); for water equations, from Glück are used Equation (5.14) and Equation (5.15).

The enthalpy of the solution depends on the refrigerant concentration in the solution ξ , and on the temperature or pressure and is calculated using Equation (5.22), Equation (5.23), Equation (5.24), Equation (5.25) and Equation (5.26) for ammonia/water mixes and Equation (5.27) for water/lithium bromide.

To calculate the performance figure, a view of the energy flows in the generator is sufficient, since power has only to be supplied there. The heat to be supplied to the generator contains, apart from the enthalpy of the expelled refrigerant vapour, the heat of refrigerant desorption, which is equal to the absorption heat released in the absorber. The EER during a single-lift absorption process is therefore always under 1.0 due to non-ideal heat recovery, and in commercial machines is typically between 0.5 and 0.8.

The rich solution from the absorber, preheated by a heat exchanger, is supplied to the generator (specific enthalpy $h_{r,g}$). The specific heating energy of the generator related to the vapour mass flow is termed q_g . From the generator, heat is removed by the expelled vapour (h_v), which for ammonia/water systems must first be freed in a rectifying column from any solvent still present. The rectification heat (q_r) freed by this condensation is part of the dissipated heat. Furthermore, heat is dissipated from the generator by the discharge of the hot poor solution ($h_{p,g}$).

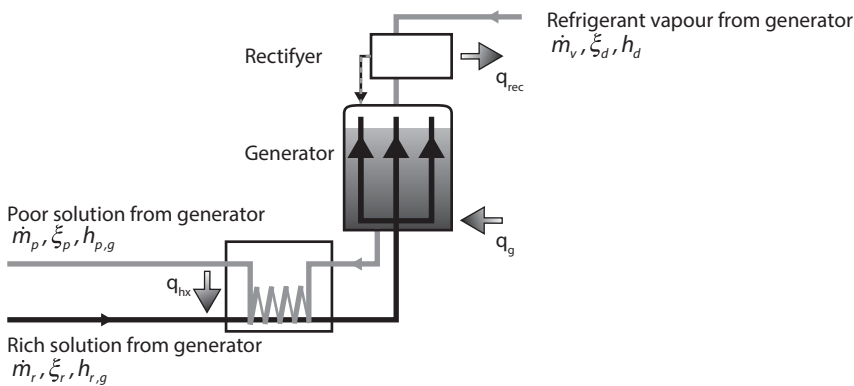


Figure 5.20 Energy balances of the generator with rectifier and heat exchanger between the poor hot solution and rich cold solution.

The energy balance related to the vapour mass flow $\dot{m}_v$ thus reads:

$$\frac{\dot{m}_r}{\dot{m}_v} h_{r,g} + q_g = q_{rec} + h_v + \frac{\dot{m}_a}{\dot{m}_v} h_{a,g} \quad (5.58)$$

Using the specific backflow relation between rich solution flow and refrigerant vapour mass flow

$$f = \frac{\dot{m}_r}{\dot{m}_v}$$

there follows:

$$\begin{aligned} q_g &= q_{rec} + h_v + f(h_{p,g} - h_{r,g}) - h_{p,g} \\ &= q_{rec} + h_v + \left(\xi_v - \xi_p \right) \frac{h_{p,g} - h_{r,g}}{\xi_r - \xi_p} - h_{p,g} \end{aligned} \quad (5.59)$$

Instead of the enthalpy difference between $h_{p,g}$ and $h_{r,g}$ at the generator entry and exit, the enthalpy difference of the poor and rich solution at the absorber entry and exit $h_{p,a}$ and $h_{r,a}$ can also be used.

$$q_g = q_{rec} + h_v + \left(\xi_v - \xi_p \right) \frac{h_{p,a} - h_{r,a}}{\xi_r - \xi_p} - h_{p,a} \quad (5.60)$$

Rectification heat q_{rec}

The rectification calorific losses depend on the refrigerant concentration at which the expelled gas leaves the generator, i.e. which equilibrium between solution and vapour is present. The purest possible vapour mass flow $\dot{m}_v$ leaving the rectifier is the vapour mass flow $\dot{m}_v^1$ produced in the generator minus the liquid quantity condensed in the rectifier $\dot{m}_\ell$

$$\dot{m}_v = \dot{m}_v^1 - \dot{m}_\ell \quad (5.61)$$

with the refrigerant concentration

$$\dot{m}_v \xi_v = \dot{m}_v^1 \xi_v^1 - \dot{m}_\ell \xi_\ell \quad (5.62)$$

From this results for the mass flows

$$\dot{m}_v^1 = \dot{m}_v \frac{\xi_v - \xi_\ell}{\xi_v^1 - \xi_\ell} \quad (5.63)$$

The lower the original refrigerant vapour concentration ξ_v^1 in the generator, the higher the condensing liquid mass flow $\dot{m}_\ell$ and the rectification heat become.

$$\dot{m}_\ell = \dot{m}_v \frac{\xi_v - \xi_v^1}{\xi_v^1 - \xi_\ell} \quad (5.64)$$

The energy q_{rec} released from the rectifying column, related to the pure vapour mass flow $\dot{m}_v$, therefore results from the difference between the initial vapour enthalpy $\dot{m}_v^1 h_v^1$ and pure vapour $\dot{m}_v h_v$ and the condensed liquid enthalpy $\dot{m}_\ell h_\ell$.

$$\begin{aligned} q_{rec} &= \frac{1}{\dot{m}_v} (\dot{m}_v^1 h_v^1 - \dot{m}_\ell h_\ell - \dot{m}_v h_v) \Leftrightarrow \\ q_{rec} + h_v &= \frac{\xi_v - \xi_\ell}{\xi_v^1 - \xi_\ell} h_v^1 - \frac{\xi_v - \xi_v^1}{\xi_v^1 - \xi_\ell} h_\ell \\ &= \frac{\xi_v - \xi_\ell}{\xi_v^1 - \xi_\ell} (h_v^1 - h_\ell) + h_\ell \end{aligned} \quad (5.65)$$

To the enthalpy of the liquid h_ℓ , therefore, the enthalpy is added which results from the gradient

$$\frac{h_v^1 - h_\ell}{\xi_v^1 - \xi_\ell}$$

multiplied by the concentration difference of the pure vapour to the liquid $\xi_v - \xi_\ell$. From the combined enthalpy-concentration diagram, the enthalpy of the pure vapour and the rectification heat can be read off.

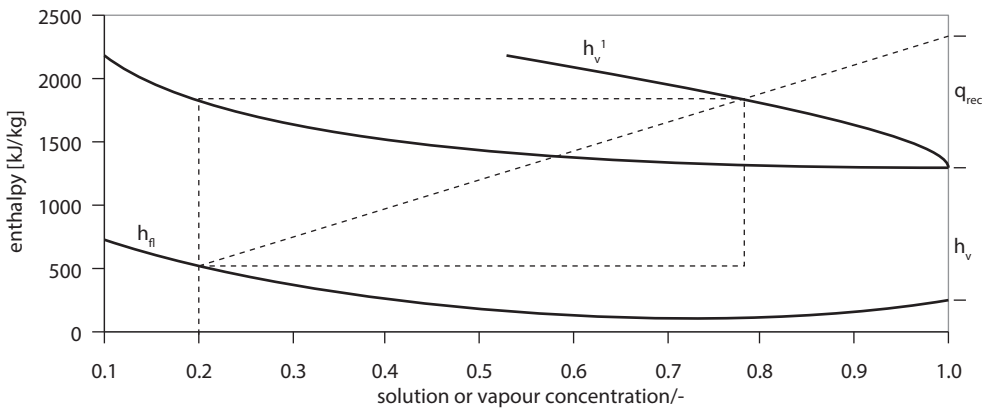


Figure 5.21 Determination of the rectification heat q_{rec} at a concentration of ammonia-water solution of 20% and an initial vapour concentration of 79.5% at 20×10^5 Pa.

From the gradient of the enthalpy difference as a function of the concentration difference, the result is, at a vapour concentration of 1.0, i.e. pure ammonia, the sum of the vapour and rectification enthalpy q_{rec} .

Example 5.11

Calculation of the rectification heat using Equation (5.65) for the two concentrations of solution 0.2 and 0.5 at 20×10^5 Pa total pressure, if the exiting vapour is to have an NH_3 concentration of 100%.

Solution concentration/-	$\xi_{v1}/-$	$h_{v1}/\text{kJ kg}^{-1}$	$h_{l1}/\text{kJ kg}^{-1}$	$h_v/\text{kJ kg}^{-1}$	$q_{rec}/\text{kJ kg}^{-1}$
0.2	0.74	1867	553	1295	1206
0.5	0.98	1439	181	1295	195

At low solution concentrations it is far more important to supply the rectification heat to the process again than at high concentrations with high initial vapour purity.

The temperature in the rectifier results from the desired vapour purity. The pure vapour is in equilibrium with a liquid whose temperature must be all the lower, the purer the vapour has to be.

Absorption heat

To complete the energy balance of the generator, the absorber balance

$$(\xi_v - \xi_p) \frac{h_{p,a} - h_{r,a}}{\xi_r - \xi_p} - h_{p,a} = h_\lambda$$

still has to be created, which results from the enthalpy difference between the poor and rich solution at the absorber entry and exit, and which is also described in the literature as h_λ . For the calculation, giving the degassing width $\xi_r - \xi_p$ and the desired vapour purity h_λ is sufficient. The enthalpy of the solution is calculated for the low-pressure level in the absorber-using equation .

Example 5.12

Determination of the enthalpy h_λ for degassing widths $\xi_r - \xi_p = 0.25 - 0.20$ or $0.52 - 0.50$ at a low pressure level of 5×10^5 Pa for an ammonia-water system.

$\xi_r/-$	$\xi_p/-$	$h_r/\text{kJ kg}^{-1}$	$h_p/\text{kJ kg}^{-1}$	$h_\lambda/\text{kJ kg}^{-1}$
0.25	0.20	206	286	994
0.52	0.50	-85.7	-73.7	372.2

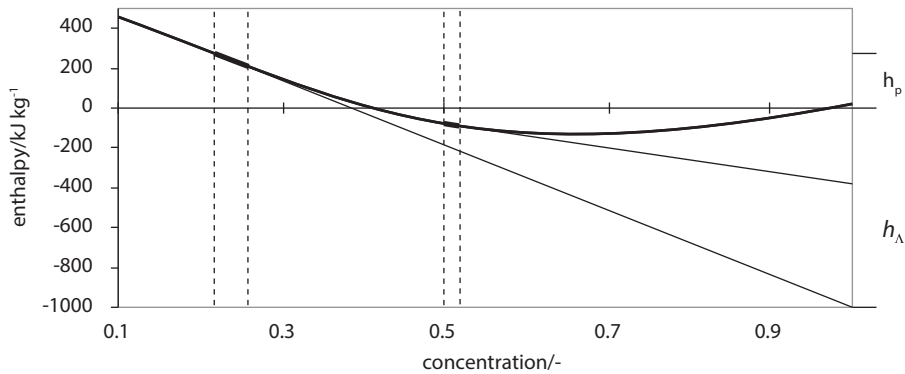


Figure 5.22 Enthalpy h_λ of the low and high concentrations of solution.

Energy efficiency ratio

Taking into account the rectification heat q_{rec} (if necessary), the enthalpy of the pure vapour h_v and the enthalpy balance at the absorber, the EER of the absorption chiller can now be calculated for different evaporator temperatures and pressure levels. The concentration of solution in the absorber is adjusted to the evaporator pressure level, i.e. it is so high that at evaporator pressure, refrigerant can still be absorbed. If low evaporator temperatures are to be achieved, the concentration of solution has to be lowered. For example, with a 20% poor ammonia solution, absorption can still take place at a pressure level under 10^5 Pa, i.e. at an evaporator temperature of -30°C . To accelerate the absorption process, the vapour pressure over the solution in the absorber should be higher than the equilibrium vapour pressure. In the following example, an ammonia-water absorption process is considered at an evaporator temperature of -10°C (corresponding to a pressure level of 2.75×10^5 Pa), and the solution is degassed to 20% refrigerant concentration. Alternatively, a process with a higher evaporator temperature of $+5^\circ\text{C}$ is analysed, which enables higher concentrations of solution.

Nr	$\xi_p/-$	$\xi_r/-$	Evaporator temperature $T_e/^\circ\text{C}$	Evaporation enthalpy $h_g/\text{kJ kg}^{-1}$	Vapour pressure evaporator/Pa
1	0.2	0.25	-10	1296	2.75×10^5
2	0.5	0.52	+5	1258	4.85×10^5

Table 5.4 Boundary conditions for two absorption processes with low and high evaporator temperatures.

Nr.	Rectification heat $q_{\text{rec}}/\text{kJ kg}^{-1}$	Enthalpy of pure vapour $h_v/\text{kJ kg}^{-1}$	Absorption enthalpy $h_\lambda/\text{kJ kg}^{-1}$	Generator heat $q_g/\text{kJ kg}^{-1}$	EER/-
1	966	1290	994	$1206 + 1295 + 994$ $= 3495$	0.4
2	195	1295	372	$195 + 1295 + 372$ $= 1860$	0.69

Table 5.5 Enthalpies and performance figures of two absorption processes.

The performance figures can be improved if both the rectification heat and the heat of absorption during the process are recovered and used, with the usability of the heat dependent on the temperature level of the freed heat. The heat freed in the rectifier at high temperatures can, for example, be used to preheat the cold rich solution flowing from the absorber. The heat freed in the absorber develops especially at high solvent concentrations at low temperatures, and is often released unused to the environment. In single-lift absorption refrigeration systems, real performance figures of approximately 0.5 - 0.7 can thus be achieved.

5.7 Static absorption cooling model

For the simulation of a real absorption cooling machine, only the external water distribution circuits with their entry temperatures and mass flows are known. The internal temperatures of the refrigerant and the solution must be calculated from the heat transfer characteristics of the heat exchanger. This method will be demonstrated in the following example of the LiBr-water absorption cooling machine, which does not need a rectifier, because the lithium bromide does not evaporate in the generator. The model description essentially follows the approach of (Ziegler, 1998), who developed a so-called characteristic equation for absorption cooling machines.

Internal temperatures are labelled with an upper case T , external temperatures with a lower case t .

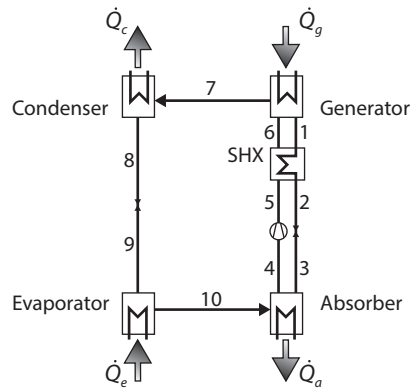


Figure 5.23 System schematic with process point numbering and external heat flows in the generator, condenser, evaporator and absorber.

Evaporator

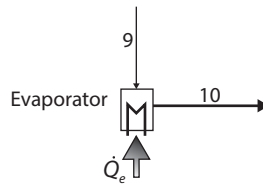


Figure 5.24 Uptaken heat flow in the evaporator.

The cooling capacity of the evaporator can be described by the mean temperature difference between the mean external $\bar{t}_e$ and the internal temperature T_e multiplied by the overall heat transfer coefficient of the heat exchanger UA_e . This heat flux is equal to the internal evaporator enthalpy difference q_e multiplied by the vapour mass flow rate $\dot{m}_v$.

$$\dot{Q}_e = (\bar{t}_e - T_e) UA_e = \dot{m}_v q_e \quad (5.66)$$

$$q_e = h_{10} - h_9 \quad (5.67)$$

q_e is defined as the enthalpy difference between the produced water vapour enthalpy h_{10} and the water enthalpy when entering the evaporator h_9 .

Absorber

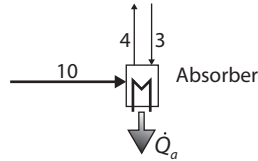


Figure 5.25 Released absorption heat in the absorber.

In the absorber, the transferred heat $UA_a(T_a - \bar{t}_a)$ between the external heat rejection circuit with mean temperature $\bar{t}_a$ and internal temperature T_a equals the enthalpy $\dot{m}_v h_{10}$ generated by the absorption of refrigerant (state point 10) plus the enthalpy from the weak solution from the generator $(\dot{m}_r - \dot{m}_v)h_3$ (state point 3) minus the rich solution enthalpy (4) returned to the generator. The weak solution mass flow is obtained from the difference of rich solution entering the generator and the expelled vapour mass flow.

$$\begin{aligned}\dot{Q}_a &= (T_a - \bar{t}_a)UA_a = \dot{m}_v h_{10} + (\dot{m}_r - \dot{m}_v)h_3 - \dot{m}_r h_4 \\ &= \dot{m}_v (h_{10} - h_3) + \dot{m}_r (h_3 - h_4) = \dot{Q}_e A_E + \dot{Q}_{ax}\end{aligned}\quad (5.68)$$

The fact that the cool rich solution returning to the generator (4) does not pre-cool completely the hot solution from the generator (3) in the solution heat exchanger causes an additional heat input to the absorber, which is named $\dot{Q}_{ax}$.

$$\dot{Q}_{ax} = \dot{m}_r (h_3 - h_4) \quad (5.69)$$

The factor A_E describes the enthalpy difference between refrigerant vapour h_{10} and weak solution h_3 normalised with q_e .

$$A_E = \frac{(h_{10} - h_3)}{q_e} \quad (5.70)$$

Generator

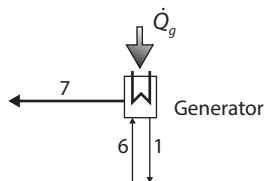


Figure 5.26 Supplied generator heat and state points in the generator.

The required generator heat is calculated from the enthalpy difference needed to heat the cool rich solution from the absorber (6) to the generator temperature level of the hot exiting weak solution

enthalpy (1) $(\dot{m}_r - \dot{m}_v)h_1$ plus the enthalpy needed for the evaporated refrigerant exiting to the condenser (7).

$$\begin{aligned}\dot{Q}_g &= (\bar{t}_g - T_g)UA_g = \dot{m}_v h_7 + (\dot{m}_r - \dot{m}_v)h_1 - \dot{m}_r h_6 \\ &= \dot{m}_v(h_7 - h_1) + \dot{m}_r(h_1 - h_6) = \dot{Q}_e G_E + \dot{Q}_{gx}\end{aligned}\quad (5.71)$$

$\dot{Q}_{gx}$ designates again the loss in the solution heat exchanger:

$$\dot{Q}_{gx} = \dot{m}_r(h_1 - h_6) \quad (5.72)$$

The factor G_E gives the normalised enthalpy differences between the exiting refrigerant vapour and the weak solution leaving the generator:

$$G_E = \frac{(h_7 - h_1)}{q_e} \quad (5.73)$$

Condenser

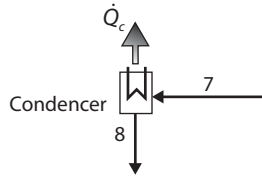


Figure 5.27 Released condensation heat in the condenser.

The heat transfer in the condenser equates to the released condensation heat of the refrigerant.

$$\dot{Q}_c = (T_c - \bar{t}_c)UA_c = \dot{m}_v(h_7 - h_8) = \dot{Q}_e \frac{h_7 - h_8}{q_e} = \dot{Q}_e C_E \quad (5.74)$$

with

$$C_E = \frac{h_7 - h_8}{q_e} \quad (5.75)$$

With the introduction of the factors A_E , C_E and G_E as well as the solution heat exchanger losses, all the heat transfer equations can now be expressed as a function of the evaporator power.

$$\dot{Q}_e = UA_e(\bar{t}_e - T_e) \quad (5.76)$$

$$\dot{Q}_e C_E = UA_c(T_c - \bar{t}_c) \quad (5.77)$$

$$\dot{Q}_e A_E - \dot{Q}_{ax} = UA_a(T_a - \bar{t}_a) \quad (5.78)$$

$$\dot{Q}_e G_E - \dot{Q}_{gx} = UA_g(\bar{t}_g - T_g) \quad (5.79)$$

Another equation for the internal temperatures is given via the solution field, which describes the slope of the isosteres in the $\log p$ over $1/T$ diagram.

The ratio of isostere slopes is described by the so-called Dühring factor B , which for the water-lithium bromide material pair lies between 1.1 and 1.2 (Kohlenbach et al., 2004).

$$(T_c - T_e)B = T_g - T_a \quad (5.80)$$

If the internal temperatures from Equation (5.76) and Equation (5.79) are applied until eliminated and inserted into Equation (5.80), the following solution arises for the average external temperatures.

$$\bar{t}_c - \bar{t}_e + \dot{Q}_e \left(\frac{C_E}{UA_c} + \frac{1}{UA_e} \right) B = \bar{t}_g - \bar{t}_a - \dot{Q}_e \left(\frac{G_E}{UA_g} + \frac{A_E}{UA_a} \right) - \frac{\dot{Q}_{gx}}{UA_g} - \frac{\dot{Q}_{ax}}{UA_a} \quad (5.81)$$

The external temperature difference between generator and absorber, and also condenser and evaporator are defined as the so-called double characteristic temperature difference.

$$\Delta\Delta t = (\bar{t}_g - \bar{t}_a) - (\bar{t}_c - \bar{t}_e)B \quad (5.82)$$

The losses caused by the solution heat exchangers are denoted with $\Delta\Delta t_{min,E}$.

$$\Delta\Delta t_{min,E} = \frac{\dot{Q}_{gx}}{UA_g} + \frac{\dot{Q}_{ax}}{UA_a} \quad (5.83)$$

Through the summary of all other factors in the slope s_E arises a simple linear equation, which can be used for a first approximation to calculate the cooling capacity from absorption machines with the constant factors $\Delta\Delta t_{min,E}$ and s_E .

$$s_E = \left[\left(\frac{C_E}{UA_c} + \frac{1}{UA_e} \right) B + \left(\frac{G_E}{UA_g} + \frac{A_E}{UA_a} \right) \right]^{-1} \quad (5.84)$$

$$\dot{Q}_E = s_E (\Delta\Delta t - \Delta\Delta t_{min,E}) \quad (5.85)$$

If the mean temperatures are simply calculated as the mean value of the inlet and outlet temperatures, then the mean temperatures can be replaced by the entry temperatures.

$$\begin{aligned} \bar{t}_i &= \frac{(t_{i,in} + t_{i,out})}{2} \\ t_{out,i} &= 2\bar{t}_i - t_{in} \end{aligned} \quad (5.86)$$

For example, the evaporator Equation (5.85) can be replaced by the following equation from which the mean external temperature can be eliminated.

$$s_E (\bar{t}_G - \bar{t}_A - B \cdot \bar{t}_C + B \cdot \bar{t}_E - \Delta\Delta t_{min,E}) = 2\dot{m}_E c_{H_2O} (-\bar{t}_E + t_{E,in}) \quad (5.87)$$

Using an iteration, the precision of the equation can be improved with changing external temperatures. For this, after the calculation of all factors and the cooling capacity, the internal temperatures are determined anew with Equation (5.76), Equation (5.77), Equation (5.78) and Equation (5.79) and with this all enthalpies.

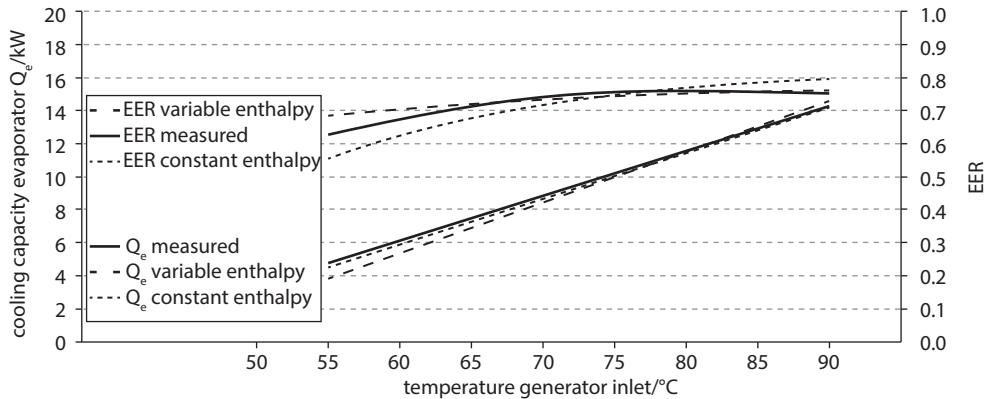


Figure 5.28 Comparison of the cooling capacity and EERs of a 10 kW LiBr/H₂O cooling machine simulated with constant enthalpies and enthalpies adjusted with iterations.

5.8 Parameter identification for the static absorption cooling machine model

The problem with the characteristic equation is that for most of the commercial absorption cooling machines, the heat transfer capacity UA , the solution mass flow and the heat recovery efficiency of the solution heat exchanger are not known. A simple parameter identification procedure allows the determination of these values as a function of the external entry temperature of the evaporator, heat rejection and generator cycles, as well as the cooling capacity of the machine. The parameters of the model are the temperature differences between external and internal mean temperatures at the evaporator, absorber, generator and condenser, the heat recovery efficiency of the solution heat exchanger and the degassing width. Another three parameters are given via the external temperature differences between entry and exit at the cold water, cooling water and heating cycle.

Firstly, the mean temperatures in the external cycles are calculated with the assumption of a linear temperature curve. For the heat rejection of the absorber and condenser, usually only the entry temperature in the absorber and the exit temperature of the condenser are known when components are in series connection. In a first approximation, it is assumed that the temperature increase from the absorber and condenser cooling cycle are equal.

$$\bar{t}_a = \left(t_{a,in} + \frac{\Delta t_{ac}}{4} \right) \quad (5.88)$$

$$\bar{t}_c = \left(t_{a,in} + \frac{\Delta t_{ac}}{2} + t_{c,out} \right) \times 0.5 \quad (5.89)$$

With the mean external temperature and the temperature difference between external and internal temperature given as a parameter, the mean internal temperature for each component is calculated. With the given internal temperatures and heat recovery efficiency of the solution heat

exchanger, all internal enthalpies and solution concentrations can be calculated. The refrigerant mass flow is determined from the given cooling capacity and the enthalpy difference between refrigerant vapour and fluid.

$$\dot{m}_{H_2O} = \frac{\dot{Q}_e}{h_{10} - h_9} \quad (5.90)$$

The mass flow of the rich solution can be calculated with the refrigerant mass flow and the degassing width.

$$\dot{m}_{rs} = \frac{X_{weak}}{X_{weak} - X_{rich}} \dot{m}_{H_2O} \quad (5.91)$$

The heat transfer capacities of the individual components are determined by the following:

$$UA_e = \frac{\dot{Q}_e}{\bar{t}_e - T_e} \quad (5.92)$$

$$UA_a = \frac{h_3(\dot{m}_{rs} - \dot{m}_{H_2O}) + h_{10}\dot{m}_{H_2O} - h_4\dot{m}_{rs}}{T_a - \bar{t}_a} \quad (5.93)$$

$$UA_c = \frac{(h_7 - h_8)\dot{m}_{H_2O}}{T_c - \bar{t}_c} \quad (5.94)$$

$$UA_g = \frac{h_1(\dot{m}_{rs} - \dot{m}_{H_2O}) + h_7\dot{m}_{H_2O} - h_6\dot{m}_{rs}}{\bar{t}_g - T_g} \quad (5.95)$$

The values CE , GE and AE as well as the loss coefficient $\dot{Q}_{gx}$ and $\dot{Q}_{ax}$ are calculated next. With these, the heat rejection capacity at the condenser and absorber, as well as the heat capacity at the generator can be determined.

$$\dot{Q}_c = \dot{Q}_e C_E \quad (5.96)$$

$$\dot{Q}_a = \dot{Q}_e A_E - \dot{Q}_{ax} \quad (5.97)$$

$$\dot{Q}_g = \dot{Q}_e G_E - \dot{Q}_{gx} \quad (5.98)$$

Now the new external exit temperature at the absorber can be calculated:

$$\bar{t}_{a,out} = \frac{UA_a T_a + UA_c T_c - \dot{Q}_a - \dot{Q}_c - \frac{UA_a \bar{t}_{a,in} - UA_c \bar{t}_{c,out}}{2}}{\frac{UA_a + UA_c}{2}} \quad (5.99)$$

From the calculated capacities and external temperature differences, all external mass flows can then be determined.

$$\dot{m}_g = \frac{\dot{Q}_g}{c_{H_2O}(t_{g,in} - t_{g,out})} \quad (5.100)$$

$$\dot{m}_a = \frac{\dot{Q}_a}{c_{H_2O}(t_{a,out} - t_{a,in})} \quad (5.101)$$

$$\dot{m}_c = \frac{\dot{Q}_c}{c_{H_2O}(t_{c,out} - t_{a,out})} \quad (5.102)$$

$$\dot{m}_e = \frac{\dot{Q}_e}{c_{H_2O}(t_{e,in} - t_{e,out})} \quad (5.103)$$

In an iteration process, the external absorber and condenser temperatures are slightly increased and the mass flows are calculated anew. The iteration is finished when the absorber mass flow equals the condenser mass flow. The model delivers all UA values of the heat exchanger, as well as the mean internal temperatures of each component, the Dühring parameter, the mass flow of the rich solution and the three external mass flows at the evaporator, absorber/condenser and generator.

The assumed temperature difference between external and internal cycle are now iterated until the Dühring parameter is almost 1.2 and the external mass flows correspond to values stated by the manufacturer.

Absorption technology and solar plants

The temperature level necessary in the generator results, as already shown, from the operating conditions of the condenser and absorber (water or air-cooled) and from the evaporation temperatures or the concentration of solution of the system. The solar plant often has to supply a temperature level over 100°C via a heat exchanger. Conventional heat storage with buffer stores is necessarily affected by high losses due to the high-temperature difference from the environment of around 70 - 100°C. Cold water storage causes less losses, due to the lower temperature difference from the environment, particularly since energy already converted is then stored. The storage capacity is limited, however, due to the small usable temperature range of a few kelvins (10 K at most, using cooled ceilings). Of future interest are concepts for storing condensed refrigerant in the chiller itself, which can then be evaporated if there is insufficient solar irradiation.

With the given temperature level of the solar circuit, the efficiency of the solar plant and the surface-related performance are determined. Via the real performance figure of the cooling process, the necessary surface of the solar plant can be determined at a given cooling load with a maximum irradiation of 1 000 W m⁻².

The buffer store volume on the primary side or the cold water side should be designed for short-term energy storage of 1 - 3 days.

5.9 Open cycle desiccant cooling

Desiccant cooling systems (DCSs) are a mature technology for air-conditioning buildings and are particularly suitable for the application of thermal solar energy, due to the low driving temperatures of around 60 - 80°C.

The technology is based on the principle of outside air dehumidification by an adsorbent such as silica gel or lithium chloride. After pre-cooling the dried fresh air with maximally humidified room exhaust air, subsequent evaporative cooling produces the desired supply air temperatures of 16 - 18°C. The desiccant cooling process can be continuously operated with slowly rotating sorption wheels or with liquid desiccants, where the outside air humidity taken up in the adsorbent is transferred to the exit air heated by supply of solar or waste heat.

For process reasons, with open cycle sorption technology, no chilling water with the usual supply temperatures of 6 - 10°C can be produced. The chilling carrier in open sorption is the moistened air, which is injected directly into the space. Due to the limited dehumidifying performance of the adsorbents used (about 6 g of water per kilogram of dry air), in very humid climates, double-stage DCSs must be used or the systems have to be coupled with compression or absorption chillers, to avoid direct supply air humidification.

Sorption systems are also very efficiently used purely to dehumidify outside air, so that the very energy-intensive dew point dehumidification in a compression chiller is not occurring. The removal of sensible heat in the space can then take place via surface cooling (usually cooled ceilings).

As an air-led system with cooling load removal only by cooled outside air, the application is particularly suitable if there is a high fresh-air requirement in the building. In winter the sorption system with a sorption wheel and a heat exchanger can be used as a highly efficient heat recovery system, and the thermal solar plant can be used for heating support.

Thermal solar energy or waste heat is used to heat regeneration air. In closed exhaust air discharge, the exhaust air is warmed up after the heat recovery device by the thermal collector, led through the sorption wheel after being heated, and expelled as waste air.

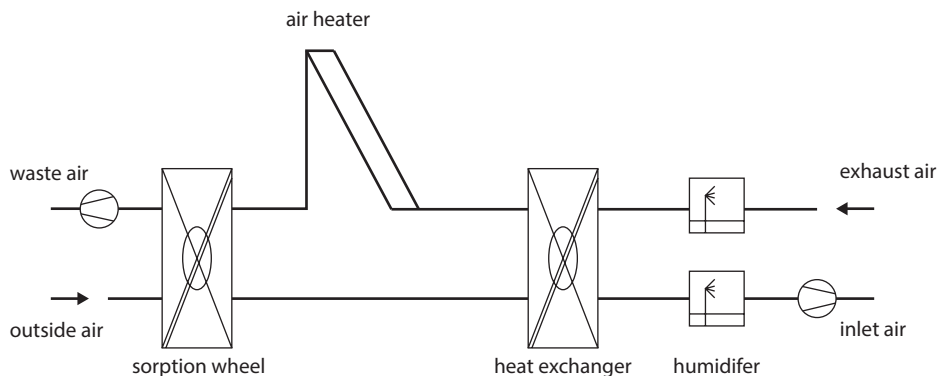


Figure 5.29 Open cycle desiccant cooling with closed exhaust air discharge.

During open exit air discharge the exhaust air is expelled after the heat recovery device as waste air. In the collector, outside air is sucked in, warmed up to regeneration temperature and expelled

after the sorption wheel. This version is usually selected for practical reasons, when either the space exhaust air is too strongly contaminated or the air circulation possibilities are spatially limited, since here an air duct is saved from the cooling machine to the collector. However, the outside temperature, which is the input temperature into the collector, is usually lower than the temperature after the heat recovery device so that more heating energy is required.

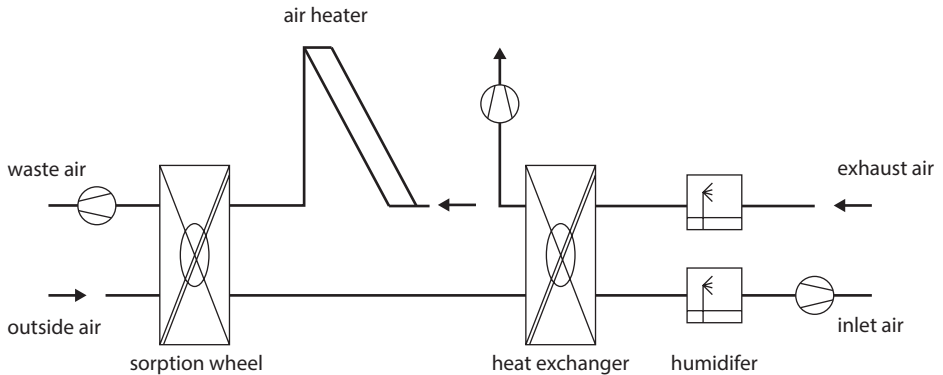


Figure 5.30 Open cycle desiccant cooling with open exhaust air discharge.

The process steps in the sorption wheel, heat recovery device, inlet air and exhaust air humidifier, and also the regeneration air heater can be understood through the enthalpy-humidity or Mollier diagram.

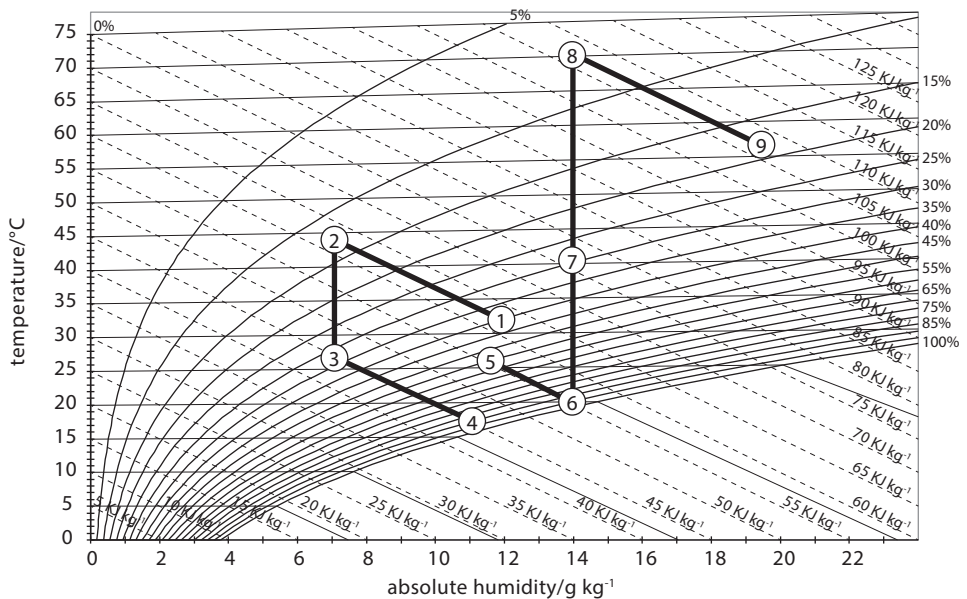


Figure 5.31 Process steps of desiccant cooling systems in the enthalpy-absolute humidity (h-x) diagram.

Outside air (1) is dried in the sorption wheel (2), pre-cooled in the heat recovery device with the additionally humidified cool space exhaust air (3) and afterwards brought to the desired supply air status by evaporative cooling (4). The space exhaust air (5) is maximally humidified by evaporative cooling (6) and warmed in the heat recovery device by the dry supply air (7). In the regeneration air heater the exhaust air is brought to the necessary regeneration temperature (8), takes up in the sorption wheel the water adsorbed on the supply air side, and is expelled as warm, humid exhaust air (9). Typical temperature and humidity conditions at design condition of 32°C outside temperature and 40% relative humidity are shown in the system scheme:

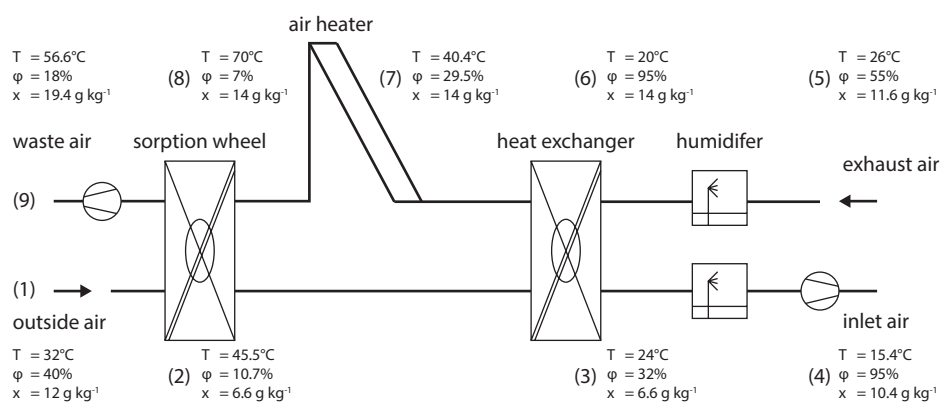


Figure 5.32 Temperature and humidity conditions in a desiccant cooling system.

The low supply temperatures in the above example are only achieved with very high heat recovery efficiencies of the heat exchanger (here 80%), high dehumidifying efficiency (here 80%) and high humidifying efficiency of the humidifier (here 95%). In real plants these are rarely reached. Measurements by the author from two demonstration projects in Germany (Althengstett) and Spain (Mataró) as well from Chinese partners showed that all the heat recovery efficiencies are often lower than given by manufacturers. Therefore, supply temperatures of 19.7°C result when under the same operating conditions.

	Mataró, Spain	Althengstett, Germany	Shanghai, China
Dehumidifying efficiency	80%	80%	88%
Humidifying efficiency	86%	85%	82%
Heat recovery efficiency	68%	62%	68%

Table 5.6 Measured component values of sorption systems in Spain, Germany and China.

Coupling with solar thermal collectors

Due to the low regeneration air temperatures for dehumidifying the sorption wheel, solar thermal collectors are particularly suitable for this air-conditioning technique. Both with air-led and with water-based flat-plate collectors, temperatures of 70°C can easily be achieved. Air collectors can be integrated simply and on large surfaces into the building shell, since leakages are unproblematic

and no frost protection is necessary. Solar air collectors are flowed through directly with outside air or with preheated exhaust air after the heat recovery device, and heated to the required regeneration temperature. Fluctuations in the regeneration air temperature due to changing solar irradiation are balanced by an auxiliary heating system or by the storage masses of the air-conditioned space.

With the use of water-based thermal collectors, the solar heat is transferred via an air-to-water heat exchanger to the exhaust air. Apart from direct heating of the exhaust air, buffer stores are used here to store the solar heat. At a required air outlet temperature of 70°C (i.e. around 40°C temperature difference from the surroundings in the summer), efficiencies of approximately 60% are achieved with air collectors at full irradiance (1000 W m⁻²); with selectively coated absorbers even 70%. Here the air collectors are flowed through at surface-related flow rates of 35 - 40 m³ m⁻² h⁻¹.

Water-based collectors, which today almost exclusively use selectively coated absorbers, need around 5 - 10K higher fluid outlet temperatures to achieve the same regeneration temperature of 70°C by means of the air-water heat exchanger. The efficiency of a flat-plate collector is comparable at these temperatures to that of an air collector.

From these efficiencies and typical EERs of a DCS of 0.9, the minimum collector area requirement can be estimated. At full irradiance around 600 W m⁻² of usable collector energy and thus 540 W m⁻² of cooling output are produced. A minimum of 1.85 m² of collector surface is necessary per kW of cooling output. With a kilowatt of cooling output, some 20 m² of office surface can be air-conditioned in a typical administrative building with 50 W m⁻² cooling load occurring.

To produce the necessary regeneration heat, combinations of back-ventilated photovoltaic facades and solar air collectors can also be used. Such energy facades supply additional electricity depending on the facade orientation and climate, for example, about 80 kWh m⁻² in the Spanish library project, apart from thermal energy.

Costs

In general, investment in the cooling machine of an air-conditioning system is only about 30% or less of the total cost. In cost comparisons between compression chillers and sorption-supported air-conditioning systems, therefore, the pure device costs may not be compared, since in the air-led sorption system dehumidifying, humidifying and heat recovery functions as well as supply and exhaust air fans are already integrated. The DCS device costs can be divided into thirds, for components, machine construction costs and control.

In a cost allocation for a system designed with 100 m² of solar air collectors and an air capacity of 18 000 m³ h⁻¹ with total costs of 185 000 Euro, it can be seen that even today the DCS device costs dominate the total price.

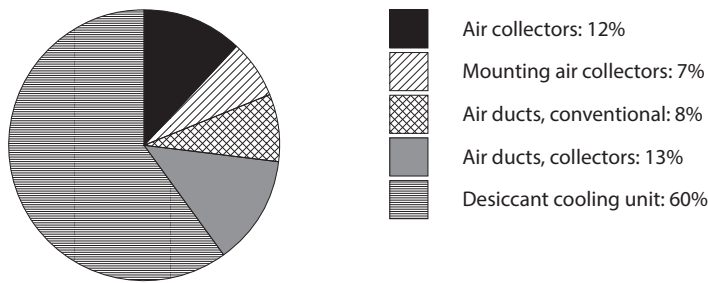


Figure 5.33 Cost allocation of a solar powered desiccant cooling air-conditioning system ($18\,000\text{ m}^3\text{ h}^{-1}$) for a production hall in Althengstett, Germany.

5.10 Physical and technological bases of sorption-supported air conditioning

Technology of sorption wheels

Continuously operating adsorption air dehumidifiers are slowly rotating, hygroscopic storage masses which are flowed through on one side by outside air and on the other by heated regeneration air. As solid sorption materials, silica gel and hygroscopic salts such as lithium chloride are commonly used. They are applied in continuously operating systems to a rotating substrate or used as fixed bed systems for intermittent operation. As a substrate for silica gel rotors, glass or ceramic fibres are used, for LiCl a cellulose matrix. Typical dehumidifying performances at regeneration temperatures of 70°C are around $4 - 6\text{ g kg}^{-1}$ of dry air.

To obtain optimal dehumidifying performance, the number of revolutions of the sorption wheel must be adapted to the regeneration air temperature and to the respective humidity conditions. Too high regeneration temperatures warm up the rotary sorption wheel after desorption so strongly that the sorption material at first can hardly take up moisture on the supply air side, but has to be cooled down first (so-called heat inhibition).

The separation between regeneration air and outside air is ensured by sealing strips, which leave a minimal gap between the seal and the sorption rotor. Maximum leakage rates between process and regeneration air should be under 3%. Such values have been measured even for small rotors of 1 m diameter. Pressure drops should not be above 150 Pa. By diverting a small section of dried outside air into the regeneration air flow, the typically 3 - 5% of exhaust air remaining in the storage mass is reduced by flushing to 0.5%, and in addition heat inhibition is reduced by cooling of the storage mass.

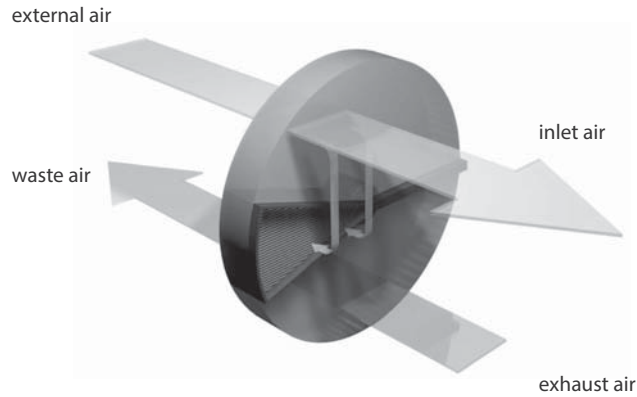


Figure 5.34 Schematic representation of a sorption rotor with flushing zone.

For winter operation, the sorption wheel is adjusted from dehumidifying operation with 10 - 15 revolutions per hour to enthalpy recuperation with revolutions of 10 min^{-1} .

Air-status calculations

When calculating the sorption process, the temperature and humidity conditions have to be determined for each process step. For this, the functional relations between absolute (x) and relative (φ) humidity, temperature T and total pressure p have to be determined: $x(\varphi, T, p)$ or $\varphi(x, T, p)$. For the energy balances, the enthalpies $h(x, T)$, i.e. the amounts of heat at constant pressure p , are balanced. The water vapour content of the air describes the mass ratio of the water vapour m_w to the dry air m_a .

$$x = \frac{m_w}{m_a} \left[\frac{\text{kg}}{\text{kg}} \right] \quad (5.104)$$

To calculate the adsorption process, a new temperature and relative humidity must be calculated for each value of the absolute humidity and enthalpy, the relative humidity φ being defined as the ratio of the water vapour partial pressure p_w [Pa] to the temperature-dependent saturation vapour pressure $p_s(T)$ [Pa].

$$\varphi = \frac{p_w}{p_s(T)} \quad (5.105)$$

If water vapour with pressure p_w and dry air with pressure p_a are regarded as ideal gases, following Dalton's law, the partial pressures add up to the total pressure p .

$$p = p_a + p_w \quad (5.106)$$

By the ideal gas equations, the water vapour content of the air can then be calculated as a function of the total pressure as well as of the water vapour partial pressure.

$$\begin{aligned} p_w V &= \frac{m_w}{M_w} RT \\ p_a V &= (p - p_w) V = \frac{m_a}{M_a} RT \end{aligned} \quad (5.107)$$

From this, results can be found for the absolute water content of the air with the mol mass of water vapour $M_w = 18 \times 10^{-3} \text{ kg mol}^{-1}$ and the mean mol mass of air $M_a = 28.97 \times 10^{-3} \text{ kg mol}^{-1}$:

$$x = \frac{m_w}{m_a} = \frac{M_w}{M_a} \frac{p_w}{p - p_w} = 0.622 \frac{p_w}{p - p_w} = 0.622 \frac{\phi p_s}{p - \phi p_s} \left[\frac{\text{kg}}{\text{kg}} \right] \quad (5.108)$$

Conversely, the relative humidity ϕ can be calculated as a function of the saturation water vapour pressure, the total pressure and the absolute water vapour content (in kg of water vapour kg^{-1} dry air):

$$\phi = \frac{x}{x + 0.622} \frac{p}{p_s(T)} \quad (5.109)$$

The density of the humid air ρ depends on the total pressure p [Pa], the water vapour pressure p_w [Pa] and the temperature T [K]:

$$\rho = \frac{m_a + m_w}{V} = 10^{-4} \left(34.8 \frac{p}{T} - 13.2 \frac{p_w}{T} \right) \left[\frac{\text{kg}}{\text{m}^3} \right] \quad (5.110)$$

The enthalpy of the air-vapour mixture h [kJ kg^{-1}] is composed of the sensible heat of the dry air $c_a T$ and the water vapour $x c_w T$, and of the evaporation enthalpy of the water $x h_e(T)$. The temperature-dependent evaporation enthalpy usually clearly dominates the enthalpy total.

$$h = h_a + x h_w = (c_a + x c_w) T + x h_e(T) \quad (5.111)$$

with

heat capacity of water vapour c_w	1.875 $\text{kJ kg}^{-1} \text{K}^{-1}$
heat capacity of dry air c_a	1.004 $\text{kJ kg}^{-1} \text{K}^{-1}$

Dehumidifying potential of sorption materials

The water vapour uptake and release by the sorption material are described by the corresponding sorption isothermal. At low temperatures, hygroscopic sorption materials take up high water-vapour quantities even at low water-vapour partial pressure of air flowing through. The highest dehumidifying rates at low outside humidity are achieved with zeolites, which require regeneration temperatures of 150 - 200°C and are thus not suitable for solar operation. More crucial for sorptive air conditioning is the water vapour uptake at medium to high humidities of the outside air, at which silica gel can take up 0.2 - 0.3 kg per kilogram of water vapour (maximum loading is around 0.5 kg kg^{-1}). The water vapour uptake of cellulose material soaked in lithium chloride depends strongly on the degree of soaking and can achieve higher values than silica gel.

At high temperatures, water vapour receptiveness decreases, and regeneration of the sorption material takes place.

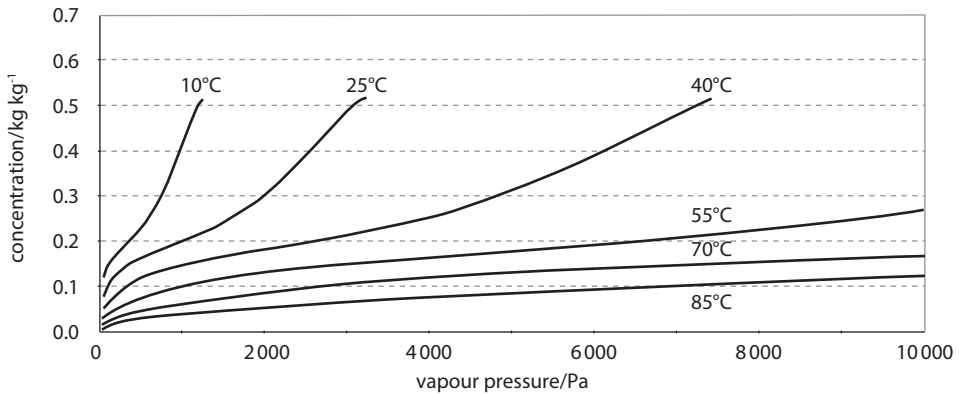


Figure 5.35 Measured concentration of water adsorbed on silica gel (sorption isothermal) as a function of water vapour pressure within air (Henning, 2001).

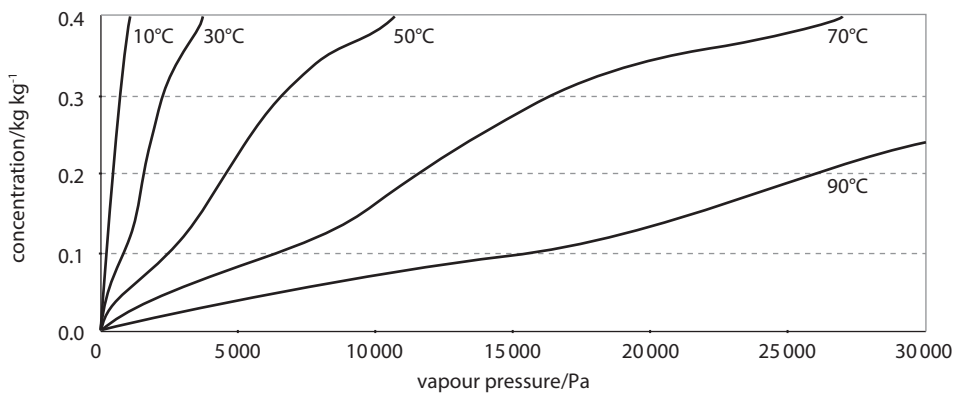


Figure 5.36 Calculated sorption isothermals of silica gel as a function of water vapour pressure.

If the load is represented as a function of relative humidity, i.e. a coordinate transformation of the x-axis is carried out, the sorption isotherms almost coincide. High water vapour partial pressure at high temperatures corresponds to a nevertheless low relative humidity and the load is small. Low partial pressure at a low temperature corresponds to a high relative humidity and the associated high load value shifts to the right.

Even at a very low relative humidity of approximately 5%, the minimum load is about 0.1 kg of water per kilogram of silica gel. This minimum load value is determined by the relative humidity of the regeneration air, and the outside air to be dried cannot fall below it.

Assuming that in a simple dehumidifying model the sorption isotherms coincide, represented as a function of the relative humidity, the outside air humidity at a given loading

value can be lowered to a minimum of the regeneration air humidity. An exact knowledge of the functional process of the sorption isotherms is then not necessary.

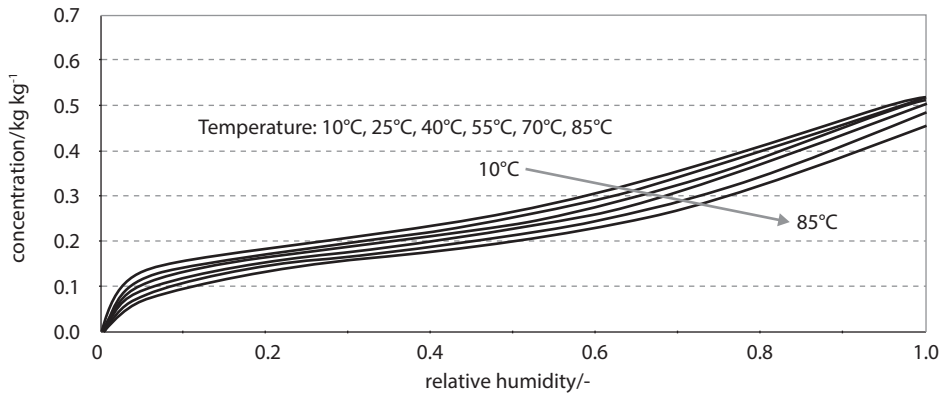


Figure 5.37 Measured sorption isotherms as a function of the relative humidity (Henning, 2001).

For the final state of the dried air in the simple model, the relative humidity, i.e. the regeneration humidity, is known, as well as the enthalpy (enthalpy of the outside air with isenthalpic dehumidification). Although both depend on temperature and absolute humidity, the equations cannot be solved analytically and the final state must be iterated. For this the absolute water vapour content of the outside air x_o is reduced until the final state corresponds to the given humidity condition of the regeneration air. As a check, the new temperature and relative humidity values are determined after each dehumidifying step, until supply air humidity and regeneration humidity correspond.

Since in the rotating sorption wheel no stationary status occurs, and the sorption isotherms do not coincide exactly, this minimum value of the supply air humidity with a vapour content x_{dry}^{ideal} is not achieved in practice. Non-ideal dehumidifying at an effective supply air humidity value x_{dry}^{eff} is covered by the dehumidifying efficiency η_{dh} .

$$\eta_{dh} = \frac{x_o - x_{dry}^{eff}}{x_o - x_{dry}^{ideal}}$$

$$x_{dry}^{eff} = x_o - \eta_{dh} (x_o - x_{dry}^{ideal}) \left[\frac{g}{kg} \right] \quad (5.112)$$

Apart from dehumidifying efficiency, deviations of the sorption wheel parameters from the nominal conditions can also be covered by correction factors. Detailed investigations of correction factors of LiCl sorption rotors have been carried out. The most important influence parameters for dehumidifying behaviour are rotor speed, air speed and the flow rate ratio between the regeneration air and outside air. For rotor speeds under the rated speed of $n = 22 \text{ h}^{-1}$, the dehumidifying performance falls constantly to approximately 80% at $n = 7 \text{ h}^{-1}$ (correction factor c_1). A reduction of the air flow velocity (at the same flow rate ratio) from a nominal rate of $v = 2.5 \text{ m s}^{-1}$ to 1.5 m s^{-1} leads to an increase in dehumidifying performance of up to 15% (correction c_2). If only the regeneration air

speed is lowered at processing-air nominal speed, the dehumidifying performance likewise rises by up to 20% (with flow rate reduction to 60%) (correction c_3).

Measurements on different sorption technologies (LiCl and silica gel) showed that the optimum rotation speed of the sorption wheel is dependent on the technology. A titanium silicate rotor has its optimal dehumidification at higher rotation speeds of 80 - 100 rotations per hour, whereas the LiCl rotor is best at low rotation rates of about 20 h⁻¹.

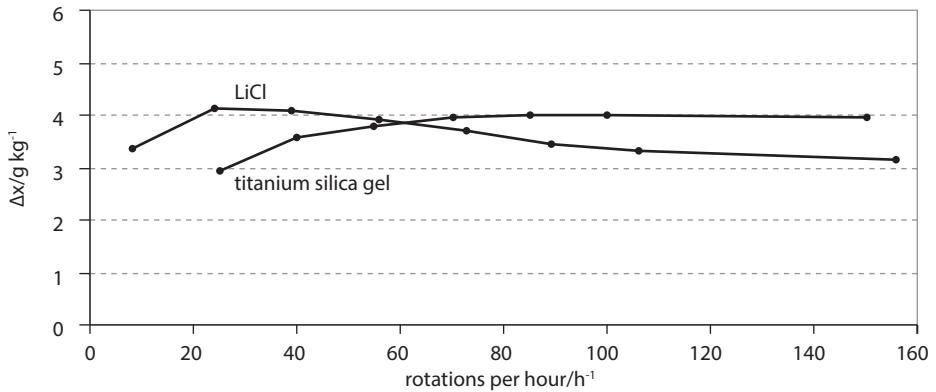


Figure 5.38 Comparison of dehumidification rates as a function of sorption wheel rotation speed.

Taking these corrections into account, the result is the effective supply air water content x_{dry}^{eff} :

$$x_{dry}^{eff} = x_o - \eta_{dh} c_1 c_2 c_3 (x_o - x_{dry}^{ideal}) \left[\frac{g}{kg} \right] \quad (5.113)$$

It is generally recommended, therefore, to use sorption rotors with as large cross section areas as possible, so that at a given flow rate the flow speeds remain small and dehumidifying performance becomes high.

Calculation of the sorption isotherms and isosteres of silica gel

For a more exact view of the sorption behaviour, first calculate the load concentration C of the sorption material as a function of the relative humidity for a constant temperature:

$$C|_{T=const} = f(\varphi)$$

In the Brunauer–Emmett–Teller theory, the water vapour receptiveness of silica gel is described by accumulation of water vapour in multi-molecular layers. For the 40°C isotherm, the following approximation for the load concentration C_0 is:

$$\frac{C_0}{C_m} = \frac{\varphi}{1-\varphi} + \frac{2(b-1)\varphi + 2(b-1)^2\varphi^2 + (Nb^2 + Nh - N^2b^2)\varphi^N \dots}{2(1+2(b-1)\varphi + (b-1)^2\varphi^2 + (b^2 + h - 2b - Nb^2)\varphi^N \dots} \dots$$

$$\frac{\dots + (2b + N^2b^2 + 2Nb - 2b^2 - Nb^2 - 2h - 2Nh)\varphi^{N+1} + (Nh + 2h)\varphi^{N+2}}{\dots + (Nb^2 + 2b - 2b^2 - 2h)\varphi^{N+1} + h\varphi^{N+2}} \quad (5.114)$$

with the coefficients $C_m = 0.11$, $b = 11$, $N = 7.2$, $h = 19\,000$ for the combination silica gel–water. The equation is valid up to a maximum load of $C_{max} = 0.4 \text{ kg kg}^{-1}$.

The conversion of the sorption isotherms to other temperatures is made by analysing the vapour pressure over the sorbents at a given load concentration C . If the adsorption process is regarded as a phase transition, the vapour pressure above the sorption material and from this the relative humidity can be calculated from the Clausius–Clapeyron equation, which normally describes the equilibrium between vapour pressure and fluid in a one-phase system as a function of the temperature.

The adsorption heat h_{ads} as a total of binding enthalpy (h_b) and evaporation enthalpy (h_e) corresponds here to the enthalpy difference between vapour and fluid in the one-phase system. The binding enthalpy decreases with rising load concentration C ; the evaporation heat of the water falls with rising temperature.

$$h_{ads}(C, T) = h_b(C) + h_e(T) \quad (5.115)$$

The adsorption heat and the binding heat as a difference of evaporation heat and adsorption heat can be approximated by quadratic functions of the load concentration. Up to a load concentration of $C = 0.1934 \text{ kg kg}^{-1}$, these equations are valid:

$$h_{ads} = h_b + h_e = h_o(1 + aC + bC^2)$$

$$h_b = h_{ads} - h_e = h_o(1 + aC + bC^2) - h_e \quad (5.116)$$

with the constants $a = -2.34$, $b = 6.05$ and $h_o = 3172 \text{ kJ kg}^{-1}$ for silica gel. The evaporation enthalpy h_e of water is selected for the calculation of the binding enthalpy as a constant with 2453 kJ kg^{-1} (20°C). Above a load concentration of $C = 0.1934 \text{ kg kg}^{-1}$, the binding enthalpy is set at $h_b = 0$ and the adsorption enthalpy is equal to the evaporation enthalpy h_e . At $C = 0$, i.e. perfectly dry material, the binding enthalpy is 719 kJ kg^{-1} .

Other authors (Henning, 1994) have produced approximation formulae with clearly higher binding enthalpies, which at zero loading lie between 1000 and 2000 kJ/kg. Accordingly, the binding enthalpy can be calculated with the simple linear equation

$$h_b = 1000 \frac{\text{kJ}}{\text{kg}} - C \times 4000 \frac{\text{kJ}}{\text{kg}}$$

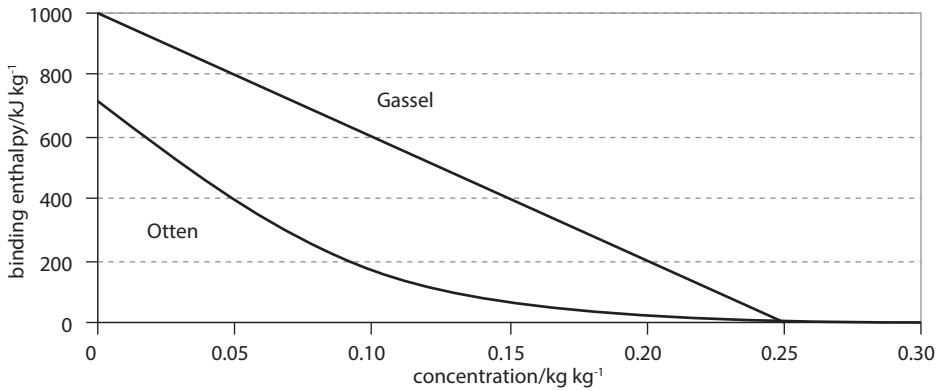


Figure 5.39 Binding enthalpy of water vapour to silica gel as a linear or quadratic function of the load concentration C .

Only at high load concentrations and binding enthalpy of zero is the change in status of air isenthalpic. At a low concentration, heating of the outside air is greater, due to the dehumidifying process through the uptake of the binding heat, than in a purely isenthalpic change in status.

For water vapour partial pressure p_w over the sorbents, the following equation is valid, based on Clausius–Clapeyron:

$$\frac{dp_w}{dT} = \frac{h_{ads}(C, T)}{R_w T^2} p_w \Leftrightarrow \frac{d \ln p_w}{d(1/T)} = \frac{-h_{ads}(C, T)}{R_w} \quad (5.117)$$

with $R_w = 461 \text{ J kg}^{-1} \text{ K}^{-1}$ as the specific gas constant of water vapour.

If the vapour pressure p_w is plotted logarithmically against the inverse temperature $1/T$, then the heat of adsorption for a given load concentration results from the gradient. The load-dependent vapour pressure curves are called isosteres ($C = \text{const}$). The smaller the load concentration of the silica gel, the lower is the vapour pressure over the sorbents at a given temperature. For the completely saturated sorbents, the binding enthalpy is zero, and the well-known Clausius equation for the saturation vapour pressure of water is obtained as a function of the temperature.

$$\frac{dp_s}{dT} = \frac{h_e}{R_w T^2} p_s \quad (5.118)$$

The integration of the Clausius equation over a temperature range T_1 to T_2 enables the calculation of the isosteres, if the pressure p_1 at temperature T_1 is known. To simplify the integral, the adsorption enthalpy is assumed to be temperature-independent.

$$\begin{aligned} \int_{p_{w1}}^{p_{w2}} \frac{dp_w}{p_w} &= \int_{T_1}^{T_2} \frac{h_{ads}(C)}{R_w T^2} dT \\ \ln\left(\frac{p_{w2}}{p_{w1}}\right) &= \frac{h_{ads}(C)}{R_w} \left(\frac{1}{T_1} - \frac{1}{T_2}\right) \\ p_{w2} &= p_{w1} \exp\left(\frac{h_{ads}(C)}{R_w} \left(\frac{1}{T_1} - \frac{1}{T_2}\right)\right) \end{aligned} \quad (5.119)$$

From the well-known 40°C isotherms, the vapour pressure p_{w1} can be determined as the still unknown parameter of the isosteres equation for each load concentration. For a given load concentration C_0 , the relative humidity φ_0 of the 40°C isotherms (i.e. $T_1 = 313 \text{ K}$) is determined iteratively from Equation (5.114). From the relative humidity, the vapour pressure p_{w1} can then be calculated to $p_{w1} = \varphi_0 \times p_s(40^\circ\text{C})$. For the load concentration $C = C_0$, the constants of the vapour pressure equation p_{w1} and T_1 are then known.

Example 5.13

Calculation of the isostere parameters of water vapour over silica gel at a load concentration of $C = 0.1 \text{ kg kg}^{-1}$.

For $C = 0.1 \text{ kg kg}^{-1}$ the adsorption heat is $h_{ads} = 2622 \text{ kJ kg}^{-1}$. The relative humidity resulting in $C_0 = 0.1 \text{ kg kg}^{-1}$ at 40°C is 18.75%.

From this the vapour pressure $p_{w1} = \varphi_0 \times p_s(40^\circ\text{C}) = 0.1875 \times 7384 \text{ Pa} = 1384.5 \text{ Pa}$ results, so the complete isostere can be calculated with the following equation:

$$p_w = 1384.5 \exp\left(\frac{2622}{0.461} \left(\frac{1}{313} - \frac{1}{T}\right)\right)$$

The two Clausius equations for pure water and water-silica gel are divided by each other at the same temperature T , so that from the relation of the vapour pressure p_w to the saturation vapour pressure p_s , the relative humidity and finally the functional connection of relative humidity and load concentration can be determined.

$$\frac{dp_s}{p_s} = \frac{h_e}{h_{ads}} \frac{dp_w}{p_w} \quad (5.120)$$

Equation (5.120) is integrated

$$\begin{aligned} \int_{p_{s,0}}^{p_s} \frac{dp_s}{p_s} &= \int_{p_{w,0}}^{p_w} \frac{h_e}{h_{ads}} \frac{dp_w}{p_w} \\ \frac{p_s}{p_{s,0}} &= \left(\frac{p_w}{p_{w,0}}\right)^{\frac{h_e}{h_{ads}}} = \left(\frac{\varphi p_s}{p_{w,0}}\right)^{\frac{h_e}{h_{ads}}} \end{aligned} \quad (5.121)$$

and solved for the relative humidity φ :

$$\varphi = \frac{p_{w,0} \frac{h_b + h_e}{h_e} p_s}{p_{s,0} \frac{h_b}{h_e} p_s} = \text{const } p_s^{\frac{h_b(C)}{h_e(T)}} \quad (5.122)$$

This equation provides the sought connection of the sorption isotherms between relative humidity, temperature (via the saturation vapour pressure p_s) and load concentration C (via the binding enthalpy h_b).

With the same loading, i.e. the same binding enthalpy h_b , conversion is thus possible from a single known sorption isotherm (with temperature T_0) to other temperatures. For simplification the evaporation heat is set temperature-independently.

$$\frac{\varphi_0}{\varphi} = \frac{p_s(T_0)^{\frac{h_b(C_0)}{h_e(T_0)}}}{p_s(T)^{\frac{h_b(C_0)}{h_e(T)}}} \approx \left(\frac{p_s(T_0)}{p_s(T)} \right)^{\frac{h_b(C_0)}{h_e}} \quad (5.123)$$

For each relative humidity φ_0 of the known 40°C sorption isotherm $C_0(\varphi_0)$ from Equation (5.114), the associated relative humidity φ at temperature T can be determined. This corresponds to the load concentration $C = C_0$, and the value pair (C, φ) is obtained.

$$\varphi = \varphi_0 \times \left(\frac{p_s(T)}{p_s(T_0)} \right)^{\frac{h_b(C_0(\varphi_0))}{h_e}} \quad (5.124)$$

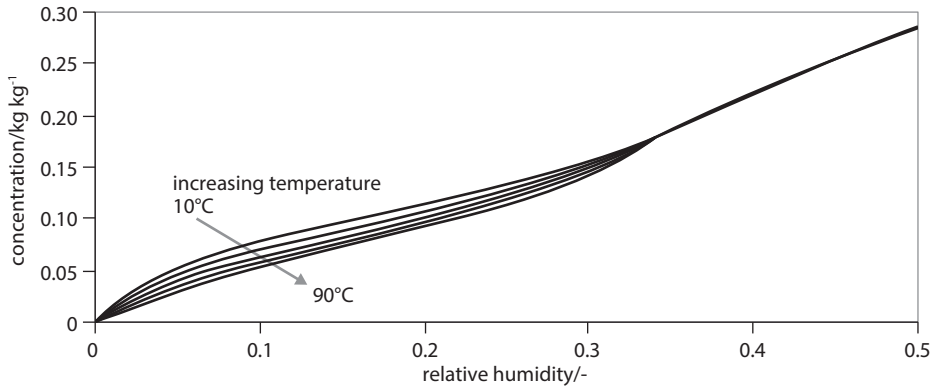


Figure 5.40 Sorption isotherms as a function of the relative humidity for temperatures between 10°C and 90°C, calculated from the known 40°C isotherm.

As soon as the binding enthalpy becomes zero, i.e. for load concentrations greater than $0.1934 \text{ kg kg}^{-1}$, all isotherms coincide exactly.

In practice, the problem is mostly that for a given humidity φ , the associated load concentration C must be determined. Thus, for example, the regeneration humidity φ_{reg} gives the minimum load concentration to which the outside air can finally be dehumidified. To determine

this concentration, Equation (5.124) must now be solved for the relative humidity φ_o , for which the load concentration C_o is known from the 40°C isotherm.

$$\varphi_o = \varphi \times \left(\frac{p_s(T_o)}{p_s(T)} \right)^{\frac{h_b(C_o(\varphi_o))}{h_e}} \quad (5.125)$$

Since the load concentration C_o depends on φ_o , an implicit equation is obtained, which must be solved iteratively. After the end of the iteration the load concentration $C = C_o$ is known, which results from the given relative humidity φ and temperature T .

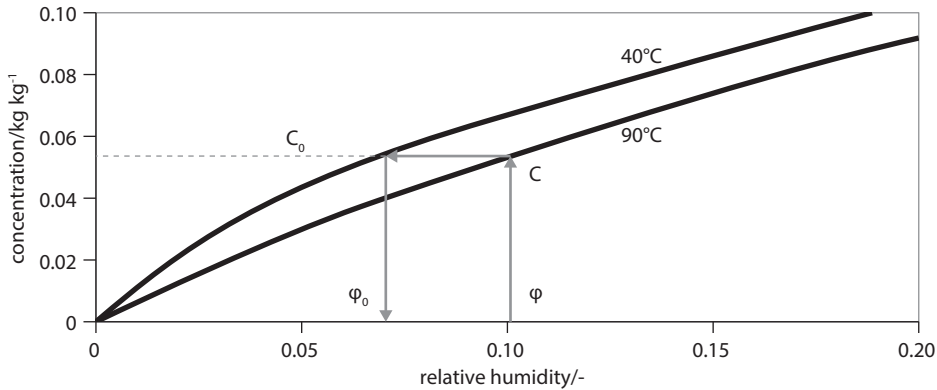


Figure 5.41 Determination of the load concentration C for a given humidity φ over the known sorption isotherm at 40°C, by determining the relative humidity φ_o at the same load $C_o = C$.

Thus with the extended model the relative humidity φ_{reg} of the regeneration air is given for the determination of the dehumidifying performance of a sorption rotor. Via the iterative method from Equation (5.125) the load concentration C_{reg} of the regenerated sorption material is determined. If outside air is now gradually dehumidified, the associated load concentration is determined for each new temperature and relative humidity φ_{dry} of the dried outside air after the dehumidifying step, until C_{dry} is equal to C_{reg} .

Calculation of the dehumidifying performance of a sorption rotor

The climatic boundary conditions chosen for an example system calculation correspond to German standards and are 32°C outside air temperature, 40% relative humidity or 12 g kg⁻¹ of water vapour, at a total pressure of 1.013×10^5 Pa. In the following the air statuses of the dried outside air are to be calculated, when regeneration air is at 70°C and contains 14 g kg⁻¹ of water vapour. First the relative humidity φ of the regeneration air is calculated at 70°C using Equation (5.109) with the saturation vapour pressure p_s from Equation (5.16), which determines the minimum relative humidity of the supply air to be dehumidified.

$$\varphi = \frac{14 \times 10^{-3} \frac{\text{kg}}{\text{kg}}}{14 \times 10^{-3} \frac{\text{kg}}{\text{kg}} + 0.622 \frac{\text{kg}}{\text{kg}}} \times \frac{101300 \text{ Pa}}{31158 \text{ Pa}} = 0.07$$

The enthalpy of the regeneration air based on Equation (5.111) with the evaporation enthalpy based on Equation (5.13) is given by:

$$h_{reg} = \left(1.004 \frac{\text{kJ}}{\text{kgK}} + 14 \times 10^{-3} \frac{\text{kg}}{\text{kg}} \times 1.875 \frac{\text{kJ}}{\text{kgK}} \right) 70^\circ\text{C} + 14 \times 10^{-3} \frac{\text{kg}}{\text{kg}} \times 2334 \frac{\text{kJ}}{\text{kg}} = 104.8 \frac{\text{kJ}}{\text{kg}}$$

The value is not required for the calculation of the supply air status.

The enthalpy of the outside air determines the temperature of the air after dehumidifying, when the process is isenthalpic.

$$h_o = \left(1.004 \frac{\text{kJ}}{\text{kgK}} + 12 \times 10^{-3} \frac{\text{kg}}{\text{kg}} \times 1.875 \frac{\text{kJ}}{\text{kgK}} \right) 32^\circ\text{C} + 12 \times 10^{-3} \frac{\text{kg}}{\text{kg}} \times 2425 \frac{\text{kJ}}{\text{kg}} = 62 \frac{\text{kJ}}{\text{kg}}$$

The iterative calculation of the outside air drying process is solved with fast convergence with the Regula falsi algorithm. In this, the initial water content $x_{o,0}$ of the outside air (relative humidity $\varphi_{o,0}$) in the first iteration step is reduced by say 50%, and the temperature T_1 and new relative humidity φ_1 are calculated via the given constant enthalpy at the new water content x_1 . At the above outside air status of 32°C , 12 g kg^{-1} of vapour content, dehumidification is 6 g kg^{-1} in the first step. The temperature of the dried outside air is calculated using Equation (5.111) at 46.7°C . Via the saturation vapour pressure of $p_s = 10445 \text{ Pa}$, based on Equation (5.16), the relative humidity φ_1 at 6 g kg^{-1} vapour content based on Equation (5.109) is obtained as 9.3%. The evaporation enthalpy is kept constant during iteration. The relative humidity φ_1 is above the regeneration humidity of 7%, so further dehumidification is necessary.

To determine the next water content x_2 , the ratio of the humidity difference $\varphi_{o,0} - \varphi_1$ to the water-content difference $x_{o,0} - x_1$ of the first dehumidifying step (i.e. the gradient of the humidity function) is equated to the ratio of the humidity difference between the outside air and regeneration air $\varphi_{o,0} - \varphi_{reg}$ to the water-content difference $x_{o,0} - x_2$. The new water content x_2 is thus equated approximately to the absolute humidity of the final state.

$$\frac{\varphi_{o,0} - \varphi_1}{x_{o,0} - x_1} = \frac{\varphi_{o,0} - \varphi_{reg}}{x_{o,0} - x_2} \quad (5.126)$$

This results in the new water content of the next iteration x_2 .

$$x_2 = x_{o,0} - (\varphi_{o,0} - \varphi_{reg}) \frac{x_{o,0} - x_1}{\varphi_{o,0} - \varphi_1}$$

$$x_2 = 12 \times 10^{-3} \frac{\text{kg}}{\text{kg}} - (0.4 - 0.07) \frac{12 \times 10^{-3} \frac{\text{kg}}{\text{kg}} - 6 \times 10^{-3} \frac{\text{kg}}{\text{kg}}}{0.4 - 0.093} = 5.55 \times 10^{-3} \frac{\text{kg}}{\text{kg}} \quad (5.127)$$

The following is generally valid:

$$x_n = x_{n-1} - (\varphi_{n-1} - \varphi_{reg}) \frac{x_{n-2} - x_{n-1}}{\varphi_{n-2} - \varphi_{n-1}} \quad (5.128)$$

For the new water content $x_2 = 5.55 \text{ g kg}^{-1}$, the new relative humidity $\phi_2 = 0.081$ is obtained via the new temperature $T_2 = 47.8^\circ\text{C}$; it is still over the limiting regeneration humidity ϕ_{reg} . As the next water content, x_3 is obtained:

$$x_3 = 5.55 \times 10^{-3} \frac{\text{kg}}{\text{kg}} - (0.081 - 0.07) \frac{6 \times 10^{-3} \frac{\text{kg}}{\text{kg}} - 5.55 \times 10^{-3} \frac{\text{kg}}{\text{kg}}}{0.093 - 0.081} = 5.14 \times 10^{-3} \frac{\text{kg}}{\text{kg}}$$

The water content x_3 of 5.14 g kg^{-1} corresponds to a relative humidity of 7.1% at an outlet temperature of 48.8°C , i.e. in the second iteration the humidity of the regeneration air state is almost achieved.

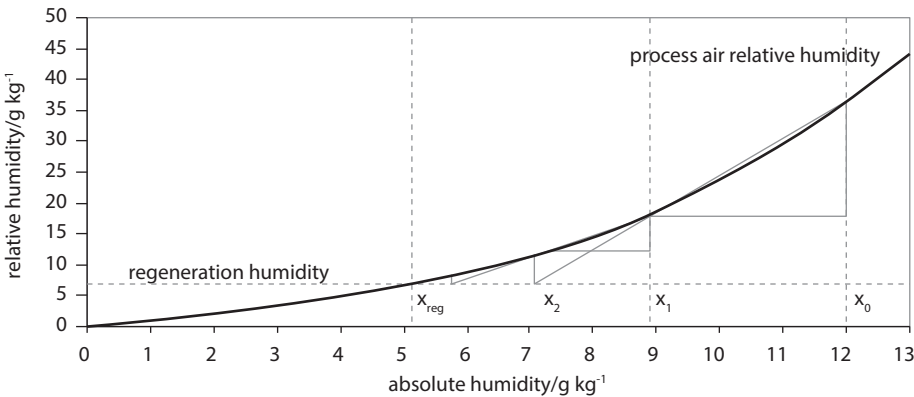


Figure 5.42 Relative humidity as a function of absolute humidity with constant enthalpy of the outside air of 62 kJ kg^{-1} for the calculation of dehumidifying.

The value of the regeneration relative humidity is fixed at 7% and gives the final state of the outside air to be dried. For better illustration of the Regula Falsi algorithm, in the first dehumidifying step only 3 g kg^{-1} has been dehumidified. Nevertheless, in the second iteration the final humidity is almost achieved. The final value of the iteration is $x_\infty = 5.1 \text{ g kg}^{-1}$, i.e. of the initial value $x_0 = 12 \text{ g kg}^{-1}$, the outside air has been dehumidified by 6.9 g kg^{-1} . The non-ideal dehumidifying is now considered by means of the dehumidification efficiency, which is set at 80%. Using Equation (5.112) the residual moisture is then 6.5 g kg^{-1} and the associated temperature of the dried outside air is 45.5°C . The associated relative humidity is 10.6%.

The status of the regeneration air after water vapour take-up in the sorption wheel can be calculated similarly. The total humidity taken up by the silica gel, ideally 6.9 g kg^{-1} (non-ideally 5.5 g kg^{-1}), is now given off with isenthalpic humidification of the regeneration air. The absolute humidity rises from 14 g kg^{-1} to 20.9 g kg^{-1} (or 19.5 g kg^{-1}).

Temperature of the damp exhaust air:	53.7°C	(56.9°C)
Relative humidity:	22%	(17.9%)

5.11 The technology of heat recovery

In sorption-supported air-conditioning systems, predominantly regenerative rotary heat recovery devices are used. These achieve, depending on the incident-flow velocity, good heat recovery

efficiencies between 70% and 90%. As a heat accumulator, corrugated aluminium plate is often used, wound into heat exchanger wheels of up to 5 m diameter. Volume flow rates are between 1 000 and 150 000 m³ h⁻¹, with pressure losses of 50 - 200 Pa. In partial-load operation the heat recovery efficiency rises due to the reduced incident-flow velocities, i.e. slightly oversized heat exchangers lead to better system efficiencies. The rotation rate of the heat recovery wheels is between 5 and 15 revolutions per minute, though an increase in the storage mass stream leads to an improvement of the heat recovery efficiency.

The hygroscopic sorption wheels are likewise used in winter for heat recovery and achieve, with a corresponding increase in the figure of revolutions, similar heat recovery efficiencies to the non-hygroscopically coated storage masses. At typical air incident-flow velocities of 3 m s⁻¹ and a supply-air to exhaust-air ratio of 1, heat recovery efficiencies between 75% and 85% are achieved, depending on the depth of the sorption rotor. Since the model for the rotary regenerative heat-transfer agent is based on a cross-flow heat exchanger, the calculation is executed first for recuperators (partitioned wall heat exchangers).

Recuperators

In a recuperator, heat is transferred from the warmer fluid (with mass flow $\dot{m}_1$ and thermal capacity c_1) convectively, with a heat transmission coefficient of h_{c1} , to the partition and after thermal conduction by the partition material convectively with h_{c2} to the colder fluid (with mass flow $\dot{m}_2$ and thermal capacity c_2).

For a high transmission rate of a heat exchanger, as high a heat transition coefficient U as possible is necessary. The heat transition coefficient is dominated by the convective transition resistances. The thermal resistance $R = s/\lambda$ of the plate material (with plate thickness s [m] and heat conductivity of the plate λ [W m⁻¹ K⁻¹]) is normally negligible. The heat transfer coefficient of a recuperator with an even partition surface is given by

$$U = \left(\frac{1}{h_{c1}} + R + \frac{1}{h_{c2}} \right)^{-1} \quad (5.129)$$

The proportion of the radiation to the heat transfer is negligible due to the virtually identical temperatures of the individual partition surfaces.

The heat transfer coefficient by convection h_c is determined in practice by model tests. These test results can then be transferred to other geometrically and hydrodynamically similar heat transfer conditions.

$$h_c = \frac{Nu(Re, Pr)\lambda}{L} \quad (5.130)$$

Nu: Nußelt number/-

λ : heat conductivity of the fluid/W m⁻¹ K⁻¹

The characteristic length L depends on the respective geometry of the heat exchanger. The most important Nußelt correlations as a function of the Reynolds and Prandtl numbers (Re, Pr), of the overall length of the heat-transferring gap/tube/channel l , and of the characteristic length L , are summarised in Table 5.7. The calculation of the material properties of air as well as of the Reynolds and Prandtl numbers can be found in Section 4.9, Calculation of the available thermal power of solar air collectors on page 281.

Flow condition	Nußelt correlation	Characteristic length	Reference
Geometry: gap/channel			
Laminar ($Re < 2300$)	$Nu = \left[7.541^3 + 1.841 \sqrt{Re Pr \frac{L}{l}} + \left(\frac{2}{1 + 22 \cdot Pr} \right)^{1/6} \left(Re Pr \frac{L}{l} \right)^{0.5} \right]^{1/3}$	$L = 2h$	Al Amouri (1994)
Turbulent ($Re > 8000$)	$Nu = 0.116 \cdot (Re^{2/3} - 125) \cdot Pr^{1/3} \left[1 + \left(\frac{L}{l} \right)^{2/3} \right]$	$L = \frac{4 \cdot A_{free}}{C}$	Gregorig (1959)
Geometry: tube			
Laminar ($Re < 2320$)	$Nu = \left(49.0 + 4.17 \cdot Re \cdot Pr \cdot \frac{L}{l} \right)^{1/3}$	$L = d_i$	Hering et al. (1997)
Turbulent ($Re > 2320$)	$Nu = 0.116 \cdot (Re^{2/3} - 125) \cdot Pr^{1/3} \left[1 + \left(\frac{L}{l} \right)^{2/3} \right]$	$L = d_i$	Gregorig (1959)
h	distance between heat-transferring surfaces (gap/channel)		
A_{free}	free area for flow		
C	circumference		
d_i	inside diameter of the pipe		

Table 5.7 Relevant Nußelt correlations for heat exchanger calculations.

The amount of heat $\dot{Q}$ given off by the warmer fluid is taken up completely by the colder fluid flow, ignoring heat losses to the external environment, and is calculated from the heat transfer rate of the heat exchanger. This results from the product of the two-dimensional elements dA , the heat transfer coefficient U and the locally varying temperature difference $T_1 - T_2$ between the two fluid flows.

$$\dot{Q} = \int U \times (T_1(x, y) - T_2(x, y)) dA = \dot{m}_1 c_1 (T_{1,in} - T_{1,out}) = \dot{m}_2 c_2 (T_{2,out} - T_{2,in}) \quad (5.131)$$

Normally only the inlet temperatures of the warm and cold fluids into the heat exchanger are known, for example, in the sorption system the temperature of the warm dried supply air ($T_{1,in}$) and the temperature of the colder space exhaust air ($T_{2,in}$). To calculate the transferred heat as a function of the inlet temperatures, the heat recovery efficiency η_{hx} is introduced, which is also known as an operational characteristic and is defined as the ratio of actually transferred power to maximum transferred power.

$$\begin{aligned} \dot{Q} &= \dot{m}_1 c_1 (T_{1,in} - T_{2,in}) \Phi = \dot{m}_1 c_1 (T_{1,in} - T_{1,out}) = \dot{m}_2 c_2 (T_{2,out} - T_{2,in}) \\ \Phi &= \frac{\dot{m}_1 c_1 (T_{1,in} - T_{1,out})}{\dot{m}_1 c_1 (T_{1,in} - T_{2,in})} = \frac{\dot{m}_2 c_2 (T_{2,out} - T_{2,in})}{\dot{m}_1 c_1 (T_{1,in} - T_{2,in})} \end{aligned} \quad (5.132)$$

Only at identical thermal capacity streams of the two sides $\dot{m}_1 c_1 = \dot{m}_2 c_2$ are the two temperature-difference ratios (defined by the heat recovery efficiency) the same.

$$\Phi = \frac{(T_{1,in} - T_{1,out})}{(T_{1,in} - T_{2,in})} = \frac{(T_{2,out} - T_{2,in})}{(T_{1,in} - T_{2,in})} \quad (5.133)$$

The heat recovery efficiencies of the most important recuperators (same-, counter- and cross-flow heat exchangers) depend functionally on the ratio of heat transfer performance UA and thermal capacity stream $\dot{C} = \dot{m}c$, which are called NTU (number of transfer units).

$$NTU = \frac{UA}{\dot{C}} \quad (5.134)$$

The heat recovery efficiency for a counter-current heat exchanger with the thermal capacity stream $\dot{C}_1 < \dot{C}_2$ is:

$$\Phi = \frac{1 - e^{-\left(1 - \frac{\dot{C}_1}{\dot{C}_2}\right) \frac{UA}{\dot{C}_1}}}{1 - \frac{\dot{C}_1}{\dot{C}_2} e^{-\left(1 - \frac{\dot{C}_1}{\dot{C}_2}\right) \frac{UA}{\dot{C}_1}}} \quad (5.135)$$

At identical mass flows on the warm and cold sides, the heat recovery efficiency is at most 0.8, with a high ratio of transmission rate UA to thermal capacity stream (> 4). The heat recovery efficiency for $\dot{C}_1 = \dot{C}_2$ is

$$\Phi = \frac{\frac{UA}{\dot{C}_1}}{1 + \frac{UA}{\dot{C}_1}} \quad (5.136)$$

The operational characteristic improves with unequal thermal capacity streams.

Direct current heat exchanger:

$$\Phi = \frac{1 - e^{-\left(1 + \frac{\dot{C}_1}{\dot{C}_2}\right) \frac{UA}{\dot{C}_1}}}{1 + \frac{\dot{C}_1}{\dot{C}_2}} \quad (5.137)$$

In cross-flow heat exchangers the directions of flow of the two fluids run perpendicular to each other. The heat recovery efficiency is obtained by an infinite series which depends on $UA/\dot{C}$.

Pure cross-current plate heat exchanger ($\dot{C}_1 < \dot{C}_2$):

$$\Phi = \frac{1}{\frac{UA}{\dot{C}_2}} \sum_{n=0}^{\infty} \left(1 - \exp\left(-\frac{UA}{\dot{C}_1}\right) \sum_{p=0}^n \frac{\left(\frac{\dot{C}_1}{\dot{C}_2}\right)^p}{p!} \right) \times \left(1 - \exp\left(-\frac{UA}{\dot{C}_2}\right) \sum_{p=0}^n \frac{\left(\frac{\dot{C}_2}{\dot{C}_1}\right)^p}{p!} \right) \quad (5.138)$$

In the infinite series it is sufficient to calculate the terms $n = 0$ to $n = 5$.

A pure cross-flow heat exchanger is defined by the fact that no lateral mixing of the individual fluid lines is possible, and occurs in practice with heat exchangers whose heat-transferring surface consists of flat or corrugated plates (plate-type heat exchangers). Typical gap widths for a plate-type heat exchanger are between 5 - 10 mm.

If in a tube heat exchanger the fluid in the pipes is flowed around perpendicularly by another fluid over the whole cross section, a mixing of the fluid lines of the outside fluid can occur transverse to the direction of flow, and a so-called one-side agitated cross-flow heat exchanger is the result. The larger the number of the tubing rows, the stronger is the approximation to the pure cross current.

One-side agitated cross current: shell and tube heat exchanger:

Current $\dot{C}_1$ remains unmixed, current $\dot{C}_2$ is agitated (with $\dot{C}_1 < \dot{C}_2$)

$$\Phi = \frac{1 - \exp\left(-\frac{\dot{C}_1}{\dot{C}_2} \left(1 - e^{-\frac{UA}{\dot{C}_1}}\right)\right)}{\frac{\dot{C}_1}{\dot{C}_2}} \quad (5.139)$$

Current $\dot{C}_2$ remains unmixed, current $\dot{C}_1$ is agitated

$$\Phi = 1 - \exp\left(-\frac{\dot{C}_2}{\dot{C}_1} \left(1 - e^{-\frac{UA}{\dot{C}_2}}\right)\right) \quad (5.140)$$

Example 5.14

Calculation of the heat recovery efficiency of a counter-current plate-type heat exchanger for a sorption system with 20 000 m³ h⁻¹ of both supply air and exhaust air flow rate.

Geometry			
<i>H</i>	1.5	m	Height of recuperator
<i>B</i>	1.5	m	Width of recuperator
<i>l</i>	1.5	m	Length of channel
<i>n</i>	250	-	Number of plates
<i>s_{pl}</i>	0.0002	m	Thickness of individual plates
<i>λ_{pl}</i>	229	W m ⁻¹ K ⁻¹	Heat conductivity of plate material
<i>d_g</i>	0.0058	m	Gap width (distance between individual plates)
<i>A_c</i>	1.09	m ²	Free cross section of recuperator (one direction)
<i>A_{pl}</i>	2.25	m ²	Area of individual plates (length × width)
<i>A_{free}</i>	0.009	m ²	Free flow cross section (one channel)
<i>d_h</i>	0.012	m	Hydraulic diameter
<i>L</i>	0.012	m	Characteristic length
<i>A_{hx}</i>	562.50	m ²	Heat-transferring surface area

	Warm air	Cold air		
V/t	5.56	5.56	m^3/s	Volume flow
T	45.50	20.00	$^{\circ}\text{C}$	Temperature
λ_{air}	0.0258	0.0251	$\text{W m}^{-1} \text{K}^{-1}$	Heat conductivity air
ρ_{air}	1.0933	1.1884	kg m^{-3}	Density air
$c_{p,\text{air}}$	1008.3	1007.0	$\text{J kg}^{-1} \text{K}^{-1}$	Heat capacity air
u_{air}	1.76×10^{-5}	1.52×10^{-5}	$\text{m}^2 \text{s}^{-1}$	Kinematic viscosity air
v_g	5.11	5.11	m/s	Mean gap velocity
Pr	0.751	0.723	-	Prandtl number
Re	3345	3878	-	Reynold number
Nu	10.805	13.176	-	Nußelt number
h_c	24.25	28.75	$\text{W m}^{-1} \text{K}^{-1}$	Convective heat transfer coefficient
U	13.16		$\text{W m}^{-1} \text{K}^{-1}$	Heat transfer coefficient
C_1	6128.87		W K^{-1}	Smaller heat capacity flow
C_2	6653.58		W K^{-1}	Larger heat capacity flow
Φ	0.56		-	Heat recovery efficiency, calculated
$T_{1,\text{out}}$		31.25	$^{\circ}\text{C}$	Calculated exit temperature
$T_{2,\text{out}}$	33.13		$^{\circ}\text{C}$	Calculated exit temperature

Regenerative heat exchangers

The degree of heat transfer of a regenerative heat exchanger depends on the incident-flow velocity v_a of the air and on the number of revolutions n of the wheel. As with the recuperators, the heat recovery efficiency increases with the rising ratio $UA/\dot{C}$, i.e. at a given transfer rate with a falling incident-flow velocity.

The concept for a rotary heat exchanger is based on the heat transfer of cross-flow heat exchangers. In this, the storage mass is simulated as a plate flowed around on both sides by air and moving perpendicularly to the air flow direction, thus cross current. After heat uptake in the warm phase, the storage mass of the wheel moves into the cool air section and gives off the heat taken up. With this model, the steady-state temperature gradient is obtained iteratively. The heat recovery efficiency improves, the greater the storage mass stream, i.e. the number of revolutions of the wheel.

For calculating the local temperature gradients in the regenerator and the outlet temperatures of the air, an imaginary partition is introduced between the air flow and the storage mass stream flowing perpendicularly to it; heat transfer takes place through this. The partition is therefore situated in the level of the thin flow channels of the regenerator and is limited by the building depth of the regenerator.

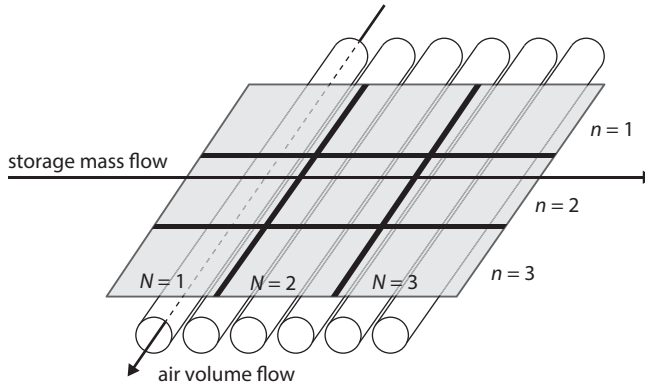


Figure 5.43 Imaginary partition surface between air flow and the storage mass stream, which is given by the mass of the channel walls.

To calculate the temperature distribution on the entry side of the air flow, the partition surface is divided into N sections and subdivided on the entry side of the storage mass stream into n sections. The surface of a partition element results from the total area $A_g/(Nn)$.

The heat flow $\dot{Q}$ transferred by a partition element results then from the thermal capacity stream of the air $\dot{C}_a = \rho \dot{V} c_{p,a}$ per segment N and the temperature difference between element entry $T_{a,1}$ and exit $T_{a,2}$, or from the thermal capacity stream of the storage mass $\dot{C}_s$ per segment n and the associated temperature difference $T_{s,2} - T_{s,1}$.

$$\dot{Q} = \frac{\dot{C}_a}{N} (T_{a,1} - T_{a,2}) = \frac{\dot{C}_s}{n} (T_{s,2} - T_{s,1}) \quad (5.141)$$

The heat removal from the air, or the heat uptake of the storage mass in the warm phase, is equal to the convective heat flow between the mean air temperature and the mean storage mass temperature of the respective element. For a partition element of surface $A_g/(Nn)$ and a heat transfer coefficient between air and storage mass of h_c , the result is:

$$\dot{Q} = h_c \frac{A_g}{Nn} \left(\frac{T_{a,1} + T_{a,2}}{2} - \frac{T_{s,1} + T_{s,2}}{2} \right) \quad (5.142)$$

From Equation (5.141) and Equation (5.142), there follows for the outlet temperatures:

$$\begin{aligned} T_{a,2} &= T_{a,1} - E_w (T_{a,1} - T_{s,1}) \\ T_{s,2} &= T_{s,1} + F_w (T_{a,1} - T_{s,1}) \end{aligned} \quad (5.143)$$

E_w and F_w are abbreviations for the following expressions:

$$E_w = \frac{\frac{h_c A_g}{\dot{C}_a} \frac{1}{n}}{1 + \frac{1}{2} \left(1 + \frac{\dot{C}_a}{\dot{C}_s} \frac{n}{N} \right) \frac{h_c A_g}{\dot{C}_a} \frac{1}{n}}$$

$$F_w = E_w \frac{\dot{C}_a}{\dot{C}_s} \frac{n}{N}$$
(5.144)

The calculation of the cold phase takes place in a similar way. Since the volumetric air flow rates of the inlet and exit air need be not identical, the thermal capacity stream of the air and of the convective heat transmission coefficient can change on the cold side.

The calculation process

The calculation begins with the warm phase.

As the first partition element (1,1) the cold storage mass element ($N = 1$) is selected, which first comes in contact with the entering warm air flow ($n = 1$). The second partition element ($N = 1, n = 2$) is then situated in the direction of flow of the warm air being cooled. Depending on the number of subdivisions n , first all temperatures $T_{1,n}$ of the storage mass elements $N = 1$ are calculated. Next the second air flow channel $N = 2$ with all subdivisions n is calculated. The result is, for example, for N and n from 1 to 3 the following calculation order:

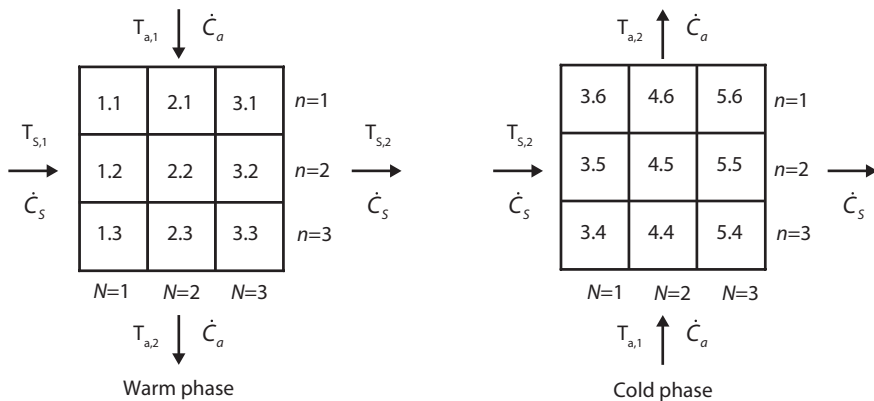


Figure 5.44 Calculation order for regenerative heat exchanger.

The thus determined temperature profile of the storage mass stream on exiting the warm period equals the entry profile into the cold period. With the temperature profile of the storage mass stream on exiting the cold phase, calculation of the warm phase again begins from the start, until

the outlet temperature profile of the storage mass stream no longer changes and the transferred amounts of heat are equal:

$$\begin{aligned}\dot{Q} &= \dot{Q}' \\ \dot{C}_a(T_{a,1} - T_{a,2}) &= \dot{C}'_a(T'_{a,2} - T'_{a,1})\end{aligned}\tag{5.145}$$

The operational characteristic can be calculated with the calculated average values of the outlet temperatures of the two gas flows. If the warmer gas flow represents the smaller thermal capacity stream, then this applies:

$$\Phi = \frac{\bar{T}_1 - \bar{T}_2}{\bar{T}_1 - \bar{T}'_1}\tag{5.146}$$

If the colder gas flow represents the smaller thermal capacity stream, then this applies:

$$\Phi = \frac{\bar{T}'_2 - \bar{T}'_1}{\bar{T}_1 - \bar{T}'_1}\tag{5.147}$$

Example 5.15

Calculation of the heat recovery efficiency of a regenerator with a wheel diameter of 90.5 cm, and a flow rate of 3000 m³ h⁻¹. Entry temperatures are 45.5° and 20°C

Regenerator				
n	10		min ⁻¹	Rotations per min
l	0.3		m	Depth of wheel
D	0.905		m	Diameter of wheel
m	40		kg	Rotating storage mass
A_{warm}/A_{cold}	50		%	Area ratio of warm air sector
Matrix				
d_i	0.0019		m	Inner diameter of capillary tubes (corresponds to hydraulic diameter)
A_{free}/A_{total}	91		%	Area ratio of free (open) cross section to total cross section
A_{hx}	366.3		m ²	Heat exchanging total surface of regenerator (internal surface)
c_s	870		J kg ⁻¹ K ⁻¹	Specific heat capacity of storage mass
A_{free}	0.2911	0.2911	m ²	Free cross section

	Warm air:	Cold air:		
λ_a	0.0279	0.0260	W m ⁻¹ K ⁻¹	Heat conductivity air
ρ_a	1.0938	1.1890	kg/m ³	Density of air flowing through regenerator
$c_{p,a}$	1008.3	1007.0	J kg ⁻¹ K ⁻¹	Heat capacity air
u_a	1.76×10^{-5}	1.52×10^{-5}	m ² s ⁻¹	Kinematic viscosity
V/t	0.83	0.83	m ³ s ⁻¹	Volume flow
T	45.50	20.00	°C	Temperature
V	2.86	2.86	m s ⁻¹	Mean flow velocity
Pr	0.696	0.698	-	
Re	309	358	-	
Nu	3.796	3.817	-	
h_c	55.69	52.27	W m ⁻² K ⁻¹	Heat transfer coefficient
A_g	183.15	183.15	m ²	Heat exchange area
C_a	918.66	997.35	W K ⁻¹	Heat capacity flow of air
C_s	5800.00		W K ⁻¹	Heat capacity flow of storage mass
$T_{air, mean}$	23.39	40.36	°C	
Φ	0.87		-	Heat recovery efficiency, calculated
T_{a1}		23.39	°C	
T_{a2}'	40.36		°C	

5.12 Technology humidifier

To achieve the evaporative cooling effect central to sorption-supported air conditioning, only humidification systems which bring liquid water but not vapour into the air can be used. The selection of the humidification system depends both on the water quality available at the location and on capital outlays, humidifier overall lengths and pressure losses. While spray humidifier systems produce very good humidification efficiencies with small pressure losses (95 - 100% at pressure losses of around 50 Pa), the capital outlays and the overall lengths are clearly higher than in simple contact humidifiers. These get along with a length of barely 60 cm, and achieve humidification efficiencies of over 90%, though the pressure losses vary according to manufacturers' data between 50 and 150 Pa. Water quality is an important issue, both for hygienic and cost reasons. Contact evaporators in contrast to spray evaporators do not transmit aerosols into the process air stream, but still require low mineral contents of the water.

The humidification efficiency η_h is defined as the ratio of the achieved absolute humidity increase (from x_{in} to x_{out}) to the maximum possible humidification to 100% relative humidity (corresponding to a maximum absolute humidity x_{max}).

$$\eta_h = \frac{x_{in} - x_{out}}{x_{in} - x_{max}} \quad (5.148)$$

To calculate the maximum absolute humidity x_{max} , it is easiest to assume isenthalpic humidification, gradually increase the absolute humidity, calculate the new temperature and relative humidity,

until 100% relative humidity is reached. More precise is the consideration of the sensible heat of the water added to the air.

To achieve the evaporative cooling effect for the sorption supported air conditioning, only humidifying systems can be utilised that introduce water into the air rather than steam. The choice of humidifying systems is dependent on both the local available water quality as well as on investment costs, humidifier construction length and pressure losses. While spray nozzle humidifying systems exhibit very good humidifying efficiencies with minimal pressure losses (95 - 100% with pressure losses of around 50 Pa), the investment costs and the construction lengths are significantly higher than with simple trickle humidifiers. These work with about 60 cm construction length and reach humidifying efficiencies of over 90%, whereas the pressure losses vary between 50 and 150 Pa according to manufacturer specifications.

For the modelling of an evaporative humidifier, Pietruschka (2010) developed a simple numerical model, which represents the humidifier characteristics very well.

For the model development the contact matrix is divided into finite elements as shown in Figure 5.45.

The mass of the water in the contact matrix is normally significantly lower than the amount of air transported through the humidifier. To simplify the model it is assumed that the water temperature is equal to the water temperature cooled by the evaporative humidification. Furthermore, the water mass and heat capacity of the matrix in the energy balance is neglected.

The energy balance describes the cooling of the air via evaporation of the water:

$$\begin{aligned} \dot{Q}_{air,i} &= h_{ev,i} \cdot \dot{m}_{w,ev,i} \\ \dot{m}_{air,i} \left(c_{p,air} (T_{air,in,i} - T_{air,out,i}) \right. \\ &\quad \left. + c_{p,vap} (T_{air,in,i} \cdot x_{air,in} - T_{air,out,i} \cdot x_{air,out}) \right) = h_{ev,i} \cdot \dot{m}_{w,ev,i} \end{aligned} \quad (5.149)$$

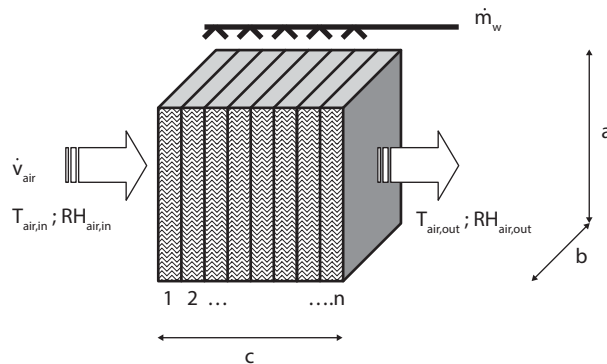


Figure 5.45 Segmentation of the humidifier's contact matrix.

The mass balance describes the increase of the water content of the air via evaporation:

$$\begin{aligned}\dot{m}_{air,i} \cdot \Delta x_{air,i} &= \dot{m}_{w,ev,i} \\ \Delta x_{air,i} &= x_{air,out} - x_{air,in}\end{aligned}\quad (5.150)$$

The water mass flow $\dot{m}_{w,ev,i}$ in Equation (5.149) is replaced using the mass balance from Equation (5.150). With this it results that

$$\left(c_{p,air} (T_{air,in,i} - T_{air,out,i}) + c_{p,vap} (T_{air,in,i} \cdot x_{air,in,i} - T_{air,out,i} \cdot x_{air,out,i}) \right) = h_{ev} \cdot \Delta x_{air,i} \quad (5.151)$$

$$T_{air,out,i} = T_{air,in,i} \left(\frac{c_{p,air} + c_{p,vap} \cdot x_{air,in,i}}{c_{p,air} + c_{p,vap} \cdot x_{air,out,i}} \right) + \frac{h_{ev} \cdot \Delta x_{air,i}}{c_{p,air} + c_{p,vap} \cdot x_{air,out,i}} \quad (5.152)$$

Following the mass transfer of water vapour from the liquid water film [kg s⁻¹] to the adjacent air flow depends on the mass transfer coefficient β , the wetting ε of the surface and the surface area A_{eff} . The driving force in the transport process is the difference in vapour pressure between the air and the water film interface. The mass transfer in flow direction y can be calculated from the following equation:

$$\dot{m}_{w,ev} = A \varepsilon \frac{\beta \cdot M}{R_m T} [p_{H_2O}(T_{air,in}) - p_{S,H_2O}(T_{water,in})] \quad (5.153)$$

The mass transfer coefficient β [m s⁻¹] depends on the flow pattern in the contact matrix channels and can be calculated from the heat transfer coefficient $h_{c,a}$ using the following equation:

$$\beta_A = \frac{h_{c,a}}{c_{p,a} \rho_a} \left(\frac{a_a}{\delta_a} \right)^{-(1-n)} \quad (5.154)$$

The heat transfer coefficient $h_{c,A}$ can be calculated as a function of the Nusselt number Nu and the characteristic length L . The characteristic length for rectangular channels is two times the distance between the adjacent walls.

$$h_{c,a} = \frac{Nu \cdot \lambda}{L} \quad (5.155)$$

If the matrix structure is assumed to be in the form of rectangular air channels, the Nusselt number can be calculated from the following correlation.

$$Nu = \left(7.541^3 + 1.841 \cdot \sqrt{Re \cdot Pr \cdot \frac{L}{l}} + \left(\frac{2}{1 + 22 \cdot Pr} \right)^{1/6} \cdot \left(Re \cdot Pr \cdot \frac{L}{l} \right)^{0.5} \right)^{1/3} \quad (5.156)$$

With the Prandtl number Pr and the Reynolds number Re defined as:

$$Pr = \frac{v_{air} \cdot \rho_{air} \cdot c_{p,air}}{\lambda_{air}}$$

$$Re = \frac{v_{air} \cdot L}{\nu_{air}}$$

Consideration of water storage effects in the structure of the contact matrix

The inertia of the humidifiers is mainly caused by water storage in the matrix structure. At system startup it takes some time until the matrix is fully wetted, which reduces the humidification efficiency. At system shutdown the water supply is cut off, but the matrix structure is still wet and the humidifier continues operation as if the water supply was turned on. When the matrix structure begins to dry, the humidification efficiency decreases slowly. For typical contact matrix humidifiers the drying process takes up to 1 hour until the matrix is completely dry. To consider this effect in the simulation model, a water storage capacity of the matrix structure $m_w(t)$ is used together with an effective water flow rate of the fresh or recirculation water. At system startup the matrix is dry ($m_w(t)=0$) and the water pump is switched on, which causes an effective water flow rate $\dot{m}_{w,pump}$, and the matrix structure starts to be wetted.

$$m_w(t) = m_w(t-1) + (\dot{m}_{w,pump} - \dot{m}_{w,ev}) \cdot \Delta t \quad (5.157)$$

The stored water $m_w(t)$ is limited to the maximum value defined for the matrix, structure. The reduced humidification efficiency is considered by a reduced available matrix surface area for the mass transfer.

$$A_{eff} = A \cdot \left(\frac{m_w(t)}{m_{w,max}} \right) \quad (5.158)$$

After shut down of the humidifier pump no water is supplied to the matrix, and the matrix structure starts to dry until $m_w(t) = 0$. The water storage effect is calculated for each segment separately. Thus, a realistic drying process from humidifier inlet to humidifier outlet is reached.

In this quite simple model no diffusion effects in the matrix structure are considered; therefore the final drying process is slightly faster than in reality. The overall agreement between measured and predicted performance of the humidifiers is good.

Limits and application possibilities of open sorption

Desiccant cooling air conditioning is thus suited to temperate and warm climates with not too high air humidities (under about 15 g per kg of dry air). Only in extremely dry climates can the desired air conditioning be achieved without a sorption wheel, via pure evaporative cooling.

Under design criteria of 32°C and 40% relative humidity, supply air statuses under 15°C and 9.5 g kg⁻¹ of humidity content can be achieved with a sorptive cooling system with a dehumidifying performance of 6 g kg⁻¹, but without a sorptive drying process barely 20°C and 13 g kg⁻¹. This humidity content is already clearly over the maximum admissible supply air value of 11.6 g kg⁻¹. Real sorption systems with humidifier efficiencies under 95% and heat recovery efficiencies

of between 70% and 75%, achieve under the above design criteria supply air statuses of around 17 - 19°C.

In a Mediterranean climate with mean monthly maximum temperatures of 36°C and 13 g kg⁻¹ humidity content, a supply air status of only 20°C and 14 g kg⁻¹ humidity content can be achieved without drying the air. The use of a sorption wheel, however, enables supply air statuses of 16°C and about 11 g kg⁻¹.

Concerning control of the system, the different outside air statuses must always be considered in order to achieve maximum energy efficiency. When cooling begins, first only the heat recovery device and exhaust air humidifier are switched on, and only at higher cooling loads are the sorption wheel, regeneration air heater and supply air humidifier used.

5.13 Design limits and climatic boundary conditions

Demands on room temperatures and humidities

With the models described for sorption wheels, heat recovery devices and humidifiers, the attainable supply air statuses for different outside air statuses can be calculated. Thus both the limitations of pure evaporative humidification and of sorption technology can be determined. At very high outside air humidities, procedure combinations with closed cycle refrigerant plants for supply air cooling must be examined.

According to the relevant norms for thermal comfort, a rise in the perceived temperature up to 27°C is allowed at high outside temperatures in the summer and with only briefly occurring high thermal loads. The upper boundary of the humidity content is set at 11.5 g kg⁻¹ or a maximum of 65% relative humidity. At a design room temperature of 26°C (which corresponds to the space exhaust air status), 11.5 g kg⁻¹ of absolute humidity results in a relative humidity of 55%.

For supply air conditioning a further specification applies, that the temperature difference between supply and room air must exceed $\Delta T = 10\text{ K}$ to prevent draughts and thermal discomfort. If for energy-saving reasons the room temperature is set at 26°C, the upper boundary of the comfort field, a minimum supply air temperature of 16°C results.

The maximum supply air humidity depends on the humidity loads of the space which are to be removed. If dehumidification is not necessary, the supply air can be moistened to the maximum admissible room value of 11.5 g kg⁻¹. From the outside air status of 32°C, 12 g kg⁻¹ humidity, only 0.5 g kg⁻¹ of humidity must then be removed, an energetically favourable change in status. In air-conditioning of administrative buildings, 8.45 g kg⁻¹ of supply air humidity is assumed to allow a removal of humidity loads, an accordingly energy-intensive process (case 2).

Regeneration temperature and humidity

The regeneration temperature determines the maximum dehumidification of the outside air. At high outside temperatures of over 30°C, 70°C regeneration air temperature can be achieved easily with flat-plate air or water collectors. The relative humidity of the regeneration air depends on the process: if the collector is flowed through by space exhaust air, the humidity is given by the exhaust air humidity plus the evaporative humidification of the exhaust air. For cost reasons, space gain and contamination of exhaust air, however, sucking in outside air for the collector, is often selected (only one air duct from the collector is necessary). The absolute humidity of the regeneration air is then equal to the outside air humidity.

Calculation of supply air status with different climatic boundary conditions

To evaluate the application possibilities of open adsorption air-conditioning systems, attainable supply air temperatures must be compared with the demands on the room air status. In the following, the temperatures are to be calculated for a design value of 32°C, 40% humidity (a temperate climate) and a damp climate with 35°C, 50% humidity for a sorption system with closed exit air discharge.

The methodology for the calculation of the air statuses is described as follows:

1. Specification of the outside air status: temperature and humidity.
2. Specification of the space exhaust air status: admissible temperature and humidity.
3. Specification of the regeneration air temperature.
4. Calculation of the exit air humidity after the evaporative humidifier as a function of the humidification efficiency (from manufacturers' data).
5. Determination of the regeneration humidity: equal to the exit air humidity in closed systems or equal to the outside air humidity with open suction of regeneration air.
6. Iterative calculation of the supply air status (temperature and humidity) after sorptive dehumidifying, as a function of the relative humidity of the regeneration air, taking into account a dehumidifying efficiency.
7. Calculation of the supply air temperature after the heat recovery device, as a function of the heat recovery efficiency.
8. Calculation of the supply air temperature and humidity after the evaporation humidifier, as a function of the humidification efficiency.
9. Checking the room air humidity and possibly reducing the supply air humidification.

For the two outside air statuses mentioned, the process is calculated in the following examples. The parameters to be given by the planner, such as efficiencies of humidification, dehumidifying, heat recovery, etc. are printed in *italics*.

1. Specification of outside air status	
Status A:	32°C, 40% relative humidity (12 g kg ⁻¹)
Status B:	35°C, 50% relative humidity (17.8 g kg ⁻¹)
2. Specification of space exhaust air status:	
	26°C, 11.5 g kg ⁻¹ (55%)
3. Specification of regeneration temperature:	
	70°C
4. Calculation of the exhaust air humidity	
Humidification efficiency:	95%

The maximum absolute humidity of the exhaust air at 100% relative humidity and isenthalpic humidification is 14.1 g kg⁻¹, i.e. in total 2.6 g kg⁻¹ could be added to the exhaust air. At a humidification efficiency of 95%, 2.5 g kg⁻¹ can be effectively added. The humidification takes place adiabatically and the temperature after humidification can be calculated from the enthalpy of the exhaust air (54.9 g kg⁻¹). For the vapour content of 11.5 g kg⁻¹ + 2.5 g kg⁻¹ = 14 g kg⁻¹ after humidification, the result is a new temperature of 20°C.

5. Determination of the regeneration air humidity

The relative humidity of the regeneration air at 70°C and 14 g kg⁻¹ is 7%. The relative humidity of the regeneration air is independent of the outside air status for closed exhaust air circulation. If outside air has been sucked in, the result for status A with 12 g kg⁻¹ of vapour content is a relative humidity of 6%, and for status B with 18 g kg⁻¹, 9.1%.

6. Iterative calculation of the supply air status after the drying process

Dehumidification efficiency:	80%

Status A: The relative humidity of the regeneration air determines the maximum dehumidification: to bring the supply air to ideally 7% relative humidity, 6.9 g kg⁻¹ must be adsorbed from the outside air. The effective dehumidifying performance is about 5.5 g kg⁻¹ and the temperature is 45.5°C, so a relative humidity of the dried supply air of 10.6% or absolute humidity of 6.5 g kg⁻¹ results.

Status B: The effective dehumidifying performance is about 7.6 g kg⁻¹ (residual moisture 10.2 g kg⁻¹) at a temperature of 54°C.

7. Calculation of the supply air temperature after the heat recovery wheel

Heat recovery efficiency:	80%

Status A: The supply air temperature after the heat recovery device is 25.1°C.

Status B: Supply air temperature 26.8°C.

8. Calculation of the supply air temperature and humidity after the humidifier

Humidification efficiency:	95%

Status A: With a humidification of the dried, pre-cooled supply air to 95% of the maximum, a supply air status of 15.4°C and 10.4 g kg⁻¹ humidity can be achieved.

Status B: The supply air status with humidification of 3.1 g kg⁻¹ is 19°C and 13.3 g kg⁻¹ humidity.

9. Comparison of maximum admissible room air humidity with given supply air statuses

Status A: The calculated 10.4 g kg⁻¹ of absolute humidity is related to the desired room air temperature of 26°C, resulting in a relative humidity of 50%, thus under the limit value of 55%.

Status B: The relative humidity of the supply air related to the room temperature is 63%, thus higher than the desired maximum value of 55%. An increase in the regeneration air temperature to 80°C would improve the supply air status to 18.3°C and 12.5 g kg⁻¹ humidity, which is however still above the humidity limit value for the room air status. Only if the direct humidification of the supply air is reduced can the humidity condition be met. However, the supply air temperatures are then so high that the cooling is insufficient: at a regeneration air temperature of 80°C the result is, for the space status with 55% humidity, a supply air temperature of 20.6°C. For such humid climates it is often necessary to further cool the supply air without adding extra humidity.

5.14 Energy balance of sorption-supported air conditioning

Usable cooling power of open sorption

Sorption-supported air-conditioning systems are driven with pure fresh air. The cooling capacity $\dot{Q}_c$ is therefore calculated from the enthalpy difference between the outside air status h_o and the supply air status h_{in} . The removable cooling load from the room $\dot{Q}_l$, however, is given by the enthalpy difference between supply air h_{in} and space exhaust air h_r , with the space exhaust air temperature usually several kelvins under the outside temperature. What proportion of the cooling capacity produced is usable depends in particular on the required dehumidifying performance as well as on the necessary fresh air flow rate, which must in every case be cooled from the outside air status, even in conventional air-conditioning systems.

$$\begin{aligned}\dot{Q}_c &= \rho \dot{V} (h_o - h_{in}) \\ &= \rho \dot{V} \left((c_a + x_o c_v) T_o + x_o h_e - ((c_a + x_{in} c_v) T_{in} + x_{in} h_e) \right)\end{aligned}\quad (5.159)$$

$$\begin{aligned}\dot{Q}_l &= \rho \dot{V} (h_r - h_{in}) \\ &= \rho \dot{V} \left((c_a + x_r c_v) T_r + x_r h_e - ((c_a + x_{in} c_v) T_{in} + x_{in} h_e) \right)\end{aligned}\quad (5.160)$$

For the three most important applications of sorption-supported air conditioning the cooling power can be determined by Equation (5.159) and the removed cooling load from the room by Equation (5.160).

Cases:

1. Pure cooling of the outside air with minimum dehumidifying to 11.5 g kg^{-1} .
2. Cooling of the outside air to 16°C supply air temperature with dehumidifying to 8.5 g kg^{-1} .
3. Pure dehumidifying of the room air to 8.5 g kg^{-1} without additional cooling, i.e. the supply air temperature equals 26°C .

The enthalpy of the outside air remains constant at 62 kJ kg^{-1} here, at design criteria of 32°C and 40% relative humidity (12 g kg^{-1}). The enthalpy of the space exhaust air is 26°C , with 55% relative humidity (11.5 g kg^{-1}) at 54.9 kJ kg^{-1} .

Case 1 - Pure cooling with minimum dehumidifying:

With the inlet air humidity specification of 11.5 g kg^{-1} , at 95% humidification and good heat recovery efficiencies of 80%, a minimum supply air temperature of 17°C is possible. The enthalpy difference between the outside air and supply air is

$$h_o - h_{in} = 62 - 45.7 = 16.3 \text{ [kJ kg}^{-1}\text{]}$$

From this a cooling capacity for $1000 \text{ m}^3 \text{ h}^{-1}$ flow rate results:

$$\rho \dot{V} (h_o - h_{in}) = 1.18 \text{ kg/m}^3 \times 1000 \text{ m}^3 / 3600 \text{ s} \times 16.3 \times 10^3 \text{ J/kg} = 5343 \text{ W}$$

With a flow rate of $1000 \text{ m}^3 \text{ h}^{-1}$, however, only a sensible cooling load of 3 kW can be removed.

$$\rho \dot{V} (h_r - h_{in}) = 1.19 \text{ kg/m}^3 \times 1000 \text{ m}^3 / 3600 \text{ s} \times (54.9 - 45.7) \text{ kJ/kg} = 3041 \text{ W}$$

Thus if only a sensible cooling load is to be removed, without a fresh air requirement existing, the sorption system must produce 1.8 times more cold than is needed as cooling output; an energetically unfavourable application.

Case 2 - Cooling with dehumidification:

If humidity loads of the space must be removed (here, for example, 3 g kg⁻¹ from 11.5 g kg⁻¹, to 8.5 g kg⁻¹), the energy expenditure for air conditioning clearly becomes higher. At such high air-drying performance of the sorption wheel, the supply air temperature must now be limited to a minimum value, since the usual 95% humidification would produce supply air temperatures far below 16°C.

The enthalpy difference rises to

$$h_o - h_m = 62 - 37.3 = 24.7 \text{ [kJ kg}^{-1}\text{]}$$

and thus the cooling capacity to 8.1 kW per 1000 m³ h⁻¹ flow rate.

The cooling load of the space now consists of sensible heat and latent heat of dehumidifying, and the enthalpy difference is

$$h_r - h_m = 54.9 - 37.3 = 17.6 \text{ [kJ kg}^{-1}\text{]}$$

The total cooling power is still 1.4 times higher than the cooling load removal from the room of 5.8 kW per 1000 m³ h⁻¹, so here too a high fresh air requirement offers a favourable initial position for open sorption cooling.

Case 3 - Pure dehumidifying:

To dehumidify the room air to 8.5 g kg⁻¹, i.e. by 3 g kg⁻¹, without cooling, the outside air at 32°C and 12 g kg⁻¹ must be dehumidified by 3.5 g kg⁻¹.

$$h_o - h_m = 62 - 47.3 = 14.7 \text{ [kJ kg}^{-1}\text{]}$$

The necessary cooling performance per 1000 m³ h⁻¹ volumetric air flow is 4.9 kW.

If dehumidifying were carried out with recirculating air, i.e. not fresh air but space exhaust air, the enthalpy difference would be reduced to

$$h_r - h_m = 54.9 - 47.3 = 7.6 \text{ [kJ kg}^{-1}\text{]}$$

The removable cooling load is around 2.5 kW.

The different energy expenditures on the basis of the constant outside air status are summarised in Table 5.8. The outside air status is given with 32°C, 40% relative humidity and an enthalpy of 62 kJ kg⁻¹.

Case	1	2	3
Inlet air status:			
Temperature	17°C	16°C	26°C
Absolute humidity	11.5 g kg ⁻¹	8.5 g kg ⁻¹	8.5 g kg ⁻¹
Enthalpy inlet air/kJ kg ⁻¹	45.7	37.3	47.3

Enthalpy difference outside air to inlet air/kJ kg ⁻¹	16.3	24.7	14.7
Cooling power/kW per 1000 m ³ h ⁻¹	5.3	8.1	4.9
Enthalpy difference room exhaust air to inlet air/kJ kg ⁻¹	9.2	17.6	7.6
Removable cooling load/kW per 1000 m ³ h ⁻¹	3.0	5.8	2.5

Table 5.8 Total cooling power and cooling load removal of open sorption-supported air conditioning with different applications.

An optimal field of deployment for sorption-supported air conditioning is found in applications with a high fresh air requirement. At high space-cooling loads with dehumidifying needs but little fresh air requirement, a combination of sorption systems with closed cycle chillers is suitable for separating dehumidifying from load removal. From the ratio of cooling power to regeneration heat, the energy performance figures of open sorption can be determined in what follows.

Coefficients of performance and primary energy consumption

To make available a kilowatt-hour of cold, compression refrigerator with an EER of 3 requires a total of 0.33 kWh of electricity. The EER is generally defined as the ratio of produced cooling power to the supplied power, either electrical power or heat.

$$COP = \frac{\dot{Q}_{cooling}}{\dot{Q}_{supply}} \quad (5.161)$$

At an average primary energy conversion efficiency η_{con} for electricity production of 35%, 0.95 kWh of primary energy is used for 0.33 kWh of electricity. The primary energy efficiency η_{pe} as a ratio of the produced cooling energy to the supplied primary energy results from the product of EER and the conversion efficiency of the respective energy carrier.

$$\eta_{pe} = \frac{\dot{Q}_{cooling}}{\dot{Q}_{primary\ energy}} = COP \times \eta_{con} \quad (5.162)$$

In electrical compression chillers, η_{pe} is around $3.0 \times 0.35 = 1.05$. If compression chillers are operated in a full air-conditioning system, reheating is often necessary after dehumidifying due to the low dew point temperature, and the mean primary energy efficiency falls to 0.6; i.e. for a kWh of cold, 1.7 kWh of primary energy are used.

With sorption-supported air-conditioning systems, both thermal energy for regeneration and electricity for fans and auxiliary aggregates such as humidifier pumps and wheel drives must be supplied. Firstly, the purely thermal EER should be considered, i.e. the ratio of cooling power produced to the necessary regeneration heat.

$$EER = \frac{\dot{Q}_{cooling}}{\dot{Q}_{regeneration}} = \frac{h_{outside/exhaust} - h_{supply}}{h_{reg} - h_{after\ HX/outside}} \quad (5.163)$$

If total cooling power from outside air is considered, the enthalpy difference between outside and supply air ($h_{outside} - h_{supply}$) must be selected. If only room loads are removed, the difference between room exhaust air and supply air ($h_{exhaust} - h_{supply}$) is to be used.

The regeneration power in the denominator is calculated as the enthalpy difference between entry into the (solar) regeneration air heater and exit from the heater with enthalpy h_{reg} . For closed exhaust air systems, the entry air into the heater is the exit air after humidification and the heat exchanger (h_{HX}); for open exhaust systems, outside air is used with enthalpy $h_{outside}$. The enthalpy after heating depends on the regeneration temperature necessary for the respective application.

As an example, the respective performance figures for the three applications of open sorption can be calculated, for closed exhaust air systems. As a boundary condition, an outside air status of 32°C and 40% relative humidity is selected. The specification is the desired supply air status (cooling with or without drying); the regeneration temperature is calculated as a function of the air to be supplied.

Case	1	2	3
Supply air			
Temperature	17°C	16°C	26°C
Humidity supply air	11.5 g kg ⁻¹	8.5 g kg ⁻¹	8.5 g kg ⁻¹
Temperature regeneration air/°C	53.3	95.1	48.3
Enthalpy regeneration/kJ kg ⁻¹	88	129.7	83.1
Enthalpy exhaust after HX/kJ kg ⁻¹	70.9	79.2	69.3
Enthalpy increase regeneration/kJ kg ⁻¹	17.1	50.5	13.8
EER from outside air/-	0.93	0.48	1.05
EER room load/-	0.53	0.35	0.55

Table 5.9 Performance figures of open sorption-supported air conditioning with a closed exhaust air system.

With a falling regeneration temperature, less dehumidifying occurs and the necessary amount of heat falls. At very low outside air humidities, air conditioning can take place by energy-neutral evaporative cooling alone, so the thermal performance figure becomes infinite in extreme cases without dehumidifying.

However, there is a low limit to the regeneration air temperature dependent on external humidity. For example, at an air status of 32°C and 40%, the performance figure ($EER_{thermal}$) does rise with a falling regeneration temperature. If the regeneration temperature falls below 52°C, there is so little dehumidification that in the supply air humidifier humidifying to 95% can no longer take place (due to the maximum admissible room air humidity). This results in an inadmissible rise in the supply air temperatures.

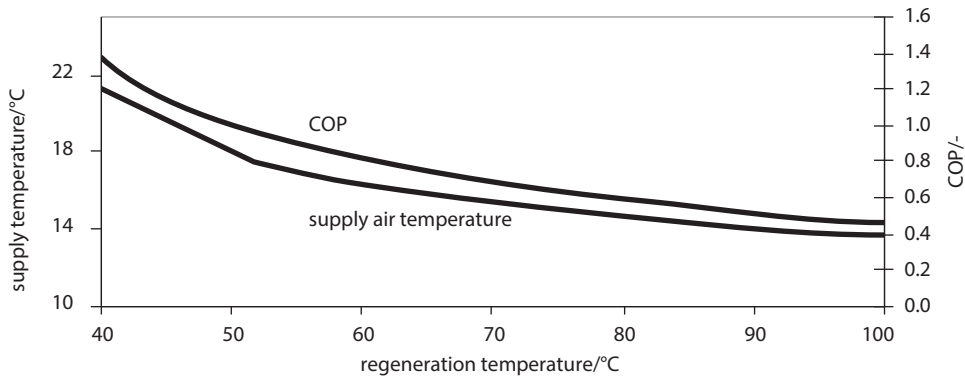


Figure 5.46 Supply air temperatures and cooling performance figures (EER) as a function of the regeneration air temperature at constant outside air statuses of 32°C and 40% relative humidity.

Although energetically very interesting, pure evaporative cooling is limited to dry outside air statuses and is only possible for a limited number of hours of operation. By using thermal solar energy, however, the regeneration heat can likewise be produced primary-energy neutrally at full sorption operation.

For a total energy balance, the additional pressure losses through the sorption wheel, heat recovery device and humidifier, and the associated electrical power increase, must be considered. At a typical flow velocity of 3 m s⁻¹ in the sorption system, pressure losses of about 150 - 200 Pa result in the sorption wheel and heat recovery device respectively, and in the humidifier between 100 and 250 Pa, depending on the design. The total of supply-side and exhaust side pressure losses is between 800 and 1300 Pa. For a 100 m² air collector field as a regeneration air heater, pressure losses of about 250 Pa can be expected.

From the total pressure losses Δp the electrical power P_{el} of the fans is calculated as a function of the fan efficiency η . At an efficiency of a large fan of 70%, the result is thus an electrical power demand of 417 - 615 W per 1000 m³ h⁻¹ of volumetric air flow, with total pressure losses between 1050 and 1550 Pa. In addition there are about 100 W per 1000 m³ h⁻¹ for electric drives of the components (circulation pumps, wheel drive, etc.).

$$P_{el} = \frac{\dot{V} \Delta p}{\eta} = \frac{1000 \text{ m}^3 / 3600 \text{ s} \times 1050 \text{ Pa}}{0.7} = 417 \text{ W}$$

Altogether, therefore, the result is connected electrical loads of some 500 - 700 W per 1000 m³ h⁻¹ of flow rate, i.e. about 1.4 - 2 kW primary energy requirement. Thus a cooling capacity of between 4.9 and 8.1 kW can be produced, depending on the application, i.e. the electrical primary energy efficiency is between 2.4 and 5.8. This value contains the pressure losses both for the heat recovery and the humidification function. These must also be considered during conventional cooling by compression refrigerant plants as part of a full air-conditioning system. If the heat is supplied either primary energy-neutrally by solar energy or waste heat is used, the desiccant cooling process is primary-energetically clearly superior to electrical compression refrigerant plants.

5.15 Closed cycle adsorption cooling

Technology and areas of application

Closed adsorption chillers operate with similar material pairs as in open cycle adsorption systems, for example, silica gel and water, but with the refrigerant water being led in the closed cycle. At low pressure, heat is extracted from the environment by evaporation of the water (i.e. usable cold is produced). The compression of the water vapour to the pressure in the condenser necessary for liquefaction takes place via a thermal compressor: the water vapour is first adsorbed on silica gel (suction function) and afterwards desorbed by the heat supply and brought to the necessary pressure. Cold water at 5 - 15°C inlet temperature is produced by the closed refrigerant circulation, and can then be distributed in the building with small pipe cross sections, a substantial advantage over purely air-led systems with large air pipes. Cooled ceilings with high cooling inlet temperatures of around 15°C can be operated with performance figures of 0.7, due to the low-temperature rise.

Up to recent years adsorption refrigerant plants were mainly constructed for large power ranges between 50 and 500 kW, and due to the small numbers of machines manufactured require high capital costs (around 500 Euro per kW of cooling capacity in large systems). In comparison to closed absorption systems on a water/lithium bromide basis, the substantial advantage of adsorption technology lies in the possible use of low heating temperatures down to 70°C. With solar energy or waste heat, usable cold can be produced with performance critically depending on the heat rejection temperature. In the last decade, several manufacturers also produced low power machines with 5 - 15 kW cooling power.

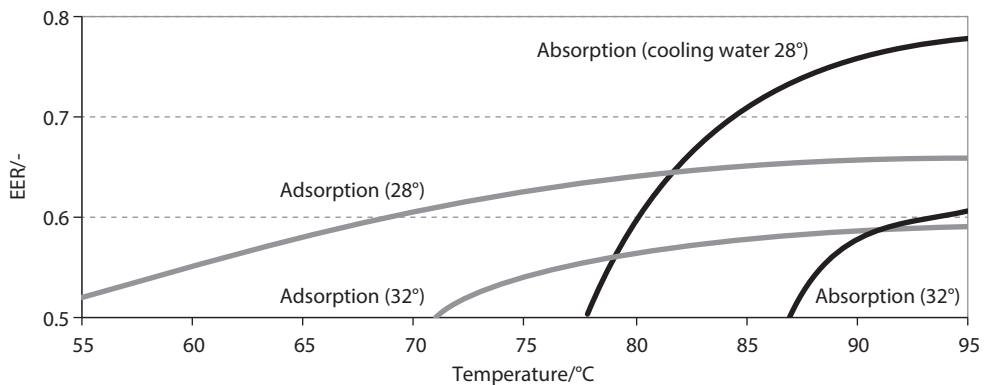


Figure 5.47 Coefficients of performance of closed adsorption and absorption chillers as a function of generator temperature.

For the design, the temperature boundary conditions for the hot water circuit (supply and return temperature), the heat rejection circuit (supply and return) and the cold water circuit must be known. Generally it is favourable to operate the chiller at high cold water temperatures, since per kelvin rise in temperature the power rises by around 8%.

For high performance figures, low cooling water temperatures for adsorber and condenser heat removal produced by a cooling tower are of special importance. In the cooling tower system,

in contrast to LiBr water absorption systems, no lower temperature limit is necessary since no crystallisation of the sorbent can occur. At too low temperatures in the evaporator (under 4°C), the evaporator has to be warmed up by the heating water circuit to avoid freezing of the evaporator heat exchanger.

Between adsorption of the refrigerant water and desorption of the sorption material silica gel, switching takes place cyclically between two chambers, so that quasi-stable operating conditions are achieved. The adsorption plant is controlled via the cycle duration as a function of the cold water inlet temperature at the evaporator: at too high inlet temperatures the adsorption process is terminated and switched to the chamber with dry sorption material. Due to the higher refrigerant flow rates into this chamber, the cooling capacity rises. If the cold water return temperature falls below a desired value, the cycle is extended, the adsorption rate sinks with increasing saturation of the sorption material and the cooling capacity falls.

Before initial operation, the four process chambers (evaporator, condenser and two sorption chambers) are evacuated with a small vacuum pump to an operating pressure of approximately 1000 Pa. The pump is operated briefly every 60 operational hours to remove desorbed gases from the materials or leakage air from the armatures.

The cold supply temperature fluctuates during a cycle by about ± 3 K around the desired value. A cold store of approximately 1/40th of the hourly cooling flow rate effectively buffers the fluctuations.

Costs

The capital outlays for a thermally operated adsorption chiller is clearly higher than the cost of a conventional air-cooled cold water set. For a 350 kW system, a capital outlay of approximately 160 000 Euro must be expected, compared with 60 000 Euro for a compression refrigerant plant. With a high annual operating time (6000 full-use hours yearly) and waste-heat use or already available solar thermal plants with low heat costs (0.01 Euro kWh⁻¹), however, economical operation can be achieved, according to manufacturers' information, due to the low operating cost (GBU mbH, 1998).

Operational principle

An adsorption chiller usually consists of two or more chambers filled with silica gel which are used alternately for water vapour adsorption and desorption, and enable a quasi-continuous process. The heat of adsorption, or the heat necessary for desorption, is removed or supplied by heat exchangers, whose ribs are tightly packed with silica gel for good thermal contact.

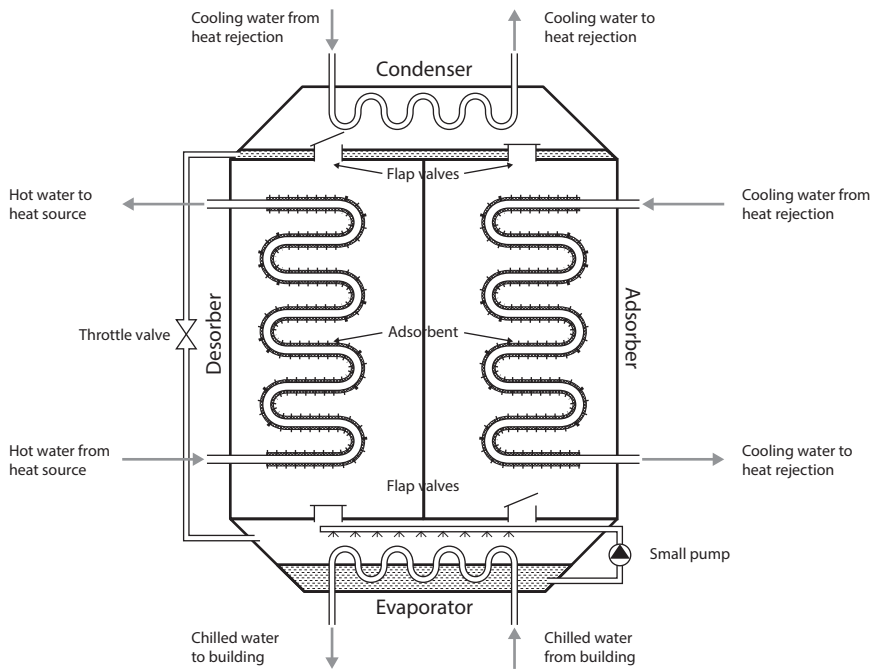


Figure 5.48 Structure of an adsorption chiller.

The two adsorbent chambers contain heat exchangers surrounded by the adsorption material, for example, silica gel, so heat of adsorption can be removed and heat supplied for desorption. The heat exchangers used in the condenser and the evaporator are circular tubes (usually copper or steel). The heat exchanger construction of the two adsorber beds differs, depending of the manufacturer, according to the technique used to bring the adsorbent on the heat exchangers. Nyshiyodo applies circular-tube and circular-fin configuration while Mayewaka uses circular-tube and a rectangular-fin structure for the construction of their heat exchangers. For both manufacturers, the adsorbent (silica gel) is then packed between the fins to obtain a large adsorbent surface area and good heat transfer. The German company Sortech developed a new coating technique with epoxy resin to bring the adsorbent directly on the circular-tube heat exchangers.

Four flap valves are used to connect or disconnect the two adsorber beds from the evaporator and the condenser. These valves can be pressure controlled (opened or closed according to pressure difference) without additional mechanical work or time controlled (the valves are closed or opened mechanically after a certain time). A throttle valve or a capillary tube is usually used to allow the liquid refrigerant to flow from the high pressure condenser to the low pressure evaporator. A small circulation pump is used to pump the liquid water of the lower part of the evaporator and spray it on the heat exchanger for a better evaporating process. Since water is used as refrigerant, the pressure inside the vessel has to be kept below atmospheric pressure and a vacuum pump has to be used for the initial start of the process and periodically during operation.

The process consists of two strokes and a short switching phase between the two strokes. In stroke 1 the lower valve flap of one of the two silica gel chambers (Bed 2) is opened to the evaporator (on the right in Figure 5.48), and the water vapour produced in the evaporator is adsorbed on the

dry, pre-cooled silica gel. The upper valve flap to the condenser of this chamber is closed. Loading takes place at low evaporator pressure (e.g. 1000 Pa at 5°C), and the freed adsorption enthalpy is removed by the cooling water. The possible water vapour load of the silica gel rises with falling coolant temperature $T_{c, in}$, which thus determines the end of adsorption. In the second chamber (left in Figure 5.48), the valve flap is closed to the evaporator and the flap to the condenser opened during the first stroke. By heat supply, the water vapour collected in the previous stroke is expelled and liquified at condenser pressure.

In stroke 2 the valve flaps operate in exactly opposite directions. The adsorbed water vapour of the first chamber is now expelled by heat supply into the condenser (valve flap to the condenser open, to the evaporator closed). In the second chamber the dried silica gel is used to adsorb water vapour from the evaporator.

Between the strokes, there is a switching phase of approximately 20 seconds for heat recovery, both chambers being flowed through alternately with cooling or heating water. The heating water is used to preheat the adsorption chamber of the preceding stroke, and the cooling water pre-cools the hot desorption chamber. A typical cycle lasts 400 seconds, so including the switching phase a stroke lasts 7 minutes.

Energy balances and pressure conditions

The process steps of a closed adsorption chiller can be clarified in an isostere (constant concentration) diagram, in which the vapour pressure of the refrigerant water p_r is plotted logarithmically as a function of the inverse temperature $1/T$ [K⁻¹]. The diagrams result from the Clausius equation of the adsorption of water vapour by silica gel already discussed for open sorption systems.

$$p_r = p_{r1}(C) \exp \left(\frac{h_{ads}(C)}{R_D} \left(\frac{1}{T_1} - \frac{1}{T} \right) \right) \quad (5.164)$$

The parameters of the Clausius equation p_{r1} (vapour pressure at temperature T_1) and the adsorption enthalpy h_{ads} depend on the load concentration C and are calculated iteratively from the sorption isotherm at 40°C.

The adsorption enthalpy results from the total concentration-dependent binding enthalpy and assumed temperature-independent evaporation enthalpy at 20°C. The vapour pressure of the refrigerant water ($C = 1$) is calculated most precisely with the saturation vapour pressure formula for water.

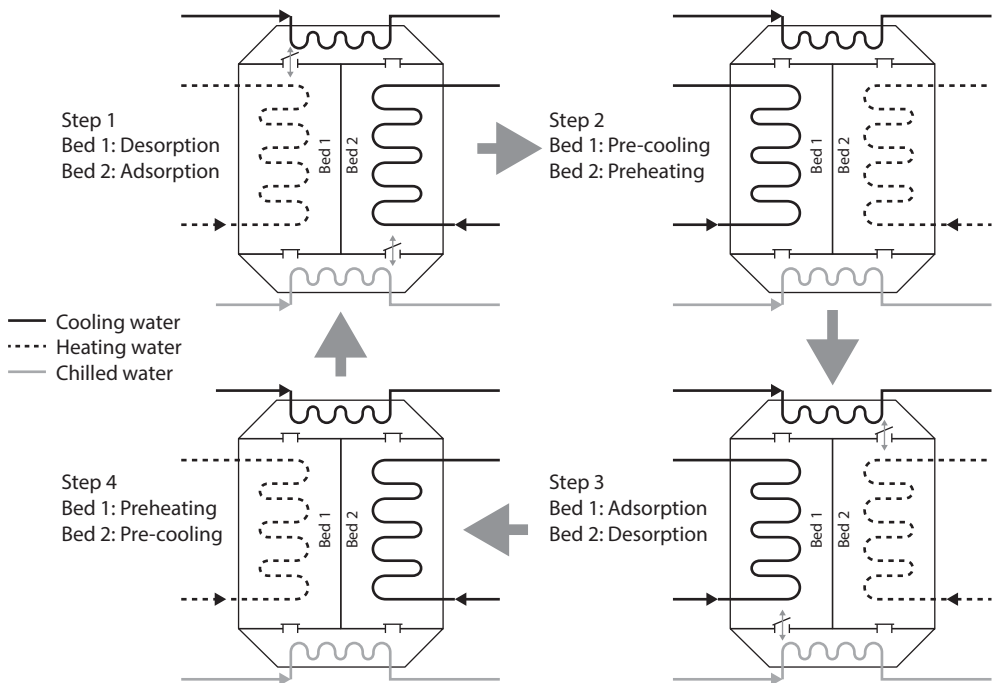


Figure 5.49 Working principle of adsorption chillers.

Load concentration C $/\text{kg}_{\text{H}_2\text{O}} \text{kg}_{\text{sor}}^{-1}$	Adsorption enthalpy h_{ads} $/\text{kJ kg}^{-1}$	vapour pressure p_{r1} at $40^\circ\text{C}/\text{Pa}$
0.05	2849	465
0.10	2622	1392
0.15	2490	2185
0.20	2453	2738
0.25	2453	3259
0.30	2453	3879
0.35	2453	4896
0.40	2453	6445

Table 5.10 Adsorption enthalpy and vapour pressure p_{r1} as a function of the load concentration of water on silica gel $(/\text{kg}_{\text{H}_2\text{O}} \text{kg}_{\text{sor}}^{-1})$.

The vapour pressure over the sorption material rises with temperature and the load concentration. In the isostere diagram, therefore, the straight line of highest pressure is the saturation vapour pressure of water at maximum load concentration $C = 1$. On this saturation vapour pressure curve are the status points 1 and 2 of the closed adsorption process. The pure refrigerant water is injected in the evaporator, evaporated there at low pressure (1), compressed by the thermal adsorption/desorption process and finally liquefied in the condenser at a higher pressure (2). The higher the

evaporator temperature, the higher the vapour pressure of status point 1, which determines the low pressure level of the adsorption cooling process.

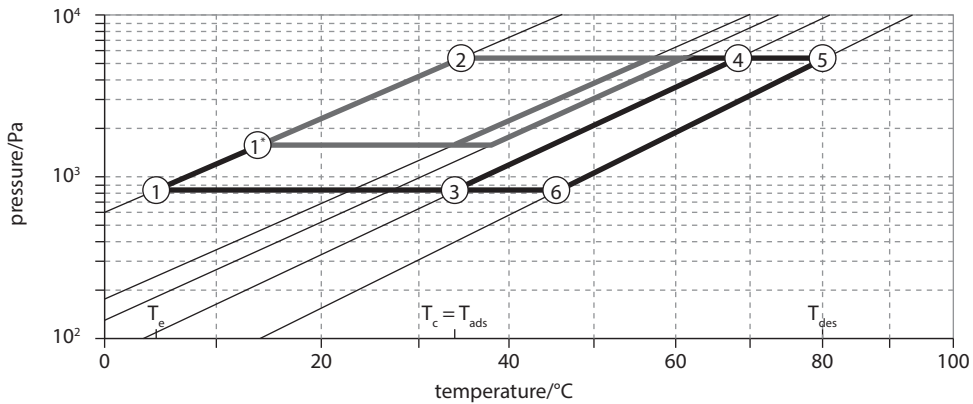


Figure 5.50 Water vapour pressure over silica gel as a function of the load concentration C (isosteres).

Illustrated are two processes with a very low evaporator temperature (4°C, status point 1) for cold water production and a higher evaporator temperature (14°C, status point 1*) for cooled ceiling applications. The high pressure level (status points 2, 4 and 5) is determined by the temperature of the condenser T_c , which also corresponds to the temperature of the adsorbent T_{ads} , if the cooling water is connected in parallel to both components. The maximum process temperature is the desorption temperature T_{des} , at which the refrigerant from the silica gel is expelled (status point 5). From the process comparison it is evident that at low evaporator temperatures (status point 1), high desorption temperatures are necessary (here 80°C), while at high evaporator temperatures desorption temperatures under 60°C suffice.

The pressure conditions and energy balances are represented for the evaporator and condenser first, and afterwards the cyclic process of adsorption and desorption of water vapour by silica gel is analysed.

Evaporator

In the evaporator the saturation vapour pressure p_e prevails; it depends on the design temperature of the chilled water circuit (status point 1 on the $\ln p - 1/T$ curve for pure water in Figure 5.50). Since heat is removed from the external circuit through a heat exchanger, the temperatures in the evaporator must be about 2 - 5 K below the external cold water inlet temperature.

The circulating chilling water quantity $\dot{m}_{ch}$ results from the desired cooling capacity of the machine. The evaporation enthalpy of the refrigerant water at evaporator temperature T_e is used to cool the chilled water circuit from return temperature $T_{r,ch}$ at the evaporator entrance to supply temperature $T_{s,ch}$ at the evaporator exit (useful energy), and to pre-cool the liquid, still warm refrigerant from the condenser (not useful for cold production). The outlet liquid refrigerant temperature from the condenser T_c is around 5 K over the cooling water return temperature, depending on the heat exchanger sizing of the cooling water circuit. The mass flow of the

refrigerant water $\dot{m}_r$ is calculated from the sum of the useful cooling power and the required condenser pre-cooling power. The performance balance of the evaporator therefore reads:

$$\dot{m}_r(h_v - h_l) = \dot{m}_{ch}c_p(T_{r,ch} - T_{s,ch}) + \dot{m}_r c_p(T_c - T_e) \quad (5.165)$$

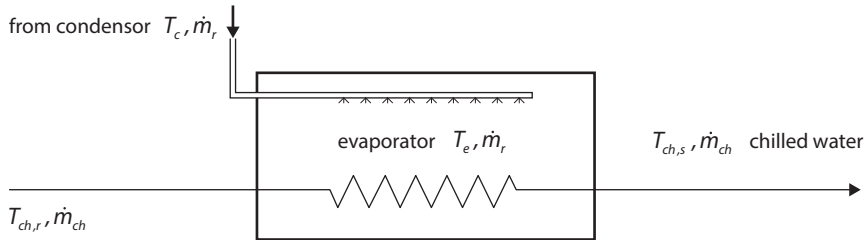


Figure 5.51 Temperatures and mass flows at the evaporator.

Example 5.16

An adsorption chiller is to be operated with 200 kW of cooling capacity at a cooling water supply temperature of 29°C (return 33°C). The cold water and evaporation mass flows necessary for this, plus the heat of evaporation, are to be calculated. The cold water set is to produce 6°C supply temperature and 12°C return temperature. In the evaporator a temperature of 4°C must be delivered at a temperature difference at the heat exchanger of 2 K.

For simplicity, calculation is based in all cycles on the material values of pure water. The chilling water mass flow results from:

$$\dot{m}_{ch} = \frac{\dot{Q}_{ch}}{c_p(T_{r,ch} - T_{a,ch})} = \frac{200 \text{ kW}}{4.190 \frac{\text{kJ}}{\text{kgK}} 6 \text{ K}} = 7.955 \frac{\text{kg}}{\text{s}} \hat{=} 28.6 \frac{\text{m}^3}{\text{h}}$$

The outlet temperature from the condenser is about 38°C, at a cooling water return temperature of 33°C and a temperature difference over the heat exchanger of 5K.

The evaporation mass flow is then calculated by dissolution of Equation (5.164). The evaporation enthalpy of water at 4°C is 2492 kJ kg⁻¹.

$$\dot{m}_r = \frac{\dot{m}_{ch}c_p(T_{r,ch} - T_{s,ch})}{(h_v - h_l) - c_p(T_c - T_e)} = \frac{7.955 \frac{\text{kg}}{\text{s}} 4.190 \frac{\text{kJ}}{\text{kgK}} 6 \text{ K}}{2492 \frac{\text{kJ}}{\text{kg}} - 4.19 \frac{\text{kJ}}{\text{kgK}} (38 - 4) \text{ K}} = 0.086 \frac{\text{kg}}{\text{s}}$$

To pre-cool the still warm condenser water, therefore,

$$\dot{m}_r c_p(T_c - T_e) = 0.086 \frac{\text{kg}}{\text{s}} 4.19 \frac{\text{kJ}}{\text{kgK}} (38 - 4) \text{ K} = 11.6 \text{ kW}$$

of cooling capacity must be used, which has to be produced by the evaporator in addition to the 200 kW of cooling capacity for cold water production.

The above energy balance of the evaporator assumes that the refrigerant mass flow $\dot{m}_r$ is constant. However, in a real adsorption machine with limited amount of adsorbent, the removal of refrigerant from the evaporator by the suction of the adsorption chamber slows down with time, as the adsorption material starts to saturate. Only at short cycle times can the effect be ignored, otherwise the dynamics of the adsorption process need to be considered.

Condenser

In the condenser (status point 2 in Figure 5.50) the desorbed refrigerant water $\dot{m}_r$ is condensed at high pressure, and the condensation heat has to be removed by a cooling water circuit with mass flow $\dot{m}_{cool,1}$. The lowest attainable condenser temperature T_c is at a cooling water inlet temperature of $T_{cool,1}$ plus the temperature difference ΔT_{hx} , typically 5 K, necessary at the heat exchanger.

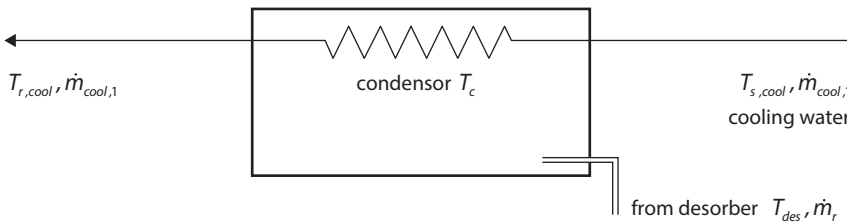


Figure 5.52 Temperatures and mass flows at the condenser.

The pressure level in the condenser determines the switching point for the desorption process. As soon as the pressure necessary for condensation is achieved in the heating phase, the valve between the desorption chamber and condenser is opened and heated further at constant pressure to the final desorption temperature T_{des} in order to desorb the refrigerant. The lower the cooling water inlet temperature $T_{s,cool}$, the lower is the pressure necessary in the condenser.

The evaporation enthalpy and the sensible heat of the water vapour heated to the desorption temperature must be removed by the cooling water, with the cooling water with mass flow being heated to the return temperature $T_{r,cool}$. Typical temperature spreads of the cooling water are about 4 K.

$$\dot{m}_{cool,1} c_p (T_{r,cool} - T_{a,cool}) = \dot{m}_r (h_v - h_l) + \dot{m}_r c_p (T_{des} - T_{a,cool} + \Delta T_{hx}) \quad (5.166)$$

Example 5.17

Calculation of the necessary condenser cooling-water mass flow for the cooling machine of Example 5.16 for a final desorption temperature T_{des} of 90°C.

The circulating refrigerant mass flow must first be cooled from the desorber to the condenser temperature, resulting in a cooling power requirement of

$$0.086 \frac{\text{kg}}{\text{s}} 4.19 \frac{\text{kJ}}{\text{kgK}} (90 - 29 + 5) \text{K} = 23.8 \text{ kW}$$

With a condensation enthalpy of 2420 kJ kg^{-1} at 34°C condenser temperature, the total power to be removed is $208 \text{ kW} + 23.8 \text{ kW} = 231.8 \text{ kW}$ and the resulting necessary cooling water mass flow at a temperature spread of 4 K is

$$\dot{m}_{\text{cool},1} = \frac{231.8 \text{ kW}}{4.19 \frac{\text{kJ}}{\text{kgK}} 4 \text{ K}} = 13.8 \frac{\text{kg}}{\text{s}} = 50 \frac{\text{m}^3}{\text{h}}$$

The pressure level in the condenser at a condenser temperature of 34°C is 5324 Pa . This pressure must be produced by heating the sorption material during the desorption process.

The adsorption process

In the adsorber, in the ideal process, the same pressure appears as in the evaporator (status point 3). The load level of the silica gel (the isostere) is given by the temperature of the adsorbent material T_{ads} , which is at a minimum of the cooling water inlet temperature (plus heat exchanger temperature difference ΔT_{hx}). The status point thus results from the intersection of the evaporator pressure line and the coolant temperature. The lower the coolant water temperature, the more water vapour can be adsorbed.

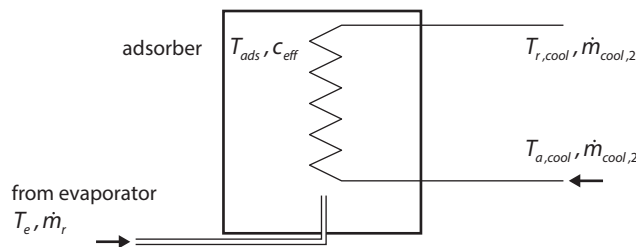


Figure 5.53 Temperatures and mass flows in the adsorber.

The pressure level in the adsorbent is determined by the vapour pressure in the evaporator, which is very low for the refrigerant water. At an evaporator temperature of 4°C , for example, the refrigerant vapour pressure p_r is only 813 Pa . At this pressure the sorption material has to adsorb the load concentration being determined by the sorption isotherm for silica gel. For this, the relative humidity in the adsorbent is calculated, which results from the ratio of evaporator pressure and saturation vapour pressure at adsorbent temperatures.

$$\varphi_{\text{ads}} = \frac{p_e}{p_s(T_{\text{ads}})} \quad (5.167)$$

From the relative humidity, the load concentration is then determined using the procedure from Section 4.1, i.e. by conversion to the known 40°C sorption isotherm.

Example 5.18

Calculate the load concentration in the adsorbent at a heat rejection supply temperature of 29°C , a temperature difference at the heat exchanger of 5 K and thus an adsorbent temperature of 34°C at an evaporator temperature of 4°C .

The saturation vapour pressure in the evaporator at 4°C is 813 Pa. Related to the saturation vapour pressure in the adsorbent at 34°C of 5324 Pa, a relative humidity of 15.3% results. The load concentration is then 0.09 kg kg⁻¹, which is very low.

If due to low evaporator pressure or high coolant temperatures, only low load concentrations are achieved, a large mass of sorption material is necessary to take up the circulating refrigerant flow rate. The adsorption machines are then correspondingly large and heavy. The necessary sorption mass can, however, only be calculated when the load concentration after desorption is known. The difference in load concentrations is called the degassing width and indicates the water vapour quantity per kg of sorption material that can be effectively adsorbed. For a high degassing width, high evaporator temperatures are favourable (and thus high vapour pressures) as well as low adsorbent temperatures (and thus high relative humidities in the adsorbent). Figure 5.43 shows the connection between load concentration and evaporator temperature at different adsorbent temperatures.

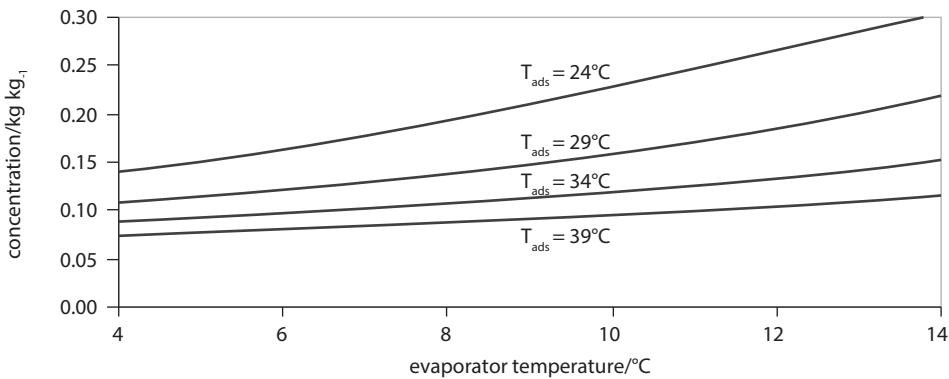


Figure 5.54 Load concentration as a function of the evaporator temperature with the adsorbent temperature as a parameter.

In the real process, a differential pressure between the evaporator and adsorbent must exist, to compensate for the pressure losses between the two chambers at the necessary flow rates. The load concentration at the end of the adsorption process is therefore lower than the value corresponding to that of the vapour pressure in the evaporator.

The heat of adsorption is removed by the cooling water $\dot{m}_{cool,2} c_p (T_{r,cool} - T_{s,cool})$ and by the heating of the cold refrigerant vapour from the evaporator to the silica gel temperature $\dot{m}_r c_p (T_{ads} - T_e)$. Additionally, the heat from the preceding desorption step, which remains after switching and pre-cooling, must be removed. The silica gel temperature equals the temperature of the cooling water at the end of the adsorption process (plus the temperature difference of the heat exchanger ΔT_{hx}). The effective thermal capacity c_p^{eff} covers both the thermal capacity of the sorbent material c_p^{sor} (of about 1.0 kJ kg⁻¹ K⁻¹) and also that of the heat exchanger c_p^{hx} related to the sorbent mass m_{sor} (e.g. 0.385 kJ kg⁻¹ K⁻¹ for copper).

$$\dot{m}_{cool,2} c_p (T_{r,cool} - T_{s,cool}) + \dot{m}_r c_p (T_{ads}(t) - T_e) = \dot{m}_r h_{ads} + c_p^{eff} m_{sor} \frac{T_{ads}(t) - T_{a,cool} + \Delta T_{hx}}{\Delta t} \quad (5.168)$$

with

$$c_p^{eff} = c_p^{sor} + \frac{m_{hx}}{m_{sor}} c_p^{hx}$$

At a typical mass ratio of heat exchanger to sorbents of approximately 2.0, an effective thermal capacity of

$$c_p^{eff} = 1 \frac{kJ}{kg} + 2 \times 0.385 \frac{kJ}{kg} = 1.77 \frac{kJ}{kg}$$

results.

Example 5.19

Calculate the necessary cooling water mass flow of the adsorbent for the above chiller under steady-state conditions, i.e. after achieving the final adsorption temperature T_{ads} (= cooling water inlet temperature +5 K, here $29^\circ\text{C} + 5^\circ\text{C} = 34^\circ\text{C}$). Under steady-state conditions, the last term of Equation (5.168), the temperature rise of the adsorbent material with time, is zero, so the effective thermal capacity needs not to be known.

The heat of adsorption is 2650 kJ kg^{-1} at the low load concentration of around 0.1 kg kg^{-1} . The outlet temperature from the evaporator is equated to the cold water return temperature of 12°C . The cooling water mass flow is

$$\dot{m}_{cool,2} = \frac{0.086 \frac{kg}{s} \left(2650 \frac{kJ}{kg} - 4.19 \frac{kJ}{kgK} (34 - 12) K \right)}{4.19 \frac{kJ}{kgK} (33 - 29) K} = 13.2 \frac{kg}{s} = 47.4 \frac{m^3}{h}$$

Heating phase

After adsorption has ended, both valve flaps of the adsorption chamber are closed and heated from the final temperature of adsorption T_{ads} until the required pressure of the condenser is achieved, at an initially constant load concentration (change in status 3 to 4 in Figure 5.50). The load concentration is given by the adsorption process and thus by the evaporator and coolant temperature.

$$\dot{m}_{heat} c_p (T_{s,heat} - T_{r,heat}) = (m_{sor} c_p^{eff} + m_{H_2O} c_p) (T_H(t) - T_{ads}) / \Delta t \quad (5.169)$$

The final temperature T_H of the heating phase at constant loading results from the condenser temperature. The higher this is, the higher is the pressure necessary to liquefy the refrigerant, and the higher the temperatures which must be produced during the heating phase and the subsequent desorption phase.

Example 5.20

Calculate the final temperature of the heating phase T_H at a condenser temperature of 34°C (cooling water $29^\circ\text{C} + \Delta T_{hx}$) and an evaporator temperature of 4°C .

The condenser temperature determines the pressure level. To be able to liquefy water at 34°C, the vapour pressure must be 5324 Pa. The evaporator temperature determines the load concentration of the adsorbent at a given adsorbent temperature (here 34°C), which, taking Example 5.18, is 0.09 kg kg⁻¹. For this isostere the temperature can now be determined with the Clausius parameters from Table 5.10, for which at the calculated load concentration the vapour pressure over the sorbents corresponds to the condenser pressure.

$$\dot{m}_{cool,2} = \frac{0.086 \frac{\text{kg}}{\text{s}} \left(2650 \frac{\text{kJ}}{\text{kg}} - 4.19 \frac{\text{kJ}}{\text{kgK}} (34 - 12) \text{K} \right)}{4.19 \frac{\text{kJ}}{\text{kgK}} (33 - 29) \text{K}} = 13.2 \frac{\text{kg}}{\text{s}} = 47.4 \frac{\text{m}^3}{\text{h}}$$

At 65°C the valve between the silica gel chamber and the condenser is opened, and liquefaction of the refrigerant begins.

The desorption process

At a constant condenser pressure, the valve flap of the desorber is opened to the condenser, and water vapour is expelled by a rise in temperature from the heating temperature T_H to the final desorption temperature T_{des} (4→5). The isostere of the minimum load concentration results from the intersection of the condenser pressure and the given desorption final temperature. In order that, at a decreasing load concentration, the necessary vapour pressure is still produced, the temperature must be increased to this final desorption temperature.

The aim of the desorption process is to reduce as far as possible the load concentration of the sorption material, which means large degassing widths. To achieve low load concentrations, the relative humidity in the desorber must be as low as possible. This results from the ratio of the vapour pressure in the desorber, which is equal to the condenser pressure, and the saturation vapour pressure at the desorption temperature T_{des} :

$$\varphi_{des} = \frac{p_c}{p_s(T_{des})} \quad (5.170)$$

At low cooling water temperatures the condenser temperatures and the vapour pressure p_c are low, and a low relative humidity in the desorber results, enabling a deep unloading of the sorption material. More effective, however, is a high desorption temperature, since the saturation vapour pressure rises exponentially with the temperature, and a rise in temperature in the desorber leads to a higher reduction of the relative humidity than a falling coolant temperature.

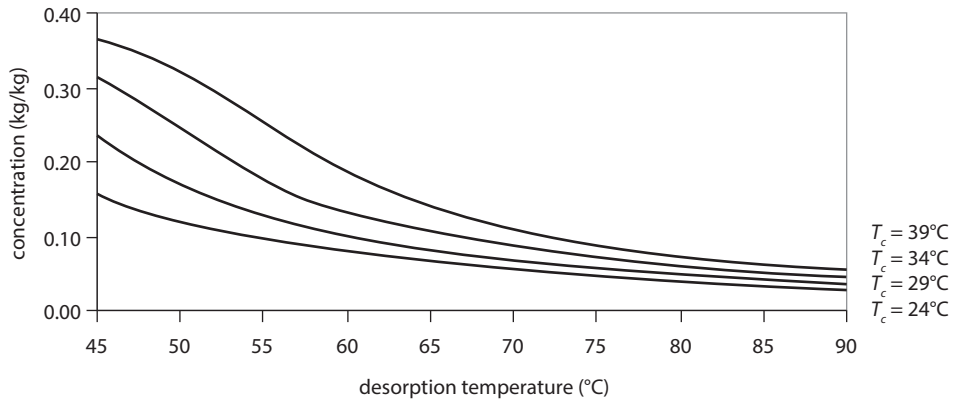


Figure 5.55 Load concentration at the end of desorption as a function of the desorption temperature at different condenser temperatures.

If a desorption temperature T_{des} is now established, the degassing width of the adsorption/desorption process can be calculated, and the mass and effective thermal capacity of the sorption material can be determined. From Figure 5.55 it is clear that even at high final desorption temperatures and low condenser temperatures, a load concentration much below 0.1 kg kg^{-1} can be attained only with difficulty. For a condenser temperature of 34°C , a value of 0.06 kg kg^{-1} at $T_{des} = 80^\circ\text{C}$ is the minimum load concentration. In the real process, the vapour pressure in the desorber must be somewhat higher than in the condenser, i.e. the desorber temperatures are somewhat higher than in the ideal process. If the adsorption machine is to be operated at very low evaporator temperatures (for example, 4°C), the degassing width ΔC as the concentration difference between adsorption C_{ads} and desorption C_{des} is correspondingly small.

Example 5.21

Calculation of the degassing width at $T_{des} = 80^\circ\text{C}$, $T_{ads} = 34^\circ\text{C}$, $T_c = 34^\circ\text{C}$ and $T_e = 4^\circ\text{C}$.

The load concentration at the end of adsorption and a vapour pressure of 813 Pa at 4°C is 0.09 kg kg^{-1} (see Example 5.18). At the end of desorption ($T_{des} = 80^\circ\text{C}$) the load concentration is 0.06 kg kg^{-1} . The degassing width $\Delta C = 0.09 \text{ kg kg}^{-1} - 0.06 \text{ kg kg}^{-1} = 0.03 \text{ kg kg}^{-1}$ is thus very low.

The necessary sorption mass m_{sor} can be calculated from the circulating refrigerant mass flow $\dot{m}_r$, the degassing width ΔC and the cycle time t_c .

$$m_{sor} = \frac{\dot{m}_r t_c}{C_{ads} - C_{des}} \quad (5.171)$$

Example 5.22

Calculation of the sorbent mass for the above chiller at a cycle duration t_c of 200 seconds. During 200 seconds, a total of 17.2 kg of water is adsorbed in the 200 kW chiller with an evaporation mass flow of 0.086 kg s⁻¹. The sorbent mass at the degassing width of 0.03 kg kg⁻¹ is 573 kg:

$$m_{sor} = \frac{0.086 \text{ kg}_{H_2O} / s \times 200 s}{0.09 \text{ kg}_{H_2O} / \text{kg}_{sor} - 0.06 \text{ kg}_{H_2O} / \text{kg}_{sor}} = 573 \text{ kg}_{sor}$$

With the known masses and thermal capacities of the sorbent material and adsorbed water, the necessary amount of heat for the desorption process can be calculated. The heating output is used to provide the desorption heat $\dot{m}_r h_{ads}$ and to heat the masses from the heating temperature T_H to the final desorption temperature T_{des} :

$$\dot{m}_{heat} c_p (T_{s,heat} - T_{r,heat}) = \dot{m}_r h_{ads} + (m_{sor} c_p^{eff} + m_{H_2O} c_p) (T_{des}(t) - T_H) / \Delta t \quad (5.172)$$

Example 5.23

Calculation of the heating power for the desorption process at an effective thermal capacity of 1.77 kJ kg⁻¹ K⁻¹ and a heating duration of 200 seconds.

$$\dot{Q}_{heat} = \underbrace{0.086 \frac{\text{kg}}{s} 2622 \frac{\text{kJ}}{\text{kg}}}_{225.5 \text{ kW}} + \underbrace{\left(573 \text{ kg} \times 1.77 \frac{\text{kJ}}{\text{kgK}} + 17.2 \text{ kg} \times 4.19 \frac{\text{kJ}}{\text{kgK}} \right) (80 - 65) \text{ K} \times \frac{1}{200 \text{ s}}}_{81.5 \text{ kW}} = 307 \text{ kW}$$

At a temperature spread of 10 K, the result is a mass flow of

$$\dot{m}_{heat} = 7.3 \frac{\text{kg}}{s} = 26.4 \frac{\text{m}^3}{h}$$

Cooling phase

After termination of desorption all valve flaps are closed, and by switching the heating water circuit to cooling water operation the silica gel chamber is cooled at constant low loading until at temperature T_C the evaporator pressure is achieved (change in status 5 to 6 in Figure 5.50). The energy balance corresponds to the heating phase.

$$\dot{m}_{cool} c_p (T_{r,cool} - T_{s,cool}) = (m_{sor} c_p^{eff} + m_{H_2O} c_p) (T_{des} - T_C(t)) / \Delta t \quad (5.173)$$

Energy efficiency ratio

The performance figure of the closed adsorption chiller can be calculated from the energy balances of the process steps discussed. The EER is defined as the ratio of chilling capacity $\dot{Q}_{ch}$ produced in the evaporator (or energy Q_{ch} produced in the cycle) to the necessary heating power $\dot{Q}_{heat}$ for the heating process and desorption (changes in status 3→4 and 4→5).

Since during the change-over process from desorption to adsorption (5→6) heat can be recovered with heat recovery efficiency η), the heating output can be reduced by this amount.

$$EER = \frac{\dot{Q}_{ch}}{\dot{Q}_{heat}} = \frac{Q_{ch}}{Q_{heat}} = \frac{Q_{ch}}{\underbrace{Q_{3 \rightarrow 4}}_{\text{heating}} + \underbrace{Q_{4 \rightarrow 5}}_{\text{desorption}} - \underbrace{\eta Q_{5 \rightarrow 6}}_{\text{cooling}}} \quad (5.174)$$

For the chiller considered so far, for 1 kilogram of evaporated refrigerant, the coefficient of performance is calculated as follows.

The usable cold results from the evaporation enthalpy of the refrigerant water:

$$Q_{ch} = h_e(T_e = 4^\circ\text{C}) = 2492 \text{ kJ}$$

During the heating phase, the adsorbent temperature T_{ads} is raised to the heating temperature T_H , which delivers refrigerant vapour at the condenser pressure. The mass of the sorbent material per kilogram of evaporated water results from the reciprocal value of the degassing width, and is 33.3 kg_{sor}.

$$\begin{aligned} Q_{3 \rightarrow 4} &= (m_{\text{sor}} c_p^{\text{eff}} + m_{\text{H}_2\text{O}} c_p)(T_H - T_{ads}) \\ &= \left(\underbrace{\frac{1 \text{ kg}_{\text{H}_2\text{O}}}{0.09 \frac{\text{kg}_{\text{H}_2\text{O}}}{\text{kg}_{\text{sor}}} - 0.06 \frac{\text{kg}_{\text{H}_2\text{O}}}{\text{kg}_{\text{sor}}}}}_{33.3 \text{ kg}} \times 1.77 \frac{\text{kJ}}{\text{kgK}} + 1 \text{ kg}_{\text{H}_2\text{O}} \times 4.19 \frac{\text{kJ}}{\text{kgK}} \right) (65 - 34) \text{ K} = 1957 \text{ kJ} \end{aligned}$$

For further heating to the final desorption temperature, an energy quantity of

$$Q_{4 \rightarrow 5} = h_{ads} + (m_{\text{sor}} c_p^{\text{eff}} + m_{\text{H}_2\text{O}} c_p)(T_{des} - T_H) = 2622 \text{ kJ} + \left(61.3 \frac{\text{kJ}}{\text{K}} \right) (80 - 65) \text{ K} = 3541 \text{ kJ}$$

is needed.

The heat given off in the cooling phase corresponds to the amount of heat during the heating phase: $Q_{5 \rightarrow 6} = Q_{3 \rightarrow 4}$.

This amount of heat can be recovered with a heat recovery efficiency η , around 70% in this case. The result is thus a performance figure for the example calculated of

$$EER = \frac{2492 \frac{\text{kJ}}{\text{kg}}}{1957 \frac{\text{kJ}}{\text{kg}} + 3541 \frac{\text{kJ}}{\text{kg}} - 0.7 \times 1957 \frac{\text{kJ}}{\text{kg}}} = 0.6$$

Better performance figures are achieved if the operating conditions are less extreme. At higher permissible cold water temperatures the adsorption machine can be run at higher evaporator temperatures. Thus the load concentration in the adsorbent rises, and at higher degassing widths the desorption temperature can be lowered. Alternatively the sorbent mass can be increased, and the circulating refrigerant mass flow adsorbed and desorbed at small degassing widths. The machines are then however very large and heavy, and for short cycle times the connected heating power must be very high.

5.16 Heat rejection and auxiliary electricity consumption

The performance of solar driven cooling systems strongly depends on the chosen heat rejection system and its control strategy. High electricity consumption caused by suboptimal control in combination with low solar fractions through insufficient system design are critical for the environmental and economic performance of sorption cooling systems, especially if they are compared to highly efficient electrical-driven compression chillers. To evaluate the overall efficiency of installed solar cooling systems within the IEA TASK 38 (International Energy Agency Solar Heating and Cooling Programme), several solar cooling systems were monitored in detail. The results clearly demonstrate that the electrical energy efficiency ratios (EER_{el}) are still low with values of up to 6 in the best case and values of below 3 in the worst case. The best primary energy ratio values obtained are 1.7 and the worst below 1.0. For comparison, systems with good compression chillers with wet cooling towers for heat rejection reach average electrical system EER of 3.0 and primary energy ratios which are slightly above 1.0.

Heat rejection systems include open or closed wet cooling towers, dry heat rejection systems, which can include water sprayers for hybrid operation, low depth geothermal heat sinks, latent heat storage, night radiative cooling or sea, lake or groundwater cooling, including buried irrigation tanks.

An analysis of 16 hotels in Hongkong showed that water cooled air-conditioning systems had about 20% less electricity consumption than air-cooled compression chillers. Similar results were obtained by replacing air cooling with direct seawater cooling towers. A detailed energy audit for one 29-storey office/commercial building in Hongkong with an electricity use of $174 \text{ kWh m}^{-2} \text{ a}^{-1}$ showed that about one-third of this electricity demand was for the chiller itself ($51 \text{ kWh m}^{-2} \text{ a}^{-1}$), 7% for the heat rejection ($12.4 \text{ kWh m}^{-2} \text{ a}^{-1}$), another one-third for the air handling units to distribute the cold ($49.2 \text{ kWh m}^{-2} \text{ a}^{-1}$) and the rest for lighting, escalators, etc. Due to the low EER of single effect absorption machines, the amount of heat rejection related to the cooling power is significantly larger than for compression chillers:

$$\frac{Q_{\text{heat rejection}}}{Q_{\text{cooling power}}} = \frac{1 + EER}{EER} \quad (5.175)$$

$$\frac{Q_{\text{heat rejection, absorption}}}{Q_{\text{heat rejection, compression}}} = \frac{EER_{\text{compression}}}{EER_{\text{absorption}}} \frac{1 + EER_{\text{absorption}}}{1 + EER_{\text{compression}}} \quad (5.176)$$

Only if the parasitic electrical consumption is very low, sorption cooling systems can have a primary energy advantage. For a highly efficient compression system with a EER of 5.0, the solar fraction of an absorption chiller with an EER of 0.6 has to be above 90% and the auxiliary energy consumption below 10% of the cooling power to save primary energy.

Cooling tower consumption values according to the German standard DIN 18599-7 are $0.018 \text{ kW}_{\text{electric}}$ per kW heat rejection for an open cooling tower with axial fan, $0.033 \text{ kW}_{\text{electric}} \text{ kW}_{\text{heat}}^{-1}$ rejection for a closed cooling tower with axial fan and $0.045 \text{ kW}_{\text{electric}} \text{ kW}_{\text{heat}}^{-1}$ rejection for a dry chiller. Monitoring results with a dry heat rejection system at the ILK Dresden with a fixed temperature difference between air entry temperature and cooling water temperature gave lower electricity consumption values of $0.008 - 0.01 \text{ kW}_{\text{electric}} \text{ kW}_{\text{heat}}^{-1}$ rejection, decreasing with increasing temperature difference. Related to the cooling power, this corresponds to $0.018 - 0.026 \text{ kW}_{\text{electric}} \text{ kW}_{\text{cooling}}^{-1}$. The heat rejection fans and circulation pump were responsible for more

than 50% of the total auxiliary electricity consumption. Experimental investigations of large wet cooling towers in the megawatt power range resulted in approximately $0.02 \text{ kW}_{\text{electric}} \text{ kW}_{\text{cooling}}^{-1}$ for forced draught towers and $0.007 \text{ kW}_{\text{electric}} \text{ kW}_{\text{cooling}}^{-1}$ for induced draught towers. Table 5.11 summarises the electrical power ranges quoted in the literature for cooling tower operation.

To provide water for the heat rejection systems also requires energy, which should be included in the total balance. Open cooling towers require between 4.3 and 7 litre water per kWh of cooling energy in single effect absorption machines (EER=0.7), depending on the water quality. Although water-cooled air conditioning is generally more effective than air cooling, some cities such as Hongkong even prohibit the use of fresh water for air conditioning, although water losses of cooling towers are only 1 - 3% of the heat rejection circuit flow rates.

The need to minimise auxiliary electricity consumption in sorption cooling systems requires optimised control strategies to reduce pumping power and cooling tower fan speed under part load conditions. The control of the cooling water supply temperature assures a stable cold water temperature at variable generator inlet temperatures. For the overall system efficiency the increase of cooling water temperature only makes sense, if the electricity consumption of the cooling tower is significantly reduced through a fan speed control and no backup system is used for heat supply. A control of the cooling water temperature only through a three-way valve is not recommendable.

Heat rejection system	Electrical power per kW thermal power	Reference
Open cooling tower with axial fan	$0.018 \text{ kW}_{\text{electric}}/\text{kW}_{\text{heat rejection}}$	DIN 18599-7
Closed cooling tower with axial fan	$0.033 \text{ kW}_{\text{electric}}/\text{kW}_{\text{heat rejection}}$	DIN 18599-7
Dry chiller	$0.045 \text{ kW}_{\text{electric}}/\text{kW}_{\text{heat rejection}}$	DIN 18599-7
Wet cooling tower 35 kW	$0.01 - 0.08 \text{ kW}_{\text{electric}}/\text{kW}_{\text{heat rejection}}$	Measurements ISE Freiburg 2009
Dry chiller 25 kW	$0.03 - 0.08 \text{ kW}_{\text{electric}}/\text{kW}_{\text{heat rejection}}$	Measurements ISE Freiburg 2012
Dry chiller	$0.008 \text{ to } 0.01 \text{ kW}_{\text{electric}}/\text{kW}_{\text{heat rejection}}$	Measurements ILK Dresden
Dry chiller	$0.018 \text{ to } 0.026 \text{ kW}_{\text{electric}}/\text{kW}_{\text{cooling}}$	Measurements ILK Dresden
Forced draught wet cooling tower Megawatt range	$0.02 \text{ kW}_{\text{electric}}/\text{kW}_{\text{cooling}}$	Measurements Saidi et al., 2011
Induced draught wet cooling tower Megawatt range	$0.007 \text{ kW}_{\text{electric}}/\text{kW}_{\text{cooling}}$	Measurements Saidi et al., 2011

Table 5.11 Heat rejection systems and electrical power requirement per kW of heat rejection or cooling power.

Modeling of Open Wet Cooling Towers

In wet cooling towers mostly latent heat transfer is used to lower the temperature of the cooling fluid. Open cooling towers generally work with water as cooling fluid which is distributed over a fill packing and forced air-cooled by a ventilator. Partial evaporation of approximately 2 - 3% of the water as well as convective heat transfer between water and ambient air result in the cooling effect. Due to the evaporation of water outlet, temperatures of the cooling water below the ambient

air temperature can be reached, which are only limited by the wet bulb temperature of the ambient air. The following analysis and modeling is taken from the PhD thesis of Dirk Pietruschka (2011).

As an example, a 35 kW open wet cooling tower is shown with its technical data in Table 5.12. The warm cooling water is distributed via spray nozzles at the top of the cooling tower on a fill packing and runs down the fill packing while it is cooled down through evaporation and convective heat transfer. After leaving the fill packing the cooled water is collected in the cold water sump from where it leaves the cooling tower again. Air is drawn in counter flow to the water from bottom to the top of the cooling tower by a motor-driven fan. To reduce water spray losses a demister at the top of the cooling tower separates the water droplets from the air stream. A flotation valve controls the water level in the cold water sump and equalises the water losses due to evaporation, spray losses and water exchange by adding fresh water from the grid.

Total heat rejection power	35 kW
Water mass flow rate	5 m ³ /h
Water supply temperature	36°C
Water return temperature	30°C
Wet bulb temperature of the ambient air	24°C
Electrical power of the ventilator motor	0.33 kW
Required power at the ventilator axis	0.17 kW
Engine and ventilator rotation	1430 min ⁻¹

Table 5.12 Technical data of a cooling tower for design conditions.

In the literature several more or less detailed models for the prediction of the performance of open wet cooling towers can be found. The first practical use of basic differential equations was first presented by Merkel in 1925, in which he combined the equations governing heat and mass transfer between water droplets and air in the tower. Webb presented a unified theoretical treatment for the thermal analysis of cooling towers, evaporative condensers and evaporative fluid chillers in 1984. A very detailed model which separately analyses the cooling effects in the fill packing, in the spray and in the rain zone of a wet cooling tower is presented by Qureshi and Zubair in 2006. This model also considers fouling effects in the packing material.

However, for most of these models, very detailed construction information especially about the implemented fill packing of the cooling tower are required for an accurate prediction of the heat and mass transfer conditions within the packing material. These details are usually not provided by the manufacturers. Therefore, a model has been developed by Pietruschka (2011), which only needs the typical information given on the data sheets of the cooling towers for design conditions like the heat rejection power, the water mass flow rate, the water supply and return temperature, the wet bulb temperature of the ambient air and the nominal electricity consumption of the ventilator motor. The model described in the following paragraph is mainly based on a method originally used for commissioning measurements on cooling towers described in the German standard DIN 1947.

The process of water cooling in an open wet cooling tower is shown for an ideal cooling tower in Figure 5.56. The water is cooled down from $t_{w,in}$ to $t_{w,out}$ while the enthalpy of the air

increases due to the increase of water content and temperature from $h_{air,in}$ to $h_{air,out}$. The overall energy balance can be described as follows.

$$\dot{m}_w c_w (t_{w,in} - t_{w,out}) = \dot{m}_a (h_{air,out} - h_{air,in}) \quad (5.177)$$

with:

$h_{air,in}$ enthalpy of inlet air/kJ kg⁻¹

$h_{air,out}$ enthalpy of outlet air/kJ kg⁻¹

Due to limited heat and mass transfer coefficients and area in the fill packing, the real behaviour of wet cooling towers is not ideal.

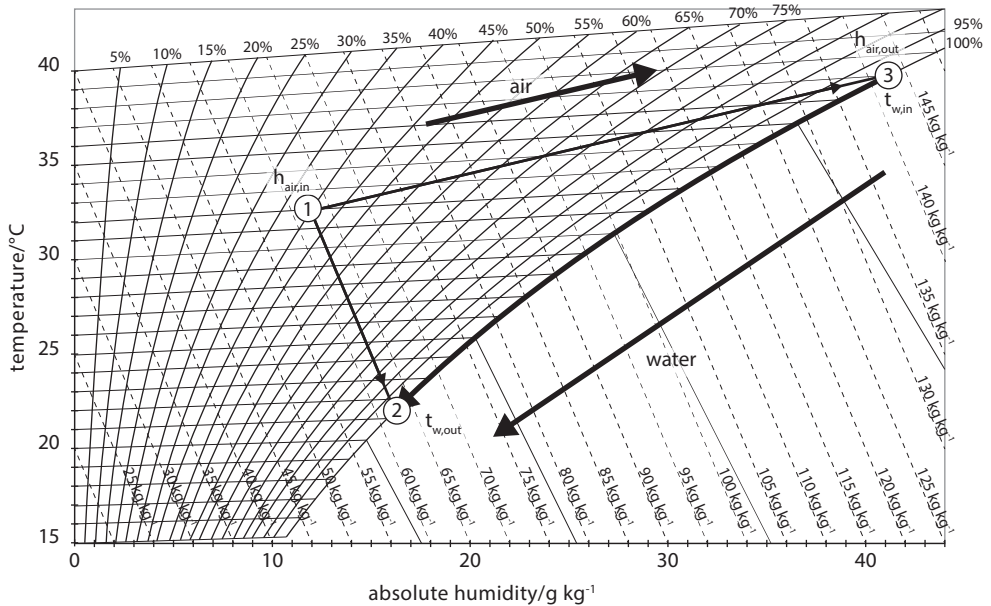


Figure 5.56 Water cooling in an ideal open wet cooling tower.

The heat rejection quality of a real cooling tower compared with an ideal cooling tower is described by the heat rejection efficiency η which is the ratio between the difference of the water inlet and outlet temperature and the difference between the water inlet temperature and the wet bulb temperature of the ambient air for a wet cooling tower and the dry bulb temperature for a dry cooling tower. In an ideal cooling tower, the water outlet temperature is equal to the wet bulb temperature of the ambient air, which results in a heat rejection efficiency of 1.0 as maximum

value. The difference between the water outlet temperature and the wet bulb temperature is called the cooling boundary difference.

$$\eta = \frac{t_{w,in} - t_{w,out}}{t_{w,in} - t_{wb}} \quad (5.178)$$

with:

$t_{w,in}$ water inlet temperature/°C
 $t_{w,out}$ water outlet temperature/°C
 t_{wb} wet bulb temperature/°C

In case of an ideal cooling tower, a minimum air flow rate of $\dot{m}_{a,min}$ is required to cool down a certain water mass flow of $\dot{m}_w$ from the inlet temperature $t_{w,in}$ to the wet bulb temperature t_{wb} . The ratio between this minimum air flow rate and the water mass flow is called the minimum air to water mass flow ratio l_{min} . Values of the minimum air to water mass flow ratio are shown in Figure 5.57 for the typical range of cooling water and wet bulb temperatures.

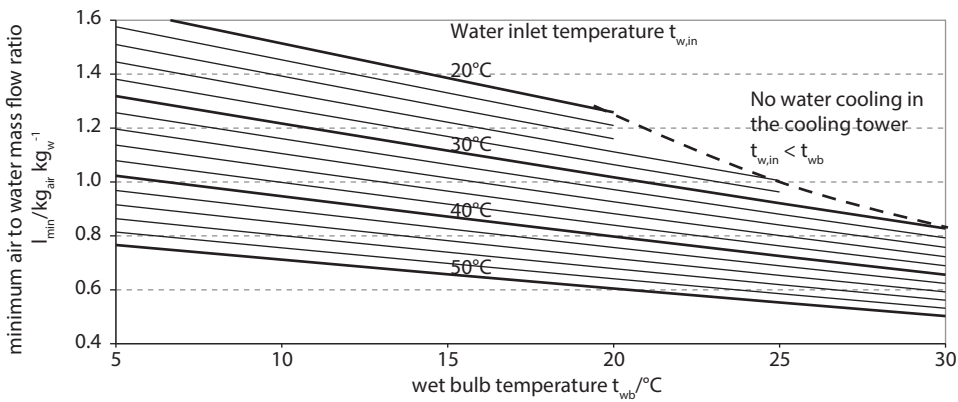


Figure 5.57 Minimum relative air amount l_{min} of cooling towers.

The ratio between the real air flow rate and the water mass flow l_0 is called the effective air to water mass flow ratio. A typical and important parameter of wet cooling towers is the dimensionless air ratio λ , which is defined as the ratio between the effective and the minimum air to water ratio.

$$\begin{aligned} \lambda_a &= \frac{l_0}{l_{min}} \\ l_{min} &= \frac{\dot{m}_{a,min}}{\dot{m}_w} \quad \left[\frac{kg_{air}}{kg_w} \right] \\ l_0 &= \frac{\dot{m}_a}{\dot{m}_w} \quad \left[\frac{kg_{air}}{kg_w} \right] \end{aligned} \quad (5.179)$$

with:

l_0	effective air to water mass flow ratio
l_{min}	minimum air to water mass flow ratio/kg _{air} kg _{water} ⁻¹
$\dot{m}_{a,min}$	minimum air mass flow rate/kg s ⁻¹
$\dot{m}_a$	actual air mass flow rate/kg s ⁻¹
$\dot{m}_w$	water mass flow rate/kg s ⁻¹
λ_a	air ratio

In the developed simulation model, the air ratio λ_a is calculated for each operation point of the cooling tower and each ambient condition separately. For the determination of the minimum air to water mass flow ratio l_{min} , first the wet bulb temperature is calculated from the following equation:

$$\begin{aligned}
 h_1 &= h_{wb} \\
 c_{pl}t_1 + x_1(r_{0^\circ C} + c_{pd}t_1) &= c_{pl}t_{wb} + x_{wb}(r_{0^\circ C} + c_{pd}t_{wb}) \\
 t_{wb} &= \frac{c_{pl}t_1 + x_1(r_{0^\circ C} + c_{pd}t_1) - r_{0^\circ C}x_1}{c_{pl} + c_{pd}x_{wb}}
 \end{aligned} \quad (5.180)$$

This equation has to be solved iteratively for the water content at wet bulb temperature x_{wb} using the additional condition that no change in the enthalpy of the air occurs. For the ideal cooling tower the outlet air temperature is equal to the water inlet temperature with a relative humidity of 100 % and the outlet water temperature is equal to the wet bulb temperature of the ambient air (see Figure 5.56). This allows the calculation of the enthalpy of the outlet air $h_{air,out}$. Using the minimum air to water mass flow ratio l_{min} of the cooling tower can be calculated from the enthalpy difference between inlet and outlet air as follows:

$$l_{min} = \frac{\dot{m}_a}{\dot{m}_w} = \frac{h_{air,out} - h_{air,in}}{c_w(t_{w,in} - t_{w,out})} \quad (5.181)$$

There is a clear correlation between the heat rejection efficiency η and the air ratio, which can be shown as a curve which is known as the cooling tower characteristic. The characteristic behaviour can be described using the equation:

$$\eta = C_k(1 - e^{-\lambda_a}) \quad (5.182)$$

C_k is the cooling tower constant, which can be determined for each cooling tower type by performance measurement. If the air flow rate of the cooling tower is given, the data on the data sheet of the cooling tower for the performance under design conditions can be used to calculate the value of C_k .

$$C_k = \frac{\eta}{1 - e^{-\lambda_a}} \quad (5.183)$$

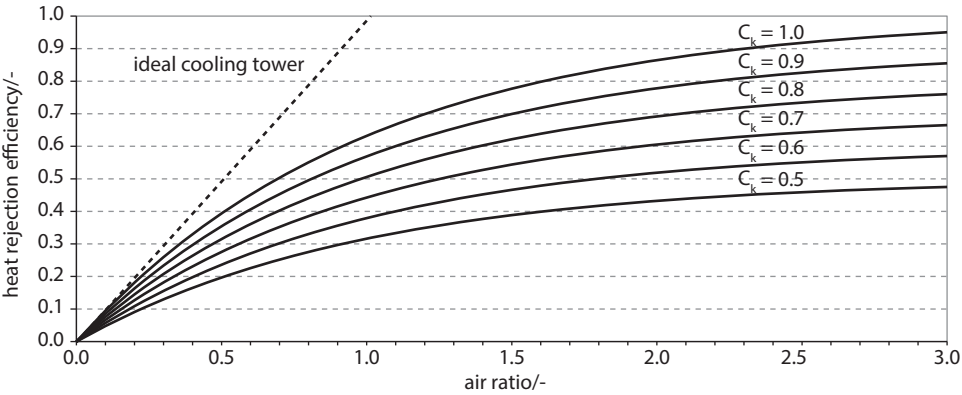


Figure 5.58 Cooling tower characteristics.

However, often only the fan speed and not the air flow rate is given on the data sheets. In this case, typical values of the cooling tower constant C_k given in Table 5.13 can be used, which have been found for open cooling towers with fill packing made of plastic and air to water counter-flow arrangement (Recknagel, Sprenger and Schramek, 1995):

Height of fill packing /m	0.3	0.5	0.7	0.9
Cooling tower constant C_k	0.65	0.81	0.93	1.0

Table 5.13 Typical values of the cooling tower constant C_k .

These values are also shown in Figure 5.59 together with a polynomial function generated from the values given for the different fill packing heights. This function can be used for the determination of the cooling tower constant C_k if no air flow rate is available.

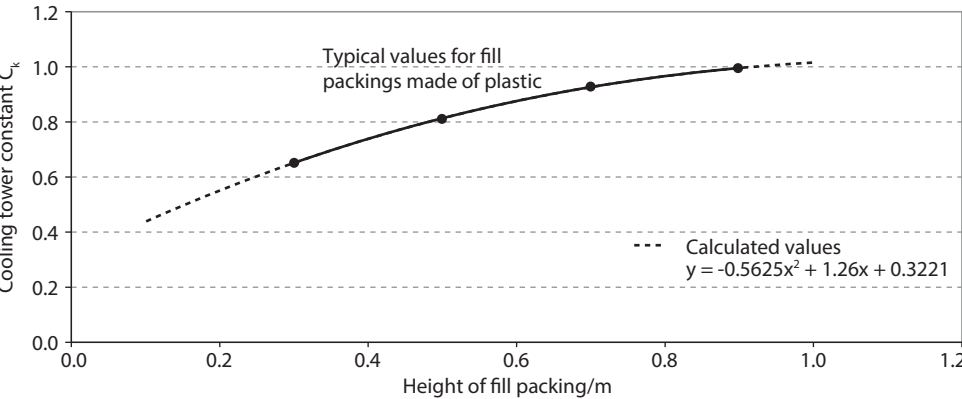


Figure 5.59 Typical values of the cooling tower constant C_k .

If the cooling tower constant C_k has been calculated, the air ratio $\lambda_{a,d}$ and the air flow rate $\dot{m}_{air,d}$ of the cooling tower can be determined for design conditions using the equations:

$$\lambda_d = -\ln\left(1 - \frac{\eta_d}{C_k}\right); \quad \dot{m}_{air,d} = \lambda_{a,d} \cdot \dot{m}_{w,d} \quad [kg/s] \quad (5.184)$$

The required value of the heat rejection efficiency η_d can be calculated for the design conditions from the water inlet and outlet and air wet bulb temperature given on the data sheet of the cooling tower:

$$\eta_d = \frac{t_{w,in,d} - t_{w,out,d}}{t_{w,in,d} - t_{wb,d}} \quad (5.185)$$

with:

- $t_{w,in,d}$ water inlet temperature for design conditions/°C
- $t_{w,out,d}$ water outlet temperature for design conditions/°C
- $t_{wb,d}$ wet bulb temperature for design conditions/°C

However, for a Northern Europe climate, wet bulb temperatures of 18 - 21°C, which result from the design conditions of cooling towers, are only given on some days of the year and are much lower on many days of the cooling period. This means that the cooling tower can be operated most of the time in part load conditions by controlling the fan speed in a range of 20 - 100 % according to the water outlet temperature. The reduction of the fan speed is directly correlated to a lower air flow rate. According to the proportional law for axial and radial fans, the following general correlation of the mass flow to the fan speed is given:

$$\dot{m}_{air,new} = \dot{m}_{air,d} \frac{n_{new}}{n_d} \quad (5.186)$$

with:

- $\dot{m}_{air,new}$ air mass flow rate at n_{new} /kg s⁻¹
- $\dot{m}_{air,d}$ air mass flow rate for design conditions/kg s⁻¹
- n_{new} fan speed of the cooling tower fan/R min⁻¹
- n_d fan speed of the cooling tower fan for design conditions/R min⁻¹

This proportional law, which is true for a wide range of Reynolds numbers, can be used with sufficient accuracy for the calculation of flow rates for variable fan speed as long as the differential pressure/volume flow chart of the cooling tower shows a quadratic characteristic.

Electricity consumption of cooling towers

If the cooling water pump is not considered, the electricity consumption of cooling towers only depends on the electricity consumption of the ventilator. For the electricity consumption of electrically driven ventilators the following general equation can be found.

$$P_{el} = \frac{\dot{V} \Delta p_t}{\eta_V \eta_M \eta_A} \quad (5.187)$$

with:

$\dot{V}$	air volume flow rate/m ³ s ⁻¹
Δp_t	total pressure increase/Pa
η_V	ventilator efficiency
η_M	motor efficiency
η_A	axis efficiency

In general the electricity consumption for design conditions is given on the data sheet of the cooling tower. If no values are given the following values for the electric power of the ventilator motors of cooling towers for design conditions can be approximately used for the calculation of the electricity consumption.

- Axial ventilators with compact design, high volume flows and small to medium pressure increases
Electric power of the ventilator motor: 6 - 10 W per kW heat rejection power
- Radial ventilators with high pressure and lower volume flows, less compact
Electric power of the ventilator motor: 10 - 20 W per kW heat rejection power

The reduction of the fan speed is directly correlated to a significantly lower electricity consumption. According to the proportional law for axial and radial ventilators, the following general correlation of the electricity power consumption to the third power of the fan speed is given:

$$P_{el, new} = P_{el, d} \cdot \left(\frac{n_{new}}{n_d} \right)^3 \quad (5.188)$$

with:

$P_{el, new}$	electric power demand at n_{new} /W
$P_{el, d}$	electric power demand under design conditions at n_d /W
n_{new}	fan speed of the cooling tower fan/R min ⁻¹
n_d	fan speed of the cooling tower fan under design conditions/R min ⁻¹

Again, this proportional law can be used with sufficient accuracy as long as the differential pressure/volume flow chart of the cooling tower shows a quadratic characteristic.

Water consumption of cooling towers

If no detailed information is available, for the water consumption, the following statistically developed hourly values can be approximately used per kW heat rejection power according to the German standard DIN 4710:

Water consumption through evaporation

2 kg h⁻¹

Water consumption through spray losses	1 kg h ⁻¹
Water consumption through water exchange required to avoid an increase of salt concentration	3 kg h ⁻¹

The water consumption through evaporation can be calculated from the following equation:

$$\dot{W}_{ev} = \dot{m}_{air, d} (x_{air, in} - x_{air, out}) \quad (5.189)$$

However, the developed cooling tower model only provides the enthalpy of the outlet air but not the temperature and water content. To calculate the water consumption it is assumed that the outlet air is humidified to 90% relative humidity, which is quite near to the real value for most of the operation conditions. With the knowledge of the enthalpy and the relative humidity, the outlet air temperature and the resulting water content $x_{air, out}$ can be calculated. The water consumption through spray losses strongly depends on construction details of the cooling tower and should be given by the producer. In some cases, wind speed and direction additionally influence this type of water loss. The water consumption through water exchange required to avoid an increase of salt concentration depends on the water consumption through evaporation and on the water hardness.

$$\dot{W}_{ex} = \frac{\dot{W}_{ev}}{\frac{d_z}{d} - 1} \quad (5.190)$$

with:

$\dot{W}_{ex}$	water consumption through water exchange/kg s ⁻¹
$\dot{W}_{ev}$	water consumption through water evaporation/kg s ⁻¹
d_z	allowed hardness of the circulated water/°dH (1°dH = 0.18 mol m ⁻³)
d	hardness of the fresh water/°dH

For the validation of the developed model, measured performance data of an open wet cooling tower is used. For the cooling tower only very few construction details were available. The following performance data of the cooling tower given for one point of operation was used.

Total power	40 kW
Water mass flow rate	0.69 kg s ⁻¹
Water supply temperature	35°C
Water return temperature	21°C
Wet bulb temperature of the ambient air	15°C

Table 5.14 Performance data of a cooling tower.

Information about the electric power consumption and the air mass flow rate were not available. According to provided photographs the fill packing height was assumed to be 0.6 m. With this little information, the cooling performance of the cooling tower has been calculated with the developed model for different wet bulb temperatures of the ambient air and for variable fan speeds. The water inlet temperature and mass flow were kept constant at 35°C and 0.69 kg s⁻¹. The results are

shown in Figure 5.60 and compared to measured performance data. As visible from this figure, the simulated performance of the cooling tower fits excellently to measured performance for the whole range of the wet bulb temperature and from medium to maximum fan speed. Only for very low fan speed the performance of the cooling tower is slightly underpredicted by the simulation model compared to the measured data. A reason for this could be that the proportional law used to calculate the air volume flow rates at different fan speeds does not fit for very low fan speeds. Altogether, a very efficient model has been developed that accurately predicts the performance over a wide range of operation and needs only very little information which is typically available on the data sheet of the cooling towers.

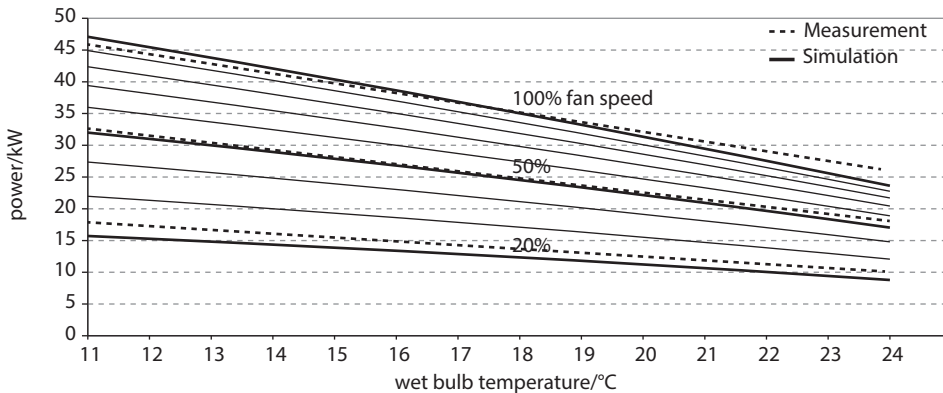


Figure 5.60 Measured and calculated performance of a cooling tower. The inlet water temperature was 35°C and the water mass flow rate was 0.69 kg s⁻¹.

Dry Heat Rejection Systems

Dry heat rejection systems are typically based on an air to liquid heat exchanger, in most cases with forced air flow by a fan. Dry heat rejection systems are not able to cool down the liquid below ambient air temperature. Depending on the flow rate and the size of the heat exchanger, the temperature spread between ambient air and liquid outlet temperature is between 3 K for very large heat exchangers and 10 K for small heat exchangers. Some of the available dry heat rejection systems operate with constant fan speeds, others offer variable fan speeds to adapt the air flow to the required temperature spread of the liquid in dependence of the ambient air temperature. Two types of fan speed control are available, systems with controllable fan speed stages and systems with, for example, frequency inverters for continuous fan speed control.

The heat rejection system has an important influence on the efficiency of solar cooling systems. The temperature reached significantly influences the efficiency of the absorption chiller, and the electricity consumption of the heat rejection system has a decisive influence on the primary energy efficiency reached. For a detailed analysis of solar-driven cooling systems it is therefore important to have a simulation model of dry heat rejection systems, which is able to predict properly the outlet temperature of the cooling water and the electricity consumption of the fan. Very detailed models of the heat exchanger on the other hand require detailed information of the heat rejection system which most of the producers are not willing to provide. Therefore, a simulation model has been developed which requires only the information on the data sheet.

For the model development a dry heat rejection system is analysed, which has been installed at the SolarNext AG office building in Rimsting, Germany and is optionally used as dry heat rejection system for their solar cooling plant. The construction details and the technical data of the system as provided on the data sheet are shown in Figure 5.61 and Table 5.15.

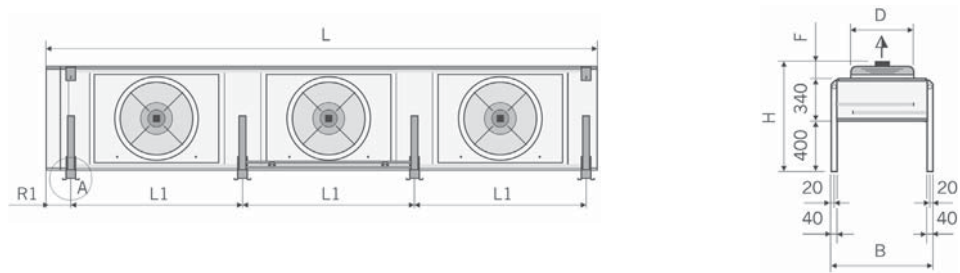


Figure 5.61 Construction details of the GFH052B/3 dry heat rejection system of the GÜNTNER GmbH (Source www.guenthner.de).

Type	L /mm	B /mm	H /mm	R /mm	C /mm
GFH052A/3	3375	895	950	80	875

Description	Max air flow	Reduced air flow
Nominal power	81.6 kW	65.3 kW
Air volume flow rate	23 200 m ³ h ⁻¹	17 500 m ³ h ⁻¹
Ethylene glycol flow rate	13.9 m ³ h ⁻¹	11.1 m ³ h ⁻¹
Pressure drop liquid side	0.5 bar	0.4 bar
Consumed electrical power P_{el}	2.1 kW	1.6 kW

Table 5.15 Dimensions and technical data of a dry heat rejection system.

All values given in Table 5.15 are for a ethylene glycol water mixture with 40% glycol, a water inlet temperature of 40°C and a water outlet temperature of 35°C at 25°C air inlet temperature.

Mathematical model

The behaviour of dry heat rejection systems/air heaters with variable air temperatures and air volume flow rates is typically shown in characteristic diagrams, where the heat transfer efficiency is shown depending on the air volume flow rate for fixed air and fluid inlet temperatures. For fixed

air and fluid flow rates, fluid temperatures and air inlet temperature, the heat exchanger efficiency of dry heat rejection systems/air heaters is defined as

$$\Phi = \frac{(T_{air, out} - T_{air, in})}{\left(\frac{T_{fluid, in} + T_{fluid, out}}{2} - T_{air, in} \right)} \quad (5.191)$$

For constant average fluid temperature the heat exchanger efficiency is also given by

$$\Phi = 1 - e^{-\frac{UA}{\dot{V}_{air} \rho_{air} c_{p, air}}} \quad (5.192)$$

In the developed model the heat exchanger efficiency is calculated for the design conditions using Equation (5.191). If Equation (5.192) is solved for heat transfer power UA we get

$$UA = -\ln(1 - \Phi) \dot{V}_{air} \rho_{air} c_{p, air} \quad (5.193)$$

and the UA value of the heat exchanger can be calculated. The air velocity at design conditions is calculated from the air volume flow rate divided by the heat exchanger inlet area (width of heat exchanger multiplied by the length of the heat exchanger). The air density ρ_{air} and the specific thermal capacity $c_{p, air}$ are calculated from the inlet air temperature and humidity at design conditions. The UA value of water to air heat exchangers follows in general the relation

$$UA = C_{HX} v_{air}^n A \quad (5.194)$$

with C being a characteristic constant of the heat exchanger and $n = 0.4 \dots 0.6$.

From Equation (5.194), the heat exchanger characteristic C_{HX} can be calculated from the design conditions and is assumed to be constant for all operation points. A value of 0.4 is used for n in the simulation model.

For the calculation of the water outlet temperature of other operation points, first the new air velocity v_{air} is calculated from the new air flow rate. Then the new UA value and the heat exchanger efficiency are determined. In the model it is assumed that the water mass flow rate does vary only by $\pm 30\%$ and therefore the water-based heat transfer coefficient is considered to be constant. For the calculation of the water outlet temperature the energy balance between air and water node is used:

$$\begin{aligned} \dot{Q}_{h, air} &= \dot{Q}_{c, fluid} \\ \dot{m}_{air} c_{air} (T_{air, out} - T_{air, in}) &= \dot{m}_{fluid} c_{fluid} (T_{fluid, in} - T_{fluid, out}) \end{aligned} \quad (5.195)$$

Equation (5.191) is solved for the air outlet temperature which is then replaced in Equation (5.195) and the water outlet temperature can be calculated as follows:

$$T_{fluid, out} = \frac{T_{air, in} - T_{fluid, in} \left(0.5 - \frac{\dot{m}_{fluid} c_{fluid}}{\Phi \dot{m}_{air} c_{air}} \right)}{\left(0.5 + \frac{\dot{m}_{fluid} c_{fluid}}{\Phi \dot{m}_{air} c_{air}} \right)} \quad (5.196)$$

The electricity consumption of the ventilator is calculated as follows:

$$P_{el,new} = P_{el,max} \left(\frac{\dot{V}_{new}}{\dot{V}_{max}} \right)^3 \quad (5.197)$$

The model was validated against some operation points provided by the producer on the data sheet for the same heat exchanger but different air flow rates. All data points are for the nominal test conditions mentioned above. The results of the model are shown in Table 5.16.

Water flow rate /m ³ h ⁻¹	Air flow rate /m ³ h ⁻¹	Cooling power (data sheet) /kW	Cooling power (model) /kW	Error /%
13.9	23200	81.6	81.6	0.00
11.0	17500	65.3	64.5	1.23
10.4	15600	61.3	58.8	4.01
8.1	11600	47.5	45.3	4.63
6.9	9540	45.0	42.9	4.67
4.1	5420	24.0	22.7	5.54

Table 5.16 Model validation for a dry heat rejection system.

The results clearly demonstrate that the developed model is able to predict the cooling power of the regarded dry heat rejection system for a wide range of operation with high accuracy. The error between predicted cooling power and the cooling power on the data sheet slightly increases with decreasing air and water flow rate from 1.23% at 75% air flow rate to 5.5% error at 23% air flow rate.

Optimising the control strategy for heat rejection

In the following different heat rejection systems with optimised control strategies are discussed for maximum primary energy savings.

For the development of improved heat rejection systems, detailed analyses on the effect of different control options on the primary energy efficiency are performed using a case study of a 15 kW solar-driven absorption cooling system (ACM) installed at an office building in Southern Germany. Detailed dynamic simulation models of the whole solar cooling system were developed and validated against measured performance data. The developed system models do not only describe the thermodynamic processes but also include the electricity consumption of the ACM, all pumps and of the heat rejection system. These models are used to analyse the effect of different control options of the solar cooling system on the overall system performance and the primary energy efficiency reached.

To compare the efficiency of the system with varied control strategies three different coefficients of performance are used:

1. The standard thermal EER_{th} as the ratio of cold produced to generator heat input;
2. The electrical EER_{el} , which considers the electricity consumption of the ACM, the cooling tower and all pumps including the cold distribution pump;
3. The total primary energy ratio (PER) which is defined as the provided cooling energy divided by the sum of consumed electricity and auxiliary thermal energy multiplied by a primary energy factor (PEF) of 2.7 for electricity (in Germany) and 1.1 for the gas boiler:

$$PER_{auxheat} = \frac{Q_{cool}}{Q_{el}PEF_{el} + Q_{h,add}PEF_{gas}} \quad (5.198)$$

If auxiliary cooling is provided by a compression chiller, the primary energy ratio is obtained from the auxiliary cooling demand multiplied by the primary energy factor of the compression chiller system.

$$PER_{auxcool} = \frac{Q_{cool}}{Q_{el}PEF_{el} + Q_{cool,aux}PEF_{compression}} \quad (5.199)$$

System simulations and validation

The analysed solar-driven absorption cooling system has been set up and installed as a test facility to cool and to heat the office building of the SolarNext company in Rimsting, Germany (47.88° North, 12.33° East). The system includes a market available 15 kW absorption chiller, two 1 m³ hot water storage tanks, one 1 m³ cold storage tank, 37 m² flat-plate collectors and 34 m² solar vacuum tube collectors all facing south with an inclination of 30°. The open wet cooling has a nominal cooling capacity of 35 kW at 30°C water supply and 36°C cooling water return temperature at ambient conditions with a wet bulb temperature of 24°C. Additionally, dry heat rejection is possible.

For the distribution of the cooling energy, chilled ceilings and fan coils are used with 16°C supply and 18°C return temperature and an automated supply temperature increase for dew point protection.

The cooling load of the single story office building with 566 m² of conditioned space is 8.9 MWh/a (16 kWh m⁻² a⁻¹) and the maximum cooling load is 18 kW.

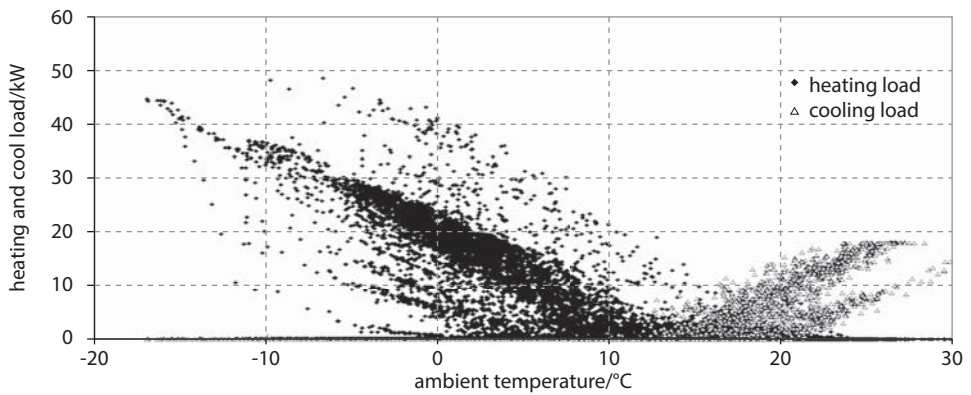


Figure 5.62 Ambient temperature dependent cooling and heating load distribution of the case study office building in Rimsting, Germany.

The installed absorption chiller is able to provide the required maximum cooling power of 18 kW if the temperature set point of the heat rejection system is reduced by 3 K from 30°C design conditions to 27°C. Table 5.17 shows the installed components and their electricity consumption.

Component description	Type	Nominal volume flow /m ³ h ⁻¹	Pressure drop /10 ⁵ Pa	Electrical power demand /W
Absorption chiller with solution pump	EAW WEGRACAL SE 15			300
Wet cooling tower	Axima EWK 035 (1420 m ⁻¹ design fan speed)	5.0		330
Absorber/Condenser pump	Wilo-IP-E 40/115- 0,55/2-3 PN10 Wilo-VeroLine-IP-E	5.0	1.20	550
Generator pump	High efficiency pump Wilo-Stratos ECO 25/1-5 PN10	2.0	0.25	56
Evaporator pump	High efficiency pump Wilo-Stratos 25/1-6 PN 10	1.9	0.35	52
Primary solar pump	WILO Stratos 30/ 1-12 PN 10	0.388	0.45	160
Secondary solar pump	WILO TOP-S 30/ 7	0.388	0.35	130
Total				1578

Table 5.17 Electrical consumption of main components of the absorption chiller system.

The distribution of electrical power has been compared with results from other low power machines (10 kW and 19.4 kW) as well as with medium power adsorption and very large compression chiller system. In all cases over 50% of the total electricity consumption is used in the heat rejection circuit. The only difference between the monitored 15 kW and a new 19.4 kW machine from the

EAW company is an improved machine design with only one solution pump; the solar pumps are assumed to consume the same.

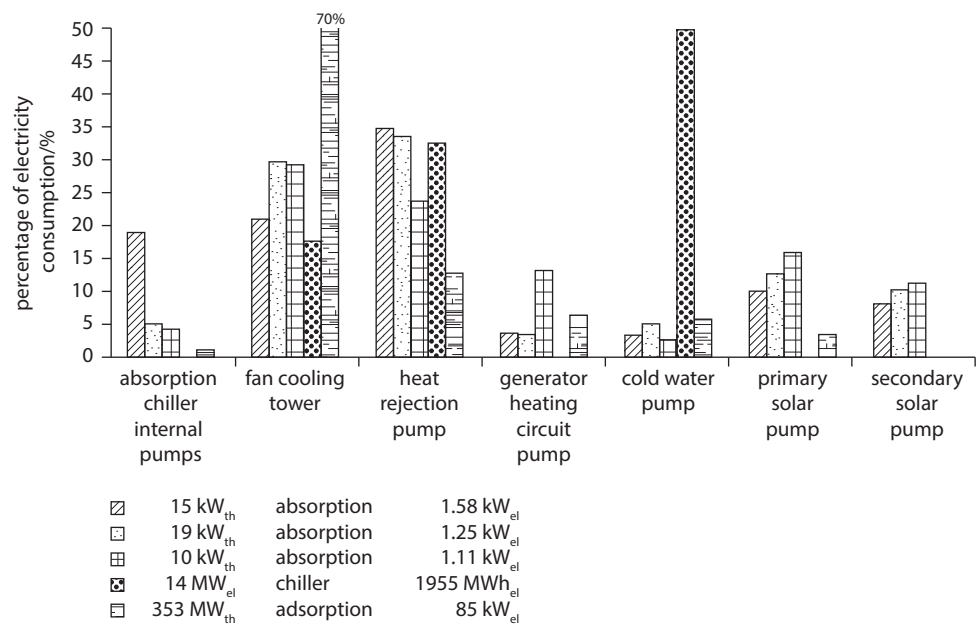


Figure 5.63 Percentage distribution of auxiliary electricity consumption for LiBr/Water absorption chillers, a large vapour compression system and an adsorption chiller.

Simulation results and discussion

A detailed dynamic simulation model of the installed system, which also considers the electricity consumption of all installed components (fans, pumps, etc.), has been developed in the simulation environment INSEL. The component models used include dynamic models for the solar collectors and the hot and cold storage tank. No inertia is considered for the absorption chiller, the piping, the wet cooling tower and the dry heat rejection.

Measurement data of the solar-driven absorption chiller in summer 2007 was used to validate the developed simulation model of the installed system. A comparison of the simulated and measured outlet temperatures of the generator, condenser and evaporator of the ACM and of the collector field shows that the performance of the installed system is well described by the developed simulation model (see Figure 5.64).

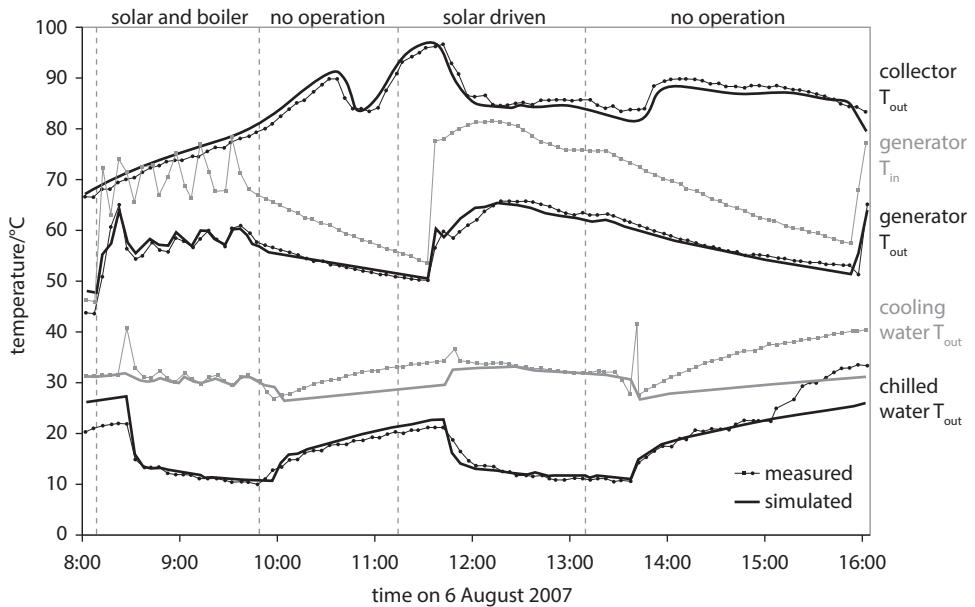


Figure 5.64 Measurements and simulation of the absorption chiller system with vacuum tube collectors and hot storage.

The deviation between the predicted and measured solar heating power of the collector circuit (dynamic model) is below 1% for the analysed day. For the absorption chiller larger differences only occur at system startup and system shutdown, which is due to the omitted inertia of the absorption chiller in the model. Otherwise the deviation between simulated and measured generator and evaporator power is below 4% and below 3% for the heat rejection power. During start up and shut down the deviation between measured and predicted performance increases to 16% for the generator, 12% for the evaporator and 10% for the heat rejection circuit of the absorber and condenser. These rather large deviations can be partly attributed to the fluctuations of the generator inlet temperature for the combined operation mode with auxiliary heating. These fluctuations result from a bad hydraulic integration of the auxiliary heater and a poorly controlled three-way mixing valve in the original system setup (optimised during the heating period 2008/2009). If only the purely solar driven part is considered, the deviations between measurement and simulation are reduced to 12% for the generator, 10% for the evaporator and 6% for the heat rejection circuit of the absorber and condenser.

Analysed control options for the heat rejection systems

The validated simulation model is used to analyse the effect of different control strategies for the installed ACM, the solar thermal system and the cold distribution on the overall performance of the solar cooling system.

The pumps of the solar collectors are considered as On/Off controlled in all cases. The collector pump is set in operation as soon as the collector temperature is 10 K above the temperature at hot storage bottom and is switched off again if the collector outlet temperature is 5 K above the temperature at hot storage bottom or if the temperature in the upper part of the

hot storage increases above 95°C. The minimum operation time of the collector pump is set to 2 minutes. The cold water supply temperature was set to 16°C. To simplify the control, the cases with variable generator inlet temperature are without temperature control at the generator inlet and with constant evaporator mass flow rate. The ACM is turned off if the generator inlet temperature drops below 65°C or the evaporator outlet temperature decreases below 6°C.

Analysed cases	Control options										
	Cooling tower			$t_{a,in}$ /°C				$t_{g,in}$ /°C variable		Chilled water dist. pump	
	Type	3-way valve	Fan speed	27	24	21	90	70-90	70-95	ΔT control	
										yes	no
Case 1	Wet	•		•			•				•
Case 2	Wet		•	•			•			•	
Case 3	Wet		•	•				•		•	
Case 3.1	Wet		•		•			•		•	
Case 3.2	Wet		•			•		•		•	
Case 4	Dry	•		•					•	•	
Case 5	Dry		•	•					•	•	
$t_{a,in}$	Absorber inlet temperature, either controlled by a 3-way valve or by fan speed control of the cooling tower. Values below 27°C (30°C for dry cooling tower) are only provided as long as reachable at the given ambient conditions.										
$t_{g,in}$	Generator inlet temperature, either constant or variable according to the temperature in the hot and chilled water storage tank.										

Table 5.18 Analysed control options of the absorption cooling system.

An additional Case 6 has been defined and analysed as reference system for a compression chiller with an electrical EER of 4.0 at 27°C heat rejection temperature and high cold water supply temperatures. The electricity consumption for heat rejection and cold distribution is considered separately. The compression chiller is combined with a dry heat rejection system with constant fan speed.

Simulation results and discussion

The electrical performance of the system strongly depends on whether the cold water distribution pump of the building and the ventilator of the cooling tower are controlled according to part load conditions or not. The electrical EERs vary between 6 and 11.5 for the cases with wet cooling tower and between 4 and 8 for the cases with dry heat rejection. The compression chiller system reaches an overall electrical EER of 3.2 (see Figure 5.65).

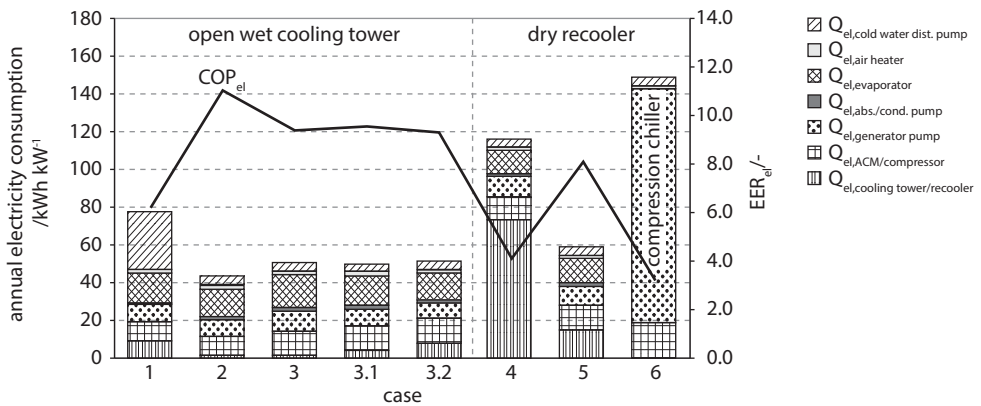


Figure 5.65 Annual electricity consumption and electrical EER.

The lowest electricity consumption of 42 kWh per kW of cooling power and therefore the highest electrical EER of slightly above 11 is obtained for Case 2 with a controlled cold water distribution pump and cooling tower fan but with the absorption chiller operated at constant generator inlet temperature. If the generator inlet temperature is allowed to vary between 70 and 90°C according to the temperature in the hot and cold storage tank (Case 3), the lower generator temperatures and correspondingly lower cooling power lead to longer operating hours to provide the same cooling energy and therefore increases the electricity consumption. The thermal EER decreases very slightly from 0.75 in Case 2 to 0.74.

However, at the same time the solar fraction is significantly increased from around 70% in Cases 1 and 2 to 83% in Case 3. A further increase of the solar fraction up to 88% can be achieved, if the heat rejection temperature set point is decreased from 27 to 24°C in Case 3.1 and to 21°C in Case 3.2 (see Figure 5.66). The reduced heat rejection temperature set points lead to an increase in the electricity demand of the cooling tower due to higher fan speeds but at the same time reduce the operating hours of the whole cooling system due to the increased thermal EER and cooling capacity, which balances the additional electricity demand.

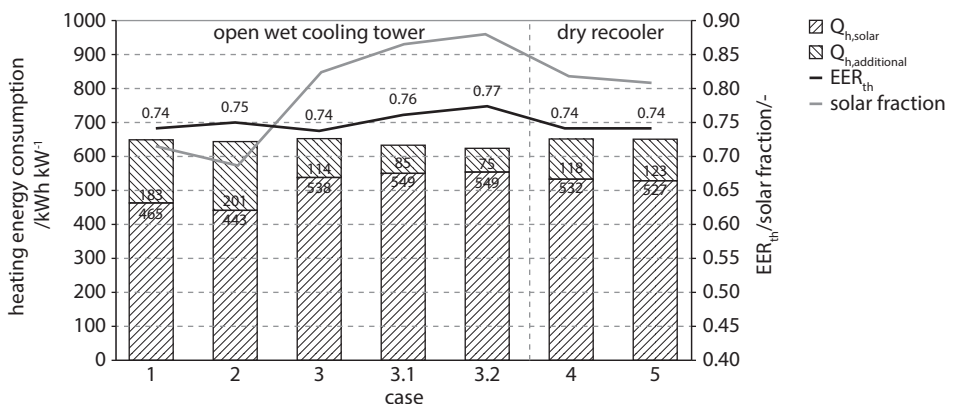


Figure 5.66 Heating energy consumption and solar fraction.

The values of the primary energy ratio, which consider the electricity and additional heating energy consumption, vary between 1.1 and 2.2, with the lowest value for Case 4 with dry heat rejection and without fan speed control and the highest value for Case 3.2 with the lowest set point for the heat rejection temperature including fan speed control and a variable generator inlet temperature (see Figure 5.67). The reference system with the compression chiller reaches a primary energy ratio of 1.1 which is nearly half of the value of the best absorption chiller case (Case 3.2) but already better than the worst absorption chiller case with dry heat rejection with constant fan speed (Case 4). This clearly indicates the importance of an energy efficient control and design of solar cooling systems and the requirement of further optimised hydraulic systems with reduced pressure drops and the utilisation of highly energy efficient pumps.

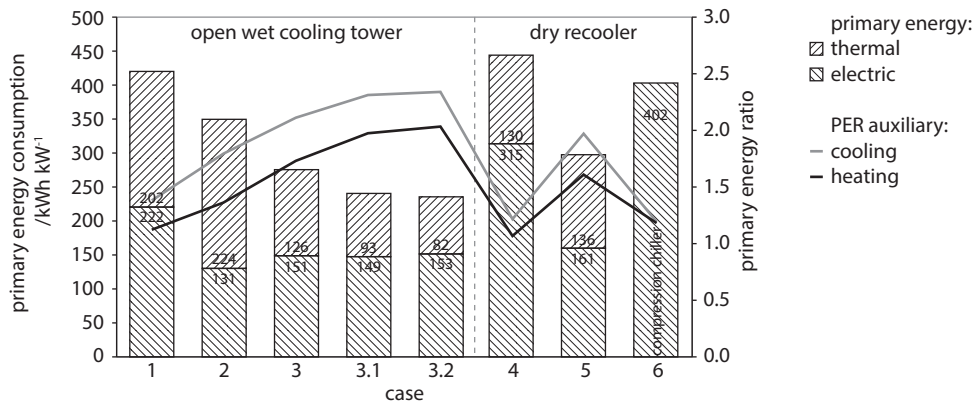


Figure 5.67 Primary energy consumption and primary energy ratio.

As visible from Figure 5.67 the additional heating energy consumption significantly increases the primary energy consumption although a high solar fraction of 70% and above is reached. This is due to the relatively low thermal EER of single effect absorption chiller. If the additional heating energy would be replaced by additional cooling provided by a highly efficient electrically driven compression chiller with a primary energy ratio of 1.1 (including electricity energy consumption of pumps and heat rejection system), 20 - 30% higher primary energy ratios can be reached with a maximum of 2.5. In this case the absorption chiller only provides cooling energy as long as sufficient heating energy is provided by the solar system and the remaining cooling energy is provided by the compression chiller.

Optimisation potential

The remaining main electricity consumers for Case 3 with optimised control are the absorber and condenser pump (37%), the collector pumps (28%), and the absorption machine itself (23%, mainly solution pump, see Figure 5.68). To further improve the overall efficiency of the solar cooling system, the electricity consumption of these three components needs to be reduced significantly. For the cooling circuit of the absorber and condenser, attempts need to be made to further reduce the pressure drop of the heat exchangers and of the spray nozzles (alternative distribution system) of the open wet cooling tower. A reduction in electricity consumption of at least 30% could be reached by these measures.

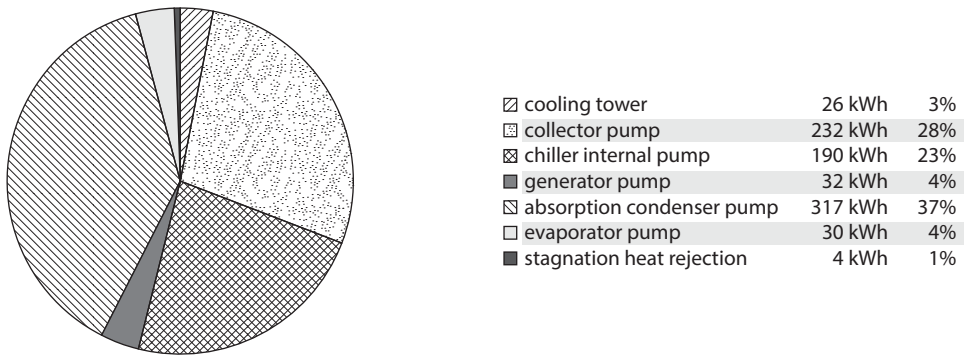


Figure 5.68 Electricity consumption of Case 3.

For the solar system the electricity consumption could be significantly reduced by up to 50% and more, if the solar system would be operated with pure water. In this case the pressure drop and heat losses of the heat exchanger and the secondary collector pump could be avoided completely. However, in regions with danger of frost, a special frost protection control and additional temperature sensors need to be implemented in the system. If danger of freezing is detected by the control system from the temperature sensors, the control system switches the collector pump on for a short time to pump warm water from the hot storage bottom into the collector field. According to recent analyses on a large solar cooling installation in Southern Germany, the heating energy losses caused by frost protection related to the annual solar energy production of the collector field are only about 3%.

For the absorption chiller, the electricity consumption is mainly caused by the integrated water and solution pumps. In the system design of the EAW WEGRACAL SE 15 with two containers, the absorber and generator are mounted at the same level. Therefore, two solution pumps are required, one for the concentrated and one for the diluted solution. A new chiller system design has been developed which integrates all components in one container with the generator and absorber mounted on different height levels. For this new design only one solution pump is required, which reduces the electricity consumption of the absorption chiller by at least 30%.

A 30% electricity reduction would improve the electrical EER of Case 3 to a value of 13. This would result in an increase of the primary energy ratio to a value of 2.7 for auxiliary heating and to 3.2 for auxiliary cooling.

To achieve the 30% savings in auxiliary electricity, vertical ground heat exchangers are an interesting option, as the parallel connection of many tubes leads to low pressure drops. Here only the circulation pump of the geothermal heat exchanger needs to be considered – the electricity needs for the cooling tower fans, which are between 17% and 30% of the total auxiliary electricity, are saved.

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6 Geothermal heating and cooling

Figure 6.1 (previous page): The Elektror office building in Scharnhauser Park near Stuttgart, Germany, with a renewable heating and cooling system based on biomass cogeneration (Photo: M. Seubert).



Figure 6.2 Installation of a horizontal low depth heat exchanger in Wüstenrot (Photo: Ursula Pietzsch).

Low depth geothermal energy is well suited for heating and cooling applications in buildings, as the underground temperature has a constant level around the annual average air temperature at a depth of about 15 m and then increases again by about 3 K per 100 metres. As this temperature level is above ambient air temperature in winter, the geothermal energy can be used as a heat source for heating applications. In summer, the geothermal temperature levels are usually lower than ambient air temperatures and the heat exchangers can be used as a heat sink for cooling applications.

The geothermal heat flux from the 3000 to 6000 K hot centre of the earth is very small at 0.06 W m^{-2} . Annually, 1021 Joule are transferred to the surface, 1000 times less than solar energy. However, in some regions with geothermal anomalies the temperature gradient can be as high as 10 K per 100 m.

Energy can be extracted from the ground to supply heat pump evaporators at higher temperature levels than ambient air in winter and thus improve the seasonal performance factor (SPF) of the heat pump. Vice versa, the geothermal heat exchangers can be used for heat rejection of the chiller condensation energy at lower temperature levels than ambient air or cooling towers, which again improves the chiller energy efficiency ratio (EER). However, although there are hundreds of thousands ground source heat pumps installed worldwide, there is very little experience reported with geothermal heat rejection for cooling machines.

Direct use of geothermal energy for building cooling or heating is the most efficient way to use low depth geothermal heat exchangers, as only electricity for energy distribution (pumps and fans) is required.

If air is used as the heat carrier medium, the heat exchangers are typically pipes buried beneath the surface, through which air is drawn into the building by electric fans. Even in warmer regions, the annual average ground temperature is lower than peak ambient air temperatures, so that pre-cooling of ventilation air is possible. A summary of application fields of low depth geothermal energy use is shown in Table 6.1.

Water or brine filled ground heat exchangers are either designed in a horizontal configuration with a shallow depth of about 2 m or installed as vertical loops. Vertical ground heat exchangers are usually constructed by inserting one or two high-density polyethylene U-tubes in vertical boreholes of 75 - 150 mm diameter. Vertical loops are usually connected in parallel to reduce pressure drop. In Europe, double U-tubes are common, whereas in the USA single tubes prevail. Double U-tubes reduce the borehole resistance by 30 - 90% and increase heat transfer.

The water cooled through contact with the earth is then distributed in the building using either activated concrete slabs with buried pipes or via air-based ventilation systems, in which the air is cooled by the water or brine (in an additional heat exchanger).

When horizontal heat exchangers at shallow depths are used, soil temperature may be influenced by the surface conditions. At greater depths, the ground may be cooled in the winter season if a heat pump extracts energy from the soil. Unbalanced heat extraction from the ground can reduce temperature levels by about 6 K within 5 years for heat pump operation only. If only heat is rejected to the ground from a cooling machine, after more than 10 years the soil temperature can reach 35°C and is no longer suitable for air conditioning. Only balanced heat fluxes kept ground temperatures constant over a 30 year period.

Heat source and heat sink	Use of heat source and heat sink	Distribution in building	Use of secondary energy	Application and temperature level
Geothermal heat exchanger	Indirect Heat pump with geothermal heat exchanger as heat source Cooling machine with geothermal heat exchanger as heat sink	Water	Heating	Domestic hot water: 60°C
				Central heating supply: 70/50°C (high-temperature heat)
				Central heating supply: 45/38°C (medium temperature heat)
			Cooling	Floor heating supply: 35/28°C (low-temperature heat)
				Building air conditioning: 12 - 18°C (low load)
	Direct	Air	Cooling	Building air conditioning: 6 - 12°C (high load)
				Cold storage warehouse: -3 - 0°C
			Heating	Deep-freezing warehouse: -30°C
				Supply air preheating: 2 - 15°C
				Pre-cooling of supply air for ventilation: 10 - 25°C
	Direct	Water	Cooling	Radiative panel/concrete core cooling: 14 - 21°C

Table 6.1 Applications of geothermal energy use.

Field studies in Germany showed that the use of a geothermal heat source improves the seasonal performance factor (SPF) from 2.6 for an ambient air heat pump to 3.3 in existing buildings (49 systems). For new buildings with low-temperature heating systems the SPF increases from 2.9 to 3.9 (74 systems monitored) (Miara et al., 2011).

The electricity consumption is thus reduced by 21% in existing and 26% in new buildings. Per kWh of geothermal energy, 0.19 kWh of electricity or 0.51 kWh of primary energy are saved in existing buildings and 0.18 kWh of electricity or 0.49 kWh of primary energy in new buildings. This is an important fact to note: only about 50% of the geothermal energy extracted effectively saves primary energy; i.e. the geothermal energy cannot fully be counted for primary energy reduction.

When comparing geothermal energy with solar thermal support of a heat pump, an analysis of six solar supported heat pump systems in Austria gave average SPF increases from 2.0 to 2.6 when including all auxiliary electricity consumption. The analysis showed that the heat pumps often produced much higher temperature levels than required for floor heating systems or domestic hot water. A combined geothermal and solar thermal heat pump system with good temperature control resulted in a SPF of 3.94.

Eklöf and Gehlin (1996) carried out thermal response test measurements at a borehole cooling installation for telephone switch stations in Sweden located in Drevikstrand and Bromma. In Drevikstrand, four boreholes were placed in a line with about 5 m spacing in average. The systems were designed to cool the circulating fluid from 22 to 16°C. With a measured high thermal conductivity of about $4 \text{ W m}^{-1} \text{ K}^{-1}$ and a rather low borehole resistance of 0.09 K m W^{-1} , the mean heat rejection capacity was determined as 30 W m^{-1} . In Bromma, 13 boreholes of 130 - 160 m depth were placed in an irregular pattern. Similar soil conductivities and even lower

borehole resistances were measured. The required temperature levels were lower at 20°C return and 14°C supply. The measured heat rejection power was significantly lower at 17 W m⁻¹.

Detailed monitoring results of maximum and average heating and cooling power of ground heat exchangers are still rare and designers often work with average values given in standards such as the German VDI 4670. In the following results from five monitoring projects are discussed to show real performance of today's ground heat exchangers in different system environments and to suggest optimisation measures.

6.1 Direct geothermal energy use for cooling and preheating of buildings

Vertical earth brine to air heat exchangers

Five vertical borehole heat exchangers of 80 m each supply cooling and heating energy to an air-based distribution system for a seminar room in an office building in Freiburg, Germany (178 m² surface area, maximum volume flow 5100 m³ h⁻¹). The geothermal cooling system is only operated when needed, i.e. if a lecture or seminar takes place. In addition direct summer cooling is provided to 157 m² activated concrete floor in the lobby. This combination has proven to be crucial to achieve reasonably long operating hours and thus an economical performance of the system. Only 233 hours for fresh air cooling were counted for the seminar room in 2005 compared to 2289 hours for the floor activation and 856 hours for winter air preheating.

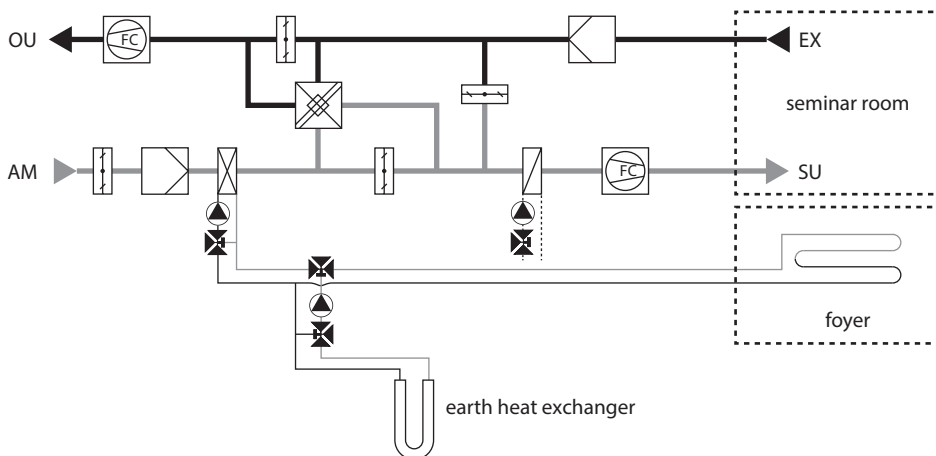


Figure 6.3 System schematics of the geothermal energy use for ventilation and concrete core cooling in the Freiburg SIC building.

The measured average soil temperature is close to 16°C during the summer. The brine temperature rises steadily as soon as the pump is turned on, reaching a spread of nearly 3 K during the day. However, the soil recovers quickly during the nights when the pump is shut off. Still a rising trend of the soil temperature is clearly visible from below 15°C to almost 17°C within a week. Supply air temperatures during operation were between 18 and 22°C (see Figure 6.4).

The coefficients of performance for heating and cooling are excellent due to the low pressure drop across the heat exchanger and the low power consumption of the geothermal heat

exchanger pump of 170 W. The average coefficient of performance for cooling is 21.6 and 18.8 for heating.

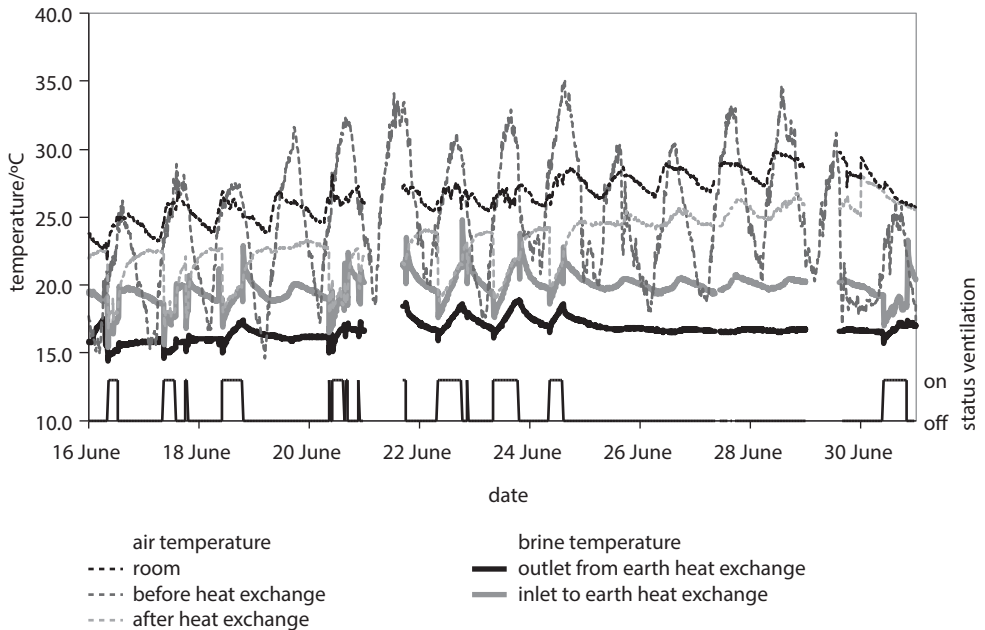


Figure 6.4 Summer cooling of ventilation air in the Freiburg building.

The thermal power dissipated by the heat exchangers is rather low with a maximum of about 24 W per metre, in case the ventilation system is operating. Also during the winter months with ambient temperatures below -10°C , the power extracted was a maximum of 16 W per metre.

If the floor cooling system is used, the thermal power dissipated by the earth heat exchangers drops to 5–10 W m^{-1} . Two factors account for the reduced power: the temperature difference between the floor cooling system and the soil is smaller than for the ventilation system with ambient air temperatures up to 35°C . Furthermore the heat transfer area between the floor and room air is rather low with 157 m^2 . At about 20 W m^{-2} cooling power, only 3 kW cooling load can be taken up by the activated floor. The temperature spread between ground heat exchange return and supply is at 3 K on average and thus slightly higher than in the case of ventilation use, as the flow rates decrease by a factor five due to higher pressure drops in the concrete floor. Increasing the pump power could decrease this temperature spread and thus the surface temperature by 1 or 1.5 K further, but this would not significantly increase the cooling power distributed in the building. A larger concrete floor activation area would improve the geothermal heat exchanger performance.

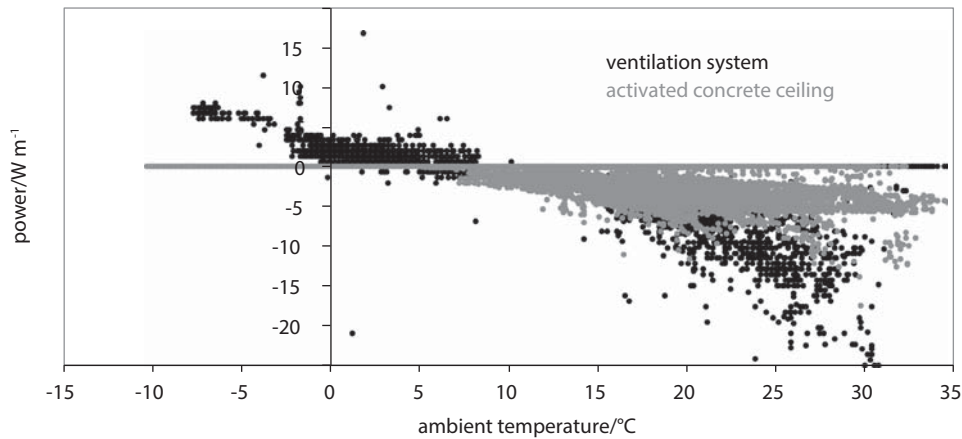


Figure 6.5 Measured geothermal power of the vertical heat exchangers in the Freiburg SIC building during 2006 as a function of ambient temperature.

If the power delivered by the geothermal heat exchanger is plotted against the brine inlet temperature, the scattering of power values reduces. The inlet temperatures only vary between 16.5 and 24°C during summer. Again, the lower power of the activated floor distribution are due to the low flow rates and limited area of activated floors.

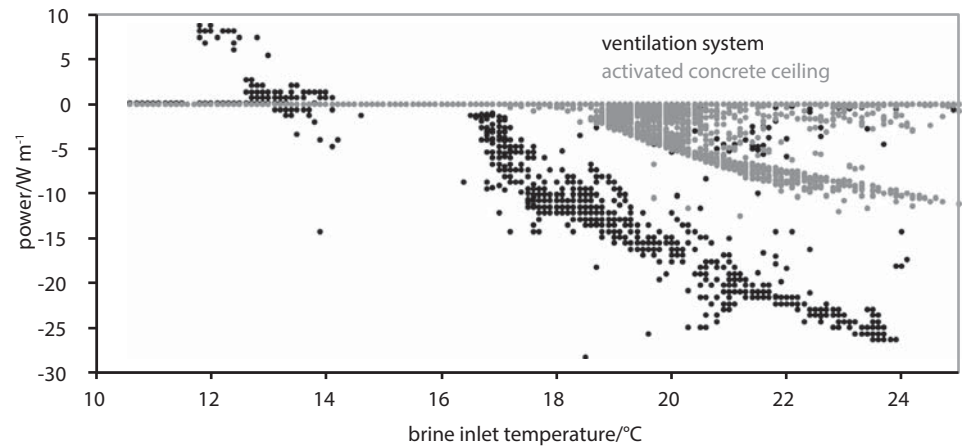


Figure 6.6 Measured geothermal power in 2006 as a function of brine inlet temperature to the ground.

Due to significantly longer operating hours of both the ventilation system of the seminar room and the floor cooling system in 2006, the cooling energy delivered by the geothermal heat exchangers nearly doubled from 2.76 MWh in 2005 to 4.87 MWh. This corresponds to 7 - 12 kWh per metre heat exchanger and summer. The low COP for heating in 2006 is due to the fact that the geothermal system was often operated at a low-temperature difference between ambient air and ground heat exchanger. Table 6.2 summarises the measured results.

Year	Cooling /kWh	Heating /kWh	COP		cooling, seminar	Hours	
			cooling /-	heating /-		cooling, floor	heating
2005	2759	2846	20.1	12.8	233	2289	856
2006	4873	246	13.5	5.2	327	2911	289

Table 6.2 Summary of monitoring results for 400 m of vertical heat exchangers in the Freiburg SIC building.

Foundation pile heat exchangers

Ground heat exchangers using foundation piles are a cost-efficient solution for heating or cooling of water in integrated U-tubes.

Measurements were done in a German office building constructed in 2008 in Ostfildern. Here 104 foundation piles were constructed with U-shaped tubes of 4 - 8 metres depth each, resulting in a total length of 624 m. The piles are rather close together (between 2.5 and 7 metres).

The office building with 2962 m² gross area has a heating and thermal absorption cooling system and is supplied by a biomass cogeneration fired district heating network. Heat and cold are distributed via concrete core activation combined with some finned radiators at the air outlets. The ventilation system with a volume flow rate of 6550 m³ h⁻¹ uses air preheated or pre-cooled by the geothermal heat exchangers.

Figure 6.7 shows the heat exchange of the foundation piles in Watt per metre as a function of ambient temperature during one year (2010). The total power of each pile is within the range of 7 kW for heating and 14 kW for cooling, which corresponds to a maximum of 24 W m⁻¹ at 36°C ambient temperature and 14 W m⁻¹ at -5°C.

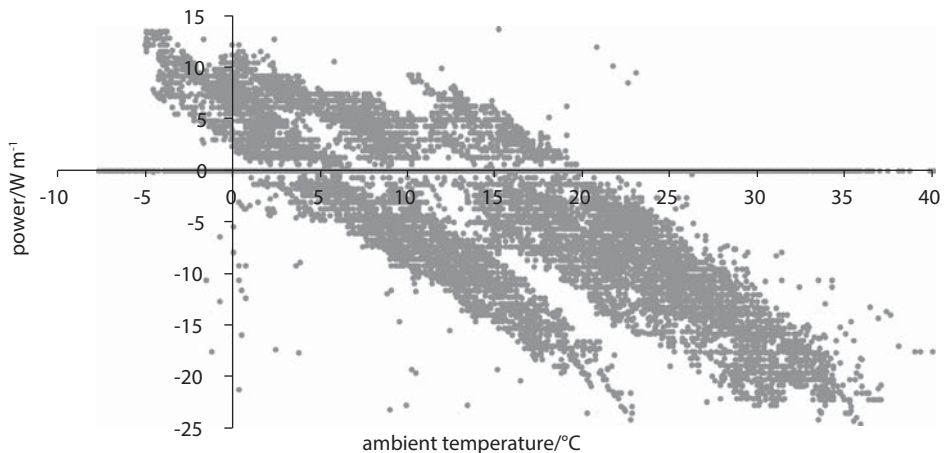


Figure 6.7 Thermal power exchanged with the ground by the foundation pile heat exchangers connected to the building ventilation system.

If the power is plotted versus the inlet brine temperature to the foundation pile heat exchanger, the temperature range reduces in summer from 15.5 to 24.5°C, similar to the application in the SIC

building with vertical heat exchangers. The power transferred is comparable to the vertical heat exchangers.

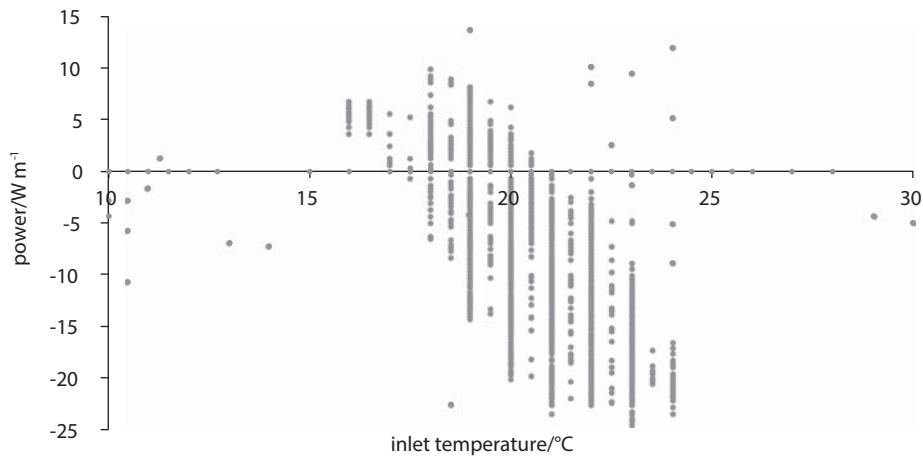


Figure 6.8 Power exchanged with the geothermal foundation pile heat exchanger as a function of brine inlet temperature.

Figure 6.9 shows the measured temperatures on the ventilation air and brine side before and after the geothermal heat exchanger in July 2010. The ambient air is cooled down for the most of July by about 5 K, while the brine temperature supplied to the brine to air heat exchanger increases from 19 to 21°C after two extremely hot weeks. The temperature difference between supply and return temperature within the geothermal heat exchanger is only 1 - 2 K. The heat recovery efficiency of the heat exchanger during the hot days of July was between 46% and 54%.

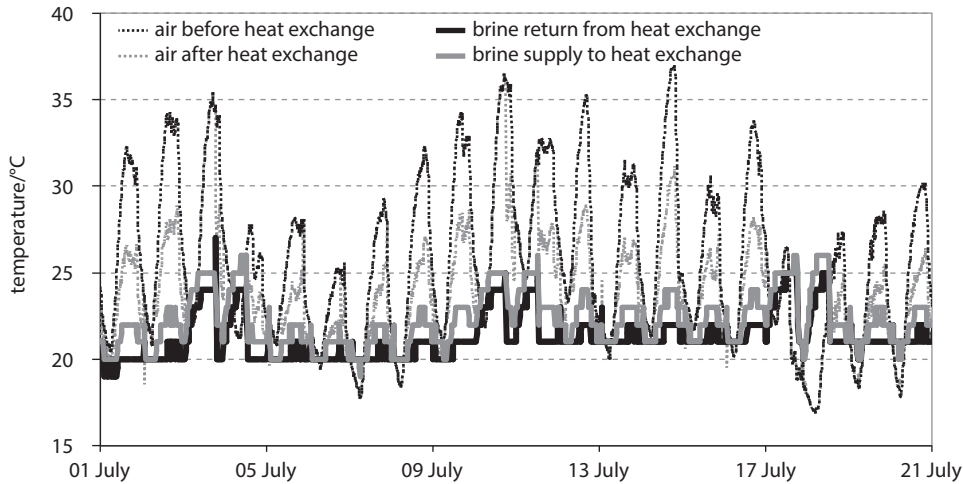


Figure 6.9 Temperature levels of air and brine before and after the heat exchanger.

Earth to air heat exchangers

In one of the first German passive standard office buildings in Weilheim an earth heat exchanger cools the fresh air supplied to the building. It is positioned around the building and consists of two polyethylene pipes with a diameter of 0.35 m and a length of 90 m each. The pipes are laid in a mean depth of 2.80 m at a mutual distance of 0.9 m (see Figure 6.10). By ventilating ambient air through the system, the air is cooled in summer and heated in winter.

The earth to air heat exchanger of the office building is mainly designed for winter preheating of ambient air. It reduces the heating demand for fresh air ventilation and prevents freezing of the cross-flow heat exchanger of the heat recovery system. During summer the earth to air heat exchanger helps meeting the cooling loads of the building. A main design goal was to achieve a small pressure loss. At the nominal volume flow of $1900 \text{ m}^3 \text{ h}^{-1}$ and a pressure drop of 175 Pa, an air exchange rate of 0.6 h^{-1} is provided.

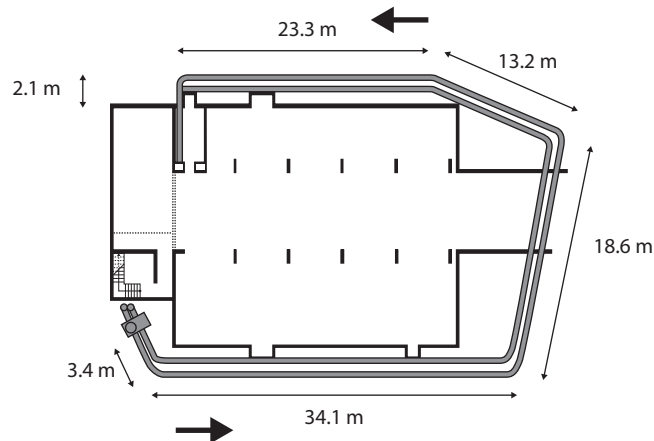


Figure 6.10 Earth to air heat exchanger in the Weilheim office building.

Inside the earth to air heat exchanger tubes, temperature sensors are placed, each 9 m. At inlet ambient air temperatures for the earth to air heat exchanger between 9 and 16°C , overlaps occur between heating defined for outside temperature below 15°C and cooling operation fixed at outside temperatures above 15°C . This means that sometimes the air is heated during the cooling period (summer) and vice versa—a control strategy which could certainly be improved.

However, the performance of the earth to air heat exchanger is excellent due to the direct contact of the ventilation air with the ground. A mean cooling power of $6 - 10 \text{ W m}^{-1}$ is obtained, increasing with increasing mean summer temperatures, and $6 - 9 \text{ W m}^{-1}$ mean heating energy. The maximum cooling power is 45 W m^{-1} , while the maximum heating power achieved is 34 W m^{-1} , resulting in annual cooling energy levels of 25 kWh m^{-1} and 17 kWh m^{-1} for heating (see Figure 6.12).

The annual coefficients of performance were calculated from the sum of heating and cooling energy divided by the consumed electrical energy. They achieve excellent values of 50, 35 and 38 for 3 years of measurements (see Figure 6.12). However, the earth heat exchanger cannot fully remove the daily cooling load, as the hygienically required fresh air volume flow limits

the cooling power, which can be supplied by such systems. From the average internal loads of $131 \text{ Wh m}^{-2} \text{ day}^{-1}$ in the south office, the earth heat exchanger efficiently provided $24 \text{ Wh m}^{-2} \text{ day}^{-1}$ in the summer period 2003, i.e. 18%.

Horizontal earth brine to air heat exchangers

An alternative to the direct air-based ground heat exchange is the indirect cooling of air by circulating a brine solution through the ground and then using an air to brine heat exchanger. This system has the advantage of avoiding any hygienic problems, which might occur in air-based systems, if condensation water is not reliably removed. Furthermore, pressure drops are usually lower in such liquid-based systems. A simple horizontal absorber placed round the perimeter of a rehabilitated office building was experimentally analysed by the author.

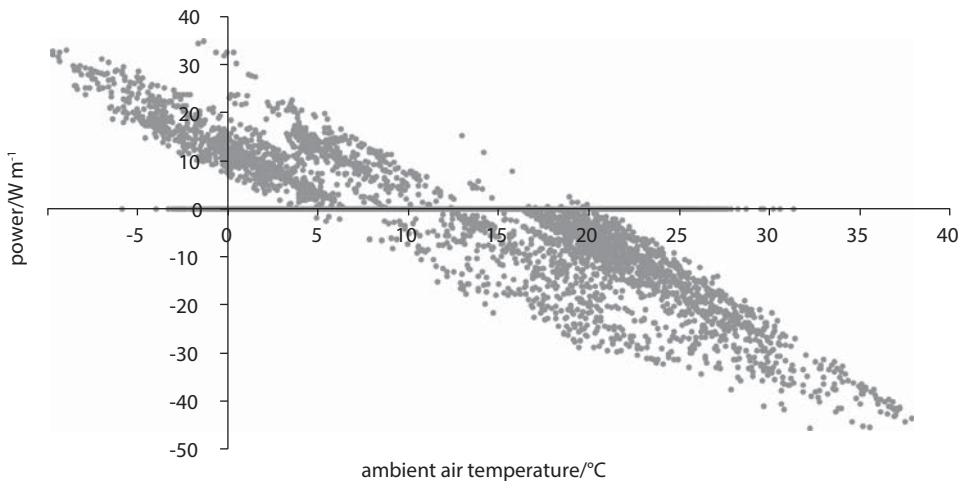


Figure 6.11 Power of earth to air geothermal heat exchanger in the Weilheim office building as a function of ambient air temperature, which corresponds to the inlet temperature to the ground heat exchanger.

For air preheating and cooling of the Eboeck office building in Tübingen, Germany, five horizontal earth to brine heat exchangers with a length of 100 m each are installed shallow under the soil surface (about 1.2 m depth). During summer they are used for cooling of supply air. The volume flow of air during daytime is about $1750 \text{ m}^3 \text{ h}^{-1}$.

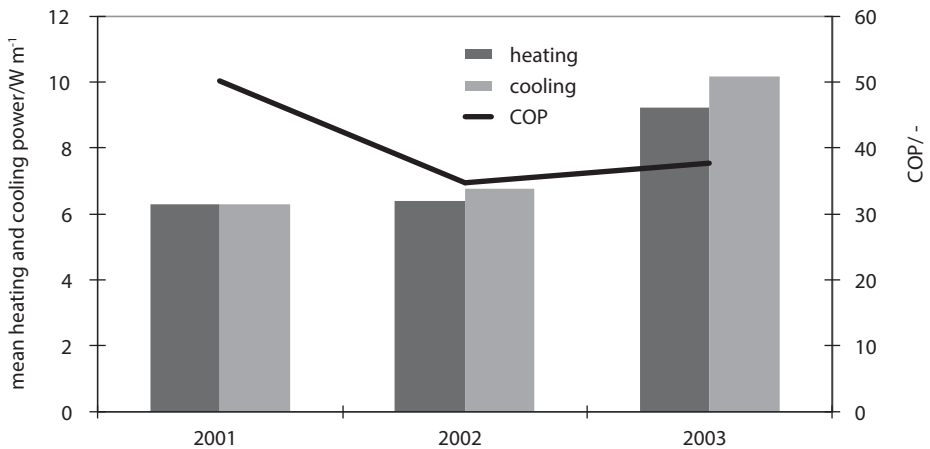


Figure 6.12 Measured mean annual heating and cooling power for the earth to air heat exchanger in the Weilheim building.

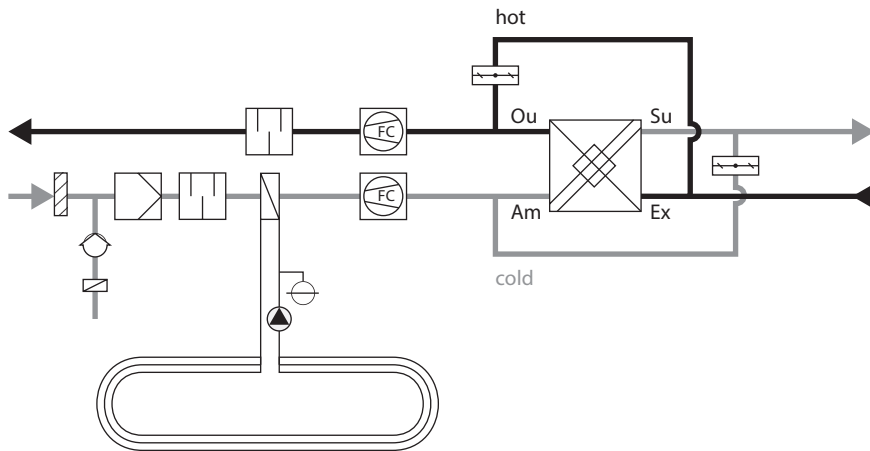


Figure 6.13 Integration of shallow geothermal heat exchanger in mechanical ventilation system in the Eboek office building.

The earth temperature levels were measured for 2 years directly in between the tubes at 1.2 m depth and a distance of 0.5 m from the building and compared to the undisturbed soil temperatures in 1.2 m and 2.0 m depth. When the earth heat exchanger is operating, for example, for 2 months during summer 2006, the daily mean earth temperature increases by about 2 - 3 K. The regeneration of the soil temperature is very fast both in summer and winter.

During a hot 14-day measurement period in June 2005, an average cooling power of 1.5 kW with a maximum of 3.5 kW was measured. Due to the close proximity and low depth of the tubes, the maximum heat dissipation per metre of tube is only 7 W.

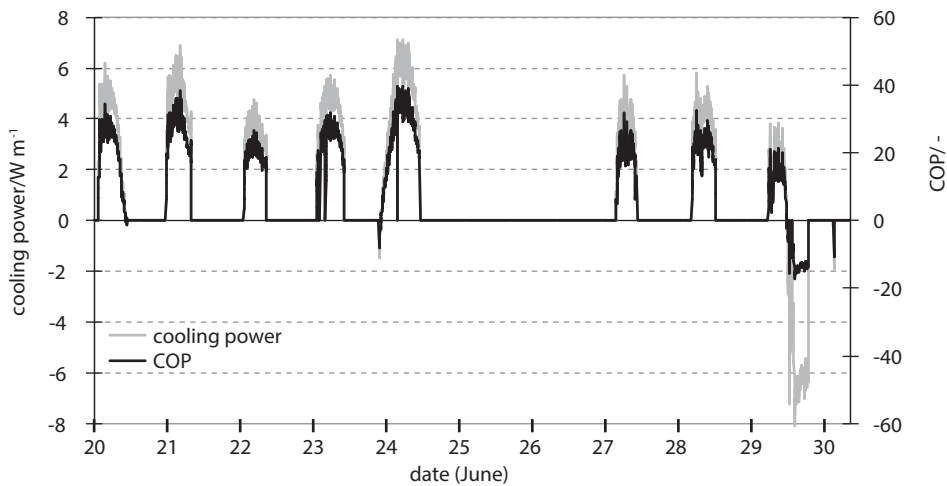


Figure 6.14 Measured cooling power per metre of horizontal heat exchanger and electrical coefficient of performance.

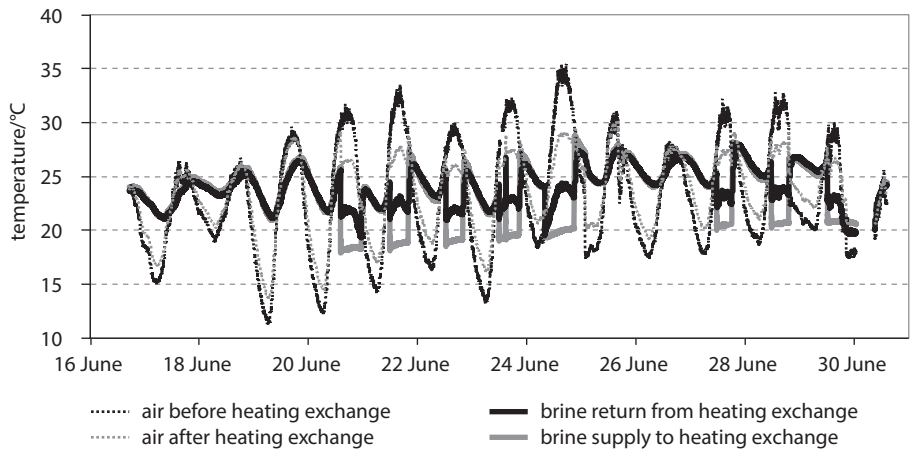


Figure 6.15 Temperature levels of geothermal heat exchanger to mechanical ventilation system.

The pressure drop due to the brine-air heat exchanger amounts to only 12 Pa. The installed fan needs an electrical power of 30 W to overcome this drop, whereas the brine pump consumes about 60 W. This results in maximum coefficients of performance of 40 and an average COP of 18.4 (see Figure 6.14). It can be seen that a phase of heating occurred during the last 2 days due to the relatively warm soil temperature. This high soil temperature is mainly due to the shallow depth of the pipes.

The ambient air can be cooled down by as much as 7 K in the heat exchanger. However, supply air to the building is still at 28°C during midday. The soil temperature rises up to nearly 20°C at the end of June (see Figure 6.15). The logarithmic average temperature difference of brine and air across the heat exchanger is about 6.3 K.

The power rejected to the ground increases with increasing ambient air temperature, but at a much lower slope as for the direct heat exchange of ambient air with the ground.

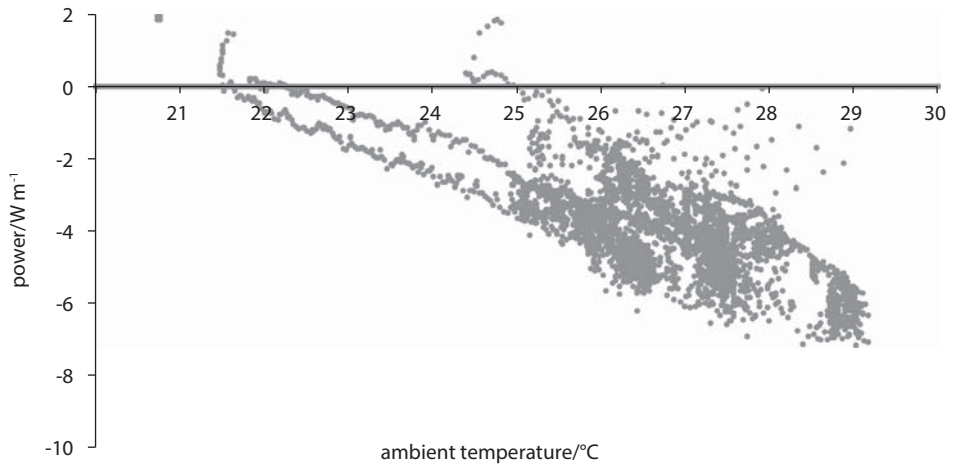


Figure 6.16 Heat rejected to the ground as a function of ambient air temperature for the horizontal heat exchanger.

Again, the correlation is better between inlet temperature to the ground and power delivered to the geothermal heat sink. The range of inlet temperature levels is quite narrow between 20 and 24°C (see Figure 6.17).

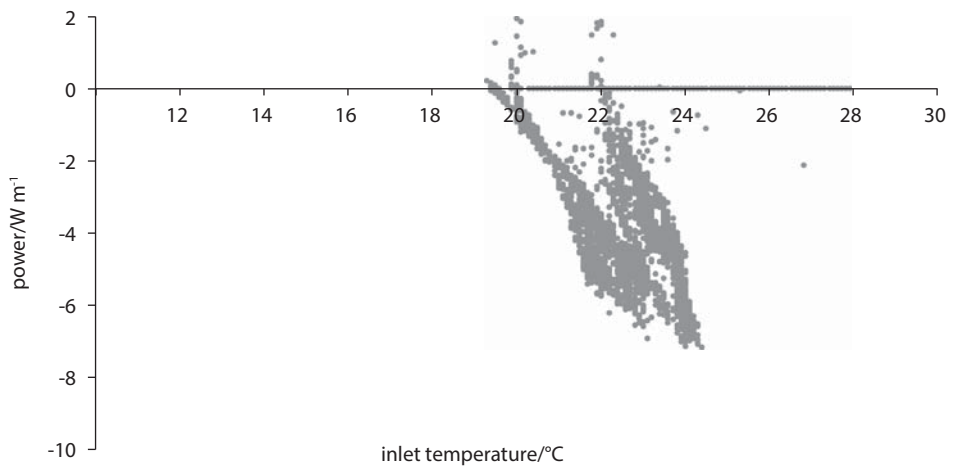


Figure 6.17 Heat rejection to the shallow horizontal geothermal heat exchangers as a function of brine inlet temperature level.

6.2 Indirect geothermal energy use

Heat pump applications

A low-energy building with geothermal energy supply has been designed and implemented in the Southern German City of Ostfildern. The building was constructed in 2006 with a gross surface area of 475 m² and is used as a Youth Centre. The level of insulation of the building envelope almost reaches passive house standard.

The heating supply is based on a heat pump in connection with a geothermal heat exchanger. Two boreholes with 134 m depth were drilled and polyethylene U-tubes with a diameter of 1½" were installed. The double-U-borehole heat exchangers are filled with a heat transfer medium (monoethylen-glycol-water mixture, concentration ≤ 25%), which is circulated with a high-efficiency pump with a measured electrical power of 300 W.

The geothermal circuit is connected to the heat pump (Vitocal 300 Type BW from Viessmann) with a maximum output power of 16.6 kW. The thermal energy goes into a storage tank where temperatures up to +35°C are provided for the floor heating and the ventilation.

The Youth Centre has excellent low heating consumption values of around 6700 - 7000 kWh a⁻¹, which corresponds to 14 kWh m⁻² a⁻¹ (2008), 15 kWh m⁻² a⁻¹ (2009) and 20 kWh m⁻² a⁻¹ (2010). The heating season of this energy efficient building is short and when the average ambient temperature rises above 5°C, the heat consumption is very low. The gap in the data, at the end of January, is due to a power failure-related data-recording interruption.

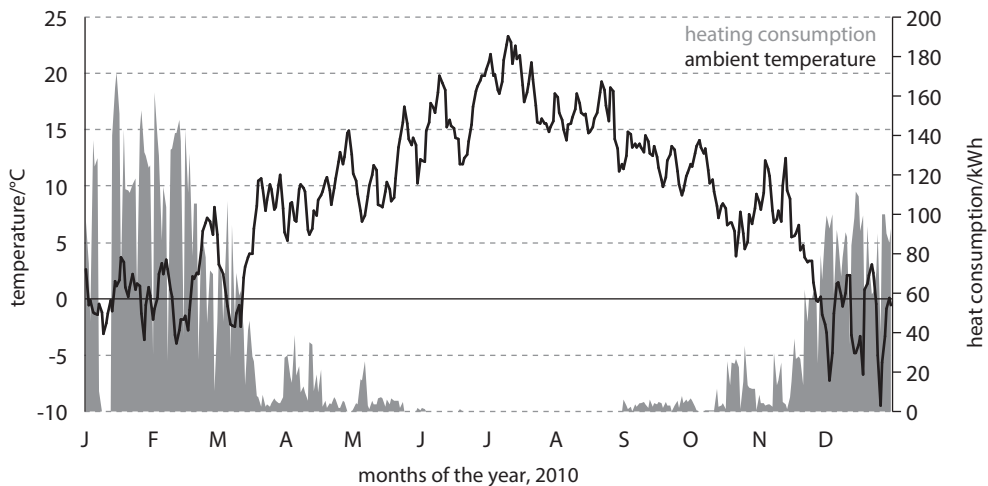


Figure 6.18 Monitoring results for heating of the low-energy building.

The monthly seasonal performance factor (SPF) of the heat pump was determined from measurement data taken in 2010. The SPF is a quality criterion of a heat pump and describes the ratio of benefits (heating energy) to effort (electrical input energy). The pure SPF of the heat pump (without auxiliary power), as it is often measured by the manufacturers for specific operating points, is 3.94. This value is significantly higher than the SPF of the whole heat pump system at 2.63, which includes the electric power for the geothermal brine pump.

From the analysis it could be seen that there is a significant potential in reducing the running times of the brine pump. Especially in the transitional period (March, April, May, September, October), the supply pump of the borehole is nearly always in use, although the compressor of the heat pump is not working and therefore there is no need to supply the heat pump. With proper control of the brine pump based on the compressor running times, a much higher SPF for the heat pump system can be achieved.

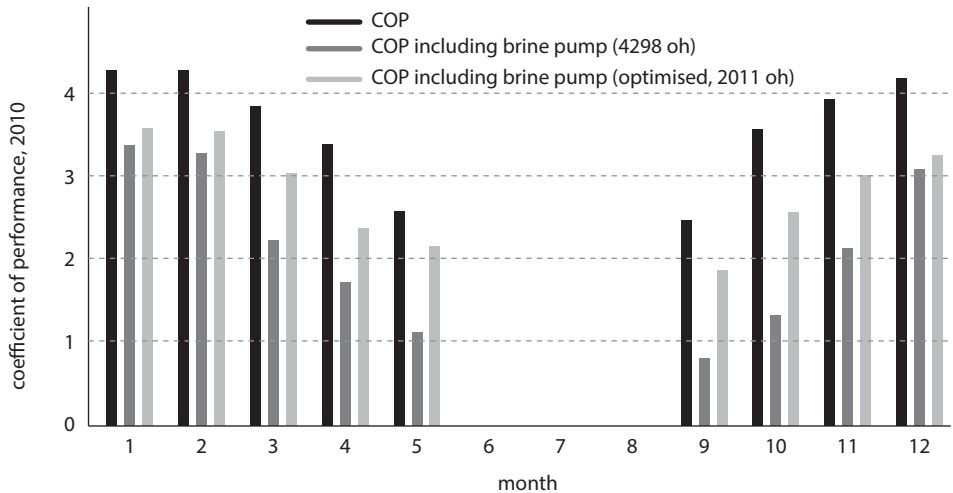


Figure 6.19 Heat pump seasonal performance factor with brine pump electricity (oh = operating hours).

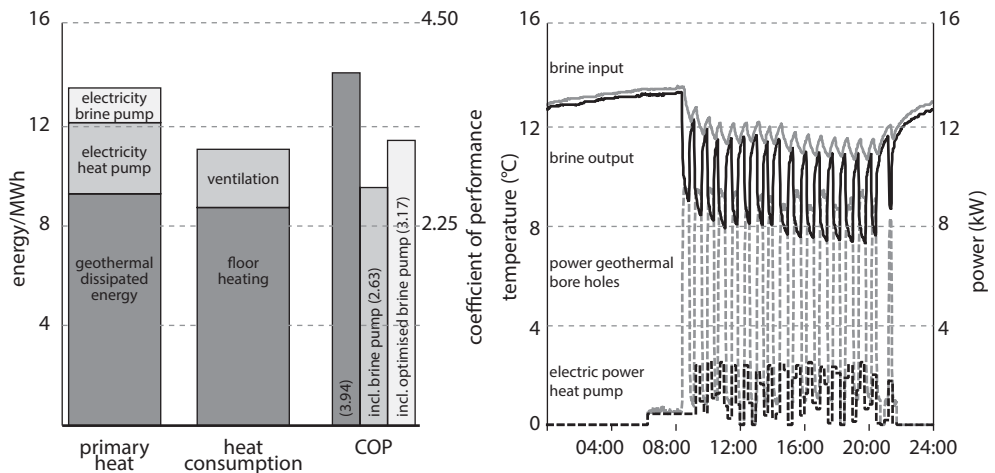


Figure 6.20 Energy consumption of heat pump system with energy delivered to the building and seasonal performance factor (left), temperature levels of geothermal heat exchanger together with power extracted and electrical power consumption of the heat pump (right).

This can also be seen in the annual performance factor for the optimised brine pump (see Figure 6.20). An optimised operation of the heat pump system, for which the operating hours of the brine pump are significantly reduced, results in an annual SPF of 3.17. The energy input includes the energy on the primary side, which is obtained from the boreholes as geothermal energy and the electrical energy of the compressor. In addition, the amount of electric energy for the brine pump is shown. The brine pump energy was calculated from the measured operating time and the connection power of the brine pump.

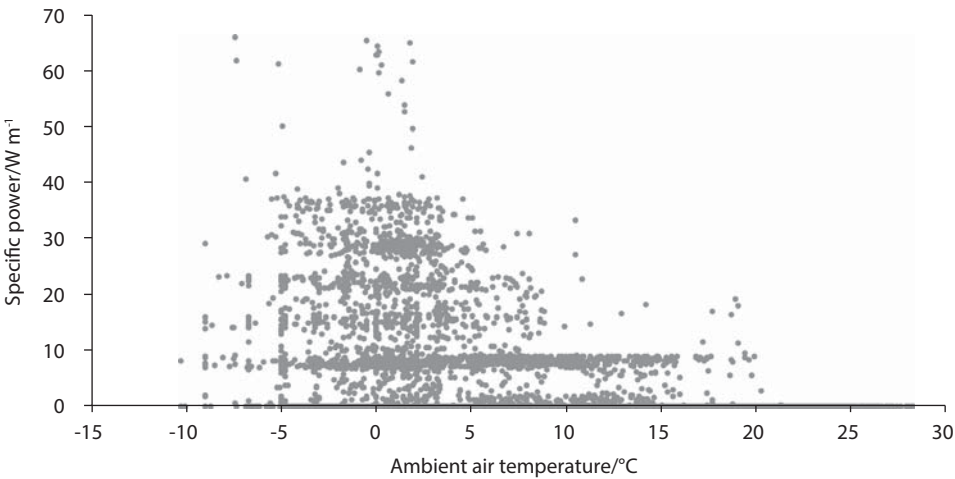


Figure 6.21 Power delivered by the geothermal heat exchangers as a function of ambient air.

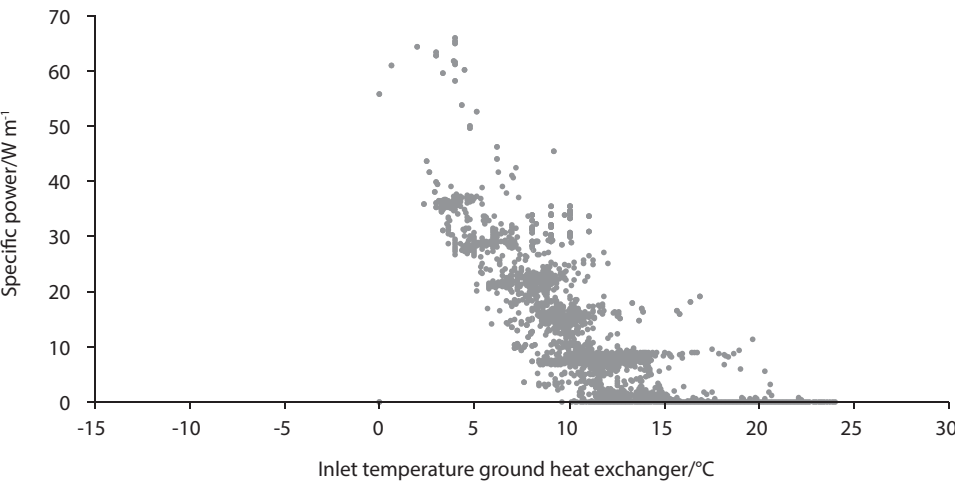


Figure 6.22 Power delivered as a function of brine inlet temperature to the geothermal heat exchanger.

The specific extraction power of the borehole for the year 2010, depending on outside temperature, typically reaches up to about 40 W m^{-1} (see Figure 6.21). The scatter plot shows that brine power over 40 W m^{-1} , corresponding to an absolute power of 10 kW (with a tube length of 268 m), is exceeded only rarely.

The correlation of specific power and temperature is better when related to the inlet brine temperature to the ground heat exchanger (see Figure 6.22). It is clearly the temperature difference between brine and soil which determines the power that can be transferred.

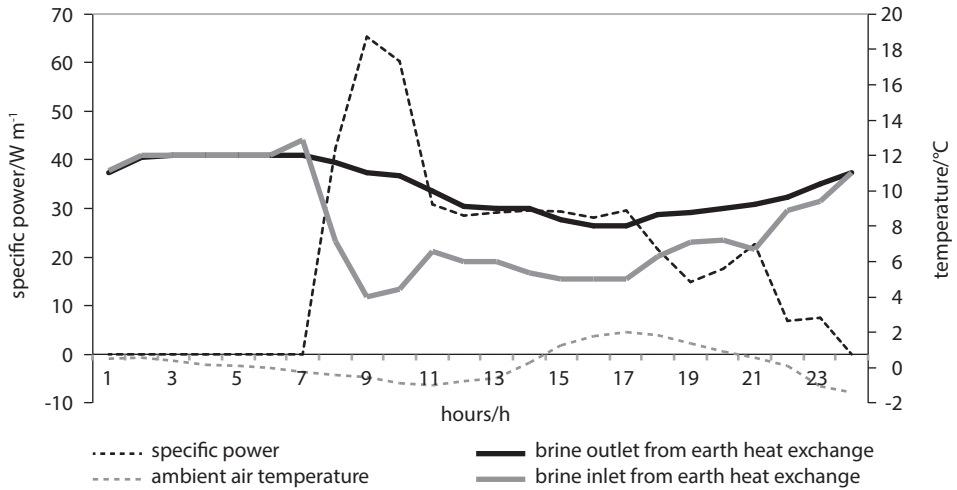


Figure 6.23 Time dependence of geothermal power extracted with heat pump use.

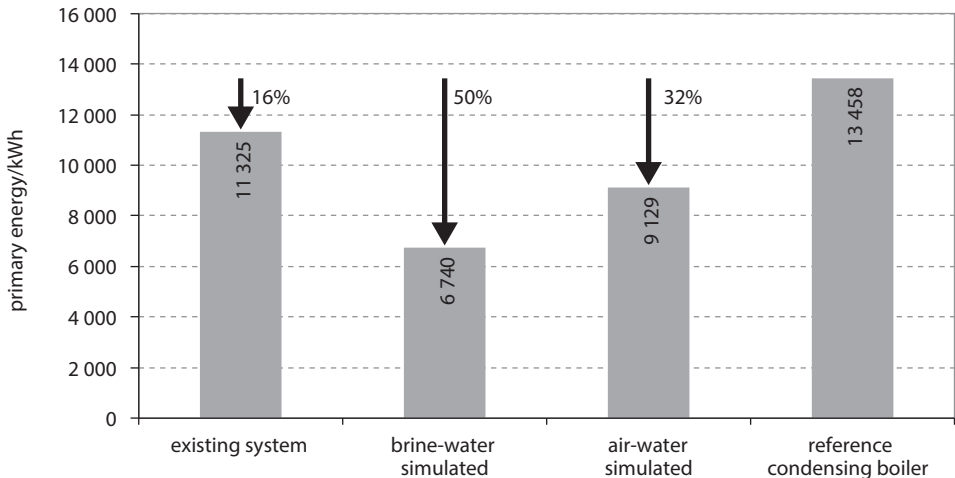


Figure 6.24 Measured and simulated primary energy performance for the existing system, a reference condensing boiler and optimised heat pumps with air or brine as input.

The temperature difference between supply and returns is mostly about 4 - 5 K (see Figure 6.20). A larger temperature difference is possible, which occasionally occurs with power levels up to 66 W m^{-1} , corresponding to an absolute power of about 18 kW.

The few performance peaks of the borehole occur mainly in the morning, when the heat pump turns on for the first time at full power (see Figure 6.23).

The total primary energy consumption of the existing system has been compared with a reference condensing boiler system. Full system simulations were carried out to simulate the performance for an optimised system with reduced brine pump operation and for an air/water heat pump. All heat pump systems perform better than the condensing boiler. In the best case, 50% primary energy can be saved.

6.3 Geothermal heat exchangers for chiller heat rejection

Efficient heat rejection is crucial for the overall primary energy balance of cooling systems, as the heat rejection system dominates the auxiliary electrical energy consumption. High electricity consumption in combination with low solar fractions are detrimental to the environmental and economic performance of solar absorption cooling systems with high heat rejection power levels, especially if they are compared with highly efficient electrical-driven compression chillers.

Geothermal heat exchangers are an interesting option for heat rejection, as the electricity needs for cooling tower fans, which are typically between 17% and 30% of the total auxiliary electricity consumption, are saved. Only the circulation pump of the geothermal heat exchanger needs to be considered. Another advantage of geothermal heat exchangers is the parallel connection of many tubes, which leads to low pressure drops and thus lower electricity consumption of the heat rejection circulation pump.

A study was carried out for two designs of geothermal heat exchangers, which had to reject a maximum power of 35 kW: either 10 double U-tubes with 120 m depth each, which corresponds to 30 W m^{-1} heat rejection or a smaller system with 7 double U tubes at 42 W m^{-1} heat rejection, i.e. 840 - 1200 m total length. The geothermal heat rejection system was compared to wet and dry cooling towers with the same maximum heat rejection design power.

The tubes have 32 mm external diameter and they are all in parallel, so that flow velocities are very small: for the 7 tube system, the flow velocity at $5 \text{ m}^3 \text{ h}^{-1}$ nominal volume flow is only 0.19 m s^{-1} , for the 10 tube system only 0.13 m s^{-1} . The resulting Reynolds numbers are 1770 and 1239 and the pressure drop is 98 Pa m^{-1} for the 7 tube and 68 Pa m^{-1} for the 10 tube system, resulting in a total pressure drop of $0.82 \times 10^5 \text{ Pa}$ for both systems.

This is lower than the pressure drop of a circulation pump for a wet cooling tower design with 35 kW at 6 K temperature difference ($36^\circ\text{C}/30^\circ\text{C}$) and a flow rate of $5 \text{ m}^3 \text{ h}^{-1}$. Here the pressure drop is $1.2 \times 10^5 \text{ Pa}$, which requires a pump of 550 W at 30% pump efficiency.

As the ground temperature level is usually much lower than the ambient or wet bulb air temperature in summer, the temperature differences between inlet and outlet can be significantly increased and the flow rates further reduced. At 12 K temperature difference, the pressure drop is only $0.41 \times 10^5 \text{ Pa}$.

To account for part load conditions with less heat rejection, it is recommendable to frequency control the circulation pump, i.e. to lower the mass flow rates. The average mass flow rate reduces by more than 50% and the total pressure drop is only $0.17 \times 10^5 \text{ Pa}$ for the best case. The boundary conditions of all cases are summarised in Table 6.3.

Inlet/outlet temperature difference			Standard (6K)				Large (12K)			
Length of vertical heat exchanger			1200 m		840 m		1200 m		840 m	
Heat transfer to ground at design conditions			30 W m ⁻¹		42 W m ⁻¹		30 W m ⁻¹		42 W m ⁻¹	
Mass flow*			con	var	con	var	con	var	con	var
Scenario number			1.1.1	1.1.2	1.2.1	1.2.2	2.1.1	2.1.2	2.2.1	2.2.2
Reynolds number	Max		1239	1493	1770	2133	620	746	885	1066
	Average		1239	524	1770	749	620	262	885	375
Velocity of fluid		/m s ⁻¹	0.13	0.06	0.19	0.08	0.07	0.03	0.08	0.04
Pressure drop of pipes	Total	/10 ⁵ Pa	0.82	0.35	0.82	0.35	0.41	0.17	0.41	0.17
	Specific	/Pa m ⁻¹	68	29	98	41	34	14	19	21
$Q_{el} \text{ pump}/P_{th}$ heat rejection		/kWh kW ⁻¹	10.6	4.1	10.6	4.1	3.7	2.4	3.7	2.4
$Q_{el} \text{ pump}/P_{th}$ cooling power		/kWh kW ⁻¹	20.6	8.0	20.6	8.0	7.2	4.7	7.2	4.7

*) con = constant, var = variable

Table 6.3 Summary of boundary conditions for geothermal heat exchangers and electrical energy requirement for the circulation pump.

To check whether the ground heat exchangers can deliver sufficiently low temperatures for heat rejection, even when the shortest version is chosen with 42 W m⁻¹ design power, the outlet temperatures have been simulated for variable mass flow and a large temperature difference of 12 K. Even at the end of summer the outlet temperature is still very low at 13°C, which the rejected heat then heats up to 26°C maximum (see Figure 6.25). The heat rejection time series was calculated for the cooling requirement of a small office building in Southern Germany with a rather low cooling energy demand of 16 kWh m⁻² a⁻¹.

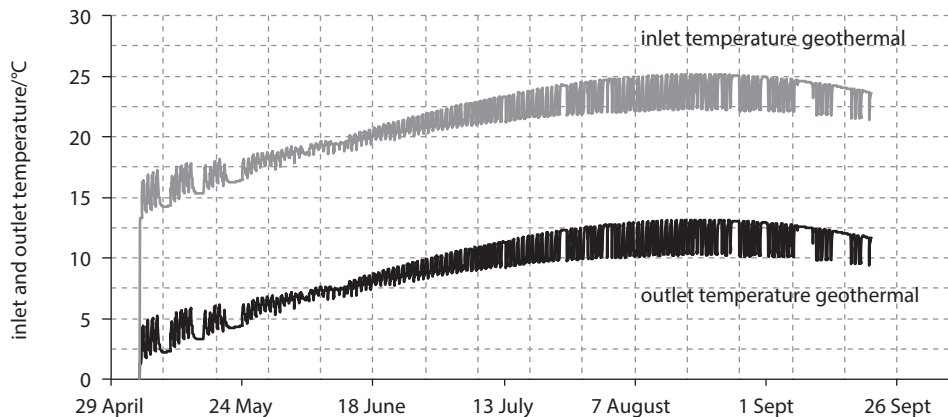


Figure 6.25 Inlet and outlet temperature of vertical geothermal heat exchanger with heat rejection from an absorption cooling system at 12 K temperature difference and variable mass flow.

This shows that even less tubes would be sufficient for heat rejection. However, the request of standards such as VDI 4640 would be violated, which states that the inlet temperatures to the ground should not be more than 11 K higher than the soil temperature.

The ground temperature levels are not significantly influenced by the heat rejection and increase by less than 1 K below 15 m (see Figure 6.26).

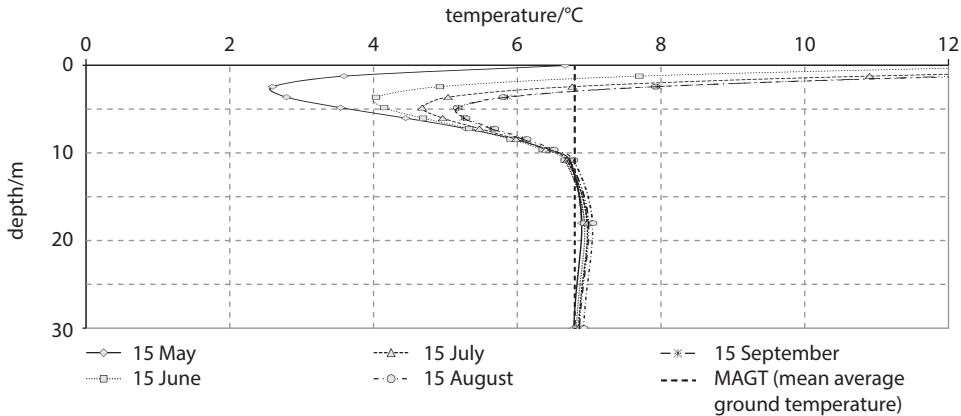


Figure 6.26 Development of ground temperature levels 200 cm from the geothermal tubes for maximum heat rejection.

The primary energy ratio of the solar absorption cooling system improves by at least 13% using geothermal heat rejection, even for small temperature differences and constant mass flow, as the electricity need for the cooling tower is no longer needed. In addition, a large reduction of pumping power is possible, if the temperature difference chosen is larger (12 K) and the mass flow is adjusted to the heat rejection power.

The total summer electricity consumption for the geothermal circulation pump is between 2.4 and 10.6 kWh per kW nominal heat rejection power. For the best case this is about one-third of the consumption for the wet cooling tower. The total savings in auxiliary electricity are about 30%.

6.4 Modeling of geothermal heat exchangers

Modeling thermal response

The main task of the design process is to determine the required length of tubing for the required thermal energy to be extracted or rejected. Modeling the thermal response of the surrounding soil requires information about the ground temperature distribution, the moisture content, groundwater movement, freezing or thawing of the soil and the geometry of the heat exchanger. Furthermore, experimental results showed that the temperature distribution around soil tubes is usually not symmetric. The soil surrounding the borehole is often considered as homogeneous with a mean thermal conductivity λ and mean diffusivity $a = \lambda / \rho c$. Simulation models are typically separated in “inner” solutions for the heat transfer between the fluid and the perimeter of the borehole including the mutual influence of the U-tubes and the outer region between the edge of the borehole and the ground.

The simplest model for the inner borehole is to calculate the heat flux per unit length of the borehole q_b from the resistance R_b , the mean fluid temperature $T_f(t)$ and a mean borehole temperature $T_b(t)$. The thermal resistance contains the convective resistance between fluid and pipe wall, the conductive resistance of the wall and borehole filling material.

$$q_b = \frac{T_f(t) - T_b(t)}{R_b} \quad (6.1)$$

As the steepest temperature gradients occur at the pipe to soil or backfill material interface, several authors developed detailed models for the near temperature field. Zeng and others (2003) developed analytical models for a range of tube configurations within the borehole. Piechowski (1999) solved heat and moisture transport equations for horizontal pipes embedded in soil. Piechowski states that the soil temperature around a pipe drops by 30 - 40% within a few centimetres, indicating the influence of precise near field models.

To obtain the mean temperature of the borehole and to calculate the complete soil temperature field $T(r, z, t)$, the heat conduction equation has to be solved as a function of time t , depth z and distance from the borehole r .

$$\frac{1}{\alpha} \frac{\delta T}{\delta t} = \frac{\delta^2 T}{\delta z^2} + \frac{\delta^2 T}{\delta r^2} + \frac{1}{r} \frac{\delta T}{\delta r} \quad (6.2)$$

The height dependence z of the temperature is often ignored and the simplest solution is obtained, if a step function constant heat input q is applied at the origin $r = 0$. The solution is known as Kelvin's infinite line source theory and is used to analyse thermal response test data.

The temperature at the boundary of the borehole is given by

$$T(r_b, t) - T(r_b, t=0) = \frac{q}{4\pi\lambda} \int_{\frac{r_b^2}{4\alpha t}}^{\infty} \frac{e^{-u}}{u} du = \frac{q}{4\pi\lambda} \left(\ln \left(\frac{4\alpha t}{r_b^2} \right) - \gamma \right) \quad (6.3)$$

γ is Euler's number (0.5772), r_b is the borehole radius and q is the constant heat injection rate per metre of borehole. The solution of the exponential integral is an approximation, with a maximum error of 10% for

$$\frac{\alpha t}{r_b^2} \geq 5$$

and 2.5% for

$$\frac{\alpha t}{r_b^2} \geq 20$$

(Gehlin, 2002). For typical boreholes, an error of 2.5% is reached between 10 and 20 hours.

The equation can be also used to calculate the temperature field at other distances r from the borehole. The accuracy of the solution thus increases when the thermal front extends, which is faster for high thermal diffusivities, i.e. high thermal conductance and low heat capacity of the soil. The solution assumes a constant temperature along the axis and an infinite length of the borehole. As the axial temperature change is much smaller than the radial temperature gradients and the length of the boreholes much larger than the radius, the errors introduced are small.

The borehole thermal resistance causes a temperature difference between the mean fluid in the geothermal heat exchanger and the borehole wall. The fluid temperature field with the additional temperature drop across the resistance can then be calculated using the undisturbed soil temperature as a starting value for $T_f(t=0) = T_{undist}$.

$$T_f(t) - T_f(t=0) = \frac{q}{4\pi\lambda} \left(\ln \left(\frac{4at}{r_b^2} \right) - \gamma \right) + qR_b \quad (6.4)$$

The measured mean fluid temperature as the arithmetic average between inlet and outlet temperature is then plotted against the logarithm of time to obtain the mean conductivity of the soil.

$$T_f(t) - T_{undist} = \frac{q}{4\pi\lambda} \ln(t) + q \left(\frac{1}{4\pi\lambda} \left(\ln \left(\frac{4a}{r_b^2} \right) - \gamma \right) + R_b \right) \quad (6.5)$$

$$T_f(t) = k \ln(t) + m \quad (6.6)$$

The effective soil conductivity is calculated from the slope k as follows:

$$\lambda = \frac{q}{4\pi k} = \frac{Q}{H} \frac{1}{4\pi k} \quad (6.7)$$

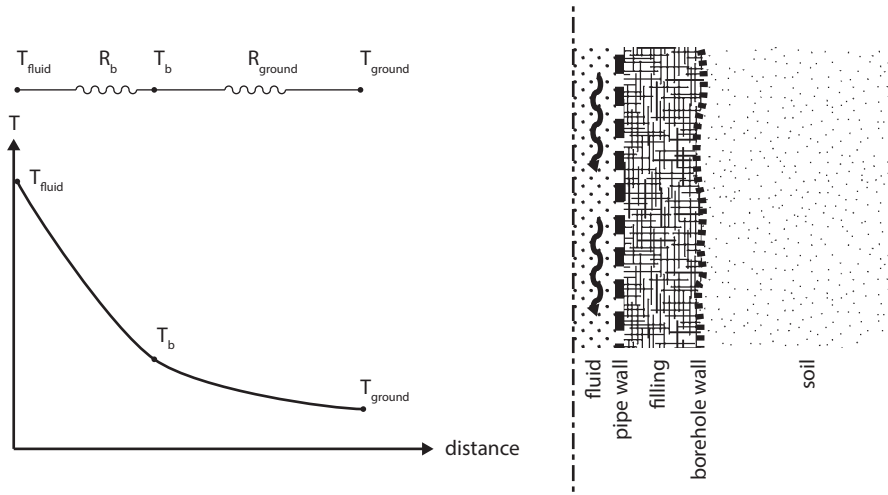


Figure 6.27 Temperature levels and resistances of borehole heat exchangers.

Knowing the thermal conductivity and the heat injected per metre of borehole depth H , the borehole resistance can then be calculated using Equation (6.4).

$$R_b = \frac{T_f(t) - T_{undist}}{q} - \frac{1}{4\pi\lambda} \left(\ln \left(\frac{4at}{r_b^2} \right) - \gamma \right) \quad (6.8)$$

Measured typical borehole resistances are around 0.1 m K W^{-1} for double U-tubes and up to 0.2 m K W^{-1} for single U-tubes in bentonite, where no special precautions are taken to keep the tubes close to the borehole wall (Gehlin, 2002).

Example 6.1

Calculation of the effective heat conductivity and borehole resistance from measurements of a vertical heat exchanger in Stuttgart, Germany. The double U-tubes are 80 m deep and are inserted in a 0.32 m borehole. The flow rate was 17 litres per minute and the injected power on average 3.4 kW. The undisturbed initial temperature of the ground was 12.5°C .

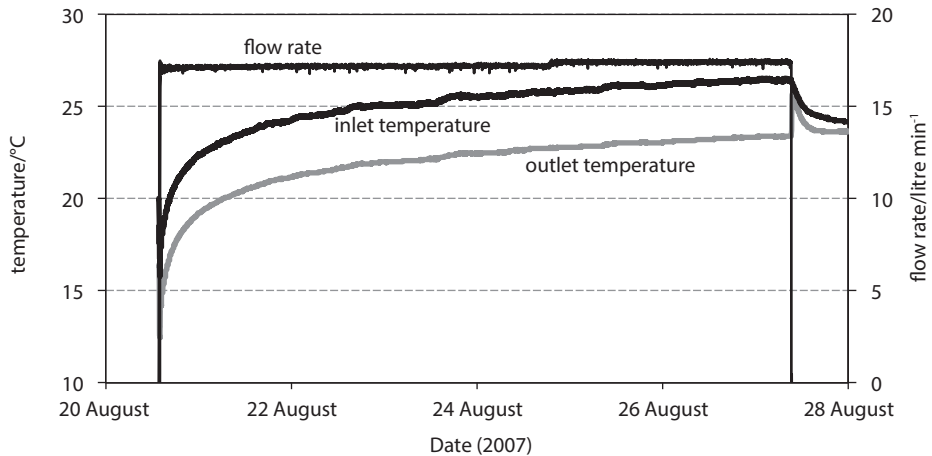


Figure 6.28 Measured temperature data and flow rates.

The measured temperature data are then plotted against the natural logarithm of time in seconds and the slope of the mean temperature k is determined, which in this case is 1.5352.

As a result the effective heat conductivity is $2.2 \text{ W m}^{-1} \text{ K}^{-1}$.

$$\lambda = \frac{3400 \text{ W}}{80 \text{ m}} \frac{1}{4\pi \cdot 1.5352 \text{ K}} = 2.2 \frac{\text{W}}{\text{m K}}$$

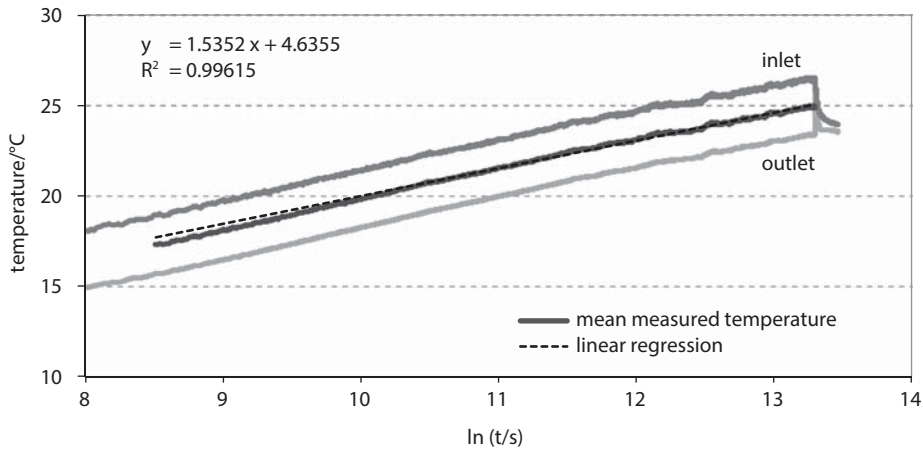


Figure 6.29 Measured inlet and outlet temperatures of a vertical geothermal heat exchanger during 6 days as a function of the natural logarithm of time.

The thermal diffusivity $a = \lambda / \rho c$ for an estimated heat capacity of $2.5 \text{ MJ m}^{-3} \text{ K}^{-1}$ is $8.8 \times 10^{-7} \text{ m}^2 \text{ s}^{-1}$. The resulting borehole resistance at the end of the response test (at 561720 seconds after the start) with a mean fluid temperature of 25°C is 0.71 m K W^{-1} .

$$R_b = \frac{25 - 12.5}{42.5} - \frac{1}{4\pi \cdot 2.2} \left(\ln \left(\frac{4 \times 8.8 \times 10^{-7} \times 561720}{0.16^2} \right) - 0.57722 \right) = 0.71$$

If the heat flux is applied at the perimeter of the borehole r_b , the solution to the so-called cylindrical heat source method is given by Carslaw and Jaeger (1947). Both solutions are similar for $a t / r_b^2 > 20$. For a sandy soil with a thermal diffusivity of $4.74 \times 10^{-7} \text{ m}^2 \text{ s}^{-1}$ and a 0.1 m radius, this corresponds to nearly 5 days. However, neither the line source nor the cylindrical source methods saturate at large timescales, which is a major drawback. Also parameters like heat capacity or ground water flow cannot be quantified.

An extension of the line source approach has been proposed by Cleasson and Eskilson (1987). This includes the height dependence of the temperature field. The integrals in the solution of the temperature field are called g-functions, which have been mostly computed numerically and then used in tabulated form. Recently, analytical solutions have been developed for the g-functions by Lamarche and Beauchamp (2007). If the g-function is known, the temperature at the perimeter of the borehole can be easily calculated.

$$T_b(t) - T(t=0) = \frac{q}{2\pi\lambda} g\left(\frac{t}{t_s}, \frac{r_b}{H}\right)$$

with

$$t_s = \frac{H^2}{9\alpha}$$

and H as the depth of the borehole. t_s is the time, when the transient process ends and the temperature field becomes stationary. The power injected into the borehole then equals the heat given off to the atmosphere. The timescale is very long: for the sandy soil described above and a 100 m deep borehole, it takes 74 years to reach steady-state conditions at constant heat injection rate.

Using the g function approach, Lamarche and Beauchamp calculated the time, when the height dependence (so-called axial effect) becomes important for a 100 m borehole length as $t \sim t_s/20$. For the wet sandy soil described above, this takes 3.7 years. Thermal interference between two boreholes separated by 5 m becomes relevant for $t > t_s/150$, that is, 0.5 years.

Numerical models

Numerical models offer more flexibility in the temperature field calculation for arbitrary geometries and time-varying heat fluxes and inlet temperature levels.

Numerical heat transfer models for geothermal heat exchangers were developed and implemented in the simulation environment INSEL (www.insel.eu). They are all based on the heat conduction equation with explicit finite difference solutions. The simplest vertical heat exchanger model uses a rectangular discretisation geometry (see Figure 6.30). Horizontally, 9 zones have been defined and vertically, 30 elements are calculated for each time step with only one-dimensional heat transfer considered (see Figure 6.30). To determine the mutual influence of a field of heat exchangers, a three-dimensional model was developed with a parallelogram geometry for the discretisation (Figure 6.31).

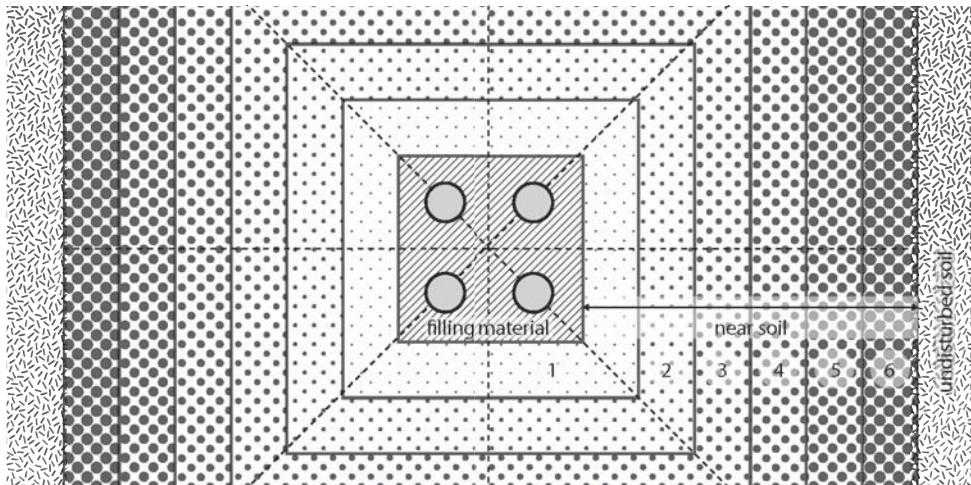


Figure 6.30 Rectangular discretisation of geothermal heat exchanger model.

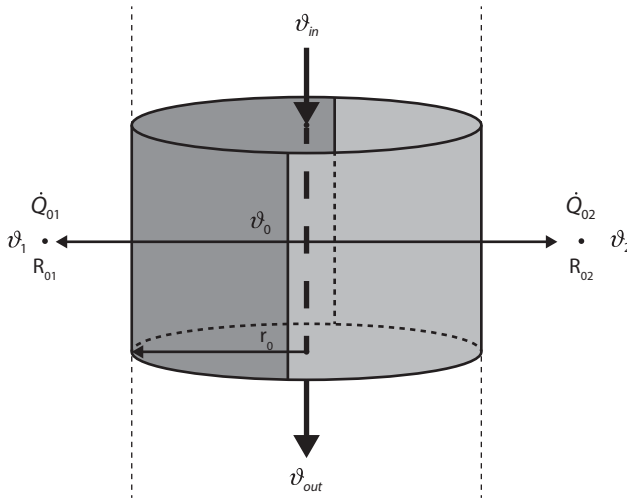


Figure 6.31 Temperatures and heat fluxes from fluid to filling material.

The mean fluid temperature ϑ_0 of each segment is assumed to be the mean value between inlet ϑ_{in} and outlet temperature ϑ_{out} .

$$\vartheta_0 = \frac{\vartheta_{in} + \vartheta_{out}}{2} \quad (6.9)$$

The volume of the fluid element is obtained from the radius of the tube r_0 [m] and the height of the volume element l_0 [m]

$$V_0 = \pi r_0^2 l_0 \quad (6.10)$$

The heat flux between the fluid and the adjacent external concrete filling $\dot{Q}_{01/02}$ is calculated from the thermal resistance $R_{01/02}$ between fluid and filling material [$\text{m}^2 \text{K W}^{-1}$], the contact surface area $A_{01/02}$ and the temperature levels of the filling material $\vartheta_{1/2}$.

$$\begin{aligned} \dot{Q}_{01} &= \frac{A_{01}}{R_{01}} (\vartheta_1 - \vartheta_0) \\ \dot{Q}_{02} &= \frac{A_{02}}{R_{02}} (\vartheta_2 - \vartheta_0) \end{aligned} \quad (6.11)$$

The transferred heating or cooling power from the fluid $\dot{Q}_0$ can then be calculated.

$$\dot{Q}_0 = \frac{V_0 \rho_0 c_{p0}}{\Delta t} (\vartheta_{in} - \vartheta_{out}) \quad (6.12)$$

with

ρ_0	fluid density [kg m^{-3}]
c_{p0}	specific heat capacity of the fluid [$\text{J kg}^{-1} \text{K}^{-1}$]
Δt	time [s]

The energy balance for a volume element is then given by

$$\dot{Q}_0 + \dot{Q}_{01} + \dot{Q}_{02} = 0 \quad (6.13)$$

where

$$\frac{A_{01}}{R_{01}} \left(\vartheta_1 - \frac{\vartheta_{in}}{2} - \frac{\vartheta_{out}}{2} \right) + \frac{A_{02}}{R_{02}} \left(\vartheta_2 - \frac{\vartheta_{in}}{2} - \frac{\vartheta_{out}}{2} \right) + \frac{V_0 \rho_0 c_{p0}}{\Delta t} (\vartheta_{in} - \vartheta_{out}) = 0$$

Given temperatures are ϑ_1 , ϑ_2 , ϑ_{in} . The unknown temperature is ϑ_{out} .

$$\left(\frac{A_{01}}{2R_{01}} + \frac{A_{02}}{2R_{02}} + \frac{V_0 \rho_0 c_{p0}}{\Delta t} \right) \vartheta_{out} = \frac{A_{01}}{2R_{01}} \vartheta_1 + \frac{A_{02}}{2R_{02}} \vartheta_2 - \left(\frac{A_{01}}{2R_{01}} + \frac{A_{02}}{2R_{02}} + \frac{V_0 \rho_0 c_{p0}}{\Delta t} \right) \vartheta_{in}$$

$$a = \frac{A_{01}}{R_{01}} \left(\frac{A_{01}}{2R_{01}} + \frac{A_{02}}{2R_{02}} + \frac{V_0 \rho_0 c_{p0}}{\Delta t} \right)^{-1}$$

$$b = \frac{A_{02}}{R_{02}} \left(\frac{A_{01}}{2R_{01}} + \frac{A_{02}}{2R_{02}} + \frac{V_0 \rho_0 c_{p0}}{\Delta t} \right)^{-1}$$

$$c = \left(\frac{A_{01}}{2R_{01}} + \frac{A_{02}}{2R_{02}} - \frac{V_0 \rho_0 c_{p0}}{\Delta t} \right) \left(\frac{A_{01}}{2R_{01}} + \frac{A_{02}}{2R_{02}} + \frac{V_0 \rho_0 c_{p0}}{\Delta t} \right)^{-1}$$

$$\vartheta_{out} = a\vartheta_1 + b\vartheta_2 - c\vartheta_{in} \quad (6.14)$$

The difference between simulated and measured temperatures are slightly higher in summer, when the seminar ventilation system is switched off and only low volume flows are used for the cooling of an activated floor in the foyer (see Figure 6.32, summer).

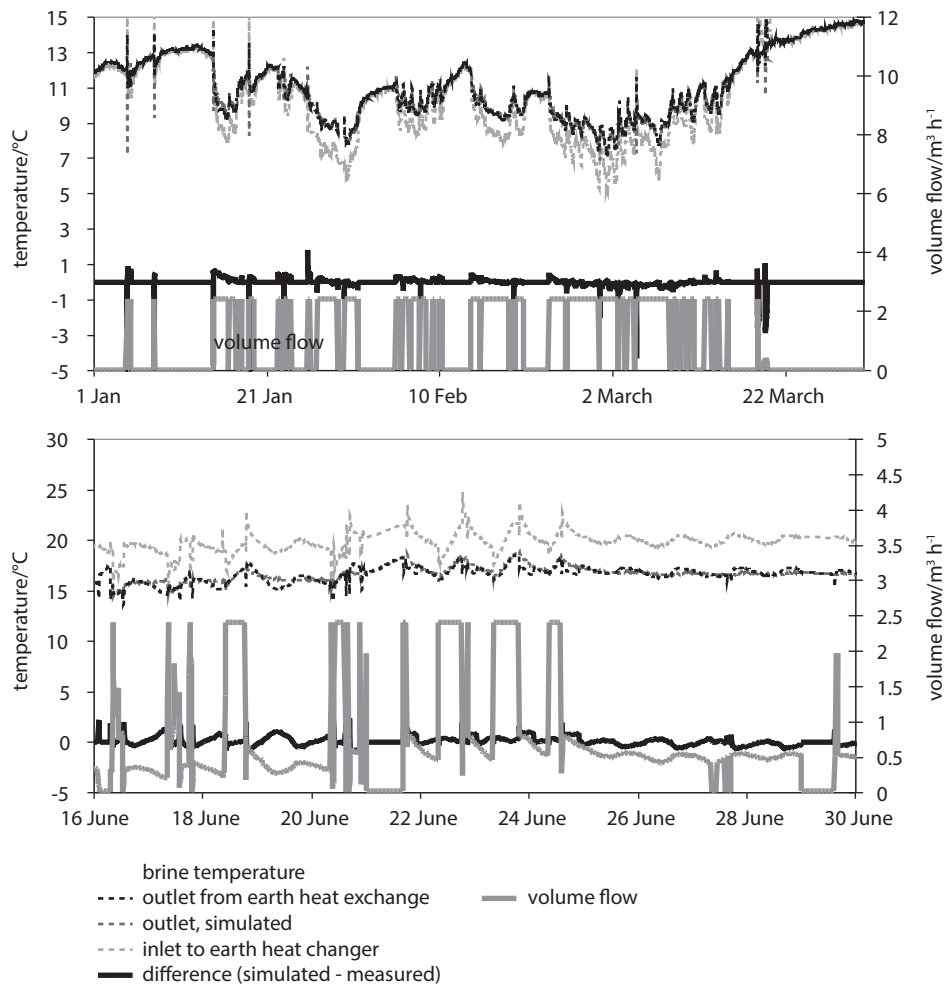


Figure 6.32 Measured brine inlet and measured and simulated outlet temperatures of the ground heat exchanger in the Freiburg SIC building in 2005, winter (top) and summer (bottom).

The three-dimensional validated model was used for parameter studies concerning the backfill and soil heat conductivity, mass flow rates and distance between the geothermal heat exchanger. The boundary conditions are those of the SIC building in Freiburg. The soil heat conductivities range from $0.5 \text{ W m}^{-1} \text{ K}^{-1}$, representing dry clay, to $2.5 \text{ W m}^{-1} \text{ K}^{-1}$ for water-saturated sand. As can be seen from Figure 6.33, the soil heat conductivity has a major influence on the cooling performance. Compared to the standard moraine soil, the energy output is lessened by 39% if the underground consists of dry clay, verifying the statements of Sanner and Rybach (1997), who show that the specific energy extraction rate increases with increasing soil heat conductivity. Zhang and Murphy (2003) support this finding but stress that the effectivity of thermal storage may decrease.

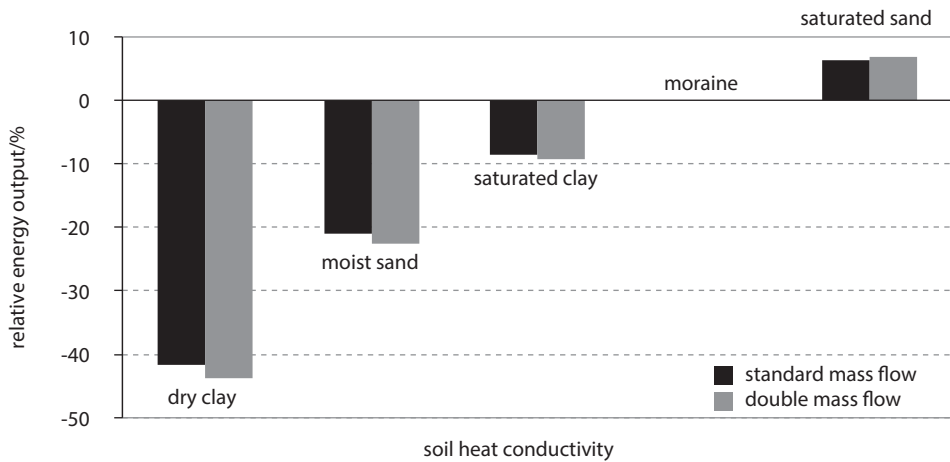


Figure 6.33 Influence of soil conductivity on heat rejection capacity of vertical heat exchangers.

Borehole backfill heat conductivities of 0.8 (light concrete), 1.6 (Bentonite) and 3.2 $\text{W m}^{-1} \text{K}^{-1}$ (high performance backfill) were investigated in the next step. Although light concrete is rarely used as backfill material, it was included in this study for comparison. From Figure 6.34 it can be seen that the effect on the energy output is relevant. However, it can be expected that the influence decreases if the heat conductivity of the surrounding soil is low.

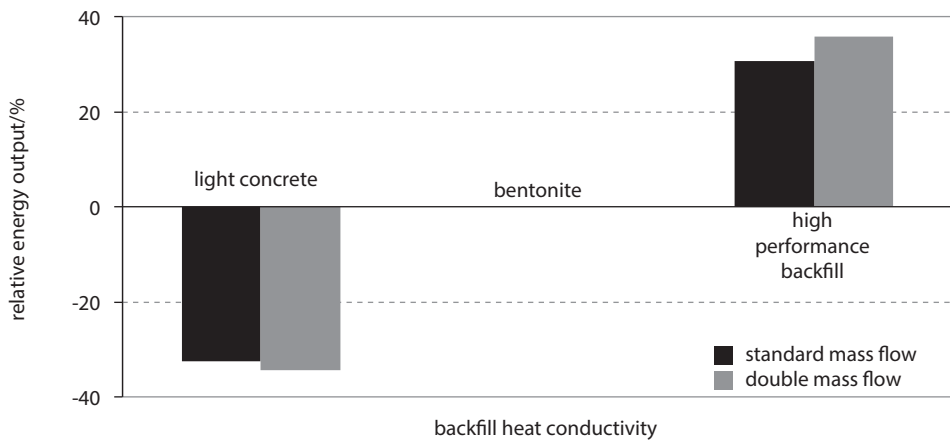


Figure 6.34 Influence of backfill heat conductivity on the heat exchange performance.

Thermal response tests have been conducted at the Stuttgart University of Applied Sciences to determine the effective heat conductivity for different backfill materials. Two borehole heat exchangers with lengths of 80 m were built, one with standard backfill with a heat conductivity of 1.6 $\text{W m}^{-1} \text{K}^{-1}$ and the other with Stuewatherm backfill with a heat conductivity of 2.0 $\text{W m}^{-1} \text{K}^{-1}$. The borehole heat exchangers are tested with a thermal power of 3 kW. The heat conductivities of

the complete system (tube, backfill and surrounding soil), as derived from the thermal response tests, are $1.94 \text{ W m}^{-1} \text{ K}^{-1}$ and $2.17 \text{ W m}^{-1} \text{ K}^{-1}$, respectively. The use of high performance backfill leads to a 12% higher total heat conductivity. The thermal response tests were also used to validate the before mentioned numerical model and showed good agreement. Figure 6.35 shows the measured and simulated mean values of input and output fluid temperatures of both heat exchangers.

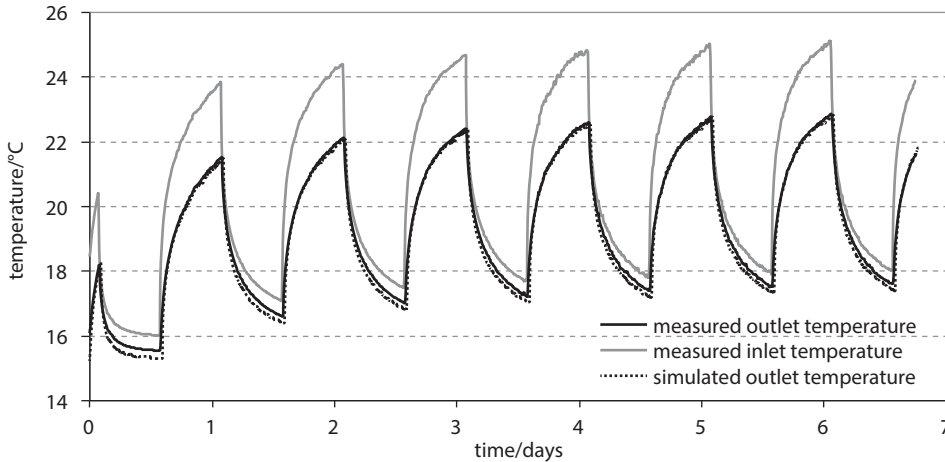


Figure 6.35 Measured and simulated temperature levels during the thermal response test.

Regarding the mutual distance of vertical ground heat exchangers, Zhang and Murphy (2003) state that the spacing of the boreholes is of major significance regarding the thermal effectiveness and that the single heat exchangers can be regarded as isolated from each other if the distance is greater than 6 m.

This finding is backed up by the current numerical parameter study (Figure 6.36). Starting with the standard distance of 6 m, the distances are subsequently decreased to 3 m and 1.5 m, keeping the total soil volume of the geothermal system constant. This results in a two times, respectively four times, greater number of heat exchangers with decreasing distance. Although the overall energy output rises, that of each heat exchanger is reduced due to the mutual influence.

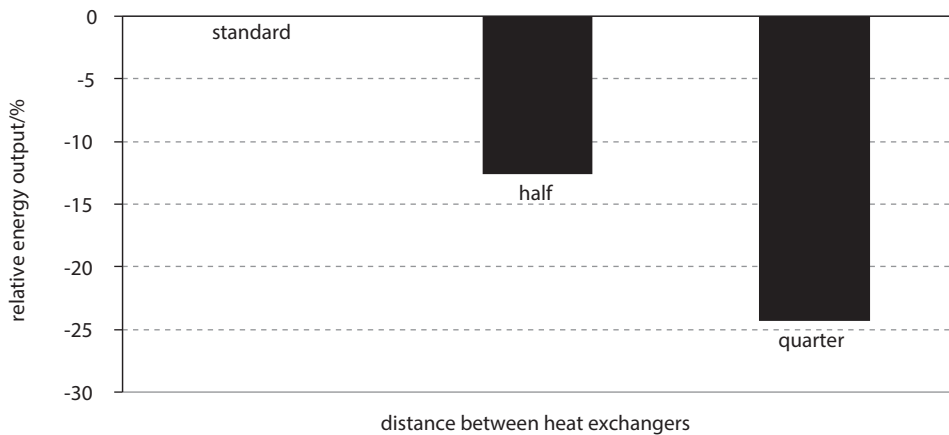


Figure 6.36 Influence of geothermal heat exchanger mutual distance on performance.

An increase of mass flow inside the tubes improves the energy withdrawal by up to 9% for high soil heat conductivities. At lower soil heat conductivities the improvement due to mass flow reduces down to zero.

Influence of climate on the cooling performance of geothermal heat exchangers

A field of five vertical ground heat exchangers with a length of 80 m each with a total volume flow of $2.4 \text{ m}^3 \text{ h}^{-1}$ is simulated in different climatic boundary conditions in order to investigate the climate influence on cooling performance. The average ambient temperatures of several cities in warm climates represent the undisturbed soil temperatures around the heat exchangers. Each system is operated for 4 months, 12 hours every day. An average sand type soil was assumed as surrounding soil. The inlet temperature into the ground heat exchangers is set to a constant of 22°C , representing the output of a thermally activated concrete slab from the building. Table 3 shows the investigated locations, the average ambient temperature (T_{amb}), the maximum outlet temperature of the ground heat exchangers (T_{max}) as well as the mean (P_{mean}), minimum (P_{min}) and maximum (P_{max}) available cooling power. The costs are derived from total annual costs of investment, operation and maintenance of 8.9 Euros per metre taken from real project data in Germany.

Location	T_{amb} / $^\circ\text{C}$	T_{max} / $^\circ\text{C}$	P_{mean} / W m^{-1}	P_{min} / W m^{-1}	P_{max} / W m^{-1}	Energy / MWh	Cost / € kWh^{-1}
Madrid	13.9	19.4	20.0	17.5	54.0	11.6	0.31
Sevilla	18.8	21.0	8.0	7.0	21.3	4.6	0.78
Bangkok	28.1	-	-	-	-	-	-
Crete	19.1	21.1	7.3	6.3	19.3	4.1	0.86
Athens	18.3	20.8	9.0	8.0	24.7	5.3	0.68

Table 6.4 Influence of location on geothermal HX performance for direct cooling applications.

Obviously, the climatic conditions of Bangkok are not suitable for direct geothermal cooling, as the inlet temperature from the building is lower than the ground temperature. Active chillers are needed here in order to reach the desired temperatures. In the Mediterranean climate of Crete on the other hand, the outlet temperature of the ground heat exchangers does not exceed 21°C at the end of the summer period. The direct use of geothermal energy via thermally activated concrete slabs is thus possible, although the achievable cooling power is relatively low. In all locations, the ground temperature regenerates to between 0.5 and 1.5°C above the undisturbed temperature at the end of each 12 hours operation cycle, rising only slightly during the 4 months period. This aspect is important regarding the long-term performance of any geothermal system (Pahud et al., 2002).

In order to determine the influence of the temperature level of the direct cooling system on the performance of the geothermal system, the same geothermal simulation as above was executed with several different inlet temperatures. Crete with a mean ambient air temperature of 19.1°C was chosen as location. Inlet temperatures between 20 and 24°C were chosen to represent the combined operation with a floor cooling system, whereas the inlet temperatures of 35°C and 40°C represent the replacement of a standard cooling tower by the borehole heat exchangers.

T_{inlet} /°C	T_{max} /°C	P_{mean} /W m ⁻¹	P_{min} /W m ⁻¹	P_{max} /W m ⁻¹	Energy /MWh	Cost /€ kWh ⁻¹
20.0	19.7	2.3	2.0	6.0	1.3	2.78
21.0	20.4	4.8	4.0	12.7	2.7	1.32
22.0	21.1	7.3	6.3	19.3	4.1	0.86
23.0	21.7	9.8	8.5	26.0	5.6	0.64
24.0	22.4	12.3	10.5	32.7	7.0	0.51
35.0	29.8	39.7	34.5	106.0	22.8	0.16
40.0	33.2	52.2	45.4	139.4	30.0	0.12

Table 6.5 Influence of inlet temperature on energy extraction and costs of geothermal heat exchangers.

From Table 6.5 it can be seen, that the maximum outlet temperature T_{max} , mean thermal power P_{mean} , minimum thermal power P_{min} , as well as maximum specific thermal power P_{max} rise approximately proportional to the inlet temperature. This leads to the conclusion that the ground temperature around the heat exchanger regenerates almost completely during the downtime of the circulation pump and no saturation is reached.

High-temperature differences between fluid inlet to the geothermal heat exchanger and ground temperature are therefore advisable, which makes the systems very suitable in combination with ventilation air cooling. Regarding the operation of the borehole heat exchangers to replace a cooling tower, it can be stated that geothermal heat exchangers are an interesting alternative concerning effectiveness and cost.

6.5 Economics of geothermal heat exchangers

The annuity method allows to combine one-time payments/investment and recurring payments with help of the annuity factor during a period under consideration. The comparison shown here is limited to capital and consumption related costs. For all cases with combined heating

and cooling applications, i.e. all year operation, the saved heating costs are credited as benefits. Planning, maintenance and any other costs (e.g. insurances) are excluded.

Detailed capital costs were available for the two 134 m boreholes of the heat pump system in Ostfildern.

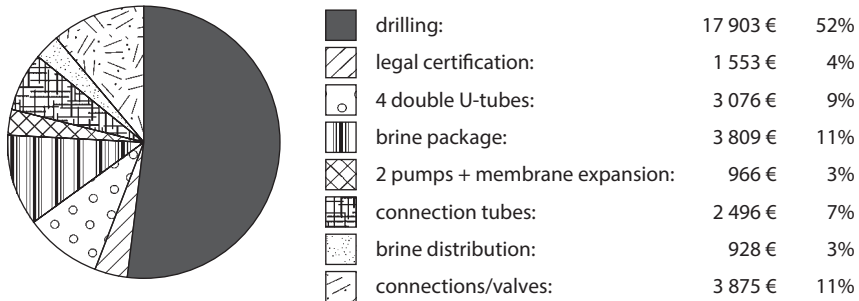


Figure 6.37 Investment costs of the vertical heat exchangers at the youth centre installation in Ostfildern.

The investment costs correspond to 129 Euros per metre of earth heat exchanger with the drilling costs accounting for about of the total costs. Other project examples (e.g. systems from Spain) have significantly lower specific investment costs (about 50 - 60 Euro m⁻¹).

A study of the Ministry of the Environment, Climate Protection and the Energy Sector Baden-Württemberg from 2011 showed a linear correlation between investments and ground heat exchanger length with a slope of 58 Euros per metre.

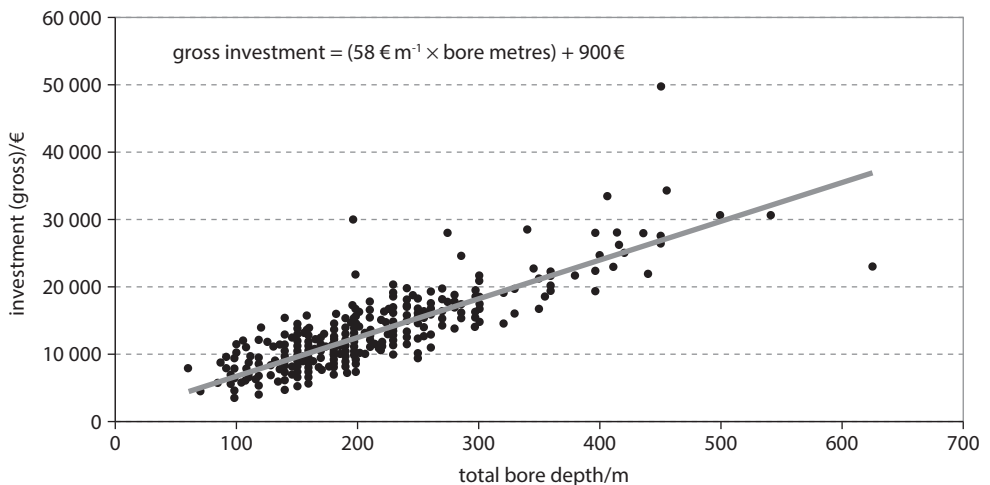


Figure 6.38 Gross investment costs for vertical earth heat exchangers.

The assumed investment costs for an air to ground heat exchanger are about 38 Euros m⁻¹.

Operation costs are related to costs for energy and utilised resources. The required auxiliary electrical energy for the brine-circulating pump and ventilators was calculated for cooling and heating during 1 year.

Because the geothermal plant is operating throughout the year, the heat gains during winter are taken as revenues with a cost saving of 9.29 ct kWh⁻¹ for district heating. Energy price increases of 3% annually are also considered.

For the annuity calculation, an interest rate of 4% was assumed. The service life of an earth heat exchanger system was considered with 40 years and in another case study with 80 years.

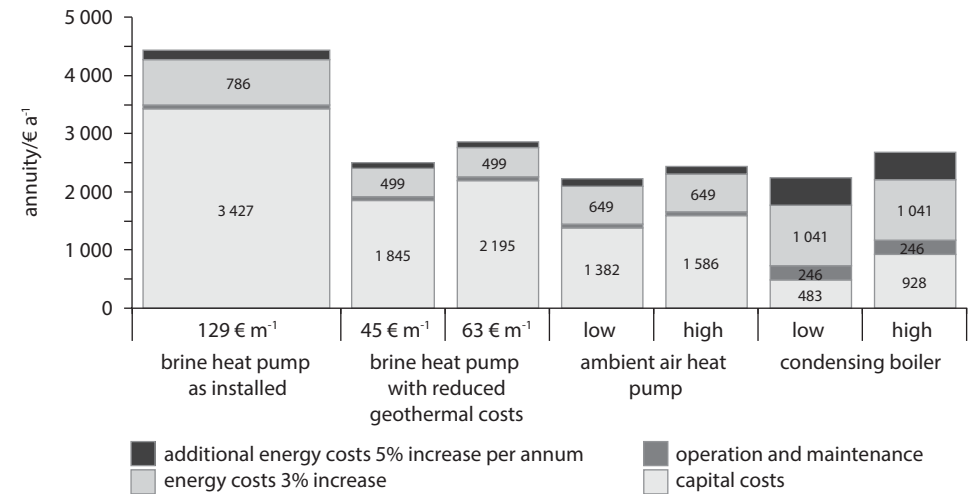


Figure 6.39 Annuity of geothermal heat pump with high costs per metre of vertical heat exchanger compared to cheaper geothermal installations, ambient air heat pumps and condensing boilers.

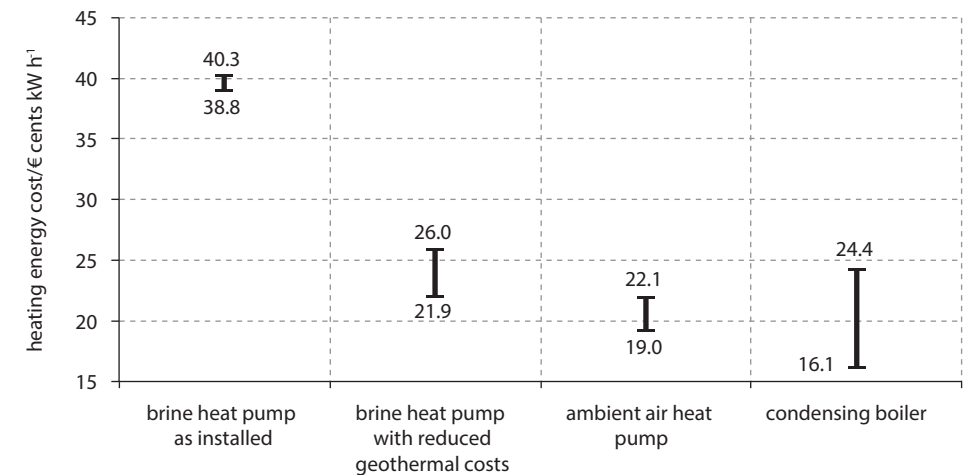


Figure 6.40 Range of total heating energy costs for the youth centre project. For the brine heat pump as installed, only the rate of energy price increase was varied from 3% to 5%; for all other cases the investment costs were varied.

When the annual costs are related to the annual heating energy consumption in the nearly passive standard building, the specific costs vary between 16 cents per kWh for the least cost case of a condensing boiler up to about 40 cents per kWh for the existing geothermal heat pump system. Costs only become comparable to the condensing boiler costs, if the geothermal heat exchanger has significantly lower investment costs. In buildings with higher heat demand, the advantage of geothermal systems with lower operation costs increases.

To compare cooling costs, three of the ground heat exchange applications were compared: the vertical ground heat exchanger from the Freiburg office building, the foundation pile heat exchanger from the Ostfildern office building project and the air to ground heat exchanger from Weilheim.

The specific costs for the vertical heat exchangers were 58 Euros per metre and for the air to earth heat exchanger 38 Euros per metre. For the foundation pile heat exchanger costs, investment costs were taken from the office building project in Ostfildern, Germany. The total brut costs of the foundation piles were 89 798 Euros, i.e. 143 Euros m^{-1} . The geothermal tubes had a share of 14 033 Euros of the total costs, which corresponds to 15%. The resulting geothermal heat exchanger costs were 22.5 Euros per metre. Not included in this cost group were the connection pipes in the building, the heating energy counter and the heat exchangers in the ventilation system. For the cost comparison 30 Euros m^{-1} were used as total costs for foundation pile geothermal heat exchangers.

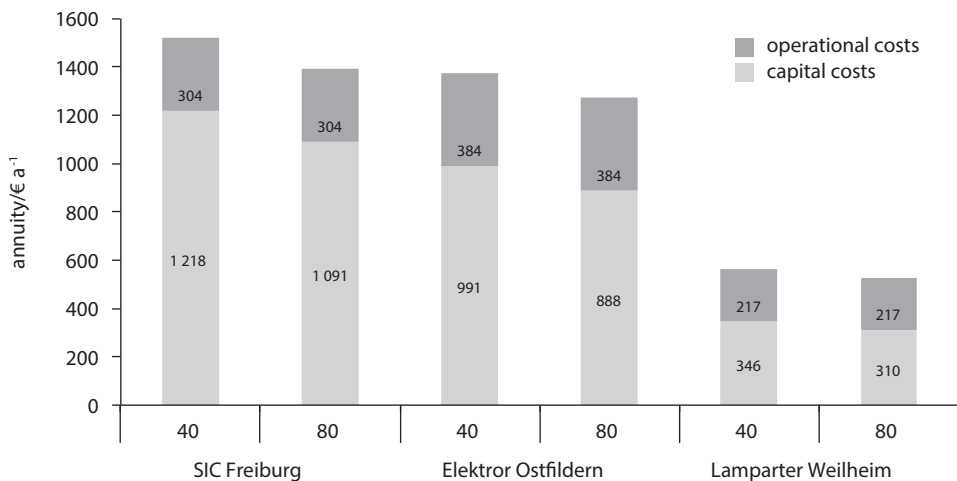


Figure 6.41 Total annual costs for the cooling applications with lifetimes of the ground heat exchanger between 40 and 80 years.

This economic comparison shows that in case of doubling of the serviceable life there are no significant cost reductions. The air to ground heat exchanger has the lowest overall investment and operational costs. If the annual costs are divided by the produced cooling energy, a specific cooling-price for each of those systems is obtained.

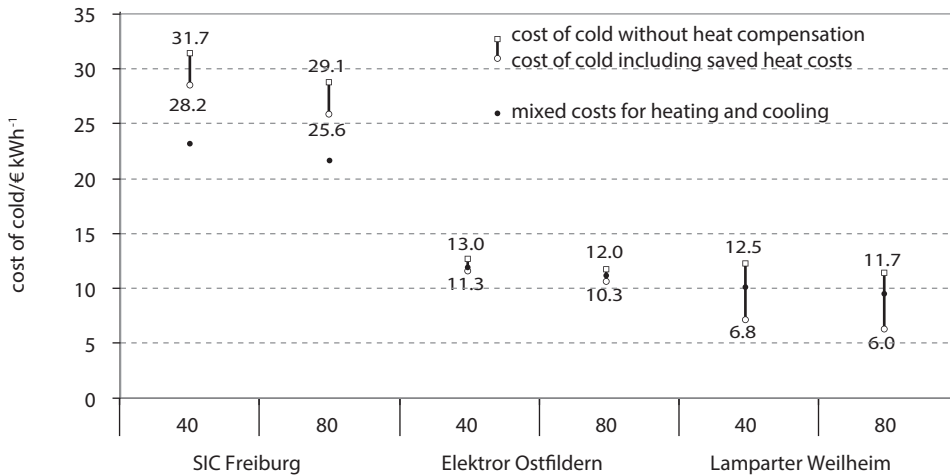


Figure 6.42 Specific cooling-price for to the individual variants.

The cooling costs are further reduced by the consideration of the saved heat energy. As already mentioned, the examined geothermal heat exchangers operate all year and thus cover in winter a part of the heat requirement. The total costs of cooling were calculated including the savings of operational costs of the heating system, i.e. by deducing the saved kilowatt hours of heating multiplied by the specific gas price. In addition the annuity was divided by the sum of heating and cooling energy to obtain a mixed cost price for heating and cooling.

The cooling costs vary between 6 cents per kilowatt hour for the air to ground heat exchanger, which is a very economic solution, and 10 - 13 cents for the foundation pile heat exchangers, which is still competitive with conventional cooling equipment. The high costs of the vertical heat exchangers and the rather low operation hours lead to high cooling costs between 25 and 32 cents per kilowatt hour in the Freiburg example.

6.6 Performance summary on geothermal heat exchangers

In summary, low-depth geothermal heat exchangers can be used for direct or indirect heating and cooling of buildings.

Air to earth heat exchangers used for pre-cooling and preheating of ventilation air have excellent ratios of produced cold to used electricity. Air-based systems are shown to have annual COPs between 35 and 50. Vertical ground heat exchangers reached coefficients of performance of about 20.

In buildings with direct use of the geothermal heat exchanger for ventilation air cooling or concrete core activation, the cooling energy production was higher than heating energy production, even in the moderate German climate conditions. Both heating and cooling power increase with increasing temperature difference between fluid inlet and soil temperature, with the highest measured power of 66 W m^{-1} obtained for a winter heat pump operation at 4°C inlet temperature, i.e. about 9 K temperature difference to the soil. For cooling the highest power of 45 W m^{-1} was reached for a rather shallow earth to air heat exchanger at 2 m depth only, but where the inlet air temperature was 35°C and the temperature difference of the air to the ground 15 K. By comparison, the 80 deep vertical heat exchangers only had 26 W m^{-1} maximum cooling power,

Type of heat exchanger	Heat and cold distribution	Building	Total length of HX /m	Heating energy /kWh m ⁻¹	Cooling energy /kWh m ⁻¹	Max heating power /W m ⁻¹	Mean heating power /W m ⁻¹	Mean COP _{el} heating cooling	Max cooling power /W m ⁻¹	Mean cooling power /W m ⁻¹
Vertical HX (80 m deep)	Ventilation system/ Concrete floor activation	SIC, Freiburg	400	7	12	16	3	12.8 20	26 10	1.5
Vertical HX (134 m deep)	Heat pump	Youth Centre Ostfildern	268	36	-	66	13	-	-	-
Horizontal HX (1.2 m deep)	Ventilation system	Ebök Tübingen	500	-	-	-	-	-	7	3
Foundation piles (4 to 8 m deep)	Ventilation system	Elektror Ostfildern	624	4	17	14	6	12.5 22	24	10
Earth air HX (2 m deep)	Ventilation system	Lamparter Weilheim	180	17	25	34	9	-	45	10

Table 6.6 Summary of performance measurements.

as the maximum inlet temperature to the ground was only 22°C due to the high heat exchanger temperature spread between brine circuit and ventilation air. Nearly the same performance was obtained for shallow foundation pile heat exchangers with a maximum cooling power of 24 W m⁻¹.

The measured overall energy yield for direct heating and cooling is rather low with values between 4 and 17 kWh m⁻¹ for heating and between 12 and 25 kWh m⁻¹ for cooling. For indirect use with heat pumps 36 kWh m⁻¹ were measured in a low-energy building. Combined with high investment costs, this low-energy yield results in high costs per kWh cooling and heating produced, especially for vertical heat geothermal heat exchangers. Only for low cost geothermal systems, which are integrated into foundation piles or are installed as shallow horizontal heat exchangers, the cooling costs including benefits from heat sales are lower than 0.1 Euro kWh⁻¹ and thus economically attractive.

The mean heating and cooling power measured was always significantly lower than quoted in standards such as the German VDI 4640. Here design values for geothermal heat extraction for small and medium size heat pump applications (up to 30 kW) are given with 20 W m⁻¹ as the lowest mean value and up to 100 W m⁻¹ for conductive ground with groundwater flow.

The use of geothermal heat exchangers for heat rejection of cooling machines was also investigated and very good performance was obtained when compared to cooling towers due to low pressure drops. The auxiliary electrical energy consumption for solar absorption cooling systems can be reduced by 30% if geothermal heat exchangers are used. The results of performance measurements are summarised in Table 6.6.

Numerical simulation models have been developed and validated against the experimental data and reproduce the measured values well. They can now be used for planning of geothermal heat exchangers and for parameter studies. The simulation results show that performance is improved, if the backfill material and obviously the soil have higher conductivity values (for example, in regions with ground water flow). The parameter studies showed that the earth heat exchangers can be directly used for building cooling also in warmer Mediterranean climates, although the power dissipation level drops at higher soil temperatures. In a Southern Mediterranean climate, even after a 4 months summer period, the maximum outlet temperature of the heat exchanger was not higher than 21°C, so that activated concrete ceilings can still be operated efficiently.

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7 Photovoltaics

Figure 7.1 (previous page): Back contact polycrystalline solar cells “Sunweb”, developed by Solland and produced by Schott (Photo: Eric Duminil).



Figure 7.2 PVT facade of the experimental SDE House on location in Madrid (photo: Jan Cremers, HFT Stuttgart), see also 9.6 Photovoltaic thermal collectors for heating and cooling generation on page 576.

Photovoltaics (PV) means the direct conversion of short-wave solar irradiance into electricity. Today's market is dominated by semiconductor solar cells on the basis of crystalline silicon, but new technologies based on plastics, organic materials or thin-film cells with diverse semiconductor combinations like cadmium telluride (CdTe), amorphous silicon (a-Si) or copper indium diselenide (CIS) might increase their market shares from currently 15% up to 30% in the next decades.

Photovoltaic modules as direct current (DC) producers are mainly connected to the 230 V low-voltage electricity grid or to the medium or high voltage lines for high power systems in the multi-megawatt range. The inverters necessary for this are available on the market with a wide input voltage and power range with efficiencies of over 95%. Today about 70% of installed power is fed into the low voltage grid, 30% into the 20 kV medium voltage grid.

The increasing number of installations of photovoltaic power plants in the multi-megawatt range requires active energy management for photovoltaic systems. This concerns the regulation of active power, the provision of idle power and specific error characteristics (fault ride through). The generating plants PV are to be involved in both the slow, static voltage support as well as the dynamic voltage support during voltage dips. Grid stabilisation becomes more and more an issue, as the installed power gets close to the electricity demand levels.

Since 2000, the photovoltaic energy fed to the grid in Germany is financially covered by the Renewable Energy Law and the extra costs are allocated to all electricity customers. This very successful German law has been applied in many countries. Since 2009, the own consumption of PV energy is treated preferentially, as more than 30 GW of installed PV power in Germany lead to a high load for the electrical distribution grid.

With 6578 GWh in 2009, the photovoltaic covered 1.1% of total electricity consumption in Germany, increasing to 3.2% in 2011 and 4.6% in 2013. In Europe, the photovoltaic power generation is estimated to reach 8 - 12% in 2020 and 15 - 25% in 2030.

Photovoltaic systems are characterised by an extremely modular structure, since in principle each module, connected with an inverter, can act as a producer of alternating current (AC). By encapsulating the extremely thin semiconductor cells in a glass-glass or glass-plastic combination, photovoltaic modules are particularly suitable for integration in buildings, since the usual glazing constructions can be used. Only the presence of cables differentiates a photovoltaic module from the use of conventional glazing. The components necessary for the systems engineering, such as circuit-breakers, fuses and inverters, can be placed anywhere in the building (or even outside), and take up very little space.

The almost limitless dimensions and designs of photovoltaic modules, the selectable module colour design and the possibility of partial transparency offer special architectural possibilities, in particular for facades. The thermal aspects of integrating photovoltaics in buildings are considered in a separate chapter.

7.1 Structure of grid-connected systems

A mains-connected photovoltaic system consists of a solar generator, an inverter, and switching and protection elements. The solar generator consists of PV modules with direct current (DC) power between 100 and 250 W for standard modules, and up to around 600 W for large-scale modules. The voltage is, depending on module type, between about 24 and 55 V, the current 5 - 9 A.

The interconnections of PV modules depend on the DC voltage level of the system, which can be in the low voltage class below 120 V up to power levels of 5 kW, requiring

little effort for safety. To work with low current levels and to use highly efficient transformer-less inverters, mostly higher DC voltage ranges of several hundred volts are selected.

The cables of the modules connected in series are combined into parallel strings in a PV junction box containing over-voltage protection and possibly string diodes. From there the DC main cable leads to the inverter. If each string is provided with its own inverter the junction box is omitted, a concept which has even been implemented even in large systems (e.g. a 1 MW building-integrated photovoltaic system was built with 569 1.5 kW inverters in Herne, Germany).

A DC circuit-breaker before the inverter enables disconnection of the system for maintenance work at the inverter, but it is not necessary for safe functioning of the PV system. Monitoring the mains voltage and disconnection of the PV system during mains power loss is usually integrated in the inverter, but the PV system must also be manually disconnectable from the mains after the inverter. Up to 4.6 kW power, single-phase supply to the mains is normal, and only with more power are three-phase inverters used.

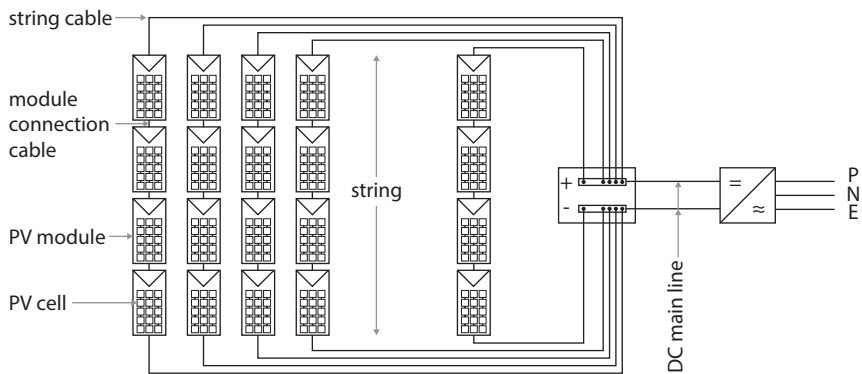


Figure 7.3 Grid connected photovoltaic system with one inverter for all module strings.

The inverter converts the photovoltaically produced DC into AC, which in building-integrated systems is mainly fed into the 230 V low-voltage mains.

Module-integrated small inverters increase the modularity of PV systems and make possible the usual AC wiring, but they require more material to produce and at low power have rather low efficiencies.

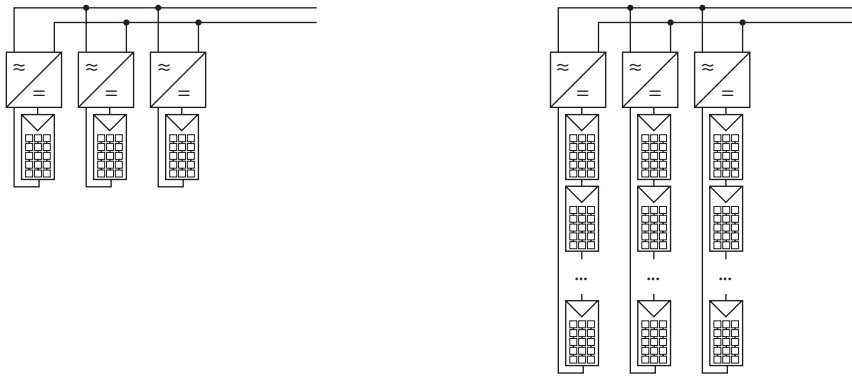


Figure 7.4 Module inverter with one inverter per module (left) and PV system with string inverters (right).

String inverters are often used today to simplify DC interconnecting and to decouple the individual module strings from each other. The high DC input voltages, due to the series interconnecting of all modules in the string, result in higher requirements for electrical safety than systems below 120 V_{DC}. In the power range up to 5 kW, a variety of devices are available, with very good transformation efficiencies up to 98%. The inverter world record with such fast-switching silicon-carbide devices is at 99.03% (press release Fraunhofer ISE, 2009).

In order to reduce materials and costs and to lower the weight and size of the inverters, transformerless inverters are increasingly used, their power electronics supplying a sinusoidal current at a DC voltage level above the mains voltage.

7.2 Solar cell technologies

Today's solar cell market is still dominated by crystalline silicon technologies, with a market share of over 80%. While the best small-dimension laboratory cells (2 cm × 2 cm) achieve efficiencies of some 25%, the worldwide highest module efficiency is about 23%. There are monocrystalline modules with up to 19.5% efficiency and polycrystalline modules with around 14% efficiency on the market.

Thin-film cells based on amorphous silicon are manufactured both for consumer products of small power and also as modules for power plants or building applications. By using extremely thin double- or triple-stacked diodes, the problem of light-induced efficiency degradation can be reduced. Amorphous silicon cells are applied both to flexible metal substrates and to coated glass and achieve stabilised efficiencies of 5 - 8%. The combination of amorphous/microcrystalline tandem cells as a structure is a well-developed manufacturing technology to deliver stabilised efficiencies from 8% to 11%, in the laboratory up to 13%.

Thin-film technologies based on CdTe are mass produced with an efficiency of 7 - 10%. Copper indium diselenide modules (CIS or CIGS with gallium) are available with 9 - 12% efficiency. Laboratory efficiencies of CIGS solar cells of 20.3% have been achieved. Still in development or pilot production are organic solar cells with 5% laboratory efficiency or dye cells with 11% laboratory efficiency and 3 - 5% in the pilot production.

7.3 Module technology

Standard modules are encased by polymer layers or cast resin between the front glazing and the back substrate (either glass or a plastic-aluminium material combination). The commonest encapsulation material is a co-polymer of ethylene and vinyl acetate (EVA), which is put on both sides between the cell and substrate, and is polymerised by heating to 140 - 160°C after evacuation in a vacuum laminator (to prevent air bubbles). The degradation of the polymer by UV radiation can largely be prevented by the addition of stabiliser materials. Special or large-scale modules are usually encased using cast-resin technology (TPU, Acrylate) or thermoplastic silicones with elastomeric properties (TPSE) and are manufactured nowadays up to 12 m² in size.

The outside glazing of a PV module consists of iron-poor glass, which is pre-stressed for sufficient mechanical stability either thermally (3 - 4 mm glass strength) or chemically (2 mm glass strength). Iron-poor white glass transmitted irradiance up to 96%. As a back glass substrate in glass-glass modules, safety glass is normally used.

Frameless modules are fastened with profile systems or glued to a frame construction as a structural glazing system. Modules with aluminium frames are often used if building integration is not required, and are mounted on top of the roof or in front of cold facades.

Besides building-glass constructions with photovoltaics, a variety of photovoltaic roof tile systems are available, in which a special plastic frame construction takes on roof tile functions such as impermeability to driving rain and rain run-off, and which are easy to fit on roof slats and connect to adjoining standard tiles.

7.4 Building integration and costs

For photovoltaic systems (small and medium power), the material costs in the form of modules, inverters and cabling are at 85% versus 15% installation costs. For large systems the installation costs may decrease to 8%.

The modules continue to dominate the total cost of about 50 - 60%, followed by the inverter with about 10 - 15%, and the protection elements, including wiring, metres, etc. Since the early 1980s, the system costs dropped drastically and are currently between 1000 and 2000 Euro per kW, or even lower in larger installations.

PV module costs tends to rise with sophisticated building integration solutions, since special modules with particular semi-transparency or colouring and no standard dimensions are often selected. The investment costs for building-integrated solutions are currently around 20 - 30% higher than standard systems. The investment cost is estimated at about 750 Euros m⁻² for a PV facade (including assembly) and are thus about 7% higher than that of a stone facade and around 20% higher than a glass or ceramic facade. In comparison with a facade of polished stone, the investment costs of a PV facade are about 60% lower.

A study of 30 German facade systems shows very strong project-related costs between 8 000 and 20 000 Euros kW⁻¹, with the modules including fixing responsible for 67% of total costs, the inverter is only 6.7% (NRW Energy Agency, 2008).

Due to the high cost of building-integrated photovoltaics, the European market share is only 2%, although it is often promoted by higher feed-in tariffs.

Taking as an example the back-ventilated structural glazing construction of a public library in Mataró in Spain (see Figure 9.6 and Figure 9.7) installed in the 1990s, the cost structure can be examined in more detail and compared with other facade constructions. The cost allocation of the back-ventilated PV facade includes 2.5 m² glass-glass special modules glued to an aluminium

profile construction with double glazing at the back, the DC cabling, the inverter and the mains-connected safety engineering with total costs of 1167 Euros m^{-2} .

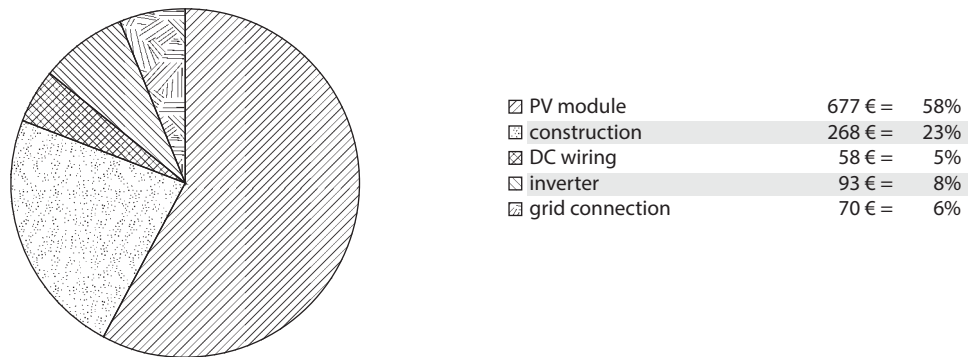


Figure 7.5 Cost allocation of a 245 m^2 PV warm facade as a percentage of the total costs of 1167 € m^{-2} . The PV facade was constructed in 1995.

In addition to the back-ventilated facade, 325 m^2 PV roof sheds with 50 W frameless standard modules are integrated in the building. The sheds are also back-ventilated and glued to the profiles as a structural glazing construction, the back double glazing being replaced by an insulated panel. The total cost of the roof shed construction was 1051 Euros m^{-2} , with the PV modules dominating the costs here too. For comparison, the cost structure of a conventional curtain facade with laminated glass (6 mm + 6 mm) and of a cold facade with PV standard modules are shown. The surface-related system engineering cost of standard modules are somewhat higher than in the facade special modules, due to higher electrical efficiencies.

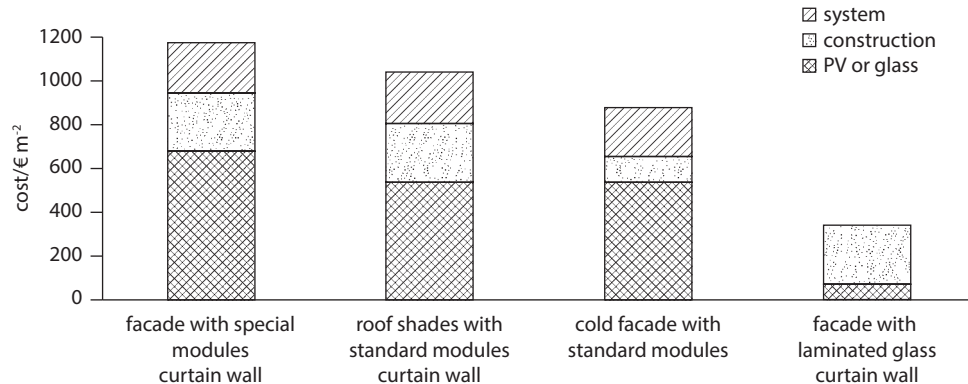


Figure 7.6 Costs for different facade systems with and without photovoltaic modules built in 1995.

7.5 Energy production and the performance ratio of PV systems

The energy yield of a photovoltaic system is indicated in kWh of electricity supplied, and related to the installed peak power in kW_p under standard test conditions (STC), i.e. 1000 W m⁻² irradiance and a module temperature of 25°C. The measured yields of grid connected systems under German climatic conditions are between 800 and 1100 kWh kW_p⁻¹.

To be able to identify a radiation-independent characteristic figure of a PV system, the specific AC energy yield Y_f (final yield) is related to the theoretically possible yield without equipment and system losses. The so-called performance ratio P_R is defined in the IEC standard 61724.

The specific capture losses L_c includes divergences between the efficiency of the generator and module efficiency, lower module power, mismatch losses and shadowing losses. The system losses L_s includes the ohmic losses of the DC components and the conversion losses of the inverter.

The performance ratio PR is the ratio of the final yield Y_f and the theoretical yield Y_r . The PR is a level of quality assessment of grid-connected PV systems, which is largely independent on the location (solar radiation).

$$P_R = \frac{Y_f}{Y_r + L_c + L_s} \quad (7.1)$$

Today's performance ratios are between 80% and 90% annually, with strong fluctuations on shorter timescales such as monthly values. The performance ratio can be easily calculated by comparing the measured AC power $P_{AC, measured}$ at irradiance $G_{measured}$ over the desired period, with the possible DC power at STC conditions $P_{DC, STC}$. The PR is thus a ratio of efficiencies.

$$PR = \frac{\int_{t_1}^{t_2} P_{AC, measured} dt}{P_{DC, STC} \int_{t_1}^{t_2} G_{measured} dt} \quad (7.2)$$

The wide value range and the size of the losses suggest a more exact loss analysis and performance monitoring of PV systems.

For this, the performance ratio can be subdivided into PV module losses, DC interconnecting losses and AC transformation losses. Like the performance ratio, the actual DC generator energy or power produced is related to the nominal power under standard test conditions and is irradiance-corrected. The module ratio (MR) is defined by

$$MR = \frac{P_{module-DC, measured}}{P_{module-DC, STC}} \cdot \frac{G_{STC}}{G_{measured}} \quad (7.3)$$

and contains the power losses of the module due to temperatures above 25°C, deviation from the rated power at low irradiance and possible shading of individual cells in the module.

The array ratio AR contains both the module losses and also power losses due to the DC wiring of the generator and adjustment losses by interconnecting non-identical modules.

$$AR = \frac{P_{generator-DC, measured}}{P_{generator-DC, STC}} \cdot \frac{G_{STC}}{G_{measured}} \quad (7.4)$$

The performance ratio PR contains finally the losses of the inverter by the transformation of DC into AC.

In most performance assessments an annual degradation of performance is assumed of about 0.25 - 0.5% for crystalline technology and 0.6 - 1% for thin-film technology. However, many systems, which have been monitored for 20 years, did not show any degradation at all.

Energy amortisation times

The use of photovoltaic systems is only worthwhile in energy economic terms if the energy expenditure in producing the entire plant is clearly lower than the energy quantity produced during its life span.

The energy amortisation time of photovoltaic systems is dominated by the energy expenditure in cell production. The primary energy expenditure to produce a kW of installed photovoltaic power (about 6 - 8 m² of surface depending on module efficiency) is about 2500 kWh kW_p⁻¹ (Palz and Zibetta, 1991). The installed photovoltaic power has increased worldwide by a factor 300 since the early 1990s, and the energy required for the production has reduced by a factor 3.

Depending on the location and irradiance, the energy payback time varies between 2 years for Southern European locations and 4 years for Scandinavian locations. In Germany typical energy amortisation times are 2.5 years (Stucki and Frischknecht, 2010).

A recent life cycle study of the Energy Research Centre in the Netherlands calculated an energy payback period between 1.7 and 4.6 years, depending on the used technology of the PV modules. Apart from the module, the type of frame and the inverter/wiring/assembly system determines the amortisation period. The module itself dominates with 90% of the total system energy content (Wild-Scholten and Alsema, 2006)

Manufacturer warranties for PV modules now cover 10 - 20 years, and the life expectancy is well over 25 years. A very long PV module life span is possible, as encapsulation permanently protects the cells from harmful environmental influences, in particular humidity. Initial industrial tests to recycle solar cells from modules have shown that the solar cells themselves display no sign of degradation even after 20 years, and can be encased again as new modules with very small losses in performance.

7.6 Physical fundamentals of solar electricity production

In photovoltaic cells, solar irradiance is converted directly into electricity. The short-wave irradiance is absorbed by the solar cell and produces free electrical charge carriers in the conduction and valence bands.

Suitable materials on a semiconductor basis for solar cells have energy gaps between the valence and conduction bands, which are adapted to the photon energy content of solar irradiance. A compromise must be found between high current generation with a small gap, where even the long wavelength infrared solar irradiance can be absorbed, and high voltage production with large gaps. The highest efficiencies can be obtained with band gaps between 1.3 and 1.5 electron Volt (eV), e.g. indium phosphide with 1.27 eV, gallium arsenide with 1.35 eV or CdTe with 1.44 eV. The currently most frequently used crystalline silicon has a rather low band gap of 1.124 eV.

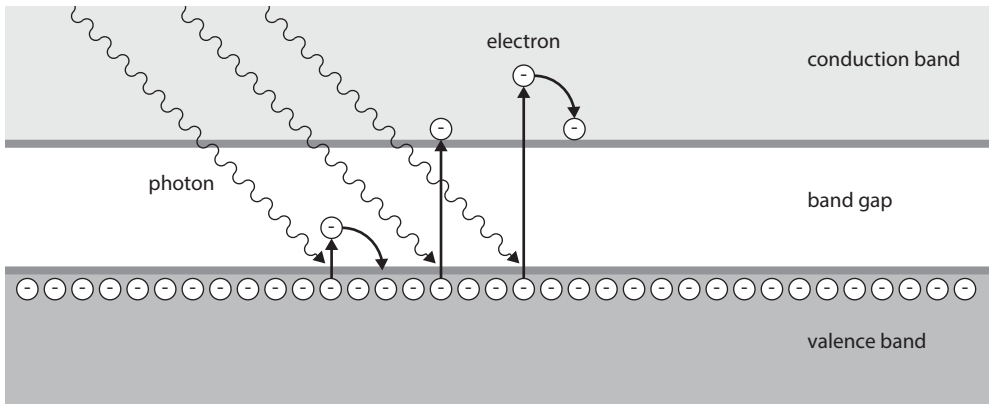


Figure 7.7 Band gaps and charge carrier production in a photovoltaic cell.

The limitation of the theoretical maximum efficiency of a solar cell to 44% is caused mainly by the width of the solar spectrum. Starting from an energy given by the band gap, electrons are lifted from the valence band into the conduction band. Higher-energy photons of the solar spectrum are likewise absorbed, but transfer the surplus energy relative to the gap into thermal energy. Furthermore, there is always a part of the long-wave solar irradiance in the infrared that is not absorbed.

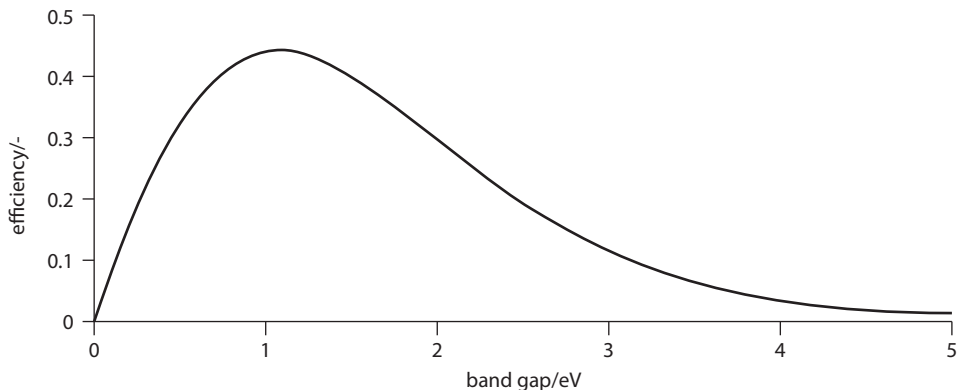


Figure 7.8 Theoretically possible efficiencies of solar cells.

Example 7.1

Calculation of the maximum wavelength of the solar spectrum still absorbed by the following solar cell materials: crystalline silicon with a band gap $E_{gap} = 1.124 \text{ eV}$, CdTe with $E_{gap} = 1.5 \text{ eV}$ and amorphous silicon with $E_{gap} = 1.7 \text{ eV}$.

The conversion of the band-gap energy E_{gap} into the wavelength λ by means of Planck's equation results in the longest wavelength of the radiation still absorbed:

$$E_{gap} = h\nu = \frac{hc}{\lambda} [J] = \frac{hc}{9\lambda} [eV]$$

with

$$h = 6.626 \times 10^{-34} Js, \quad c = 2.99792 \times 10^8 \frac{m}{s}, \quad q = 1.6021 \times 10^{-19} As$$

$$\Rightarrow E_{gap} = \frac{1.24 \times 10^{-6}}{\lambda} [eV] \text{ with } \lambda [m]$$

Photons of greater wavelength are below the band-gap energy and cannot be absorbed. Crystalline silicon therefore absorbs to $\lambda < 1.1 \times 10^{-6} m$, CdTe to $\lambda < 0.826 \times 10^{-6} m$ and amorphous silicon to $0.729 \times 10^{-6} m$ only in the visible, due to the high gap. The complete infrared radiation of sunlight is not absorbed by amorphous silicon and is partly responsible for the low efficiencies of this material. The gap of amorphous silicon can be reduced, however, by the addition of germanium.

The electrical field within the space-charge-zone is produced by doping the semiconductor material with foreign atoms. If the tetravalent silicon is endowed with pentavalent phosphorus, free electron charge carriers (negative or n-doping) develop. Then if the other side of the silicon wafer is doped with trivalent boron atoms (positive or p-doping), an electron deficiency develops there. The free electrons diffuse into the p-doped area and thus create an electrical field; the p-doped section now displays an electron excess in the boundary layer and is negative, the n-doped section has an electron deficiency and is positively charged. The electric field is in equilibrium with the diffusion process, preventing further electrons of the n-doped layer from moving to the p-layer. In equilibrium the p/n junction has a constant electrochemical potential, i.e. the average energy of electrons has a constant value in the band gap as described by the Fermi level.

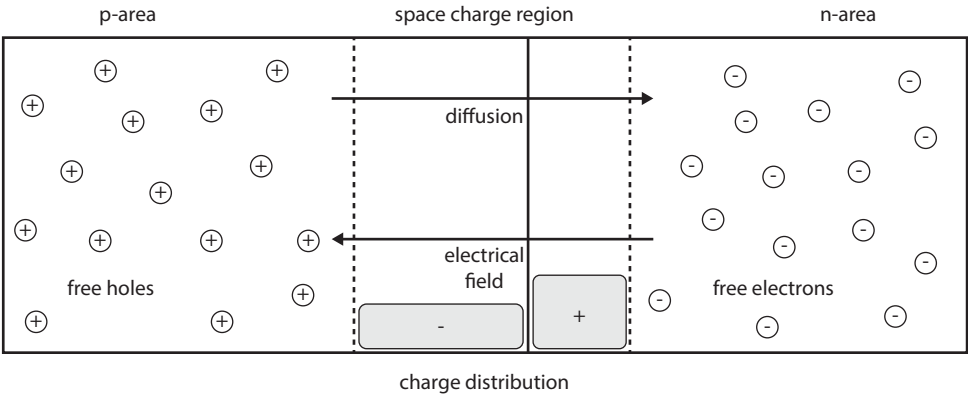


Figure 7.9 Electrical field and movement of the charge carriers.

As the Fermi level is close to the valence band in the p layer and close to the conduction band in the n-layer, a constant Fermi level in the p/n junction means that the conduction and valence bands are curved across the space charge region, which indicates an electrostatic potential across this region. The potential barrier prevents electrons diffused into the n-region from back diffusion up the barrier and holes as majority carriers in the p-region to diffuse back into the n-region.

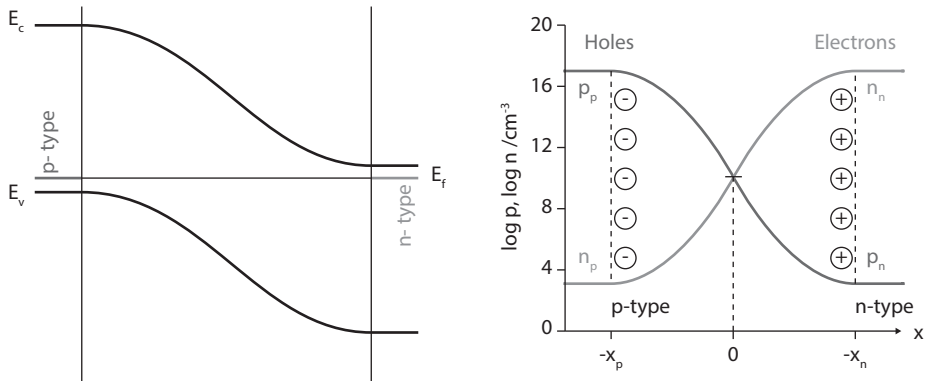


Figure 7.10 Band gap structure of a p/n junction under equilibrium conditions (left) and equilibrium concentrations of electrons and holes in a p/n junction for silicon doped with 10^{17} donors and acceptors per cm^3 (right); the width of the space charge region is $0.147 \mu\text{m}$ and the built-in voltage is 836 mV .

The changing electric potential $-q\phi(x)$ results from the charge distribution of fixed donor and acceptor states. The electron affinity μ_{n0} describes the binding energy of an electron at the bottom of the conduction band and is 4.01 eV for silicon. In addition to the fixed electron affinity, the chemical potential has a concentration dependent part $\Delta\mu_n(x)$, which is given by the difference between conduction band and Fermi level.

The overall electrochemical potential of the electrons η_n is given by the sum of the electrical plus the chemical potential, i.e. by the difference between Fermi level and electron energy in the vacuum, which is constant under dark equilibrium conditions or under illuminated short circuit conditions.

If free charge carriers (electrons and holes) are now produced by photon absorption, they start to diffuse. Once an electron has diffused into the n-layer, i.e. is found outside the space charge region, it cannot diffuse back due to the potential barrier. Electrons diffusing to the n-area are the majority carriers and can leave at the contacts, whereas they remain minority carriers at the p-side and recombine. The photocurrent thus flows in the opposite direction to the forward current of a p/n diode!

The increase of electron density in the n-area leads to an increase in the electrochemical potential (i.e. the Fermi level splits in so-called quasi-Fermi levels), and the increase of holes decreases the hole potential. The difference gives the voltage level.

If the outer contacts of the solar cell are short-circuited, the electro-chemical potential between p- and n-layer, i.e. the difference between the Fermi levels, reduces to zero.

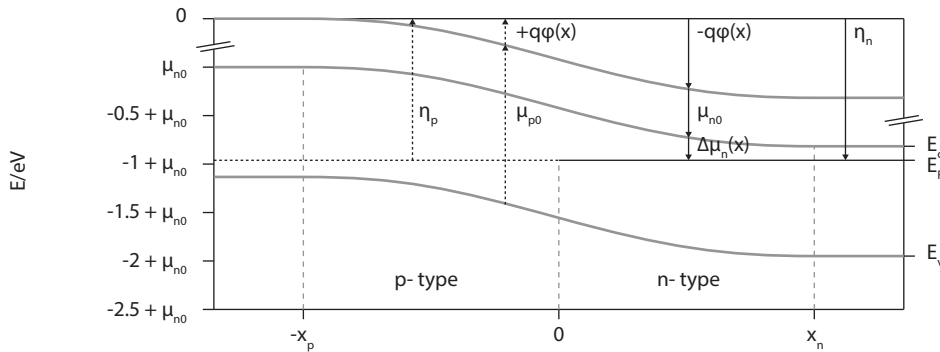


Figure 7.11 Band gap structure across the p/n junction under short circuit conditions.

The photocurrent produced is always a DC.

7.7 Current-voltage characteristics

Photovoltaic cells are photocurrent generators whose electrical characteristic is determined by the superposition of a diode characteristic and a voltage-independent photocurrent, which is a valid assumption for crystalline technologies. The voltage produced is reduced by series resistances of the material and the metallic contacts, and part of the photocurrent produced flows off due to shunt resistances. The current usable in the outer electrical circuit results from the photocurrent minus the diode and shunt losses. The voltage across the internal photodiode is obtained from the outside voltage measured at the contacts plus the voltage drop over the series resistance.

Characteristic values and efficiency

In an open outer electrical circuit (total value of current $I = 0$) the maximum voltage is generated; this is called the open circuit voltage, V_{oc} . If the external circuit is short-circuited (voltage $V = 0$) the maximum so-called short circuit current I_{sc} is obtained.

The electrical power P of the solar cell is calculated by the product of the current and voltage.

$$P = I \times V \quad [W] \quad (7.5)$$

The performance curve indicates a sharp maximum near the open circuit voltage; this is called the maximum power point (MPP).

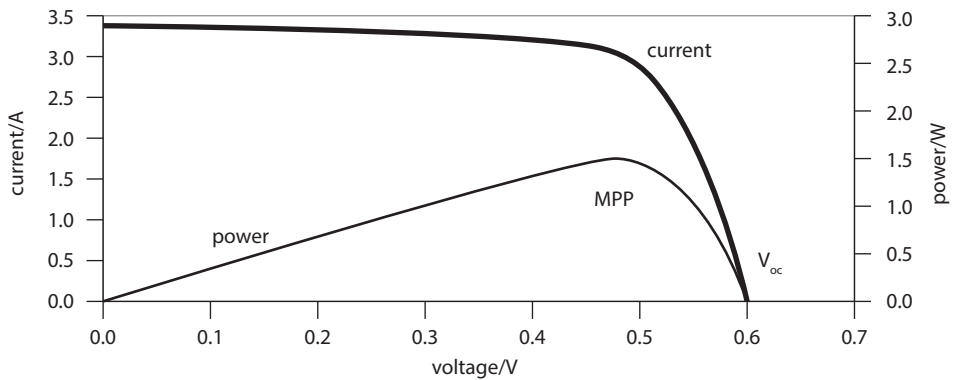


Figure 7.12 Current-voltage characteristic (bold) and power curve (thin) of a single monocrystalline solar cell with 100 cm² surface.

All characteristics were calculated using the simulation environment INSEL (Schumacher, 1991). The efficiency of a solar cell is calculated from the relation of the surface-related power produced at the MPP point and the solar irradiance G .

$$\eta = \frac{P_{MPP}}{A \times G} \quad (7.6)$$

Depending on the reference selected for the surface A , the cell efficiency or the module efficiency is calculated.

Curve fittings to the current-voltage characteristic

The best curve fitting to measured current-voltage curves of a crystalline solar cell is obtained from the mathematical description of a circuit diagram characterised by the parallel connection of two diodes with the diode saturation currents I_{01} and I_{02} , and the diode factors n_1 and n_2 , the so-called two-diode model. In the circuit diagram a current source produces an irradiance-dependent photocurrent I_{ph} , part of which flows off at the diodes due to charge carrier recombination. The current loss caused by low resistance at the edges of the solar cell is characterised by the shunt resistance R_p , which lies parallel to the diodes and the current source. If the voltage at the contacts of the solar cell, i.e. at the consumer, is called V , then the somewhat higher voltage $V + I/R_s$ applies to all the parallel components. R_s is the series resistance of the solar cell across which a voltage drop develops proportionally to the current I .

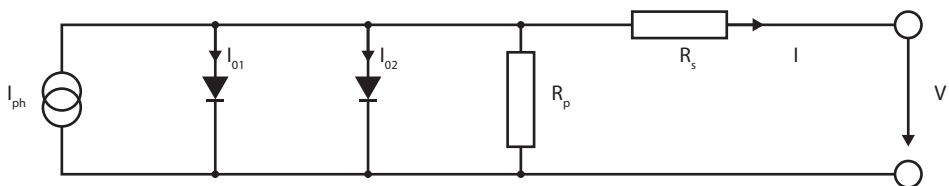


Figure 7.13 Alternate circuit diagram of a solar cell in the two-diode model.

The two-diode model is represented by an implicit equation of the current, which can only be solved iteratively.

$$I = I_{ph} - I_{01} \left(\exp \left(\frac{q(V + IR_s)}{n_1 k T} \right) - 1 \right) - I_{02} \left(\exp \left(\frac{q(V + IR_s)}{n_2 k T} \right) - 1 \right) - \frac{V + IR_s}{R_p} \quad (7.7)$$

n_1, n_2	diode factors [-]
I_{01}	saturation current from base (p-silicon) and emitter (n-silicon) [A]
I_{02}	saturation current in the space-charge zone [A]
I_{ph}	photocurrent of the solar cell [A]
R_s	series resistance [Ω]
R_p	shunt resistance [Ω]
k	Boltzmann constant [$1.38046 \times 10^{-23} \text{ J K}^{-1}$]
q	elementary charge [$1.602 \times 10^{-19} \text{ C}$]
T	temperature [K]

Diode factors and saturation currents

The two diode equations with the saturation currents I_{01} and I_{02} and diode factors n_1 and n_2 describe diffusion and recombination characteristics of the charge carriers in the material itself (index 1) and in the space-charge zone (index 2). Photo-generated charge carriers diffuse as minority carriers into the space charge region and then pass into the majority carrier region, i.e. electrons into the n-layer and holes into the p-layer. The existing potential barrier prevents back diffusion of the majority carriers.

If the charge carriers lose their energy during diffusion by dropping back again directly into the valence band or recombining via defect levels in the band gap, the saturation currents rise and the usable photocurrent is reduced.

The saturation current I_{01} depends on the diffusion coefficient and on the life span of the photo-generated charge carriers outside the space-charge zone. From the exponential dependency of the charge carrier concentration on the voltage, the diode factor $n_1 = 1$ results. The temperature dependence of the saturation current I_{01} is non-linear.

$$I_{01} = C_{01} T^3 \exp \left(-\frac{E_{gap}}{kT} \right) \quad (7.8)$$

with

C_{01}	Temperature coefficient obtained from parameter fitting [A K^{-3}]
E_{gap}	Band gap [J or eV], e.g. crystalline silicon: $1.124 \text{ eV} = 1.6 \times 10^{-19} \times 1.124 \text{ J} = 1.8 \times 10^{-19} \text{ J}$

The second diode describes the recombination of charge carriers in the space-charge zone. The saturation current I_{02} rises with charge carrier density and with the recombination rate. The diode

factor $n_2 = 2$ can be deduced by recombination of charge carriers at defect levels in the centre of the band gap. The temperature dependence of I_{02} is given by the following equation:

$$I_{02} = C_{02} T^{5/2} \exp\left(-\frac{E_{gap}}{2kT}\right) \quad (7.9)$$

The values of the temperature coefficients are obtained from parameter fits to the current-voltage characteristics. The value range for a cell with a surface of 100 cm^2 is typically between 150 and 180 A K^{-3} for C_{01} , and between 1.3 and $1.7 \times 10^{-2} \text{ A K}^{5/2}$ for C_{02} .

Series resistance

The series resistance R_s consists of the internal resistance of the solar cell and the contact resistance. Since the current density increases with irradiance, the power losses at the series resistance also rise. The surface-related series resistance $r_{s,cell}$ of an industrially manufactured solar cell with a surface of 100 cm^2 is typically about 10^{-4} to $10^{-5} \Omega \text{ m}^2$, approximately half being caused by metallisation at the front and the remainder by the internal cell resistance and other contact resistances. The entire series resistance of a solar module results from the surface-related cell resistance $r_{s,cell} [\Omega \text{ m}^2]$ multiplied by the number of cells N_s connected in series. The total series resistance $R_{s,total} [\Omega \text{ m}^2]$, often given as an absolute value, is obtained by:

$$\begin{aligned} R_{s,total} &= \frac{r_{s,cell} \times N_s}{A_{cell}} \quad [\Omega] \\ r_{s,total} &= r_{s,cell} \times N_s = R_{s,total} A_{cell} \quad [\Omega \text{ m}^2] \end{aligned} \quad (7.10)$$

In particular the series resistance influences the gradient of the current-voltage characteristic near the open circuit voltage and can be determined in a simplified model by the gradient of the characteristic at the open circuit voltage V_{oc} .

$$R_s \approx \left. \frac{dV}{dI} \right|_{V=V_{oc}} \quad (7.11)$$

With rising series resistance, the gradient near the open circuit voltage decreases and the current-voltage curves flatten. A module series resistance of 0.036Ω with 36 cells in series corresponds to a cell resistance of $10^{-5} \Omega \text{ m}^2$; 36Ω correspond to a cell resistance of $10^{-2} \Omega \text{ m}^2$.

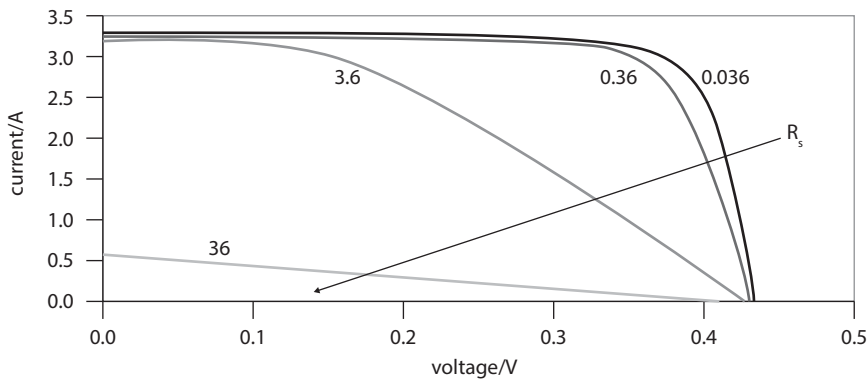


Figure 7.14 Influence of the series resistance $[\Omega]$ on the current-voltage characteristic of a module with 36 cells.

Shunt resistance

The parallel or shunt resistance R_p represents the leakage current, which is lost mainly in the p/n interface of the diode and along the edges. For a parallel connection of cells, the reciprocal resistances add up, i.e. for N parallel-connected cells and the same shunt resistances for each cell ($R_{p,1} = R_{p,2} = \dots = R_{p,np}$) the total resistance is as follows:

$$\begin{aligned} \frac{1}{R_{p,total}} &= \frac{1}{R_{p,1}} + \frac{1}{R_{p,2}} + \dots + \frac{1}{R_{p,np}} = \frac{N_p}{R_{p,cell}} \\ R_{p,total} [\Omega] &= \frac{R_{p,cell} [\Omega]}{n_p} = \frac{r_{p,cell} [\Omega m^2]}{A_{cell} [m^2] \times n_p} \\ r_{p,total} [\Omega m^2] &= \frac{r_{p,cell} [\Omega m^2]}{N_p} = R_{p,total} [\Omega] A_{cell} [m^2] \end{aligned} \quad (7.12)$$

Typical values of commercial silicon cells are between 0.1 and 10 Ωm^2 . The module manufacturers often give an absolute value in Ω for the shunt resistance.

The shunt resistance influences the gradient of the current-voltage characteristic within the area of the short circuit current and can be determined approximately from the gradient of the characteristic at the point of the short circuit current.

$$R_p \approx \left. \frac{dV}{dI} \right|_{V=0} \quad (7.13)$$

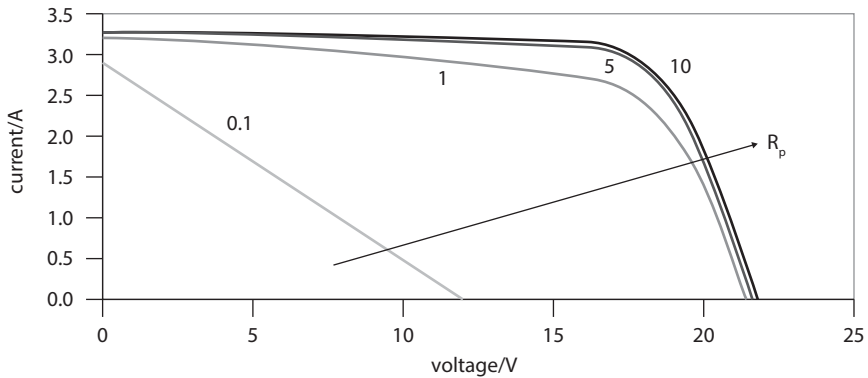


Figure 7.15 Influence of the shunt resistance $[\Omega]$ on the current-voltage characteristic for a module with 36 cells in series.

At a cell surface of 100 cm^2 , a module resistance of 0.1Ω corresponds to a cell resistance of $10^{-3} \Omega \text{ m}^2$, 10Ω correspond to a cell resistance of $10^{-1} \Omega \text{ m}^2$.

Example 7.2

On the data sheet of a large-scale module with 3 parallel strings of 54 cells each (100 cm^2 surface), a series resistance of 0.3Ω and a shunt resistance of 150Ω are indicated. From these values the surface-related cell resistances are to be calculated.

Series resistance:

$$r_{s, \text{cell}} [\Omega \text{ m}^2] = \frac{R_{s, \text{total}} [\Omega] \times A_{\text{cell}} [\text{m}^2]}{N_s} = \frac{0.3 \Omega \times 10^{-2} \text{ m}^2}{54} = 5.55 \times 10^{-5} \Omega \text{ m}^2$$

The surface-related total resistance of the module results from the product of the cell resistance and the number of cells in series, and is $3 \times 10^{-3} \Omega \text{ m}^2$.

Shunt resistance:

$$\begin{aligned} R_{p, \text{total}} [\Omega] &= 150 \Omega = \frac{r_{p, \text{cell}} [\Omega \text{ m}^2]}{A_{\text{cell}} [\text{m}^2] \times N_p} \\ r_{p, \text{cell}} [\Omega \text{ m}^2] &= 150 \Omega \times 10^{-2} \text{ m}^2 \times 3 = 4.5 \Omega \text{ m}^2 \\ r_{p, \text{total}} [\Omega \text{ m}^2] &= R_{p, \text{total}} [\Omega] A_{\text{cell}} [\text{m}^2] = \frac{r_{p, \text{cell}} [\Omega \text{ m}^2]}{N_p} = 1.5 \Omega \text{ m}^2 \end{aligned}$$

Photocurrent

The photocurrent of a solar cell depends not only on the wavelength-dependent absorption coefficients but also on the diffusion and recombination characteristics of the material. With

sufficient accuracy the photocurrent can be approximated as linear-dependent on irradiance and temperature.

$$I_{ph} = \frac{G}{G_{STC}} \left(I_{ph,STC} + \alpha_i (T_{PV} - T_{PV,STC}) \right) \quad (7.14)$$

G_{STC} Irradiance under standard test conditions [1000 W m⁻²]

$T_{PV,STC}$ Module temperature under standard test conditions [25°C]

In the data sheets of the module manufacturers, the temperature coefficient of the current per kelvin is indicated as a percentage or in absolute units [AK⁻¹]. The temperature coefficient α_i is usually very small and positive (about 0.03 - 0.04% of the short circuit current per kelvin). Neglecting the very small diode current, the photocurrent can be equated to the short circuit current I_{SC} .

Both the diode currents and the photocurrent are temperature dependent. This leads, particular in thin-film technologies, to non-constant temperature coefficient of the MPP power. The temperature coefficients are dependent on the absolute level of irradiance and temperature. For a quick estimate a constant temperature coefficient of the power loss is useful. According to the manufacturers, the temperature coefficient of the power varies between -0.3% and -0.5% per kelvin for crystalline technologies, -0.19% and -0.24% per kelvin for amorphous silicon, 0.25% per kelvin for CdTe and -0.3% and -0.36% per kelvin for CIS.

Parameter adjustment from module data sheets

To simplify parameter adjustment, the two-diode model can be reduced to a one-diode model in which, according to the Shockley theory, recombination in the space-charge zone is neglected, so the second diode term is omitted. The current-voltage characteristic is thereby simplified to the following implicit equation:

$$I = I_{ph} - I_0 \left(\exp \left(\frac{q(V + IR_s)}{nkT} \right) - 1 \right) - \frac{V + IR_s}{R_p} \quad (7.15)$$

This equation contains six free parameters. In the data sheets of the module manufacturers, however, usually only three operating points are indicated on the current-voltage characteristic (short circuit current at $V = 0$, open circuit voltage for $I = 0$, and the MPP values I_{MPP} , V_{MPP}) plus the temperature coefficients of the voltage α_v and the current α_i . The sixth equation follows from the fact that the MPP describes the maximum output of the module, i.e. the derivation of the power to the voltage (at the MPP) is zero:

$$dP / dV \big|_{V=V_{MPP}} = 0 \quad (7.16)$$

To avoid numerical problems in calculating the photocurrent using the exponential term, all currents are normalised to the respective cell surfaces, thus calculated with current densities i .

The current-voltage equation to fit the parameters is given with:

$$i = i_{ph} - i_s \left(\exp \left(\frac{q(V + ir_s)}{n_s kT} \right) - 1 \right) - \frac{V + ir_s}{r_p} \quad (7.17)$$

The current density is obtained from the irradiance G and temperature T :

$$i_{ph} = (c_{ph} + c_t T) G \quad (7.18)$$

and the saturation current of the Shockley-diode with:

$$i_s = c_s T^3 \exp\left(-\frac{E_g}{kT}\right) \quad (7.19)$$

The six free parameters to be determined are the irradiance and temperature dependence of the photocurrent c_{ph} , c_t , the Shockley diode factor n_s , the coefficient of the saturation current c_s and the serial and parallel resistances r_s and r_p .

Parameters 1 and 2 (c_{ph} and c_t):

Neglecting the series resistance, the photocurrent and the short-circuit current at $V = 0$ are equal. For N_p parallel cells and a cell area A_c , the photocurrent i_{ph} is given with:

$$i_{ph} \approx \frac{I_{sc}}{N_p A_c} = (c_{ph} + c_t T_{STC}) G_{STC} \quad (7.20)$$

The first two parameters of the photocurrent are obtained directly from the short circuit current given in the data sheet and the temperature coefficient a_i according to Equation (7.14). If a temperature change of 1 K is assumed for a constant irradiance, this leads to:

$$(c_{ph} + c_t (T_{STC} + 1)) G_{STC} = \frac{I_{sc} + a_i}{N_p A_c} \quad (7.21)$$

If Equation (7.20) is subtracted from Equation (7.21), we obtain the temperature coefficient of the photocurrent c_t .

$$c_t = \frac{a_i}{N_p A_c G_{STC}} \quad (7.22)$$

If this result is inserted into Equation (7.20), it leads to the irradiance coefficient of the photocurrent c_{ph} .

$$c_{ph} = \frac{1}{G_{STC}} \left(\frac{I_{sc, STC} - a_i T_{STC}}{N_p A_c} \right) \quad (7.23)$$

Parameter 3 (n_s):

The Shockley diode factor n_s is obtained from the diode equation and the temperature dependence of the voltage a_v . Because of the numerical sensitivity of the exponential function, it is advisable to calculate the equation for a single cell, i.e. the voltage is divided by the numbers of cells connected in series.

$$V_{oc} = \frac{V_{oc, STC} + a_v (T - T_{STC})}{n_s} \quad (7.24)$$

At the point of the open circuit voltage, the current is equal to zero, so that:

$$0 = i_{ph} - i_s \left(\exp \left(\frac{qV_{oc}}{n_s kT} \right) - 1 \right) - \frac{V_{oc}}{r_p} \quad (7.25)$$

With today's typically very high parallel resistances, the last term can be neglected. The exponential term is much larger than 1, so the deducted saturation current is also negligible (kT/q is about 25 mV at room temperature, while the open circuit voltage of a cell is about 0.5 V). This results in:

$$V_{oc} = \frac{n_s kT}{q} (\ln(i_{ph}) - \ln(i_s)) \quad (7.26)$$

With $dV_{oc}/dT = a_v/N_s$ and the differential rule $(uv)' = u'v + uv'$ it leads to:

$$\alpha_v := \alpha_v / N_s = \frac{n_s k}{q} \left(\ln(i_{ph}) - \ln(i_s) + T \left(\frac{1}{i_{ph}} \frac{di_{ph}}{dT} - \frac{1}{i_s} \frac{di_s}{dT} \right) \right) \quad (7.27)$$

with

$$\frac{di_{ph}}{dT} = c_t G$$

and

$$\frac{di_s}{dT} = 3c_s T^2 \exp \left(-\frac{E_g}{kT} \right) + c_s T^3 \exp \left(-\frac{E_g}{kT} \right) \frac{E_g}{kT^2} = i_s \left(\frac{3}{T} + \frac{E_g}{kT^2} \right)$$

it results in

$$\alpha_v = \frac{V_{oc}}{T} + \frac{n_s k}{q} \left(\frac{T c_t G}{i_{ph}} - 3 - \frac{E_g}{kT} \right)$$

and finally

$$n_s = \frac{q \left(\alpha_v - \frac{V_{oc}}{T} \right)}{k \left(\frac{T c_t G}{i_{ph}} - 3 - \frac{E_g}{kT} \right)} \quad (7.28)$$

Parameter 4 (c_s):

The equation of the open circuit voltage Equation (7.25) leads directly to the coefficient of the saturation current according to Equation (7.19).

$$c_s = \left(i_{ph} - \frac{V_{oc}}{r_p} \right) \left(T^3 \exp \left(-\frac{E_g}{kT} \right) \left(\exp \left(\frac{qV_{oc}}{n_s kT} \right) - 1 \right) \right)^{-1} \quad (7.29)$$

Since the parallel resistance r_p is not yet known, it is first assumed with a value of $0.1 \Omega \text{ m}^2$, so that c_s can be calculated. In further iterations c_s is newly calculated.

Parameter 5 (r_s):

The power density p (W m^{-2}) results from the current-voltage-characteristic according to Equation (7.17)

$$p = i_{ph}V - i_s V \left(\exp \left(\frac{q(V + i_r r_s)}{n_s kT} \right) - 1 \right) - \frac{V^2 + i_r V r_s}{r_p}$$

From the first derivative of the power at the MPP follows:

$$\begin{aligned} \left. \frac{dp}{dV} \right|_{V_m} = 0 &= i_{ph} - i_s \left(\exp \left(\frac{q(V_m + i_m r_s)}{n_s kT} \right) - 1 \right) \\ &\quad - i_s V_m \frac{q}{n_s kT} \left(\exp \left(\frac{q(V_m + i_m r_s)}{n_s kT} \right) - 1 \right) - \frac{2V_m + i_m r_s}{r_p} \end{aligned}$$

Since this equation is zero, it can be subtracted from any equation, e.g. of the current equation at the MPP:

$$i_m = i_{ph} - i_s \left(\exp \left(\frac{q(V_m + i_m r_s)}{n_s kT} \right) - 1 \right) - \frac{V_m + i_m r_s}{r_p}$$

This results in

$$i_m = \frac{i_s q V_m}{n_s kT} \left(\exp \left(\frac{q(V_m + i_m r_s)}{n_s kT} \right) - 1 \right) + \frac{V_m}{r_p}$$

and thus for the series resistance

$$r_s = \frac{n_s kT}{i_m q} \ln \left(1 + \frac{n_s kT}{i_s q V_m} \left(i_m - \frac{V_m}{r_p} \right) \right) + \frac{V_m}{i_m} \quad (7.30)$$

The MPP voltage V_m is specified here for a single cell.

Parameter 6 (r_p):

Now, all other parameters are known, the parallel resistance of the current-voltage equation at the MPP can be calculated directly:

$$r_p = \frac{V_m + i_m r_s}{i_{ph} - i_s \left(\exp \left(\frac{q(V_m + i_m r_s)}{n_s kT} \right) - 1 \right) - i_m} \quad (7.31)$$

Finally, the Shockley parameter needs to be iterated one or several times.

Example 7.3

The six parameters of the one-diode model should be determined for a module with the following characteristics:

Nominal power	85	W
MPP-current I_m	4.9	A
MPP-voltage V_m	17.4	V
Short circuit current I_{SC}	5.48	A
Open circuit voltage V_{oc}	22.0	V
Temperature coefficient of the current α_I	3.51	mA K ⁻¹
Temperature coefficient of the voltage α_V	-80	mV K ⁻¹
NOCT	47.2	°C
Cell area A_c	0.015625	m ²
Number of cells in series n_s	36	
Number of parallel strings n_p	1	
Module area	0.649	m ²
Weight	7.7	kg
Schottky diodes (9 A, 45 V)	2	

Table 7.1 Datasheet of a BP 485 module.

Parameter 1 and 2:

The irradiance coefficient of the photocurrent is:

$$c_{ph} = \frac{1}{1000 \text{ W m}^{-2}} \cdot \frac{5.48 \text{ A} - 0.00351 \text{ A K}^{-1} \times 298.15 \text{ K}}{1 \times 0.015625 \text{ m}^2} = 0.2837 \text{ V}^{-1}$$

and the temperature coefficient:

$$c_I = \frac{0.00351 \text{ A K}^{-1}}{1 \times 0.015625 \text{ m}^2 \times 1000 \text{ W m}^{-2}} = 0.2246 \times 10^{-3} \text{ V}^{-1} \text{ K}^{-1}$$

Parameter 3:

The Shockley diode factor is determined from Equation (7.28). The numerator is:

$$1.602 \times 10^{-19} \text{ As} \left(-\frac{0.08 \text{ V K}^{-1}}{36} - \frac{22 \text{ V}}{36 \times 298.15 \text{ K}} \right) = -6.844 \times 10^{-22} \text{ J K}^{-1}$$

and the denominator is:

$$1.381 \times 10^{-23} \text{ J K}^{-1} \left(\frac{298.15 \text{ K} \times 0.22464 \times 10^{-3} \text{ V}^{-1} \text{ K}^{-1} \times 1000 \text{ W m}^{-2}}{5.48 \text{ A} / 0.015625 \text{ m}^2} - 3 - \frac{1.602 \times 10^{-19} \text{ As} \times 1.12 \text{ V}}{1.381 \times 10^{-23} \text{ J K}^{-1} \times 298.15 \text{ K}} \right) \\ = -6.406 \times 10^{-22} \text{ J K}^{-1}$$

such that:

$$n_s = \frac{-6.844}{-6.406} = 1.068$$

Parameter 4:

The coefficient c_s is the saturation current calculated from the following terms.

$$T^3 \exp\left(-\frac{E_g}{kT}\right) = 298.15^3 K^3 \times \exp\left(-\frac{1.602 \times 10^{-19} \text{ As} \times 1.12 \text{ V}}{1.381 \times 10^{-23} \text{ JK}^{-1} \times 298.15 \text{ K}}\right) \\ = 3.15 \times 10^{-12} K^3$$

and

$$\exp\left(\frac{qV_{oc}}{n_s kT}\right) - 1 = \exp\left(\frac{1.602 \times 10^{-19} \text{ As} \times 22 \text{ V} / 36}{1.068 \times 1.381 \times 10^{-23} \text{ JK}^{-1} \times 298.15 \text{ K}}\right) - 1 \\ = 4.663 \times 10^9$$

Such that:

$$c_s = \left(\frac{5.48 \text{ A}}{0.015625 \text{ m}^2} - \frac{22 \text{ V} / 36}{0.1 \Omega \text{ m}^2} \right) \frac{1}{3.15 \times 10^{-12} K^3 \times 4.663 \times 10^9} \\ = (350.72 - 0.6111 / r_p) \times 68.081 = 23461 \text{ Am}^{-2} K^{-3}$$

Parameter 5:

The series resistance is calculated from

$$\frac{n_s kT}{q} = \frac{1.068 \times 1.381 \times 10^{-23} \text{ JK}^{-1} \times 298.15 \text{ K}}{1.602 \times 10^{-19} \text{ As}} = 27.450 \times 10^{-3} \text{ V}$$

and

$$i_s = 23461 \text{ Am}^{-2} K^{-3} \times 298.15^3 K^3 \exp\left(\frac{-1.602 \times 10^{-19} \text{ As} \times 1.12 \text{ V}}{1.381 \times 10^{-23} \text{ JK}^{-1} \times 298.15 \text{ K}}\right) \\ = c_s \times 3.1498 \times 10^{-12} = 7.390 \times 10^{-8} \text{ Am}^{-2}$$

such that:

$$r_s = \frac{27.450 \times 10^{-3} \text{ V}}{4.9 \text{ A} / 0.015625 \text{ m}^2} \ln \left(1 + \frac{27.450 \times 10^{-3} \text{ V}}{7.390 \times 10^{-8} \text{ Am}^{-2} \times 17.4 \text{ V} / 36} \left(4.9 \text{ A} / 0.015625 \text{ m}^2 - \frac{17.4 \text{ V} / 36}{0.1 \Omega \text{ m}^2} \right) \right) \\ - \frac{17.4 \text{ V} / 36}{4.9 \text{ A} / 0.015625 \text{ m}^2} \\ = 8.7532 \times 10^{-5} \times \ln \left(1 + \frac{5.6793 \times 10^{-2}}{i_s} \left(313.6 - \frac{0.4833}{r_p} \right) \right) - 1.5412 \times 10^{-3} = 1.468 \times 10^{-4} \Omega \text{ m}^2$$

Parameter 6:

The parallel resistance is calculated with

$$\begin{aligned} V_m + i_m r_s &= 17.4V / 36 + 1.468 \times 10^{-4} Wm^2 \times 4.9A / 0.015625m^2 \\ &= 0.5294V \end{aligned}$$

and

$$\begin{aligned} i_s &\left(\exp \left(\frac{q(V_m + i_m r_s)}{n_s kT} \right) - 1 \right) \\ &= 7.390 \times 10^{-8} Am^{-2} \times \left(\exp \left(\frac{0.5294V}{27.450 \times 10^{-3} V} \right) - 1 \right) \\ &= 17.556 Am^{-2} \end{aligned}$$

such that:

$$\begin{aligned} r_p &= \frac{0.5294V}{5.48A / 0.015625m^2 - 17.556 Am^{-2} - 4.9A / 0.015625m^2} \\ &= 0.0271 \Omega m^2 \end{aligned}$$

The first iteration is given with:

$$\begin{aligned} c_s &= (350.72 - 0.6111 / 0.0271) \times 68.081 = 22342 Am^{-2} K^{-3} \\ i_s &= 22342 \times 3.1498 \times 10^{-12} = 7.037 \times 10^{-8} Am^{-2} \\ r_s &= 8.7532 \times 10^{-5} \times \ln(1 + 8.0706 \times 10^5 \times 295.76) - 1.5412 \times 10^{-3} \\ &= 1.473 \times 10^{-4} \Omega m^2 \\ V_m + i_m r_s &= 313.6 \times 0.4833 + 1.473 \times 10^{-4} = 0.5295V \\ i_s &\left(\exp \left(\frac{q(V_m + i_m r_s)}{n_s kT} \right) - 1 \right) \\ &= 7.037 \times 10^{-8} Am^{-2} \times \left(\exp \left(\frac{0.5295V}{27.450 \times 10^{-3} V} \right) - 1 \right) \\ &= 16.779 Am^{-2} \end{aligned}$$

and

$$r_p = \frac{0.5295V}{350.72 - 16.779 - 313.6} = 0.0260 \Omega m^2$$

The value is already very close to the first value of $0.0271 \Omega m^2$. The results are summarised in the following table.

Irradiance coefficient of the photocurrent c_{ph}	0.2837	V ⁻¹
Temperature coefficient of the photocurrent c_t	0.2246×10^{-3}	V ⁻¹ K ⁻¹
Shockley diode factor n_s	1.068	
Shockley diode saturation coefficient c_s	22 342	A m ⁻² K ⁻³
Series resistance r_s	1.473×10^{-4}	Ω m ²
Parallel resistance r_p	0.026	Ω m ²

Table 7.2 Parameter of single-diode-model (BP485).

With these parameters, it is possible to calculate current-voltage-characteristics at different irradiances and temperatures.

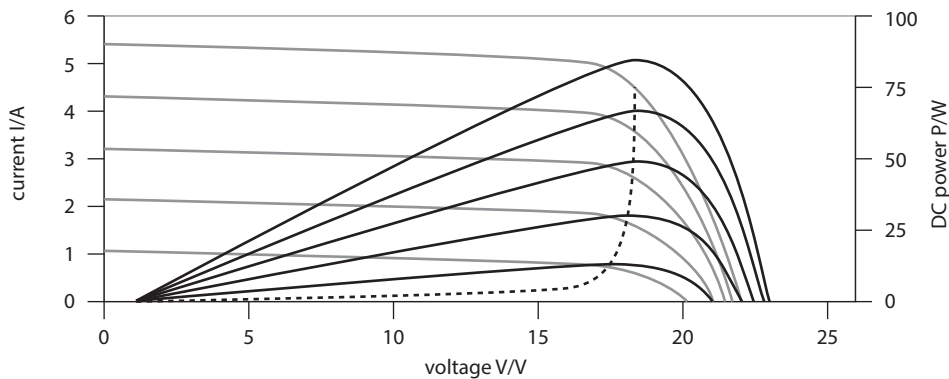


Figure 7.16 Calculated current-voltage characteristics and resulting power at a fixed module temperature of 25°C and irradiance values of 200, 400, 600, 800 and 1000 W/m². The exponential curve shows the location of the maximum power points (MPP) for various characteristics.

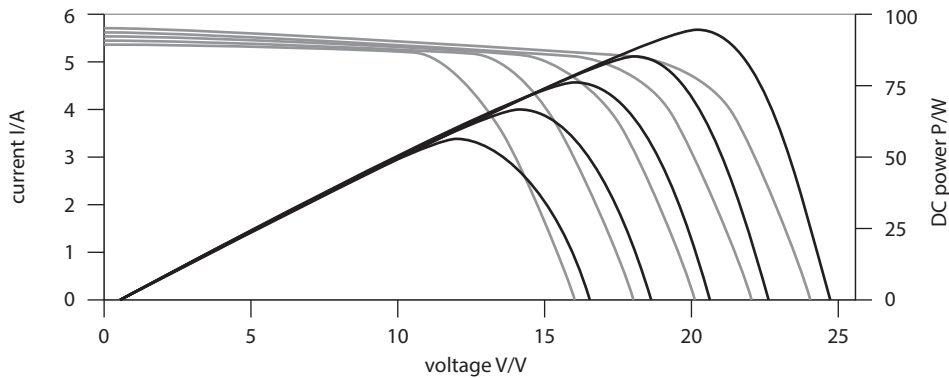


Figure 7.17 Calculated current-voltage characteristics and resulting power at a fixed irradiance level of 1000 W/m² and module temperature of 0°C, 25°C, 50°C, 75°C and 100°C.

Furthermore, the calculated parameters are compared to the module data sheet specifications.

Parameters:	Calculated	Data sheet	
Nominal power	85.43	85.0	W
MPP voltage V_m	17.54	17.4	V
MPP current I_m	4.87	4.9	A
Short circuit current I_{SC}	5.44	5.48	A
Open circuit voltage V_{oc}	22.04	22.0	V

Table 7.3 Comparison of calculated and extracted data from the module parameters of the BP module 485.

With the parameter identification procedure, it is possible to determine automatically the characteristics of all commercially available modules. This method is implemented in the simulation environment INSEL (www.insel.eu). As an example, the spread of the series resistances of the cells of 2000 market available modules is displayed.

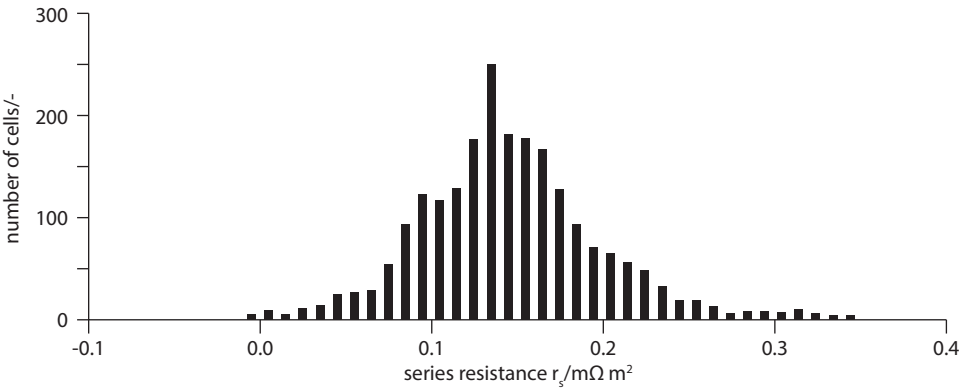


Figure 7.18 Distribution of series resistances of 2000 modules from the Photon magazine database.

Curve fitting of the two-diode model

For an adjustment of the six unknown parameters of the two-diode model to measured data, the error function of the measured currents at a given voltage, irradiance and temperature and of the simulated currents must be minimised.

$$\Phi = \sqrt{\sum_{i=1}^N (I_{i,meas} - I_{i,calc})^2} \tag{7.32}$$

With little mathematical effort, i.e. a linear regression, only the parameters $I_{ph,STC}$ and a_I can be calculated from the linear irradiance-dependence and temperature-dependence of the short circuit current. The remaining parameter set can only be determined by gradient, raster or genetic procedures (Pukrop, 1997).

Example 7.4

Calculation of the parameters $I_{ph,STC}$ and a_1 of a Siemens M55 module from the following table of values measured.

Irradiance G / $W\ m^{-2}$	Module temperature T_{PV} / $^{\circ}C$	Short circuit current I_{SC} / A
220	30	0.74
510	41	1.72
850	54	2.88
1000	60	3.40

To carry out a linear regression of the temperature dependence of the short circuit current using Equation (7.14), the irradiance G must be eliminated from the equation, so that only the temperature difference from standard test conditions remains as an independent parameter. The short circuit current is thus divided by the irradiance, and the regression for I_{SC}/G against $T_{PV}-T_{PV,STC}$ is carried out.

The linear regression produces as axis intercept $I_{ph,STC}/G_{STC} = 0.0034\ A\ W^{-1}\ m^2$, i.e. a short circuit current $I_{ph,STC}$ of 3.4 A, and as a gradient $I_{ph,STC}/G_{STC}\ a_1 = 1 \times 10^{-6}\ A\ W^{-1}\ m^2\ K^{-1}$, i.e. $a_1 = 3 \times 10^{-4}\ K^{-1}$ (+0.03%/K).

Current/irradiance $I_{ph,STC}/G_{STC}$ / $A\ W^{-1}\ m^2$	Temperature difference $T_{PV}-T_{PV,STC}$ / K
3.364×10^{-3}	5
3.373×10^{-3}	16
3.388×10^{-3}	29
3.400×10^{-3}	35

Only by simplifying, i.e. linearisation of the single or two-diode model equation, all six parameters can be calculated by the least squares minimisation. The current-voltage characteristics, for example, described by the two-diode model equation cannot be linearised, since the series resistance occurs in both the exponential and the Shockley-diode as well as in the linear term of the parallel resistance.

$$i = i_{ph} - i_s \left(\exp \left(\frac{q(V + i r_s)}{kT} \right) - 1 \right) - i_r \left(\exp \left(\frac{q(V + i r_s)}{2kT} \right) - 1 \right) - \frac{V + i r_s}{r_p} \quad (7.33)$$

A linearisation is only possible if the series resistance is assumed to be known. Then, Equation (7.33) is transformed into:

$$c_s a_1 (V, T, i, r_s) + c_r a_2 (V, T, i, r_s) + \frac{1}{r_p} a_3 (V, T, i, r_s) = b \quad (7.34)$$

with

$$\begin{aligned}
 a_1 &= T^3 \exp\left(\frac{-E_g}{kT}\right) \left(\exp\left(\frac{q(V+ir_s)}{kT}\right) - 1 \right) \\
 a_2 &= T^{5/2} \exp\left(\frac{-E_g}{2kT}\right) \left(\exp\left(\frac{q(V+ir_s)}{2kT}\right) - 1 \right) \\
 a_3 &= V + ir_s \\
 b &= i_{ph} - i
 \end{aligned}$$

If a data set of N data tuple (G_j, T_j, V_j, i_j) , $j = 1, \dots, N$ is available from measured I-V characteristics, a $3 \times N$ matrix can be formed.

$$A = \begin{pmatrix} a_{1,1} & a_{2,1} & a_{3,1} \\ a_{1,2} & a_{2,2} & a_{3,2} \\ \vdots & \vdots & \vdots \\ a_{1,N} & a_{2,N} & a_{3,N} \end{pmatrix}$$

Then the three unknown parameters c_s , c_r and r_p can be expressed as a vector $x = (c_s, c_r, 1/r_p)^T$ and the right side of the equation is also a vector $b = (b_1, b_2, K b_N)^T$.

In matrix notation, it leads to an overdetermined equation system, which is solved for the sought vector x .

$$Ax = b$$

The best solution for x is that which minimises the squared error sum.

$$\chi^2 = \sum_{j=1}^N \left(\sum_{i=1}^3 a_{ij} x_i - b_j \right)^2 \quad (7.35)$$

Using a genetic algorithm, the series resistance is varied until the solution is found.

If the current-voltage or power characteristics of an SM55 module with characteristic values from the parameter identification process are compared with the exact calculations of the two-diode model, then a good agreement of the characteristics can be seen. The one-diode model overrates the performance by 0.6 - 1.4 W, which at high irradiances amounts to 1.5%, and at low irradiances 15%.

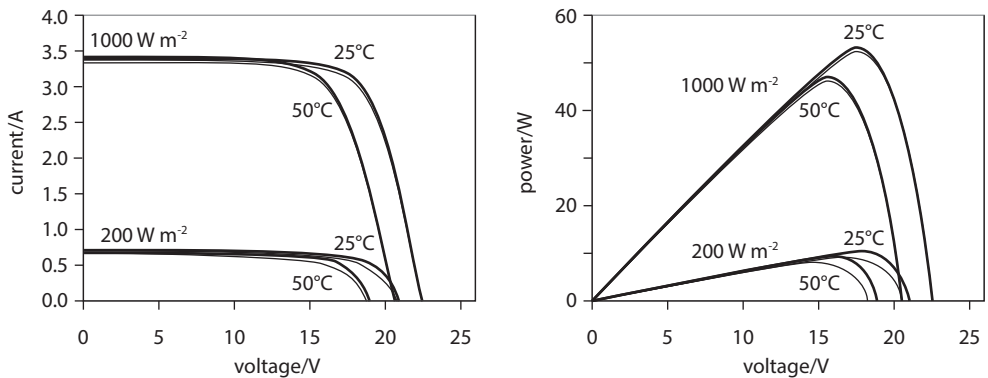


Figure 7.19 Comparison of the current-voltage and of the power characteristic of a SM55 module using the parameter identification procedure (bold curves) and the two-diode model (thin curves).

I-V characteristic model for thin-film modules

The IV characteristics of thin-film modules differ from crystalline modules, which is due to several factors.

First the temperature effect of some thin-film cells tends to be lower than for crystalline modules, which means that power losses are lower for high-temperature levels.

Second, due the high band gap and therefore narrow wavelength interval for absorption (especially for amorphous silicon cells), the influence of the spectrum on the photocurrent is higher than for broadband absorbers such as crystalline silicon or CIS. For pure diffuse light with a very high proportion of short-wave blue irradiance in the spectrum, the performance is up to 20% higher than for an AM 1.5 spectrum.

Furthermore, in some thin-film modules, the efficiency decreases with irradiance due to low charge carrier mobility and high internal resistances. Since the nominal power is measured at STC conditions, i.e. high radiation, at low irradiance the efficiency increases. This means that the low-light performance is, for example, better in the case of amorphous silicon modules than for crystalline silicon modules.

For crystalline modules, the efficiency at 10% of the maximum STC radiation ($\approx 100 \text{ W m}^{-2}$) decreases by about 5%. For CdTe the low-light performance is worse than for crystalline modules: at 10% of the radiation the efficiency drops more than 10% below the nominal efficiency. For amorphous silicon the low light efficiency might even be higher than at STC.

To describe the performance of thin-film modules correctly over the whole irradiance and temperature range, neither the single-diode model nor the two-diode model are suitable. To this end more than six free parameters in the IV curve model are necessary. Especially the voltage

dependence of the short circuit current has to be considered, which is parameterised by the free parameter m and the internal voltage V_{bi} .

$$i_{ph}^+ = i_{ph} \left(1 - \left(m(V_{bi} - (V + i r_s)) \right)^{-1} \right) \\ = (c_g + c_t T) G \left(1 - \frac{1}{m(V_{bi} - (V + i r_s))} \right) \quad (7.36)$$

To describe thin-film modules over the entire irradiation and temperature range, more free parameters are needed, such as the band gap E_{gap} and the temperature dependence of the diode current, which contains the properties of two cell types in the case of tandem cells.

Then the diode term contains four free parameters: c_0 , c_1 , E_{gap} and n .

$$i_0 = c_0 T^{c_1} \exp \left(- \frac{E_{gap}}{n k T} \right) \quad (7.37)$$

Altogether, the IV curve model has now 10 free parameters. To fit the IV curve, a simplex algorithm according to Melder Nead is used.

As an example, measured characteristics of an amorphous/microcrystalline tandem cell module are fitted with the extended model, at irradiances of 50, 100, 200, 300 up to 1000 W/m² with three temperature levels of 10, 25 and 40°C.

The data sheet specifications for the tandem module are:

Parameter	Data sheet value	
Nominal power	128	W
MPP voltage V_m	45.4	V
MPP current I_m	2.82	A
Short circuit current I_{SC}	3.45	A
Open circuit voltage V_{oc}	59.8	V
Temperature coefficient of the current α_i	0.07	% K ⁻¹
Temperature coefficient of the voltage α_V	-0.24	% K ⁻¹

Table 7.4 Specifications of a tandem module.

Figure 7.20 shows the very good agreement between measured and calculated IV characteristics.

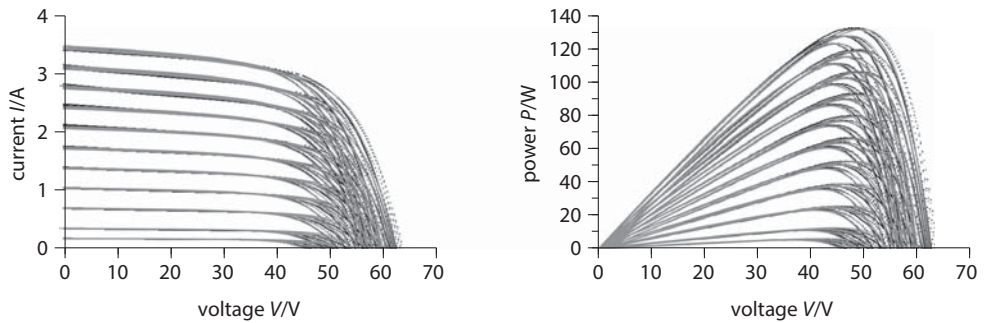


Figure 7.20 Measured (black) and calculated (grey) I - V characteristics and power of an amorphous/microcrystalline tandem module.

Spectral- and temperature-related effects of thin-film technologies

Some thin-film technologies have narrow spectral responses, show a temperature-related reversible degradation and generally have lower temperature-related power losses than crystalline technologies.

For amorphous silicon, degradation takes place due to dissociation of bonds in the material. This so-called Staebler Wronski effect is reversible at high temperatures. Complete annealing is achieved for temperature levels above 120°C , but partial annealing takes place at temperature levels of $60 - 80^{\circ}\text{C}$. Therefore, building-integrated amorphous silicon modules without back ventilation can have a 6 - 9% higher energy yield despite slightly negative temperature coefficients.

Typical annual efficiency oscillations due to the spectral effect are 8 - 20%. The spectral effect and the Staebler Wronski degradation effect have a similar phase (maximum annealing and "best" blue spectrum in summer and highest degradation and worst spectral response in winter). Simulation studies (Skoczek et al., 2011) show that in colder climates the spectral effect dominates, whereas in warm regions the temperature-related power losses and annealing are more important to explain the power oscillations of amorphous modules.

The spectral effect can be modeled using the spectral simulation model described in the solar resource chapter. The spectral response or quantum efficiency of a solar cell describes the wavelength dependent absorption of a solar cell. As an example the measured spectral response of two cells of an amorphous/microcrystalline tandem cell are shown.

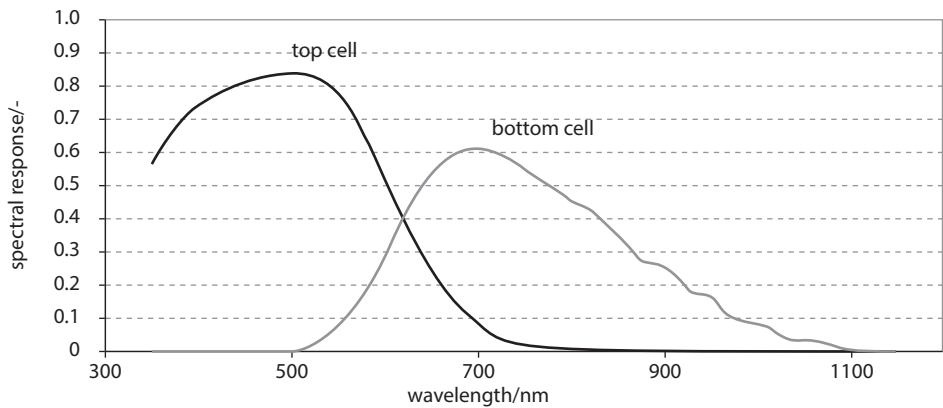


Figure 7.21 Measured spectral response of the two cells of a tandem solar cell.

To show the impact of seasonal spectrum change on the power output, monthly mean irradiance weighted spectra were calculated for the summer month June and the winter month December for a Southern German location (see Figure 3.18, Simulated mean monthly spectral irradiance as a function of wavelength for the German location Munich, on p. 168).

To analyse the current changes for different spectral irradiances, the monthly spectrum was multiplied, i.e. weighted, with the quantum efficiency of the top and bottom. To calculate the current density of the bottom cell, the incident spectrum is first multiplied with the quantum efficiency of the top cell for each wavelength, which gives the absorption of the top cell. Only the remaining transmitted spectrum is used for the bottom cell and multiplied with the quantum efficiency of the bottom cell.

The integration over the wavelength range of 0.35 - 1.15 μm gives the average monthly intensities. For example, for December in the Northern German location of Hamburg, the average irradiance is 45.7 W m^{-2} , of which the top cell absorbs 16 W m^{-2} or 35%. The current generated by this absorbed irradiance is obtained by calculating the number of photons for the given wavelength. A high-energy short-wave photon generates an electron hole pair, which then loses energy by thermalisation to the bandgap. Recombination losses are not considered.

The remaining 29.7 W m^{-2} are transmitted to the bottom cell, which absorbs 9.2 W m^{-2} or 31%. Although less power is absorbed, the current density is comparable, as the low-energy absorbed photons generate electron-hole pairs close to the bandgap without losing so much energy by thermalisation.

If an AM1.5 spectrum would be used for all hours of the month, the ratio of weighted spectrum to incident spectrum for the top cell would be 36% and 31% for the bottom cell.

The results of all spectral simulations are summarised in the table below. In the summer months the current densities are always higher than the current calculated with the AM1.5 spectrum. In winter, the currents are lower than for an AM1.5 spectrum. In general the bottom cell is less sensitive to spectral changes than the top cell.

Location	Top		Bottom	
	June	December	June	December
Hamburg	5.52	-3.04	2.56	-0.32
Munich	4.14	-12.15	1.28	-1.28
Barcelona	6.63	-12.43	2.88	1.28
Milano	6.63	-8.29	2.56	0.64
Antalya	8.29	-5.52	2.56	0.96

Table 7.5 Deviation in percent of monthly current from the current obtained with the AM1.5 spectrum for different European locations.

The temperature coefficient of single, double or triple junction amorphous silicon modules is rather low with -0.1 - 0.3 % per kelvin. Detailed measurements of amorphous silicon modules on an outdoor test site in Southern Germany (Lechner, 2012) gave temperature coefficients of 0.2% K⁻¹ for amorphous silicon and 0.29% K⁻¹ for CdTe.

For double junctions with a Si and microcrystalline materials the temperature coefficients are close to crystalline technologies with 0.26 - 0.43% K⁻¹.

Simple explicit model for system design

The IV curve obtained from the parameter identification can only be solved iteratively, even if the shunt resistance is assumed as infinite. Only by ignoring the series resistances a simple explicit relation between current and voltage is obtained from Equation (7.17) which then allows fast system design.

$$i = i_{ph} - i_s \left(\exp \left(\frac{qV}{n_s kT} \right) - 1 \right) \quad (7.38)$$

Of the four parameters c_t , c_{ph} , n_s and c_s , the temperature and irradiance coefficient of the photocurrent can be directly calculated using Equation (7.22) and Equation (7.23). The temperature coefficient of the saturation current c_s and the diode parameter n_s can be determined by three different methods:

1. For the parameter determination, the temperature dependence of the current a_i and the short circuit current equation for the photocurrent parameters c_t and c_{ph} are used. Furthermore, the temperature coefficient of the voltage a_V is used for the diode parameter n_s , and the open circuit voltage point for the temperature coefficient of the saturation current c_s according to Equation (7.29). The parameters are calculated using the above equations of the identification method, while the series resistance is set at zero and the parallel resistance is infinite. In this case the MPP condition is not used and the MPP performance is overestimated at all irradiances. However, the temperature dependence of the power and the open circuit voltage correspond well with real characteristics.

2. As above, qV is used for the diode parameter n_s . a_i and the short circuit current equation are used for the photocurrent parameters. However, the saturation current i_s and thus c_s are no longer calculated from the open circuit voltage condition, but at the MPP point:

$$i_s = \frac{i_{ph} - i_{MPP}}{\exp\left(\frac{qV_{MPP}}{n_s kT}\right) - 1} \quad (7.39)$$

This condition results in substantially better agreement for the power values; however the open circuit voltage does not correspond with the real characteristic.

3. Alternatively, the parameters can be determined from three operating points (with short circuit current, open circuit voltage and the MPP condition) and from the temperature dependence of the current. Here, however, the condition of a significant temperature dependence of the voltage is not used. The coefficient of the saturation current is calculated from the open circuit condition.

$$i_s = i_{ph} \left(\exp\left(\frac{qV_{oc}}{n_s kT}\right) - 1 \right)^{-1}$$

The diode parameter n_s is now obtained from the MPP condition. To solve the equations analytically, the subtrahend -1 is neglected when compared to the exponential term in the equation of the saturation current.

$$\begin{aligned} i_{MPP} &= i_{ph} - \frac{i_{ph}}{\exp\left(\frac{qV_{oc}}{n_s kT}\right)} \exp\left(\frac{qV_{MPP}}{n_s kT}\right) \\ \Rightarrow n_s &= \frac{q(V_{MPP} - V_{oc})}{kT \ln\left(1 - \frac{i_{MPP}}{i_{ph}}\right)} \end{aligned} \quad (7.40)$$

The use of the MPP condition results in correct performances at 25°C. At other irradiances and above all different module temperatures, however, the performance is calculated with errors over 20%, so this calculation method is not recommended.

In the following, a comparison of the IV curve calculation is carried out using the exact two-diode model with series and parallel resistance and the three methods of parameter identification based on the simple explicit model. For this, current-voltage characteristics and power curves have been calculated at 1000 W m⁻² irradiance and a module temperature of 50°C. Method 1 clearly overestimates the MPP performance, with the open circuit voltage near to the two-diode value. Method 3 underestimates the power and calculates the voltage level with a large error. The second method uses both the temperature dependence of the voltage and the MPP condition and results in by far the best curve fit. At 1000 W m⁻² irradiance and a module temperature of 50°C, the following performance curves are obtained for a standard module (calculated from manufacturer's data based on the simple explicit models and the two-diode model):

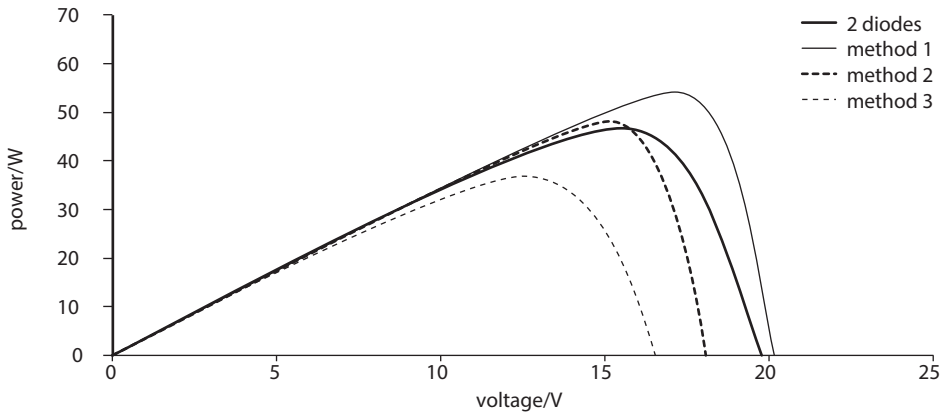


Figure 7.22 Comparison of the power curves of a SM55 module.

I-V characteristic addition and generator interconnecting

To calculate the current-voltage characteristic of a photovoltaic generator, the characteristics of the individual modules have to be added. Here the voltage contributions V_{ij} of the modules i of the string j concerned are added in serial connection at a common current. Since the currents through each module are equal in serial interconnecting, the current is limited by the short circuit current of the worst module (or the current passed through the bypass diodes).

In parallel interconnections of the strings, the generator characteristic results from adding the respective string currents I_j at a given voltage between zero and the string voltage.

Example 7.5

Calculation of PV generator power and voltage in the MPP point for a system with 36 modules of 53 W MPP power (17.4 V MPP voltage), which is divided into nine strings, each with 4 modules connected in series.

- Serial interconnecting: For the four series connected modules the MPP voltages are added at equal current ($I_{MPP} = 53 \text{ W} / 17.4 \text{ V} = 3.05 \text{ A}$).

$$V_{MPP, string} = V_{MPP, module 1} + V_{MPP, module 2} + \dots + V_{MPP, module 4} = 69.6 \text{ V}$$

- Parallel interconnection: For the nine strings joined in parallel, the string currents (in each case 3.05 A) are added at equal voltage ($V_{MPP, String} = 69.6 \text{ V}$).

$$I_{MPP, Generator} = I_{MPP, String 1} + I_{MPP, String 2} + \dots + I_{MPP, String 9} = 27.45 \text{ A}$$

The system thus has an MPP operating point under standard test conditions of 69.6 V and 27.45 A, i.e. a DC power of 1910 W.

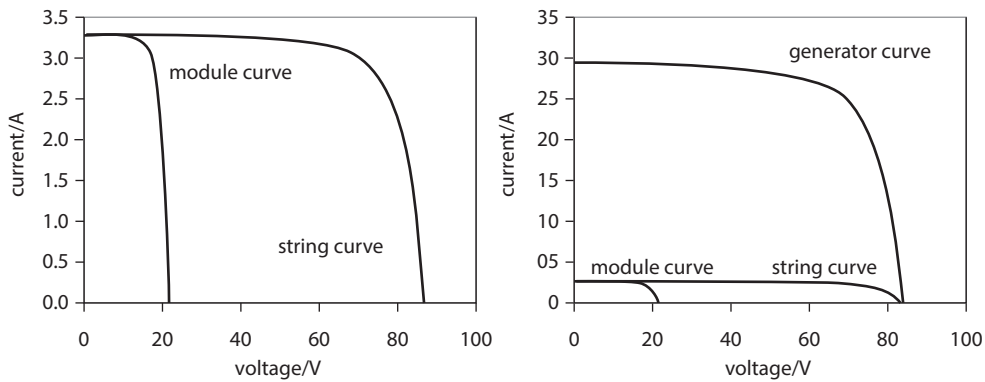


Figure 7.23 I-V characteristic of an individual module with thirty-six cells in series and addition of four series connected module IV curves to the string characteristic (left), and addition of nine parallel PV strings, each with four modules in series, to the generator characteristic (right).

7.8 PV performance with shading

Bypass diodes and backwards characteristics of solar cells

If individual cells of a module are shaded, they produce a lower photocurrent than the unshaded cells. Since during a series connection equal currents flow through all the cells, starting from zero current at open circuit voltage, the small photocurrent of the shaded cells at first limits the total current. The high photocurrents produced by the unshaded cells can be let through only within the backward voltage range of the shaded cells, with the shaded cells taking up power. Depending on the number of further cells connected in series, the backward voltage can become so high that the breakdown voltage in the shaded cell is achieved and irreversible damage is caused.

The breakdown voltage V_{Br} marks the backward voltage of a diode, at which an exponential current rise (avalanche breakdown with exponent m) takes place, in commercial solar cells between -10 V and -30 V. The power dissipation in the shaded cell can become so high locally that the cell and plastic encapsulation are damaged. The characteristic of a solar cell within the backward voltage range is described by an extension term in the one- or two-diode model:

$$I = I_{ph} - I_0 \left(\exp \left(\frac{q(V + IR_s)}{nkT} \right) - 1 \right) - \frac{V + IR_s}{R_p} \left(1 + \left(\frac{a}{1 - \frac{V + IR_s}{V_{Br}}} \right)^m \right) \quad (7.41)$$

with a as the empirical fit parameter. The parameters for the description of the characteristic in the backward voltage region scatter very strongly and are not a constituent of the usual data sheet specifications.

As an example, the measured breakdown voltage V_{Br} of a polycrystalline cell is -15 V, the exponent m is 3.7 and the fit parameter a is 0.1. The current increase in the backward voltage region resulting from these values is very steep: thus the maximum photocurrent of unshaded

cells of a standard module of 3.3 A is let through already at 8.3 V, which, as will be shown, puts in question the usual application of one bypass diode per 18 cells.

Firstly, a module characteristic without external bypass diodes will be shown, in order to demonstrate the problems of the power uptake of the shaded cells and the over proportional power losses of the module. From the total of 36 serial connected cells of the module, one cell is shaded with a remaining diffuse irradiance of 200 W m^{-2} , while all the other cells are irradiated with 1000 W m^{-2} . From the characteristic of the shaded cell and from the characteristic of the 35 unshaded cells already added, the module characteristic can be constructed.

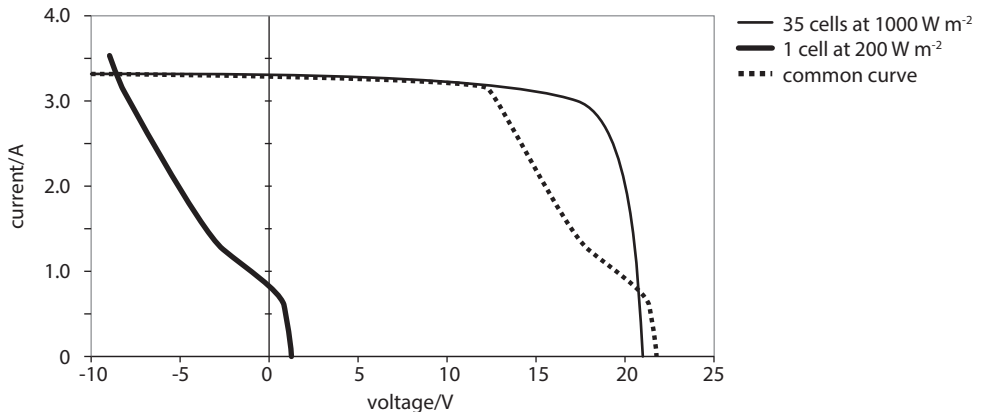


Figure 7.24 Construction of a module characteristic with a shaded cell (200 W m^{-2}) and 35 unshaded cells in series.

At a total current value $I = 0$, the open circuit voltage of the shaded cell and the unshaded cell are added up to a total voltage, which does not differ from an unshaded module. If the total value of current is now slowly increased, all the cells remain in the positive voltage range up to the short circuit current of the shaded cell. The shaded cell can only let through higher currents than their short circuit current at negative voltages, with the total voltage constantly decreasing. At a total voltage $V_{\text{total}} = 0$, the shaded cell is most strongly negatively polarised (here -8.3 V), while the unshaded cells display further positive voltages (namely $+8.3 \text{ V}$). The power uptake of the shaded cell results from the product of the current and voltage, and is $-8.3 \text{ V} \times 3.3 \text{ A} = -27.4 \text{ W}$. The power losses of the total module due to a single shaded cell are 47%! If the backward current of the solar cell did not rise so strongly (as for a 'better' diode), the common characteristic would flatten even more strongly.

To reduce the high negative voltages and the power uptake, bypass diodes with reverse current passage direction are connected in parallel to the photoelectric cells, ideally one bypass diode per solar cell. If the currents are then added at equal voltages, the characteristic hardly changes for positive voltages, apart from the extremely low saturation current of the diode. As soon as the shaded cell assumes values of around -0.5 V to -0.7 V , however, the bypass diode switches and the current can rise exponentially.

If this characteristic is connected in series with the 35 unshaded solar cells, only small performance losses of the module are detected, since only very small negative voltages occur at the shaded cell. However, for technological production problems the concept of cell-integrated

bypass diodes diffused into the solar cell is not widespread. Connecting all the cell contact cables to mount external bypass diodes is too complex and expensive. As a compromise, the use of one bypass diode per 18 cells has become accepted, i.e. in a 36-cell module a total of 3 cell access wires must be led out into the external wiring box.

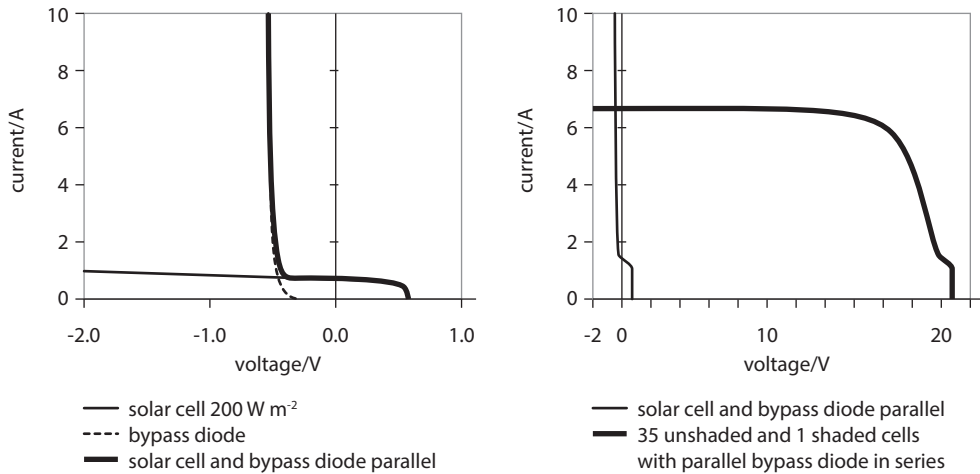


Figure 7.25 Parallel connection of a solar cell with a bypass diode (left) and series connection of 35 unshaded cells with the shaded cell with a parallel bypass diode.

In the case of one bypass diode per 18 cells, a module characteristic with 36 cells is constructed in what follows, with one cell shaded.

First the cell string consisting of 18 cells is considered with the shaded cell and the parallel bypass diode. The characteristic in the positive voltage range is dominated by the backwards characteristic of the shaded cell. For a total voltage of zero, there is a high negative voltage of -8.3 V at the shaded cell. The parallel bypass diode would only switch at a negative total voltage, and thus does not yet influence the cell string characteristic. If the shaded cell string is connected in series with the remaining unshaded cell string, the result is a common characteristic, which adopts the full current values of the unshaded cells below half the total voltage. The shaded cell string then goes into the backward voltage area and either the bypass diode switches, or the backward current is already so high that the photocurrent is allowed through.

With the steep backwards characteristic simulated here the bypass diode does not switch; at half the total voltage the shaded cell string has a voltage level of 0 V (+8.3 V of the unshaded seventeen cells and -8.3 V at the shaded cell), and the full current of the unshaded cells is allowed through. Only with flatter backwards characteristics does the bypass diode contribute to the limitation of the power uptake of the shaded cells.

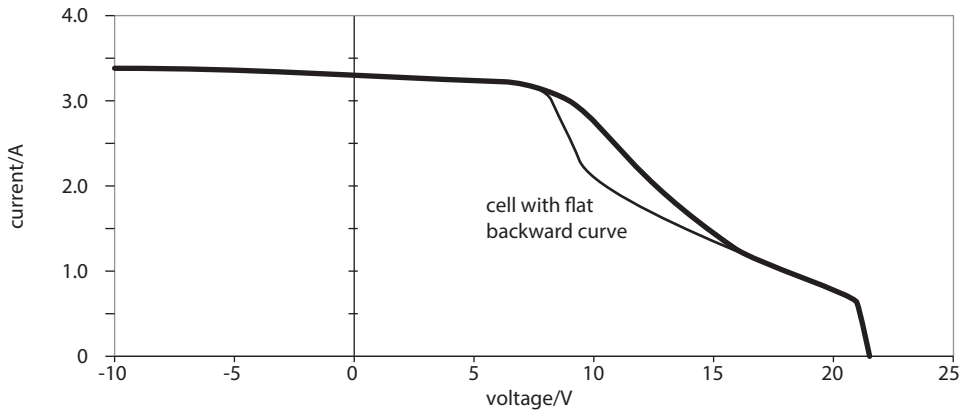


Figure 7.26 Module characteristic with shading of an individual cell with one bypass diode per 18 cells.

Only if a very flat backwards characteristic is assumed (here with a breakdown voltage of -25 V) does the bypass diode switch below the kink in the characteristic.

In thin-film modules, there is usually only one bypass diode for the whole module, i.e. there is no protection for partial shading of individual cells. The breakthrough characteristic of thin-film modules usually shows a gradual current increase at negative voltages, which means that there is no sharp breakthrough voltage. As shaded cells thus let through currents at not too high negative voltage levels, the modules are not damaged by shading.

7.9 Simple temperature model for PV modules

With high irradiance, the PV module temperature is usually above the STC temperature of 25°C , even at low ambient temperatures. The electrical power P_{MPP} of a PV module decreases linearly with the module temperature, the proportional decrease being calculated from the total of the voltage and current coefficients.

$$\frac{\alpha_p}{P_{MPP}} = \frac{\left. \frac{d(VI)}{dT} \right|_{MPP}}{P_{MPP}} = \frac{\frac{dV}{dT} I_{MPP} + \frac{dI}{dT} V_{MPP}}{V_{MPP} I_{MPP}} = \frac{\alpha_V}{V_{MPP}} + \frac{\alpha_I}{I_{MPP}} \quad (7.42)$$

With average power loss coefficients of $0.3 - 0.4\%$ per kelvin, at usual module temperatures of 50°C a performance decrease of as much as $7.5 - 10\%$ can be expected, which makes a simple model necessary to determine the module temperature. The model applies at first only to the simplest case of free-standing modules, for which simple assumptions can be made for the convective heat transfer and the long-wave radiation exchange with the sky and the ground. The thermal behaviour of special building integration solutions (double glazing, roof integration, etc.) and associated performance reduction is considered in a separate chapter.

In the steady-state case, a simple energy balance results in which the absorbed solar radiation $\dot{Q}_G$ minus the electrical power P_{el} and the thermal losses by radiation $\dot{Q}_r$ and convection $\dot{Q}_c$ must equal zero.

$$\dot{Q}_G - P_{el} - \dot{Q}_r - \dot{Q}_c = 0 \quad (7.43)$$

The absorbed solar radiation is a product of PV module surface A , irradiance G and the effective absorption coefficient α , which includes both the irradiance transmission through the glass cover and the absorption of the solar cell material. For this effective absorption coefficient, values between 0.7 and 0.9 can be used.

$$\dot{Q}_G = \alpha GA \quad (7.44)$$

For long-wave radiation exchange between the module front and the sky, and the module rear and the ground, a simplified temperature difference is assumed in each case between the module and the ambient temperature T_o , and a simplified form factor between the module surface and an infinitely large surrounding surface, so

$$\dot{Q}_r = 2h_r A (T_{PV} - T_o) \quad (7.45)$$

with

$$h_r = \sigma \varepsilon (T_{PV}^2 + T_o^2) (T_{PV} + T_o) \quad (7.46)$$

an emission coefficient of glass of $\varepsilon = 0.88$, and σ as the Stefan-Boltzmann constant ($\sigma = 5.6697 \times 10^{-8} \text{ W m}^{-2} \text{ K}^{-4}$). The temperatures are to be given in kelvin.

The convective heat flow is dominated at the module front by forced convection by wind forces ($h_{c,w}$ as a function of the wind velocity v_w); at the module rear, depending on the installation situation, more by free laminar or turbulent convection ($h_{c,free}$ as a function of the temperature difference between the module and the environment). For the front and rear, a simplified common heat transmission coefficient h_c is formed for free and forced convection:

$$h_c = \sqrt[3]{h_{c,w}^3 + h_{c,free}^3} \quad (7.47)$$

with the simple approximations

$$\begin{aligned} h_{c,w} &= 4.214 + 3.575 v_w \\ h_{c,free} &= 1.78 (T_{PV} - T_o)^{1/3} \end{aligned} \quad (7.48)$$

The heat flux by convection is then given by:

$$\dot{Q}_c = 2h_c A (T_{PV} - T_o) \quad (7.49)$$

The complete energy balance is thus described by

$$\begin{aligned} \alpha GA - P_{el} - h_r A (T_{PV} - T_o) - h_c A (T_{PV} - T_o) &= 0 \\ \Rightarrow T_{PV} &= \frac{(\alpha - \eta_{el}(T_{PV}))G}{h_r(T_{PV}) + h_c(T_{PV})} + T_o \end{aligned} \quad (7.50)$$

with the electrical power $P_{el} = \eta_{el} GA$ used. Since both the heat transfer coefficients and the electrical performance depend on the module temperature, the equation must be solved iteratively.

Example 7.6

Calculation of the PV module temperature and electrical power losses at 800 W m^{-2} irradiance, an ambient temperature of 10°C and a wind velocity of 3 m s^{-1} . The electrical efficiency under standard test conditions is 12% and the temperature coefficient of the performance is $0.4\% \text{ K}^{-1}$. The optical effective absorption of the module is set at 80%.

- First iteration: Assumption of a module temperature of 50°C and calculation of h_r , h_c , η_{el} .

$$\begin{aligned} h_r &= 5.57 \text{ W m}^{-2} \text{ K}^{-1} \\ h_{c,w} &= 14.9 \text{ W m}^{-2} \text{ K}^{-1} \\ h_{c,free} &= 6.09 \text{ W m}^{-2} \text{ K}^{-1} \\ h_c &= 15.23 \text{ W m}^{-2} \text{ K}^{-1} \\ \eta_{el} &= 10.8\% \end{aligned}$$

From this we can calculate the new module temperature:

$$T_{PV} = 36^\circ\text{C}$$

- Second iteration: Heat transfers and electrical performance at 36°C :

$$\begin{aligned} h_r &= 5.18 \text{ W m}^{-2} \text{ K}^{-1} \\ h_{c,w} &= 14.9 \text{ W m}^{-2} \text{ K}^{-1} \\ h_{c,free} &= 5.27 \text{ W m}^{-2} \text{ K}^{-1} \\ h_c &= 15.11 \text{ W m}^{-2} \text{ K}^{-1} \\ \eta_{el} &= 11.47\% \end{aligned}$$

Module temperature after the second iteration:

$$T_{PV} = 37^\circ\text{C}$$

- Third iteration:

$$\begin{aligned} h_r &= 5.21 \text{ W m}^{-2} \text{ K}^{-1} \\ h_{c,w} &= 14.9 \text{ W m}^{-2} \text{ K}^{-1} \\ h_{c,free} &= 5.34 \text{ W m}^{-2} \text{ K}^{-1} \\ h_c &= 15.31 \text{ W m}^{-2} \text{ K}^{-1} \\ \eta_{el} &= 11.42\% \end{aligned}$$

Module temperature after the third iteration:

$$T_{PV} = 36.7^\circ\text{C}$$

The electrical efficiency is 11.44%; i.e. the performance reduction in relation to the STC condition is 4.7%.

7.10 Systems engineering

Photovoltaic systems engineering includes on the DC side connecting the PV modules to the PV generator the sizing of the DC main line and the associated safety engineering (lightning protection and fault current recognition). The connection of the generator via an inverter to the public low-voltage mains for small systems or directly to the medium voltage level of 20 kV for larger generators requires the design of voltage levels and an optimum matching of inverter DC power and nominal PV power.

DC connecting: Cable sizing

The DC main line leads the entire generator current from the parallel-connected strings to the inverter. The percentage DC power losses p_R by ohm resistance R , related to the rated power, should be below 1%. The resistance R is calculated from the product of the specific conductor resistance

of copper $\lambda = 0.0178 \Omega \text{ mm}^2 \text{ m}^{-1}$, the entire line length (back and forth the length l , i.e. a total of $2l$) and the cable diameter A_q [mm^2].

$$R = \frac{\lambda 2l}{A_q}$$

For a simple line length between the PV branch and inverter l [m], the cable diameter A_q is thus obtained from the admissible DC power loss ΔP_{DC} over the resistance R , related to the DC performance at rated voltage V_n [V] and rated current I_n [A]:

$$\begin{aligned} p_R &= \frac{\Delta P_{DC}}{P_{DC}} = \frac{I_n^2 R}{P_{DC}} = \frac{P_{DC}^2}{V_n^2} \frac{R}{P_{DC}} = \frac{P_{DC}}{V_n^2} \frac{\lambda 2l}{A_q} \\ A_q &= \frac{2l \lambda P_{DC}}{p_R V_n^2} \end{aligned} \quad (7.51)$$

Example 7.7

Calculation of the cable diameter of a 2 kW system with a rated voltage level of 60 V_{DC} or 240 V_{DC} for a main-line length between roof and cellar of $l = 10 \text{ m}$. The admissible power loss p_R is to be 1%.

$$\begin{aligned} 60\text{V}: A_q &= \frac{2l \lambda P_{DC}}{p_R V_n^2} = \frac{2 \times 10 \times 0.0178 \times 2000}{0.01 \times 60^2} = 19.8 \text{ mm}^2 \\ 240\text{V}: A_q &= \frac{2 \times 10 \times 0.0178 \times 2000}{0.01 \times 240^2} = 1.2 \text{ mm}^2 \end{aligned} \quad (7.52)$$

Even at the relatively low power of 2 kW, the cable diameters of DC lines for the small system voltage are very large!

System voltage, ground connection and electrical safety

If the PV open circuit voltage remains under 120 V_{DC} and an inverter with an isolating transformer is used, the PV system conforms to safety category III based on VDE 0100/IEC 364, and additional measures for the protection of individuals are unnecessary. At higher DC voltages the modules must conform to safety category II, i.e. they must be tested for insulation defects at a voltage of 2000 V plus four times the open circuit voltage. In addition, the module cabling must be designed so as to best prevent short circuits and ground faults. This can be most easily achieved by separate positive and negative lines with double insulation.

The solar generator potentials are either not grounded, i.e. floating, and symmetrically biased against ground, or the positive or negative pole is grounded. Some high efficiency inverters overlay half of the AC voltage level on the DC side. This might lead to rather high capacitive current flows, especially if the modules have stainless steel frames.

The isolation resistance between a cell and the module frame is typically about 10 megaohm, so that in the case of free floating generators small currents flow in the nanoampere range. Sometimes these leakage currents can damage modules, especially thin-film technologies, so that it is recommended to ground the negative pole of the generator.

In some topologies of transformer-less inverters, a ground connection of the negative pole can be directly provided. In inverters with transformers, special grounding kits connect the negative DC pole directly to ground or via a fuse.

String diodes and short-circuit protection

In short-circuit situations or low voltages within a string due to shading, a high current can flow into this string when there are other strings connected in parallel. This is a current for which the module cables are not designed. This fault current flows in the opposite direction to the photocurrent and can thus be blocked by a string diode connected in series to the modules of a string. The voltage drop over the string diode of about 1 V leads, however, to a constant power loss, and checking the diode function is maintenance-intensive. Especially for low power systems string diodes and fuses are often not installed.

With partial shadings within a string the current can be drastically reduced, but the voltage remains almost unchanged even at low irradiance. In the MPP point, therefore, the shaded string also makes a rather small contribution to the total current value, and no fault currents flow. The common characteristic for four parallel-switched strings, each with four modules in series, is represented with 1000 W m^{-2} irradiance on three unshaded strings and one shaded string with an irradiance of 200 W m^{-2} . The shaded string is first operated within the area of positive power; only near the open circuit voltage of the common characteristic do small negative currents occur.

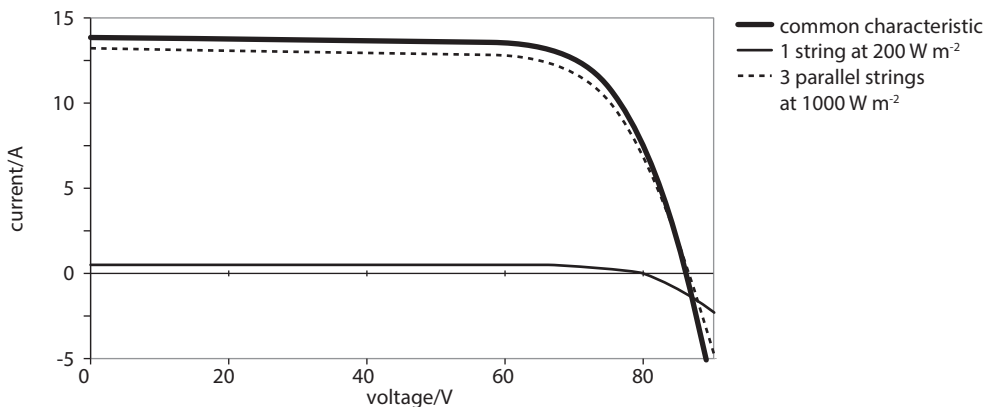


Figure 7.27 Operating points of a PV system with four parallel strings each with four modules in series and one shaded string.

Short circuits within a string are more serious than shading. PV generators can be operated potential-free, i.e. neither the positive nor the negative voltage level is connected with mass, and the inverter sometimes has a galvanic separation from the mains. A simple short circuit therefore causes no fault current at first, since only the potential level at the short-circuit point is brought down to ground level. Only a second short circuit within a string can short-circuit one or more modules, so the MPP and open circuit voltage of the string now drops significantly.

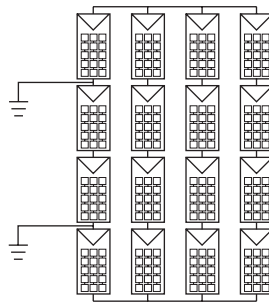


Figure 7.28 Double short circuit in a module string.

If the generator field is now operated at higher voltages than the open circuit voltage of the partly short-circuited string, the non-short-circuited modules are operated in far forward voltage and take up power. When this happens, the currents can become very high. For example, in a generator with four strings with four modules each in series in a string, two modules are to be short-circuited. In the MPP point, the string with the short circuits is also operated within the area of positive power; however, the short circuit string takes up power near the open circuit voltage (here about -15 A at 60 V at 1000 W m^{-2} irradiance).

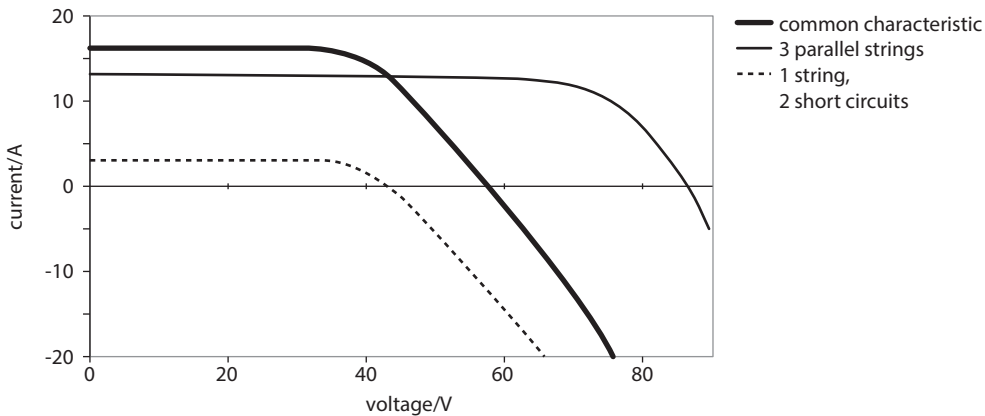


Figure 7.29 Characteristics of a generator with two short circuits in a string.

The risk of a double short circuit is, however, negligible with short-circuit and ground fault prevention (single line cables conforming to safety category II). Insulation monitoring can detect a simple short circuit promptly, so timely elimination of faults is possible without consequences. Protection from high fault currents can also be ensured by fuses, which are preferable to the string diodes due to smaller power losses.

Grounding of the metallic construction of the PV roof fixation tends to worsen indirect lightning protection. The usual overvoltage protection with varistors can be attached to the DC side of the inverter.

Inverters: Operational principle

In the inverter the produced DC is commuted periodically by controllable power-electronic semiconductor switches to an alternate current with the frequency of the electrical grid. Today there are about 50 different topologies available on the market, which differ by the use of transformers, the design of the electronic circuit and the grounding strategy.

In the simplest case, the positive photocurrent per period of the 50 Hz oscillation is switched during the half oscillation period to the one pole of a transformer and during the second oscillation half to the other pole, so a rectangular alternating voltage develops. However, a rectangular voltage wave form has such a high harmonic content that disturbances to electrical devices can emerge, and the maximum admissible harmonic currents in the public low-mains system (according to the European standard EN 61000-3-2) may be exceeded. The smallest harmonic content of the current to be fed in is obtained by a high frequency commutation of the DC voltage, in which the pulse width of the switching operation is varied in such a way that after smoothing of the current a largely sinusoidal wave develops, a widespread inverter concept known today as pulse-width modulation.

Apart from the maximum values of the harmonic currents, which are limited, the quality of the alternating voltage produced is described by the distortion factor k , which is defined as the relation of the effective values of the voltage harmonics to the effective values of the basic and harmonic oscillations.

$$k = \sqrt{\frac{\sum_{n=2}^{\infty} V_n^2}{\sum_{n=1}^{\infty} V_n^2}} \quad (7.53)$$

Usual distortion factors are between 1.5% and 5%. A further quality factor is the power factor L , which is defined as the relation of the active power to the apparent power and is thus a measure for the reactive power of an inverter.

$$L = \frac{P_{AC, active}}{P_{AC, apparent}} \quad (7.54)$$

The power factor should be as close as possible to 1. Apart from the adjustment efficiency, which will be discussed later, and the transformation efficiency of an inverter, the quality factors of harmonic content, power factor and electromagnetic compatibility (EMC) should be considered when selecting a product.

Electrical safety and mains monitoring

During disconnection from the mains, decentrally feeding photovoltaic systems must be separated immediately from the mains, to be sure of avoiding danger to people or further mains faults. A power failure is detected by under- or over-voltage relays or frequency monitoring integrated in the inverter. A so-called island formation, in which (despite mains failure) photovoltaic systems produce power, while users simultaneously consume this power, can nevertheless occur with this

passive network monitoring. Active mains-monitoring methods are a more reliable solution, with small disturbances such as pulsed voltages or frequency deviations imposed on the mains. In the case of frequency deviation, the inverter constantly tries to modify the feeding frequency and is synchronised with the mains frequency again with each zero crossover of the mains voltage. In the case of power failure, on the other hand, the frequency rises or falls constantly until it is outside a given band and the inverter switches off. The current modifications caused by modifications of the voltage level or the phase are amplified sufficiently by a feedback loop for the inverter to detect the power failure.

Inverter efficiencies: MPP tracking and adjustment efficiency

Similar to AC production, an inverter has the function of operating the DC PV generator at the point of maximum power; i.e. the effective input impedance must be constantly adapted to the irradiance and temperature-dependent point of maximum power. The MPP regulation is often based on periodic driving through of a voltage ramp until the power measured has achieved the optimum.

While power variations for thin-film modules are typically $\pm 5\%$, the MPP voltages fluctuate more by about $\pm 10\%$. This has consequences on the system design. As the fill factors are generally lower than for crystalline technologies, the differences between MPP voltage and open circuit voltage are usually larger. While for crystalline technologies the ratio of open circuit voltage to MPP voltage is between 1.18 and 1.27, it reaches 1.27 up to 1.38 for thin-film technologies, with the highest values for amorphous silicon technologies. The search algorithm of an inverter MPP tracker for the best voltage level thus requires a different and mostly larger voltage band for thin-film systems.

The adjustment efficiency η_{adj} is defined as the relation of actual power at a given MPP regulation and the maximum possible DC power with ideal MPP operation and is close to 100% in modern inverters.

$$\eta_{adj} = \frac{P_{DC,real}}{P_{DC,ideal}} \quad (7.55)$$

Transformation efficiency

The most important characteristic of an inverter is its transformation efficiency, which is defined by the relation of AC power output to DC input power. The power output P_{AC} results from the difference between the DC input power P_{DC} and the energy dissipation P_I .

$$\eta = \frac{P_{AC}}{P_{DC}} = \frac{P_{DC} - P_I}{P_{DC}} \quad (7.56)$$

The energy dissipation is made up of the input power-independent losses (internal current supply and magnetisation losses p_{own}), the losses in the semiconductor switches I_{switch} which are linear-dependent on the power output, and the Ohm cable losses r_{ohm} which rise as the square of the AC performance. It can be represented with good accuracy by a second-order polynomial.

To obtain a unit less representation of the coefficients, all absolute performances are related to the DC-rated nominal power P_n of the inverter.

$$p_l = \frac{P_l}{P_n} = \frac{p_{own}}{P_n} + v_{switch} \frac{P_{AC}}{P_n} + r_{ohm} \left(\frac{P_{AC}}{P_n} \right)^2 = p_{own} + v_{switch} p_{AC} + r_{ohm} p_{AC}^2 \quad (7.57)$$

The energy dissipation is represented as a function of AC power output p_{AC} , since for a positive DC input power first the own losses of the inverter are covered (i.e. $p_l = p_{own}$) and no switch or Ohm losses yet occur. The efficiency and thus the power output must be zero.

$$\eta = \frac{(P_{DC} - P_l) / P_n}{P_{DC} / P_n} = \frac{p_{AC}}{p_{DC}} = \frac{p_{DC} - (p_{own} + v_{switch} p_{AC} + r_{ohm} p_{AC}^2)}{p_{DC}} \quad (7.58)$$

With this description of the energy dissipation, however, both entry and exit powers occur in the efficiency term. The aim must be to calculate the efficiency of the inverter as a function of the input power supplied by the PV generator.

p_{AC} is therefore replaced by ηp_{DC} and the equation is solved for η .

$$\begin{aligned} \eta &= 1 - \frac{p_{own}}{p_{DC}} - v_{switch} \eta - r_{ohm} \eta^2 p_{DC} \\ \Rightarrow \eta &= \frac{1 + v_{switch}}{2r_{ohm} p_{DC}} \pm \sqrt{\frac{(1 + v_{switch})^2}{(2r_{ohm} p_{DC})^2} + \frac{p_{DC} - p_{own}}{r_{ohm} p_{DC}^2}} \end{aligned} \quad (7.59)$$

Only the positive term of the quadratic equation supplies physically meaningful values. The three loss coefficients p_{own} , v_{switch} and r_{ohm} can be calculated from three efficiency values η_1 , η_2 and η_3 given by most manufacturers in technical data sheets for power ratios of the rated power, for example, $p_1 = 10\%$, $p_2 = 50\%$ and $p_3 = 100\%$. The algebraic manipulation of the three equations for the three unknown quantities results in (Schmidt and Sauer, 1996):

$$p_{own} = \frac{p_1 p_2 p_3 (\eta_1^2 p_1 (\eta_2 - \eta_3) + \eta_1 (\eta_3^2 p_3 - \eta_2^2 p_2) + \eta_2 \eta_3 (\eta_2 p_2 - \eta_3 p_3))}{\eta_1^2 p_1^2 - \eta_1 p_1 (\eta_2 p_2 + \eta_3 p_3) + \eta_2 \eta_3 p_2 p_3 (\eta_2 p_2 - \eta_3 p_3)} \quad (7.60)$$

$$v_{switch} = \frac{\eta_1^2 p_1^2 (\eta_2 p_2 - \eta_3 p_3 - p_2 - p_3) + \eta_1 p_1 (\eta_3^2 p_3^2 - \eta_2^2 p_2^2) + \dots}{(\eta_1 p_1 - \eta_2 p_2)(\eta_1 p_1 - \eta_3 p_3)(\eta_3 p_3 - \eta_2 p_2)} \quad (7.61)$$

$$r_{ohm} = \frac{\eta_1 p_1 (p_2 - p_3) + \eta_2 p_2 (p_3 - p_1) + \eta_3 p_3 (p_1 - p_2)}{\eta_1^2 p_1^2 - \eta_1 p_1 (\eta_2 p_2 + \eta_3 p_3) + \eta_2 \eta_3 p_2 p_3 (\eta_3 p_3 - \eta_2 p_2)} \quad (7.62)$$

Example 7.8

Calculation of the inverter characteristic values for two commercial inverters with the following manufacturer data:

Inverter power	1.8 kW	1.4 kW
$\eta_1: P_{DC}/P_n = 0.1$	79.4	83.0
$\eta_2: P_{DC}/P_n = 0.5$	89.9	91.9
$\eta_3: P_{DC}/P_n = 1.0$	88.9	89.8

The characteristic values calculated using the above equations result in:

Inverter power	1.8 kW	1.4 kW
P_{own}	0.016575	0.015505
V_{switch}	0.045513	0.010553
r_{ohm}	0.067941	0.095879

With these three characteristic values the efficiency can be calculated as a function of the power ratio p_{DC} using Equation (7.58).

At low DC input power, the efficiency is dominated by the consumption of the power electronics; at high input power Ohm losses account for the decrease in efficiency. The maximum inverter efficiency is thus often around half the rated power.

At DC input power above the rated power of the inverter, the efficiency depends on the overload behaviour of the inverter. Many inverters can be used for some time when overloaded, and regulate only on overheating of the electronics away from the MPP point. In a simple approximation, in the case of overload, i.e. $p_{DC} > 1$, the AC power output is held constant at the level of the rated power output given at $p_{DC} = 1$:

$$p_{AC}|_{p_{DC}>1} = (p_{DC}\eta)|_{p_{DC}=1} = \eta|_{p_{DC}=1} = const \quad (7.63)$$

The transformation efficiency thus drops sharply with increasing overload.

$$\eta|_{p_{DC}>1} = \frac{p_{AC}|_{p_{DC}=1}}{p_{DC}} = \frac{\eta|_{p_{DC}=1}}{p_{DC}} \quad (7.64)$$

The following efficiency characteristics of the two inverters were calculated up to the DC-rated power ($p_{DC} = 1$) with the parameters from Example 7.8 and above the rated power with Equation (7.64).

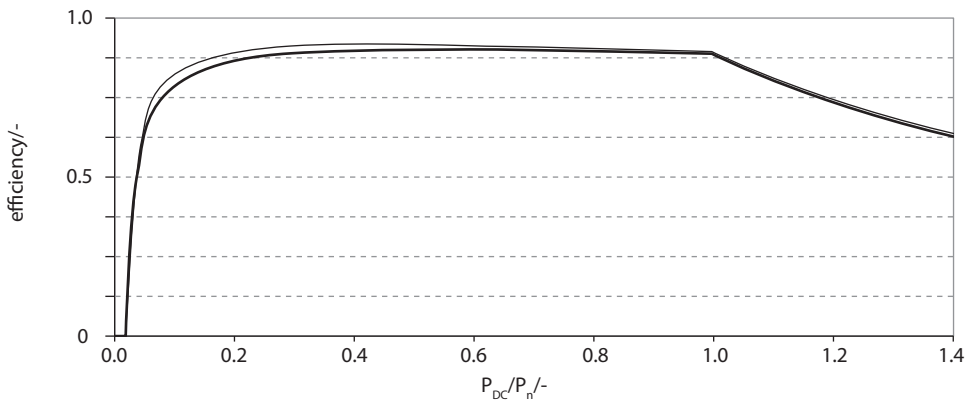


Figure 7.30 Transformation efficiency of two inverters as a function of the DC input power normalised on rated power.

To enable a standardised comparison between inverters, the so-called European efficiency has been introduced, which weights the partial load efficiencies with average radiation conditions in Central Europe.

$$\eta = 0.03\eta_{p_{DC}=5\%} + 0.06\eta_{p_{DC}=10\%} + 0.13\eta_{p_{DC}=20\%} + 0.1\eta_{p_{DC}=30\%} + 0.48\eta_{p_{DC}=50\%} + 0.2\eta_{p_{DC}=100\%} \quad (7.65)$$

The European efficiency cannot, however, take into account either the influence of module orientation under different radiation conditions or the influence of different PV generator to inverter power ratios. To determine the real annual efficiency of the inverter, these influences must be analysed.

Power sizing of inverters

To determine the annual efficiency of an inverter, the frequency distribution of DC power generated has to be known. Then the different partial load efficiencies of the inverter, multiplied by the energy-weighted frequency and normalised with the DC annual energy, produce the annual efficiency.

Given that under northern European climatic conditions irradiances over 900 W m^{-2} seldom occur and the module temperature at high irradiance values is usually over 25°C and thus causes power losses, it is not necessary even with south-facing roofs to design the inverter for the nominal power of the PV generator under standard test conditions. On south-facing facades, which often occur within the building integration area, irradiance values over 700 W m^{-2} play no role in energy terms, so an inverter designed for a PV generator-rated power is clearly oversized. Since undersizing the inverter reduces the system costs, in the next section a calculation will be made of how the rated power can be reduced without large efficiency losses for a given module orientation.

For the performance sizing of an inverter for a mains-connected PV system, the following procedure is suggested:

1. First the energy-weighted frequency of the irradiance is determined for the respective module orientation: hourly irradiance values are divided into irradiance classes with mean irradiance G_i and class width ΔG_i , the absolute frequency of the irradiance per class in hours per year ($n_{h,i}$) is determined, and finally the frequency is energy-weighted with the mean irradiance value

of the respective class. Thus the energy irradiated annually on the surface in kWh m^{-2} in each irradiance interval ΔG_i is obtained. It is sufficient to divide the irradiance of 0 - 1000 W m^{-2} into 10 classes with 100 W m^{-2} class width.

2. The irradiated energy per irradiance interval $G_i n_{h,i}$ is then converted into electrical energy $P_{DC} n_{h,i}$ using the efficiency of the PV generator η_{PV} and the generator surface A_{PV} , with the frequency distribution unchanged.

$$P_{DC,i} n_{h,i} = \eta_{PV} G_i n_{h,i} A_{PV} \quad [\text{kWh}] \quad (7.66)$$

The influence of the module temperature is neglected when calculating the annual inverter efficiency.

3. For an inverter with a given rated DC power P_n , the standardised DC input power $p_{DC,i} = P_{DC,i}/P_n$ is then determined for each irradiance class, and the inverter efficiency $\eta_{inv}(p_{DC,i})$ is calculated. This results in the AC energy:

$$P_{AC,i} n_{h,i} = P_{DC,i} n_{h,i} \eta_{inv}(p_{DC,i}) \quad (7.67)$$

4. The mean annual efficiency of the inverter is then obtained from the totalled AC energy divided by the total of the DC energy.

$$\eta_{inv,year} = \frac{\sum_{i=1}^n \eta_{inv,i} P_{DC,i} n_{h,i}}{\sum_{i=1}^n P_{DC,i} n_{h,i}} = \frac{\sum_{i=1}^n P_{AC,i} n_{h,i}}{\sum_{i=1}^n P_{DC,i} n_{h,i}} \quad (7.68)$$

Example 7.9

As an example, for a south-facing facade and a south-facing roof inclined at 45° , the mean annual efficiency of the 1.8 kW inverter shown above for a PV generator in Stuttgart with 1.8 kW of rated DC power (12% module efficiency and 15 m^2 surface) is to be calculated, and compared with the Euro efficiency.

1. First the energy-weighted frequency distribution is calculated from hourly time series of the irradiance. Although the low irradiance values occur most frequently, the most energy-relevant irradiance intervals are situated in the middle irradiance area between 400 and 600 W m^{-2} for a south-facing roof or 300 and 400 W m^{-2} for a south-facing facade.

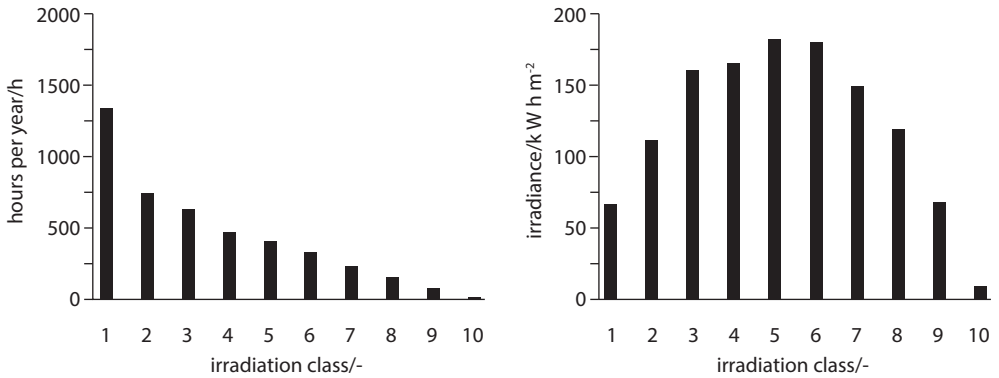


Figure 7.31 Frequency of the irradiance $n_{h,i}$ and energy-weighted irradiance for a south-facing roof in Stuttgart inclined at 45°.

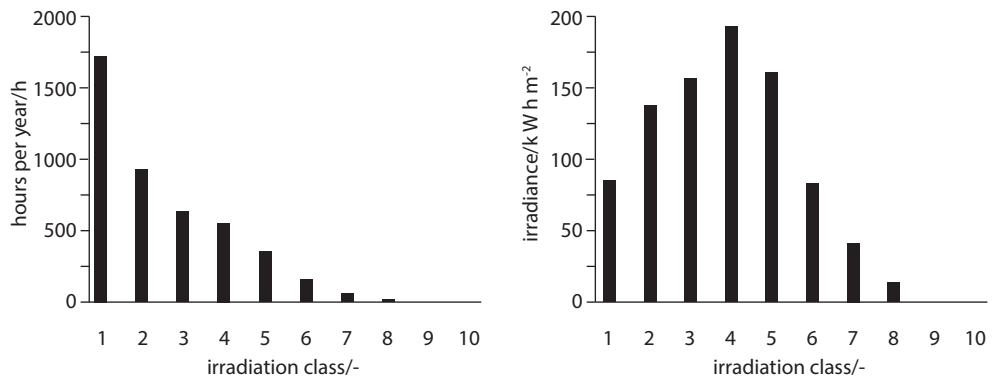


Figure 7.32 Frequency of the irradiance $n_{h,i}$ and energy-weighted irradiance for a south-facing facade.

- In the second step the PV energy is calculated per irradiance class using the constant efficiency of 12% and the surface of 15 m². Here class 4 is the most energy-relevant class at the south roof with irradiance levels between 400 and 500 W m⁻², i.e. a mean irradiance of 450 W m⁻². The result is with 404 h absolute frequency a DC energy of

$$P_{DC} n_{h,4} = G_4 n_{h,4} \eta_{PV} A_{PV} = 0.45 \frac{\text{kW}}{\text{m}^2} \times 404 \text{ h} \times 0.12 \times 15 \text{ m}^2 = 327 \text{ kWh} \quad (7.69)$$

- From the DC energies, the mean PV performance for each class is calculated, related to the inverter rated power and converted by the inverter efficiency into AC energy. For the class 4 above of the south-facing roof, the mean PV performance is given by 327 kWh/404 h = 0.63 kW; i.e. related to the inverter with 1800 W nominal power the result is a relative p_{DC} power of 0.35. The associated efficiency at this partial load is 0.88, so the AC energy of this class is given by

$$P_{AC} n_{h,4} = 327 \text{ kWh} \times 0.88 = 266 \text{ kWh} \quad (7.70)$$

4. The mean yearly efficiency results from the total of the AC energy divided by the totalled DC energy. It is 87.5% for the south-facing roof and 85.9% for the south-facing facade. The Euro-efficiency is 86%, independent of orientation.

If the inverter-rated power is now varied in relation to the PV generator power, the mean yearly efficiency can be calculated using this procedure, with only steps 3 and 4 changing. For the south-facing facade, the south-facing roof and a horizontal PV generator orientation, the inverter-rated power varied from 30% of the PV generator power, which is extremely undersized, to 200% of the PV generator performance, i.e. greatly oversized. The efficiency characteristic of the inverter takes into account the case of overload by limitation of the power output to the nominal inverter power (by means of regulation away from the MPP point). The results show that the optimal sizing of the inverter for module angles of inclination between the horizontal surface and 45° for a south-facing roof is 80% of the PV-rated power (yearly efficiency of 88%, location Stuttgart).

For a south-facing facade the optimal sizing is 60% of the PV generator power (88% yearly efficiency). Also evident are the relatively flat maxima particularly in the overload area; i.e. oversizing is of no consequence, while undersizing below the optimum leads to relatively large performance losses.

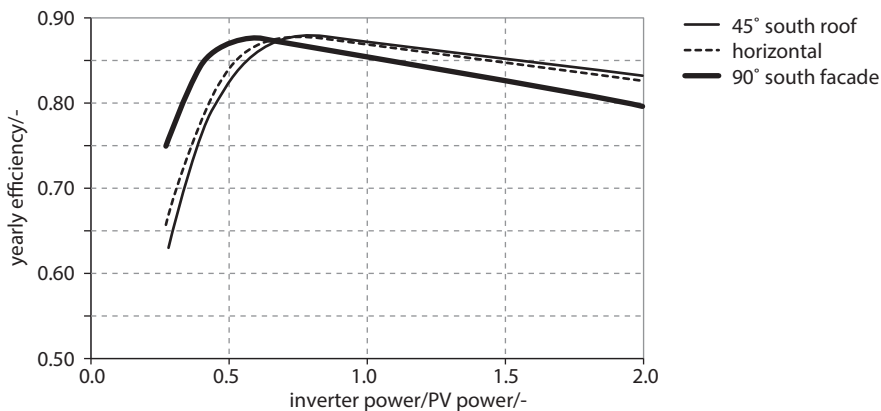


Figure 7.33 Yearly efficiencies of an inverter as a function of the power ratios of the inverter and PV generator in a German climate.

For the inverter nominal power, therefore, a relation to the PV generator power of 60 - 130% with less than 2% efficiency losses is possible for south-facing roofs with random inclinations. For south-facing facades, inverter power ratios of 45 - 100% of the PV generator power are recommended for this type of inverter.

For modern inverters with high efficiency under part load conditions the trend is to not to underdimension the inverter so much. To account for PV generator power above STC conditions, which happens for irradiance conditions above 1000 W m^{-2} , the recommended inverter nominal power relative to the PV generator power on inclined surfaces is between 0.9 and 1.2.

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8 Compression chillers and heat pumps

Figure 8.1 (previous page): Youth centre L-Quadrat in Ostfildern with passive energy standard and ground source heat pump (Photo: Barta).



Figure 8.2 Heat pump with vertical ground heat exchangers in the youth centre Ostfildern (Photo: Barta).

As renewable electricity fractions have increased strongly in the last decade, heating or cooling using electrically driven heat pumps or compression chillers offer new possibilities for renewable energy supply.

Power generated by photovoltaic (PV) modules has become so cheap that electrical compression cooling using PV power has become an interesting option for solar cooling. Primary energy efficiencies are comparable to solar thermal cooling and depending on energy prices, feed-in tariffs and investment costs, PV cooling systems can be at the same level or even cheaper than solar thermal systems. Heat pumps can supply heating and domestic hot water most efficiently, if the supply temperature levels are low.

8.1 Overview of heat pump and chiller technologies

Heat pumps or chillers can be basically divided into two types; sorption heat pumps, in which the cold vapour is compressed by heating a solvent, which has absorbed the refrigerant vapour, and the compression heat pump, which is currently the predominant technology used in heat pumps and air conditioning.

In compression heat pumps, the suction of the gaseous refrigerant from the evaporator and subsequent compression is carried out by an electrically or combustion-driven mechanical compressor.

The term 'heat pump' describes only the machine itself. A decisive factor for efficiency and costs of a heat pump system is the temperature level of the environmental heat source, from which heat can be extracted, with efficiency rising with temperature level of the heat source.

Heat pumps or chillers can be characterised in accordance with DIN EN 14511, which specifies first the heat transfer medium for the outdoor heat exchanger and second, the heat transfer medium of the indoor heat exchanger.

Heat transfer medium		
Outdoor heat exchanger	Indoor heat exchanger	Classification
Air	Air	Air/Air Heat Pump or Air-cooled Air conditioner
Water	Air	Water/Air Heat Pump or Water-cooled Air conditioner
Brine	Air	Brine/Air Heat Pump or Brine-cooled Air conditioner
Air	Water	Air/Water Heat Pump or Air-cooled Chiller
Water	Water	Water/Water Heat Pump or Water-cooled Chiller
Brine	Water	Brine/Water -Heat Pump or Brine-cooled Chiller

Table 8.1 Classification according to DIN EN 14511 for the most common heat pumps or chiller systems.

Heat sources for heat pumps

Instead of ambient air used in air to air or air to water heat pumps it is advisable to use heat sources with higher temperature levels, as this improves the heat pump efficiency. Vice versa for cooling machine operation it is advisable to reject heat at a lower temperature level than hot ambient air in summer.

Horizontal ground heat exchangers at low depth (below 2 m) allow to extract heat of about $10 - 35 \text{ W m}^{-1}$ depending on the soil conductivity. Vertical heat exchangers with about 100 m depth allow a heat extraction between 20 and 70 W m^{-1} (German technical guideline VDI 4640). Also heat extraction from waste water or solar absorbers is beneficial to the performance of heat pumps.

Refrigerants and compressor technologies

Commercial chillers or heat pumps use refrigerants to convey heat from the low-temperature level used for cold production to the high-temperature level, where it condenses and releases heat.

Numerous types of refrigerants are available and they vary in terms of energy efficiency, stability and safety classifications, ozone depletion potential (ODP), and global warming potential (GWP). When selecting a chiller or heat pump, the temperature requirements and refrigerant's characteristics must be appropriately matched, and the operating temperatures and pressures involved should also be considered. There are several environmental factors that concern refrigerants and also affect the future availability for chiller applications, which is a key consideration in applications where a large chiller may last for 25 years or more. All refrigerants are characterised by two numbers: the ODP and GWP. The ODP values range from 0 to 1: the closest the ODP value is to 1, the more harmful the refrigerant is for the ozone layer. The GWP compares the amount of heat trapped in the atmosphere by a certain mass of the gas in question to the amount of heat trapped by a similar mass of carbon dioxide (whose GWP is standardised to 1). A substance's GWP depends on the timespan over which the potential is calculated, as a gas which decays fast may initially have a large effect but for longer time periods becomes less important. Methane has a GWP of 25 over 100 years, but 72 over 20 years; the refrigerant R134a has a 20 year GWP of 3400.

Refrigerants most commonly used in refrigeration systems can be classified into four groups:

- hydrocarbons
- halocarbons
- zeotropes and azeotropes
- inorganic refrigerants

Refrigerants (R) belonging to the hydrocarbon group are ethane, propane, butane and isobutane. They have been in use since the early 19th century and together with ammonia were the most widely used refrigerants before the introduction of chlorinated fluorocarbons in the 1930s. Other naturally occurring substances and inorganic refrigerants are CO_2 , ammonia, water and air. The group number 7 denotes inorganic references followed by the molar mass. For example, R-717 is ammonia (NH_3) with a molar mass of 17 g mol^{-1} . They do not have an ozone depletion potential and have no or negligible GWP.

Refrigerants belonging to the halocarbon group are derivatives of the hydrocarbons obtained by substituting chlorine or fluorine for the hydrogen atoms in methane and ethane. As chlorine and fluorine are both halogens, this group of refrigerants is called the halogenated hydrocarbons or halocarbons. Common refrigerants in this group of organic refrigerants are

R-11 (or CFC-11), R-12 (or CFC-12), R-13 (CFC-13) and R-22 (CHF_2Cl or HCFC-22). CFCs are generally characterised by a high ODP value close to 1.0, because they contain chlorine. HCFCs have ODP values between 0.02 (R-123) and 0.11 (R-141b). Complying with the Montreal Protocol, these (hydro) chlorofluorocarbons are being phased out in many countries due to damage to the ozone layer.

An azeotrope is a mixture of two substances which cannot be separated into its components by distillation. It evaporates and condenses as a single substance and its properties are completely different from its constituents. For example, azeotrope R-500 is a mixture composed of 73.8 per cent R-12 and 26.2 per cent R-152. A zeotrope is a mixture whose composition in the liquid phase differs to that in the vapour phase. Therefore these mixtures do not boil at constant temperature.

The most popular refrigerant used today for cooling of buildings R134a has zero ODP, but a massive GWP of 1430. The UK BREEAM sustainable building rating system includes points for refrigerants with a GWP of less than ten. According to a recent analysis by the engineering company Ove Arup, this BRE standard leaves just a handful of options such as ammonia (R717), carbon dioxide (R744) and hydrofluoroolefins (HFOs), known collectively as R1234.

Almost all the other discovered refrigerants that have a GWP lower than ten with a zero ODP are flammables derived from propane, pentane, butane, propylene, ethane or isopropane.

Ammonia (R717) is an excellent refrigerant with zero ODP and GWP. Because it is poisonous at high concentrations, it is mainly used in large-scale applications. Carbon dioxide (R744) has to be used at very high pressures with less efficiency. Being much safer, it is a good choice for small-scale heat pumps with zero ODP and a GWP of just one. HFO (R1234) is a new refrigerant with a very low flammability, a GWP of six and zero ODP.

The evaporated refrigerant vapour has to be compressed to be able to condense at higher temperature levels. There are four basic types of compressors used in vapour compression chillers: reciprocating compressors, scroll compressors, screw compressors, and centrifugal compressors are all mechanical machines that can be powered by electric motors, steam, or gas turbines.

Reciprocating compressors are positive displacement compressors, which use combinations of cylinder unloading and on/off compressor cycling of single or multiple compressors to compress the refrigerant vapour. Scroll compressors use two interleaving scrolls to move refrigerant into successively smaller chambers. Either one of the scrolls is fixed, while the other orbits eccentrically, or the compression motion is co-rotating the scrolls synchronously, but with offset centres of rotation thereby compressing pockets of vapour between the scrolls.

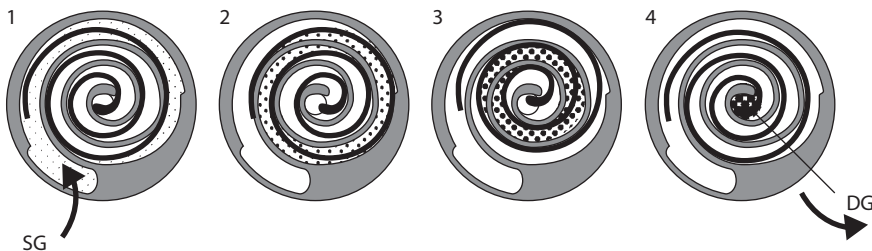


Figure 8.3 Scroll compressor with one fixed scroll and suction gas (SG) compressed to the high pressure discharge (DG) to the condensor.

Screw-driven compressors use a rotary type positive displacement mechanism. The gas compression process of a rotary screw is a continuous sweeping motion, so there is very little pulsation or surging of flow, as occurs with piston compressors.

8.2 Energy efficiency of heat pumps and chillers

There are various efficiency metrics for heat pumps to make them comparable to the efficiency levels of conventional boilers. The energetic efficiency of heat pump technology depends on a large number of factors, in particular those that affect the conditions of operation. For chillers the heat source is equivalent to the useful cold produced and the heating supply temperature corresponds to the condensation heat rejection temperature.

The manufacturer-provided characteristic parameter of a heat pump is the coefficient of performance (COP). The COP of a heat pump is defined by the quotient of the benefit (heat of condensation $\dot{Q}_{cond}$) and the expenditure (input power P_{elec}) for a given operating point:

$$COP = \frac{\dot{Q}_{cond}}{P_{elec}} \quad (8.1)$$

For chillers the efficiency is characterised by the energy efficiency ratio (EER), which is the ratio of produced cold (heat transferred to the evaporator $\dot{Q}_{eva}$) to the electrical expenditure.

$$EER = \frac{\dot{Q}_{eva}}{P_{elec}} \quad (8.2)$$

The COPs of heat pumps or EERs of chillers are determined by independent and accredited heat pump testing laboratories for certain operating points. In test bench measurements according to EN 14511 or EN 255-3, the electrical power consumption of the heating circulation pump and the source-side delivery pump (e.g. in brine/water heat pumps), are not always taken into account. The COP is a quality criterion of the heat pump; the higher the coefficient, the more efficient the heat pump is.

Air to water heat pumps analysed in Switzerland (test conditions: A2/W35) have displayed a continuous rise in COP. The average COP value of about 2.3 in 1993 improved by the end of 2004 to around 3.5. The measured values since the year 2000 are scattered between 3.02 to 4.42. Test conditions are characterised by the source and sink type (B for brine, W for water, A for air) and the temperature levels of source and sink.

The trend of the brine (B) to water (W) heat pumps (test condition B 0°C/W 35°C) shows a continuous increase in the development of COP figures until 2000, when the average value of an initial 3.9 improved to 4.4. Since 2000, the average COP values have not changed significantly.

The continuous rise in efficiency of air/water heat pumps is due to national quality labelling and to market competition. The described increases in the COP values are mainly due to the introduction of scroll compressors.

Seasonal performance

The Seasonal Performance Factor (SPF) or Seasonal Coefficient of Performance (SCOP) is defined as a seasonal average coefficient of performance. The calculation method takes into account part-load conditions, and other types of energy consumption, such as when the unit is in standby mode.

When determining the SCOP, the different methods of accounting for auxiliary power (pumps, control) render the comparisons with other systems difficult. There are also considerable differences between power measurements on test rigs and in actual operation.

The reference value of the SCOP is calculated for the reference annual heating demand, which is determined for conditions specified in DIN EN 14825 and is used for labelling, comparison and certification purposes. In practice, the seasonal performance factor (SPF), describes the annual efficiency of a heat pump. It is defined as the ratio of an entire year of delivered energy to the heating system plus the hot water heating energy, to the total electric power usage (including auxiliary energy).

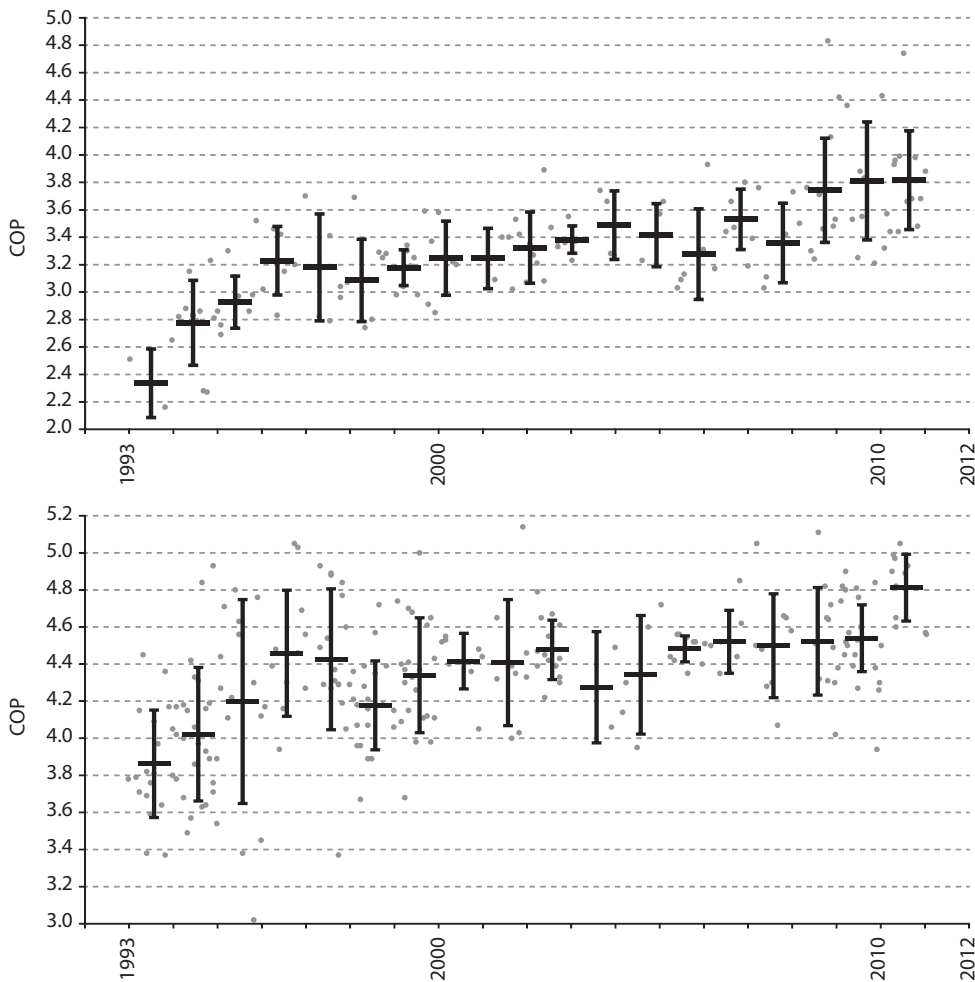


Figure 8.4 COP trend of air/water heat pumps (top) and brine/water heat pumps (bottom) from the heat pump test centre Buchs in Switzerland.

About 100 heat pumps (primarily air/water and brine/water heat pump systems) were measured and evaluated in existing buildings by the German Fraunhofer Institute of Solar Energy (Russ and Miara, 2010). The heat pumps are used to cover the heat and hot water requirements for different dwelling types using different hydraulic concepts. These differing requirements for heat demand and supply temperatures are reflected in the results, in terms of heat generated, hours of operation and ultimately in the seasonal performance factors. Due to the different system configurations and the different building types, the results show a large spread.

For the Brine/Water heat pumps with horizontal earth heat exchangers, an average COP of 3.2 is achieved. When vertical borehole heat exchangers are used as a heat source, an average COP of 3.3 is reached and there are large differences in the individual projects. The SPF is strongly dependent on the temperature difference between the heat source and heat sink, and also on the absolute level of the supply temperature of the heat sink. The larger the share related to domestic hot water (e.g. summer), the worse the performance factor is. This is due to the higher temperatures required in domestic hot water heating.

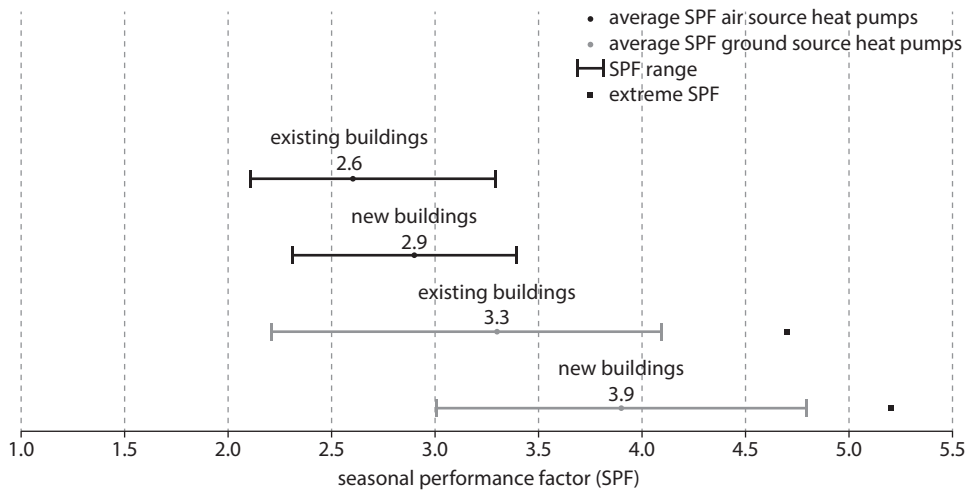


Figure 8.5 Seasonal performance factors determined by Fraunhofer ISE broken down by type of heat pump and application in existing or new buildings.

Primary energy efficiency

To compare heat pumps energetically with conventional heating systems, it is necessary to take into account the conversion efficiency of primary energy. The primary energy efficiency is defined as the ratio of useful energy to the primary energy:

$$\eta_{pr} = \frac{\text{useful energy}}{\text{primary energy}} \quad (8.3)$$

Useful energy is the energy that is available to the consumer for heating or cooling purposes after conversion of final energy carriers delivered to the building such as gas, electricity, pellets or others. Primary energy Q_p denotes the energy needed to produce these final energy carriers, for example, to produce electricity from gas, coal or others. Primary energy is calculated from the final

energy Q_f which is composed of both useful energy Q_{use} and conversion losses, using a primary energy factor f_p .

$$f_p = \frac{Q_p}{Q_f} \quad (8.4)$$

$$Q_f = Q_{use} + \text{conversion losses} \quad (8.5)$$

The factor includes the losses incurred in the provision of energy source (for example, production, transportation, refining, drying or storage).

The primary energy factor for electricity in Germany according to DIN 18599-1, Appendix A, including supply chains and distribution, is currently 2.7, which is expected to drop to a value of about 2.1 by 2030. If the electricity is produced mainly by renewable energies, such as in Switzerland or Norway, then heat pumps represent renewable heat generation that holds great potential for greenhouse gas savings.

Grade of quality

The grade of quality η_{HP} is calculated as the ratio of the COP of the heat pump or compression chiller in the ideal Carnot process COP_{Carnot} and indicates the deviation of the actual process from the ideal process.

$$\eta_{HP} = \frac{COP}{COP_{Carnot}} \quad (8.6)$$

The Carnot COP is the ratio of the theoretically achievable specific heating or cooling capacity q_0 to the specific technical work W_t and is dependent on the condensation temperature T_{cond} and the evaporation temperature in kelvin. For heat pump operation the relation is:

$$COP_{Carnot} = \frac{q_0}{W_t} = \frac{T_{cond}}{T_{cond} - T_{eva}} \quad (8.7)$$

The higher the grade of quality, the less irreversibility occurs in the real process.

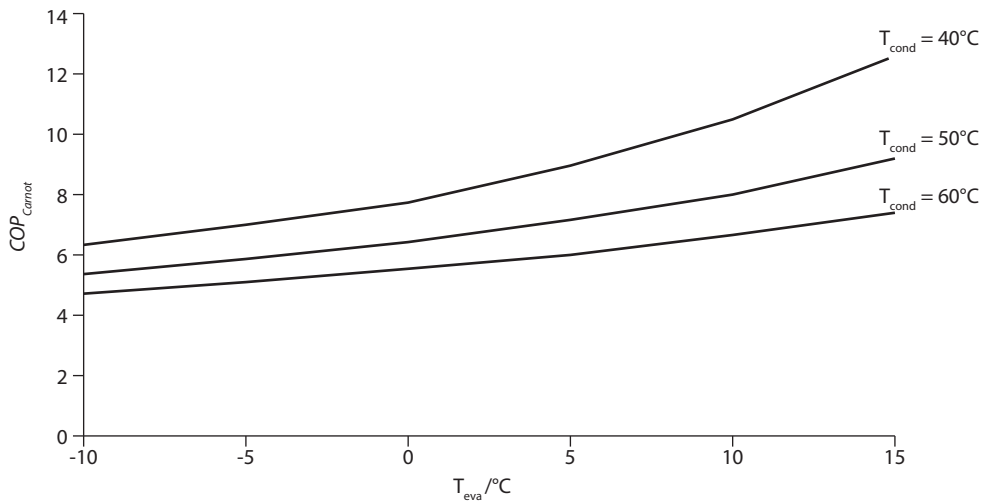


Figure 8.6 Heat pump Carnot coefficients of performance as a function of condenser and evaporator temperature.

8.3 Heat pump and compression chiller modelling

The following section describes the mathematical-physical model with the manufacturer-specific details regarding individual components.

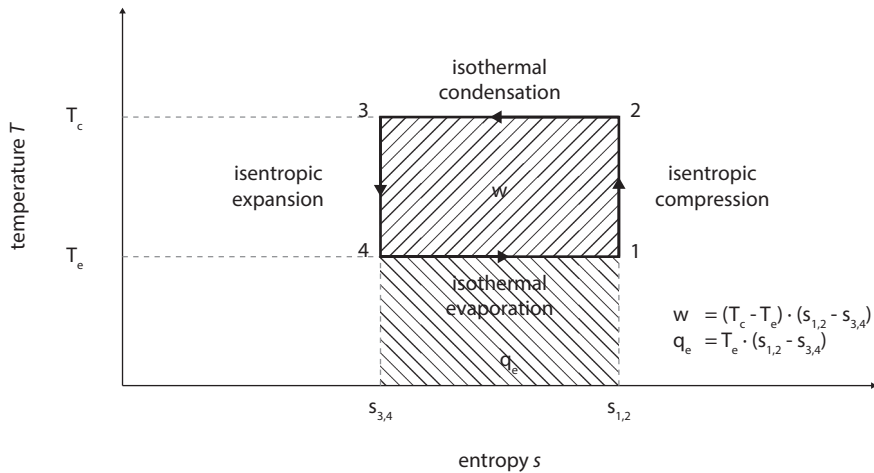


Figure 8.7 The ideal Carnot process in the temperature-entropy (T - s) diagram.

General thermodynamic refrigeration process

The anticlockwise running thermodynamic cycle is the reverse cycle of a thermal engine and provides the basis for the heat pump model. The ideal Carnot process is based on the following state changes:

- Isothermal evaporation ($4 \rightarrow 1$)
- Isentropic compression ($1 \rightarrow 2$)
- Isothermal condensation ($2 \rightarrow 3$)
- Isentropic expansion ($3 \rightarrow 4$)

Expressing the evaporation heat and the electrical work in terms of entropy changes, the COP of the cooling process is obtained.

$$\begin{aligned}
 COP &= \frac{q_c}{W} \\
 &= \frac{T_e(S_{1,2} - S_{3,4})}{T_c - T_e(S_{1,2} - S_{3,4})} \\
 &= \frac{T_e}{T_c - T_e}
 \end{aligned} \tag{8.8}$$

Example 8.1

At 0°C evaporation and 35°C condensation, a real refrigerating system has a refrigerating capacity of 100 kW and power demand of 21 kW. What is the COP and the Carnot efficiency of the refrigerating system?

$$\begin{aligned}
 COP_{real} &= \frac{100}{21} = 4.8 \\
 COP_{Carnot} &= \frac{273}{35} = 7.8 \\
 \eta_{Carnot} &= \frac{COP_{real}}{COP_{Carnot}} = \frac{4.8}{7.8} = 0.62
 \end{aligned}$$

Theoretical dry process

The Carnot process is not feasible for a compression chiller with a mechanical compressor, as at the end of evaporation the vapour is still wet, which might damage the compressor. In the following four steps, the theoretical dry process for compression cycles (sometimes called 'Plank' process) is defined.

1 Evaporation process

An isentropic compression of the refrigerant as wet steam (point 1) causes damage in the compressor. Therefore, evaporation to at least the right limit curve between wet and dry

vapour is required. The start of the compression will lie on the saturated vapour curve (point 1) for the theoretical dry process.

2 Isentropic compression

The isentropic compression to the pressure p_c takes place in the dry saturated area (point 2'), so that through the course of the isobars in this area, the final compression temperature T_2 is higher than the condensation temperature T_2 or T_{Cond} .

3 Heat removal to condensation temperature and condensation

Prior to the transfer of the condensing heat, there is a heat removal from the dry saturated steam with required temperature reduction from the compression end temperature to the condensing temperature. During condensation further heat rejection takes place.

4 Isenthalpic expansion

Instead of an isentropic expansion, an isenthalpic expansion by a simple throttle device takes place (e.g. an expansion valve).

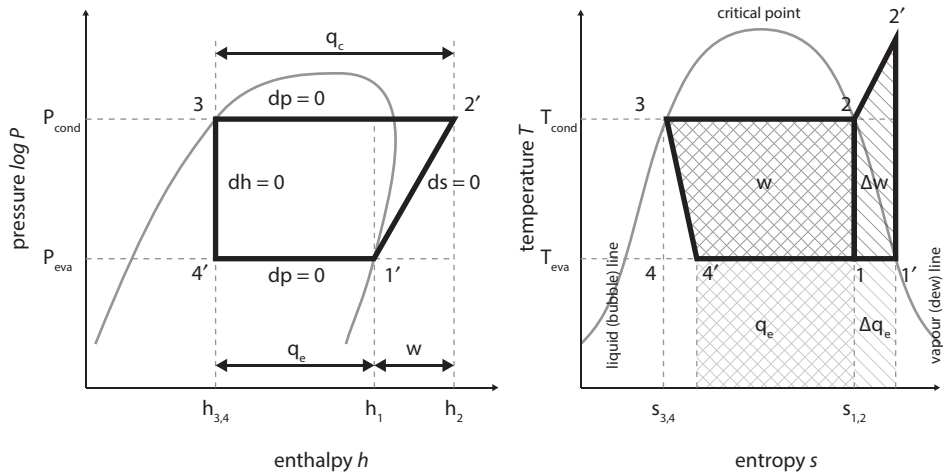


Figure 8.8 Theoretical dry process in a $\log p$ - h and T - s diagram.

The dry process can thus be characterised as follows:

- Saturated vapour suction (1')
- Isentropic compression (1' $\rightarrow$ 2')
- Isobaric heat dissipation (2' $\rightarrow$ 3)
- Isothermal heat dissipation (2 $\rightarrow$ 3)
- Isenthalpic expansion (3 $\rightarrow$ 4')
- Isothermal heating (4' $\rightarrow$ 1')

Process with superheating and subcooling

In the process of a real cold vapour machine, the evaporation of the refrigerant takes place not only to the right limit curve with subsequent saturated vapour suction, but includes overheating so that dry, superheated vapour is sucked into the compressor. Superheating means the heating of the dry saturated steam above the evaporation temperature T_{eva} near the right limit curve. There are two reasons for process design with superheating: first, it must be ensured that a complete evaporation

of the refrigerant over the entire flow cross section is made in order to prevent the entry of liquid droplets into the compressor, which could lead to mechanical damage; second, the superheating temperature difference $\Delta T = T_1 - T_{eva}$ can be used as a control variable for the evaporator control.

The liquefaction (condensation) to the left boundary curve can be followed by subcooling ΔT_{sub} . Subcooling means the cooling of the refrigerant to a temperature below the condensation temperature at saturation pressure. The reason for subcooling is to ensure the complete liquefaction. If one considers the processes in the flow cross section of the liquefaction (condensation), then the liquid refrigerant increases on the outer wall, while in the interior of the tube, a larger vapour portion remains. To ensure the conduction of heat from the vapour through the liquid layer on the wall, a subcooled liquid temperature is advantageous in order to achieve complete liquefaction. The irregular working of the throttle valves, as a result of unwanted evaporation should also be avoided. With subcooling, unwanted evaporation of the refrigerant is reduced, and the refrigerant enters with a lower vapour content to the evaporator, resulting in an increase of the cooling capacity.

In the real process irreversibilities occur during for all state changes.

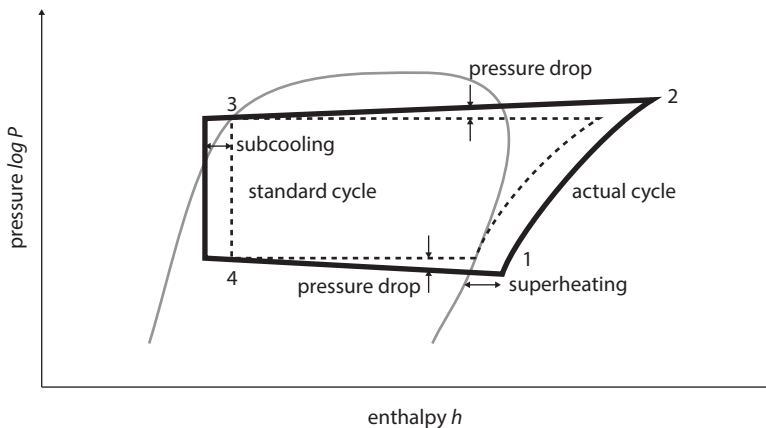


Figure 8.9 Real process compared with the standard cycle in the $\log p - h$ diagram.

The following state changes occur in real processes:

- Evaporation with pressure drop
- Superheating with pressure drop
- Compression above the condensing pressure
- Removal of the super heat with pressure drop
- Liquefaction (condensation) with pressure drop
- Subcooling with pressure drop
- Expansion with pressure drop

Thermodynamic properties of the compressor

The compressor is the core component of the vapour compression engines. Its task is to compress the evaporated refrigerant from the evaporation to the condensing pressure and also takes over the pumping function of the refrigerant circuit.

The crucial characteristic of the compressor and thus the entire heat pump is the isentropic efficiency η_{is} , which describes the compressor quality.

The polytropic compression is calculated by introducing the isentropic compression efficiency, which describes the ratio of enthalpy difference for ideal compression with the higher real enthalpy change caused by compression:

$$\eta_{is} = \frac{h_{2,is} - h_1}{h_{2,polytropic} - h_1} \quad (8.9)$$

Example 8.2

Refrigerant 134a enters the compressor of a refrigerator as superheated vapor at 0.14 MPa and -10°C with a flow rate of 0.05 kg s^{-1} and leaves at 0.8 MPa and 50°C . The refrigerant is cooled in the condenser to 26°C and 0.72 MPa and is throttled to 0.15 MPa.

Determine (a) the cooling power and the power input to the compressor, (b) the isentropic efficiency of the compressor and (c) the COP.

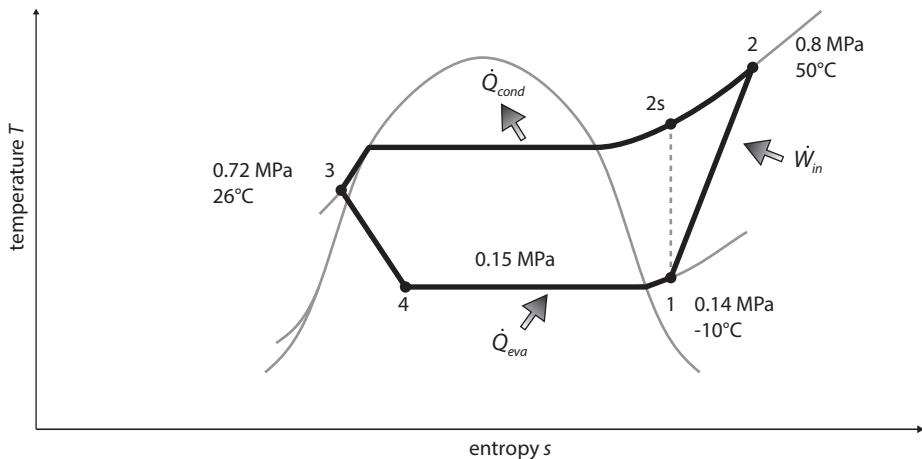


Figure 8.10 Isentropic and polytropic compression in the $\log p - h$ diagram for the example conditions.

Properties of R-134a :

State 1: Superheated with $p_1 = 0.14 \text{ MPa}$ and $T_1 = -10^\circ\text{C}$, $h_1 = 243.40 \text{ kJ kg}^{-1}$

State 2: Superheated with $p_2 = 0.8 \text{ MPa}$ and $T_2 = 50^\circ\text{C}$, $h_2 = 284.39 \text{ kJ kg}^{-1}$

State 3: Saturated liquid with $p_3 = 0.72 \text{ MPa}$ and $T_3 = 26^\circ\text{C}$, $h_3 = 85.75 \text{ kJ kg}^{-1}$

State 4: Throttling, $h_4 = h_3 = 85.75 \text{ kJ kg}^{-1}$

a) Heat removal from refrigerated space and work input:

Evaporator 4-1, with no work input $w = 0$:

$$q_{eva} = h_1 - h_4 \quad \dot{Q}_{eva} = \dot{m}q_{eva} = 0.05 \text{ kg s}^{-1} (243.40 - 85.75) \text{ kJ kg}^{-1} \\ = 7.88 \text{ kW}$$

Compressor 1-2, adiabatic compression $q = 0$

$$w_{in} = h_1 - h_2 \quad \dot{W}_{in} = \dot{m}w_{in} = 0.05 \text{ kg s}^{-1} (284.39 - 243.40) \text{ kJ kg}^{-1} \\ = 2.05 \text{ kW}$$

b) Isentropic efficiency of compressor:

$$\eta_{is} = \frac{h_{2s} - h_1}{h_2 - h_1}$$

State 2s: Superheated with $p_2 = 0.8 \text{ MPa}$ and $s_{2s} = s_1 = 0.9606 \text{ kJ kg}^{-1} \rightarrow h_{2s} = 281.05 \text{ kJ kg}^{-1}$

$$\eta_{is} = \frac{281.05 - 243.40}{284.39 - 243.40} = 0.919 = 91.9\%$$

The coefficient of performance:

$$COP = \frac{\dot{Q}_{eva}}{\dot{W}_{in}} = \frac{7.88 \text{ kW}}{2.05 \text{ kW}} = 3.84$$

The example was found in various internet presentations on engineering thermodynamics. Many more examples can be found in the web.

The isentropic compression efficiency can be approximated using a cross-term second-order polynomial with the following structure:

$$\eta_{is, poly} = a_0 + a_1 \cdot T_{cond} + a_2 \cdot T_{cond}^2 + a_3 \cdot T_{eva} + a_4 \cdot T_{eva}^2 + a_5 \cdot T_{cond} \cdot T_{eva} \quad (8.10)$$

The coefficients of this correlation can be obtained if the compressor performance data are known. Some manufacturers provide these data with polynomials according to EN129001. With these polynomial functions, the coefficients a_0 to a_5 can be calculated.

As an example, the isentropic efficiency of a scroll compressor is calculated as a function of the condenser and evaporator pressure for various condenser temperatures. The refrigerant is R-407C. The evaporator temperature varied in each case from -16 to +16°C.

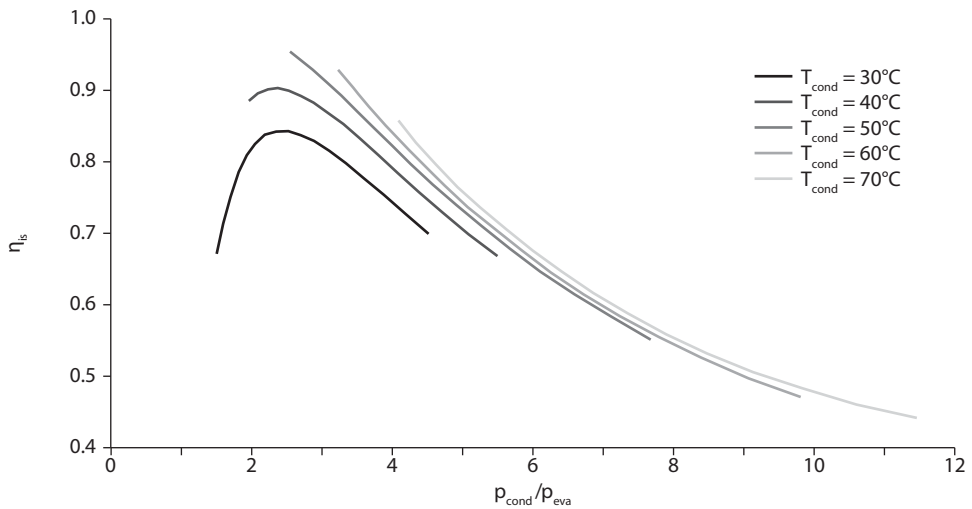


Figure 8.11 Isentropic efficiency, dependent on the condenser temperature.

The isentropic efficiency has a maximum at a pressure ratio between 2 and 4. The position of this maximum depends on the design features of the compressor, which is optimised for a specific operating point. The pressure ratio between 2 and 4 covers the usual temperature differences occurring between the evaporator and condenser in the heating mode. A nearly isentropic

compression occurs only if no heat exchange with the environment and no friction during the compression process occur.

Simulation model for heat pumps and chillers

A steady-state physical heat pump model based on the theory described above has been developed and implemented in the simulation environment INSEL (www.insel.eu).

The inputs required for the model are the inlet fluid temperature at the evaporator side (°C), the mass flow rate at the evaporator side (kg s^{-1}), the inlet fluid temperature at the condenser side (°C), the mass flow rate at the condenser side (kg s^{-1}), the mode (heating/cooling), the set point temperature for heating or cooling (°C), the superheating temperature difference ΔT_{sh} in evaporator (K), and the subcooling temperature difference ΔT_{sc} in condenser (K). The parameters needed for the model are shown in the simulation block screenshot.

The model can be used to simulate both heat pumps and vapour compression chillers. The set point temperature for heating or

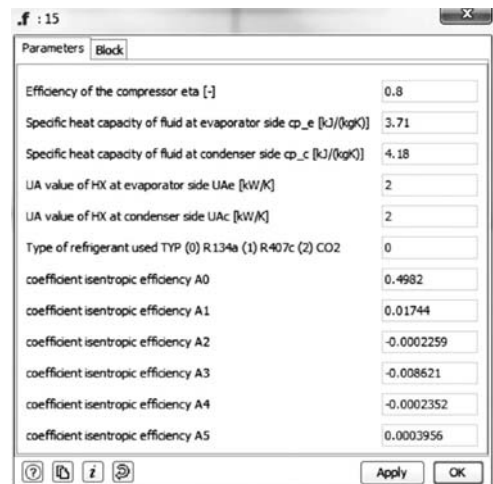


Figure 8.12 Screenshot of the parameter list in the simulation environment INSEL.

cooling is an input to the model. The UA values of both heat exchangers are assumed to be constant and must be provided by the user as parameters. The isentropic efficiency of the compressor is calculated using the cross-term correlation depending on condenser and evaporator temperatures. Outputs of the model are:

- Outlet fluid temperature at the evaporator side (°C)
- Outlet fluid temperature at the condenser side (°C)
- Evaporator temperature (°C)
- Condenser temperature (°C)
- Power at condenser side (kW)
- Power at evaporator side (kW)
- Mechanical work of the compressor (kW)
- Electrical power of the compressor (kW)
- Coefficient of performance (-)
- Isentropic compression efficiency (-)

Cycle calculations

The basis of the cycle calculation are the equations of state for the respective refrigerants. In the following $\log p-h$ diagram, the sequence of state points for the refrigerant R-407C are shown.

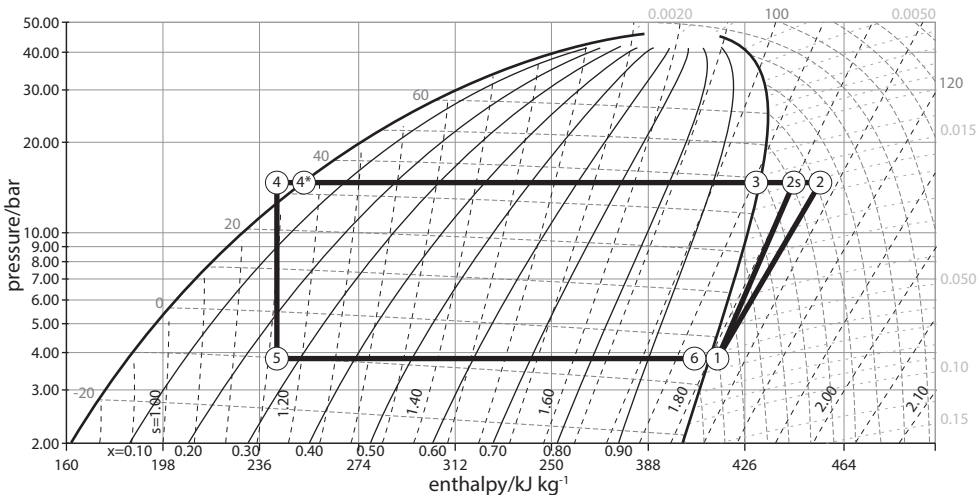


Figure 8.13 $\log p-h$ diagram of the refrigerant R407-C with marked state points of the refrigeration process.

The diagram shows the limiting curves of the wet vapour area, with the two horizontal isobars for the evaporation and condensation. The left limiting curve has the vapour content $x=0$, and separates the liquid area from the wet vapour area. In the liquid area, the isotherms run almost vertically. The right limit curve ($x=1$) separates the wet steam area from the pure vapour area, the region of the superheated steam.

First the enthalpies and entropies of all the points are calculated based on the given temperature levels and corresponding pressure levels in the evaporator and condenser.

Calculated points	Description	Formulae used
Point 1	Entry of the compressor with super-heated vapour	$T_1 = T_6 + \Delta T_{sh}$ $h_1 = h(R407C; T = T_1; p = p_{eva})$ $s_1 = s(R407C; T = T_1; p = p_{eva})$
Point 2s	End of the isentropic compression	$s_{2s} = s_1$ $s_{2s} = s(R407C; T = T_{2s}; p = p_{cond})$ $h_{2s} = h(R407C; T = T_{2s}; p = p_{cond})$
Point 2	End of the polytropic compression	$\eta_{is} = \frac{h_{2s} - h_1}{h_2 - h_1}$ $h_2 = h(R407C; T = T_2; p = p_{cond})$
Point 3	Entry of the condenser	$h_3 = h(R407C; T = T_3; x = 1)$ $p_{cond} = p(R407C; T = T_3; x = 1)$
Point 4*	Exit of the condenser with saturated liquid (before sub cooling)	$h_4 = h(R407C; T = T_{4,sl}; x = 0)$ $p_{cond} = p(R407C; T = T_{4,sl}; x = 0)$
Point 4	Entry of the expansion valve (after subcooling)	$T_4 = T_{4,sl} - \Delta T_{rc}$ $h_4 = h(R407C; T = T_4; p = p_{cond})$
Point 5	Exit of expansion valve, entry to evaporator	$h_5 = h_4$ $T_5 = T(R407C; p = p_{eva}; h = h_5)$
Point 6	Exit of the evaporator before super-heating	$T_6 = T(R407C; p = p_{eva}; x = 1)$ $h_6 = h(R407C; p = p_{eva}; x = 1)$

with

T_n	fluid temperature of the secondary or primary circuit [K]
p_n	pressure at the corresponding state [Pa]
h_n	enthalpy at point corresponding state [kJ kg ⁻¹]
s_n	entropy at the corresponding state point [kJ kg ⁻¹ K ⁻¹]
c_p	specific heat capacity [kJ kg ⁻¹ K ⁻¹]
η_{is}	isentropic efficiency [-]
x	vapour content of the wet steam area ($x = 0 \dots 1$)
UA_n	area times U value (properties of a heat exchanger)
$\dot{m}$	mass flow [kg s ⁻¹]
$\dot{Q}$	thermal power [kW]
P	electric power [kW]

To calculate the thermal power on the evaporator and condenser site the temperature levels need to be known. The model is simplified in the sense that the evaporator or condenser temperatures are calculated from the mean values of the temperatures of the corresponding state points.

For zeotropic refrigerant mixtures such as R407C, an extra computational effort is required to determine these mean condenser/evaporator temperature, since the evaporation and condensation do not occur at a constant temperature.

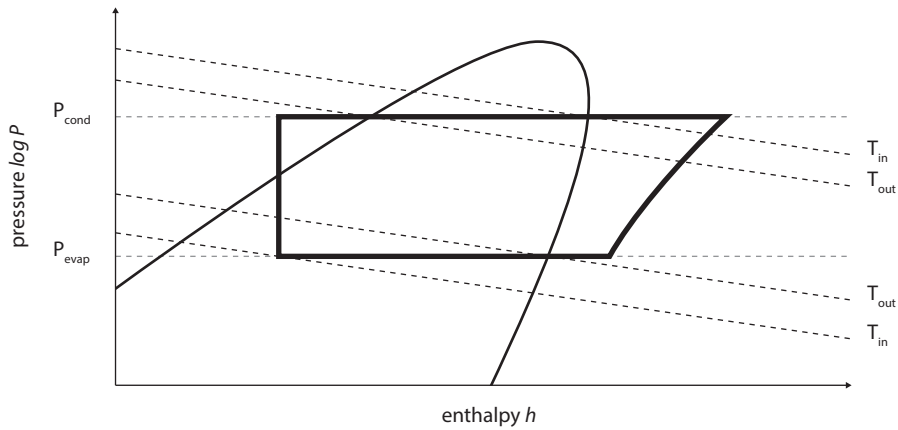


Figure 8.14 Pressure – enthalpy diagram for zeotropic mixtures with a lower evaporation temperature at the inlet of the evaporator and a higher condensation temperature at the inlet of the condenser.

In order to relate external heat transfer fluid temperatures with refrigerant temperatures, the two heat exchangers are treated as simple heat exchangers with phase change on one side using the NTU method.

It is assumed that the cooling machine can always supply the set point temperature. For the given inputs, the model calculates the electricity needed to reach the set point temperature.

Evaporator

$$T_{eva} = \frac{T_5 + T_6}{2} \quad (8.11)$$

$$\frac{T_{eva} - T_{W\,eva,out}}{T_{eva} - T_{W\,eva,in}} = \exp \left[\frac{-U A_{eva}}{\dot{m}_{eva} c_{p1}} \right] \quad (8.12)$$

$$\dot{Q}_{eva} = \dot{m}_{eva} c_{p1} (T_{W\,eva,in} - T_{W\,eva,out}) \quad (8.13)$$

$$\dot{Q}_{eva} = \dot{m}_{ref} (h_1 - h_5) \quad (8.14)$$

Condenser

$$T_{cond} = \frac{T_{4sl} + T_3}{2} \quad (8.15)$$

$$\frac{T_{cond} - T_{Wcond,in}}{T_{cond} - T_{Wcond,out}} = \exp \left[\frac{-U A_{cond}}{\dot{m}_{eva} c_{p2}} \right] \quad (8.16)$$

$$\dot{Q}_{cond} = \dot{m}_{cond} c_{p2} (T_{Wcond,in} - T_{Wcond,out}) \quad (8.17)$$

$$\dot{Q}_{cond} = \dot{m}_{ref} (h_2 - h_4) \quad (8.18)$$

Electrical Power

$$\eta_{is,en} = 0.4982 + 0.01744 T_{cond} + 0.0002259 T_{cond}^2 + 0.008621 T_{eva} \\ + 0.0002352 T_{eva}^2 + 0.0003956 T_{cond} T_{eva} \quad (8.19)$$

$$P_{elec} = \dot{m}_{ref} (h_2 - h_1) \quad (8.20)$$

$$P_{elec,con} = \frac{P_{elec}}{0.8} \quad (8.21)$$

$$COP = \frac{\dot{Q}_{cond}}{P_{elec,con}} \quad (8.22)$$

To calculate the connected electrical power, a mechanical efficiency and a motor drive efficiency of 90% each were assumed. The polynomial coefficients for the isentropic efficiency were calculated from the data provided by the compressor manufacturer for a 16 kW heat pump with a scroll compressor. For this machine the type of compressor used is known, as well as the two heat exchangers. Using the polynomial functions from Copeland software and a small programme written in EES, the isentropic efficiency can be calculated and then correlated to determine the parameters a_0 to a_5 .

The comparison between measurement and simulation shows good agreement. The following diagram depicts the typical daily operation of the heat pump when there is heat demand. The outlet water temperature at the condenser has been taken as set point and the model calculates the outlet brine temperature at the evaporator side, as well as the electrical power needed by the heat pump.

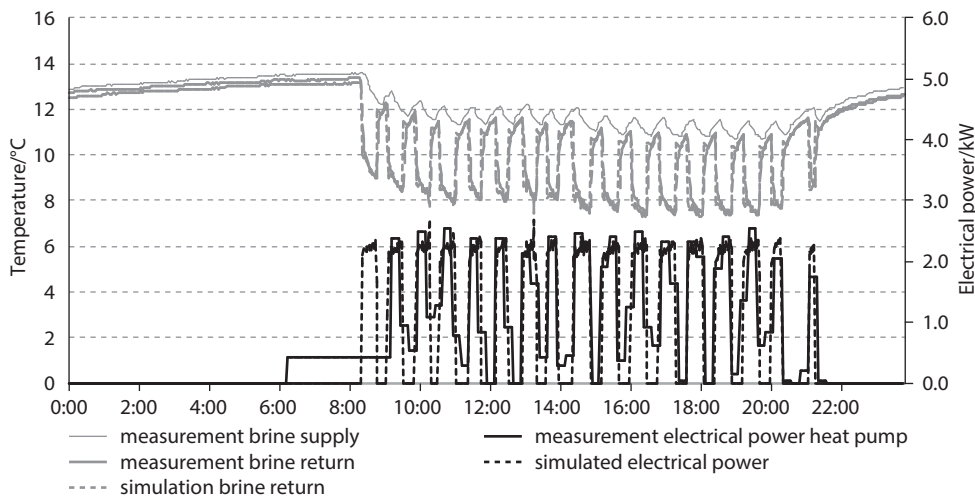


Figure 8.15 Comparison measurement/simulated values of a geothermal heat pump with 16 kW thermal power (21 February 2011).

Other days have been simulated and compared with measurement data. The three days in February correspond to typical winter days with heat demand, whereas the days in March and April correspond to spring days with lower heat demand. The COP only includes the electricity consumption for the compressor and internal controls.

Day	Q_{heat} /kWh	Q_{brine} /kWh	$Q_{brine\ sim}$ /kWh	P_{elec} /kWh	$P_{elec\ sim}$ /kWh	COP /-	COP sim /-
22 Feb 2011	86.3±3.2	74.8±4.0	76.9	18.2±0.9	17.6	4.7±0.4	4.9
23 Feb 2011	98.3±3.5	83.8±4.3	86.4	20.5±1.0	20.4	4.8±0.4	4.8
24 Feb 2011	80.9±3.1	69.1±3.9	73.1	16.9±0.8	16.7	4.8±0.4	4.8
20 March 2011	10.6±0.4	9.2±0.5	12.0	2.4±0.1	2.1	4.4±0.4	4.9
01 April 2011	17.7±0.7	14.4±0.8	16.1	4.0±0.2	3.6	4.4±0.4	4.9

Table 8.2 Comparison simulation/measurements.

The geothermal energy extraction for the five measurement days, the electricity consumption of the heat pump and the resulting COPs are shown in the next diagram. Measurements and simulations fit well.

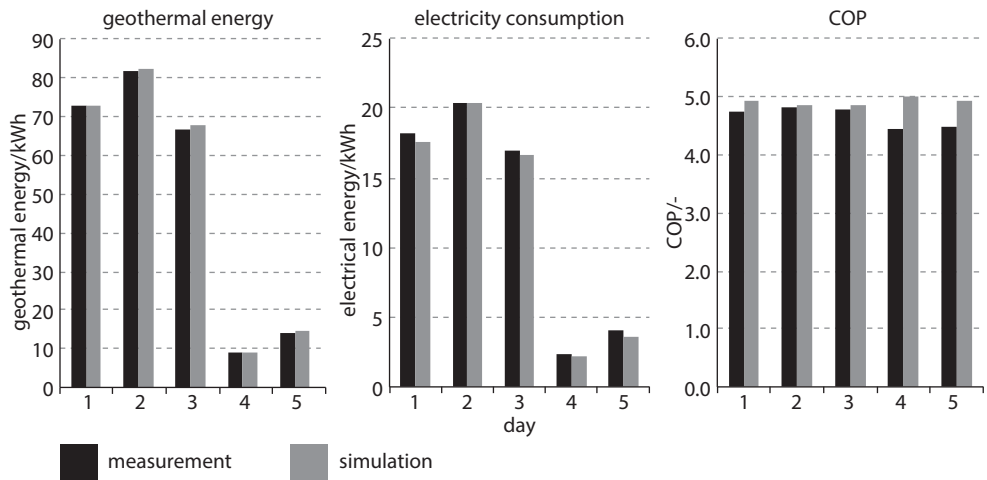


Figure 8.16 Comparison measurement/simulation for five different days of heat pump operation.

Furthermore, the model was used to simulate the performance of air cooled tandem scroll compressor cooling systems, which were used for a system comparison between photovoltaic and solar thermal cooling with cooling capacities of 30 kW, 40 kW and 50 kW. The refrigerant was again R407C as a zeotropic mixture with 52% R134a, but the isentropic efficiencies were calculated using R134a only.

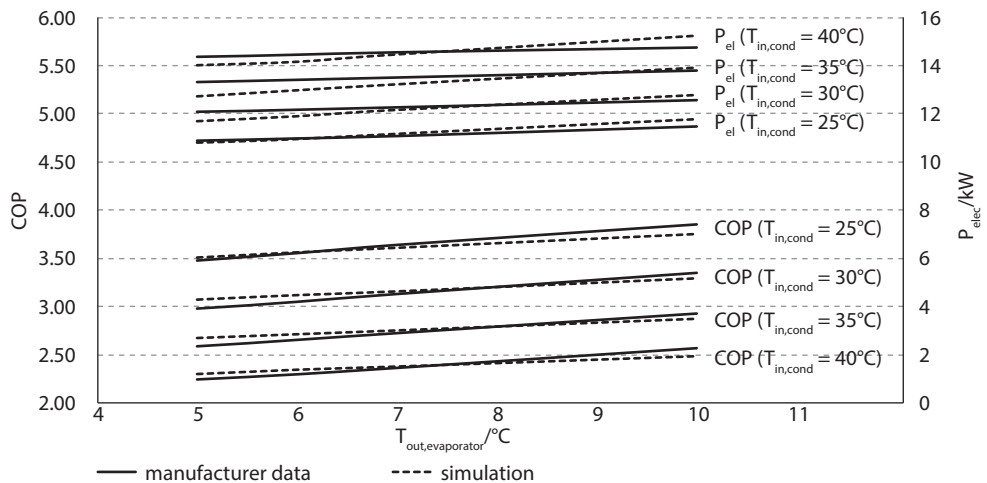


Figure 8.17 Simulated and manufacturer data of COP and electrical power of a compression chiller of 40kW cooling power.

8.4 Case studies for photovoltaic compression versus thermal cooling

A comparison was carried out between air cooled vapour compression chillers in the power range of 30 to 50 kW powered by the grid or by a PV system and a solar thermal-driven single effect absorption chiller for three different locations in Europe. A second comparison was done for two hot southern climates in Egypt and Cyprus with very different energy price structures.

Comparing photovoltaic cooling and single effect thermal chillers

Cooling loads were simulated on an hourly basis for a small three-storey office building defined in the IEA Task 25. The building is orientated with its main axis east-west and its characteristics are given in Table 8.3.

The description 'low' and 'high' are related to the cooling load scenarios, which are dominated by the external loads if there is no sun protection. Movable sun protections are closed if the radiation of the facade is higher than 300 W m^{-2} and open if the radiation on the facade is lower than 250 W m^{-2} .

The properties of construction are shown in Table 8.4 with representative U values for different wall types and windows corresponding to Palermo, Madrid and Stuttgart. A summary of the cases analysed is shown in Table 8.5.

The maximum cooling loads and annual cooling energy for all cases are summarised in Figure 8.18. If there is no night ventilation strategy with higher air exchange rates, then buildings with higher average U values have lower cooling demand (Case 1 with U values of the walls of $1.1 \text{ W m}^{-2} \text{ K}^{-1}$ compared to Case 2 with a U value of $0.41 \text{ W m}^{-2} \text{ K}^{-1}$). The highest cooling energy demand occurs, when no sun protection is provided, no night ventilation is used and the building is well insulated. Note that the maximum cooling load (in W m^{-2}) does not vary as much with building case and location, whereas the annual cooling energy requirement varies by more than a factor 3 as a function of shading system and internal loads.

System definition and performance results

The system comparison was carried out between a photovoltaic system with 21 kW peak power (100 modules, 210 W each) coupled to compression chillers between 30 and 50 kW depending on the load situation. The solar thermal cooling systems consist of either a flat-plate collector field (FPC) or compound parabolic concentrating vacuum tube collectors (CPC) system coupled to a 30 kW single effect absorption chiller.

Percentage of openings on external walls

Facade orientation	Glazed area/%
North	37
West	-
East	10
South	37

Properties of zones

Number of floors	3
Average height of floors/m	3.2
Longest facade (south)/m	21.3
Total reference surface/m ²	930

Infiltration and ventilation

Infiltration: air changes per hour/h ⁻¹	0.2
Mechanical ventilation: ventilation ratio (occupied)/h ⁻¹	0.34
Relative humidity set point/%	50
Air temperature of delivered air/°C	20

Internal gains, lighting and movable sun protection

	Cooling loads low	high
Specific gains, equipment, people and lighting/W m ⁻²	25	34
Occupation rate/occupants m ⁻²	0.034	0.1
Sun protection	Activated (low load scenario)	No sun protection (high load scenario)

Table 8.3 Building characteristics.

	Palermo		Madrid		Stuttgart	
Building type	Type 1	Type 4	Type 2	Type 4	Type 3	Type 4
Type of construction	1.10	0.41	0.66	0.41	1.10	0.41
U value/W m ⁻² K ⁻¹						
Type of window	Single	Triple	Double	Triple	Double	Triple
layer thickness/ mm	6	4/8/4/8/4	4/16/4	4/8/4/8/4	4/16/4	4/8/4/8/4
U window/W m ⁻² K ⁻¹	5.73	2.26	3.21	2.26	2.76	2.26
g value window [-]	0.837	0.678	0.72	0.678	0.4	0.678

Table 8.4 Properties of construction.

		Building type	$U_{\text{wall}}/\text{W m}^{-2} \text{K}^{-1}$	Internal loads	Sun protection	Annual cooling load (kWh m^{-2})	Annual heating load (kWh m^{-2})	Maximum cooling load (W m^{-2})
Case 1	Palermo	1	1.1	Low	Yes	46	18	38
Case 2	Palermo	4	0.41	Low	Yes	61	0	32
Case 3	Palermo	1	1.1	High	No	94	2	54
Case 4	Palermo	4	0.41	High	No	141	0	39
Case 5	Madrid	2	0.66	Low	Yes	34	11	34
Case 6	Madrid	4	0.41	Low	Yes	36	5	33
Case 7	Madrid	2	0.66	High	No	88	0	36
Case 8	Madrid	4	0.41	High	No	97	0	34
Case 9	Stuttgart	3	1.1	Low	Yes	8	56	24
Case 10	Stuttgart	4	0.41	Low	Yes	17	20	23
Case 11	Stuttgart	3	1.1	High	No	31	25	30
Case 12	Stuttgart	4	0.41	High	No	54	2	30

Table 8.5 Cases considered for the simulations.

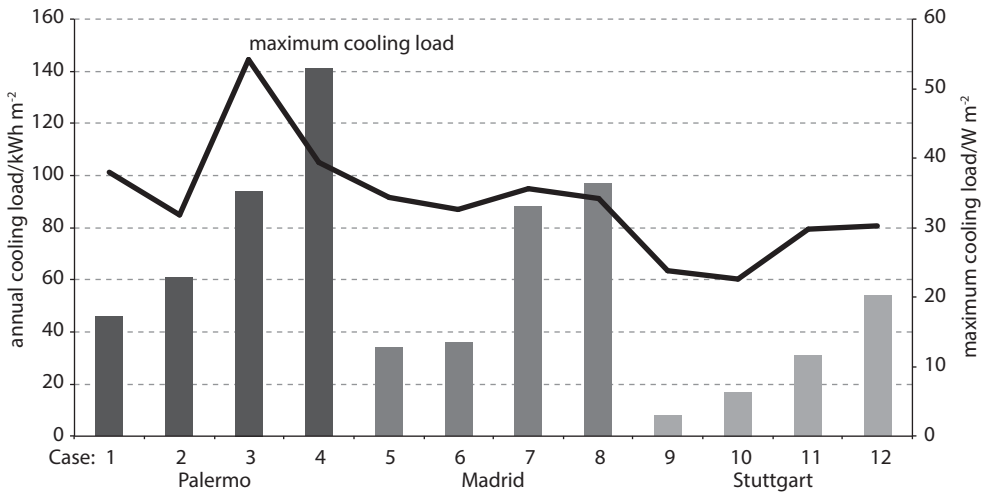


Figure 8.18 Cooling energy and maximum cooling load for buildings with different external and internal loads in three European locations.

The surface area of the active solar energy system is between 10% and 13% of the total office building surface.

Component	Characteristic	FPC	CPC	PV
Solar collector/module	total surface/m ²	112.5	93.2	125
Photovoltaic inverter	DC power/kW			2 × 10.3
Solar heat exchanger	heat transfer coefficient/kW K ⁻¹	12	9.5	
Solar storage	volume/litres	5000		
Cooling tower	air flow rate/m ³ h ⁻¹	9000		
	fan electric consumption/kW	0.89		
Thermal chiller	type	Absorption		
	nominal cooling power (kW)	30		
	nominal COP	0.7		
Cold storage tank	volume/litres	1000		
Electric chiller (Backup for thermal cooling systems)	nominal cooling power/kW	10.5	10.5	50
	nominal COP	3.5		
DHW Consumption profile	hot water temperature/°C	45		

Table 8.6 Characteristics of the components selected for thermal or electrical cooling for the location Palermo.

The simulation of the photovoltaic cooling system was done in the simulation environment INSEL (www.insel.eu). For each hourly time step, the current voltage characteristic of the generator is calculated and the maximum power point determined. This DC power is then used as an input to the inverter model, which simulates the conversion efficiency to AC power.

The performance results of the photovoltaic system for the three different locations are shown in Table 8.7.

	Palermo	Madrid	Stuttgart
Total energy yield/kWh kW _p ⁻¹	1561	1590	1033
Tilted irradiance (25°)/kWh m ⁻² a ⁻¹	1821	1840	1207
Horizontal irradiance/kWh m ⁻² a ⁻¹	1658	1655	1083

Table 8.7 Irradiance and photovoltaic electricity production for the different locations.

First the photovoltaic contribution to the cooling energy demand was calculated for the three different locations. The solar fraction is between 40% and 50% depending on the cooling demand load file. The remaining electricity is exported to the grid—the more so, the lower the total cooling energy demand.

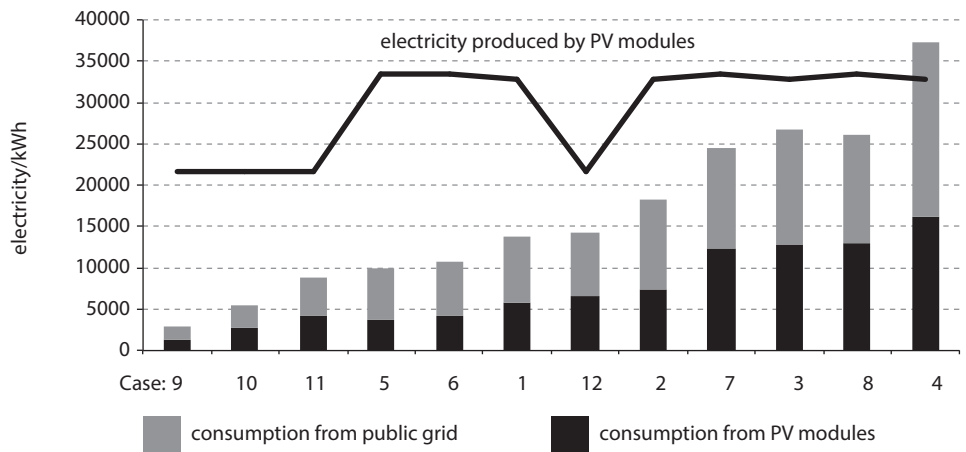


Figure 8.19 PV electricity produced and electricity consumed for cooling delivered by the PV system or by the public grid.

The solar fractions to the total cooling demand are very similar for the solar electric and the single effect solar thermal cooling system and are between 40% and 50% for most cases.

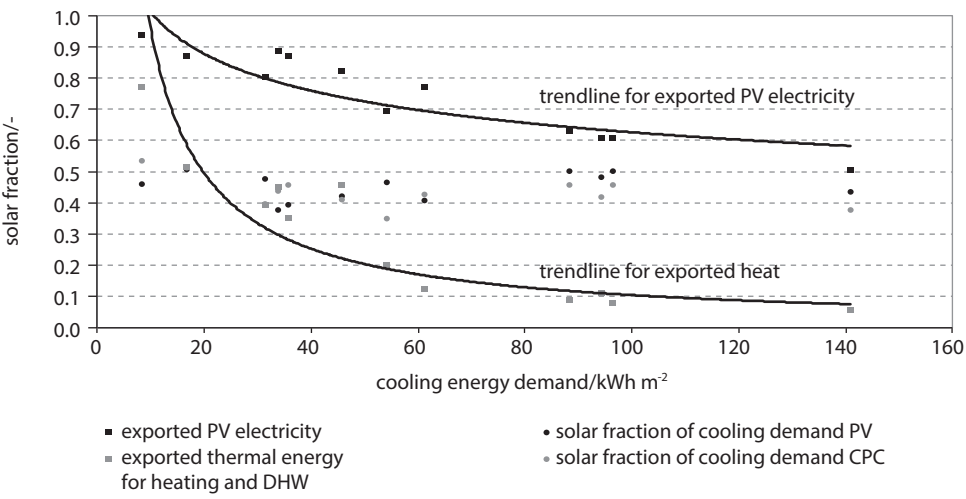


Figure 8.20 Solar PV and thermal (CPC) fraction used for cooling and exported PV electricity or CPC collector thermal energy used for heating and domestic hot water.

The generated solar electricity or heat not used for cooling can be either exported to the grid or used for domestic hot water or heating energy production. For buildings with low cooling energy demand, the exported PV electricity is very high between 80% and 90% of all PV electricity produced. The heating energy used within the building for domestic hot water or heating support very much depends on the climate and the building standard. Only in one case is the 'exported' solar thermal heat high, at nearly 80% for the Stuttgart location with high heating demand. In all

other cases, the exported fraction is either between 40% and 50% for moderate cooling and heating loads and drops to around 10% or less in locations with nearly no heating demand.

The auxiliary electrical energy consumption is about 20% higher for the solar thermal cooling system, as more heat has to be rejected for a single-effect thermal chiller when compared to an air cooled compressor. Therefore, the primary energy savings for cooling with the solar fraction between 40% and 50% are around 28 and 36% for a CPC thermal cooling system and between 40% and 48% for the PV cooling system.

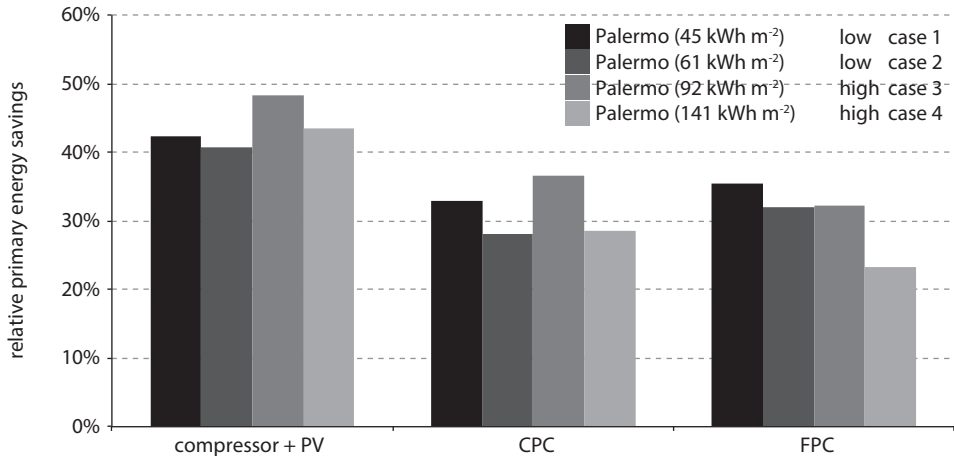


Figure 8.21 Primary energy savings of solar cooling systems compared to a reference electrical compression chiller system powered by the electrical grid.

Cost comparison

The parameters corresponding to the cost performance of the system are calculated following the procedure from the IEA Task 25. No funding subsidies are included.

Table 8.8 shows different cases considered for the calculation of cost. Three different cases are considered for the calculation of the annual electricity cost of the photovoltaic cooling system, depending on the connection of the system to the public grid and the feed-in tariff, which was varied between no feed-in compensation in Case A, medium tariffs in Case B (18.5 to 24 Eurocents kWh⁻¹) to very high tariffs in Case C (33 to 42 Eurocents kWh⁻¹) and low tariffs in Case D (18.5 to 21 Eurocents kWh⁻¹).

In the case of the thermal solar cooling system, Case A does not consider any benefit for the DHW and heating produced by the system. For Cases B and D, a benefit is calculated multiplying the gas tariff by the amount of heat corresponding to the domestic hot water and heating produced by the system. Here a low (around 5 Eurocents kWh⁻¹) and high (about 10 Eurocents kWh⁻¹) gas tariff was compared.

Case C is thus the most advantageous for PV cooling (very high feed-in, no compensation of heat produced), Case A the worse (no feed-in), Case D is the most advantageous for thermal cooling (low feed-in tariff, high benefit for heat produced).

Case	Photovoltaic cooling system		Thermal solar cooling system
	Feed-in tariff for electricity		Tariff of gas
A	no benefit for exported PV electricity		no benefit for heating and DHW
B	Germany	0.2455 € kWh ⁻¹	0.0572 € kWh ⁻¹
	Italy	0.2085 € kWh ⁻¹	0.0542 € kWh ⁻¹
	Spain	0.1855 € kWh ⁻¹	0.0506 € kWh ⁻¹
C	Germany	0.42 € kWh ⁻¹	-
	Italy	0.33 € kWh ⁻¹	-
	Spain	0.4 € kWh ⁻¹	-
D	Germany	0.19 € kWh ⁻¹	0.114 € kWh ⁻¹
	Italy	0.2085 € kWh ⁻¹	0.108 € kWh ⁻¹
	Spain	0.1855 € kWh ⁻¹	0.101 € kWh ⁻¹

Table 8.8 Cases for the calculation of electricity cost of the photovoltaic cooling system.

The assumptions listed in Table 8.9 were made to calculate the investment costs of the system.

In all cases, the chiller itself causes only about 20% or less of the total investment. The solar energy system dominates the costs with about 30 – 50% of the total costs followed by the installation costs. Note that the assumed price for photovoltaics of 2500 € kW⁻¹ has dropped further in some countries to below 1000 Euro kWp⁻¹.

The absolute capital costs are lowest for the compression chiller reference system, followed by the photovoltaic cooling system, the flat-plate thermal collector and the CPC collector absorption cooling system with a 30 kW chiller see (Figure 8.22)

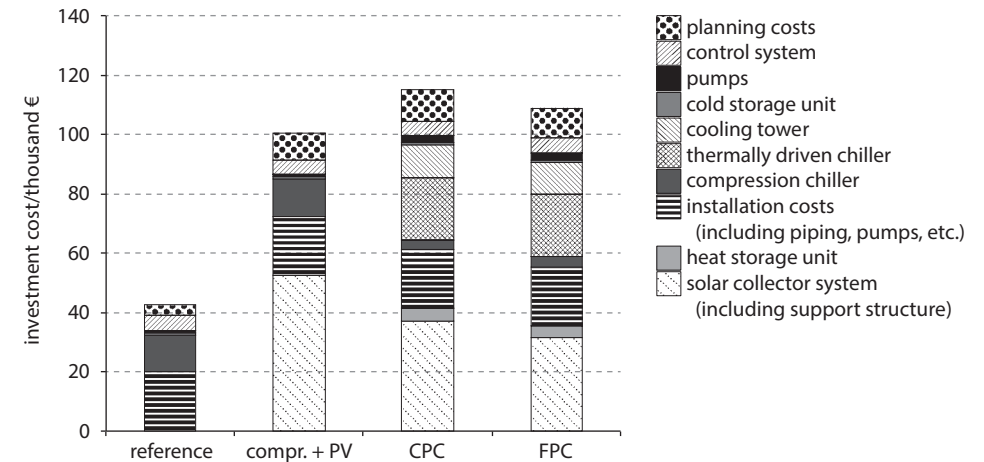


Figure 8.22 Investment costs for Case 1 in Palermo ($U = 1.1 \text{ W m}^{-2} \text{ K}^{-1}$) with a 30 kW compression chiller system compared to a 30 kW absorption chiller system.

		Stuttgart	Madrid	Palermo
PV modules	€ kW _{peak} ⁻¹	2500		
Flat-plate solar collectors FPC	€ m ⁻²	280		
Parabolic concentrators CPC	€ m ⁻²	400		
Evacuated tubes ETC	€ m ⁻²	350		
Heat storage unit	€ m ⁻³	800		
Backup heater	€ kW ⁻¹	120		
Installation of hydraulic system (cooling)	€	20000		
Installation of hydraulic system (heating)	€	17000		
Compression chiller	€ kW ⁻¹	310		
Absorption chiller	€ kW ⁻¹	700		
Cold storage unit	€	700		
Cooling tower	€ kW ⁻¹	150		
Solar pumps P ₁ , P ₂	€	800		
Delivery pumps P ₃ , P ₅	€	800		
Chiller pumps P ₄ , P ₆	€	800		
Control cost	€	5000		
Planning costs	% of inv. cost	10%		
Interest rate		6%		
Yearly maintenance cost of solar energy system	% of inv. cost	1.0%		
Yearly maintenance cost of other components	% of inv. cost	1.5%		
Electricity cost - energy	€ kWh ⁻¹	0.2455	0.1855	0.2085
Electricity cost - installed power (peak loads)	€ kW ⁻¹	75		
Water price	€ m ⁻³	3.88	1.77	1.21
Expected lifetime of solar energy system	years	20		
Lifetime of compression system	years	8		
Lifetime of other components	years	15		
Conversion factor electricity	kWh _{el} kWh _{primary} ⁻¹	0.36		
Conversion factor gas	kWh _{gas} kWh _{primary} ⁻¹	1		
CO ₂ emission rate (electricity)	kg kWh ⁻¹	0.5	0.55	0.8

Table 8.9 Cost and emission assumptions.

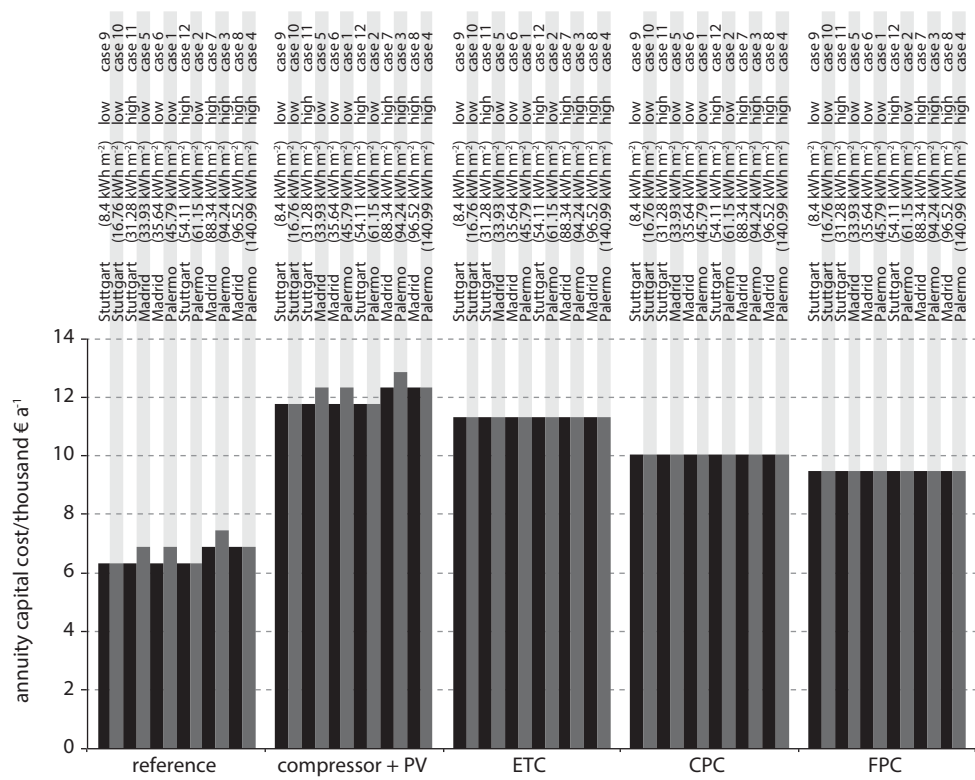


Figure 8.23 Annuity of capital costs for all cases.

If the annuity of the capital costs is calculated, the situation changes, as the lifetime of a compression chiller used in the reference and PV scenario is assumed to be lower (8 years) than an absorption chiller (15 years) and the solar components (20 years). The annuity for the compression chiller is 16% compared to 8.7% for the other components. Now the solar thermal systems are preferable to the PV system, but still more expensive in capital costs than the reference. The electrical compression system costs vary from case to case, as different sizes of compression chillers between 30 and 50 kW were chosen depending on the location and its maximum cooling load.

The cost of maintenance and inspection is considered as 1% of the investment cost for the solar collector and heat storage unit, and as 1.5% of the investment cost for the other components.

The annual operation and maintenance costs grow with the annual cooling load, since more electricity is used to operate the pumps, compressor, ventilator, cooling tower and back-up. In the case of the photovoltaic cooling system, the income generated by the feed-in tariff can lead to negative operation costs especially for low cooling load cases.

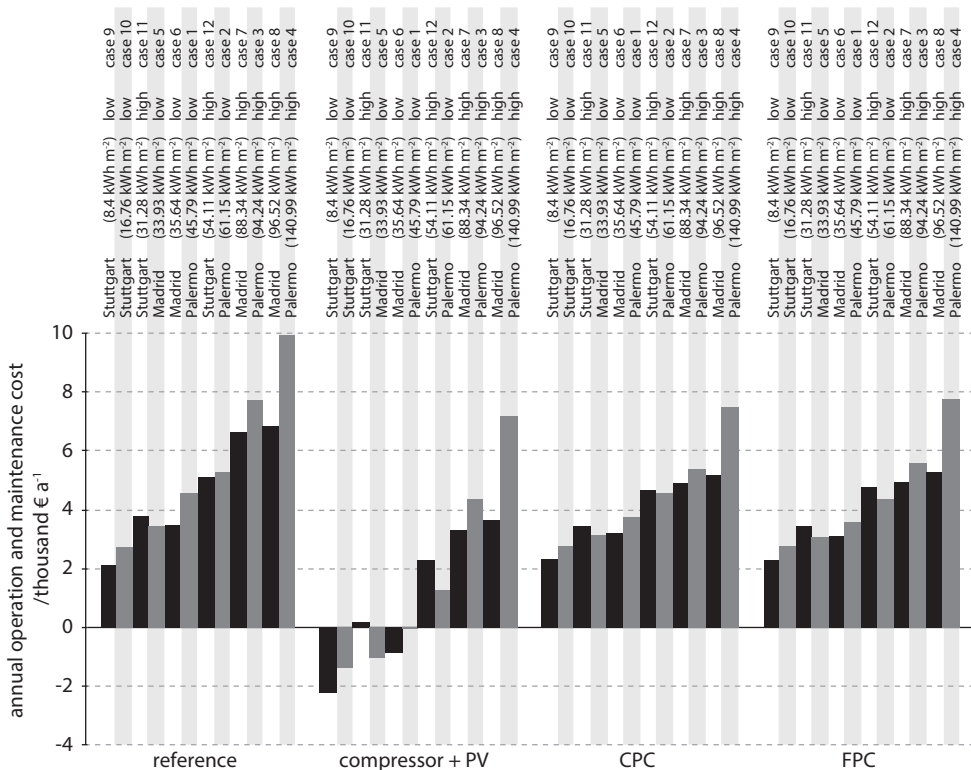


Figure 8.24 Annual operation and maintenance costs for conditions of case B, i.e. a net metering situation, where the same price is paid for exported and consumed electricity.

$$\begin{aligned}
 \text{operation and maintenance costs} &= \text{maintenance and inspection costs} \\
 &+ \text{electricity costs} \\
 &+ \text{water costs} \\
 &- \text{income from electricity or heat generation}
 \end{aligned} \quad (8.23)$$

The total annual costs of the PV compression system with net metering feed-in tariffs are lower than the solar cooling systems for cases with low cooling load, i.e. high fractions of exported energy. For lower fractions of exported energy, i.e. high cooling loads, the thermal system performs better in terms of costs. In some cases the thermal cooling systems with flat-plate collectors are even cheaper than the chosen reference system costs (for the location Palermo with high cooling energy demand).

$$\begin{aligned}
 \text{total annual costs} &= \text{capital costs} \\
 &+ \text{operation costs} \\
 &+ \text{maintenance costs}
 \end{aligned} \quad (8.24)$$

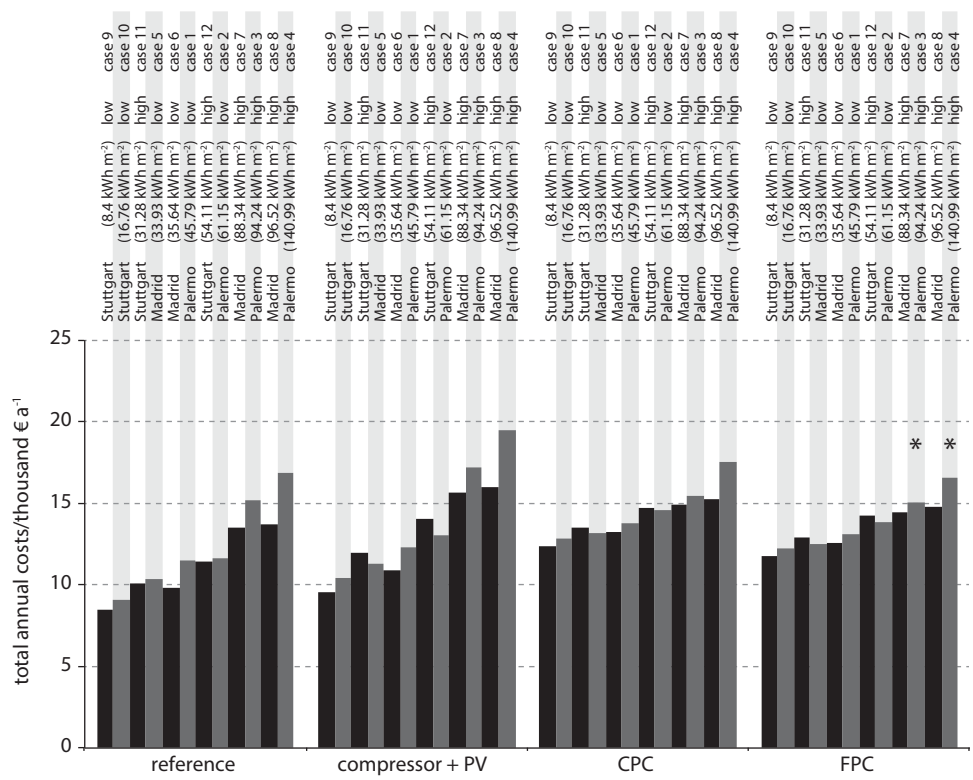


Figure 8.25 Total annual cost for case B with feed-in tariffs equivalent to domestic electricity prices. The stars mark cases where solar thermal cooling is cheaper than the reference system.

The cost of the cooling energy decreases as expected with increasing load hours. PV and solar thermal costs are comparable for moderate feed-in tariff conditions. PV cooling is only cheaper for very high feed-in tariffs (Case C) and high exported energy fractions, i.e. low cooling demand.

$$\text{cost of cooling production} = \frac{\text{annual total costs of solar system}}{\text{total cooling energy produced}} \quad (8.25)$$

In addition, the costs per kWh of saved primary energy were calculated, which also decrease inversely with annual cooling load. For the Case A, since no feed-in tariff is considered, the cost of saved primary energy of the photovoltaic cooling system are high and very near to the values of the thermal solar cooling system (FPC and CPC). The Cases B and C correspond to a photovoltaic cooling system considering two prices for the feed-in tariff. Since the feed-in tariff received in Case C is higher than in Case B, the cost of saved primary energy are lower and negative for low cooling loads, because in Case C the energy supplied by the photovoltaic modules covers the total cost and a benefit is additionally received.

$$\text{cost of saved primary energy} = \frac{\text{annual extra cost of solar system}}{\text{annual primary energy savings}} \quad (8.26)$$

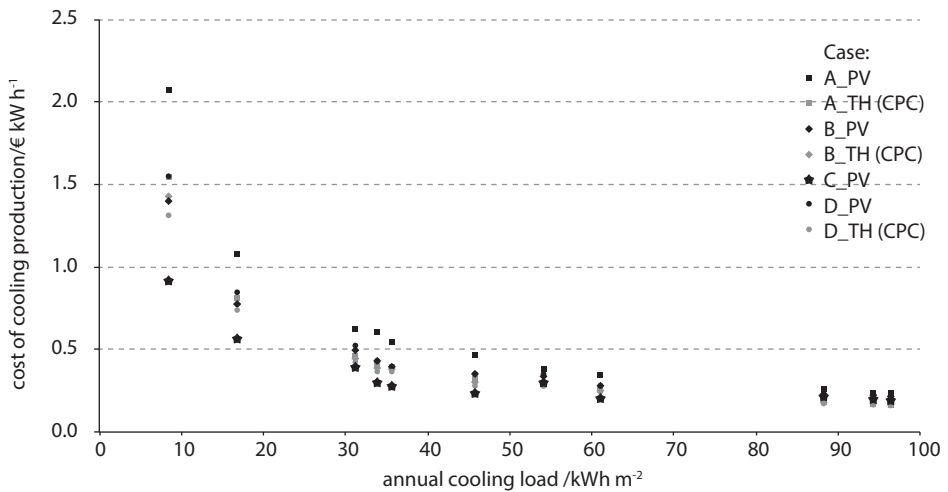


Figure 8.26 Cooling energy costs for different feed in tariffs.

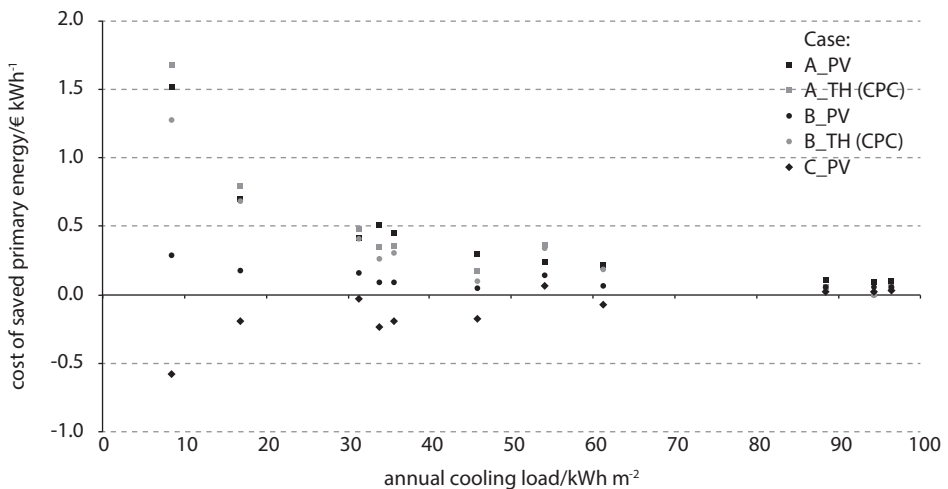


Figure 8.27 Costs of saved primary energy as a function of the annual cooling energy demand.

Comparing photovoltaic cooling and multi effect thermal cooling systems

A solar cooling case study was done for a large office building in Cairo/Egypt with a total useful floor area of 15 100 m² and a conditioned volume of 55 116 m³. Double glazed windows with sun protecting coating are considered for the fully glazed facades with a U value of 1.16 W m⁻² K⁻¹ and g value of 0.265. Additional shading is provided by a roof overhang of 2.5 m in the upper floors of the south, southeast and southwest facing facades. For all opaque building elements like external walls, roof and floors, an insulation of 20 cm is considered resulting in U values of 0.18 W m⁻² K⁻¹. The resulting maximum cooling load of the building is 800 kW (52 W m⁻²) and the annual cooling

energy demand is 1970 MWh a⁻¹ (130 kWh m⁻² a⁻¹). Due to the necessity of dehumidification in summer, the temperature level of the cold water circuit is 7°C/14°C.

Solar cooling systems

The limiting factor for the size of the solar cooling systems is the available and usable roof area, which is only 2 000 m², i.e. 13% of the total air conditioned surface. For the system design, simulations were performed for single effect, double effect and triple effect absorption chillers. The single effect absorption chiller was combined with efficient vacuum tube collectors with an optical efficiency of 0.65, a linear heat transfer coefficient of 1.585 W m⁻² K⁻¹ and a temperature dependent quadratic heat transfer coefficient of 0.002 W m⁻² K⁻². The maximum possible collector size at horizontal orientation is 2050 m² gross collector area, which is equal to a collector aperture area of 1350 m². For the double effect absorption chiller, linear concentrating Fresnel collectors are considered. The optical efficiency of the Fresnel collectors is 62% with a linear heat transfer coefficient of 0.1 W m⁻² K⁻¹ and a temperature dependent quadratic heat transfer coefficient of 0.00043 W m⁻² K⁻². For the linear Fresnel collectors, the maximum collector aperture area is 1320 m² (60 collectors with 4 m length and 8 m width).

To evaluate the optimum system configuration the size of the hot water storage and the capacity of the absorption chillers were varied and the optimum system design found for each of the solar thermal cooling systems was selected. For the PV driven compression chiller, the available and useful roof area of 2000 m² allows the installation of 1200 m² mono crystalline PV modules with an optimum inclination of 25° towards the south. Dynamic annual simulations were performed for the following four system configurations:

- 1: Single effect absorption cooling machine (ACM) 422 kW (THERMAX, ProChill LT12C), 7°C/12.2°C cold water, wet cooling tower, vacuum tube collector field for hot water supply. 2025 m² gross collector area, 1350 m² collector aperture area, 3.3 kW electricity consumption solar pump, 20 m³ hot water storage and 10 m³ cold water storage.
- 2: Double effect ACM 500 kW (Jiangsu, Shuangliang) 7°C/12°C cold water, wet cooling tower, linear concentrating Fresnel collectors, 2050 m² gross collector area including spaces between the rows, 1320 m² collector aperture area, 3.2 kW electricity consumption solar pump, 20 m³ pressurised hot water storage (max. 200°C) and 10 m³ cold water storage.
- 3: Triple effect ACM 563 kW vapour driven (250°C) (Kawasaki Sigma Ace CF01-10-0001), 7°C/15°C cold water, wet cooling tower, linear concentrating Fresnel collectors for steam supply (max. 250°C at 3.9 MPa) 1280 m² gross collector area including spaces between the rows, 880 m² collector aperture area, 1.8 kW electricity consumption solar pump, no hot water storage and 10 m³ cold water storage.
- 4: Compression Chiller 795 kW (Quantum A090 3C12 with R-134a as refrigerant), 7°C/12°C cold water, integrated direct dry heat rejection, electrical COP of 2.9 at 100%, 3.9 at 75%, 4.9 at 50% and 6.5 at 25% cooling capacity. A 10 m³ cold water storage is considered
- 5: PV system: 875 modules with 180 W peak power, 25° inclination towards the south, 1206 m² total module area, 156 kW_p total installed power at maximum power point, with 150 kW inverter.

For the thermal cooling systems additional cooling is provided by an electric compression chiller with an average electrical COP of 2.8. This includes the electricity consumption of the compression chiller, of the dry heat rejection system and of all connected pumps. For heat rejection of the thermally driven water-LiBr absorption chillers, wet cooling towers are considered with

frequency inverters for fan speed control at part load conditions. Compared with the single effect absorption chiller, the required heat rejection energy is much lower for the triple effect chiller, but due to the high mass flow rate in the absorber/condenser circuit, a bigger cooling tower is required.

Solar cooling results

The fraction of the thermally driven absorption chillers on the overall cooling energy demand of the building together with the solar energy system efficiency is shown in Figure 8.28. The lowest thermal cooling fraction of 37% is obtained for the single effect absorption chiller, since no backup heating is used in this case. This system reaches the highest overall solar thermal system efficiency of 40%. The much lower solar energy system efficiency of 27 - 31% of the concentrating collector results mainly from the fact that these collectors can only use the direct solar radiation. In the annual average, the direct beam radiation in Cairo is only 60% of the total solar radiation. The system with the double effect absorption chiller and an auxiliary system heating reaches 91% thermal cooling fraction of the total annual cooling load, since only the peak loads above 500 kW need to be covered by the compression chiller. The triple effect absorption chiller reaches a higher maximum cooling power of 563 kW and is therefore able to cover 93% of the annual cooling load of the building.

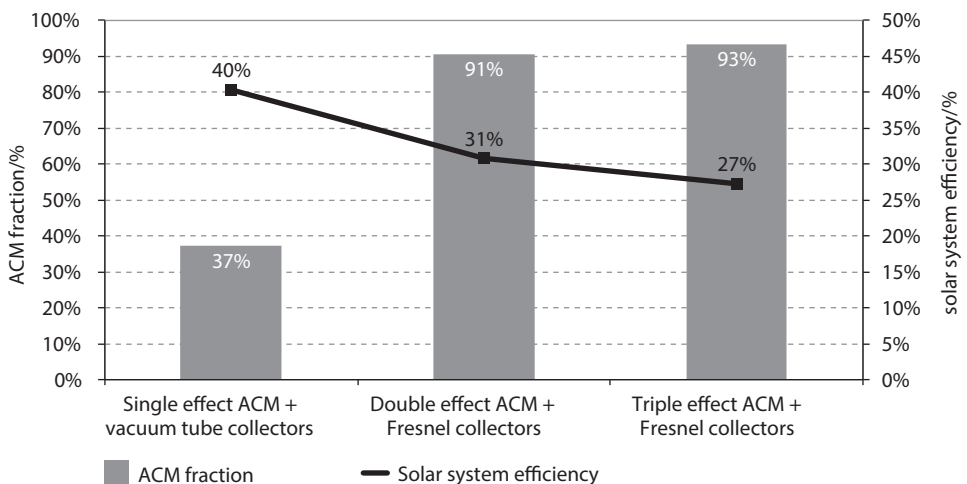


Figure 8.28 Fraction of the ACM on the cooling load and solar energy system efficiency.

The solar heating energy and the additional heating energy provided to the absorption chillers are shown in Figure 8.29 together with the average thermal COP of the chillers, which are 0.7 for the single effect, 1.31 for the double effect and 1.83 for the triple effect chiller. Due to the higher thermal COP, the double and triple effect chillers require much lower heating energy than the single effect system. Although the double effect system covers 91% instead of 37% (single effect) of the annual cooling energy demand, the required total heating energy demand is only 30% higher than the solar heating energy demand of the single effect chiller. The triple effect chiller requires even 4% less heating energy compared to the single effect chiller although it covers 93% instead of 37% of

the annual cooling load. The size of the of the solar collector system for the triple effect chiller is 33% smaller than the collectors for the double effect absorption chiller (880 m² instead of 1320 m²).

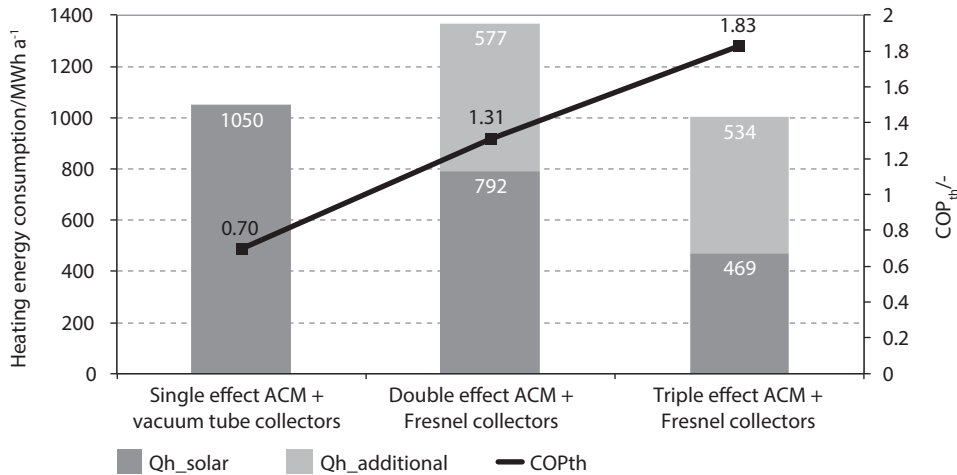


Figure 8.29 Solar heating, additional heating and average thermal COP of the multi-effect thermal cooling systems.

The partial load control strategies for all thermal cooling systems were optimised in the simulation model so that a high ratio of cooling energy to auxiliary electricity consumption was obtained. This is mainly achieved by reducing the cooling tower ventilation power under part load conditions. As a result, the electrical COP is higher than 10 for all three systems (see Figure 8.30).

Figure 8.31 shows the primary energy consumption of the four analysed solar cooling systems compared to the primary energy consumption of a reference system with an efficient compression chiller. The resulting average primary energy ratio (PER) as the quotient of total cooling energy provided and total primary energy consumed is also shown in this graph.

The overall best energetic performance is reached for the triple effect absorption chiller, which reaches a primary energy ratio of 1.6, i.e. 12% more than the single effect system. If the local electricity grid is considered as ideal storage, i.e. all the excess PV electricity can be exported, the PER of the PV driven compression chiller is only slightly lower at 1.59 compared to the best thermal cooling system (PV total electricity production corresponds to 33% of cooling electricity demand). If only the produced electricity that can be directly used by the chiller is considered (22% PV solar fraction of total cooling electricity, 11% can not be used), the PER decreases to 1.37, which is even worse than the single effect absorption cooling system. This is due to the chosen simple dry heat rejection system for the compression chiller, whereas the absorption chiller uses a well-controlled wet cooling tower. Although the heat rejection energy is higher for the single effect absorber, the auxiliary energy consumption is slightly less than for the compression chiller.

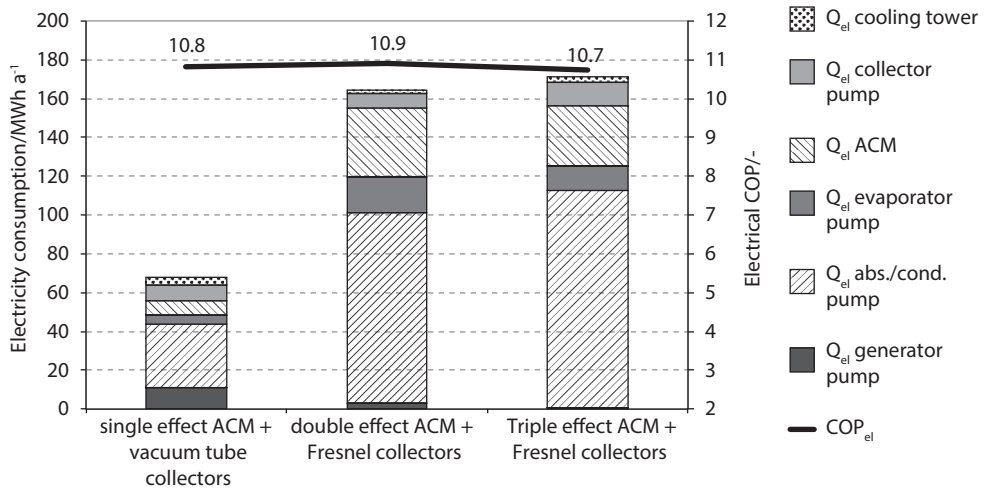


Figure 8.30 Auxiliary electricity consumption and electrical coefficient of performance for the three absorption chillers investigated.

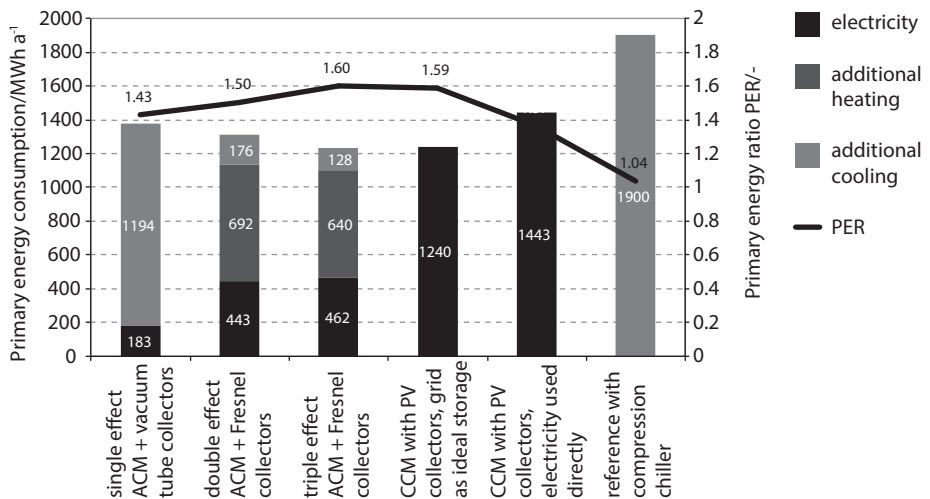


Figure 8.31 Primary energy consumption and average primary energy ratio (PER).

Compared to an efficient standard compression cooling system only fed by the local grid, all analysed cooling systems reach significantly higher primary energy ratios of +38% in case of the single effect absorption chiller up to +54% in case of the triple effect chiller with Fresnel collectors. This highlights the main advantage of efficiently designed and controlled solar cooling systems.

Cost comparison

The following assumptions were taken for the economic evaluation of the thermal and electrical cooling systems:

Case	Chiller type	Specific chiller costs	Heat rejection	Solar heating system	Specific solar energy system costs
Case 1	Single Effect	250 € kW ⁻¹	20 € kW ⁻¹	Vacuum tube	350 € m ⁻² (Aperture)
Case 2	Double Effect	300 € kW ⁻¹	20 € kW ⁻¹	Fresnel	500 € m ⁻² (Aperture)
Case 3	Triple Effect	500 € kW ⁻¹	20 € kW ⁻¹	Fresnel	500 € m ⁻² (Aperture)
Case 4-6	Compression chiller	200 € kW ⁻¹	Integrated air cooled	PV	3 000 € kW _p ⁻¹

Table 8.10 Assumptions for the economic evaluation.

Note that the costs for photovoltaic systems has dropped further to 1000 Euros kW_p⁻¹ in some countries.

For the piping, 20% of equipment costs were added. For the overall installation and system integration, 30% of the total system costs were added. The integration costs are usually the highest cost risk factor. The total system costs are highest for the double effect absorption chiller system due to the large and expensive collector field required and lowest for the compression chiller without solar input.

The distribution of operational costs shows comparable or cheaper costs for the solar thermal cooling systems when compared to the reference system or the PV cooling. Only when benefits from selling PV electricity to the grid are obtained, will the PV cooling system have lower operational costs.

The operational costs strongly depend on the energy prices in the countries of installation. Whereas in Egypt the energy prices are currently heavily subsidised and are only 0.0174 Euros kWh⁻¹ for gas and 0.062 Euros kWh⁻¹ for electricity, they are at 0.101 Euros kWh⁻¹ for gas in Cyprus and 0.23 Euros kWh⁻¹ for electricity. Water costs are 1.6 Euros m⁻³ in Egypt and 4 Euros m⁻³ in Cyprus.

As a result, the overall cooling costs in Egypt are about half of the cooling costs in Cyprus. Capital costs dominate the overall costs for countries with very low energy prices and vice versa for higher energy prices.

In countries with high energy prices, the PV cooling system is already more economic than the reference electrical compression system, especially if the excess electricity can be sold to the grid. The solar thermal cooling systems are more expensive, but offer the highest primary energy savings (the triple effect machine).

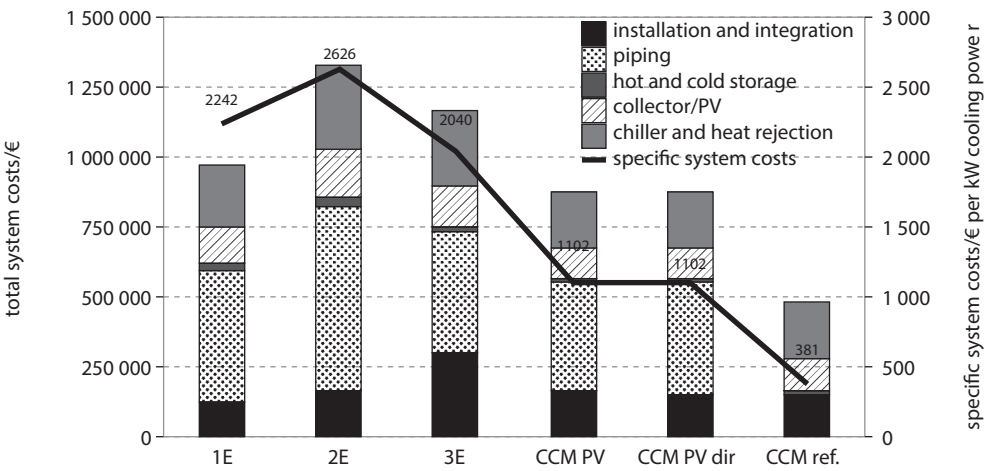


Figure 8.32 Total investment cost for all systems analysed.

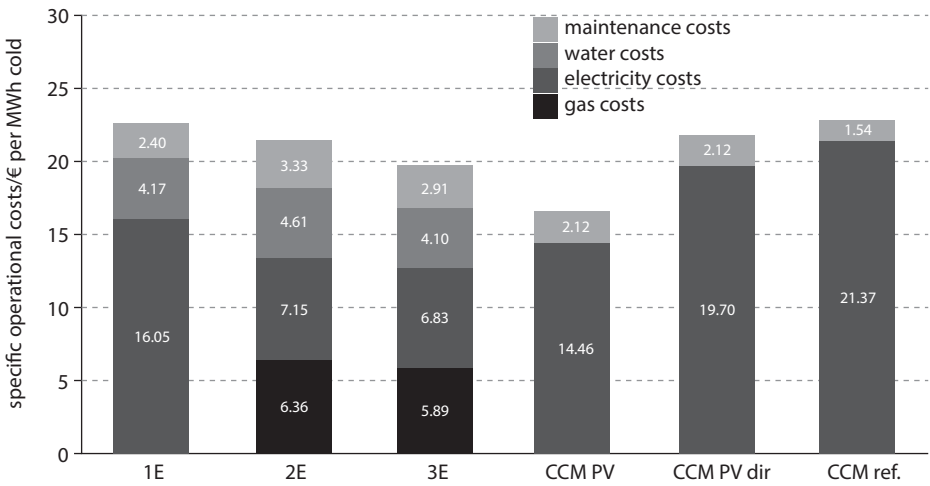


Figure 8.33 Specific operational costs for Egypt with low energy prices.

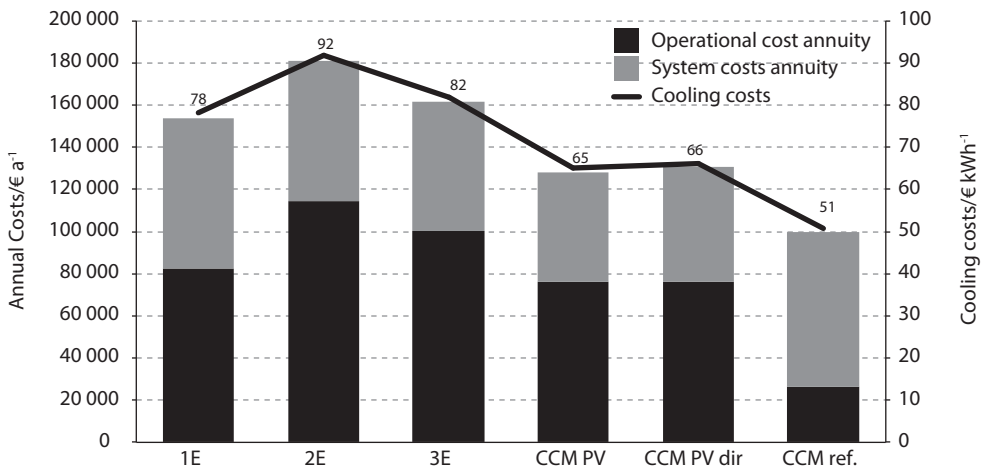


Figure 8.34 Annual costs for cooling in Egypt with very low energy prices.

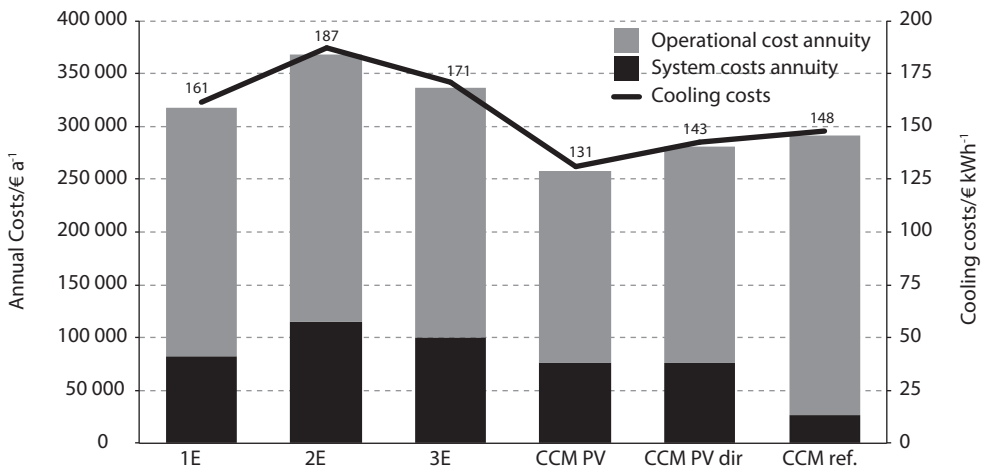


Figure 8.35 Annual cooling costs for Cyprus with higher energy prices.

8.5 Conclusions on case studies for photovoltaic and thermal cooling

In conclusion, the study of single effect absorption chillers for office buildings in various European locations showed that solar thermal cooling can be cost efficient today for high cooling load applications when compared to a compression chiller system with or without photovoltaic energy. Backup cooling is always recommended for single effect machines to achieve good primary energy efficiencies. The auxiliary energy consumption has to be minimised by efficient components and control strategies.

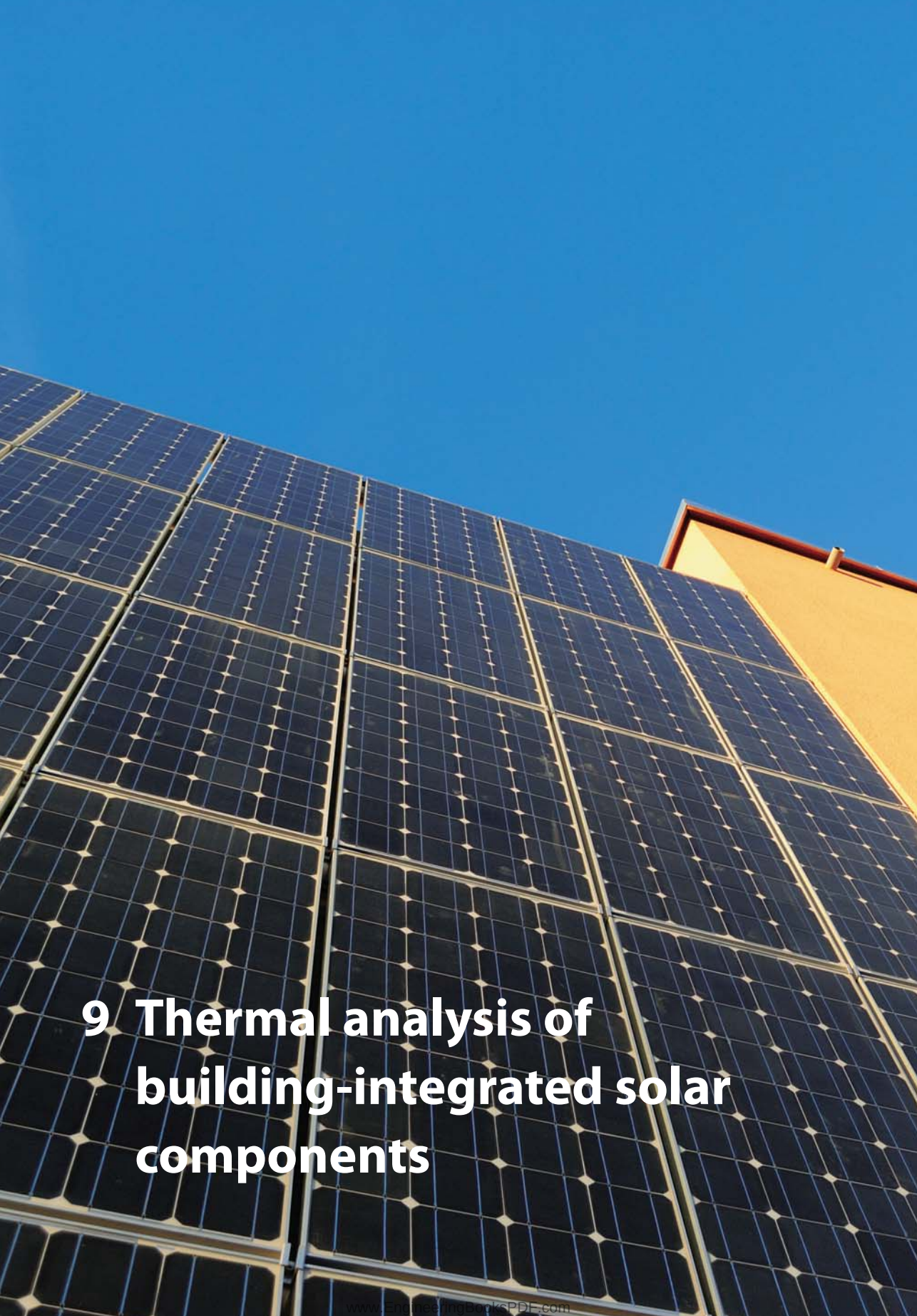
The results for an office building in Cairo or Cyprus locations show the overall best performance with a primary energy ratio of 1.6 was reached for a triple effect chiller with backup

heating (1st choice) and backup cooling (second choice). Double effect absorption chillers with backup heating (1st choice) and backup cooling (second choice) have only a slightly higher primary energy ratio than single effect absorption chillers with backup cooling only. The PV driven compression chillers reach comparable primary energy ratios, if the PV electricity that is not used for cooling can be exported to the grid and thus, additionally saves primary energy. Otherwise, the primary energy ratio of this system is lower than the analysed thermal cooling systems. However, it could be shown that all analysed solar cooling systems reach 32% to 54% higher primary energy efficiencies than standard systems with compression chillers.

The economic performance strongly depends on the boundary conditions chosen. In countries with very low, mostly subsidised energy prices, the higher capital costs dominate the annual cooling costs and an electric compression system is always cheaper than solar options. Higher energy prices favour the solar powered options, both photovoltaic and solar thermal cooling systems.

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9 Thermal analysis of building-integrated solar components

Figure 9.1 Title image: PV facade of the ParkHaus in Scharnhauser Park near Stuttgart (photo: Seubert).



Figure 9.2 Stuttgart Solar Decathlon building home+ with a mono crystalline PVT roof and coloured poly crystalline PV facade elements (photo: Jan Cremers, HFT Stuttgart).

The substitution of classical building materials with active solar components represents a particularly interesting multi-functional use of solar technology in buildings. Active solar components produce electricity or heat, which is transferred to a fluid such as air or water in a controllable way. In addition, with building integration, heat flows to the building occur which can be described by heat transfer coefficients and total energy transmission factors. Different options for building integration exist, which can be characterised by the overall optics, i.e. whether the building-integrated solar component is opaque or semi-transparent, and by the integration type, i.e. whether the solar component is ventilated, integrated into a sandwich element with insulation or into a glazing structure.

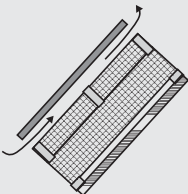
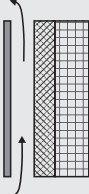
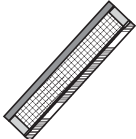
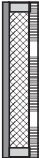
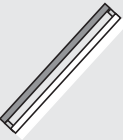
Optics	Roof	Facade	Integration type
Opaque			Ventilated
Opaque			Sandwich
Semi-transparent			Glazing

Table 9.1 Possibilities for building integration.

By using ventilated double facades with photovoltaic panels as an example, a methodology can be developed for thermal characterisation that enables computations of the heating energy and cooling load of a building with integrated solar components.

Previous thermal analysis of active solar components (air and water collectors, and photovoltaic) has been based on the assumption of thermal separation from the building, i.e. the calorific losses of the solar radiation absorber were computed on both sides against the ambient air temperature T_a . With building-integrated solar components, in particular warm facades, the assumption of a solar element surrounded by outside air is no longer applicable. While most thermal flat-plate collectors have sufficient back insulation (with insulating thickness > 6 cm),

partial transparent photovoltaic modules, for example, are often only separated by a further glazing from the room (with room air temperature T_i) for architectural reasons.

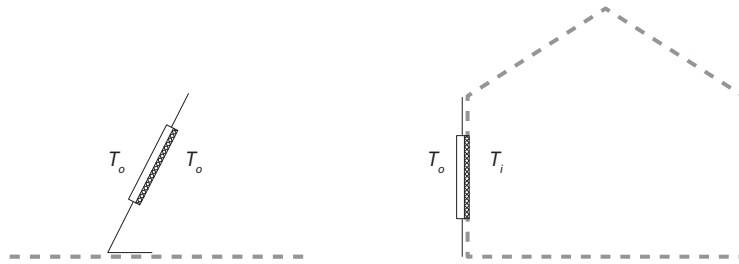


Figure 9.3 Freestanding and building-integrated collectors.

Due to this thermal coupling, heat gains for the space occur contributing in winter to the heating energy demand, but in summer can cause overheating problems. With photovoltaic modules in double-glazing construction, of special interest are the surface temperatures of the module (to determine the electrical power) and of the glazing on the room side (to determine the effective total energy transmittance characterised by the g value).

With back-ventilated PV double facades, the heat supplied by the modules can serve as useful thermal energy for preheating outside air. At the same time, the heated gap air can recover transmission heat losses of the room.

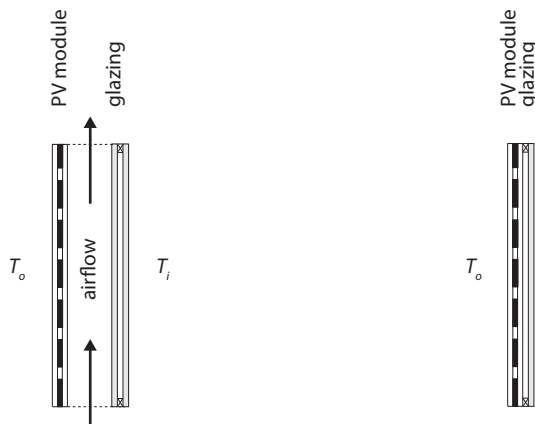


Figure 9.4 PV module in a back-ventilated cavity facade with double-glazing on the room side and a PV module integrated in double-glazing construction.

First an overview is given of typical temperature conditions of building-integrated solar components, mainly photovoltaic modules, in different building-integration versions. The empirical equations obtained from measured data enable fast analysis of the temperature influence on the annual electrical efficiency and on the thermal behaviour of the building.



Figure 9.5 (previous page): Exterior and interior views of a semi-transparent photovoltaic facade with insulating glass (courtesy of Firma Grammer, Stadt Regensburg, Hochbauamt, photo: Peter Ferstl).



Figure 9.6 Back-ventilated cavity photovoltaic facade of a public library in Mataró (photo: Ursula Eicker).



Figure 9.7 Back-ventilated cavity photovoltaic facade of a public library in Mataró (photo: Ursula Eicker).

Subsequently, a more detailed thermal model of a back-ventilated photovoltaic facade is developed which describes heat flows and temperature conditions in conventional double-glazing facades. With this model, the monthly thermal energy gains of a PV cavity facade can be calculated.

9.1 Empirical thermal model of building-integrated photovoltaic

Solar modules are generally characterised by high optical absorption coefficients in the wavelength area of short-wave solar radiation. In photovoltaic modules, however, only a small part of the absorbed irradiance, about 10 - 20%, is converted into electricity, with mainly heat being produced. This heat can be used for heating purposes, but it reduces the electrical power of the PV generator due to the module's rise in temperature. Crucial for the temperature levels are, at a given solar irradiance, the convective heat transfer mechanisms at the front and rear of the module, which depend mainly on the wind velocity.

The convective and radiant heat transfer of the module rear side is especially influenced by the installation situation. For detailed calculations of the temperatures, the respective relevant Nußelt correlations must be determined in each case; they depend on geometry, heat flow density, degree of turbulence etc.

For a rough estimate of the temperature conditions in different installation situations, it is sufficient to use linear regressions of temperature rises against the irradiance derived from measurements. The linear connections between the module temperature and irradiance neglect the strong dispersion of the measured values, in particular due to wind influences, but they lead to a sufficiently exact estimate of the electrical power losses and the mean temperature rise at a given irradiance. A thermal model has been developed by Sauer (1995), validated at building-integrated components, and regression analyses covering all relevant integration possibilities have been performed for twelve German climatic test reference years for different installation situations:

Nr	Module mounting	Ventilation
1	Free standing module	Optimum ventilation
2	Roof mounted module, large distance between module and roof tiles	Optimum ventilation
3	Roof mounted, mean distance between module and roof tiles	Good ventilation
4	Roof mounted, small distance between module and roof tiles	Limited ventilation
5	Roof integrated	Without ventilation
6	Cold facade with large air gap	Good ventilation
7	Cold facade with small air gap	Limited ventilation
8	Facade integrated	Without ventilation

Table 9.2 Installation situations of building-integrated solar elements.

The gradient of the linear regression curves shows the rise in the temperature difference between the module and environment $\Delta T = T_{Module} - T_o$ per $W\ m^{-2}$ of irradiance increase ΔG . From this, the

temperature rise at 1000 W m^{-2} irradiance can then be calculated, to compare the installation situations.

$$(T_{\text{Module}} - T_o) \Big|_{1000 \text{ W/m}^2} = \frac{\Delta T}{\Delta G} \times 1000 \frac{\text{W}}{\text{m}^2} \quad (9.1)$$

The gradient obtained as an average of the test reference years varies from a minimum of $0.019 \text{ K (W m}^{-2})^{-1}$ for a freestanding module, up to $0.052 \text{ K (W m}^{-2})^{-1}$ for a facade without back ventilation, so at 1000 W m^{-2} irradiance, module temperatures of $19 - 52 \text{ K}$ above the ambient temperature appear.

The mean temperature rises at 1000 W m^{-2} irradiance are represented together with the minima and maxima of the twelve German meteorological test reference years for all eight installation situations. From this the regression coefficient for each installation situation can be read off. The fluctuations for a given installation situation are mainly caused by different wind velocities at the locations. In addition, the relative electrical power losses are represented compared to those of the freestanding module. In the most unfavourable version – the non-back-ventilated facade – $7.5\% - 10\%$ less electricity is produced annually than by the freestanding module, due to temperature effects alone. The electrical energy losses are calculated relative to the annually produced energy of a freestanding module.

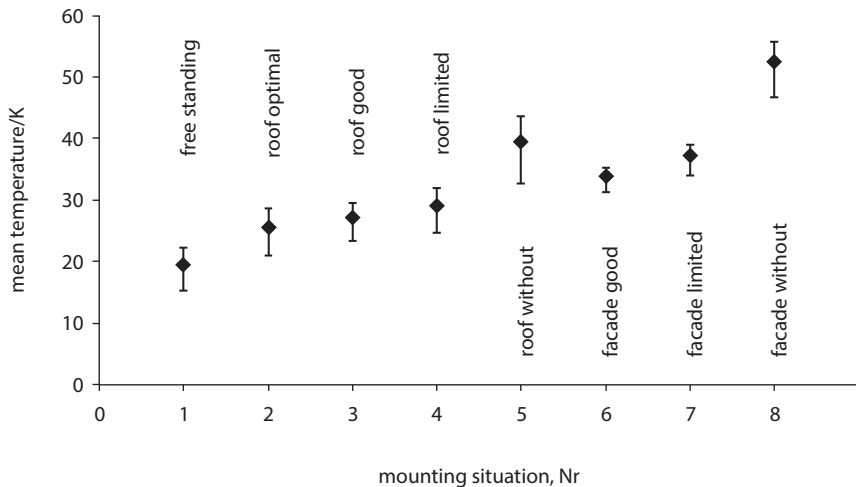


Figure 9.8 Mean annual temperature rise of a PV module in different building integration solutions with optimal, good and limited ventilation or without ventilation.

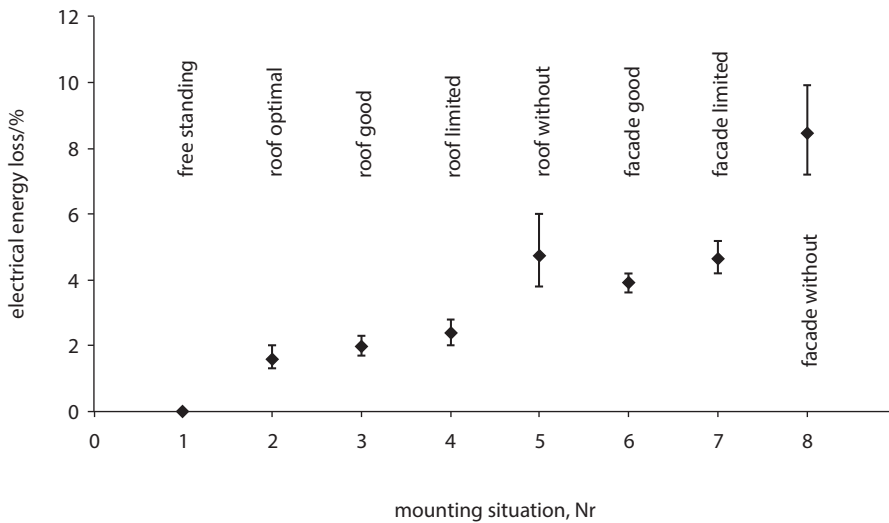


Figure 9.9 Yearly electrical energy loss of a PV module in different building integration solutions.

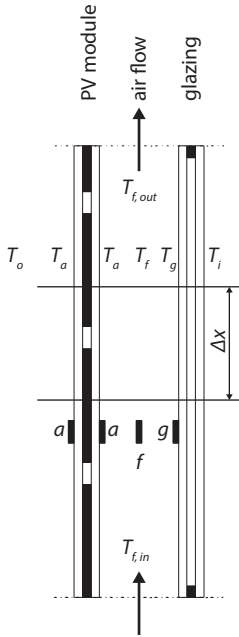
9.2 Energy balance and stationary thermal model of ventilated double facades

While the empirical regression equations for building-integrated photovoltaic elements provide sufficient accuracy for module temperature calculation and electrical performance analysis, a more exact model was created to analyse the effective heat transfer and thermal use of the produced PV module heat. This model permits different temperatures as a boundary condition of the integrated solar component (ambient temperature and room temperature), and takes into account the heat flows from the absorbing solar element or from the room into a ventilation gap.

Energy balances for each temperature node are set up to calculate the temperatures, and the set of equations produced thus is solved. Since most solar components have only small thermal masses, a stationary energy balance is sufficiently exact. The calculation methodology, which can be generalised for different installation situations, is explained by the example of a back-ventilated photovoltaic heated facade (Fux, 2006).

The structure of the facade corresponds to a typical double facade construction, with the photovoltaic module constituting the outer shell, back-ventilated with outside air. The back-ventilation can be either free convection or fan-operated. In cavity facades, the back-ventilation gap typically has dimensions of between about 0.1 and 1 m, so in contrast to commercial air collectors, low flow velocities can generally be expected. The gap dimensions and flow rates influence in particular the convective heat transfer in the air gap and thus the thermal efficiency, which in general is far lower than in turbulent flow air collectors.

For the energy balance, three temperature nodes are considered: node *a* for the absorber (here the PV module), node *f* for the fluid (here air) and node *b* for the gap closing glazing to the room. Due to the thinness of the PV laminates, typically two glass sheets with 4 mm and 6 mm, only one temperature node is used for the photovoltaic module. A stationary energy balance is set up for the three temperature nodes.



T_o	ambient temperature/°C
T_a	absorber (= PV module) temperature/°C
T_f	fluid temperature/°C
T_b	glass temperature at the back of the gap/°C
T_i	room temperature/°C
h_c	convective heat transfer coefficient/W m ² K ⁻¹
h_r	radiant heat transfer coefficient/W m ² K ⁻¹
α_{PV}	effective absorption coefficient of PV cells
τ_{PV}	effective transmission coefficient of the PV module
α_b	absorption coefficient of the rear (glazing)
U_b	U value between the gap rear (node b) and room/W m ² K ⁻¹
U_f	U value between the PV module (node a) and environment/W m ² K ⁻¹
c	heat capacity of air/J kg ⁻¹ K ⁻¹
ρ	density of air/kg m ⁻³
$\dot{V}$	flow rate/m ³ s ⁻¹
L	gap height/m

Figure 9.10 Temperature nodes and descriptions of a building-integrated back-ventilated photovoltaic facade.

Node a

The PV module as a thermal absorber absorbs solar irradiance G with the effective absorption coefficient α_{PV} , which includes absorption by the solar cell, reflection losses and remaining transmittance of the PV glass pane, and is typically about 80%. The calorific losses of the absorber are divided into a heat flow to the environment via the facade front heat transfer coefficient U_f , a convective heat flow from the absorber to the gap air with the heat transfer coefficient h_{ca} , and into a radiation exchange with the gap rear, the heat transfer coefficient being h_r . The electrical power is deducted in the balance from the absorbed irradiance.

$$G\alpha_{PV} - U_f(T_a - T_o) - h_{ca}(T_a - T_f) - h_r(T_a - T_b) - \dot{Q}_{el} = 0 \quad (9.2)$$

Node f

By the convective heat transfer of the two gap confinement surfaces (absorber and gap rear) with temperatures T_a and T_b and gap width b , the fluid temperature is increased over the distance dx .

$$c\rho\dot{V}\frac{dT_f}{dx} = h_{ca}b(T_a - T_f) + h_{cb}b(T_b - T_f) \quad (9.3)$$

For the gap width b , the unit length $b = 1 \text{ m}$ is used here.

Node b

The solar irradiance transmitted by the PV module, with the transmission coefficient t_{PV} , is absorbed with the absorption coefficient of the back (glazing) α_b . The calorific losses consist of the radiation exchange with the PV module, the convective heat transfer to the fluid with the heat transfer coefficient h_{cb} and of the losses to the room, with the heat transfer coefficient U_b .

$$G(\tau_{PV}\alpha_b) - h_r(T_b - T_a) - h_{cb}(T_b - T_f) - U_b(T_b - T_i) = 0 \quad (9.4)$$

The front heat transfer coefficient of the facade U_f is calculated as usual from the thermal resistance of the PV module with layer thickness s_{PV}/m and heat conductivity $\lambda_{PV}/W\ m^{-1}\ K^{-1}$, and from the outside heat transfer coefficient $h_a/W\ m^{-2}\ K^{-1}$. The U value of the rear U_b consists of the thermal resistance of the gap rear $R_b/m^2\ K\ W^{-1}$ (e.g. $0.3\ m^2\ K\ W^{-1}$ for non-coated double glazing), and of the internal thermal resistance between the surface and room $1/h_i$:

$$U_f = \frac{1}{\frac{s_{PV}}{\lambda_{PV}} + \frac{1}{h_a}} \quad (9.5)$$

$$U_b = \frac{1}{R_b + \frac{1}{h_i}} \quad (9.6)$$

Equation (9.2) and Equation (9.4) are used to represent the absorber temperature T_a and the glass temperature T_b as a function of T_f , T_i and T_o , and then to use them in Equation (9.3).

$$T_a = \frac{(B\alpha_{PV} + h_r\tau_{PV}\alpha_b)G + h_rU_bT_i + BU_fT_o + (Bh_{ca} + h_rh_{cb})T_f - B\dot{Q}_{el}}{AB - h_r^2} \quad (9.7)$$

$$T_b = \frac{(A\tau_{PV}\alpha_b + h_r\alpha_{PV})G + AU_bT_i + h_rU_fT_o + (Ah_{cb} + h_rh_{ca})T_f - h_r\dot{Q}_{el}}{AB - h_r^2} \quad (9.8)$$

From this results a differential equation for the fluid temperature T_f which can be solved by separation of the variables.

$$c\rho V \frac{dT_f}{dx} = D_1T_i + D_2T_o - D_3\dot{Q}_{el} + D_4G - D_5T_f \quad (9.9)$$

The general solution is

$$T_f(x) = \frac{1}{D_5} (D_1T_i + D_2T_o - D_3\dot{Q}_{el} + D_4G - Ce^{-Zx}) \quad (9.10)$$

with the integration constant C dependent on the boundary condition and the constants D_1 to D_5 given by

$$D_1 = \frac{h_{ca}h_rU_b + h_{cb}AU_b}{AB - h_r^2}$$

$$D_2 = \frac{h_{cb}h_rU_f + h_{ca}BU_f}{AB - h_r^2}$$

$$D_3 = \frac{h_{ca}B + h_{cb}h_r}{AB - h_r^2}$$

$$D_4 = \frac{h_{ca}h_r\tau_{pV}\alpha_b + h_{ca}B\alpha_{pV} + h_{cb}h_r\alpha_{pV} + h_{cb}A\tau_{pV}\alpha_b}{AB - h_r^2}$$

with $A = U_f + h_{ca} + h_r$, $B = U_b + h_{cb} + h_r$ and $Z = \frac{D_5}{c\rho V}$.

The integration constant C results from the boundary condition at the gap inlet; here the ambient temperature T_o :

Boundary condition, with $x = 0$, $T_f = T_o$

$$C = D_1T_i + (D_2 - D_5)T_o - D_3\dot{Q}_{el} + D_4G \quad (9.11)$$

Thus the special solution results in

$$T_f(x) = (1 - e^{-Zx}) \frac{D_1T_i + D_2T_o - D_3\dot{Q}_{el} + D_4G}{D_5} + T_o e^{-Zx} \quad (9.12)$$

The mean fluid temperature of the entire flow channel is obtained by integrating Equation (9.12) over the entire gap length L .

$$\begin{aligned} \bar{T}_f &= \frac{1}{L} \int_0^L T_f(x) dx \\ &= \frac{D_1T_i + D_2T_o - D_3\dot{Q}_{el} + D_4G}{D_5} \left(1 + \frac{1}{LZ} (e^{-ZL} - 1) \right) + \frac{T_o}{LZ} (1 - e^{-ZL}) \end{aligned} \quad (9.13)$$

With the mean fluid temperature, the mean absorber and rear temperatures of the gap can also be calculated, using Equation (9.7) and Equation (9.8).

The coefficients D_1 to D_5 depend on the heat transfer coefficients for convection and radiation in the gap, and on the heat transfer coefficients to the environment or to the interior. Since these are also temperature-dependent, the gap temperatures can only be determined iteratively.

Before the effective heat transfer coefficients (U and g values) are determined from the temperatures, a brief examination will be made of the necessary heat transfer coefficients and relevant Nußelt correlations for the heat transfer in double facades with large gap dimensions.

9.3 Heat transfer coefficients for the interior and facade air gap

The heat transfer coefficient inside h_i depends on the room geometry, the heating system, the type of ventilation and other parameters. For most applications, the assumption of a constant heat transfer coefficient is sufficiently exact. The European standard EN 832 states that a mean thermal resistance of $1/h_i = 0.13 \text{ m}^2 \text{ KW}$ can be used inside.

The convective heat transmission coefficients in the air gap h_{ca} and h_{cb} depend on the facade geometry and on the flow type (free or forced). With cavity facade geometries with gap dimensions over 10 cm, the flow rates are generally small (under 0.5 m s^{-1}), so the free convection proportion cannot be neglected even with fan-driven forced ventilation. The derivation of heat transfer coefficients from first principles is extremely complex, so that mostly empirical correlations for Nußelt numbers as a function of flow velocity and fluid properties are used. Experiments by Schwab (1993) on asymmetrically heated double facades with a height to gap width ratio of 50 showed a steady increase of the heat flux density with rising Reynolds number. If one of the plates was cooled below the inlet air temperature, the heat flux densities on the hot plate even decreased with rising Reynold number. The correlations used below fit experiments on a 6.5 m high double facade with 0.14 m gap distance, where the 'hot' plate was a photovoltaic module and the back of the air gap was a double glazing.

Under mixed convection conditions, the Reynolds number consists of a free and a forced convection part.

$$Re = \sqrt{Re_{free}^2 + Re_{for}^2} \quad (9.14)$$

The free convection proportion results from the temperature-induced density variation over the gap height L .

$$Re_{free} = \sqrt{\frac{Gr}{2.5}} \quad (9.15)$$

$$Gr = \frac{g\beta'L^3[(\bar{T}_f - \bar{T}_{a,b})]}{\nu^2} \quad (9.16)$$

$$Re_{for} = \frac{vL}{\nu} \quad (9.17)$$

With flow conditions over an individual plate, the boundary layer flow at the lower panel edge can begin laminary and after a certain length become turbulent, the transition point being about $Re = 2 \times 10^5$ (Merker and Eiglmeier, 1999). The mean Nußelt number is formed from a laminar and a turbulent proportion.

$$Nu = \sqrt{Nu_{lam}^2 + Nu_{turb}^2} \quad (9.18)$$

For Prandtl numbers between $0.6 < Pr < 10$, integration over the local Nußelt number produces the laminar proportion:

$$Nu_{lam} = 0.664 \sqrt{Re} \sqrt[3]{Pr} \quad (9.19)$$

The turbulent proportion is calculated from an empirical correlation derived from numerical integration of the boundary layer equation:

$$Nu_{turb} = \frac{0.037 Re^{0.8} Pr}{1 + 2.443 Re^{-0.1} \left(Pr^{\frac{2}{3}} - 1 \right)} \quad (9.20)$$

with

$$Pr = \frac{vc\rho}{\lambda}$$

From the mean Nußelt number, the heat transmission coefficients result as usual in

$$h_c = \frac{Nu \lambda}{L}$$

Example 9.1

Calculation of the two convective and of the radiation heat transfer coefficient of a 6.5 m high back-ventilated PV facade with 14 cm gap depth, for PV module temperatures of 50°C and a back gap temperature (e.g. glazing) of 30°C. The flow velocity of the gap air is 0.3 m s⁻¹ and the fluid temperature is 40°C.

The material values for the left, warm PV side are calculated with the mean temperature between the surface and the gap air, i.e. here 45°C; for the right side similarly from the surface temperature of the glazing and the mean gap air temperature (i.e. 35°C).

Material values of the gap air:

Mean temperature of	45°C (PV)	35°C (glass)	
Kinematic viscosity ν	17.546×10^{-6}	16.60×10^{-6}	m ² s ⁻¹
Heat conductivity of air λ	0.02758	0.02554	W m ⁻¹ K ⁻¹
Density ρ	1.095	1.130	kg m ⁻³
Heat capacity c	1008.25	1007.75	J kg ⁻¹ K ⁻¹
heat expansion coefficient β'	0.00314	0.0032	K ⁻¹
Re_{for}	111 134	117 486	
G_r	2.8×10^{11}	3.123×10^{11}	
Re_{free}	334 329	353 435	
Pr	0.75	0.74	
Re	352 316	372 450	
Nu_{turb}	863	894	
Nu_{lam}	325	332	
Nu	922	954	

From these result the heat transfer coefficients $h_{ca} = 3.66 \text{ W m}^{-2} \text{ K}^{-1}$, $h_{cb} = 3.75 \text{ W m}^{-2} \text{ K}^{-1}$ and $h_r = 3.07 \text{ W m}^{-2} \text{ K}^{-1}$.

If the temperatures are not given, the heat transfer coefficients and temperatures must be determined iteratively. Firstly, temperatures at the gap confinement surfaces are assumed, so that the heat transfer coefficients can be calculated and then the mean temperatures recalculated. With the new temperatures, heat transfer coefficients are again calculated, etc.

Example 9.2

Calculation of the mean temperatures and air outlet temperature of a 6.5 m high, 1 m wide and 0.14 m deep back-ventilated photovoltaic facade under the following boundary conditions:

Irradiance on the facade	$G = 800 \text{ W m}^{-2}$
Ambient temperature	$T_o = 10^\circ\text{C}$
Room air temperature	$T_i = 20^\circ\text{C}$
Wind velocity	$v_w = 3 \text{ m s}^{-1}$
Flow velocity in the gap	$v = 0.3 \text{ m s}^{-1}$
Absorption coefficient of the PV module	$\alpha_{pv} = 0.8$
Transmission coefficient of the PV module	$\tau_{pv} = 0.1$
Absorption coefficient of the back glazing	$\alpha_b = 0.05$
Layer thickness of the PV module	$s_{pv} = 0.01 \text{ m}$
Heat conductivity of the PV module	$\lambda_{pv} = 0.8 \text{ W m}^{-1} \text{ K}^{-1}$
Heat resistance of the back glazing	$R_b = 0.18 \text{ m}^2 \text{ K W}^{-1}$
Heat transfer coefficient inside	$h_i = 8 \text{ W (m}^2 \text{ K)}^{-1}$
Electrical efficiency	$\eta_{el} = 0.12$
Emission coefficient	$\epsilon = 0.88$

Solution:

The temperatures from Example 9.1 are used as initial values, resulting in the heat transfer coefficients of the first example. With these heat transfer coefficients the mean fluid temperature is calculated and inserted into the equations of the heat transfer coefficients. After four iterations, the error in the temperatures is less than 0.1°C and the coefficients are:

Heat transfer coefficient absorber fluid	$h_{ca} = 5.1 \text{ W (m}^2 \text{ K)}^{-1}$
Heat transfer coefficient glazing fluid	$h_{cb} = 3.5 \text{ W (m}^2 \text{ K)}^{-1}$
Heat transfer coefficient for radiation	$h_r = 2.9 \text{ W (m}^2 \text{ K)}^{-1}$
Thus a mean fluid temperature results	$T_f = 19.6^\circ\text{C}$.

If the mean fluid temperature is inserted into Equation (9.7) and Equation (9.8), this results in

Mean PV module temperature	$T_a = 40.9^\circ\text{C}$
Glazing temperature on the gap side	$T_b = 26.5^\circ\text{C}$

Fluid outlet temperature at 6.5 m height

$$T_{f,out} = 27.2^{\circ}\text{C}$$

The fluid outlet temperature after 6.5 m height is 27.2°C , i.e. the PV facade has warmed up the ambient air by 17.2 K. The temperature distribution over the facade height for the above boundary conditions explains the temperature rise of the fluid and of the gap confinement surfaces. The surface temperature of the back glazing on the room side is also raised slightly above the room temperature despite low outside temperatures of 10°C .

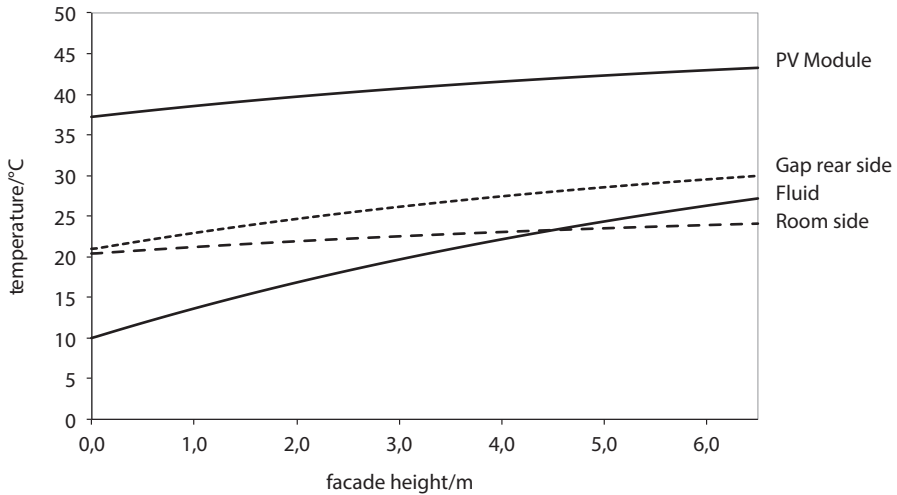


Figure 9.11 Rise in temperature of the ambient air entering through a back-ventilated PV facade. The boundary conditions are 800 W m^{-2} irradiance, 3 m s^{-1} wind velocity, 10°C outside temperature, 20°C room air temperature and a flow velocity in the gap of 0.3 m s^{-1} .

9.4 Building-integrated solar components (U and g values)

To determine the influence of building-integrated solar components on the thermal behaviour of the building, it is a good idea to use the usual component characteristic values such as the heat transfer coefficient U and the total energy transmission factor g . These characteristic values are constant in conventional components. Since, however, the energy flows and temperature levels of absorbing solar elements depend greatly on irradiance and ambient temperature, the U and g values must be determined time-dependently.

Since energy gains from warm-air use by back-ventilated facades are also to be considered, the U and g values are divided into two components, which differentiate between transmission and ventilation heat flows (Bloem et al., 1997). U_{trans} characterises the entire calorific losses from the interior, U_{vent} the calorific losses from the inside to the air gap, which can be recovered by the back ventilation, g_{trans} the solar radiation gains transmitted directly into the interior, and g_{vent} the absorbed radiation gains contributing to the heating up of the air volume flow.

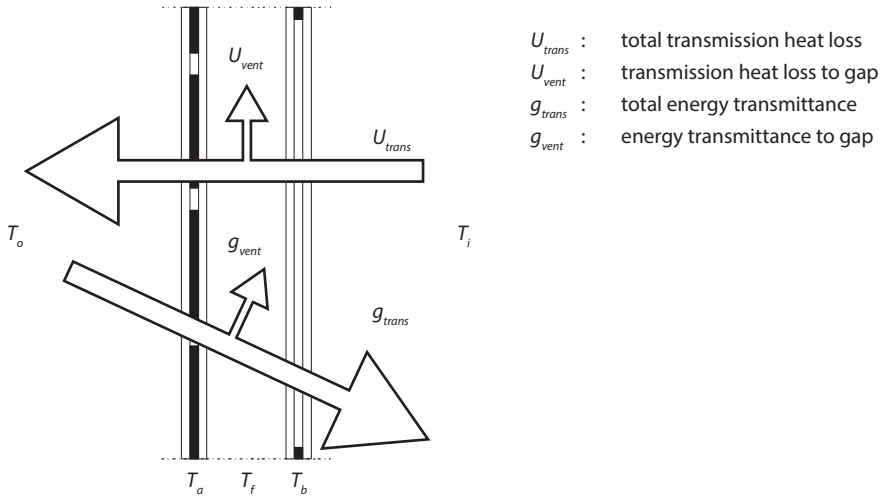


Figure 9.12 Characteristic values for the thermal characterisation of a back-ventilated facade.

The transmission heat flow $\dot{Q}_{trans}$ describes the effective heat losses from the room as the difference of transmission heat losses and direct solar gains, calculated similarly to the procedure for the effective U values of windows. The ventilation heat flow $\dot{Q}_{vent}$ contains that part of the transmission heat losses that can be recovered by the back ventilation (related to the temperature difference between the room and environment) plus the heat gains of the air gap due to absorption of solar radiation.

$$\dot{Q}_{trans} = U_{trans}(T_i - T_o) - g_{trans}G \quad (9.21)$$

$$\dot{Q}_{vent} = U_{vent}(T_i - T_o) + g_{vent}G \quad (9.22)$$

The heat transfer coefficient U_{trans} is defined via the heat transfer coefficient of the rear U_b , which is known and calculated from the mean temperature $\bar{T}_b$. For this definition, the heat flow from the room $U_b(T_i - \bar{T}_b)$ is normalised to the temperature difference between the room and the environment.

$$U_b(T_i - \bar{T}_b) = U_{trans}(T_i - T_o) \quad (9.23)$$

$$U_{trans} = \frac{U_b(T_i - \bar{T}_b)}{(T_i - T_o)} \quad (9.24)$$

The direct energy transmission coefficient g_{trans} contains the optical transmission of the facade and consists of the transmission of the PV module and the back glazing. For example, g_{trans} results, for a facade with 15% glass proportion of the PV module and a back double-glazing, from the

product of the glass proportion of 15%, the single glazed PV module transmission of approximately $\tau_{PV-glass} = 90\%$ and the transmission of the double glazing of approximately $\tau_{gap, rear} = 80\%$.

$$g_{trans} = \frac{A_{glass}}{A_{module}} \tau_{PV-glass} \tau_{gap, rear} = 0.15 \times 0.9 \times 0.8 = 0.108 \quad (9.25)$$

The useful energy $\dot{Q}_{vent}$ of the heated air within the gap is calculated directly from the mean surface temperatures of the gap confinement surfaces and from the mean fluid temperature.

$$\dot{Q}_{vent} = h_{ca}(\bar{T}_a - \bar{T}_f) + h_{cb}(\bar{T}_b - \bar{T}_f) \quad (9.26)$$

If the mean temperatures from the energy balance equations are used, $\dot{Q}_{vent}$ can be calculated as a function of the room and ambient temperatures, irradiance and the produced electricity,

$$\dot{Q}_{vent} = \left(\frac{1-e^{-ZL}}{ZL} \right) (T_i D_1 - T_o (D_5 - D_2)) + \left(\frac{1-e^{-ZL}}{ZL} \right) (G D_4 - \dot{Q}_{el} D_3) \quad (9.27)$$

with L describing the height of the facade.

In agreement with Equation (9.22), $\dot{Q}_{vent}$ is divided into a solely temperature-dependent term, described by U_{vent} , and an irradiance-dependent term, characterised by g_{vent} . Since the PV electrical efficiency is essentially irradiance-dependent, it is integrated into the g_{vent} term. The normalisation to the temperature difference between the room and environment thus produces for U_{vent} :

$$U_{vent} = \frac{\left(\frac{1-e^{-ZL}}{ZL} \right) (T_i D_1 - T_o (D_5 - D_2))}{T_i - T_o} \quad (9.28)$$

The irradiance-dependent term is normalised to the irradiance G . A sufficiently exact approximation in thermal terms results from assuming a constant electrical efficiency η_{el} for the PV module.

$$g_{vent} = \left(\frac{1-e^{-ZL}}{ZL} \right) \left(D_4 - \frac{\dot{Q}_{el} D_3}{G} \right) = \left(\frac{1-e^{-ZL}}{ZL} \right) (D_4 - \eta_{el} D_3) \quad (9.29)$$

Example 9.3

Calculation of the component characteristic values U_{trans} , U_{vent} and g_{vent} of the photovoltaic double facade, with the boundary conditions of the last example.

U_{trans}	$= -2.14 \text{ W (m}^2 \text{ K)}^{-1}$
U_{vent}	$= 1.07 \text{ W (m}^2 \text{ K)}^{-1}$
g_{vent}	$= 0.178$

Thus the ventilation gains at an irradiance of 800 W m^{-2} amount to

$$\dot{Q}_{vent} = U_{vent} (T_i - T_o) + g_{vent} G = 1.07 \underbrace{\frac{\text{W}}{\text{m}^2 \text{ K}} (20 - 10) \text{ K}}_{10.7} + 0.178 \times 800 \underbrace{\frac{\text{W}}{\text{m}^2}}_{142.4} = 153.1 \frac{\text{W}}{\text{m}^2}$$

The gains from the absorption of solar radiation at a solar efficiency g_{vent} of 17.8% dominate.

Since the back-glazing temperature is higher than the room temperature T_i , the transmission coefficient U_{trans} is negative and, in effect, heat gains are supplied to the room: $\dot{Q}_{trans} = -97.4 \text{ W m}^{-2}$.

9.5 Warm-air generation by photovoltaic facades

With the above method, hourly U and g values can be calculated, and hence hourly energy balances for the facade system can be drawn up. Weighted monthly average values can then be used for heating-energy calculations based on the monthly balance procedure of EN 832. The thermal gains of the back-ventilated facade consist of direct solar gains (described by the constant g_{trans} value), indirect solar gains from solar radiation absorption and subsequent heat transfer to the gap air (g_{vent}), plus the heat recovered from the interior (U_{vent}). The irradiance-weighted g_{vent} value is given by

$$\bar{g}_{vent} = \frac{\sum_{j=1}^{\text{hours per month}} g_{vent,j} G_j}{\sum_{j=1}^{\text{hours per month}} G_j} \quad (9.30)$$

The useful energy from heat recovery is calculated by the temperature difference-weighted value $\bar{U}_{vent}$.

$$\bar{U}_{vent} = \frac{\sum_{j=1}^{\text{hours per month}} U_{vent,j} (T_i - T_{o,j})}{\sum_{j=1}^{\text{hours per month}} (T_i - T_{o,j})} \quad (9.31)$$

The entire transmission heat loss of the room is expressed by a mean heat transfer coefficient $\bar{U}_{trans}$.

$$\bar{U}_{trans} = \frac{\sum_{j=1}^{\text{hours per month}} U_{trans,j} (T_i - T_{o,j})}{\sum_{j=1}^{\text{hours per month}} (T_i - T_{o,j})} \quad (9.32)$$

	G_m /kWh m ⁻²	T_o /°C	n_h	$\bar{g}_{vent} = \eta_{th}$	$\bar{U}_{vent}(\bar{T}_i - \bar{T}_o)n_h$ /kWh m ⁻²	$\bar{U}_{trans}(\bar{T}_i - \bar{T}_o)\eta_h$ /kWh m ⁻²	Q_{trans} /kWh m ⁻²	Q_{vent} /kWh m ⁻²
Jan	82	9.75	744	0.142	8.05	11.00	2.16	19.67
Feb	88	9.95	672	0.130	7.03	9.30	-0.20	18.50
Mar	106	11.26	744	0.128	6.98	8.08	-3.37	20.58
Apr	94	12.92	720	0.124	5.18	5.78	-4.32	16.80
May	80	16.16	744	0.126	2.91	1.80	-6.88	13.02
Jun	80	20.06	720	0.122	0.17	-3.69	-12.32	9.90
Jul	88	23.65	744	0.118	-2.29	-9.06	-18.61	8.15
Aug	100	23.46	744	0.129	-2.20	-9.60	-20.36	10.64
Sep	108	21.26	720	0.134	-0.71	-6.84	-18.48	13.75
Oct	107	17.01	744	0.130	2.26	-0.52	-12.12	16.25
Nov	90	12.70	720	0.137	5.42	6.16	-3.57	17.73
Dec	84	10.75	744	0.146	7.10	9.25	0.12	19.45
Sum/ Mean	1107	15.7	8760	0.13	40	22	-98	184

Table 9.5 Climatic boundary conditions, ventilation gains and transmission heat losses of the back-ventilated south-facing facade in Barcelona.

Since in the monthly energy balance procedure only solar gains are taken into account, $\bar{U}_{vent}$ can also be deduced from the total transmission heat loss coefficient $\bar{U}_{trans}$, so an effective transmission heat loss $\bar{U}_{trans, eff} = \bar{U}_{trans} - \bar{U}_{vent}$ remains.

With these characteristic values, the contribution of back-ventilated facades to building heating energy can be calculated and compared with conventional facade systems. In summer the thermal load of the PV facade for the interior is easily calculated from the total of direct solar

gain (g_{trans}) and indirect gain (U_{trans}). Any useful heating energy is calculated as usual from the rise in temperature of the gap air via g_{vent} and U_{vent} .

With the above procedure, monthly component characteristic values for a back-ventilated PV facade for a Mediterranean (Barcelona) and a German (Stuttgart) climate have been calculated. The total ventilation gains Q_{vent} result from the sum of the ventilation gains by irradiance and the heat flow from the room into the air gap, multiplied by the number of hours in the month n_h . The monthly irradiance G_m is given by the sum of hourly values in kWh m^{-2} .

$$Q_{vent} = \bar{g}_{vent} G_m + \bar{U}_{vent} (\bar{T}_i - \bar{T}_o) n_h \quad (9.33)$$

The mean ventilation g value $\bar{g}_{vent}$ can be interpreted directly as the solar thermal efficiency of the back-ventilated facade. The monthly transmission heat loss Q_{trans} is calculated from the total heat flow from the room, minus the direct solar gains.

$$Q_{trans} = \bar{U}_{trans} (\bar{T}_i - \bar{T}_o) n_h - \bar{g}_{trans} G_m \quad (9.34)$$

The g_{trans} value only takes into account the optical transmission of the glazing system and is constant here at 0.108. The thermal efficiency at the low flow velocity of 0.3 m s^{-1} is 13% on average. From the total transmission heat losses of 50 kWh m^{-2} in the heating season, 40 kWh m^{-2} can be recovered by feeding back the heated gap air (interior temperature constant at 20°C).

In Stuttgart, the average thermal efficiency is somewhat higher at 15%. Despite the high U value of the non-coated double-glazing of $4 \text{ W (m}^2 \text{ K)}^{-1}$, the transmission heat loss of the PV facade system is in effect clearly smaller due to the thermal energy gains, and the real U_{trans} value varies between 1.5 and $1.8 \text{ W (m}^2 \text{ K)}^{-1}$. Of the total transmission heat losses in the heating season, 142 kWh per m^2 of facade system, 99 kWh m^{-2} can be recovered.

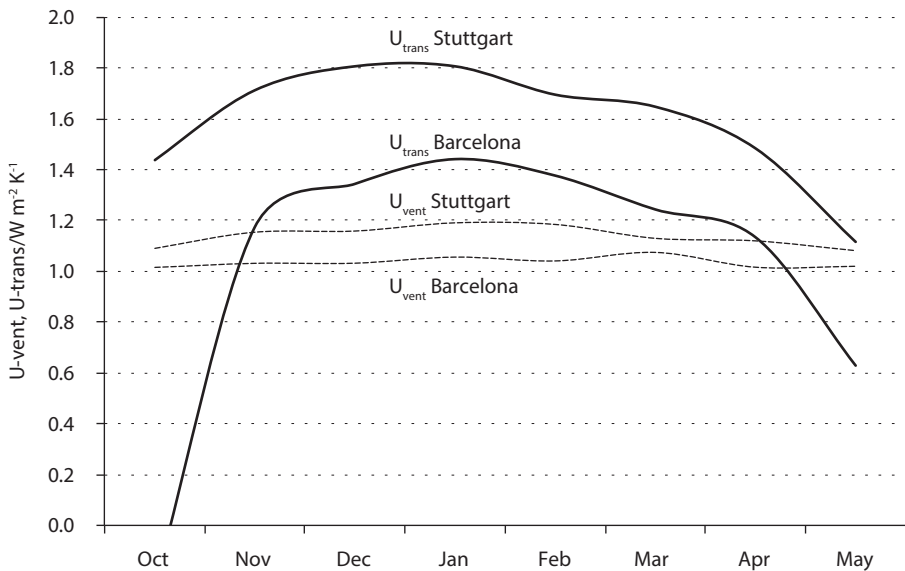


Figure 9.13 Mean monthly characteristic values of the back-ventilated photovoltaic facade with conventional double-glazing as the gap rear ($U = 4 W (m^2 K)^{-1}$) in Stuttgart and Barcelona.

With the methodology described, different facade types can be compared in energy terms, with the thermal gains depending greatly on the thermal separation of the gap rear from the building. If the thermal separation is improved, the transmission heat losses to the gap fall, but so do the ventilation heat gains. At the same time, however, the summer load of the room is reduced, so the best possible thermal separation is always to be recommended.

While a back-ventilated PV facade produces solar ventilation gains of between 83 and $93 kWh(m^2a)^{-1}$ in Stuttgart, depending on the gap rear construction, the solar ventilation gains of a vitreous cavity facade are only $15 kWh(m^2a)^{-1}$. The heat recovered from the room depends only on the quality of the thermal separation between the room and the back-ventilation gap, and is $27 kWh m^{-2}$ for cavity facades with and without PV when heat-protecting glass is used. The direct solar gains are very high in fully vitreous facade types, between 250 and $285 kWh m^{-2}$, and must in all cases be controlled with shading devices. With commercial g values for external sun protection of about 20% , overheating problems in the summer must be expected in such all-glass facades.

9.6 Photovoltaic thermal collectors for heating and cooling generation

Photovoltaic installations integrated in buildings can be used not only for combined electricity and heating production but also for cooling. For this purpose, the long-wave radiation at night is used to cool a fluid that circulates below the PV module.

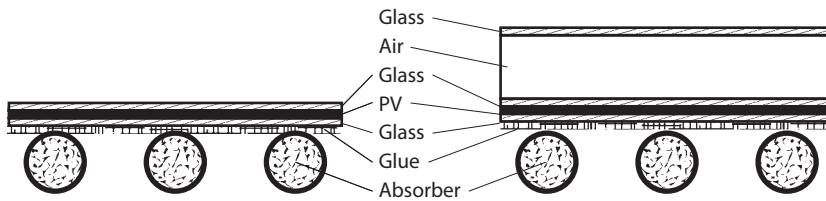


Figure 9.14 Construction of a PVT Collector for cooling purposes, and for combined electrical and heating production with an extra glass sheet to reduce thermal losses.

Despite the numerous publications about theoretical and experimental installations of PVT collector systems, very few applications are market-ready and operating experience is scarce. The international engineering agency IEA lists over 50 PVT projects in the last twenty years, of these less than 20 are water-based (Hansen, 2007). Most of these projects were realised in Europe, mainly the Netherlands and Great Britain. Furthermore, there are several projects in Thailand, where large-scale amorphous solar cells were installed in PVT water collectors on hospitals and government buildings (Jaikla et al., 2005).

Currently several manufacturers are working on the production and commercialisation of different PVT systems, e.g. the Israeli producer Millennium Electric with an uncovered PVT collector for combined energy production of electricity, water and air; the Dutch manufacturer PVTwins with an uncovered hybrid collector; Menova Energy from Canada with a concentrated system called PowerSpar, etc. Within the framework of the European student competition Solar Decathlon 2010, a PVT collector for cooling was developed and implemented at the University of Applied Sciences Stuttgart.

Performance calculation of a PVT collector

Heat losses of the absorber to the surroundings as well as the collector efficiency and the heat removal factors F' and F_R are calculated as described in chapter 3.2 Sun-Earth geometry on page <?>. The long-wave emissions to the sky as the useful cooling power replaces the heating power through absorbed radiation.

For an uncovered PVT collector, the heat losses of the absorber through the glued PV module to front and back are calculated as parallel-aligned thermal resistances. A simplified model for a PV module and the thermal absorber assumes the same temperature node (Erell and Etzion, 2000).

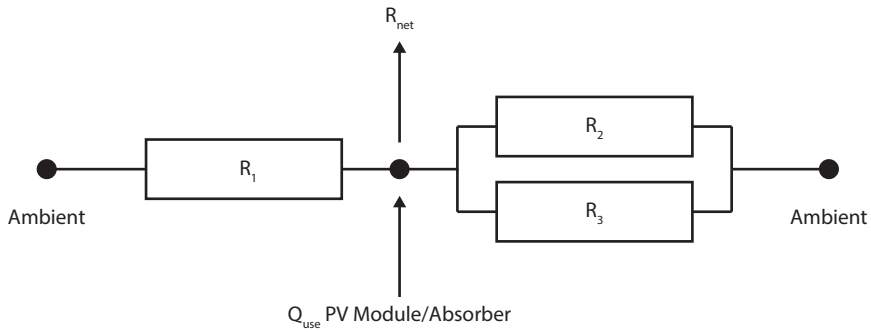


Figure 9.15 Thermal resistance for an uncovered PVT collector.

Where,

- R_1 : thermal resistance resulting from free and forced convection between the PV module and the ambient air
- R_2 : thermal resistance through long-wave radiation between the absorber and the surroundings/roof
- R_3 : thermal resistance through free convection between the back of the absorber and the surroundings/roof

The total resistance of the front and back results for the parallel resistances in:

$$U_t = \frac{1}{R_f} + \frac{1}{R_b} \quad (9.35)$$

Here the resistance of the front consists solely of convective resistances – the long-wave radiation exchange is considered as useful energy.

$$R_f = R_1$$

The resistance of the back surface R_b is calculated from the parallel radiative and convective resistances R_2 and R_3 :

$$R_b = \frac{R_2 R_3}{R_2 + R_3}$$

To calculate the efficiency factor of the collector F' , the cooling unit's efficiency F must be defined, which depends on the external radius of the thermal collector, on the distance between the tubes as well as the absorber sheet thickness and their thermal properties. The heat removal factor F_R can then be calculated with known F' .

$$F' = \frac{1/U_t}{W \left[\frac{1}{h_f \pi D_i} + \frac{1}{\lambda_{kon, eff}} + \frac{1}{((W-D)F+D)U_t} \right]}$$

As opposed to solar thermal heat generation of the absorbed radiation, the long-wave radiative exchange R_{net} between the PV module (temperature T_{PV}) and the night sky (temperature T_{sky}) is used for the effective performance balance. If the fluid input temperature is higher than the ambient temperature T_o , effective cooling is produced – the heat loss is positive.

$$\dot{Q}_n = AF_R \left(R_{net} + U_t (T_{f,in} - T_o) \right) \quad (9.36)$$

with

$$R_{net} = h_r (T_{PV} - T_h)$$

and

$$h_r = \sigma \varepsilon_1 (T_{PV}^2 + T_h^2) (T_{PV} + T_h)$$

The calculation of the heat transfer coefficient and the effective performance must be iterative, but converge quickly if the initial values are chosen sensibly.

An exact calculation is possible, if the thermal resistance between the PV module and absorber are considered.

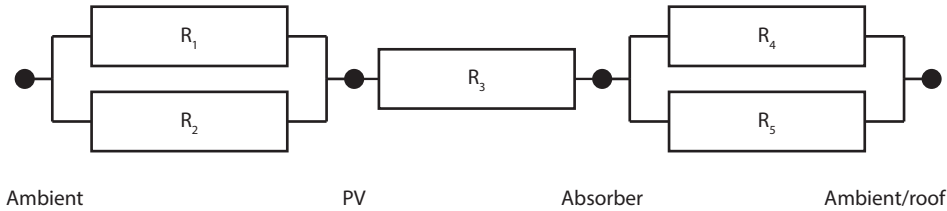


Figure 9.16 Thermal circuit diagram of the uncovered and non-insulated PVT collector.

Where

- R_1 : Thermal resistance of longwave radiation from the PV cover towards the sky
- R_2 : Thermal resistance of mixed convection (free + forced) from the PV cover to the ambient
- R_3 : Thermal resistance of conduction through the PV module and glue
- R_4 : Thermal resistance of longwave radiation from the absorber towards the roof
- R_5 : Thermal resistance of mixed convection from the absorber to the ambient/roof

The parallel thermal resistances for radiation and convection R_1 and R_2 are calculated between the PV module and the ambient temperature, meaning the longwave radiation exchange must be standardised to the temperature difference of the surroundings.

$$h_r = \sigma \epsilon_{PV} \frac{(T_{PV}^4 - T_{sky}^4)}{T_{PV} - T_{amb}} \quad (9.37)$$

If the PV temperature is near the ambient, the heat transfer coefficient approaches infinity, and thus the first model is to be used.

The calculation of convective and long-wave radiation resistances has already been examined in detail in Chapter 4. The calculation of the thermal resistance R_3 of the PV module including the glue layer is new here.

Layer	Thickness /mm	Thermal conductivity /W m ⁻¹ K ⁻¹
Glass	3	0.95
EVA	0.5	0.23
Antireflection layer	$(0.06 - 0.1) \times 10^{-3}$	1.38
Silicon	0.25	148
EVA	0.5	0.23
Tedlar	0.1	0.36
Glue Epoxy/Aluminium	0.1	0.85

Table 9.3 Construction and thermal properties of a PV module including glue layer.

As a result the total resistance $R_2 = 0.0079 \text{ m}^2 \text{ K W}^{-1}$. The total heat transfer coefficient can thus be calculated iteratively (see Figure 9.8).

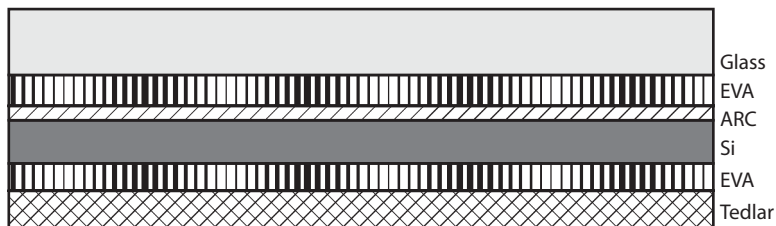


Figure 9.17 Construction of a photovoltaic module with a glass/Tedlar composite.

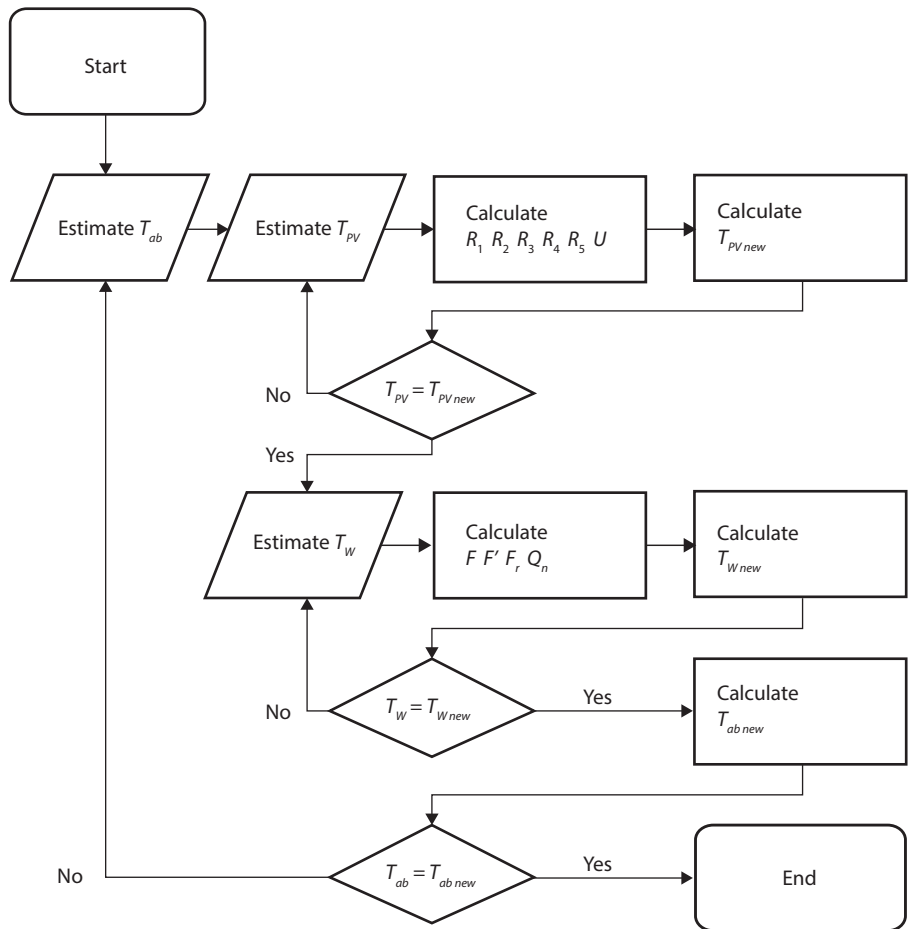


Figure 9.18 Flow chart of the iterative calculation of thermal resistance and efficiency factors.

Example 9.4

Calculation of the thermal performance of a 1 m² collector with a PV module as in Table 9.1, with a 0.2 mm aluminium absorber ($\lambda = 220 \text{ W m}^{-1} \text{ K}^{-1}$). The tube distance is 8 cm, the copper tubes of the PVT absorbers have 12 mm exterior and 10 mm interior diameter. The sky temperature is 5°C with an air temperature of 20°C at night. The input temperature of the water is 25°C at a mass flow of 40 kg m⁻² h⁻¹. The emissivity of the glass PV module is 0.85 and the emissivity of the absorber is 0.02 at roof surface emissivity of 0.85.

Model with common temperature nodes		Model with different temperature nodes between PV and absorber	
R_1	$= 0.02986 \text{ m}^2 \text{ K W}^{-1}$	R_1	$= 0.0238 \text{ m}^2 \text{ K W}^{-1}$
R_2	$= 8.672 \text{ m}^2 \text{ K W}^{-1}$	R_2	$= 0.3252 \text{ m}^2 \text{ K W}^{-1}$
R_3	$= 0.4106 \text{ m}^2 \text{ K W}^{-1}$	R_3	$= 0.007679 \text{ m}^2 \text{ K W}^{-1}$
		R_4	$= 0.4106 \text{ m}^2 \text{ K W}^{-1}$
		R_5	$= 8.672 \text{ m}^2 \text{ K W}^{-1}$
T_{PV}	$= 22.35^\circ\text{C}$	T_{PV}	$= 21.83^\circ\text{C}$
		T_{ab}	$= 22.46^\circ\text{C}$
F	$= 0.954$	F	$= 0.7708$
F'	$= 0.9083$	F'	$= 0.6175$
F_R	$= 0.8597$	F_R	$= 0.4905$
U_t	$= 5.672 \text{ W m}^{-2} \text{ K}^{-1}$	U_t	$= 36 \text{ W m}^{-2} \text{ K}^{-1}$
h_r	$= 4.553 \text{ W m}^{-2} \text{ K}^{-1}$	h_r	$= 42 \text{ W m}^{-2} \text{ K}^{-1}$
R_{net}	$= 78.98 \text{ W m}^{-2} \text{ K}^{-1}$	Q_{rad}	$= 76.4 \text{ W m}^{-2} \text{ K}^{-1}$
Q_{use}	$= 92.3 \text{ W m}^{-2}$	Q_{use}	$= 88.24 \text{ W m}^{-2}$

The PVT collector developed by the University of Applied Sciences Stuttgart as part of the Solar Decathlon Project was simulated using this algorithm. The PVT module consists of monocrystalline cells in a frameless glass construction dimensioned $1.194 \text{ m} \times 2.324 \text{ m}$ and an electric performance of 410 W_p per Module. The electric module efficiency is around 14.7%. In total there are 38 m^2 of horizontal PVT collectors on the roof. The collectors were used for cooling production at night with a mass flow of $40 \text{ kg m}^{-2} \text{ h}^{-1}$ driven with an 85 W electric pump.

Firstly, the measurements of a prototype model were carried out in Stuttgart, taking place however in November with already low ambient temperatures between 5 and 15°C . The sky temperature was calculated to as low as -15°C under clear conditions. With a long-wave radiation performance between 100 and 120 W m^{-2} , the temperature decrease was 3 K .

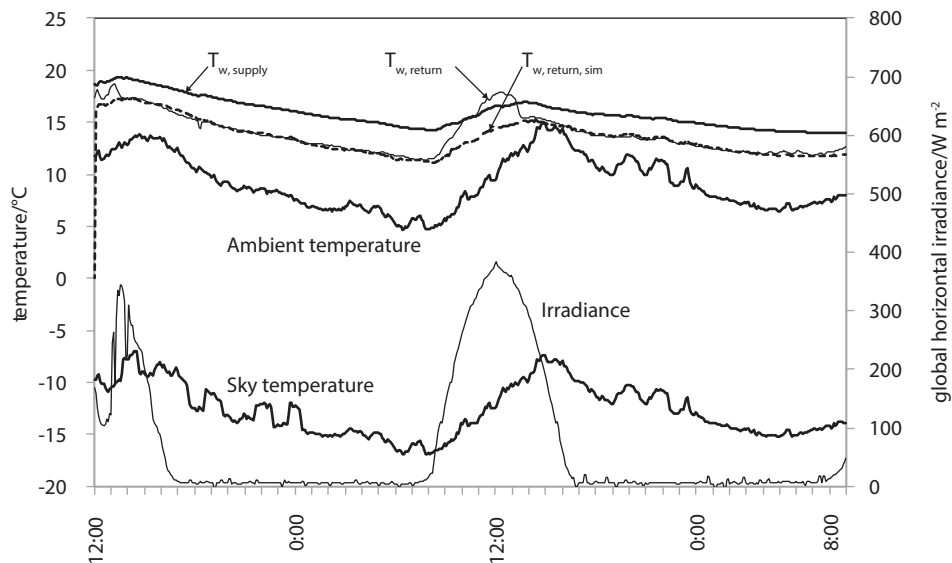


Figure 9.19 Measured and simulated temperatures of the PVT collector from 18 to 20 November 2009 in Stuttgart.

In June 2010, the performance of the entire system with PVT and storage tank was tested and measured on location in Madrid. The nightly cooling cycles of 9 - 11 hours achieved an average cooling power of 38 and 65 W m⁻² with very good ratios of cooling energy to electrical consumption between 17 and 30.

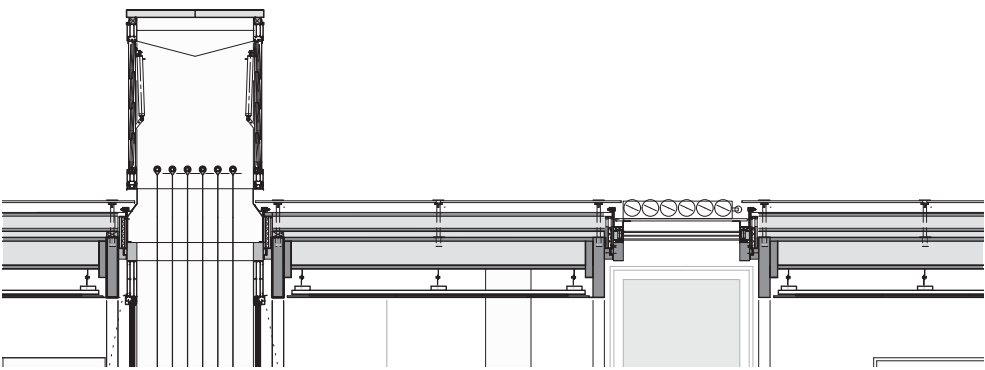


Figure 9.20 Roof construction of the Stuttgart Solar Decathlon building with PVT collectors (HFT Stuttgart).

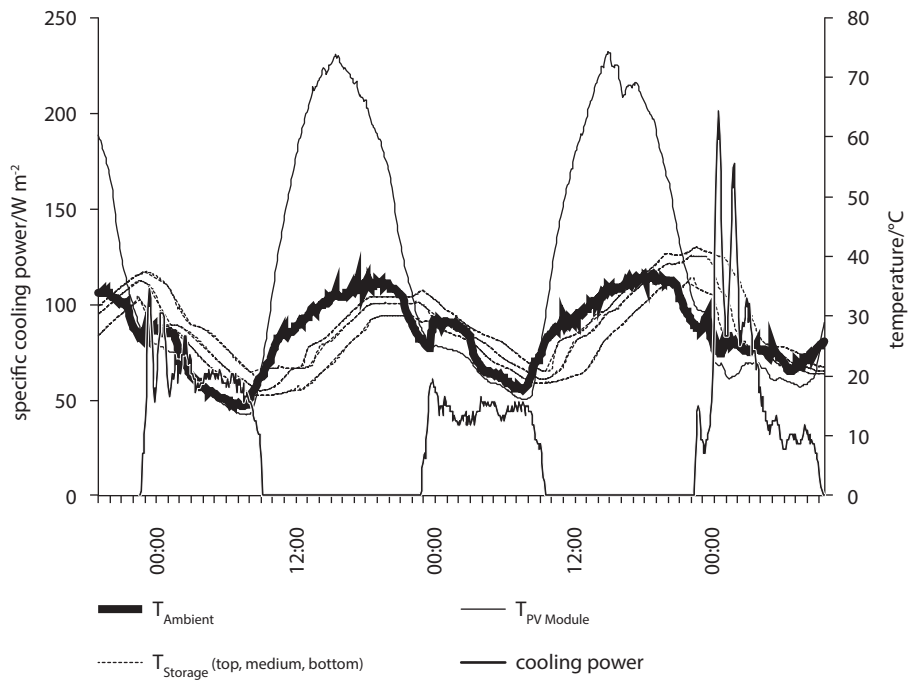


Figure 9.21 Temperature measurements and obtained cooling power from 22 - 25 June 2010 in Madrid when the PVT was connected to the storage tank.

An overview of the commissioning results during the competition time is displayed in Table 9.4.

Date	Energy dissipated /kWh	Operation time /h	Mean cooling power /W m ⁻²	E _{elec} pump /kWh	COP _{elec}
19 - 20 June	23.7	9.5	65.6 ± 3.159	0.81	29.3
21 - 22 June	13.4	9.2	38.5 ± 3.85	0.78	17.2
22 - 23 June	24.6	10.6	61.1 ± 2.831	0.90	27.3
23 - 24 June	17.0	10.4	43.0 ± 2.886	0.89	19.2
24 - 25 June	25.7	10.8	62.4 ± 2.779	0.92	27.9

Table 9.4 Summary of measurement results from the Solar Decathlon building in Madrid.

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